621.85::534.83

Paper No. 190-16

Some Methods to Reduce Noise in Toothed Belt Drives*

By

Masanori KAGOTANI**, Toshio AIDA***, Tomio KOYAMA⁺Susumu SATO⁺⁺, Takeshi HOSHIRO⁺⁺⁺

The noise level of toothed belt drives is affected by the interference of belt and pulley teeth at the point where meshing on the driving pulley begins. A preferable amount of interference can be obtained by a suitable selection of the pitch difference between belt and pulley. The smaller the radius of the rounded tip corners of the pulley teeth, the lower the noise of the belt. Vibration due to the impact of the belt and pulley teeth at the point where meshing begins is almost never transmitted to the pulley.

1. Introduction

There have been very few systematic studies on the noise produced by toothed belt drives. However, one study by Kubo, *et al.*^{(1),(2)} investigated the characteristics of such noise, the mechanisms which generate it, and proposed several effective methods to reduce it.

This report discusses the effects of interference at the point where meshing begins on noise reduction in toothed belt drives. Then, the results of the effects of the impact of meshing on pulley vibration, and the relation between the belt noise and the radius of the rounded tip corner of the teeth are described.

2. Testing apparatus and experimental methods

2.1 Experimental apparatus Figure 1 shows a testing machine of the power circulating type used to measure noise and vibration in toothed belt drives. The machine is driven by a non-step speed variator ^⑭ with a 4 kw induction motor and the speeds of the pulley vary from 400 rpm to 3600 rpm. Both of the driving and driven pulleys are the L-type and both have 36 teeth. Arbitrary tension is applied to the test belt ^① by torque coupling ^⑤. Sliding bearings and torsion bars are used to reduce and isolate noise and vibration. In order to isolate the ambient noise, the test portion was placed in a simple anechoic chamber. However there was no way in which to isolate the noise which radiated from the cast iron bed ^⑧ and the concrete foundations ^⑨. The walls ^⑪ of the

anechoic chamber are made of a wood plate 50 mm thick which contains an air layer. The inner surface is lined with an identical thickness of glass wool. The sound isolation is more than 35 dB at frequencies above 400 Hz and sound absorption above 200 Hz is sufficient. In order to further isolate the ambient noise, a microphone was placed near the test belt.

Figure 2 shows a block diagram of the system used to measure belt noise and the circumferential vibration acceleration of the driving pulley. A one inch condenser microphone ^① was located at a distance of 180 mm (x=100 mm, y=100 mm, z=100 mm) from the driving pulley and was directed to the point this where the meshing began. The circumferential vibration acceleration of the driving pulley was detected by a 2 gf vibration pickup ^② which was screwed to the side of the pulley 40 mm from the center. The vibration signal was measured through a slip ring ^④. The frequency characteristic of the vibration pickup was flat between 1 and 21000 Hz. Both noise and vibration signals were recorded on data recorders ^⑦ and ^⑧, and the data were analyzed by a frequency analyzer ^⑩. The slip ring was set so that it measured only vibration.

① Test belt	⑨ Concrete foundation
② Power circulating belt	⑩ Anechoic chamber
③ Driving pulley	⑪ Wood wall
④ Driven pulley	⑫ Absorbing material
⑤ Loading device	⑬ Flat belt
⑥ Torsion bar	⑭ Non-step speed variator with motor
⑦ Plain bearing	⑮ Pump unit
⑧ Cast iron bed	

Fig.1 Toothed belt noise testing rig of the power circulating type

* Received 9th July, 1979.

** Assistant, Osaka Industrial University, 3-1-1, Nakagaito, Daito-shi, Osaka.

*** Professor, Osaka Industrial University.
+ Lecturer, Osaka Institute of Technology, 5-16-1, Omiya, Asahi-ku, Osaka.

++ Professor, Kyoto University, Yoshidahonmachi, Sakyo-ku, Kyoto.

+++ Manager of Power Transm. Eng. div., Mitsubishi Belting Ltd., 4-1-21, Hamazodori, Nagata-ku, Kobe.

2.2 Test belt and pulley dimensions

Figure 3 shows the dimensions of the test belt and the pulleys and the pitch difference, Δt . The test belt was a L-type polychloroprene rubber belt with 92 teeth and a width of 19 mm. Its load-carrying cords were made of fiberglass. The pitch difference, Δt , between belt and pulley pitches is given by (see Ref. (3))

$$\Delta t = t_B - (2R_p + 2c) \left(\sin \phi + \frac{\pi \beta}{180} \right) \dots \dots \dots (1)$$

The pulley pitch can be changed by shifting the hob for the manufacture. The scatter of pressure angle, α , of each pulley was within an error 5 % from angle α of pulley No.6 taken as a standard.

2.3 Experimental methods Noise and vibration levels were measured every 20 rpm for low speeds, every 50 rpm for middle speeds, and every 100 rpm for high speeds. The initial tension of the test belt was set at a little less than half the sum of the tight side and slack side tensions. The slack side tension was generally set at about 5 kgf.

Method of measurement of belt tension was as in Koyama⁽⁴⁾.

3. Noise reduction methods

The sound pressure wave forms of toothed belt drives are similar to those of roller chain drives,^{(5),(6)} as shown in Fig. 4. The most powerful component in their noise is the sound of the impact of meshing which is generated at the point of contact with the driving pulley. It seems that the belt noise is a point source. There are two methods to reduce noise in toothed belt drives. The first is to reduce both the peak level $P_{\max}$ and the damping time of the impact sound. The second method is to insulate and absorb belt noise by using enclosure and sound absorptive materials. The meshing impact noise of toothed belt drives is thought to be generated when the top land of a pulley tooth collides with the bottom land of a belt tooth⁽¹⁾. In the next section, the effect of vibration caused by meshing impact on the vibration of the pulley will be discussed.

- ① Microphone
- ② Acc. vibration pickup
- ③ Precision sound level meter
- ④ Slip ring
- ⑤ Preamplifier
- ⑥ Measuring amp.
- ⑦ Data recorder
- ⑧ Data recorder
- ⑨ Electromagnetic oscillograph
- ⑩ Frequency analyzer
- ⑪ High speed level recorder

Fig.2 Block diagram of apparatus for measuring vibration and noise

Figure 5 shows belt noise and circumferential vibration acceleration of the driving pulley. The ordinate is the noise level in dB(A) of the test belt (upper part of the figure) and the circumferential vibration acceleration level in dB(A) of the driving pulley (lower part of the figure), and the abscissa is the pulley speed in rpm. Vibration acceleration level, A_{ee} , is expressed by

$$A_{ee} = 20 \log \frac{a}{a_0} \dots \dots \dots (2)$$

where a_0 is standard acceleration : $a_0 = 0.1 \text{ cm/s}^2$. The driving pulley tested was No.6. In the case without the test belt (b), the tested belt was taken away from the rig and the shafts were driven by the power circulating belt only. The noise levels at pulley speeds under 900 rpm are constant due to the effects of ambient noise. At speeds above 900 rpm, the noise level increases as pulley speed increases due to the increasing radiation of sound for example, off the iron bed and the test stand. The vibration level in

Z	m_B	h_B	s_B	r_B	t_B	b	t'	c
92	1.4	1.95	3.25	0.5	9.525	19.0	0.75	0.45

(i) Form and dimensions of test belt

No.	$2R_p$ mm	m_p mm	h_p mm	r_p mm	2α deg.	2β deg.	2ϕ deg.	Δt mm
1	108.03	3.05	2.68	0.85	40.0	3.70	6.30	+0.022
2	108.14			0.85		3.71	6.29	+0.012
3	108.25			0.85		3.72	6.28	+0.003
4	108.37			0.85		3.72	6.28	-0.008
5	108.47			1.25		2.74	7.26	-0.018
6	108.48			0.85		3.73	6.27	-0.018
7	108.49			0.55		3.88	6.12	-0.018
8	108.61			0.85		3.73	6.26	-0.028
9	108.72			0.85		3.73	6.26	-0.038

(ii) Form and dimensions of test pulley

Fig.3 Forma and dimensions of test belt and pulley

Fig.4 Sound wave form of toothed belt drives (non-resonant)

the case of (b) increases with pulley speed because of such factors as the vibration of the bearings and the foundation, the effect of gravity acceleration as the pulley rotates, the error of transmission, and the whirling of the shafts. In the case of (a), the tight side tension T_0 is equal to 70 kgf. Some clear peaks were observed in the vibration level. This comes from the resonance of the transverse vibration of the belt with the meshing frequency. Out of the regions of resonance, when the pulley speed is under 2000 rpm, the level of (a) is 5~10 dB(A) higher than that of (b), but over 2000 rpm, both are almost the same. This means that the meshing impact between belt and pulley does not affect pulley vibration very much. Also, the circumferential vibration acceleration of the toothed pulleys is lower than that of the gears⁽⁷⁾. It appears that the vibration of toothed pulley is also lower than that of roller chain sprockets, because the toothed belt mass is generally smaller. On the other hand, the noise level of (b) is a constant 54 dB(A) at pulley speeds under 900 rpm. At speeds above 900 rpm, it reaches a maximum 70 dB(A) because of sound radiation from such sources as mentioned above. The noise level of (a) is more than 20 dB(A) greater than that of (b). The above suggests that the noise of toothed belt drives is generated from the belt itself and that sound radiation from pulleys is negligible. This means that the vibration from the impact of belt and pulley is almost never transmitted from the pulleys to other parts through the shafts and bearings, and that belt noise can be easily decreased by

Fig.5 Noise and circumferential vibration of pulley

insulation and absorption. Moreover, as the predominant frequency of belt noise is high⁽¹⁾, sufficient noise reduction can be expected even if only light insulation and absorption materials are used as the enclosure.

As the purpose of this study was to find methods to reduce the noise produced by the impact of meshing, various pulleys were measured for noise. On the basis of the results obtained from these experiments, possible methods are discussed below.

4. Experimental results and discussion

4.1 The effect of the velocity of meshing impact on belt noise Figure 6 shows the relationship between noise level and pulley speed. Pulley No.8 was tested with the tight side tension T_0 set at 30 kgf. Line (a) of Fig. 6 represents normal test conditions. Noise level found to increase almost linearly with an increase in the pulley speed and it takes some peak values at the point where meshing frequency coincides with higher order natural frequencies of the belt's transverse vibration. For example, the peak at 600 rpm is caused by the resonance of the third mode natural frequency (360 Hz) with the meshing frequency. Line (b) shows noise level when the loop of the vibrating tight side of the belt is pressed lightly by the outer race of the small roller bearing. When resonance is curtailed in this way, the curve of the noise level coincides with line (c) a straight line. A straight line (d) shows the relation between sound pressure level and pulley speed, where sound pressure is assumed to be proportionate to both meshing impact velocity and the number of instances of such impact within a limited time. Line (d) can be expressed by the following Eqs.

$$P = 20 \log \frac{n}{400} + 10 \log \frac{n}{400} + 60 \quad \dots\dots\dots (3)$$

$$P = 20 \log \left(\frac{n}{400} \right)^{1.5} + 60 \quad \dots\dots\dots (4)$$

where P is the sound pressure level in dB, i.e. 60 dB at 400 rpm, and n is the pulley speed in rpm. The first term on the right-

Fig.6 Effect of meshing impact on belt noise on belt noise

hand side of Eq.(3) shows the effect of meshing impact velocity and the second term the effect of the number of instances of such impact.

The impact velocity, v_i , between the bottom land of the belt and the top land of the pulley is expressed by ⁽¹⁾

$$v_i = 2|v_b| \sin \phi \dots \dots \dots (5)$$

where v_b is the peripheral speed of the belt and ϕ is the angle between the center of the rounded tip corner of the pulley tooth to the center of the pulley (Fig.3). Although the incline of (d) is 1.5, both (b) and (c) are 1.7. This is because meshing impact velocity can not be expressed as simply as in Eq.(5) due to the effect of the transverse vibration phase of the belt and the bending rigidity. The harmonics of the meshing frequency at high speeds coincide with the damped natural frequency of the belt noise (Fig.4), and the mechanical properties of polymers generally increase as the speed of deformation increases ⁽⁹⁾.

(i) No.1 pulley, $\Delta t = +0.022$ mm

(ii) No.8 pulley, $\Delta t = -0.028$ mm

Fig.7 Effect of pulley speed on belt noise

When meshing frequency coincides with the transverse vibration of the belt, the predominant belt noise frequency is different from that of the noise produced when there is no resonance. As the noise in resonance radiates from the entire length of the tight side of the belt, we cannot treat it as a point source. The noise level increment in a resonant condition is therefore greater than that of v_i . Figure 6 also shows this. As already stated, noise created by resonance that can be easily prevented is not a serious problem.

4.2 The effects of tight side tension and pitch difference on belt noise

Figure 7 shows the relationship between tight side tension T_0 and belt noise. In the case of (i), the pitch difference between the belt and pulley is positive where pulley No.1 is used, and in the case of (ii), it is negative where pulley No.8 is used. Both noise level curves increase in proportion to pulley speed with some peak levels at the resonant point and the inclines are about 1.7, as in Fig.6. When T_0 is less than 15 kgf, the effect of resonance on the noise level is not so marked. This is because the natural frequencies of transverse vibration of belt are low under low levels of tension, thus the value of dB(A) is also low. Moreover the force of the impact of meshing is weaker on account of the bending rigidity of the belt.

Though the noise levels of (i) increase with increasing tension T_0 , the noise levels above 50 kgf of (ii) do not change as noticeably.

Figure 8 shows the relationship between tight side tension T_0 and noise level. The pulley speed is 1000 rpm but the noise level increment accompanying resonance is not taken into consideration. Dotted line (a) represents a case of positive pitch difference. The noise level increases with increased tight side tension T_0 . In case of (b), noise level increases rapidly below 30 kgf and its incline is greater than that of (a). The noise level shows the maximum value at 30~40 kgf, but from 40 to 80 kgf, it does not increase even if the tension increases. Above $T_0 = 80$ kgf, the noise level increases with an incline

Fig.8 Effect of tight side tension on belt noise

equal to (a). Comparing the noise levels of (a) and (b), (b) is higher when T_0 is less than 60 kgf. When T_0 is greater than 60 kgf, (a) is higher. This means that if a suitable pitch difference is selected depending on the change in operating tension, a certain amount of noise reduction is possible.

Figure 9 shows the relationship between pitch difference and noise level. In case of high tension, such as (a), noise level shows the minimum value at $\Delta t = -0.02 \sim -0.03$ mm, and when pitch difference increases to positive direction, noise level increases gradually. In case of low tension, (b), decreases when pitch difference increases to positive direction, but when there is too much of a positive pitch difference, the noise level remains high. By comparing (a) and (b), it is evident that if positive pitch difference is selected at low tension and negative pitch difference at high tension, good results about noise reduction will be obtained.

Since belt noise is usually generated when the top land of pulley teeth collide with the bottom land of the belt, the meshing behavior of the belt and pulley at the meshing point should be considered.

Figure 10 shows the meshing state of both belt and pulley at the point where the belt tooth begins to mesh with the driving pulley tooth. Good results in noise reduction may be obtained if the collision velocity v_i is reduced by interference by the belt tooth flank with the pulley tooth flank before the top land of the pulley tooth collides with the bottom land of the belt. The interference quantity $\lambda(x)$ is defined as the length of overlap of a No.3 pulley tooth and a belt tooth in x -direction. Though interference begins before the tooth position shown in Fig.10 is reached, in order to simplify the comparison, the relation between $\lambda(x)$ at the

Fig.9 Effect of pitch difference on belt noise

Fig.10 Meshing between belt and pulley

position shown in Fig.10 and the tight side tension and pitch difference is discussed.

Figure 11 shows the relationship between $\lambda(x)$ and T_0 . $\lambda(x)$ was calculated as in an earlier paper⁽⁹⁾. Such values as the modulus of elasticity of belt, E , the compliance of belt tooth, $f(K)$, and the coefficient of friction between belt and pulley, μ , in Koyama,⁽¹⁰⁾ were also used here. If the values of $f(K)$, E , μ , and slack side tension are constant, $\lambda(x)$ increases with an increase of the tight side tension T_0 . In the case of low T_0 and a negative Δt , $\lambda(x)$ is equal to 0. Therefore in the case of Fig.11(b), the top land of tooth No.3 on the driving pulley collides directly with the bottom land of the belt tooth until $T_0 = 32$ kgf, but above this level, the tooth cannot mesh without interference.

The results in Fig.8 can be considered looking at Fig.11. In the case of $\Delta t = +0.022$ mm, the noise level at $T_0 = 10 \sim 30$ kgf almost never changes. This is because the increment of impact force due to an increase in the tight side tension is mitigated by a decrease in impact velocity v_i due to an increase in interference quantity $\lambda(x)$. Above a tension of 30 kgf, noise reduction is not very effective because interference $\lambda(x)$ is greater than 0.2 mm. In the case of $\Delta t = -0.028$ mm, the noise level increases rapidly until $T_0 = 30$ kgf when interference begins. The noise level at the tight side tension range of 30 kgf to 80 kgf has a tendency to decrease slightly as tension increases. In this range, $\lambda(x)$ increases from 0 to 0.2 mm and as a result of this interference, the increment of the impact force is mitigated. At levels over 80 kgf, noise level increases with same incline at $\Delta t = +0.022$ mm, and in this case, $\lambda(x)$ is greater than 0.2 mm. As the interference $\lambda(x)$ affects noise level, the latter is given in terms of $\lambda(x)$.

Figure 12 shows the relationship between the noise levels of four kinds of pulleys and $\lambda(x)$. Pulley speed is 1000 rpm, and the noise level increment caused by resonance is omitted. The noise levels for all pulleys increase with $\lambda(x)$ in the high tension range ($\lambda(x) > 0.3$ mm). Noise levels in the low

Fig.11 Relation between tight side tension and interference

tension range ($\lambda(x) < 0.3$ mm) depend on pitch difference. If we select a suitable pitch difference and make $\lambda(x)$ 0.1~0.2 mm, noise reduction can be adequately obtained.

The above results demonstrate that the interference between the tooth flanks of both belt and pulley gives good results in terms of reducing noise in toothed belt drives. A $\lambda(x)$ of 0.1~0.2 mm is adequate for a L-type polychloroprene rubber belt.

As $\lambda(x)$ is affected by such factors as tension, pitch difference, the number of meshing teeth and dimensions and materials of belt, the calculation is not simple. However, if positive pitch difference at low tension levels and negative pitch difference at high tension levels are used good results can be obtained.

In the case of the interference at the meshing point, the belt and pulley collision velocity equation may differ from Eq.(5). Though the first contact point between belt tooth and pulley tooth can be decided from Koyama⁽⁹⁾, the collision velocity at the first contact point cannot be given by Eq.(5). Such problems will be left for future research.

Fig.12 Relation between belt noise and interference

Fig.13 Effect of radius of rounded tip corner of pulley tooth on belt noise

4.3 The effect of rounded tip corner of pulley on belt noise Figure 13 shows the effect of the radius of rounded tip corner of pulley teeth r_p on belt noise. The tight side tension T_0 is equal to 30 kgf and the pitch difference Δt is equal to -0.018 mm. The radius r_p of (a) is 1.25 mm, (b) 0.85 mm, and (c) 0.55 mm. The noise level of (a) is slightly higher than those of either (b) and (c).

With a change in the radius of rounded tip corner, the angle 2ϕ change (Fig.3). The collision velocity equation, i.e., Eq.(5), depends on angle ϕ . If the calculation of v_i is based on the angle in (c), the noise levels of (a) and (b) seem to increase respectively by $20\log(\sin 3.63^\circ / \sin 3.06^\circ) = 1.5$ dB, $20\log(\sin 3.31^\circ / \sin 3.06^\circ) = 0.7$ dB. Strictly speaking, as the contact point of the belt and the pulley must be decided as in Koyama⁽⁹⁾ and the interference condition at the point of meshing varies with the radius of the rounded tip corner, it is difficult to believe that all the differences between the noise levels shown in Fig.13 depend on the v_i differences as caused by angle ϕ . Nevertheless, the noise level seems to decrease when the radius of the rounded tip corner is reduced.

5. Conclusions

The following conclusions have been obtained from the above research on noise reduction in toothed belt drives:

- (1) The noise level of toothed belt drives is in proportion to about 1.7th power of the pulley speed.
- (2) To reduce noise in pulleys, possible methods include: (i) causing both tooth flanks of the belt and the pulley to interfere with each other at a sufficient level at the meshing point on driving pulley before the top land of the pulley tooth collides with the bottom land of the belt. This is possible, if a suitable pitch difference is selected. (ii) decreasing the size of the radius of the rounded tip corner of pulley teeth.
- (3) The solid borne sound caused by the meshing impact of belt and pulley teeth is almost never transmitted pulley and other parts.

The authors wish to thank Mr. I. Yuruzume of the Mechanical Engineering Laboratory, for pointing out omissions in the present paper.

References

- (1) Kubo, A., *et al.*, Bull. of JSME, Vol.14, No.75, 1971, p.991.
- (2) *ibid.*, p.998.
- (3) Koyama, T., *et al.*, Bull. of JSME, Vol.23, No.181, 1980, p.1240.
- (4) *ibid.*, Vol.22, No.169, 1979, p.988.
- (5) Okoshi, M., *et al.*, Journal of JSPE (in Japanese), Vol.26, No.2, 1960, p.79.
- (6) Koyama, T., *et al.*, Preprint of JSME (in Japanese), No.774-7, 1977, p.62.
- (7) Aida, T., *et al.*, Trans. of JSME (in Japanese), Vol.34, No.268, 1968, p.2235.
- (8) Kawata, Y., *et al.*, Zairyokodokogaku handbook (in Japanese), 1966, p.805, Asakura Shoten.
- (9) Koyama, T., *et al.*, Bull. of JSME, Vol.23, No.181, 1980, p.1235.
- (10) *ibid.*, Vol.22, No.169, 1979, p.982.