Vibration reduction effect of one-way clutch on belt-drive systems

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ABSTRACT

In order to study the vibration reduction effect of the one-way clutch on belt-drive systems, especially on the transporting belt, an experimental platform is built. The strongest resonant areas of the midpoint of the transporting belt with and without the one-way clutch are compared. The experimental results clearly show the vibration reduction effect of the one-way clutch on the resonance of the belt-drive system. In order to determine the optimum vibration reduction parameters of the one-way clutch, a two-pulley belt-drive dynamic model coupled is established. The commonalities between numerical simulation and experimental research are the inclusion of a translating belt, a driving pulley, a driven pulley and an accessory pulley. The difference is that there is no connection between the physical parameters of the numerical simulation and experimental studies. By establishing the relationship between the longitudinal vibration and transverse vibration of the translating belt spans, the coupled governing equations of the belt-drive system are derived to describe the coupling of the transverse vibration of the transporting belt and rotation of the pulleys. The natural frequency of the belt is determined by using the amplitude spectrum of the free vibration response of the system. Numerical results illustrate that the two main parameters of the one-way clutch, the wrap spring and the pre-load have little effect on the natural vibration frequency of the belt. However, these two main parameters are very sensitive to the vibration reduction effect of the one-way clutch. Therefore, this study is very helpful to understand the optimal design of the one-way clutch.

1. Introduction

Belt-drive dynamic systems are commonly used in mechanical engineering. Large unwanted transverse vibrations always exist in these systems (Mote and Wu, 1985; Sze et al., 2005; Chen and Yang, 2006; Marynowski, 2010; Yang et al., 2010; Čepon et al., 2011; Huang et al., 2011; Ding et al., 2012; Yao et al., 2012; Yang et al., 2017; Tang et al., 2018). Therefore, much attention has been paid to understanding the vibration characteristics of the belt-drive systems (Zhang and Zu, 1999; Beikmann et al., 1996), especially for the vibration characteristics of the transporting belt (Li et al., 2003; Ding and Chen, 2008; Lim et al., 2010; Ghayesh et al., 2010; Yang and Zhang, 2014; Zhang et al., 2015; Ding et al., 2017; Wang and Zu, 2017; Wang, 2018). Marynowski and Kapitaniak reviewed the study of the dynamics of axially transporting continua before 2014 (Marynowski and Kapitaniak, 2014). One-way clutches have been widely used to reduce the vibration of belt-drive systems (Wang and Mote, 1986; Leamy and Perkins, 1998; Čepon and Boltezar, 2009; Scurtu et al., 2012; Lewicki et al., 2013). To help the design of the one-way clutches, it is imperative to understand the influences of the system parameters on the vibration reduction effect of the clutches.

By controlling torque transmitted in only one direction, one-way clutches effectively eliminate the resonance peaks of belt-drive systems. Based on a two-degree of freedom model, Zhu and Parker established a non-linear dynamics model for a two-pulley belt-drive system coupled with one-way clutch (Zhu and Parker, 2005, 2006). By utilizing a one-degree-of-freedom for engaged state and two, uncoupled degrees of freedom for the disengaged state, Mockensturm and Balaji studied damping effect of one-way clutches on the rotation of the pulleys (Mockensturm and Balaji, 2005). Also by using a discrete system, Cheon investigated the vibration characteristics of spur gear pairs with a one-way clutch (Cheon, 2007). Since the moving belt is modeled as two massless springs, the damping effect of the one-way clutch on the transverse vibration of the moving belt cannot be reached in the above-mentioned works. By using a piece-wise linear system, Zhu and Parker analyzed the vibration reduction effect of a one-way clutch on a three-pulley serpentine drive system (Zhu and Parker, 2008). However, wrap spring of the clutch has been ignored. Therefore, influences of the
The principle diagram of the experiment shown in Fig. 1. CKZ-A35100 one-way clutch was used, as shown in Fig. 2(a). Fig. 3 describes with its replacement, respectively. In the experimental study, the CKZ-clutch parameters on the vibration reduction effect of the one-way clutch on the dynamic behavior of the belts in belt-drive systems have not been clear.

Ding and Zu built a two-pulley belt-drive system, and an accessory pulley is coupled to the driven pulley by a one-way clutch. The translating belt is modeled as axially moving string spans(Ding and Zu, 2013) and beam spans(Ding and Zu, 2014). The stable steady-state responses of the belt-drive system are obtained by the Galerkin method (Ding and Zu, 2013) and the differential quadrature and integral quadrature methods(Ding and Zu, 2014), respectively. They found that the one-way clutch clearly eliminated resonance peaks of the transverse vibration of the transporting belt and the rotation of pulleys. Based on non-trivial boundary conditions, characteristics of steady-state responses of a belt-drive system are studied for the driven pulley excitation(Ding and Li, 2014) and dual excitations(Ding, 2015). Although resonance responses of the two-pulley belt-drive system with a one-way clutch have been examined, the influences of the parameters of the clutch on the vibration reduction effect of the one-way clutch still need to be clear. Moreover, the above-mentioned modeling of pulley-belt coupling relationships is all based on quasi-static assumptions. By averaging the axial tension in the translating belt, the coupling vibration of the belt and the pulleys is achieved.

In the above-mentioned literature, the dynamics of a belt-drive system coupled with one-way clutches have only been studied theoretically, but has not been verified by experimental research. Therefore, in order to achieve the purpose of studying system dynamics through an experimental approach, this work builds an experimental platform for a belt-drive system with a one-way clutch. Furthermore, without the quasi-static stretch assumption, the relationship between the longitudinal and transverse vibration of the translating belt is established to realize the coupling vibration between the belt and the pulleys. For modeling the dynamical system, the viscous damping coefficient of the transverse vibrations of the moving belt is considered. Natural frequencies and periodic resonance responses are numerically obtained by using the differential and integral quadrature methods. Influences of clutch parameters on the vibration reduction effect are numerically studied.

2. Experimental setup

Figs. 1 and 2 show the experimental platform of two-pulley belt-drive system coupled with a one-way clutch and the one-way clutch with its replacement, respectively. In the experimental study, the CKZ-A35100 one-way clutch was used, as shown in Fig. 2(a). Fig. 3 describes the principle diagram of the experiment shown in Fig. 1. CKZ-A35100 one-way clutch used in packaging, transportation, metallurgy, mining and other mechanical systems, by preventing reverse or beyond the operation, to avoid the mechanical system of large resonance. In order to compare the damping effect of the one-way clutch on the system, a replacement without one-way function is adopted, as shown in Fig. 2(b). One thing should be noted that, due to the installation requirements, the replacement must be consistent with the one-way clutch size in the assembly. This results in an unavoidable quality difference between the two experimental elements. The standard mass of the CKZ-A35100 model one-way clutch is 3.6 Kg. However, the mass of the replacement is 4 Kg. There is a mass difference of 0.4 Kg. This may cause the resonance area to be slightly different. Without one-way function, the resonance of the system will occur in advance. This is because the slightly larger mass causes the natural frequency of the belt-drive system to become slightly smaller. In order to get closer to engineering practice, this experiment uses three-phase asynchronous motor as the only driving source. Through the frequency converter to control the output frequency of the motor, and control the drive pulley evenly accelerated or evenly slow down the rotation. In the belt-drive system, the most widely used belt is a multi-wedge belt. In this experiment, a multi-wedge belt is used. The model is KK-USA-Poly-V-Belt-PL1210. The main material is black rubber. The wedge is 4.7 mm, the belt is 10 mm high, the wedge is 400, and the wedge is 6. As shown in Figs. 1 and 3, the axle box is arranged on one side of the driven pulley. Outside the axle box, a wheel load is installed to provide a load for the experimental system. The encoder is connected to the measurement display controller for real-time observation of the rotational speed of the driven pulley. The driven system is designed to be movable so that the initial tension of the belt can be adjusted. A laser displacement sensor is mounted on the test bench to measure the transverse vibrations of the center of the multi-wedge belt.
3. Experimental test

In the experiment, the Bruel & Kjaer (BK) data acquisition system is adopted. The initial tension on the wedge belt can be determined by measuring the vibration frequency of the static belt. An impact excitation is applied to the wedge belt, and the attenuation vibration of the wedge belt is measured, as shown in Fig. 4 (a). Then the time-domain data are measured by the fast Fourier transform, and the frequency-domain diagram shown in Fig. 4 (b) is obtained. As shown in Fig. 4(b), the fundamental frequency of the static wedge belt is 60 Hz. It is not difficult to understand that the larger the fundamental frequency, the greater the initial tension of the corresponding wedge belt. However, the state with a fundamental frequency equal to 60 Hz is used as the condition for this experiment. By entering the control curve in the frequency converter to control the motor to accelerate evenly or evenly slow down the rotation. In this experiment, the control motor speed range is 600–2200 rpm. Transporting speed of the corresponding wedge belt is 3.927–14.399 m/s. The resonant response curve of the vibration displacement at the midpoint of the wedge belt can be obtained by setting the sweep up and downwards. With the change of rotational speed, Fig. 5(a) shows the resonant amplitude response curve of the belt in the belt-drive system coupled with a one-way clutch. Fig. 5(a) shows only the resonant regions corresponding to the maximum vibration displacement. The experimental results in Fig. 5(a) clearly demonstrate that the hardening nonlinearity is involved in the belt-drive system. In particular, in the gradual acceleration sweep curve, the hardening nonlinear characteristic is more pronounced.

4. Dynamics modeling

Fig. 6 shows the mechanic model of a two-pulley belt-drive system coupled with a one-way clutch. For describing the driven pulley and the driving pulley, \( \theta_i(t) \) \( (i = 1, 2) \) are used to represent the angular displacements. The driven pulley and the driving pulley are assumed to be same, with the rotational inertia \( J \) and the radius \( r \). Furthermore, \( \theta_a(t) \) and \( J_a \) are angular displacement and rotational inertia of the accessory pulley, respectively. \( M \) is the pre-load of the accessory pulley and the driven pulley. Properties of the translating belt are assumed to be uniform with density \( \rho \), modulus \( EA \), where \( A \) is the belt’s cross-sectional area and \( E \) is the belt’s young modulus. Moreover, the belt’s translating speed \( c \) and the initial static axial tension of the belt \( P_0 \) are considered to be constant and uniform. \( l \) is the length of above and below belt spans. \( P_i(x,t) \) \( (i = 1, 2) \) is the total axial tension in the belt span \( i \).
The wrap spring is disconnected when the angular displacement of
\[ (\theta_1 - \theta_2) \] is much smaller than the trans-
\[ \omega \] vibration frequency of the driving pulley. By applying the di-
volution of the driving pulley is assumed to be a harmonic excitation. Therefore, governing equations of rotational vibration of the pulleys are derived as
\[ \ddot{\theta}_1 + c_3 \dot{\theta}_1 = \tau \left( \frac{E_A}{l} \int_0^t w_{1,2} \, dx + \frac{M}{l} (\theta_2 - \theta_1) \right) \]
\[ - \frac{E_A}{l} \int_0^t w_{1,2} \, dx + \frac{M}{l} (\theta_1 - \theta_2) \]
\[ + M - f (\delta \delta) K_d (\delta_1 - \delta_0), \]
\[ J_0 \ddot{\theta}_2 + c_2 \dot{\theta}_2 = [P_1 (t) - P_2 (t)] r, \]
\[ J_0 \ddot{\theta}_1 + c_3 \dot{\theta}_1 = [P_1 (t) - P_2 (t)] r + M - f (\delta \delta) K_d (\delta_1 - \delta_0), \]
\[ f (\delta \delta) = \begin{cases} -1 & \delta \delta = \delta_1 - \delta_0 > 0 \\ 0 & \delta \delta = \delta_1 - \delta_0 \leq 0 \end{cases} \]
where the dot above the symbol represents the derivative of time. Since the longitudinal displacement \( u(x,t) \) is much smaller than the trans-
verse displacement \( w(x,t) \) and \( \Theta \approx \rho A \), the second equation of Eq. (2) can be replaced by
\[ u(x, 0) = \int_0^t w_{1,2} \, dx + x_0 \Theta (t) + C_1 (t) \]
\[ C_2 (t) = \dot{C}_0 (t) + \frac{1}{2} \frac{E_A}{l} \int_0^t w_{1,2} \, dx, \]
\[ C_3 (t) = \frac{1}{2} \frac{E_A}{l} \int_0^t w_{1,2} \, dx, \]
\[ C_4 (t) = \frac{1}{2} \frac{E_A}{l} \int_0^t w_{1,2} \, dx. \]
where \( C_0 (t) \) and \( C_2 (t) \) are functions of the time \( t \). Based on the boundary conditions of the longitudinal vibration of the translating belt spans, the first line of Eq. (3), the following equations are derived
\[ \rho A w_{1,x,t} + 2 c w_{1,x,t} + c_2 w_{1,x,t} + \rho A w_{2,x,t} + \rho A w_{2,x,t} - \frac{E_A}{l} w_{2,x,t} = 0, \]
\[ u_1 (x, t) = u_2 (x, t) = 0, \]
\[ w_{1,0,x,t} (0, t) = w_{2,0,x,t} (0, t) = 0, \]
where a comma preceding \( x \) or \( t \) denotes partial differentiation with respect to \( x \) or \( t \), \( t \) is the moment of inertial, \( c_2 \) is the viscous damping coefficient, and is introduced for concentrating energy dissipative in the transverse vibration of the translating belt. An accessory pulley acts as a load connected with a driven pulley by a wrap spring with stiffness \( K_d \). The wrap spring is disconnected when the angular displacement of the driven pulley is less than that of the accessory pulley. Therefore, one-way clutch disengages (Zhu and Parker, 2006). The equations of rotational vibration of the driving pulley, the driven pulley and the accessory are derived as
\[ J_0 \ddot{\theta}_1 + c_3 \dot{\theta}_1 = [P_1 (t) - P_2 (t)] r + M - f (\delta \delta) K_d (\delta_1 - \delta_0), \]
\[ J_0 \ddot{\theta}_2 + c_2 \dot{\theta}_2 = [P_1 (t) - P_2 (t)] r, \]
\[ J_0 \ddot{\theta}_2 + c_2 \dot{\theta}_2 = [P_1 (t) - P_2 (t)] r + M - f (\delta \delta) K_d (\delta_1 - \delta_0), \]
\[ f (\delta \delta) = \begin{cases} 1 & \delta \delta = \delta_1 - \delta_0 > 0 \\ 0 & \delta \delta = \delta_1 - \delta_0 \leq 0 \end{cases} \]
where the dot above the symbol represents the derivative of time. Since the longitudinal displacement \( u(x,t) \) is much smaller than the trans-
verse displacement \( w(x,t) \) and \( \Theta \approx \rho A \), the second equation of Eq. (2) can be replaced by
\[ u(x, 0) = \int_0^t w_{1,2} \, dx + x_0 \Theta (t) + C_1 (t) \]
\[ C_2 (t) = \dot{C}_0 (t) + \frac{1}{2} \frac{E_A}{l} \int_0^t w_{1,2} \, dx, \]
\[ C_3 (t) = \frac{1}{2} \frac{E_A}{l} \int_0^t w_{1,2} \, dx, \]
\[ C_4 (t) = \frac{1}{2} \frac{E_A}{l} \int_0^t w_{1,2} \, dx. \]
V-belt. Moreover, Table 1 shows the geometrical and physical parameters of the V-belt. The nonlinear coupled ordinary differential equation (12) of two-pulley belt-drive systems are numerically solved by using the fourth order Runge-Kutta method (Shangguan et al., 2013). In this study, a V-belt is adopted. Fig. 7 shows the shape of the cross-sectional area of the V-belt. Three different pre-loads of the accessory pulley and the driven pulley on the time history and the natural frequency of free vibration of the belt does not change with pre-load. Therefore, in the following study, for determining the influence of the pre-load of the accessory pulley and the driven pulley and the stiffness the wrap spring of the clutch on the vibration reduction effect of the clutch on the moving belt, the excitation frequency is set as 541 rad/s (=2πω1).

In order to determine the damping effects of the one-way clutch on the resonance of the transporting belt, Fig. 11 shows the steady-state response amplitude of the transporting belt coupled with and without one-way clutch, changing with the wrap spring stiffness of the one-way clutch. As shown in Fig. 11, without one-way function, the wrap spring has little effect on the resonance of the belt. Moreover, for a relatively small stiffness of the wrap spring, the one-way function cannot reduce the resonance of the belt. For a certain area, means around 2500–5000 N m/rad, the vibration reduction effect is very sensitive to the spring stiffness. However, the vibration reduction effect does not change with further growing spring stiffness.

Fig. 12 presents the influences of the pre-load of the accessory pulley and the driven pulley on the time history of the transporting belt. The numerical results in Fig. 12 clearly show that the resonance amplitude of the transporting belt does not vary with the change of the pre-load. Moreover, there is an optimal pre-load for reducing the resonance of the transporting belt. One thing should be mentioned that, only in a specified area of the pre-load, means 2 Nm to 6 Nm, the one-way clutch can reduce the resonance of the belt. Otherwise, the one-way clutch reduces the resonance of the translating belt. Therefore, in the following investigation, for determining the influences of the pre-load of the accessory pulley and the driven pulley on the resonance of the translating belt, the excitation frequency is around 540 rad/s. The numerical results in Fig. 8 shows that resonance of the above belt span takes place while the excitation frequency is around 540 rad/s. The numerical results in Fig. 8 shows that resonance of the above belt span takes place while the excitation frequency is around 540 rad/s. The numerical results in Fig. 8 shows that resonance of the above belt span takes place while the excitation frequency is around 540 rad/s. Therefore, based on the numerical results in Fig. 9, the stiffness of the wrap spring has negligible effect on the time history and the natural frequency of the translating belt.

Fig. 10 describes the influences of the pre-load of the accessory pulley and the driven pulley on the time history and the natural frequency of the translating belt. For a certain area, means around 2500–5000 N m/rad, the vibration reduction effect is very sensitive to the spring stiffness. However, the vibration reduction effect does not change with further growing spring stiffness.

Table 1 Properties of the pulley-belt coupled with a one-way clutch.

<table>
<thead>
<tr>
<th>Item</th>
<th>Notation</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Radius of pulleys</td>
<td>r</td>
<td>0.04520 m</td>
</tr>
<tr>
<td>Radius of accessory</td>
<td>r_a</td>
<td>0.08890 m</td>
</tr>
<tr>
<td>Rotation inertia of the driven pulley</td>
<td>J</td>
<td>0.001607 kg m^2/rad</td>
</tr>
<tr>
<td>Rotation inertia of accessory</td>
<td>J_a</td>
<td>1.62 J</td>
</tr>
<tr>
<td>Torsion damping coefficient of driven pulley</td>
<td>c_0</td>
<td>0.02 N m s/rad</td>
</tr>
<tr>
<td>Torsion damping coefficient of accessory</td>
<td>c_a</td>
<td>0.02 N m s/rad</td>
</tr>
<tr>
<td>Length of span i</td>
<td>l</td>
<td>0.55180 m</td>
</tr>
<tr>
<td>Young’s modulus</td>
<td>E</td>
<td>2 × 10^11 N/m^2</td>
</tr>
<tr>
<td>Linear viscous damping</td>
<td>c_v</td>
<td>2 × 10^10 N s/m</td>
</tr>
<tr>
<td>Cross-section area</td>
<td>A</td>
<td>5.671 × 10^{-9} m^3</td>
</tr>
<tr>
<td>Area moment of inertia</td>
<td>I</td>
<td>2.775 × 10^{-4} m^4</td>
</tr>
<tr>
<td>Density</td>
<td>ρ</td>
<td>1150 kg/m^3</td>
</tr>
<tr>
<td>Static tension</td>
<td>P_0</td>
<td>150 N</td>
</tr>
</tbody>
</table>

Fig. 7. The form of the cross-sectional of the V-belt.

Fig. 8. The maximum steady-state response of the above belt span via sweep frequency.
way function may increase the resonance of the transporting belt. Therefore, the pre-load of the one-way clutch should be carefully designed.

6. Conclusions

In this work, the vibration reduction effects of the one-way clutch on the resonance of the transporting belt in the belt-drive system are studied. The experimental platform is established with two pulleys and a wheel load. Mechanistic model with the driving pulley, the driven pulley and an accessory pulley is also built. The responses of a two-pulley belt-drive system coupled with a one-way clutch are obtained by using the differential and integral quadrature methods. The stable frequency response of the transverse vibration of the transporting belt is

Fig. 9. The time histories and frequencies of the free vibration of the moving belt with different $K_d$. 

(a) The time history with $K_d=2000$ Nm/rad  
(b) The amplitude spectrum for (a)  
(c) The time history with $K_d=4000$ Nm/rad  
(d) The amplitude spectrum for (c)  
(e) The time history with $K_d=6000$ Nm/rad  
(f) The amplitude spectrum for (e) 


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presented to find the resonance area. Through similar mechanical model and different physical parameters, experimental study and theoretical analysis both show that the one-way clutch has a vibration reduction effect.

The time history and the amplitude spectrum of the free vibration of the belt-drive system demonstrate that the natural frequency of the transporting belt almost does not change with the change of the stiffness of the wrap spring of the clutch and the pre-load of the accessory and the driven pulley. Moreover, the resonance amplitude of the transporting belt is presented with the changing wrap spring stiffness and pre-load. Numerical results show that the vibration reduction effects of the one-way clutch are very sensitive to the change of the spring stiffness and the pre-load of the clutch. By presenting the tendency of the influences of these parameters, the best ranges for vibration reduction of these parameters are determined. Therefore, this work will contribute to design the one-way clutch.

Fig. 10. The time histories and frequencies of the free vibration of the transporting belt with different M.
Fig. 11. Effects of the wrap spring stiffness of the clutch on the steady-state response of the belt span.

Fig. 12. The effects of the pre-load of accessory on steady-state responses of the above belt span.

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Appendix A. Supplementary data

Supplementary data related to this article can be found at http://dx.doi.org/10.1016/j.euromechsol.2018.04.004.

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