

Vibrations and Instability in Automotive Front End Accessory Drive Belt System

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ABSTRACT

The objective of this paper is to explore the dynamic features of vibrations and instability in automotive front end accessory drive belt system. To address this issue, a non-linear model of a moving beam with friction coupling on elastic foundation is developed. The model takes into account of a variety of nonlinear effects, including the contact stiffness nonlinearity, nonlinearities associated with the friction, tensioned parametric excitation, as well as the beam geometry nonlinearity. Different nonlinear properties are examined, and numerical simulations are conducted to investigate the system and to correlate with the experimental results. The major contributors to the dynamic contact instability are illustrated and are attributed to modal coupling and/or negative damping.

INTRODUCTION

The belt transmission systems have been widely used in industrial and automotive systems for motion and power transmission in the last 100 years. However, due to the vast breadth and depth of the relevant dynamics and

tribology issues, the fully understanding has been the classic yet critical topic for mechanical engineering community. Early studies on belts were mostly confined to obtaining the coefficient of friction and natural frequencies and modes by simple analysis.

For dynamics aspect, analytical models for axially moving systems have been extensively studied in the last 50 years. Sack [1], Miranker [2], Mote [3], Ulsoy [4] were among the pioneers in this research area. Ulsoy et al [5-11] examined nonlinear transverse belt vibration responses in the system composed of tensioner. Their study shows dynamic tensions in the spans are prescribed by torque variations in the adjacent driven accessories. They also studied instability phenomena of a belt tensioner system, including linear parametric instability, and made qualitative comparisons between numerical and experimental data. The dynamic tensions parametrically excite transverse vibration, leading to instabilities. They considered the possibility of parametric instability and presented a mechanism which may cause large transverse span vibration due to tension fluctuations. Abrate [12] discussed models available for analyzing the free and forced axial,

transverse and torsional vibrations of belts. The effects of initial tension, transport velocity, bending rigidity, support flexibility, large displacement and belt and pulley imperfections are discussed. Wickert *et al* [13-20] further investigated the nonlinear features of belt drive system. They modified the classical modal analysis method by casting the equations of motion for a traveling string into a canonical form. They studied experimentally the dynamic behavior of power transmission belt in the presence of two in-phase pulley eccentricities, validating the analytical model proposed in reference. They developed a modal perturbation solution for a continuous, non-autonomous and gyroscopic system with geometric non-linearity. Parker *et al* [21-24] developed a spatial discretization of this model extended to multiple pulleys. By incorporating the belt bending stiffness, they presented advanced model. Modeling the belt as a moving beam shows that transverse vibration of every span is linearly coupled with the rotations of the two adjacent pulleys at its ends. This coupling behavior is more consistent with observed vibration than prior models. The degree of span-pulley coupling depends on the equilibrium curvature of the span. Further, a coupling indicator is defined for each span to quantify coupling strength. Their attention is focused on analytical and numerical methods to calculate equilibrate and the effects of design parameters on the equilibrium and coupling indicator. Perkins, Beikmann *et al.* [25-33] showed that there exists a linear mechanism coupling rotational vibration and transverse vibration of the belt spans adjacent to the tensioner arm. They treated the belt as a moving string and studied a prototypical three-pulley model, which captured a linear coupling mechanism between the tensioner rotation and the transverse vibrations of the two spans adjacent to the tensioner. Other spans are decoupled from pulley rotations in the linear model. They demonstrated that non-linear coupling of rotational and transverse vibration arises from finite stretching of the belt. While such coupling is often small, it can become greatly magnified under conditions leading to an internal resonance. Since both quadratic and cubic non-linearity arise due to the non-linear coupling, and serpentine belt systems are hybrid systems with continuous and discrete components, it is challenging to perform the non-linear analysis. In their study, the two-to-one internal resonance is investigated using a numerical method. The non-linear vibration model was discretized using the linear eigenfunctions and the coupled vibration response was calculated using a Runge-Kutta method. They also developed a non-linear model to describe the longitudinal response of automotive serpentine belts and to predict the belt slip due to the dynamic tension fluctuation. They also proposed a general model for the rotational response of the entire serpentine belt drive system and applied the results to predict the onset of belt slip. Zhang, Zu *et al.* [34-36] established belt drive with linear viscous damping and gave a complex modal

analysis of the serpentine belt drive system. They studied the free and forced vibration of viscoelastic moving belts using the direct multiple scales method. They derived the closed-form solution of eigenfunctions of a prototypical serpentine belt drive system. Hawker [37] investigated the natural frequencies of damped drive systems with a dynamic tensioner. Kraver *et al.* [38] developed a complex procedure to solve both underdamped and overdamped systems. Zhu and Guo [39] studied free and forced vibration of an axially moving string with an arbitrary velocity profile. Ozkaya and Pakdemirli [40] investigated analytical solutions for the axially accelerating string. Ozkaya and Pakdemirli [41] studied the vibrations of an axially moving beam with time-dependent velocity. Zen and Müftü [42] discussed friction induced transverse vibrations of an axially accelerating string. Giampaolo and Müftü [43] studied the stability of an axially accelerating string subjected to frictional guiding force. Riedel and Tan [44] detailed coupled forced response of an axially moving strip with internal resonance. Barker, Oliver and Breig [45] detailed a dynamic analysis of belt drive tension forces during rapid engine acceleration. Kwons and Ih [46] studied the vibrational power flow in the moving belt passing through a tensioner. Pellicano and Bertuzzi [47] investigated the primary and parametric non-linear resonance of a power transmission belt. Kim and Lee [48] analyzed the non-linear vibration characteristics of a belt-driven system. Belt drive dynamics belong to the class of hybrid continuous/discrete dynamics systems, and all of the above discussed dynamics issues fall in the low frequency range. For instance, automotive multi-rib V belt drive system is one of the typical kind of belt drive systems, which has been widely used in automotive industry. Such kind of system exhibits complex dynamic behavior, including rotational vibrations of pulleys with the belt spans serving as coupling springs, and transverse and longitudinal vibrations in the various belt spans. Over the last decade, a tremendous progress has been made to understand and improve belt drive dynamic properties of this type. The related issues have been successfully addressed by industry and academics with specific softwares [49-50] and standard commercial software like SimDrive [51]. For this kind of efforts, the general solution of the governing nonlinear equations provides the coupled longitudinal and transverse response of the translating belt drive system. Typical simulation outputs include pulley hubloads, belt impact dynamic forces, and belt slip rates at the pulleys.

On the other hand, the belt tribology as the critical area concerning with power transmission capacity has been studied by a numerous researchers. The pioneers include Swift [52], Gough [53], Grosch [54], Childs *et al.* [55-60], Gerbert and Gerbert [61-66]. Childs and Gerbert conducted extensive fundamental researches on belt creep, slip, and other tribological topics. Yu *et al.*

[59-60] use experimental and finite element approach investigated the performance of running of V-ribbed belts. Frank [67] presented a mechanics and tribology analysis of belt drive. Rubin [68] gave an exact mechanics and tribology solution for steady motion of an extensible belt in multipulley belt drive systems. Dolan and Worley [69] provided a closed form approximation to the solution of V-belt force and slip equations. Alciatore and Traver [70] presented multiple pulley belt drive mechanics with both creep theory and shear theory. Fujita et al. [71] established the relation between the quantitative evaluation of the surface texture of rubber under the reciprocating sliding condition and the frictional characteristics, they provide an approach for the extraction of features from the digital images. Myshkin, Petrokovets, and Kovalev [72] discussed the general tribology features of polymers, including adhesion, friction, wear, and mass-transfer, Amijima, et al. briefed the tribological features of V-ribbed belts. Persson [74] summarized the advances on the theory of rubber friction. Hansson [75] gave out geometry conditions for good power capacity. SAE has established Standard for friction testing [76].

Moreover, the friction and dynamics interaction and instability problems have rarely addressed until last decade [77-87]. The studies of V-ribbed belt instability induced load carrying capability loss and squeal have not been received enough attention until recently [88-99]. Actually, this kind of high frequency dynamics and vibration problems associated with the belt drive system are very critical in applications. The friction-induced high frequency instability could result in the loss of load carry capability, worsen slippage and accelerate wear, as well as generate squeal. In real application, the belt drive system failures have been estimated and judged by the loss of power transmission capability (overslip), wear and noise. But the noise and the instability induced load carry capability have not been fully understood until recently. Connell and Rorrer [88] investigated the friction-induced vibrations in V-ribbed belt applications. Connell and Meckstroth [89] studied the Influence of pulley profile on automotive accessory drive noise. Meckstroth [90] presented a dynamic test procedures with varied condition to quantify the variations in load carrying capability of different belts and pulleys under different conditions. Meckstroth, Deneszcuk, and Skrobowski [91] investigated the accessory drive belt pulley entry friction and belt misalignment noise. Dalgarno, Moore and Day [92] investigated the tangential slippage and squeal of V-ribbed belts with experiments and finite element analysis, they attributed the noise from the excitation of the fundamental vibration of belts. Fujii et al. [93] analyzed the accessory belt lateral vibration in automotive engines. Sheng et al. [94] proposed a model and experimental investigation of belt slip and misalignment noise in automotive accessory belt drive system. Okura and Tatsumi [95] analyzed the

squeal radiated from serpentine belt for accessories drive system. In these vibration and instability analyses, the squeal other than the load carrying capability loss has been emphasized. Recently, Sheng, et al. [96-99] further characterized the fundamental properties of belt friction-induced noise and instability by extensive experiments, including dry and wet conditions, tangential and radial type slip. They also presented different mathematical models to clarify friction-induced instability and squeal generation mechanisms. Some of their models are contradictory with the conventional ones that have been used in the last decade. This paper is intended to further improve related model by developing the understanding on the contact instability and a variety of nonlinear properties associated with it. To address the salient nonlinear properties associated with the contact instability, a new model is proposed to take into account of a variety of nonlinear effects, including the contact stiffness nonlinearity, nonlinearity due to tensioned parametric excitations, friction-induced nonlinearity including stick-slip, negative damping and modal coupling, as well as the beam geometry nonlinearity. After a coordinate transformation and some mathematical modifications, the governing equation is rewritten into an extended operator form that retains the mathematical structure of the mathematical equation. Galerkin discretization is readily applied in this extended operator context. Although the reformulation initially seems complicated, the key ideas are such that the method is straightforward to implement. Different nonlinear properties are examined respectively. Numerical simulations using Matlab are conducted to investigate the nonlinear properties and to correlate with the experimental results in terms of spectrum signatures. The major contributors to the dynamic contact instability are illustrated and are attributed to modal coupling and/or negative damping. The principles involved here can be applied to other kind of V and flat belt.

EXPERIMENTAL OBSERVATIONS OF CONTACT INSTABILITY

V-ribbed belts have been used as power transmission elements since the 1950s when they were developed to provide a combination of high flexibility and high power transmission capacity for drives where small diameters were required. Fig 1 shows a typical automotive engine with the V-ribbed belt drive system in its front end. In this system, the V-ribbed belt is used to transmit power from driving pulley to driven pulleys. The driving shaft is crankshaft, and the driven shafts include the shafts of alternator/generator, air-con, power steering, water pump, idler and tensioner. Fig.2 shows the schematic of typical V-ribbed belts. The main belt constituents are polyester cords as the tension member, a cushion elastomer compound which envelops the cords, and a rib elastomer compound which forms the belt ribs, and

which may be stiffened through the use of short fiber reinforcement in the axial direction. The flexibility of the belts arises from the small section depth, and the power transmission capacity from the V-shaped ribs, which develop traction. Given that a belt has the power transmission capacity required by a drive, the belt over-slip, wear and noise are the indexes by which V-ribbed belt may be defined as failure. In some application, it has been observed that tangential slip of the belt relative to the pulley can lead to contact instability, which further worsen the system by accelerate over-slip and develop squeal. This situation is particularly critical for some belt designs when its instability occurs at low slip, since small slip are generally unavoidable due to the operation change and system vibrations. Once the contact instability takes place, it is likely to reduce the interface friction and accordingly to help further develop slip. Fig. 3 is the typical result of the belt load carry capability vs slippage measured on a dry belt system. The belt slip s is defined as $s = \Delta\omega / \omega_r$ where ω_r is the speed of the driver pulley and $\Delta\omega$ is the difference in speed between the driver and driven pulleys. In the test the torque ranges from 10 to 20 N m, the tight side tension on the driven/driver pulleys ranges 10-45 kgf/rib and the belt speed ranges from 100 to 2000 rpm. It shows that the transmitted torque load exhibits a sharp drop after certain slip value denoted as point Q, which is roughly equivalent to 5%. In the belt test two distinct regions with different behaviors were observed: 1. when the belt slip is below 5% the drive was stable; at slip levels of higher than 5% the drive was unstable. No oscillations was observed at slip levels below Q or 5%, whereas at slip level higher than Q or 5%, sustained and severe oscillations are recorded. Fig. 4 is the recorded acceleration waveform and its spectrum measured from a idler hub before point Q, Fig.5 is the measured acceleration waveform and the spectrum of the idler hub just after point Q. Fig.6 is the measured acceleration waveform and the spectrum of the idler hub after point Q, which shows that the system exhibits sustained and severe oscillations. The acceleration dominates at a primary oscillation with high frequency and its harmonics. A strong squeal are audible when the load loss takes place. Fig.7 is the measured results of another belt system. The high frequency oscillation is associated with the drop of load carrying capability. It is nature to infer that the loss of load carrying capability is due to the contact instability, which reduces the interface friction. The apparent coefficient of friction yielded from Euler equation also exhibits the similar drop trend as the torque. But this apparent feature is not inherent and is not the root cause of the radical loss of load carrying capability. Actually, this type of loss of load carrying capability is very sensitive to the application conditions. Slightly change the condition could eliminate the occurrence of the instability and the load loss. For instance, reducing the

wrap angle could eliminate it for a given system when the instability takes places. This kind of instability and load loss is tightly linked with system dynamics instead of solely with friction feature. As the instability onsets at very low slippage rate, it is particularly critical to applications where small slips are usually always unavoidable. This kind of instability and load loss is different from the conventional type of the load carry capability loss that occurs at very high slippage rate, as documented by Mechtroth [90]. The later is not a big concern as large slips rarely has chance to take place in applications.

MODELING OF BELT-PULLEY INTERFACE FRICTION AND CONTACT DYNAMICS

1) FRICTION MODEL

In order to estimate belt system torque carrying capacity, the friction must be known, not only with a new belt and pulley, but with aged belts and pulleys and over a broad range of environmental and application conditions as well. Conventionally, the coefficient of friction (COF) has been used to characterize friction, and it is usually measured with stationary, semi-stationary or dynamic test equipments. Such equipments are usually unable to provide a very repeatable, single coefficient of friction for the entire belt length, over the full range of applied and environmental conditions. The coefficients of friction of belts have been calculated using classical Euler belt equation. While this equation may not capture all the subtleties of the belt slip condition, it provides a good dynamic average characterization of a belt/pulley limit torque situation, and has been used as SAE standard and used by whole automotive industry. This equation applies only at a slip condition. This equation implies that the coefficient of friction is not a function of pulley diameter, speed or number of ribs. The validity of these assumptions has been confirmed by testing on this stand especially for break-in pulleys and belts. Consider a belt driven pulley shown in Fig. 8, the general relationship is given by

$$T_h - \rho AV^2 = (T_l - \rho AV^2) e^{\frac{\mu\beta}{\sin(\theta/2)}} + I\dot{\omega} / R \quad (1)$$

Where ρA is the length specific mass of belt, V is the circumferential belt speed, T_h is tight side tension, T_l is low side tension, I is the inertia of pulley, β is belt wrap angle (radians), μ is the coefficient of friction of the belt to pulley, excluding the V angle wedge effect. θ is the V wedge angle. Since this is for one rib of a V-ribbed belt, multiplying the tension by the number of ribs in the left and right terms will not change the equation, thus the above equation are independent of the number of ribs, and the total tension of the belt can be used in this

equation. If the wedging effect of the V angle is included in the coefficient of friction, the equation becomes:

$$\mu_e = \frac{1}{\beta} \ln \left(\frac{T_h - \rho AV^2 - I\dot{\omega}/R}{T_l - \rho AV^2} \right) \quad (2)$$

In which $\mu_e = \mu / \sin(\theta/2)$ is a larger number that includes wedging effect. An average or effective COF can be obtained between belt and pulley, but this value is often subject to wide variations. Such inconsistency is due to an absence of uniform sliding velocity and cleanliness, inconsistent surface topography, and a complex distribution of interface pressure between belt and pulley, particularly the system dynamics effects. In industrial engineering design, the COF has been assumed to be constant, but the actually measured COF are highly velocity or slip dependent [90,94-5]. Of particularly interested feature is the negative slope of COF vs slip, as this specific friction feature is linked with negative damping and is likely to results in instability or self-excited vibrations. Based on [90, 94-95], the negative slope is always the case for wet belt system, whereas it only takes place at very high slip for most of dry belt. In [98], the wet belt interface is formulated to be the unsteady mixed lubrication and some boundary lubrication. For wet belt applications, the full fluid lubrication is unlikely to take place due to the relatively low velocity, low viscosity of water and the open interface. The lubrication takes place in the mixed or boundary regimes, so that the frictional resistance is developed due to a combination of fluid shear over entire contact and solid friction at localized asperity contacts. In mixed lubrication, the load is carried primarily by the fluid with solid-to-solid interaction also contributing significantly to the friction. In boundary lubrication, the load is carried primarily by the solid asperities with the fluid shear still contributing significantly to the friction. Furthermore, under typical application conditions, COF is a function of the surface velocity, curvature and normal contact load on the mating surfaces. A unified principle that can represent the interface under all conditions has eluded most researchers. Many alternative formulations have to be applied based on different application regimes.

It is noted that the belt load carry capability usually has a loss at high slippage rate, which is different from the problem addressed here. At very high slippage rate, the loss of torque usually does not accompany with high frequency oscillation, and it has been attributed to be the viscoelastic friction effect between two contacting rough surfaces. Many existing analyses have demonstrated that the dynamic interaction of two surface exhibit complicated phenomena, particularly some surface parameters and preload can lead to negative friction torque/speed characteristic. Negative slope in friction torque/speed relation exists at higher preload. It is made

more severe by the presence of elastic tangential force. Lower preload tend to diminish or completely remove the negative slope in the friction torque-sliding speed relation. The elastic contact factor, angular speed, normalized average radius of asperity summit curvature, have strong influence on equivalent negative slope of the torque vs speed curve [87]. Farhang and Lim [87] detailed an analysis of a visco-elastic account of interaction at the micron scale, and its statistical quantification through the approximate analytical representation of the statistical expectation of contact force and the introduction of the contact force into the macro-scale dynamics of the two-disk system. Steady-state analysis of the system establishes the relation between friction torque and speed and supports observed behavior of many mechanical systems with friction. As a result of coupling of the macro-system's dynamics and contact, there are combinations of parameters at the micro- and macro-scale that yield negative slope in friction torque/sliding speed relation, a well-known source of dynamic instability. When the belt operates near their maximum load capacity, very high contact pressure occurs at the mesh interface. This may lead to the commonly encountered load loss regime.

It is generally very difficult to derive an applicable explicit and unified friction formula despite we can elaborate the processes. For the dry and wet cases respectively, the following models [98] are used to model COF by fitting to classic Ford belt friction data given in [90]. The power law model is used to model dry friction:

$$F_D(\dot{x}) = F_0 + F_1 \dot{x}^\alpha - F_2 \dot{x}^\beta \quad (3)$$

in which F_0 is the minimum level of Coulomb friction, and α and β (ranging from 1/2 to 2) are empirical parameters. F_1, F_2 are dynamic friction parameters reflecting the increase feature at low slip and loss feature at very high slip.

The following power model is used to model wet friction:

$$F_W(\dot{x}) = F_f + f_s = F_c + (F_0 - F_c) e^{-|\dot{x}/\dot{x}_s|^\delta} + F_v \dot{x} \quad (4)$$

in which F_0 is the level of static friction, F_c is the minimum level of Coulomb friction, and $\dot{x}_s$ and δ (ranging from 1/2 to 2) are empirical parameters. F_v is the viscous friction parameters.

By appropriately choosing the parameters, the fitted dry and wet COF curves are plotted in Fig.8-9, each has two represent cases, a and b. To incorporate the friction models into motion equation, the fitted friction laws can be further represented by polynomial functions of the slip velocity: