San Diego State University

ME - 314 Group Design Project

Conceptual Accessory Gearbox Design

Group 7
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Table of Contents:

List of Figures:

List of Tables:

Abstract

This project entails the conceptual design of an accessory drive gearbox for a commercial aircraft, intended to be mounted below the jet engine compressor case. The gearbox is designed to drive components including the engine fuel pump, oil pump, hydraulic pump, and electrical generator. Power is transmitted to the gearbox via an accessory drive shaft connected to the engine compressor spool through a compressor power transfer shaft, adhering to the packaging and weight specifications.

The gear train components were sized and analyzed according to the American Gear Manufacturers Association (AGMA) standards. Specific design parameters include the use of either spur or helical gears with a 20° pressure angle, adherence to a No 10 quality standard, with a reliability of 98%. The system's operational requirements dictate that it must withstand moderate shock loading, maintain uniform power delivery, and operate below a temperature of 250℉.

The gearbox supports various component torques and drive speeds with specific tolerances. For instance, the oil pump operates at 8500 rev/min with a maximum torque of 259.5 in-lbf counterclockwise, while the electrical generator operates at 6795 rev/min delivering a torque of 185.5 in-lbf clockwise, suitable for its operational needs. The fuel pump is driven at a rate of 13950 rev/min having a maximum torque of 135.5 in-lbf clockwise. The hydraulic pump requires a maximum torque of 270.1 in-lbf counterclockwise, while being driven at a speed of 3500 rev/min. With the overall weight of the gearbox and components constrained to 260 lbf, material selection, sizing, and component layout were optimized for weight reduction and functionality.

The gear train and housing design were validated against specified design factors, meeting the minimum design factor of 1.15 for the shafts and 1.3 for gear bending stresses. Engineering software tools such as SolidWorks were used to develop schematic representations. Analytical calculations were performed using Excel spreadsheets. The design process makes sure to meet all specified requirements while providing a reliable, economic, and efficient mechanical solution for aircraft accessory drives.

Design Requirements

The design project for the accessory gearbox has requirements for each system component. The accessory drive shaft must have a minimum design factor of 1.15. The gears need to maintain a minimum design factor of 1.30 for bending and 1.15 for contact. The shafts for the four accessory drive components (oil pump, fuel pump, hydraulic pump, and electrical generator) each require a minimum design factor of 1.10. With the gearbox housing and components having a structural weight of 260 lbf.

During the accessory gearbox design, the following assumptions were made: a pressure angle of 20° for spur or helical teeth, uncrowned teeth, a No. 10 quality standard for all gears, a reliability of 98%, precision enclosed units, moderate shock loading, uniform power distribution, operating temperatures below 250°F, and a service life requirement equivalent to 10^8 cycles for all gear train components. All gearing should be adjusted during the assembly of the accessory gearbox.

The compressor transfer shaft operates at a speed of 14,000 rpm counterclockwise with a torque of 450.18 ± 0.5 in·lbf, driven by an engine compressor. Meanwhile, the accessory drive shaft provides a torque of 643.1 ± 0.5 in·lbf at a speed of 9,800 rpm counterclockwise. The bevel gears on the transfer and accessory drive shafts have 28 teeth with a pitch diameter of 3.5 inches and 40 teeth with a pitch diameter of 5.0 inches.

Calculations were performed in Excel, MATLAB, and manually for verification. Detailed calculations are presented in the Appendix. Figure 1 provides a schematic of the gearbox installation from four different perspectives.

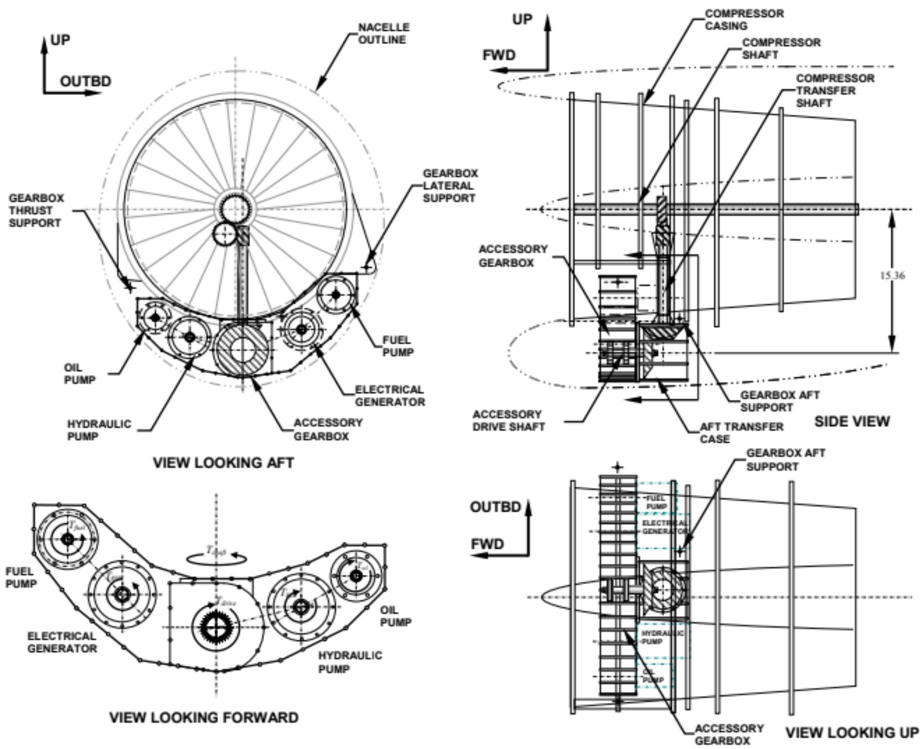


Figure 1: Gearbox Installation

Several constraints had to be addressed in the gear train design. A significant challenge was ensuring that the shafts and gears fit and operate within the gearbox housing. Furthermore, another challenge assigned to the engineering team was to not exceed the weight limit of the gearbox case and its components. This factor is crucial to the success of the gear train design and helped narrow down the material choices for the gears and the shafts. Additionally, the gears and shafts had to be designed around the component shafts due to the predetermined placement of the oil pump, fuel pump, electrical generator, and hydraulic pump, as illustrated in Figure 1. For gears that mesh together, their diametral pitches must be equal, or else some gear interference and even catastrophic failure might occur. With an operating temperature below 250°F, the materials for the gears and the shafts were picked meticulously. Furthermore, the minimum design factors required for the shafts and gears restrict the allowable diameter sizes, adding another hurdle for the engineering team to jump through to complete the gear train design.

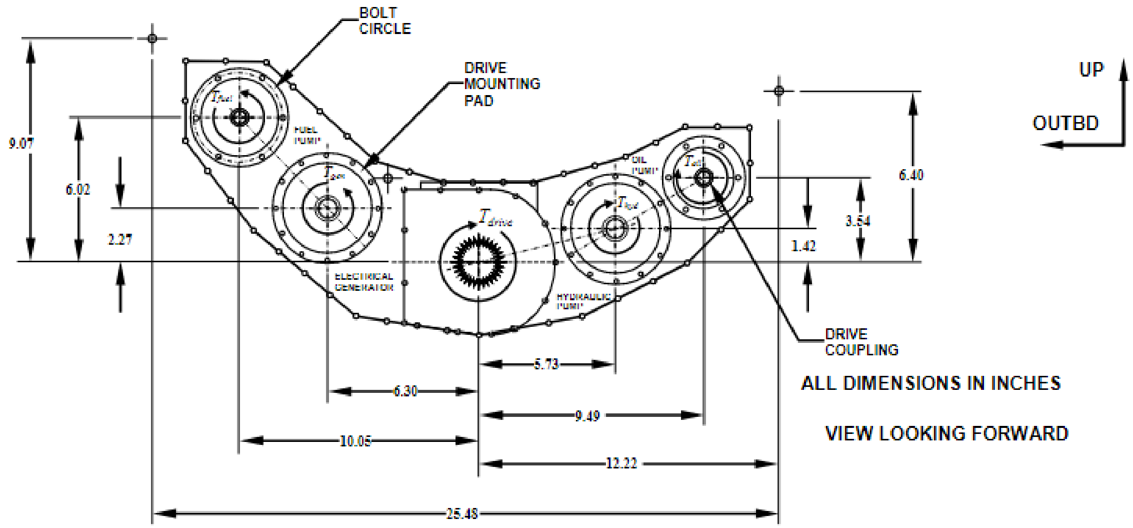


Figure 2: Accessory Component Locations

Table 1: Component Specifications							
Component	**Weight**	**Drive Speed (± 10)**	**Mating Drive**	**Max Torque**	**Aft CG**	**Bolt Pattern**	**Bolt Circle**
Oil Pump	27.00 lbf	8500 rev/min	8 - Spine	259.5 in·lbf ccw	3.78 in	6 × 10 - 24 - UNC - 3A	2.97 - in
Fuel Pump	28.00 lbf	13950 rev/min	8 - Spine	135.5 in·lbf cw	2.48 in	6 × 10 - 24 - UNC - 3A	3.68 - in
Electrical Generator	45.00 lbf	6795 rev/min	12 - Spine	185.5 in·lbf cw	4.13 in	12 × 8 - 32 - UNC - 3A	4.25 - in
Hydraulic Pump	32.00 lbf	3500 rev/min	12 - Spine	270.1 in·lbf ccw	4.46 in	12 × 8 - 32 - UNC - 3A	4.25 - in

Gear Train Design and Layout

The gear train design includes 19 gears distributed across 11 shafts, of which 5 are primary: the accessory drive shaft, oil pump shaft, fuel pump shaft, hydraulic pump shaft, and electrical generator shaft. The accessory drive shaft needed a minimum design factor of 1.15, while the four other primary shafts each required a minimum factor of 1.1. After performing calculations based on the chosen dimensions and materials, these requirements were met and can be seen in Table Table 8 and Table 9.

When designing the gears, the following specifications were used: a 20° pressure angle for spur or helical teeth, uncrowned teeth, a No. 10 quality standard, and 98% reliability. The gears also met the required design factors of 1.3 for bending and 1.15 for pitting (contact). The selected gear material was annealed 31CrMoV9 at 400°F and the chosen shaft material was AISI 4140 Q&T 400°F.

To maximize space within the gearbox housing, some shafts used compound gears to connect components. This design choice also improved the design factor by efficiently transferring torque from pinion to gear. Gears were secured on shafts using retaining rings, keyways, grooves, and bearing shoulders.

Designing the gear train for the accessory gearbox required careful consideration of multiple parameters. One of the primary challenges was achieving the required drive speed for each accessory component while ensuring that all parts fit within the gearbox housing and that gear contacts were optimal.

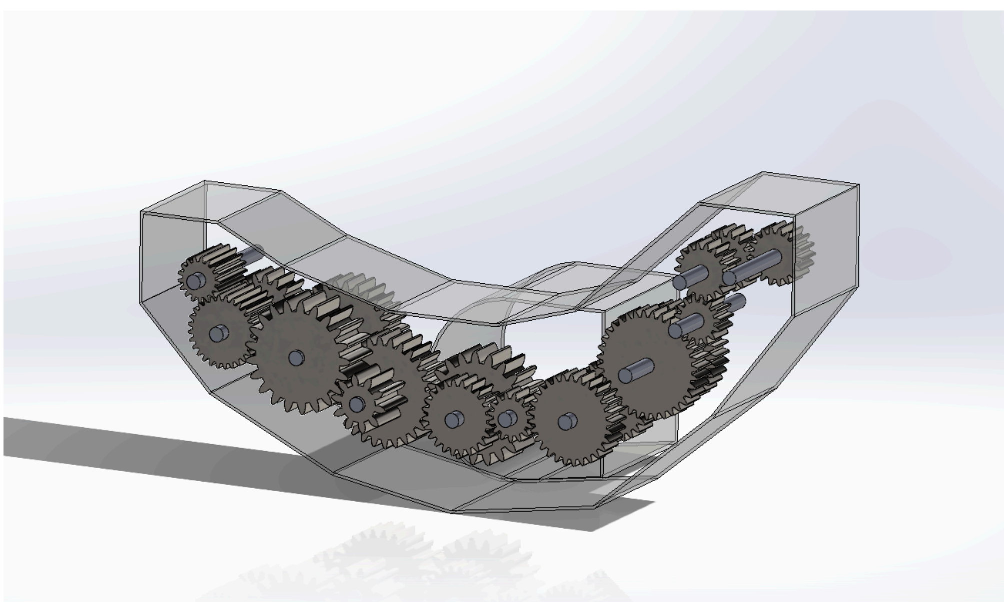


Figure 3: Gear Train Solidworks Layout

Gear Design and Selection

The completed gear train comprises 19 individual gears mounted on 11 separate shafts. All 19 utilized are spur gears with uncrowned teeth and a pressure angle of 20° pressure angle. The chosen gear material is 31CrMoV9 at 400°F, offering an ultimate strength of 236 kpsi, and a yield strength of 212 kpsi. This material was selected for its high strength, and excellent wear resistance. An example of one gear is shown in Figure 4, and the geometry of each gear can be found in Table 2.

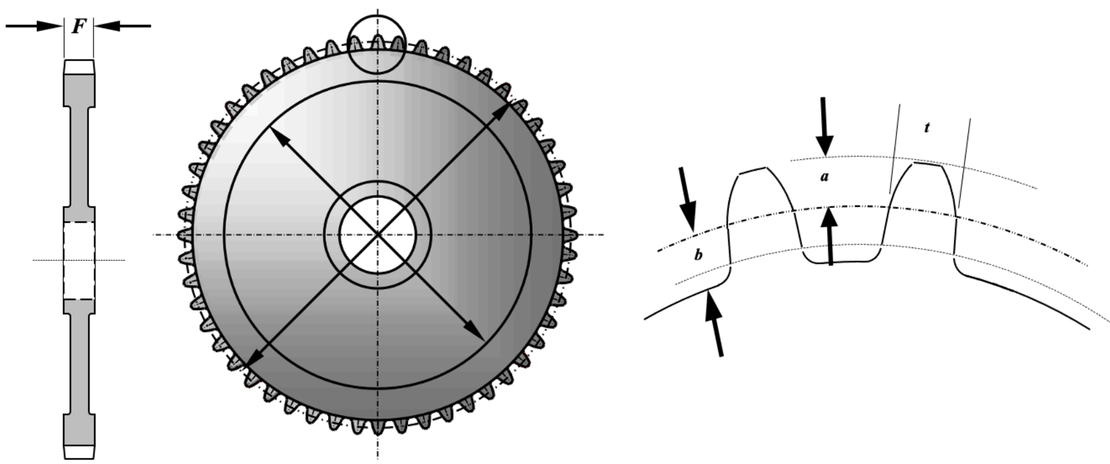


Figure 4: Gear Geometry

The table below contains key dimensions for all the gears used in the gear train system, encompassing values for the number of teeth, diametral pitch, addendum, dedendum, clearance, tooth thickness, whole depth, and face width. The diametral pitch is consistent for gears that mesh together, along with their respective addendum and dedendum values, which are calculated based on the gear's pitch diameter. Clearance is determined by subtracting the dedendum of the driver gear from the pinion gear's addendum. The depth, column 9, is the sum of the addendum and dedendum of the gear’s tooth. Face width was selected to ensure the gears fit within the gearbox housing while meeting safety factors of 1.3 for bending and 1.15 for contact. More details about these safety factors can be found in the gear analysis section.

Table 2: Gear Geometry

Gear	N	P_d	d (in)	a (in)	b (in)	c (in)	t (in)	h (in)	F (in)
2	24	10	2.4	0.127	0.0158	0.000093	0.157	0.235	1
3	15	10	1.5	0.127	0.0158	0.000093	0.157	0.235	1
4	29	10	2.9	0.127	0.0158	0.000093	0.157	0.235	1
5	26	10	2.6	0.127	0.0158	0.000093	0.157	0.235	1

Table 2: Gear Geometry									
Gear	N	P_d	d **(in)**	a **(in)**	b **(in)**	c **(in)**	t **(in)**	h **(in)**	F **(in)**
6	31	10	3.1	0.127	0.0158	0.000093	0.157	0.235	1
7	35	10	3.5	0.127	0.0158	0.000093	0.157	0.235	1
8	15	10	1.5	0.127	0.0158	0.000093	0.157	0.235	1
9	18	10	1.8	0.127	0.0158	0.000093	0.157	0.235	1
10	19	10	1.9	0.127	0.0158	0.000093	0.157	0.235	1
11	18	10	1.8	0.127	0.0158	0.000093	0.157	0.235	1
12	21	7	3.0	0.127	0.0158	0.000093	0.224	0.336	1.428
13	28	7	4.0	0.127	0.0158	0.000093	0.224	0.336	1.428
14	10	6	1.667	0.127	0.0158	0.000093	0.262	0.392	1.667
15	21	6	3.5	0.127	0.0158	0.000093	0.262	0.392	1.667
16	28	8	3.5	0.127	0.0158	0.000093	0.196	0.294	1.25
17	19	8	2.375	0.127	0.0158	0.000093	0.196	0.294	1.25
18	28	12	2.333	0.127	0.0158	0.000093	0.131	0.196	2
19	17	12	1.42	0.127	0.0158	0.000093	0.131	0.196	2

Gear Analysis - Gear Loading

Each of the 19 gears in the assembly will be forcefully loaded in bending and torsion from the accessory bevel gear. The reactionary forces along the pressure line between the teeth of the pinions and the teeth of the meshing gears was used to calculate the total loading, W. Only tangential and radial components of force are considered due to the utilization of spur gears which have negligible axial forces and the use of bearing shoulders that prevent the gears from axial shift. All gears have a pressure angle of 20°, operate under 250℉, and are required to have a service life of 10^8 cycles. The loading of each gear is summarized in Table 3 below.

Table 3: Gear Loading										
Gear	***N***	P_d	***d* (in)**	***N* (rpm)**	***V* (ft/min)**	m_G	K_B	W^t **(lbf)**	W^r **(lbf)**	***W* (lbf)**
2	24	10	2.400	9800.000	6157.522	2.083	1.00	584.649	212.795	622.170
3	15	10	1.500	15680.000	6157.522	1.600	1.00	584.649	212.795	622.170
4	29	10	2.900	8110.345	6157.522	1.115	1.00	584.649	212.795	622.170
5	26	10	2.600	8110.345	5520.537	1.115	1.00	652.108	237.348	693.959
6	31	10	3.100	6802.225	5520.537	0.839	1.00	652.108	237.348	693.959
7	35	10	3.500	6802.225	6232.864	0.839	1.00	346.304	126.044	368.529
8	15	10	1.500	15871.858	6232.864	2.333	1.00	346.304	126.044	368.529
9	18	10	1.800	13226.548	6232.864	0.833	1.00	346.304	126.044	368.529
10	19	10	1.900	13226.548	6579.134	0.833	1.00	328.078	119.410	349.133
11	18	10	1.800	13946.528	6572.146	1.056	1.00	328.078	119.537	349.504
12	21	7	3.000	9800.000	7696.902	2.083	1.00	467.719	170.236	497.736
13	28	7	4.000	7350.000	7696.902	0.750	1.00	467.719	170.236	497.736
14	10	6	1.667	7350.000	3207.043	0.750	1.00	1122.526	408.566	1194.567
15	21	6	3.500	3500.000	3207.043	0.476	1.00	1122.526	408.566	1194.567
16	28	8	3.500	3500.000	3207.043	0.476	1.00	1122.526	408.566	1194.567
17	19	8	2.375	5157.895	3207.043	1.474	1.00	1122.526	408.566	1194.567
18	28	12	2.333	5157.895	3150.779	1.474	1.00	1142.571	415.862	1215.899
19	17	12	1.420	8495.356	3158.192	1.643	1.00	1139.889	414.886	1213.045

Gear Analysis - Bending Stresses

The bending stress analysis was conducted according to the American Gear Manufacturers Association (AGMA) standards. The design requirements specify that all gears must meet a minimum safety factor of 1.300 for bending and endure a service life of 10^8 cycles. The loading is classified as moderate shock, with uniform power distribution. The bending stress on the gears resulted from interactions with neighboring gears. The chosen material, annealed 31CrMoV9 at 400°F, was selected for all gears due to its favorable mechanical properties and its ability to meet the 98% reliability standard. Material properties for 31CrMoV9 400°F were used to calculate the bending stress for each of the gears. The maximum bending stress encountered in the gear train is found at the oil pump gear, with a bending stress of 45,973.58 psi. To save cost on the production of this gearbox, the grade of a gear was dependent on how high the calculated factor of safety was in respect to the required factor of safety. For example, if the gear in question receives a lot of torque and stress, grade 2 was utilized to increase the factor of safety and improve on the gear box design. Some gears function both as the pinion and the gear, so the smallest values between them were used in the tables.

Table 4: Gear Bending Stress Analysis Results

Gear	***N***	***W^t* (lbf)**	**n_d**	**K_0**	**K_V**	***Y***	**K_S**	**K_m**	**K_B**	***J***	***σ* (psi)**
2	24	584.649	1.847190061	1.25	1.2998	0.337	1.023623753	1.160147733	1	0.32	35,252.14
3	15	584.649	2.401426826	1.25	1.2998	0.29	1.019519183	1.160147733	1	0.25	44,941.80
4	29	584.649	2.664963133	1.25	1.2999	0.356	1.025126695	1.1416656	1	0.35	31,766.08
5	26	652.108	2.402975361	1.25	1.2866	0.394	1.089948472	1.034320994	1	0.39	25,224.35
6	31	652.108	2.424032572	1.25	1.2866	0.346	1.059439723	1.034320994	1	0.375	25,499.03
7	35	346.304	3.386287469	1.25	1.3014	0.37425	1.026498535	1.1600026	1	0.3675	18,253.17
8	15	346.304	2.274451406	1.25	1.3014	0.29	1.019519183	1.1600026	1	0.25	26,649.72
9	18	346.304	2.732210683	1.25	1.3014	0.309	1.021251353	1.1600026	1	0.295	22,622.88
10	19	328.078	3.026178345	1.25	1.3083	0.314	1.021689955	1.144560337	1	0.3075	20,425.26
11	18	328.078	2.896438917	1.25	1.3083	0.309	1.021251353	1.144560337	1	0.3	20,926.90
12	21	467.719	4.778307115	1.25	1.3289	0.328	1.305769	1.15288316	1	0.33	17,373.53

Table 4: Gear Bending Stress Analysis Results											
Gear	***N***	**W^t (lbf)**	**n_d**	**K_0**	**K_V**	**Y**	**K_S**	**K_m**	**K_B**	**J**	**σ (psi)**
13	28	467.719	4.929509811	1.25	1.3289	0.353	1.366098	1.15288316	1	0.358	16,754.61
14	10	1122.526	2.370889231	1.25	1.2274	0.213	1.067943387	1.126387869	1	0.213	35,014.73
15	21	1122.526	3.590533032	1.25	1.2274	0.328	1.080348073	1.126387869	1	0.32	23,577.40
16	28	1122.526	2.084789514	1.25	1.2257	0.353	1.049659885	1.151348487	1	0.345	40,606.22
17	19	1122.526	1.87249652	1.25	1.2257	0.314	1.046377747	1.151348487	1	0.315	44,334.42
18	28	1142.571	2.114793154	1.25	1.2180	0.353	1.053290427	1.241595177	1	0.35	38,907.80
19	17	1139.889	1.805733573	1.25	1.2180	0.303	1.048995811	1.241595177	1	0.295	45,973.58

Table 5: Gear Bending Strength							
Gear	***N***	***d* (in)**	**Y_N**	**K_T**	**R**	**K_R**	**σ_{ALL} (psi)**
2	24	2.40	0.9960666236	1	0.98	0.9549225461	81158.6
3	15	1.50	0.9767774606	1	0.98	0.9549225461	81158.6
4	29	2.90	0.9960666236	1	0.98	0.9549225461	81158.6
5	26	2.60	0.9767774606	1	0.98	0.9549225461	59257.3
6	31	3.10	0.9960666236	1	0.98	0.9549225461	59257.3
7	35	3.50	0.9960666236	1	0.98	0.9549225461	59257.3
8	15	1.50	0.9767774606	1	0.98	0.9549225461	59257.3
9	18	1.80	0.9960666236	1	0.98	0.9549225461	59257.3
10	19	1.90	0.9960666236	1	0.98	0.9549225461	59257.3

Table 5: Gear Bending Strength							
Gear	N	d **(in)**	Y_N	K_T	R	K_R	σ_{ALL} **(psi)**
11	18	1.80	0.9767774606	1	0.98	0.9549225461	59257.3
12	21	3.00	0.9767774606	1	0.98	0.9549225461	81158.6
13	28	4.00	0.971788421	1	0.98	0.9549225461	81158.6
14	10	1.67	0.9767774606	1	0.98	0.9549225461	81158.6
15	21	3.50	0.9960666236	1	0.98	0.9549225461	81158.6
16	28	3.50	0.9960666236	1	0.98	0.9549225461	81158.6
17	19	2.38	0.9767774606	1	0.98	0.9549225461	81158.6
18	28	2.33	0.9681400969	1	0.98	0.9549225461	81158.6
19	17	1.42	0.9767774606	1	0.98	0.9549225461	81158.6

Gear Analysis - Contact Stresses

AGMA standards were used to perform all contact stress analyses to compute the pitting resistance of each gear. The contact stresses endured by each gear considers the elastic coefficient, tangential transmitted load, overload factor, dynamic factor, size factor, load-distribution factor, surface condition factor, pitch diameter of the gear, face width of the gear, and geometry factor for pitting resistance. The results of the analysis of the gear contact stresses are summarized in Table 6. The completion of the analysis proves that the minimum design factor of 1.15 is met with the selected material, geometry, and loading.

Table 6: Gear Pitting (Contact) Stress Analysis Results											
Gear	N	W^t **(lbf)**	n_d	K_0	K_V	Z_N	K_S	K_m	C_f	I	σ **(psi)**
2	24	584.649	1.334429516	1.25	1.2998	0.9727074142	1.023623753	1.160147733	1	0.114627939	186,292.54
3	15	584.649	1.303750018	1.25	1.2998	0.948436889	1.019519183	1.160147733	1	0.1146327939	185,918.67

Table 6: Gear Pitting (Contact) Stress Analysis Results

Gear	***N***	W^t **(lbf)**	n_d	K_0	K_V	Z_N	K_S	K_m	C_f	I	σ **(psi)**
4	29	584.649	3.335973555	1.25	1.2999	0.9727074142	1.025126695	1.1416656	1	0.105913245	74,519.26
5	26	652.108	3.250735873	1.25	1.2866	0.948436889	1.089948472	1.034320994	1	0.1124830103	68,018.44
6	31	652.108	3.381585167	1.25	1.2866	0.9727074142	1.059439723	1.034320994	1	0.1124830103	67,059.73
7	35	346.304	4.037469703	1.25	1.3014	0.9727074142	1.026498535	1.1600026	1	0.1124878269	56,165.92
8	15	346.304	3.950180617	1.25	1.3014	0.948436889	1.019519183	1.1600026	1	0.1124878269	55,974.65
9	18	346.304	3.573159417	1.25	1.3014	0.9727074142	1.021251353	1.1600026	1	0.08765285587	63,464.34
10	19	328.078	3.574144684	1.25	1.3083	0.9727074142	1.021689955	1.144560337	1	0.08253693043	63,446.84
11	18	328.078	3.485712626	1.25	1.3083	0.948436889	1.021251353	1.144560337	1	0.08253693043	63,433.22
12	21	467.719	1.673838447	1.25	1.3289	0.948436889	1.305769	1.15288316	1	0.06887010104	144,811.74
13	28	467.719	1.625669274	1.25	1.3289	0.9421820942	1.366098	1.15288316	1	0.06887010104	148,119.26
14	10	1122.526	2.415797978	1.25	1.2274	0.948436889	1.067943387	1.126387869	1	0.1088584897	100,335.98
15	21	1122.526	2.46335	1.25	1.2274	0.97270	1.08034	1.12638	1	0.1088	100,917.03
16	28	1122.526	1.231595465	1.25	1.2257	0.9727074142	1.049659885	1.151348487	1	0.09574261688	201,847.34
17	19	1122.526	1.202747172	1.25	1.2257	0.948436889	1.046377747	1.151348487	1	0.09574261688	201,531.52
18	28	1142.571	1.301666036	1.25	1.2180	0.9727074142	1.053290427	1.241595177	1	0.1159064916	190,981.60
19	17	1139.889	1.271782883	1.25	1.2180	0.948436889	1.048995811	1.241595177	1	0.1159064916	190,591.86

Table 7: Gear Pitting Strength							
Gear	N	d (in)	Y_N	K_T	R	K_R	σ_{ALL} (psi)
2	24	2.40	0.9960666236	1	0.98	0.9549225461	244049
3	15	1.50	0.9767774606	1	0.98	0.9549225461	244049
4	29	2.90	0.9960666236	1	0.98	0.9549225461	244049
5	26	2.60	0.9767774606	1	0.98	0.9549225461	222622
6	31	3.10	0.9960666236	1	0.98	0.9549225461	222622
7	35	3.50	0.9960666236	1	0.98	0.9549225461	222622
8	15	1.50	0.9767774606	1	0.98	0.9549225461	222622
9	18	1.80	0.9960666236	1	0.98	0.9549225461	222622
10	19	1.90	0.9960666236	1	0.98	0.9549225461	222622
11	18	1.80	0.9767774606	1	0.98	0.9549225461	222622
12	21	3.00	0.9767774606	1	0.98	0.9549225461	244049
13	28	4.00	0.971788421	1	0.98	0.9549225461	244049
14	10	1.67	0.9767774606	1	0.98	0.9549225461	244049
15	21	3.50	0.9960666236	1	0.98	0.9549225461	244049
16	28	3.50	0.9960666236	1	0.98	0.9549225461	244049
17	19	2.38	0.9767774606	1	0.98	0.9549225461	244049
18	28	2.33	0.9681400969	1	0.98	0.9549225461	244049
19	17	1.42	0.9767774606	1	0.98	0.9549225461	244049

Shaft Analysis and Sizing

The gear train design incorporates 11 shafts, with five primary ones all made from 4140 Q&T 400°F. This material was chosen for its tensile strength of 257 kpsi and yield strength of 238 kpsi, meeting the required design factors. The quenching and tempering process improves strength, hardness, and fatigue resistance. The gearbox features a drive shaft powered by an engine compressor spool, alongside four accessory shafts. A total of 19 gears are mounted on these shafts, made of 4140 Q&T 400°F. Shaft lengths were determined based on their positions relative to the central drive shaft and gear locations.

The shaft design ensures proper alignment and gear positioning while preventing undue wear on the shaft and gear. Bearing shoulders are used to prevent axial gear movement on the shaft, positioned closest to the shaft's center, where the diameter is largest. The accessory drive, electrical generator, and hydraulic pump shafts have bearing shoulders since they contain compounded gears. These gears are separated by a step distance to avoid interference. The oil and fuel shafts contain only one gear each, so instead of bearing shoulders, they use keyways and retaining rings. The retaining rings are placed on both sides of the gears to secure them, while bearing shoulders and retaining rings work together on the other three shafts to firmly position the gears.

Shaft Loading - Accessory Drives

The shafts rotate and experience a continuous bending moment, which results in fully reversed bending stress. The accessory drive shaft undergoes both bending and torsion due to the input torque and a downward force, generating a bending moment on the shaft. The shaft loading analysis started by examining the forces on the spur gear, revealing the tangential and radial components of the gear's forces. These components were used to calculate the gear forces, which facilitated the shaft reaction analysis. By summing moments in the x and y directions, the reactions at each end of the shaft were determined. The bending moment for each gear on the five shafts was then calculated. Shear and bending moment diagrams were also created to identify the maximum bending moment. Tables 7 and 8 contain the calculated forces, reactions, and bending moments for each shaft and gear. However, after further analysis it was determined that the oil pump shaft underwent high stress and forces which led to failure under the current conditions. To resolve this issue, the oil pump shaft surface factor (k_a) was upgraded to a ground finish of 0.8361 to improve the endurance strength (S_e) to ultimately pass the factor of safety required. Apart from the oil shaft, all other shafts have a machined finish resulting in a surface factor (k_a) of 0.6205.

Table 8: Accessory Drive Shaft Loading											
Shaft	F_{1x} **(lbf)**	F_{1y} **(lbf)**	F_{2x} **(lbf)**	F_{2y} **(lbf)**	R_{Ax} **(lbf)**	R_{Ay} **(lbf)**	R_{Ox} **(lbf)**	R_{Oy} **(lbf)**	M_A **(in·lbf)**	M_B **(in·lbf)**	M_C **(in·lbf)**
Drive	-287.575	-607.493	185.650	-461.818	348.187	183.188	615.500	182.634	642.024	2849.314	3333.281

Table 9: Component Shaft Loading										
Shaft	F_{1x} **(lbf)**	F_{1y} **(lbf)**	F_{2x} **(lbf)**	F_{2y} **(lbf)**	R_{Ax} **(lbf)**	R_{Ay} **(lbf)**	R_{Ox} **(lbf)**	R_{Oy} **(lbf)**	M_A **(in·lbf)**	M_B**(in·lbf)**
Oil	823.882	890.334	N/A	N/A	-346.898	-374.878	-476.984	-515.457	1404.578	N/A
Fuel	183.031	-297.746	N/A	N/A	154.132	-250.733	28.90	-47.013	220.739	N/A
Hyd	230.799	-1172.059	213.592	1175.317	-98.137	-1226.363	-115.345	1121.013	1504.454	1384.068
Elect	-240.759	650.857	149.483	-336.851	176.567	-420.686	-213.676	567.021	605.946	4505.641

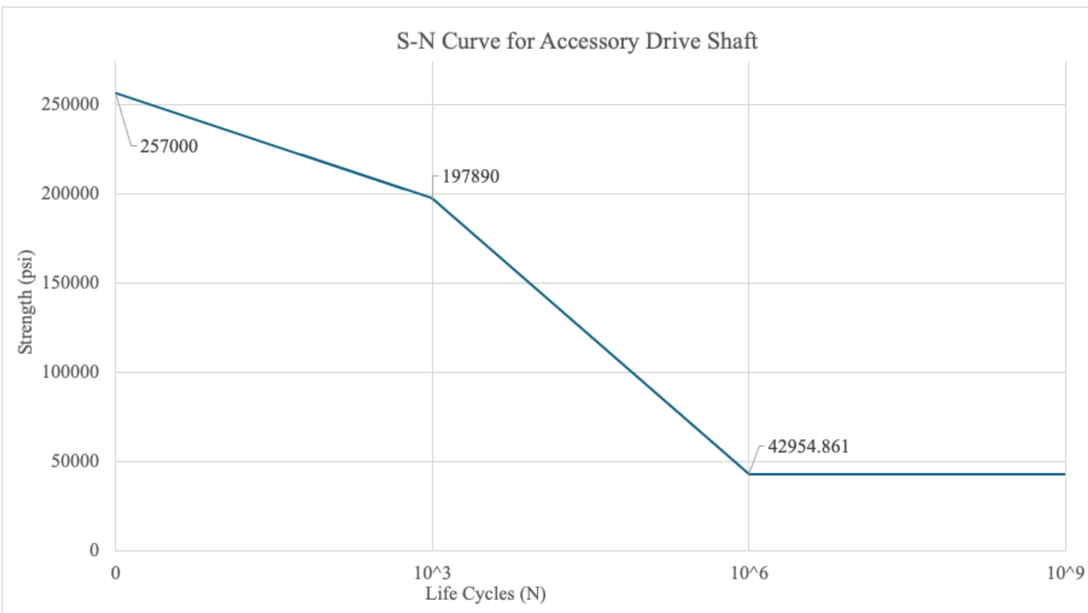


Figure 5. S-N Curve for Accessory Drive Shaft

Shaft Geometry

The gear train design is composed of 11 shafts, including the accessory drive shaft and four component shafts that link the oil pump, fuel pump, hydraulic pump, and electric generator to the gear train. Figure 6 illustrates the placement of the gears relative to the drive shaft, with the oil pump positioned on the left and the fuel pump on the right. The schematic guided the dimensioning of the shafts and the placement of each gear.

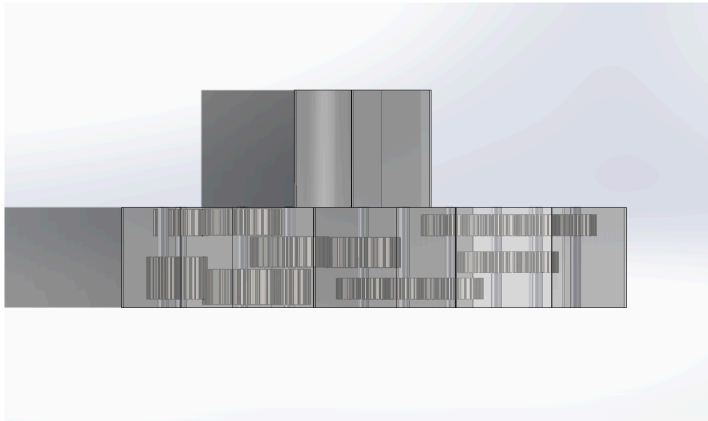


Figure 6: Finalized Gear Locations

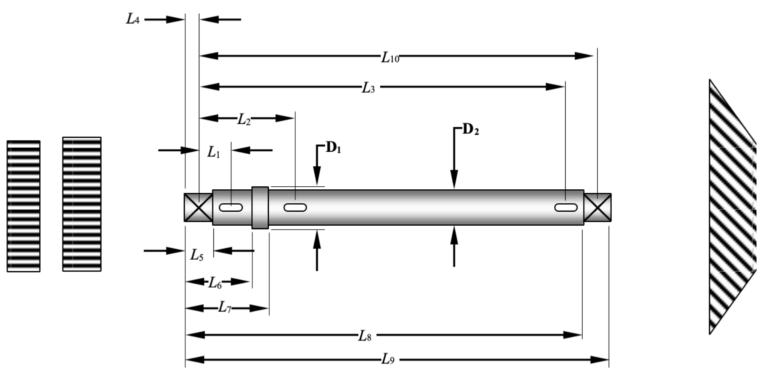


Figure 7: Accessory Drive Shaft Schematic

Figure 7 provides an illustration of the accessory drive shaft, with the sizing details summarized in Table 10. This table outlines the positions of the gears relative to the bevel gear, along with the shaft and bearing shoulder diameters. According to the design, the shaft should have its largest diameter at the center and smaller diameters toward the ends, with the bearing shoulder diameter exceeding that of the shaft itself. The DE-Goodman criterion was used to determine the minimum shaft diameters at the ends. The final diameters listed in Tables 10 and 11 were selected to ensure they meet the required safety factors of 1.10 and 1.15.

Table 10: Accessory Drive Shaft Sizing Results

Component	L_1 **(in)**	L_2 **(in)**	L_3 **(in)**	L_4 **(in)**	D_1 **(in)**	D_2 **(in)**
Drive Shaft	1.000	3.715	5.3125	10.290	2.000	2.500

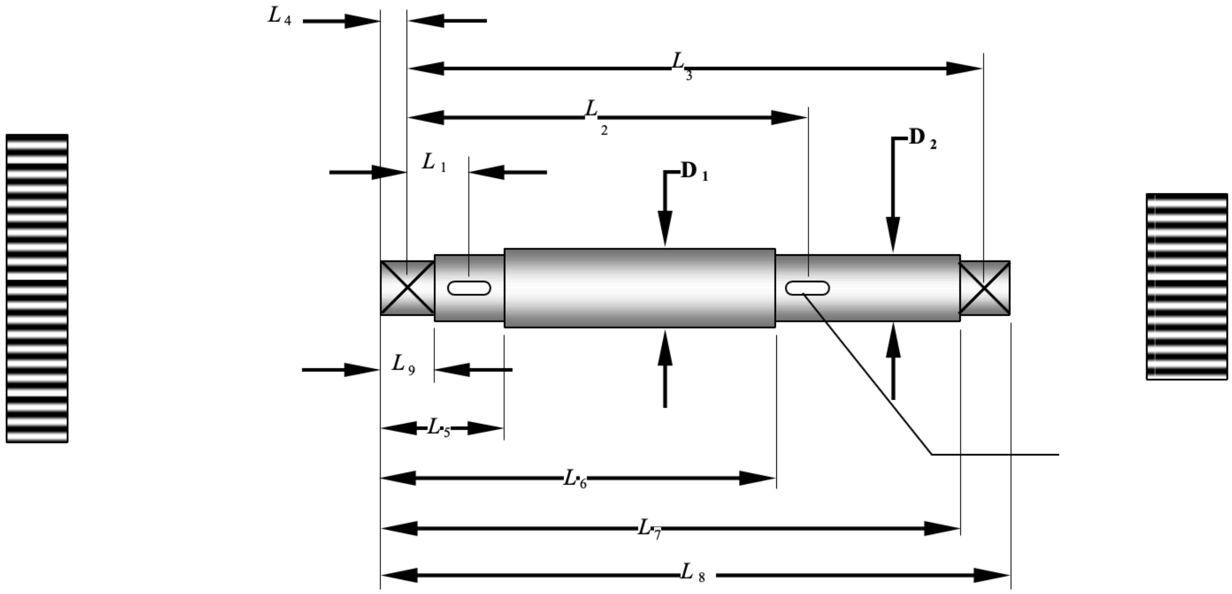


Figure 8: Component Shaft Schematic

Table 11 provides a summary of the geometry of the component drive shafts, with labels that can be referenced in Figure 8. These lengths were crucial for calculating shaft loading. The hydraulic pump and electrical generator shafts each have two gears mounted, whereas the oil pump and fuel pump shafts have only one gear each, requiring a single diameter across the entire shaft. Gear diameter was also considered to ensure proper mounting on each shaft.

Table 11: Component Shaft Sizing Results						
Component	**L_1 (in)**	**L_2 (in)**	**L_3 (in)**	**L_4 (in)**	**D_1 (in)**	**D_2 (in)**
Oil Pump	2.00	4.750	N/A	N/A	1.050	1.250
Fuel Pump	4.00	4.750	N/A	N/A	1.00	1.50
Hyd Pump	1.3350	3.6250	4.750	N/A	2.250	2.500
Elec Gen	1.00	4.00	4.750	N/A	2.000	2.500

Table 12 presents the design and fatigue safety factors. These results are derived from various parameters, including the method used to mount and secure the gears to the shafts. The oil and fuel pump shafts used bearing shoulders since only one gear was mounted on each. Instead of using, keyways and retaining ring grooves were employed to secure the gears to those two shafts. The drive shaft, hydraulic pump, and electric generator shafts used all three securing methods.

The DE-Gerber criterion was applied to calculate the fatigue safety factor, while von Mises stress was determined to assess yielding and determine the design safety factor. Each shaft design achieved acceptable safety factors, meeting the project's requirements of 1.10 and 1.15.

Table 12: Factor of Safety Results

Shaft	Drive Shaft	Oil Pump	Hydraulic Pump	Electrical Generator	Fuel Pump
Design FOS	12.804	4.684	32.547	13.183	19.310
Bearing Shoulder Fatigue FOS	7.275	3.882	4.160	3.769	14.686
Keyways Fatigue FOS	4.898	2.363	15.435	5.267	9.206
Grooves Fatigue FOS	2.153	1.177	7.045	2.265	4.213

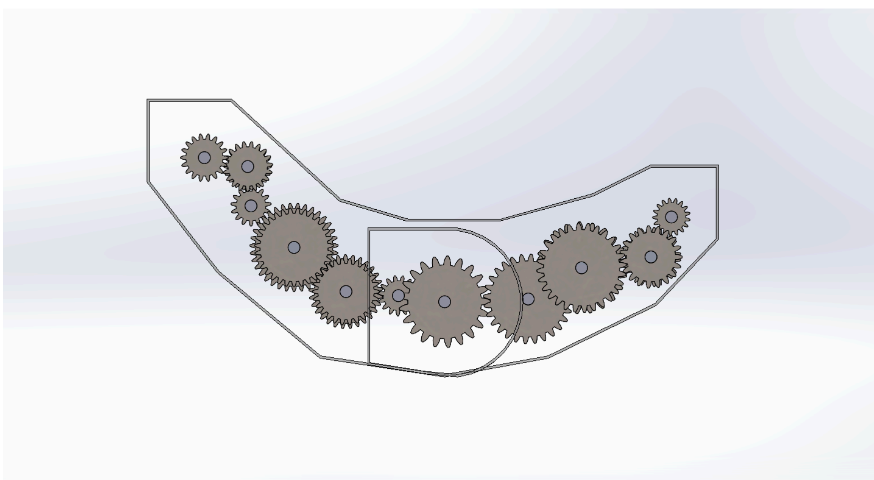


Figure 9: View Looking Forward of Gear Train Solidworks Model

Conclusion

The design of the accessory gearbox demands a range of engineering expertise to plan, size, and calculate the necessary gear train components. Utilizing these skills, the accessory gearbox was designed to meet the specified requirements and dimensions. The 11 shafts, made from AISI 4140 Q&T 400°F, feature varying diameters, ensuring the design factor meets the minimum requirements of 1.15 for the accessory drive shaft and 1.1 for the component shafts. The shafts surpass the design factor requirements significantly due to our consideration of groove fatigue during the design process. The 19 gears, arranged according to Figures 3, Figure 6 and Figure 9 are made from annealed 31CrMoV9 at 400°F with different pitch diameters and face widths to meet the design factors of 1.3 in bending and 1.15 in contact while fitting within the gearbox housing's limited space. Table 13 compares the final design results of the accessory gearbox with the project requirements.

Table 13: Final Results Comparison

Component	Numerical Values	Project Requirements
Accessory Drive Shaft - Yield Design Factor	12.804	1.150
Oil Pump - Yield Design Factor	4.684	1.100
Fuel Pump - Yield Design Factor	19.310	1.100
Hydraulic Pump - Yield Design Factor	32.547	1.100
Electric Generator - Yield Design Factor	13.183	1.100
Gear - Bending Design Factor	1.806 - 4.93	1.300
Gear - Contact Design Factor	1.203 - 4.037	1.150
Accessory Drive Shaft - Drive Speed	9800 rev/min	9800 rev/min
Oil Pump - Drive Speed	8495.356 rev/min	8500 ± 10 rev/min
Fuel Pump - Drive Speed	13946.528 rev/min	13950 ± 10 rev/min
Electric Generator - Drive Speed	6802.225 rev/min	6795 ± 10 rev/min
Hydraulic Pump - Drive Speed	3500.000 rev/min	3500 ± 10 rev/min

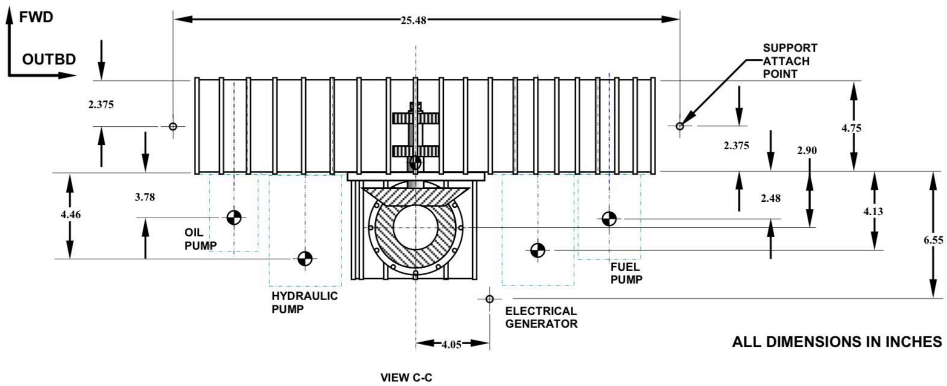
References

[1] Mechanical Engineering Design, ME314 lecture slides.

[2] Budynas, Richard G. & Nisbett, J. Keith. *Shigley's Mechanical Engineering Design, 9th edition.* McGraw Hill. Published 2011.

[3] Mechanical Engineering Design, ME314 class examples.

Appendix:



VIEW LOOKING UP

Figure 10: Bottom View of Gearbox Housing

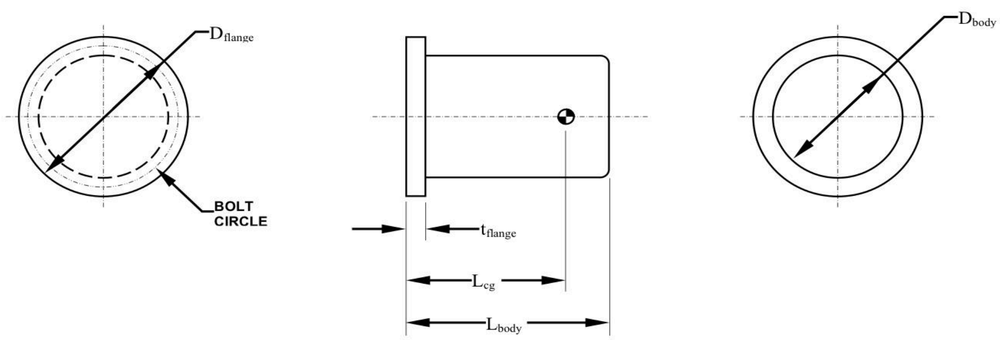


Figure 11: Details of Accessory Components

Table 14: Component Dimensions							
Component	**Bolt Circle**	**D_{flange} (in)**	**D_{body} (in)**	**L_{body} (in)**	**L_{eg} (in)**	**t_{flange} (in)**	**Case Material**
Oil Pump	2.97-in	3.47	2.55	4.075	3.78	0.115	Aluminum
Fuel Pump	3.68-in	4.11	3.26	4.575	2.48	0.115	Aluminum
Hydraulic Pump	4.25-in	4.60	3.75	5.855	4.46	0.125	Aluminum
Electric Generator	4.25-in	4.60	3.75	5.855	4.13	0.125	Aluminum

Table 14: Nomenclature

Symbol	Description
P_d	Diametral Pitch [Teeth/in]
N	Number of Teeth
d	Pitch Diameter
a	Addendum [radial distance between top land and pitch circle]
b	Dedendum [radial distance between bottom land and pitch circle]
c	Clearance [amount which dedendum of gear exceed addendum of mating gear]
h	Whole depth [sum of addendum and dedendum]
t	Thickness
L	Length
D	Diameter
F_w	Face Width
n	Revolutions
V	Pitch-line Velocity
m_g	Speed Ratio
K_B	Rib-thickness Factor
W_t	Tangential Force Component
W_r	Radial Force Component
W	Total Force
R	Reaction
M	Moment

Table 15: Nomenclature

Symbol	Description
K_0	OverLoad Factor
K_V	Dynamic Factor
K_s	Size Factor
K_m	Load Distribution Factor
K_T	Temperature Factor
K_R	Reliability Factor
Y	Lewis Form Factor
J	Bending Strength Geometry Factor
I	Surface Strength Geometry Factor
C_f	Surface Condition Factor
Y_N	Bending Stress Cycle Factor
Z_N	Contact Stress Cycle Factor

<u>Gear Train Component Analysis using AGMA Methods</u>:

The main goals in designing the gear train are to determine the correct dimensions, factoring in the gearbox size and specified parameters. The gears must meet a No. 10 quality standard using 21 Cr Mo V9 steel at 400℉ where gears 2 and 3 will be the focus of the calculations on the accessory drive shaft. As specified in the deliverables, the accessory drive shaft transmits a torque of 643.1 in·lbf at a rotational speed of 9800 rev/min in a counter-clockwise direction. This will be the basis of the design.

Table 16: Calculations and Specifications for Gear Design

Variable	Value
Applied Torque | $T_{applied}$	$643.1\ in.lbf$
Given Speed | V_G	$9800.000\ \frac{REV}{min}$
Given Drive Speed | V_{DS}	$6795.000\ \frac{REV}{min}$
Pressure Angle | θ	20.000°
Gear Train Value | e	0.6941
Calculated Drive Speed | V_{DSC}	$6802.224\ \frac{REV}{min}$
Diametral Pitch | P_D	$10.0\ \frac{teeth}{inch}$
Number of Teeth | N_p	$N2 = 24 \mid N3 = 15$
Pitch Diameter | D_p	$D_{p2} = 2.4\ in \mid D_{p3} = 1.5\ in$
Addendum | A_D	$0.1\ in$
Face width | F_p	$1.0\ in$
Dedendum | D	$0.135\ in$
Tooth Thickness | t	$0.157\ in$

Calculations:

The following displays calculations for the number of teeth, pitch diameters, addendum, dedendum, tooth thickness, face width, and the speed of gears. In turn, the bending and contact stresses are found in order to determine the safety factors.

Gear Train Analysis of Components [AGMA]

Practical dimensions while considering sizing of the gearbox and provided parameters.

- No. 10 quality Standard
- Material Selection: 31 Cr Mo V9 annealed Steel @ 400 °F
- Calculations follow the gear N_2 and N_3 of accessory drive shaft.
- It is given that the accessory drive shaft delivers a torque of 643.1 ± 0.5 in. lbf at a speed of 9800 rev/min (ccw)

Design of Gear Train

- Objectives:
  - number of teeth
  - pitch Diameters
  - Addendum
  - Dedendum
  - Tooth thickness
  - Face width
  - Speed of gears

- Speed [V_G]: (given) 9800.0 rev/min
- Pressure Angle [Φ]: (given) 20°
- Applied Torque [$T_{applied}$]: (given) 643.1 ± 0.5 in · lbf
- Drive Speed [V_{DS}]: (given) 6975.0 rev/min
- Gear Train Value [e]: $e = \frac{N2}{N3} \frac{N3}{N4} \frac{N5}{N6} = \frac{24}{15} \frac{15}{19} \frac{26}{31} = 1.6 \cdot 0.517 \cdot 0.838 = 0.6941$
- Calculated Drive Speed [V_{DSC}]: $e \cdot V_G = 0.6941 \cdot 9800 = 6802.225$ rev/min
- Diametral Pitch [P_d]: 10 $\frac{\text{teeth}}{\text{in}}$
- Number of Teeth [N]: Selected N2 = 24

- Pitch Diameter [D_p] : $D_{P(2)} = \frac{N2}{P_d} = \frac{24}{10} = 2.4$ in

- Addendum [A_D] : $A_D = \frac{1}{P_d} = \frac{1}{10} = 0.1$ in

- Dedendum [D_D] : $D_D = \frac{1.35}{P_d} = \frac{1.35}{10} = 0.135$ in

- Tooth Thickness [t] : $t = \frac{\pi \cdot A_D}{2} = \frac{\pi(0.1)}{2} = 0.157$ in

Any adjustments of chosen values are depicted in the AGMA factor of Safety calculation.

Gear Analysis [Bending]

This Section displays the calculations for the allowable Contact stress and AGMA bending Stress in order to ultimately determine resulting factor of Safety.

- Material Selection : 31 Cr Mo V9 annealed Steel @ 205°C
  - Tensile Strength [KPSI] : 236.0
  - Yield Strength [KPSI] : 212.0
  - Elongation [%] : 10.0
  - Reduction in Area [%] : 41.0
  - Brinell Hardness [H_B] : 601.0

- Pressure Angle [ϕ] : 20°
- Diametral Pitch [P_d] : 10 $\frac{\text{Teeth}}{\text{in}}$
- Pitch Diameter [D_p] : $D_{P(2)} = 2.4$ in $D_{P(3)} = 1.5$ in (Excel)
- Brinell Hardness [B_H] : 601.0
- Speed [V_G] : 9800 rev/min
- Pinion Speed [nP] : $nP = \frac{D_{P(2)} \cdot V_G}{D_{P(3)}} = \frac{2.4 \cdot 9800}{1.5} = 15680$ rev/min

- Speed Ratio [mG] : $mG = \dfrac{D_{P(2)}}{D_{P(3)}} = \dfrac{2.4}{1.5} = 1.6$

- Pitch Line Velocity [V_{PL}] : $V_{PL} = \dfrac{\pi \cdot D_{P(3)} \cdot nP}{12} = \dfrac{\pi(1.5)(15680)}{12} = 6157.5216 \ \dfrac{ft}{min}$

- Horse power [Hp] : $H_P = \dfrac{V_G \cdot T_{applied}}{5252} = \dfrac{9800\,(643.1)}{5252} = 1199.996 \ hp \xrightarrow{T(\frac{1}{2}) \to N2} \dfrac{9800\,(321.55)}{5252} = 599.998 \ hp$

- Tangential Force [W_t] : (Pinion) $W_t = \dfrac{6000 \cdot H_P}{V_{PL}} = \dfrac{6000\,(599.998)}{6157.5216} = 584.649 \ lbf$

- Pinion Bending Strength Geometry Factor [J_P] : 0.250 Figure 14-6 using $\frac{}{15}\rfloor^{24}$ and interpolation

- Gear Bending strength Geometry Factor [J_G] : 0.320 Figure 14-6 using $\frac{}{24}\rfloor^{15}$ and interpolation

- Surface Strength Geometry Factor [I] : $I = \dfrac{\cos(\phi) \cdot \sin(\phi)}{(2 \cdot mH)\left(\dfrac{mH}{mG+1}\right)} = \dfrac{\cos(20^\circ) \cdot \sin(20^\circ)}{(2 \cdot 1)\left(\dfrac{1.6}{1.6+1}\right)} = 0.1146$

- Pinion Elastic Coefficient [E_P] (C_P) : $E_P = 2300 \sqrt{PSI}$

- Gear Elastic Coefficient [E_G] (C_G) : $E_G = 2300 \sqrt{PSI}$

Lewis form factor Pinion = 0.296

Lewis form factor Gear = 0.377

- Overload Factor [K_o] : 1.250 moderate shock

- Pinion Size Factor [K_{sP}] : $K_{sP\,(15)} = 1.192\left(\dfrac{F_P \cdot \sqrt{Y_P}}{P_d}\right)^{0.0535} = 1.192\left(\dfrac{1 \cdot \sqrt{0.296}}{10}\right)^{0.0535} = 1.0195$

- Gear Size Factor [K_{sG}] : $K_{sG\,(24)} = 1.192\left(\dfrac{F_P \cdot \sqrt{Y_G}}{P_d}\right)^{0.0535} = 1.192\left(\dfrac{1 \cdot \sqrt{0.377}}{10}\right)^{0.0535} = 1.023$

- Dynamic Factor Variable [B] : $0.25(12 - Q_v)^{2/3} = 0.25(12-10)^{2/3} = 0.3968$

- Dynamic Factor Variable [A] : $A = 50 + 56(1 - B) = 50 + 56(1 - 0.3968) = 83.7792$

- Dynamic Factor Variable [K_v] : $K_v = \left(\dfrac{A + \sqrt{V_{PL}}}{A}\right)^B = \left(\dfrac{83.7792 + \sqrt{6157.5216}}{83.7792}\right)^{0.3968} = 1.2998$

- Face Width [F_P] : $F_P = 1$ in

- Surface Condition Factor [C_f] : $C_f = 1$ unity

- Uncrowned Teeth [C_{mc}] : $C_{mc} = 1.0$

- $C_{pf} = \frac{F}{10 \cdot D_{p(2)}} - 0.025 = 0.04166$ $\qquad F_p \leq 1\,in$

- $C_{pm} = 1.1$ (Straddle-mounted pinion, $S_1/S \geq 0.175$)

- $C_{ma} = 0.127 + (0.0158)(1.000in) + (-0.930 \times 10^{-4})(1.000in)^2 = 0.1427$

- $C_e = 0.8$ (Gearing to be adjusted at assembly) unity

- Load Distribution Factor $[K_m]$: $K_m = C_{mf} = 1 + C_{mc}(C_{pf}C_{pm} + C_{ma}C_e) = 1.16$

- Rim Thickness Factor $[K_b]$: $K_b = 1$ unity

- AGMA Bending Stress Calculations for Pinion: $\sigma_{pinion} = W^t(K_o)(K_v)(K_{sp})\left(\frac{P_d}{F_p}\right)\left(\frac{K_m \cdot K_b}{J_p}\right)$

$$= 584.649(1.25)(1.2998)(1.0195)\left(\frac{10\ teeth/in}{1}\right)\left(\frac{(1.16)\quad(1)}{0.250}\right)$$

$$\boxed{\sigma_{pinion} = 44{,}941.8\ PSI}$$

- AGMA Bending Stress Calculations for Gear : $\sigma_{Gear} = W^t(K_o)(K_v)(K_{sG})\left(\frac{P_d}{F_p}\right)\left(\frac{K_m \cdot K_b}{J_G}\right)$

$$= 584.649(1.25)(1.2998)(1.023)\left(\frac{10\ teeth/in}{1}\right)\left(\frac{(1.1594)(1)}{0.320}\right)$$

$$\boxed{\sigma_{Gear} = 35{,}252\ PSI}$$

- AGMA Bending Stress Calculations : $\sigma_{allow} = 108.6(H_B) + 15890\ psi = \boxed{81{,}158.6\ PSI}$ (Figure 14-3) ✓

<u>Gear Analysis [Bending] Continued</u>

This Section will display calculations for AGMA bending Stress factor of Safety.

- Bending Stress Cycle factor for pinion $[Y_{NP}]$: $Y_{NP} = 1.3558(N)^{-0.0178} = 1.3558(10^8)^{-0.0178} = 0.9767$

- Bending Stress Cycle factor for gear $[Y_{NG}]$: $Y_{NG} = 1.3558\left(\frac{N}{3}\right)^{-0.0178} = 1.3558\left(\frac{10^8}{3}\right)^{-0.0178} = 0.9961$

- Reliability Factor $[K_R]$: $K_R = 0.9549$ (interpolation)
  (given : 0.98) $\qquad (Y_Z)$

- Temperature Factor $[K_T]$: 1 unity

Safety Factor for bending Pinion $[FOS_P]$: $\dfrac{\frac{(\sigma_{allow})\, Y_{NP}}{K_t\, K_R}}{\sigma_{pinion}} = \dfrac{\frac{(81{,}158.6)(0.9767)}{(1)(0.9549)}}{44{,}941.8} = \boxed{1.85}$ ✓

Safety Factor for bending Gear $[FOS_G]$: $\dfrac{\frac{(\sigma_{allow})\, Y_{NG}}{K_t\, K_R}}{\sigma_{Gear}} = \dfrac{\frac{(81{,}158.6)(0.9961)}{(1)(0.9549)}}{35{,}252} = \boxed{2.40}$ ✓

Gear Analysis [Contact]

This Section will display calculations for allowable contact stress.

- Contact Stress Cycle Factor for Pinion $[Z_{NP}]$: $Z_{NP} = 1.4488(10^8)^{-0.023} = 0.9484$
- Contact Stress Cycle Factor for Gear $[Z_{NG}]$: $Z_{NG} = 1.4488\left(\frac{10^8}{3}\right)^{-0.023} = 0.9727$
- Reliability Factor $[K_R]$: $K_R = 0.9549$
- Surface Strength Geometry Factor $[I]$: $I = \dfrac{\cos(\phi)\cdot\sin(\phi)}{(2\cdot mH)\left(\frac{mH}{mG+1}\right)} = 0.1146$
- Overload Factor $[K_o]$: 1.250 *moderate shock*
- Dynamic Factor Variable $[K_V]$: $K_V = \left(\dfrac{A + \sqrt{V_{PL}}}{A}\right)^B = \left(\dfrac{83.7792+\sqrt{6157.5216}}{83.7792}\right)^{0.3968} = 1.2998$
- Pinion Size Factor $[K_{sp}]$: $\underset{(15)}{K_{s_p}} = 1.192\left(\dfrac{F_P\cdot\sqrt{Y_P}}{P_d}\right)^{0.0535} = 1.192\left(\dfrac{1\cdot\sqrt{0.296}}{10}\right)^{0.0535} = 1.0195$
- Gear Size Factor $[K_{sG}]$: $\underset{(24)}{K_{sG}} = 1.192\left(\dfrac{F_P\cdot\sqrt{Y_G}}{P_d}\right)^{0.0535} = 1.192\left(\dfrac{1\cdot\sqrt{0.377}}{10}\right)^{0.0535} = 1.023$
- $C_e = 0.8$ (Gearing to be adjusted at assembly) unity

- Face Width [F_p] : $F_p = 1$ in

- Pinion Elastic Coefficient [E_p] : $E_p = 2300 \sqrt{PSI}$
  C_p

- Tangential Force [W_t] : $W_t = \frac{6000 \cdot H_p}{V_{PL}} = \frac{6000\,(599.998)}{6157.5216} = 584.649$ lbf
  (Pinion)

- AGMA Contact Stress Pinion [σ_{pinion}] : $\sigma_{pinion} = C_P \sqrt{W^t K_o K_v K_{sp} \frac{K_m}{d_p F_p} \frac{C_f}{I}}$

$$= 2300 \sqrt{(584.649)(1.25)(1.2998)(1.0195) \frac{1.16}{(1.5)(1)} \cdot \frac{1}{0.1146}}$$

$$\boxed{\sigma_{pinion} = 185{,}828.45 \text{ PSI} \Rightarrow 185{,}918.67 \text{ (excel)}}$$
(rounding)

- AGMA Contact Stress Pinion [σ_{Gear}] : $\sigma_{Gear} = C_P \sqrt{W^t K_o K_v K_{sG} \frac{K_m}{d_G F_G} \frac{C_f}{I}}$

$$= 2300 \sqrt{(584.649)(1.25)(1.2998)(1.023) \frac{1.16}{(1.5)(1)} \; \frac{1}{0.1146}}$$

$$\boxed{\sigma_{Gear} = 186{,}292.54 \text{ PSI}}$$

Gear Analysis [(Contact] Continued

This Section displays the Safety factor for pitting failure.

- Reliability Factor [K_R]: $K_R = 0.9549$

- Temperature Factor [K_T]: 1 unity

- Contact Stress Cycle Factor for Pinion [Z_{NP}]: $Z_{NP} = 1.448(10^8)^{-0.023} = 0.9484$

- Contact Stress Cycle Factor for Gear [Z_{NG}]: $Z_{NG} = 1.448\left(\frac{10^8}{3}\right)^{-0.023} = 0.9727$

- Brinell Hardness [B_H]: 601.0

- Contact Fatigue for Pinion [Sc_p]: $Sc_p = 349(H_B) + 34300 = 244049$ Grade 2

- Hardness Ratio Factor [C_H]: $H_P = 1.0$

(Same material)

- Hardness Ratio Factor [C_H]: $H_G = 1.0$

- AGMA Contact Factor of Safety Pinion [FOS_P]: $FOS_P = \dfrac{\frac{Sc_p Z_{NP} H_P}{K_R \; K_T}}{\sigma_{pinion}}$

$$= \frac{\frac{(244049)(0.9484)(1)}{(0.9549)(1)}}{185{,}918.67} = \boxed{1.30} \checkmark$$

- AGMA Contact Factor of Safety Gear [FOS_G]: $FOS_P = \dfrac{\frac{Sc_G Z_{NG} H_G}{(K_R)(K_T)}}{\sigma_{Gear}}$

$$= \frac{\frac{(244049)(0.9727)(1)}{(0.9549)(1)}}{186{,}292} = \boxed{1.33} \checkmark$$

<u>Accessory Drive Shaft Analysis:</u>

In the following sections the calculations for the drive shaft reactions based on loadings are used to determine the bending moments for the entire shaft based on gears 2, 12 and the bevel gear. Each of the reactions at the centerpoint of those gears are broken into components to allow for each moment on the shaft to be calculated. With these moments the maximum bending stress can be determined.

Table 17: Calculations and Specifications for Accessory Drive Shaft

Variables	Value
Input Torque | T_{input}	643.1 in·lbf
X-Distance of Mating Gears [2&3] | x_1	1.933 in
Y-Distance of Mating Gears [2&3] | y_1	0.256 in
Tangential Force | W_{t1}	584.649 lbf
Radial Force | W_{r1}	284.283 lbf
O to Gear 2 | L_1	1.000 in
O to Gear 12 | L_2	3.715 in
O to Bevel Gear | L_3	5.312 in
O to A | L_4	10.29 in
Accessory Drive Shaft Selected Diameter | d	2.000 in
Pressure Angle | ϕ	20.0°
Pitch Diameter of Bevel | d_p	3.50 in
Face Width of Bevel | F	1.53 in
Pitch Angle | γ	35.000°

Accessory Drive Shaft Analysis

Calculation of the Shaft's loadings, reactions, and bending moments.

The Shaft is Subject to constant bending moment.

Bending Stress is completely reversed

$$\sigma_{min} = \sigma_{max}$$

$$M_m = 0$$

$$T_a = 0$$

- Distance from Center-to-Center of Mating Gears $[X_{(1)}]$:

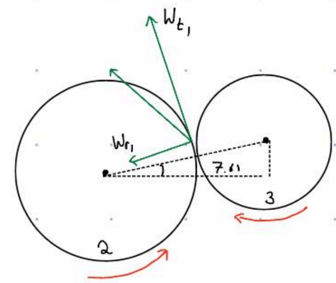
$$\text{Gears } [2-3]: \quad X_1 = \frac{(D_G + D_P)\cos(\theta)}{2} = \frac{(2.4 + 1.5)\cos(0.31)}{2} = 1.933 \text{ in}$$

- Distance from Center-to-Center of Mating Gears $[Y_{(1)}]$:

$$\text{Gears } [2-3]: \quad Y_1 = \frac{(D_G + D_P)\sin(\theta)}{2} = \frac{(2.4 + 1.5)\cos(0.31)}{2} = 0.256 \text{ in}$$

- Tangential Force $[W_{t_1}]$: Components

$$\theta_1 = \tan^{-1}\left(\frac{Y_1}{X_1}\right) = \tan^{-1}\left(\frac{0.256}{1.933}\right) = 0.131 \text{ rad}$$

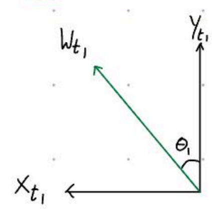


$$X_{t_1} = W_{t_1}\sin\theta_1 = 584.649\sin(0.131\text{ rad}) = -76.37 \text{ in}$$

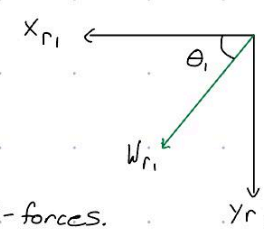
$$Y_{t_1} = W_{t_1}\cos\theta_1 = 584.649\cos(0.131\text{ rad}) = -579.61 \text{ in}$$

$$X_{r_1} = W_{r_1}\cos\theta_1 = 212.795\cos(0.131\text{ rad}) = -210.96 \text{ in}$$

$$Y_{r_1} = W_{r_1}\sin\theta_1 = 212.795\sin(0.131\text{ rad}) = -27.89 \text{ in}$$



Sum of X and Y components for Gear 2:



$$F_{x_1} = X_{t_1} + X_{r_1} = -76.37 - 210.96 = -287.57 \text{ lbf}$$

$$F_{y_1} = Y_{t_1} + Y_{r_1} = -579.61 - 27.89 = 607.5 \text{ lbf}$$

Same Process for Gear 12 for x-forces and y-forces.

Calculation on <u>bevel Gear</u> for tangential and radial Components.

· Midpoint Pitch radius [r_{av}]:

$$r_{av} = \frac{d_p - F\sin(\gamma)}{2} = \frac{3.50 - 1.53\sin(35)}{2} = 1.311 \text{ in}$$

· Tangential Component [W_t]:

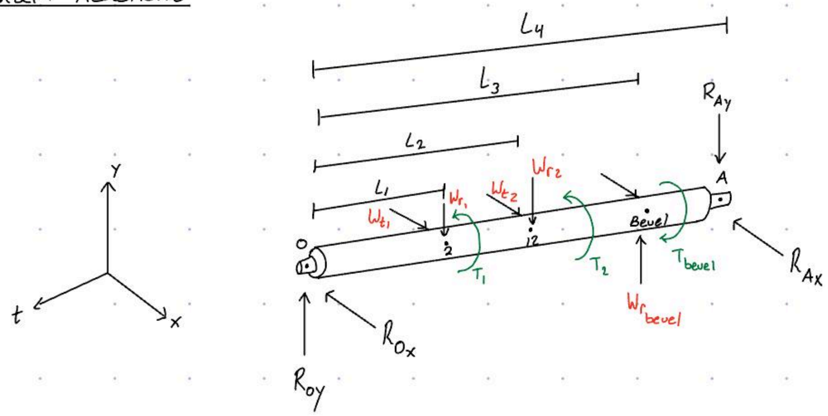
$$W_{t_{bevel}} = \frac{T_{input}}{r_{av}} = \frac{643.1}{1.311} = 490.464 \text{ lbf}$$

· Radial Component [W_r]:

$$W_{r_{bevel}} = W_{t_{bevel}} \tan\phi \cos\gamma = 490.54 \tan(20°)\cos(35°) = 146.09 \text{ lbf}$$

(146.230 Excel)

<u>Shaft Reactions</u>



$L_1 = 1.00$ in

$L_2 = 3.715$ in

$L_3 = 5.3125$ in

$L_4 = 10.29$ in

Using the previous Calculation on forces, the <u>reaction forces</u> Can be Determined for the Shaft analysis.

$$\sum M_{Ox} = 0 = -F_{x_1}(L_1) + F_{x_2}(L_2) + W_{t_{bevel}}(L_3) - R_{Ax}(L_4)$$

$$0 = -(287.57)(1) + (185.65)(3.715) + (490.54)(5.3125) - R_{Ax}(10.29)$$

$$R_{Ax} = 347.35 \text{ lbf} \quad (348.187 \text{ Excel})$$

$$R_{Ax} = \frac{-(-287.575)(1) + 185.65(3.715) + 490.462(5.3125)}{10.290}$$

$$\sum M_{Oy} = 0 = F_{y_1}(L_1) + F_{y_2}(L_2) + W_{r_{bevel}}(L_3) - R_{Ay}(L_4)$$

$$0 = -607.5(1) + -461.818(3.715) + 146.230(5.3125) - R_{Ay}(10.29)$$

$$R_{Ay} = 183.188 \text{ lbf}$$

Reactions about point A in order to find $\underline{R_{ox}}$, $\underline{R_{oy}}$.

$$\sum F_x = 0 = -R_{ox} - F_{x_1} + F_{x_2} + W_{t_{bevel}} - R_{Ax}$$

$$0 = -R_{ox} - (-287.57) + 185.65 + 490.464 - 348.187$$

$$R_{ox} = 615.5 \text{ lbf}$$

$$\sum F_y = 0 = R_{oy} + F_{y_1} - F_{y_2} + W_{r_{bevel}} - R_{Ay}$$

$$0 = R_{oy} + 607.5 - (-461.818) + 146.23 - 183.188$$

$$R_{oy} = 182.634 \text{ lbf}$$

Shaft Bending Moment

Bending moments about each gear can now be calculated by using the x and y components.

Gear 2:

$$M_{x_2} = R_{ox}(L_1) = 615.5\,(1) = 615.5 \text{ lbf}$$

$$M_{y_2} = R_{oy}(L_1) = 182.634\,(1) = 182.634 \text{ lbf}$$

Gear 12:

$$M_{x12} = R_{ox}(L_2) + F_{x_2}(L_2 - L_1) = (615.5 \cdot 3.715) + 185.650\,(3.715 - 1) = 2790.62 \text{ in}\cdot\text{lbf}$$

$$M_{y12} = R_{oy}(L_2) + F_{y_2}(L_2 - L_1) = 182.634(3.715) + (-461.818)(3.715 - 1) = -575.35 \text{ in}\cdot\text{lbf}$$

Bevel

$$M_{Bx} = R_{ox} L_3 + F_{x_2}(L_3 - L_1) = 615.5(5.3125) + (-287.575)(5.3125 - 1) + 185.65(5.3125 - 3.715)$$

$$M_{Bx} = 2326.249 \text{ in}\cdot\text{lbf}$$

$$M_{By} = R_{oy} L_3 + F_{y_1}(L_3 - L_1) + F_{y_2}(L_3 - L_2) = (182.634)(5.3125) + (-607.5)(5.3125 - 1) + (-461.818)(5.3125 - 3.715)$$

$$M_{By} = -2387.326$$

Bending Moments

$$M_2 = \sqrt{M_{x_2}^2 + M_{y_2}^2} = \sqrt{(615.5)^2 + (182.634)^2} = \boxed{642.024 \ in \cdot lbf}$$

$$M_{12} = \sqrt{M_{x_{12}}^2 + M_{y_{12}}^2} = \sqrt{(2790.62)^2 + (-575.35)^2} = \boxed{2849.314 \ in \cdot lbf}$$

$$M_B = \sqrt{M_{B_x}^2 + M_{B_y}^2} = \sqrt{(2326.249)^2 + (-2387.326)^2} = \boxed{3333.281 \ in \cdot lbf}$$ Critical ✓

From the calculations we can see that the maximum moment on the shaft is on gear 12 at 3333.281 in · lbf occuring.

Therefore, the maximum bending stress can be found.

Bending Stress

$$\sigma = \frac{Mc}{I} = \frac{32(3333.281)}{\pi(2.0)} = \boxed{4244.07 \ PSI}$$

These specific calculations were performed for each of the accessory drive components in order to determine loading reactions and bending moments for the ultimate goal of obtaining a satisfactory Factor of Safety.

Accessory Drive Shaft Diameter Sizing, Fatigue Analysis, and Factors of Safety:

In order to develop the correct S-N curve for the drive shaft, all factors were selected based on the material, diameters, and maximum bending moment. With that, a proper fatigue factor of safety could be calculated.

Table 18: Calculations and Selections for Fatigue analysis and S-N Curve

Variables	Value
Correction Factor | f	0.781
constant | a	93.9
constant | b	0.1101
Endurance Limit | S'_e	100000 psi
Surface Factor | k_a	0.6205
Size Factor | k_b	0.816
Loading Factor | k_c	1
Temperature Factor | k_d	1.025
Reliability Factor | k_e	0.8275
Miscellaneous-Effects Factor | k_f	1
Actual Endurance Limit | S_e	42954.861 psi
Fatigue Strength | S_f	713.6 kpsi

Accessory Drive Diameter Analysis

Each shaft was designed with a diameter that met the design requirements, taking into consideration the parameters of the objective.

The required factor of safety is $\eta_d = 1.15$.

- Acessory Drive Shaft Diameter $[d_a]$ $d_a = 2.0$ in

Material: 4140 Steel Alloy Quenched and Tempered at 400°F

- Tensile Strength $[S_{ut}]$: 257 kpsi
- Elongation: 8%
- Brinell Hardness: 510
- Yield Strength $[S_y]$: 238 kpsi
- Reduction: 28%

- Endurance Limit $[S_e']$: $S_e' = 100$ kpsi for $S_{ut} > 200$ kpsi
- Surface Factor $[K_a]$: $K_a = a(S_{ut})^b = 2.7(S_{ut})^{-0.265} = 0.6205$ (Michined)
- Size Factor $[K_b]$: $K_b = 0.879(d^{-0.107}) = 0.816$
- Loading Factor $[K_c]$: $K_c = 1$ (Equation 6-25)
- Temperature Factor $[K_d]$: $K_d = 1.0250$ (Estimated)
- Reliability Factor $[K_e]$: $1 - 0.08(z_a)$ $K_e = 0.8275$
- Miscellaneous - Effects Factor $[K_f]$: $K_f = 1.0$ (Assumed)
- Stress Concentration Factor Bending $[K_t]$: $K_t = 5.0$ (Table 7-1)
- Stress Concentration Factor Torsion $[K_{ts}]$: $K_{ts} = 3.0$ (Table 7-1)

It is now known that the maximum bending moment is $M_a = 3333.281$ in·lbf

- Torque Between Gears $[T_m]$: $T_m = W_{t_2}\left(\frac{d_2}{2}\right) = 584.649\left(\frac{2.4}{2}\right) = 701.578$ in·lbf

- Accessory Drive Shaft Diameter [d]: $d = 2.0$ in

- Selected Larger Diameter [D]: $D = 2.5$ in

- Ratio $\left[\frac{D}{d}\right]$: 1.25

<u>Endurance Limit</u>

$$S_e = K_a K_b K_c K_d K_e K_f S_e'$$

$$= (0.6205)(1)(0.816)(1.0250)(0.8275)(1.0)(100000)$$

$$S_e = 42954.861 \text{ PSI}$$

<u>Fatigue Strength Equation</u>

∵ Correction factor $f = 0.781$

$$a = \frac{(f \cdot S_{ut})^2}{S_e} = \frac{(0.781 \cdot 257)^2}{429.54} = 93.9$$

$$b = -\frac{1}{3}\log\left(\frac{f \cdot S_{ut}}{S_e}\right) = -\frac{1}{3}\log\left(\frac{0.781 \cdot 257}{429.54}\right) = 0.1101$$

$$S_f = a N^b = 93.9(10^8)^{0.1101} = 713.6 \text{ KPSI}$$

Using <u>S_{ut}</u> <u>$f \cdot S_{ut}$</u> <u>S_e</u> The S-N curve was developed

DE - Goodman

This is used to determine the minimum diameter.

Reversible and continuous

$$d = \left(\frac{16n}{\pi} \left\{ \frac{1}{S_e} \left[4(K_f M_a)^2 + 3(K_{fs} \overset{0}{\cancel{T_a}})^2 \right]^{1/2} + \frac{1}{S_{ut}} \left[4(K_f \overset{0}{\cancel{M_m}})^2 + 3(K_{fs} T_m)^2 \right]^{1/2} \right\} \right)^{1/3}$$

$$= \left(\frac{16(1.15)}{\pi} \left\{ \frac{1}{42954.861} \left[4(1 \cdot 3333.281)^2 \right]^{1/2} + \frac{1}{257000} \left[3(2.2 \cdot 701.578)^2 \right]^{1/2} \right\} \right)^{1/3}$$

$$\boxed{d = 0.646 \text{ in}}$$

Fatigue Factor of Safety

The Drive Shaft Utilizes:

- Bearing Shoulder
- Groove (Snap ring)
- Keyway

- $S_{ut} = 257000$ PSI
- $S_y = 238000$ PSI
- $S_e = 42954.861$ PSI
- Diameter of accessory drive $[d_a]$: $d_a = 2.0$ in
- Shoulder larger diameter selected $[D]$: $D = 2.5$ in
- Ratio of Bearing Shoulder $\left[\frac{r}{d}\right]$: 0.05
- Ratio for Bearing Shoulder $\left[\frac{D}{d}\right]$: 1.25

- Stress Concentration Factor for Torsion $[K_{tsB}]$: $K_{tsB} = 1.650$
  - Based on Table 3-12 Torsion PDF
- Bearing Shoulder [r] : $r = 0.10$ (well rounded fillets)

$$d = \left(\frac{8nA}{\pi S_e} \left\{ 1 + \left[1 + \left(\frac{2BS_e}{AS_{ut}} \right)^2 \right]^{1/2} \right\} \right)^{1/3}$$

Neuber Constant for Bending and Torsion

$$a = 0.246 - 3.08(10^{-3})S_{ut} + 1.51(10^{-5})S_{ut}^{2} - 2.67(10^{-8})S_{ut}^{3}$$

$$= 0.246 - 3.08(10^{-3})257000 + 1.51(10^{-5})257000^{2} - 2.67(10^{-8})257000^{3}$$

$$a = 0.0115$$

$$a_s = 0.19 - 2.51(10^{-3})S_{ut} + 1.35(10^{-5})S_{ut}^{2} - 2.67(10^{-8})S_{ut}^{3}$$

$$= 0.19 - 2.51(10^{-3})257000 + 1.35(10^{-5})257000^{2} - 2.67(10^{-8})257000^{3}$$

$$a_s = 0.002$$

Fatigue Stress-Concentration Bearing Shoulder

Notch Sensitivity for Bending Torsion

$$q = \frac{1}{1 + \frac{\sqrt{a}}{\sqrt{r}}} = \frac{1}{1 + \frac{0.025}{\sqrt{0.10}}} = 0.965$$

$$q_s = \frac{1}{1 + \frac{\sqrt{a_s}}{\sqrt{r}}} = \frac{1}{1 + \frac{0.002}{\sqrt{0.10}}} = 0.994$$

- $K_t = 1.950$

Neuber Equation

$$K_{f_B} = 1 + q(K_{t_B} - 1) = 1 + 0.965(1.950 - 1) = 1.917$$

$$K_{fs_B} = 1 + q_s(K_{ts_B} - 1) = 1 + 0.994(1.650 - 1) = 1.646$$

DE - Gerber FOS Bearing Shoulder

$$\frac{1}{n} = \frac{8 \cdot A_B}{(\pi d^3 S_e)} \left\{ 1 + \left[1 + \left(\frac{2 B_B S_e}{A_B S_{ut}} \right)^2 \right]^{1/2} \right\}$$

$$A_B = \sqrt{4(K_f M_a)^2 + 3(K_{fs} T_a)^2} = \sqrt{4(1.917 \cdot 3333.281)^2} = 12776.982$$

$$B_B = \sqrt{4(K_f M_m)^2 + 3(K_{fs} T_m)^2} = \sqrt{3(1.646 \cdot 701.579)^2} = 2000.089$$

$$\frac{1}{n} = \frac{8(12776.982)}{\pi(2.0)^3(42954.861)} \left\{ 1 + \left[1 + \left(\frac{2(2000.089)(42954.861)}{(12776.982)(257000)} \right)^2 \right]^{1/2} \right\}^{-1}$$

$$\boxed{n_B = 7.275}$$

KeyWay Fatigue Analysis DE - Gerber FOS

$$q = \frac{1}{1 + \frac{\sqrt{a}}{\sqrt{r}}} = \frac{1}{1 + \frac{0.0115}{\sqrt{0.025}}} = 0.932$$

$\because r = 0.025$

$$q_s = \frac{1}{1 + \frac{\sqrt{a_s}}{\sqrt{r}}} = \frac{1}{1 + \frac{0.002}{\sqrt{0.025}}} = 0.988$$

$$K_f = 1 + q(K_t - 1) = 2.063$$

$$K_{fs} = 1 + q_s(K_{ts} - 1) = 2.975$$

$K_t = 2.14$ (Table 7-1 and $\frac{r}{d}$ value)

$K_{ts} = 3$

$A_B = \sqrt{4(K_f M_a)^2 + 3(K_{fs} T_a)^2} = \sqrt{4(2.063 \cdot 3333.281)^2} = 13749.864$ $\because T_a = 0$

$B_B = \sqrt{4(K_f M_m)^2 + 3(K_{fs} T_m)^2} = \sqrt{3(2.975 \cdot 3333.281)^2} = 3615.287$ $\because M_m = 0$

$$\frac{1}{n} = \frac{8A}{\pi d^3 S_e}\left\{1 + \left[1 + \left(\frac{2 B S_e}{A S_{ut}}\right)^2\right]^{1/2}\right\}$$

$$= \frac{8(13749.864)}{\pi(2)^3(42954.861)} \cdot \left\{1 + \left[1 + \left(\frac{2(3615.287)(42954.861)}{(13749.864)(257000)}\right)^2\right]^{1/2}\right\}$$

$\boxed{n_K = 4.898}$

Groove Fatigue Analysis DE - Gerber FOS

$q = \dfrac{1}{1 + \frac{\sqrt{a}}{\sqrt{r}}} = \dfrac{1}{1 + \frac{0.0115}{\sqrt{0.02}}} = 0.925$ $\because r = 0.025$

$q_s = \dfrac{1}{1 + \frac{\sqrt{a_s}}{\sqrt{r}}} = \dfrac{1}{1 + \frac{0.002}{\sqrt{0.02}}} = 0.986$

$K_f = 1 + q(K_t - 1) = 4.698$

$K_{fs} = 1 + q_s(K_{ts} - 1) = 2.972$

$K_t = 5$ (Table 7-1 and $\frac{r}{d}$ value)

$K_{ts} = 3$ (Table 7-1)

$$A = \sqrt{4(K_f M_a)^2 + 3(K_{fs} T_a)^2} = \sqrt{4(4.698 \cdot 3333.281)^2} = 31322.439 \qquad \because T_a = 0$$

$$B = \sqrt{4(K_f M_m)^2 + 3(K_{fs} T_m)^2} = \sqrt{3(2.972 \cdot 701.579)^2} = 3611.769 \qquad \because M_m = 0$$

$$\frac{1}{n} = \frac{8A}{\pi d^3 S_e} \left\{ 1 + \left[1 + \left(\frac{2 B S_e}{A S_{ut}} \right)^2 \right]^{1/2} \right\}$$

$$= \frac{8(31322.439)}{\pi (2)^3 (42954.861)} \cdot \left\{ 1 + \left[1 + \left(\frac{2(3611.769)(42954.861)}{(31322.439)(257000)} \right)^2 \right]^{1/2} \right\}$$

$$\boxed{n_G = 2.153}$$

Design FOS for Accessory Drive Shaft

Using Von Mises stress to calculate fos in order to check Yield FOS.

Maximum Von Mises stress [σ'_{max}]: $\sigma' = \left[\left(\frac{32 K_f (M_m + M_a)}{\pi d^3} \right)^2 + 3 \left(\frac{16 K_{fs} (T_m + T_a)}{\pi d^3} \right)^2 \right]^{1/2}$

$\because K_f$ from Groove Calculation

$$= \left[\left(\frac{32(4.698)(0 + 3333.281)}{\pi (2)^3} \right)^2 + \left(\frac{16(2.972)(701.579 + 0)}{\pi (2)^3} \right)^2 \right]^{1/2}$$

$$\boxed{\sigma' = 20072.613}$$

Yield Factor of Safety

$$n_y = \frac{S_y}{\sigma'_{max}} = \frac{257000}{20072.613} = \boxed{12.804}$$

Team Reporting Spreadsheet

ME314 TEAM REPORTING WORKSHEET - SPRING 2024									
TEAM #									
		% COMPLETE							
SHAFT ANALYSIS & SIZING	RESPONSIBLE STUDENT	WEEK 1	WEEK 2	WEEK 3	WEEK 4	WEEK 5	WEEK 6	WEEK 7	WEEK 8
SHAFT SUPPORT REACTIONS	Sean and Maxx	N/A	N/A	N/A	N/A	N/A	30	75	100
SHEAR & BENDING MOMENT DIAGRAMS	Edgar	N/A	N/A	N/A	N/A	N/A	0	50	100
STRESS CONCENTRATION FACTORS	William and Maxx and Sean	N/A	N/A	N/A	N/A	N/A	20	45	100
STATIC ANALYSIS	Jacob	N/A	N/A	N/A	N/A	N/A	33	67	100
FATIGUE ANALYSIS	William and Maxx	N/A	N/A	N/A	N/A	N/A	0	40	100
S-N CURVES	Isaac and William	N/A	N/A	N/A	N/A	N/A	0	40	100
MATERIAL SELECTION	Sean	N/A	N/A	N/A	N/A	N/A	100	100	100
		% COMPLETE							
GEARTRAIN ANALYSIS & SIZING	RESPONSIBLE STUDENT	WEEK 1	WEEK 2	WEEK 3	WEEK 4	WEEK 5	WEEK 6	WEEK 7	WEEK 8
GEARTRAIN LAYOUT	Everybody	N/A	N/A	N/A	N/A	N/A	50	90	100
GEAR FORCES	Sean and Maxx	N/A	N/A	N/A	N/A	N/A	25	50	100
GEAR BENDING STRESS ANALYSIS	Everybody	N/A	N/A	N/A	N/A	N/A	25	50	100
GEAR CONTACT STRESS ANALYSIS	Everybody	N/A	N/A	N/A	N/A	N/A	25	50	100
GEAR RATIOS	Everybody	N/A	N/A	N/A	N/A	N/A	50	100	100
MATERIAL SELECTION	Everybody	N/A	N/A	N/A	N/A	N/A	100	100	100
		% COMPLETE							
FINAL REPORT	RESPONSIBLE STUDENT	WEEK 1	WEEK 2	WEEK 3	WEEK 4	WEEK 5	WEEK 6	WEEK 7	WEEK 8
DESIGN REQUIREMENTS REVIEW	Everybody	N/A	N/A	N/A	N/A	N/A	50	75	100
SHAFT SIZING & ANALYSIS	Everybody	N/A	N/A	N/A	N/A	N/A	0	50	100
GEARTRAIN ANALYSIS	Everybody	N/A	N/A	N/A	N/A	N/A	50	75	100
GEAR STRESS ANALYSIS	Everybody	N/A	N/A	N/A	N/A	N/A	33	67	100
GRAPHICS/TABLES	Isaac and Jacob	N/A	N/A	N/A	N/A	N/A	0	50	100
FINAL REPORT	Everybody	N/A	N/A	N/A	N/A	N/A	50	75	100
TEAM ATTESTATION	SIGNATURE	LEVEL OF EFFORT (Score of 1-5)							
Maxx Pastore	Maxx Pastore	N/A	N/A	N/A	N/A	N/A	4	5	5
Isaac Rodriguez	Isaac Rodriguez	N/A	N/A	N/A	N/A	N/A	4	4	5
Joshua Hammond	Joshua Hammond	N/A	N/A	N/A	N/A	N/A	4	4	5
Edgar Diaz	Edgar Diaz	N/A	N/A	N/A	N/A	N/A	4	5	5
Sean Hedgecock	Sean Hedgecock	N/A	N/A	N/A	N/A	N/A	5	5	5
Jacob Artolachipe	Jacob Artolachipe	N/A	N/A	N/A	N/A	N/A	4	4	5
William Fischer	William Fischer	N/A	N/A	N/A	N/A	N/A	4	5	5