A Study on Strength of Toothed Belt*
(7th report, Wear of Polyurethane Belt under No Load)

By

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No load fatigue test of L-type polyurethane toothed belt for general industries was done.

The direction of working flank is decided by the initial tension and the pitch difference. The initial tension in which the wear is least is a little lower than the tension at which the direction of the working flank changes. The tendency of the relation between belt wear and the initial tension coincides with that of the relation between the friction power of the belt and the pulley at the incomplete meshing region and the initial tension.

Key Words: Machine Element, Synchronous Belt, Fatigue Test,
No Load, Polyurethane Belt, Wear

1. Introduction

There are many cases when toothed belts are used under no load or nearly no load condition e.g. toothed belts for revolution transmission of office automation instruments, computer terminals, point setting of machine tools, or idle pulleys. Generally it is considered that toothed belts used under these conditions are hardly damaged at all and so there have been very few studies done on the life-span of toothed belts under no load conditions.

The authors made a theoretical analysis and performed experiments on the load distribution of toothed belts under no load conditions to study the meshing of the belt teeth and the pulley teeth⁽¹⁾. This is an important factor for estimating a belt's life span. In this paper, the effect of the magnitude of the initial tension on the wear of toothed belts was tested by using L-type polyurethane belts with load-carrying cords of wire rope. The wear of the belts was studied in relation to the amount of the interference between belt and pulley. Polyurethane belts were used because they are generally used for transmission of a light load, and the wear of the belts was

clearly apparent.

From this study, datum for the determination of suitable initial tension was obtained.

2. Experimental Apparatus and Experimental Procedure

2.1 Experimental apparatus

Figure 1 shows the no load fatigue test apparatus for toothed belts used in this experiment. A 5 kw 3 phase induction motor ① drives a driving shaft with both of the pulleys 1 ② at 48.3 s^{-1} . Two specimens ③ were tested at the same

① 5 kW Motor, ② Pulley 1, ③ Test belt,
④ Pulley 2, ⑤ Guide way, ⑥ Ball bearing,
⑦ Screw for initial tension,
⑧ Worm and worm wheel, ⑨ Total counter,
⑩ Anechoic Chamber, ⑪ Air conditioner

Fig.1 No load fatigue test apparatus for toothed belt

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time. Pulley 2 ④ slides in parallel on the guide ways ⑤ with ball bearings ⑥. Test belts receive an arbitrary initial tension produced by the parallel movement of the shaft with the pulley 2. This apparatus was placed in a wooden anechoic chamber 900 mm wide, 1200 mm deep, and 1700 mm high. The temperature in the chamber was kept at $24 \pm 2^\circ\text{C}$ by using an air conditioner ①. The temperature of the pulley teeth was measured in a hole of 2 mm diameter and 10 mm depth drilled in a tooth on the outside surface of the disc by using thermocouples.

Figure 2 shows the apparatus for testing the wear of belt teeth. Specimens taken from the test belt were fixed on the lever. A friction disc made of JIS-SS41 had an outside diameter of 109 mm and a width of 25 mm. The disc speed was varied from 0.2 m/s to 2 m/s and the load P was arbitrarily changed. The temperature could be varied from the room temperature to 200°C by using a heater. The temperature of the disc outside surface was measured by using a thermocouple in a small hole 2 mm in diameter and 10 mm deep drilled 2 mm on the outside edge of the disc.

2.2 Experimental procedure

The initial belt tension T_i was obtained by the first mode natural frequency of transverse vibration of the test belt. The belt tension was checked one hour after the test started and then every 12 hours. At that time the dust from the wear of the belt was cleared away, and the pulleys were adjusted so that the eccentricity of the test belt was minimum. In this experiment, the relation between the number of cycles N and the difference between the masses M of the test belts before and after the fatigue test was determined. The number of cycles N is determined by the following equation:

$$N = \frac{2 \times (\text{Number of pulley teeth}) \times (\text{Total number of revolutions of pulley})}{\text{Number of belt teeth}}$$

By using the wear test apparatus, the relation between the temperature of the disc surface and the mass of wear of belt was determined. The load P was kept constant at 39 N and the sliding speed

Fig.2 Wear test apparatus for belt tooth

was kept constant at 0.2 m/s. Specimens 5 pitches in length were taken from the belt for the testing. The test time was 3 hours and the difference of the mass of the specimen before and after the test was considered to indicate the mass of the wear.

A balance of 200 gs capacity and 1 mg sensitivity was used for the measurement of mass. As the polyurethane belt is highly hygroscopic, specimens were kept in a desiccator for 24 hours before measurement of the mass to eliminate moisture.

2.3 Test belts and pulleys

Figure 3 shows the forms and dimensions of the test belts and pulleys used in this experiment. The belts were standard polyurethane belts used in general industries and available on the market. They had 80 teeth and were 19 mm wide; a 3x3 steel wire rope was used for the load-carrying cords. Though the belts had some scatters of length, which were within the limit of length determined by ISO 5296-1978(E).

The material of the pulleys was soft steel (JIS-SS41), and the pulleys were hobbled. Pulleys with 3 different outside diameters were used. The No.2 pulley is available on the market.

The pitch difference Δt between the belt pitch t_b and the pulley pitch t_p is given as follows:

$$\Delta t = t_b - t_p = t_b - 2(R_p + a)(\sin\phi + \beta)$$

where, ϕ and β are the angles of pulley shown in Fig.3(b). a is called P.L.D.

t_b	a_b	d_b	h_b	m_b	r_b	g_1	g_2	b	t_c	$2a$ rad
9.525	80	1.4	1.9	3.25	0.5	0.75	0.28	19.0	1.0	0.698

(a) Forms and dimensions of test belts

No.	a_p	t_p	$2R_p$	a	r_p	h_p	m_p	$2a$ rad	2ϕ rad	2β rad	Δt
1		9.534	108.33						0.1150	0.0595	-0.009
2	36	9.544	108.44	0.48	0.85	2.68	3.01	0.698	0.1149	0.0597	-0.019
3		9.558	108.62						0.1147	0.0599	-0.033

(b) Forms and dimensions of test pulleys

Fig. 3 Forms and dimensions of test belts and pulleys

(Pitch Line Differentials); this is the distance from the belt pitch line on the pulley to the outside circle of the pulley. This is a very important value which has a great effect on the life-span of the belt.

Pulley 1 and pulley 2 have the same outside diameters in every experiment.

3. Experimental Results and Discussions

3.1 State of belt wear

Fatigue tests of toothed belt drives under applied loading show some features of fracture, what is breaking of the load-carrying cords, shearing of belt teeth, wear of belt teeth, etc. (2) In the case of no load fatigue test, the wear of belt teeth was the only feature.

Figure 4 is a sketch of the state of belt wear. Pulley No.3 was used in this experiment. As $\Delta t = -0.033$ mm, the belt pitch is shorter by 0.033 mm than the pulley pitch. $N = 3 \times 10^7$, about 200 hours, were required for the test.

Figure 4(a) shows that in the case of $T_i = 600$ N, the front tooth flanks of the belt (the tooth flanks on the belt running direction) wore out. The depth of wear is about 0.3 mm. There is a shallow groove near the root of every tooth caused by the interference of the pulley tooth tip. Wear marks can also be found on the bottom land of the tooth groove.

(b) shows that in the case of $T_i = 800$ N, the front tooth flanks of the belt were only slightly worn. A little burnishing caused by the contact of the pulley top land was found on the bottom land of the tooth groove.

(c) shows that in the case of $T_i = 1100$ N, the back tooth flanks of the belt (the opposite side tooth flanks in the belt running direction) were worn out. The depth of wear was about 0.8 mm. Slight wear marks were found on the bottom land of the tooth groove.

As shown above, the place and amount

Fig. 4 State of wear of belts ($\Delta t = -0.033$ mm), $N = 3 \times 10^7$ (200 hours)

of wear differ according to the magnitude of initial tension. These differences can be considered to be caused by 1) the interference between belt teeth and pulley teeth, 2) the tooth load, or 3) the amount of slippage of the belt teeth and the pulley teeth in the incomplete meshing region.

Figure 5 shows the amount of interference λ_k between the belt teeth and the pulley teeth. Table 1 shows the numerical values used in the calculation. f_k is the compliance of the belt tooth; b_L , the back-lash; μ , the coefficient of friction between the belt tooth groove and the pulley tip cylinder and SE indicates the modulus on the creep of the belt wound on the pulley. The calculation was the same as described in the paper (1).

When the back tooth flank of the belt receives interference, λ_k is indicated as plus, and when the front tooth flank of the belt receives interference, it is indicated as minus.

The abscissa of Fig. 5 indicates the tooth number. Tooth number 3 is the first tooth in the complete meshing region and the following teeth are numbers 4, 5, etc in the direction of belt's motion. The complete meshing of the belt tooth and the pulley tooth ends at the tooth number 21. The spaces between the tooth numbers 2 and 3, and between 21 and 22 are the incomplete meshing region. At the onset of meshing in the incomplete meshing region, the front tooth flank of the belt contacts with the pulley tooth flank. The region where both flanks make a sliding contact is in the space from the starting point of the contact to ϕ degrees before tooth number 3. The back tooth flank of the belt makes a sliding contact with the pulley tooth flank in the space from the

Table 1 Numerical values used in calculation

SE kN	265	μ	0.65
f_k $\mu\text{m/N}$	1.83	b_L mm	0.36

Fig. 5 Calculated results of amount of interference

starting point of the contact tooth number 3.

The solid line in Fig.5 shows the amount of interference at $T_i=600$ N. The interference occurred on the front tooth flanks of the belt. This coincides with the experimental result. The maximum value of the interference is shown at the tooth number 3 with λ_k about 0.12 mm. After tooth number 19, λ_k becomes 0. This means that the belt and the pulley teeth do not contact with each other in this part. It is considered that the wear of the belt teeth occurs in the process of the sliding contact between the point a half pitch before tooth number 3 and the point 1/3 pitch before tooth number 3. We considered that the magnitude of the wear is affected by the amount of interference, the tooth load and the length of sliding contact in this process. In the regions from the point 1/3 before the tooth number 3 to the tooth number 3 and from tooth number 3 to tooth number 18, the belt teeth are not worn because of the lack of sliding contact.

In the case of $T_i=800$ N, the front of all the belt teeth receive almost equal interference. The sliding contact region at the onset of meshing is from 2/5 pitch before tooth number 3 to 1/3 before tooth number 3. This region is very short and the maximum amount of interference is smaller than that in the case of $T_i=600$ N. There is an incomplete meshing region at the end of meshing. However, considering the state of wear and the pulley temperature mentioned later, this incomplete meshing region has little effect on the wear of belt teeth.

In the case of $T_i=1100$ N, the interference occurs at the back tooth flank of the belt. The experimental results also prove this. The maximum amount of interference occurs at the incomplete meshing region between tooth number 2 and tooth number 3, and it is about 0.11 mm. The sliding contact part is between the point 4/5 pitch before

tooth number 3 and tooth number 3. This part is the largest.

In all the three cases, the interference occurs only on one side of the tooth. This coincides with the experimental results. In general, at low tension range the interference occurs on the front tooth flank, and at high tension it occurs on the back tooth flank. The tension when the interference change from one side to the other side is equal to the tension when the extended belt coincides with the pulley pitch.(3) That is, when $t_b'=t_p$, the tooth flank receiving interference changes to the other tooth flank. In this case, t_b' is the belt pitch extended by the tension. This tension is called the tension at the change of mating flank, T_c , hereafter.

3.2 Amount of wear

Figure 6 shows the relation between the amount of the wear of the belt and the number of cycles N , where $T_i=450$ N is kept constant and Δt is taken as a parameter. The dark marks indicate the wear of the back tooth flank of the belt, and the white marks indicate the wear of the front tooth flank of the belt. In every figure on the amount of wear hereafter the same system of marks are used.

The amount of the wear of the belt is greatly affected by Δt even though T_i is kept constant. In the case of $\Delta t=-0.019$ mm, the amount of wear is small and the rate of the amount of wear against the number of cycles is very small after $N=2 \times 10^7$. In the case of $\Delta t=-0.009$ mm, the amount of wear is the largest and the rate of wear is constant regardless of the number of cycles. In the case of $\Delta t=-0.033$ mm, the rate of wear shows the same tendency as when $\Delta t=-0.009$ mm up to the point of $N=1.5 \times 10^7$. When the number of cycles is larger than $N=1.5 \times 10^7$, the rate of wear shows the same tendency as when $\Delta t=-0.019$ mm.

Fig.6 Relation between amount of wear of belt and number of cycles (T_i =constant)

Fig.7 Relation between amount of wear of belt and number of cycles (Δt =constant)

Figure 7 shows the relation between the amount of the wear of the belt and the number of cycles, where $\Delta t = -0.033$ mm is constant and T_i is a parameter. This figure shows the same results as in Fig. 6.

From these figures we can see that the wear of the back tooth flank of the belt increases linearly with an increase in the number of cycles. In the case of wear of the front tooth flank, the rate of wear decreases with an increase of the number of cycles.

During the fatigue test, the pulley pitch t_p decreased. This was due to 1) the initial elongation of the load-carrying cords; 2) the decrease of the distance from the bottom land of the belt groove to the center of the load-carrying cords caused by wear and compressive residual deformation; 3) better meshing resulting from the wear of belt teeth and so on.

Figure 8 shows the relation between the tension at change of mating flank T_c and the number of cycles N . In every case, T_c decreases gradually with an increase of N , and the rate of decrease of T_c becomes very small after $N = 2 \times 10^7$. The rate of decrease of T_c at $T_i = 450$ N, when the wear is on the front tooth flank, is larger than that of T_c at $T_i = 1100$ N. This is because, in the former case, the tooth tip of the pulley makes contact with the bottom land of the belt tooth groove before the interference between the belt tooth and the pulley tooth occurs; and, while the sliding contact between both of the teeth flanks occurs the bottom land of the belt tooth groove is worn out by the pulley tooth tip. Therefore, the wear of this portion is greater than in the later case. This can also be determined from the state of belt wear. In the case of $T_i = 450$ N, T_c , which was equal to 1035 N at the start of testing, decreased to 790 N at $N = 0.75 \times 10^7$ and decreased to 650 N at $N = 2.25 \times 10^7$. The change in T_c coincides with the change in the amount of wear at $T_i = 450$ N as shown in Fig. 7. That is, when N is over 1.5×10^7 , the value of T_c approaches the value of T_i , and the amount of wear becomes very small at this point. In the case of $T_i = 1100$ N, the change of T_c is

Fig. 8 Relation between T_c and N

comparatively small, and so the meshing state does not change much with time. Therefore, the amount of wear increases linearly.

Figure 9 shows the difference of T_c before and after the test. The number of cycles N is 3×10^7 . Although the center pitch length of all the new test belts was within the pitch length tolerance by ISO 5296-1978(E), there was some scatter of length from 890 N to 1035 N which is expressed in T_i . T_c of the belts in any tension after the test decreases compared with that before the test. The dotted line shows ΔT_c , the difference between T_c before and after the test. ΔT_c is small when the initial tension T_i is close to T_c before the test. This means that in the tension range where the interference between the belt teeth and the pulley teeth is small, the wear at the bottom land of the belt tooth is small and consequently the change of T_c is small.

Figure 10 shows the calculated results of the interference λ_k of the belt teeth before and after the test. All the conditions used in the calculations were the same except that the difference between Δt before the test and Δt after the test is 0.014 mm at $T_i = 450$ N and 0.005 mm at $T_i = 1100$ N. This is based on the measurement of ΔT_c as shown in Fig. 9.

In the case of $T_i = 450$ N, the maximum amount of the interference before the test is about 0.16 mm, and the contact point of the belt tooth with the pulley tooth at the onset of meshing is 3/5 pitch before the tooth number 3. The maximum amount of interference after the test is 0.08 mm and the contact point is 2/5 pitch before the tooth number 3. Namely, the maximum amount of the interference becomes half after the test and the sliding contact almost disappears. This is considered to be the reason why the progress of the wear is much slower when N is greater than 1.5×10^7 .

Fig. 9 Relation between change of T_c and T_i

In the case of $T_i = 1100$ N, the maximum amount of the interference increases about 20% after the test compared with that before the test. However, as the length of the sliding contact hardly increases at all, the progress of the wear is not shown in Fig.7.

From the above, it is considered that the amount of interference λ_k between the belt tooth and the pulley tooth and the tooth load Q_k have a great effect on the wear of the belt tooth. An example of the calculated results on λ_k and Q_k is shown in Fig. 11.

Figure 11 shows the relation between the initial tension T_i and the amount of the interference λ_3 as well as the tooth load Q_3 on the first tooth (tooth number 3) in the complete meshing region. In each case, both λ_3 and Q_3 are discontinuous at the tension T_c . If the tension is higher than T_c , the back teeth flanks interfere with the pulley teeth,

Fig.10 Change of amount of interference between before and after test

Fig. 11 Relation between amount of interference and tooth load at tooth number 3 and initial tension

and λ_3 and Q_3 increase linearly with an increase of T_i . If the tension is lower than T_c , the front teeth flanks interfere with the pulley teeth, and λ_3 and Q_3 increase with a decrease of T_i . If T_i is lowered even more, λ_3 and Q_3 will decrease. However, in such a low tension range the belt teeth and the pulley teeth do not get a normal meshing, because the belt teeth climb onto the pulley teeth. For example, in the case of $\Delta t = -0.019$ mm, $\lambda_3 = -0.075$ mm and $Q_3 = 43$ N at $T_i = 300$ N.

Figure 12 shows the relation between the tooth surface temperature τ of the pulley 1 and the initial tension T_i . In this experiment, one test belt was used and T_i was varied. τ was measured one hour after the beginning of the test when the surface temperature becomes stable.

τ is the lowest at $T_i = 800$ N, and it becomes higher when T_i is over or below 800 N. We can easily assume from Figs.7 and 10 that the rising temperature of the pulley is caused by the interference in the incomplete meshing region. The reason why T_i at the minimum τ is a little lower than $T_i = 900$ N is that T_c is reduced during the test and that τ is influenced by the frictional heat of ball bearings at high tension range.

Figure 13 shows the relation between m , the mass of wear of belt tooth tested

Fig.12 Relation between temperature of pulley tooth and initial tension

Fig.13 Relation between temperature of disc surface and amount of wear of belt tooth

by the experimental apparatus shown in Fig. 2, and the temperature of the disc surface τ' . m is the mass of wear of the belt tooth per 1 km of sliding contact with the disc. In general, m has a tendency to increase with an increase of τ' , but the wear of the belt tooth is independent of the disc temperature within the range of 40-60 °C, which is the range shown in Fig. 12. Consequently, it is considered that the wear of belt shown in Figs. 6, 7, and 14 is not affected very much by the pulley temperature.

In Fig. 14, the pitch difference Δt was taken as a parameter and this figure shows the relation between the wear of the belt M and the initial tension T_i . M was measured at $N=3 \times 10^7$. When $M \approx 0.1$ g, the belt teeth wore little. However, when $M \approx 3$ g, the teeth were worn out to about 30% of their thickness.

The three lines in Fig. 14 show the calculated results of the friction power $W = \mu' Q_{\max} v_m$ of the interference between the belt teeth and the pulley teeth at the incomplete meshing region. μ' is the kinetic coefficient of friction between the belt teeth and the pulley teeth. $Q_{\max}$ is the maximum tooth load at the incomplete meshing region and is equal to Q_3 . v_m is the mean sliding velocity which can be obtained by dividing the length of sliding contact on the belt tooth flank by the time. We do not consider this W as a theoretical value of the amount of wear, but we measure it to compare the effect of the initial tension on the wear of the belt with the rate of the frictional work at the incomplete meshing region.

T_c of each Δt is indicated by the discontinuous lines (vertical lines). In this case of $\Delta t = -0.009$ mm and -0.019 mm, there is a tension range where $W=0$, due to the lack of the sliding contact in this tension range. As mentioned above, the calculation was made using the

standard belt dimensions, even though T_c is not constant due to the scatter of the belt pitch and it decreases with the progress of the wear of the belt. The experimental results show that in the case of each Δt the front tooth flank of the belt wore out in the tension range lower than T_c , and the opposite side wore out in the tension range higher than T_c . However, in the case of $\Delta t = -0.009$ mm at $T_i = 200$ N, the back tooth flank wore out because the belt pitch was longer than the standard belt pitch, and in the case of $\Delta t = -0.033$ mm at $T_i = 900$ N, the same result was obtained because T_c decreases during the experiment.

The amount of the wear of the belt becomes minimum when T_i is set a little lower than T_c . This is because T_c decreases and becomes equal to T_i owing to the wear on the bottom land of the tooth groove, the fit of meshing, the elongation of the load-carrying cords of the belt, and so on. Consequently, the amount of the interference or the tooth load in the incomplete meshing region is very little.

In the case of the wear of the back tooth flank, the mass of the wear M increases with an increase of T_i . This result coincides with the tendency of the calculated result of W . In the case of the wear of the front tooth flank of the belt, the tendency of M at the low tension does not coincide with W . For example, in the case of $\Delta t = -0.019$ mm and $\Delta t = -0.033$ mm, M increases suddenly at $T_i = 250$ N and at $T_i = 400$ N, respectively. Although W is calculated assuming that the belt pitch line wound on the pulley does not change according to the magnitude of T_i , actually the pitch line does change according to T_i . The change of the belt pitch line is caused by the interference between the belt tooth and the pulley tooth at the onset of meshing, the bending rigidity of the belt and so on. The change of the belt pitch line is

Fig. 14 Relation between amount of wear of belt and friction power at incomplete meshing region and initial tension

remarkable at low tension range. This change takes the form of the belt teeth climbing onto the pulley teeth. In the low tension range where this phenomenon appears, the pressure angle of the belt tooth changes with the progress of the wear of the belt tooth, and consequently, the belt teeth climb even farther on the pulley teeth. $T_L=250$ N of $\Delta t=-0.019$ mm and $T_L=400$ N of $\Delta t=-0.033$ mm are the limits at which the unusual wear occurs. In the tension range lower than these limits the belt tooth can not get normal meshing with the pulley tooth, and consequently, the load-carrying cords are broken or the belt teeth are worn out beyond recognition before reaching $N=3 \times 10^7$.

From Fig. 14 it is clear that the initial tension must be set at 10-20 % lower than the value of T_C for the minimum wear of the belt. Except for the case of $\Delta t \approx 0$ the belt can not be used at the tension range one-half of T_C because of the unusual wear of the belt teeth. When belts on the market are used in combination with pulleys on the market, T_C shows different values as shown in Fig. 9, because both of them have different values of pitch. T_C is determined as follows: 1) apply a certain initial tension on the belt, 2) make the pulley rotate several times in one direction, 3) change the initial tension inturn and repeat the action of 2), and 4), find out the tension where belt tooth flank in contact with the pulley tooth flank switches to the opposite side.

4. Conclusions

The following results have been obtained from the no load fatigue tests of L-type polyurethane toothed belts for general industries.

(1) Belts are worn out by the interference at the onset of meshing, even under a no load condition.

(2) The front tooth flank of the belt wears out at low tension, the back tooth flank wears out at high tension.

The tension where the mating tooth flank switches is obtained when the extended belt pitch equals the pulley pitch.

(3) The tension where a mating tooth flank switches decreases with an increase of the number of cycles. The amount of decrease of this tension is small when the initial tension is close to the original tension where the mating tooth flank switches.

(4) The qualitative tendency of the relation between the amount of wear of belt and the belt tension coincides with the relation between the frictional power at the incomplete meshing region and the initial tension except for the tension range where the belt teeth climb onto the pulley teeth.

(5) In order to minimize the amount of wear, it is necessary to set the initial tension 10-20 % smaller than the tension at the change of mating tooth flank. This tension coincides with the minimum value of the frictional power when the tooth flank wears out a little.

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