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Determining the Gear Design Parameters Which Have Strong Correlation with Gear Volume and Contact Ratio for Helical Gear Design Optimization

(Menentukan Parameter Reka Bentuk Gear yang Mempunyai Korelasi Kuat dengan Isipadu Gear dan Nisbah Sentuhan untuk Pengoptimuman Reka Bentuk Gear Heliks)

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ABSTRACT

The market has a significant demand for low-cost, compact size, extended fatigue life and high load-carrying capacity helical gears. One way to manufacture such gears is by minimizing gear volume and maximizing contact ratio. Most researchers focus on gear design optimization, but minimal study has been conducted to identify design variables that significantly impact gear volume and contact ratio. Hence, it is crucial to determine gear design parameters that strongly correlate with gear volume and contact ratio beforehand. A single-stage helical gear train of local electric multiple units (EMU) is used as the gear model, and the calculations are done based on ISO 6336. Seven parameters have been chosen as independent variables; module, number of teeth of pinion, gear ratio, helix angle, tooth thickness factor, normal pressure angle and addendum factor. Graphs of total gear volume and contact ratio against independent variables are plotted to examine their relationship. The results indicate that normal module, number of teeth of the pinion, and gear ratio strongly correlate with the total gear volume. Meanwhile, the number of teeth of pinion, helix angle, tooth thickness factor, normal pressure angle and addendum factor strongly correlate with contact ratio. Therefore, these design parameters must be considered to improve the accuracy of the gear design optimization model, which aims to optimize gear volume and contact ratio.

Keywords: Helical gear; gear volume; contact ratio; gear design optimization; correlation

INTRODUCTION

The helical gear is one of the most fundamental components in the modern industrial sector. In a power transmission system, it is utilized to transmit motion and power between shafts via teeth. It has been extensively used in various mechanical engineering areas because helical gear meets the performance, precision, and specific power demands required by modern mechanical designs (Younes et al. 2022). Furthermore, helical gear has a high load-carrying capacity, operates at a higher speed, and is generally smooth and quiet in operation (Godwin Raja Ebenezer et al. 2017).

The gear design technique seems straightforward at first glance since the components are regularly seen, and the strength estimates may be computed based on the many gear standards available (Rai & Barman 2019). However, it is a complex process involving many design variables, graphs, empirical calculations, tables, and other aspects depending on the recommended gear standards (Rai & Barman 2018; Tang et al. 2020). Improper settings of design parameters of helical gear may lead to defects like tooth trace errors, tooth profile errors, excessive roughness, and even severe issues, including tooth surface scuffing, tooth size issues, and abnormal hob wear, which will raise costs and energy consumption. Technicians in helical gear manufacturing typically rely on their previous experience when setting design parameters. They infrequently modify and optimize these parameters based on conditions and requirements. This results in higher manufacturing costs, lower processing efficiency, and difficulty managing gear precision (Wu et al. 2021). Hence, creating the gear modification parameter for the optimization model and solving it via the optimization algorithm using a computer simulation technique to replace the actual experiment is an efficient strategy to achieve the optimum design of gear modification parameters (Tang et al. 2020).

There is a significant demand in the market for gears that are low cost, compact size, extended fatigue life and high load-carrying capacity (Rai & Barman 2018). One way to manufacture gears with low prices and compact size is by minimizing their total volume or total weight, which is directly proportional to the volume of gear (Maputi & Arora 2019; Sudhagar & Raman 2015). Several studies have been carried out to reach this objective. Under constrained conditions such as teeth's bending strength, shaft torsional strength, and gear dimensions, Yokota et al. (1998) utilized Genetic Algorithm to optimize a gear pair's weight, which allowed them to find the optimal weight for the gear pair. Thompson et al. (2000) studied the trade-off relationship between the volume of multi-stage spur gear units and the surface fatigue life of gears. Chong et al. (2002) established the automatical preliminary design of multi-stage gear drives by using the simulated annealing approach to reduce the geometrical volume of a gearbox. Mendi et al. (2010) employed a genetic algorithm to optimize the spur gear's module, shaft diameter, and rolling bearing to minimize gearbox volume. Savsani et al. (2010) developed an optimization system for a spur gear train using particle swarm optimization and simulated annealing techniques. Zhang et al. (2010) developed mathematical models for planetary gear reducers and used reliable grey PSO to minimize the gear volume. Golabi et al. (2014) presented actual graphs obtained from an optimization process aiming to optimize the volume/weight of the gearbox while taking the number of stages and their ratio into consideration. It is possible to use the graphs to determine the optimal number of stages, normal modules, and gear face width for a particular gearbox design. Wang et al. (2015) deduced the formula for the clearance volume of gear and improvised the formula of gear volume by considering the clearance volume. The formula is used to optimize the gear volume of single-stage spur gear. Miler et al. (2017) examined the effects of profile shift on the volume optimization of spur gear pairs, while Zhao et al. (2021) reduced the transmission ratio of rotary gear reducer to minimize its volume by using Genetic Algorithm. Yaw (2020) created an algorithm based on the Artificial Immune System (AIS) to determine the best combination of parameters for designing a two-stage planetary gear train system with the minimum volume.

On the other hand, maximizing a gear's contact ratio is one method to manufacture gears with extended fatigue life and high load-carrying capacity (Marimuthu & Muthuveerappan 2016). Since the gear tooth is never subjected to full load, the bearing capacity and overall mesh stiffness are considerably enhanced (Ding et al. 2017; H. Kang et al. 2017). Another benefit of gears with a high contact ratio is that movement is conveyed more evenly and consistently, as this implies reduced vibration, shock, and noise, all of which contribute to an increase in the gear transmission's overall performance (Godwin Raja Ebenezer et al. 2019; H. Kang et al. 2017). Kang and Choi (2008) optimized the helix angle for the helical angle as the helix angle positively correlates with a high contact ratio of a helical gear pair. Rackov et al. (2013) used the Generalized Particle Swarm Optimization algorithm to maximize the value of the contact ratio by optimizing the addendum heights and the correction factor of pinion. Huang et al. (2014) devised a multi-objective optimization model that maximized the contact ratio while minimizing the sensitivity of the contact ellipse to the misalignment errors based on the tooth contact analysis method with alignment considerations. Ren (2014) applied a reliability-based optimal design method in transmission design and established a multi-objective optimization model for automatic mechanical transmission to minimize the volume of gear and maximize the contact ratio simultaneously. Bozca (2017) on the other hand, set the minimization of tooth bending stress as the objective function in his optimization model of an automotive transmission gearbox. It has been discovered that raising the contact ratio reduces tooth bending stress. Kapelevich & Shekhtman (2017) analyzed the asymmetric tooth gears while considering the effective contact ratio, which is influenced by bending and contact tooth deflections. This is done to identify the optimal solution for high-performance gear drives that meet strict criteria for high load capacity, high efficiency and minimal transmission error. Rajesh et al. (2021) optimized the profile shift coefficient to maximize contact ratio since non-standard gears outperform standard gears regarding tooth strength, contact ratio, and several other factors.

According to the literature, researchers primarily focus on gear design optimization to optimize the gear volume and contact ratio. However, only a minimal study has been conducted to identify the design variables that may significantly impact the value of gear volume and contact ratio in the optimization model. Besides that, the accuracy of the volume model is poor since most researchers used a simplified method to calculate gear volume (Wang et al. 2019). On top of that, methods of calculating a high transverse contact ratio helical gears have never been developed extensively. This is likely because helical gear transmission has never been thoroughly researched (Pedrero et al. 2007; Sánchez et al. 2014). Therefore, to improve the accuracy of gear design optimization, it is crucial to determine which gear design parameters have a strong correlation with gear volume and contact ratio first before constructing the gear design optimization model. In this paper, a local EMU gear train (single-stage helical gear) is taken as a sample to examine the correlation between gear design parameters towards the helical gear volume and contact ratio. The gear volume and contact ratio calculations are done based on ISO 6336. Graphs of volume and contact ratio against gear design parameters are plotted. Then from the graphs, the correlation between gear design parameters towards gear volume and contact ratio is determined.

METHODOLOGY

THE FORMULA OF GEAR VOLUME

Due to the complexity of the helical gear's design, it is not easy to precisely quantify its overall volume. Furthermore, if the exact equation of total volume is used, the mathematical optimization model becomes more complex, making it difficult to compute the optimal solution. The gear volume equation is often derived from the pitch diameter to simplify the computation. However, to verify the significance of the profile shift coefficient towards the gear volume, the pitch diameter is substituted with the tip diameter, as shown in equation (1)-(2) (Rai & Barman 2019).

$$V_1 = \frac{\pi {d_{a1}}^2 b}{4} \tag{1}$$

$$V_2 = \frac{\pi {d_{a2}}^2 b}{4} \tag{2}$$

From Equation (1)-(2), b is the facewidth, V_1 and V_2 are volume of pinion and wheel respectively, while d_{a1} and d_{a2} is the tip diameter of pinion and wheel respectively. The equation related to the addendum diameter and face width of the helical gear is shown in Equation (3)-(5).

$$d_a = d + 2h^*_{an} m_n \tag{3}$$

$$d = \frac{m_n z}{\cos\beta} \tag{4}$$

$$b = \psi_d d_1 \tag{5}$$

From Equation (3)-(5), d, h^*_{an}, m_n, Z, β and ψ_d are pitch diameter, addendum factor, module, number of teeth and ratio of facewidth to pitch diameter, respectively. By adding Equation (1)-(2) while substituting Equation (3)-(5) into Equation (1)-(2), the total volume of pinion and gear wheel is as shown in equation (6)-(7).

$$V = \frac{\pi\left(\frac{m_n z_1}{\cos\beta} + 2h^*_{an} m_n\right)^2 \psi_d\left(\frac{m_n z_1}{\cos\beta}\right)}{4} + \frac{\pi\left(\frac{m_n z_2}{\cos\beta} + 2h^*_{an} m_n\right)^2 \psi_d\left(\frac{m_n z_1}{\cos\beta}\right)}{4} \tag{6}$$

$$= \frac{\pi\psi_d\left(\frac{m_n z_1}{\cos\beta}\right)\left[\left(\frac{m_n z_1}{\cos\beta} + 2h^*_{an} m_n\right)^2 + \left(\frac{m_n z_2}{\cos\beta} + 2h^*_{an} m_n\right)^2\right]}{4} \tag{7}$$

The relationship between the number of teeth of the pinion and the number of teeth of the wheel is shown in Equation (8).

$$z_2 = i z_1 \tag{8}$$

From Equation (8), i is the transmission ratio of helical gear. By substituting Equation (8) into Equation (7), the total volume of the pinion and gear wheel is as shown in Equation (9)-(10).

$$V = \frac{\pi\psi_d\left(\frac{m_n z_1}{\cos\beta}\right)\left[8{m_n}^2 {h^*_{an}}^2 + \frac{4(1+i){m_n}^2 z_1 h^*_{an}}{\cos\beta}\right]}{4} \tag{9}$$

$$\therefore V = \pi\psi_d \frac{{m_n}^3 z_1}{\cos\beta} h^*_{an}\left[2h^*_{an} + \frac{(1+i)z_1}{\cos\beta}\right] \tag{10}$$

In this paper, the helical gear volume V is calculated by using Equation (10). From this equation, it can be seen that there are six independent parameters involved in calculating the total volume V; number of teeth of pinion z_1, normal module m_n, gear ratio i, helix angle β, tooth thickness factor ψ_d and addendum factor h^*_{an}.

THE FORMULA FOR CONTACT RATIO

The contact ratio of helical gear ε_r is the sum of the transverse contact ratio ε_a and longitudinal contact ratio ε_β. The formula is as shown in Equation (11).

$$\varepsilon_r = \varepsilon_\alpha + \varepsilon_\beta \tag{11}$$

The transverse contact ratio and longitudinal contact ratio are shown in Equation (12)-(13).

$$\varepsilon_\alpha = \frac{1}{2\pi}[z_1(\tan\alpha_{at1} - \tan\alpha_t) + z_2(\tan\alpha_{at2} - \tan\alpha_t)] \tag{12}$$

$$\varepsilon_\beta = \frac{b\sin\beta}{\pi m_n} \tag{13}$$

By substituting Equation (12)-(13) into Equation (11), the formula of contact ratio ε_r becomes as shown in Equation (14).

$$\varepsilon_r = \frac{z_1}{2\pi}(\tan\alpha_{at1} - \tan\alpha_t) + \frac{iz_1}{2\pi}(\tan\alpha_{at2} - \tan\alpha_t) + \frac{b\sin\beta}{\pi m_n} \tag{14}$$

From Equation (14), α_t, α_{t1} and α_{t2} are transverse pressure angle, transverse working pressure angle of pinion and transverse working pressure angle of wheel, respectively. The formula of α_t, α_{t1} and α_{t2} is as shown in Equation (15)-(17).

$$\alpha_t = \tan^{-1}\left(\frac{\tan\alpha_n}{\cos\beta}\right) \tag{15}$$

$$\alpha_{t1} = \cos^{-1}\left(\frac{d_{b1}}{d_{a1}}\right) \tag{16}$$

$$\alpha_{t2} = \cos^{-1}\left(\frac{d_{b2}}{d_{a2}}\right) \tag{17}$$

From Equation (15)-(17), α_n, d_{b1} and d_{b2} is the normal pressure angle, base diameter of pinion and base diameter of wheel, respectively. The formula of d_{b1} and d_{b2} are as shown in Equation (18)-(19).

$$d_{b1} = d_1\cos\alpha_t \tag{18}$$

$$d_{b2} = d_2\cos\alpha_t \tag{19}$$

By substituting Equation (18)-(19) into Equation (16)-(17), the formula of contact ratio ε_r becomes as shown in Equation (20)-(22).

$$\varepsilon_r = \frac{z_1}{2\pi}\left\{\tan\left[\cos^{-1}\left(\frac{m_n z_1}{\cos\beta}\times\frac{\cos\alpha_t}{\left(\frac{m_n z_1}{\cos\beta}\right)+2h^*_{an}m_n}\right)\right]-\frac{\tan\alpha_n}{\cos\beta}\right\} + \frac{iz_1}{2\pi}\left\{\tan\left[\cos^{-1}\left(i\frac{m_n z_1}{\cos\beta}\times\frac{\cos\alpha_t}{\left(i\frac{m_n z_1}{\cos\beta}\right)+2h^*_{an}m_n}\right)\right]-\frac{\tan\alpha_n}{\cos\beta}\right\}+\frac{1}{\pi}z_1\psi_d\tan\beta \tag{20}$$

$$\varepsilon_r = \frac{z_1}{2\pi}\left\{\tan\left[\cos^{-1}\left(\frac{z_1\cos\alpha_t}{z_1+2h^*_{an}\cos\beta}\right)\right]-\frac{\tan\alpha_n}{\cos\beta}\right\}+\frac{iz_1}{2\pi}\left\{\tan\left[\cos^{-1}\left(\frac{iz_1\cos\alpha_t}{iz_1+2h^*_{an}\cos\beta}\right)\right]-\frac{\tan\alpha_n}{\cos\beta}\right\}+\frac{1}{\pi}z_1\psi_d\tan\beta \tag{21}$$

$$\varepsilon_r = \frac{z_1}{2\pi}\left\{\tan\left[\cos^{-1}\left(\frac{z_1\cos\beta\sqrt{\cos^2\beta+\tan^2\alpha_n}}{(z_1+2h^*_{an}\cos\beta)(\cos^2\beta+\tan^2\alpha_n)}\right)\right]-\frac{\tan\alpha_n}{\cos\beta}\right\} + \frac{iz_1}{2\pi}\left\{\tan\left[\cos^{-1}\left(\frac{iz_1\cos\beta\sqrt{\cos^2\beta+\tan^2\alpha_n}}{(iz_1+2h^*_{an}\cos\beta)(\cos^2\beta+\tan^2\alpha_n)}\right)\right]-\frac{\tan\alpha_n}{\cos\beta}\right\}+\frac{1}{\pi}z_1\psi_d\tan\beta \tag{22}$$

The contact ratio of helical gear ε_r is calculated by using Equation (22). From this equation, it can be seen that there are seven independent parameters involved in calculating the contact ratio ε_r; normal module m_n, number of teeth of pinion z_1, gear ratio i, helix angle β, tooth thickness factor ψ_d, normal pressure angle α_n, addendum factor h^*_{an}.In this paper, a single-stage helical gear train of a local EMU is taken as a sample to examine the relationship between the aforementioned independent variables (m_n, z_1, i, β, ψ_d, α_n, h^*_{an}) towards the total gear volume V and contact ratio ε_r. The parameters of the gear are shown in Table.

TABLE 1. Parameters of a single-stage helical gear train of local EMU

No	Parameters	Symbol	Value
1	Module		6
2	Number of Teeth of Pinion		24
3	Gear Ratio		3.875
4	Helix Angle		20°
5	Tooth Thickness Factor		0.8
6	Normal Pressure Angle		26
7	Addendum Factor		1.08

RESULTS AND DISCUSSION

CORRELATION BETWEEN INDEPENDENT VARIABLES WITH GEAR VOLUME AND CONTACT RATIO

Only one parameter will be assigned as a variable to verify its influence, while other parameters will be set as constant. Graphs of volume and contact ratio against gear design parameters are shown in Figure 1. Seven graphs are plotted since there are seven independent variables. The x-axis, left, and right sides of the y-axis represent each independent variable, gear volume and contact ratio, respectively. The blue dots and orange dots represent the gear volume value and contact ratio, respectively.

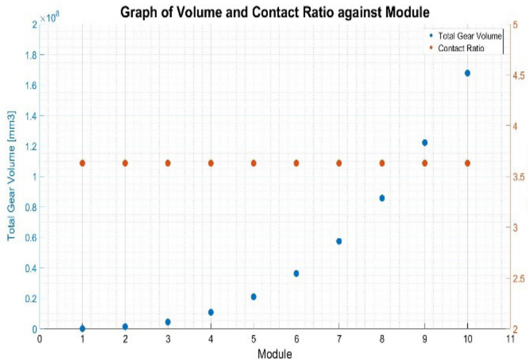
From Figure 1, it is found that the normal module , number of teeth of pinion z_1 and gear ratio i have a positive correlation with the gear volume because as the value of these parameters increases, the gear volume is also increased. On the other hand, helix angle β, tooth thickness factor ψ_d, normal pressure angle α_n, addendum factor h^*_{an} do not correlate with gear volume because the value of gear volume did not change even though these parameters increased. Therefore, it can be said that the value of helical gear volume is greatly influenced by normal module m_n, number of teeth of pinion z_1 and gear ratio i. When optimizing the volume of helical gear, it is sufficient to consider only module , number of teeth of pinion and gear ratio as design variables, while other parameters can be omitted.

On the other hand, the number of teeth of pinion z_1, helix angle β, tooth thickness factor ψ_d and addendum factor h^*_{an} have a positive correlation with the contact ratio. This is because as the value of these parameters increased, the contact ratio also increased. Meanwhile, the normal pressure angle α_n has a negative correlation with the contact ratio since the value of the contact ratio decreased when the value of the normal pressure angle α_n decreased. On the other hand, module m_n and gear ratio i has no correlation with contact ratio as the value of contact ratio did not change even though the value of these parameters have been increased. Hence, it can be said that the value of the contact ratio of helical gear is greatly influenced by number of teeth of pinion z_1, helix angle β, tooth thickness factor ψ_d and addendum factor h^*_{an}. When optimizing the contact ratio of helical gear, it is sufficient to consider the only number of teeth of pinion z_1, helix angle β, tooth thickness factor ψ_d, addendum factor h^*_{an} and normal pressure angle α_n as design variables, while other parameters can be neglected.

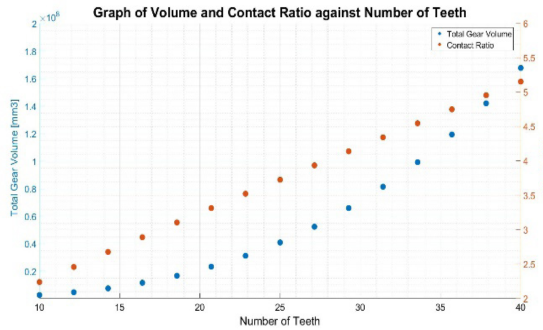
Table 2 summarises the correlation between gear design parameters and dependent variables. When constructing a multi-objective gear design optimization model which optimizes both gear

TABLE 2. Correlation between independent variables with gear volume and contact ratio

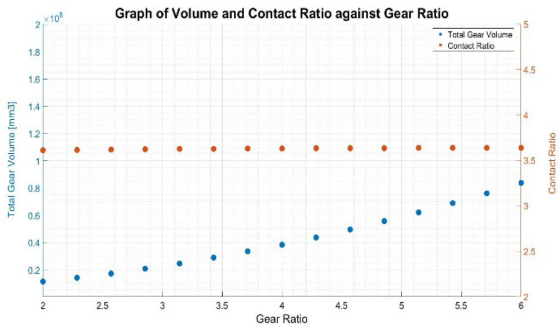
No.	Independent Variables	Symbol	Correlation with Gear Volume	Correlation with Contact Ratio
1	Module	m_n	Positive	-
2	Number of Teeth of Pinion	z_1	Positive	Positive
3	Gear Ratio	i		
4	Helix Angle	β	-	Positive
5	Tooth Thickness Factor	ψ_d	-	Positive
6	Normal Pressure Angle	α_n	-	Negative
7	Addendum Factor	$h*an$	-	Positive



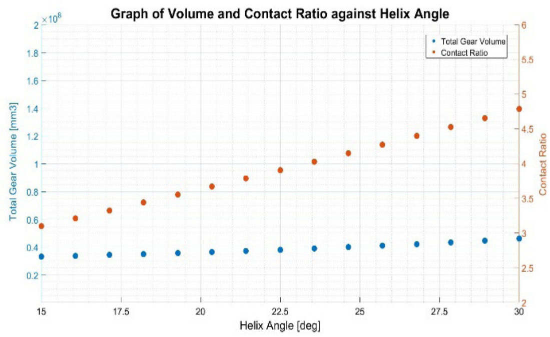
(a) Module



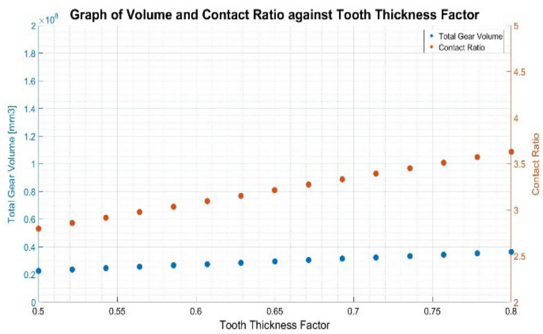
(b) Number of Teeth of Pinion



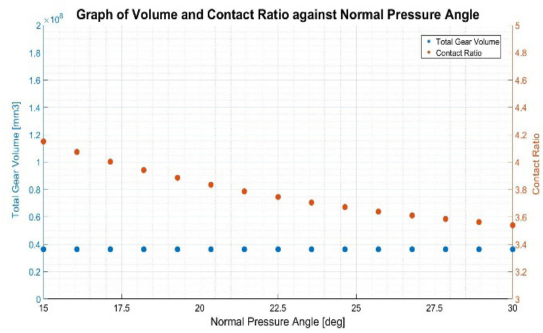
(c) Gear Ratio



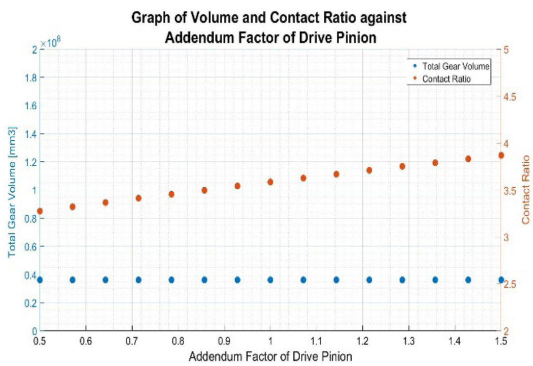
(d) Helix Angle



(e) Tooth Thickness Factor



(f) Normal Pressure Angle



(g) Addendum Factor

FIGURE 1. Temporal analysis of TA muscle according to pedal task

volume and contact ratio simultaneously, all of these parameters must be considered to achieve an optimal solution.

CONCLUSION

A single-stage helical gear train of a local EMU was taken as a sample to examine the correlation between gear design parameters towards the helical gear volume and contact ratio. The gear volume and contact ratio calculations are done based on ISO 6336. Graphs of volume and contact ratio against gear design parameters were plotted. Then from the graphs, the correlation between gear design parameters towards gear volume and contact ratio was determined.

When examining the correlation between gear design parameters and gear volume, it is found that normal module , number of teeth of pinion and gear ratio have a positive correlation with the gear volume, while helix angle , tooth thickness factor , normal pressure angle , addendum factor do not correlate with gear volume. Hence, it can be said that the value of helical gear volume is greatly influenced by normal module , number of teeth of pinion and gear ratio . When optimizing the volume of helical gear, it is sufficient to consider only module , number of teeth of pinion and gear ratio as design variables, while other parameters can be omitted.

On the contrary, when examining the correlation between gear design parameters and contact ratio, it is found that the number of teeth of pinion , helix angle , tooth thickness factor and addendum factor have a positive correlation with the contact ratio. Meanwhile, the normal pressure angle has a negative correlation with the contact ratio while the module and gear ratio does not correlate with the contact ratio. Therefore, it can be said that the value of the contact ratio of helical gear is greatly influenced by number of teeth of pinion , helix angle , tooth thickness factor and addendum factor . When optimizing the contact ratio of helical gear, it is sufficient only to consider number of teeth of pinion , helix angle , tooth thickness factor , addendum factor and normal pressure angle as design variables, while other parameters can be neglected.

Finally, when constructing a multi-objective gear design optimization model that optimizes both gear volume and contact ratio simultaneously, all of these parameters must be considered to achieve an optimal solution.

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DECLARATION OF COMPETING INTEREST

None

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