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Araştırma Makalesi / Research Article

Design and Analysis of an Automotive Differential by Changing the Final Reduction Ratio

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Abstract

Differential gear system is called the system used due to the different amounts of distance traveled by the wheels of the vehicles while cornering. Differential is designed according to the road handling performance and some features of the vehicles. Differential types differ due to needs. In fact, they all have the same function. In the differential gear system, hypoid bevel gears are preferred instead of spur gears. The differential mechanism is a gear system that transmits the movement of the drive shaft to the wheel, reduces its speed and increases its torque, and ensures the ease of the first movement in the vehicle. In the hypoid gear type, since the gear axes are perpendicular to each other, the center of gravity of the vehicle is moved closer to the ground, improving the handling of the vehicle. The use of any of the bevel gear groups of the ring gear and pinion gear is preferred because it reduces vibration and noise caused by friction. Commonly available automobile differentials have a maximum reduction ratio of 6. This is because designing an automotive differential with a reduction ratio greater than 6 results in a bulky design that is impossible to position with the limited space available. Also, increasing the size of the differential can lead to excessive unwanted weight. Most land vehicles have differentials with reductions of 3 or 4. Commercially speaking, it is almost impossible to find a differential with a reduction greater than 6. Most manufacturers introduce an additional single-speed gearbox, but this would complicate the design and increase service costs. The aim of this study is to design and analyze a differential with different reduction ratios. The article includes a force-based analysis performed in Adams and Ansys Simulation, as well as all calculations to prove the success of the design. In this study, changes were made in the cycle ratio by keeping the ring gear constant and taking into account the changes in the number of pinion gears. As a result of mathematical calculations and analysis, a decrease was observed in the normal module value as the cycle ratio increased. A quieter working environment was provided by using hypoid gears instead of spur gears in the design. As the distance between the axes increased, the mirror helix angle decreased and the efficiency decreased. As a result of the analyses, it was possible to intervene earlier in problems that may be observed during and after the production process.

Keywords: Differential gearbox, gear, speed-torque, final reduction ratio

1. Introduction

When the vehicle rounds the bend, all wheels tend to rotate at different speeds. When rounding, the wheels on the inside of the bend take a shorter distance than the wheels on the outside. Wheels that travel shorter distances have slower speeds (Boos and Koepf, 1992). This means that in a bend, for example, if you want to turn right, the rear wheel on the right should turn less and the rear wheel on the left should turn more (compared to a rear-wheel drive). If the two wheels rotate equally, there will be a strain on the rotation of the vehicle and the axles will break. The differential is a mechanical gear system that senses this necessity and transmits power according to the speed balance between the two wheels on one axle (Leske and Schaffler, 1994). The differential transmits engine power to the wheels. The rotational movement of the engine is adjusted according to the speed in the transmission and transferred to the differential. In rear-wheel drive vehicles, the differential movement is transferred to the wheels by rotating them 90 degrees (Matsuna et al., 2000).

The differential gear box ensures that the wheel on the inside rotates slower than the wheel on the outside when the vehicle is rounding the bend. This makes rounding safer (Wang et al., 2019). During rotation, the wheels on the inside and outside rotate around a certain circle axis. As shown in Figure 1, the wheel on the outside on the rotation axis will rotate on the circle axis with a larger diameter than the wheel on the inside. If there were no differential, the wheels on the inside and outside would rotate at the same speed and drag would occur on the wheels (Kryger, 2007; Erjavec and Ken, 2015).

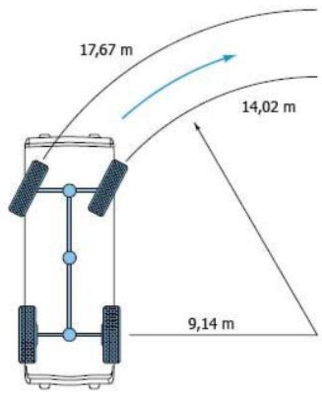

Figure 1. The condition of the vehicle's wheels when rounding the bend (Kryger, 2007).

1.1. Differential working principle

While the vehicle is traveling straight, the ring gear, which receives the movement from the gear ring, rotates the differential gear box. Along with the box, the axle and spider gears inside also rotate. Gears cannot rotate around their own axes (Demir and Başeğmez, 2022).

When the vehicle is rounding a bend, the wheel on the outside of the vehicle must turn faster than the wheel on the inside. Because the wheels on the outside have to cover a longer distance (Heinz, 1999). During a bend, the wheel on the outside rotates faster due to the path it will take, which causes the wheel axle on the outside to rotate faster (Heinz, 2002).

From the moment the vehicle starts to round a bend, the load on the wheels on the inside and the fact that the wheels on the outside have to cover a greater distance than the other wheels cause the internal gear axle speed to be reduced. For example, let's imagine that the car's wheel on the inside stops on a sharp bend. When the wheel in the bend stops, the axle gear will also remain stationary (Ordu, 2018; Dündar, 2022). When the axle gears are fixed, the spider gears, which perform the rotation with the ring, rotate around their axes by rolling on the fixed axle gear. Thanks to this working system, the spider gears rotate on their own axis together with the rotating box around the ring and rotate around the fixed axle gear (Hancock et al., 2005). By rotating the spider gears around their own axes, the axle gear on the wheel on the outside is enabled to rotate faster than the spider gear. Because the axle gear on the wheel on the outside performs the rotation around its axis due to the rotation performed by the differential gears and the spider gears.

1.2. Differential and traction power

There are two factors that determine the value of torque that can be applied to the wheels. One of them is defined as equipment and the other as traction power (Çetinkya, 2023). Traction power is high on smooth roads. The torque applied to the wheels can be limited by the gearbox and engine. When traction is weak on slippery roads, the torque produced is limited to the maximum extent to prevent the wheels from slipping on slippery surfaces.

Increasing the speed of the vehicle during wheel slippage in a bend causes the wheel to slip more quickly and at a higher speed. The highest torque value is limited to the highest value that will not cause the wheel to slip in its current position. However, if the wheels with high traction reach a very low amount of torque that can be applied to the wheel with low traction on slippery road, the vehicle has difficulty in moving. When any of the front or rear wheels lose contact with the ground, the wheels will try to rotate in the air and the motion transmission will be interrupted. To solve such problems, LSD (limited slip differential) and locked differential systems have been developed (Bahçekapılı, 2008).

1.3. Final reduction ratio

The reduction ratio of the gears is the ratio of the torque transmitted to the shaft (FR vehicles rear-wheel drive) or to the pinion gear driving the output shaft (FF vehicles front-wheel drive) to the engine torque and vehicle speed, which can vary depending on the transmission. In other words, the reduction ratio of the differential (final gears) is called the final reduction ratio (Huchtkoetter and Klein, 1996; Schmid et al., 2013).

The final reduction ratio is determined by taking into account the power of the vehicle engine, the weight of the vehicle, the acceleration capability of the vehicle. The final reduction ratio is selected between 3-5 for speed-oriented vehicles and between 5-8 for heavy vehicles such as vans and trucks preferred for load carrying (Erjavec and Ken, 2015).

1.4. Impact of differential ratio differences on the vehicle

Automobile manufacturers determine the differential ratio according to the area of use, depending on the expected usage characteristics of the vehicles (Topaç, 2010). For example, different types of differentials are used in off-road vehicles and speed-oriented vehicles. Depending on the conditions in which the vehicle will be used, vehicle type and engine power, differential ratios can start from 2:1 and go up to 12:1. However, the vehicles we encounter in everyday life usually have a ratio of 3:1 to 4:1. Automobile manufacturers produce automobiles between these ratios in order to maintain a balance between the high performance and low fuel consumption criteria desired by the user (Behrooz and Crolla, 2012).

The higher the difference in rotation between the gears, the lower the gear ratio will be (e.g. 4.7:1)

- Good acceleration
- Traction increase
- Low max. speed
- High fuel consumption
- Mostly cargo carrier vehicles and Off-Road vehicles
- As the difference in rotation amount between gears decreases, the gear ratio will increase (Ex. 2.3:1).
- Bad acceleration
- Poor traction

- Low max. speed
- Low fuel consumption

Engines with low power may not be able to handle it and it is mostly used in high power engines (Wang et al., 2019).

The aim of this study is to design and analyze differentials with different reduction ratios. The article includes a force-based analysis performed in Adams and Ansys simulation, as well as all calculations to prove the success of the design.

2. Material and Method

2.1. Dimensioning of hypoid gears

Hypoid gears are frequently used in vehicle mechanisms, and hypoid gears have similar characteristics to bevel gears. They transmit power and movement between shafts that are 90˚ to each other and whose axes do not intersect. The surface of the gear makes a sliding movement in the direction of the width of the gear. The teeth of hypoid gears are curved and hardened by heat treatment (Akkurt, 2000; Arslan, 2011).

Three methods are used for opening curvilinear bevel gears: Gleason, Klingelnberg and Oerlikon. In the Oerlikon method, in Figure 2, the tool head rotates around its own axis and this center is continuously rolled on a circle that is concentric with the center of the plan gear. In this way, all gears are opened.

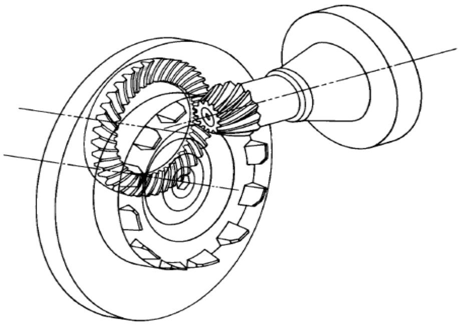

Figure 2. Machining of hypoid gear by oerlikon method (Xuan, 2024)

By changing its position on the axis, the value of the contact surfaces on the outer sidewall is obtained. In short, large forces are transferred. The sizing of the large diameter hypoid gear is realized according to the data of the machine. To achieve a noiseless movement with minimum wear, the outer sidewalls of the gears must be in contact with the ring section of the gears. (Schmid et al., 2013).

2.2. Hypoid gear calculation

The hypoid gear design consists of two parts. During the initial design, values are selected based on similar characteristics found in the literature. Geometric values are obtained according to the initial design data. During the second design, the data and geometric values from the first design are checked to ensure that the calculation data are safe (Waldron and Kinzel, 2016; Altındağ, 2020).

The hypoid gear design consists of two parts. During the initial design, values are selected based on similar characteristics found in the literature. Geometric values are obtained according to the initial design data. During the second design, the data and geometric values from the first design are checked to ensure that the calculation data are safe (Moldovean, 2007).

<u>Calculations</u>

<u>Surface Pressure (B_{zul})</u>

Surface pressure was calculated as

$$B_{z_{ul}} = f_B \frac{k_0 \cdot y_G \cdot y_H}{C_s \cdot S_G^{0,5}} = 0,20 \frac{5 \cdot 1 \cdot 0,895}{1,5 \cdot 1,6^{0,5}} = 0,396$$

Pinion average segment circle diameter (d_{m1})

$d_{m1} \geq 113 \sqrt{\frac{P \cdot F_d}{n \cdot B_{zul}}} \geq$

$113 \sqrt{\frac{11{,}2 \cdot 1{,}5}{875 \cdot 0{,}396}} \geq 41{,}19$ value is found.

Ring gear angle (δ_2)

$$tg\delta_2 = \frac{dm_2}{2a} - \frac{dm_2}{2a} \cdot \frac{z_1}{z_2} + \frac{z_1}{z_2}$$

found as

$tg\delta_2 = \frac{193}{2.35} - \frac{193}{2.35} \cdot \frac{8}{47} + \frac{8}{47} = 2{,}458 \quad \delta_2 = 67{,}86°$

$\sin \varphi \approx \frac{2 \cdot a}{dm_2} \approx \frac{3.35}{193}$ found as $\varphi : 21{,}22°$

Mismatch angle (φA)

Calculated as $\varphi A : arctg\ (tg\ \varphi\ .\ sin^2 \cdot \delta_2) : arctg\ (\ tg\ 21.22\ sin^2 67.86\ )$

$\varphi A : 18.42°$

Pinion gear cone angle (δ_1)

$\delta_1 : \arcsin(cos\ \delta_2\ .\ cos\ \varphi A$

$: arc \sin(\cos 67{,}86\ .\ \cos 18{,}42)$

calculated as $\delta_1 : 20{,}95°$

Contact angle (φP)

φP

$: arctg\ (\ tg\varphi . \sin \delta_2)$

$: arctg\ (\ tg\ 21.22 \sin 67{,}86)$

found as $\varphi P : 19{,}78°$

Spiral angles ($\beta 1\ ve\ \beta 2$)

$tg\beta 1 : (\ i \left(\frac{dm_1}{dm_2}\right) - \cos \varphi\ P) / \sin \varphi P$

$tg\quad \beta 1 : (\ 5{,}875 \left(\frac{41{,}19}{193}\right) - \cos 19{,}78) / \sin 19{,}78$

found as $\beta 1 : 42{,}75°$

$\beta 2 :\ \beta 1 - \ \varphi P \quad : 42{,}75 - 19{,}78$

found as $\beta 2 : 22{,}97°$

Normal module (m_{mn})

The modulus is the first factor that determines gear strength, the ability to transfer power. There are different ways to determine the module of gears. For example, one way is to determine the module after determining the number of gears. After the approximate determination of parameters such as the material, the number of teeth on the gear and the tooth width of the gear, the module is calculated. In helical gear type, the normal module value is determined by dividing the average pitch circle diameter by the number of d_{m2} teeth and then multiplying the helix angle of the gear (Arslan, 2011).

Calculated as $m_{mn} : \cos \beta 2\ .\ \frac{dm_2}{z_2}$: cos 22,97 . $\frac{193}{47}$

m_{mn} : 3,78 mm

Ring gear tooth width (b_2)

$b_2 \leq 0{,}18\ .\ d_{m2}$

$b_2 \leq 0{,}18\ .\ 193 \qquad b_2 : 33$ mm

Pinion gear tooth width (b_1)

$b_1 \approx \frac{b_2}{\cos \varphi\ P} + 3.\ m_{mn}\ .\ tg\varphi P$

$b_1 \approx \frac{33}{\cos 19{,}78} + 3.\ 3{,}78\ .\ tg 19{,}78$

$b_1 \approx 39.14 \qquad b_1$: 39 mm

If we check the normal module formula;

it is found as $m_{mn} : \cos \beta 1.\ (d_{m1} / z_1) : \cos 42{,}75\ (\ 41{,}19 / 8) : 3{,}78$ mm

2.3. Calculation of tooth dimensions

The formulas

$h_k : 1.\ m_{mn}$

$h_f : 1{,}25.\ m_{mn}$

$h : h_k + h_f$ are used.

According to the formulas above ;

h_k : 3,78 mm , h_f: 4,725 mm and h: 8,505 mm are found.

2.4. Equivalent spur gear calculations for hypoid gear wheels

Rolling circle diameter (d_e)

$$de_1 = \frac{d_{m1}}{\cos\Delta\propto . \cos\delta_1}$$

$$de_1 = \frac{41{,}19}{1. \cos 20{,}95} = 44{,}10\ mm$$

$$re_1 = \frac{d_{e1}}{2} = \frac{44{,}10}{2} = 22{,}05\ mm \text{ bulunur.}$$

$$de_2 = \frac{d_{m2}}{\cos\Delta\propto . \cos\delta_2\ .\ cos^2\varphi p}$$

$$de_2 = \frac{193}{1.\cos 67{,}86\,.\,cos^2 19{,}78} = 578{,}34\ mm$$

$$re_2 = \frac{d_{e2}}{2} = \frac{578{,}34}{2} = 289{,}17\ mm$$ bulunur.

Equivalent tooth numbers (z_e)

$$ze_1 = \frac{z_1}{\cos\Delta\propto .\cos\delta_1}$$

$$ze_1 = \frac{8}{1.\cos 20{,}95} = 8{,}56\ mm$$

$$ze_2 = ze_1\,\frac{de_2}{de_1}$$

$$ze_2 = 8{,}56\ \frac{578{,}34}{44{,}1} = 112{,}25\ mm$$

Geometric connections (α_e)

$$\text{tg}\ \propto_e = \frac{\text{tg}\ \propto_n}{\cos\beta 1} = \frac{\text{tg}\ 20}{\cos 42{,}75}$$

found as $\propto_e = 26{,}36$ °

Step (t_e)

$t_e = me\,.\,\pi\,.\cos\alpha e$ = 5,14 . π . cos 26,36

$t_e = 14{,}46\ mm$

Equivalent number of teeth in the taxonomy circle (Z_n)

As the helix angle increases, the required number of teeth (z1) decreases proportionally to avoid root cutting.

Z_{1n} = (Z_{1e} . cos $\Delta\propto$) / ($\cos^2 \beta_g$. cos β_1)

Z_{1n} = (8,56 . 1) / (0,593 . cos 42,75) = 19,75

Z_{2n} = (Z_{2e} . cos $\Delta\propto$) / ($\cos^2 \beta_g$. cos β_2)

Z_{2n} = (112,25 . 1) / (0,593 . cos 22,97) = 205,59

Diameters of equivalent taxonomy circles (d_n)

While calculating the equivalent taxomony circle diameters, the equivalent taxomony diameters of the ring gear and pinion gear are obtained by multiplying the normal module and the number of teeth.

$d_{1n} = m_{mn}$. Z_{1n} = 3,78 . 19,78 = 74,65 mm

found as $d_{2n} = m_{mn}$. Z_{2n} = 3,78 . 205,59 = 777,13 mm

$r_{1n} = d_{1n}$ / 2 = 74,65 / 2 = 37,32 mm

found as $r_{2n} = d_{2n}$ / 2 = 777,13 / 2 = 388,56 mm

Equivalent tooth top circle radii (r_k)

$r_{k1} = r_{1n} + h_{k1}$ = 37,32 + 4,3 = 41,62 mm

$r_{k2} = r_{2n} + h_{k2}$ = 388,56 + 3,1 = 391,66 mm bulunur.

Equivalent tooth bottom circle radii (r_f)

$r_{f1} = r_{1n} - h_{f1}$ = 37,32 – 4 = 33,32 mm

$r_{f2} = r_{2n} - h_{f2}$ = 388,56 – 5,2 = 383,36 mm

2.5. Positioning of spider and axle gears

The arrangement of the spider and axle gears inside the differential gearbox and connected to the ring gear is shown in Figure 3.

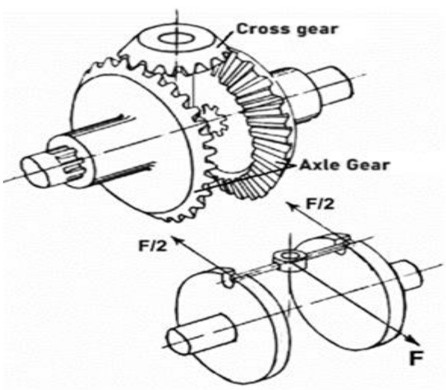

Figure 3. Arrangement of axle and spider gears in the differential box

If any of the axle gears in the fixed differential box is rotated forward at a speed of 10 rpm, the axle gear of the same size will rotate in the reverse direction at a speed of 10 rpm. During movement at constant speed, the torques transmitted to the wheels are equal. As can be seen in Figure 3 the axle gears were replaced with notched discs placed on them, one of the spider gears was embedded by being centered vertically, and the notches whose ends were opened were replaced with a rod placed to meet each other.

3. Results and Discussion

3.1. Evaluation of hypoid gear wheel data

Mesh elements and node numbers were selected to be suitable for both realistic analysis results and processing speed. Mesh convergence was performed to select the appropriate mesh. In this way, the development of the values was observed as the mesh became smaller. Meanwhile, no significant difference was observed in the measurements after the 0.8 mm mesh size. For this reason, 0.8 mm mesh size was determined as the optimum mesh size. Quadratic elements were used for mesh elements to be closer to reality. The mesh process has been completed with Tetra10 elements as Quadratic elements.

In line with the calculations, the first data entries and results are shown in Table 3.1 below. The number of pinion gears is 8 and the number of ring gears is 47. According to the number of gears, the conversion ratio was found to be 5.8.

In line with the data, the design was made using the solidworks drawing program.

Table 1. Data input and results

1.DATA	
* DATA INPUT	
Number of pinion gears (z1)	8
Number of face gears (z2)	47
Distance between axes (a)	35
* RESULTS	
Conversion rate (i)	5,875
Module in front section (me)	5,14
Section circle diameter (dm1)	42
Normal modulus (mp)	3,78
Step (te)	14,46
Pinion tooth width (b1)	39
Mirror tooth width (b2)	33
Overdental size (hk)	3,7
Tooth root size (hf)	4,6
* GEAR ANGLES	
Pitch engagement ratio	2,22
Contact angle	19,78°
Pinion cone angle (δ_1)	20,95°
Mirror cone angle (δ_2)	67,86°
Helix angle β_1	42,75°
Helix angle β_2	22,97°

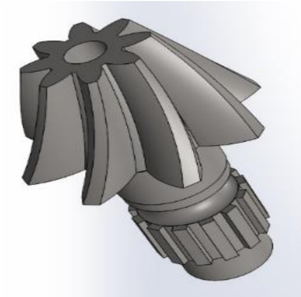

Figure 4. Pinion gear view in Solidworks environment

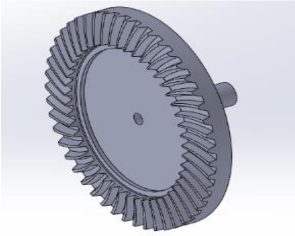

Figure 5. Ring gear view in Solidworks environment

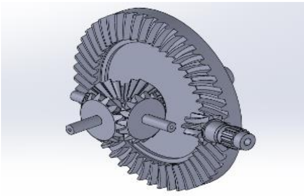

Figure 6. Assembly view in Solidworks environment

After drawing the design parts, each part was assembled as a solid model in the assembly interface. After the assembly data is entered to the parts, the parts are moved and motion animation is performed.

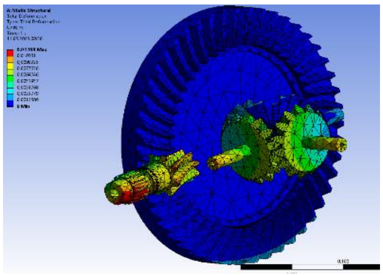

Figure 7. Total deformation analysis view

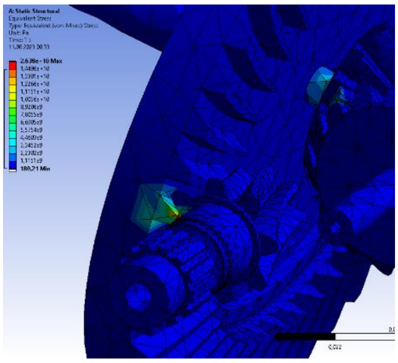

Figure 8. Stress analysis between ring and pinion

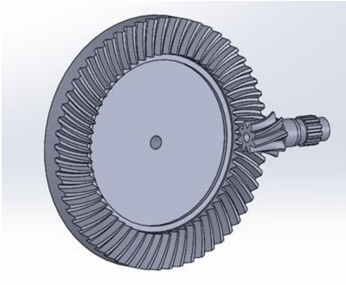

Figure 9. Data ring-pinion view

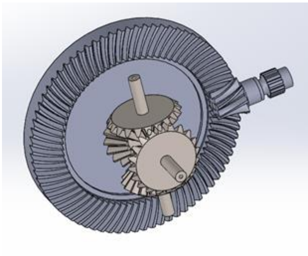

Figure 10. Data ring-pinion view

Table 2. Data input and results

2.DATA	
***DATA INPUT**	
Number of pinion gears (z1)	8
Number of face gears (z2)	58
Distance between axes (a)	42
*** RESULTS**	
Conversion rate (i)	7,25
Module in front section (me)	7,57
Section circle diameter (dm1)	42
Normal modulus (mp)	2,06
Step (te)	14,23
Pinion tooth width (b1)	38
Mirror tooth width (b2)	33
Overdental size (hk)	2,06
Tooth root size (hf)	2,57
*** GEAR ANGLES**	
Pitch engagement ratio	5,65
Contact angle	23,61°
Pinion cone angle (δ_1)	23,39°
Mirror cone angle (δ_2)	64,72°
Helix angle β_1	74,22°
Helix angle β_2	50,61°

Regarding the 2nd data input, the number of pinion gears is kept constant and the number of ring gears is selected as 58. The cycle ratio is calculated from the ratio of the number of ring gears to the number of pinion gears. The 2nd data cycle rate is calculated as 7.25.

Table 3. Data input and results

3.DATA	
* DATA INPUT	
Number of pinion gears (z1)	8
Number of face gears (z2)	75
Distance between axes (a)	45
* RESULTS	
Conversion rate (i)	9,37
Module in front section (me)	5,12
Section circle diameter (dm1)	42
Normal modulus (mp)	1,86
Step (te)	11,35
Pinion tooth width (b1)	39
Mirror tooth width (b2)	33
Overdental size (hk)	1,86
Tooth root size (hf)	2,32
* GEAR ANGLES	
Pitch engagement ratio	6,21
Contact angle	25,26°
Pinion cone angle (δ_1)	24,09°
Mirror cone angle (δ_2)	63,69°
Helix angle β_1	68,73°
Helix angle β_2	43,47°

In the 3rd data input, the pinion gear is kept constant. The ring gear is selected as 75. Cycle ratio is calculated as 9.37. The number of pinion gears is fixed according to the data. The number of ring gears is increased in each data set to increase the cycle ratio. The normal modulus value decreased with the increase in the cycle ratio. Depending on the cycle ratio, the step clutch ratio increased with the increase in the number of ring gears.

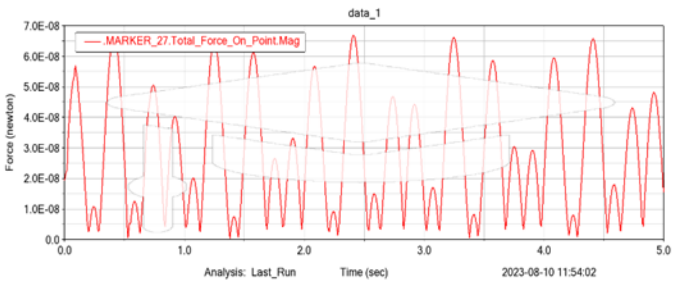

Figure 11. 1st data total force slope analysis

Figure 12. Data total force slope analysis

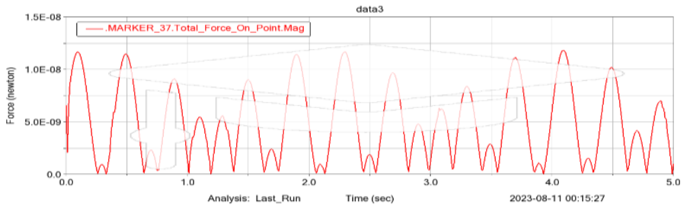

Figure 13. Data total force slope analysis

Figure 11 shows the total force slope analysis of the differential gearbox based on the results of data 1. By increasing the number of teeth in the ring gear, the differential gearbox cycle ratio increased.

5. Conclusions

For good driving ability and fuel consumption, differentials must be designed according to the vehicles' performance, road holding and some special situations. Different types of differentials are available for a wide variety of needs.

If there was no differential, the wheels on the drive axle would rotate at the same speed. When the vehicle rounds the bend, the tires on the inside wear out, and without a differential, the road holding would be poor because the wheels could not rotate at different speeds.

As a result of the design and analysis, the following results are given:

- The design of the hypoid gear mechanism was carried out in line with mathematical calculations.
- In line with the calculations, the design was carried out in the Solidworks program.
- By keeping the number of pinion gears constant, the cycle ratio was changed thanks to the number of ring gears.
- The normal modulus value decreased with the increase in the cycle rate.
- Depending on the cycle rate, the step clutch ratio increased with the increase in the number of ring gears
- An increase in the contact angle was also observed when the number of ring gears increased.
- The use of hypoid gears instead of spur gears provides a quieter working environment.
- As the distance between the axes increases, the ring helix angle decreases. With the decrease in the helix angle, an increase in the tooth length is observed.
- An increase in the modulus value was observed with the increase in the distance between the axes. With the increase in the normal modulus value, the load carrying capacity also increases.
- As the distance between the axes increases, the yield decreases.
- As a result of the analyzes carried out, it provided the opportunity to intervene earlier in the problems to be observed during and after the production process.

Declaration of Author Contributions

The authors declare that they have contributed equally to the article. All authors declare that they have seen/read and approved the final version of the article ready for publication.

Declaration of Conflicts of Interest

All authors declare that there is no conflict of interest related to this article.

Acknowledgment

This study was produced from the first author's master's thesis.

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