Modified sprocket tooth profile of roller chain drives

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A B S T R A C T

Roller chain drive is widely used in timing mechanism of gasoline engine. However, its polygonal action and meshing impact effect resulted from the non-conjugated meshing feature may damage the synchronization and uniformity of transmission. In this paper, new sprocket tooth profile is developed to reduce polygonal action and meshing impact under high speed. A new conjugated profile is derived by modifying involute profile to guarantee that the moving distance of chain is equal to the arc length of pitch circle that a sprocket rotates at the same time and the center line of chain at tight side is always tangent to the pitch circle. An asymmetrical modification method for sprocket tooth profile is also proposed. A multi-body dynamic model of the timing mechanism of a gasoline engine with the intake and exhaust sprockets is developed. The fluctuation and meshing impact of the chain are analyzed under different rotational speeds. The results show that newly developed sprocket profile can efficiently reduce meshing impact and friction of chain. The stability of chain transmission under high speed can be improved.

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1. Introduction

Roller chain drive is recognized to be one of the most effective forms of power transmission in mechanical systems. It has a basic feature with a constant ratio because of no slippage or creep. So, it is widely used as a timing mechanism and oil bump system in a car engine. The noise and vibration of such systems still are problems as higher speed, lighter weight and higher quality are required in today’s car.

Roller chain drive is generally suitable for the transmission under a slower speed due to its polygonal action and meshing impact. Many researches on dynamic behaviors of a roller chain are carried out experimentally and theoretically for the improvement of the roller chain drive. Ariartnam and Asokanthan [1] studied the periodic fluctuation of power transmitting chains brought about by factors such as polygonal action and eccentricity of sprockets by treating the chain as a traveling uniform heavy string. Veikos and Freudenstein [2] proposed a discrete model of chain by lumped masses connected by linear springs which considered the coupling between the longitudinal and transversal vibrations. Choi and Johnson [3] presented a dynamic model for the analysis of the performance of roller chain drive with a tensioner based on the axially moving material model with consideration of the effects of polygonal action, impact, and the periodic span length changes. Low [4] presented a computer-aided analysis for the selection of roller chain drives based on the mechanic equations of the drives used in mechanical power transmission systems. Liu et al. [5] modeled a coupled chain and sprocket system interacting with local impacts and derived the impulse function. Sheu et al. [6] proposed a kinematic model considering friction based on the vector loop approach for analyzing the kinetostatics and mechanical efficiency of roller drives. Pedersen [7,8] developed a dynamic model of a roller chain drive including the impact with guide-bars, the inner friction and polygonal action of roller chains. Yang et al. [9] established the dynamic equation of sleeve roller chain by using finite element method and analyzed the acceleration response of the sleeve roller chain.

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A new sprocket tooth profile is also investigated by some researchers to minimize the polygonal action and meshing impact. Rong [10] reported that it is necessary to study the tooth profile of a sprocket, shape of chain link, elasticity of material and meshing dynamics for the decrease of vibration and noise of chain drive. Xue and Wang [11] studied the impact velocity between chain and sprocket as the sprockets have an involute tooth profile. Zhang et al. [12] investigated the approximate conjugate meshing of the chain drive by using a roller pin. However, the structure of such chain is complicated and the manufacturing cost is high. Wang and Zhang [13,14] developed a sprocket profile by integrating the involute and beeline and investigated the meshing impact and velocity waving of the chain as the sprocket has the kind of tooth profile.

The meshing process between chain and sprocket is generally non-conjugate, which results in polygonal action and meshing impact during the meshing process. The polygonal action means that the pitch line of the chain is a tangent or secant line of the pitch circle of the sprocket alternately. The pitch line of the chain keeps moving up and down. The instantaneous transmission ratio between driving and driven sprockets is variable. Those features are able to cause an uneven chain velocity, transversal and longitudinal vibration, noise and meshing impact, and damage the synchronization and evenness of the transmission, so that the application of the roller chain drive because the transmission speed of the timing mechanism in the car engine can be up to 7000–8000 r/min. Only can a specially designed roller chain achieve a conjugate action. We try to achieve an approximate conjugate action by the modification of the sprocket tooth.

In this article, a new type of sprocket tooth profile is theoretically developed in order to improve the operational performance of roller chains under high-speed situation. The tooth profile of the sprocket can ensure that the meshing between sprocket and chain roller is an approximate conjugate action. The multi-body dynamic model of the timing mechanism in the engine with intake and exhaust sprockets is established. Then, an asymmetrical modification method for the sprocket tooth profile is proposed and the dynamic characteristic of the asymmetrically modified sprocket is analyzed.

2. Meshing mechanism of roller chain

2.1. Sprocket profile equation

The center line of the roller chain is a tangent or secant line of the sprocket pitch circle alternately due to the polygonal action. The periodical variation of the chain center line affects directly the chain velocity and instantaneous angular velocity of the driven sprocket. To reduce the velocity variations, it is necessary to ensure that the arc length of the pitch circle in one pitch angle is equal to the pitch of the chain, or, that the moving distance of the chain at any moment is equal to the arc length of the pitch circle to be turned, as shown in Fig. 1. For realizing that, we let the contacting pitch line, that is, the line to contact with pitch circle tangentially, move an offset \( k \) transversally away from the pitch circle of the sprocket and then attain a roller pitch line (see Fig. 2). Assuming that the sprocket is fixed, the contacting pitch line rolls around the pitch circle of the sprocket without sliding. A new sprocket tooth profile is one equidistant curve of the trajectory curve of a roller center.

Referring to Fig. 1, as the roller moves from positions 1 to 2, the movement distance of the chain is equal to the chain pitch \( p \). The diameter of the pitch circle of the new tooth profile of the sprocket is expressed as

\[
d = \frac{pz}{\pi}
\]

where \( z \) denotes the number of teeth of the sprocket and \( p \) the pitch of the chain.

The offset \( k \) of the roller pitch line relative to the contacting pitch line is

\[
k = \frac{p}{2 \sin \phi} - \frac{d}{2}
\]

where \( \phi \) denotes the pitch angle of the sprocket.

![Fig. 1. Schematic diagram of meshing between sprocket and roller.](image-url)
In the Cartesian coordinate system XOY, lines $L_1$ and $L_2$ represent the contacting pitch line and roller pitch line respectively, as shown in Fig. 2. Line $L_1$ rolls around the pitch circle without sliding. When line $L_1$ rotates by an angle $\theta$, line $L_1$ moves to $L_3$ and line $L_2$ to $L_4$. Point A on line $L_1$ moves to point C. Point B on line $L_2$ moves to point D. According to the principle of generating involute curve, the moving trajectory of any point C on $L_3$ is a family of involute curves, and the equation is

$$\begin{align*}
  x &= r(\theta \cos \theta - \sin \theta) \\
  y &= r(\theta \sin \theta + \cos \theta)
\end{align*}$$

Line CD is normal to line $L_4$ at any position. Therefore, the coordinates of point $D(X,Y)$ on line $L_4$ and point $C(X,Y)$ on line $L_3$ meet the following equations

$$\begin{align*}
  X &= x - k \sin \theta \\
  Y &= y + k \cos \theta
\end{align*}$$
To substitute Eq. (3) into Eq. (4), the trajectory equation of the roller center can be obtained as

\[
\begin{align*}
X(\theta) &= r\theta \cos \theta - (r + k) \sin \theta \\
Y(\theta) &= r\theta \sin \theta + (r + k) \cos \theta.
\end{align*}
\tag{5}
\]

New tooth profile curve is an equidistant curve of the roller center trajectory. The curve will be derived then by solving the enveloping curve of circles.

2.2. New tooth profile for the sprocket

We take the fixed frame \( \{o; i, j\} \) and the dynamic frame \( \{P; i_p, j_p\} \), as shown in Fig. 3. Curve V is the trajectory curve of the roller center and U is an equidistant curve of V. Assume that origin P is on the trajectory curve V of the roller center and the direction of \( \vec{i}_p \) is outward along the O–P line. The initial position of the dynamic frame is \( P_0; \vec{i}_{p0}, \vec{j}_{p0} \) on the \( \vec{j} \) axis. Draw a circle with the center of \( P_0 \). The circle is fixed on the dynamic frame. A circle family is formed by moving the circle center along curve V. At the same time, point \( M_0 \) on initial position circle moves to point \( M \). The position of point \( M \) is decided by rotational angle \( \beta \). The position of \( P \) is decided by rotational angle \( \alpha \). According to Eq. (5), the vector equation of trajectory curve V of the roller center is expressed by parameter \( \theta \) as

\[
\vec{r} (\theta) = X(\theta) \vec{i} + Y(\theta) \vec{j}
\tag{6}
\]

And the equation of the circle family can be written as

\[
\vec{r}' = \vec{r} (\theta) + h (\cos \beta \vec{i}_p + \sin \beta \vec{j}_p)
\tag{7}
\]

where \( h \) is the distance between curves V and U in Fig. 3, i.e., roller radius.

The vector transforming relationship between dynamic frame and fixed frame is

\[
\begin{align*}
\vec{i}_p &= -\sin \alpha \vec{i} + \cos \alpha \vec{j} \\
\vec{j}_p &= -\cos \alpha \vec{i} - \sin \alpha \vec{j}.
\end{align*}
\tag{8}
\]
The value of angle $\alpha$ in Eq. (8) is determined by an angle $\theta$, i.e., $\alpha(\theta) = \theta - \arctan \frac{r}{c_1 \theta}$. However, no matter what the function relationship changes, the transforming relationship in Eq. (8) is always tenable. Letting $\alpha = \theta$ and combining Eq. (7) with Eq. (8), we have

$$\vec{r}(\theta) = [X(\theta) - h \sin(\theta + \beta)] \hat{i} + [Y(\theta) + h \cos(\theta + \beta)] \hat{j}. \quad (9)$$

The coordinate of any point $(X', Y')$ can be derived based on Eq. (9) as

$$\begin{align*}
X'(\theta) &= X(\theta) - h \sin(\theta + \beta) \\
Y'(\theta) &= Y(\theta) + h \cos(\theta + \beta).
\end{align*} \quad (10)$$

Substituting Eq. (5) into Eq. (10), the partial derivative to parametric variables $\theta$ and $\beta$ is derived as

$$\begin{align*}
\frac{\partial x}{\partial \theta} &= -r \theta \sin \theta - k \cos \theta - h \cos(\theta + \beta) \\
\frac{\partial x}{\partial \beta} &= -h \cos(\theta + \beta) \\
\frac{\partial y}{\partial \theta} &= r \theta \cos \theta - k \sin \theta - h \sin(\theta + \beta) \\
\frac{\partial y}{\partial \beta} &= -h \sin(\theta + \beta).
\end{align*} \quad (11)$$

According to the principle of solving an envelope curve, there is

$$\frac{\partial x}{\partial \theta} \frac{\partial y}{\partial \beta} - \frac{\partial y}{\partial \theta} \frac{\partial x}{\partial \beta} = 0. \quad (12)$$

Combining Eq. (11) with Eq. (12), we get

$$\tan \beta = -\frac{r \theta}{k}. \quad (13)$$

We also have

$$\begin{align*}
\sin \beta &= \pm \frac{r \theta}{\sqrt{k^2 + (r \theta)^2}} \\
\cos \beta &= \mp \frac{k}{\sqrt{k^2 + (r \theta)^2}}.
\end{align*} \quad (14)$$

By combining Eqs. (5) and (10) with Eq. (14), the equidistant curve equation of the roller center trajectory is derived as

$$\begin{align*}
X'(\theta) &= \left( r \theta \mp \frac{hr \theta}{\sqrt{k^2 + (r \theta)^2}} \right) \cos \theta - \left( r + k \mp \frac{hk}{\sqrt{k^2 + (r \theta)^2}} \right) \sin \theta \\
Y'(\theta) &= \left( r \theta \mp \frac{hr \theta}{\sqrt{k^2 + (r \theta)^2}} \right) \sin \theta + \left( r + k \mp \frac{hk}{\sqrt{k^2 + (r \theta)^2}} \right) \cos \theta.
\end{align*} \quad (15)$$

where “$\mp$” expresses that the trajectory curve of the roller center has two equidistant curves, which are located in the internal and external sides of the roller center trajectory respectively. We only use the internal equidistant curve of the roller center trajectory in this paper. So, “$-$” will be used in Eq. (15).

2.3. Modification of tooth profile

Usually, there is a big impact effect as the chain link meshes into and out of the sprocket. The impact energy constrains the working ability of the roller drive under a high speed situation. The impact energy $E_A$ and impact speed $V_A$ can be expressed as [15].

$$E_A = \frac{m p}{2000} V_A^2 \quad (16)$$
where $\gamma$ is the pressure angle of meshing and $m$ is the chain mass, kg/m.

According to Eqs. (16) and (17), the factors to affect meshing impact energy and speed include meshing pressure angle $\gamma$, pitch $p$ and rotary speed $n$. To reduce pressure angle $\gamma$ is helpful to decrease impact speed $V_A$ and impact energy $E_A$. Therefore, we try to modify the tooth profile to reduce pressure angle.

Because of the unidirectional transmission of the timing chain drive mechanism, the practical working tooth flank is one side of the sprocket tooth. So, the tooth profile can be asymmetrically modified in order to reduce the pressure angle $\gamma$. That is, the working tooth flank is the new developed tooth profile on which the roller can engage with the sprocket to cut down the meshing impact force between rollers and sprocket. Not-working tooth flank is the standard tooth profile.

3. Analyzed roller chain and its model

3.1. Analyzed chain mechanism

As an example, the chain-drive timing mechanism in high speed petrol engine is analyzed which is composed of a crank shaft sprocket 1, an intake sprocket 2, an exhaust sprocket 6, a roller chain 3, a hydraulic tensioner 4 and a fixed guide 5, as shown in Fig. 4. The roller chain of the mechanism has a pitch of 9.525 mm and the number of links in the ring of chain is 114. The tooth number of the crank shaft sprocket is 19. The tooth numbers of both the intake and the exhaust sprockets are 38. The center distance between the intake and exhaust sprockets is 133.35 mm. The analyzed sprockets adopt the GB1244 tooth profile, the DIN8196T1-87 tooth profile and the new sprocket tooth profile, respectively. The offset $k$ is calculated to be 0.0668 according to Eq. (2). The detailed parameters are listed in Table 1.

The analyzed tooth profiles have a lot of difference. The tooth channel radius of the profile on the standard GB1244 is $r_1 = 0.5025d_1 + 0.05$ where $d_1$ is the diameter of the roller. The channel radius on the standard DIN8196T1-87 is between $r_{1\text{ min}} = 0.505d_1$ and $r_{1\text{ max}} = 0.505d_1 + 0.069\sqrt[3]{d_1}$. Moreover, the radius of the working profile on the GB1244 is $r_2 = 1.3025d_1 + 0.05$, and the radius on the DIN8196T1-87 is between $r_{2\text{ min}} = 0.12d_1(\sqrt{z} + 2)$ and $r_{2\text{ max}} = 0.008d_1(\sqrt{z^2} + 180)$.

3.2. Modeling approach for roller chain

With the development of CAE technology, more and more simulation software can provide friendly pro-process module nowadays. We make use of software Pro/Engineer to establish the 3D physical models of all components in roller chain drives. The 3D sprocket model is shown in Fig. 5. The finite element model for roller chain drives is then developed by FEM software.

The roller chain drive is modeled as a lumped mass system. The mechanical components in the roller chain drive are defined as independent elements respectively. Such as, chain element is represented as a separate rigid body with 3 degrees of freedom. Sprocket elements are represented by mass and moment of inertia. The meshing process is described by stiffness, damping and
backlash. Chain elements are assumed to be elastically connected to each other, and they are elastically connected to sprockets. The whole train drive system is represented using different elements in a series connection. Fig. 6 shows the simulation model for the timing chain transmission system in Fig. 4.

The chain drives are modeled as a multi-body dynamic system and software is developed in which the Lagrange multiplier method is adopted to solve the dynamic problem. In the software, all parameters required in the model are set in accordance with the practical chain drive system. The dynamic behavior of the chain drives can be analyzed, such as, the meshing impact force, the load fluctuation of valve camshaft, the friction and impact between chain and sprocket, the friction between chain and tensioner, the friction and impact within the fixed guide, and the preload of tensioner.

4. Dynamic behavior of roller chain

4.1. Meshing impact effect

Impact force occurs when the roller chain engages with the sprocket especially at high speed. The transient peaks of the impact force are present during chain starts to mesh into the sprocket. The impact force is one of the main sources of vibration and noise existed in the timing chain drive mechanism. It may also result in the stretch and fatigue of chain drives.

Fig. 7 shows the meshing impact force of the timing chain transmission system (see Fig. 4) with different sprocket tooth profiles as the crank shaft sprocket rotates at the speeds of 3000 and 6000 r/min. It is found that the impact force exhibits higher value when the chain begins to mesh with the sprocket, and smaller when the roller completely meshes with the sprocket. The impact forces are much larger as the German standard sprocket DIN8196T1-87 and the standard GB1244 sprocket are used. The impact force is the smallest as a newly developed tooth profile is adopted. For example, the impact force peak value of the crank shaft sprocket reaches 518.7 N for DIN8196T1-87, 369.4 N for GB1244 and 321.0 N for the new tooth profile at 3000 r/min

<table>
<thead>
<tr>
<th>Parameters</th>
<th>Size</th>
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</thead>
<tbody>
<tr>
<td>Teeth number of intake and exhaust sprockets</td>
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</tr>
<tr>
<td>Teeth number of crank shaft sprocket</td>
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</tr>
<tr>
<td>Pitch (mm)</td>
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</tr>
<tr>
<td>Diameter of roller</td>
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</tr>
<tr>
<td>Teeth thickness (mm)</td>
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</tr>
<tr>
<td>Diameter of graduated circle (mm)</td>
<td>115.21</td>
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<tr>
<td>Diameter of addendum circle (mm)</td>
<td>120.77</td>
</tr>
<tr>
<td>Diameter of guided arc (mm)</td>
<td>6.35</td>
</tr>
<tr>
<td>Tightening force (N)</td>
<td>400</td>
</tr>
<tr>
<td>Load (N m)</td>
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</tr>
<tr>
<td>Roller radius h (mm)</td>
<td>3.175</td>
</tr>
</tbody>
</table>

Fig. 5. 3D model of sprocket.
Fig. 6. The simulation model for timing chain transmission system.

Fig. 7. Impact force between roller and sprocket (curve 1—between roller and crank shaft sprocket; curve 2—between roller and intake camshaft sprocket; curve 3—between rollers and exhaust camshaft sprocket). (a) On GB1244 at 3000 r/min; (b) on GB1244 at 6000 r/min; (c) on DIN8196T1-87 at 3000 r/min; (d) on DIN8196T1-87 at 6000 r/min; (e) with new profile at 3000 r/min; (f) with new profile at 6000 r/min.
respectively. The peak values of the intake sprocket reach 310.0 N for DIN8196T1-87, 239.4 N for GB1244 and only 91.1 N for the new tooth profile at 3000 r/min. The peak values of the intake sprocket are 376.4 N for DIN8196T1-87, 259.4 N for GB1244 and only 128.6 N for the new tooth profile at 6000 r/min.

Fig. 8. Impact force with different speeds.

Fig. 9. Friction between chain and transmission elements (curve 1—with crank shaft; curve 2—hydraulic; curve 3—intake camshaft sprocket; curve 4—exhaust camshaft sprocket). (a) On GB1244 at 3000 r/min; (b) on GB1244 at 6000 r/min; (c) on DIN8196T1-87 at 3000 r/min; (d) on DIN8196T1-87 at 6000 r/min; (e) with new profile at 3000 r/min; (f) with new profile at 6000 r/min.
The peak value variation of the meshing impact force between roller and sprocket is shown in Fig. 8 as the intake/exhaust sprocket speed varies from 1000 to 8000 r/min. It can be found that the peak value with the new sprocket tooth profile is lower than with the other profile sprockets. The peak value of the meshing impact force tends to get higher as the chain speed increases. For example, the peak value is 735.1 N at the speed of 8000 r/min and 518.7 N at 3000 r/min when the sprocket is designed on DIN8196T1-87. When it is on GB1244, the peak value is 626.3 N at the speed of 8000 r/min and 385.2 N at 3000 r/min. As the new tooth profile is used, the peak value is 550 N at the speed of 8000 r/min and 308.6 N at 3000 r/min.

4.2. Chain friction

Decreasing the friction within the timing mechanism in the engine is very useful for reducing noise of the engine and improving service life of the timing chain. Friction between chain and transmission elements is calculated by the software after all parameters for chain and transmission elements are given. Only a friction coefficient must be specified. As an approximation, friction coefficient is set to be 0.05 and it is used for all the elements. Fig. 9 shows the variation of friction between chain and transmission elements as the rotational speeds of the camshaft are 3000 and 6000 r/min respectively. We can find that the friction is smallest as the sprocket has the new tooth profile. The friction peak value between chain and hydraulic tensioner with the new tooth profile at the speed of 3000 r/min is 4.9 N, which is reduced by 28% compared with using the standard GB1244 sprocket and by 43% compared with the DIN8196T1-87 sprocket. The friction peak value with the new tooth profile at 6000 r/min is 3.1 N, which is reduced by 63% compared with GB1244 and by 64% compared with DIN8196T1-87. The friction between chain and tensioner with the new tooth profile is lowest because the profile is helpful to reduce the tension of train. Therefore, it is an efficient way to improve the transmission efficiency and the working life of the timing chain by using the new tooth profile sprocket.

4.3. Fluctuation of angular speed

The instant angular speed of the driven sprocket is variable all the time due to the polygonal action, which may damage the synchronization and uniformity of transmission seriously. The variation of angular speed of the driven sprocket can be expressed by nonuniformity coefficient $K_k$ as

$$K_k = 2 \frac{\omega_{2\text{ max}} - \omega_{2\text{ min}}}{\omega_{2\text{ max}} + \omega_{2\text{ min}}}.$$  

Fig. 10 shows the nonuniformity coefficient of the angular speed of the driven sprocket as the speed of the crank shaft sprocket changes from 1000 to 8000 r/min. It is found that the angular speed fluctuation with the new tooth profile sprocket is smaller at low rotational speed than with the sprocket profile designed on GB1244 and DIN8196T1-87. As the speed is over 2000 r/min, the speed fluctuations of the sprocket with three kinds of tooth profiles are very close.

5. Features with modified tooth profile

5.1. Tension force of chain

The tension force of the chain reflects the stress state of the chain. The excessive chain tension may result in insecurity, vibration and noise of chain drives. Fig. 11 shows the tension variation of the chain at the rotational speeds of 3000 and 6000 r/min when sprockets have different tooth profiles. The tension of the tight side of the chain fluctuates around 600 N as the sprocket is based on the standard GB1244, and around 800 N as on DIN8196T1-87. As a new asymmetric tooth profile sprocket with pressure angle
Fig. 11. Tension of chain with different sprocket tooth profiles, (a) on GB1244 at 3000 r/min; (b) on GB1244 at 6000 r/min; (c) on DIN8196T1-87 at 3000 r/min; (d) on DIN8196T1-87 at 6000 r/min; (e) with new profile at 3000 r/min; and (f) with new profile at 6000 r/min.

Fig. 12. Friction between chain and tensioner.
9.73° is used, the tension fluctuates around 450 N, and it is lower by 25% than on the standard GB1244 and by 43% than DIN8196T1-87. So, it is beneficial to reduce chain tension and improve stress state by using an asymmetric tooth profile sprocket on the condition of equivalent power and load.

5.2. Friction of chain

Fig. 12 shows the friction between chain and hydraulic tensioner at 3000 r/min when the sprocket uses a modified tooth profile with pressure angle 9.73°. The friction can reflect the size of transversal fluctuation of the chain and it affects the life of the chain. Transient peaks of friction appear when chain link moves into and out of the tensioner. The peak value of friction is 6–8 N at the beginning of contacting tensioner and 4–6 N at the ending. The friction reduces obviously to 1–2 N after link contacts with tensioner entirety.

The variation of average friction between chain link and hydraulic tensioner is shown in Fig. 13 as the speed of intake sprocket varies from 1000 to 8000 r/min. With the increase of rotational speed, the tension of train tends to decrease so that the friction between chain and tensioner is reduced. As the sprocket uses the modified new tooth profile, the friction is lowest in comparison with the sprockets based on the standard GB1244 and DIN8196T1-87. This is because the chain tension caused by dynamic load cuts down greatly.

5.3. Fluctuation of the chain

Fig. 14 shows the fluctuation of the chain to connect the intake and exhaust sprockets, which is the variation of chain displacement along the transversal direction. The sprockets with the new tooth profile and the asymmetrically modified profile are respectively adopted as the intake and exhaust sprockets. The fluctuation of the chain is obviously reduced by using the asymmetrically modified tooth profile sprocket at the speeds of 3000 and 6000 r/min.

Fig. 15 shows that the variation of transversal vibration of chain link to connect the intake and exhaust sprockets with different tooth profiles as the rotational speed of the crank shaft varies from 1000 to 8000 r/min. When the rotational speed is below 5500 r/min which is the common used speed in the car engine, the transversal vibration of the modified sprocket is averagely lower than sprockets designed on the standards GB1244 and DIN8196T1-87. As the speed is over 6500 r/min, the transversal vibration of chain link tends to increase with the modified new tooth sprocket.

6. Conclusions

A new sprocket tooth profile is presented which can effectively reduce the dynamic effect and meshing impact of chain drives. The tooth profile is obtained by modifying the traditional involute tooth profile. It can guarantee that the moving distance of the chain at any moment is equal to the arc length of the pitch circle to be turned, and that the roller center line of the tight side is always tangent to the pitch circle of the sprocket. The new tooth profile enables rollers to mesh with the sprocket gradually so that the impact force between rollers and sprocket is reduced.

The multi-body dynamic model of the chain-drive timing mechanism with intake and exhaust sprockets in a petrol engine is established. The meshing impact between tight side link and sprocket, the friction between chain link and transmission components, and the fluctuation of link to connect two camshafts are obtained. The analyzed results indicate that the meshing impact and the friction are reduced obviously by using the new tooth profile sprocket under different rotational speeds.

An asymmetry tooth profile of the sprocket is put forward which aims at the feature of one-side transmission of the timing chain drive mechanism. The method to modify and get the asymmetry tooth profile is discussed. The analysis for the dynamic behavior of the timing chain drive mechanism with the asymmetry tooth profile is carried out. The results indicate that the dynamic behavior of the timing chain drive mechanism with the asymmetry tooth profile is improved compared with the traditional involute tooth profile.
A feature of the sprocket with asymmetry teeth indicates that the friction between link and transmission components is obviously decreased and the tension in chain link, fluctuation of the chain, and the speed fluctuation of the camshaft are improved.

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Y. Wang et al. / Mechanism and Machine Theory 70 (2013) 380–393

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