Three Dimensional Solid Modeling of Gears using Solidworks Secondary Development

Mingxin Li

Xinxiang Polytechnic College, Xinxiang, Henan, 453000, China

Abstract — We study three dimensional parametric agile design of gears and related software platforms in terms of: parameterization, modularization, and the software being easy to use, expand and maintain. We base our study on SolidWorks platform, aiming to generate 3-D solid models of gears automatically in terms of: i) inputting parameters to improve the modeling speed, ii) build up foundations for structure design, iii) do dynamic simulation, and iv) interference checks and v) use finite element analysis to improve design efficiency. Our results include: i) 3-D accurate description for involute gear is realized, ii) accurate specification of involute gear in SolidWorks has been achieved, iii) parameterized agile modeling system modules of involute cylindrical spur gear, helical gear and internal gear have been developed using SolidWorks as support platform, and iv) success in adopting object oriented technology for involute coordinate equations.

Keywords- solidworks; gear; modeling; secondary development; involute

I. INTRODUCTION

With the development of computer assisted design (CAD) technology, the design of engineering and products is gaining more and more attention. Gear modeling technology based on PRO-E has been widely applied in the fields of ships, machinery, automobiles, electronics, construction, textile and chemical etc. The proportion of CAD technology applied in machinery products is becoming bigger and bigger, therefore, program research in this field has become the focus. The main object of three dimensional CAD system design is three dimensional models, therefore in the specific engineering product design, it needs to take actual three dimensional structures as basis and optimizes various models. PRO-E software is with great three-dimensional design ability, can build up three-dimensional CAD model and better reflect products processing and manufacturing process as well as time products constructing process. In three dimensional design, it needs to construct CAM and CAD functions from aspects of curve surface design, assembly design, finite element design, sheet metal design and numerical control machining etc, at the same time, it can make simulation for three dimensional model of products to ensure meeting the demands of specific application. Product data management is also an important modeling content needs to be considered in three dimensional CAD design process, which is mainly used to meet the development and design of complicated products. In three dimensional CAD design process, it needs to consider about product quality control and management and ensure various data in product life cycle design process meets the demands of model design, including drawings and technical documents etc. Three dimensional CAD model design needs to make data sharing for CAPP, CAD, CAM and other systems and ensure the products to meet standard design demand and realize safe management of drawings, processing codes, process card and technical material.

II. GEAR MODELING

A. Semi-analytic Method Based on Contact Mechanics

For spherical roller, based on Hertz contact theory [7], loading on roller $Q$ and total amount of elastic deformation $\delta$ between roller and raceway has following relationship:

$$Q = Kn_1 \delta^{1.5} \quad \delta \geq 0,$$

$Kn$ is deformation constant, which has relationship with material and raceway mean curvature, for steel gear:

$$Kn = 3.13 \times 10^5 \left( \delta_i^{1/3} \sum \rho_i^{1/3} + \delta_e^{1/3} \sum \rho_e^{1/3} \right) \quad (1)$$

In which $\sum \rho_i^{1/3}$ and $\sum \rho_e^{1/3}$ are curvature and function of inner and outer raceways respectively. When loading $Q$ is known, the maximum contact stress between roller and raceway can be attained easily with adoption of Hertz theory:

$$P_0 = \frac{3Q}{2\pi ab} \quad (2)$$

In which a,b are the long axis and short axis of contact ellipse and can be attained by calculation of loading $Q$ and curvature and function.

For cylindrical roller with limited length, there is no concise relationship as Hertz formula between loading and amount of elastic deformation between roller and rollaway. Closed analytic solutions are difficult to be attained under most of the situations. At present, calculation adopts similar formula or empirical formula most, for example for steel gear, the empirical formula given by Palmgren[8] is:
\[ \delta = 3.84 \times 10^{-3} \frac{Q^{0.9}}{l^{0.8}} \]  

This formula proves that there is no relationship between amount of elastic deformation and diameter of rollaway; obviously the above formula has certain limitation. The corrected Palmgren formula consider about the diameters of roller and rollaway as well as the concavity of curvature [9]:

\[ \delta = 4.83 \times 10^{-5} \frac{Q^{0.9}}{l^{0.74} D_w^{0.1}} (1 \pm k)^{0.1} \]  

But different from spherical roller, the distribution of contact stress between cylindrical roller and rollaway is relatively complicated, which is not as concise of formula (2) and it needs to adopt numerical method to solve. The most common method adopted at present is to simplify contact in a further way. Assume contact force distributes evenly along roller plain line direction, while the horizontal distribution is based on Hertz:

\[ p_y = p_{0y} \sqrt{1 - \left(\frac{x}{a_y}\right)^2} \]  

In which \( p_{0y} \) is the biggest contact stress on plain line, the width of contacting area can be confirmed after the distribution of contact stress on plain line is confirmed. In this way, two-dimensional contact problem can be changed into one dimension problem, which greatly decreases the difficulty of solution and amount of calculation and adopts simple iterative approach for solution.

For gear with single roller row, such as universal rolling gear and rotary support of four point contact type with single row balls etc, when the external load is known, it can make use of geometrical relationship to list equilibrium equation and attain the contact loading \( Q \) borne by each roller after solution and then attain the maximum contact stress between roller and rollaway. Without loss of generality, assume radial load \( F_r \), axial load \( F_a \) and moment \( M \) function together, inner and outer rings produce relative displacements \( \delta_r, \delta_a \) and \( \theta \). Based on the geometric relationship between ring and roller, amount of elastic transformation and contact load between roller and rollaway expressed by \( \delta_r, \delta_a \) and \( \theta \) can be attained through calculation and solve equilibrium equation.

\[
\begin{align*}
F_r - \sum_{i=1}^{Z} Q_{a\theta} &= 0 \\
F_a - \sum_{i=1}^{Z} Q_{a\theta} &= 0 \\
M_m - \sum_{i=1}^{Z} M_{a\theta} &= 0
\end{align*}
\]  

Three basic unknown quantities \( \delta_r, \delta_a \) and \( \theta \) can be attained and then attain contact loading \( Q \) borne by each roller and corresponding maximum contact stress. Equation (6) is relatively complicated non-linear equation, which can be solved with Newton iteration method.

For gear with double-row roller, such as matched bearing, shaft bearing, double row ball slewing reducer as well as three row roller slewing bearing (actually there is only two-row roller bearing). Basic unknown quantities are still relatively displacement of, \( \delta_a \) and \( \theta \). After correction of geometric relationship, the attained equilibrium equation is also with the form of equation (6), which can be solved with the same method.

B. Finite Element Method

As a general mechanical calculation method, finite element method has been applied in gear analysis and optimization, but due to the particularity of gear structure, this method also faces difficulties that can’t be solved in analyzing and optimizing gear performance:

(1) Great difficulty in accurate calculation, under external loading, no matter for spherical roller or cylindrical roller, the contact area between roller and rollaway is very finite in the overall size of gear, which has brought about certain difficulty to finite element mesh and accurate calculation of contact stress. It must be meshed carefully (picture 1) and study grid convergence to confirm the accuracy of results.

(2) Great difficulty in establishing calculation model, generally, multiple contacts exist between roller and rollaway; in addition, boundary conditions and loadings need to add auxiliary means to implement under the function of moment; moreover, the constraint conditions of roller also need to be considered carefully, all these factors have brought about certain difficulty to establish accurate calculation model.

(3) A large number of calculation and low efficiency, due to belong to classic non-linear problem, a large number of calculation is needed to solve the contact stress between roller and rollaway with finite element method, especially for structures with complicated contact relationship with slewing bearing etc, it needs more time for calculation and it is difficult to satisfy design and analysis demand.
Therefore, considering about the speed and accuracy demand of engineering application, finite element method is more suitable to solve problem with small amount of calculation and simple model, such as research on contact relationship between single roller and rollaway, stiffness calculation of shafting and calculation of sliding bearing oil film pressure etc, but it is not suitable for the analysis and calculation of overall performance of gear and also not suitable for making fast iterative optimization for gear parameter.

C. Dynamic Load Capacity and Dynamic Load Curve

At present, dynamic load capacity and dynamic load curve are calculated based on Lundberg—Palmgren fatigue life theory [6]. When radial load Fr, axial load Fa and moment M are known, the load Pi(i=1,2,3…Z) function on each roller can be calculated based on above calculation method of rolling element load distribution. Equivalent rolling load of inner rollaway is:

\[ L_{i0} = \left( \frac{Q_{ei}}{Q_{e}} \right)^{e} \]  

Because the fatigue damages of this rollaway are independent events. Based on multiplication rule, the failure probability of the whole slewing bearing equals to the product of each failure probability, such as for single row four point contact ball type rotary support, when fatigue damage happens to any rollaway of these four rollaway, it can be regarded that fatigue damage happens to the whole gear; and then the fatigue life of the whole set slewing bearing can be calculated based on the fatigue life of each rollaway.

\[ \frac{1}{L_{10b}} = \left( \frac{1}{L_{101}} \right)^{e} + \left( \frac{1}{L_{102}} \right)^{e} + \left( \frac{1}{L_{103}} \right)^{e} + \ldots \]  

III. GEAR MODEL OPTIMIZATION OF PRO/E TOOL

A. Maximum Model Practice of Tooth Height Model Practice

At present, dynamic load capacity and dynamic load curve are calculated based on Lundberg—Palmgren fatigue life theory [6]. When radial load Fr, axial load Fa and moment M are known, the load Pi(i=1,2,3…Z) function on each roller can be calculated based on above calculation method of rolling element load distribution. Equivalent rolling load of inner rollaway is:

\[ Q_{ei} = \left[ \frac{1}{Z} \sum_{i=1}^{Z} P_{i}^{e} \right]^{1/3} \]  

Equivalent rolling load of outer rollaway is:

\[ Q_{eo} = \left[ \frac{1}{Z} \sum_{i=1}^{Z} P_{o}^{e} \right]^{1/3} \]  

For ball support, dynamic capacity (that is the corresponding load when life is one million turns) is:

\[ Q_{c} = A \left( \frac{2f}{2f-1} \right)^{0.41} \left( \frac{1 + \gamma}{1 + \gamma_{3}} \right)^{0.5} \left( \frac{D_{c}}{D_{w}} \right)^{0.5} F(D_{w})Z^{0.3} \]  

In the formula, A is constant, \( f = \pi / Dw \), r is groove curvature radius and D is diameter of roller. And then the rating life of single rollaway is:

\[ L_{10} = \left( \frac{Q_{ei}}{Q_{c}} \right)^{e} \]  

2) Initial value estimation based on ISO formula
Estimate the initial value for gear modification based on ISO formula and then make parameters division for established model. Gear control process can start from master gear and passive gear and calculate maximum practice, minimum practice and gear loading. If make calculation of shape modification from the top of the gear, and then the total model modification is:

\[ \Delta_{\text{max}} = \frac{K \cdot F_t}{\varepsilon_a \cdot C \cdot b} \]  (14)

In the formula, \( K_d \) is coefficient of work condition
\( F_t \) is peripheral force N, \( b \) is working tooth width, mm;
\( \varepsilon_a \) is transverse contact ratio and \( C_y \) is composite stiffness N/(mm·μm)

For external gear of tooth bar cutter machining with 20°profile angle, its single tooth stiffness \( C \) makes GUI module calculation of MATLAB as following:

When unit load \( F_t \cdot \frac{K}{b} \) is bigger or equal to 100N/mm;

\[ C' = 0.8 \cdot C_{th} \cdot C_{R} \cdot \cos \beta \]  (15)

When unit load \( F_t \cdot \frac{K}{b} \) is smaller than 100N/mm;

\[ C' = \frac{0.8 \cdot C_{th} \cdot C_{R} \cdot \cos \beta \cdot (F_t \cdot \frac{K}{b})}{b} \cdot \frac{100}{100} \]  (16)

In the formula, \( C_{th} \) is the coefficient of wheel structure; \( C_R \) is theoretical single tooth stiffness, N/(mm·μm), this is the theory flexibility of q for a pair of gears.

GUI modular calculation formula of MATLAB of q is:

\[ q = 0.04723 + 0.15524 + 0.03275 - 0.00633 \cdot \frac{0.1165}{\varepsilon_a} - 0.00192 \cdot \frac{0.2418}{\varepsilon_a} + 0.00529 \cdot \frac{0.00183}{\varepsilon_a} \]  (17)

The unit of q is mm·μm/N.

In the formula, \( z_{v1} \) and \( z_{v2} \) are virtual number of teeth of small gear and big gear, \( z_{v1} = z_{v2} \cdot \cos \beta \cdot \cos \beta \cdot \beta \); \( \beta_b \) is base helix angle.

Mesh stiffness \( C_y = C \cdot (0.25 + 0.75 \varepsilon_a) \)

3) Initial value estimation based on H.Sigg formula

It can be learnt from model modification tooth profile tolerance zone recommended by H.Sigg as well as the GUI module calculation formula of MATLAB that the total model modification of straight gear is:

\[ \Delta_{\text{max}} = (4 + 0.05 \cdot \frac{F_t}{b}) \pm 4 \]  (18)

\[ \Delta_{\text{max}} = (11.5 + 0.05 \cdot \frac{F_t}{b}) \pm 3.5 \]  (19)

Total model modification of helical gear is:

\[ \Delta_{\text{max}} = (4 + 0.04 \cdot \frac{F_t}{b}) \pm 4 \]  (20)

\[ \Delta_{\text{max}} = (9 + 0.04 \cdot \frac{F_t}{b}) \pm 3.5 \]  (21)

\( \Delta_{\text{max}1} \) and \( \Delta_{\text{max}2} \) are total model modifications from the top of active and passive gear, μm.

4) Initial value estimation based on Rolls-Royce company formula

Based on model modification formula of Rolls-Royce Company, the total model modification of gear top and gear bottom of straight gear is:

\[ \Delta_{\text{max}1} = \Delta_{\text{max}2} = 20 + \frac{0.042 \cdot F_t}{(Y_b \cdot \frac{b}{b}) \min b} \]  (22)

Total model modification of gear top and gear bottom of helical gear is:

\[ \Delta_{\text{max}1} = \Delta_{\text{max}2} = 18 + \frac{0.036 \cdot F_t}{(Y_b \cdot \frac{b}{b}) \min \cos \beta \cdot b} \]  (23)

In which, \( F_t \) is peripheral force, N; \( b \) is width at the bottom of the tooth, mm; \( b \) is the working tooth width, mm; \( \beta \) is angle of pitch circle.

\( Y \) is coefficient of tooth form, \( Y = \frac{(s_y)^2}{(6 \pi \cdot \frac{k m}{h_F})} \),

in which \( s_y \) is chord tooth thickness of dangerous section, \( h_F \) is bent arm when loading functioning at top of tooth.

5) Recommendation type initial value estimation based on BS436-1970 and TOCT13755-68

Tooth shape model modification method recommended by BS436-1970 is mainly for turbine gear. It can be learnt from this method that modify the original tooth top of turbine gear along the direction parallel to the line, maximum model modification \( \Delta_{\text{max}} = 0.008 m_n \), and maximum model height \( h_{max} = 0.47 m_n \), \( m_n \) is normal modulus.
Maximum model modification $\Delta_{\text{max}}$ recommended by 6 level gear can be attained based on TOCT13755-68 tooth shape model modification method:

When $m_n = 2 \sim 2.75$, $\Delta_{\text{max}} = 0.01 m_n$

When $m_n = 3 \sim 4.5$, $\Delta_{\text{max}} = 0.008 m_n$

When $m_n = 5 \sim 10$, $\Delta_{\text{max}} = 0.006 m_n$

**B. Model Length Optimization of Tooth Height Model Practice**

In the process of gear practice, it needs to make optimization by starting from practice model and based on the requirements of length of practice; therefore from the perspective of length, the practice can be divided into short gear model practice and long gear model practice. The length of practice needs to meet starting point B1 to starting point of single tooth engagement C or from end point of double teeth meshing B2 to end point of single tooth meshing D (as shown in picture 2).

Theoretical analysis and experimental research have pointed out that long and short model modifications have following features:

1. In the gear practice process, it needs to make force analysis from maximum modification load, ensure the length of practice within the reasonable error range can be controlled and rotation error of long model practice is smaller than short model practice;
2. If gear load capacity is only half or even less than rated load, usually in the empty load state, it needs to control the impact force of gear and reduce the noise of gear.

Optimization modeling of tooth height practice length is as shown in picture 2:

The parametric modeling of involute cylindrical spur gear is accomplished completely. User clicks Regenerate in the menu, system will let user select if need to input parameter, after inputting parameter, the system will generate involute cylindrical spur gear automatically wanted by the user, as shown in picture 3.

**IV. CONCLUSIONS**

The system has made secondary development for three dimensional modeling software SolidWorks by taking Windows7 operation system as development platform and VisualBasic6.0 as programming tool. The system mainly includes two parts of gear parameter design and three dimensional modeling, where parameter design module mainly makes design calculation and strength checking for parameters of the gear; three-dimensional modeling module mainly realizes three dimensional modeling of involute gear. The research of this system has improved efficiency and accuracy of gear design and modeling, which is good for shortening the products design time, improving product quality and building up foundations for further simulation assembly, motion simulation and finite element analysis.

**REFERENCES**


MINGXIN LI: THREE DIMENSIONAL SOLID MODELING OF GEAR BASED ON SOLIDWORKS SECONDARY …


