Tool path active design for tooth modification of face-hobbed spiral bevel gears

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Abstract. An advanced design and manufacturing method called tool path active design for face-hobbed spiral bevel gears is mainly discussed in this paper. The method bases on the forming mechanism of extended epicycloids and utilizes an integrated cutter head without tilting, meanwhile, all its controlled parameters can be directly calculated according to the given meshing performances and modified conditions. Starting with the cutting principle of face-hobbing method, a tooth flank mending plan of altering tool path is developed, which detailed expressions are also given in functions. Then, the meshing performances which are used to determine the modification parameters are introduced and their formulas are also derived. Finally, the proposed method is validated by a numerical example of tooth contact analysis for a face-hobbed hypoid gear pair. As a further explanation, a numerical example of tooth contact analysis is done to validate the feasibility of the proposed method.

1. Introduction
The face-hobbed spiral bevel gears are divided into two types: Klingelnberg and Oerlikon. Although both of them utilize the face-hobbing method, which is a timed continuous indexing method, the structures of machines and tools they adopt differ widely. In the Klingelnberg’s system, two-part head cutter and machine without tilt and swivel mechanisms are adopted. While, the integrated cutter head and machine with tilt and swivel mechanism are adopted by Oerlikon methods: Spiroflex and Spirac.

In recent years, a series of new achievements of face-hobbing mechanism have been achieved. An advanced design method based on the cutter with bicircular profile and the corresponding tooth contact analysis was presented by Simon in [1]; polynomial functions were applied to induce variations in the cradle radial setting for the pinion tooth surfaces corresponding to reduced transmission error amplitudes in [2] and [3], a novel method for load distribution calculation is applied to investigate the influence of tooth modifications on loaded tooth contact in [4], the optimization methodologies were developed to minimize tooth contact pressures, angular displacement error of driven gear and transmission error in [5]; the influence of misalignment on EHD lubrication in face-hobbed spiral bevel gears was discussed in [6], and optimal tooth modifications were presented in [7]; a method which controlled the execution of motions on the CNC hypoid generation using the relations on the cradle-type machine was proposed in [8]. Kawasaki studied the effect of transmission performance by substituting straight cutter blade for arc cutter blade [9-10]. Fan established the Kinematics model of face-hobbing indexing and divided the cutter blade profile into four parts [11], however, it was not involved that how to divide them. A universal motion concept (UMC) using cutter head with four-segment blade profile was developed by Fan in [12], and tooth surface error correction was presented based on UMC in [13]. An optimal design method of cutter blade profile for face-
hobbed spiral bevel gears to eliminate the tooth edge contact in case of heavy load and considered
misalignment in [14]. However, the condition of cutter axis tilting when using integrated cutter always
exists. Meshing characteristics and design of anti-backlash are also discussed by Deng in [15-17], and
an assessment method of an antibacklash is proposed.

Theoretically, tooth surfaces of face-hobbed spiral bevel gear pair generated by a complementary
generating crown gear are conjugate [12]. However, in application, they need to be modified because
of existence of machining error, assembly error and deformations. In order to get the tooth “crown”,
Klingelnberg alters the eccentric distance between inside and outside cutter to remedy the tooth
longitudinal curvature, and Oerlikon tilts cutter head [18]. Whether adjusting cutter eccentric distance
or tilting cutter head, the calculation is complex and needs to repeat several times. At present, both
Klingelnberg and Oerlikon have developed their own software to calculate machine settings and NC
codes automatically, but the above problems still exist, especially for traditional machine with cradle.

To remedy this problem, combining with the advantages of the two systems, this paper develops a
new method called tool path active design which utilizes an integrated cutter head without tilting. Its
controlled parameters can be directly calculated according to the given meshing performance
parameters, and the calculation of machine settings is also simplified.

2. Tool path active design method

To explain the cutting principle of face-hobbed spiral bevel gears, the imaginary generating crown
gear is usually introduced, which is a special bevel gear with 90° pitch angle. The tooth surfaces of the
generating gear are kinematically formed by the traces of the cutting edges of the cutter blades [12], as
shown in Figure 1. For face-hobbing method, each tooth trace of the generating gear in the pitch plane
is a segment of epicycloid, which can be taken as the trace of a cutting edge point when the roll circle
fixed on the cutter head purely rolls on the base circle fixed on the generating gear. The speed ratio of
cutter and generating gear is constant during cutting. However, if the roll circle rolls on the base circle
with sliding, the epicycloidal tooth trace could be changed, as shown in Figure 2. Based on this
thought, the method of tool path active design for remeedying tooth surface is developed. By this
method, the speed ratio of cutter and generating gear during cutting is changed periodically, the aim of
which is to realize the tooth surface modification by controlling the correction of tooth trace,
combining with the blade circular arc profile.

In the existing cutting methods, the speed ratio of cutter and generating gear is constant, which is
equal to the reciprocal value of their gear ratio.
\[ \theta_c = -\frac{z_0}{z_p} (\theta_k - \theta_{k0}) \]  
(1)

Where, \( \theta_c \) is rotation of generating gear, \( \theta_k \) is rotation of cutter head, \( \theta_{k0} \) is initial position of cutter head, \( z_0 \) is cutter starts and \( z_p \) is the number of generating gear teeth.

However, in this method, the rotation of generating gear is a time-varied function, which is expressed by

\[ \theta_c = -\frac{z_0}{z_p} (\theta_k - \theta_{k0}) + c_2 \cdot (\theta_k - \theta_{km}) + c_4 \cdot (\theta_k - \theta_{km})^4 + \cdots + c_{2n} \cdot (\theta_k - \theta_{km})^{2n} \]  
(2)

Where, \( c \) is control parameters for tooth trace modification, \( \theta_{km} \) is cutter position corresponding to the tooth surface reference point.

Usually, the meshing performances of gear pair are all second order tooth contact characteristics. Only the factor \( c_2 \) could influence the second order parameters of tooth surface, and the other \( C \) factors are used to tooth longitudinal chamfering, the starting position of which is determined by \( \theta_{km} \). Therefore, the tooth surface is divided into three segments along face width, and the cutter blade traces are also divided accordingly.

In practice, there are a few groups of cutter blades in a cutter head which rotates continually. Consequently, the practical function of generating gear is a periodic function whose period is \( 2\pi/z_0 \), which is named tool path function. That is to say each group of the cutter is a period of the function. Every period is also divided into four intervals: cutting interval for inside blade, free transition interval from inside to outside blade, cutting interval for outside blade, and free transition interval from outside to inside blade, which are represented by a, b, c and d respectively in Figure 3. T1 is the effective cutting time interval of inside blade, and T2 corresponds to outside blade.

![Figure 3. One period of the practical function.](image)

As shown in Figure 3, the cutter blade must turn into the desired trace in the advance of cutting the workpiece and delay to drop out the desired trace after cutting, which means that time interval “\( a \)” must be longer than T1, and “\( c \)” must be longer than T2. Theoretically, the free transition intervals “\( b \)” and “\( d \)” are indispensable and as long as possible, in order to make sure that cutter blades can turn into the predetermined traces smoothly. The function of free transition is not given in this paper, because the machine can deal with it automatically.

3. The meshing performance of gear pair
The actual tooth traces in this method depend on the meshing performance of gear pair which needs to be cut. The major meshing performance parameters commonly include angular acceleration of gear \( \Omega_{h2} \), the direction angle from the tangent of tooth trace to the tangent of tooth contact-point trace \( \Theta_i \) and the length of tooth instantaneous contact ellipse \( l \). Their formulas are deduced as follows in detail.
3.1. Assumptions and fundamental deductions
Assume that the tooth surfaces of pinion and mating gear have been known, which includes radial vector, normal vector, principal curvatures and their directions; and all vectors used in this section have been transmitted to the gear coordinate.

First of all, an orthogonal frame \( (t, \tau, n) \) is defined on the gear tooth surface, as shown in Figure 4. To be convenient, \((i, j, k)\) is used as a substitution.

![Figure 4](image)

**Figure 4.** The definition of the orthogonal frame \( (t, \tau, n) \).

As shown in Figure 4(b), \( M' \) is a point in the gear tooth surface \( \Sigma' \), and \( \Pi \) is the common tangent plane when pinion’s tooth surface \( \Sigma'' \) and mating gear’s \( \Sigma' \) are meshing in point \( M' \). The relative velocity \( \mathbf{v}^{12} \) and relative angular velocity \( \mathbf{\omega}^{12} \) [19] are expressed respectively as

\[
\begin{align*}
\mathbf{\omega}^{12} &= \mathbf{\omega}_i + \mathbf{\omega}_j + \mathbf{\omega}_k \\
\mathbf{v}^{12} &= \mathbf{V}_i + \mathbf{V}_j + \mathbf{V}_k
\end{align*}
\]

(3)

The real tooth surfaces are point-contact, and the instantaneous contact points compose the tooth contact-point trace. The radius vector and normal vector of any point along the trace in \( \Sigma' \) are denoted as \( \mathbf{r}^{(1)} \) and \( \mathbf{n}^{(1)} \). Similarly, for the corresponding point in \( \Sigma'' \), they are denoted as \( \mathbf{r}^{(2)} \) and \( \mathbf{n}^{(2)} \). When the two points become meshing points, there are

\[
\begin{align*}
\mathbf{r}^{(1)} &= \mathbf{r}^{(2)} = \mathbf{r} \\
\mathbf{n}^{(1)} &= \mathbf{n}^{(2)} = \mathbf{n}
\end{align*}
\]

(4)

Take a derivative with both of two sides of equations at the same time, there are

\[
\begin{align*}
\mathbf{v}^{(12)} &= \frac{d\mathbf{r}^{(2)}}{dt} - \frac{d\mathbf{r}^{(1)}}{dt} \\
\mathbf{\omega}^{(12)} \times \mathbf{n}^{(1)} &= \frac{d\mathbf{n}^{(2)}}{dt} - \frac{d\mathbf{n}^{(1)}}{dt}
\end{align*}
\]

(5)

Where, \( d\mathbf{r}^{(1)} \), \( d\mathbf{r}^{(2)} \) are the variation of radius vector along the tooth contact-point trace in \( \Sigma' \) and \( \Sigma'' \) respectively, \( d\mathbf{n}^{(1)}, d\mathbf{n}^{(2)} \) are the variation of normal vector along the tooth contact-point trace in \( \Sigma' \) and \( \Sigma'' \) respectively.

As shown in Figure 4, both \( d\mathbf{r}^{(1)} \) and \( d\mathbf{r}^{(2)} \) are in the common tangent plane \( \Pi \), they can be expressed as

\[
\begin{align*}
d_1\mathbf{r}^{(1)} &= d_1x\mathbf{i} + d_1y\mathbf{j} \\
d_2\mathbf{r}^{(2)} &= d_2x\mathbf{i} + d_2y\mathbf{j}
\end{align*}
\]

(6)

\( d\mathbf{n}^{(1)} \) and \( d\mathbf{n}^{(2)} \) [20] can be expressed
\[
\begin{align*}
\begin{bmatrix} d_1 \n_1 \\ d_1 \n_2 \end{bmatrix} &= -\left( k_1^1 d_1 x + \tau_{g_1}^1 d_1 y \right) \mathbf{i} - \left( \tau_{g_1}^1 d_1 x + k_1^1 d_1 y \right) \mathbf{j} \\
\begin{bmatrix} d_2 \n_1 \\ d_2 \n_2 \end{bmatrix} &= -\left( k_2^2 d_2 x + \tau_{g_2}^2 d_2 y \right) \mathbf{i} - \left( \tau_{g_2}^2 d_2 x + k_2^2 d_2 y \right) \mathbf{j}
\end{align*}
\]

(7)

Where, \( k_1^1, k_1^1, \tau_{g_1}^1 \) and \( k_2^2, k_2^2, \tau_{g_2}^2 \) are the normal curvatures and geodesic torsion in direction of \( x \) and \( y \) in tooth surface \( \Sigma^1 \) and \( \Sigma^2 \) respectively.

Solving the Equations (5), (6) and (7), there are

\[
\begin{align*}
\frac{d_x}{dt} &= \left( k_1^1 S_y - \tau_{g_1}^1 S_x \right) \\
\frac{d_y}{dt} &= \left( k_1^1 S_y - \tau_{g_1}^1 S_x \right) \\
\frac{d_x}{dt} &= \frac{d_x}{dt} - V_x \\
\frac{d_y}{dt} &= \frac{d_y}{dt} - V_y \\
where
\begin{align*}
S_x &= \tau_{g_x}^1 V_x + k_1^1 V_y - \omega_x \\
S_y &= k_1^1 V_x + \tau_{g_x}^1 V_y + \omega_x \\
K &= k_1^1 k_2^1 - \left( \tau_{g_x}^1 \right)^2 \\
k_x^1 &= k_1^1 - k_2^1 \\
k_y^1 &= k_1^1 - k_2^1 \\
r_{g_x}^1 &= \tau_{g_x}^1 - \tau_{g_x}^2
\end{align*}
\]

(8)

3.2. Angular acceleration \( \Omega_{12} \)

Angular acceleration is the gradient of gear ratio, which represents the stability of the gearing. When two surfaces are meshing, the normal components of the velocity of their meshing points must be equal, so the angular acceleration \( \Omega_{12} \) can be derived as

\[
\Omega_{12} = \frac{d \left( \frac{\omega_1}{\omega_2} \right)}{dt} = \frac{1}{\omega_2} \left( \frac{d \omega^{(12)} \cdot \n_1}{dt} + \frac{d \omega^{(12)} \cdot \n_2}{dt} - \mathbf{v}^{(12)} \cdot \left( \frac{d \n_1}{dt} + \frac{d \omega^{(1)} \times \n_1}{dt} \right) \right)
\]

According to Equations (6), (7) and (8), assuming that the angular velocity \( \omega_x = 1 \), \( \Omega_{12} \) can be written as

\[
\Omega_{12} = \frac{K \cdot B_1 + B_2}{K \cdot B_1} \begin{align*}
B_1 &= \left( \omega^{(12)} \times \n_1 \right) \cdot \left( \omega^{(1)} \times \n_1 \right) - \mathbf{v}^{(12)} \cdot \left( \omega^{(1)} \times \n_1 \right) - S_y V_x - S_x V_y \\
B_2 &= S_y ^2 k_y^1 - 2 S_y S_y \tau_{g_x}^2 + S_y^2 k_y^2 \\
B_3 &= \omega_2 \left( \mathbf{a}^{(1)} \cdot \n_1 \right)
\end{align*}
\]

Since the tooth surfaces of face-bobbed spiral bevel gear pair without modification are conjugated, the angular acceleration of them is zero. Consequently, the value of \( \Omega_{12} \) is quite small in the tooth reference point, less than \( 10^{-4} \) level. Generally, the transmission error, which is the error between the actual and theoretical position of the gear, is introduced to control the angular acceleration.

3.3. The tangent angle of tooth contact-point trace \( \theta_t \)

The tooth contact-point trace always satisfies: from the toe-top to the heel-root in the tooth concave flank and from the toe-root to the heel-top in the convex flank, as shown in Figure 5. Therefore, it is an obtuse angle in concave side and an acute angle in convex side.

The tangent direction of the tooth contact-point trace is identical to the movement speed direction of the meshing point in the tooth surface, so, \( \theta_t \) can be expressed by

\[
\tan \theta_t = \frac{\frac{d_x}{dt} V_y - \frac{d_y}{dt} V_x}{\frac{d_x}{dt} V_y + \frac{d_y}{dt} V_x} = \frac{k_1^1 S_y - \tau_{g_x}^1 S_x - KV_y}{k_1^1 S_y - \tau_{g_x}^1 S_x - KV_x}
\]
3.4. The length of instantaneous contact ellipse

When the point contact tooth surfaces undertake load, a deformation occurs on the tooth contact point and an instantaneous contact pattern appears in the tooth surface. Regardless of the third and higher order components, the instantaneous contact pattern becomes an ellipse. The length of major axis of the ellipse is called the length of instantaneous contact ellipse $l$. It depends on the induced normal curvatures of tooth surfaces and the suffered load. However, in theoretical analysis, if elastic deformation $\delta$ is given, the length $l$ only depends on the induced normal curvatures of tooth surfaces.

Assuming that the induced normal curvatures of tooth surfaces in direction $t$ and $\tau$ are $k_{1t}, k_{1\tau}, r_{1t}, r_{1\tau}$, according to the Euler formula and Bertrand formula on curvature [21], the main values of the induced normal curvatures of the contact point can be expressed by

$$
\begin{align*}
k_{1t}^{12} &= k_{1t}^{12} \cos^2 \phi_{t} + 2r_{x}^{12} \sin \phi_{t} \cos \phi_{t} + k_{1\tau}^{12} \sin^2 \phi_{t}, \\
k_{2t}^{12} &= k_{1t}^{12} \sin^2 \phi_{t} - 2r_{x}^{12} \sin \phi_{t} \cos \phi_{t} + k_{1\tau}^{12} \cos^2 \phi_{t},
\end{align*}
$$

Where, $\phi_t$ - The angle from the principle direction to the direction $t$.

As shown in Figure 6, the elastic deformation $\delta$ is quite minute (Gleason 0.00635mm). The length of instantaneous contact ellipse $l$ can be expressed by

$$
l = 2 \frac{k_{1t}^{12} \sin \theta}{k_{1t}^{12}} \approx \frac{2}{k_{1t}^{12}} \theta = 2 \frac{2\delta}{k_{1t}^{12}}
$$

Where, $k_{1t}^{12}$ is the smaller value between $k_{1t}^{12}$ and $k_{2t}^{12}$.

Tool path active design method corrects one or both of the gear pair by combining tool-path planning with the blade circular arc profile optimizing [14]. It can directly calculate the C factors of the tool path function and the curvature radiusses of the blade profiles according to the required meshing performance in the reference point M. The control parameters are composed of blade profile curvature radius $\rho_1, \rho_2$ and the C factors $c_{21}, c_{22}$. Theoretically, four meshing performance parameters can be ensured. However, only the pinion tooth surfaces are usually corrected, the control parameters can be utilized become $\rho_2$ and $c_{22}$. In this case, only two meshing performance parameters can be assured.

4. Numerical experiment

A numerical experiment is done for a pair of spiral bevel gears with offset in this section. The main parameters of the gear pair: shaft angle 90°, offset 25 mm, teeth number 8 for pinion 8 and 39 for gear, face width 40 mm, outer pitch diameter 265 mm, and mean spiral angle of gear is 34.63°. Nominal radius of cutter is 106.5 mm and cutter starts is 5. The machine settings are calculated and are listed in Table 1.
Table 1. Machine settings.

|                        | Pinion | Gear  |
|------------------------|--------|-------|
| Cutter radial distance/mm | 115.117 |       |
| Cutter center roll position/° | 66.993 | -54.624 |
| Workoffset/mm           | 25.053 | 0     |
| Mach. center to cross point/mm | -1.234 | 0     |
| Machine root angle/°     | 14.341 | 75.332 |

Firstly, reference points of both flanks are defined in the middle of the gear tooth surfaces, then, the reference points of pinion tooth which satisfy the theoretical gear ratio can be calculated. This procedure is also the computational procedure of control parameters according to the desired meshing performance of gear pair. The meshing performance usually includes three aspects as involved in Section 3, but there are only two modification parameters in a pinion tooth surface. In order to test the corrected results visually, the angular acceleration and the length of contact ellipse are selected to calculate the pinion modification parameters, and the former is represented by the maximum transmission error.

In practice, the pinion concave flank and the gear convex flank are used as driving side of the gear pair, and other two flanks are coasting side. In this experiment, the cutter blade profile of gear is a circular arc whose curvature radius is 1200mm, and the meshing performance parameters are given that: the maximum transmission error is 60μrad for both sides; the length of contact ellipse l is 10 mm for driving side and 12 mm for coasting side in reference point. Meanwhile, the tooth longitudinal chamfering is also introduced, and a length of 10 mm in toe and heel for chamfering is given for driving side and 15 mm for coasting side. The calculated pinion modification parameters and tool path function factors are listed in Table 2.

Table 2. Pinion modification parameters and tool path function factors.

|                        | Pinion concave (outside blade) | Pinion convex (inside blade) |
|------------------------|-------------------------------|------------------------------|
| Blade curvature radius/mm | 700                           | 900                          |
| C2 factor              | -0.006362                     | 0.006545                     |
| C4 factor-MI           | -1.315476                     | 0.5248678                    |
| C4 factor-ME           | -1.315476                     | 0.5248678                    |
| Cutter position for ref. point-M/rad | 0.8632               | 0.8626                      |
| Cutter position for cha. point-M/rad | 0.9474              | 0.9260                      |
| Cutter position for cha. point-ME/rad | 0.7437            | 0.8089                      |

Applying them to Equation (3), the functions of tool path for both sides can be determined respectively, and they compose one period of the practical tool path function by one of them moves a phase angle of $\pi/\pi_0$.

On the basis of machine settings and tooth modification parameters, the mathematical cutting modules for both gear and pinion can be established, then tooth surfaces with modification or not can also be acquired. The ease-off topographies of pinion tooth surfaces are shown in Figure 7. Figure 7(a) is the case without any modification; 7(b), 7(c) and 7(d) are the cases with a single modification parameter; 7(e) and 7(f) are the real modification results but 7(e) without chamfering.

Figure 7(a) shows that the mating tooth surfaces of the face-hobbed spiral bevel gear pair with offset are not conjugate, and a “crown” of tooth surface develops automatically, which is beneficial to gear transmission. Meanwhile, there is a line in the surface which approximately satisfies the meshing function in the case of theoretical gear ratio. Figure 7(b) shows that the cutter blade profile only can correct tooth flank in direction of tooth height. Figure 7(c) and 7(d) show that the changing tool path can directly affect the tooth longitudinal crown and the 4th order parameters can also realize the tooth
chamfering. Figure 7(e) and 7(f) indicate that the tool path design method can realize the tooth surfaces’ crowning and chamfering as we expect.

(a) Pinion ease-off without any modification  
(b) Pinion ease-off with blade circular arc only  
(c) Pinion ease-off with 2nd order modification only  
(d) Pinion ease-off with 4th order chamfering only  
(e) Actual pinion ease-off without chamfering  
(f) Actual ease-off of pinion tooth  

Figure 7. Pinion tooth ease-off topographies for Concave.

In order to evaluate the meshing performance of the gear pair which is derived from above, the TCA method is applied to two cases: one with modification but chamfering and another with modification and chamfering, which are shown in (e) and (f) of Figure 7. The TCA results are shown in Figure 8 and Figure 9 respectively.

As shown in Figure 8 and Figure 9, the maximum transmission error is 58.03μrad for driving side, and 61.36μrad for coasting side, and the tooth chamfering introduced does not affect the transmission error. It is less than design value 1.97μrad for driving side and 1.36μrad more for coasting side. The difference of the transmission error between the TCA and the design results from the rounding of curvature radius of cutter blade circular arc profile. And the errors are also acceptable in practice. The figures labeled as a) are the tooth contact patterns without considering tooth surfaces’ overlap, and figures labeled as b) are the actual contact patterns. Both a) and b) indicate that the contact patterns are all in the middle zone of the tooth surface as our design, and tooth chamfering affects nothing on the contact patterns but a bending at both ends.
Figure 8. TCA results without chamfering.

Figure 9. TCA results with chamfering.
5. Conclusions
This paper mainly discusses a new design and manufacturing method which is called too path active design, its features and advantages are listed below:

1. This method utilizes integrated cutter heads without tilting, which makes the machine without tilt mechanism. Compared with Klingelnberg, it simplifies the cutter head, and compared with Öerlikon, it simplifies the milling machine. Applied to free-form CNC machine, it just needs a four-axis controlling, meanwhile, the machine stiffness is greatly enhanced because of saving a swing axis for workpiece.

2. This method corrects the tooth surfaces by changing tool path and curvature radius of blade circular arc profile, it can directly calculate modification factors according to the meshing performances.

3. The machine settings of this method are simplified because of cutter without tilting. They just depend on the gear geometry, and only need to be calculate once regardless of tooth surface modification.

Meanwhile, there is a constraint for this method: it must be assured that only one cutter blade can be working at any time when cutting, which means that the cutter starts will be less than Öerlikon cutter.

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