# OPTIMISATION OF GEAR GEOMETRICAL PARAMETERS USING KISSSOFT

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***Abstract:***

*In this study, optimisation of speed gears for a tractor transmission was performed with KISSsoft software. Optimisation was carried out under three constraints. These constraints are input power-torque, volume for system in transmission and gear ratio for each speed. The purpose of this study was to optimize the module, face width, gear quality, centre distance, number of teeth, helix angle, addendum modification coefficient and pressure angle for each speed considering the constraints. Tooth bending stress, tooth contact stress, contact ratio and specific sliding were considered for evaluation during optimisation. Strength calculation of gear pairs which were optimized and defined all geometrical parameters with KISSsoft were also calculated with mathematical model indicated in ISO 6336. Then, the results were compared.*

*KEYWORDS: GEARS, GEOMETRICAL PARAMETERS, OPTIMISATION, KISSSOFT*

## *1. Introduction*

Gears are used for many mechanical system in particular to automotive at the present time. Gears can be designed more reliable, lighter, quieter with optimisation studies. Also, gears can be more competitive in terms of cost during optimisation. In this study, optimisation of speed gears for a tractor transmission was performed with KISSsoft software. Input power-torque, volume for system in transmission and gear ratios for each speed were considered during optimisation. Module, facewidth, gear quality, centre distance, number of teeth, helix angle, addendum modification coefficient and pressure angle of eight quantity of gears for four speed were defined considering tooth bending stress, tooth contact stress, contact ratio and specific sliding during optimisation. Strength calculation of gear pairs which were optimized and defined all geometrical parameters with KISSsoft were also calculated with mathematical model indicated in ISO 6336. Then, the results were compared.

## *2. Calculating the load capacity of helical gears*

Gears face the tooth bending stress and the tooth contact stress during power-torque transfers. Therefore, some damages can occur on gears. The damages which can arise from stress on gears should be considered during design phase.

### *2.1. Tooth bending stress*

Distribution of forces on gears are shown in Fig.1. The tooth bending stress according to ISO 6336 standard is calculated as the following

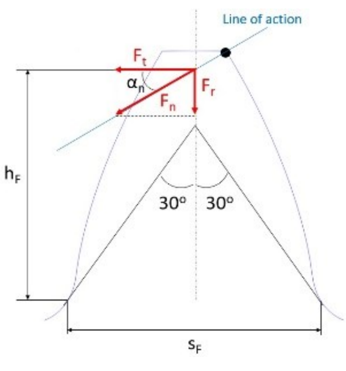


Fig.1 Tooth bending stress

The real tooth-root stress, $\sigma_F$ is calculated as the following[1-6]:

$$\sigma_F = \frac{F_t}{bm_n} Y_F Y_S Y_\varepsilon Y_\beta K_A K_v K_{F\beta} K_{F\alpha} \quad (1)$$

where $F_t$ is the nominal tangential load [N], $b$ is the facewidth [mm], $m_n$ is the normal module [mm], $Y_F$ is the form factor [-], $Y_S$ is the stress correction factor [-], $Y_\varepsilon$ is the contact ratio factor [-], $Y_\beta$ is the helix angle factor [-], $K_A$ is the application factor [-], $K_V$ is the dynamic factor [-], $K_{F\beta}$ is the face load factor [-], $K_{F\alpha}$ is the transverse load factor [-].

The permissible bending stress, $\sigma_{FP}$ is calculated as the following:

$$\sigma_{FP} = \sigma_{Flim} Y_{ST} Y_N Y_\delta Y_R Y_X \quad (2)$$

where $\sigma_{Flim}$ is the nominal stress [N/mm$^2$], $Y_{ST}$ is the stress correction factor [-], $Y_N$ is the life factor [-], $Y_\delta$ is the relative notch sensitivity factor [-], $Y_R$ is the relative surface factor [-], $Y_X$ is the size factor [-].

The safety factor for bending stress, $S_F$ is calculated as the following:

$$S_F = \frac{\sigma_{FP}}{\sigma_F} \quad (3)$$

### *2.2. Tooth contact stress*

Surface pressure which occurs on gears is calculated according to ISO 6336 standard as the following [1-6]:

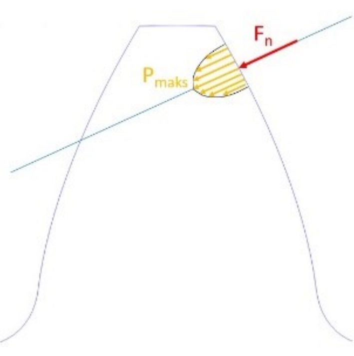


Fig.2 Tooth contact stress

The real contact stress, $\sigma_H$ is calculated as the following:

$$\sigma_H = \sqrt{\frac{F_t}{bm_n}\frac{u+1}{u}} Z_H Z_E Z_\varepsilon Z_\beta \sqrt{K_A K_V K_{H\beta} K_{H\alpha}} \quad (4)$$

where u is gear ratio [-], $Z_H$ is the zone factor [-], $Z_E$ is the elasticity factor [$\sqrt{N/mm^2}$], $Z_\varepsilon$ is the contact ratio factor [-], $Z_\beta$ is the helix angle factor [-], $K_{H\beta}$ is the face load factor, $K_{H\alpha}$ is the transverse load factor [-].

The permissible contact stress, $\sigma_{HP}$ is calculated as the following:

$$\sigma_{Hp} = \sigma_{Hlim} Z_N Z_L Z_V Z_R Z_W Z_X \quad (5)$$

where $\sigma_{Hlim}$ is the allowable stress [N/mm²], $Z_N$ is the life factor [-], $Z_L$ is the lubrication factor [-], $Z_V$ is the velocity factor [-], $Z_R$ is the roughness factor [-], $Z_W$ is the work hardening factor [-], $Z_X$ is the size factor [-].

The safety factor for contact stress, $S_H$ is calculated as the following:

$$S_H = \frac{\sigma_{Hp}}{\sigma_H} \quad (6)$$

## 3. Optimisation with KISSsoft

In this scope of work, four speed gears of a tractor transmission were optimized via KISSsoft software. Input power is 50 kW and torque is 238 Nm for speed gears which is optimized in this study. These four speed gears have ratios like in Table 1 with tolerances of %4.

Table 1 Ratios

| Speed | Ratio |
|---|---|
| 1 | 3,1 |
| 2 | 1,9 |
| 3 | 1,1 |
| 4 | 0,7 |

Maximum volume which can be used in transmission for these speed gear group as Fig.3.

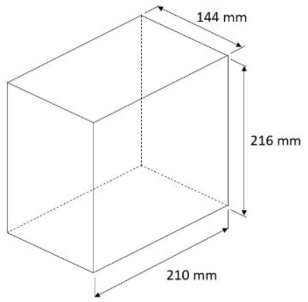


Fig.3 Volume

Firstly face width, quality and module range were defined so that optimum gear pairs can be determined for volume constraint. In beginning, other gear geometrical parameters were accepted as constant. Then, other gear parameters were optimized. Material of gears was preferred as 16MnCr5. Distance for synchroniser ($s_1$ and $s_3$), assembly distance ($s_2$) and volume constraint were considered for determining of gear face width.

$$b_1 + s_1 + b_2 + s_2 + b_3 + s_3 + b_4 = 210\ mm \quad (7)$$

Then, all possible gear pairs according to ratios were calculated with KISSsoft considering input values and constraints.

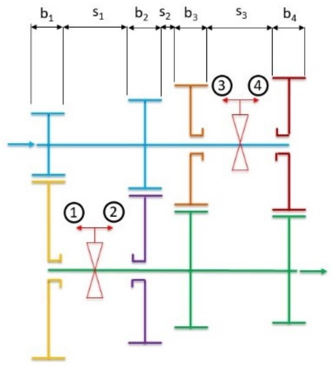


Fig.4 Speed Gears

Optimal face width, quality and module range for gear pairs were determined via graphics like Fig.5 which contain the results of all gear pairs calculated by KISSsoft according to module, minimum root safety and minimum flank safety .

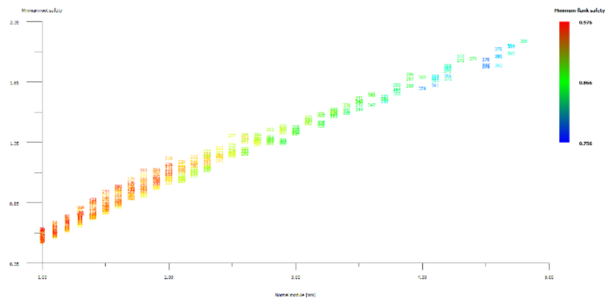

Fig.5 Module, $S_F$, $S_H$

Then, proper centre distance was specified for speed gears group. After determining of face width, quality and module range of gears, all results of gear pairs which were calculated by KISSsoft for different centre distance values were placed in graphics like Fig.6 according to tip diameters of gear pairs and centre distance in order to specify the optimal centre distance.

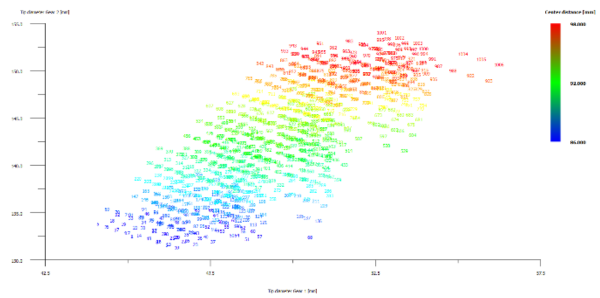

Fig.6 Tip diameter of gear pairs, centre distance values

After determining of centre distance from in graphics like Fig.6 and module from module range which was defined in graphics like Fig.5, then optimisation of number of teeth of gear pairs was applied. Proper number of teeth of gear pairs were determined considering maximum specific sliding and contact ratio of gear pairs.

After determining of face width, quality, module, centre distance and number of teeth, optimisation of helix angle was applied. Helix gears create additional axial forces in system according to spur gears. Therefore, contact ratio and axial forces were considered during determining of helix angle.

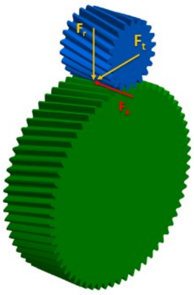

Fig.7 Forces on gear

Finally, addendum modification coefficient and pressure angle of gears were specified via KISSsoft. Addendum modification coefficient and pressure angle have effect on gear profile so contact ratio, specific sliding and safety factors were considered during optimisation.

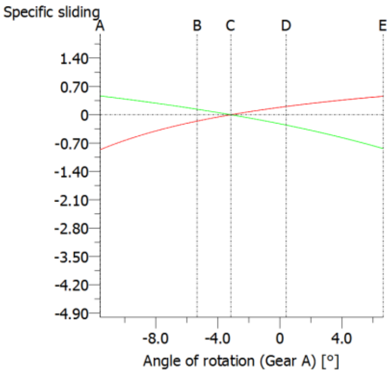


Fig.8 Specific sliding

Optimisation results of face width, quality, module, centre distance, number of teeth, helix angle, addendum modification coefficient and pressure angle for four speed gears groups are like Table 2 below.

Table 2 Optimisation results

| | 1.Speed | 2.Speed | 3.Speed | 4.Speed |
|---|---|---|---|---|
| Number of teeth $z_1$ | 20 | 30 | 40 | 57 |
| Number of teeth $z_2$ | 64 | 55 | 45 | 41 |
| Module [mm] | 2 | 2 | 2 | 1,75 |
| Pressure angle [°] | 20° | 20° | 20° | 20° |
| Helix angle [°] | 13° | 15° | 17° | 15° |
| Addendum modification coefficient $x_1$ | 0,6 | 0,3 | 0,1 | 0 |
| Addendum modification coefficient $x_2$ | 0,9419 | 0,2201 | -0,0416 | 0,1297 |
| Facewidth [mm] | 40 | 25 | 20 | 20 |
| Quality | 6 | 7 | 8 | 8 |

Optimal four speed gears group which were defined all geometrical parameters like Table 2 was placed in volume as Fig.9. According to Fig.9, ıt seems that there is no any problem about volume constraint. All gears can be assembled properly in volume.

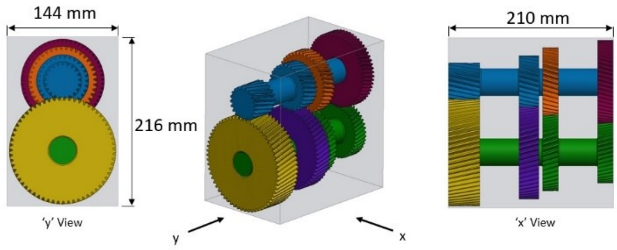


Fig.9 Speed gears in volume

### *4. Results and discussion*

Tooth bending stress, tooth contact stress and safety factor of pinion gears for each speed which were optimized via KISSsoft were also calculated according to mathematical model in ISO 6336. The results of pinion gears for both KISSsoft and mathematical model are like Table 3 below.

Table 3 Results

| | 1.Pinion | 2.Pinion | 3.Pinion | 4.Pinion |
|---|---|---|---|---|
| Tooth-root stress $\sigma_F$ from KISSsoft [N/mm$^2$] | 543,96 | 569,46 | 576,47 | 570,03 |
| Tooth-root stress $\sigma_F$ from mat. Model [N/mm$^2$] | 546,97 | 607,52 | 625,93 | 561,37 |
| Safety factor for bending stress $S_F$ from KISSsoft | 1,41 | 1,34 | 1,32 | 1,34 |
| Safety factor for bending stress $S_F$ from mat. Model | 1,46 | 1,32 | 1,28 | 1,43 |
| Contact stress $\sigma_H$ from KISSsoft [N/mm$^2$] | 1363,60 | 1286,05 | 1292,8 | 1197,23 |
| Contact stress $\sigma_H$ from mat. Model [N/mm$^2$] | 1345,09 | 1264,66 | 1239,2 | 1226,15 |
| Safety factor for contact stress $S_H$ from KISSsoft | 1,10 | 1,13 | 1,13 | 1,22 |
| Safety factor for contact stress $S_H$ from mat. model | 1,12 | 1,19 | 1,21 | 1,22 |

Regarding to results in Table 3, tooth root stress of pinion 1 (546,97 N/mm$^2$) according to mathematical model is bigger (%0,5) than KISSsoft result (543,96 N/mm$^2$). For root safety factor, results of mathematical model (1,46) is bigger (%3,5) than the results of KISSsoft (1,41). Tooth contact stress of pinion 1(1345,09 N/mm$^2$) according to mathematical model is smaller (%1,4) than the results of KISSsoft (1363,60 N/mm$^2$). For flank safety factor, mathematical model result(1,12) is bigger (%1,8) than KISSsoft result (1,1).

Regarding to results for pinion 2, mathematical model result is bigger (%6,7) than KISSsoft result for tooth root stress. Tooth contact stress according to mathematical model is smaller (%1,7) than KISSsoft result. For root safety factor, mathematical model result is smaller (%1,5) than KISSsoft result and flank safety factor of mathematical model is bigger (%5,3) than KISSsoft result.

Regarding to results for pinion 3, tooth root stress according to mathematical model is bigger (%8,5) and tooth contact stress according to mathematical model is smaller (%4,1). For root safety factor, result of mathematical model is smaller (%3) and flank safety factor of KISSsoft result is smaller (%7) than mathematical model.

Regarding to results for pinion 4, tooth root stress according to mathematical model is smaller (%1,5) and tooth contact stress is bigger (%2,4) than KISSsoft result. The root safety factor of mathematical model is bigger (%6,7) than KISSsoft result. For flank safety factor, results are same.

According to results, ıt seems that there are maximum %8 differences between KISSsoft and mathematical results. There are some reasons for these differences. KISSsoft considers the tolerances and deformation of gears during calculation. Besides, KISSsoft specifies some correction coefficients according to own experiences. Although there are some differences between results, ıt seems that safety factors for both method are suitable according to target values. Therefore, KISSsoft can be used reliably for strength calculation of gears.

## 5. Conclusion

Four speed gears of a tractor transmission were optimized by KISSsoft software. During optimisation, input power-torque, ratios and maximum volume were considered as constraint. Facewidth, centre distance, module, quality, number of teeth, helix angle, addendum modification coefficient and pressure angle of gear pairs were specified with optimisation. Then, tooth root stress, tooth contact stress and safety factors were also calculated according to mathematical model in ISO 6336. After calculations, the results were compared. Regarding to tooth root stresses, maximum differences is %8,5 for pinion 3. Regarding to tooth contact stresses, maximum differences is %4,1 for pinion 3. Regarding to root safety factors, maximum differences is %6,7 for pinion 4. Regarding to flank safety factors, maximum differences is %7 for pinion 3. According to this study, both KISSsoft results and mathematical model results are within the range of target value.

Also, the results below were determined during optimisation:

i. Increasing the module has a positive effectiveness on root safety factor and negative effectiveness on flank safety factor.
ii. Increasing the face width of gears has a positive effectiveness on flank safety factor.
iii. Increasing the centre distance results decreased tooth contact stress.
iv. Increasing the number of teeth results increased contact ratio.
v. Increasing the helix angle results increased both contact ratio and axial forces.
vi. Increasing the addendum modification coefficient has a positive effectiveness on root safety factor.

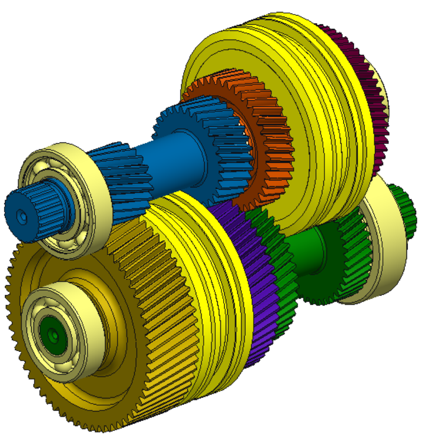

Fig.10 Concept design


### Acknowledgements

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## 6. Literature


[1] ISO 6336-1: calculation of load capacity os spur and helical gears, part: basic principle, introduction and general influence factors.

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[3] ISO 6336-3: calculation of load capacity of spur and helical gears, part 3: calculation of tooth bending strength.

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