Simulation of Meshing of Herringbone Double Circular-Arc Helical Gears using Multibody Dynamic Analysis Software

C K Lee¹, C L Hsieh¹, J C Ruan² and C Y Wang²

1 Department of Industrial Engineering and Management, Cheng Shiu University, 840 Chengcing Rd., Kaohsiung, Taiwan (R.O.C.)
2 Department of Mechanical Engineering, Cheng Shiu University, 840 Chengcing Rd., Kaohsiung, Taiwan (R.O.C.)
E-mail: ckle@gcloud.csu.edu.tw

Abstract. This paper aims to complete the simulation of meshing of a pair of herringbone double circular-arc helical gears by using the multibody dynamic analysis software, SolidWorks motion. Three types of edge modifications, including no edge modification, chamfered edge modification, and rounded edge modification, are applied respectively to the pair of gears. The output angular velocities and contact regions corresponding with the three types of edge modifications are analyzed. Moreover, whether the center distance assembly error affects the output angular velocity and contact regions are analyzed. The results obtained by the simulation of meshing show that rounded edge modification is better than chamfered edge modification, and chamfered edge modification is better than no edge modification. The center distance assembly error has no effect on the output angular velocity and has little effect on the contact regions. Therefore, double circular-arc helical gears are not sensitive to center distance assembly errors.

1. Introduction

Compared with involute gears and single circular-arc helical gears, double circular-arc helical gears have much higher contact strength and tooth bending strength, especially the stepped double circular-arc helical gears. The main disadvantage of helical gears is that the axial component force will be generated when the gears mesh. The axial force is caused by the helix angle. The larger the helix angle, the greater the axial force. The application of herringbone gear is the way to eliminate the axial component of force. The herringbone double circular-arc helical gears have the advantages of both double circular-arc helical gears and herringbone gears. Litvin and Lu [1] investigated the generation, geometry, meshing, and contact of double circular-arc helical gears. Litvin and Lu [2] modified the geometry of double circular-arc helical gears for absorption of transmission errors caused by misalignment. Lu et al. [3] investigated the conditions of load share under a load and determined the real contact ratio for aligned and misaligned double circular-arc helical gears. Wang and Chen [4] applied an adjustable bearing to compensate for offset and angular misalignment for double circular-arc helical gears. Litvin et al. [5] proposed a new version of Novikov-Wildhaber helical gears with lower noise and vibration caused by errors of alignment, the possibility of grinding and application of hardened materials, and lower stresses. Yang [6] proposed stepped triple circular-arc gears, of which contact stress was investigated via finite element analysis. Zhang et al. [7] proposed modified double circular-arc helical gears by tooth end relief with helix to reduce transmission errors and stresses. Wang [8] designed a planetary gear reducer with double circular-arc helical gear. Qu et al. [9] investigated the load sharing of double circular-arc helical gears based on finite element analysis. Li et
al. [10] proposed a new theoretical approach for the analysis of the impact of the double circular arc gear meshing. Gao et al. [11] proposed a novel point contact gear drive with pinion circular arc teeth and gear parabolic curve teeth, of which the gear stress distribution and the contact deformation were investigated by using finite element analysis. Liang et al. [12] studied the generation and meshing of a double helical gear transmission with curve element constructed tooth pairs. At present, no literature has been found to perform the simulation of meshing and analyze the influence of edge modification on the output angular velocity for a pair of herringbone double circular-arc helical gears by using multibody dynamic software.

2. Mathematical modeling of double circular-arc helical gear

Based on the coordinate transformation theory and the theory of gearing, the mathematical model of double circular-arc helical gear can be created. According to standard GB/T12759-1991, the profile of the basic rack of double circular-arc helical gear is as shown in figure 1. The mathematical modeling of double circular-arc helical gear has three steps.

Step 1: Placing a coordinate system $S_n(x_n, y_n, z_n)$ to the profile and representing the mathematical model of the profile in $S_n(x_n, y_n, z_n)$ by equation (1).

$$r_n^{(i)}(\theta) = \begin{bmatrix} x_n^{(i)}(\theta) & y_n^{(i)}(\theta) & 0 & 1 \end{bmatrix}^T$$

(1)

Step 2: Placing a coordinate system $S_i(x_i, y_i, z_i)$ to the basic rack and representing the mathematical model of the basic rack in $S_i(x_i, y_i, z_i)$ by equation (2).

$$r_i^{(i)}(\theta, u) = M_{m_n}(u) r_n^{(i)}(\theta) = R_y(-\beta) T_z(u) T_r^{(i)}(\theta)$$

(2)

Step 3: Placing a coordinate system $S_z(x_z, y_z, z_z)$ to the double circular-arc helical gear and representing the mathematical model of the double circular-arc helical gear in $S_z(x_z, y_z, z_z)$ by equation (3). The second equation in equation (3) is obtained by using the theory of gearing.

$$\begin{align*}
\begin{bmatrix} r_z^{(i)}(\theta, u, \phi) \\ u = -\rho \phi \csc \beta + \cot \beta \left[ x_n^{(i)}(\theta) + \frac{y_n^{(i)}(\theta)^2}{x_n^{(i)}(\theta)} \right] 
\end{bmatrix} = M_{z_1}(\phi) r_z^{(i)}(\theta, u)
\end{align*}$$

(3)
3. CAD modeling of herringbone double circular-arc helical gear

To complete the CAD modeling of a pair of herringbone double circular-arc helical gears, three kinds of software are used one by one. Firstly, the mathematical software, Mathematica, is applied to calculate the coordinates of cross-section curves of surfaces of one single tooth groove. Secondly, the CAD system, Pro/Engineer, is applied to create the surfaces of one single tooth groove. In Pro/Engineer, the cross-section curves are plotted using the datum curve function. The surfaces of one single tooth groove are created using the boundary blend function. The surfaces are exported to SolidWorks using an IGES format file. Thirdly, the CAD system, SolidWorks, is applied to create the solid model of a pair of herringbone double circular-arc helical gears. In SolidWorks, the excess surfaces are trimmed using the surface trimming function. The discrete surfaces are combined into one single tooth groove surface using the surface knitting function. The whole tooth groove surfaces are created using the surface copying function. The gear blank cylinder is created using the extrusion function. All teeth are formed using the surface cutout function. The solid model of herringbone double circular-arc helical gear is obtained using the solid mirroring function. Figure 2 shows the obtained solid model of a pair of herringbone double circular-arc helical gears. The small gear has 8 teeth. The big gear has 17 teeth. The gear module is 20 mm. The helix angle is 15 degrees. The pressure angle is 24 degrees.

![Figure 2. The solid model of a pair of herringbone double circular-arc helical gears.](image)

4. Settings for the simulation of meshing

The simulation of meshing of the pair of herringbone double circular-arc helical gears is completed by the multibody dynamic software, SolidWorks motion, which uses ADAMS as its solver. The settings for the simulation are as follows: In the contact property manager, the contact type is set to solid bodies, the materials are set to steel (greasy), and the contact force is set to impact model. The time function curve of torque applied to the big gear shaft hole is set to STEP5(time,0,0,1,5000) N-mm, indicating that the torque is gradually loaded from 0 to 5000 N-mm in the time range from 0 to 1 second, and then keeps 5000N unchanged. The time function curve of angular velocity applied to the small gear shaft hole is set to STEP5(time,1,0,2,3) deg/s, indicating that the angular velocity is gradually loaded from 0 to 3 deg/s in the time range from 1 to 2 seconds, and then remains 3 deg/s unchanged. The 3D contact resolution is set to using precise contact. The accuracy is set to 0.00001. The integrator type is set to SI2_GSTIFF, which provides better accuracy of velocities and accelerations.

5. Simulation of meshing

5.1. Original design

For the original design, the edges on the end faces are not modified, as shown in Figure 3. The output angular velocity for the design is shown in Figure 4. The angular velocity has a high variation at 11.13
seconds. At this moment, the contact regions on the no edge modification small gear are shown in Figure 5. Edge contact phenomenon can be observed at this moment.

\[\text{Figure 3. The edges on the end faces are not modified.}\]

\[\text{Figure 4. The output angular velocity.}\]

\[\text{Figure 5. The contact regions on the no edge modification small gear.}\]

5.2. Improved design 1
For the improved design 1, the edges on the end faces are chamfered, as shown in Figure 6. The output angular velocity for the design is shown in Figure 7. The angular velocity has a moderate variation at 11.32 seconds. At this moment, the contact regions on the chamfered edge modification small gear are shown in Figure 8. Edge contact phenomenon can also be observed at this moment.

\[\text{Figure 6. The edges on the end faces are chamfered.}\]

\[\text{Figure 7. The output angular velocity.}\]

\[\text{Figure 8. The contact regions on the chamfered edge modification small gear.}\]
5.3. Improved design 2
For the improved design 2, the edges on the end faces are rounded, as shown in Figure 9. The output angular velocity for the design is shown in Figure 10. The angular velocity has a low variation at 11.48 seconds. At this moment, the contact regions on the rounded edge modification small gear are shown in Figure 11. Edge contact phenomenon can also be observed at this moment.

![Figure 9](image1.png) **Figure 9.** The edges on the end faces are rounded.

![Figure 10](image2.png) **Figure 10.** The output angular velocity.

![Figure 11](image3.png) **Figure 11.** The contact regions on the rounded edge modification small gear.

5.4. Influence of center distance assembly error
To analyze the influence of center distance assembly error, a center distance assembly error of 0.05 mm is applied intentionally to the gear shafts, as shown in Figure 12. The pair of gears with rounded edge modification is used as the object of analysis. The output angular velocity for the pair of gears is shown in Figure 13. The contact regions on the small gear are shown in Figure 14. It shows that the center distance assembly error has no effect on the output angular velocity and has little effect on the contact regions.

![Figure 12](image4.png) **Figure 12.** A center distance assembly error of 0.05 mm is applied to the gear shafts intentionally.

![Figure 13](image5.png) **Figure 13.** The output angular velocity.
Figure 14. The contact resins on the small gear having rounded edge modification and center
distance assembly error.

6. Conclusion
This paper has completed the simulation of meshing of a pair of herringbone double circular-arc
helical gears by using the multibody dynamic analysis software, SolidWorks motion. The gear pair
with no edge modification is very sensitive to the edge contact phenomenon. By modifying the sharp
edges as chamfered edges, the problem can be improved to a certain extent. By modifying the sharp
edges as rounded edges, the problem can be greatly improved. Besides, the center distance assembly
error does not influence the output angular velocity of the gear pair and has little influence on the
contact regions. Although the edge modification technology can reduce the sensitivity of the gear pair,
it needs to spend extra time to process the sharp edges. Therefore, it is suggested to directly apply the
longitudinal crowning technology to save the extra edge modification time.

References
[1] Litvin F L and Lu J 1993 Math. Comput. Model 18 31-47
[2] Litvin F L and Lu J 1995 Comput. Meth. Appl. Mech. Eng. 127 57-86
[3] Lu J, Litvin F L and Chen J S 1995 Math. Comput. Model 21 13-30
[4] Wang C Y and Chen C K 2001 Proc. Inst. Mech. Eng. C. 215 759-71
[5] Litvin F L, Fuentes A, Gonzalez-Perez I, Carnevali L and Sep T M 2002 Comput. Meth. Appl.
Mech. Eng. 191 5707-40
[6] Yang S C 2009 Mech. Mach. Theory 44 1019-31
[7] Zhang H, Hua L and Han X H 2010 Mech. Mach. Theory 45 46-64
[8] Wang C Y 2012 AMM 152-154 1595-1600
[9] Qu W, Peng X, Zhao N and Guo H 2012 Structural Engineering and Mechanics 43 439-48
[10] Li Y, Wu B and Zhu Lin 2015 The Open Mech. Eng. J. 9 160-7
[11] Gao Y, Chen B, Tan R and Zhang Y 2016 J. Adv. Mech. Des. Syst. Manuf. 10 JAMDSM0009
[12] Liang D, Luo T and Tan R 2018 Math. Probl. Eng. 2018 1539849