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Analysis and Design of Sprocket Driven Transmission System

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Abstract: ***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 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. Also the center line of chain at tight side is always tangent to the pitch circle. The fluctuation 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 and also the stability of chain transmission under high speed can be improved.***

Keywords: Roller chain drive, variable speeds, Modification, Sprocket.

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.

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 is used to be limited in a lower speed. However, it is necessary to develop a new type of tooth profile with a conjugate action so as to reduce the polygonal action and impact of the roller chain drive. Only a specially designed roller chain can achieve a conjugate action. We try to achieve an approximate conjugate action by the modification of the sprocket tooth.

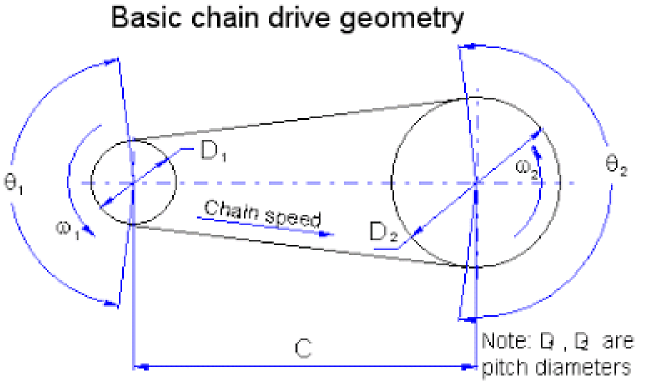
In this article, a new type of sprocket tooth profile is theoretically developed in order to improve the operational performance of roller chains under high-speed situation. The tooth profile of the sprocket can ensure that the meshing between sprocket and chain roller is an approximate conjugate action.

2. Methodology for Modelling

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 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.

In the software, all parameters required in the model are set in accordance with the practical chain drive system.



3. Materials Selection

3.1Cast Iron Sprockets

Cast Iron is the most common and economical material for flat wire belt sprockets, they are accurately cast from high grade iron. Other diameters can be provided on special order.

3.2Plastic Sprockets

All plastic sprockets are fully machined and meet USDA and FDA guidelines for food contact.

- UHMW Polyethylene can withstand continuous temperatures up to 180 degrees F. Stock sprockets are UHMW.
- High Temp UHMW Polyethylene can withstand continuous temperatures up to 220 degrees F.
- Nylon sprockets provide 2-3 times the strength of UHMW and can withstand higher temperatures.

3.3Steel Sprockets

- FL sprockets have no flange and a hub sticking out one side to allow debris to fall through the belt.
- MT sprockets are made either from a solid piece of steel or have a flange welded at the base of the teeth, for belt support. Flangeless and -MT sprockets have their teeth hardened to a Rockwell 50-55 on the C Scale. All other steel sprockets can have their teeth hardened on request.

3.4 Stainless Steel Sprockets

Stainless sprockets are either investment cast from 18-8 stainless or fully machined from T-303 SS or T-316 SS. Fully machined flangeless (-FL) or machined tooth (-MT) sprockets made from various stainless steels are also available.

3.5 Finalising the Material

The material **EN8 Steel** as named in British standards was selected due to its better machinability and hig tensile strength.

4. Equations

4.1 Sprocket Metric Dimensions (Chain 520)

4.1.1 Dimensions

1) Sprocket thickness (t_s) – 5.7658 mm
2) Number of teeth (z) – 38
3) Pitch (p) – 15.88 mm
4) Addendum circle diameter(d_a)
   $= d + 1.25p - d_r = 220$ mm
5) Pitch circle diameter (d_p) – 210 mm
6) Dedendum circle diameter (d_d) – 200 mm
7) Tooth height above pitch polygon –
   $k_{max} = 0.625p – 0.5\ d_r + (0.8/z)*\ p = 5.1793$ mm
   $k_{min} = 0.5(p – d_r) = 2.86$ mm
8) Root circle diameter (d_f) = d_d = 200 mm
9) Tooth angle (τ) = 360 / z = 9.4737°

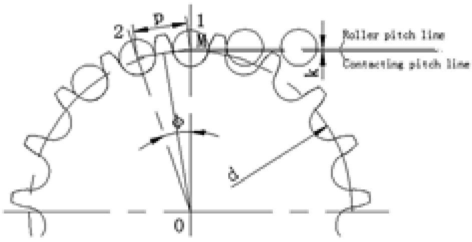
4.1.2 Dimensions (CHAIN 520)

1) Chain size = 520
2) Links = 108
3) Type = O-RING
4) Pitch = 15.88 mm
5) Width = 6.35 mm
6) Roller diameter (d_r) = 10.16 mm
7) Gap (g) = pitch – roller diameter = 5.72 mm

4.2 Formulas Related To Transmission

$F_x = (T_e) * N_{tf} * \eta / r$ ….(in kg or newtons)
$W_{DRIVE} / W_{WHEELS} = N_f$ …..(final drive ratio)
$W_d = N_f * W_w$
$W_e = N_t * W_d$
$W_{ENGINE} / W_{DRIVE} = N_t$ …..(Transmssion ratio)
Hence , $W_w = W_e / N_t * N_f$
$= (rpm) * 2\pi/60 * (1/\ N_t * N_f)$ …..(rad/sec)
Corresponding ground speed V_x is given by ,
$V_x = W_w * r$ (m/s)
$N_{tf} = N_t * N_f$
$\eta_{tf} = \eta_t * \eta_f$

W_d = speed of driving sprocket (rad/sec)
W_e = speed of engine (rad/sec)
N_t = transmission ratio
N_f = final drive ratio



4.3 Reference Engine Specifications

Displacement	249.6 cc
Maximum Power	26.5 HP @ 8500 rpm
Maximum Torque	22.9 Nm @ 7000 rpm
Number of Cylinders	1
Number of Gears	6
Seat Height	784 mm
Ground Clearance	145 mm
Kerb/Wet Weight	164 kg
Fuel Tank Capacity	13 litres
Top Speed	135 kmph

4.4 Torque and Force Calculations

4.4.1 Base Formula

Torque at wheels ($F_w * r_w$) = max torque from engine * primary gear reduction ratio * secondary reduction ratio * sprocket ratio* transmission efficiency

= 22.7*2.808*3.33*2.714*0.98

T_w = 565 Nm

R_w = 0.254 m (wheels) = 0.105 m(sprocket)

F_w = 1135.0264 N (wheel) = 2745.33 N (sprocket)

4.4.2 Calculations for Different Gears

Assumptions

1) Throttle air intake diameter is 38mm
2) Transmission efficiency is 0.97
3) Max torque is obtained from at 7000 rpm

For 1st gear –
T_w = 560 Nm
F_s = 5333.33 N

For 2nd gear –
T_w = 356 Nm
F_s = 3390.48 N

For 3rd gear –
T_w = 264 Nm
F_s = 2514.28 N

For 4th gear –
T_w = 220 Nm
F_s = 2095.24 N

For 5th gear –
T_w = 188 Nm
F_s = 1790.48 N

For 6th gear –
T_w = 161.6 Nm
F_s = 769.52 N

5. Modeling and Analysis

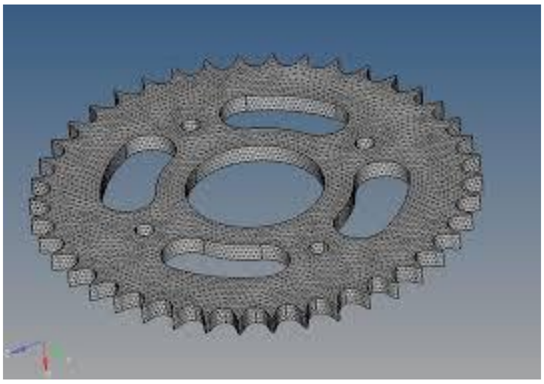


Figure: CAD Model of Sprocket

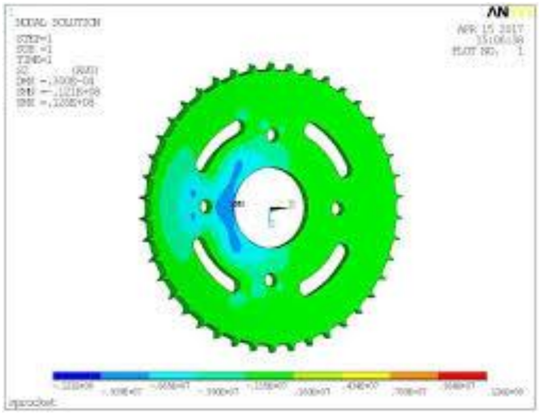


Figure: Nodal Solution

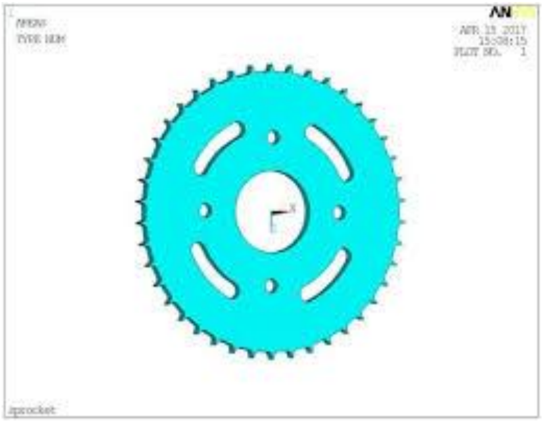


Figure: Thermal Analysis

Figure: Nodal Solution Von Mises Stresses

5.1 Tension force on sprocket

The tension force of the sprocket reflects the stress state of the sprocket. The excessive tension on the sprocket 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.

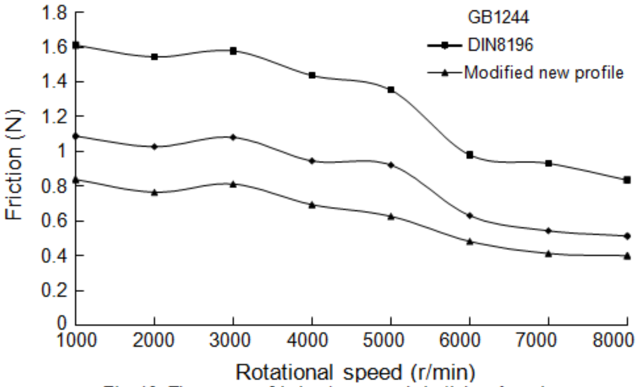


Fig. 13. The average friction between chain link and tensioner.

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 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 chain drive mechanism.

The method to modify and get the asymmetry tooth profile is discussed. The analysis for the dynamic 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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