

Experimental and Calculation Analysis of Rotational Vibration for an Engine Front End Accessory Drive System

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ABSTRACT

Experimental methods for measuring static and dynamic characteristics of an engine Front End Accessory Drive System (FEADS) are presented. The static performance of a FEADS is the static tension of the belt, and the dynamic properties of a FEADS are transverse vibration of belt, and rotational vibration performances that include rotational response of pulleys and tensioner arm, dynamic tension of belt span, slip factor between belt and pulley. A mathematical model and calculation method for rotational vibration analysis of a 8 pulley-belt FEADS is established. In the model, creeping effect of a belt on pulley wrap arc, viscous damping and dry friction of a tensioner are considered. In calculation of dynamic performances of the FEADS, the excitation torques with multi-frequency components from crankshaft torsional vibration are obtained from the measurement. The static tension and steady-state tension in belt spans, natural frequency of a FEADS, rotational response of driven pulleys and tensioner arm, dynamic tension of belt span, slip factor between belt and pulley are estimated with the proposed method. The calculated and tested rotational response of driven pulley and tensioner arm, and tension of each belt span are compared well with experiment, which validate the presented model and calculation method. The modeling method and experimental procedure are instructive for designing a FEADS.

INTRODUCTION

LITERATURE REVIEW

A Serpentine Belt Accessory Drive System (SBADS) with a multi-rib belt have a thin cross-section compared with a V-belt, thereby reducing bend stiffness of belt and increasing

contact area between belt and pulleys. An important component used in a SBADS is the automatic spring-loaded tensioner, which generally consists of a torsional spring, a tensioner arm pivoted about a fixed-end point, a tensioner pulley pinned to the arm, and a damping mechanism based on dry friction or viscous effects. The tensioner maintains constant tractive tension of belt despite accessory torques and belt speed changes. The advantages for a SBADS include compactness, ease of belt replacement, length of belt and accessory bearing life, and automatic tension loss compensation. Therefore, SBADS is widely used in automotive engine Front End Accessory Drive System (FEADS) to transmit crankshaft power to accessories.

A typical automotive engine FEADS generally consists of a crankshaft pulley, a serpentine belt, a tensioner, and several accessories such as alternator, air conditioner, power steering pump, and water pump etc.[1]. Under operating conditions, the crankshaft speed fluctuations are the major source of vibrations in the system. This leads to undesirable noise and excessive slip, which can hamper the belt and bearing life. In order to meet customer expectation in NVH (Noise, Vibration and Harshness) and reliability, various researchers have been investigating the dynamic behaviors and characteristics for a SBADS.

Dynamic behaviors for a SBADS can be categorized into two kind of vibrations: (1) rotational vibration of pulleys and tensioner arm (with the belt spans acting as axial springs), and (2) transverse vibration (normal to belt travel) of the belt spans. According to these behaviors, two categories of mathematical models have been presented in previous studies. The first category is rotational vibration model [1, 2, 3, 4, 5, 6] for estimating the rotational vibration performances concluding rotational response of pulleys and tensioner arm.

dynamic tension of belt span, slip factor between belt and pulley. The second category is coupled vibration model, which consists of rotational vibration of pulleys and tensioner arm, and transverse vibration of belt spans. The coupled vibration model is used for calculating the transverse vibration of the belt spans and rotational vibration of pulleys. If the belt bending stiffness is ignored or considered, the coupled model is defined as string-coupled model [7, 8, 9, 10, 11, 12, 13, 14] and beam-coupled model [15, 16, 17, 18, 19, 20], respectively.

A prototypical rotational vibration model for a FEADS is established by Hwang et al. [1] in 1994. They investigate the rotational vibration responses and mode characteristics of the system, and apply the results to predict the slip factor between belt and pulley. In the model, the overall equations of motion are linearized, and an “equivalent viscous damping” is adopted to estimate the tensioner damping. Later, Kraver et al. [2] extend a complex modal procedure to analyze the frequency response characteristics of a SBADS assuming viscous belt and tensioner damping. Their results are compared against that of Hwang [1] and found to be in good agreement. But the belt creeping in the system is not considered. Researches in [1][2] concentrate on linear analysis for a SBADS. However, dry friction of a tensioner represents a strong nonlinearity that may produce rich dynamic response. In order to reveal the nonlinear mechanism of a Coulomb-damped tensioner, Leamy et al. [3] follow the model in [1] and establish a nonlinear rotational vibration model with the tensioner Coulomb friction considered. The Incremental Harmonic Balance (IHB) method is utilized to calculate periodic solutions for the nonlinear equations of motion, and the stick-slip motion of a tensioner arm is captured. But only a single-frequency excitation is considered in the system. Then, Cheng et al. [4] discuss the impact of the dry friction torque of a tensioner on dynamic characteristics for a SBADS. The system is excited with multi-frequency components. Two kinds of periodic responses for both the nonstick and stick-slip motions are obtained. But the creeping effect of the belt spans on pulley wrap arc is not considered, and the analytical results are not verified by experiments.

To investigate the dynamic properties of the string-coupled model for a SBADS, Ulsoy et al. [7] in 1985 firstly consider the coupling between the belt transverse motion and rotational motion of tensioner arm. In 1996, Beikmann et al. [8] establish a prototypical string-coupled model involving a driving pulley, a driven pulley and a tensioner pulley. They examine the linear coupling of the belt and pulleys for free vibration from infinitesimal belt stretching. Calculation results indicate that only the two belt spans adjacent to the tensioner couple with the pulley rotations. Their subsequent study [9] investigates nonlinear coupling of the transverse and rotational motions resulting from finite belt stretching. Following this three-pulley typical system, Zhang et al. [10]

implement modal analysis by dividing the entire system into two subsystems, and a closed form characteristic equation is obtained. In their subsequent study [11], they use the direct multiple scales method to study nonlinear coupling, which is demonstrated to generate better results than that of the discretization multiple scales method. Moreover, Parker [12] expresses the equations of motion for a SBADS in an extended operator form. He utilizes perturbation method to calculate the eigensolutions and dynamic response of the system. Eigensensitivities to the system parameters including belt tension and speed are obtained. For a SBADS with n pulleys, generic and explicit formulae about sensitivities of both frequency and steady harmonic response are established by Hou et al. [13]. Optimizations are conducted on tensioner parameters so as to suppress dynamic response of the system. Other relevant researches on the dynamic performances of the string-coupled model are summarized by Chen [14] in 2005.

For the beam-coupled model, because the nonlinear equations of motion for belt spans are established with fourth-order differential equations, it is difficult to obtain the closed-form solution of the dynamic response for a SBADS. A prototypical beam-coupled model is established by Kong et al. [15] in 2003. The steady-state (where the belt velocity is a constant) solutions are obtained by a numerical iteration method with the boundary conditions of the belt spans. In 2004, they develop a singular perturbation solution based on Boundary Value Problem (BVP) solvers to obtain the numerically exact steady-state solutions [16]. The steady-state responses and pulley-belt coupling are shown to be sensitive to the system geometry, belt bending stiffness and belt spans tension. In their subsequent work [17], the beam-coupled model is linearized about the steady-state, and then is reformulated to make Galerkin discretization readily applicable. Through examining the eigensolutions from the discretized equations, the dynamic characteristics due to incorporating belt bending stiffness are discussed. Additionally, to explore the dynamic features of vibrations and instability for a FEADS, Dukkipati et al. [20] develop a nonlinear model taking into account of the contact stiffness nonlinearity, tension parametric excitation, and geometry nonlinearity. In their investigation, the numerical simulations are conducted to investigate the system and to correlate with the experimental results.

MOTIVATION AND OBJECTIVES

Rotational vibration is the basic vibration type for a FEADS. Its dynamic characteristics impact belt tension, belt and bearing life, and efficiency of power transmission [21]. In the aforementioned researches, it is clear that the viscous damping or dry friction torque of a tensioner is considered in modeling rotational vibration in most studies. But the influence of the belt creeping on the dynamic performances of a FEADS is not fully understood, and the excitation from crankshaft torsional vibration is usually regarded as a single

frequency component. Additionally, in prior studies, most of the simulation results are not validated by experiments. Therefore, those deficiencies significantly motivate the present work.

The chief objectives of this study are to present the experimental methods for the static tension and dynamic characteristics for a 8 pulley-serpentine belt FEADS, and to the understanding of the FEADS characteristics by experimental data. Other contributions of this paper are to establish a rotational vibration model for the FEADS, and to validate the model by comparing simulation and experiment. In the model, creeping effect of a belt on pulley wrap arc, viscous damping and dry friction of a tensioner, and multi-harmonic excitations components are considered. Newton-Raphson iterative method is used to calculate the static tension and steady-state tension of each belt span. The natural frequency and the rotational periodic response of the FEADS are obtained with complex modal solution procedure. To the end of this paper, the effect of the belt creeping on the dynamic characteristics of the FEADS is presented.

DYNAMIC PERFORMANCE MEASUREMENT OF A FEADS

TEST RIG AND PROCEDURE

Test rig

Fig. 1 shows one generic test rig for measuring the dynamic performances of a FEADS in a four-cylinder engine. The engine is motored with a FEV dynamometer test cell, and a Rotec data acquisition and analysis system is used for measurement and data analysis. The schematic of the FEADS is given in Fig. 2. It is observed that the crankshaft pulley rotates clockwise and a single multi-rib belt called serpentine drives all pulleys. An automatic tensioner is used for improving the system dynamic performance by automatically compensating for tension changes of the belt spans that contact tensioner pulley.



Figure 1. Generic test rig for measuring dynamic performances of a FEADS.

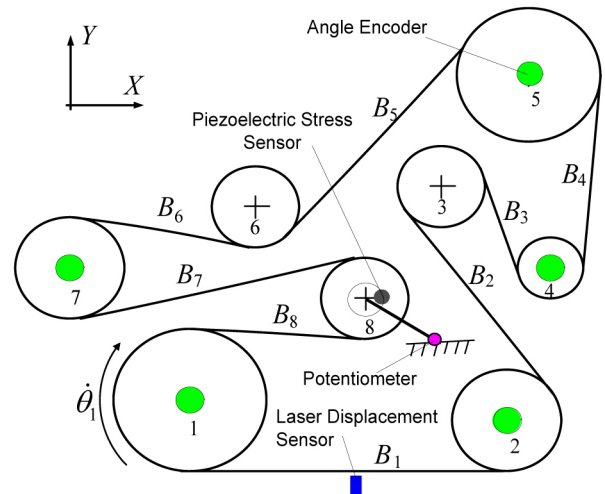


Figure 2. Schematic of the FEADS in Figure 1.

1 CS pulley, 2 AC pulley, 3 IDL1, 4 ALT pulley, 5 PS pulley, 6 IDL2, 7 WP pulley, 8 TEN pulley.

In Fig. 2, the pulleys and belt spans are numbered counter-clockwise, with the crankshaft pulley as the first pulley and the span for pulleys 1 to 2 as the first span. The i th belt span is notated as B_i . All the pulleys in the FEADS are: Crankshaft pulley (CS, No.1 in Fig. 2), Air Conditioner pulley (AC, No.2 in Fig. 2), Alternator pulley (ALT, No.4 in Fig. 2), Power Steering pump pulley (PS, No.5 in Fig. 2), Water Pump pulley (WP, No.7 in Fig. 2), tensioner (TEN) pulley (No.8 in Fig. 2), and two idler pulleys (IDL, No.3 and No.6 in Fig. 2). It notes that two idler pulleys and one TEN pulley are driven from backside of the belt while all the other accessory pulleys are driven by the inside of the belt.

Measurement of belt static tension

The static tension of the belt can be directly obtained by a tensiometer, or be indirectly calculated with the hubload of the TEN pulley.

(1). Direct measurement

Ignoring the friction at the pulley shaft, the shaft torque of each driven pulley is zero when the system is in the static status. Therefore, the static tension of each belt span is the same.

The serpentine belt used in the FEADS is a new belt. The FEADS should operate several minutes before the measurement to ensure that the property of the belt is uniform. Then, a tensiometer is used to measure static tension of the B_1 , and the average of three tested results is regarded as the belt static tension of direct measurement.

(2). Indirect measurement

A piezoelectric stress sensor installed at the center of the *TEN* pulley is used to measure its hubload. Note that the piezoelectric stress sensor should be installed in the wrap angle bisector of the *TEN* pulley support.

The static tensions of the B_7 and B_8 are the same and equal to the belt static tension due to zero-torque of the *TEN* pulley shaft. With the measured hubload in static status, and according to the geometric relations between the *TEN* pulley and the belt spans adjacent to it, the static tension of belt (denoted by T_s) can be calculated as

$$T_s = \frac{1}{2} \cdot \frac{T_{hub}}{\cos\left(\frac{\psi_1(\theta_{t0}) - \psi_2(\theta_{t0})}{2}\right)} \quad (1)$$

where T_{hub} denotes the measured hubload of the *TEN* pulley; θ_{t0} is the static angle of the *TEN* arm between the positively oriented X-axis and the arm in the static status, as shown in Fig. 3; ψ_1 (ψ_2) represents the contact angle between the positively oriented X-axis and the longitudinal direction of the B_7 (B_8).

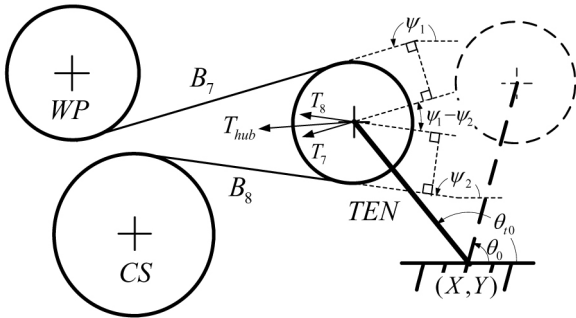


Figure 3. Tensioner assembly geometry.

Defined θ_t as the rotational angle of the *TEN* arm, which is positive in a counter-clockwise rotation. It can be observed in Fig. 3 that, the angle ψ_1 (ψ_2) depends on the *TEN* arm angle θ_t , the radius of WP and TEN (CS and TEN) pulleys R_7 and R_8 (R_1 and R_8), and the *TEN* pulley center location (X_8 , Y_8):

$$\begin{aligned} \psi_1(\theta_t) &= \pi + \arctan\left[\frac{R_7 + R_8}{L_7}\right] - \arctan\left[\frac{Y_7 - Y_8}{X_8 - X_7}\right], \\ \psi_2(\theta_t) &= \pi - \arctan\left[\frac{R_1 + R_8}{L_8}\right] + \arctan\left[\frac{Y_8 - Y_1}{X_8 - X_1}\right]. \end{aligned} \quad (2)$$

with

$$\begin{aligned} X_8 &= X + L_t \cos(\theta_t), \\ Y_8 &= Y + L_t \sin(\theta_t), \end{aligned} \quad (3)$$

and

$$\begin{aligned} L_7(\theta_t) &= \sqrt{(X_7 - X_8)^2 + (Y_7 - Y_8)^2 + (R_7 + R_8)^2}, \\ L_8(\theta_t) &= \sqrt{(X_1 - X_8)^2 + (Y_1 - Y_8)^2 + (R_1 + R_8)^2}. \end{aligned} \quad (4)$$

where, L_7 (L_8) denotes the length of the B_7 (B_8); L_t is the length of the *TEN* arm, and (X , Y) represents the location of the fixed-end point for the arm, as shown in Fig.3.

Measurement of belt dynamic tension

In the FEADS, only the dynamic tension T_7 (T_8) of the B_7 (B_8) can be indirectly calculated with the dynamic hubload of the *TEN* pulley. Assuming the dynamic tensions of the B_7 and B_8 are the same, and ignoring the influence of the angular fluctuation of the *TEN* arm. With the measured dynamic hubload of the *TEN* pulley, the dynamic tension of the B_7 (B_8) can be calculated by Eq. (1).

Measurement of rotational response

Each pulley except *TEN* pulley and two idler pulleys is equipped with an angle encoder to measure its rotational speed. Then, the angular velocity of each pulley is calculated by the measured rotational speed.

The movement of the *TEN* arm is obtained using a potentiometer equipped in the fixed-end of the arm. Note that the center of the potentiometer should be installed at the same axis with the rotational center of the *TEN* arm.

Excitation of the FEADS is the fluctuation of the crankshaft rotational speed, which is measured by an angle encoder installed in the outer ring center of the crankshaft torsional damper.

The slip ratio is defined as the relative velocity between belt and pulley, and divided by the belt velocity [21-22]. The expression of the slip ratio between belt and the i th pulley can be given by

$$\Gamma_i = \frac{V_B - V_i}{V_B} * 100\% \quad (5)$$

where Γ_i is the slip ratio between the belt and the i th pulley; $V_i = \dot{\theta}_i \cdot R_i'$, is the velocity of the i th pulley at the pitch diameter, here, $\dot{\theta}_i$ and R_i' are the angular velocity and the pitch radius of the i th pulley, respectively; the average velocity of the belt, $V_B = (\dot{\theta}_{idler1} \cdot R_{idler1}' + \dot{\theta}_{idler2} \cdot R_{idler2}')/2$, is determined by the angular velocities $\dot{\theta}_{idler1}$ and $\dot{\theta}_{idler2}$, and the pitch radii R_{idler1}' and R_{idler2}' of both idler pulleys before and after the i th pulley.

Measurement of belt transverse displacement

The transverse displacement of a point in the belt span is measured using a laser displacement sensor. Generally, the midpoint of the belt span with maximum tension or the longest belt span has the largest transverse displacement. According to the rotational direction of the CS pulley, the tension of the B_1 and B_8 are maximum and minimum, respectively. For this system, the transverse displacement at the midpoint of the B_1 is measured.

MEASUREMENT RESULTS AND ANALYSIS

Static tension

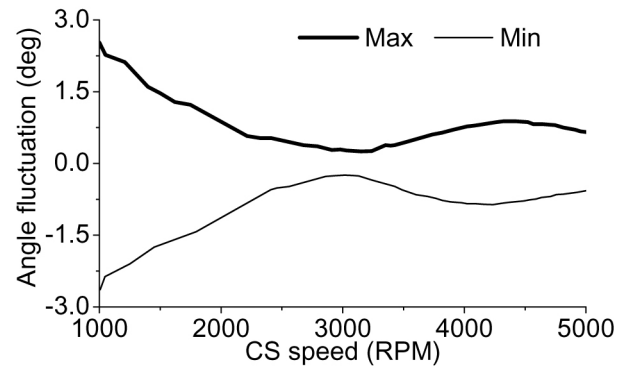
The measured tensions of the B_1 in three times are 540 N, 570 N, and 550 N, respectively. The average value, 553.3 N of these tensions is regarded as the belt static tension. It can be seen that 30 N tension difference exists between these tensions measured by tensiometer.

The measured hubload of the TEN pulley is 1080 N in the static status, and the static tension of the belt is 547 N calculated by Eq. (1). The tension difference is only 6.3 N between the static tensions of the direct and indirect measurements. Therefore, to avoid the trouble caused by calculation, the average value of several measurements of the static tensions by tensiometer is regarded as the belt static tension in engineering.

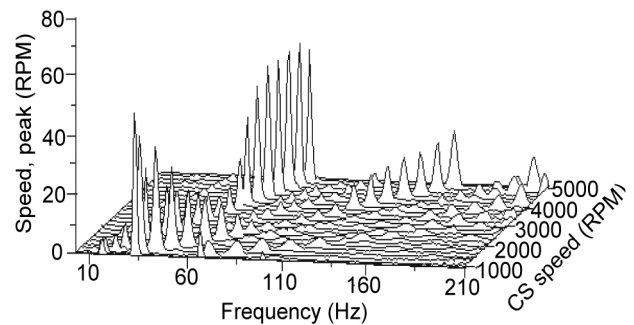
Rotational vibration of pulleys

In dynamic test, only ALT pulley bears accessory torque (5.5 N.m), while other driven pulleys are unloaded. The engine runs from 1000 RPM to 5000 RPM in 30 seconds with a constant acceleration. The input crankshaft load (62.5 KW) is controlled by a dynamometer.

Fig. 4 illustrates the rotational vibration of the CS pulley versus CS speed. It clearly indicates that the large excitation occur at the lower CS speed.



(a). Angular fluctuation amplitudes.



(b). Waterfall plot of the angular velocity fluctuations.

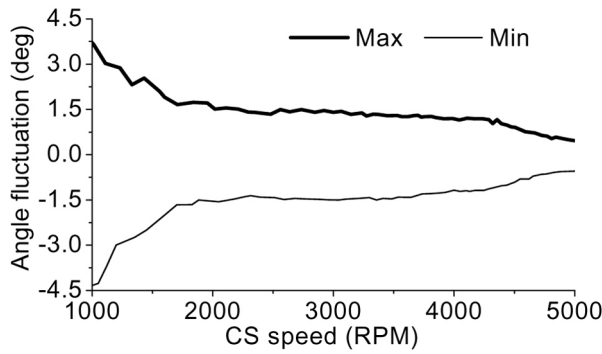
Figure 4. Rotational vibration of the CS pulley versus CS speed.

The rotational responses of AC and ALT pulleys versus CS speed are shown in Fig. 5 (a) and (b), respectively. It is clearly illustrated that rotational vibration responses of pulleys are larger when CS speed is lower due to the large excitation, and the angular fluctuation amplitude of the ALT pulley is the largest in the driven pulleys (about 8°).

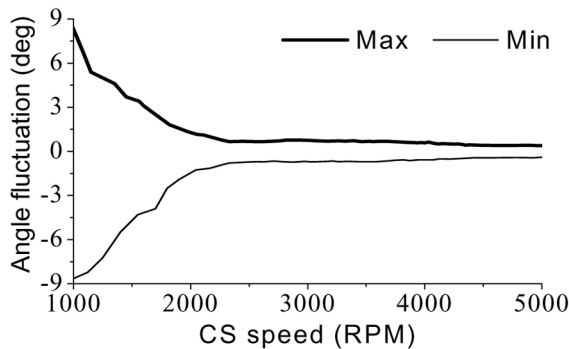
Rotational vibration of the TEN arm

The angular fluctuation amplitudes of the TEN arm versus CS speed are shown in Fig. 6 (a).

Fig. 6 (a) shows that the angular fluctuation amplitudes of the TEN arm are larger when CS speed is lower due to the large excitation, but become smaller with the greater CS speed. However, the maximum angular fluctuation amplitude for the TEN arm is about 3° , which is within 5° and meets the engineering requirements.

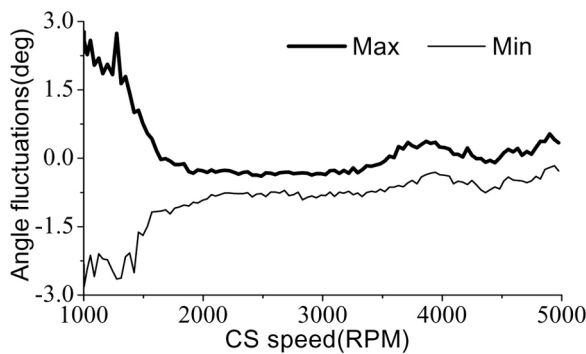


(a). Angular fluctuation amplitudes of the AC pulley.

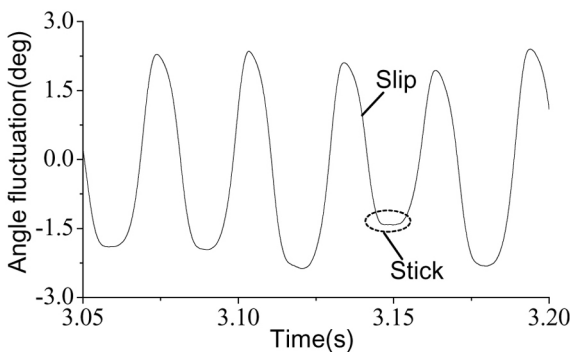


(b). Angular fluctuation amplitudes of the ALT pulley.

Figure 5. Rotational responses of the AC pulley and ALT pulley versus CS speed.



(a). Angular fluctuation amplitudes versus CS speed.



(b). Angular fluctuations versus time (1000 RPM).

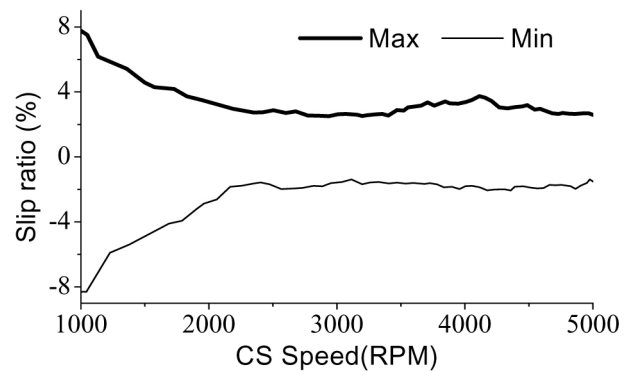
Figure 6. Rotation vibration of the TEN arm.

When CS speed is 1000 RPM, angular fluctuation of the *TEN* arm versus time is shown in Fig. 6 (b). Its curve shapes indicate that the stick-slip motion of the *TEN* arm is captured.

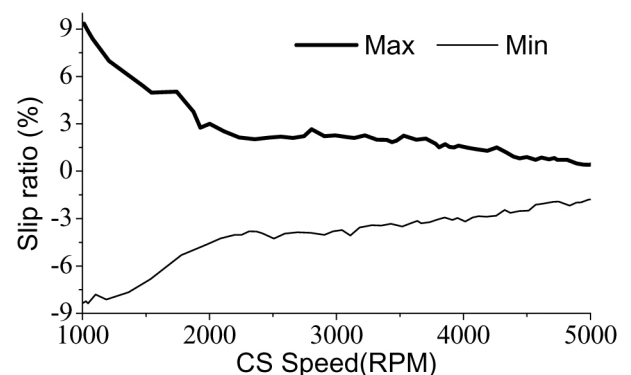
Slip ratio between belt and ALT (CS) pulley

The *ALT* pulley is the component with the highest inertia (including the inertia of the *ALT* rotor) of all driven pulleys, and with a certain torque during operation. Therefore, the *ALT* pulley may occur slipping because the large tension difference of the two belt spans (B_3 and B_4) adjacent to it. Additionally, the *CS* pulley may occur slipping because the largest tension difference of the tight belt span (B_1) and the slack belt span (B_8).

In the FEADS, the *PS* and *AC* (*AC* and *WP*) pulleys without external torque are regarded as two idler pulleys before and after the *CS* (*ALT*) pulley. The slip ratio between the belt and the *CS* (*ALT*) pulley versus *CS* speed is shown in Fig. 7. It clearly indicates that the slip ratios are larger in lower *CS* speed due to the large excitation.



(a). Slip ratio between the belt and the CS pulley.



(b). Slip ratio between the belt and the ALT pulley.

Figure 7. Slip ratio between the belt and the CS (*ALT*) pulley versus CS speed.

Hubload of the TEN pulley

Fig. 8 illustrates the hubload of the *TEN* pulley versus *CS* speed. It is clearly shown that the hubload is in a relatively