Design of an Advanced 500-hp Helicopter Transmission*

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The emphasis on the design of the NASA/Bell Helicopter Textron (BHT) 500-hp advanced technology demonstrator transmission, which subsequently will be referred to as the 500-hp transmission, was placed on designing a 500-hp version of the OH-58C 317-hp transmission that would have the capabilities for a long, quiet life at a minimum increase in the cost, weight, and space that usually increases along with power increases. This was accomplished by implementing advanced technology that has been developed during the last decade and making improvements dictated by field experience.

These advanced technology components, concepts, and improvements and their anticipated effect on the 500-hp transmission are

- High contact ratio spur gear teeth in the planetary will tend to reduce the noise level and increase gear life.
- Spiral bevel gears made of cleaner gear steels, stronger gear steels, vacuum carburized, shot peened for increased gear tooth pitting fatigue life, as well as gear tooth bending fatigue strength, and lubricated with Aeroshell 555 oil will save weight and space and increase gear life.
- Bearings made of cleaner steels in conjunction with improved analytical tools required to predict the effect of these cleaner steels and other variables on the fatigue life of a bearing will save weight and space and increase the reliability.
- The investment cast planet carrier, hot isostatic pressed, as a replacement for the planet carrier made of two-piece construction of forging and plate stock will reduce cost.
- The investment cast stainless-steel top case, also hot isostatic pressed and shot peened, will provide maximum corrosion resistance, excellent fatigue strength, no significant creeping, and good strength-to-weight ratio, all at a reasonable cost.
- The cantilever-mounted planetary ring gear has no working spline to generate wear debris; it isolates the meshing teeth from the housing for noise reduction; and it provides a soft mount for a more uniform load distribution among the planets.
- The sun gear still has its working spline; however, it is crown hobbed and hardened and will be submerged in a bath of flow-through oil, which should prevent the spline from wearing.
- The straddle-mounted bevel gear will enable higher torques to be transmitted without destroying the tooth contact patterns.

Figure 1 shows the 500-hp transmission, and figure 2 shows the 317-hp transmission; the latter is shown without the dummy mast and the mast ball bearing. Not shown on the 500-hp transmission, but a part of it, is the oil pump and hydraulic pump drive quill identical to that shown on the 317-hp transmission.

The system of units used for the principal measurements and calculations for the data in this paper are the customary units and these are shown in parentheses.

High Contact Ratio Spur Gear Planetary

High contact ratio (HCR) spur gearing is not new; however, the first application of this concept by BHT was the transmission for the Model 222, designed in 1974. There are approximately 50 Model 222 helicopters in service at this writing, with the HCR spur gears in the planetary. The Model 222

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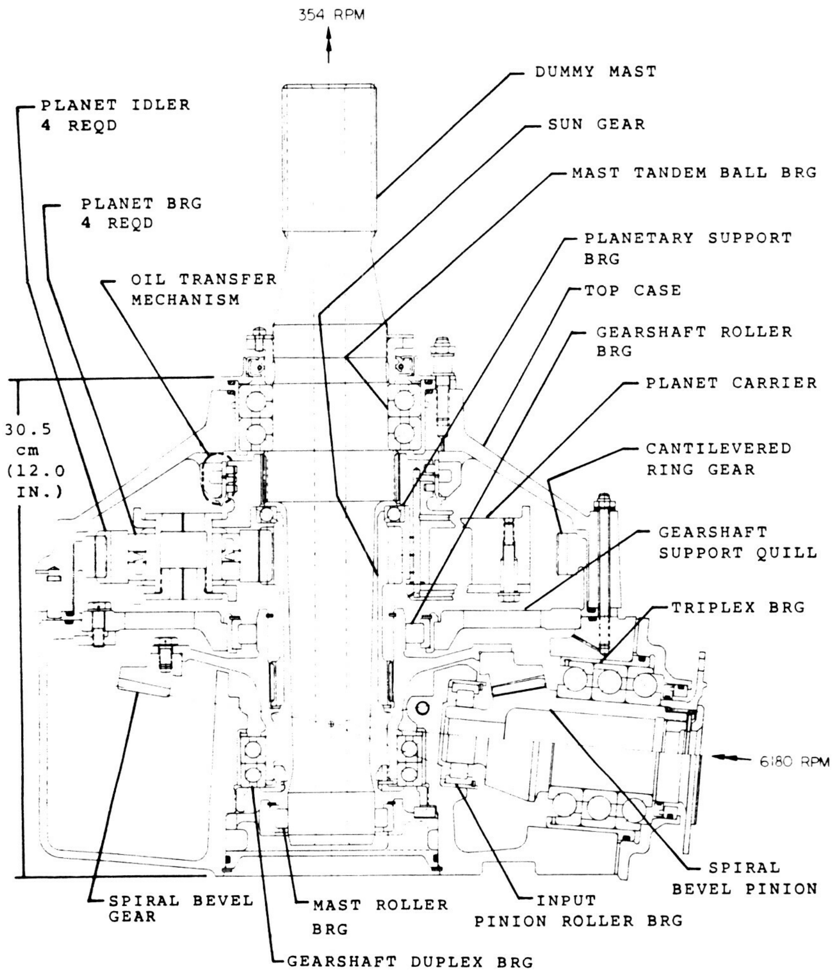

Figure 1. - NASA/BHT 500-hp transmission.

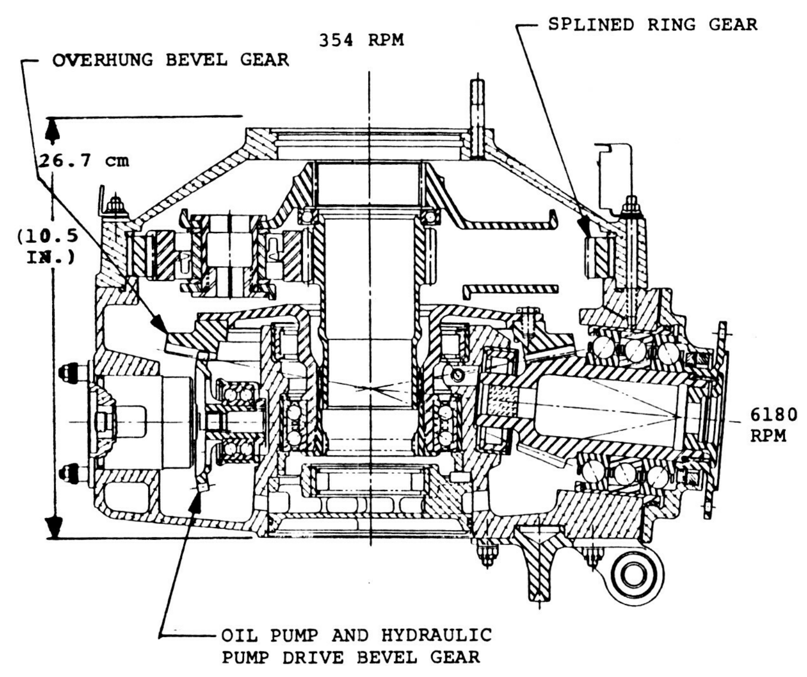

Figure 2. - OH-58C 317-hp transmission.

planetary is the same size and ratio as the final stage standard (STD) spur gear planetary in BHT's XV-15 tilt rotor aircraft transmission designed in 1969. The reason for selecting the HCR design over the STD design for the Model 222 is summarized in table 1. A study performed by BHT showed that when both planetaries are sized to transmit the same torque with identical gear tooth root bending fatigue stresses (per AGMA with all K-factors equal to 1), the HCR spur gear planetary is lighter, generates less noise, and has a longer life. The tooth numbers and the ratio of the Model 222 planetary made it easy to transform an STD design into an HCR design.

To advance the technology of planetary gearing, NASA Lewis contract NAS3-21595 specified that the planetary for the NASA/BHT 500-hp transmission be of an HCR design. NASA Lewis was aware that BHT had the HCR planetary designed, developed, and in production for the Model 222, and they were equally aware that it would not be feasible to transform all STD planetary designs into HCR designs because of the effects that the numbers of teeth and the ratio have on the transformation. Table 2 shows some basic data for the STD planetaries in the XV-15 transmission and the 317-hp transmission. Notice that the sum of the numbers of teeth of the sun gear and planet idler is 91 for the XV-15 transmission and 62 for the 317-hp transmission. Referring to figure 3, one can readily see the effects of the sum of the numbers of teeth on the profile contact ratio. The profile contact ratio of a gearset is defined as the number of times the base pitch is contained in the active length of line of action. Therefore, the maximum profile contact ratio on a particular gearset exists when the active length of line of contact is equal to the total length of line of contact, that is, when the active length of contact extends from the tangency point on the base circle of one gear to the tangency point on the base circle of the mating gear. Figure 3 shows that the maximum profile contact ratio on the 48 × 43 gearset (48-tooth sun gear driving a 43-tooth planet) is 5.27 as compared with only 3.59 for the 27 × 35 gearset.

Figure 3 also shows the active line of action for a contact ratio of 2.000 on each gearset. Even though both active lines of contact are centered about the pitch point (that is, the approach action zone and the recess action zone are equal), it can be seen that more flexibility exists for the 48 × 43 gearset should it be desirable to shift the active portion of the line of action more into the recess action zone. Any significant amount of shifting of the active line of contact on the 27 × 35 gearset will result in large increases in sliding velocities since the ends of the active line of contact (the first and last points of contact) are relatively close to the base circles of the two gears.

It can be shown that the maximum profile contact ratio for different gearsets varies directly as the sum of their numbers of teeth. It can also be shown that there are only two variables that affect the maximum profile contact ratio: the sum of the numbers of teeth of the gearsets and their pressure angle. The increase in either variable causes an increase in the maximum profile contact ratio.

The discussion to this point has made no mention of the physical limitations that usually prevent a gearset from having the maximum profile contact ratio. These limitations are pointed teeth (zero top land width), teeth too thin on one member or the other, and root clearance. At the higher

TABLE 1. COMPARISON OF A STANDARD AND A HIGH CONTACT RATIO SPUR GEAR PLANETARY

	WT, Kg(lb)	EFF, %	NOISE, db	LIFE
Standard (XV-15 Final Stage)	18 (40)	99.7	X	X
High Contact Ratio (Model 222)	17 (38)	99.4	X-9.5	2X

Both planetaries are sized to transmit the same torque with the same gear tooth bending stresses.

TABLE 2. BASIC DESIGN DATA FOR THE STANDARD PLANETARIES IN THE XV-15 TRANSMISSION AND THE 317-HP TRANSMISSION

XV-15 TRANSMISSION

	Sun	Planet	Planet	Ring
Number of teeth	48	43	43	138
Diametral pitch	11.375	11.375	11.875	11.875
Pressure angle, deg	24.95	24.95	18.82	18.82
Pitch diameter, cm(in.)	10.7183 (4.2198)	9.6017 (3.7802)	9.1976 (3.6211)	29.5176 (11.6211)

317 HP TRANSMISSION

	Sun	Planet	Planet	Ring
Number of teeth	27	35	35	99
Diametral pitch	8.8571	8.8571	9.1429	9.1429
Pressure angle, deg	24.60	24.60	20.19	20.19
Pitch diameter, cm(in.)	7.7429 (3.0484)	10.0371 (3.9516)	9.7234 (3.8281)	27.5034 (10.8281)

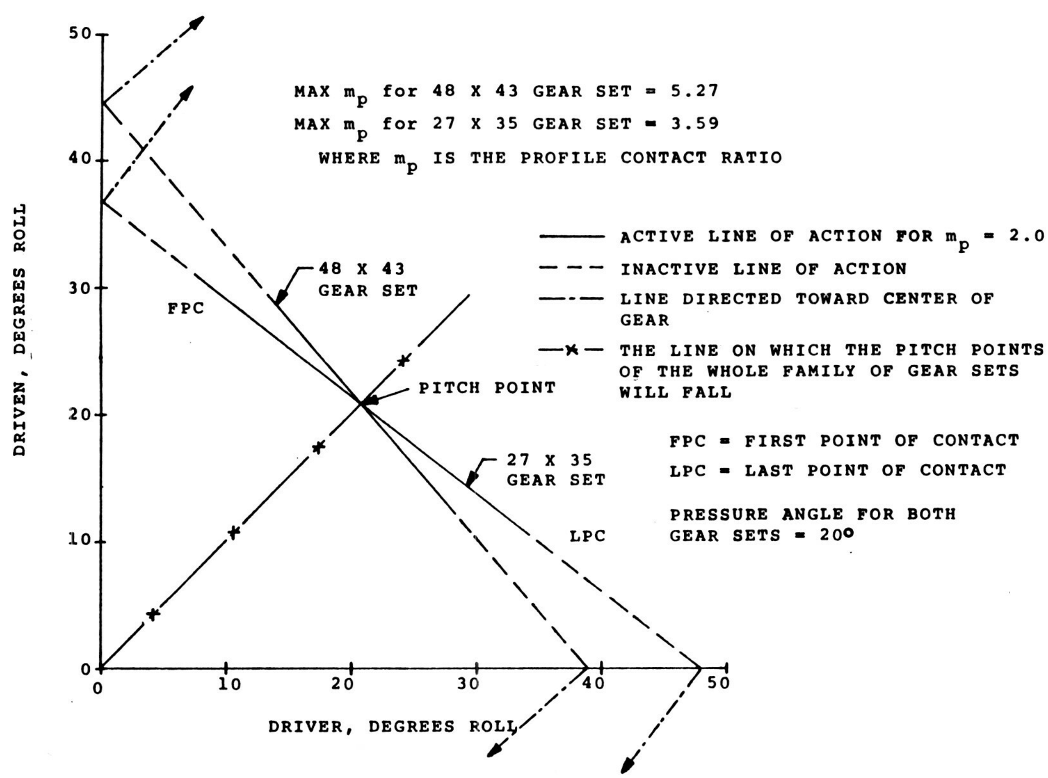

Figure 3. - Relationship of tooth contact ratios on gearsets of different numbers of teeth but with equal pressure angles.

pressure angles, the teeth become pointed. When the teeth on one member are increased in thickness to prevent them from becoming pointed, the root clearance is reduced, and the teeth on the mating gear become thinner and weaker.

Table 3 shows the design data for the selected planetary gears for the 500-hp transmission. The gears are shown in figure 1. Several iterations were made in the selection process: two at a pressure angle of 18° and one each at pressure angles of 20°, 22°, and 24.95°. The 20° pressure angle resulted in the best compromise between top land width, tooth thickness, and root clearance.

Table 4 shows the stress analysis (per AGMA) of the selected planetary gears, and for comparison the stress analysis of the planetary gears in the 317-hp transmission is also shown. The significance of this comparison is that for the same weight (the corresponding gears have equal face widths) both the planetaries appear to be capable of meeting their performance requirements. Even though the 500-hp planetary is more highly stressed, it does not exceed the allowables used at BHT. But the comparison between an HCR and STD gear planetary can be made only if they are sized to some common baseline such as the same torque with equal tooth bending stresses or the same torque with equal weights. The latter method will be used in this case as the 500-hp planetary of HCR design is compared with the 317-hp planetary of STD design. Their weights are equal, and, to make their torques equal, the 317-hp planetary will be stressed at the 500-hp level. The results can be seen in table 4. The tooth bending stress of 367 MPa (53 200 psi) on the planet of the 500-hp transmission is the most severe of the bending stresses because of the reversed bending that is inherent in a planet idler. This value is 1 percent higher than the bending stress of 363 MPa (52 600 psi) on the 317-hp planet operating at 500 hp; however, it is still below the allowable used at BHT.

The Hertz stress of 1420 MPa (205 900 psi) on the sun-planet mesh of the 317-hp transmission operating at 500 hp is the most severe of the compressive stresses. This value is 10 percent higher than the Hertz stress of 1286 MPa (186 500 psi) on the 500-hp planet. Transforming these compressive stress values into gear tooth pitting lives, using the relationship that the life is inversely proportional to the Hertz stress raised to the 5.22 power, shows that the L_{10} life of the 317-hp planetary sun-planet mesh operating at 500 hp is only 60 percent of the L_{10} life of the same mesh in the 500-hp planetary.

TABLE 3. BASIC DESIGN DATA FOR THE HCR PLANETARY IN THE 500-HP TRANSMISSION

	Sun	Planet	Planet	Ring
Number of teeth	27	35	35	99
Diametral pitch	8.8571	8.8571	9.1429	9.1429
Pressure angle, deg	20	20	14.0682	14.0682
Pitch diameter, cm (in.)	7.7429 (3.0484)	10.0371 (3.9516)	9.7234 (3.8281)	27.5034 (10.8281)
OD/ID, cm (in.)	8.623 (3.395)	10.668 (4.200)	10.668 (4.200)	27.292 (10.745)
Root diameter, cm (in.)	7.036 (2.770)	9.080 (3.575)	9.080 (3.575)	28.537 (11.235)
Face width, cm (in.)	3.492 (1.375)	3.178 (1.251)	3.178 (1.251)	2.540 (1.000)
Top land width, cm (in.)	.114 (.045)	.076 (.030)	.076 (.030)	.372 (.147)
Contact ratio	2.086		2.086	
Number of planets		4		

TABLE 4. STRESS ANALYSIS OF THE PLANETARY GEARS IN THE 317 HP AND 500 HP TRANSMISSIONS AND IN THE 317 HP TRANSMISSION OPERATING AT 500 HP

	Tangential load, N (lb)	Bending stress, MPa (psi)	Hertz stress, MPa (psi)
500 hp Trans.			
Sun	14,176 (3,187)	253 (36,700)	1,286 (186,500)
Planet/Sun	14,176 (3,187)	325 (47,200)	1,286 (186,500)
Planet/Ring	14,629 (3,289)	367 (53,200)	859 (124,600)
Ring	14,629 (3,289)	381 (55,200)	859 (124,600)
317 hp Trans.			
Sun	8,989 (2,021)	201 (29,100)	1,131 (164,000)
Planet/Sun	8,989 (2,021)	230 (33,400)	1,131 (164,000)
Planet/Ring	9,274 (2,085)	214 (31,000)	698 (101,300)
Ring	9,274 (2,085)	285 (41,300)	698 (101,300)
317 hp Trans. Oper. @ 500hp			
Sun	14,176 (3,187)	316 (45,800)	1,420 (205,900)
Planet/Sun	14,176 (3,187)	363 (52,600)	1,420 (205,900)
Planet/Ring	14,629 (3,289)	336 (48,800)	878 (127,000)
Ring	14,629 (3,289)	449 (65,100)	878 (127,300)

(This comparison in life is no doubt dependent on the sliding velocity; however, since the effect of sliding velocity on pitting life is unknown at this time, no attempt will be made to adjust the pitting life ratio. The maximum sliding velocity at the sun-planet mesh is 234 cm/s (92 in/s) on the 500-hp HCR design and 163 cm/s (64 in/s) on the 317-hp STD design.)

Thus, it appears that an HCR spur gear planetary in the 500-hp transmission will have 1.68 times the gear tooth pitting life of a STD spur gear planetary in the 317-hp transmission operating at 500 hp. But the anticipated weight saving did not materialize. In fact, if the two planetaries had been sized for the same torque and the same tooth bending stress, the 500-hp HCR planetary would have been heavier than the 317-hp STD planetary operating at 500-hp. As shown in table 1, there was a weight saving in transforming the XV-15 STD planetary into the model 222 HCR planetary. This is attributed to the large numbers of teeth in the sun and planet gears and to the favorable ratio of the same gearset.

Spiral Bevel Gears

When a new spiral bevel gearset is required, the normal routine is to design it, test it, and put it into service. Any design or manufacturing deficiencies uncovered during the testing phase are corrected and the testing phase repeated. Little has been done in the last decade to change this routine except in the area of developing finite-element analysis and stress measurement techniques to aid in designing large spiral bevel gears where the tendency is to minimize gear tooth backup material and the web thickness in order to save weight. Thus, except for specialized applications, the real technological advancements in recent years that have affected, or that have the potential to affect, spiral bevel gears have to be the use of cleaner gear steels, stronger gear steels, vacuum carburization, and shot peenning for increased gear tooth pitting fatigue life as well as gear tooth bending fatigue strength, and the use of Aeroshell 555 oil for increased gear tooth pitting fatigue life.

The bevel gears in the 500-hp transmission take advantage of all five of these technological advancements as the gears increase in power capabilities with no change in size. The bevel gears will be made from two different types of steels: VIM VAR 9310 and VIM VAR EX-00053. Both steels will be produced by the vacuum induction melt (VIM) process followed by the vacuum arc remelt (VAR) process. The gears will be shot peened after finish grind, and the gears wil be lubricated with Aeroshell 555 oil.

The bevel gears in the 500-hp transmission are identical in size to those in the 317-hp transmission. Figure 1 shows the bevel gears and table 5 shows the stress analysis data on the same bevel gears operating in both transmissions. Shown also are the allowables that are generally used as guidelines at BHT. As can be seen, the bending stress on the pinion at 500 hp is 327 MPa (47 408 psi), 13 percent above the allowable of 290 MPa (42 000 psi) for spiral bevel gears made of VAR 9310 and conventionally (atmospheric) carburized, ground, and shot peened. There are two methods by which the 13 percent increase is planned to be overcome. One is using VIM VAR 9310, vacuum carburized, ground, and shot peened. The other is using VIM VAR EX-00053, vacuum carburized, ground, and shot peened. BHT is actively engaged in an internal R&D program to determine the magnitude of these benefits by performing single-tooth bending fatigue tests on 10-pitch spur gear teeth on a Sonntag machine. Table 6 shows how this will be accomplished. All tests are scheduled to be completed by the end of 1982.

In the process of designing the bevels for the 500-hp transmission, it was assumed that the combination of VIM VAR 9310 steel and vacuum carburizing would account for the tooth bending stress being 13 percent above the allowable for VAR 9310 and conventional carburizing. Data exist that allow substantial increases in gear tooth bending fatigue strength design limits for gears made of VAR 9310 versus air melt steels. Coleman (ref. 1) states that testing performed by Gleason Works justifies an increase of 78 percent in the allowable tooth bending stress for carburized VAR 9310 gears over carburized air melt steel gears. The degree of cleanliness going from air melt to VAR is no doubt much larger than going from VAR to VIM VAR; however, only 13 percent increase in strength is required.

Relative to vacuum carburizing, BHT had some fine pitch helical gears vacuum carburized and metallurgically evaluated. A higher carbon gradient was easily obtained with vacuum carburizing. This provides for more R_c 60 depth in the root fillet area of the teeth where the maximum bending stresses are present. Thus, it is anticipated that there will be some gain in tooth bending fatigure strength by vacuum carburizing. This gain would be in addition to the primary reason that vacuum carburizing is becoming so popular: cost and energy savings, which is clearly a technological advancement.

Should the combination of VIM VAR 9310 and vacuum carburizing not produce the 13 percent more bending fatigue strength required, backup bevel gears made of the new stronger VIM VAR EX-00053 gear steel are being provided. Although not thoroughly tested yet, indications are that conventional carburized VIM VAR EX-00053 is stronger than conventional carburized VAR 9310 (ref. 2). Whether the indicated increase in strength is due to material difference or steel processing difference (VAR versus VIM VAR) is not yet known; however, it is assumed at this time that the VIM VAR EX-00053 is stronger than the VIM VAR 9310. In addition to its strength, EX-00053 is a moderately high hot hardness material (tempered at 533 K (500° F)) and exhibits excellent fracture toughness properties (especially at normal transmission operating temperatures of 366 K (200° F)): 262 MPa $\sqrt{m}$ (238 ksi $\sqrt{in}$) for EX-00053 as compared with 85 MPa $\sqrt{m}$ (77 ksi $\sqrt{in}$) for CBS 600; 125

MPa $\sqrt{m}$ (114 ksi $\sqrt{in}$) for CBS 1000; 60 MPa $\sqrt{m}$ (55 ksi $\sqrt{in}$) for Vasco X-2M and 110 MPa $\sqrt{m}$ (100 ksi $\sqrt{in}$) for 9315. These data were taken from figure 25 in reference 3. The 9310 steel was not included in the Terratek work; however, the 9315 is probably close enough for a good comparison.

TABLE 5. STRESS ANALYSIS DATA ON THE SAME SPIRAL BEVEL GEARS AT TWO POWER LEVELS: 317 HP AND 500 HP

	Bending stress, MPa (psi)	Hertz stress, MPa (psi)	Calculated temp. rise, K (°F)
317 hp: Pinion	207 (30,057)	1,474 (213,728)	410 (277)
Gear	171 (24,868)		
500 hp: Pinion	327 (47,408)	1,851 (268,421)	449 (348)
Gear	270 (39,223)		
Allowables:			
VAR 9310	241 (35,000)	1,551 (225,000)	443 (338)
VAR 9310, Shot peened	290 (42,000)	1,668 (242,000)	443 (338)
VAR 9310, Shot peened, Aeroshell 555 oil	290 (42,000)	2,275 (330,000)	?

The allowable hertz stress values shown are the values for L_{10} life of 10^9 cycles.

For other L_{10} lives, use the following relationship:

$$L_{10} \propto \left(\frac{1}{\text{Hertz stress}}\right)^{5.22}$$

TABLE 6. BHT IR&D PROGRAM FOR SINGLE TOOTH BENDING FATIGUE TESTS ON VARIOUS MATERIALS AND PROCESSES (ALL SPECIMENS ARE SHOT PEENED)

Gear specimen material	Carburizing method	Carburizing temp., K (°F)	Quenching medium	Shot peen intensity, cm (in.)
VAR 9310	Conventional	1200 (1700)	Oil	.020A (.008A)
VAR 9310	Vacuum	1200 (1700)	Oil	.020A (.008A)
VIM VAR 9310	Conventional	1200 (1700)	Oil	.020A (.008A)
VIM VAR 9310	Vacuum	1200 (1700)	Oil	.020A (.008A)
VIM VAR 9310	Vacuum	1311 (1900)	Oil	.020A (.008A)
VIM VAR EX-00053	Conventional	1144 (1600)	Air	.020A (.008A)
VIM VAR EX-00053	Conventional	1144 (1600)	Oil	.020A (.008A)
VIM VAR EX-00053	Conventional	1144 (1600)	Oil	.033A (.013A)
VIM VAR EX-00053	Vacuum	1144 (1600)	Oil	.020A (.008A)

Referring again to table 5, one can see that the Hertz stress of 1851 MPa (268 421 psi) on the 500-hp gearset is significantly above the allowable normally used at BHT. There are two areas of advanced technology achievements that are being used to overcome the deterimental effects of the high compressive stresses: shot peening and Aeroshell 555 oil. Neither of these technological achievements are new; however, the effect of each on the gear tooth pitting fatigue life in helicopter transmissions has been only recently determined. This was done by a joint effort of NASA Lewis and BHT. With regard to shot peening, it has been known for many years that shot peening the root fillet areas of carburized gear teeth increases the tooth bending fatigue endurance limit by about 20 to 50 percent, but the effects of shot peening on tooth pitting fatigue life were not known. Thus, in 1980, NASA Lewis provided the ground-carburized spur gear specimens made of VAR 9310 steel. BHT did the shot peening on the roots and the working surfaces of the gear teeth, and NASA Lewis performed the tooth pitting tests on their gear test rigs. The results showed a 50 percent increase in the L_{10} life over unpeened gears of the same lot.

With regard to Aeroshell 555 oil, it has also been known for many years that the relatively low pitting life on helicopter transmission gears was primarily the result of being forced, for logistical reasons, to use either the MIL-L-7808 or MIL-L-23699 synthetic lubricants developed specifically for turbine engine use. These engine oils caused the gear designer to demand smoother and smoother ground tooth surfaces over the years, and finally, honed teeth in attempting to maintain separation of gear tooth contacts in low-speed final drive applications. In recent years in the commercial market where the customer is more concerned about transmission overhaul intervals (TBO) than whether the transmission and the engine use the same oil, the trend is shifting to oils of the same viscosity but containing friction modifiers in special additive packages. Aeroshell 555, a 5×10^{-6} m^2/s (5 cS) synthetic oil, falls into this category. In 1980 NASA Lewis again provided the carburized and ground spur gear specimens made of VAR 9310 steel, BHT furnished four different types of oil, and NASA Lewis again performed the gear tooth pitting tests. One of the oils was Aeroshell 500, which is qualified to the MIL-L-23699 specification. This oil was used as the reference oil. The results of the tests showed Aeroshell 555 oil the best with a 500 percent increase in the L_{10} life over the Aeroshell 500 reference oil. Transforming this 500-percent increase in life into a higher allowable Hertz stress (as shown in table 5) was done using the relationship that the life is inversely proportional to the Hertz stress raised to the 5.22 power. Thus, the allowable Hertz stress of 2275 MPa (330 000 psi) is significantly higher than the 1851-MPa (268 421-psi) operating stress shown for the 500-hp condition. Treating this 500-percent increase in life as a life adjustment factor of 5 at the L_{10} level also shows that the use of Aeroshell 555 oil is sufficient to overcome the increase in Hertz stress:

$$\frac{L_{10}\ 317\ \text{hp}}{L_{10}\ 500\ \text{hp}} \propto \left(\frac{500\ \text{hp}}{317\ \text{hp}}\right)^{5.22/z} = 3.29$$

which is less than 5.

The last column in table 5 (the calculated temperature rise of the tooth contacts of the bevel gearset, an indication of the gear tooth scoring tendency) shows that the value of 449 K (348° F) for the 500-hp gearset is 6 K (10° F) higher than the allowable of 443 K (338° F), which is the highest ever used at BHT. The allowable was established on a bevel gearset operating in MIL-L-23699 oil. Since Aeroshell 555 oil contains friction modifiers in its additive package that should reduce the amount of heat generated at the tooth contacts, thereby reducing the temperature rise, no problem is anticipated in exceeding the allowable shown by only 1.3 percent K (3.0 percent °F).

Thus, it appears that the advancements in technology of cleaner steels, stronger steels, vacuum carburizing, shot peening, and better oils can combine to allow a spiral bevel gearset, originally designed for 317 hp, to operate safely at 500 hp. The Hertz stress allowable increased in greater proportion than the Hertz stress such that the 500-hp bevel gearset has a 2.25 longer L_{10} life than the 317-hp bevel gearset. However, in the tooth bending category, the story is different. Battles (ref. 2) shows the rotating beam mean strength of oil-quenched EX-00053 to be 21 percent higher than that of AMS 6265 (VAR 9310). If this turns out to be true on gear teeth, then the allowable tooth bending stress for shot peened EX-00053 spiral bevel gears will be 350 MPa (50 800 psi) (which is larger than the 327-MPa (47 408-psi) bending stress on the 500-hp bevel pinion), but the tooth bending margin of safety of 0.17 on the 317-hp bevel pinion has dropped down to 0.07 on the 500-hp bevel pinion. If vacuum carburizing EX-00053 could raise its tooth bending strength 9 percent, then the allowable would be 381 MPa (55 300 psi), and the margin of safety would go back up to 0.17. Thus, whether or not the above advancements in technology will cover the 58 percent increase in tooth bending stress from the 317-hp transmission to the 500-hp transmission, without reducing the margin of safety will depend largely on the effects of vacuum carburizing.

Bearings

Probably the two most important recent technological advancements in the area of rolling element bearings are cleaner steels and improved analytical tools to predict the effect of these cleaner steels and other variables on the fatigue life of a bearing. An example of a cleaner bearing steel for the aircraft industry is the contemporary M-50 tool steel produced by the vacuum induction melt (VIM) process followed by the vacuum arc remelt (VAR) process. This bearing steel is usually referred to as VIM VAR M-50 steel. Some tests on high-speed ball bearings made of VIM VAR M-50 and operating in a 5×10^{-6} m^2/s (5 cS) type II oil qualified to the MIL-L-23699 specification have shown a 7.33 increase in the L_{10} life over similar bearings made of VAR M-50 (ref. 4).

An example of an improved analytical tool considered by BHT to be a significant technological advancement in the discipline of predicting bearing lives is the engineering design guide titled "Life Adjustment Factors for Ball and Roller Bearings" (ref. 5). This design guide was sponsored by the Rolling-Elements Committee of the Lubrication Division of the ASME and offers a unified approach to considering the many variables that affect the lives of bearings. The use of this guide enables the relationship between the designer and the user to be one of complete understanding. This is not to say that all bearings will operate as predicted, but it does mean that variables such as the kind of material, material processing, lubrication, speed, and shaft misalinement can be addressed in a meaningful and consistent manner.

Both of these technological advances have been used in the design of the 500-hp transmission. (See fig. 1 for the identification and location of the bearings.) Most of the bearings in this transmission are the same as used in the 317-hp transmission (fig. 2) except for a material change from a VAR steel to VIM VAR M-50 tool steel. The planetary support bearing and the mast roller bearing were not changed. Since there is very little difference between the weights of the OH-58C planetary and the 500-hp planetary, there was no need to change the planetary support bearing. The mast roller bearing was not changed because it has adequate life in its present condition. Table 7 shows the bearing lives for the 317-hp transmission and table 8 shows the bearing lives for the 500-hp transmission. All the bearing lives are calculated using a computer program from A. B. Jones, shown unadjusted and then adjusted in accordance with the aforementioned engineering design guide.

As previously mentioned, some tests indicate that a factor of 7.33 may be used when going from VAR M-50 to VIM VAR M-50. The design guide allows the use of a factor of 3 for VAR M-50. This would indicate that a factor of $7.33 \times 3 = 22$ may be used for VIM VAR M-50. However, in performing the life calculations shown in table 8, a conservative approach was taken: the 22 factor was divided by 2. This gives a life adjustment factor for processing of 11 for VIM VAR M-50.

Since all the torque-dependent bearings are not in the same physical position relative to each other in the two transmissions and thus the loads are not proportional to the power transmitted, the most meaningful method of comparing the lives of the two transmissions in terms of bearing lives only is to compare the mean lives of the two transmissions. This is best accomplished by summing the individual bearing lives and determining the mean life of the sum.

The individual lives of the bearing in each transmission may be summed as follows:

$$\frac{1}{L} = \left[\sum_{i=1}^{k} N_i \left(\frac{1}{L_i} \right)^e \right]^{1/e}$$

where

k total number of different bearings

N total number of like bearings of kind i

e dispersion exponent (Weibull slope)

TABLE 7. SUMMARY OF BEARING LIVES FOR 317-HP TRANSMISSION AT 270 HP AND 6180 RPM INPUT

Bearing	Unadjusted L_{10} life, hr	Life adjustment factors D	E	F	G	H	Total	Adjusted L_{10} life, hr
Triplex	1,238	2	3	2.00	1	1	12.00	14,856
Input pinion roller	3,343	2	3	.74	1	1	4.44	14,843
Gearshaft duplex	659	2	3	.3	1	1	1.80	1,186
Gearshaft roller	1,961	2	3	.34	1	1	2.04	4,000
Planet roller	3,723	2	3	.30	1	1	1.80	6,701
Planetary support	26,602,000	2	3	.30	1	1	1.80	47,883,600
Mast tandem	1,849	2	3	.30	1	1	1.80	3,328
Mast roller	22,429	2	3	.30	1	1	1.80	40,372

TABLE 8. SUMMARY OF BEARING LIVES FOR 500-HP TRANSMISSION AT 427 HP AND 6180 RPM INPUT

Bearing	Unadjusted L_{10} life, hr	Life adjustment factors D	E	F	G	H	Total	Adjusted L_{10} life, hr
Triplex	362	2	11	1.90	1	1	41.80	15,132
Input pinion roller	630	2	11	.66	1	1	14.52	9,148
Gearshaft duplex	477	2	11	.30	1	1	6.60	3,148
Gearshaft roller	935	2	11	.33	1	1	7.26	6,788
Planet roller	654	2	11	.30	1	1	6.60	4,316
Planetary support	26,602,000	2	3	.30	1	1	1.80	47,883,600
Mast tandem	1,399	2	11	.30	1	1	6.60	9,233
Mast roller	6,695	2	3	.30	1	1	1.80	12,051

D = Material factor
E = Processing factor
F = Lubrication factor
G = Speed factor
H = Misalignment factor

Total = D x E x F x G x H

The dispersion exponent e is given the value 4, based on test experience at BHT using MIL-L-7808 and MIL-L-23699 oils and VAR steels. Because of the VAR bearing steels and MIL-L-23699 oil in the 317-hp transmission and the VIM VAR bearing steels and Aeroshell 555 oil in the 500-hp transmission, it may be logical to use different values of e; however, BHT has few test data on this subject, and the results are inconclusive. Thus, the dispersion exponent of 4 is used for both transmissions in this analysis.

After the lives have been summed as shown above, the L_{10} life will be transformed into the characteristic life, $L_{63.2}$, from a Weibull plot by extending the L_{10} life up slope e to the $L_{63.2}$ position as follows:

$$\log L_{63.2} = \frac{\log \log \dfrac{1}{1-0.632} - \log \log \dfrac{1}{1-F(t)}}{e} + \log t$$

where

t percent failed life L

$F(t)$ probability of failure prior to time t

If $t = L_{10}$ life, then $F(t) = F(L_{10}) = 0.10$. The mean life is now calculated (ref. 6) as

$$\bar{x} = L_{-63.2}\Gamma\left(1 - \frac{1}{e}\right)$$

where

$\bar{x}$ mean life

Γ gamma function

If $e = 4$, then

$$\Gamma\left(1 + \frac{1}{4}\right) = 0.9064$$

If $e = 1$, then

$$\Gamma\left(1 + \frac{1}{1}\right) = 1.000$$

The mean life is the mean time between failures (MTBF) or the time to one failure.

Table 9 shows the results of these calculations and how they compare for the two transmissions. As expected, the mean life of 3703 hr for the 500-hp transmission is greater than the 1872 hr for the 317-hp transmission.

This large increase in the mean life is primarily caused by two factors: the processing factor of 11 for VIM VAR M-50, and the reduced load (reduced in proportion of the 58 percent increase in torque) on the gear shaft duplex bearing as the result of straddle mounting the spiral bevel gear mesh that lets the gearshaft roller bearing carry more of the load. By virtue of the method used to sum the lives of all the bearings, the mean life of the system of bearings can never be greater than that of the bearing with the shortest adjusted life. In both transmissions the shortest adjusted life bearing is the gearshaft duplex bearing. Thus, the change from the overhung mounting on the 317-hp transmission to the straddle mounting on the 500-hp transmission not only stiffened the bevel gear mesh but also increased the mean life of the bearings.

Since the data substantiating the large life adjustment factor of 11 are limited, shown also in table 9 are the bearing life data for the 500-hp transmission using a life adjustment factor for processing of 3.91 instead of 11. This 3.91 factor was obtained recently in some limited tests performed by BHT on some low-speed, highly loaded roller bearings made of VIM VAR M-50. (This work was performed under contract DAAJ02-76-C-0046, Advanced Transmission Components

TABLE 9. COMPARATIVE LIFE DATA OF THE 317-HP AND 500-HP TRANSMISSIONS IN TERMS OF BEARING LIVES ONLY

Hour	317-hp Trans. E = 3	500-hp Trans. E = 11	E = 3.91
Sum of L_{10} lives	1177	2328	828
Characteristic life, $L_{63.2}$	2065	4086	1453
Mean life (MTBF)	1872	3703	1317

E = Life adjustment factor for steel processing

Investigation, with the Applied Technology Laboratory of the U.S. Army Research and Technology Laboratories (AVRADCOM), Fort Eustis, Virginia.) Thus, in the event that the 3.91 factor is correct, the system of bearings in the 500-hp transmission has a mean life of 1317 hr which is significantly less than the 1872 hr shown for the 317-hp transmission, but only 12 percent less than the 1500 hr existing on the contemporary transmissions in service today.

Investment-Cast Planet Carrier

Advanced technology items are seldom accompanied with cost reductions; however, the investment-cast planet carrier is a distinct exception. The current method of making planet carriers at BHT is of a two-piece construction of forging and plate stock, bolted together as shown in figure 1. The investment cast planet carrier concept is currently being developed for the new Jet Ranger IV helicopter. Using the investment-cast one-piece construction concept to replace the two-piece planet carrier on the Jet Ranger IV, the recurring cost savings are estimated to be 50 percent. The carrier represents 14 percent of the cost of the transmission. Thus, for the Jet Ranger IV helicopter, the cost of the transmission would be reduced by a one-piece investment-cast carrier. For a $30 000 transmission, this represents a cost savings of $2100.

The investment-cast planet carrier for the 500-hp transmission will be similar to the carrier shown in figure 1, except that it will be of a one-piece construction. It is the same size as the carrier in the Jet Ranger IV transmission. The primary difference between the two carriers is that the 500-hp carrier also serves as the oil distributor to the sun-planet gear meshes, the planet bearings, and the sun gear spline. The many oil passages required in the distribution of the oil make the casting concept superior to the forging-plate concept.

The investment-cast carrier concept was selected for the 500-hp transmission as a means of expanding the concept to include the oil distribution tasks. The two-piece planet carrier on the Model 222 has practically the same oil distribution task as required on the 500-hp transmission, and, because it is machined from a forging, the oil distribution feature becomes expensive. The casting concept should lessen this expense significantly.

The anticipated payoff is high; however, the risk is not without significance. There are some differences between the forging-plate and the casting concept that must be investigated thoroughly before putting the cast carrier into service. These differences are related to material and material processing differences: VAR 4340 forging and air melt 4140 plate for the two-piece carrier, and investment-cast 15-5 PH stainless steel for the one-piece carrier. The following potential problem areas exist and are being investigated during the development of the investment-cast carrier for the Jet Ranger IV transmission:

- Impact resistance
- Fracture toughness
- Notched and smooth fatigue strength
- Stress corrosion cracking
- Fretting and wear
- Assembly abuse (pressing planet shafts in and out)
- Torsional spring rate of the carrier

The first three parameters are being checked on cast specimens configured for the particular test involved. The last four parameters will be checked on an actual investment-cast carrier on bench test. The last four parameters will also be evaluated on the investment-cast carrier for the 500-hp transmission as it is being tested.

To get the cast carrier to perform more like the original two-piece wrought carrier, the cast carrier will be subjected to the hot isostatic pressing (HIP) process before final heat treatment. The HIP process should increase the castings' impact resistance and fracture toughness, but this has not been verified yet. It does have an effect on the smooth rotating beam (R. R. Moore) fatigue strength of cast 15-5 PH specimens. (See fig. 4.) It increases the mean reversed bending endurance limit on smooth specimens, where no stress raisers exist, by 23 percent. It has little or no effect on the unidirectional endurance limit of "as-cast" rectangular flexures. This was expected because of the inherent stress raisers at the corners of the flexures. The HIP process appears to be ineffective on stress raisers but in combination with shot peening, the "as-cast" rectangular flexures show an endurance limit approximately equal to that for smooth round specimens when the difference in reversed bending and unidirectional bending is resolved with the aid of a Goodman diagram. Thus, the investment-cast 15-5 PH planet carrier for the 500-hp transmission will be subjected to the HIP process and shot peened for maximum strength and reliability. All these benefits can be had for 50 percent of the cost of the two-piece machined construction.

Investment-Cast Stainless-Steel Housings

The top case of the 500-hp transmission will be made of investment-cast 15-5 HP stainless steel. The non-load-carrying walls will be cast as thin as practical (about 0.254 cm (0.100 in.)) and then chemically milled to a thickness of 0.152 cm (0.060 in.). The top case will then be treated in the same manner as the investment-cast planet carrier: it will be subjected to the HIP process and shot peened. The result will be an advanced technology transmission housing with maximum corrosion resistance, excellent fatigue strength, no significant creeping, and good strength-to-weight ratio, all at a reasonable cost and a low risk in demonstrating each of the attributes.

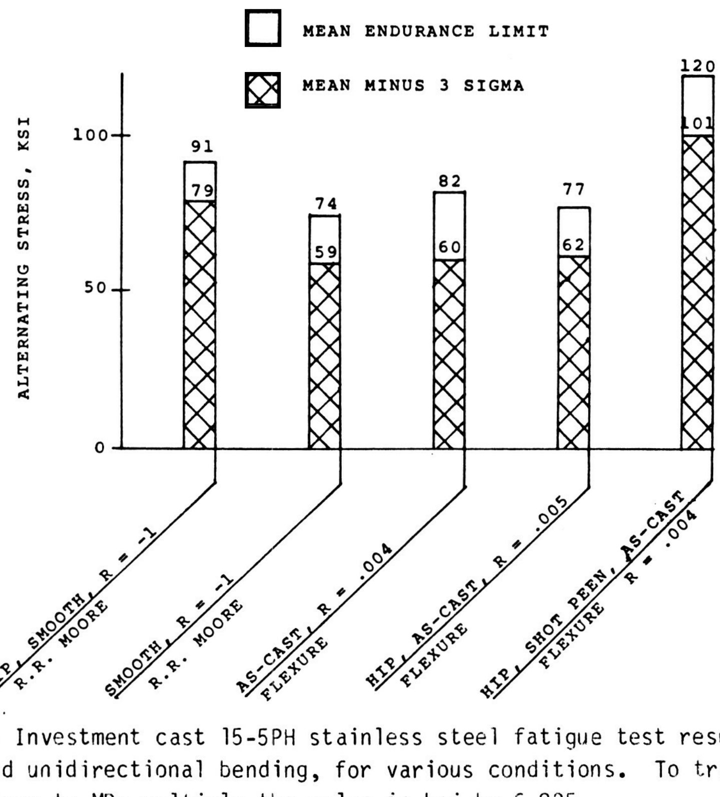

Figure 4. - Investment cast 15-5PH stainless steel fatigue test results, reversed and unidirectional bending, for various conditions. To transform these values to MPa multiply the value in ksi by 6.895.

Most contemporary transmission housings are made of either magnesium or aluminum alloy castings. Both of these materials creep rather rapidly at operating temperatures. This is especially detrimental to spiral bevel gear housings where the gears must be located accurately relative to each other to maintain proper tooth contact patterns. Both of these materials also have low fatigue strengths, and magnesium alloy castings are very susceptible to corrosion.

Other types of housings that some helicopter manufacturers are proposing and developing for the 1980's are the metal matrix composite, the graphite filament polyimide resin system, and the stainless-steel weldment. The composite and resin housings are costly and appear to be many years away from a usable product that can be as highly reliable as existing cast housings. The welded stainless-steel housing appears to be a viable concept except for the many welds required to connect the stiffening ribs and cylinders to each other and to the thin sheet metal shell. The approach being taken by BHT for advanced technology transmission housings, the investment-cast stainless-steel housing, selectively chemically milled for weight reduction, hot isostatic pressed, and shot peened, should be equivalent to or exceed all the capabilities of the welded stainless-steel housing and be more favorable in cost but void of welds. Recent developments in the ability to investment cast large housings combined with chemical milling permits this direction to be taken in providing advanced technology transmission housings for the 1980's. Heretofore, nothing has been able to out-perform magnesium castings as a transmission housing in the area of weight savings. But for a certain size of housing, Fitzgerald (ref. 7) has reported as much as a 23 percent weight saving in going from a cast magnesium housing to the stainless-steel weldment. Obviously, the size of the weight saving will depend on the size and complexity of the housings; however, the fact that the potential is there adds impetus to the other benefits predicted for a stainless-steel housing, especially one that is investment cast, hot isostatic pressed, chemically milled, and shot peened, instead of welded.

Miscellaneous Improvements

Three other distinct improvements were made in upgrading the 317-hp transmission to the 500-hp transmission: the straddle-mounted spiral bevel gear, the positive flow-through oiling to the sun gear spline, and the cantilever-mounted ring gear. These items/concepts are shown in figure 1. They represent improvements dictated by field experience and logic. The wear of the sun gear spline and the ring gear spline has been shown to be sporadic and not conducive to long transmission operating lives. The overhung spiral bevel gear performed satisfactorily for the 317-hp transmission; however, the additional rigidity inherent in the straddle-mounted design will no doubt be required to maintain proper tooth contact patterns for the 500-hp transmission during normal operation and especially during overtorque testing.

The straddle-mounted bevel gear concept is already in use on the 206L–1 435-hp transmission. At first glance this 435-hp rating is only 13 percent less than 500 hp; however, because the difference in reduction ratio in the spiral bevel gearset between the 317-hp transmission and the 206L–1 transmission, the 500-hp rating represents a 32 percent increase in torque on the gear member of the set. The penalty for straddle mounting adds to the increased height and weight of the transmission.

The sun gear spline is a working spline in that it must accommodate the off-axis operation of the bevel gear shaft and the planetary. In a perfect nonloaded position, the bevel gear shaft and the sun gear have a common axis of rotation, but, due to tolerances and deflections, a condition of misalinement exists that causes the spline connection to work. In the 317-hp transmission, both the external spline on the sun gear and the mating internal spline in the bevel gear shaft, are of core hardness, noncrowned (straight), heavily shot peened, and oil-mist lubricated. The wear problem exists primarily because the wear particles are not washed out of the contact area.

Based on experience on the XV–15 tilt rotor aircraft, the splined connection on the 500-hp transmission will have the sun gear spline crown hobbed, nitrided, and the white layer blasted off with garnet, and the bevel gear shaft internal spline will be of core hardness and noncrowned. The oil for the positive oiling feature is being supplied from an orifice in the investment-cast planet carrier as shown in figure 1. Note that there is a dam at both ends of the splined area of such height that the working sufaces of the spline teeth are always submerged in oil. In addition to this, since the spline area is being constantly fed with oil, the oil flows through, flushing out any wear particles that have been generated. This prevents the accumulation of wear particles that tend to act as a lapping compound and cause excessive wear. Thus, it is anticipated that the crowned and hardened male spline working inside a core hardness straight female spline submerged in a bath of flow-through oil will prevent the spline from wearing.

The cantilever-mounted ring gear is used in the 500-hp transmission primarily to eliminate the spline wear that is inherent in the ring gear-to-case spline connection on the 317-hp transmission. The spline wear is not excessive for the life requirements of the 317-hp transmission; however, it would be for an advanced technology long-life transmission. The cantilevered mounted ring gear is also used in the 500-hp transmission for two other reasons: noise reduction and more uniform load distribution. The cantilevered mounting tends to isolate the ring gear from the housing and thus reduces any structure-borne noise emanating from the meshing gear teeth, and the cantilevered mounting provides a soft mounting to equalize load sharing among the four planets. The soft mounting minimizes the effect of the positional and runout inaccuracies associated with the location of the mast, mast spline, planet carrier spline, planet pinions, ring gear teeth, and ring gear mounting flange.

Size and Weight Comparison

The size and weight of the 500-hp transmission has been alluded to in nondiscrete terms throughout this paper. This section will address the subject in detail by comparing the weights and heights of the 500- and 317-hp transmissions. Figure 1 shows that the 500-hp transmission is 30.50 cm (12.00 in.) high. This is 3.81 cm (1.50 in.) higher than the 317-hp transmission. The increase in height is primarily due to the addition of the bevel gear shaft support quill to straddle mount the bevel gear, the oil transfer mechanism for transferring oil from the top case to the rotating planet carrier, and the

TABLE 10. WEIGHT COMPARISON OF 317-HP AND 500-HP TRANSMISSIONS, KG (LB)

	317-HP Trans.	500-HP Trans.
Gearshaft	2.6 (5.7)	3.4 (7.6)
Gearshaft support	0 (0)	1.3 (2.9)
Planet carrier	4.6 (10.2)	6.0 (13.2)
Ring gear	2.3 (5.0)	3.5 (7.7)
Mast ball brg.	.9 (1.9)	1.3 (2.8)
Top Case	3.8 (8.4)	4.6 (10.2)
Remaining parts less mast	39.5 (87.2)	39.5 (87.2)
TOTAL	53.7 (118.4)	59.6 (131.6)
Weight increase	-	5.9 (13.2)
Percent weight increase	-	11%
Weight includes lubricating oil		
Lb/hp	.37	.26

mast tandem ball bearing. The 317-hp transmission does not have the support quill and the oil transfer mechanism, and it has a single, instead of a tandem, ball bearing on the mast (not shown in fig. 2).

The diameter of the 500-hp transmission is the same as that for the 317-hp transmission. Thus, the 3.81-cm (1.50-in.) growth in height is the only external size change for the 58-percent increase in power capability. Table 10 shows the weights of the two transmissions. The weight of the 500-hp transmission is only 11 percent greater than that of the 317-hp transmission; 11 percent greater for a 58-percent increase in power capability. In pounds per horsepower this represents a 29-percent reduction. The weights for the 317-hp transmission are actual; whereas, those for the 500-hp transmission are calculated. The only assumption made is that the investment-cast stainless-steel top case can be designed to weigh no more than a comparable magnesium top case. In the section "Investment Cast Stainless-Steel Housings," reference was made to a 23-percent weight saving in changing from a cast magnesium housing to a stainless-steel weldment. This was accomplished on a 34-kg (75-lb) magnesium housing. The 500-hp transmission top case is much smaller (4.6 kg (10.2 lb), as originally designed in magnesium), which makes it more difficult to exact a weight saving; however, it is anticipated that by careful design, the top case can be made of investment-cast stainless steel for the same weight required for the cast magnesium top case.

Thus, this size and weight comparison shows just how close the objective came to being met, the objective being that by using existing advanced technology components, the 317-hp transmission could be made into a 500-hp transmission with no change in size or weight. Except for the 3.81-cm (1.50-in.) increase in height and the 5.9-kg (13.2-lb) increase in weight, the objective was met.

References

1. Coleman, Wells: A New Perspective on the Strength of Bevel Gear Teeth. AGMA 229.13, Oct. 1969.
2. Battles, R. A.: Development of Helicopter Transmission Components for the 1980's. American Helicopter Society HPS-10, Nov. 1979.
3. Cutler, R. A.; et al.: Elevated Temperature Fracture Toughness and Fatigue Testing of Steels for Geothermal Applications. TR80-87 under DOE Contract No. EY-76-C-02-0016, Terratek, University Research Park, 420 Wakara Way, Salt Lake City, Utah, Nov. 1980.
4. Bamberger, E. N.; Zaretsky, E. V.; and Signer, H.: Endurance and Failure Characteristic of Main Shaft Jet Engine Bearing at 3 Million DN. ASME 76-Lubs-16, May, 1976.
5. Bamberger, E. N.; et al.: Life Adjustment Factors for Ball and Roller Bearings. ASME Engineering Design Guide, Sept., 1971.
6. Johnson, L. G.: The Statistical Treatment of Fatigue Experiments. Elsevier Publishing Company, New York, New York, 1964.
7. Fitzgerald, P. C.; and Gardner, G. F.: Advanced Transmission Component Development and Test Program. American Helicopter Society HPS-8, Nov. 1979.