USAAMRDL-TR-75-59B



# INVESTIGATION OF ADVANCED HELICOPTER STRUCTURAL DESIGNS

Volume II - Free Planetary Transmission Drive

 Sikorsky Aircraft
Division of United Technologies Corporation Stratford, Conn. 06602
Stratford, Conn. 06602
May 1976

**Final Report** 

Approved for public release; distribution unlimited.



**Prepared** for

EUSTIS DIRECTORATE U. S. ARMY AIR MOBILITY RESEARCH AND DEVELOPMENT LABORATORY Fort Eustis, Vo. 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 was conducted by Boeing-Vertol under the terms of Contract DAAJ02-74-C-0066.

Numerous design concepts, material selections, and manufacturing techniques were investigated for the various helicopter components (e.g., body group, main rotor, and transmission). 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 Application 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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Pransmission concept. This type of drive, compared with conventional two-stage planetary transmissions, promises several advantages, including 10% lighter weight, 16.5% lower cost in 500-unit quantities, 1/2 percent greater efficiency, almost twice the reliability, and greater tolerance to loss of lubrication.

The free planet transmission concept is defined as a planetary gear arrangement in which the planets are not constrained by being secured to a spider or planetary carrier. This lack of constraint is achieved by satisfying certain geometric relationships with conventional compound planetary gearing.

This study commenced with a survey of current work in free planet transmissions and a review of the actual hardware used in a 500-HP development test. This review indicated the attractiveness of the free planet transmission concept.

During preliminary design of a free planet transmission for the Medium Utility Transport (MUT) aircraft, design requirements were established for power, speeds, rotations, and size of envelope. Various drive train arrangements were considered, including high-speed bevel gear inputs, high-speed helical gear inputs, and dual high-speed spur gear inputs. The high-speed spur gear arrangement was selected as best for the MUT aircraft. A free planet transmission was also examined for a UTTAS-type drive arrangement.

Design enalysis for the free planet was developed for gear tooth stress, axial length, pinion shaft design, and roller ring loads. This design analysis was the basis for development of a computer program for selecting the free planet design. The gear geometry parameters were varied, and some simple parametric curves were developed. Preliminary design layouts were made of both a conventional two-stage planetary and a free planetary transmission.

A final design layout was made of the free planet transmission, and cost, weight, survivability/vulnerability, and reliability characteristics were determined. Through use of a helicopter design model, the impact of the free planet transmission on aircraft performance was determined.

It is recommended that a free planet transmission be built for a helicopter application in which a single-engine 20,000 to 30,000 rpm input is available at a design power of 400 to 500 HP. This would permit verification of the concept and demonstration of its projected improvements.

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## PREFACE

The Program reported herein was conducted during a seven-month period from 20 January 1975 to 20 July 1975 for the Eustis Directorate, U. S. Army Air Mobility Research and Development Laboratory (USAAMRDL), Fort Eustis, Virginia, under Contract DAAJ02-74-C-0061.

USAAMRDL technical direction was provided by Mr. L. Thomas Mazza and Mr. E. Rouzee Givens of the Eustis Directorate, Technology Applications Division.

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## INTRODUCTION

Transmissions employing conventional planetary gear drives have performed satisfactorily for many years. Continuing research and development has been aimed at improving the performance of such transmissions. One promising development resulting from this effort has been the free planet transmission. This transmission can be classified as a quasi-compound planetary employing a sun gear, planet spindle assemblies, ring gears, and rolling rings.

The free planet concept was developed in the 1960's by The Curtiss-Wright Corporation. The concept covers broadly those planetary gear arrangements in which the planets are not constrained by being secured to a carrier. This lack of constraint was achieved by balancing moments and forces in the various planes through use of conventional compound planetary gearing.

Initial work was done with the Curtiss-Wright Power Hinge,<sup>R</sup> which has been used satisfactorily for fixed-wing aircraft flap actuation systems. Figure 1 illustrates the power hinge concept.

In 1970, Sikorsky Aircraft Division conducted an engineering design study to evaluate advances in VTOL aircraft drive train technology (Reference 1). Included in this evaluation was a free planet transmission for a 4,000-HP helicopter drive train. The configuration investigated offered potential advantages over a conventional planet by:

- (1) eliminating planet bearing power losses and failures,
- (2) having low planetary weight,
- (3) permitting high reduction in two compound stages of high efficiency,
- (4) providing sufficient flexibility and self-centering to give good load distribution between planet pinions,
- (5) effectively isolating planetary elements from deflections of housing, and
- (6) increasing operating time after loss of lubricant, since there were no planet bearings.

In 1972 and 1973, Curtiss-Wright designed, fabricated, and tested two 500-HP, 20-to-1 reduction ratio single-stage free planet transmissions (Reference 2). Testing was accomplished through a regenerative arrangement and indicated high mechanical efficiency, good load distribution, and potential advantages in weight reduction, reliability, survivability, and cost.

In January 1975, Sikorsky Aircraft and Boeing Vertol received contract modifications to a program for Advanced Helicopter Structural Design Investigation, to evaluate free planet transmission preliminary designs for helicopter application. This section of the report presents the results of the Sikorsky evaluation.



#### PRINCIPLE OF OPERATION OF FREE PLANET DRIVE

The free planet transmission concept covers a variety of planetary gear configurations employing no planet carriers or conventional planet mounting bearings. The free planet transmission is so configured that the forces on three orthogonal planes through the planet pinion shaft are in equilibrium.

The concept is best illustrated by first studying a simple conventional compound planetary drive (Figure 2). This design requires bearings to react loads in the tangential and radial planes. The forces in the transverse plane are already in equilibrium as a result of the reduction ratio. The bearing load in the tangential plane is approximately 5 to 6 times that in the radial plane.

The free planet design evolved from recognition of the advantages afforded through elimination of the bearings, which react loads in the tangential plane. The bearings could be eliminated by separating the gears in an axial direction.

Figure 3 is a schematic of a free planet drive that requires no conventional rolling element. The planet gear faces are spaced axially to enable the gear tooth forces to keep the planet spindles in equilibrium. The gears must be so spaced that they lie along the balance line. The gear tooth separating and centrifugal forces are reacted by and balanced out by cylindrical rings concentric with the sun gear axis. The planet spindles have diameters that roll freely on the cylindrical rings. The planet spindles are free in the sense that they are constrained only by the gear meshes and the free-floating cylindrical support rings.

One can verify that all forces and moments add up to zero about any point in or parallel to the three planes.

The free planet drive concept, as it has evolved, can be summarized as follows:

- (1) The reduction ratio requirement and maximum diameter define the forces and the geometry in the transverse plane.
- (2) Free-floating rings react the loads in the radial plane.
- (3) Skewing moments in the tangential plane are eliminated by spacing the gears axially.





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TANGENTIAL PLANE









### PREVIOUS TEST EXPERIENCE

Demonstration free planet test hardware designated FP500 and FP501 was designed, fabricated, and tested by Curtiss-Wright. The FP501 unit is shown in Figure 4. The unit is a compound planetary gear assembly consisting of three gear meshes. The first plane of the planet gear meshes with the sun gear. The second plane meshes with the output internal ring gear. The third plane meshes with the stationary internal ring gear. The first and second planes of the planet gears are splined, piloted, and locked to a quill shaft by a nut and cup lock. The third plane is splined, double piloted, and locked to the second plane of the planet gear by a nut and cup lock. The gears are timed so that the second and third planes are aligned. The FP501 unit is the same as the FP500 except that it has no quill, and torque is transmitted through the hollow support shaft.

### Static Test Results

Static tests indicated good load distribution between planet spindles and gear tooth load patterns. The stiffer FP501 unit resulted in a wider spread of load distribution at lower loads. Gear meshing patterns indicated full face contact, which led to the conclusion that there was no end loading, thus verifying the predicted self-alignment under static conditions.

#### Dynamic Test Results

A 50-hour endurance test was run in a back-to-back, or regenerative, test facility at a rated speed of 8,000 rpm and power of 500 HP. During routine inspection after 26.75 hours, fretting and wear of the gear pilots were observed. Corrective action was taken to permit completion of the 50 hours of testing. Splines, gear, and shaft pilot were cleaned and plated to give a tighter fit. Final teardown after 50 hours of testing indicated that the free planet components were in excellent condition and there was no further deterioration of the splines and shaft pilots (Reference 2).

The FP501 unit (without quill shaft and with pinion torque transmitted through the hollow support shaft) was subjected to 9-3/4 hours of testing in a regenerative facility. With the exception of the sun gear, all gears exhibited a tooth pattern that indicated full face width engagement. The sun-to-pinion mesh appeared to be end loaded. This may have been the result of a helix error ground into the sun gear, bending of the pinion shaft, or tilting of the pinion shaft in a tangential plane. The reason for the end loading can only be assessed if sun gear and pinions are inspected in detail and load sharing and vibration levels for the pinions are determined. Because load sharing was not measured dynamically in any of the testing, no statement can be made now of the presence of this phenomenon.

### End-Loaded Gear Problem

One plausible explanation of the end-loaded sun pinion mesh may be lack of machining tolerance tight enough to prevent tilting of the pinion shaft in the tangential plane.



Figure 4. Curtiss Wright FP501 Test Unit

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Figure 5 shows the magnified effect of loose indexing tolerance. The indexing tolerances of the FP501 free planet test unit (pinion-to-pinion indexing tolerance of the reaction and output ring gear meshes) were ±.003 inch.

### Sikorsky Test Results

Another Eustis Directorate program, Roller Gear Drive Development Testing, had a similar indexing problem. Initial testing with roller gear drive was accomplished with parts held to  $\pm .0002$  inch. The final tolerance was  $\pm .001$ inch. Load sharing was measured dynamically, and the conclusion was drawn that an indexing tolerance greater than  $\pm .003$  inch is excessive. On the basis of the conclusions drawn by Curtiss-Wright in Design and Development Testing of Free Planet Transmission Concept, one can agree that the free planet transmission is a promising concept, but more testing is needed with respect to the statement that "the force balance principle . . . appears to be sound . . . dynamically." (Reference 2).

## Frosted Zone Problem

Another potential problem uncovered during examination of the Curtiss-Wright FP501 was a waviness, or frosted zone, on the pinion bearing journal diameters. They are shown in Figure 6.

The frosted zone may be the result of (1) vibration of the pinion in a tangential plane due to poor indexing tolerances, (2) lack of roundness in grinding bearing journal diameters during manufacture, or (3) skidding during various drive conditions of the rings on the pinion bearing journal diameters. Frosted zones can be shown to be a minor problem. Whatever the initial cause, they do not necessarily cause stress concentrations large enough to make the surface distress self-propagating. In fact, a smoothing over and plastic spreading occur on the higher unpeeled surface. Figure 7 is an excellent example from the Roller Gear Drive R&M Test. Pinion S/N33 was used in a bench test and completed 200 hours of testing. Examination revealed a frosted zone on the lower roller. This pinion was then used in the R&M test. Examination at the end of 22.5 hours of R&M testing revealed that the frosted zones and peeling evident after the 200-hour test had disappeared. This self-healing phenomenon was noted earlier by Franklin Institute Research Laboratories in"Derivation of a Fatigue Life Model for Gears,"USAAMRDL TR-72-14, which observed that shallow spalling of rolling contacting elements did not propagate deeper. The report concluded, "...a form of compliance may be responsible for the fact that cracks at the bottom of the shallow spall did not propagate under the Hertzian stresses or from lubricant-induced hydraulic pressure propagation."











Figure 7. Roller Gear Drive First-Row Pinions

## TYPES AND ARRANGEMENTS OF FREE PLANET DRIVES

The free planet drive can take many forms and arrangements. Each type has its own advantages and disadvantages. Schematics of some of the various types of free planets are shown in Figures 8 through 10. The possibilities of designing free planets have no theoretical limitation, but only two possible configurations appear practical. One is the power hinge used for flap actuation, and the other is the three-gear pinion free planet design, which is the subject of this study.

#### Power Hinge

Initial work in the design of free planet transmissions was performed with the power hinge, which is similar to the free planet transmission design selected for the present study. To achieve equilibrium, forces and moments must add up to zero about any point in or parallel with three planes. This is illustrated in Figure 1 for the forces acting on the power hinge. In a radial plane, the radial separating forces resulting from action of the gear teeth are reacted by free-floating rings. A force balance in the radial plane indicates

$$F_{e_{rad}} + F_{c_{rad}} - F_{s_{rad}} = F_{sup}$$

The use of two output ring gears in symmetry prevents the planets from skewing and, as shown, the summation of moments about any point in the radial plane is zero.

In the transverse plane of Figure 1, the sum of the forces is zero, since

$$F_{c} + F_{c} - F_{c} = 0$$

The moment in the transverse plane to produce the desired reduction ratio is also zero, since

$$F_{c} s + F_{c} x - F_{c} (xz + x) = 0$$

It is also clear that forces and moments acting in parallel with the tangential plane are also balanced by this symmetrical arrangement.

#### Other Types

Other types of free planet drives are reported in References 2 and 3.

Figures 8 through 10 show some of the possible free planet drive concepts. All would work, but they appear to be unnecessarily complicated. The unsymmetrical free planet design shown in Figure 8 was advanced in the Advanced Technology VTOL Drive Train Configuration Study. It is a possible solution, but no hardware has been built to demonstrate this concept.







Figure 9. Free Planet Schematics - Four Load



TANGENTIAL FORCE PLANE





## Three Gear Pinion Free Planet

The free planet drive shown in Figure 11, which is the subject of this report, consists of a sun gear input, free planet pinion with three spur gears per shaft, output ring gear on central free planet pinion, and fixed ring gear on outer free planet pinion. This type of arrangement for a helicopter drive offers a reduction ratio range from 5-to-1 up to 30-to-1. The force balance criterion is much the same as in the power hinge example discussed previously.







Figure 11. Three Gear-Free Planet Schematics

#### PRELIMINARY DESIGN

#### Design Requirements

The baseline aircraft is a medium-sized utility transport (MUT), the same as that used in the Advanced Helicopter Structural Design Investigation Study. This baseline aircraft design uses UTTAS technology and provides internal volume for crew, litters, passengers, cargo, estimated fuel, and equipment. Table 1 lists MUT baseline aircraft data. Figure 12 shows the aircraft general arrangement. Table 2 lists the speed and maximum horsepower design requirements for the free planet transmission.

The transmission gears and shafts are designed for infinite life. Bearings are designed for 3,000 hours B.10 life minimum at the power and speeds listed in Table 34. Accessory drives are located in the rear cover of the main transmission.

#### Envelope, Rotation, and Ratio Restrictions

The paper engines for the baseline MUT had an output speed of 30,000 rpm and develop 925 HP per engine. Since the MUT was designed for a main rotor speed of 340 rpm, an overall reduction ratio of 98.6:1 was required.

With the location of the engines and overall reduction ratio established, the number of reduction stages needed to deliver power from the engines to the main rotor shaft was examined. The fewer the reduction stages, the lighter the weight. Three configurations were examined that would deliver power with a minimum number of reduction stages.

#### Design Envelope Limits

At the start of preliminary design, limits were established for the transmission envelope. Since the main rotor shaft must pass through the center of the sun gear, the minimum possible diameter was set at 6.0 inches. The maximum ring gear diameter was established at 31.0 inches because of the size limitation of the quench press used during case hardening of the gear teeth. The free planet design parameters established are listed in Table 3.

In general, the lightest transmission will result when the highest possible reduction ratio is located in the final reduction stage. Therefore, the earlier reduction stages should have reduction ratios as low as possible.







Figure 12. MUT Aircraft General Arrangement



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	TABLE 1.	MUT BASELINE DATA S	SHEET		
DESIGN ATTRIBUTES					
GENERAL		MAIN ROTOR		TAIL ROTOR/FAN	
DESIGN G W (LB)	0,171.0	RADIUS (FT)	20.50	RADIUS (FT)	01.4
PAYLOAD (LB)	960.0	CHORD (FT)	1.322	CHORD (FT)	.535
WEIGHT EMPTY (LB)	6618.0	NO. OF BLADES	4.0	NO. OF BLADES	4.0
FUEL (LB)	1389.0	ROTOR SOLIDITY	.0819	ROTOR SLDTY/AF	.1547
HOVER POWER (SHP)	0.8711	TIP SPEED (FPS)	730.0	TIP SPEED (FPS)	0.007
HOVER + CLIMB HP	1261.0	ASPECT RATIO	15.511	ASPECT RATIO	8.231
MAIN ROTOR DESIGN HP	1048.0	CT/SIGMA	.0850	CT/SIGMA	.1089
TAIL ROTOR CANT (DEG)	20.00	MAIN ROTOR LIFT	9239. <sup>4</sup>	TAIL ROTOR LIFT	231.9
M R DISC LOADING (PSF)	7.00	FIGURE OF MERIT	.7555	FIGURE OF MERIT	.7147
MAIN G B DESIGN HP	1564.0	BLADE AREA (SQ FT)	108.4	BLADE AREA (SQ FT)	9.4

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Location	Speed (rpm)	Power (hp max)
Input Drives		
Dual Engine Single Engine	30,000 30,000	1,850 925
Main Rotor Tail Takeoff Total Tail Rotor Takeoff	340	1,454 170 120
Accessory Drives		
Generator (Two) Tachometer Generator Servo Hydraulic Pump Aux Servo Hydraulic Pump Utility Hydraulic Pump Lubrication Pump	8,100 3,900 4,200 4,000 4,200 6,000	30 1 4 4 7 1

TABLE 2. FREE PLANET TRANSMISSION DESIGN REQUIREMENTS

TABLE 3. FREE PLANET TRANSMISSION DESIGN PARAMETERS Minimum Sun Gear Diameter 6.00 inches Maximum Ring Gear Diameter 32.0 inches Output RPM 340 Input RPM 12,000, - 15,000 Reduction Ratio 10 to 20:1 Gear Allowable Compressive Stress\* 130,000 psi Gear Allowable Bending Stress (One Way) 55,000 psi Minimum Bearing Life (B.10 Life) 3,000 hours Roller Allowable Compressive Stress 150,000 psi \* Using AGMA calculation method

### TRANSMISSION LAYOUT ARRANGEMENTS FOR MEDIUM UTILITY TRANSPORT

### Dual High-Speed Bevel Gear Inputs

The first arrangement, shown in Figure 13, employs a freewheel unit driven by the engine, whose output drives a bevel gear set. The output bevel gear of this set, which is concentric with the main rotor shaft, is the combining gear for both engines. It drives the final reduction stage and the tail takeoff. The final stage is a free planet reduction unit with output to the main rotor shaft.

The use of two engines on the MUT in a vee arrangement creates the problem of large overall width of the aircraft, which impacts on air transportability. The excessive width problem will be further aggravated in the future by the addition of IR suppressors.

If the aircraft required a single-engine arrangement, this bevel gear input drive would offer the lightest weight, smallest number of parts, and full use of the high ratio capability of the free planet drive unit. A schematic of such an arrangement is shown as Figure 14.







Figure 14. Single High-Speed Bevel Gear Input

#### Dual High-Speed Crossed Helical Gear Inputs

The second configuration considered is shown in Figure 15. The first stage of this system is a crossed helical mesh, which permits parallel engine mounting and a wide center distance between engines. The driven helical gear transmits power through a freewheel unit to the second-stage combining spiral bevel mesh. The driven gear of the spiral bevel mesh is concentric with the main rotor shaft and drives both the tail takeoff and free planet reduction unit. This configuration was rejected, because crossed helical gears are inefficient for high-power, high-torque applications.







### Dual High-Speed Spur Gear Inputs

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The third possibility is shown in Figure 16. In this configuration, the engine drives through a freewheel unit to the first stage combining spur gear mesh. The output gear of the spur gear mesh drives a single spiral bevel mesh which turns the corner. The output bevel gear is concentric with the main rotor shaft and drives the tail takeoff and free planet reduction unit.

This design was selected, since the bearing problem on a high-speed 30,000rpm spur mesh is much easier to solve than on a 30,000-rpm bevel mesh. Through the use of idler gears, the necessary spacing is obtained between the engines, and pads are provided for an accessory drive.





## FREE PLANET TRANSMISSION CONCEPT FOR UTTAS-SIZE AIRCRAFT

A preliminary design study of the application of the free planet concept to a UTTAS-size twin turbine helicopter indicates that the concept is unsuitable, primarily because the high reduction ratio it offers cannot be fully used without significant compromises in engine location.

To take full advantage of the free planet concept, only one other stage of gearing is needed between the engine and the main rotor. For a twin-engine helicopter using front-drive engines, one possible solution is to locate the engines horizontally aft of the main gearbox in a vee configuration, so their input bevel pinions mesh with a common bevel gear. The fundamental objection to this arrangement is the effect on aircraft balance, since the engines must be located farther forward than on a smaller aircraft, such as MUT or ASH. This situation is expected to be aggravated in the future by the addition of IR suppressors. When additional factors such as engine inlet ducting, easy accessibility, and ballistic survivability are considered, the present UTTAS engine arrangement which requires two bevel gear stages is hard to beat. When the overall UTTAS reduction ratio is spread over three stages of speed reduction to meet the geometry constraints, only a simple planetary is required for the output stage.
#### FREE PLANET UNIT DESIGN

To select a free planet unit to meet the constraints of the MUT aircraft, consideration has to be given to reduction ratio, gear teeth stress, axial length, pinion shaft design, and roller ring design. Evaluation of the interactive effects of these attributes and simplification of the selection process made it necessary to develop a computer program, shown in Appendix A.

#### Reduction Ratio

The reduction ratio for the free planet is determined through use of the equivalent system method. The equivalent system and the free planet configuration are shown in Figure 17. The equivalent system has the same relative pitch-line velocities as the actual system, and all gears rotate



Equivalent System



on fixed centers. This simplifies the task of determining the reduction ratio of the system. A minus sign indicates that the member turns in the opposite direction from the input. In the equivalent system, all gears are on fixed center. The speed of rotation of any shaft can be found in terms of the speed of rotation of any other shaft:

$$-rpm_{c} \frac{R_{3}}{P_{3}} = (rpm_{out} - rpm_{cage}) \left\{ \frac{R_{2}}{P_{2}} \right\} = (rpm_{in} - rpm_{cage}) \left\{ -\frac{S}{P_{1}} \right\}$$

Simplifying and solving for rpm /rpm results in

 $RR = \frac{rpm_{in}}{rpm_{out}} = \begin{cases} 1 + \frac{R_3P_1}{P_3S} \\ \frac{1 - \frac{R_3P_2}{P_3R_2}}{\frac{P_3R_2}{P_3R_2}} \end{cases}$ 

#### Gear Tooth Stress Analysis

The dynamic bending stresses and compressive stresses for the gear teeth of the drive train are calculated and compared with an allowable stress.

#### Bending Stress Equation

The basic equation for determining the bending stress at the root of a tooth in a spur or bevel gear is as follows:

$$f_{b} = \frac{W_{t}K_{o}}{K_{v}} \cdot \frac{P_{d}}{F} \cdot \frac{K_{s}K_{m}}{J}$$

where

- $W_{+}$  = tangential tooth load
- $K_{a}$  = overload factor
- $K_{\perp}$  = dynamic factor
- $P_d$  = diametral pitch
- F = face width
- $K_{c}$  = size factor
- $K_m = load$  distribution factor
- J = geometry factor

All the free planet transmission drive gears are case carburized and ground to close tolerances to minimize dynamic effects. The dynamic factor as a result is, therefore, taken as 1.0.

The overload factor makes allowances for the roughness or smoothness of operation of the driving and driven members. Again, this factor is taken as 1.0.

The load distribution factor accounts for the combined effects of deflection of mountings and misalignment of gears. For bevel gears, the load distribution factor is less critical and is taken as 1.10. For the free planet, which is considered to be less critical than a planetary drive, the load distribution factor is taken as 1.10. The load distribution factor for conventional spur or helical gears is taken to be 1.30.

The size factor reflects nonuniformity of material properties and is taken as 1.0 for aircraft spur gears. For bevel gears, the size factor is a function of the diametral pitch. The geometry factor evaluates the shape of the tooth, the position at which the most damaging load is applied, stress concentration due to geometric shape, and load sharing. In bevel gears the geometry factor is taken for the mean normal section of the tooth.

## Compressive Stress Equation

The contact stress for steel spur gears can be calculated by

$$f_{c} = \frac{(21)(10^{6}) W_{t}}{(\sin 2)(F)} \left\{ \frac{1}{d_{p}} + \frac{1}{d_{s}} \right\} + \text{for external} - \text{for internal}$$

For bevel gears, the contact stress is given by

$$f_{c} = K_{p} \sqrt{\frac{2T_{p} K_{o}}{K_{v}} \cdot \frac{1}{F d_{p}^{2}} \cdot \frac{K_{s} K_{m} K_{f}}{I}}$$

where

 $K_{p} = elastic coefficient$   $K_{o} = overload factor$   $K_{v} = dynamic factor$   $T_{p} = pinion torque$   $d_{p} = pinion pitch diameter$   $K_{s} = size factor$   $K_{m} = load distribution factor$   $K_{f} = surface condition factor$  I = geometry factor

## Allowable Stresses

Table 4 gives the allowable stresses for carburized and ground steel.

The difference in allowable stresses for spur and bevel gears is due mainly to the different size factors used.

Components	Allowable Stress
Spur Gears - One-Way Bending	$F_{b} = 55,000$
Bevel Gears - One-Way Bending	$F_{b} = 30,000$
Spur Gears	$F_{c} = 130,000$
Bevel Gears	$F_{c} = 200,000$

# TABLE 4. ALLOWABLE GEAR BENDING AND CONTACT STRESSES

## Axial Length Determination

One necessary condition for the free planet transmission is that the sum of the moments about any point in a radial plane equal zero. This can be accomplished simply during design selection by spacing the pinion shaft gears so that they lie along the balance line shown in Figure 18.



Figure 18. Axial Length Pinion Shaft Schematic

The geometry is shown in Figure 19.



Figure 19. Axial Length Pinion Shaft Geometry



# Pinion Shaft Design

The pinion shaft of the free planet transmission is designed to carry both torque and bending. The torque in the pinion shaft is the result of the differences in torque of the reaction ring gear, output ring gear, and sun gear torques. The resulting loads on the pinion shaft in both the tangential and transverse directions are shown in Figure 20.





The torque between points 1 and 2 is given by

$$T_{1-2} = W_{s} \left\{ \frac{DP_{1}}{2} \right\}$$
$$T_{2-3} = W_{c} \left\{ \frac{DP_{3}}{2} \right\}$$

From a torque balance in the transverse plane, it can also be shown that

$$W_{c} \left\{ \frac{DP_{3}}{2} \right\} = W_{a} \left\{ \frac{DP_{2}}{2} \right\} + W_{s} \left\{ \frac{DP_{1}}{2} \right\}$$

In the tangential plane, the bending moment on the pinion between points 1 and 2 is equal to

$$M_{1-2} = W_s x$$

The maximum moment on the pinion shaft between points 1 and 2 is equal to

$$M_{1-2} = W_{s} \left( a + b - \frac{FR2P2}{2} \right)$$
MAX

To design for a fatigue design condition, the torque is considered steady, and the moment is vibratory and subject to complete reversal.

$$f_{\underline{vib}} = \frac{M_{1}Z}{Z}$$
$$f_{s} = \frac{T_{1-2}}{Z_{p}} = \frac{T_{1-2}}{ZZ}$$

As shown in Reference 4, the margin of safety using the maximum shearing stress theory of failure is equal to

$$MS = \frac{1}{\sqrt{\frac{K_{t}f_{v}}{Fen}^{2} + 4\left(\frac{f_{s}}{F_{ty}}\right)^{2}}} - 1 = 0$$

$$f_{\frac{y=2}{1-2}} = \frac{\frac{W_{s}(a+b-\frac{FR2P2}{2})}{Z}}{Z}$$

$$f_{s} = \frac{W_{s} DP_{1}}{(2)(2)(2)}$$

For an open section,  $K_t = 1.0$ .

Since the pinion shaft is made of 9310 steel AMS 6265 with a core hardness of Rockwell C 30-45, the material properties are:

 $F_{tu} = 136,000$   $F_{ty} = 115,000$   $F_{en} = 44,500$  SEF = .72Pf = .70

$$\left\{\frac{K_{t}fv}{F_{en}}\right\}^{2} + \left\{\frac{f_{s}}{F_{ty}}\right\}^{2} = 1$$

$$\left\{\frac{W_{s}\left(a+b-\frac{FR2P2}{2}\right)}{22400 Z}\right\}^{2} + \left\{\frac{W_{s}DP_{1}}{(2)115000}\right\}^{2} = 1$$

$$Z = \left\{\frac{W_{s}\left(a+b-\frac{FR2P2}{2}\right)}{22400}\right\}^{2} + \left\{\frac{W_{s}DP_{1}}{230,000}\right\}^{2}$$

where

1

FR2P2 = face width of output ring pinion mesh

$$T = \frac{FR3P3}{2} + .10 + \frac{FR2P2}{2}$$
$$a = \frac{DP2(T)}{DP3 - DP2}$$
$$b = \frac{DP1}{DP3} (a + T)$$
$$v_{s} = \frac{T_{s} (2)}{(DS1)(Nopin)}$$

The flexibility of the pinion shaft or the wind-up is determined from

$$\boldsymbol{\varrho} = \frac{\mathrm{T}}{\mathrm{Z}_{\mathrm{p}}\mathrm{Gr}_{\mathrm{o}}} = \frac{\mathrm{T}}{\mathrm{2}\mathrm{Z}\mathrm{Gr}_{\mathrm{o}}}$$

The size of the pinion can be determined for the torsional and bending loads. In addition, the torsional wind-up can be determined.

## Roller Ring Loads

Equilibrium was considered only in the tangential and transverse planes. To establish equilibrium in the radial plane, roller rings are required to react the centrifugal loads and the gear-separating loads. Figure 21 is a free body representation of the pinion shaft in the radial plane.



Figure 21. Roller Ring Loading

Summing the moments about point  $R_2$ ,

$$W_{c}(\tan \theta_{3})(K) + W_{B}(\tan \theta_{2})(K + T) - F_{iner}P + R_{1}(K + T + a + b - m)$$
  
-  $W_{s} \tan \theta_{1}(K + T + a + b) = 0$ 

$$F_{\text{inertia}} = \text{mr}^{2}$$
$$= \left(\frac{\text{Weight}}{32.2}\right) \left(\frac{\text{radius}}{12}\right) \left\{ \left(\frac{2}{60}\right) \text{ RPM} \right\}^{-2}$$

Solving for R<sub>1</sub>,

$$R_{1} = \frac{1}{K + T + a + b - m} \left\{ W_{s} \tan \theta_{1} (K + T + a + b) + F_{iner} P - W_{c} K \tan \theta_{3} - W_{b} (K + T) \tan \theta_{2} \right\}$$

Similarly, summing the moments about point  ${\rm R}_1$  and solving for  ${\rm R}_2$ 

$$R_{2} = \frac{1}{K + T + a + b - m} \left\{ W_{c} \tan Q_{3}(T + a + b - m) + W_{B} \tan Q_{2}(a + b - m) + W_{s} \tan Q_{1}(a + b - m) + W_{s} \tan$$

For the case of a start-up condition or operation at close to zero speed,  $R_2$  may be negative, requiring the addition of a roller on the inside roller diameters.

#### Free Planet Design Selection Model

A computer model was developed to aid in selection of a free planet unit for the baseline MUT. Appendix A is a listing of the actual program used. Figure 22 is a simplified flow chart of the program logic.

With the criteria already established, few acceptable designs were anticipated, but thousands were found suitable. Development of other criteria narrowed the selection. Table 5 is a list of the criteria used.

Figures 23 and 24 summarize some of the computer-generated results. Figure 23 is a map of possible free planet designs plotted on the basis of reduction ratio and overall unit height. Higher ratios lead to larger overall height, but this height increase can be limited by operating on what can be called the efficient frontier. Figure 24 illustrates the effect of changing the sizes of the sun gear and ring gears. The lowest weight is achieved with the lowest reduction ratio and lowest height. For a given reaction ring gear size and required reduction ratio, the smaller the sun gear, the lower the overall unit weight and height. Contrary to what might be expected for a given sun gear and reduction ratio, decreasing the reaction ring diameter increases gearbox height and weight.

Therefore, the flatter the free planet package, the lighter the overall weight. The decrease in weight continues until the size and weight of the roller ring diameter increase faster than the weight of the gearing decreases. The effect of increased roller ring weights was not significant in the free planet designs investigated during this study.













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	LIMI	TS
ITEM	LOW	HIGH
iametral Pitch#	5, <b>#</b> 9 <b>#</b> 1	1 12
umber of Pinions	5	12
umber of Teeth Sun Gear Sl	18	200
lameter Sun Gear		► 6.0
umber of Teeth Pinion Pl	18	
ameter Pinion Pl		
umber of Teeth Pinion P2	18	
ameter Pinion P2	L Dro &	Dp1 > Dpo
imber of Teeth Ring Geer Autnut P2	60	200
amoer or reconstruct output R2	00	200
umbow of Mooth Divis D2	٦Q	< <b>[ •</b> ∪
ander of Teeth Pinion P3	10	
Lameter Finion F3	( )	000
umber of Teeth Ring Gear Reaction R3	60	200
lameter Ring Gear Reaction R3		27.0
eduction Ratio	+10.0	+40.0
ace Width of P2	.6	DP2
cial Length	< 25.0 and	3 D <sub>R3</sub>
$\frac{N_{S1}}{N_{P1}} = Whole Number + Irreduce$	cible Fraction	
$\frac{N_{R2}}{N_{P2}} = Whole Number + Irreduce$	cible Fraction	
$\frac{N_{R3}}{N_{P3}} = Whole Number + Irreduce$	cible Fraction	
ual Pinion Spacing Criteria for Each	h Mesh	
$\frac{N_{S1} + N_{R1}}{No \text{ of Pinion}} = Whole Numbers$		



Figure 23. Map of Range of Free Planet Solutions



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## Conventional Two-Stage Planetary Transmission

A conventional two-stage planetary transmission was designed in order to compare a free planet transmission with conventional drive train technology. The conventional two-stage planetary drive consists of an input spur mesh bevel gear mesh, and two planetary reduction stages, one of which drives the main rotor. Figure 25 is a preliminary layout of the conventional planetary transmission. Power is transmitted to an input spur gear from each engine, which in turn drives a spur combining gear. This spur combining gear is located on the bevel pinion shaft and drives the bevel output gear, which is concentric with the main rotor shaft. The output of the bevel gear drives the sun gear of the first-stage planetary gear set. The planetary reduction stage has an input sun gear, stationary (bolted to housing) ring gear, and output cage. The output cage of the firststage planetary drives the sun gear of the second-stage planetary. The cage output of the second-stage planetary is splined to the main rotor shaft and transmits power to the main rotor. The main rotor shaft is supported at the top of the main gearbox by a cylindrical roller bearing and at the bottom by a tandem\_mounted, split in...er race ball bearing set. Rotor loads are reacted through the main rotor shaft bearings to a cast magnesium housing bolted to the airframe. Figure 26 is a schematic of the drive train, showing the rotational speeds. The weight, which reflects a parametric weight for overrunning clutch and lubrication system, of the conventional planetary, is 737 pounds.









#### Free Planet Transmission

The free planet transmission consists of an input spur mesh from each engine, a bevel gear mesh, and the free planet reduction unit that drives the main rotor. Figure 27 is a schematic of the drive train, and Figure 28 is a preliminary layout of the free planet transmission.



Figure 27. Preliminary Free Planet Drive Train Schematic

Power is transmitted from each of the two aft-mounted engines through an input spur gear to a spur combining gear. The spur combining gear is located on the bevel pinion shaft and drives the bevel output gear, which is concentric with the main rotor shaft. The spline output of the bevel gear transmits power to the sun gear of the free planet reduction unit. The free planet consists of an input sun gear, five planet pinion assemblies with three pinion gears for each shaft, three roller rings, and two ring gears. One ring gear provides the reaction torque. The other ring gear serves as the output member and transmits power to the main rotor. Table 6 supplies the free planet unit geometry.

The main rotor shaft, which is driven by the output ring gear, is supported at the top of the gearbox by a cylindrical roller bearing and at the bottom by a tandem-mounted, split inner race ball bearing set. Rotor loads are reacted through the main rotor shaft bearings to a cast magnesium gearbox housing. The weight for this preliminary free planet transmission includes a parametric weight for the overrunning clutch and lubrication system and is 705 pounds.



	table 6	. FREE PI GEOMETF	ANET UNIT GEA	R TEETH Y SELECTION)	
ITEM	NUMBER OF TEETH	PITCH DIAMETER	DIAMETRAL PITCH	NUMBER OF PINIONS	REDUCTION RATIO
NS NP1	50 68	5.0 6.8	10.0		
NP2 NR2	27 110	3.84 15.64	7.034	5	16 <b>.13</b> 7
NP3 NR3	58 165	6.40 18.20	9.068		

The weight of the free planet transmission is within 5% of the weight of a conventional planetary transmission. The computer model was then used with parametric curves to design a lower ratio free planet transmission with larger diameter ring gears.

## FINAL FREE PLANET DESIGN

The final free planet transmission design is shown in Figure 29. The transmission is lower in height. It has a lower reduction ratio, larger sun gear diameter, and larger ring gear diameters. Figure 30 is a schematic showing the drive train speeds of the final design.

Table 7 supplies the gear geometry.

The gear tooth loads for an output speed of 340 rpm with 1,454 HP to the main rotor are

$${}^{T}_{out} = \frac{(63025)(1454)}{340} = 269520$$
$${}^{T}_{in} = \frac{269520}{10.851} = 24800$$

The tangential tooth load on the sun pinion mesh is

$$W_{ts} = \frac{2 \text{ Tin}}{d_s M}$$
  
=  $\frac{(2)(24800)}{(7.0)(5)}$ 

1

The tangential tooth load on the output ring gear mesh is

$${}^{W}t_{R_{2}} = \frac{2 \text{ Tout}}{d_{R2} M}$$
  
=  $\frac{(2)(269520)}{(19.0)(5.0)}$   
 ${}^{W}t_{R_{2}} = 5674$ 

The reaction ring gear torque is found from

$${}^{W}t_{R_{3}} = {}^{W}t_{R_{2}} - {}^{W}ts = 4256$$



Figure 29. Layout of Final Free Planet Transmission



Figure 30. Schematic of Final Free Planet Transmission

	TABLE7.GEAR GEOMETRY FREE PLANET TRANSMISSION REDUCTION RATIO10.857						
Member	Number of Teeth	Diametral Pitch	Number of Pinions	Pitch Diameter			
Sun	35	5.0		7.0			
P <sub>1</sub>	39			7.8			
P2	21	5.0	5	4.2			
R <sub>2</sub>	95			19.0			
P_3	41	5.0		8.20			
R <sub>3</sub>	115			23.0			

The speed of the input sun gear is

$$N_{sun} = N_{out} RR$$

$$N_{sun} = (340)(10.857)$$

$$N_{sun} = 3691 RPM$$

The rotational speed of the pinion shaft about its own center is

$${}^{N_{P}}{}_{1}{}^{P}{}_{2}{}^{P}{}_{3} = \frac{1}{1 + \frac{P_{1}R_{3}}{P_{3}S}} N_{sun}$$

$${}^{N_{P}}{}_{1}{}^{P}{}_{2}{}^{P}{}_{3} = \frac{1}{\frac{(39)(115)}{1 + (41)(35)}} (3691)$$

$${}^{N_{P}}{}_{1}{}^{P}{}_{2}{}^{P}{}_{3} = 895 \text{ RFM}$$

"sing these data, the gear tooth stresses and face width were calculated as outlined in the Gear Tooth Stress Analysis section. This information is summarized in Table 8.

	TABLE	8. GEAR FACE WIDT	TH AND GEAR STRESSES	
Memb	per	Face Width	Compressive Stress	Bending Stress
Sun P <sub>1</sub> P <sub>2</sub>	Output	.550 .545 1.50	145,000 145,000	32,800 32,050 49,700
R <sub>2</sub>	Ring Mesh	1.47		40,550
P3 R3	Reaction Ring Gear Mesh	.80 .75	115,000	55,000 55,000

## Weight Analysis

Comparison of the free planet transmission with a conventional two-stage planetary indicates that a 9% weight saving is achieved by the free planet design. The baseline conventional two-stage planetary design weighs 737 pounds, while the free planet transmission weighs 671 pounds. Table 9 contains a component weight breakdown for both transmissions.

The weights of the baseline main gearbox, main rotor shaft, and other drive system components were estimated from statistical weight trending equations based on Sikorsky's drive system design philosophy.

TABLE 9. WEIGHT COMPARISON OF FREE PLANET WITH CONVENTIONAL TWO-STAGE PLANETARY TRANSMISSION					
Component	Free Planet Weight, Lb	Conventional Two Stage Planetary Weight, Lb			
Main Rotor Shaft	62.0	83.0			
Planet Assembly	144.0	159.0			
Gears	41.0	58.0			
Tail Takeoff and Accessories	37.0	37.0			
Bearings	37.0	50.0			
Freewheel Units	10.0	10.0			
Housings and Sump	200.0	195.0			
Lubrication System	98.0	103.0			
Seals, Spacers and Retainers	22.0	22.0			
Supports	11.0	11.0			
Miscellaneous	9.0	9.0			
	671.0	737.0			

## Cost Analysis

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The cost of the free planet transmission is 16.5% less than that of a conventional two-stage planetary transmission in quantities of 500 or more. Table 10 gives the estimated manufacturing cost for transmissions built in quantities of 1, 50, 100, and 500 units. Table 11 is a breakdown of cost by components for a prototype free planet transmission and conventional planetary transmission.

TABLE 10.COST COMPARISON OF FREE PLANET TRANSMISSION WITH TWO_STAGE CONVENTIONAL PLANETARY TRANSMISSION					
Quantity (units)	Free Planet (dollars)	Conventional Planetary (dollars)			
1 50 100 500 Prototype Tooling Production Tooling (50 units or more)	157,500 84,600 76,500 59,700 254,800 645,000	184,200 101,800 91,500 71,600 252,000 631,000			

FREE PLANET (dollars)		CONVENTIONA (dol	AL PLANETARY llars)	
	49,700		49,000	
42,000 4,900 2,800		39,200 5,600 4,200		
	14,000		16,800	
	9,800		11,200	
	4,200		5,600	
	70,000		89,600	
	9,800		11,200	
	157,500		183,400	
	254,800		252,000	
	FREE F (dolla 42,000 4,900 2,800	FREE PLANET (dollars) 49,700 4,900 2,800 14,000 9,800 4,200 70,000 9,800 157,500 254,800	FREE PLANET (dollars)       CONVENTIONA (dol $49,700$ $49,700$ $49,700$ $39,200$ $4,900$ $5,600$ $2,800$ $4,200$ $14,000$ $9,800$ $4,200$ $70,000$ $9,800$ $157,500$ $254,800$ $254,800$	

# TABLE 11. COST BREAKDOWN FOR PROTOTYPE FREE PLANETAND CONVENTIONAL PLANETARY TRANSMISSION

#### Survivability/Vulnerability Analysis

Assessment of survivability characteristics indicates that the free planet transmission offers improved operation after loss of lubrication compared with the conventional transmission. The free planet transmission is more vulnerable to 23mm AP threats than the conventional transmission.

For operation after loss of normal oil supply, the free planet transmission design offers several improvements. The most important of these is elimination of planet gear bearings, which are the major source of failure after loss of lubrication. The gears in the free planet unit are also more tolerant of cil loss. Fewer parts are used, so there are fewer sources of heat. The system is more efficient, so less heat is generated. The transmission density is low, so more cooling air is available.

The higher vulnerability of the free planet to 23mm threats compared with the conventional transmission reflects the vulnerability of roller rings. Protection of these rings would be difficult. To stop all threats would require massive rings which would be very heavy. One way to prevent this problem is to permit the projectile to perforate the ring without stopping the projectile or absorbing the energy.

## Efficiency

The efficiency of the free planet transmission was compared with that of the conventional two-stage planetary drive. Overall transmission efficiency of the free planet transmission was 97.3%. Overall efficiency of the conventional planetary drive was 96.8%. This improvement in efficiency for the free planet transmission translates into 7.5 HP.

The efficiency of the free planet system was calculated by the equivalent system method. The term "equivalent" refers to the fact the tooth mesh losses in the two systems are the same. In the equivalent system approach, an artificial rotation is imposed on the complete planetary gear train to effectively stop the planet carrier. Relative motions of all members of the planetary train are unchanged. However, the planet gears are idlers in the equivalent system, and the entire train can be considered a conventional gear train with fixed axes of rotation. The only change is in the pitch-line velocities of the gears. They are now equal to the velocities of engagement of the gears in the planetary train.

By this approach, efficiency, speed, and power flow relationships can be determined for the free planet system.

The planetary gear train and the equivalent fixed-axes gear train are shown in Figure 31.

The reduction ratio of planetary gear trains is calculated as follows:

$$R = \frac{1 + \frac{N_B}{N_A} \frac{N_E}{N_D}}{1 - \frac{N_C}{N_F} \frac{N_E}{N_D}}$$
$$= \frac{1 + \frac{39}{35} \frac{115}{41}}{1 - \frac{21}{95} \frac{115}{41}} = \frac{76}{7} = 10.8571$$

The cage speed ratio is

$$\frac{\frac{W_{c}}{W_{i}}}{W_{i}} = \frac{1}{1 + \frac{N_{B}}{N_{A}} \frac{N_{E}}{N_{D}}}$$
$$= \frac{1}{1 + \frac{39}{35} \frac{115}{41}} = \frac{287}{1184} = .2424$$



Figure 31. Free Planet Actual and Equivalent Fixed System Schematic

In Figure 31, the external torques acting on the shafts of the equivalent system are the same as those acting on the same shafts in the planetary system. The speeds in the equivalent system have been reduced by the carrier speed of the actual system. The product of the torque and the speed is a measure of the power transmitted. A positive value of this product indicates that the shaft is a driver. A negative value indicates that the shaft is driven. Thus, the equivalent gear train has two driving members and one driven member, and the power flow is from drivers to driven.

To determine system losses, the power developed at the two driving members must first be established. In the actual system, output power is

$$Po = TW, \eta_{\bullet}$$

where  $m_{\bullet}$  is the overall planetary efficiency. In the equivalent system, the output shaft is a driver and its power input is

$$\frac{76}{7} \left\{ \frac{3381}{22496} \right\} \text{TW}_{i} = \frac{483}{296} \text{TW}_{i}$$

The power input of the other driver is

The power losses in the equivalent system can now be calculated:

Losses = 
$$\frac{483}{296}$$
 TW<sub>1</sub> $(1 - \epsilon_{FC} \epsilon_{DE}) + \frac{897}{1184}$  TW<sub>1</sub>  $(1 - \epsilon_{AB} \epsilon_{DE})$ 

where the E's are the appropriate fixed center gear mesh efficiencies. The overall efficiency for the planetary gear then is

$$\eta_{\bullet} = \frac{TW_{i} - Losses}{TW_{i}}$$

where the losses are the same as in the equivalent system.  $\mathcal{T} = \frac{\mathrm{TW}_{i}}{296} - \frac{\frac{483}{296}}{1184} = \frac{\mathrm{TW}_{i}}{\mathrm{TW}_{i}} \left(1 - \varepsilon_{AB} \varepsilon_{DE}\right) - \frac{\frac{897}{1184}}{\mathrm{TW}_{i}} = \frac{\mathrm{TW}_{i}}{\mathrm{TW}_{i}} \left(1 - \varepsilon_{AB} \varepsilon_{DE}\right)$ 

Simplifying and solving for  $m_{s}$ ,

$$\eta_{\bullet} = \frac{1 - \frac{897}{1184} \left(1 - \boldsymbol{\epsilon}_{AB} \boldsymbol{\epsilon}_{DE}\right)}{1 + \frac{483}{296} \left(1 - \boldsymbol{\epsilon}_{FC} \boldsymbol{\epsilon}_{DE}\right)}$$

$$\mathcal{E} = 1 - \left\{ \frac{1 \pm \frac{M_1}{M_2}}{\beta_a + \beta_r} \right\} \left\{ \begin{array}{c} \boldsymbol{\beta} & a^2 + \boldsymbol{\beta} & r^2 \\ \boldsymbol{\beta} & a^2 +$$

where

 $\boldsymbol{\beta}_{a}, \boldsymbol{\beta}_{r} = \text{arc of approach and recess}$ f = average coefficient of friction

For mesh A - B shown in Figure 31:

$$\epsilon_{AB} = 1 - \left\{ \frac{1 + \frac{N_A}{N_B}}{\beta_a + \beta_r} \right\} \left\{ \frac{\beta_a^2}{\beta_a^2} + \frac{\beta_r^2}{\beta_a^2} \right\} \frac{f}{2}$$

$$\beta_a = \frac{\sqrt{\frac{R_{0B}^2}{R_{0B}^2} - \frac{R_{bB}^2}{R_{bB}^2} - \frac{R_B \sin \Phi}{R_B}}{\frac{R_B}{R_{bA}^2}}$$

$$= \sqrt{\frac{8.2^2 - 7.2063^2}{6.4672} - 7.8 \sin 22.5}$$

$$= .14346$$

$$r = \sqrt{\frac{R_{0A}^{2} - R_{bA}^{2} - RA \sin \phi}{R_{bA}}}$$

$$= \sqrt{\frac{7.4^{2} - 6.4672^{2} - 7.0 \sin 22.5}{6.4672}}$$

- .14191

ß

$$V = \frac{D_A \pi N_A}{12}$$

$$= \frac{T \pi}{12} \left(\frac{897}{1184} \frac{76}{7}\right) 340 = 5125.1 \text{ fpn}$$

$$V_S = V \cos \varphi \left\{1 + \frac{N_A}{N_B}\right\} \frac{\beta_A + \beta_T}{4}$$

$$= (5125.1) \cos 22.5 \left(1 + \frac{35}{39}\right) .07134$$

$$= 641 \text{ fpm}$$

$$f = \frac{2}{3} \left\{\frac{.050}{641} + .002 \sqrt{641}\right\}$$

$$= .03376$$

$$\mathcal{E}_{AB} = 1 - \left\{\frac{1 + \frac{35}{39}}{.28537}\right\} (.04072) \left(\frac{.03376}{2}\right)$$

.9954

=

	TABLE 12.	EFFICIENCY DATA	FOR FREE	PLANET	TRANSMISSION		
MESH	B <sub>a</sub>	<sup>B</sup> R	fpm	's	f	E	
A - B C - F D - E	.14346 .04918 .14593	.14191 .06384 .12306	5121 2759 5388	641 254 215	.03376 .02124 .01957	.9954 .9978 .9991	
	•14995	•12300		~	.019)1	•9991	
The overall free planet efficiency is

$$= \frac{1 - \frac{897}{1184} (1 - (.9954)(.9991))}{1 + \frac{483}{296} (1 - (.0078)(.9991))}$$

= .9908

The analytics: technique for calculating efficiency was compared with test data from the Curtiss-Wright 500 HP FP501 test unit. The analytical and test data agreed within .1%, which is well within the error of experimental measurement. The 19.2425:1 reduction ratio of the FP501 test unit had a measured efficiency of 98.8% of full speed and rated torque.

The calculated overall free planet efficiency for the final design configuration is 99.08%. The conventional two-stage planetary unit has an efficiency of 98.5%.

Total transmission losses were estimated from a knowledge of the type of gear mesh, design horsepower, and power transmitted by each gear mesh. Experience has demonstrated that tooth mesh and bearing losses can be estimated conservatively as 1/2% per mesh. A further 3/4% of total power transmitted must then be added to account for churning losses in the entire transmission.

The losses in the free planet and conventional planetary are presented in Table 13.

TABLE 13. LOS OF PLA	S, SOURCES AND FREE PLANET ANI NETARY DRIVES	EFFICIENCY CONVENTIO	( DNAL TWO-STAGE	
	FREE PLAI	VET	TWO-STAGE PLA	NETARY
	LOSS PERCENT	LOSS HP	LOSS PERCENT	LOSS HP
MAIN ROTOR DRIVE-1450 HP				
EPICYCLIC BEVEL SPUR CHURNING	•92 •50 •50 •75	13.34 7.25 7.25 10.87	1.5 .50 .50 .75	21.75 7.25 7.25 10.87
TAIL ROTOR DRIVE-113 HP				
BEVEL SPUR CHURNING	• 50 • 50 • 75	•55 •55 •83	• 50 • 50 • 75	• 55 • 55 • 83
ACCESSORY DRIVE-30 HP				
BEVEL SPUR CHURNING	.50 1.00 .75	.15 .60 .23	.50 1.00 .75	.15 .60 .23
TOTAL LOSSES		41.62		50.03
EFFICIENCY		97.3		96.8

## Reliability Analysis

The reliability of the free planet transmission was compared with that of a conventional two-stage planetary transmission. The analysis indicated a 2-to-1 improvement in reliability of the free planet over the conventional two-stage planetary.

## Background

The reliability data used covered Sikorsky main gearboxes with a cumulative total of over 300,000 flight hours. The design criteria for gear tooth stresses and design bearing lives were taken as identical in the free planet and the conventional planetary. Failure rates for the component elements (sun gears, planet pinions, ring gears, bearings) were taken as identical for the two systems.

Both the free planet and conventional two-stage planetary were considered to have the same failure modes as those experienced by production planetary designs. The measure of reliability is the removal rate caused by a validated failure before a gearbox reaches its scheduled removal time for overhaul, so the removal rate does not include scheduled removals.

The failure modes experienced in the sample of operational main transmissions include gear tooth fracture, spalling, scoring, and wear, in addition to planet pinion bearing failures. Of these failure modes, only gear tooth fracture and planet pinion bearing failures result in gearbox removals. This permits the use of gear tooth bending stress and bearing failures in the reliability analysis.

MEMBER FA	
31)	ILURE RATE ilures/cycle)
Sun Gear Planet Pinion 1 Ring Gear Pinion Bearings	$31.5 \times 10^{-12}$ 77.5 x 10 <sup>-12</sup> Negligible

The failure rates based on flight test data are shown in Table 14.

These failure rates are expressed in failures per cycle to take into account the number of mesh points of the gears as well as the frequency of loading.

## <u>Analysis</u>

The gear tooth stress levels and resulting failure rates are not precisely those of the free planet transmission or conventional planetary transmission. The rates have been adjusted to provide a valid comparison. This adjustment reflects the assumption that the ratio of actual failure rate to probability of failure, based on working stress level, is constant for each type of gear. Therefore, a constant can be defined for sun gears, planet pinions, and ring gears. In establishing this constant from past operational designs, the fatigue bending endurance strength of the gear teeth was taken as the endurance strength of the core material for a part with a ground surface.

The endurance limit was determined as follows:

Material Core Hardness		9310 CEVM Steel R <sub>c</sub> 30-45
Endurance Limit $E_{m}$ (Ground surface)		55,000 psi
Size Effect Factor (SEF)		0.8
Mean Endurance Limit	=	(SEF)(E <sub>m</sub> )
	=	(.8)(55,500)

This mean endurance limit of 44,000 psi is used in the design of gearboxes and represents a 50% probability of failure. Extensive test experience has established that the standard deviation of fatigue data for steel is 10% of the mean strength for components made of the same and of similar steel alloys. The probability of failure for any other stress level can now be determined easily from standard tables of the Gaussian distribution. These tables give probabilities as a function of the number of standard deviations.

The quantity a fis defined as

a = (<u>Mean Endurance Limit)-(Working Stress</u>) (Mean Endurance Limit) (Standard Deviation)

Figure 32 represents the basic concepts of the analytical approach.

As an example of this analysis for the free planet transmission, consider the sun-pinion mesh. The sun gear failure rate from Table  $1^4$  is 31.5 x  $10^{-12}$  failures per cycle, based on a maximum working stress level of 30,500 psi. Based on a normal statistical distribution for a mean endurance limit of 55,000 psi and a standard deviation of 10% of the mean stress, the probability of failure at this stress is .0009. Defining a constant, K, as the ratio of failure rate to the probability of failure gives



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$$K_{s} = \frac{\lambda s}{P_{s}}$$
  
=  $\frac{31.5 \times 10^{-12}}{.0009}$   
=  $3.5 \times 10^{-8}$ 

Similarly, in the case of a planet pinion failure, the failure rate is based on a maximum stress of 34,800 psi (P<sub>p</sub> = .0150), which leads to

$$K_{\rm p} = 1.5 \times 10^{-8}$$

The total failure rates of the free planet transmission and the conventional two-stage planetary can be found by combining the individual component failure rates with the number of components.

For the free planet transmission,

$$\lambda_{\text{Free Planet}} = \lambda_{\text{sun}} + (m_{\text{sun/pinion}} \lambda_{\text{sun/pinion}}) \\ + (m_{\text{pinion/ring}} \lambda_{\text{pinion/ring}}) \\ + (m_{\text{output pinion/ring}} \lambda_{\text{output pinion/ring}}) \\ + (m_{\text{upper roller}} \lambda_{\text{rollers}}) \\ + (m_{\text{lower roller}} \lambda_{\text{rollers}}) \\ + \lambda_{\text{ball bearing}} \\ + (m_{\text{pinion shaft}} \lambda_{\text{shaft}})$$

The term ball bearing is the failure rate of the bearing that supports the dead weight of the free planet. This failure rate was assumed to be negligible, since the bearing has a calculated life of over 12,000 hours.

For the conventional two-stage planetary design,

 $\lambda_{\text{planetary}} = \lambda_{\text{sun}}^{(2)} + m_{\text{pinion}} (\lambda_{\text{sun/pinion}} + \lambda_{\text{pinion/ring}})$ 

- +  $\mathcal{M}$  thrust washers  $\lambda$  thrust washers
- + M plates 2 plates
- + *m* bearings  $\lambda$  bearings
- + **~**<sub>ring</sub> (2)

For the comparative analysis of a free planet with a conventional planetary, the endurance limit for current designs is 71,700 psi for ground surfaces. The a coefficients of  $\sigma$  to obtain the probabilities from the Gaussian distribution are calculated with this new endurance limit. All the historic data, as well as the actual comparative data, are summarized in Table 15.

The only potential problem that has not been addressed in this analysis is failure of rollers and journal of the free planet design. The rollers are expected to experience a pitting mode of failure, which would not result in gearbox removal.

The electron beam weld in the pinion shaft design is a fabrication technique recently introduced at Sikorsky. Electron beam welding was used for fabrication of rolling elements and gears on the Roller Gear Drive Development Program. In the roller gear drives built for that program, the design of the electron beam weld was the source of almost all problems. The welds were subsequently redesigned, and quality control procedure were improved to the point where after 79.5 hours of testing there were no weld failures.

The failure rates predicted for the free planet and conventional planetary designs are as follows:

Conventional Two-Stage Planetary

First Stage	$33.701 \times 10^{-6}$
Second Stage	$11.837 \times 10^{-6}$
Total	45.538 x 10 <sup>-6</sup>
Free Planet	$25.890 \times 10^{-6}$

The predicted mean time between failures (MTBF) is defined as the total flight hours on all parts, both satisfactory and failed, divided by the number of anticipated failures. MTBF is the reciprocal of the total failure rate. Therefore, for the two systems, the comparative predicted MTBF's are:

Conventional	Two-Stage	Planetary	System	MTBF =	22,000	hours
Free Planet				MTBF =	38,600	hours

The preceding analysis has provided failure rates on components experiencing stress cycles at the greatest rate. The total failure rate is also arfected by the number of components in each system. Since the free planet has far fewer components, and the criteria used were comparable for both designs, the results indicate that the MTBF of the free planet design is approximately twice that of the conventional planetary. The validity of these analyses can be determined only by testing in an environment representing the operational use as closely as possible. The real proof of the predicted reliability can be demonstrated only with a sample chosen from operational use.

	TABLE 15.	FALTIR A TWO-	E RATES COMPA STAGE CONVENT	LRISON O	F FREE PLAN IETARY DRIVE	ET AND UNIT		
UNIT	FATIGUE BENDING STRESS	ದ	PROBABILITY OF FAILURE	кто <sup>8</sup>	CYCLES PER HOUR X 10-6	NUMBER PER HOUR X10 <sup>6</sup>	NUMBER OF COMPONENTS	TOTAL FAIL.RATE FAILURE/6 HOUR X10
FREE PLANET Sun Sun Pinion(Sun) Pinion(Sun) Ring(Output) Pinion(Reaction) Bearing TOTAL UNIT	32,800 32,050 49,700 40,550 50,000 55,000	5.43 5.53 2.33 2.33 2.33 2.33	10 <sup>-7</sup> 10 <sup>-7</sup> .0010704 .0012220 .0012220	3.5 1.5 1.5 Negl. -	.167796 .150588 .150588 .033288 .033288 .150588	Negl. Negl. 2.418 Negl. Negl. Negl.	ユララユラユユ	- 12.090 13.800 - 25.89
CONVENTIONAL TWO-STAGE FLANETAR First Stage Sun Pinion(Sun) Pinion(Ring) Ring Bearings	ב 11,200 14,200 14,200 14,200 14,200	6.09 3.75 3.84 3.86	10 <sup>-7</sup> .00035259 .00025057	Negl.	.147000 .242340 .242340 .066420 .242340	Negl. 1.282 0.911 Negl. 4.05		8.974 6.377 28.350
Second Stage Sun Pinion(Sun) Pinion(Ring) Ring Bearings TOTAL UNIT	30,000 44,900 44,200 52,000	5.82 3.028 3.84 2.75	10 <sup>-7</sup> .00036605 .0002615 .0029798	3.5 1.5 Negl.	.046020 .073320 .073320 .020400 .073320	Negl. 0.403 0.068 Negl. 1.22	してくして	2.821 0.476 8.540 55.50

#### Helicopter Design Modeling - Analysis

The Sikorsky HDM (Helicopter Design Model) was used to evaluate the effect of reductions in transmission weight and cost on the MUT. HDM is a rapid, efficient tool for design iteration and evaluation of baseline helicopters and advanced concept helicopters.

## Background

Preliminary design of an aircraft is an iterative procedure involving configuration, weights, and performance. An initial configuration is developed from such design constraints as payload, volume, number of crew, number of engines, limit on rotor size, and mission equipment.

HDM is a digital computer program that provides the designer with the following outputs: rotor geometry, component weight bieakdown, mission analysis, engine and gearbox sizing, speed capability, and cost. These cutputs provide the designer with the refinements needed for each design iteration. A closed solution is achieved when the configuration, performance, weights, mission requirements, and system design specifications are consistent. Thus, HDM plays an important part in closing the design loop and furnishes insight into design sensitivities at the preliminary level to a degree never previously realizable. Aside from the derivation of the design point aircraft, the extensive trade-off and optimization capability of HDM enables the designer to trend away from the baseline configuration.

The program is available on the UNIVAC 1110 facility at our corporate research laboratories in Hartford, Connecticut. The program has been the primary preliminary design tool for the following contracts and proposals:

> U.S. Army Advanced Antitorque Study U.S. Army HLH Proposal U.S. Army UTTAS Proposal NASA/Army Rotor Systems Research Aircraft Predesign Study U.S. Army Structural Armor Fuselage Study U.S. Army ABC Operational Configuration Study U.S. Navy VTOL Escort Study U.S. Army AAH Proposal

For the present study, HDM was modified to suit the design constraints for a medium-size utility helicopter (MUT) and to obtain the desired level of detail in weights equations, engine and gearbox sizing criteria, and aerodynamic performances.

HDM has four basic loops LO, Ll, L2, L3, as shown in Figure 33. L0 is used to derive the gross weight needed to achieve the required payload. If gross weight is specified, payload is calculated. The calculations within L0 form the nucleus of the program. Ll, L2, and L3 enable trending, for a single set of input data, of the three primary design contraints:  $bl de loading (C_m / )$ . Elements of the drive system may be sized on the



Figure 33. Helicopter Design Model Flow Chart

basis of a design performance requirement, such as percent over-rating above the design hover input power. Knowledge of rotor power and PCTPR defines total power required from the engines, thus enabling selection of engine type and size. If required rotor geometry (radius and chord) is specified, CTSIG and DL are calculated. If a particular tail rotor geometry is specified, PCTPR is calculated. CTSIG, DL, and PCTPR may be selected as single inputs or as a required range (initial, final, and incremental values) so that repeated passes are made around the appropriate loop (L1, L2 or L3) to create a matrix of design points. For each range of any of these three variables, the interpolated value needed to produce the aircraft selected, based on user perference for minimum weight, minimum cost, maximum productivity, etc. Thus, if ranges of values are desired for CTSIG, DL, and PCTPR, the program identifies the combination of values needed to optimize the helicopter design. The user may request printouts at various levels of definition and at varying frequency through the calculation. For example, he may request a complete detailed weight breakdown for every pass around LO, or a summary weight statement on completion of optimization.

Life-cycle cost of a military helicopter is a summation of the costs of development, production, ground support equipment, crew training, maintenance, spares, and fuel. The composition of each of these items depends on the particular project under study. Development and production costs for the baseline MUT helicopter were statistically trended and were, in general, a function of the component weights already calculated. Outputs from this subroutine are production cost, flyaway cost, and life-cycle cost. Cost modeling was limited to flyaway cost for the purpose of this study. Flyaway cost was based on production of 500 aircraft and is stated in 1974 dollars. Table 16 is the MUT baseline data sheet, Table 17 presents MUT baseline weights, and Table 18 presents MUT baseline costs.

For the MUT aircraft with the free planet transmission, the aircraft was resized for two different cases: (1) with the same payload as the baseline aircraft and (2) with the same gross weight as the baseline aircraft. In each of these cases, the dollars per pound and the weights of the total transmission system were changed to reflect the improvements with the free planet transmission. For the case with the same payload as the baseline aircraft, Table 19 is the summary data sheet for the resized MUT aircraft, Table 20 is the summary weight statement, and Table 21 is a life-cycle cost summary. Similarly, for the case with the same gross weight as the baseline aircraft, Table 22 is the summary data sheet for the resized MUT aircraft, Table 23 is the summary weight statement, and Table 24 is a life-cycle cost summary.

700.0 8.231 .1089 231.9 7147 7 9.4 4.0 .535 4.40 .1547 TAIL ROTOR/FAN BLADE AREA (SQ.FT. TIP SPEED (FPPS) FIGURE OF MERIT TAIL ROTOR LIFT ROTOR SLDTY/AF NO. OF BLADES ASPECT RATIO RADIUS (FT) CHORD (FT) CT/SIGMA MUT BASELINE DATA SHEET 20.50 1.322 4.0 .0819 730.0 15.511 .0850 9239.4 .7555 108.4 BLADE AREA (SQ.FT.) MALIN ROTOR FIGURE OF MERIT MAIN ROTOR LIFT TIP SPEED (FPS) ROTOR SOLIDITY NO. OF BLADES ASPECT RATIO RADIUS (FT) CHORD (FT) CT/SIGMA TABLE 16. 20.00 7.00 1048. 960. .8711 9471. 6618. 1389. 1261. 1564. M.R. DISC LOADING (PSF) TAIL ROTOR CANT (DEG) MAIN ROTOR DESIGN HP MAIN G.B. DESIGN HP GENEFAL DESIGN ATTRIBUTES WEIGHT EMPTY (LB) HOVER POWER (SHP) DESIGN G.W. (LB) HOVER + CLIMB HP PAYLOAD (LB) FUEL (LB)

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All a data for the particular and

TABLE 17 . SUMM BASE	ARY WEIGHT STATEMENT LINE MUT	
GROUP	WEIGHT	% GW
MAIN ROTOR GROUP WING GROUP TAIL GROUP TAIL ROTOR/FAN TAIL SURFACES BODY GROUP ALIGHTING GEAR FLIGHT CONTROLS ENGINE SECTION PROPULSION GROUP	820. 0. 152. 47. 105. 1055. 380. 638. 100. 1907.	8.65 .00 1.60 .49 1.11 11.14 4.01 6.74 1.06 20.14
ENGINES AIR INDUCTION EXHAUST SYSTEM LUBRICATING SYSTEM FUEL SYSTEM ENGINE CONTROLS STARTING SYSTEM AUXILIARY PROPULSION PROPELLERS DRIVE SYSTEM	422. 40. 297. 0. 269. 25. 19. 0. 835.	4.46 .42 3.13 .00 2.84 .26 .20 .00 8.82
AUXILIARY POWER UNIT INSTRUMENTS HYDRAULICS ELECTRICAL GROUP AVIONICS ARMAMENT GROUP FURNISHINGS AIR CONDITIONING AND ANTI-ICE AUXILIARY GEAR	0. 135. 0. 247. 460. 53. 422. 48. 60.	.00 1.43 .00 2.61 4.86 .56 4.\6 .51 .63
VIBRATION SUPPRESSION TECHNOLOGY SAVINGS CONTINGENCY WEIGHT EMPTY FIXED USEFUL LOAD PILOT COPILOT OIL-ENGINE -TRAPPED FUEL TRAPPED	76. 0. 66. 6618. 504. 235. 235. 14. 6. 14.	.80 .00 .70 69.88 5.32
MISSION EQUIPMENT OTHER FUL. PAYLOAD FUEL-USABLE GROSS WEIGHT	0. 0. 960. 1389. 9471.	10.14 14.66

ITEM	DO	DLLARS
DEVELOPMENT COST PER AIRCRAFT	******	88959.
PROTOTYPE COST PER PRODUCTION AIRCRAFT		22291.
RECURRING PRODUCTION COST	529160.	
GFE AVIONICS	40000.	
ENGINE COST	89378.	
(FLYAWAY COST)	(658539.)	
INITIAL SPARES	206147.	
GROUND SUPPORT EQUIPMENT	39512.	
INITIAL TRAINING AND TRAVEL	52601.	
ACQUISITION COST		956800.
FLIGHT CREW	457200.	
FUEL + OIL	298324.	
REPLENISHMENT SPARES	893368.	
ORG + D/S + G/S MAINT	369534.	
DEPOT MAINTENANCE	322775.	
RECURRING TRAINING	274509.	
MAINTENANCE OF GSE	20769.	0(0() 70
OPERATING COST		2636479.
LIFE-CYCLE COST		3704529.
PRODUCTIVITY		.01088
FLEET LIFE CYCLE COST		1652264512.

TABLE 18. LIFE-CYCLE COST SUMMARY - BASELINE

Eł	ABLE 15.	DATA SHEET MUT AIRCRAFT RESIZED FREE PLANET TRANSMISS	TO SAME PA	YLOAD WITH AND COST	
DESIGN ATTRIBUTES					
GENERAL		MAIN ROTOR		TAIL ROTOR/FAN	
DESIGN G.W. (LB)	9335.	RADIUS (FT)	20.35	RADIUS (FT)	4.37
PAYLOAD (LB)	960.	CHORD (FT)	1.312	CHORD (FT)	.531
WEIGHT EMPTY (I.B)	6495.	NO. OF BLADES	4.0	NO. OF BLADES	4.0
FUEL (I.B)	1376.	ROTOR SOLIDITY	.0819	ROTOR SLDTY/AF	.1547
HOVER POWER (SHP)	1162.	TIP SPEED (FPS)	730.0	TIP SPEED (FPS)	700.0
HOVER + CLIMB HP	1243.	ASPECT RATIO	15.511	ASPECT RATIO	8.229
MAIN ROTOR DESIGN HP	1033.	CT/SIGMA	.0850	CT/SIGMA	.1087
TAIL ROTOR CANT (DEG)	20.00	MAIN ROTOR LIFT	9106.4	TAIL ROTOR LIFT	228.5
M.R. DISC LOADING (PSF)	7.00	FIGURE OF MERIT	.7555	FIGURE OF MERIT	.7146
MAIN G.B. DESIGN HP	1542.	BLADE AREA (SQ.FT)	106.8	BLADE AREA (SQ.FT)	9.3

	SUMMARY WEIGHT STATEMENT NUT AIRCRAFT RESIZED TO SAME PAYLOAD	
GROUP	WEIGHT	% GW
MAIN ROTOR GROUP WING GROUP TAIL GROUP TAIL ROTOR/FAN TAIL SURFACES BODY GROUP ALIGHTING GEAR FLIGHT CONTROLS ENGINE SECTION PROPULSION GROUP ENGINES AIR INDUCTION EXHAUST SYSTEM LUBRICATING SYSTEM FUEL SYSTEM AUXILIARY PROPULSION PROPE DRIVE SYSTEM AUXILIARY POWER UNIT INSTRUMENTS HYDRAULICS ELECTRICAL GROUP AVIONICS	808. 0. 149. 46. 104. 1049. 375. 630. 100. 1819. 418. 40. 295. 0. 267. 25. 19. 25. 19. 247. 460.	8.65 .00 1.60 .49 1.11 11.23 4.02 6.75 1.07 19.49 4.48 .43 3.16 .00 2.86 .27 .20 .00 8.10 .00 8.10 .00 1.45 .00 2.65 4.93
ARMAMENT GROUP FURNISHINGS AIR CONDITIONING AND ANTI-ICE AUXILIARY GEAR VIBRATION SUPPRESSION TECHNOLOGY SAVINGS CONTINGENCY	53. 422. 48. 60. 75. 0. 65.	.57 4.52 .51 .64 .80 .00 .70
WEIGHT EMPTY	6495.	69.58
FIXED USEFUL LOAD PILOT COPILOT OIL-ENGINE -TRAPPED FUEL TRAPPED MISSION EQUIPMENT OTHER FUL.	504. 235. 235. 14. 6. 14. 0. 0.	5.40
PAYLOAD FUEL-USABLE	960. 1376.	10.28 14.74
GROSS WEIGHT	9335.	

TABLE 21. LIFE-CYCLE CO MUT AIRCRAFT	OST SUMMARY RESIZED TO SAME PAY:	LOAD
ITEM	DOLLARS	
DEVELOPMENT COST PER AIRCRAFT PROTOTYPE COST PER PRODUCTION AIRCRAFT RECURRING PRODUCTION COST GFE AVIONICS ENGINE COST (FLYAWAY COST) INITIAL SPARES GROUND SUPPORT EQUIPMENT INITIAL TRAINING AND TRAVEL ACQUISITION COST FLIGHT CREW FUEL + OIL REPLENISHMENT SPARES ORG + D/S + G/S MAINT DEPOT MAINTENANCE RECURRING TRAINING MAINTENANCE OF GSE	520992. 40000. 88383. (649375.) 203373. 38962. 52496. 457200. 295453. 878892. 365023. 317258. 273899. 20468.	88078. 21981. 944206.
OPERATING COST		2608194.
LIFE-CYCLE COST		3662459.
PRODUCTIVITY		.01109
FLEET LIFE CYCLE COST		1831229328.

Ę	ABLE 22 .	DATA SHEET MUT AIRCRAFT RESIZED FREE PLANET TRANSMIS	) TO SAME P SSION WEIGH	AYLOAD WITH T AND COST	
DESIGN ATTRIBUTES					
GENERAL		MAIN ROTOR		TAIL ROTOR/FAN	
DESIGN G.W. (LB)	.1746	RADIUS (FT)	20.50	RADIUS (FT)	4.40
PAYLOAD (LB)	1027.	CHORD (FT)	1.322	CHORD (FT)	.535
WEIGHT EMPTY (LB)	6552.	NO. OF BLADES	4.0	NO. OF BLADES	h.0
FUEL (LB)	1389.	ROTOR SOLIDITY	.0819	ROTOR SLDTY/AF	.1547
HOVER POWER (SHP)	1178.	TIP SPEED (FPS)	730.0	TIP SPEED (FPS)	700.0
HOVER + CLIMB HP	1261.	ASPECT RATIO	15.51l	ASPECT RATIO	8.231
MAIN ROTOR DESIGN HP	1048.	CT/SIGMA	.0850	CT/SIGMA	.1089
TAIL ROTOR CANT (DEG)	20.00	MAIN ROTOR LIFT	9239.1	TAIL ROTOR LIFT	231.9
M.R. DISC LOADING (PSF)	7.00	FIGURE OF MERIT	.7555	FIGURE OF MERIT	7147.
MAIN G.B. DESIGN HP	1563.	BLADE AREA (SQ.FT)	108.4	BLADE AREA (SQ.FT)	4.6

TABLE 23.	SUMMARY WEIGHT STATEMENT MUT AIRCRAFT RESIZED TO SAME GRC	SS WEIGHT
GROUP	WEIGHT	% GW
MAIN ROTOR GROUP WING GROUP TAIL GROUP TAIL ROTOR/FAN TAIL SURFACES BODY GROUP ALIGHTING GEAR FLIGHT CONTROLS ENGINE SECTION PROPULSION GROUP ENGINES AIR INDUCTION EXHAUST SYSTEM LUBRICATING SYSTEM FUEL SYSTEM ENGINE CONTROLS STARTING SYSTEM AUXILIARY PROPULSION PROPEL DRIVE SYSTEM AUXILIARY POWER UNIT INSTRUMENTS HYDRAULICS ELECTRICAL GROUP AVIONICS ARMAMENT GROUP FURNISHINGS AIR CONDITIONING AND ANTI-ICE AUXILIARY GEAR VIBRATION SUPPRESSION TECHNOLOGY SAVINGS CONTINGENCY	$\begin{array}{c} 820. \\ 0. \\ 152. \\ 47. \\ 105. \\ 1055. \\ 380. \\ 638. \\ 100. \\ 1841. \\ 422. \\ 40. \\ 297. \\ 0. \\ 269. \\ 25. \\ 19. \\ 269. \\ 25. \\ 19. \\ 269. \\ 25. \\ 19. \\ 269. \\ 25. \\ 19. \\ 247. \\ 40. \\ 247. \\ 460. \\ 53. \\ 422. \\ 48. \\ 60. \\ 76. \\ 0. \\ 66. \end{array}$	$\begin{array}{c} 8.65 \\ .00 \\ 1.60 \\ .49 \\ 1.11 \\ 11.14 \\ 4.01 \\ 6.74 \\ 1.06 \\ 19.44 \\ 4.46 \\ .42 \\ 3.13 \\ .00 \\ 2.84 \\ .26 \\ .20 \\ .00 \\ 8.12 \\ .00 \\ 1.43 \\ .00 \\ 2.61 \\ 4.86 \\ .56 \\ 4.46 \\ .51 \\ .63 \\ .80 \\ .00 \\ .69 \end{array}$
WEIGHT EMPTY FIXED USEFUL LOAD PTLOT COPILOT OIL-ENGINE -TRAPPED FUEL TRAPPED MISSION EQUIPMENT OTHER FUL. PAYLOAD FUEL-USABLE GROSS WEIGHT	6552. 504. 235. 235. 14. 6. 14. 0. 0. 0. 1027. 1389. 9471.	69.18 5.32 10.84 14.66

TABLE 24. LIFE-CYCLE COST SUMMARY MUT AIRCRAFT RESIZED TO SAM	E GROSS WEIGHT
ITEM DOLLA	RS
DEVELOPMENT COST PER AIRCRAFTPROTOTYPE COST PER PRODUCTION AIRCRAFTRECURRING PRODUCTION COST525581.GFE AVIONICS40000.ENGINE COST89376.(FLYAWAY COST)(654957.)INITIAL SPARES205322.GROUND SUPPORT EQUIPMENT39297.INITIAL TRAINING AND TRAVEL52545.ACQUISITION COST457200.FLIGHT CREW457200.FUEL + OIL298318.REF LENISHMENT SPARES885527.ORG + D/S + G/S MAINT367091.DEPOT MAINTENANCE319787.RECURRING TRAINING274179.MAINTENANCE OF GSE20606.	88483. 22170. 952121.
OPERATING COST	2622708.
LIFE-CYCLE COST	3685481.
PRODUCTIVITY	.01175
FLEET LIFE CYCLE COST	1842740608.

#### CONCLUSIONS

The free planet transmission concept offers significant improvements over contemporary helicopter transmissions.

The most promising application, from an aircraft point of view, is a drive system for a single-engine, low-horsepower (500 - 600 HP), high-speed engine input. This permits a relatively high reduction ratio free planet unit (14 to 20:1 reduction ratio) and leads to significant reduction in the number of parts in the drive system. If free planet drive is considered for a UTTAS type engine drive train configuration, little improvement can be expected. A utility transport aircraft does not lend itself to design of a high enough ratio free planet drive unit. Engine location requires at least two gear meshes before the free planet unit is reached.

A 9% weight reduction is achieved with the free planet design. The free planet main transmission weighs 671 pounds, compared with 737 pounds for a two-stage conventional planetary design.

The costs of a free planet transmission and conventional planetary transmission are comparable in low quantities. A cost saving of 16.5% can be achieved for a free planet for a production quantity of 500 units.

An improvement in efficiency of over one-half of 1% is achieved through the use of the free planet transmission. Overall main gearbox efficiency is 97.3% for the free planet design and 96.8% for the conventional two-stage planetary. This difference in efficiency translates to a power available difference of 7.5 HP when transmitting 1450 HP to the main rotor.

An improvement in reliability of almost two-to-one is achieved through the use of the free planet design. The predicted MTBF of the free planet unit is 38,600 hours. The conventional two-stage planetary unit has a predicted MTBF of 22,000 hours.

The free planet is more tolerant of loss of lubrication than a conventional planetary design. Improvement is needed in the free planet bearing rings to reduce sensitivity to 23mm AP threats.

Further testing is needed to verify the self-alignment hypothesis and loadsharing characteristics under dynamic conditions.

#### RECOMMENDATIONS

Further work in the free planet concept for production helicopters should be conducted in an application in which single-engine, high-speed input is available in the 20,000 to 30,000 rpm range while transmitting 400 to 500 HP. This would permit a high-ratio, simple, lightweight transmission.

The self-alignment hypothesis and load-sharing characteristics under dynamic conditions should be verified by strain gaging the planet pinion shafts that orbit the sun gear and rotate about their own centers. Changing the planet pinion indexing tolerances will permit assessment of the actual pinion indexing requirements and may permit significant reduction in unit cost of a free planet transmission.

The effect of loss of normal lubricant supply should be assessed experimentally to verify the expected improvement through the elimination of conventional planetary bearings.

Reliability testing of a free planet unit and conventional two-stage planetary should be conducted to verify the projected 2-to-1 improvement in reliability of the free planet design.

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## APPENDIX A

# FREE PLANET SELECTION COMPUTER PROGRAM LISTING

С	PLALET PINION WITH THREE SFUR GEARS PER SHAFT, CUTFUT RING		
с	GEAR ON CENTRAL FREE PLANE	ET PINION, AND FIXED RING GEAR ON	
C	OUTER FREE PLANET PINION		
С			
- <b>C</b>	A.A.A.A.A.A.		
C			
C	PROGRAM WILL DETERMINE NUM	BERS OF TEETH, DIAMETRAL PITCH	
C	AND PITCH DTAMETERS FOR AL	L MEMBERS, GEAR TOOTH FACE WIDTHS	
r	AS WELL AS FREE PLANET UNT	TS OVERALL LENGTH	
č			
-e	and a second	an an a state a state a second se	
Ċ	NO	IMENCLATURE	
С			
С	NOPIN	= NUMBER OF PINIONS	
С	NSI	= NUMBER OF TEETH IN SUN GEAR - INTIAL	
С	NS	= NUMBER OF TEETH IN SUN GEAR VARIABLE	
T.	PITCH3 -	= DIAMETRAL PITCH OF FIRST ROW	
C		SUN-PINION MESH	
6		C.C.J.A.IO.IJ.IT	
č	n¢	316171611111212114 - DTANETED OF CIIM	
~	03	- DTANETER OF SUM MYNYMM	
č	NPIMAX	- UTARLICH OF SUN RINIRUN	
<u> </u>		DAU PINIAN	
č	NP 1	- NUMBER OF TEETH TH FTDST DOL PTATON	
c	DPI	= DIAMETER OF FIRST ROW PINTON	
č	MR3MIN	= NUMBER OF TEETH ON STATIONARY	
C		REACTION RING SEAR - THITAL	
C	MD 3MAH	- NIMAFR OF TEFTE ON STATTONAGE	
Ċ		REACTION DING SEAD - MAYTMIN	
č	MR 3	= NUMBER OF TEETH ON STATTONARY	
č		REACTION RING SEAR	
c	NP3 MAX	= NUMBER OF TEETE ON PINION THAT	
C		MATES STATIONARY REACTION RING	
<b>.</b>		GEAR - MAXINUM	
C	NP3	= NUMBER OF TEETH ON PINION THAT	
C		MATES STATIONARY REACTION RING	
C		GEAR	
C	PITCH3	= DIAMETRAL PITCH OF THIRD ROW STITION	
C		ARY RING AND PINION MESH	
	DRI		
	DF 3	MING GEAR	
C C	UP 5	- DIAMETER UP THIND ROW PINION	
č	THETAX	- NUMBER OF LEEIN ON SECOND ROW RING	
Ċ	NAMEN	- NUMBER OF TEETS ON CECOND DOU OT HE	
č	uutulu	- NUMBER OF ILLII ON SECURD KUN KING CEAD - NININUM	
	NP3	- NIMBLE OF LELT ON CCUMP DON	
~	1172	- NUMBLE UP ILLIA UN SECUNU EUN	
с С	NO THAT	RING OLAR - Mimord of Terty on Eccond Gou	
	T T CTAR	- NUMBER OF ILLIN UN SECUND RUN	
č	NPt	- NUMBER OF TEETH ON SECOND DON OTHION	
~		- NUMBER UP TELLS UN SECURU RUM PINIUN	
~	UNC	- VINHEIER UP SECOND RUN OUTPUT KING	
c	092	T ATANETER OF CEADNO DOL DINTON	
č		- DIRREICH UN SCOUND KUN FIRIUR	
C C	n M		
č			
Ċ	FP1P1	- FACE MIDIN RASED ON COMPRESSIVE	
č	T N JF J	- FREE WIDTH DRILD UN CONFREIJITE	

С FR2P2 = FACE WIDTH BASED ON COMPRESSIVE C STRESS OF OUTPLT RING PINION MESH С F SUNP1 FACE WIDTH BASED ON COMPRESSIVE С STRESS OF SUN FINION MESH ¢ AXIAL LENGTH OR HEIGHT OF FREE AXIAL C PLANET MEASURED FROM PINION С EXTREMITY REAL NAPIN INTEGER COUNT 2 FORMAT 43F 16.01 5 FORMAT (1H +1X+\*NS++2X+\*NP1++1X+\*NP2++1X+\*NR2++1X+\*NP3\*+1X+ \*NR3 \*+4x+\*DS \*+4x+\*DP1 \*+4x+\*DP2 \*+4x+\*DR2 \*+4x+\*LP3 \*+4x+\*DR3\*+ 1 2 6 X . 2 \*RR \*+6X+\*N \*+\*X+ + \*SP1 \*+1X+\*R3P3 \*+1X+\*R2P2 \*+1X+\*LENGT +\*+1X+ 3 \*1 R0M\*+2X+\*2 R0M\*+2X+\*3 R0M\*1 • FORMAT (1443+3X+\*NUMBERS OF GEAR TEETH\*+6X+5X+\*DIAMETERS OF GEAR\*+ 1\*TEETH MEMBERS\*+3X+\*REDUCTION\*+2X+\*PIN\*+1X+ \*GEAR FACE\* 2 \*WIDTH \*+1X+ \*AXIAL \*+2X+\*DIAMETRAL PITCH\*) READ (5+2) DSMIN. HP. RPHOUT WRITE (E+2) DSMIN+HP+RPHOUT COUNT : 1 DO 10 NOPIN = 6+12 WRITE (6+N) WRITE (6.5) COUNT = COUNT + I NSI = 118/ NOPINI + NOPIN DO 20 NS = NSI+200+NOPIN COUNT = COUNT + 1---00 30 J = 5+12 PITCHI = J COUNT = COUNT + 1 IF (J GO TO 30 .29. 111 IF (J .EQ. 9 ) GO IO 30 05 = NS / PITCH1 -IF IDS .LT. DSMIN 1 GO TO 20 NPIMAX = 3+NS 00 40 NP1 = 18+NPIMAX THE FOLLOWING CARDS CHECK THAT ONLY COMPOUND PLANETARIES С ARE CHOSEN THAT HAVE HUNTING TEETH WHICH MEANS FOR EACH С £ HESH WE HUST SHOW THAT С C NS/NP = WHOLE NUMBER +UNREDUCIBLE FRACTION С NR/NP = WHOLE NUMBER + UNREDUCIBLE FRACTION C .... .... CHECK FOR HUNTING TEETH OF PI-SI MESH С ÷ \*\*\*\* LL = NS = NP1 NI = L1/N1 3 KI = L1-K1+N1 **K b** IF EK4 -11 1+5+7 K2 ---------GO TO 9 = N3 7 11 N1 = K4 60 10 3 6 K2 = 1 CONTINUE IF (K2 .E9. 01 GO TO ND COUNT = COUNT +1 = NP1 / PITCHI DP1 NAPIN = NOPIN IF ( 140-/ ASIN ( DP1 + 2./ PITCH1 )/(DS + OP1))).LT. NAPINI GO TO 30 NR3MIN = (NS + 2+ NPL) / 2 IF INRIHIN .LT. 60 F NRIHIN = 60 NR3MIN = (NR3MIN / NOPIN ) • NOPIN NR3MAX = 5 •NR3MIN

IF (NR3MAX +GT.200 ) NR3MAX = 200 DO 50 NR3 = NR3HIN+NR3HAX+NOPIN NP3MAX =[NR3+21/5 COUNT = COUNT +1 DO 60 NP3 = 18+NP3MAX COUNT = COUNT +1 С .... .... С CHECK FOR HUNTING TEETH OF R3-P3 MESH С .... .... = NR3 11 NI = NP3 413 К1 = L1/N1 KĄ = L1 - K1+N1 IF 1K4- 11 401+404+402 401 = 0 K2 GO 10 105 40.2 = N1 LL = K4 N1 GO TO 403 404 K 2 = 1 CONTINUE 405 IF (K2 .EQ. D) GO TO 66 PITCH3 = (NR3 - NP3)/(DS + DP1) IF (PITCH3 .61.17.0) GO TO 60 K2 \_\_\_\_\_Z 0 ł GO 10 9 = N1 7 L 1 N 1 = K4 GO 10 3 = 1 6 K 2 9 CONTINUE IF (K2 .E9. 0) GO TO 40 COUNT = COUNT +1 = NP1 / PITCH1 DP1 NAPIN = NOPIN IF ( 1 100./ ASIN ( DP1 + 2./ PITCH1 )/(DS + DP1))).LT. NAPINI GO TO 3U NR3MIN = (NS + 2+ NPL) / 2 1 IF (NR3MIN .LT. 60 J NR3MIN = 60 NR3MIN = (NR3MIN / NOPIN ) • NOPIN NR3MAX = 5 • NR3MIN IF (NR3MAX +G7.200 ) NR3MAX =200 DO 50 NR3 = NR3MIN+NR3MAX+NOPIN NP3MAX =(NR3+2)/5 COUNT = COUNT +1 DO 60 NP3 = 18+NP3MAX COUNT = COUNT +1 С .... .... C CHECK FOR HUNTING TEETH OF R3-F3 MESH С .... .... = NR3 11 = NP3 NI NE3 K1 = L1/N1 = L1 - K1+N1 Kł 1K4- 11 401+404+402 TE 401 ĸz = 0 GO TO 905 402 L1 = N1 = KN NI GO TO 403 404 = 1 K2 CONTINUE 405 IF (K2 .EQ. 0) GO TO GC PITCH3 = (NR3 - NP3)/(DS + DP1) IF (PITCH3 .GT.12.0) GO TO GO IPICH3 = PITCH3

```
RPICH3 = IPICH3
          IF ( PITCH3 .GT. (RPICH3+.001))
IF ( PITCH3 .LT.(RPICH3-.F01))
                                               GO TO 60
                                               60 10 60
                   = NR3 / PITCH3
= NP3 / PITCH3
          DR 1
         DP3
          IF (1093 -12.+ 093); .LT.
                                          (DCMIN-1.01) 60 10 60
          IF (DR3 .CT. 13.C)
                                          GO TO 60
    MAPIN
                  = NOPIN
          NR2MAX
                  = 2 • NR3
          NR2MIN
                  = NR3 / 2
          IF & NR2MIN .LT. 60 1
                                    NP2MIN = 60
          NRZMIN = ENRZMIN / NOPINJ+ NOPIN
    D 0
         10 NR2
                  = NR2HIN+NR2HAX+NOPIN
                                                    COUNT
                                                               = COUNT +1
          NP2MAX = [NR2+21/5
                  = 18+NP2MAX
    00
        80 NP2
                                                    COUNT
                                                               = COUNT +1
              ....
                                                   ....
          CHECK FOR HUNTING TEETH OF R2-P2 MESH
              ....
                                                   ....
                  = NR2
          11
                    = NP2
          N 1
                    = L1/N1
303
          K1
          K 4
                    = L1-K1+N1
          IF
               (K4 -1) 301+304+302
301
          K 2
                    = 0
          GO TO
                 305
                  = N1
30.2
          LI
          N1
                   = K4
          GO TO
                  30 3
                  = 1
3.010
          K 2
30.5
          CONTINUE
         IF (K2 .EG. D) CO TO 80
PITCH2 = (NR2 - NP2 )/(DS + DP1 )
IF (PITCH2 .GJ. 12.0)GO TO 8C
          IPICH2 = PITCH2
          RPICH2
                  = IPICH2
          IF I PITCH2 .GT. ( RPICH2 . DP 1 ))
                                               GO TO 80
          IF ( FITCH2 .LT. (RPICH2-.EC) ))
                                               GO IO 80
                  = NR2/ PITCH2
          082
          DP2
                  = NP2/ PITCH2
         IF (1DR2 -(2. •DP2)) .LT.
IF (DR2 .GT. 19.0)
                                           105HIN- 1.011 30 TO 80
                                          GO TO 80
                   = NOPIN
    NAPIN
                             ((DP2 + 2./PIICH2) /(DP2- DP2)).LT.
         IF 66 180./ASIN
          NAPIN)
   1
                                         CO TO 10
    AS
                   = NS
    APZ
                   = NP2
    AR2
                   = NR2
                   ÷
    AR3
                    NR3
    AP1
                   = NP1
    AP3
                   = NP3
          IF (ENR3+NP2).EQ. (NP3+NR2)) GO TO 201
                  = ( 1 + (AR3+ AP1)/(AP3+ AS))/ (1 - ( AR3 + AP2) /
          RR
                     1 AP3 + AR211
   1
          GO TO 202
201
          RR
                      0.0
202
          CONTINUE
    IF
         L RR .LE. 0.01
                                          GO TO 80
          IF IT ABSERRILLT. 10.1. OR.
                                           tABStRRI.GT. 30.011 GO TO 80
          RPMIN
                 = RR+ RPMOUT
          TOROS
                   = ( 6 30 2 5. • HP )/ RPHIN
          TORGRZ
                 = (63025. +HP)/RPHOUT
                  = 12.0.TORUS)/(DS+ NAPIN)
= 12.0.TOROR2)/(DR2+ NAPIN)
          NTSUN
          HTR2
```

С

C

С

```
WTR3
                                                                = WTR2 -WTSUN
                                     FSUNPI = ({21.+WTSUN}+
                                                                                                                                          (1./DS+1./DP1))/ ((SIN(2.+22.5))
                 1 + 1
                                  145. **211
                                                              = ((2).+WTR3)+()./CP3-1./DR3))/ ((SIN(2.+22.5))+()45.
                                     FR3P3
                 1 ++211
                                    FR2P2
                                                              = (+21.+HTR2)+(1./[P2-1./DP2]) / (( SIN(2.+22.5))+(14
                 15.++211
                                     IF ( FR2P2 .GT. (.50+DP2)) GO TO 80

IF ( DP2.GT.DP3).AND.( DP1 .LT. DP2 ) GO TO 101

T = FR3P3 /2.0 +.10 + FR2P2 / 2.

A = (0P2+T )/(DP3-DP2)
                                     B = {DP1/DP31+(A+T)
IF((A+9)+LT+(T+,1)) G0 TO 101
                                                            = T + A + B + FR3P3/2. + FSUNPI/24
                                     AXIAL
                                                                                                                                     GO TO 102
                                                                = 0.0
      101
                                     AXIAL
       102
                                     CONTINUE
                                     IF 4 AXIAL .67+ 13+01 60 10 PO
                                     IF ( AXIAL .GT. (3. + DR3)) GO TO 80
                    WRITE (6,90) NS.NP1.NP2.NR2.NP3.NR3.DS.DP1.CP2.DR2.CP3.DR3.RR.
                                                                      FSUNPI+FR3P3+FR2P2+AXIAL+
PITCH1+PITCH2+PITCH3
                                        NOPIN.
                 5
                Ē
           30 FORMAT (1H +I4+514+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,2+2X+F5,
                 2
                                               F6+3+1X+F6.31
          BO CONTINUE
           TO CONTINUE
          60 CONTINUE
          50 CONTINUE
          48 CONTINUE
          30 CONTINUE
          20 CONTINUE
          10 CONTINUE
                   WRITE (6+1000) COUNT
   1000 FORMATCH ... THE NUMBER OF INTERATIONS IN ARRIVING AT A SOLUTION IS
                1++1201
STOP
                   END
9 X G L
     5.0
                                     1385.0
                                                                           345.0
aFIN
```

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1.2.2.