

ТОРАЙҒЫРОВ УНИВЕРСИТЕТІНІҢ
ҒЫЛЫМИ ЖУРНАЛЫ

НАУЧНЫЙ ЖУРНАЛ
ТОРАЙҒЫРОВ УНИВЕРСИТЕТА

**ҚАЗАҚСТАН ҒЫЛЫМЫ
МЕН ТЕХНИКАСЫ**

2001 ЖЫЛДАН БАСТАП ШЫҒАДЫ



**НАУКА И ТЕХНИКА
КАЗАХСТАНА**

ИЗДАЕТСЯ С 2001 ГОДА

ISSN 2788-8770

№ 2 (2025)

ПАВЛОДАР

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ТОРАЙГЫРОВ УНИВЕРСИТЕТ**
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СВИДЕТЕЛЬСТВО

о постановке на переучет периодического печатного издания,
информационного агентства и сетевого издания
№ KZ51VPY00036165

выдано

Министерством информации и общественного развития
Республики Казахстан

Тематическая направленность

Публикация научных исследований по широкому спектру проблем
в области металлургии, машиностроения, транспорта, строительства,
химической и нефтегазовой инженерии, производства продуктов питания

Подписной индекс – 76129

<https://doi.org/10.48081/VJZF6109>

Импакт-фактор РИНЦ – 0,216

Импакт-фактор КазБЦ – 0,406

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DESIGN AND OPTIMIZATION OF GEOMETRIC PARAMETERS OF GEARS FOR AN ELECTRIC VEHICLE TRANSMISSION

This study focuses on the design and optimization of the geometric parameters of gears in electric vehicle (EV) transmissions to enhance performance, efficiency, and noise reduction. The optimization process was conducted using KISSsoft, considering constraints such as input speed and torque, transmission system volume, and gear ratios for each driving mode. A novel aspect of this research is the integration of transmission efficiency analysis with a variable efficiency coefficient into the optimization workflow, which has been demonstrated to significantly influence the optimal design of multi-speed EV transmissions.

Throughout the optimization, key factors such as tooth bending stress, contact stress, contact ratio, and specific sliding were analyzed. Additionally, the impact of module variations on gear profile geometry, overlap ratio, tooth count, and gear pair mass was investigated. Strength calculations for the optimized gear pairs, determined for all geometric parameters using KISSsoft, were performed in accordance with the mathematical model outlined in ISO 6336, and the results were subsequently compared. The findings emphasize the critical role of profile modifications and precise gear design in optimizing gear performance, ultimately contributing to improved efficiency, reliability, and service life.

Keywords: gear design optimization, electric vehicle transmission, KISSsoft, gear profile modification, ISO 6336.

Introduction

In recent years, the automotive industry has been transitioning from internal combustion engine (ICE) vehicles to electric vehicles (EVs) due to increasingly stringent fuel efficiency and emission regulations worldwide. To meet these evolving standards and enhance EV performance, extensive research and development efforts are being directed toward the creation of advanced electric vehicle transmission systems, aiming to improve energy efficiency, driving dynamics, and overall sustainability [1].

The development of electric vehicles is progressing rapidly, with related technologies continuously advancing and gradually reaching maturity. As key components of the

pure electric vehicle drive system, the drive motor and transmission require careful parameter matching and design optimization to maximize the vehicle's driving range under identical battery conditions [2]; [3]. Electric vehicles have traditionally relied on single-speed gearboxes due to their straightforward design and high efficiency. Nevertheless, there is growing interest in multi-speed gearboxes for electric vehicles because of their potential advantages. These include improved efficiency of the electric traction machine and inverter, enhanced vehicle acceleration, better gradeability, higher top speed, and possible reductions in the overall mass and volume of the traction system [4]. Multi-speed gearboxes may be particularly advantageous for performance electric vehicles, light to heavy-duty trucks, and buses, given their substantial torque and power demands. It is worth noting that the ideal gearbox design can differ based on the specific use case and driving conditions [5].

For example, [6] shows that optimization outcomes vary between simulations using the New Europe Drive Cycle and Urban Dynamometer Driving Schedule, underscoring the necessity for thorough design and optimization of the entire powertrain system. Furthermore, [7] indicates that single speed gearboxes configurations offer efficiency benefits compared to multi-speed gearboxes, but stresses the importance of considering both application-specific machine design and topology to fully realize the technological potential.

This paper introduces an innovative power solution for electric vehicles that incorporates an electric motor and a planetary gear system. The optimization process focuses on two primary objectives: the volume and transmission efficiency of the two-speed gearbox. The suggested mechanism is particularly suitable for EV drive systems, owing to its uncomplicated design and capacity for smooth gear shifts at high velocities. Additionally, it is adaptable for use in in-wheel drive systems, which require an extremely compact configuration. This arrangement addresses speed disparities through the utilization of open gear shift components, effectively reducing clutch resistance losses.

Materials and methods

This study focuses on selecting the optimal transmission for an electric vehicle by considering its specific characteristics (table 1) and those of the electric motor, utilizing the KISSsoft software [8] for analysis.

Table 1 – Basic requirements for EV

Item	Parameter
Seats	1
Vehicle weight, kg	200
Power reserve, km	100
Maximum speed, km/h	60
Maximum gradeability, %	>20
Wheelbase, mm	1750
Wheel size	R12

The vehicle is equipped with an electric motor rated at 10 kW (table 2), operating at a nominal voltage of 72 V DC. To reduce size and weight, DC propulsion motors generally function at high rotational speeds. As a result, a reduction gearbox is necessary to align the motor's speed with the wheel rotation [9]; [10]; [11].

Table 2 –Technical characteristics of the electric motor

Item	Parameter
Constant voltage	72 V
Rated power	10 kW
Rated torque	50 Nm
Rated base speed	2000 rpm
Rated maximum speed	5000 rpm

The determination of the overall transmission ratio must satisfy the vehicle's performance criteria, ensuring both adequate gradeability for ascent and the capability to attain the maximum velocity [12]. The corresponding expression is:

$$i_o = \frac{\pi \cdot n_{\max} \cdot d_t}{v_{\max}} = 8.2$$

where, $n_{\max}$ – maximum speed of the electric motor ($n_{\max} = 5000$ rpm)

d_t – wheel diameter ($d_t = 522$ mm = 0.522 m)

$v_{\max}$ – maximum speed ($V_{\max} = 60$ km/h = 1000 m/min)

The gear ratio of the first stage can be determined using the following formula [13]:

$$i_1 = 0,9365 \cdot i_o^{0.667} = 3.8$$

$$i_2 = \frac{i_o}{i_1} = 2.1$$

where, i_1 – first stage gear ratio

i_2 – second stage gear ratio

After entering output parameters of electric motor and gear ratios into KISSsoft, program designed 3D model and kinematic scheme. In figure 1. shows kinematic scheme of EV transmission consists of two sequential gear pairs that progressively reduce or increase speed and torque.

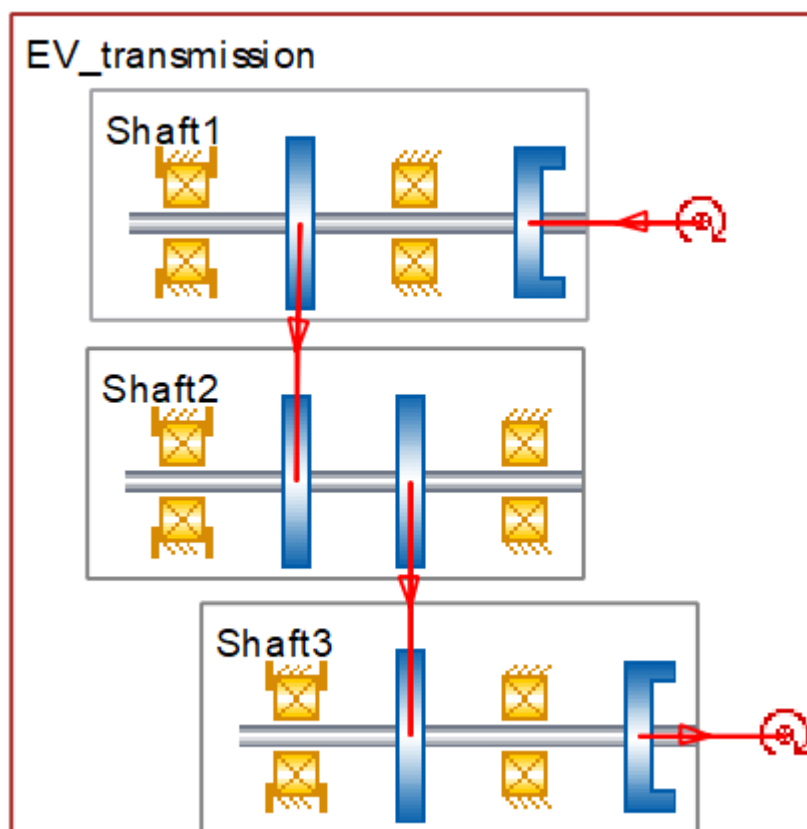


Figure 1 – Kinematic scheme of EV transmission

The preliminary calculation of the gearbox dimensions is carried out using specialized software, determining key parameters such as center distance, tooth width, gear diameters, and gear ratio while considering permissible loads. The optimization process focuses on achieving one of several key objectives: minimizing weight to reduce energy consumption, increasing torque capacity for high-load transmission, or enhancing specific power to balance compactness, strength, and efficiency.

The design process for the first stage of the gearbox in KISSsoft begins with model preparation, which includes defining gear, shaft, and bearing parameters, as well as evaluating their deformations. The nominal gear ratio of the first stage ($i_1=3.8$) is then specified, dictating both speed and torque, and the operational context of the transmission is selected. An initial calculation is performed to determine center distances, gear module, gear diameters, and tooth width, as well as to estimate component weight for structural mass optimization. Subsequently, six configurations with maximum specific power are analyzed, leading to the selection of the optimal design (Figure 3). This final configuration ensures high power density while maintaining minimal center distance, thereby enhancing the gearbox's compactness, mechanical strength, and overall efficiency.

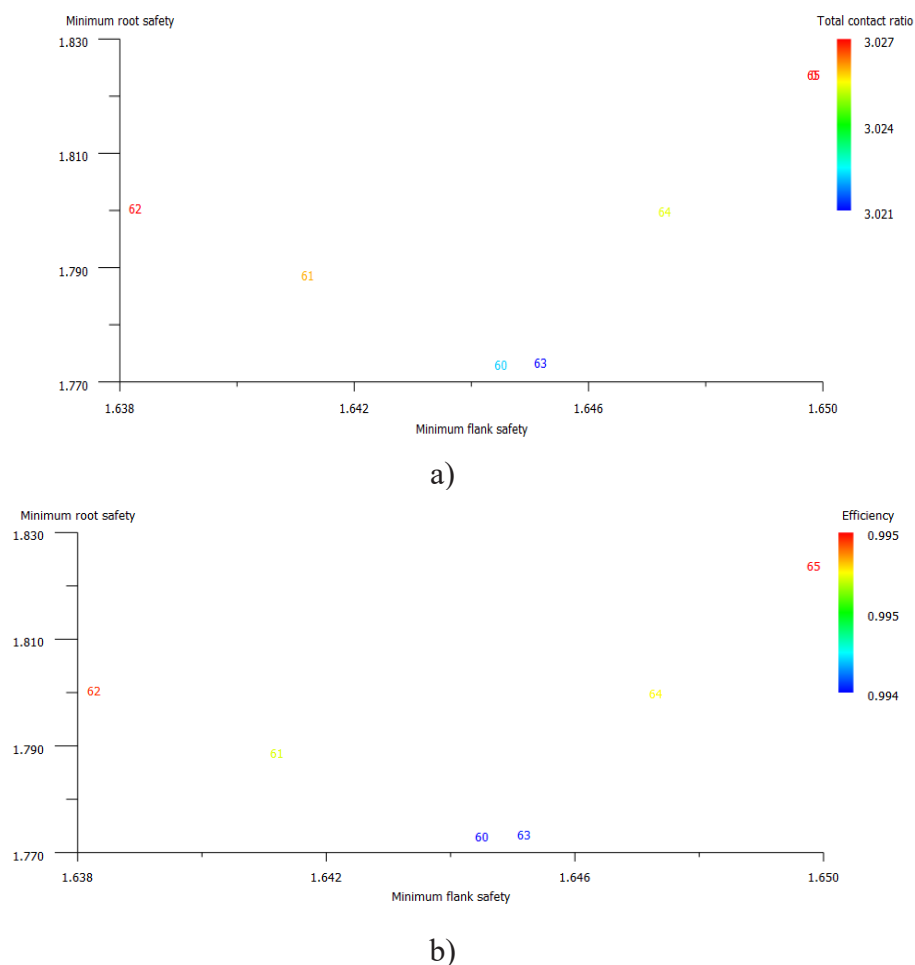


Figure 2 – Graphs of the dependence of minimum root safety on minimum flank safety for the general contact ratio (a) and for the efficiency (b) of the first speed gear pair

Geometry																							
Tooth geometry symmetric																							
Normal module	m_n	1.0000 mm	↔																				
Normal pressure angle	α_n	20.0000 °	↔																				
Gear 1		helix right hand	↕																				
Helix angle at reference circle	β	9.0000 °	↔																				
Center distance	a	61.0000 mm	↔																				
<table border="1"> <thead> <tr> <th></th> <th>Gear 1</th> <th>Gear 2</th> <th></th> </tr> </thead> <tbody> <tr> <td>Number of teeth</td> <td> 61 </td> <td> 60 </td> <td></td> </tr> <tr> <td>Facewidth</td> <td> 25.0000 </td> <td> 25.0000 </td> <td>mm +</td> </tr> <tr> <td>Profile shift coefficient</td> <td> -0.1293 </td> <td> -0.1209 </td> <td>↔ ↔ ↗</td> </tr> <tr> <td>Quality (ISO 1328:2013)</td> <td> 6 </td> <td> 6 </td> <td>↗</td> </tr> </tbody> </table>					Gear 1	Gear 2		Number of teeth	61	60		Facewidth	25.0000	25.0000	mm +	Profile shift coefficient	-0.1293	-0.1209	↔ ↔ ↗	Quality (ISO 1328:2013)	6	6	↗
	Gear 1	Gear 2																					
Number of teeth	61	60																					
Facewidth	25.0000	25.0000	mm +																				
Profile shift coefficient	-0.1293	-0.1209	↔ ↔ ↗																				
Quality (ISO 1328:2013)	6	6	↗																				
Material and lubrication																							
Gear 1	Case hardening steel	18X2H2M, case-hardened, ISO 6336-5 Figure 9/10 (MQ), Core hardness $\geq$ 25HRC Jominy J=12mm +																					
Gear 2	Case hardening steel	18X2H2M, case-hardened, ISO 6336-5 Figure 9/10 (MQ), Core hardness $\geq$ 25HRC Jominy J=12mm +																					
Lubrication	Oil bath lubrication	ISO-VG 220 + ↔ +																					

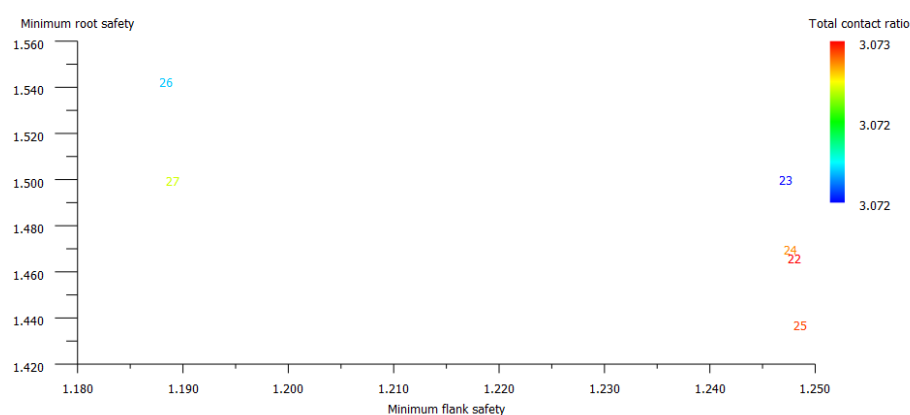
Figure 3 – Geometric parameters of the first speed gear pair

The graph (Figure 2.a) represents the relationship between minimum root safety and minimum flank safety for various gear meshing configurations. The color scale

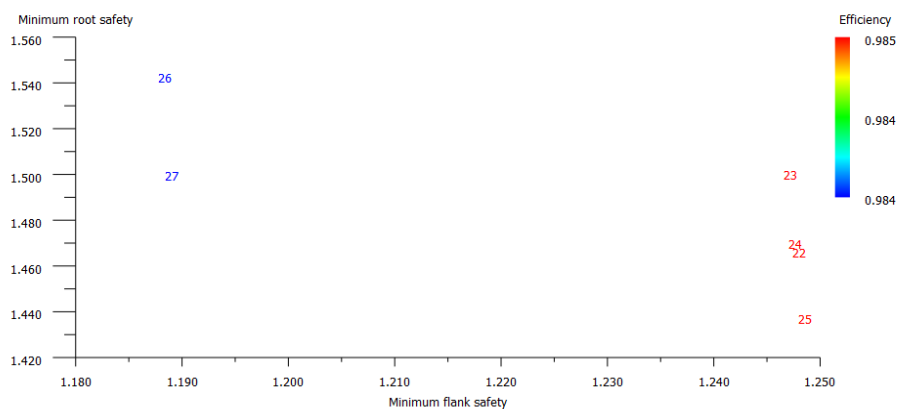
indicates the total contact ratio, which describes the number of tooth pairs engaged simultaneously. A higher contact ratio ensures smoother load transmission and reduces impact forces. Unlike the previous graph, the color scale here represents efficiency, highlighting energy losses within the transmission system. Higher efficiency values correspond to lower energy losses during power transmission.

As a result, the most optimal configuration is represented by point 65, as it demonstrates high flank safety, ensuring resistance to surface wear, high root safety, preventing fatigue failure, a high contact ratio that guarantees smooth load distribution, and superior efficiency that minimizes energy losses.

A similar analysis is conducted for the second gear stage (Figure 4), leading to the selection of point 23 (Figure 5) as the optimal configuration due to its favorable geometric parameters. Additionally, the software generates a 3D model of the transmission system (Figure 6).



a)



b)

Figure 4 – Graphs of the dependence of minimum root safety on minimum flank safety for the general contact ratio (a) and for the efficiency (b) of the second speed gear pair

Geometry		Gear 1		Gear 2		Details...	
Tooth geometry		symmetric					
Normal module	m_n	1.6250	mm	↔	Number of teeth	z	28
Normal pressure angle	α_n	20.0000	°	↔	Facewidth	b	35.0000
Gear 1	helix right hand				mm	+	
Helix angle at reference circle	β	9.0000	°	↔	Profile shift coefficient	x	0.0976
Center distance	a	70.0000	mm	↔	Quality (ISO 1328:2013)	A	6
Material and lubrication							
Gear 1	Case hardening steel	18X2H2M, case-hardened, ISO 6336-5 Figure 9/10 (MQ), Core hardness $\geq 25\text{HRC}$ Jominy J=12mm					
Gear 2	Case hardening steel	18X2H2M, case-hardened, ISO 6336-5 Figure 9/10 (MQ), Core hardness $\geq 25\text{HRC}$ Jominy J=12mm					
Lubrication	Oil bath lubrication	ISO-VG 220					

Figure 5 – Geometric parameters of the second speed gear pair

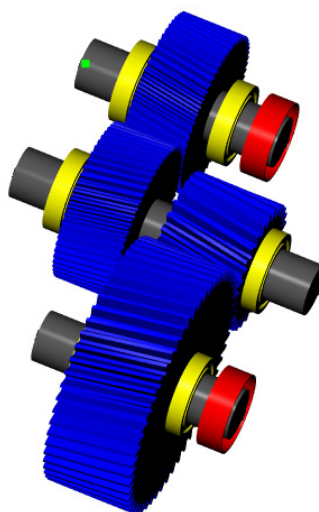


Figure 6 – 3D model of the electric vehicle gearbox

Next, we optimize the gear transmission by implementing deep-tooth profile gears [14] using KISSsoft. This approach significantly enhances transmission performance by enabling precise calculation of meshing parameters.

Within the software, we conduct a contact analysis using the face load factor ($K\beta$), which quantifies the load distribution variation along the tooth width. The calculation is performed without considering manufacturing deviations, in accordance with Annex E of the ISO 6336-1 standard.

Face load factor		
Method	Calculation without manufacturing deviation ISO 6336-1 Annex E	
Face load factor	$K_{H\beta}$	1.8160

Figure 7 – Load distribution factor along the tooth length $K_{H\beta}$ calculated without considering manufacturing deviations, in accordance with Appendix E of ISO 6336-1

In Figure 7, the face load factor $K_{H\beta}=1.8160$ indicates that the load distribution along the gear tooth width is uneven. This means that in the most heavily loaded areas, the overload reaches 81.6 % compared to a uniform load distribution.

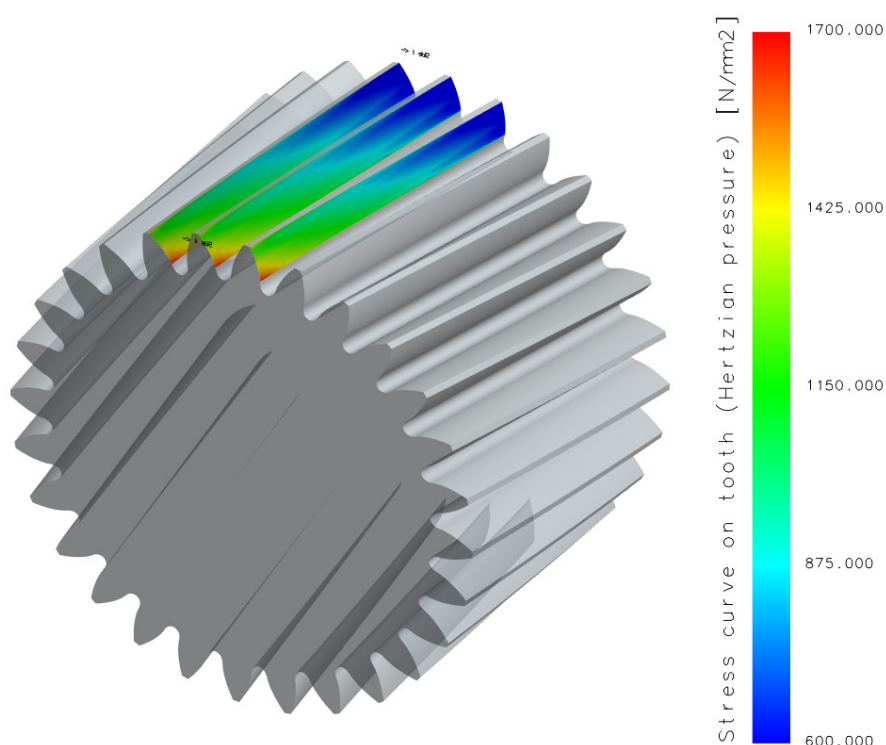


Figure 8 – Contact stress distribution

Figure 8 illustrates the distribution of contact stresses (Hertzian pressure) on the gear teeth. Due to the high face load factor $K_{H\beta}$, a zone of increased stress is observed on the tooth surface, located closer to the beginning of contact, with a gradual decrease towards the tooth base. This indicates an uneven load distribution, which may be attributed to gear meshing geometry, misalignment errors, or insufficient tooth profile modifications.

To reduce the $K_{H\beta}$ factor, we will recalculate the values considering manufacturing deviations according to ISO 6336-1 Annex E. Additionally, we will apply further modifications (Figure 9) aimed at optimizing the load distribution along the tooth width.

Additional modifications

Variant for calculation: +

Gear	Flank	Modification type	Value [μm]	Factor 1	Factor 2	Status
Gear 1	both	Flank line crowning	5.0000	1.0000		active
Gear 1	both	Helix angle modification, parallel (value)	-11.0000			active
Gear 2	both	Flank line crowning	5.0000	1.0000		active
Gear 2	both	Helix angle modification, parallel (value)	-11.0000			active

Figure 9 – Variants of additional gear modifications in KISSsoft software

Face load factor

Method:

Face load factor	$K_{\text{H}\beta}$	1.1807
Toothing tolerance	$ \Sigma f_{\text{H}\beta} $	10.7000 μm
Axis alignment tolerance	$ f_{\text{per}} $	6.6000 μm

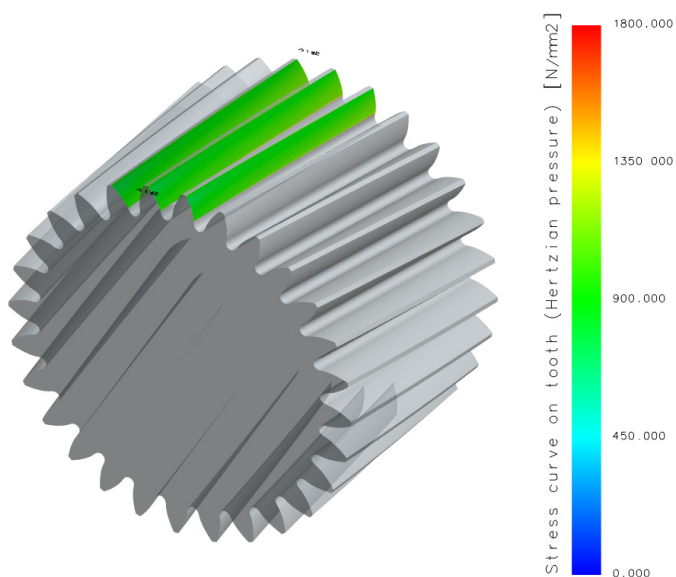
Figure 10 – (a) Load distribution factor along the tooth length $K_{\text{H}\beta}$ calculated considering manufacturing deviations according to ISO 6336-1 Annex E. (b) Contact stress distribution

Figure 9 presents a table of additional gear modifications in KISSsoft. In this configuration, both gears utilize the Flank line crowning modification with a value of 5 μm , which introduces a slight rounding along the tooth width. This helps to reduce stress concentrations and compensate for potential alignment errors. Additionally, both gears include the Helix angle modification, parallel (value) with a value of -11 μm , which adjusts the helix angle of the tooth line. This modification influences the

load distribution among the teeth, reducing local overloads and improving meshing conditions.

After recalculating all parameters, the software produced the results shown in Figure 10a. Comparing this value with the previous one (1.8160, without manufacturing deviations), it is evident that considering real manufacturing tolerances has significantly reduced the load factor. This indicates that after implementing the adjustments, the gear transmission now distributes the load more evenly along the tooth width.

Result and discussion

The study optimized an electric vehicle (EV) transmission using KISSsoft software, focusing on load distribution and design efficiency through a step-by-step approach:

Preliminary Design: Key parameters – center distance, gear module, and tooth width – were calculated, setting the first-stage gear ratio to balance speed and torque.

Configuration Selection: Six configurations were analyzed, selecting point 65 for the first stage and point 23 for the second, based on safety, contact ratio, and efficiency.

Stress Analysis: Initial calculations revealed uneven load distribution, with a face load factor of 1.8160.

Profile Modification: Flank line crowning (5 μm) and helix angle correction (-11 μm) reduced stress concentration.

Recalculation: Considering manufacturing deviations (ISO 6336-1 Annex E), the load factor dropped to 1.1807.

Optimization: The final design improved durability, reduced noise, increased efficiency, and maintained compactness. This structured method highlights KISSsoft as a key tool for optimizing gear design and managing real-world deviations.

This structured approach, combining deep-tooth profile gears, profile modifications, and helix angle corrections, significantly improved the transmission's reliability and operational efficiency, underscoring KISSsoft software as a vital tool for gear design optimization and real-world deviation management.

Conclusion

The analysis of gear transmission performance utilizing KISSsoft software examined various factors affecting the reliability and efficiency of meshing. The transmission error graph analysis revealed significant deviations, indicating fluctuations in gear operation that may result in vibrations and increased noise levels. The initial load distribution factor along the tooth length, $K\beta=1.8160$, indicated an 80 % overload, highlighting a substantial concentration of load in one engagement area.

To mitigate this coefficient, a tooth profile modification was implemented, including flank line crowning of 5 μm for both gears and a helix angle correction of -11 μm to compensate for alignment errors. Subsequent to these modifications, the load distribution factor decreased to $K\beta=1.1807$, indicating a more uniform load distribution.

The initial Hertzian contact stress distribution exhibited high local concentrations near the tooth edges. Following optimization, the stresses became more evenly distributed, confirming the efficacy of the tooth profile modification. The linear load distribution across the tooth width initially demonstrated a significant shift in peak load

due to alignment errors (f_{par}) and total profile deviations (Σf_H). However, the optimized gear geometry balanced the load distribution, thereby reducing the risk of local overloads.

Consequently, the utilization of deep-tooth profile gears, profile modifications, and helix angle corrections significantly reduced the load distribution factor and stress concentration, thereby improving gear durability, mitigating wear risk, and enhancing the smoothness of meshing. The application of KISSsoft software for gear transmission optimization proves to be an effective tool for improving the reliability and operational characteristics of the mechanism.

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Received 17.03.25.

Received in revised form 17.03.25.

Accepted for publication 05.06.25.

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17.03.25 ж. баспаға түсті.

17.03.25 ж. түзетулерімен түсті.

05.06.25 ж. басып шығаруға қабылданды.

ЭЛЕКТР КӨЛІКТЕРДІҢ БЕРІЛІСІНЕ АРНАЛҒАН ТІСТІ ДӨҢГЕЛЕКТЕРДІҢ ГЕОМЕТРИЯЛЫҚ ПАРАМЕТРЛЕРІН ЖОБАЛАУ ЖӘНЕ ОҢТАЙЛАНДЫРУ

Бұл зерттеу өнімділікті, тиімділікті және шуды азайтуды арттыру үшін электр көліктерінің (ЭК) берілістеріндегі берілістердің геометриялық параметрлерін жобалауға және оңтайландыруға бағытталған. Оңтайландыру процесі кіріс жылдамдығы мен айналу моменті, беріліс жүйесінің көлемі және әрбір жүргізу режимі үшін беріліс коэффициенттері сияқты шектеулерді ескере отырып, KISSsoft көмегімен жүргізілді. Бұл зерттеудің жаңа аспектісі - ауыспалы тиімділік коэффициенті бар беріліс тиімділігін талдауды оңтайландыру жұмыс процесіне біріктіру, ол көп жылдамдықты ЭК берілістерінің оңтайлы дизайнына айтарлықтай әсер ететіні дәлелденді.

Оңтайландыру барысында тістің иілу кернеуі, жанасу кернеуі, контакт қатынасы және ерекше сырғу сияқты негізгі факторлар талданды. Сонымен қатар, модульдердің вариацияларының беріліс профилінің геометриясына, қабаттасу коэффициентіне, тістер санына және беріліс жұбының массасына әсері зерттелді. KISSsoft көмегімен барлық геометриялық параметрлер

үшін анықталған оңтайландырылған беріліс жұптары үшін беріктік есептеулері ISO 6336 стандартында көрсетілген математикалық үлгіге сәйкес орындалды және нәтижелер кейін салыстырылды. Қорытындылар профильді модификациялаудың және берілістің нақты дизайнының беріліс өнімділігін оңтайландырудағы маңызды рөлін атап көрсетеді, сайып келгенде, тиімділікті, сенімділікті және қызмет ету мерзімін жақсартуға ықпал етеді.

Кілтті сөздер: беріліс дизайнын оңтайландыру, электр көлігінің трансмиссиясы, KISSsoft, беріліс профилін өзгерту, ISO 6336.

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Поступило в редакцию 17.03.25.

Поступило с исправлениями 17.03.25.

Принято в печать 05.06.25.

ПРОЕКТИРОВАНИЕ И ОПТИМИЗАЦИЯ ГЕОМЕТРИЧЕСКИХ ПАРАМЕТРОВ ЗУБЧАТЫХ КОЛЕС ДЛЯ ТРАНСМИССИИ ЭЛЕКТРОМОБИЛЯ

В данном исследовании основное внимание уделяется проектированию и оптимизации геометрических параметров зубчатых передач в трансмиссиях электромобилей (ЭМ) с целью повышения производительности, эффективности и снижения шума. Процесс оптимизации проводился с использованием KISSsoft с учетом таких ограничений, как входная скорость и крутящий момент, объем трансмиссионной системы и передаточные числа для каждого режима движения. Новым аспектом данного исследования является интеграция анализа эффективности трансмиссии с переменным коэффициентом эффективности в рабочий процесс оптимизации, что, как было продемонстрировано, оказывает существенное влияние на оптимальную конструкцию многоскоростных трансмиссий электромобилей.

В ходе оптимизации анализировались такие ключевые факторы, как напряжение изгиба зубьев, контактное напряжение, контактное отношение и удельное скольжение. Кроме того, было исследовано влияние изменений модуля на геометрию профиля зубчатого колеса, коэффициент перекрытия, количество зубьев и массу зубчатой пары. Расчеты прочности оптимизированных зубчатых пар, определенные для всех геометрических параметров с использованием KISSsoft, выполнялись в соответствии с математической моделью, изложенной в ISO 6336, а затем результаты сравнивались. Результаты подчеркивают важную роль модификации профиля и точного проектирования зубчатых передач в оптимизации рабочих характеристик передач, что в конечном итоге способствует повышению эффективности, надежности и срока службы.

Ключевые слова: оптимизация конструкции зубчатых передач, трансмиссия электромобиля, KISSsoft, модификация профиля зубчатых передач, ISO 6336.

Теруге 16.05.25 ж. жіберілді. Басуға 30.06.25 ж. қол қойылды.
Электрондық баспа
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Шартты баспа табағы 32,68 Таралымы 300 дана. Бағасы келісім бойынша.
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