Experimental identification of the corrective effect of a non-circular pulley: application to timing belt drive dynamics

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Abstract

The work presented in this paper aims at showing experimentally how an oval pulley can generate a corrective effect on a timing belt drive subjected to a 2nd order periodic excitation (fluctuating load torque). For that purpose, a simplified timing belt drive of a 4-cylinder car engine is reproduced on a test stand. The oval pulley is mounted on the crankshaft axis. Experiments are conducted for different phasing angles of the oval pulley at several driving speeds. Pulley angular vibrations and span tension fluctuations are monitored. The results obtained from these experiments are analyzed in the angular and angular frequency domains. The results are compared with experiments performed on a reference case (equivalent circular crankshaft pulley). The study focuses on the effect of the oval pulley phasing angle on the amplitude of the 2nd order harmonics governing the angular response of the transmission.

1 Introduction

In an automotive engine, the Timing Belt Drive (TBD) is a key component in charge of synchronizing the valve train system and the crankshaft (figure 1). This synchronization constitutes one of the essential functions of the engine. It ensures the correct timing of the valves opening and closing with respect to the piston movement.

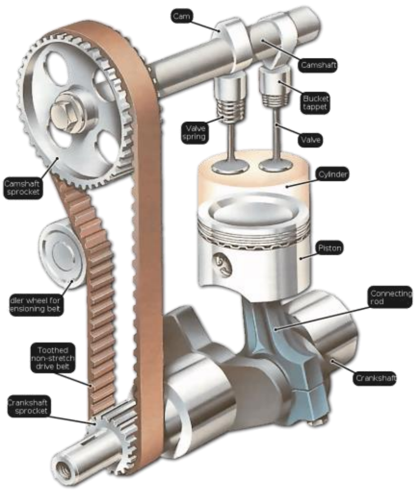


Figure 1: Principle of the valve train system of a car engine with a timing belt drive

In running conditions, TBD are exposed to a very harsh vibratory environment. Various excitation sources such as crankshaft acyclism and fluctuating load torques applied to driven pulleys, generate vibratory phenomena that may affect the TBD dynamic performances and life. In particular, two phenomena require careful monitoring and need to be minimized:

- Rotational vibrations of the transmission axes.
- Tension force fluctuations in the belt spans.

If mishandled, these two phenomena may induce increased component fatigue, power losses, noise and in the most extreme case can cause desynchronization of the TBD that could result in engine failure (piston-valve clash).

In response to ever-stricter requirements for engine efficiency and reliability, car manufacturers now commonly use TBD comprising innovative pulleys with Non-Circular (NC) pitch profile. When rotating, a NC pulley causes periodic elongations of its adjacent belt spans. Hence, it behaves like an exciter that may generate a corrective rotational excitation able to counteract the other excitation sources acting on the TBD. It is now known that for optimal profile shape and phasing in the transmission, the use of a NC pulley can improve considerably the vibratory performances of a transmission [1,2,3,4]. Nevertheless, determining the optimal design parameters of a non-circular pulley remains hard to accomplish. To achieve this, it is important to clearly understand and identify the impact of such pulleys on the dynamic behaviour of TBD.

The literature on this subject is relatively poor. Most works concern kinematics and quasi-static analyses as reported in [5,6]. In recent papers, Zhu et al. [2] and Passos et al. [3,4] propose numerical models developed to predict the dynamic behaviour of a transmission comprising NC pulleys. Using these models, the authors perform numerical studies that show how an oval crankshaft pulley can significantly reduce the rotational vibrations of a camshaft pulley in a four-cylinder engine (2nd order periodic camshaft load torque). The models are based on a discrete approach (also called 0D/1D approach) similar to that implemented by Hwang et al. [7] for poly-V belt transmissions comprising circular pulleys only. In his model, Hwang considers the belt spans as linear spring-damper elements connected to the pulleys represented by rotational inertias. In [8], Passos et al. conducted an experimental investigation on a basic transmission comprising two pulleys. Two configurations of the basic transmission were studied: one with an oval pulley mounted on the driving axis and the other with an equivalent circular pulley. The experiments were done with no excitation source (constant driving speed and constant load torque applied to the driven pulley). Instantaneous angular speed and acceleration of the driven pulley, transmission error and transmitted torques obtained for the two configurations of the transmission were analyzed and compared. The comparison provided a rigorous description of the proper effect of the oval pulley on the rotational dynamics of the transmission.

The present work aims at completing the studies presented in [3,4,8]. It shows experimentally how an oval pulley can generate a corrective effect able to counteract an excitation source such as a 2nd order periodic (H2) load torque. For that purpose, the oval pulley is mounted on the driving axis of a transmission whose architecture is similar to that of a 4-cylinder car engine timing belt drive. Usually, in this type of transmissions the camshaft pulley is subjected to a fluctuating load torque having a 2nd order periodicity. The test stand is presented in section 2. Experiments are conducted for different phasing angles of the oval pulley and at several driving speeds. Angular vibrations and span tension fluctuations are monitored. The results obtained from these experiments are analyzed and compared with those of a reference case (equivalent circular crankshaft pulley) in section 3.

2 Test stand

2.1 Studied transmission

The studied transmission is represented in figure 3 and its geometrical characteristics are given in table 1. The transmission has been designed in order to reproduce a simplified TBD of a 4-cylinder car engine. It comprises four pulleys: one oval crankshaft pulley (driving pulley), one camshaft pulley and one idler pulley on each side of the transmission. All the pulleys are circular except the crankshaft pulley that has an elliptical pitch profile. Such a pulley has two design parameters: its eccentricity e and its phasing angle ϕ_0 corresponding to the initial orientation angle between the oval pulley major axis and the y-axis (figure 3). In the present study, the eccentricity is imposed (marketed component) and the phasing angle is freely adjustable.

$$\phi_0 = \theta_{CS}(t = 0) \tag{1}$$

Three levels of stationary crankshaft speeds are considered ($\omega_{CS} \in \{600 ; 1500 ; 3000 \text{ rpm}\}$). The camshaft pulley is subjected to a fluctuating load torque C_{CAM} having a 2nd order periodicity with respect to the crankshaft rotation θ_{CS}. The measured load torque is plotted versus the crankshaft angle for the three different driving speeds in figure 2 (a, c, e). The angular frequency content of the torque is shown in figure 2 (b, d, f). Whatever the driving speed, the torque is governed by a dominant 2[nd] order harmonic and secondary even harmonics of 4[th], 6[th] and 8[th] orders. The torque is also modulated by harmonics of 0.5[th] and 1[st] orders due to geometry faults and misalignments affecting slightly the measurement. When the crankshaft runs at 3000 rpm, order 9.5 appears due to dynamic effects occurring in the mechanical device that generates the load torque (see section 2.2).

	Coordinates of the pulley centers		**Pitch profiles of the pulleys**
	X (cm)	**Y (cm)**	
Crankshaft	0	0	Ellipse: – Major axis $a = R_{CS} + e$ – Minor axis $b = R_{CS} - e$ with $\begin{cases} R_{CS} = 3.19\ cm \\ e = 0.07\ cm \end{cases}$
Idler (Tight side)	17.3	-2.8	Circle of radius $R_{IT} = 3\ cm$
Camshaft	39.3	0	Circle of radius $R_{CAM} = 2R_{CS}$
Idler (Slack side)	16.9	4.7	Circle of radius $R_{IT} = R_{IS}$

Table 1: Geometrical characteristics of the transmission

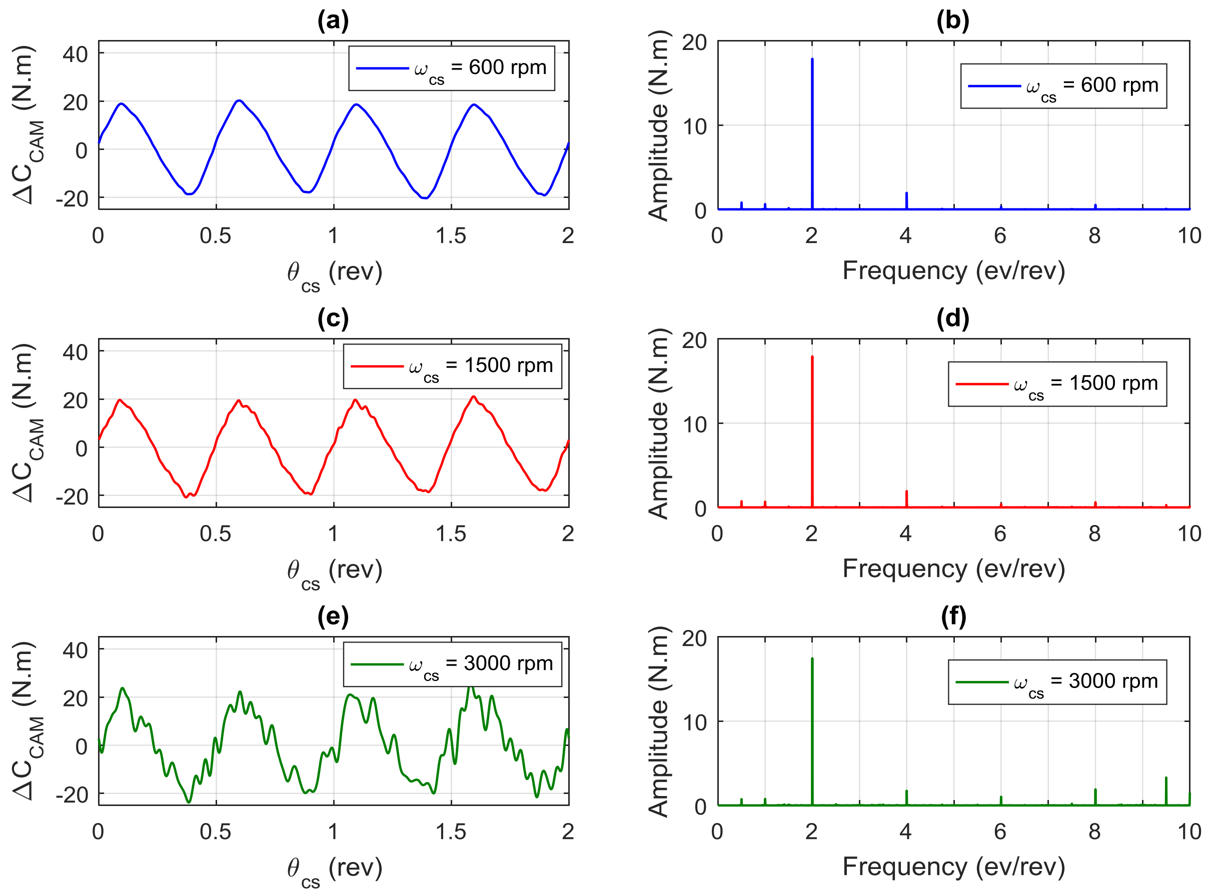


Figure 2: Variations of camshaft load torque in the angular (a, c, e) and angular frequency (b, d, f) domains

2.2 Global architecture

The global architecture of the test stand is illustrated in figures 3 and 4. It comprises two rotary shafts (one driving and one driven shaft) and two supporting parts for idler pulleys. The distances between the pulleys are freely adjustable, enabling a custom setting of the transmission geometry.

The crankshaft pulley is mounted on the driving axis coupled to a speed-controlled electric motor. The driven axis that supports the camshaft pulley of the studied transmission is coupled to the camshaft of a cylinder head fully equipped. It enables applying a real camshaft load torque to the transmission (see figure 2). This load torque has already been discussed in section 2.1.

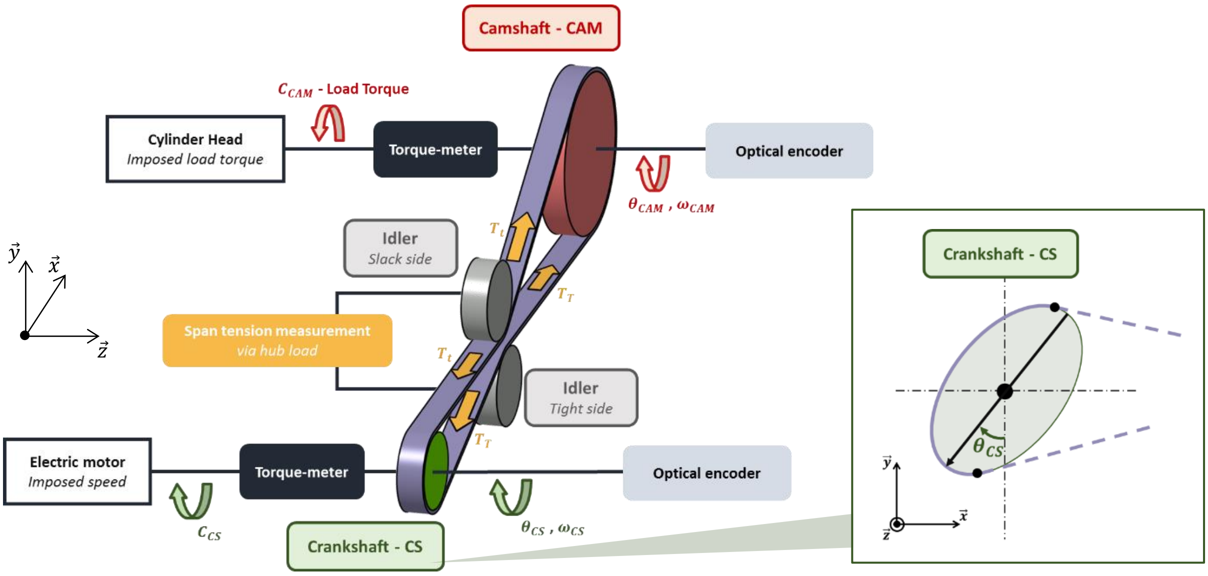
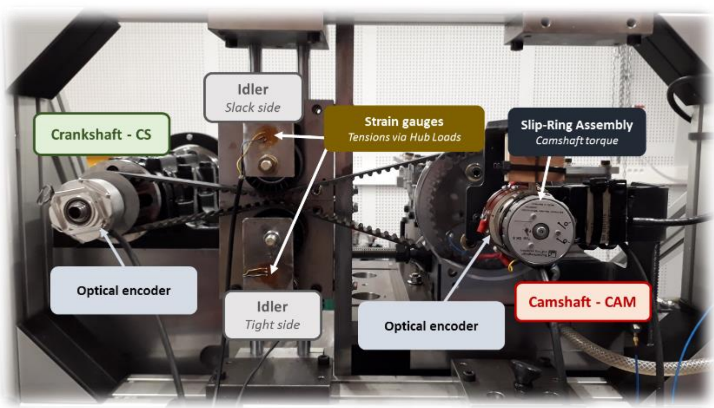
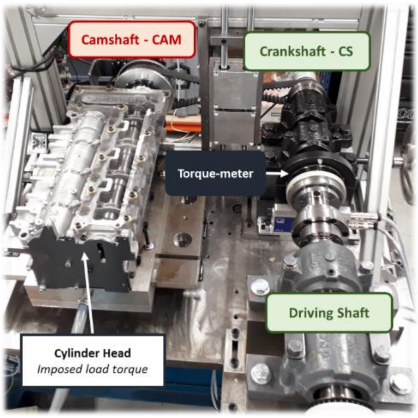


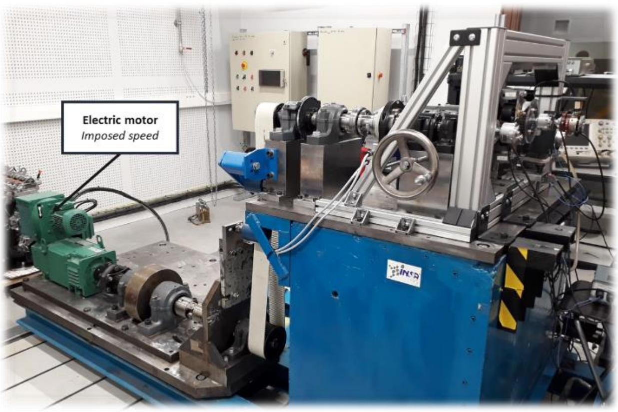
Figure 3: Scheme of the global test stand architecture



(a)



(b)



(c)

Figure 4: Face (a), top (b) and global (c) views of the test stand

2.3 Measurement devices

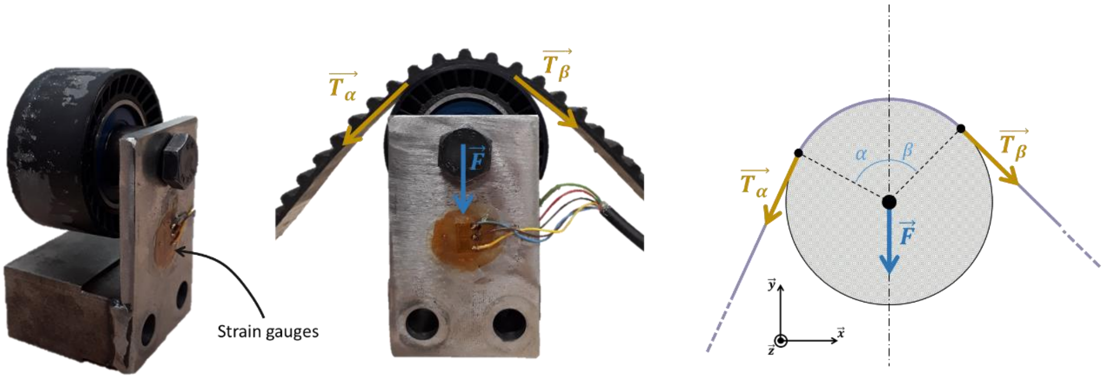
The measurement system is very similar to that used for the works presented in [8]. The system comprises usual devices employed for experimental investigations on gear transmission error and belt transmissions [9,10,11]. It enables measuring all the quantities depicting the rotational dynamics of the transmission.

Optical encoders with a resolution of 2500 pulses per revolution are mounted on the driving and driven axes for measuring their rotation angles and speeds. Torques transmitted by the driving and the driven axes are monitored. On the driving axis, a torque meter with a range of 200 N.m is placed before the driving pulley. On the driven axis, the camshaft is equipped with strain gauges for measuring the resistive load torque. The electrical connection for gauges is provided by means of a slip-ring assembly. The supporting parts used for idler pulleys are equipped with strain gauges so that to measure y-axis component of the hub load applied onto the idler axis. The belt span tension forces can be deduced from the hub load as illustrated in figure 5. On both sides of an idler, belt tensions can be considered equal ($T_\alpha = T_\beta = T$) and then:

$$T = \frac{F}{\sin\alpha + \sin\beta} \tag{2}$$

where F is the y-axis component of the hub load, T is the tension force in both belt spans and α and β are the respective orientation angles of the spans.

The data acquisition system is custom made within a N. I. PXI frame including counter boards for the optical encoders (pulse timing method [9]) and classical data acquisition boards for the other sensors.



(a) (b)

Figure 5: Idler supporting part used to measure the span tensions (a) and principle diagram (b)

3 Experimental analysis

The result analyses are performed by observing the following quantities in the angular and the angular frequency domains:

- Instantaneous angular speed variations for the camshaft pulley ($\Delta\omega_{CAM}$).
- Tension force variations in the tight and the slack sides (ΔT_T and ΔT_t)

These quantities are first studied for a reference case for which the crankshaft pulley is circular (section 3.1). Then, the transmission equipped with the oval crankshaft pulley is considered (section 3.2). A comparison study is conducted showing how the oval pulley affects the levels of angular vibrations and tension force fluctuations depending on its phasing angle and the driving speed.

3.1 Reference case: circular crankshaft pulley

The fluctuations and the frequency content of the camshaft angular speed are shown in figure 6. For all driving speeds, the variations are periodic and dominated by a 2nd order harmonic due to the dominant 2nd order harmonic governing the load torque. The amplitude of the H2 harmonic fluctuation is about 4.5 rpm when the driving speed is 600 rpm and rises to approximately 15 rpm when the crankshaft speed reaches 1500 and 3000 rpm. One can see also that the frequency spectrum contains secondary harmonics mainly due to the other camshaft torque harmonics (even orders and 9.5th order for ω_s = 3000 rpm) but also to some disturbing excitations such as resonances of the test stand and 0.5th and 1st order modulations due to misalignment and geometry faults on the crankshaft and camshaft pulleys.

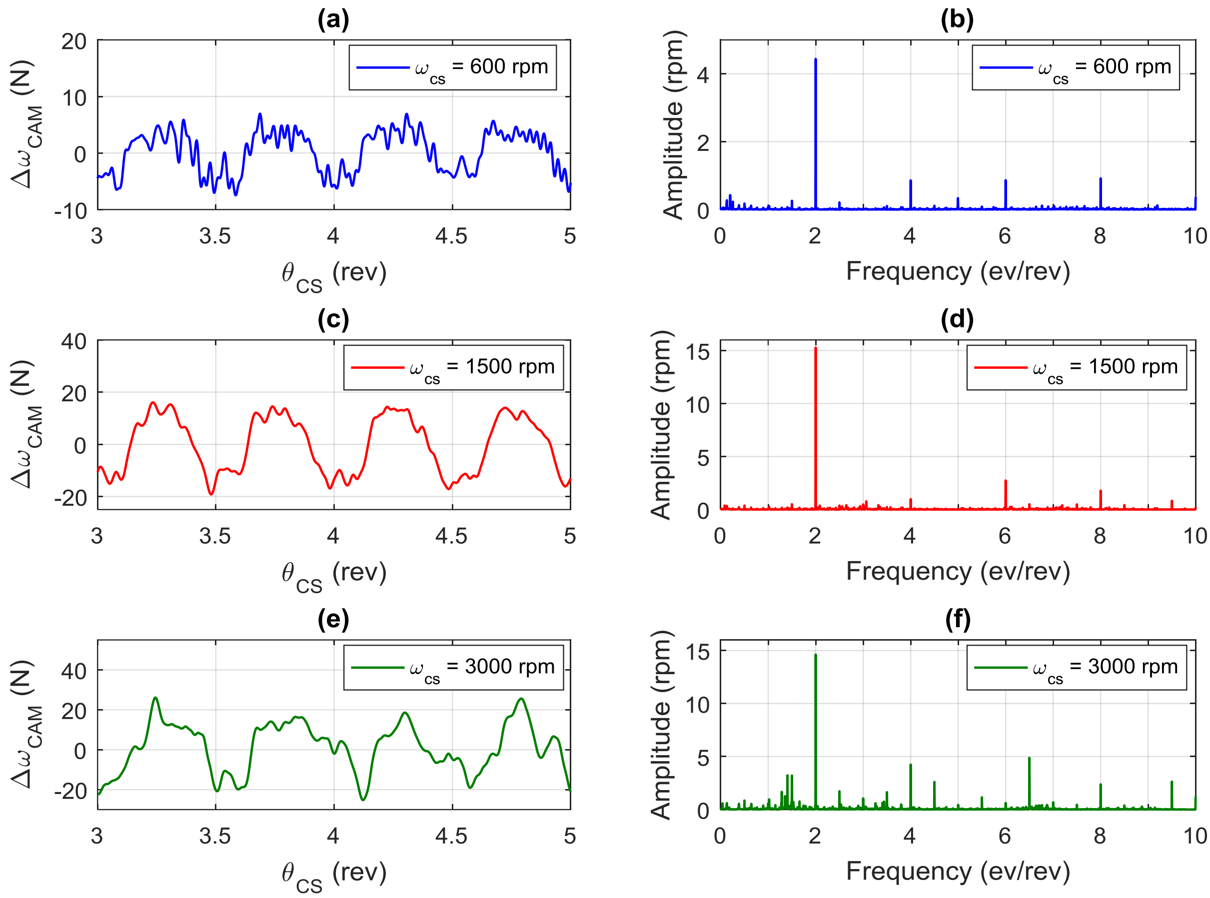


Figure 6: Camshaft angular speed variations in the angular (a, c, e) and angular frequency (b, d, f) domains

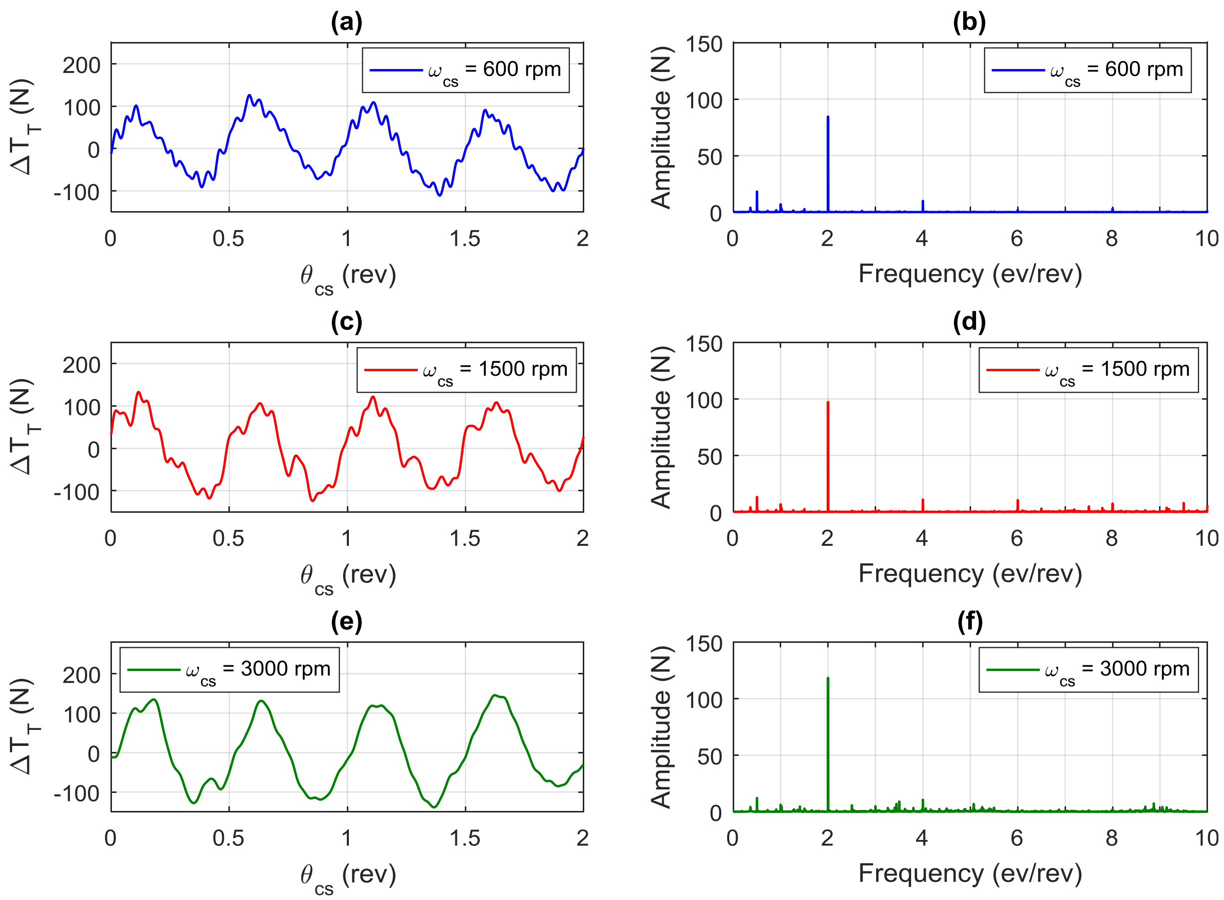


Figure 7: Tight span tension variations in the angular (a, c, e) and angular frequency (b, d, f) domains

Figure 7 and 8 show the variations and the frequency content of the belt tensions in the tight and slack spans of the transmission. The amplitude of the dominant H2 harmonic slightly increases with the driving speed. The H2 amplitude is of course always higher in the tight span than in the slack span. Moreover, one can note that the tension fluctuations in the tight and slack spans are out-of-phase, which is another common result for belt transmission subjected to fluctuating load torque.

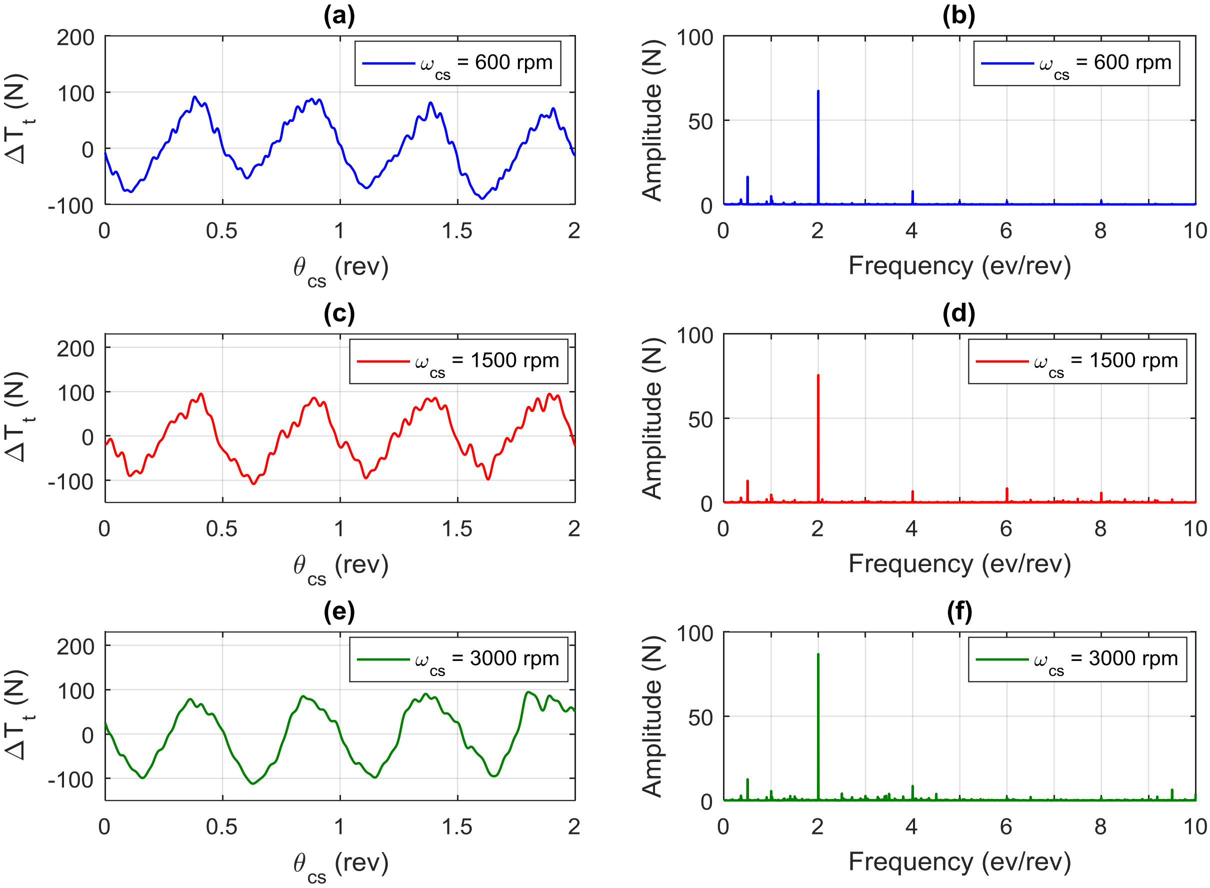


Figure 8: Slack span tension variations in the angular (a, c, e) and angular frequency (b, d, f) domains

3.2 Oval crankshaft pulley: study of the corrective effect

Results obtained for the reference case (section 3.1) have shown that the variations of the camshaft speed and the belt span tensions caused by the camshaft load torque are strongly dominated by a 2nd order harmonic (H2) for all the driving speeds. Thus, in the following, the analyses focus only on the amplitude of H2 harmonics. A parametric study has been performed in order to show how the oval pulley phasing angle impacts the H2 amplitude. Considering that an oval pulley is symmetric along its major axis, the experiments are only performed for a phasing angle ranging between 0 and 180°.

Figures 9, 10 and 11 show respectively the results obtained for the three driving speeds 600 ; 1500 and 3000 rpm. Each figure comprises three graphs on which the amplitudes of the H2 harmonics that govern the camshaft speed and tension forces in the tight and slack spans are respectively plotted versus the phasing angle of the crankshaft oval pulley. On these graphs, the H2 amplitude evolution with the phasing angle is represented with a red solid line marked with circles. Horizontal blue solid lines represent the amplitude of the H2 harmonics in the reference case (circular crankshaft pulley). Green and red hatched areas correspond to phasing angle ranges for which the oval pulley respectively involves a reduction or an increase of the H2 harmonic.

When the crankshaft runs at 600 rpm (figure 9), a speed fluctuation reduction can be obtained for a phasing angle range [0,40°]∪[150,180°] with a maximum reduction of nearly 82% for a phasing angle around 0°. In contrast, for a phasing angle in the range]40,150°[the amplitude is higher than in the reference case with a critical increase of 75% for a phasing angle equal to 90°. H2 amplitude of the tension forces in tight and slack sides are significantly reduced for non-overlapping phasing angle ranges, respectively [97,177°] and [0,82°]. Amplitudes of tension force variations are less impacted by the phasing angle of the oval pulley. For both spans, the maximum reduction and increase ratios are the same, respectively about 24 and 26%.

When the crankshaft runs at 1500 rpm (figure 10), the phasing angle ranges corresponding to an increase and a reduction of fluctuations are quite the same as for a driving speed equal to 600 rpm. The reduction and increase ratios are of the same order of magnitude and are obtained for very similar optimal and critical phasing angle values.

For a driving speed of 3000 rpm (figure 11), the angle ranges and the corresponding reduction and increase ratios differ. The speed variations are reduced when the phasing angle belongs to the range [0,30°]∪[170,180°]. The optimal and critical phasing angles are respectively 10° and 95° with a maximum reduction and increase ratios of 60% and 166% respectively. The tension force variations in the tight span are reduced for a phasing angle in the range [0,22°]∪[137°,180°]. The maximum reduction is about 50% when the phasing angle is around 10° and there is a maximum increase of nearly 67% for an angle of 95°. The tension force fluctuations induced in the slack span are reduced for a phasing angle comprised in the range [0,45°]∪[158,180°] with a minimum amplitude (-30%) and a maximum amplitude (+39%) for phasing angles of 10° and 105° respectively. It can be seen that if the phasing angle belongs to the range [0,22°]∪[170°,180°], the fluctuations are reduced for the camshaft speed and the tension forces in both spans simultaneously. In particular, for a phasing angle equal to 10° all the fluctuations are significantly reduced.

In the light of these results, one can finally note that it is possible to reach a compromise with a phasing angle around 0°. Whatever the driving speed, such a value ensures a strong reduction of the camshaft speed variations and limits the tensions variations in both spans.

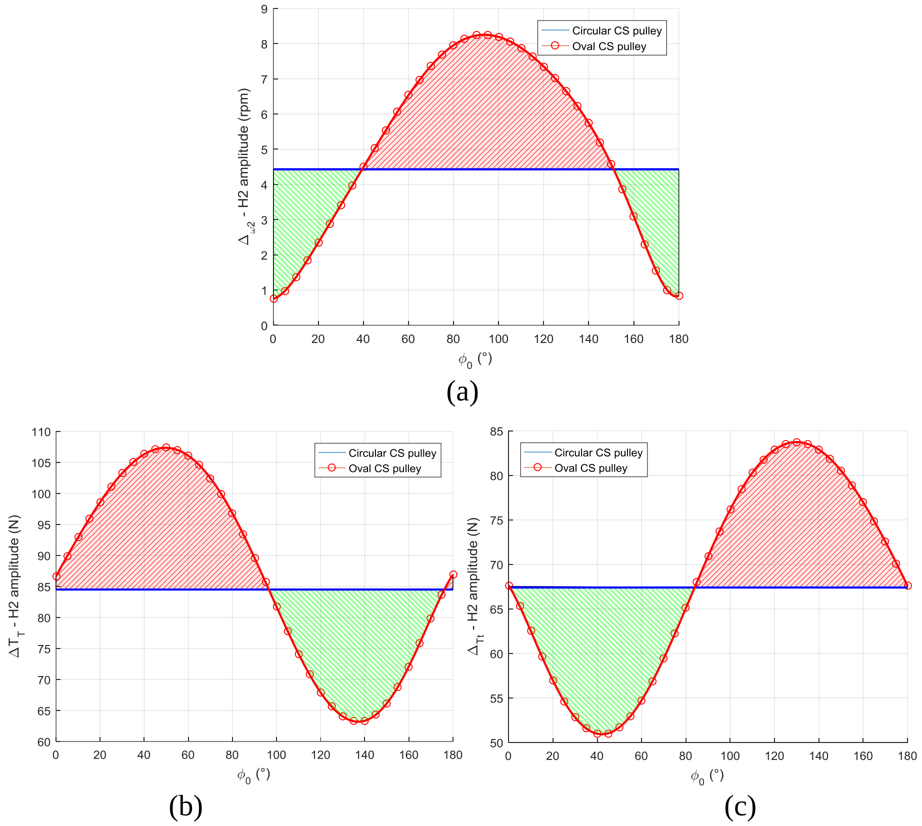


Figure 9: H2 harmonic amplitude of the camshaft speed (a) and tension forces in the tight (b) and slack (c) spans when the crankshaft runs at 600 rpm

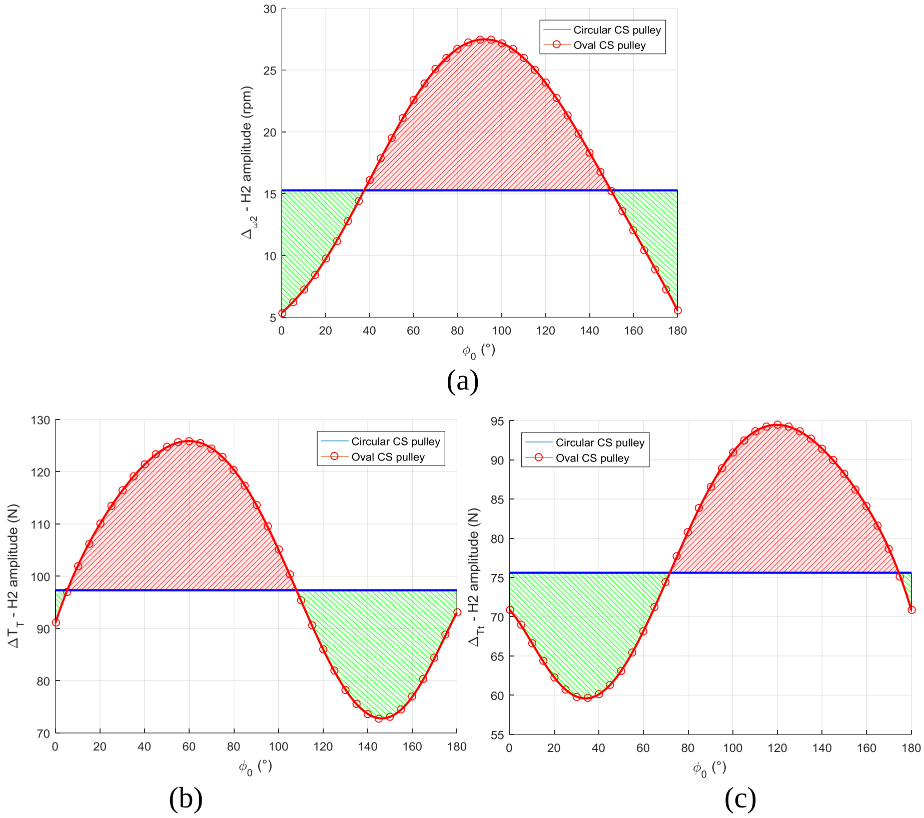


Figure 10: H2 harmonic amplitude of the camshaft speed (a) and tension forces in the tight (b) and slack (c) spans when the crankshaft runs at <u>1500 rpm</u>

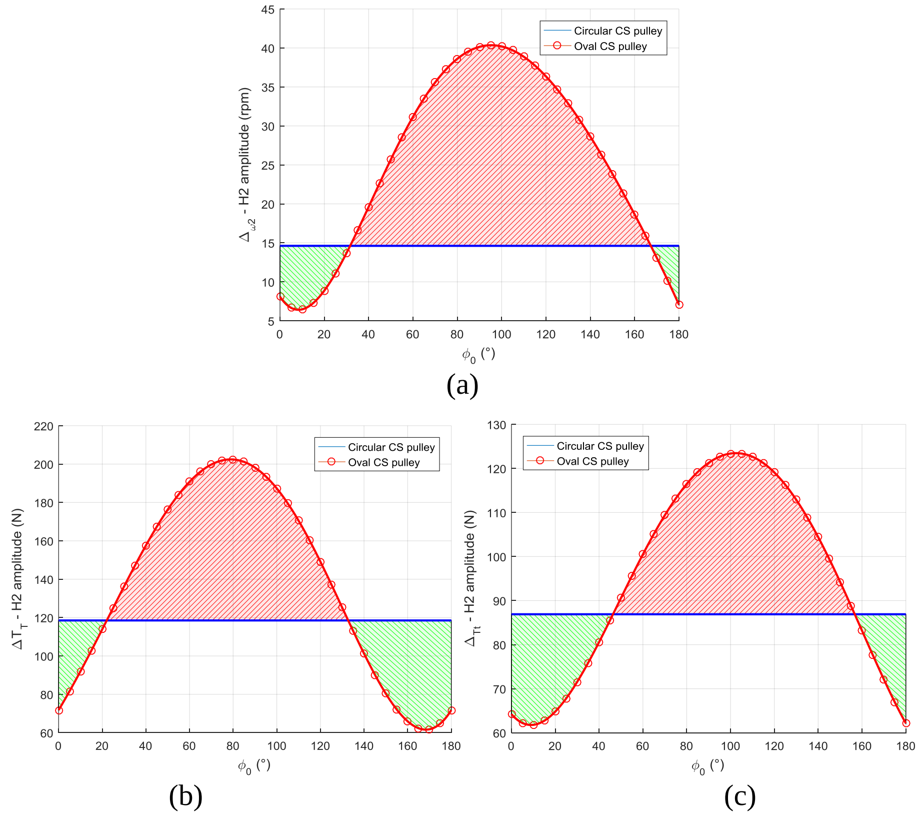


Figure 11: H2 harmonic amplitude of the camshaft speed (a) and tension forces in the tight (b) and slack (c) spans when the crankshaft runs at <u>3000 rpm</u>

4 Conclusion

The experimental results presented in this paper demonstrate to what extent, for a well-chosen phasing angle, an oval pulley has a corrective effect enabling a reduction of the H2 fluctuations affecting the camshaft angular speed and the span tension forces. These results also show how much, for wrong phasing angles, the pulley can degrade the dynamic behaviour of the transmission inducing a strong increase of speed and tension fluctuations. This is why transmissions comprising NC pulleys must be designed carefully.

In addition, one can note that the impact of an oval pulley depends on the driving speed, which make the design of the transmissions more difficult. As already discussed in previous works [3,4], it is not so easy to reduce significantly and simultaneously speed and tension force fluctuations in belt spans using a NC pulley only.

For completing the present work, it could be meaningful to extend the researches to more complex systems. Thus, future works could involve transmissions subjected to other excitation sources such as acyclism and/or transmissions comprising a dynamic tensioner. It could be interesting as well to study the case of a NC pulley having a different profile shape adapted to the correction of harmonics of other order. Also, the effect of NC pulleys on TBD acoustic radiation still remains to investigate (impact on belt meshing noise, span transverse vibrations, …).

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