Design of Planetary Gear Train for Geared Rotary Actuator

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Abstract: Geared Rotary Actuators are mechanically operable devices widely used in the aerospace industry so that controlled motion can be provided to secondary flight control surfaces. Their usage can be seen in a variety of applications including powered hinges for aircraft structural movements, they support aerodynamic loads in addition to the surface hinge moment, as an actuator in a linked mechanism where they supply actuation torque. Mostly GRAs are situated along with the bay door drives, and also can be used for controlling leading or trailing edge movements of the aircraft. They receive Power Drive Unit (PDU) torque via assembly of shafts and joints. From a mechanical standpoint, the GRAs can be termed as epicyclical gear reducer or compound planetary gearbox. In this work, the governing criteria and mathematical equations used to characterize GRA are discussed. SOLIDWORKS is used for part modeling of GRA. More focus will be on the speed reduction ratio calculation, determination of strength, and overall Factor of Safety (FOS) of the torque transmission unit. Thus an attempt is made to obtain the best performance parameters to meet the desired requirements of Geared Rotary Actuator.

Keywords: Geared Rotary Actuators, secondary flight control surfaces, powered hinge, power drive unit, SOLIDWORKS, factor of safety

1. INTRODUCTION

An Actuator is a part of a machine that is mainly responsible for moving and controlling a mechanism or a complete system. An actuator receives input as a control signal and a source of energy. A control signal is relatively low energy and maybe electric voltage or current, pneumatic or hydraulic pressure, or can be a human-generated power. When it receives a control signal, an actuator reacts by converting the signal’s energy into desired motion.

Mechanical Actuators are generally used to translate mechanical motion (often rotary) into linear motion or vice versa with the help of gearing arrangements. These actuators are typically part of a larger system which can be power drive units, mechanical interconnects, and feedback devices to control the motion of multiple devices.
1.1 Geared Rotary Actuator (GRA)

Geared Mechanical Actuator is a geared mechanism that is used to convert mechanical motion (often rotary) into rotary motion at a different combination of speeds and force. They are also used when required to change the direction of motion. Unlike hydraulic or electromechanical actuators, they receive their power from an external source. Further GRA is classified as simple planetary actuators and compound differential planetary actuators. Simple planetary actuators are most commonly found in commercial leading-edge slat applications and are bolted to structure driving rack and pinions to translate the slat surface. Compound differential planetary actuators are most commonly used in trailing edge flap designs. It offers higher ratios for torque multiplication while driving rack and pinions to translate the flap surfaces.

Geared Rotary Actuators are initially used for the positional control of wing-mounted flight control Surfaces of an aircraft. Their usage ranges from their application in powered hinges, when directly supporting the aerodynamic load forces in addition to the surface hinge moment, to actuators in a linked mechanism where they provide the actuation torque and the aerodynamic load reaction and is delegated to other load-bearing equipment. In almost all cases, the GRAs are situated along with the leading or trailing edges of an aircraft and are connected to each other via shafts and joints while transferring the Power Drive Unit torque to the GRAs.

1.2 Working of GRA

Geared Rotary Actuators (GRA’s) can be used to control the movement of leading-edge flaps of aircraft wing Structures to alter the lift characteristics of the wing. Each actuator includes an epicyclic-type reduction gearing arrangement which transmits the rotation of an input member to a pair of output members. One of which is fixed to the wing Structure, and So is usually referred to as an 'earth' and another one is coupled to the control Surface to move the control Surface relative to the wing Structure of an aircraft. Geared rotary actuators (GRA) are mechanical assemblies performing speed reduction and torque amplification from the input shaft to the actuator output and providing reaction to the control surface loads. The compound planetary gear arrangement differs from a conventional epicyclic gear arrangement by eliminating the carrier assembly for planetary gears. This is made possible by balancing the tangential teeth forces on the planetary pinions at the ring gear meshes.
The compound planetary gear set contains a sun gear receiving the input torque through the input shaft, a ring gear supplying the output torque to the driven element, two stationary ring gears symmetrically placed at the two sides of the output ring gear. The power flows from the input sun gear to the output ring gear through multiple planet gears meshing around the complete periphery of the sun gear. Each planet gear consists of three pinions: the outer end pinion gears are identical to each other and are meshing with two fixed outer ring gears. For each planet gear, the tooth load at the center mesh is reacted by the two identical forces at the fixed meshes. A full arrangement of planetary gears is used to distribute the load over a large number of gear teeth which, in addition to minimizes the stresses on the gear tooth and ensures more even load distribution which ultimately helps to increase the stiffness of an actuator.

The output element of the geared rotary actuator is radially supported by the fixed housing. Radial support occurs along two parallel circular surfaces, symmetrically located at the two sides of the actuator output. Such type of support provides the necessary radial reaction force to balance the force acting on the actuator output. While the moment of the external force about the actuator axis is balanced by the torque provided by the actuator output gear. As a result, the actuator behaves as a powered hinge. The Power Hinge can be subjected to torque, shear, and axial thrust loads. The hinge shear and thrust loads are isolated from the hinge moment (torque) by the annular ball bearings coupling the movable output ring gear to the fixed ring gears. Since the fixed ring gears are typically supported by aircraft structure, all loads originating at the output ring gear are transmitted through the annular ball bearings to the fixed ring gear and the aircraft structure. Thus, the spindle gears do not experience unbalanced loads; only torque about the axis of the actuator.

2. NUMERICAL MODEL OF GRA

As for epicyclic gear reducers, gears rotating about a fixed axis when observed from a static observer are called sun gears, while gears whose axis is not stationary are defined planets underlining their revolving motion around the sun gears. The gears making up the rotary actuators are spur gears which are sized according to the module and the number of teeth. An important feature of the teeth commonly used in the GRA planets is the rounded axial profile. This non-standard geometry accommodates the planets deformation due to the bending moment, thereby ensuring a more uniform load distribution along the teeth length that contributes to reduce localized stress. It is also important to underline that most often the gears of a GRA are cut with a large correction increasing the root thickness permitting higher load transfer in gears with a limited number of teeth (whole assembly must be compact as possible and should be light in weight). Desired Technical Specifications of this GRA are as follows:
• Torque Required at the output (to be provided by moving ring gear) = 4500 N-m at 20 RPM
• Speed available at input of sun gear = 2964 RPM
• Desired Reduction Ratio = 148.2:1
• Desired Time of Operation = less than 2 secs

2.1 Calculations for Reduction Ratio

The speed ratio between output and input of a GRA can be calculated by applying repeatedly the well-known Willis equation used for ordinary epicyclic gear reducers.

\[ r = \frac{\omega_n - \omega_p}{\omega_1 - \omega_p} \]
\[ r = \frac{Z_2Z_5 - 1}{Z_2Z_4 + 1} \] \hspace{1cm} \ldots \ldots \text{Willi’s equation}

As we are designing this GRA for reduction ratio purpose, it can be obtained by taking inverse of Gear Ratio as,

\[ \text{Reduction Ratio} = \frac{1}{\text{Gear Ratio}} = \frac{1}{r} \]

For obtaining correct value of reduction ratio we must know the values of number of teeth on sun, planetary pinion and ring gear respectively. This can be obtained by fulfilling the following conditions while selecting number of teeth. There are three essential conditions for the planetary gear train to transmit the power from input shaft to output shaft. These essential conditions are as:

- **Condition-1:**
  This expression is valid for the gear system designed as standard gear system. No. of teeth on the gear can be modified by gear design based on profile shift. Centre distance between two gears to be matched. i.e. \( a_1 = a_2 \)

  \[ Z_4 = Z_1 + 2Z_2 \]

- **Condition-2:**
  It is requisite for placing the planet gears consistently about the sun gear.

  \[ \frac{Z_1 + Z_4}{N} = 1 \]

- **Condition-3:**
  It ensures the operation of the planet gears without any interference with each other. This condition is the requisite for meeting the standard gear design where planet gears are equally placed.

  \[ Z_2 + 2 < (Z_1 + Z_2) \sin \left( \frac{180}{N} \right) \]

- **Condition-4:**
  In such compact designs while operating at such higher reduction ratios, gears in mesh may subject to higher wear and undercutting problems. So, this condition ensures that while operating, gears would not undercut.

  \[ (Z_4 + 2) < \left( \frac{Z_1' + Z_4}{2} \right) \]

These four conditions are necessary for any planetary gear train to transmit power and motion. Any of these conditions not getting satisfied will result in non-feasible planetary gear train design. After satisfying all the above conditions and performing satisfying number of iterations we finalised number of teeth values as follows,
Number of teeth on Sun Gear, \( Z_1 = 35 \)
Number of teeth on Sun Gear Spline, \( Z_1' = 34 \)
Number of teeth on Central Gear of planetary pinion, \( Z_2 = 28 \)
Number of teeth on Outer Gear of planetary pinion, \( Z_3 = 29 \)
Number of teeth on Moving Ring Gear, \( Z_4 = 91 \)
Number of teeth on Fixed Ring Gear, \( Z_5 = 92 \)

This combination of teeth satisfies all the above four mentioned conditions, as well as it gives desired high reduction ratio without any undercutting of gears.

Condition-1: **Satisfied**

\[
Z_4 = Z_1 + 2Z_2 \\
91 = 35 + (2 \times 28) \\
91 = 91
\]

Condition-2: **Satisfied**

\[
\frac{Z_1 + Z_4}{N} = 1 \\
\frac{35 + 91}{6} = 21 \\
21 = 21
\]

Condition-3: **Satisfied**

\[
Z_2 + 2 < (Z_1 + Z_2) \sin \left( \frac{180}{N} \right) \\
30 < 31.5
\]

Condition-4: **Satisfied**

\[
(Z_3 + 2) < \left( \frac{Z_1' + Z_3}{2} \right) \\
(29 + 2) < \left( \frac{34 + 29}{2} \right) \\
31 < 31.5
\]

Now, reduction ratio is given by

\[
r = \frac{Z_2Z_5}{Z_4Z_3} - 1 \\
\frac{Z_2Z_5}{Z_4Z_3} + 1
\]

\[
r = \frac{(28 \times 92)}{(29 \times 91)} - 1 = \frac{-0.006747638}{35 \times 29} + 1
\]

So, **reduction ratio** \( \frac{1}{r} = \frac{1}{-0.006747638} \approx -148.2:1 \)

Here we achieved the desired reduction ratio and with this we are able to provide 4500 N-m torque as an output via moving ring gear. As we know that, this whole compound planetary assembly is used for converting high-speed low-torque input into low-speed high-torque output with one of its ring gear connected to flight control surfaces while other remain fixed and bolted to the aircraft structure. So during operation, whole planetary gear assembly might be subjected to various stresses and hence, it should have
sufficient strength so that torque will be transmitted efficiently without failure of any parts as well as it should possess sufficient value of Factor of Safety.
From the observations it has been seen that spur gears are most likely subjected to bending and wear type of failures. Out of the two, industrial gears are mostly designed with the consideration of wear failure only. Hence, we designed gears with the consideration of wear strength criteria while determining desired value of Factor of Safety (FOS).

2.2 Material used, Dimensions of Gears and FOS Calculations

- **Material to be used** = AISI 440C / ASTM S44004 / X105CrMo17
- **Application Areas** = Antifriction Bearings and Races, Structural Uses in Corrosive environments, Heavy duty industrial gears, Transmission Shafts, Aircraft Forgings
- **Ultimate tensile strength** = 2170 MPa
- **Yield Tensile Strength** = 2090 MPa
- **Type of heat Treatment** = Martensitic Stainless steel - Tempered at 316 °C

- **Dimensions of Gears**

  - **Module (m)** = 1.75
  - **Pressure angle (α)** = 20°
  - **Centre distance,**

    \[
    \text{C.D}_1 = \left(\frac{Z_1 + Z_2}{2}\right) \times m = \frac{35 + 28}{2} \times 1.75 = 55.125 \text{ mm}
    \]

    \[
    \text{C.D}_2 = \left(\frac{Z_4 - Z_2}{2}\right) \times m = \frac{91 - 28}{2} \times 1.75 = 55.125 \text{ mm}
    \]

  - **Addendum,** \( h_a = 1 \times m = 1.75\)
  - **Tooth depth** \((h) = 2.25 \times m = 3.9375\)
  - **Minimum Rim Thickness,** \( t_r \text{(min)} = 1.2 \times \text{(Tooth Depth)} = 1.2 \times 3.9375 = 4.725 \text{ mm}\)
  
  So, minimum rim thickness while designing the gears should be at least 4.725 mm.

- **Pitch Circle Diameter \((D_p)\)**

  \[
  D_{p1} = m \times Z_1 = 61.25 \text{ mm}
  \]

  \[
  D_{p2} = m \times Z_2 = 49 \text{ mm}
  \]

  \[
  D_{p3} = m \times Z_3 = 50.75 \text{ mm}
  \]

  \[
  D_{p4} = m \times Z_4 = 159.25 \text{ mm}
  \]

  \[
  D_{p5} = m \times Z_5 = 161 \text{ mm}
  \]

- **Tip Diameter \((D_a)\)**

  \[
  D_{a1} = D_{p1} + (2 \times m) = 64.75 \text{ mm}
  \]

  \[
  D_{a2} = D_{p2} + (2 \times m) = 52.5 \text{ mm}
  \]

  \[
  D_{a3} = D_{p3} + (2 \times m) = 54.25 \text{ mm}
  \]

  \[
  D_{a4} = D_{p4} - (2 \times m) = 155.75 \text{ mm}
  \]

  \[
  D_{a5} = D_{p5} - (2 \times m) = 157.5 \text{ mm}
  \]

- **Root Diameter \((D_f)\)**

  \[
  D_{f1} = D_{a1} - (2 \times h) = 56.875 \text{ mm}
  \]

  \[
  D_{f2} = D_{a2} - (2 \times h) = 44.625 \text{ mm}
  \]

  \[
  D_{f3} = D_{a3} - (2 \times h) = 46.375 \text{ mm}
  \]

  \[
  D_{f4} = D_{a4} + (2 \times h) = 163.625 \text{ mm}
  \]
\[ D_{fs} = D_{a5} + (2 \times h) = 165.375 \text{ mm} \]

- **Factor of Safety (FOS)**

\[
FOS = \frac{S_b}{P_{eff}}
\]

\[
FOS = \frac{S_w}{P_{eff}}
\]

**Effective Load acting on Gear Tooth (Peff):**

\[
P_{eff} = \frac{C_x}{C_v} \times P_t
\]

Pitch Line Velocity (v)

\[
v = \frac{\pi \times D_p \times n}{60000} = \frac{\pi \times 52.5 \times 2964}{60000} = 9.505674 \text{ m/s}
\]

Velocity factor (C_v)

\[
C_v = \frac{3}{3 + v} = 0.2398911 \quad (as \ v < 10 \text{ m/s})
\]

By referring table no.1 value of service factor (C_s) can be taken as, 1.75.

**TABLE 1. Service Factor for speed reduction Gearboxes**

| Working Characteristics of Driving Machine | Working Characteristics of Driven Machine |
|-------------------------------------------|----------------------------------------|
|                                            | Uniform | Medium Shock | Heavy Shock |
| Uniform                                   | 1       | 1.25         | 1.75        |
| Light Shock                               | 1.25    | 1.5          | 2           |
| Medium Shock                              | 1.5     | 1.75         | 2.25        |

Now, tangential force due to rated torque can be found as: (Considering 6 planet gears)

\[
P_t = \frac{2 \times M_t}{6D_{p1}} = \frac{2 \times 30364.37}{6 \times 61.25} = 165.2482721 \text{ N}
\]

Effective load acting on gear tooth can be given as,

\[
P_{eff} = \frac{1.75 \times 165.2482721}{0.2398911} = 1205.48226 \text{ N}
\]

**Beam Strength of Gear tooth (S_b):**

(For 1 and 2 gear pair)

\[
S_b = m \times b \times Y \times \sigma_b
\]

\[
= 1.75 \times 35 \times 0.373 \times 723.33
\]

\[
= 16525.45417 \text{ N}
\]

Factor of Safety for 1-2 pair is,

\[
FOS = \frac{S_b}{P_{eff}} = \frac{16525.45417}{1205.48226} = 13.709
\]

**Wear strength of gear tooth (S_w):**
(For 1 and 2 gear pair)

\[
S_w = D_{p1} \times b \times Q \times k \\
S_w = D_{p1} \times b \times Q \times 0.16 \times \left( \frac{BHN}{100} \right)^2
\]

Where, ratio factor \( (Q) = \frac{2z_g}{Z_g + Z_p} = \frac{2 \times 35}{35 + 28} = 1.111 \)

\[
S_w = 61.25 \times 35 \times 1.11 \times 5.9536 = 14181.144 \text{ N}
\]

Factor of Safety for 1-2 pair is,

\[
FOS = \frac{S_w}{P_{eff}} = \frac{14181.144}{1205.48226} = 11.764
\]

But, in this case factor of safety is very high. Hence, as per the standards for industrial purpose, here we have used Buckingham's equation for attaining required factor of safety (FOS).

Buckingham’s equation for dynamic load \( (P_d) \)

\[
P_d = \frac{21v(c \times e \times b + P_t)}{21v + \sqrt{(c \times e \times b) + P_t}}
\]

Where, \( c \) is deformation factor and \( e \) is sum of errors between two Meshing Teeth can be determined by using table no. 2 and table no.3 respectively.

**TABLE 2. Values of Deformation Factor**

| Materials       | 14.5° full depth Teeth | 20° full depth Teeth | 20° Stub Teeth |
|-----------------|------------------------|---------------------|---------------|
| Pinion Mtl.     | Gear Mtl.              |                     |               |
| Grey CI         | Grey CI                | 5500                | 5700          | 5900          |
| Steel           | Grey CI                | 7600                | 7900          | 8100          |
| Steel           | Steel                  | 11000               | 11400         | 11900         |

**TABLE 3. Tolerances on the Adjacent Pitch**

| Grade | \( e \) (microns) |
|-------|-------------------|
| 1     | 0.80 + 0.06\( \varphi \) |
| 2     | 1.25 + 0.10\( \varphi \) |
| 3     | 2.00 + 0.16\( \varphi \) |
| 4     | 3.20 + 0.25\( \varphi \) |
| 5     | 5.00 + 0.40\( \varphi \) |
| 6     | 8.00 + 0.63\( \varphi \) |
| 7     | 11.00 + 0.90\( \varphi \) |
| 8     | 16.00 + 1.25\( \varphi \) |
| 9     | 22.00 + 1.80\( \varphi \) |
| 10    | 32.00 + 2.50\( \varphi \) |
| 11    | 45.00 + 3.55\( \varphi \) |
| 12    | 63.00 + 5.00\( \varphi \) |
To determine sum of errors between two meshing teeth (e) we used Grade 3 tolerance on adjacent pitch, which will be used further to determine dynamic load (P_d).

\[ e = 2.00 + 0.16\phi \]

Now, Total error (e) is given by,

\[ e = e_p + e_g \]

Where,

- \( e_p \): Error for Pinion
- \( e_g \): Error for Gear

Where \( \phi \) is Tolerance Factor and generally taken as,

\[ \phi = m + 0.25 \sqrt{D_p} \]

\[ P_d = \frac{21 \times 9.5056(11400 \times 0.0052 \times 35 + 165.248272)}{21 \times 9.5056 + \sqrt{(11400 \times 0.0052 \times 35) + 165.248272}} \]

\[ = 1797.0296 \text{ N} \]

For Buckingham’s Equation Effective load can be determined as,

\[ P_{eff} = (C_s \times P_t + P_d) \]
\[ = (1.75 \times 165.248272) + 1797.0296 \]
\[ P_{eff} = 2086.2 \text{ N} \]

In industrial and heavy duty applications gears are mostly subjected to fatigue and repetitive type of stresses. Hence, for such applications spur gears are mostly designed with consideration of wear failure criterion. Hence, FOS can be found as:

\[ FOS = \frac{S_w}{P_{eff}} = \frac{4673.48224}{4535.542} = 6.7975 \]

So, here we got comparatively better values of factor of safety. But still there is scope of improvement by changing parameters like face width of gears (b) up to some extent or by changing material used for manufacturing of GRA.

### 3. RESULTS

As per the given data we need to design a device which will be capable of transmitting very high torque at very low speeds. As per the given data, device must be capable of providing 4500 N-m torque as output with output gear rotating at the speed of 20 RPM. So, here we have designed carrier-less compound Planetary Gear Train which will be capable of providing high torque as an output. While designing gear set which will meet our requirements, we have performed number of iterations to find out suitable number of teeth values so that design will be compact and operable according to our requirements.

a) Using Willi’s formula for gear ratio we determined satisfactory value of reduction ratio. i.e. Reduction ratio = -148.2:1

b) Using this reduction ratio value and after satisfying all the tooth combination criteria’s obtained values of tooth combinations are as:

- \( Z_1=35, \ Z'_1=34, \ Z_2=28, \ Z_3=29, \ Z_4=91, \ Z_5=92 \)

c) While satisfying all the dimensional criteria’s of PGT, it’s also required that torque must be transmitted efficiently without any failure of the component as well as without affecting the performance of device. Hence, factor of safety is determined based on wear strength failure criteria for each transmission pair of gears using Buckingham’s dynamic load equation. It’s as follows:

- For 1-2 gear pair, determined value of factor of safety (FOS) is 6.7975
- For 2-4 gear pair, determined value of factor of safety (FOS) is 2.5292
- For 3-5 gear pair, determined value of factor of safety (FOS) is 2.0742

4. CONCLUSION

One of the prime objectives of this work is to develop a system which is highly compact and light in weight. From the analytical and numerical results we are able to obtain high reduction ratio which made this system possible to provide low-speed high-torque as output. As well as, we obtained satisfying value of Factor of safety while taking into consideration the criteria of wear failure. From the study it has been observed that Factor of Safety (FOS) can be further improved by altering various parameters. Basic parameters which also affect the FOS are Ultimate strength (Uts) well as yield strength (Yts) of material, face width of gear, module of gear and service factor of gear. On improving the value of Uts, Yts and face width of gear, the value of Factor of Safety also starts increasing. Factor of Safety can also be improved by decreasing the value of service factor. But there are some drawbacks of altering these parameters. While improving FOS, one can increase the value face width of gear up to certain extent only. Because while optimizing the value of FOS by changing these parameters, also increases size and ultimately weight of the gear unit. Similarly, on decreasing the value of service factor improves the value of FOS. But on the other hand it also affects the life of Product. Hence, one needs to select the parameters according to desired requirements of the product in such a way that it would not affect other parameters of the design.

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