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3600 HP SPLIT TORQUE HELICOPTER TRANSMISSION

Final Report

By:
Graham White
Transmission Research, Inc.
Cleveland, Ohio 44106

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NATIONAL AERONAUTICS AND SPACE ADMINISTRATION
Lewis Research Center
Cleveland, Ohio 44135

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SUMMARY

This report contains final design details of a helicopter transmission that is powered by GE twin T 700 engines each rated at 1800 hp. The work was conducted under contract NAS 3-23931, awarded in September 1983 by NASA Lewis Research Center.

Task I of the study, the final design phase, demonstrates that in comparison with conventional helicopter transmission arrangements the split torque design offers:

a) A weight reduction of 15%

b) A reduction in drive train losses of 9%

c) Improved reliability resulting from redundant drive paths between the two engines and the main shaft.

The transmission fits within the NASA LeRC 3000 hp Test Stand and accepts the existing positions for engine inputs, main shaft, connecting drive shafts, and the cradle attachment points. One necessary change to the test stand involves gear trains of different ratio in the tail drive gearbox.

Progressive uprating of engine input power from 3600 hp to 4500 hp twin engine rating is allowed for in the design. In this way the test transmission will provide a base for several years of analytical, research, and component development effor targeted at improving the performance and reliability of helicopter transmissions.

1. INTRODUCTION

Geared drive trains have proved to be the lightest and most efficient means of transmitting power from the engines of a helicopter to the main and tail rotors. Attempts to improve the effectiveness of helicopter transmissions, by achieving a higher ratio of torque output/weight, include both the examination of new drive train configurations and the introduction of advanced technology components.

Previous design efforts by Transmission Research, Inc. have demonstrated that the adoption of improved drive train arrangements based on a split-torque principle can realize greater benefits with respect to weight reduction than can the substitution of redesigned components in a conventional transmission.

As a continuation of earlier work, a preliminary design of a helicopter transmission incorporating split torque gear trains was developed under contract NAS 3-22120 and subsequently forwarded to NASA LeRC in response to RFP 3-422196. This design of transmission is radically different from those used with production helicopters in the 3000-4000 hp range because it does not have a planetary reduction unit to supply high torque to the main rotor shaft. Fundamental to split torque type of transmission is that the power and torque from each engine is divided between two parallel paths prior to recombination on a single gear that drives the transmission output shaft. It is this feature of carrying torque through two or more separate drive paths that results in the descriptive term split torque transmission.

2. POTENTIAL BENEFITS FROM A SPLIT TORQUE TRANSMISSION

Eliminating the one or two planetary reduction stages of a conventional helicopter transmission and replacing them with a single gear driven by multiple pinions can bring a number of advantageous characteristics, as listed below. The transmission developed accordingly exploits these characteristcs where possible.

1) A potential weight (dry) of about 892 lb for a 3600 hp transmission powered by two GE T700 engines.
2) Separate, redundant drive paths between each engine and the final stage of speed reduction, so extending into the transmission the reliability benefit associated with two engines.
3) A greater speed reduction ratio at the final reduction stage than possible from a planetary unit; achieved at less weight than with a planetary unit.
4) Marginally lower losses than alternative transmissions as a result of including only three fixed-axis gear reduction stages between engines and main shaft.
5) Potentially reduced noise levels as a result of having only ten gear-mesh points in drive trains to the main shaft.
6) Precisely known loads at all mesh points, in contrast to the uncertain load sharing between several planet pinions.
7) Lower overall height than alternative designs.
8) Ability to accept uprating of twin T700 engines to 4500 hp within the same diameter housings at a marginal gain in weight.
9) Ability to develop and incorporate advanced technology components on a progressive basis.

3. DESIGN SPECIFICATIONS FOR NASA 3600 HP TRANSMISSION

Development of a twin-engine, 3600 hp helicopter transmission that incorporates split-torque drive trains is based on the following outline specification:

Number of engines and type	Two; GE T700
Engine separation; front face	60 inch total
Engine power, each	1800 hp
Engine rated speed, rpm	20900 nominal
Transmission overall ratio	81.6:1
Tail drive power; mean & transient	10% & 20% engine power
Main shaft power; cont. rating	90% engine power
Main shaft speed; rpm	258
Tail drive shaft speed; rpm	4133
Housing feet location	to suit test stand
Engine input flanges	to suit test stand
Engine and tail drive waterlines	to suit test stand
Main shaft forward inclination	3°

Weight target; at least 10% below the parametric trend line

In addition to the foregoing items the transmission must fit within the confines of the NASA-LeRC 3000 hp test stand, be mounted on the existing cradle, and accommodate the present couplings and their locations.

Since the test stand is of closed-loop type, the gear ratios between the main shaft and engine inputs, and the main shaft and tail drive shaft, cannot be approximated. These ratios must be exact integer solutions that correspond with the tooth number products already incorporated in the test stand gearboxes.

Allowable design stresses

Detail design of components in the transmission is such as to keep within design allowables appropriate to current production transmissions. Further, since the transmission is a test unit, the sizing of all shafting, splines, bolted joints, and most bearings, is purposely made adequate for the uprating of each engine input from 1800 hp to 2000 hp.

Shafting	AISI 4340	
	Bending stress	20000 lb/in^2
	Shear stress	30000 lb/in^2
Gearing	AMS 6265	AGMA rating
	Spur gear compressive stress	160000 lb/in^2
	Spur gear bending stress	60000 lb/in^2
		GLEASON rating
	Spiral bevel compressive stress	220000 lb/in^2
	Spiral bevel bending stress	30000 lb/in^2

The following constants have been adopted in calculation of bearing life:

a) Bearing load prorate as a fraction of max load: 0.6

b) B_{10} material life factor: 5

4. DEVELOPMENT OF TRANSMISSION ARRANGEMENTS BASED ON INCLINED CROSS-SHAFTS

Dominating the development of a new transmission configuration in the present case is a requirement that the front-face axes of the engines have a 60 inch separation. This wide separation, a military requirement related to engine survival, has a restrictive effect on the choice of configuration for it tends to force the use of bevel gears at both the first and the second reduction stages of a transmission.

While many transmission arrangements of split-torque form can be derived that satisfy the need for only three reduction stages, low losses and low weight, few of these designs can, in addition, meet the requirement that the engine axes be parallel and widely separated. Figures 1 and 2, for instance, are examples of split-torque transmissions that have many attractions including those of reduced weight, reduced losses, and ability to accept the 1800 hp rating of each T700 engine. Unfortunately these arrangements lack the ability to accept engines that are parallel and separated by a distance (60 inches) that is markedly greater than the diameter of the combining gear.

Configurations with widely separated engines

A consequence of the 60 inch engine separation is that suitable arrangements which are worthy of detailed examination reduce to some three types and variants; none of these have previously been explored in respect of application to high ratio helicopter transmissions. The group or family of drive trains is distinguished by the presence of inclined cross-shafts that pass over, or under, a final-stage combining gear. The number of pinions that drive the combining gear provides a convenient method for classifying the different designs.

Each of the arrangements brings the following characteristics:

a) an ability to accept any position of the engines including wide separation, fore/aft positions, height, and any inclination

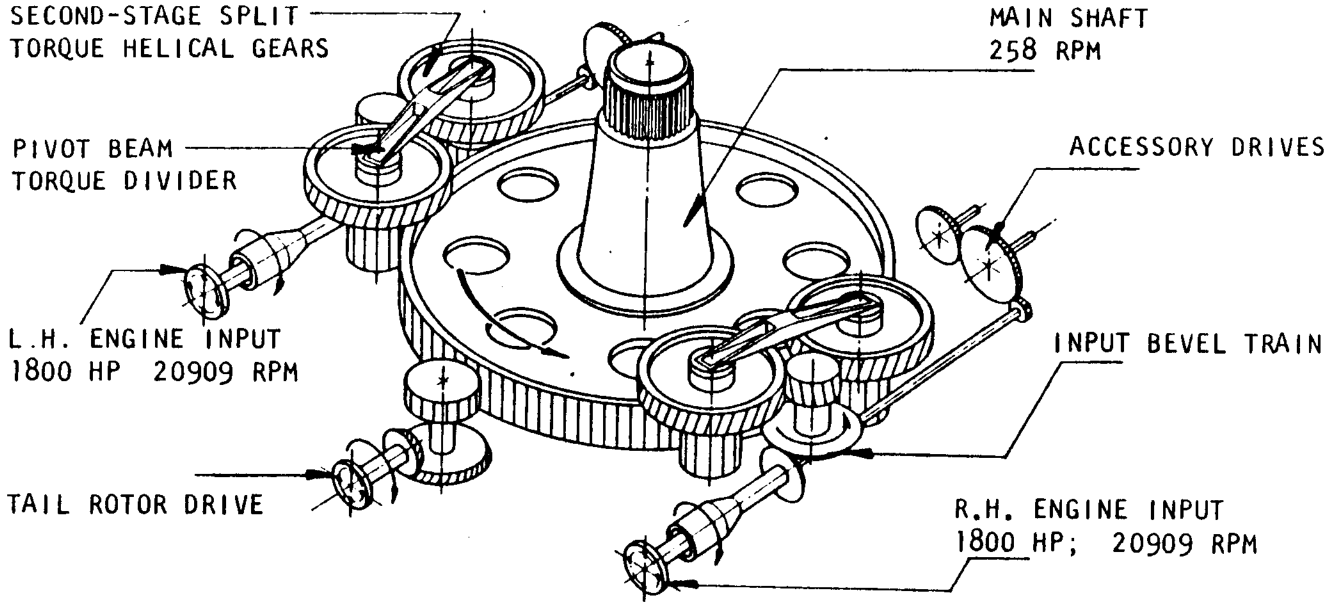


FIGURE 1

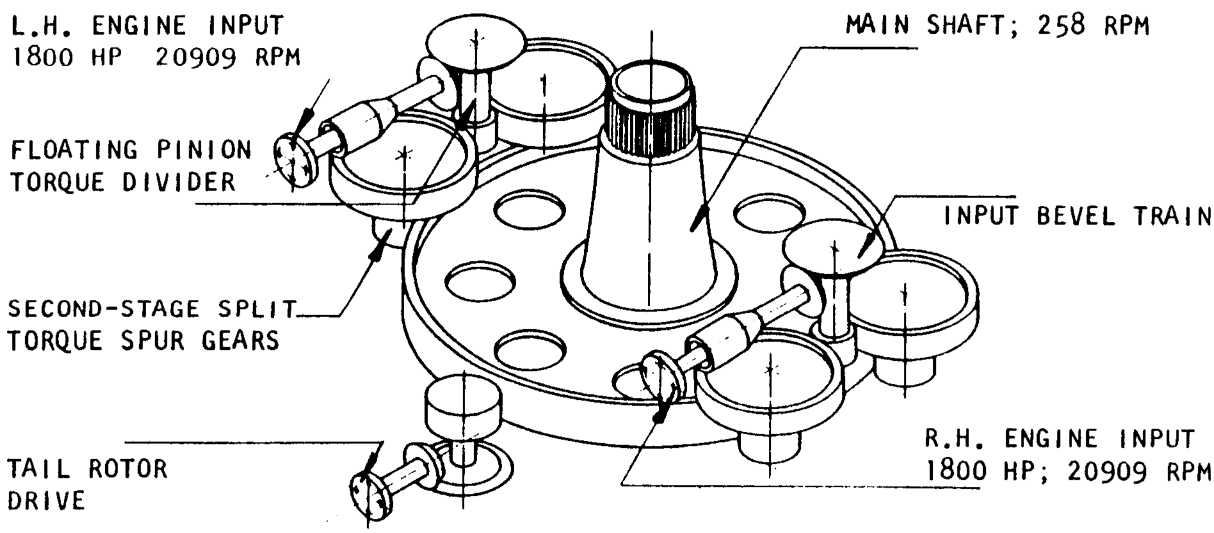


FIGURE 2

Examples of simple, split-torque helicopter transmissions that can accept the twin 1800 hp engine rating but are not suited to engine axes that are widely separated.

b) the provision of redundant, or independent, drive paths between each engine and the gear that drives the main shaft

c) the low losses and low component counts associated with three reduction stages based on fixed-axis gears

d) a higher ratio of torque output to weight than realized by conventional production designs.

Number of final-drive pinions

Transmission designs based on three, four. and five final-drive pinions are described. These arrangements have least mechanical complexity together with adequate torque capacity for the 3600 hp twin engine rating involved. Other groups of drive train arrangements are feasible that have more than four pinions around the combining gear. But these arrangements are not discussed in the present context as a consequence of being appropriate to the higher engine ratings and output torques of heavy-lift helicopters.

The actual gear train arrangements adopted to space the pinions around the combining gear are shown in Figures 3 through 5.

5. SPEED RATIO AT EACH REDUCTION STAGE

The overall speed ratio required between engines and the main shaft of the transmission is a nominal 81:1. In normal circumstances the tooth numbers in each reduction stage could be selected to provide an overall ratio which closely approximated that given in the specification. In order to make use of the NASA test stand, however, the ratios in the transmission must be an exact match with those of the test stand. This matching of the ratios in each of the three drive loops, from each engine and the tail drive to the main shaft, allows the loading mechanism to wind a torque into each loop and then be locked stationary. If the ratio in the transmission did not match those in the test stand then a continual rotation of the loading mechanism would be required to hold a given test torque.

5.1 Engine to Main Shaft Ratio

The test stand ratio between main shaft and engine input is:

$$\frac{29 \times 9^2 \times 5^2 \times 8}{31 \times 17 \times 11} \quad \text{in lowest terms}$$

Tooth stress levels at the transmission output stage show that the combining gear needs to be in the region of 32-36 inches diameter with teeth of about 6-8 diametral pitch (DP). Since the lowest common factors in the test stand ratios must also appear in the ratio of a new transmission, the test stand ratios can be listed as shown below and used as a basis for selecting teeth on the combining gear.

Tooth Number Options for Final Reduction Stage

Factor for teeth on combining gear	Combining gear teeth and pitch	Combining gear diameter, inches
29 x 8 = 232	232 x 7 DP	33.14
29 x 10 = 290	290 x 8 DP	36.25
30 x 8 = 240	240 x 7 DP	34.28
40 x 5 = 200	200 x 6 DP	33.33
45 x 5 = 225	225 x 7 DP	32.14
45 x 5 = 225	225 x 4 DP	35.43
27 x 8 = 216	216 x 6 DP	36.00
27 x 9 = 243	243 x 7 DP	34.70
27 x 10 = 270	270 x 8 DP	33.75

5.2 Engine and Intermediate Stage Ratios

In conjunction with the choice of ratio and tooth numbers at the final stage, the engine reduction and intermediate gear trains must include the remaining lowest common factors contained in the test stand ratio. As examples of this procedure, the following tooth numbers can be adopted in conjunction with final-stage ratios of 225/31 and 232/34.

Combining gear ratio	Intermediate bevel ratio	Engine reduction ratio
$\frac{225}{31} = 7.258$	$\frac{72}{22}$ or $\frac{108}{33}$	$\frac{58}{17}$ or $\frac{116}{34}$
$\frac{225}{31} = 7.258$	$\frac{58}{17}$ or $\frac{116}{34}$	$\frac{72}{22}$ or $\frac{108}{33}$
$\frac{232}{34} = 6.824$	$\frac{100}{31}$ or $\frac{90}{31}$	$\frac{81}{22}$ or $\frac{45}{11}$
$\frac{232}{34} = 6.824$	$\frac{81}{22}$ or $\frac{45}{11}$	$\frac{100}{31}$ or $\frac{90}{31}$

5.3 Tail Drive Ratio

An ideal choice of tooth numbers for the tail rotor driveshaft would give the correct speed for the tail drive shaft, while involving no change in the test stand ratios. But only two bevel gears are used to extract tail drive power from the rear combining pinion, and this one train cannot compensate for all the prime numbers in the tail drive loop of the test stand. Accordingly, tooth numbers in the transmission have been selected as:

tail drive gear teeth/pinion teeth = 58/25

which gives a tail drive speed of 4344 rpm, slightly above the ideal. In this case two helical gear trains in the tail drive loop of the test stand need changing to accommodate the new transmission.

5.4 Gear Ratios and Speeds in Transmission

Based on the foregoing possibilities for matching the ratios in the transmission with those existing in the test stand, the following sets of tooth numbers were adopted after a check on the stress levels involved.

Summary of Tooth Numbers and Speeds in Each Drive Stage

Drive Stage	Pinion or Gear	Tooth Number	Speed rpm
Engine reduction	pinion	34	20908.8
	gear	116	6128.4
Intermediate bevel, front	pinion	22	6128.4
	gear	72	1872.6
Intermediate bevel, rear	pinion	22	6128.4
	gear	72	1872.6
Combining stage	pinion	31	1872.6
	gear	225	258
Tail rotor drive	gear	58	1872.6
	pinion	25	4344.4

6. FOUR-PINION TRANSMISSION WITH INCLINED CROSS SHAFTS

Figure 3 shows an arrangement in which four pinions are spaced around a combining gear that drives the main shaft. Virtually any required position of the engines can be obtained by appropriate spacing of the final drive pinions and selection of the shaft angle for the engine reduction bevels. A detailed layout of this design is given in Fig. 4.

In each engine driveline the division of transmitted torque between the parallel sets of intermediate stage bevel gears and final drive pinions allows the overall ratio of 81.6:1 to be generated in three reduction stages at the high efficiency associated with fixed-axis gear trains. A torque-dividing unit is included to ensure that the parallel drive paths always carry their designed fraction of total torque; no change in speed is taken across this unit.

Accessory gearboxes are mounted on the front face of the main housing, while power for the tail rotor is extracted from one of the rear bevel gears.

Engine Reduction Stage

The engine reduction stage consists of a simple bevel train that accepts an engine speed of about 21,000 rpm and provides a reduction ratio of 3.4. The shaft angle of the gearset is chosen to suit the transverse offset and fore/aft position of an engine.

A coilspring type of overrunning clutch is placed between each engine input shaft and the associated bevel pinion. The clutch contributes least weight in this position as a result of being sized for the minimum torque. In the test transmission, however, the clutch is omitted and the engine reduction bevel pinions are connected directly to their input drive flange.

Intermediate Stage Bevel Gears

Torque from each engine bevel gear passes into a dividing unit which supplies equal torques to two second-stage bevel pinions that are connected in parallel. Coaxial mounting of the dual sets of bevel pinions result in the rear pinions being driven by inclined cross-shafts.

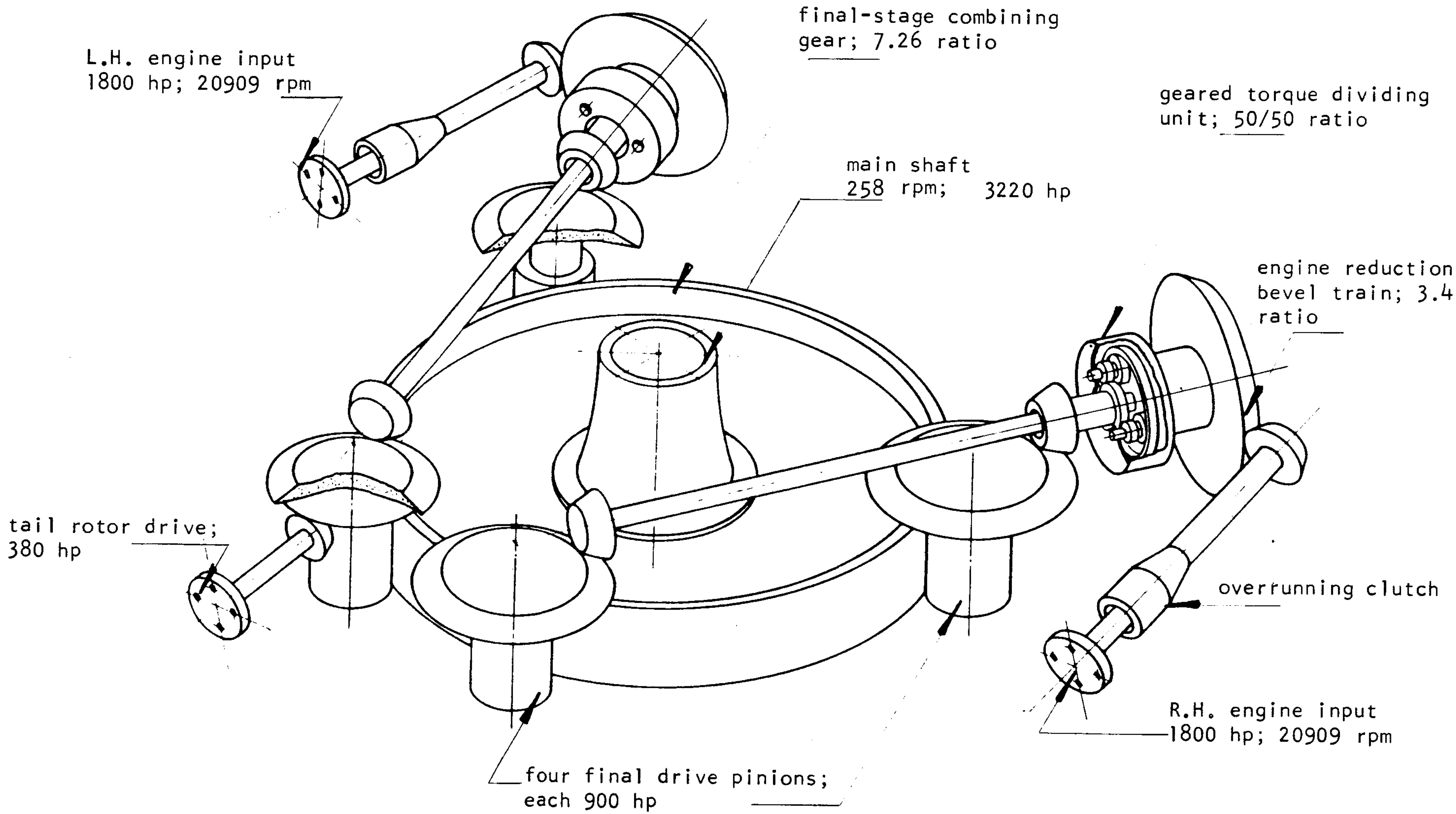


Figure 3 Arrangement of gear trains in four-pinion transmission

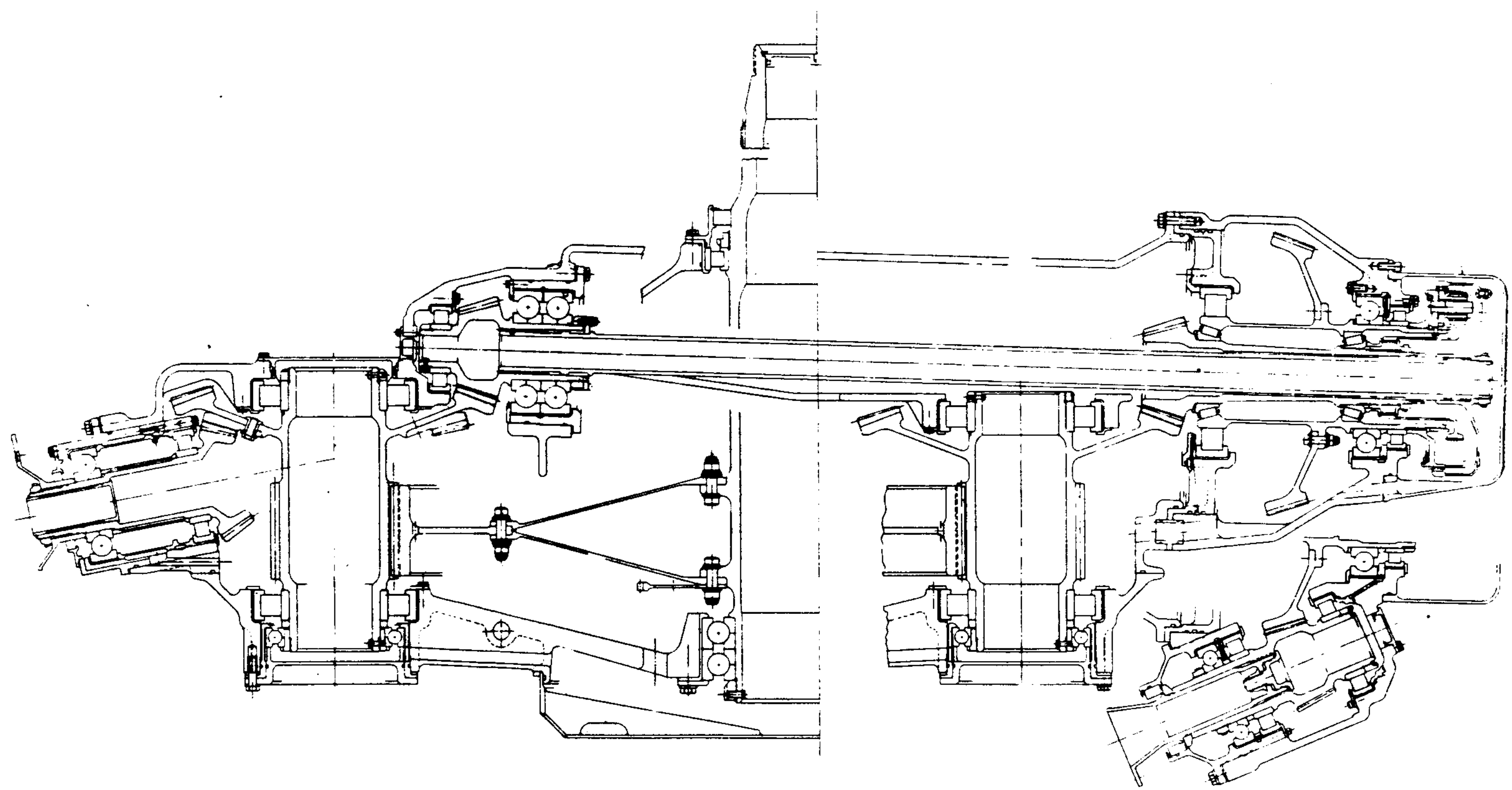


Figure 4. Detailed design layout of four-pinion split-torque transmission.

Since the bevel pinions receive identical torques and rotate at the same speed they transmit the same power; this power can rise to a nominal maximum of 900 hp to each pinion. More precisely, each pinion carries half the power supplied by an engine less power extracted by an accessory gearbox.

The bevel pinions and the gears that they drive are identical in respect of tooth loads, tooth geometry, and speed ratio, while having different types of bearing supports.

Torque dividing mechanism

The intermediate bevel pinions are not connected directly to the engine bevel gear, but are driven through a torque dividing unit that supplies two identical output torques, each being half the torque input from the engine bevel gear.

Several alternative mechanisms are appropriate but the type chosen, for reasons of lightness, reliability, and ability to accommodate axial growth, consists of a small planetary unit with stepped planet pinions that engage a ring gear and a sun gear. The ring gear supplies torque to a rear bevel pinion while the sun gear transmits an identical torque to a front pinion.

There is no motion across the gears and bearings of the torque dividing unit and so no losses are incurred. At a change in engine torque, however, an incrementally small angular adjustment of the ring gear and sun gear occurs as the long cross shaft deflects torsionally and as the housings that carry the bevel gears deform. The function of the torque divider is to keep constant the division of torque between the parallel sets of bevel gears and final drive pinions despite such torsional and structural deformations.

Final reduction stage

The final-stage pinions and associated bevel gears are of one-piece construction in order to minimize weight and allow each assembly to be carried on only two bearings.

Spacing of the four pinions around the combining gear assists in balancing the radial loads on the gear and the main shaft; the spacing chosen keeps to a low level the unbalanced load that occurs during single engine operation.

Different methods of construction for the combining gear are feasible, the lightest design comprising a steel gear rim and web that transmits torque to the main shaft through titanium support discs. Direct mounting of the combining gear on the main shaft provides a rigid support for the gear while avoiding the introduction of bearings additional to those that carry the main shaft.

Speed reduction ratios up to about 10:1 are possible when four pinions drive the combining gear, the highest ratios naturally involving a larger gear diameter and a consequent increase in housing size. In order to keep the gear and housing size comparable with planetary-type designs, and allow for future engine growth over 1800 hp, the final reduction ratio is held at 7.26:1. This ratio is much greater than the 4.7:1 available from a five-pinion planetary unit, and results in a transmission that is about 100 lb lighter.

Tail rotor drive

Power for the tail rotor is extracted from one of the rear bevel assemblies. In order to minimize height, however, and to give flexibility in choice of ratio and shaft inclination, the tail drive pinion is driven from the underside of the main bevel gear. This arrangement also has the advantage of a drive path between the main shaft and the tail drive shaft that is redundant except for the pinion that meshes with the combining gear.

At maximum power conditions the main bevel gear on which the tail drive bevel is mounted carries 900 hp. But the mean power extracted by the tail drive shaft is some 380 hp, which leaves the associated final drive pinion carrying only 520 hp to the combining gear. Transient load increases imposed by the tail rotor thereby reduce the power carried by that final drive pinion. In effect, extracting tail rotor power from the rear bevel assembly results in reduced duty on the associated final drive pinion.

A consequence of positioning the tail drive pinion as described is an offset between the tail drive flange and the vertical center plane through the main shaft. The tail drive shafting can either be taken directly to the tail rotor assembly, which itself is offset from the fuselage centerline, or the drive shafting can be angled to a conventionally placed intermediate gearbox at the base of the tail rotor pylon. In either case it appears logical to place the transmission cooler and blower in the space created by offset of the tail drive shaft.

TRANSMISSION SUMMARY: FOUR-PINION SPLIT TORQUE DESIGN

Number of reduction stages; engines to main shaft	3
Number of reduction stages; engines to tail drive	3
Drive train losses; % of input power	2.25%
Number of primary gears	19
Number of primary bearings	31
Non-redundant gears and bearings	1 & 3
Noise generating meshes	11
Engine reduction speed ratio: 116/34	3.41
Intermediate reduction ratio: 72/22	3.27
Final reduction ratio: 225/31	7.26
Overall speed ratio; engines to main shaft	81.04
Projected weight for 3600 hp rating, lb	892

7. FIVE PINION TRANSMISSION WITH INCLINED CROSS SHAFTS

Extension of the four-pinion arrangement to include a fifth pinion around the combining gear allows a complete separation of the tail drive from the main shaft drive trains (fig. 5). The fifth pinion and its accompanying bevel gear and pinion is positioned on the fore/aft centerline of the transmission; its only function is that of extracting tail rotor power from the combining gear.

Addition of the fifth pinion involves no dimensional or structural changes to the engine reduction bevels, the intermediate bevels, or the final drive pinions. Each of these gear trains remain as with the four-pinion arrangement. But now all four of the main drive pinions transmit half single-engine power, 900 hp, to the combining gear, while a wider separation of the two rear intermediate bevel gears brings a change in the shaft angle of the engine reduction bevel train.

Minor increases in drive train losses occur as a result of tail drive power passing through five meshes, in place of the three meshes with a four-pinion design. The totals of gears and bearing also rise with the five pinion arrangement and bring an incremental weight gain. Offsetting these drawbacks, however, is the principal advantage that all the drive trains that connect the engines, the main shaft, and the tail drive shaft, are independent from each other, or redundant. Further, the drive train losses and overall weight are still less than with a conventional helicopter transmission.

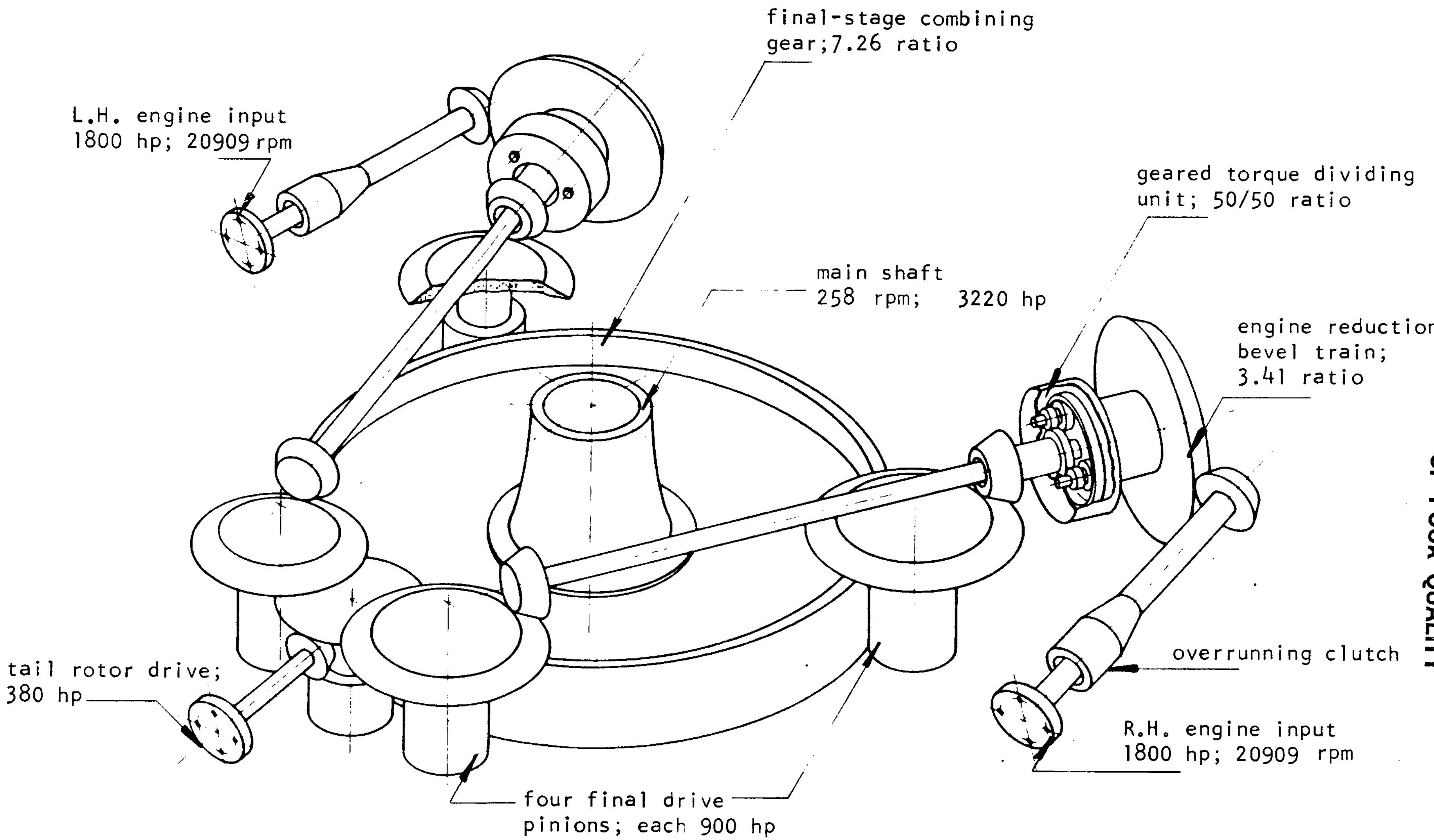


Figure 5 Arrangement of gear trains in five-pinion transmission

TRANSMISSION SUMMARY: FIVE-PINION SPLIT TORQUE DESIGN

Number of reduction stages; engines to main shaft	3
Number of reduction stages; engines to tail drive	5
Drive train losses; % of input power	2.45%
Number of primary gears	20
Number of primary bearings	34
Non-redundant gears and bearings	1 & 3
Noise generating meshes	12
Engine reduction speed ratio: 116/34	3.41
Intermediate reduction ratio: 72/22	3.27
Final reduction ratio: 225/31	7.26
Overall speed ratio; engines to main shaft	81.04
Projected weight for 3600 hp rating, lb	927

8. THREE-PINION TRANSMISSION WITH INCLINED CROSS SHAFT

An attractive simplification of the four-pinion arrangement is obtained by combining the two rear pinions, and driving the single bevel gear that results from two pinions(fig. 6). The overall effect is, first, to reduce the number of final drive pinions around the combining gear to three, and second, to place the tail drive shaft on the central axis of the transmission.

The benefits associated with reduced component totals, however, bring the penalty that torque for the main shaft has to be transmitted through three mesh points, while the forward bevel gears must carry more than half single-engine power. Paradoxically, the torque dividing unit is simplified because it does not have to provide two output torques that are equal.

Comparison with alternative designs shows the three-pinion arrangement offers the following characteristics:

a) accepts any position of the engines;
b) has the low losses associated with the four pinion design;
c) requires fewer gears and bearings than the four and five pinion designs;
d) sacrifices some level of drive train redundancy;
e) involves a weight gain at the final reduction stage as a result of having a wider facewidth;
f) provides a centrally located tail drive shaft;
g) requires intermediate stage bevel gears of higher torque capacity than the four pinion design.

Engine reduction stage

No change is involved in the engine reduction stage except for a different shaft angle than used in the four and five pinion designs.

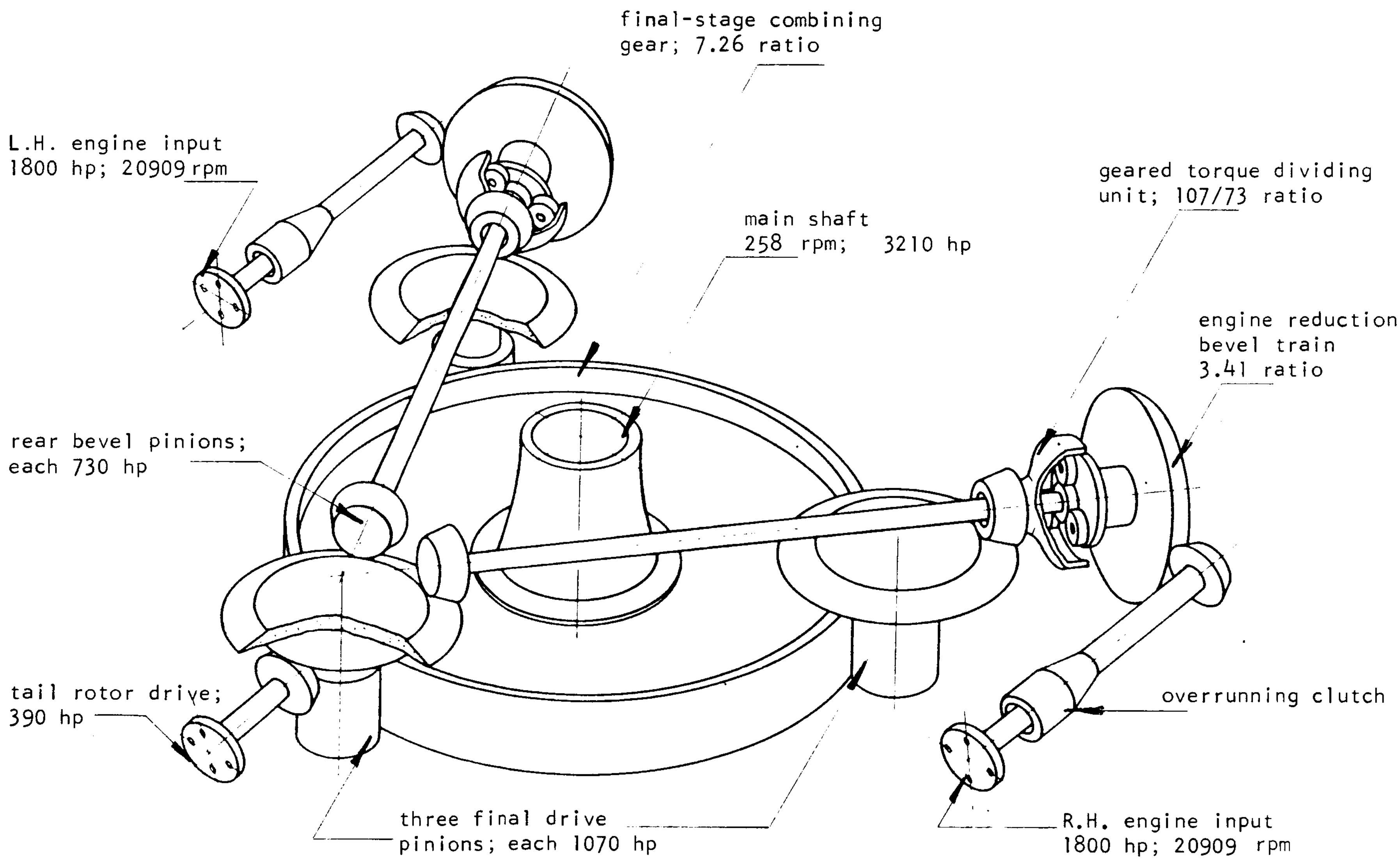


Figure 6 Arrangement of gear trains in three-pinion transmission

Intermediate stage bevel gears

The intermediate bevel gears must transmit equal torque to each of the three final drive pinions. For this condition to be realized, the rear bevel gear must receive sufficient torque for both the tail rotor and the final drive pinion. This additional torque is easily supplied as a result of the gear being driven by two pinions.

For instance, if the tail rotor drive demands a mean power level that is about 11% of the 3600 hp twin engine rating, or 390 hp, then each of the three final drive pinions must transmit 1070 hp to the main shaft. Therefore each of the forward bevel pinions must carry 1070 hp while each of the two rear bevel pinions must transmit 730 hp into their common bevel gear.

The principal change from the four and five final-pinion designs is clearly associated with the front bevel gears and the attached final drive pinions being upsized from 900 to 1070 hp.

Torque dividing mechanism

In contrast to the four pinion design, in which a torque divider supplies equal torque to two bevel pinions, the three pinion design requires two torques in the ratio of 1070/730, or 1.47, assuming 11% of engine power to the tail drive shaft.

Such a ratio is conveniently supplied by a simple sun/planet pinion/ring gear type of epicyclic train. With torque from an engine bevel gear applied to the planet carrier frame, and the ratio of ring diameter/sun diameter equal to 1.47, the lowest torque from the sun gear is given to a rear bevel pinion while the ring gear carries a higher torque to a front bevel pinion.

No internal motion occurs within the epicyclic torque divider as result of all its members turning at 6128 rpm, the speed of the engine bevel gear. But at a change of engine torque from zero to maximum the planet pinions will rotate about 10 degrees as a crossshaft deflects torsionally under the action of the torque applied to it.

Final reduction stage

A consequence of three pinions driving the combining gear, each transmitting 1070 hp, is that the facewidth of the gear must be increased in comparison with the four-pinion design if the diameter and stress levels are to remain comparable. This larger facewidth results in a small weight increase. There is also less potential for uprating the design to accommodate future engine growth since gear facewidth may have already been increased to its limit.

Otherwise the structure of the combining gear, the method of attachment to the main shaft, and the reduction ratio generated remain as with the four pinion design.

Tail rotor drive

No change is involved in the tail rotor drive pinion and its support bearings. The three pinion arrangement does have an advantage over the four pinion arrangement as a result of the tail drive shaft being centrally positioned.

TRANSMISSION SUMMARY: THREE-PINION SPLIT TORQUE DESIGN

Number of reduction stages; engines to main shaft	3
Number of reduction stages; engines to tail drive	3
Drive train losses; % of input power	2.25%
Number or primary gears	17
Number of primary bearings	29
Non-redundant gears and bearings	3 & 5
Noise generating meshes	10
Engine reduction speed ratio: 116/34	3.41
Intermediate reduction ratio: 72/22	3.27
Final reduction ratio: 225/31	7.26
Overall speed ratio; engines to main shaft	81.04
Projected weight for 3600 hp rating, lb	915

9. COMPARISON OF TRANSMISSION ARRANGEMENTS

Final selection of the transmission arrangement from the alternatives is based on a comparison of weight, drive losses, component counts and level of drive train redundancy. These factors are given in Table I; the weights for each arrangement are derived from design layouts.

A UH 60 type transmission also is included in Table I for two reasons. First, it represents a recent and advanced transmission design that is powered by T700 engines. Second, the split torque arrangement comprises a possible replacement for the UH 60 arrangement as a consequence of both designs fitting the NASA test stand, and accordingly having identical drive shaft locations and torque reaction points.

Table I shows the four pinion design to have best overall advantage for the following reasons:

* least overall weight
* reduced losses
* only one non-redundant gear
* comparable totals of gears and bearings

A further advantage of the four-pinion arrangement is that twin-engine growth to 4500 hp can be accepted within the same housing, whereas this is not the case with the three-pinion arrangement.

These factors result in selection of the four-pinion arrangement for further design work, detailing, and optional fabrication under Task IV.

TABLE I COMPARISON OF SPLIT TORQUE AND PLANETARY TRANSMISSIONS

CHARACTERISTIC	SPLIT TORQUE DESIGN			PLANETARY DESIGN
	3 Pinion	4 Pinion	5 Pinion	
Weight; lb at 3600 hp	915	892	936	1050
Losses; % at 3600 hp	2.25 81 hp	2.25 81 hp	2.35 84.6 hp	2.48 89.1 hp
Noise meshes	10	11	12	16
No. of gears	17	19	20	16
No. of bearings	29	31	36	31
Non-redundant gears & bearings	3G & 5B	1G & 3B	1G & 3B	8G & 11B
		Design Selected		UH 60 Type

9.1 Weight Comparison with Planetary Design

The weight reduction available from the four-pinion split torque design is assessed by comparison with an uprated UH 60-type planetary transmission. Previous examinations into helicopter transmission trends have demonstrated that weight varies according to (output torque)$^{.75}$ for equal overall reduction ratio.

A baseline for comparison is the UH 60 planetary transmission with a weight of 943 lb at 1560 hp per engine. This weight includes the main transmission, input sections, main shaft, sump and internal lube pumps. Not included are the accessory gearboxes, cooler and blower.

Main shaft torque is proportional to engine rating when the rotor speed remains unchanged. It follows that the dry weight of planetary type of transmission rated at 1800 hp per engine is:

Weight for 3600 hp planetary transmission $= 943(3600/3120)^{.75} = 1050$ lb

* Weight for 3600 hp split torque transmission $= 892$ lb

On the basis of transmission dry weight the split torque design therefore gives a 15% weight reduction.

* See section 19, page 66.

10. GEOMETRY OF GEAR AND SHAFT LOCATIONS

The location of the four final drive pinions around the combining gear is determined primarily by the offset and the forward position of each engine input drive flange from the main shaft. These two dimensions are predetermined from the existing test stand. In addition, small changes in the pinion positions are made so that the mesh points are out of sequence. That is, teeth on the four pinions enter mesh not simultaneously, but at intervals of one-quarter the pitch of the combining gear.

Interrelated with the position of the final drive pinions is the shaft angle of the intermediate stage bevel gears and that of the engine bevel gears. Each shaft angle has a precise value corresponding to positioning the engine input flange at its correct height, offset, and forward position relative to the main shaft. The location of gear axes and shaft angles adopted in the final design is given in the following table.

GEAR AND SHAFT LOCATION DATA

Forward shaft angle of main shaft to waterline	87°
Shaft angle of intermediate bevel gears	$88^{\circ}20'$
Shaft angle of engine bevel gears	$24^{\circ}26'30''$
Shaft angle of tail drive bevel gears	$100^{\circ}53'$
Angle between rear bevel gears and fore/aft axis	$\pm 21^{\circ}$
Angle between rear and front bevel gears	$\pm 90^{\circ}$
Height of engine bevel cone point above waterline	8.8251 in
Coordinates for engine bevel pitch cone:	
x dimension, from main shaft	32.3376 in
y dimension, from main shaft	30.000 in

Coordinates for forward final drive pinions:

x dimension, from main shaft	7.238 in
y dimension, from main shaft	18.819 in

Coordinates for rear final drive pinions:

x dimension, from main shaft	18.819 in
y dimension, from main shaft	7.224 in

11. FINAL REDUCTION STAGE

The final reduction stage has the duty of supplying torque to the main rotor shaft while contributing least weight. The use of multiple drive pinions boosts the ratio available at this critical stage as a result of allowing reductions in the diameter and facewidth of the combining gear.

With the present design there are restrictions on the choice of tooth numbers as a result of having to match the ratio in the existing test stand. It is for this reason that the 225T/31T combination is selected to give a final ratio of 7.26:1. In other circumstances, with a free choice of ratio, a slightly higher ratio would be appropriate.

A module pitch for the teeth is selected, in preference to a diametral pitch to obtain a required diameter for the gear while retaining the tooth combination already noted. Further, the mesh facewidth for the baseline design is sufficiently narrow to allow later upsizing of the transmission by increasing the facewidth of the combining gear.

Angular spacing of the pinions around the combining gear is chosen so that the four sets of pinion teeth are out of phase by one quarter of a pitch. In this way any torque pulsing effect from four teeth in identical mesh positions is avoided.

Comprehensive data for the final reduction stage gears is tabulated on pages 29 and 30.

DATA FOR FINAL STAGE GEARS

Item	Gear		Pinion(4)
No. of teeth	225		31
Speed ratio		7.2581	
Speed; rpm	258		1872.6
Center dist; in		20.1575	
Pitch; dia; in	35.4331		4.8820
Pitch & pressure angle	4 module (6.35DP)		20°
Facewidth ; in	3.75		4.00
Power, max. rated , hp	3600		900
Torque at max power , in-lb	879410		30291
Tangential load; lb		12409	
AGMA Geom. Factor	.464		.464
Compressive stress, max; lb/in^2		158700	
Bending stress, max; lb/in^2	58870		55190
K Factor; lb/in^2		771	
Scoring index (Per AGMA 2170.01)		11890	
Flash temperature; $^{\circ}F$ (Per AGMA 2170.01)		280	
Dwg. No.	ST 4012		ST 4014

Alternate design option for lower stress levels

Pressure angle		22.5°	
Compressive stress, max; lb/in^2		151350	
Bending stress, max; lb/in^2	56900		53350

TOOTH FORM DATA FOR FINAL STAGE GEARS

Item	Gear	Pinion (4)
No of teeth	225	31
Pitch dia; in	35.4331	4.8820
Outside dia; in	35.686	5.259
Addendum; in	0.126	0.189
Addendum shift; in	-.062	+.062
Base dia; in	33.296	4.587
NCT at pitch dia; in	0.225	0.270
NCT at outside dia; in	0.131	0.108
J Factor	0.464	0.464
Whole depth factor	2.400	2.400
Action line length; in	0.811	0.811
Contact ratio	1.7447	1.7447
I Factor	0.139	0.139

11.2 Combining Gear with Increased Rating

Dimensions chosen for the baseline design of combining gear are such as to allow for growth of the engines without any change in housing dimensions. An assumption is that the T700 engine will experience growth from its initial 1500 hp to about 2250 hp. The combining gear can accommodate this growth by combinations of small increases in facewidth, pressure angle and operating stresses.

A general expression relating the torque carried with pinion geometry and tooth compressive factor K is: $R = 0.5\left\{-1 + \left[1 + 4neKD^3/(2T)\right]^{0.5}\right\}$

where R = final stage reduction ratio; $R = 225/31$

n = number of final drive pinions; $n = 4$

e = mesh facewidth/pinion pitch diameter; $e = F/d$

K = contact stress factor; $K = w(R + 1)/(FdR)$; lb/in^2

w = tooth tangential load, lb

D = pitch diameter of combining gear; $D = 35.433$ in

T = torque capacity of combining gear, in-lb

P = power capacity of combining gear, hp

F = mesh facewidth, in

d = pinion pitch dia, in

Inserting the constants R, n and D appropriate to the baseline design and a main shaft speed of 258 rpm results in the equations:

$T = 1484.41\ e\ K$ and $P = 6.0766\ e\ K$

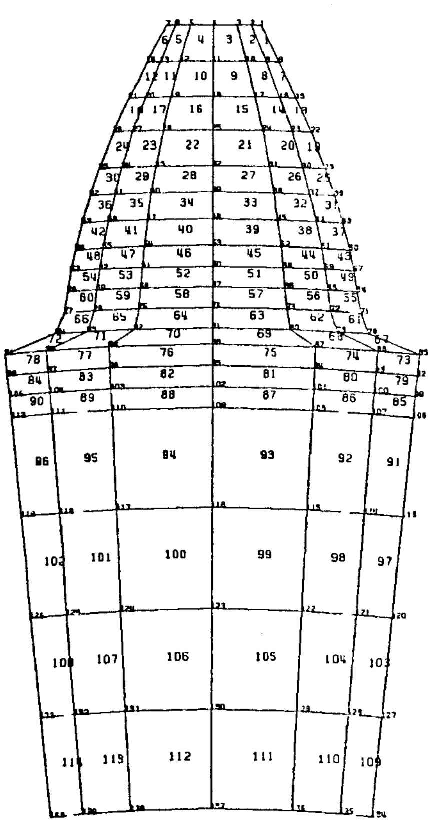
Changes to the final reduction train as a means of accommodating the power from each T700 engine rising from 1500 hp to 2250 hp are as tabulated on page 32. These changes in dimensions are appropriate to conventional involute teeth; a separate investigation into the applicability of high contact ratio teeth is also relevant, but has not been undertaken in the present study.

COMBINING GEAR DIMENSIONS FOR HIGHER RATINGS

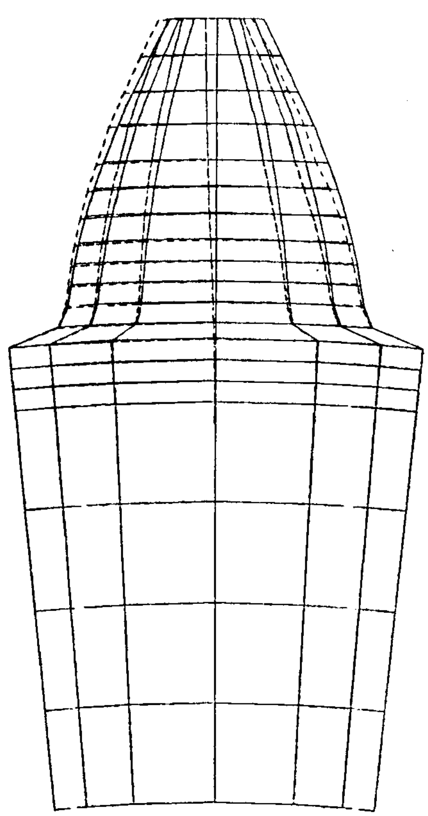
Combined power 2-T700 engines	3600 hp	4000 hp	4500 hp
Combining gear torque capacity, in-lb	879418	977132	1099273
Contact stress K factor, lb/in^2	771	804	823
Facewidth/pinion ratio e	0.77	0.82	0.90
Actual facewidth, in	3.75	4.00	4.40
Pressure angle, degrees	20	22.5	25
Compressive stress; lb/in^2	158700	154480	158190
Bending stress; lb/in^2	58870	57480	59160

Analysis of the gear tooth bending and compressive stresses, and the effect of support rim thickness, was undertaken by a finite element program. Typical results for the mesh pattern, tooth deflection and stress contours are as shown in the printouts below for a 6.35 DP gear.

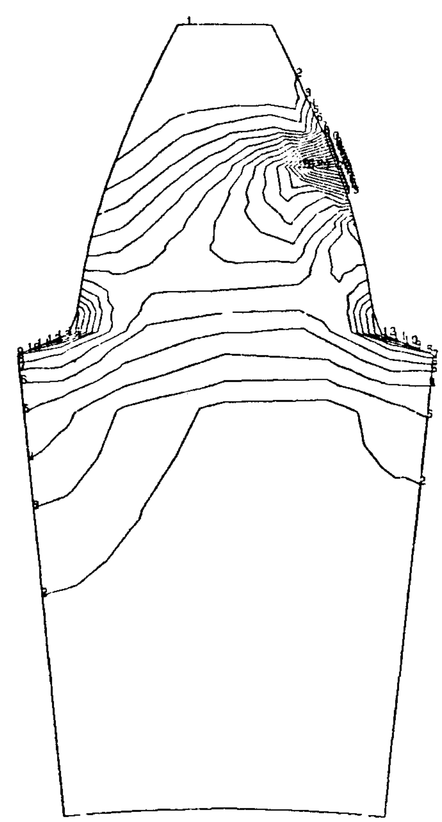
Mesh pattern



Tooth deflection



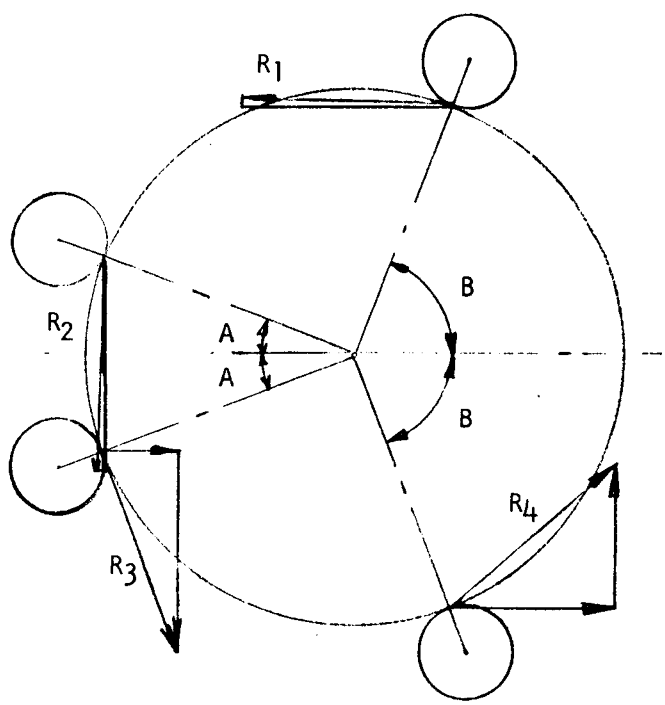
Stress contours



11.3 Net Load on Combining Gear

A consequence of the four final drive pinions not being spaced symmetrically is that the combining gear experiences a resultant radial load that is reacted on to the main shaft bearings. A second reason for a radial load is that the tooth load from one pinion, that associated with the tail drive gear, is reduced as a result of it carrying half-engine power less the tail drive power.

The resultant tooth load at the four pinion meshes is denoted by R_1 through R_4 as shown in the figure below. Horizontal and vertical components of these resultants then are added in order to determine, first, the net horizontal and vertical forces, and second, the single resultant load on the gear.



Components of tooth forces acting on the combining gear

With the gear pressure angle $\emptyset$, the horizontal components are:

From R_1: $-R_1 \cos (90 - B - \emptyset)$

From R_2: $+R_2 \cos (90 + A - \emptyset)$

From R_3: $+R_3 \cos (90 - A - \emptyset)$

From R_4: $+R_4 \sin (B - \emptyset)$

Vertical components:

From R_1: $+R_1 \sin (90 - B - \emptyset)$

From R_2: $-R_2 \sin (90 + A - \emptyset)$

From R_3: $-R_3 \sin (90 - A - \emptyset)$

From R_4: $+R_4 \cos (B - \emptyset)$

With the present design, at max. rated power of 1800 hp from each engine:

$A = 21^\circ$ $B = 69^\circ$ $\emptyset = 20^\circ$

$R_1 = R_2 = R_4 = 13205$ lb $R_3 = 7630$ lb

Horizontal component = +1539 lb

Vertical component = -10068 lb

Net resultant force on gear = 10185 lb

12. INTERMEDIATE STAGE BEVEL GEARS

Each of the four sets of intermediate stage bevel gears is rated at 900 hp. Identical gear diameters and tooth proportions are accordingly adopted for the forward and rear bevel sets in each driveline. Outline data for the gears is summarized in the following table.

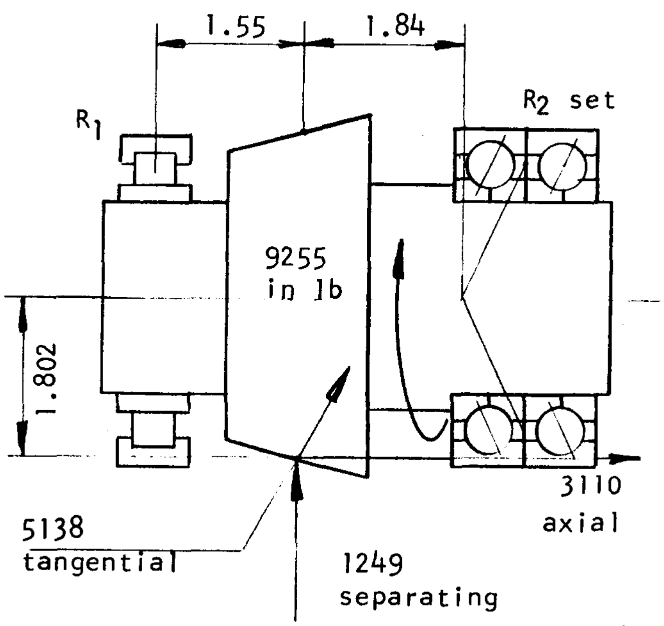
Data for Intermediate Stage Bevel Gears

Item	Pinion		Gear
No. of teeth	22		72
Ratio		3.27273	
Diametral pitch		5.3333	
Facewidth, in		1.800	
Pressure angle		20°	
Shaft angle		$88^{\circ}20'$	
Modified contact ratio		2.22	
Pitch diameter, in	4.125		13.500
Pitch angle	$16^{\circ}51$		$71^{\circ}29'$
Mean spiral angle		27°	
Mode of operation	driver		driven
Hand, rotation	LH,CW		RH,CCW
Max. power, hp		900	
Torque, in-lb	9255.6		30291
Speed, rpm	6128		1872.6
Compressive stress; lb/in^2		212700	
Bending stress; lb/in^2	28400		28500
Mean working dia; in	3.603		11.792
Scoring temperature (Gleason)	$\Delta T = 225^{\circ}F$		$T_f=200^{\circ}+225^{\circ}=425^{\circ}F$

Item	Pinion	Gear
Tangential force; lb	5138	5138
Axial force; lb	3110	1151
Separating force; lb	1249	3150
Gear web angle	arctan (1151/3150) = 20.07°	

12.1 Bearing Loads and Life for Rear Bevel Pinions

Each of the two intermediate rear bevel pinions is straddle-mounted on bearings that are spaced as shown below. Tooth forces and the bearing loads corresponding to a 900 hp rating are given on the diagram. These loads are used for determining bearing life.



At a 900 hp rating:

Pinion torque,	9255 in-lb
Tangential tooth load, T	5138 lb
Axial tooth load, A	3110 lb
Separating tooth load, S	1249 lb
Front brg. load, max.	R_1 = 2744 lb
Rear brg. loads, max. on set	R_2 radial = 3005 lb; R_2 axial = 3110 lb

As noted on p. 4, bearing load prorate factor of 0.60 and B_{10} material life factor of 5 are used for the calculations of bearing life throughout this report.

All pertinent bearing data is shown in Section 18, Bearing Data Summary, pp. 63 and 64.

All bearing locations are shown on Figure 6A, p. 65.

Bearing Sizing and Life for Rear Pinion

Front bearing R_1	
Speed, rpm	6128
Max load, lb	2744
0.6 prorated load, lb	1646
Bearing size, mm	40 x 90 x 23
Dynamic capacity, lb	17100
Computer calc. B_{10} life, hours (material life factor of 5)	9200
Rear bearing set R_2 (tandem arrangement)	
Speed, rpm	6128
Max loads on pair, lb	3005 radial, 3110 axial
0.6 prorated loads, lb	1803 radial, 1866 axial
Tandem split inner race pair, mm	60 x 130 x 62
Dynamic capacity of one brg, $\alpha = 25^{\circ}$, lb	C = 18370
Dynamic capacity of pair, $C(2)^{0.7}$, lb	$C_2 = 29840$
Bearing constants	X = .41; Y = .87
* $P = XP_r + YP_a = .41(1803) + .87(1866)$, lb	2363
Computer calc. B_{10} life, hours (material life factor of 5)	27400

* P_r denotes the prorated radial load for the thrust bearing

P_a denotes the prorated axial load for the thrust bearing

X and Y are handbook bearing constants

Gleason
MACHINE DIVISION

SPIRAL BEVEL GEAR DIMENSIONS   NO. S008340   FORM M   DATE 12/14/83   TIME 10:57

TRANSMISSION RESEARCH-CLEVELAND

	PINION	GEAR
NUMBER OF TEETH	22	72
PART NUMBER		
DIAMETRAL PITCH		5.333
FACE WIDTH	1.800"	1.800"
PRESSURE ANGLE	20D 0M	
SHAFT ANGLE	88D 20M	
TRANSVERSE CONTACT RATIO		1.291
FACE CONTACT RATIO		1.807
MODIFIED CONTACT RATIO		2.220
OUTER CONE DISTANCE		7.118"
MEAN CONE DISTANCE		6.218"
PITCH DIAMETER	4.125"	13.500"
CIRCULAR PITCH	0.589"	
WORKING DEPTH	0.330"	
WHOLE DEPTH	0.366"	0.366"
CLEARANCE	0.035"	0.035"
ADDENDUM	0.236"	0.095"
DEDENDUM	0.130"	0.271"
OUTSIDE DIAMETER	4.576"	13.560"
FACE ANGLE JUNCTION DIAMETER		
THEORETICAL CUTTER RADIUS	6.004"	
CUTTER RADIUS	6.000"	
CALC. GEAR FINISH. PT. WIDTH		0.140"
GEAR FINISHING POINT WIDTH		0.140"
ROUGHING POINT WIDTH	0.050"	0.130"
OUTER SLOT WIDTH	0.070"	0.140"
MEAN SLOT WIDTH	0.076"	0.140"
INNER SLOT WIDTH	0.070"	0.140"
FINISHING CUTTER BLADE POINT	0.040"	0.080"
STOCK ALLOWANCE	0.020"	0.010"
MAX. RADIUS-CUTTER BLADES	0.046"	0.103"
MAX. RADIUS-MUTILATION	0.068"	0.118"
MAX. RADIUS-INTERFERENCE	0.042"	0.083"
CUTTER EDGE RADIUS	0.035"	0.075"
CALC. CUTTER NUMBER	3	7
MAX. NO. BLADES IN CUTTER		16.990
CUTTER BLADES REQUIRED	STD DEPTH	STD DEPTH
GEAR ANGULAR FACE - CONCAVE		18D 43M
GEAR ANGULAR FACE - CONVEX		19D 59M
GEAR ANGULAR FACE - TOTAL		21D 11M
DRAWING NOS.	ST-4058	ST-4133
	ST-4039	ST-4020

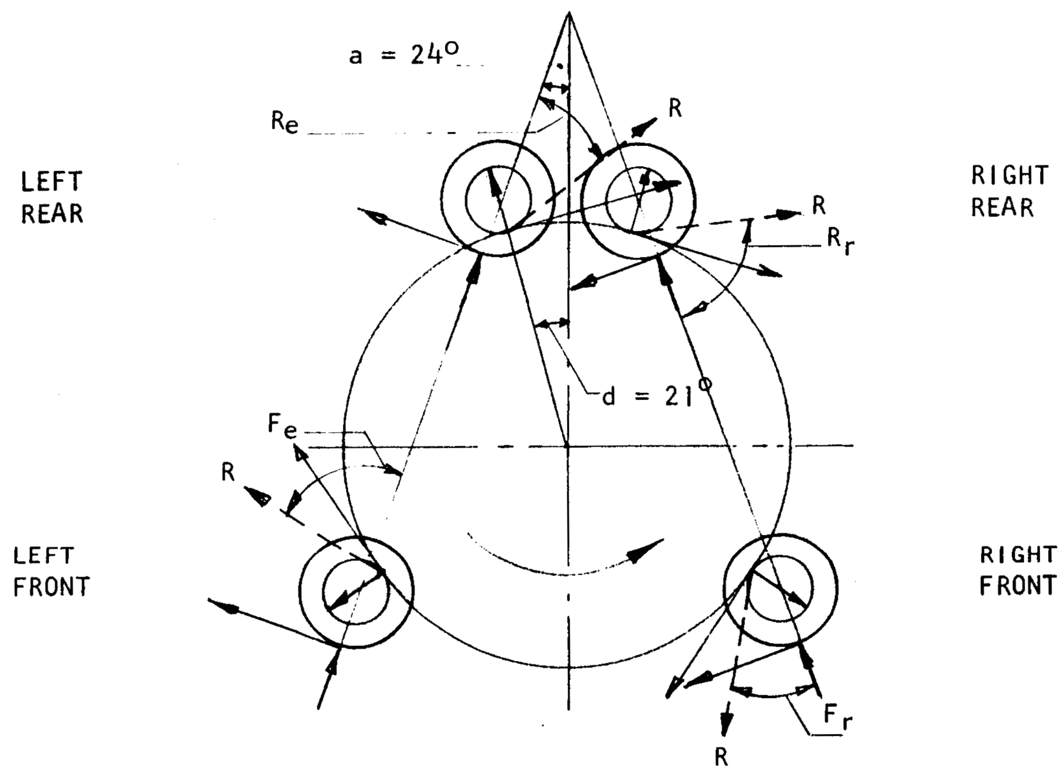
		PINION		GEAR
PITCH APEX TO CROWN		6.745"		2.170"
FACE ANG JUNCT TO PITCH APEX				
MEAN CIRCULAR THICKNESS		0.335"		0.172"
OUTER NORMAL TOP LAND		0.127"		0.088"
MEAN NORMAL TOP LAND		0.135"		0.096"
INNER NORMAL TOP LAND		0.144"		0.090"
PITCH ANGLE		16D 51M		71D 29M
FACE ANGLE OF BLANK		19D 36M		72D 36M
INNER FACE ANGLE OF BLANK				
ROOT ANGLE		15D 44M		68D 44M
DEDENDUM ANGLE		1D 7M		2D 46M
OUTER SPIRAL ANGLE				32D 29M
MEAN SPIRAL ANGLE				27D 0M
INNER SPIRAL ANGLE				21D 36M
HAND OF SPIRAL	LH		RH	
DRIVING MEMBER	PIN			
DIRECTION OF ROTATION-DRIVER	CW			
OUTER NORMAL BACKLASH	MIN	0.005"	MAX	0.007"
DEPTHWISE TOOTH TAPER	TRL			
GEAR TYPE				GENERATED
FACE IN PERCENT OF CONE DIST				25.287
DEPTH FACTOR - K				
ADDENDUM FACTOR - C1				
GEOMETRY FACTOR-STRENGTH-J		0.3385		0.3379
STRENGTH FACTOR-Q		2.79241		0.85488
EDGE RADIUS USED IN STRENGTH		0.035"		0.075"
CUTTER RADIUS FACTOR - KX		1.006		
FACTOR	MN	0.8593		
STRENGTH BALANCE DESIRED	STRS			
STRENGTH BALANCE OBTAINED	STRS			0.002
GEOMETRY FACTOR-DURABILITY-I		0.1176		
DURABILITY FACTOR-Z		2107.76		1165.11
GEOMETRY FACTOR-SCORING-G		0.003321		
SCORING FACTOR - X		0.2218		
ROOT LINE FACE WIDTH		1.800"		1.800"
PROFILE SLIDING FACTOR		0.00381		0.00608
RATIO OF INVOLUTE/MEAN CONE		1.409		
AXIAL FACTOR-DRIVER CW	OUT	0.336	OUT	0.038
AXIAL FACTOR-DRIVER CCW	IN	0.205	OUT	0.093
SEPARATING FACTOR-DRIVER CW	SEP	0.135	SEP	0.104
SEPARATING FACTOR-DRIVER CCW	SEP	0.299	ATT	0.060
DUPLEX SUM OF DEDENDUM ANG		3D 52M		
ROUGHING RADIAL		6.387"		
INPUT DATA	KTT M			
INPUT DATA	CUTM			1

DIMENSIONS FOR INTERMEDIATE STAGE BEVEL GEARS

13.1 LOADS ON INTERMEDIATE BEVEL GEAR & SPUR PINION ASSEMBLIES

Angular position of resultant load R at the four pinions that drive the combining gear. In all cases pressure angle $\phi = 22.5^{o}$.

Position of loads on the four bevel gear and spur pinion assemblies



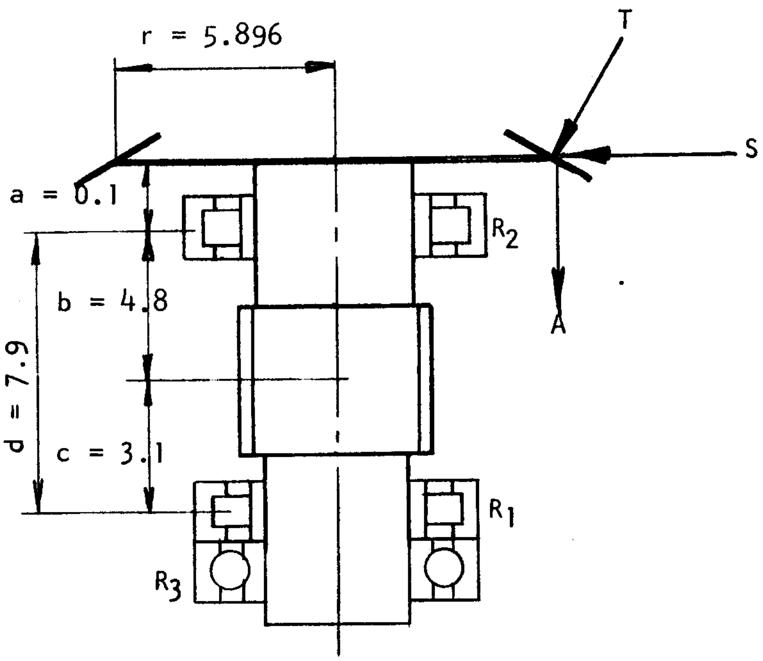
Rear left: $R_e = 180 - [a + d + 90 + \phi] = 22.5^{o}$

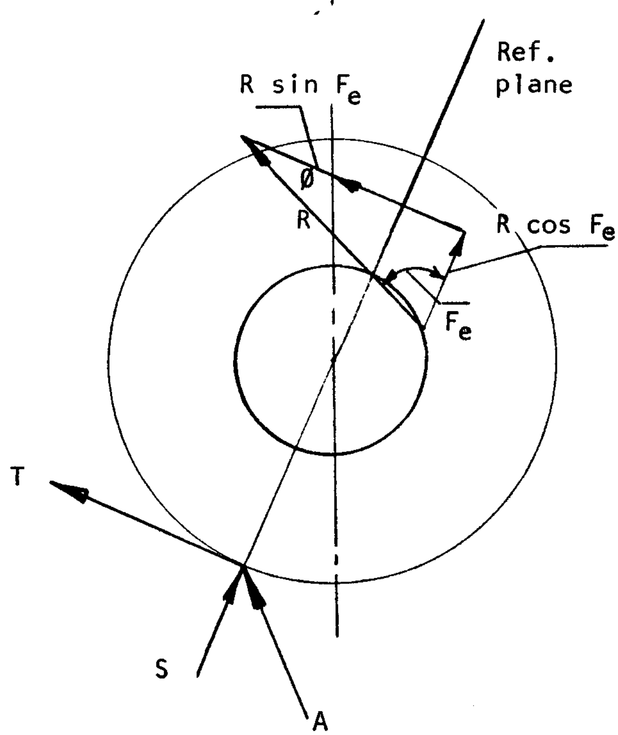
Front left: $F_e = 90 + \phi - (a + d) = 67.5^{o}$

Rear right: $R_r = 90 + \phi - (a + d) = 67.5^{o}$

Front right: $F_r = 180 - (a + d) - 90 - \phi = 22.5^{o}$

13.2 BEARING LOADS FOR INTERMEDIATE GEAR & FINAL PINION





(a) Front Left Set

F_e = 67.5° at 900 hp rating

T = 5138 lb

S = 3150 lb

A = 1151 lb

r = 5.896 in   ø = 22.5°

Spur pinion torque = 30291 in-lb

W_t = 12409 lb

R = $W_t/\cos ø$ = 13431 lb

Loads in Ref. plane

From axial load A, Moment = Ar

$\swarrow R_2 = 859$

$\nearrow R_1 = 859$

From separating load S

$\nearrow R_2 = 3190$

$\swarrow R_1 = 40$

From pinion load, resultant R

$\nearrow R_2 = 2017$

$\nearrow R_1 = 3123$

Summing loads in Ref. plane

$\nearrow R_2 = -859 + 3190 + 2017 = 4348$ [lb]

$\nearrow R_1 = 859 - 40 + 3123 = 3942$ [lb]

(b) Loads Normal to Ref. Plane

From bevel tangential load, T = 5138 lb

$\nwarrow R_2 = 5203$ $\qquad\qquad$ $\searrow R_1 = 65$

From spur pinion load of $R \sin F_e$ where $R \sin F_e = 13431 \sin 67.5^\circ = 12408$ [lb]:

$\nwarrow R_2 = 4869$ $\qquad\qquad$ $\nwarrow R_1 = 7539$

Summing loads normal to Ref. plane

$$\nwarrow R_2 = 5203 + 4869 = 10072 \text{ (lb)}$$

$$\nwarrow R_1 = -65 + 7539 = 7474 \text{ (lb)}$$

From vector sum of loads, R_2 and R_1 are:

$$R_2^{\,2} = 4348^2 + 10072^2 \qquad R_2 = 10970 \text{ (lb)}$$

$$R_1^{\,2} = 3942^2 + 7474^2 \qquad R_1 = 8450 \text{ (lb)}$$

Bearing sizing and life

	Upper (roller) bearing R_2	Lower (roller) bearing R_1	Ball Thrust bearing R_3
Speed, n, rpm	1872.6	1872.6	1872.6
Max, load, at 900 hp, lb	10970 radial	8450 radial	1151 axial
0.6 prorated load, P, lb	6582	5070	691
Bearing size, mm	95x170x32	95x170x32	105x145x20
Dynamic capacity, C, lb	44300	44300	11200
C/P value, prorated	6.730	8.737	--
* C/P_e value, prorated	--	--	18.636
B_{10} life; $10^6(C/P)^{3.33}/60n$, hours	5100	12100	--
B_{10} life; $10^6(C/P_e)^3/60n$, hours	--	--	57600
B_{10} life; material life factor of 5, hours	25500	60500	288000

$P_e = XP_r + YP_a$

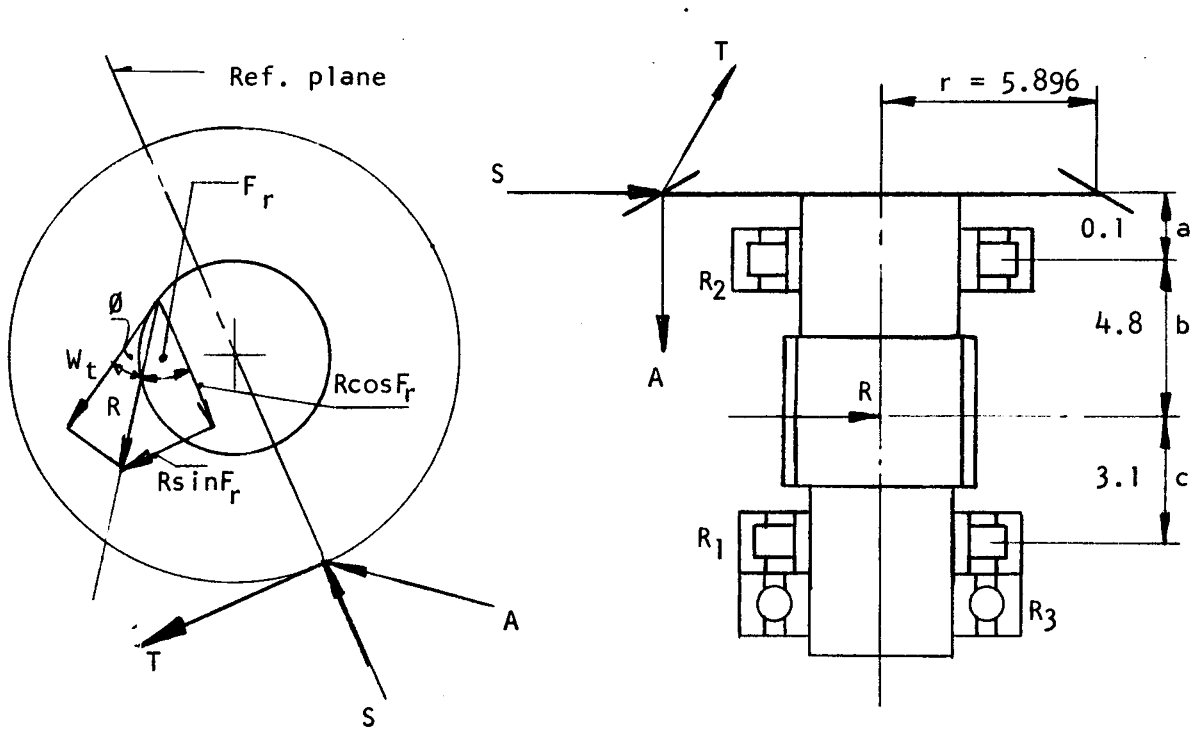
* P_e denotes the equivalent prorated load for the thrust brg.
P_r denotes the prorated radial load

$P_e = (0.87)(691) = 601$ lb

P_a denotes the prorated axial load

X and Y are handbook bearing constants (X=0.41; Y=0.87)

(b) <u>Front Right Set</u>



W_t = 12409 lb; R = 13431 lb; Ø = 22.5°

F_r = front right angle from Ref. plane = 22.5°

At 900 hp; T = 5138; S = 3150; A = 1151 [lb]

<u>Loads in Ref. plane</u>

From axial load A, Moment = Ar

R_2 = 859

R_1 = 859

From separating load S

R_2 = 3190

R_1 = 40

From pinion load, resultant R

R_2 = 4869

R_1 = 7539

Summing loads in Ref. plane

$R_2 = 859 - 3190 + 4869 = 2538$ [lb]

$R_1 = -859 + 40 + 7539 = 6720$ [lb]

Loads normal to Ref. plane

From bevel tangential load: T = 5138 lb

$R_2 = 5203$

$R1 = 65$

From spur pinion load of R sin F_r , where R sin F_r = 13431 sin 22.5° = 5140 lb:

$R_2 = 2017$

$R_1 = 3123$

Summing loads normal to Ref. plane

$R_2 = 5203 + 2017 = 7220$ [lb]

$R_1 = 3123 - 65 = 3058$ [lb]

Vector sum of loads on R_2 and R_1

$R_2^2 = 7220^2 + 2538^2$ $\qquad R_2 = 7653$ [lb]

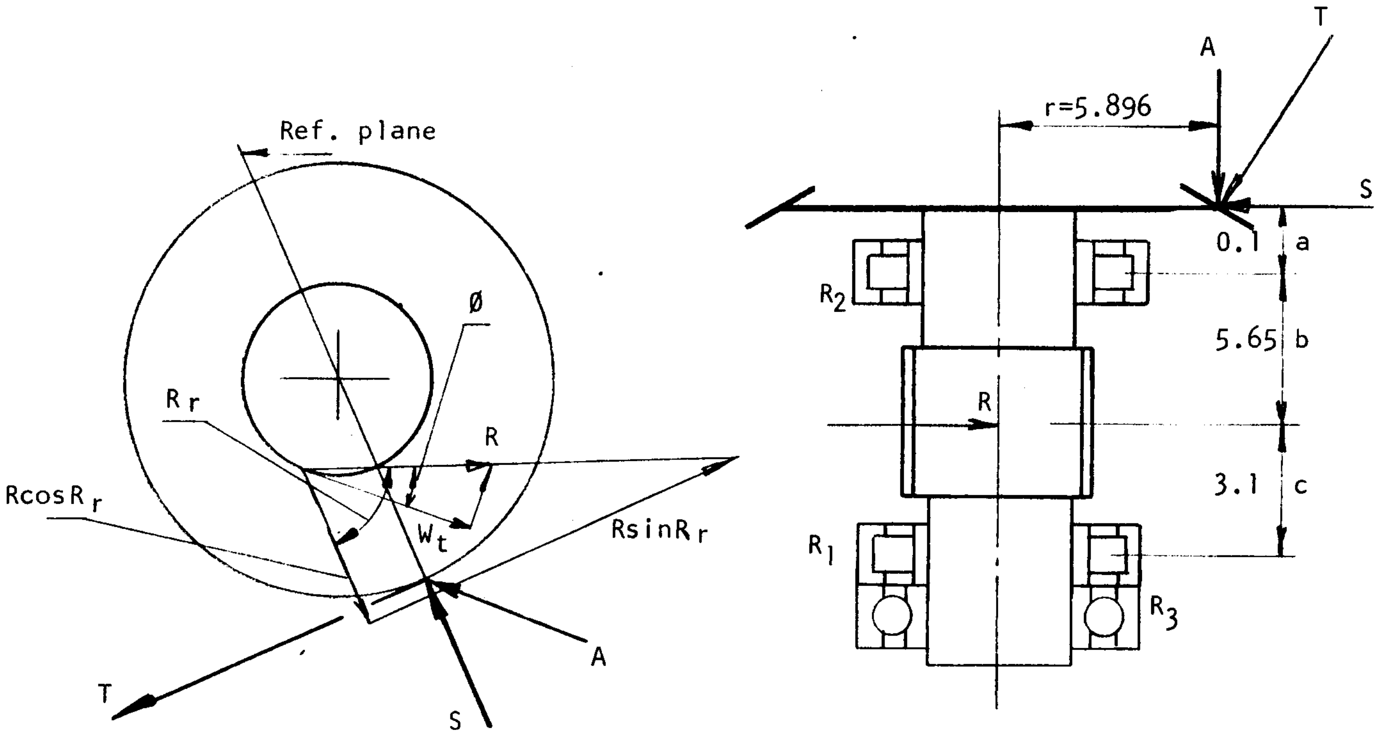
$R_1^2 = 3058^2 + 6720^2$ $\qquad R_1 = 7383$ [lb]

Bearing sizing and life

	Upper Bearing R_2	Lower Bearing R_1
Speed, rpm	1872.6	1872.6
Max load, at 900 hp, lb	7653	7383
0.6 prorated load P, lb	4592	4430
Bearing size , mm	95x170x32	95x170x32
Dynamic capacity C, lb	44300	44300
C/P value, prorated	9.647	10.00
B_{10} life; $10^6(C/P)^{3.33}/(60n)$, hrs.	16900	19000
B_{10} life; material life factor of 5 hrs.	84500	95000

Calculations and sizing for the ball thrust bearing R_3 (part No. 4023) are identical with the thrust bearing calculations shown on page 41.

(c) Rear Right Set



R_r = rear angle from Ref. plane = 67.5°

W_t = 12409 lb ; R = 13431 lb ; Ø = 22.5°

R_r = angle from Ref. plane for right rear spur pinion at 900 hp;

S = 3150; A = 1151 ; T = 5138 ; [lb]

Loads in Ref. plane

From axial load A, Moment = Ar

R_2 = 776

R_1 = 776

From separating load S

R_2 = 3186

R_1 = 36

From pinion load, resultant R

R_2 = 1821

R_1 = 3319

Summing loads in Ref. plane

$R_2 = -776 + 3186 - 1821 = 589$ [lb]

$R_1 = -776 + 36 - 3319 = 2579$ [lb]

Loads normal to Ref. plane: T = 5138 lb

From bevel tangential load

$R_2 = 5197$

$R_1 = 59$

From spur pinion load of $R \sin R_r$, where $R \sin R_r = 13431 \sin 67.5^\circ = 12409$ lb,

$R_2 = 4396$

$R_1 = 8013$

Summing loads normal to Ref. plane

$R_2 = 5197 - 4396 = 801$ [lb]

$R_1 = 8013 + 59 = 8072$ [lb]

Vector sum of loads on R_2 and R_1

$R_2^2 = 589^2 + 801^2$ $\qquad R_2 = 994$ [lb]

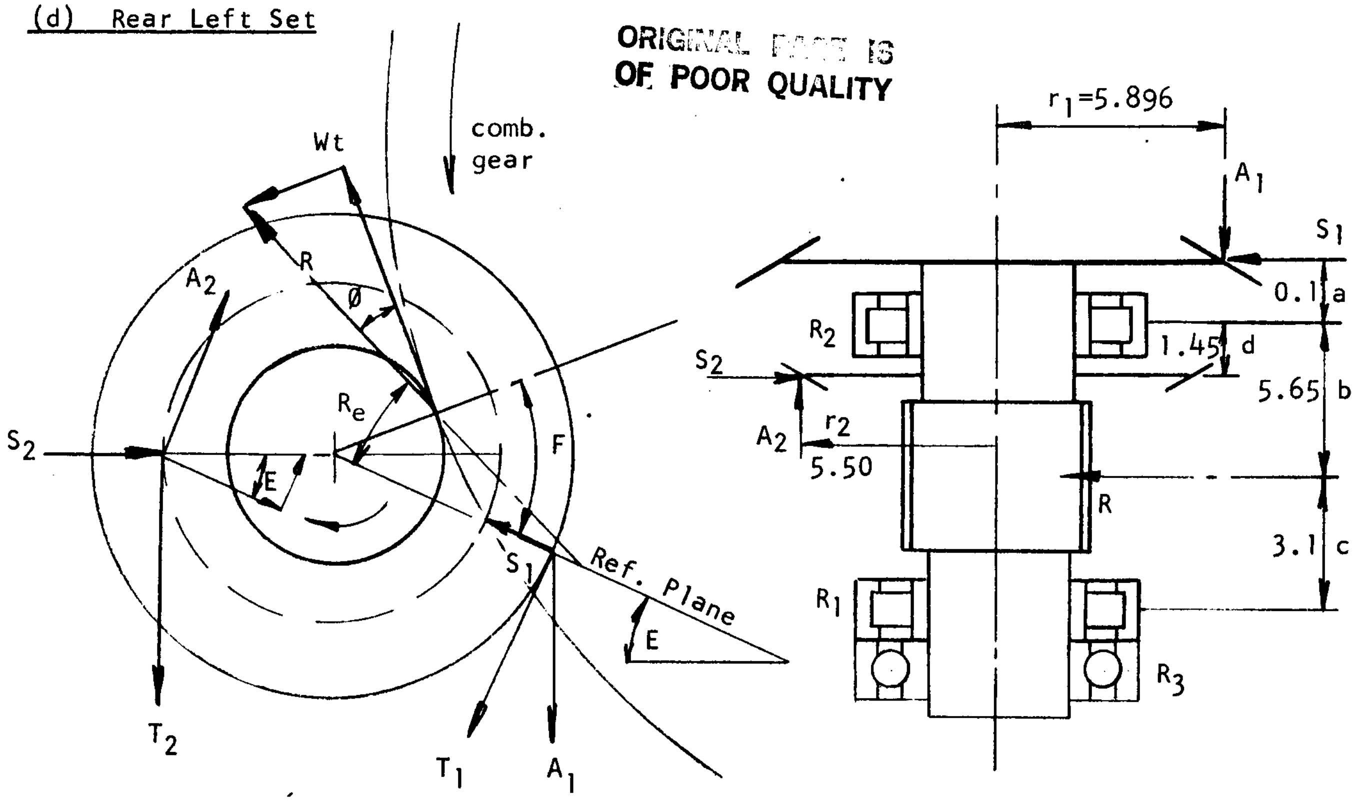
$R_1^2 = 2579^2 + 8072^2$ $\qquad R_1 = 8474$ [lb]

Bearing sizing and life

	Upper Brg. R_2	Lower Brg. R_1
Speed, rpm	1872.6	1872.6
Max load, at 900 hp, lb	994	8474
0.6 prorated load P, lb	596	5084
Bearing size, mm	95 x 170 x 32	95 x 170 x 32
Dynamic capacity C, lb	44300	44300
C/P value, prorated	74.3	8.713
B_{10} life; $10^6(C/P)^{3.33}/(60n)$, hrs.	15×10^6	12000
B_{10} life; material life factor of 5 hrs.	75×10^6	60000

Calculations and sizing for the ball thrust bearing R_3 (part No. 4023) are identical with the thrust bearing calculations shown on page 41.

(d) Rear Left Set



R_e = angle between Ref. plane and pinion/gear centerline

$R_e = 90 - Ø - 45^o = 22.5^o$ $\qquad$ $E = 24^o$; $F = 45^o$

At 900 hp from main bevel pinion $\qquad$ $T_1 = 5138$; $S_1 = 3150$; $A_1 = 1151$; [lb]

AT 380 hp to tail drive gear $\qquad$ $T_2 = 2680$; $S_2 = 3349$; $A_2 = 1698$; [lb]

and 520 hp to spur pinion $\qquad$ R = 7760 lb; $Ø = 22.5^o$

Loads in Ref. plane

From axial load A_1; Moment = $A_1 r_1$, $R_1 = 776$, $R_2 = 776$

From separating load S_1

$R_1 = 36$; $\quad R_2 = 3186$

From spur pinion load R

$R_1 = 4629$; $\quad R_2 = 2540$

From component of sep. load S_2; $\searrow R_1 = 507$, $\searrow R_2 = 2570$

From component of tan. load T_2; $\searrow R_1 = 181$, $\searrow R_2 = 909$

From component of A_2 moment; $\searrow R_1 = 975$, $\searrow R_2 = 975$

Summing loads in Ref. plane

$$\searrow R_1 = 776 - 36 + 4629 - 507 - 181 - 975 = 3706\,[\mathrm{lb}]$$

$$\searrow R_2 = -776 + 3186 + 2540 - 2570 - 909 + 975 = 2446\,[\mathrm{lb}]$$

Loads normal to Ref. plane

From tang. load T_1; $\nearrow R_1 = 65$ $\nearrow R_2 = 5197$

From spur pinion load R; $\nearrow R_1 = 1918$ $\nearrow R_2 = 1052$

From compt. of sep. load S_2; $\nearrow R_1 = 226$ $\nearrow R_2 = 1136$

From compt. pf tan. load T_2; $\nearrow R_1 = 406$ $\nearrow R_2 = 2043$

Summing loads normal to Ref. plane

$$\nearrow R_1 = 65 + 1918 + 226 - 406 = 1803\,[\mathrm{lb}]$$

$$\nearrow R_2 = 5197 - 1052 - 1136 + 2043 = 5052\,[\mathrm{lb}]$$

Vector sum of loads on R_1 and R_2 bearings

$$R_1^2 = 3706^2 + 1803^2 \; ; \; R_1 = 4121\,[\mathrm{lb}] \; ; \; R_2^2 = 2446^2 + 5052^2 \; ; \; R_2 = 5613\,[\mathrm{lb}]$$

Bearing sizing and life

	Upper Bearing R_2	Lower Bearing R_1
Speed; rpm	1872.6	1872.6
Max load at 900 hp, lb	5613	4121
0.6 prorated load P, lb	3368	2473
Bearing size, mm	95x170x32	95x170x32
Dynamic capacity C, lb	44300	44300
C/P value, prorated	13.15	17.91
B_{10} life; $10^6(C/P)^{3.33}(60n)$, hrs.	47400	132000
B_{10} life; material life factor of 5 hrs.	237000	660000

Calculations and sizing for the ball thrust bearing R_3 (part No. 4023) are identical with the thrust bearing calculations shown on page 41.

14. ENGINE REDUCTION BEVEL STAGE

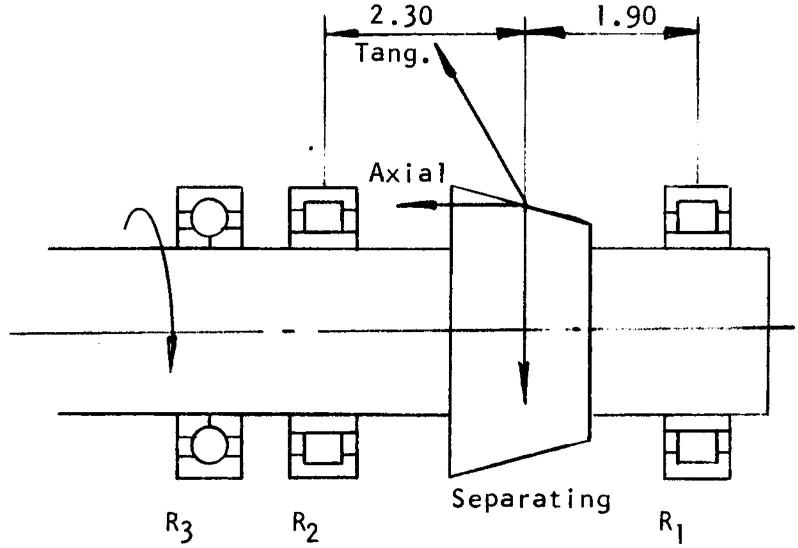
Bevel gears at the input section provide a speed reduction of 3.41:1 from an engine speed of 20909 rpm. Each gear is carried on three bearings. In the case of the input pinion all radial loads are carried on two roller bearings, leaving the ball bearing to carry only axial thrust. Each bearing has individual jets for the supply of lubricant.

Tooth stresses and the scoring factor, at maximum power, are arranged to be within those experienced in production designs.

ITEM	PINION		GEAR
No. of teeth	34		116
Speed, rpm	20909		6128.4
Diametral pitch, in		9.872	
Facewidth, in		1.800	
Pressure angle/shaft angle, degrees		20/24.441	
Modified contact ratio		2.405	
Pitch dia, in	3.444		11.750
Pitch angle, degrees	5.468		18.973
Mean spiral angle, degrees		18	
Mode of operation	driver		driven
Hand; rotation	LH;CW		RH;CCW
Mean working dia, in	3.2724		11.1648
Power, max., hp		1800	
Torque, in-lb	5426		18511
Bending stress, lb/in^2	23650		23320
Compressive stress, lb/in^2		201300	
Scoring index (Gleason)		$\Delta T = 216^oF$ $T_f = 200 + \Delta T = 416^oF$	

Tangential force, lb		3315	
Axial force, lb	+1194		-610.9
Separating force, lb	1161		1555
Gear web angle, degrees		21.45	

Tooth Loads and Bearing Loads for Input Pinion



Bearing life for engine bevel pinion bearings

For comparative purposes the bearing life of the high-speed bearings is calculated by both the catalogue method and a computerized method that includes lubrication and material factors. Data on loads, speeds, bearing sizes, capacity and B_{10} life is as given in the following table for the 1800 hp rating.

Item	Front Bearing R_1	Rear Bearing R_2	Thrust Bearing R_3
Speed, rpm	20909	20909	20909
Max load, lb	1797	1823	1194
0.6 prorated load, lb	1078	1094	716
Bearing size, mm	60x110x22	60x110x22	60x110x22
Dynamic capacity C, lb	21400	21400	11180
Computer calc. B_{10} life, hrs. (material life factor of 5)	19300	18400	50500

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MACHINE DIVISION

SPIRAL BEVEL GEAR DIMENSIONS NO. S008339A FORM M DATE 3/19/84 TIME 10:16

TRANSMISSION RESEARCH-CLEVELAND #

	PINION	GEAR
NUMBER OF TEETH	34	116
PART NUMBER		
DIAMETRAL PITCH		9.872
FACE WIDTH	1.800"	1.800"
PRESSURE ANGLE	20D 0M	
SHAFT ANGLE	24D 27M	
TRANSVERSE CONTACT RATIO		1.424
FACE CONTACT RATIO		1.939
MODIFIED CONTACT RATIO		2.405
OUTER CONE DISTANCE		18.066"
MEAN CONE DISTANCE		17.166"
PITCH DIAMETER	3.444"	11.750"
CIRCULAR PITCH	0.318"	
WORKING DEPTH	0.171"	
WHOLE DEPTH	0.190"	0.190"
CLEARANCE	0.019"	0.019"
ADDENDUM	0.114"	0.057"
DEDENDUM	0.076"	0.133"
OUTSIDE DIAMETER	3.671"	11.858"
FACE ANGLE JUNCTION DIAMETER		
THEORETICAL CUTTER RADIUS	9.000"	
CUTTER RADIUS	9.000"	
CALC. GEAR FINISH. PT. WIDTH		0.093"
GEAR FINISHING POINT WIDTH		0.090"
ROUGHING POINT WIDTH	0.040"	0.080"
OUTER SLOT WIDTH	0.054"	0.090"
MEAN SLOT WIDTH	0.055"	0.090"
INNER SLOT WIDTH	0.054"	0.090"
FINISHING CUTTER BLADE POINT	0.030"	0.050"
STOCK ALLOWANCE	0.014"	0.010"
MAX. RADIUS-CUTTER BLADES	0.031"	0.060"
MAX. RADIUS-MUTILATION	0.057"	0.085"
MAX. RADIUS-INTERFERENCE	0.021"	0.043"
CUTTER EDGE RADIUS	0.020"	0.040"
CALC. CUTTER NUMBER	0	0
MAX. NO. BLADES IN CUTTER		
CUTTER BLADES REQUIRED	STD DEPTH	STD DEPTH
GEAR ANGULAR FACE - CONCAVE		
GEAR ANGULAR FACE - CONVEX		
GEAR ANGULAR FACE - TOTAL		

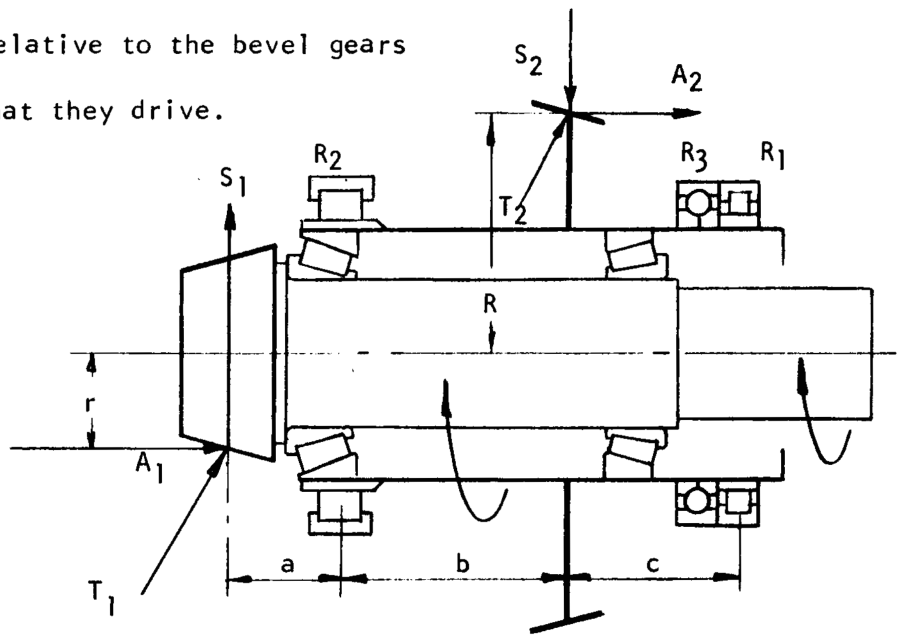
		PINION		GEAR
PITCH APEX TO CROWN		17.973"		17.065"
FACE ANG JUNCT TO PITCH APEX				
MEAN CIRCULAR THICKNESS		0.186"		0.111"
OUTER NORMAL TOP LAND		0.088"		0.062"
MEAN NORMAL TOP LAND		0.089"		0.065"
INNER NORMAL TOP LAND		0.090"		0.062"
PITCH ANGLE		5D 28M		18D 59M
FACE ANGLE OF BLANK		5D 52M		19D 11M
INNER FACE ANGLE OF BLANK				
ROOT ANGLE		5D 16M		18D 35M
DEDENDUM ANGLE		0D 12M		0D 24M
OUTER SPIRAL ANGLE				23D 1M
MEAN SPIRAL ANGLE				18D 0M
INNER SPIRAL ANGLE				12D 54M
HAND OF SPIRAL	LH		RH	
DRIVING MEMBER	PIN			
DIRECTION OF ROTATION-DRIVER	CW			
OUTER NORMAL BACKLASH	MIN	0.003"	MAX	0.005"
DEPTHWISE TOOTH TAPER	TRL			
GEAR TYPE				GENERATED
FACE IN PERCENT OF CONE DIST				9.964
DEPTH FACTOR - K				
ADDENDUM FACTOR - C1				
GEOMETRY FACTOR-STRENGTH-J		0.4535		0.4598
STRENGTH FACTOR-Q		3.96237		1.14539
EDGE RADIUS USED IN STRENGTH		0.020"		0.040"
CUTTER RADIUS FACTOR - KX		1.089		
FACTOR	MN	0.7447		
STRENGTH BALANCE DESIRED	STRS			
STRENGTH BALANCE OBTAINED	TPLD			- 0.011
GEOMETRY FACTOR-DURABILITY-I		0.1104		
DURABILITY FACTOR-Z		2605.64		1410.67
GEOMETRY FACTOR-SCORING-G		0.002027		
SCORING FACTOR - X		0.1895		
ROOT LINE FACE WIDTH		1.800"		1.800"
PROFILE SLIDING FACTOR		0.00284		0.00554
RATIO OF INVOLUTE/MEAN CONE		1.107		
AXIAL FACTOR-DRIVER CW	OUT	0.220	IN	0.033
AXIAL FACTOR-DRIVER CCW	IN	0.175	OUT	0.077
SEPARATING FACTOR-DRIVER CW	SEP	0.214	SEP	0.084
SEPARATING FACTOR-DRIVER CCW	SEP	0.252	SEP	0.046
DUPLEX SUM OF DEDENDUM ANG		0D 36M		
ROUGHING RADIAL		16.739"		
INPUT DATA	KTT M			
INPUT DATA	CUTM			

DRAWING NOS. ST-4102 ST-4059

DIMENSIONS FOR ENGINE REDUCTION BEVEL GEARS

14.1 LOADS ON ENGINE BEVEL GEAR BEARINGS

The support bearings for the engine bevel gear carry two sets of loads, one set from the intermediate bevel pinion and one set from the engine bevel pinion. These loads are in different planes. Accordingly the bearing loads from each mesh are developed separately prior to being resolved into common planes and then added. Right hand and left hand assemblies have to be examined separately on account of different positions of the input pinions relative to the bevel gears that they drive.



a = 2.10 in

b = 3.20 in

c = 4.35 in

r = 1.801 in

R = 5.582 in

(1) Tooth loads imposed by the intermediate bevel pinion at 1800 hp and a pinion torque of 9255 in-lb are:

A_1;	Axial thrust	= 9255 x .336	= 3110 lb
S_1;	Separating force	= 9255 x .135	= 1249 lb
T_1;	Tangential force	= 9255/1.801	= 5138 lb

Gear factors are obtained from Gleason dimension sheet, page 38.

(2) Bearing loads imposed by the engine bevel gear; RH engine.

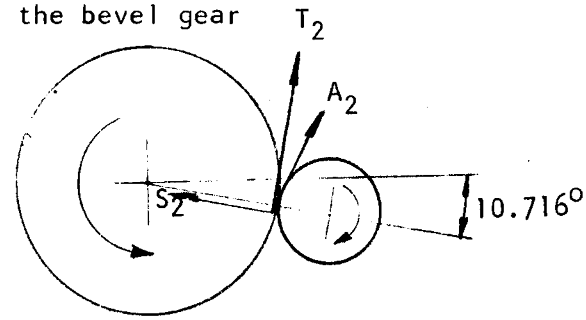
At 1800 hp input and a torque of 18510 in-lb at the bevel gear

A_2; Axial thrust = 18510 x (-0.033) = -611 lb

S_2; Separating force = 18510 x 0.084 = 1555 lb

T_2; Tangential force = 18510/5.582 = 3315 lb

R; Mean radius of gear = 5.582 in



Gear factors are obtained from Gleason dimension sheet, page 50.

(3) Net loads in horizontal/vertical plane for both bevels; RH engine.

Horizontal	Vertical
$R_2 = 854 + 1954 = 2813$ lb	$R_2 = 6567 + 81 = 6648$ lb
$R_1 = 395 + 1587 = 1982$ lb	$R_1 = 1429 - 830 = 599$ lb
$R_2{}^2 = 2813^2 + 6649^2$	$R_2 = 7219$ lb
$R_1{}^2 = 1982^2 + 599^2$	$R_1 = 2071$ lb

Bearing Loads and Life; RH Engine

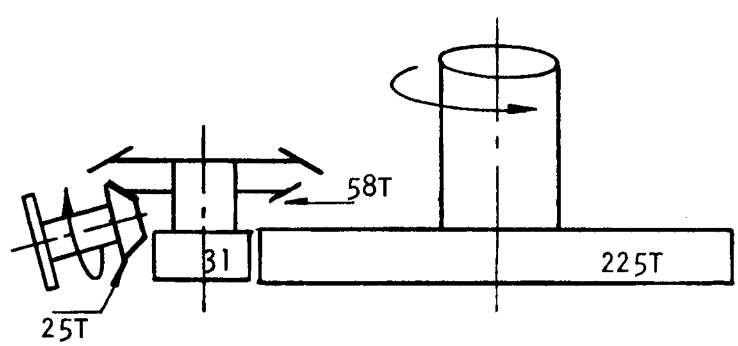
Item	Roller Bearing R_2	Roller Bearing R_1	Ball Thrust Bearing R_3
Speed; rpm	6128	6128	6128
Max load at 1800 hp, lb	7219	2071	3721
0.6 prorated load, lb	4331	1242	2232
Bearing size, mm	100x180x34	110x150x20	110x170x28
Dynamic capacity C, lb	48300	20200	19580
Computer calc. B_{10} life (material life factor of 5), hours	18000	64000	12500

15. TAIL ROTOR DRIVE

Power for the tail rotor drive shaft is extracted from a bevel pinion that is mounted underneath one of the four main-drive bevel gears. This arrangement keeps the numbers of bearings to a minimum while providing a near redundant drive path between the main shaft and the tail drive shaft.

The final drive pinion associated with the tail drive bevel gear is lightly loaded as a result of tail drive power being extracted from the same shaft. Design ratings for the tail drive gears, given in the following Table, are a steady maximum power of 380 hp and transient powers up to 865 hp. Bearing design is based on a 0.6 prorate factor applied to the steady maximum power of 380 hp.

The symmetrical positions adopted for the four final drive pinions result in the tail drive shaft being offset from the centerline of the transmission. A small angular displacement is given to the tail drive shaft to make a direct connection to the tail gear box of the NASA test stand. In an aircraft configuration this inclination would be less as a result of the tail axis being taken to the more distant pylon for the tail rotor. A less attractive alternative would be the introduction of an offset spur gear train to bring the tail drive shaft on to the transmission center line; but this change gives the incorrect direction of rotation to the tail drive shaft.



Schematic arrangement of tail drive gears

DATA FOR TAIL DRIVE BEVEL GEARS

Item	Pinion	Gear
Number of teeth	25	58
Ratio	2.320	2.320
Pitch diameter D, in	4.7414	11.00
Facewidth, in	1.500	1.500
Diametral pitch	5.27273	5.27273
Shaft angle, degrees	100.88333	100.88333
Pressure/spiral angles, degrees	20/25	20/25
Hand /rotation	LH/CCW	RH/CW
Mode of operation	driven	driver
Modified contact ratio	2.03	2.03
Power, hp	380 mean 865 peak	
Torque, max., in-lb	12549	29113
Speed, rpm	4344.4	1872.6
Compressive stress lb/in^2, peak	236800	236800
Bending stress, lb/in^2, peak	42460	42400
Pitch angle γ , degrees	24.7394	76.1439
Mean working dia = $D - F\sin\gamma$, in	4.11366	9.5437
Tangential force; peak, no prorate, lb	6101	6101
Tangential force; max. cont., no prorate, lb	2680	2680
Axial force, peak, lb	3609	1698
Separating force, peak, lb	1035	3349
Bearing prorate in terms of peak loads	0.6(380/865) = 0.264	
Torque, max. cont. (380 hp), in-lb	5513	12790
Axial force, max. continuous, lb	1585	746
Separating force, max. continuous, lb	455	1471

BEARING LOADS AND LIFE FOR PINION

Item	Rear Thrust Bearing, R_2	Front Bearing, R_1
Speed, rpm	4344.4	4344.4
Bearing loads; max. continuous based on 380 hp, lb	1100 radial 1585 axial	3598 radial -
Bearing loads; 0.6 prorated, lb	660 radial 951 axial	2159 radial -
Bearing size, mm	55x120x29	65x120x23
Dynamic capacity C, lb	15350	23300
Bearing constants	X =.41; Y =.87	-
* Equivalent load P, lb	1098	2159
C/P value, prorated	13.98	10.79
B_{10} life $10^6(C/P)^3/(60n)$, hours	10500	-
B_{10} life $10^6(C/P)^{3.33}/(60n)$, hours	-	10600
B_{10} life; material life factor of 5, hours	52500	53000

* $P = XP_r + YP_a$

P_r denotes the prorated radial load for the thrust bearing

P_a denotes the prorated axial load for the thrust bearing

X and Y are handbook bearing constants

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SPIRAL BEVEL GEAR DIMENSIONS NO. S008341A FORM M

TRANSMISSION RESEARCH-CLEVELAND #

	PINION	GEAR
NUMBER OF TEETH	25	58
PART NUMBER		
DIAMETRAL PITCH		5.273
FACE WIDTH	1.500"	1.500"
PRESSURE ANGLE	20D 0M	
SHAFT ANGLE	100D 53M	
TRANSVERSE CONTACT RATIO		1.495
FACE CONTACT RATIO		1.376
MODIFIED CONTACT RATIO		2.032
OUTER CONE DISTANCE		5.665"
MEAN CONE DISTANCE		4.915"
PITCH DIAMETER	4.741"	11.000"
CIRCULAR PITCH	0.596"	
WORKING DEPTH	0.363"	
WHOLE DEPTH	0.407"	0.407"
CLEARANCE	0.044"	0.044"
ADDENDUM	0.218"	0.145"
DEDENDUM	0.189"	0.262"
OUTSIDE DIAMETER	5.137"	11.070"
FACE ANGLE JUNCTION DIAMETER		
THEORETICAL CUTTER RADIUS	4.502"	
CUTTER RADIUS	4.500"	
CALC. GEAR FINISH. PT. WIDTH		0.130"
GEAR FINISHING POINT WIDTH		0.130"
ROUGHING POINT WIDTH	0.035"	0.110"
OUTER SLOT WIDTH	0.055"	0.130"
MEAN SLOT WIDTH	0.062"	0.130"
INNER SLOT WIDTH	0.055"	0.130"
FINISHING CUTTER BLADE POINT	0.035"	0.080"
STOCK ALLOWANCE	0.020"	0.020"
MAX. RADIUS-CUTTER BLADES	0.032"	0.096"
MAX. RADIUS-MUTILATION	0.051"	0.102"
MAX. RADIUS-INTERFERENCE	0.057"	0.089"
CUTTER EDGE RADIUS	0.030"	0.070"
CALC. CUTTER NUMBER	4	7
MAX. NO. BLADES IN CUTTER		15.293
CUTTER BLADES REQUIRED	STD DEPTH	STD DEPTH
GEAR ANGULAR FACE - CONCAVE		20D 13M
GEAR ANGULAR FACE - CONVEX		22D 8M
GEAR ANGULAR FACE - TOTAL		23D 32M
Drawing Nos.	ST 4117	ST 4111

		PINION		GEAR
PITCH APEX TO CROWN		5.054"		1.216"
FACE ANG JUNCT TO PITCH APEX				
MEAN CIRCULAR THICKNESS		0.315"		0.194"
OUTER NORMAL TOP LAND		0.132"		0.078"
MEAN NORMAL TOP LAND		0.137"		0.088"
INNER NORMAL TOP LAND		0.144"		0.080"
PITCH ANGLE		24D 44M		76D 9M
FACE ANGLE OF BLANK		27D 41M		78D 7M
INNER FACE ANGLE OF BLANK				
ROOT ANGLE		22D 46M		73D 12M
DEDENDUM ANGLE		1D 58M		2D 57M
OUTER SPIRAL ANGLE				31D 29M
MEAN SPIRAL ANGLE				25D 0M
INNER SPIRAL ANGLE				18D 29M
HAND OF SPIRAL	LH		RH	
DRIVING MEMBER	GEAR			
DIRECTION OF ROTATION-DRIVER	CW			
OUTER NORMAL BACKLASH	PIN	0.005"	MAX	0.007"
DEPTHWISE TOOTH TAPER	GIVN			
GEAR TYPE				GENERATED
FACE IN PERCENT OF CONE DIST				26.479
DEPTH FACTOR - K				
ADDENDUM FACTOR - C1				
GEOMETRY FACTOR-STRENGTH-J		0.3181		0.3186
STRENGTH FACTOR-Q		3.07616		1.32400
EDGE RADIUS USED IN STRENGTH		0.030"		0.070"
CUTTER RADIUS FACTOR - KX		1.014		
FACTOR	MN	0.9892		
STRENGTH BALANCE DESIRED	GIVN			
STRENGTH BALANCE OBTAINED	GIVN			- 0.002
GEOMETRY FACTOR-DURABILITY-I		0.1167		
DURABILITY FACTOR-Z		2016.08		1323.62
GEOMETRY FACTOR-SCORING-G		0.001871		
SCORING FACTOR - X		0.1388		
ROOT LINE FACE WIDTH		1.500"		1.500"
PROFILE SLIDING FACTOR		0.00356		0.00644
RATIO OF INVOLUTE/MEAN CONE		1.379		
AXIAL FACTOR-DRIVER CW	OUT	0.288	OUT	0.058
AXIAL FACTOR-DRIVER CCW	IN	0.124	OUT	0.105
SEPARATING FACTOR-DRIVER CW	SEP	0.082	SEP	0.115
SEPARATING FACTOR-DRIVER CCW	SEP	0.272	ATT	0.075
DUPLEX SUM OF DEDENDUM ANG		4D 55M		
ROUGHING RADIAL		5.071"		
INPUT DATA KTT	GIVN			
INPUT DATA CUTM				

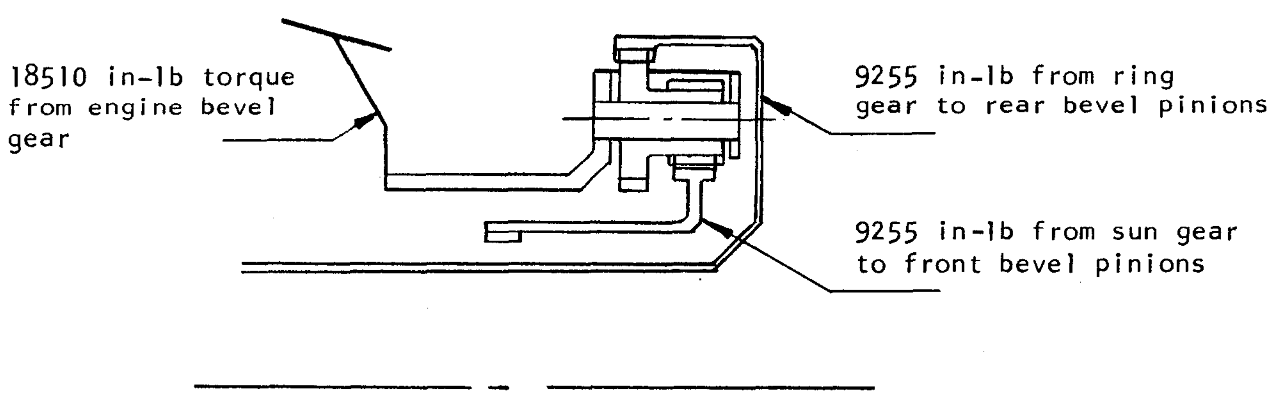
DIMENSIONS FOR TAIL DRIVE BEVEL GEARS

16. TORQUE DIVIDING UNIT

Each engine bevel gear receives a torque of 18510 in lb, at 1800 hp, and has to deliver exactly half of this torque to each of two bevel pinions. In order that these torques remain invariant, despite torsional windup of the drive shafts, and housing deflections, a torque dividing unit is positioned between the engine bevel gear and the two intermediate pinions.

A number of options are available for the torque divider, but one based on an epicyclic unit with stepped planet pinions is adopted since it:

a) avoids the axial thrusts associated with bevel gear designs
b) has straight spur teeth that can accommodate the relatively large (.050 in) axial growth and tolerances associated with the long drive shaft to each rear pinion
c) can have a one-piece carrier frame
d) is not sensitive to centrifugal loads.



Arrangement of torque divider for two equal output torques

DESIGN DATA SUMMARY FOR TORQUE DIVIDER

Item	Sun Gear	Large Pinion	Small Pinion	Ring Gear
No. of teeth	120	24	24	120
Diametral pitch	24	16	24	16
Pitch dia, in	5.000	1.500	1.000	1.500
Facewidth, in	0.75	0.25	0.7	0.45
No. of gear sets	1	10	10	1
Torque at 1800 hp, in-lb	9255	-	-	9255
Tangential tooth load, lb	360	247	370	247
Tooth compressive stress lb/in^2	144000	131200	144000	131200
AGMA J factor; 20^op.a.	.415	.486	.486	-
Root bending stress, lb/in^2	37100	42300	34000	-

Combined pinion mass	0.22 lb
Centrifugal load per planet	704 lb
Bearing load at max torque	538 lb (large pinion end)
Bearing load at max torque	432 lb (small pinion end)
Bearing static capacity	1690 lb (each needle roller)
Bearing pin dia	.4375 in
Pin bending stress with no oil hole	27760 lb/in^2 (adopted)
Pin bending stress with cross oil-hole	51350 lb/in^2 (not used)
Pin socket load	611 lb max (7500 psi)
Frame angular deflection	.0001 in
Tooth slope	.0001 in/in (approx.)

CROSS-SHAFT SIZING

Each cross shaft carries a maximum of 900 hp at a speed of 6128 rpm. The steady design torque then is 9256 in-lb.. Torsional shear stress at the various sections of the shaft is as follows:

Shaft Section	Out. dia in	In. dia in	J in^4	Kt	Torsional stress lb/in^2
U cut of LH spline	1.610	1.30	.379	1.20	23590
LH axial retainer	1.650	1.30	.447	1.20	20450
Central section	1.600	1.30	.363	1	20400
RH spline root	1.589	1.30	.345	-	21160

Critical speed for cross shaft

The support span for the cross shaft is 35.25 between the center of the LH retainer flange and the RH locating-bushing. For thin-wall shafting the first critical speed is:

$$S \text{ critical} = 19.04(10^6)D/(2.9L^2)$$

where D = 1.60 and L = 35.25 in

Then S critical = 8454 rpm

Since the running speed is 6128 rpm the cross shaft operates at 27.5% below the first critical speed.

Spline loads	L.H. spline	R.H. spline
spline condition	free	locked
torque, max; in-lb	9255	9255
pitch, dia; in	1.750	1.6875
No. of teeth and DP	21; 12/24	27; 16/32
direct shear stress, lb/in^2	2200	3760
spline contact stress, lb/in^2	3510	5957

17. LOSSES IN TRANSMISSION

Losses in the transmission system are estimated from a knowledge of the type of gear mesh, the design horsepower, and the power transmitted by each gear mesh. Past experience, backed by experimental confirmation, demonstrates that a conservative estimate of tooth mesh losses together with bearing losses, is 0.5% of transmitted power for both spur gear and bevel gear. A further 0.75% of power transmitted must then be added to account for the miscellaneous churning losses.

On the foregoing basis the losses in the transmission are as follows:

Gear Mesh	Loss Percent	Loss hp
1 input bevel	0.5	18
1 intermediate bevel (1800 hp)	0.5	9
1 intermediate bevel (1800 hp)	0.5	9
1 spur	0.5	18
Churning	0.75	27
Transmission Total Dry Losses		81 hp

(Tail drive losses included in above)

Heat Generated in Main Gearbox

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

$$Q_G = 2545\ F_{hp}\ [\text{BTU/hr}] \qquad F_{hp} = 81\text{ hp}$$

$$Q_G = 206{,}000\text{ BTU/hr}$$

With either MIL-L-7808 or MIL-L-23699 oil, an inlet temperature of 180°F and and an outlet temperature of 230°F, the required oil flow is given by:

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

where the factor $C_e = 0.60$ is an empiric factor obtained from measurements on production main gearboxes, C_p is the specific heat of the cooling oil (BTU/lb-F^o) and ρ the density of the cooling oil (lb/ft^3)

$$W_o = \frac{0.60(42.4)(81)}{(.1337)(.528)(54.4)(50)}$$

$$W_o = 10.7 \text{ gal/min}$$

LIST OF OIL JETS

DWG. NO.	LOCATION	BEARING	NO. JETS .035/.040 dia	TOTAL JETS
4099	INPUT SHAFT	FRONT ROLLER	2	4
4099	INPUT SHAFT	REAR ROLLER	2	4
4086	INPUT SHAFT	BALL	2	4
4061	INPUT BEVEL GEAR SHAFT	FRONT BALL	1	2
4081	INPUT BEVEL GEAR SHAFT	REAR ROLLER	1	2
4037	REAR INTERMEDIATE BEVEL PINION SHAFT	ROLLER	1	2
4041	REAR INTERMEDIATE BEVEL PINION SHAFT	TANDEM BALL	1	2
4026	FINAL DRIVE PINION/BEVEL GEAR SHAFT	UPPER ROLLER	1	4
4026	FINAL DRIVE PINION/BEVEL GEAR SHAFT	LOWER ROLLER	1	4
4023	FINAL DRIVE PINION/BEVEL GEAR SHAFT	BALL	1	4
4127	TAIL DRIVE SHAFT	ROLLER	1	1
4121	TAIL DRIVE SHAFT	BALL	1	1
4052	MAIN SHAFT	UPPER ROLLER	1	1
4006	MAIN SHAFT	TANDEM BALL	1	1
	INPUT SHAFT BEVEL PINION/GEAR MESH		3	6
	FRONT INTERM. BEVEL PINION/GEAR MESH		3	6
	REAR INTERM. BEVEL PINION/GEAR MESH		3	6
	FINAL DRIVE PINION/GEAR MESH		3	12
	FINAL DRIVE BEVEL PINION/TAIL DRIVE GEAR MESH		3	3
4076/4080	FRONT INTERM. BEVEL PINION TAPER ROLLERS	TAPER ROLLERS	MIST	0
4067	TORQUE DIVIDER ROLLER	NEEDLE ROLLER	MIST	0
	TOTAL			69

18. BEARING DATA SUMMARY

DWG. NO.	LOCATION	BEARING TYPE	SIZE	BRG.NO. (TYPICAL)	NO. REQ'D	SPEED RPM	MAX LOAD LBS.	DYN. CAPA-CITY, LBS.	B_{10} LIFE * HOURS
4099	Input bevel pinion, front	Cylindrical roller	60x110x22mm	212	2	20909	1797	21400	19300
4099	Input bevel pinion, rear	Cylindrical roller	60x110x22mm	212	2	20909	1823	21400	18400
4086	Input bev. pin., rear thrust	Ball, split inner ring	60x110x22mm	212	2	20909	1194	11540	50500
4061	Input bev.gear, thrust	Ball, split inner ring	110x170x28mm	1022	2	6128	3721	18890	12500
4081	Input bevel gear, rear, L.H.	Cyl. roller (Less inner ring)	100x180x34mm	220	1	6128	3511	48300	198000
4081	Input bevel gear, rear, R.H.	Cyl. roller (Less inner ring)	100x180x34mm	220	1	6128	7219	48300	18000
4135	Input bevel gear, front, L.H.	Cylindrical roller	110x150x20mm	1922	1	6128	1733	20200	114000
4135	Input bevel gear, front, R.H.	Cylindrical roller	110x150x20mm	1922	1	6128	2071	20200	64000
4080	Front intm. bev. pinion, rear	Taper roller, Timken(Less cup)	2.625x4.25x1.0"	29590	2	0	-	4210	∞
4076	Front intm. bev. pinion, front	Taper roller, Timken	60x95x24mm	JLM508748 JLM508710	2	0	-	3270	∞
4037	Rear intm. bev. pinion	Cylindrical roller	40x90x23mm	308	2	6128	2744	17100	9200
4041	Rear intm. bev. pinion	Ball, tandem split inner ring matched set	60x130x62mm	312	2(prs.)	6128	3005 rad. 3110 ax.	29840	27400

DWG. NO.	LOCATION	BEARING TYPE	SIZE	BRG.NO. (TYPICAL)	NO. REQ'D	SPEED RPM	MAX LOAD LBS.	DYN. CAPACITY, LBS.	B_{10} LIFE* HOURS
4026	Intm. bev. gear, upper	Cylindrical roller	95x170x32mm						
	a) R.H. front			219	1	1873	10970	44300	25500
	b) R.H. rear			219	1	1873	5613	44300	237000
	c) L.H. front			219	1	1873	7653	44300	84500
	d) L.H. rear			219	1	1873	994	44300	75×10^6
4026	Intm. bev. gear, lower	Cylindrical roller	95x170x32mm						
	a) R.H. front			219	1	1873	8450	44300	60500
	b) R.H. rear			219	1	1873	4121	44300	660000
	c) L.H. front			219	1	1873	7383	44300	95000
	d) L.H. rear			219	1	1873	8474	44300	60000
4023	Intm. bevel gear	Ball, split inner ring	105x145x20mm	1921	4	1873	1151	11200	288000
4127	Tail drive pinion, front	Cylindrical roller	65x120x23mm	213	1	4344	3598	23300	53000
4121	Tail drive pinion, rear	Ball, split inner ring	55x120x29mm	311	1	4344	1100 rad. 1585 ax.	14940	52500
4052	Main shaft, upper	Ball, Kaydon	7.5x9.5x1.0 "	-	1	258	-	6270	N/A
4006	Main shaft, lower	Ball, tandem, split inner ring matched set	180x250x66mm	1936	1 (pr.)	258	-	48420	N/A
4067	Torque divider	Needle roller	7/16x5/8x $\frac{1}{2}$ "	-	40	0	538	1300	∞

N/A: NOT APPLICABLE

* Bearing System B_{10} Life = 1500 hours. Based on 0.6 Prorated Load, Material Life Factor of 5.

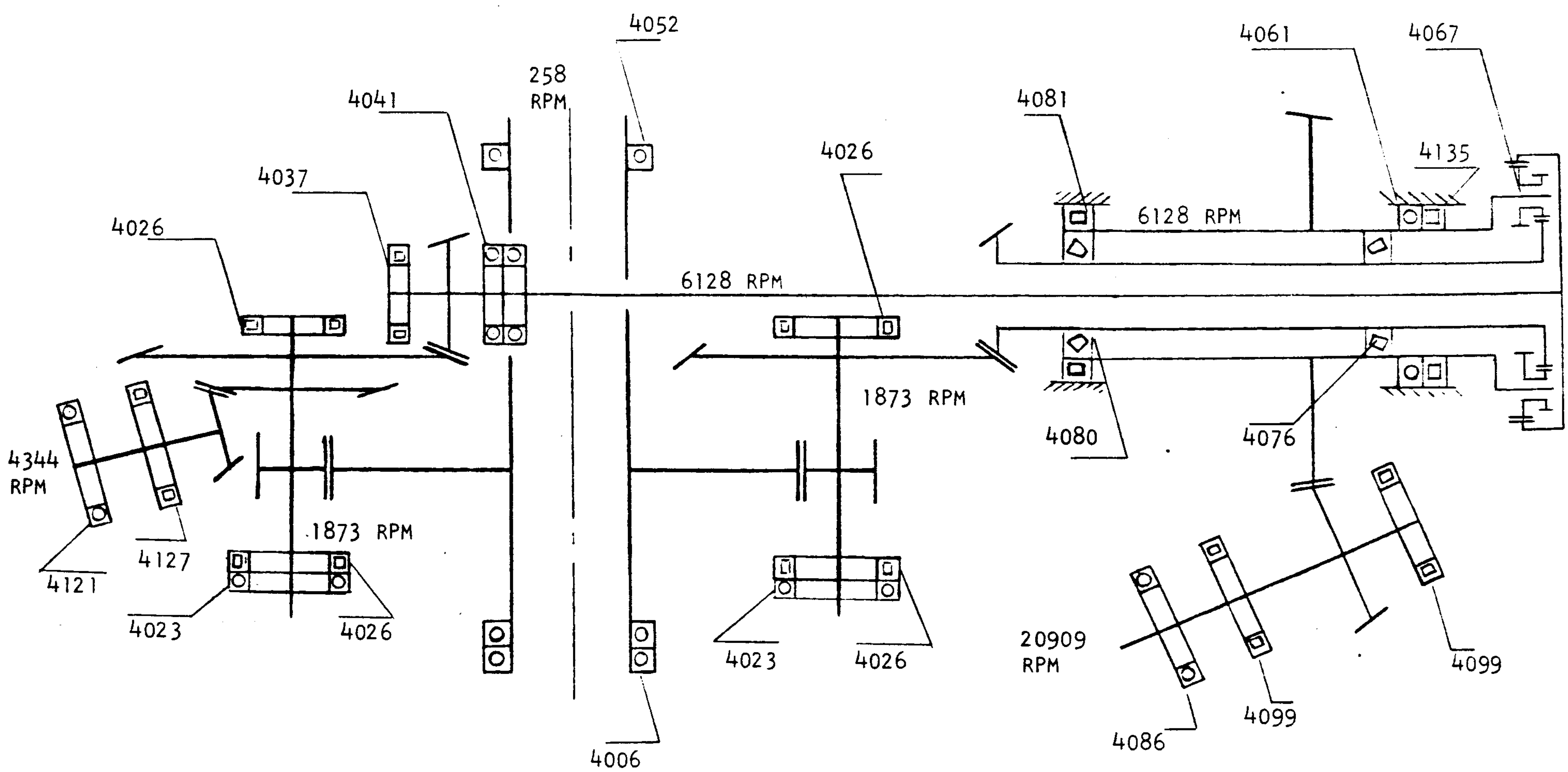


FIG. 6A BEARING LOCATION DIAGRAM

19. WEIGHT SUMMARY - 4 PINION DESIGN

	Weight lbs.	Weight lbs.
Main housings		
Main upper housing	62.0	
Main lower housing and feet	65.0	
Engine input housing assembly (2)	19.6	
Engine input housing bearing support (2)	7.8	
Sump pan and fittings	16.2	
Input shaft housing (2)	3.0	
Tail drive assembly housing	2.5	
		176.1
Housing sleeves and bearing retainers		21.4
Main shaft assembly		
Main shaft	112.2	
Top bearing	6.3	
Top locknut	1.6	
Lower locknut	2.8	
Lower bearings (2)	17.5	
		140.4
Combining gear assembly		
Gears	101.7	
Support discs (2)	21.1	
		122.8
Final pinion and bevel assembly		
Gear sets, front (2)	61.2	
Gear sets, rear (2)	64.2	
Top roller bearings (4)	24.7	
Lower roller bearings (4)	24.9	
Lower thrust bearings (4)	8.0	
		183.0
Forward bevel pinion assembly		
Bevel pinion (2)	15.2	
Pinion taper bearings (4)	5.2	
Drive sleeve and spacers (2)	3.1	
		23.5
Rear bevel pinion assembly		
Bevel pinion (2)	11.0	
Front bearings (2)	3.3	
Rear bearings (4)	14.2	
Rear bearing housings & retainer (2)	3.4	
Bearing nuts (4), & shaft retainer (2)	2.2	
		34.1

Cross-shaft drive		
Drive shaft (2)	9.3	
Drive flange to cross shaft (2)	1.9	
		11.2
Torque divider		
Carrier frame	3.1	
Ring gear (2)	4.2	
Planet pinions (20)	2.4	
Planet shaft & bearings (20)	1.7	
Oil flange and ring (2)	0.5	
Sun gear (2)	4.2	
Locknut & spacer	0.8	
		16.9
Engine bevel gear assembly		
Bevel gear & shaft (one-piece) 2	40.3	
Ball thrust bearing(2)	8.4	
Rear roller bearing (2)	12.4	
Front roller bearing (2)	4.8	
		65.9
Engine bevel pinion assembly		
Bevel pinion (2)	13.0	
Front bearing & nut (2)	4.7	
Rear roller bearing (2)	4.2	
Rear thrust bearing (2)	3.6	
Lube ring, spacer, nut (2)	0.9	
		26.4
Input shaft and clutch assembly		
Clutch coil, inner shaft, nut (2)	3.6	
Input shaft (2)	3.8	
Input flange (2)	0.8	
Seal assembly (2)	0.4	
		8.6
Tail drive assembly		
Bevel gear	5.7	
Bevel pinion	5.4	
Front roller bearing	2.5	
Rear thrust bearing	3.2	
Drive flange	1.3	
Spacers and nut	1.4	
Seal assembly	0.3	
		19.8

Miscellaneous hardware		
Bolts, inserts, seal rings, etc.	12.0	
Lubrication system, dry		
Pumps, filter, jets, plugs, filler	30.0	
Total Weight, dry		892 lbs
Addition for lubricant		
7 gallons of oil	50.4 lbs	

The 892 lb. weight shown above has been calculated as for a flight transmission. Because of various reasons, including reduced costs, the test stand transmission weighs about 8% more, see table below:

Weight Increments for Test Stand Unit

Item	Reason	Weight, lb
Main shaft	allows engine rating of 2000 hp	+12.6
Lower main shaft brgs.	standard width employed	+ 2.3
Output gear flanges	steel in place of titanium	+22.4
Output pin.& bevel assy	pinion separate from bevel	+ 8.6
Output pin. bearings	equal size bearings used	+ 5.4
Cross shaft	steel instead of alum. or composite	+ 8.4
Forward bevel pinion	not fully hollowed out	+ 1.8
Rear bevel pin. brgs.	standard width employed	+ 1.6
Eng. bevel gear assy	separate gear & shaft employed	+ 8.7

20. BENEFITS OFFERED BY SPLIT TORQUE TRANSMISSION

A consequence of the present study is that the following performance benefits can be achieved from a 3600 hp helicopter transmission that incorporates split torque gear trains and has widely separated engines.

1) Weight reduced by 15%.
   Overall weight is 892 lb

2) Drive train losses reduced by 9%
   Losses are 2.25%

3) Improvement in reliability
   Redundant drive paths from engines to combining gear

4) Reduced number of noise meshes

5) Comparable totals of gears and bearings

6) Development potential to accept twin 2250 hp engines

These comparisons are based on a standard planetary gear transmission.

21. FUTURE TECHNOLOGY DEVELOPMENTS

The design discussed is based on essentially the same level of technology as employed in current production transmissions. The new configuration, however, does allow the following advanced technology areas to be developed and progressively introduced into the test transmission.

1) Combining gear concept can be developed to include advanced involute, high contact ratio, or conformal tooth profiles.

2) Protected oil cooler and sump can be positioned within the large diameter main shaft.

3) Each engine driveline can have individual lubrication and debris detection systems.

4) Widely separated bearings and gear meshes reduce risk of secondary entrainment damage.

5) High-speed overrunning clutch development is necessary for least weight.

6) A one-piece main shaft and combining gear assembly gives a further weight reduction of 12-14 pounds.

INSTALLATION IN NASA-LeRC TEST STAND

Design of the transmission gear ratios, housing location points, and drive shaft flanges, is influenced by the need for the transmission to fit in the NASA test stand. In its present form the test stand is configured to accept a UH 60 transmission; it follows that to fit in the test stand the split torque transmission must be a near replacement design for the UH 60.

Since the test stand is of closed-loop type the speed ratios in the transmission must be an exact integer-match with those of the test stand. In this way the torque loading units of the test stand can remain stationary, as intended, once a torque is locked into the drive trains. Three torque loading units are present; these allow separate loading of transmission drive trains between each engine input and the main shaft, and between the main shaft and tail drive shaft.

Figure 7 shows the arrangement of gear trains in the test stand, together with the tooth numbers that must be accommodated. Figures 8 and 9 show the position of the test transmission in the test stand together with the location of the test stand gearboxes and the support beams.

Tooth Numbers Adopted

Tooth numbers adopted in the split torque transmission are given below and are compared with the equivalent gear train of the test stand.

1) Engine input to main shaft (both sides)

Test stand ratio	Transmission ratio
$\frac{392}{50} \times \frac{270}{119} \times \frac{145}{77} \times \frac{75}{31}$	$\frac{225}{31} \times \frac{72}{22} \times \frac{116}{34}$
or: $\frac{(612.5)(187920)}{(612.5)(23188)}$	$\frac{187920}{23188}$

2) <u>Tail drive to main shaft</u>

Test stand ratio	Transmission ratio
$\frac{392}{50} \times \frac{125}{70} \times \frac{63}{49} \times \frac{29}{31}$	$\frac{225}{31} \times \frac{58}{25}$
or: $\frac{(6860)(13050)}{(6860)(775)}$	$\frac{13050}{775}$

SUMMARY OF TEST STAND CHANGES REQUIRED FOR ACCOMMODATION OF THE 3600 HP SPLIT TORQUE TRANSMISSION

Design of the split torque transmission is such as to involve as few changes as possible to the existing test stand. Major items in the test stand that require <u>no change</u> include:

a) the splined drive flange between the transmission main shaft and the test stand shaft

b) torque loading units

c) drive flanges for each engine input and the tail drive shaft

d) drive shafting and torque measuring instrumentation for each engine input

e) transmission cradle structure and the test stand attachment points

f) external components of the lube system and the cooler.

<u>Changes that are necessary</u> include

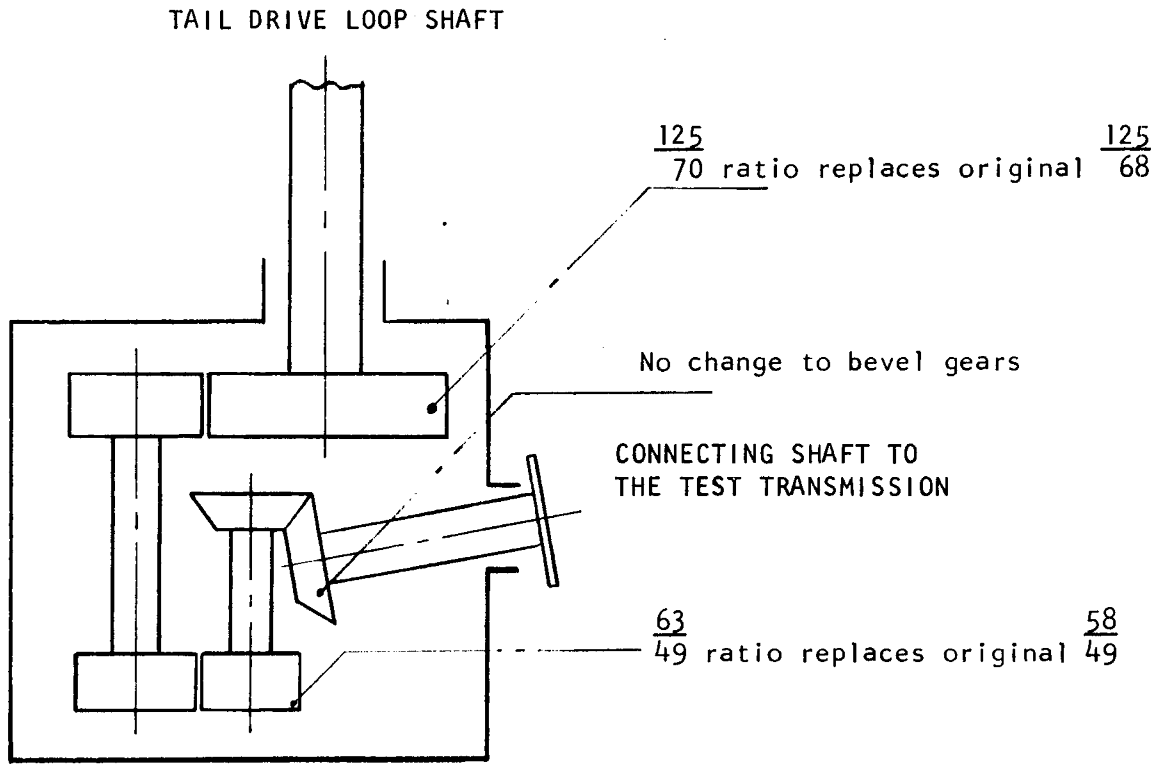
a) four small cutouts in the cradle lower face; to clear protrusions on the new transmission housing

b) repositioning of the test stand tail drive gearbox to align with the transmission drive shaft

c) replacement of two helical gear trains in the tail drive gearbox. The existing helical gear trains of ratio (125/68)(58/49) must be replaced by similar helical gears of ratio (125/70)(63/49)

The bevel gears in the gearbox remain unaltered together with all the housings, shafts, bearings, and spacers.

d) a connecting shaft of reduced length is needed between the transmission and the tail drive gearbox.



New gear ratios required in tail drive gearbox of test stand

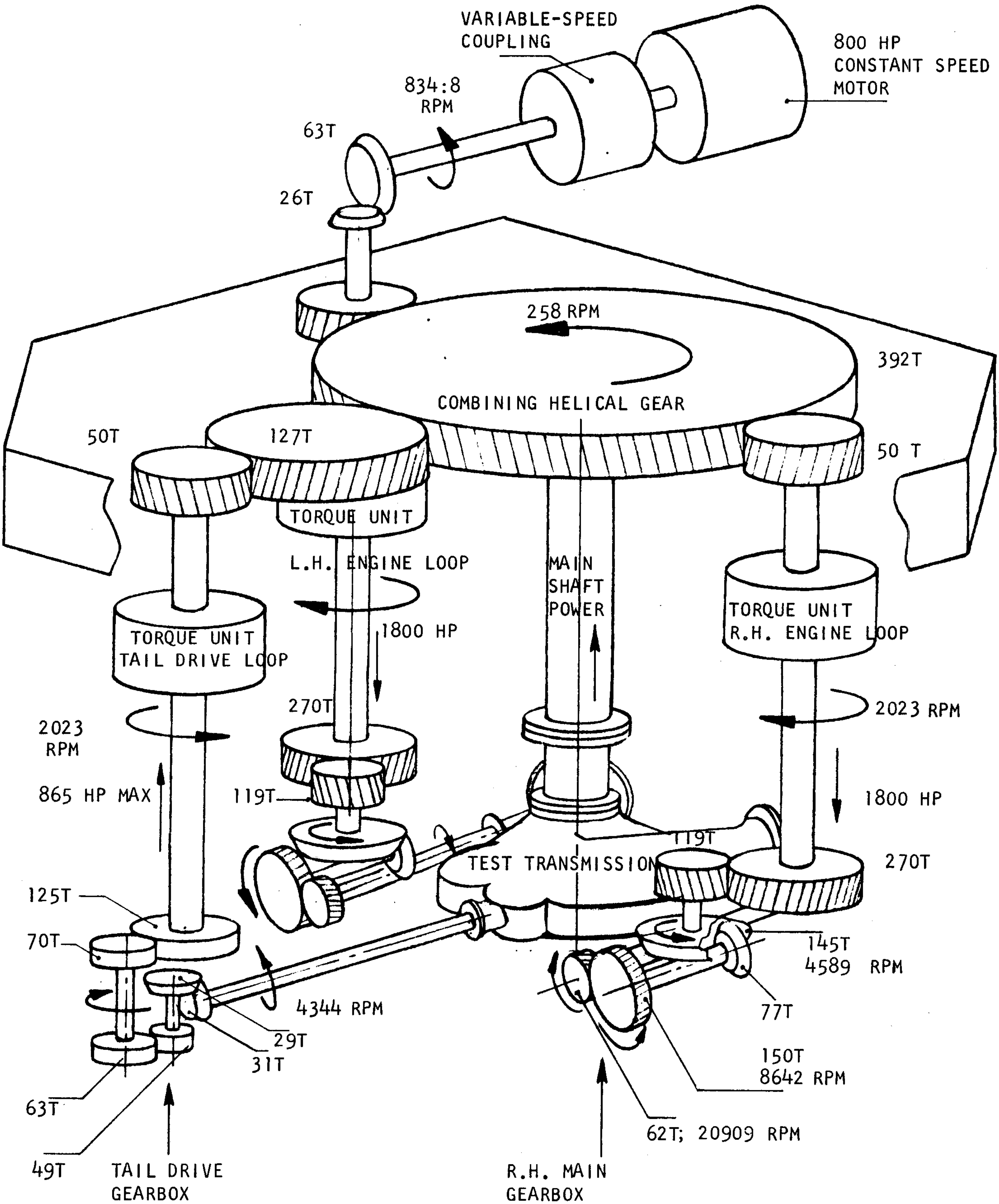


Fig. 7 Arrangement of gear trains in helicopter transmission test stand

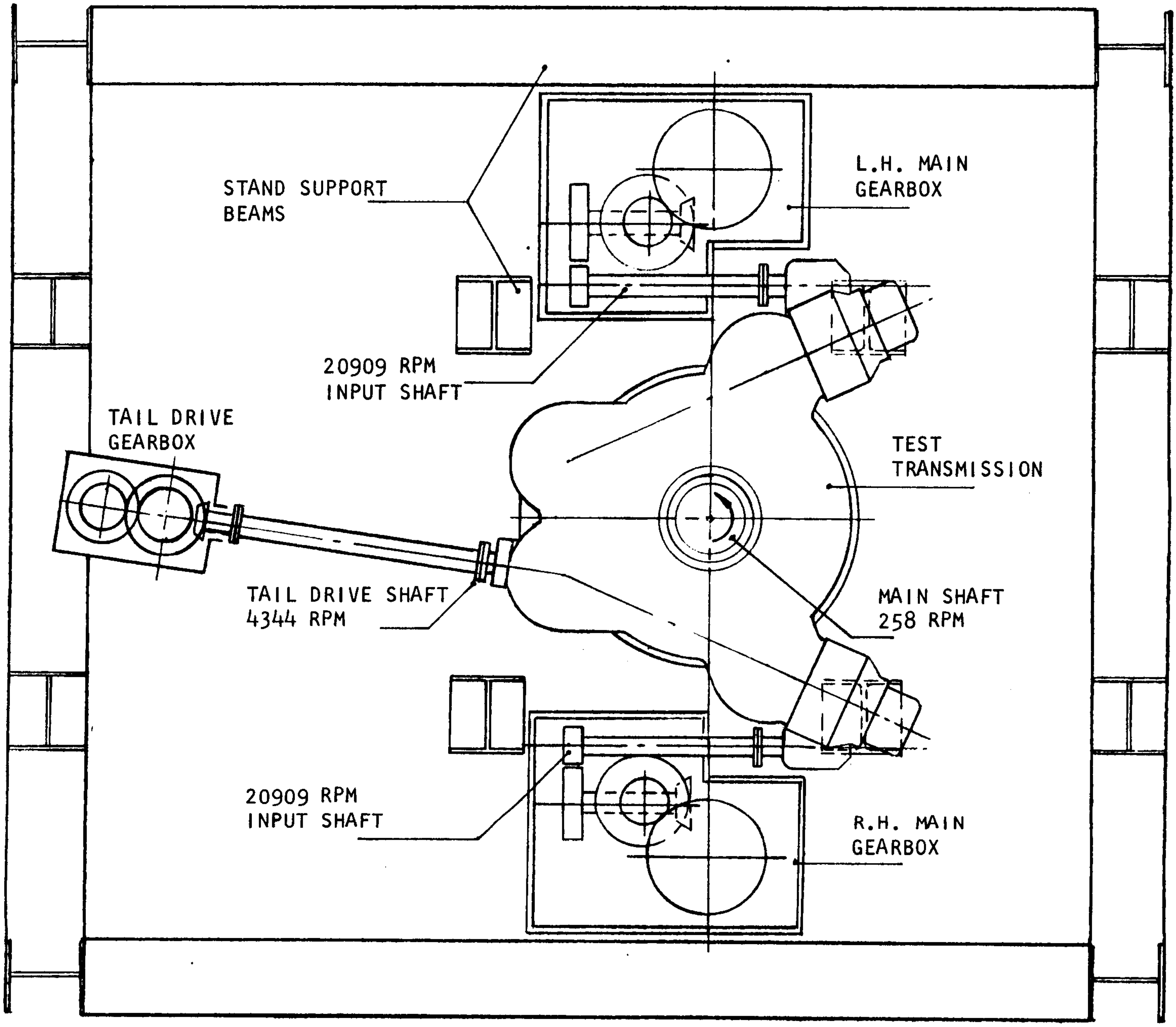


Fig. 8 Position of split-torque transmission in NASA Lewis Test Stand
-Plan View-

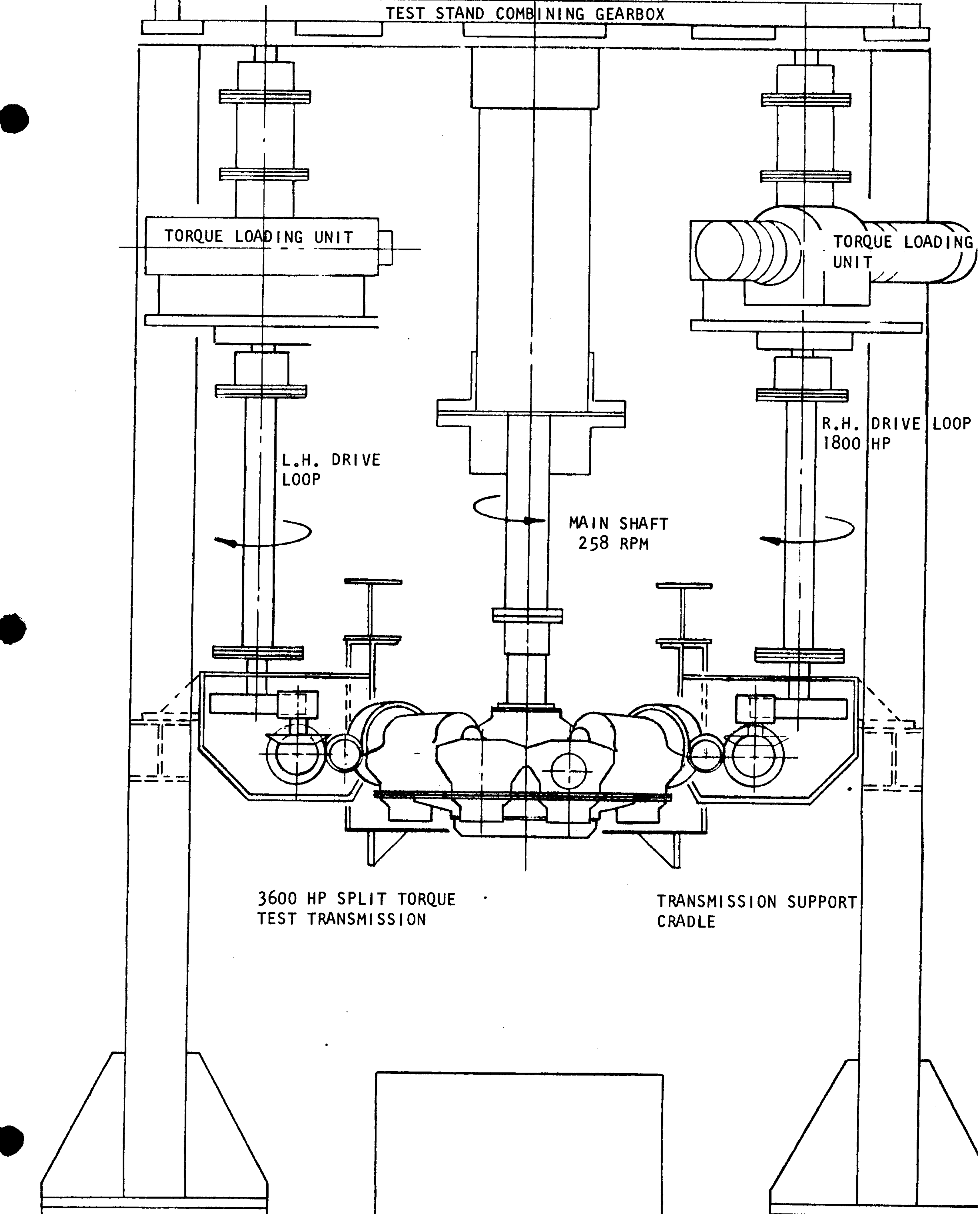


Fig. 9 End view of transmission in NASA Test Stand

23. PARTS LIST

PART NO.	DWG. SIZE	NO. REQ'D	DESCRIPTION	REMARKS
ST 4000	R		MAIN ASSEMBLY (2 SHTS.)	
ST 4001	E	1	MAIN HOUSING ASSEMBLY, UPPER (5 SHTS.)	
* ST 4001A	E	1	MAIN HOUSING, UPPER (CASTING)(7 SHTS.)	SEE ST 4001
ST 4002	E	1	MAIN HOUSING ASSEMBLY, LOWER (2 SHTS.)	
* ST 4002A	E	1	MAIN HOUSING, LOWER (CASTING)(2 SHTS.)	SEE ST 4002
ST 4003	E	1	OIL PAN	
* ST 4004	R	1	SHAFT, MAIN ROTOR	SEE ST 4011
ST 4005	C	1	LOCKNUT, LOWER BEARING, MAIN SHAFT	
ST 4006	A	1 pr.	BALL BEARING SET, MAIN SHAFT, LOWER (#1936)	
ST 4007-1	A	12	HEX. HD. CAP SCR. 3/8-24 UNF-3Ax1"LG.-STD.	
ST 4007-2	A	16	HEX. HD. CAP SCR. $\frac{1}{4}$-28 UNF-3Ax3/4 LG.-STD.	
ST 4007-3	A	26	HEX. HD. CAP SCR. 5/16-24 UNF-3Ax1" LG.-STD.	
ST 4007-4	A	80	HEX. HD. CAP SCR. 5/16-24 UNF-3Ax1$\frac{1}{4}$ LG.-STD.	
ST 4007-5	A	120	HEX. HD. CAP SCR. #10-32 UNF-3Ax$\frac{1}{2}$ LG.-STD.	
ST 4007-6	A	12	HEX. HD. CAP SCR. 5/16-24 UNF-3Ax3/4 LG.-STD.	
ST 4007-7	A	32	HEX. HD. CAP SCR. #10-32 UNF-3Ax5/8 LG.-STD.	
ST 4007-8	A	36	HEX. HD. CAP SCR. $\frac{1}{4}$-28 UNF-3Ax5/8 LG.-STD.	

PART NO.	DWG. SIZE	NO. REQ'D	DESCRIPTION	REMARKS
ST 4008	D	1	RET. PLATE, LOWER BEARING, MAIN SHAFT	
ST 4009	D	1	LINER, LOWER BEARING, MAIN SHAFT	
* ST 4010	B	60	BOLT, TENSION, COMB. GEAR ASS'Y	SEE ST 4011
ST 4011	D	1	COMBINING GEAR ASSEMBLY	
* ST 4012	D	1	SPUR GEAR, COMBINING	SEE ST 4011
* ST 4013	D	2	FLANGE, COMBINING GEAR	SEE ST 4011
* ST 4014	D	4	SPUR PINION, INTERMEDIATE	SEE ST 4011
ST 4015-1	A	1	"O" RING -1/8 SECTION - SPECIAL (OIL PAN)	
ST 4015-2	A	1	"O" RING - 1/8 SECTION - SPECIAL (SPLIT LINE)	
ST 4015-3	A	4	"O" RING - PARKER #2-163	
ST 4015-4	A	4	"O" RING - PARKER #2-161	
ST 4015-5	A	1	"O" RING - PARKER #2-176	
ST 4015-6	A	1	"O" RING - PARKER #2-273	
ST 4015-7	A	10	"O" RING - PARKER #2-049	
ST 4015-8	A	4	"O" RING - PARKER #2-162	
ST 4015-9	A	2	"O" RING - PARKER #2-276	
ST 4015-10	A	4	"O" RING - PARKER #2-277	
ST 4015-11	A	2	"O" RING - PARKER #2-169	
ST 4015-12	A	2	"O" RING - PARKER #2-019	

PART NO.	DWG. SIZE	NO. REQ'D	DESCRIPTION	REMARKS
ST 4015-13	A	2	"O" RING - PARKER #2-171	
ST 4015-14	A	2	"O" RING - PARKER #2-017	
ST 4015-15	A	4	"O" RING - PARKER #2-016	
ST 4015-16	A	8	"O" RING - PARKER #2-011	
ST 4015-17	A	2	"O" RING - PARKER #2-166	
ST 4015-18	A	4	"O" RING - PARKER #2-020	
ST 4015-19	A	4	"O" RING - PARKER #2-007	
ST 4015-20	A	2	"O" RING - PARKER #2-150	
ST 4015-21	A	2	"O" RING - PARKER #2-031	
ST 4015-22	A	5	"O" RING - PARKER #2-024	
ST 4015-23	A	2	"O" RING - PARKER #2-043	
ST 4015-24	A	2	"O" RING - PARKER #2-041	
ST 4015-25	A	2	"O" RING - PARKER #2-028	
ST 4015-26	A	2	"O" RING - PARKER #2-048	
ST 4015-27	A	1	"O" RING - PARKER #2-254	
ST 4015-28	A	3	"O" RING - PARKER #2-047	
ST 4015-29	A	1	"O" RING - PARKER #2-029	
ST 4015-30	A	1	"O" RING - PARKER #2-033	
ST 4015-31	A	10	"O" RING - PARKER #2-014	

PART NO.	DWG. SIZE	NO. REQ'D	DESCRIPTION	REMARKS
ST 4015-32	A	1	"O" RING - 3/32 SECTION-SPECIAL	
ST 4015-33	A	4	"O" RING - PARKER #2-042	
ST 4015-34	A	2	"O" RING - PARKER #2-036	
ST 4015-35	A	4	"O" RING - PARKER #2-037	
ST 4015-36	A	5	"O" RING - PARKER #2-015	
ST 4015-37	A	1	"O" RING - PARKER #2-155	
ST 4015-38	A	8	"O" RING - PARKER #2-018	
ST 4015-39	A	4	"O" RING - PARKER #2-012	
ST 4016-1	A	4	SOC. HD. CAP SCR.-$\frac{1}{4}$-28 UNF-3Ax$\frac{1}{2}$ LG.- STD.	
ST 4016-2	A	34	SOC. HD. CAP SCR.-#10-32 UNF-3Ax$\frac{1}{2}$ LG. - STD.	
ST 4016-3	A	4	SOC. HD. CAP SCR.-#8-36 UNF-3Ax$\frac{1}{2}$ LG. - STD.	
ST 4016-4	A	8	SOC. HD. CAP SCR.- $\frac{1}{4}$-28 UNF-3Ax7/8 LG.- STD.	
ST 4016-5	A	20	SOC. HD. CAP SCR.-$\frac{1}{4}$-28 UNF-3Ax5/8 LG. - STD.	
ST 4016-6	A	4	SOC. HD. CAP SCR.-#8-36 UNF-3Ax3/8 LG. - STD.	
ST 4016-7	A	6	SOC. HD. CAP SCR.-#8-36 UNF-3Ax5/16 LG. - STD.	
ST 4016-8	A	32	SOC. HD. CAP SCR.-#10-32 UNF-3Ax5/8 LG. - STD.	

PART NO.	DWG. SIZE	NO. REQ'D	DESCRIPTION	REMARKS
ST 4017	A	1	SHAFT SEAL, TAIL DRIVE FLANGE(C/R #24911)	
ST 4018	B	4	SPACER, LOWER BEARINGS, INTERMEDIATE BEVEL GEAR	
ST 4019	C	4	CAP, LOWER BEARING, INTERMEDIATE BEVEL GEAR	
ST 4020	D	2	BEVEL GEAR, INTERMEDIATE, REAR	
ST 4021	B	4	SLEEVE LOCKNUT, INTERMEDIATE BEVEL GEAR	
ST 4022	C	4	BEARING RETAINER, LOWER, INTERMEDIATE BEVEL GEAR	
ST 4023	A	4	BALL BRG., SPLIT INNER RING, INTM. BEVEL GEAR (#1921)	
ST 4024	C	4	SHIM, FITTED, INTM. BEVEL GEAR	
* ST 4025	B	4	LINER, LOWER ROLLER BRG., INTM. BEVEL GEAR	SEE ST 4002
ST 4026	A	8	CYL. ROLLER BRG., INTM. BEVEL GEAR (#NU-219)	
ST 4027	C	4	RET. PLATE, LOWER ROLLER BRG., INTM. BEVEL GEAR	
* ST 4028-1	A	60	FLANGED TENSION BOLT (SPS #EWB26-7-14)	SEE ST 4011
ST 4028-2	A	36	FLANGED TENSION BOLT (SPS #EWB26-5-10)	
ST 4028-3	A	90	FLANGED TENSION BOLT (SPS #EWB26-5-14)	
ST 4028-4	A	36	HEX. HD. SHEAR BOLT (NA6303-4)	
ST 4028-5	A	8	FLANGED TENSION BOLT (SPS #EWB26-3-4)	
* ST 4029-1	A	120	FLANGED FLEXLOC NUT (SPS #FN26-720)	SEE ST 4011
ST 4029-2	A	240	FLANGED FLEXLOC NUT (SPS #FN26-524)	

PART NO.	DWG. SIZE	NO. REQ'D	DESCRIPTION	REMARKS
ST 4029-3	A	36	FLEXLOC HEX. LOCKNUT (SPS #20FC-1032)	
ST 4029-4	A	2	FLANGED FLEXLOC NUT (SPS #FN26-820)	
ST 4029-5	A	8	FLANGED FLEXLOC NUT (SPS #FN22-1018)	
ST 4030-1	A	180	PLAIN WASHER -SPS #WP22-7 (7/16)	
ST 4030-2	A	60	PLAIN WASHER, C'SUNK -SPS #WC22-7 (7/16)	
ST 4030-3	A	156	PLAIN WASHER -SPS #WP22-5 (5/16)	
ST 4030-4	A	52	PLAIN WASHER -SPS #WP22-3 (#10)	
ST 4030-5	A	12	PLAIN WASHER (3/8 SAE STD.)	
ST 4030-6	A	44	PLAIN WASHER(SAE STD. $\frac{1}{4}$)	
ST 4030-7	A	104	PLAIN WASHER (SAE STD. 5/16)	
ST 4030-8	A	146	PLAIN WASHER (SAE STD. #10)	
ST 4030-9	A	54	MED. SPRING LOCKWASHER (#10 HIGH COLLAR,STD.)	
ST 4030-10	A	28	MED. SPRING LOCKWASHER ($\frac{1}{4}$ - STD.)	
ST 4031	C	4	RET. PLATE, UPPER BRG., INTM. BEVEL GEAR	
ST 4032	B	4	FLANGED LINER/SPACER, LOWER BRG., INTM BEVEL GEAR	

PART NO.	DWG. SIZE	NO. REQ'D	DESCRIPTION	REMARKS
ST 4033	B	2	LOCKNUT, UPPER BRG., INTM. BEVEL GEAR, REAR	
ST 4034	B	2	LOCKNUT, UPPER BRG., INTM. BEVEL GEAR, FRONT	
ST 4035	B	2	LOCKNUT, ROLLER BRG., INTM. BEVEL PINION, REAR	
ST 4036	A	2	LINER, ROLLER BRG., INTM. BEVEL PINION, REAR	
ST 4037	A	2	CYL. ROLLER BRG., INTM. BEVEL PINION, REAR (#NU-308)	
ST 4038	B	2	RET. PLATE, ROLLER BRG., INTERMEDIATE PINION, REAR	
ST 4039	D	2	BEVEL PINION, INTERMEDIATE, REAR	
ST 4040	C	2	RET. PLATE, BALL BRG., INTM. BEVEL PINION, REAR	
ST 4041	A	2 prs.	BALL BRG., SPLIT INNER RING, INTM. BEV. PIN., REAR (#312)	
* ST 4042	B	2	LINER, BALL BRG., INTM. BEVEL PINION, REAR	SEE ST 4048
ST 4043	C	2	FITTED SHIM, BALL BRG. CARRIER, INTM. BEVEL PINION, REAR	
ST 4044	B	2	LOCKNUT, BALL BRGS., INTM. BEVEL PINION, REAR	
ST 4045	B	2	RETAINER, CROSS SHAFT	
ST 4046	C	1	LOCKNUT, UPPER BRG., MAIN SHAFT	
ST 4047	A	1	SHAFT SEAL, MAIN ROTOR SHAFT (C/R #72539)	
ST 4048	D	2	BRG. CARRIER ASS'Y, INTM. BEVEL PINION, REAR	
ST 4049-1	A	1	RET. RING, SPIROLOX (RAMSEY #RR-456)	
ST 4049-2	A	2	RET. RING, SPIROLOX (RAMSEY #RR-700)	
ST 4049-3	A	2	RET. RING, SPIROLOX (RAMSEY #RR-256)	

PART NO.	DWG. SIZE	NO. REQ'D	DESCRIPTION	REMARKS
ST 4049-4	A	2	RET. RING, SPIROLOX (RAMSEY #RR-293)	
ST 4049-5	A	1	RET. RING, SPIROLOX (RAMSEY #RS-157)	
ST 4049-6	A	1	RET. RING, SPIROLOX (RAMSEY #RR-875)	
ST 4050	D	1	SEAL RETAINER, MAIN SHAFT	
ST 4051	D	1	FLANGED LINER, UPPER BRG., MAIN SHAFT	
ST 4052	A	1	BALL BRG., UPPER, MAIN SHAFT (KAYDON #KG 075 CPO)	
ST 4053	D	2	CROSS SHAFT	
ST 4054	C	1	PLUG, MAIN SHAFT	
ST 4055-1	A	4	PLUG, LEE (#156001)	
ST 4055-2	A	7	PLUG, LEE (#343001)	
ST 4055-3	A	1	PLUG, LEE (#187101)	
ST 4055-4	A	1	PLUG, LEE (#281001)	
ST 4055-5	A	5	PLUG, LEE (#156101)	
ST 4056	A	2	SUPPORTING RING, CROSS SHAFT RETAINER	
ST 4057	B	2	FITTED SHIM, INTM. BEVEL PINION, FRONT	
ST 4058	D	2	BEVEL PINION ASSEMBLY, INTM., FRONT	

PART NO.	DWG. SIZE	NO. REQ'D	DESCRIPTION	REMARKS
ST 4059	D	2	BEVEL GEAR, INPUT	
ST 4060	C	2	RET. PLATE, BALL BRG., INPUT BEVEL GEAR	
ST 4061	A	2	BALL BRG., INPUT BEVEL GEAR (#1022)	
* ST 4062	B	2	LINER, BALL BRG., INPUT BEVEL GEAR	SEE ST 4083, 4084
ST 4063	A	2	FITTED SHIM, BALL BRG., INPUT BEVEL GEAR	
* ST 4064A	D	2	COVER, TORQUE DIVIDER (CASTING)	SEE ST 4148,4199
ST 4065	C	2	RING GEAR, TORQUE DIVIDER	
ST 4066	C	20	PINION, PLANET, TORQUE DIVIDER	
ST 4067	A	40	NEEDLE BRG., TORQUE DIVIDER (TORRINGTON #B-78)	
ST 4068	B	2	RETAINER PLATE, PIN, PLANET PINION	
ST 4069	A	20	PIN, PLANET PINION	
ST 4070	C	2	SPUR GEAR, SUN	
ST 4071	C	2	CARRIER, RING GEAR	
ST 4072	A	2	LOCKNUT, RING GEAR CARRIER (WHITTET-HIGGINS #BH-08)	
ST 4073	B	2	LOCKNUT, INTM. BEVEL PINION, FRONT	
ST 4074	D	2	CARRIER, PLANET PINION	
ST 4075	B	2	LOCKNUT, BRGS.,INPUT BEVEL GEAR	
ST 4076	A	2	FRONT TAPERED ROLLER BRG.INTM. BEV. PIN., FRONT (TIMKEN)	
ST 4077	A	2	FITTED SPACER, TAPERED ROLLER BRG., FRONT	

PART NO.	DWG. SIZE	NO. REQ'D	DESCRIPTION	REMARKS
ST 4078	D	2	HOLLOW SUPPORT SHAFT, INPUT BEVEL GEAR	
ST 4079	D	2	RET. CAP, REAR ROLLER BRG., INPUT BEVEL GEAR	
ST 4080	A	2	REAR TAPERED ROLLER BRG.,INTM. BEV. PIN. FRONT (TIMKEN)	
ST 4081	A	2	CYL. ROLLER BRG., REAR, INPUT BEV. GEAR (NU-220 LESS INNER RACE)	
* ST 4082	B	2	LINER, ROLLER BRG., REAR, INPUT BEVEL GEAR	SEE ST 4130, 4131
ST 4083	E	1	INPUT HOUSING ASSEMBLY, R.H. (3 SHEETS)	
* ST 4083A	E	-	INPUT HOUSING ASSEMBLY, R.H. (CASTING)(3 SHEETS)	SEE ST 4083
ST 4084	E	1	INPUT HOUSING ASSEMBLY, L.H. (3 SHEETS)	
* ST 4084A	E		INPUT HOUSING ASSEMBLY, L.H. (CASTING)(3 SHEETS)	SEE ST 4084
* ST 4085-1	A	92	SCREW THREAD INSERT - HELI - COIL #3591-4CN-0375	SEE ST 4001,4002,4083,4084, 4140
* ST 4085-2	A	104	SCREW THREAD INSERT - HELI - COIL #3591-5CN-0625	SEE ST 4001,4083,4084
* ST 4085-3	A	175	SCREW THREAD INSERT - HELI - COIL #3591-3CN-0285	SEE ST 4001,4002,4048,4083, 4131, 4131, 4084,4089,4118,4130,
* ST 4085-4	A	2	SCREW THREAD INSERT - HELI - COIL #3591-5CN-0469	SEE ST 4001
* ST 4085-5	A	12	SCREW THREAD INSERT - HELI - COIL #3591-6CN-0750	SEE ST 4002
ST 4086	A	2	BALL BEARING, INPUT BEVEL PINION (#212)	
ST 4087	A	2	LOCKNUT, REAR BRGS., INPUT BEVEL PINION	
ST 4088	A	2	MATING RING, FACE SEAL, INPUT FLANGE	

PART NO.	DWG. SIZE	NO. REQ'D	DESCRIPTION	REMARKS
ST 4089	D	1	HOUSING ASSEMBLY, R.H., INPUT BEVEL PINION	
ST 4090	B	2	END CAP, INPUT BEVEL PINION	
ST 4091	C	2	FLANGED SHAFT, INPUT	
ST 4092	B	2	LUBE. RING, FRONT BRG., INPUT BEV. PINION	
ST 4093	B	2	CLAMPING BOLT, INPUT SHAFT	
ST 4094	A	2	DRAIN SLEEVE, INPUT HOUSING	
ST 4095	A	2	FACE SEAL, INPUT FLANGE (EG&G SEALOL - TYPE 792)	
ST 4096	C	2	FITTED SHIM, INPUT BEVEL PINION HOUSING	
ST 4097	A	2	SPACER, REAR BRG., INPUT BEVEL PINION	
ST 4098	B	2	SPACER/OILER, REAR BRGS., INPUT BEVEL PINION	
ST 4099	A	4	CYL. ROLLER BEARING, INPUT PINION (NU-212)	
* ST 4100	B	2	LINER, REAR BRGS., INPUT BEVEL PINION	SEE ST 4089, 4157
ST 4101	B	2	RET. PLATE, REAR BRG., INPUT BEVEL PINION	
ST 4102	D	2	BEVEL PINION, INPUT	
* ST 4103	A	2	LINER, FRONT BRG., INPUT PINION	SEE ST 4083, 4084
ST 4104	B	2	LOCKNUT, FRONT BRG., INPUT BEVEL PINION	
ST 4105	D	2	SPUR GEAR, PUMP DRIVE	
ST 4106	A	2	LOCK RING, FRONT BRG., INPUT BEVEL PINION	
* ST 4107-1	A	6	PULL DOWEL (JERGENS #31706)	SEE ST 4083, 4084

PART NO.	DWG. SIZE	NO. REQ'D	DESCRIPTION	REMARKS
* ST 4107-2	A	1	DOWEL (5/16 DIA,x7/8 LG. STD.)	SEE ST 4001
* ST 4107-3	A	1	DOWEL (DIAMOND)(5/16 DIA.x7/8 LG., ALTER)	SEE ST 4001
ST 4108	C	2	RET. PLATE, FRONT BRG., INPUT BEVEL PINION	
ST 4109	B	2	SPACER/RETAINER, SUN SPUR GEAR	
ST 4110	A	2	RET. PLATE, SUN SPUR GEAR	
ST 4111	D	1	BEVEL GEAR, TAIL DRIVE	
ST 4112	A	12	THREADED BUSHING, BEVEL GEAR, TAIL DRIVE	
ST 4113	C	2	COVER, INTM. BEVEL GEAR, REAR	
ST 4114	B	2	COVER, CROSS SHAFT	
ST 4115	A	2	WASHER, CARRIER, PLANET PINIONS	
ST 4116	C	1	SEAL CARRIER, TAIL DRIVE	
ST 4117	D	1	BEVEL PINION, TAIL DRIVE	
ST 4118	D	1	HOUSING ASSEMBLY, TAIL DRIVE	
ST 4119	C	1	LINER/SPACER, ROLLER BRG., TAIL DRIVE	
ST 4120	B	1	SPACER, TAIL DRIVE BRGS.	
ST 4121	A	1	BALL BEARING, BEVEL PINION, TAIL DRIVE (#311)	
* ST 4122	B	1	LINER, BALL BRG., TAIL DRIVE	SEE ST 4113
ST 4123	B	1	PLUG, TAIL DRIVE PINION	

PART NO.	DWG. SIZE	NO. REQ'D	DESCRIPTION	REMARKS
ST 4124	D	1	FLANGE, TAIL DRIVE	
ST 4125	A	1	LOCKNUT, BEVEL PINION, TAIL DRIVE	
ST 4126	C	1	SHIM, FITTED, TAIL DRIVE HOUSING	
ST 4127	A	1	CYL. ROLLER BRG., BEVEL PINION, TAIL DRIVE	
ST 4128	C	1	RET. PLATE, TAIL DRIVE BRG.	
ST 4129	C	2	SPLINED SLEEVE, SUN SPUR GEAR	
ST 4130	E	1	BEARING RETAINER HOUSING ASS'Y, R.H., INTM.	
* ST 4130A	E	1	BEARING RETAINER HOUSING, R.H. (CASTING), INTM.	SEE ST 4130
ST 4131	E	1	BEARING RETAINER HOUSING ASS'Y, L.H., INTM.	
* ST 4131A	E	1	BEARING RETAINER HOUSING, L.H. (CASTING), INTM.	SEE ST 4131
* ST 4132	A	2	BUSHING, CROSS SHAFT	SEE ST 4058
ST 4133	D	2	BEVEL GEAR, INTM., FRONT	
* ST 4134	A	30	STUD, LOCKED IN, INPUT HOUSING (MS-51833-203)	SEE ST 4083,4084
ST 4135	A	2	CYL. ROLLER BRG., FRONT,INPUT BEVEL GEAR (NU 1922)	
* ST 4136	A	2	LINER, FRONT ROLLER BRG., INPUT BEVEL GEAR	SEE ST 4083,4084
ST 4137	C	2	SPACER/OILER, FRONT BRGS.,INPUT BEVEL GEAR	
ST 4138	A	2	SPACER, BEARINGS, INPUT BEVEL GEAR	
ST 4139	A	2	LUBE PUMP	
ST 4140	D	2	HOUSING ASSEMBLY, LUBE PUMP	

PART NO.	DWG. SIZE	NO. REQ'D	DESCRIPTION	REMARKS
ST 4141	C	2	RETAINER, LUBE PUMP HOUSING	
ST 4142	A	1	LUBE JET, UPPER BRG., MAIN SHAFT	
ST 4143	A	1	LUBE JET, LOWER BRG., MAIN SHAFT	
ST 4144	B	2	LUBE JET, ROLLER BRG., INTM. BEVEL PINION, REAR	
* ST 4145	B	4	LINER, UPPER BRG., INTM. BEVEL GEAR	SEE ST 4001
ST 4146	C	2	RET. PLATE, FRONT ROLLER BRG., INPUT BEVEL GEAR	
ST 4147	A	20	SPACER, BRGS., PLANET PINION	
ST 4148	D	1	COVER, TORQUE DIVIDER, R.H.	
ST 4149	D	1	COVER, TORQUE DIVIDER, L.H.	
ST 4150	B	4	LUBE ROD, COMB. GEAR & PINION, L.H.	
ST 4151	B	4	LUBE ROD, COMB. GEAR & PINION, R.H.	
ST 4152	A	3	COVER PLATE, LUBE PORT	
* ST 4153-1	A	2	BUSHING, MTG. BOLT (UNIVERSAL ASA #H-56-22)	SEE ST 4002
* ST 4153-2	A	4	BUSHING, MTG. BOLT (UNIVERSAL ASA #H-56-34)	SEE ST 4002
* ST 4153-3	A	2	BUSHING, MTG. BOLT (UNIVERSAL ASA #H-56-48)	SEE ST 4002
ST 4154	A	8	CLAMP, MTG. BOLT	
ST 4155-1	A	2	MTG. BOLT (NAS 6710-42 ALTER)	
ST 4155-2	A	4	MTG. BOLT (NAS 6710-56 ALTER)	

PART NO.	DWG. SIZE	NO. REQ'D	DESCRIPTION	REMARKS
ST 4155-3	A	2	MTG. BOLT (NAS 6710-64 ALTER)	
ST 4156-1	A	2	FLAT HD. SOC. CAP SCREW #10-32 UNF-3A$\times\frac{1}{2}$ LG.	
ST 4157	D	1	HOUSING ASS'Y, L.H., INPUT BEVEL PINION	

(*) ASTERISK INDICATES THAT THE PART IS ALSO LISTED ON THE CORRESPONDING SUBASSEMBLY DRAWING

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16. Abstract

Final design details of a helicopter transmission that is powered by twin T-700 engines each rated at 1800 HP are presented. In comparison with conventional helicopter arrangements, the split-torque design offers:

a. A weight reduction of 15%

b. A reduction in drive trains losses of 9%

c. Improved reliability resulting from redundant drive paths between the two engines and the main shaft.

The transmission is retrofittable into the NASA - Lewis Research Center's test stand without any changes to the test stand other than a simple change in gear ratio in the tail drive gearbox. It allows for uprating of engine input power from 3600 to 4500 HP, thus providing a base for a research and development effort targeted at improving the performance and reliability of helicopter transmissions.

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