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Inside and Outside Flank Alternate Meshing Silent Chain and Experimental Evaluation of Dynamic Performance

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Abstract: A new silent chain with inside and outside flank alternating meshing mechanism and sprockets were designed, and the main technical parameters and structure type of the plate and sprocket profile were described. In order to verify the good meshing transmission performance of the new silent chain, we did the performance comparison test for inside and outside flank alternate meshing silent chain and the ANSI standard silent chain with regard to transverse displacement vibration and wear extension. The results show that the new silent chain transmits smoothly, with standard silent chain compared to significantly reduce vibration quantity of transverse displacement, and it has good wear resistance properties, which is much superior to the standard silent chain. The results also verify low impact characteristics of the new silent chain and the rationality of design method. Because of its special meshing mechanism and structure type of alternate load, the new silent chain reduces meshing impact, vibration and polygon effect when chain and sprocket meshes, and fundamentally improves silent chain transmission performance, then extends the life of the chain.

Key words: Meshing mechanism, polygon effect, silent chain, sprocket, transverse vibration, wear

INTRODUCTION

With continuous development of the machinery industry and automobile industry, the traditional chain drive was made to adapt to new requirements of high-speed heavy, variable speed and variable load in the working environment. In roller chain transmission, transmission mode was achieved by collision contact of the chain roller and sprocket tooth. The transmission mode could cause wear extension of roller chain and meshing noise (Liu *et al*., 1997; Calvo *et al*., 2006). While in silent chain drive, drive mode of chain and sprocket was achieved by progressive meshing transmission of plate profile and sprocket tooth. Such progressive meshing method could reduce the impact and noise, enhance transmission efficiency, and improve polygon effect of chain drive mechanism (Wada *et al*., 1999; Mengfan, 2008; Zhengzhi *et al*., 1984).

Transmission mode of the silent chain had a close relationship with match of plate profile and sprocket profile. It was not the same that different plate profile of the silent chain matched with the sprocket tooth. The ANSI standard silent chain (ANSI, 2007) was the outer contact meshing, plate outer face and a straight tooth sprocket meshed instantaneously, roller chain was the same, there was the polygon effect, which not only caused velocity uniformity, but also since the dynamic loading shock and link vibration make chain wear increased, it had been unable to meet today's requirements for transfer efficiency and the noise of the chain drive.

This study proposed and designed structure type and new tooth form of silent chain and sprocket with inside and outside flank alternating meshing mechanism. The new silent chain began to mesh with sprocket, inside flank of the plate first contacted with the sprocket tooth, as the plate and sprocket tooth further engaged, the outside flank of the adjacent plate began to mesh with the sprocket, until outside flank of the plate located on the sprocket alveolar. Because of its special meshing mechanism and structure type of alternate load, the new silent chain could significantly reduce the polygon effect causing shock and vibration in silent chain transmission, reduce wear and improve the transmission efficiency.

PLATE PROFILE

The new silent chain based on the ANSI standard silent chain was designed, which improved the profile of standard plate and sprocket, and made the meshing process of chain and sprocket smoother. In plate profile, the crotch height and inside flank width were changed: crotch height was reduced to increase plate width, which could enhance the plate intension and prevent damage to occur. While inside flank width was increased to make plate and sprocket in meshing contact first mesh with inside flank of the next plate, then mesh with outside flank of the plate. Such could increase the meshing contact time, reduce the impact and improve the polygon effect in the meantime. The plate of the ANSI standard silent chain was shown in Fig. 1.

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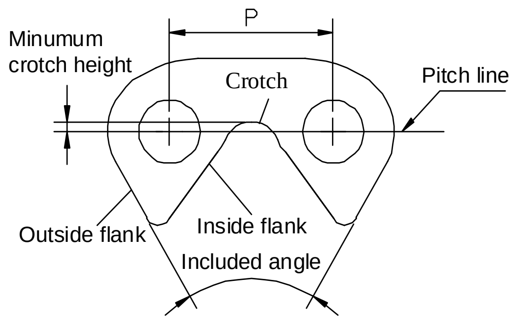

Fig. 1: The plate of the ANSI standard silent chain

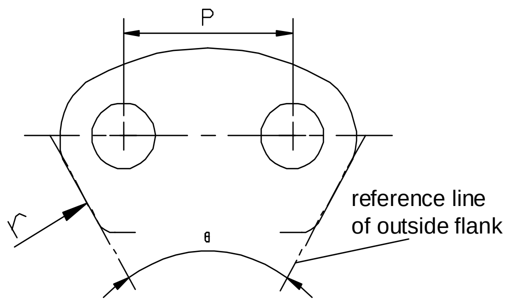

Fig. 2: Structure diagram of outside flank of the plate

Outside flank of the plate: Sprockets designed were not ANSI standard sprocket profile, instead of using negative modified involute profile. Therefore, outside flank of the plate employed a concave curve to match with the profile, and pressure angle was $\theta = 60^{\circ}$. In addition, the sprocket and chain meshed, because outside flank of the plate and sprocket profile was similar, which could increase the surface contact area of mutual engagement, thus significantly reduced meshing impact and the noise level during the entire chain drive.

If the pressure angle of sprocket was α, m was the modulus, and z was the teeth number, then involute curvature radius of sprocket was $r = r_b \tan \alpha$ (Mengfan, 2008). Where r_b was the involute base circle radius, that was $r_b = mz \cos \alpha$.

Outside flank of the standard silent chain was deemed as the reference straight line, outside flank curve of new silent chain and the reference line was circumscribed (Fig. 2). The curvature radius of the outside flank should be larger than that of the involute profile of sprockets to ensure when plate and sprocket meshed, outside flank of the plate and sprocket tooth could contact. The mathematical relationship of curvature radius for outside flank of the plate was as follows:

$$r > mz \sin \alpha$$

Inside flank of the plate: In the meshing process of sprocket and chain, to achieve the sprocket first to contact with inside flank of the plate I (Fig. 3), and then contact with outside flank of the plate II (Fig. 4), share contact force of sprocket and plate I to increase inside flank width of the plate I, so inside flank of the plate employed convex curve to match with sprockets. Inside flank of the plate projected outwardly relative to outside flank of the plate an amount δ (Fig. 3) when chain was pulled straight.

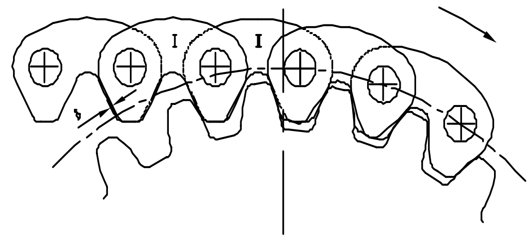

Fig. 3: Sprocket mesh with inside flank of the plate I

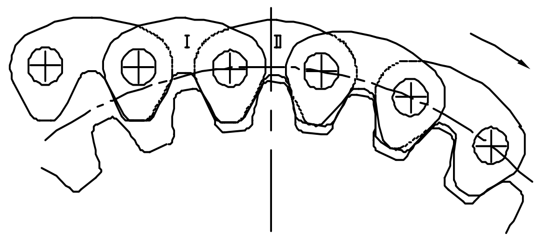

Fig. 4: Sprocket mesh with outside flank of the plate II

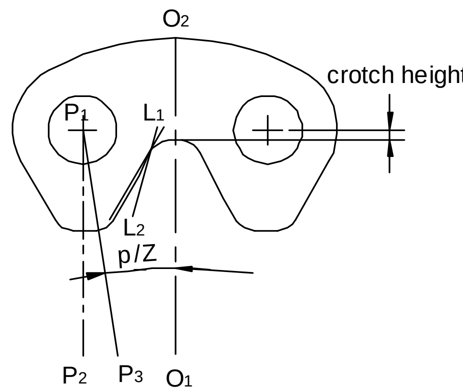

Fig. 5: Structure and size limits of inside flank of the plate

Structure and size limits of inside flank of the plate show in Fig. 5. The line L_1 and outside flank line of the plate was symmetric with the vertical line P_1P_2 through the pin hole center, the line L_2 and outside flank line of the plate was symmetric with the line P_1P_3 through the pin hole center, and the angle between the line P_1P_3 and the symmetry centerline O_1O_2 of the plate was pi/N. The N was the maximum tooth number.

When chain and sprocket meshed in transmission, in order to share contact force of plate and sprocket, improve polygon effect and make sprocket and inside flank of the plate contact firstly, inside flank of the plate should project outwardly relative to the line L_1, and it was in the semi-enclosed space of the line L_1 and L_2 to avoid generating interference case when it and sprocket meshed, therefore, at most, inside flank curve cut with the line L_2. And when the inside flank curve of the plate was closer to the line L_2, it was more obvious that the polygon effect was improved.

SPROCKET PROFILE

So far, there were yet no unified ISO standards of silent chain sprocket. National standards of transmission silent chain and sprocket were most the same as using the American standard ANSI B29.2M and were established in principle.

Silent chain sprocket formulated by the German standard DIN8190 was actually the involute profile of the negative variation coefficient (DIN 8191, 1998), its pressure angle was 30 degrees. The main advantage of using involute profile was that the involute was only formed by a curve, easy processed, not sensitive to center distance error and could make the chain which was worn and elongated and sprocket engage to get better compensation. Method of processing gear with imaginary rack cutter could solve modification coefficient of involute sprocket. When the rack and gear without addendum modification meshed, the middle line of rack n-n was the tangent to gear pitch circle.

When the rack cutter of the pitch P hypothetically was used to process involute sprocket, we need to make the middle line of rack cutter move towards the sprocket center at a certain distance χm, and make the position of the rack cutter and the plate seated on the sprocket the same, as shown in Fig. 6. Modification coefficient could be calculated from the geometric relation (Wangji, 1996):

$$\chi = \frac{\sqrt{3}}{4}\pi + \frac{\pi}{2}\cot\frac{\pi}{z} - \frac{z}{2} - \frac{2\pi f}{p}$$

where p was pitch, f the distance from a pin center to outside flank, pressure angle $\alpha = 60^{\circ}$, z the tooth number of sprocket, and m the module.

DYNAMIC TEST

Test specifications: Test chains used were both inside and outside flank alternate meshing silent chain and standard silent chain. Two kinds of silent chain pitch were P = 15.875 mm, chain link number was Lp = 94. In order to ensure the comparability of test results, technology quality indicators such as structural parameters, material and heat treatment of silent chain and sprocket used in the test were consistent. And plate material was 45 Mn, pin material was 20CrMnMo. Test chain ran 5 min in advance, driving/driven sprocket tooth number were 26, pressure oil-spray lubrication was adopted.

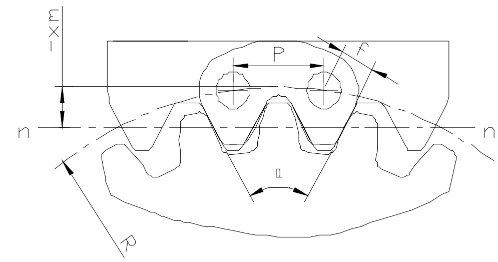

Fig. 6: Rack cutter processing method for the modification coefficient

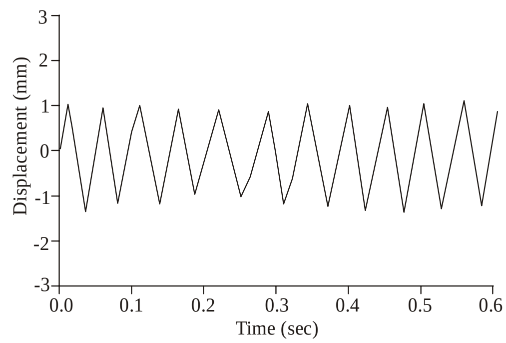

Fig. 7: Transverse vibration displacement of the new silent chain

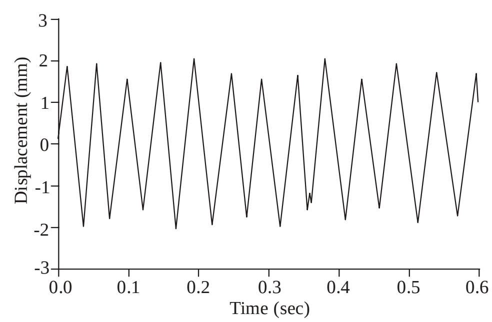

Fig. 8: Transverse vibration displacement of the standard silent chain

Transverse displacement vibration: NAIS laser displacement sensor was used to measure the size of dynamic transverse displacement of silent chain on the closed force flow test bench (Zhengzhi *et al*., 1984; Turnbull *et al*., 1992), and 0.6×1.1 mm laser spot hit on the surface of perpendicular to the plate. A spectrum analyzer was used to record and access to experimental values of chain vibration displacement in driving sprocket speed n1 = 800 r/min, and frequency-domain analysis for signal acquisition was done.

Experimental data from Fig. 7 and 8 could be seen, relatively qualitative analysis between the two chains, that transverse displacement vibration of the new silent chain was smaller and relatively stationary than that of the standard silent chain.

To describe quantitatively different frequency effects on the transverse vibration characteristics of the transmission chain under the same speed, the power spectral density analysis were performed on experimental results. Integral of power spectrum was achieved in the whole frequency domains, the results showed that chain link produced the energy size of different vibration frequencies. Power spectral density function was

$$G_x(k) = \frac{2\Delta t}{N} \left| x_k \right|^2 .$$

where Δt was ampling interval, x_k fast Fourier transform values of the Sampling point x_i, N, sampling points.

Meshing frequency was caused by the polygon effect:

$$f = \frac{nz}{60} = \frac{800 \times 26}{60} = 346.67\,Hz$$

Seen from Fig. 9, close to frequency 350 Hz, the maximum vibration amplitude of power spectral density of the new silent chain was about 0.27 mm^2*s. While in Fig. 10, the same frequency was close to 350 Hz, and the maximum amplitude of the standard silent chain was approximately 0.66 mm^2*s, Both frequency of the maximum amplitude and meshing frequency of polygon effect was generally consistent, Therefore, it was confirmed that the meshing frequency caused by the polygon effect gave rise to the maximum vibration amplitude in the chain drive system. From relatively speaking, it was still clearly shown that motion stability performance of new silent chain was better than that of the standard silent chain.

Wear-resisting performance: For silent chain, their wear-resisting performance indicator measured and evaluated was not wear loss said in tribological field but wear elongation which had a direct impact on the reliability of the chain transmission. At present, countries allowed ultimate wear elongation of silent chain to be $\epsilon = \Delta L / L \leq 3\%$ (Mengfan *et al*., 2004) (ΔL was chain wear elongation, and L was initial chain length). As the chain drive technology and chain product manufacturing technology continued to improve, some industrialized countries had reduced the chain wear elongation to $\epsilon \leq 1.5\%$.

Chain wear curve was got by the running-in test to do wear-resisting performance comparison of the new silent

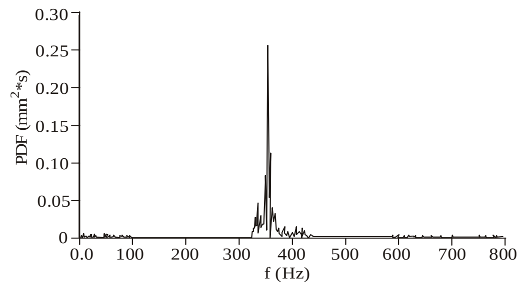

Fig. 9: Power spectral density of the new silent chain

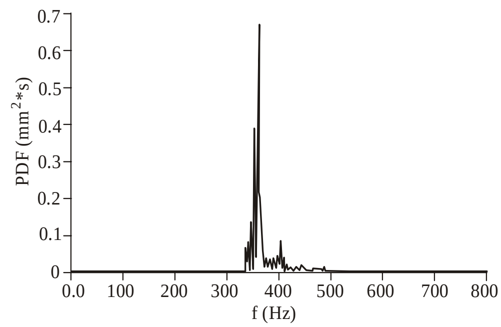

Fig. 10: Power spectral density of the standard silent chain

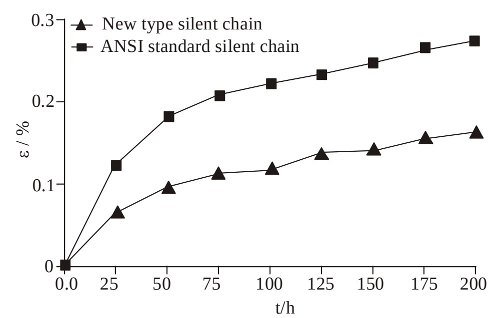

Fig. 11: Wear curves of the silent chain

chain and standard silent chain. Testing time was 200 h and chain length was measured on the Lcc-1500 digital measuring instrument every 25 h. The link number of the two tested chains was 30, and measuring load was 456N.

Wear curve from two silent chains in Fig. 11 could be seen that in the 200 h test time, wear elongation of the standard silent chain was relatively $\varepsilon = 0.279\%$, while that of new silent chain was relatively $\epsilon = 0.162\%$, and wear elongation of the former was approximately 1.7 times than that of the latter. It was worth noting that initial wear elongation of the standard silent chain in the first 50 h was 0.182%, It was approximately 65.2% of the total wear elongation. While the initial wear elongation of new silent chain was only 25 h, and wear curve slope was small, it was about 39.5% of the total wear elongation. In the subsequent 125 h, the normal wear curve slope of new silent chain was lower than that of standard silent chain, and the new silent chain done smooth transition.

To two tested silent chain, material and heat treatment of their plate and pin were the same, and we could see that in the same experimental conditions, wear-resistant characteristics of the new silent chain was significantly better than that of the standard silent chain.

CONCLUSION

From the present study of inside and outside flank alternate meshing silent chain and on correlating the results of dynamic performance evaluation, the following conclusions are made:

- New silent chain has alternate bearing characteristics that inside and outside flank of the plate can mesh with sprockets, while the standard silent chain is not available in transmission type, which is a new-type transmission, and fundamentally improve the performance of high-speed transmission in silent chain.
- New silent chain drives smoothly, its transverse displacement vibration is significantly less than that of the standard silent chain, and the test shows that it can reduce the polygon effect in silent chain drive.
- Two hundred hours wear test shows that wear-resisting performance of new silent chain is better than that of the standard silent chain. Which indicate that the new silent chain has good wear-resisting characteristics, and ensures transmission efficiency and the reliability of the silent chain drive.

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REFERENCES

ANSI B29.2M, 2007. Inverted Tooth (Silent) Chains and Sprockets, the American Society of Mechanical Engineers, New York.

Calvo, J.A., V. Diaz, J.L. San Roman and M. Ramirez, 2006. Controlling the timing chain noise in diesel engines, Int. J. Veh. Noise Vibrat., 2(1): 75-89.

DIN 8191, 1998. Tooth form for chain wheels for inverted tooth chains complying with DIN 8190 Dimensions of profile German Industrial standards, Berlin.

Liu, S.P., K.W. Wang, S.I. Hayek, M.W. Trethewey and K.H.K. Chen, 1997. A global-local integrated study of roller chain meshing dynamics. J. Sound Vibrat., 203(1): 41-62.

Mengfan, Z., Fengzeng, M.L. Tao and Y. Bing, 2004. Experimental research on the wear mechanisms and temperature and speed characteristics of a new silent chain, Tribology, 24(6): 560-563.

Mengfan, Z., 2008. The Meshing Principle of Silent Chain, China Machine Press, Beijing.

Turnbull, S.R., S.W. Nicol and J.N. Faweett, 1992. An Experimental investigation of the dynamic behavior of a roller chain drive. ASME pap n77-DET-168 for Meet, 26-30.

Wada, M.S. Ide, S. Miki and A. Ehira., 1999. Development of a Small Pitch Silent Chain for a Single-Stage Cam Drive System, SAE Technical Paper ,Warrendale, PA.

Wangji, M., 1996. Meshing design of silent chain and involute sprockets machine design, 13(7): 25-27.

Zhengzhi, F., W. Xing and C. Heng, 1984. Chain Drive, Machinery Industry Press, Beijing.