WEP Warwick Evans Publishing

International Journal of Mechanical and Electrical Engineering

ISSN: 3005-9615 (Print), ISSN: 3005-7132 (Online) | Volume 3, Number 3, Year 2024
DOI: https://doi.org/10.62051/ijmee.v3n3.16
Journal homepage: https://wepub.org/index.php/IJMEE/index

Dynamic Analysis of Helical Gears Considering Friction

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ABSTRACT

The use of high contact ratio (HCR) gears is an effective method to reduce single tooth loads, as well as vibration and noise, making the application of helical gears increasingly widespread. Although the presence of helix angles can cause significant relative sliding, the presence of more teeth in contact makes HCR gears more sensitive to surface quality. The existing literature mainly investigates geometric optimization, load distribution, or efficiency calculation of helical gears. The research on the influence of rough surfaces on the dynamic performance of helical gears is very limited. To this end, a multi degree of freedom mathematical model of static transmission error was established based on the 2-DOF system. The relative sliding friction was studied using different friction coefficients, and the influence of different friction coefficients on the gear system was analyzed in detail. The numerical results indicate that surface roughness has a relatively small impact on the dynamic transmission error of the system, but has a significant impact on the motion and friction in the direction of departure.

KEYWORDS

Helical Gear; Friction; Dynamic Response.

1. INTRODUCTION

Gears are widely used in numerous types of industrial machinery, automobiles, ships, locomotives, airplanes, and other machines. The competitive industrial marketplace and customers demand gear products with excellent reliability, high power-weight ratio, small size, and low noise. The helical gear has a transverse contact ratio. Double and triple teeth pairs contact alternatively during the mesh,, a tooth will never be under the entire load, thus the bearing capacity and the comprehensive mesh stiffness are greatly improved [1]. Another advantage of the helical gear system is that movement is transferred more uniformly and stable which results in lower vibration, shock, and noise[2] . In other words, employing the helical gear is a significant method of improving transmission performance. The helical gear has been widely used in high-speed and heavy-load fields [3-5]

An important source of power loss in gear transmission is sliding friction, which can affect the external excitation of system dynamics[6, 7]. Using variable friction models with different friction coefficients to explore their effects on wear, efficiency, and dynamics.

2. MODEL AND METHODOLOGY

2.1. Helical Gear Meshing Schematic

The three-dimensional model of the helical gear pair is shown in picture below (a), and the two-dimensional projection on the plane is shown in picture below Fig 1. To facilitate analysis, the

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transverse displacement of the gear center is enlarged. In the figure, OLA is the meshing line of the gear(δ_L); d is the center distance, a_τ is the meshing angle, and γ is the relative position angle at any time. O_1 and O_2 are the initial gear center positions of gear p and gear g, respectively. d_0 is the initial center distance.; a_0 is the initial pressure angle, we assume it is 20°.

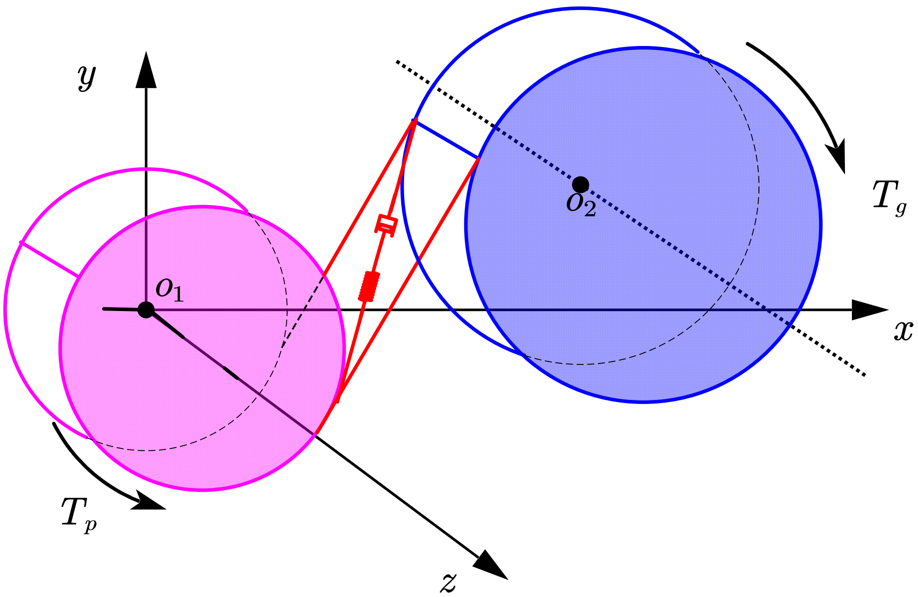

Fig 1. The three-dimensional model

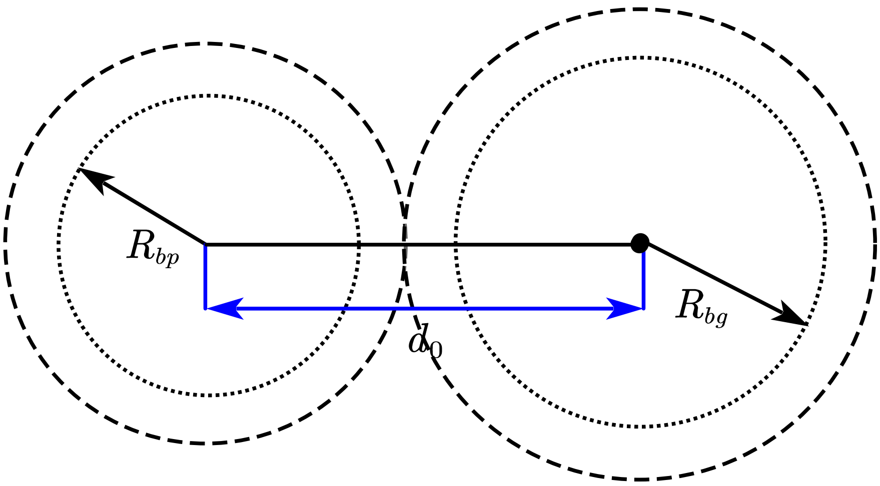

Fig 2. the two-dimensional projection on the plane

2.2. Pfriction Model

Do not number your paper: The m is the Gear quality, The T is Torsional moment which applied to a gear, The β is the helix angle at which the two gears mesh; The pressure angle on the meshing surface is a_τ; b_τ is the lateral relative position angle of the gear pair.

In the figure, we note that the initial center distance is d_0

$$d_0 = R_p + R_g \tag{1}$$

From the geometric relation we can get this relative displacement of the gear system δ_L along the line-of-action (LOA) and the dynamic transmission error δ along LOA can be expressed as δ_L is represent as Eq. (2) .

$$\delta_L = \left(\theta_1 R_1 - \theta_2 R_2\right)\cos\beta \tag{2}$$

The friction between a pair of meshing teeth is often manifested as sliding friction and rolling friction. Sliding friction refers to the relative sliding between two mating surfaces, and the friction coefficient usually refers to the sliding friction coefficient.

Over the past decades, the study of gears has shown interest in the measurement and calculation of the coefficient of friction, which is a combination of tooth profile, lubricant condition, surface finish and operating parameters.

Under the daily working conditions of this combination of these factors, the lubrication of gears enters a very common lubrication state, that is, the EHL state

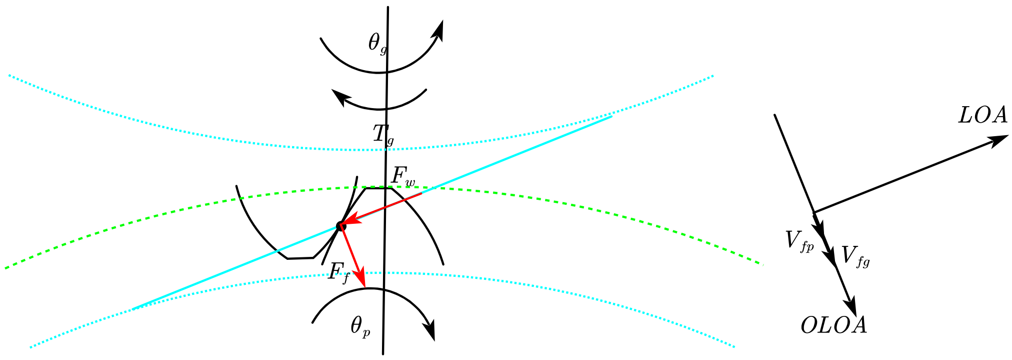

Fig 3. Friction force diagram

To calculate friction, As the meshing progresses, the friction arm of the system is constantly changing, which we quote the friction arm equation $R_f^{p,g}$

$$R_f^p = \left(R_p \sin\left(a_n\right) + l_p\right)\cos\beta \tag{3}$$

$$R_f^g = \left(R_g \sin\left(a_n\right) - l_p\right)\cos\beta \tag{4}$$

Among them

$$l_p = \sqrt{R_{bg}^2 + \rho_g^2}\,\sin\left(a_n\right) - \rho_g \tag{5}$$

$$\rho_g = \frac{\sqrt{R_{og}^2 - R_{bg}^2} - \varphi_{mg} \bmod\left(R_{bg}\theta_g, Z_{act}\right)}{\cos\left(\beta\right)} \tag{6}$$

We define the pinion speed as V_p and the speed of the large gear as V_g, and V_p is always greater than V_g according to the inversely proportional relationship between the speed and the radius, so we can get Z_{act} is the degree of tooth surface overlap; we can get the friction calculation formula:

$$F_f^{p,g} = \mu F_m^{p,g} \tag{7}$$

where μ is the coefficient of friction; F_m is the meshing force received when the gear meshes; V_p, V_g is the sliding speed of the pinion relative to the large gear which can be calculated as:

$$V_{fp} = R_f^p \dot{\theta}_p \quad (8)$$

$$V_{fg} = R_f^g \dot{\theta}_g \quad (9)$$

where W is the unit normal load; where Z is the tooth width of the gear.

$$T_f^{p,g} = F_f R_f^{p,g} \quad (10)$$

The gear dynamics equation is:

$$m\ddot{x} + c\dot{x} + kx = F \quad (11)$$

Therefore, we can derive the dynamic equation of the system as

$$J_1\ddot{\theta}_1 + R_{bp}F_\theta^2 - T_{f\theta}^1 = T_1 \quad (12)$$

$$J_2\ddot{\theta}_2 + R_{bg}F_\theta^2 - T_{f\theta}^2 = -T_2 \quad (13)$$

3. LITERATURE REFERENCES

The data used in our simulation is shown in the following gear parameter table,

Table 1. Three Scheme comparing

Gear Parameters	Pinion		Gear
Number of teeth	23		47
Mass(kg)	0.21		0.37
Module(mm)		2.5	
Pressure angle(°)		20	
Input torque(N·m)		50	
Helix angle β		8.0	
Tooth top height factor ha		1.0	

We use a friction coefficient of 0.4 and a torsional torque of TP=50N/m to discuss the comparison of the presence or absence of friction in the helical gear system. To this end, we have created a dimensionless meshing period time domain graph of the helical gear system with or without friction at a speed of 1000r/min, as shown below:

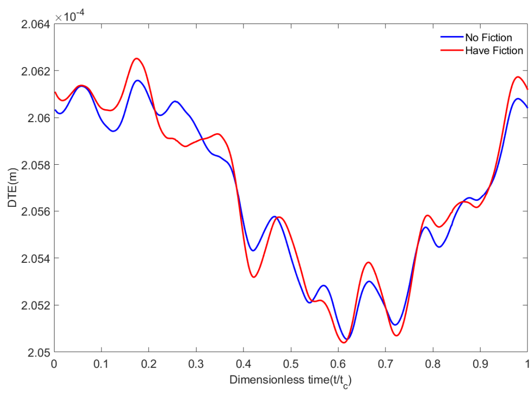

Fig 4. Friction comparison chart

We can clearly see from the graph that friction has a significant impact on the gear system during a meshing cycle, especially since friction does not only have the same effect on the system (increasing or decreasing errors). We can see from the graph that the influence of friction on transmission error varies with the change of meshing period, and the increasing and decreasing effects always appear together.

4. CONCLUSION

(1) The HCR gear has a greater relative sliding velocity than the NCR gear, and its transmission performance is more sensitive to surface roughness because more teeth are in contact at the same time. Through the introduction of fractal theory to the characterization of a rough surface, an improved MDOF model to investigate the influence of surface roughness in terms of STE is presented. A particular lubrication condition, EHL condition, with a time-varying friction coefficient is employed. Geometrical analysis shows that surface roughness has a large effect on the magnitude of the friction coefficient in the EHL condition, and numerical results indicate a small influence of roughness on DTE.

(2) The response of the gear system under friction excitation exhibits complex nonlinear dynamic characteristics. With an increasing friction coefficient l, the gear system underwent nT-periodic, quasi-periodic, and chaotic motions, succes-sively. The coupled lateral-torsional vibration had non-linear characteristics that are more complicated than that of the system with pure torsional motion due to the nonlinear effects induced by bearing support. In addition, the extent of amplification of the response in vertical displacements were always greater than that in horizontal displacements, which is mainly caused by the gravitational effect.

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