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Dynamic Analysis of Timing Silent Chain System for a V-type Engine of a Vehicle

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Abstract: Based on multi-body dynamic software RecurDyn, this paper proposes a modified form of timing silent chain system combing with the existing problem that vibration and chain tension is too large, which is applied for complicated conditions in a V-type engine, such as high speed, variable loads. The analysis of chain drive meshing characteristics is completed. Using the multi-body dynamic soft-ware RecurDyn, the dynamics characteristics of the improved system is studied, including chain tension, transmission error, chain fluctuations, equivalent spring force in different operating conditions. The study results show that chain tension, transmission error, chain fluctuation and equivalent spring force are within the scope of permission, all of them can meet the design requirement. There-fore, the design of this system is reasonable and practicable. The research results will provide a basis for assessing timing silent chain system in a V-type engine and a theoretical reference for designing and optimizing the timing silent chain system.

1. Introduction

There are mainly three forms of power transmission in the modern engine timing mechanism: timing gear, timing toothed belt, timing chain.(1) The research on timing gear starts early, so its design technology is more mature and its structure is reliable, but timing gear system whose structure is complex not only has much parts, but also its quality is large so that the larger impact is arose during the meshing process. Timing toothed belt whose quality is small and structure is simple is not suitable for larger power transmission, and its design life can not achieve synchronization with engine. Timing silent chain can meet the high speed and heavy load conditions, which can transfer large power and realize stable operation. Therefore, more and more timing silent chain systems are used for valve train of high power engine, for example, military machine, mining machine.

Currently, the valve train of V-shaped engine with large power almost uses timing gear power system.(2) Due to its complex configuration, heavy weight and low transmission efficiency, it can not transmit engine power effectively. Therefore, silent chain has been applied in V-shaped engine timing chain system by some engine makers in recent years. Domestic and foreign scholars researched on silent chain drive from many aspects. James D. Young(3) adjusted tooth sprocket meshing frequency by modifying meshing tooth sprocket profile in timing chain system, thereby reduced camshaft drive noise; Hiroshi Takagishi,et al(4) of Honda created simulation method in calculation of camshaft torque and chain load caused by crankshaft speed fluctuation; Jack S.P.Liu,et al(5) of Ford Motor Company calculated natural frequency by using theoretical method and CAE analysis, and verified through the experimental data. Taiwan scholars Chintien Huang,et al(6) put forward the tooth contact analysis of round pin jointed silent chain by kinematics analysis(7) ; Meanwhile, they researched on chordal action and transmission errors of silent chains; Xue Yunna(8) from Shandong University investigated theoretical analysis and meshing mechanism of timing chain system. ZHU Meilin, et al(9), research the V-shaped engine timing chain system from four aspects including engineering design, strength calculation, component test as well as whole engine test. But two hydraulic tensioners are installed at the slack side of the timing silent chain system. This structure leads to reduce the timing silent chain stiffness. And thus make engine produce vibration and noise. In addition, there is no research on the dynamic characteristics of the V-shaped engine timing silent chain system.

In summary, this paper proposes a modified form of the V-shaped engine timing silent chain system with one hydraulic tensioner(Fig.1), and researches dynamic characteristics analysis for the V-shaped engine timing silent chain system in

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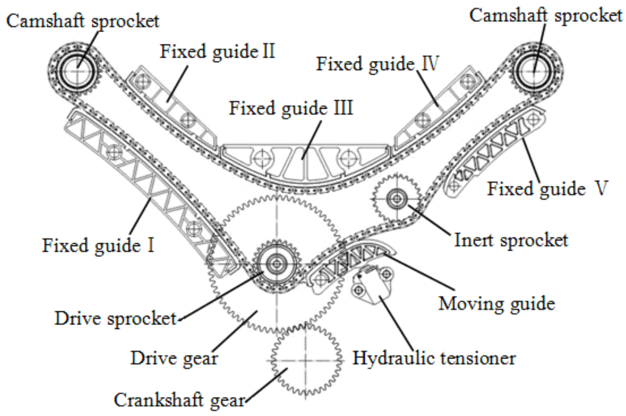

Fig. 1 Timing silent chain system of the V-shaped engine

complex operating conditions by means of the multi-body dynamics software RecurDyn. The research results have an important guiding significance to independent design and R&D capabilities of V-shaped engine timing silent chain system.

2. Simulation Model

The timing drive system of the V-shaped engine (Fig. 1) is composed of two stage transmission subsystems. The first-stage transmission is completed by a pair of spur gears with the radio of 2:1. The parameters of two gears are as follows: Drive gear modulus m=2.5, pressure angle β=12º, the number of teeth z=68, pitch circle diameter d=173.798mm; Crankshaft gear modulus m=2.5, pressure angle β=12º, the number of teeth z=34, pitch circle diameter d=86.899mm. The second-stage of the drive system which is to be studied is the toothed chain drive with the radio of 1:1. This system is installed in the front of the first-stage gear transmission system. It consists of the following components: drive sprocket, inert sprocket, two camshaft sprockets, moving guide, five fixed guides, hydraulic tensioner, timing silent chain. The four sprockets are involute sprockets in the system, whose parameters are identical other than the installing phase angle. The parameters of involute sprocket are as follows: The number of teeth z=25, pressure angle α=31.5º, pitch p=8mm, addendum circle diameter d_a=62.73mm, pitch circle diameter d_b=63.83mm, root circle diameter d_f=54.30mm. The chain plate is external meshing plate and its pitch is 8mm. Among them, the drive gear is mounted coaxially with the drive sprocket, so that the power transmission can be realized.

In the timing chain drive system model, one hydraulic tensioner is only used in the loose side, fixed guide Ⅴ realizes the chain tension between inert sprocket and camshaft sprocket. The torque transmitted to the drive sprocket by the crankshaft gear is obtained by calculated, and it is applied on the sprocket as the drive torque for the system. Due to the parameters of hydraulic tensioner are very complicated. So in the paper, the hydraulic tensioner is replaced by equivalent spring. Fig. 2 shows the timing silent chain system dynamics model of V-shaped engine based on the modern multi-body dynamics software RecurDyn. The movement of one link in a single cycle can be divided into eight stages: Drive sprocket(S1), slack side(SP),inert sprocket(S2), tight side Ⅴ(TP1), camshaft sprocket(S3), tight side Ⅱ, Ⅲ,Ⅳ(TP2),camshaft sprocket(S4) and tight side Ⅰ(TP3). In order to obtain simulating data conveniently, one link is marked.

In the model, four sprockets connect with rack by revolute joints. The connection of links is also revolute joints. The chain plate and sprocket realizes mutual constraints by contact force as well as chain plate and guide(10). In RecurDyn, the approximate result of the contact movement dynamics is that an object penetrates into another object when imposing a certain speed at the contact point, there will be corresponding normal pressure and friction at the contact pair. In the contact force model, the contact normal force is defined as a function of the amount of permeability, and the relationship between the normal force and the permeation amount are modified by indentation index. Normal force f_n (11) is calculated as follows:

$$f_n = k\delta^{m_1} + c\frac{\delta}{\dot{\delta}}\left|\dot{\delta}\right|^{m_2}\delta^{m_3} \quad (1)$$

Where

δ : Infiltration capacity

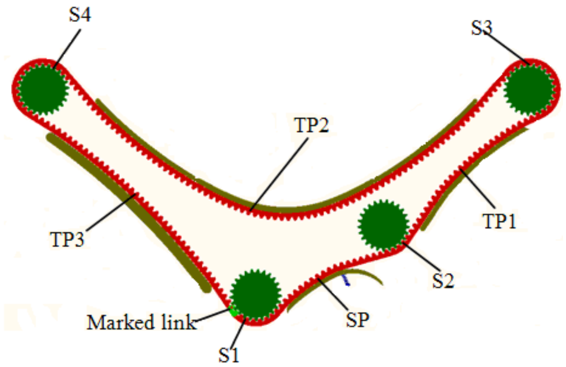

Fig. 2 Timing silent chain system dynamics model

$\dot{\delta}$	: Penetration speed
K	: Elasticity coefficient
c	: Contact damping coefficient
m_1	: Stiffness coefficient
m_2	: Damping coefficient
m_3	: Dent index

Elasticity coefficient k and contact damping coefficient c are determined by experimental. Stiffness coefficient m_1 and damping coefficient m_2 are related to the generation of non-linear contact force. Dent index m_3 is related to the generation of the indentation damping force. These parameters are related to the materials and dimensions.

The expression of the corresponding friction force f_f [11] is as follows:

$$f_f = \mu(v)\left|f_n\right|$$
$$f_f = sign(f) \times \min(\left|f_f\right|, f_{\max}) \tag{2}$$

Where

$\mu(v)$: Friction coefficient

3. Dynamic Analysis Results

This V-shaped engine timing chain system is used on a military vehicle. Its work environment is very complex. The maximum speed is 4000rpm. In this paper, three kinds of typical condition for the dynamics analysis are selected, there are 1000rpm, 2500rpm, 4000rpm. When a link finishes an operating cycle (the drive sprocket turns 3024°), the simulation results are obtained.

3.1 Chain tension force

As the main power transmission component in the process of silent chain drive, the link withstands cyclical changing force. Through the analysis of the chain tension figure lines in different conditions, the chain tension can be obtained in the process of chain movement. The effect of initial startup impact, meshing impact and polygon effect on the timing silent chain system are explored, which provides a theoretical reference for the V-shaped engine timing silent chain system further optimization. At the same time, the static strength check of timing chain is completed by using the maximum value of the chain tight side.

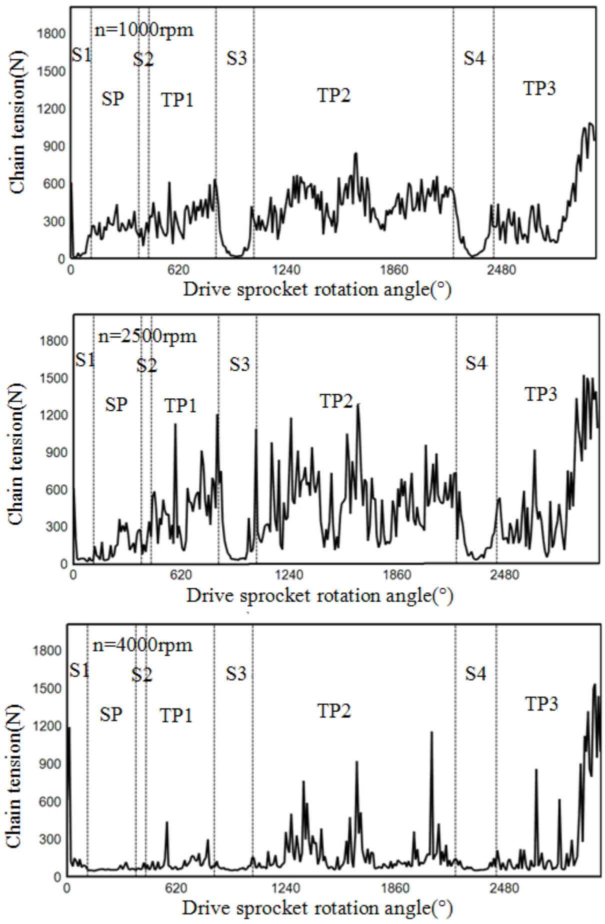

Fig. 3 The curve of the chain tension

Fig. 3 shows the changeable graph line of chain tension with the drive sprocket rotation angle in a complete running at these conditions. At first, the chain tension becomes larger, but with the speed increasing, it changes small. When the link is in S1 stage, the chain tension is maintained at 160N, and in S2, S3, S4 stages, the chain tension is less than 200N. Because the chain and the sprocket are in the meshing process, the chain tension is mainly from centrifugal force generated by sprocket. When the link is in SP stage, the link is in the slack side, the hydraulic tensioner absorbs chain fluctuations and realizes the tension of the loose side, which leads to generation of the chain tension.

When there is not rated power meter and power curve used for reference in the automotive chain calculation, the safety coefficient method is be used to calculate the static strength(2). The static strength safety coefficient of automotive chain is as follows:

$$n = Q / F \tag{3}$$

Where:

Q : The tensile strength of automotive chain

F : The tight side tension of automotive chain

Statistics shows that the suitable value of n is 14 to 18[2].

The maximum value of the timing chain tight side tension is 1520N, which is can be got from chain tension variation diagram. The tensile strength of automotive chain is $Q \geq 2.2$KN[9], the safety coefficient n of the timing silent chain is 14.47. Therefore, the timing silent chain can meet the timing requirement of reliability.

3.2 Transmission error

Due to the polygon effect and the dynamic impact load, there will be gas phase difference between intake and exhaust camshaft sprockets in the drive process of the timing silent chain system. That is transmission error(12). In order to ensure the intake and exhaust valve timely opening and closing as well as timely and output of the engine, the transmission error must

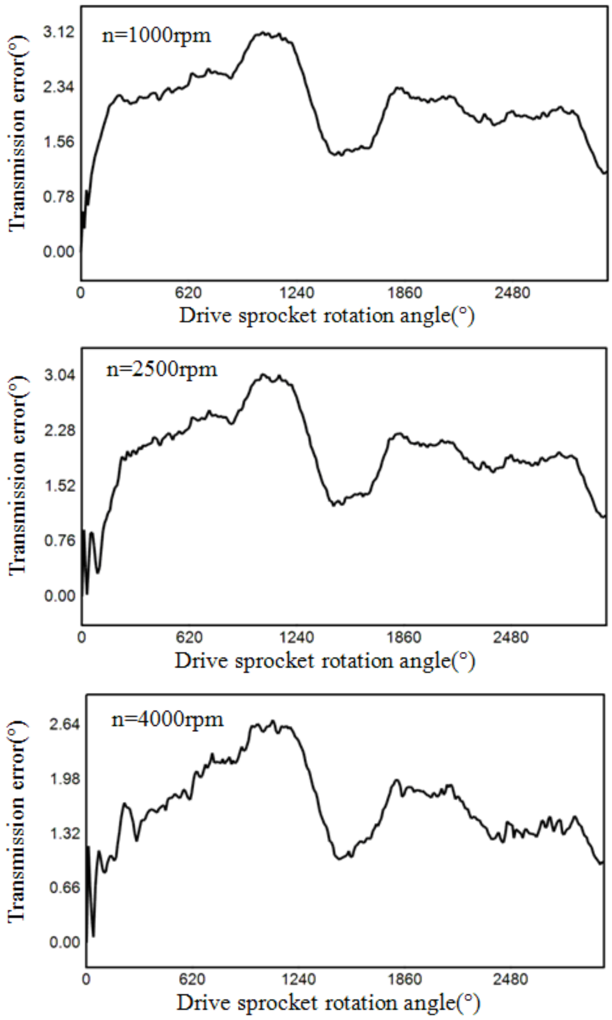

Fig. 4 Transmission error

be guaranteed within the scope permitted. Transmission error is calculated as follows:

$$\varphi = \theta_1 - \theta_2 \quad (4)$$

Where:

θ_1 : The rotation angle of intake camshaft sprocket

θ_2 : The rotation angle of exhaust camshaft sprocket

Fig. 4 shows the transmission error between the intake and exhaust camshaft sprocket at the three conditions. When the drive sprocket speed is 1000rpm, the largest transmission error is 3.120°. When the drive sprocket speed is 2500rpm, the maximum transmission error is 3.052°. When the drive sprocket speed is 1000rpm, the largest transmission error is 2.695°. It can be seen that the transmission error is gradually decreases with engine speed increasing, which is less than permit transmission error 5°, therefore it can guarantee valve timing well.

3.3 The movement locus of marked link

Because the chain is composed of many links whose pitch is determined, the engagement of chain and sprocket likes a polygon. Therefore, the effective radius appears cyclical fluctuations. which leads to the lateral and longitudinal movements of the chain. In addition, the engaged direction of chain speed changes which forms meshing impact in the meshing between the chain and sprocket. Therefrom, the polygon effect of chain is formed[13]. In the polygon effect, the timing silent chain system will produce vibration and noise in the operation process, which affects the overall performance of the engine. The movement locus of marked link is a good reflection of the polygon effect. The fluctuations of the timing silent chain in the process of running can be learned by observing the movement locus of marked link.

When the rotation speed is constant, the larger plate pitch , the more obvious the polygon effect, and vibration and noise of the whole system will increase. When the pitch is constant, the higher the speed, the more serious the impact and chain fluctuation. Theoretical value(14)of the fluctuation of the system is calculated by the following formula:

$$\psi = \frac{p \times (1/\sin(\pi/z) - 1/\tan(\pi/z))}{2} \tag{5}$$

By Eq. (5), the value of theory fluctuation is $\psi = 0.2516$mm

Fig. 5 shows the movement locus of marked link at the three conditions. When the link is in low speed, there exists minor local fluctuation, which is mainly caused by the polygon effect. As the speed increases, the trajectory tends to smooth. However when in high-speed conditions, the impact changes more serious, particularly at the junction of link and sprockets, and the junction of the link and guides. But its actual value of fluctuations is 0.1971mm, which is less than permitted value and meets the requirement.

3.4 The equivalent spring force

The function of the tensioner is for chain tension, and it compensates for the chain elongation caused by chain wear as well as reduces the vibration caused by chain load which is transmitted on the engine cylinder(15), so as not to appear these phenomena, such as abnormal engagement, out of the chain and the tooth skipping. The hydraulic tensioner is the main component of the chain tension. Meanwhile, it is the main component of the timing silent chain system. In order to study conveniently, the hydraulic tensioner is replaced by the equivalent spring in dynamics analysis of the V-shaped engine timing system(16). It will provide important theoretical reference to the subsequent hydraulic tensioner type and choice of parameters by extracting the equivalent spring force under various conditions.

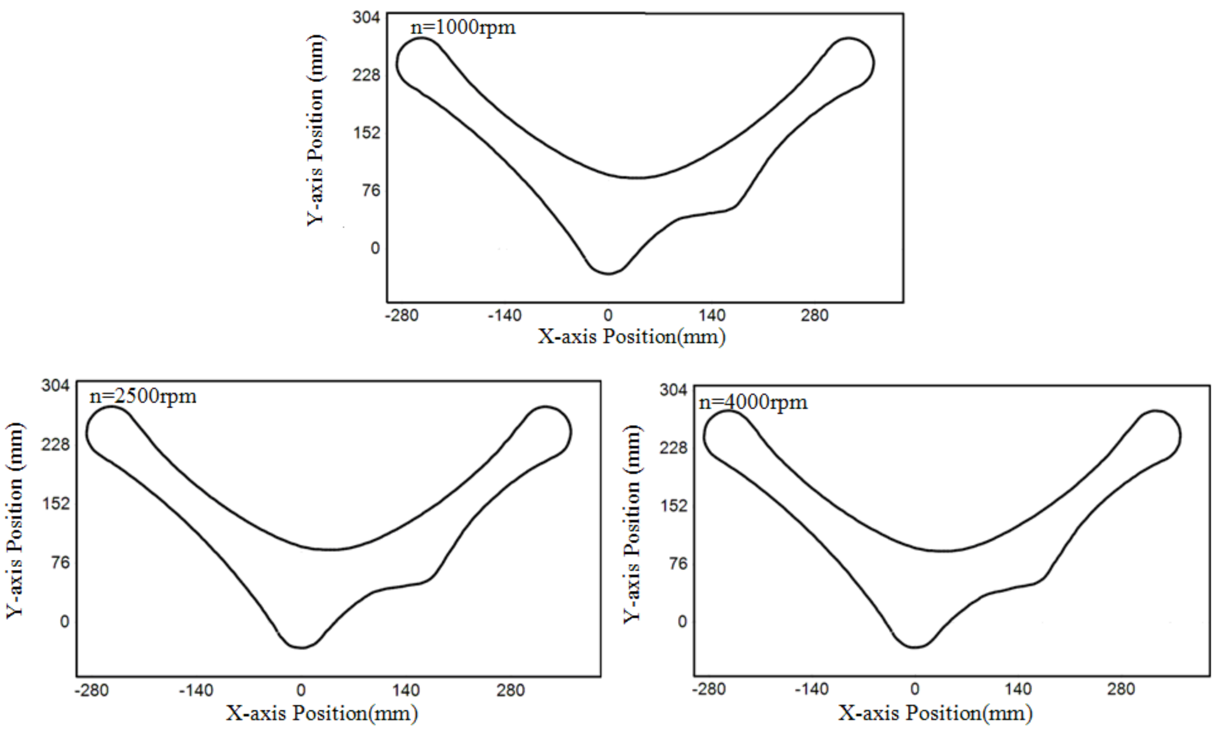

Fig. 5 The trajectory of marked link

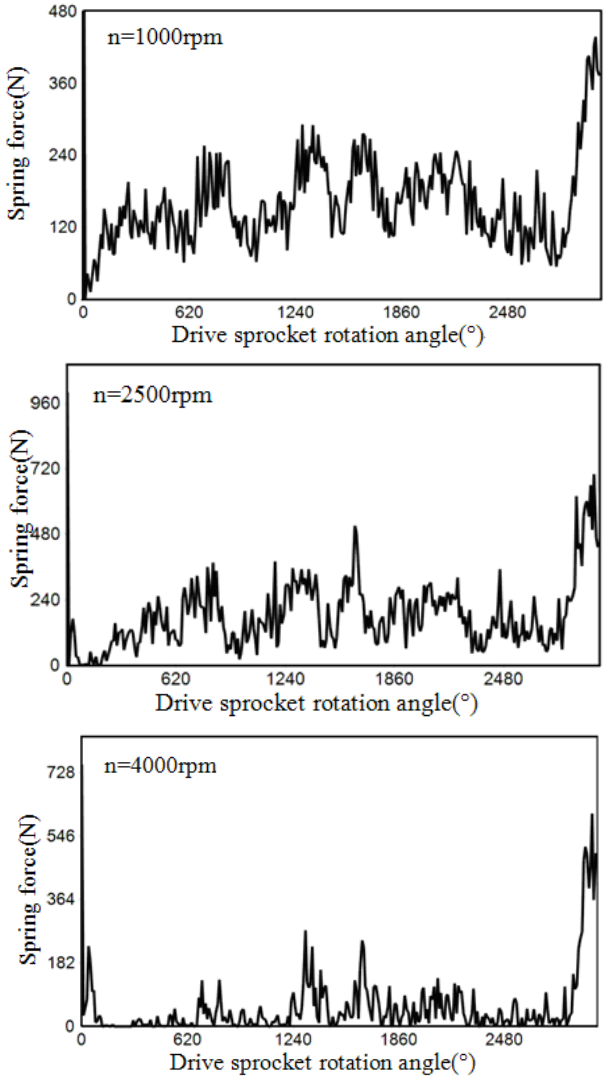

Fig. 6 The equivalent spring force

Fig. 6 shows the change chart of equivalent spring force at the three conditions. In the first condition, with drive sprocket rotation angle increasing, the spring force becomes larger in the operating cycle of one link. With drive sprocket speed increasing, the equivalent spring force firstly increases, and then tends to change small. In the third conditions, the equivalent minimum valve of spring force is sequentially as follows: 37N, 41N, 19N. The maximum value is as follows: 397N, 619N, 599N.

4. Conclusions

(1) Based on vibration and noise of the V-shaped engine timing chain system, this paper proposes to use one hydraulic tensioner and fixed guide to replace original two hydraulic tensioner to realize the chain tension. This can prevent the chain stiffness reduction and improve the system of vibration and noise effectively, which enhances and improves the original system.

(2) According the modified form of the V-shaped engine timing silent chain system, dynamic characters analysis is conducted by muli-body dynamic method. The analysis results are as follows: the effective maximum chain tension is 1520N, which is within the scope of licensed; transmission error is less than 5°, so that it can guarantee the timing chain system valve timing; the actual fluctuation value is 0.1971mm, which is less than theory fluctuation value 0.2516mm, and meets the design requirement; equivalent spring force is not greater than 619N, to meet the design requirements, and provide theoretical reference for the subsequent design of the hydraulic tensioner.

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