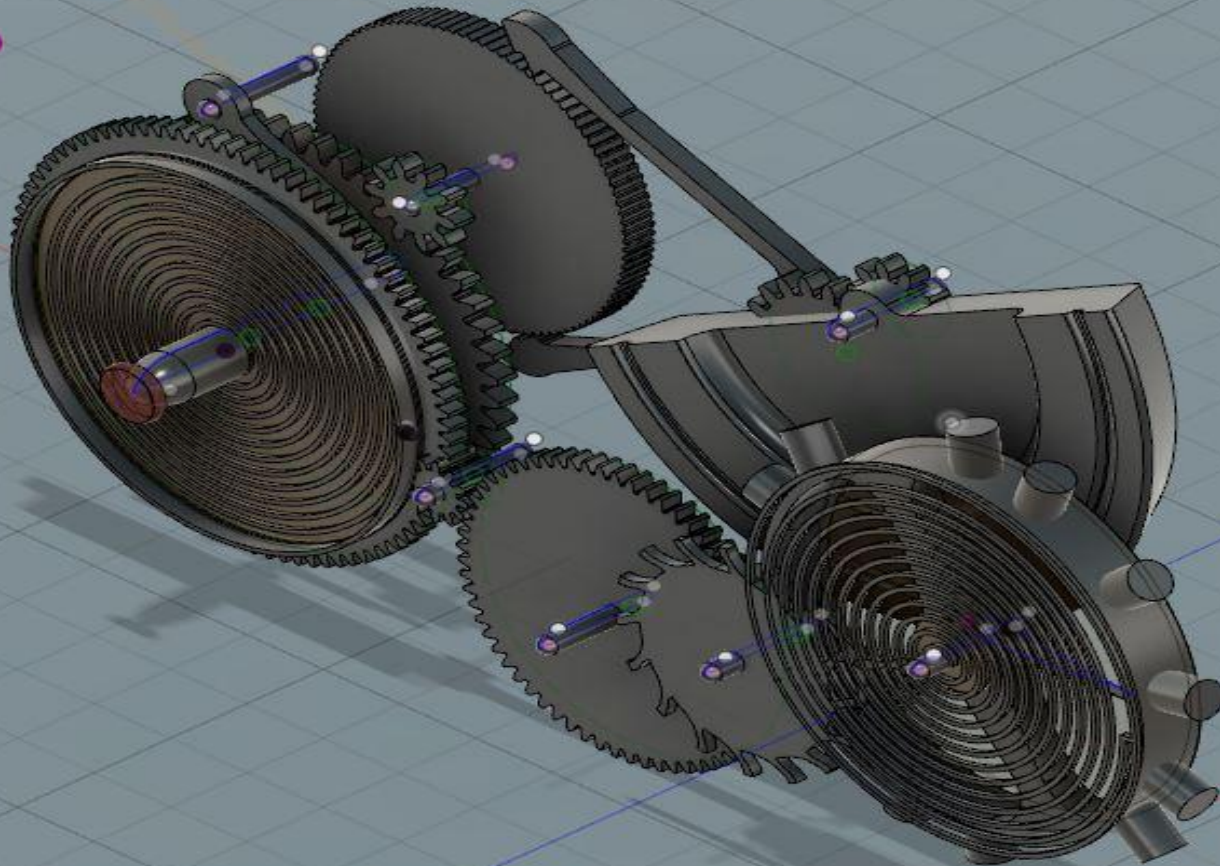


SELF WINDING MECHANISM IN MECHANICAL WATCHES

JYOTESH CHOWDARY - ME15B149
RIDHI PUPPALA- ME15B133
S N VENKATESWARAN - ME15B134

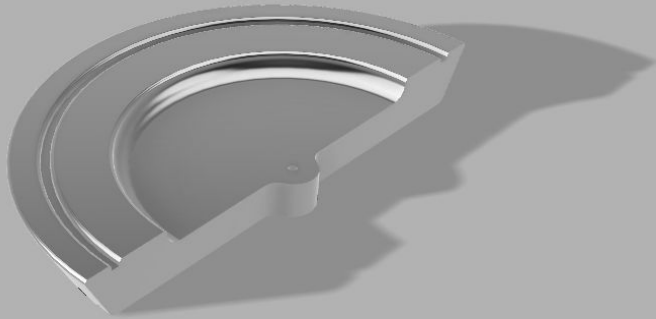
MODELLED IN FUSION 360



Design Analysis and Calculation

- Magic lever mechanism - Ratchet wheel and Lever
- Spur Gear sets (Pinion and Gear) - 4
- Spiral Torsional Spring - Hairspring
- Jewel Bearing
- Supporting shaft at the axis of hairspring for Torsion
- Self Winding rotor mass

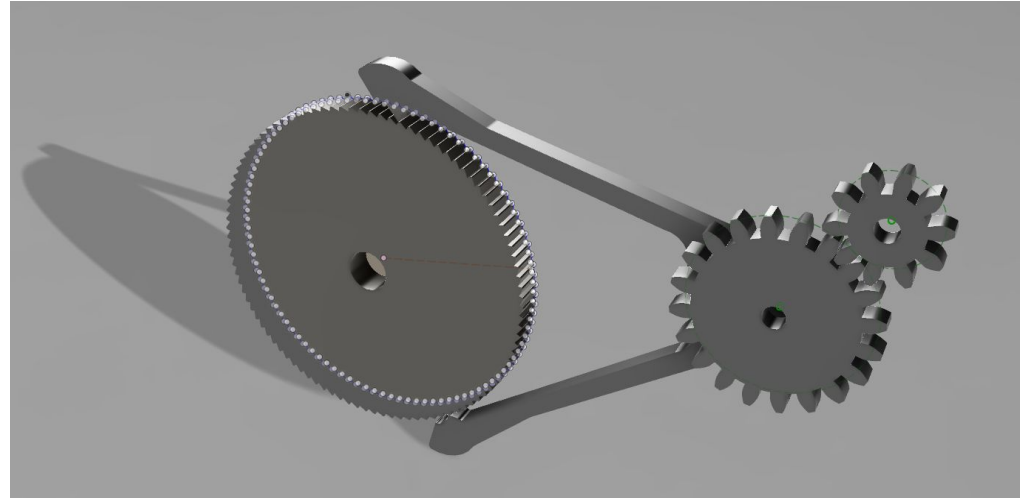
Rotor mass for Self winding



-
- For a watch of typical external diameter 40mm, we choose a semi-circular mass of diameter 36 mm.
 - Appropriate material considered is stainless steel
 - Density = 7700 kg/m^3
 - Volume = $\pi * d^2 * t / 8$ $t = 3\text{mm}, d = 36\text{mm}$
 - Mass = volume * density ~ 12 grams
 - Max Torque due to gravity = $Y_{\text{COM}} * mg = 2r / \pi * mg$
 - $T_{\text{max}} = 0.001348 \text{ N-m.}$

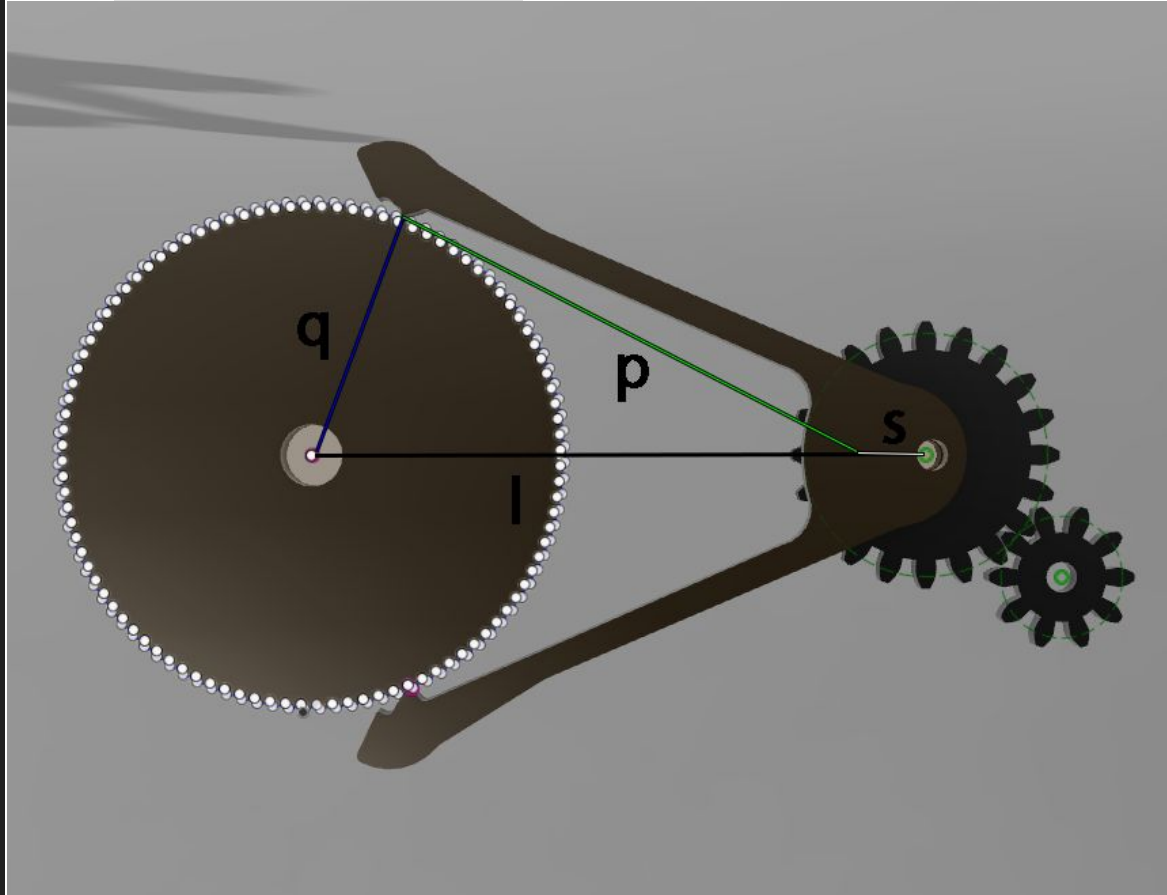
Magic Lever mechanism

- For achieving a unidirectional winding for the torsional spring while the mass rotates in any direction, we have decided to use magic lever mechanism.

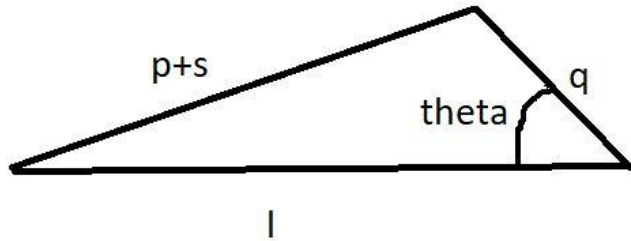


Magic Lever mechanism

FOUR BAR MECHANISM



Magic Lever mechanism



- Grashof criterion for crank rocker is $(s+l < p+q)$
- To achieve a span of around 30° on the ratchet wheel for every full rotation of the gear to which the lever is rigidly fixed.

For the above conditions

s = eccentricity of lever pivot from Gear2 centre
= 2 mm (assumed)

p = lever length \sim 15 mm

q = diameter of the ratchet wheel 8mm

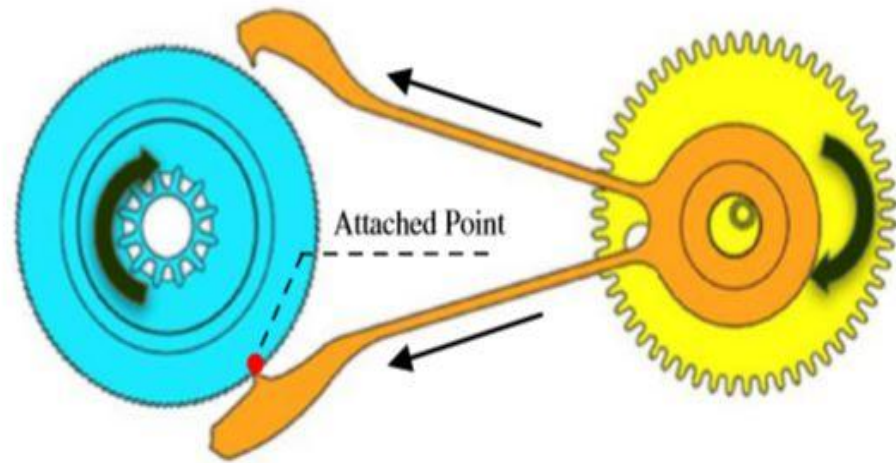
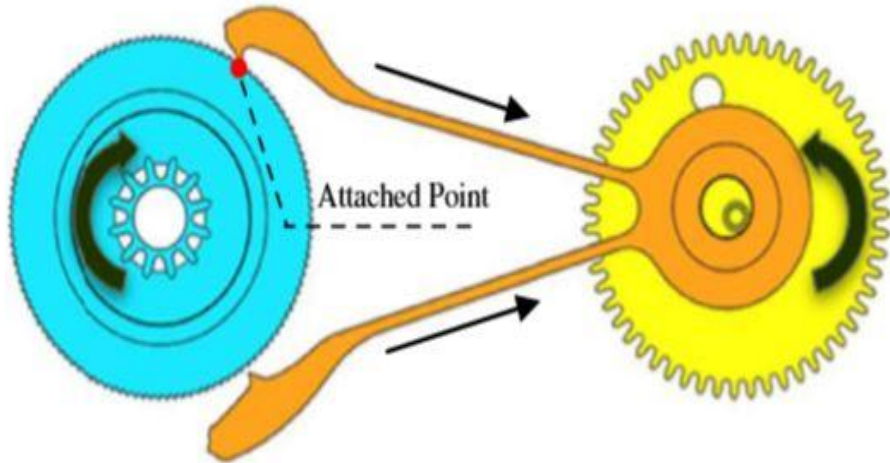
l = centre distance = 20 mm

$\theta = 56.8$ $\phi = 22.798$ ($\theta - \phi$) = 34°

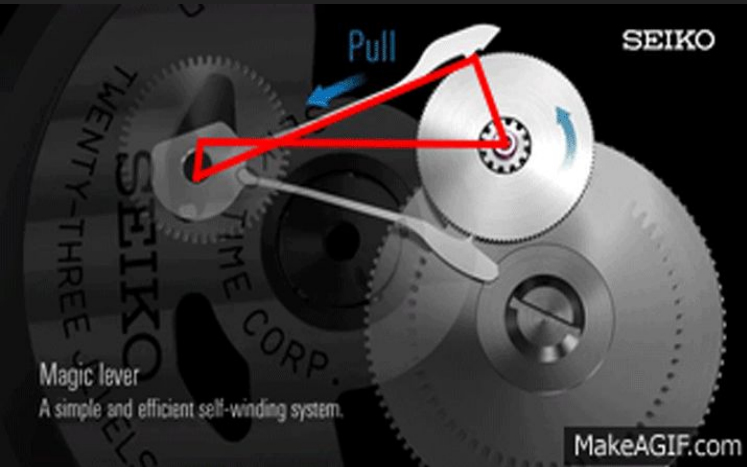
Assumed 100 teeth for a 8mm OD ratchet wheel

Magic Lever mechanism

- The lever meshes with the ratchet wheel only at one point in each direction of rotation.
- Due to different arm lengths a torque multiplication factor of (q/e) is given to ratchet wheel.



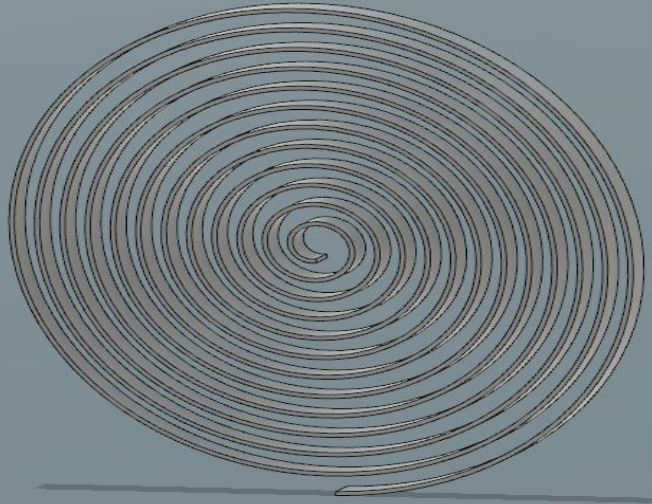
Magic Lever mechanism



PHYSICS BEHIND THE MAGIC LEVER

- Rotating mass is connected to pinion of gear set 1 of gear ratio (m_{G1}) = 2 through the same shaft
- Gear set 1 \rightarrow 10T, 20T with $m = 0.4\text{mm}$
- Lever is rigidly pivoted to Gear2 with an eccentricity of 2mm between their centres.
- Levers has 2 arm of length $\sim 15\text{mm}$ designed in such a way that only one arm meshes with ratchet wheel in a particular rotation direction
- Lever can be assumed to apply force tangentially on the wheel $F_t = T_{\text{mass}} * (q/e) * m_{G1}$
- Ratchet wheel is connected to Gear3 through the same shaft. Gear3(10T) meshes with Gear4(50T) to form gear set2($m_{G2}=5$, $m=0.4\text{mm}$)

Spiral Torsional Spring



ABSTRACT

- Main spring is enclosed in the barrel which acts as the power reserve.
- Gear4 (50) is connected to arbor of main spring through a central shaft, while the other end is connected of spring is connected to barrel that is rigidly fixed to Gear5(90T) for transfer of energy to wheels of the watch
- Torsional springs used in watches are spiral and called hairsprings.
- Material : High carbon Spring Steel,
ASTM A - 227 Hard Drawn $S_{ut} = 1015 \text{ MPa}$

Design Calculations

The formula for torque delivered by a spiral torsion spring is:

$$(1) \quad M = \frac{\pi E b t^3 \theta}{6L} \text{ lb.}\cdot\text{in (N}\cdot\text{mm)}$$

where

E = Modulus of elasticity, psi (MPa)

θ = Angular deflection in revolutions

L = Length of active material, in. (mm)

M = Moment or torque, lb.·in (N·mm)

b = Material width, in. (mm)

t = Material thickness, in. (mm)

- For spring steel $E = 207 \text{ GPa}$
- $\theta = 2$ (revolutions) for maximum winding (assumed)
- $b = 4\text{mm}$ $t = 0.5\text{mm}$ (assumed)
- $M = T_{\text{TS}} = T_{\text{mass}} * (q/e) * m_{\text{G1}} * m_{\text{G2}}$
- $T_{\text{mass}} = 1.348 * 10^{-3} \text{ Nm}$ $(q/e) = 8$ $m_{\text{G1}} = 2$ $m_{\text{G2}} = 5$
- $T_{\text{TS}} = 0.108 \text{ Nm}$
- Active length $L = 753\text{mm}$.
- Assuming $N = 12$ full revolutions.
- $R_{\text{avg}} = L / (2\pi * N) \sim 10\text{mm}$.

Design Calculations

- Stress imposed in spiral torsion springs are due to bending,deflecting beam formula can be used.
- $S = 862 \text{ MPa}$ (calculated)
- $(\text{FoS})_{\text{bending}} = (S_{\text{ut}}/S) = 1.17$
- For long lasting springs cold drawn spring steel can be used,thus increasing the $(\text{FoS})_{\text{bending}}$
- Cold drawing the material can improve the strength of the material

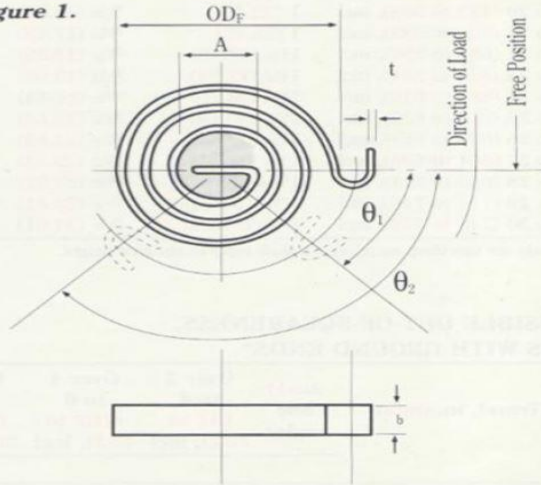
$$S = \frac{6M}{bt^2} \text{ psi (MPa)}$$

Calculations

The following formula, based on concentric circles with a uniform space between the coils, gives a close approximation of the minimum OD_F:

$$OD_F = \frac{2L}{\pi \left(\frac{\sqrt{A^2 + 1.27Lt} - A}{2t} - \theta \right)} - A \text{ in. (mm)}$$

Figure 1.



For an arbor diameter (A) = 3mm (assumed)

- OD_F ~ 25mm (calculated)
- Spring when wound by 2 revolutions will decrease in diameter.
- Barrel (casing) of 20mm ID with a thickness of 0.5mm is designed to enclose the spring.
- Outer free end of Hair Spring will be rigidly fixed to casing and the casing is rigidly fixed to Gear5(90T) while arbor is fixed to shaft that holds Gear4(50T)

Design of Spur Gear set 2 (Gear 3-4)

Assumed Data:

- Module (m) = 0.4mm.
- Pressure Angle $\phi = 20^{\circ}$
- Gear ratio $m_G = 5$
- The gear is Uncrowned and the Material is AISI 304 Stainless Steel (Hardened). The Brinell Hardness for both gear and pinion is 123 HB.

Design of Spur Gear set 2 (Gear 3-4)

Based on eccentric mass specifications and specified assumptions:

- Angular velocity of pinion is $\omega = 15$ degrees/sec.
- Undercutting is essential to avoid Interference in Gears.
- Assuming $Z_p = 10$, therefore $Z_G = 50$.
- Diameter of pinion $d_p = m_G * Z_p = 4\text{mm}$.
- Diameter of gear $d_G = m_G * Z_G = 20\text{mm}$.
- $F^t = 10.8$ N.
- Pitch line velocity, $V = r_p * \omega_p = 5.23 * 10^{-4}$ m/s.
- From table, Lewis form factor $Y_G = 0.40$.

Design of Spur Gear set 2 (Gear 3-4)

CORRECTION FACTORS

It is now required to calculate the correction factors in the AGMA stress and strength equations.

- **Overload Factor** $(K_o)_p = 1 = (K_o)_G$ (Assuming uniform conditions).
- **Dynamic Factor** $(K_v)_p = 1.001$ (for $V=5.23 \cdot 10^{-4}$ m/s and $Q_v=10$).
- **Size factor** $K_s = 1$ for both pinion and gear.
- **Rim thickness factor** $K_B = 1$ for both.
- **Surface geometry factor**, $I = 0.1333$ for both.
- **Reliability factor**, $K_R = 1.001$ for both.
- Assuming Hardness are same for both pinion and gear, **Hardness ratio factor** $C_H = 1$.
- Assuming $E_p = E_G = 200$ GPa and $\nu_p = 0.3$ for both, we get **Elastic coefficient** $C_p = 187.02$ $(\text{N/mm}^2)^{1/2}$

Design of Spur Gear set 2 (Gear 3-4)

CORRECTION FACTORS CONTD.

- **Surface condition factor (C_f)** = 1.5.
- Considering 10^7 cycles for pinion, **Stress cycle factors** for Bending are $(Y_N)_P = 1.0176$ and $(Y_N)_G = 1.047$.
- **Stress cycle factors** for Pitting are $(Z_N)_P = 1$, $(Z_N)_G = 1.03$.
- **Bending strength** for both pinion and gear is $S_t = 199.47$ MPa.
- **Pitting strength** for both pinion and gear is $S_c = 533.43$ MPa.
- For Factor of Safety calculation we are assuming face width of pinion to be 2mm and for gear to be 1mm.

Design of Spur Gear set 2 (Gear 3-4)

FACTOR OF SAFETY CALCULATION

- **Load Distribution Factor, K_H** for pinion
 $C_{mc}=1, C_{pm}=1.1, C_e=1, C_{ma}=4.402 * 10^{-3}, C_{pf}=0.025$
Therefore $K_H = 1.031902$.
- **Load Distribution Factor, K_H** for Gear
 $C_{mc}=1, C_{pm}=1.1, C_e=1, C_{ma}=4.001 * 10^{-3}, C_{pf}=0$
Therefore $K_H = 1.031902$.
- So the Factors of Safety for Bending and Pitting for pinion and Gear are given below after performing the calculation.

$$(S_F)_P = 5.531$$

$$(S_F)_G = 2.924$$

$$(S_H)_P^2 = 1.3386$$

$$(S_H)_G^2 = 3.3514$$

Design of Spur Gear set 3 (Gear 5-6)

Assumed Data:

- Module (m) = 0.25mm.
- Pressure Angle $\phi = 20^\circ$
- Gear ratio $m_G = 7.5$
- The gear is Uncrowned and the Material is AISI 304 Stainless Steel (Hardened). The Brinell Hardness for both gear and pinion is 123 HB.

Design of Spur Gear set 3 (Gear 5-6)

Based on eccentric mass specifications and specified assumptions:

- Angular velocity of gear is $\omega = 1.745 \cdot 10^{-3}$ radians/sec.
- To avoid interference, the minimum number of teeth in pinion will be around $N_p = 12$.
- Assuming $N_p = 12$ then $N_G = 90$.
- Undercutting will be done to avoid Interference in Gears.
- Diameter of pinion $d_p = m_G \cdot Z_p = 3\text{mm}$.
- Diameter of gear $d_G = m_G \cdot Z_G = 22.5\text{mm}$.
- $F_t = 9.6\text{N}$
- Pitch line velocity, $V = r_p \cdot \omega_p = 1.96 \cdot 10^{-5} \text{ m/s}$.
- From data sheets, Lewis form factor $Y_G = 0.43$ and $Y_p = 0.24$

Design of Spur Gear set 3 (Gear 5-6)

CORRECTION FACTORS

It is now required to calculate the correction factors in the AGMA stress and strength equations.

- **Overload Factor** $(K_o)_p = 1 = (K_o)_G$ (Assuming uniform conditions).
- **Dynamic Factor** $(K_v)_p = 1.002 = (K_v)_G$ (for $V = 1.96 \times 10^{-5}$ m/s and $Q_v = 10$).
- **Size factor** $K_s = 1$ for both pinion and gear.
- **Rim thickness factor** $K_B = 1$ for both.
- **Surface geometry factor**, $I = 0.1333$ for both.
- **Reliability factor**, $K_R = 1.001$ for both.
- Assuming Hardness are same for both pinion and gear, **Hardness ratio factor** $C_H = 1$.
- Assuming $E_p = E_G = 200$ GPa and $\nu_p = 0.3$ for both, we get **Elastic coefficient** $C_p = 187.02 \text{ (N/mm}^2\text{)}^{1/2}$

Design of Spur Gear set 3 (Gear 5-6)

CORRECTION FACTORS CONTD.

- **Surface condition factor (C_f)** = 1.5.
- Considering 10^7 cycles for pinion, **Stress cycle factors** for Bending are $(Y_N)_P = 1.0176$ and $(Y_N)_G = 1.0548$.
- **Stress cycle factors** for Pitting are $(Z_N)_P = 1$, $(Z_N)_G = 1.0474$.
- **Bending strength** for both pinion and gear is $S_t = 199.47$ MPa.
- **Pitting strength** for both pinion and gear is $S_c = 533.43$ MPa.
- For Factor of Safety calculation we are assuming face width of pinion to be 2mm and for gear to be 1mm.

Design of Spur Gear set 3 (Gear 5-6)

FACTOR OF SAFETY CALCULATION

- **Load Distribution Factor, K_H for pinion**
 $C_{mc}=1, C_{pm}=1.1, C_e=1, C_{ma}=4.4026 * 10^{-3}, C_{pf}=0.025$
Therefore $K_H = 1.0319$.
- **Load Distribution Factor, K_H for Gear**
 $C_{mc}=1, C_{pm}=1.1, C_e=1, C_{ma}=4.001 * 10^{-3}, C_{pf}=0$
Therefore $K_H = 1.004$.
- So the **Factors of Safety for Bending and Pitting** for pinion and Gear are given below after performing the calculation.
 $(S_F)_p = 2.147$ $(S_F)_G = 2.232$
 $(S_H)_P^2 = 0.9832$ $(S_H)_G^2 = 4.1575$
- Since the Factor of Safety for pitting coming out to be less than 1, we move to another material for pinion.

Design of Spur Gear set 3 (Gear 5-6)

FACTOR OF SAFETY CALCULATION CONTD.

- So the material that we are upgrading for pinion is AISI 316 Steel (Heat treated), grade 2 which is resistance to pitting and corrosion having a brinell hardness of 217 HB.
- So going forward with this material we get the FOS to be

$$(S_F)_p = 2.858$$

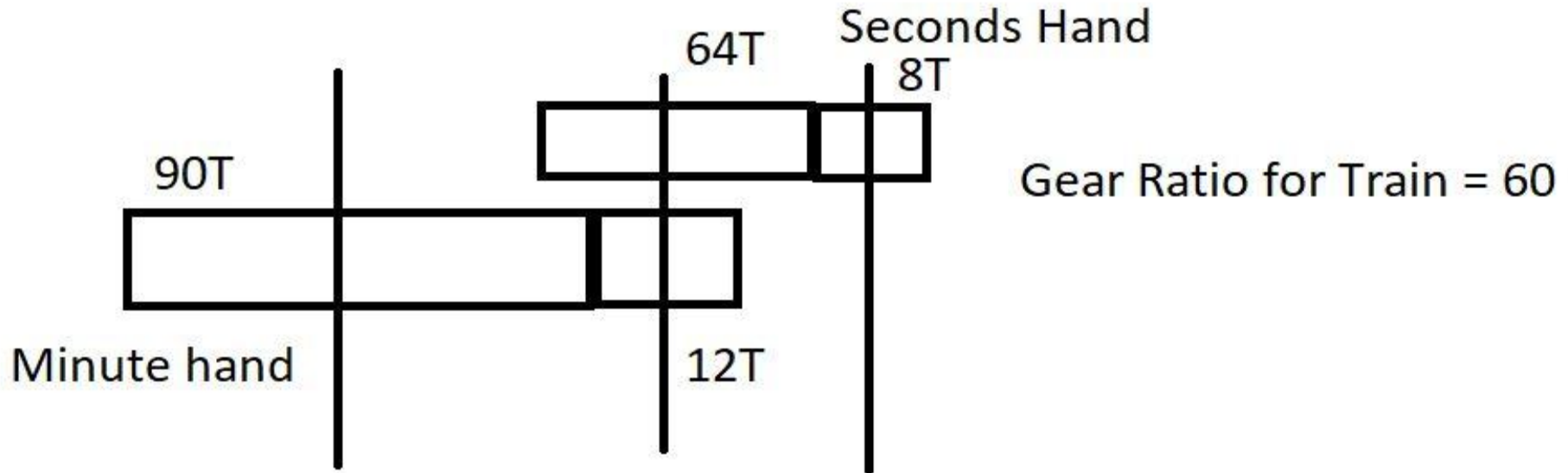
$$(S_F)_G = 2.232$$

$$(S_H)_p^2 = 1.9937$$

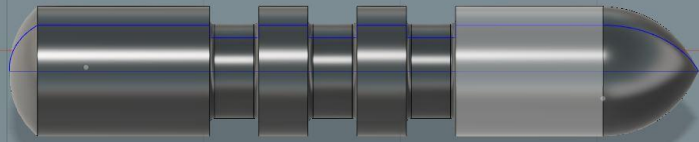
$$(S_H)_G^2 = 4.1575$$

Gear Teeth Calculation

Minute and Seconds hand are connected to gears with 90T and 8T respectively



Shaft Design



Assembly :

- The design of the shaft has been done for torsion and bending. The shaft experiences Torsion which is alternating which a constant bending moment can be considered due to low rotational speed.
- This Shaft supports Gear 4 with 50T that is coupled to Magic lever mechanism (Winding side) and Gear 6 (90 T) rigidly fixed to casing that contains the main spring. Thus experiencing two radial forces from gears and an alternating torsion from mainspring.

Assumptions :

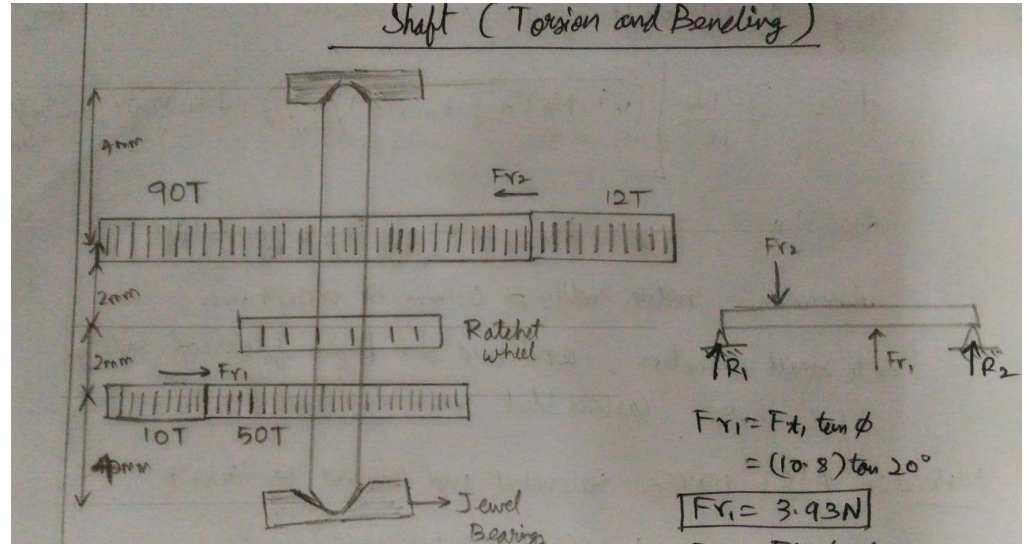
- Alternating Bending Moment (M_a) = 0
- Alternating Torsion (T_a) = 0.03Nm
- Max Torsion (T_{max}) = 0.1 Nm
- Notch Radius = 0.1mm for Step
- $K_t = K_{ts} = 3$ (Step radii)

Shaft Design- Schematic Diagram

Material :

AISI 1095 Quenched and Tempered @ 600F has been chosen keeping in mind the need for High factor of Safety and Small diameters. Carbon Steel with High S_{ut} was chosen.

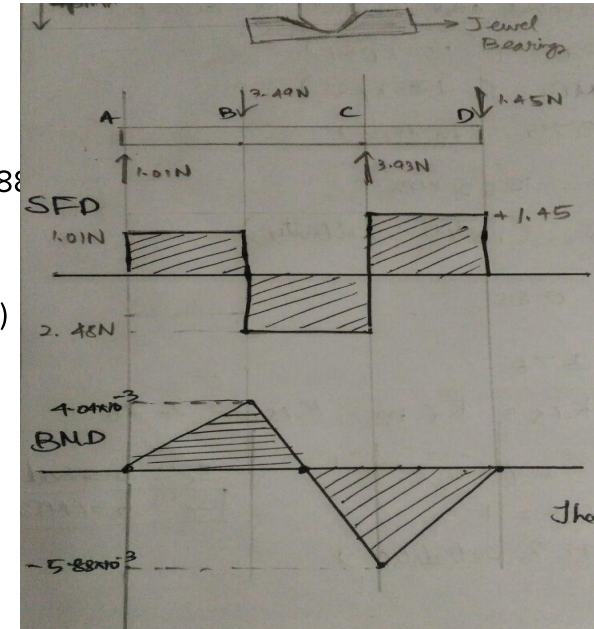
$$S_y = 814 \text{ MPa} \quad S_{ut} = 1262 \text{ MPa}$$



Shaft Design- SFD & BMD

Calculations

- For Radial forces $F_{r1} = 3.93\text{N}$
- $F_{r2} = 3.49\text{N}$
- Mean Bending Moment (M_m) = 5.88Nm
- Mean Torsion (T_m) = 0.08Nm
- $C_{\text{load}} = 1$ $C_{\text{temp}} = 1$ $C_{\text{size}} = 1$
- $C_{\text{reliab}} = 0.659$ (99.999% reliability)
- $C_{\text{surf}} = 0.9$
- $S_e = S_f = 374.25\text{MPa}$



- For notch radius $r = 0.1\text{mm}$ $S_{\text{ut}} = 1262\text{MPa}$ ---> notch sensitivity(q) = 0.88
 $K_f = K_{fs} = K_{fsm} = K_{fm} = 2.76$
- Fatigue Factor of Safety N_f is computed from ASME shaft equation
- For a $d = 1\text{mm}$, $N_f < 1$
- Therefore a $d = 2\text{mm}$ was chosen for which the $N_f = 4.682$ (OK)

JEWEL BEARING - PROPERTIES

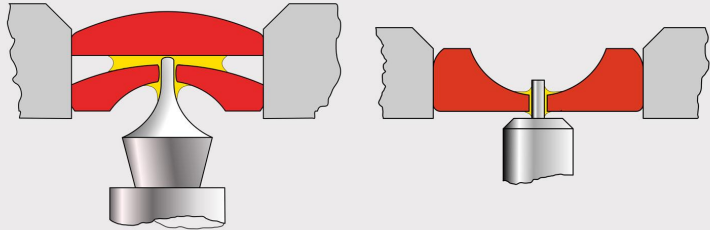


Historically Jewels used in the mechanical watches are **rubies** or **sapphires** or occasionally diamonds.

Jewels have two properties that are useful for watchmakers –

- They are **hard** and therefore **wear very slowly**,
- Secondly they can be worked to a **very smooth finish**.
These properties relate directly to their function in watchmaking – reducing friction.
- There are a lot of moving parts inside a mechanical watch and the key to an accurate and efficient movement is to **minimise friction**.
- In order to achieve this, the tolerances on wheels and gears are extremely tight, ensuring that the teeth interlock as smoothly as possible.

JEWEL BEARING



- The **doughnut shaped jewels** have their surfaces shaped and finished in such a way that the oil that is used on all of these moving parts is held where it is needed rather than spreading across the jewel surface.
- There are also **cap jewels** which affix to the end of shafts but do not have holes drilled right through them. These are often used in conjunction with hole jewels and prevent movement of the axle up and down.
- There are also **pallet jewels** – these are roughly rectangular jewels fit on the pallet fork (one on each side) and are the part that actually engages with the escape wheel to control the rate of the movement.

JEWEL BEARING - SHOCK ABSORPTION

-
- Finally there is a **roller jewel (impulse pin)** that is on the balance wheel and engages with the other end of the pallet fork to rock the fork back and forth.
 - Jewels also indirectly play a part in **shock absorption**. The most common shock system in use today is the **Incabloc system** invented in the 1930s which mounts the balance cap jewels in springs to absorb any shocks that occur and prevent damage to the movement.



JEWEL BEARING

So let's look at jewel counts for simple watches and what the jewels are:

- **7 jewels** (common in older pocket watches, but unheard of now) – this is simply a jewelled escapement – two hole jewels and two cap jewels (one each for each end of the balance wheel shaft), two pallet jewels and a roller jewel.
- **15 jewels** – the 7 from above plus hole jewels for the high speed elements of the running train (it's not needed on minute or hour wheels).
- **17 jewels** – about the minimum in a decent watch today add hole jewels to each end of the centre wheel
- **19 jewels** – adds cap jewels to the escape wheel
- **21 jewels** – cap wheels to the pallet fork
- **23 jewels** – mainspring barrel jewels (more common in pocket watches than wristwatches)

THANK YOU