DREXEL UNIVERSITY
College of Engineering

MEM431

Final Report:

Gearbox Design Analysis

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Important Information: In order to prevent myself from becoming confused with the abundance of variables, I used variables taught to me previously and might not necessarily be the same as what was taught in this course.

Introduction:

The design project assigned was to design a single reduction, spur gear-type speed reducer while adhering to the following condition: A water turbine transmits 50 kilowatts (kW) of power to a pair of gears at 4500 revolutions per minute (rpm). The output of the gear pair must drive an electric power generator at 3600 rpm. The center distance for the gear pair requires not to exceed 150 millimeters (mm). As such, there was a certain, limited, degree of freedom allowed when creating the gear and the pinion.

Calculations & Analysis:

The first step in the gearbox design was calculating the pitch diameters of both the gear and the pinion. In order to calculate this two equations were utilized with the center distance restriction being substituted into the equation; $CD = r_p + r_g = \frac{D_p + D_g}{2} = 300mm$, where CD represents center distance, r represents the radii and D the pitch diameters for the pinion and gear (also represented by subtext 1 and subtext 2). The simple gear ratio equation was used in conjunction with the center distance equation being; $\frac{N_1}{N_2} = \frac{D_g}{D_p} = \frac{4500rpm}{3600rpm}$, where N is the input and output rpm respectively. After all substitutions the pitch diameter of the pinion and gear were determined to be 133.3mm and 166mm, respectively. However for more accurate CAD dimensions and more precise figures resulting from these figures, the pitch diameters were rounded down to 132mm and 166mm. Once the pitch diameters were rounded down, the new center distance was calculated at 149.5mm. After this, the following power relation equation; $P = \frac{2\pi N_1 T_1}{60} = \frac{2\pi N_2 T_2}{60} = 50E + 3 = \frac{2\pi \cdot 4500 T_1}{60}$, (where P is power, N is rotational speed, and T is torque) was utilized to calculate the torque in the pinion and was found to be 106.1 newton-meters (Nm). Once the pinion torque was calculated, the following relation; $N_1 T_1 = N_2 T_2 \rightarrow 4500 \cdot 106.1 = 3600 T_2$, was used to calculate the torque in the gear which was found to be 132.625 Nm. Then, using relations found in the supplementary text, Machine Elements in Mechanical Design, a preemptive module was calculated with; $m = \frac{T_1}{T_2} = \frac{106.1}{132.625} = 1.25$, where m is the module of the gear and pinion and resulted in a module of 1.25. However after substituting this module into; $m = \frac{D_1}{z_1} = \frac{D_2}{z_2}$, where z is the number of teeth per gear/pinion, the number of teeth for the gear was 132.8 and 105.6 for the pinion. This number of teeth resulted in very thin teeth which made them susceptible to fracture so because of this, the module was increased to 2 which decreased the tooth count for the pinion and gear to 66 and 83 respectively.

Next, the tangential forces for each gear were calculated using; $W_{t1,t2} = \frac{T_{1,2}}{.5D_{1,2}}$, where W is the tangential force. The tangential forces were found to be 1598N and 1608N for the pinion and the gear. Using the calculated tangential forces and substituting into the following equation; $W_{r1,r2} = W_{t1,t2} tan\phi$, where phi represents the pressure angle, the radial forces were calculated to be 582N and 585N. Finally, once the radial forces were calculated, the normal force in each gear could be found using; $W_{n1,n2} = \sqrt{W^2_{t1,t2} + W^2_{r1,r2}}$, where W sub n is the normal forces, the normal forces were calculated to be 1700 N and 1711 N.

Material Selection:

In order to create a gearbox that could withstand the stresses applied by the water turbine, different materials were required for the different components. The material selected for the gear and pinion is SAE 1022 steel single water quenched and tempered at 350 degrees fahrenheit (SWQT350). This steel was selected for its relatively low cost, reasonable tensile strength for the purposes, and a high brinell hardness rating. SAE 1022 steel has a tensile strength of 931 megapascals (MPa), yield strength of 517 MPa, and a brinell hardness rating (HB) of 262. The shaft selected for use is 200mm long and made of AISI 1566 carbon steel which has a yield strength of 517 MPa and a tensile strength of 992 MPa. This material was selected for its high strength and its common use in marine engineering because of its ability to resist seawater corrosion which is pertinent for a water turbine. The bearing selected is a steel open ball bearing with an ABEC 1 rating, dynamic radial load capacity of 380 N (per bearing), and maximum speed of 37,000 rpm. This bearing easily reaches the standard required by the radial load because there are two bearings per shaft and four in total. The fasteners selected are a black-oxide class 12.9 alloy steel hexagon socket head screw and a class 10.9 steel nut. The screw has a tensile strength of 1220 MPa, yield strength of 1100 MPa, and a proof load of 970 MPa while the nut has a tensile strength of 1040 MPa, yield strength of 940 MPa, and proof load of 830 MPa. As both fasteners are only used to secure the gearbox case and the bearing housings to the gearbox case combined with the fact that there are 44 total (of each nut and screw), they provide more than enough support to secure the gearbox. The final member of the gearbox was something added in order to secure the bearings in place and it is the bearing housing which is made of 2014 aluminum and has a yield strength of 414 MPa and a tensile strength of 483 MPa.