Automotive Accessory Drive System Modelling

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INTRODUCTION

Belt driven accessories are ubiquitous in the automotive industry. The past practice was to include an additional belt for each accesory added as would be the case for the power steering pump and air conditioning compressor. This creates problems during replacement of a broken belt; particularly if it is located at the back of the belt stack since all other belts must be removed to affect its replacement. However, the automotive industry has began replacing many of the conventional multiple belt accessory drive systems with a new drive arrangement capable of powering several (if not all) of the accessories with a single belt.

The profile of this single belt, sometimes called a Poly-V belt, differs markedly from the traditional V-belt. As illustrated in Figure 1, it is much wider, thinner and uses a different profile on its primary contact surface. The drive systems that use this style of belt have been called "serpentine drives". This refers to convoluted path the belt must follow to accommodate the accessory locations and to provide sufficient contact with the accessory pulleys to transmit the driving force (especially when the flat side of the belt is used). The V-belts used in the conventional automotive accessory drives generally do not have sufficient flexibility to withstand the back bending associated with serpentine drives.

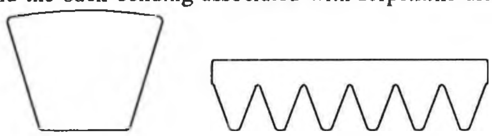


Figure 1 Comparative belt profiles having the same minimum radius of curvature.

The thinner profile of these ribbed belts means a more compliant belt with a longer flex life and less centrifugal tension loss due to belt wear and seating. Serpentine systems with the ribbed belts also provide greater drive ratio flexibility since smaller diameter pulleys can be used without sacrificing the belt's flex fatigue life.

PROBLEM IDENTIFICATION

Like their predecessors, the automotive serpentine accessory drive systems are still troubled with a number of noise and vibration problems. These are perceived as quality defects by the customer, and increased pressure has been placed on the suppliers of the accessory drive technology to reduce them. These problems frequently take the form of large amplitude belt span vibrations known as belt whip or belt flutter which can produce objectionable noise. Reducing the associated vibrations would not only decrease the generated noise levels but would also reduce belt and bearing wear.

It is generally accepted that the span vibrations can be directly attributed to two different classes of excitation. These are: (1) direct or internal - irregularities in the sheave, belt or shaft and (2) parametric or external - variations in the belt span tension. Of these two forms of excitation, the majority of the vibrations encountered in belt drive systems can be attributed to torque or tension variations, although impulses from the system irregularities can be significant. The geometric parameters of the system must be selected so that the natural frequency of vibration of the belt spans are not integral multiples of the excitation frequency caused by system irregularities. This last point is important since large amplitude vibrations of the belt span may be caused by this resonance effect.

The majority of the investigations concentrating on this research area have been associated with the determination of the natural frequencies of the span vibrations and the instability which results from belt tension fluctuations. Little emphasis has been placed on the role played by the angular motion of the accessory pulleys due to complex crankshaft oscillations and accessory loading characteristics. The magnitude and frequency of this angular motion is directly responsible for the belt tension fluctuations and hence the instability of the span vibrations. Thus, in response to those who would suggest that non-linear modelling is the appropriate direction to follow, it should be recognized that it in necessary to first determine whether that degree of modelling is required.

THEORETICAL MODEL

A lumped parameter multiple–degree–of–freedom model was developed from which the angular modal parameters (resonance frequencies and mode shapes) of an automotive serpentine accessory drive system could be estimated.

Belt Configuration Code

The general equations governing the angular motion of a serpentine belt drive system depend on the belt configuration (the path the belt follows around the accessories). A code was established to identify the belt configuration and is illustrated in Figure 2. Illustrated are the eight possible combinations of three consecutive pulleys which must be considered when analyzing the motion of a particular pulley or accessory. With this code "Y" represents a pulley which is reverse-wrapped (driven by the belt's flat surface) and "N" indicates a pulley which is not reverse-wrapped (driven by the belt's ribbed surface).

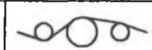
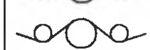
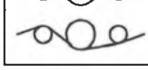
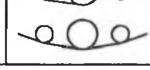
ILLUSTRATION	CONFIGURATION	ILLUSTRATION	CONFIGURATION
	N–N–N		Y–N–N
	N–N–Y		Y–N–Y
	N–Y–N		Y–Y–N
	N–Y–Y		Y–Y–Y

Figure 2 The possible belt configurations for three consecutive pulleys.

Belt Span Stiffness

It has been observed that, over the range of belt tensions typically used in automotive applications, the belt can be assumed to obey Hooke's Law (i.e., constant stiffness). The stiffness value for each span is found from the belt span geometry and modulus of elasticity.

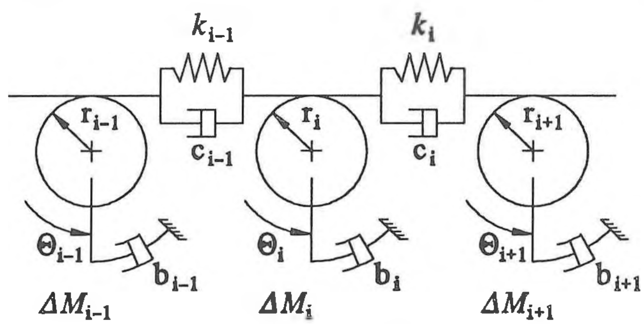


Figure 3 Lumped parameter model of connected pulleys and belts including elasticity and damping effects.

Equations of Motion
The analytical model of the accessory drive system represents the accessories and pulleys as inertias constrained to move in only one direction. These are coupled by belt strand connections which are assumed to be linear and massless springs with linear viscous dampers as illustrated in Figure 3. By applying Newton's second law to the analytical model the internal and external torques are equated and a mathematical model is obtained in the form of a system of second order ordinary differential equations. These equations govern the rotational motion and are assumed to represent deviations from the steady motion. The internal torques consist of the inertia, damping and elasticity terms while the external torques are the excitation forces and include the moments due to fluctuating accessory loads and engine torques.

A system of statically coupled ordinary differential equations describing the rotational motion of each pulley can be obtained by applying the analysis technique to each of the pulleys in the system. The resulting equations can be expressed in the standard matrix form

$$[J]\ddot{\theta} + [C]\dot{\theta} + [K]\theta = [F] \tag{1}$$

where $[J]$ is the inertia matrix, $[C]$ is the damping matrix, $[K]$ is the static or stiffness matrix and $[F]$ is the forcing function vector. The inertia matrix consists of the pulley and accessory inertia values along the diagonal while the damping and stiffness matrix elements ($C_{i,j}$ and $K_{i,j}$) are defined by the system geometry.

Unlike the typical spring–mass–damper damping and stiffness matrices, the off–diagonal elements may be positive and the elements in the upper right (i=1, j=N) and lower left (i=N, j=1) positions are not zero. Once the stiffness and damping factor for each belt span and the component damping factors have been determined the system of equations can be solved for the resonance frequencies, mode shapes and forced response.

Resonance Frequencies and Mode Shapes
The resonance frequencies and mode shapes of the accessory drive system are determined from the mathematical model; however, due to the presence of complex eigenvalues and eigenvectors, the process is slightly more complicated than with undamped systems. The homogeneous form of the general matrix equation, a set of N second order ordinary differential equations, was reduced to the set of $2N$ first order equations as illustrated in Equation (2).

$$\begin{bmatrix} 0 & J \\ J & C \end{bmatrix} \begin{Bmatrix} \ddot{\theta} \\ \dot{\theta} \end{Bmatrix} + \begin{bmatrix} -J & 0 \\ 0 & K \end{bmatrix} \begin{Bmatrix} \dot{\theta} \\ \theta \end{Bmatrix} = \begin{Bmatrix} 0 \\ 0 \end{Bmatrix} \tag{2}$$

The roots are complex with the real component being the exponential decay factor and the imaginary component being the damped resonance frequency in radians per second. For stable systems the eigenvalues are real and negative or complex with negative real parts; the complex eigenvalues occur as conjugate pairs.

EXPERIMENTAL WORK

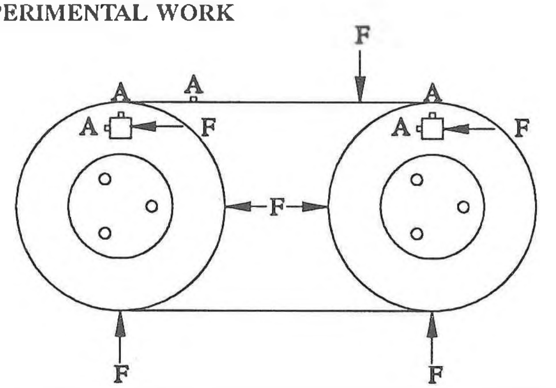


Figure 4 Experimental apparatus showing the excitation ("F") and vibration ("A") measurement locations.

A number of experiments were conducted on two- and three-pulley systems to investigate the angular vibration characteristics of simplified accessory drives. The effects of component inertia, belt tension and span length and belt tension on the belt span transverse vibration resonance frequencies. This investigation included the use of an impact hammer and accelerometer to extract modal information. Measurement locations are illustrated in Figure 4.

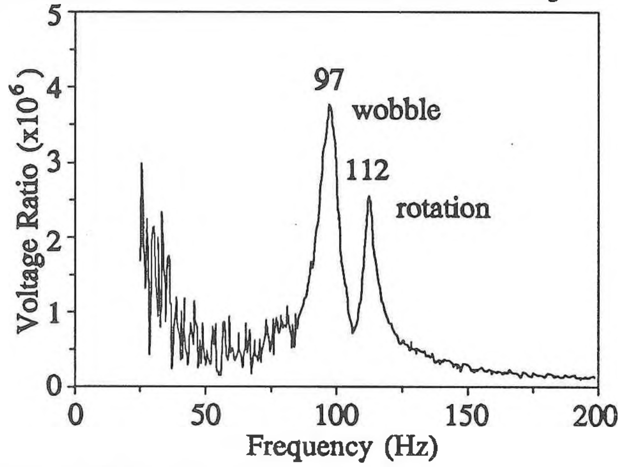


Figure 5 Sample system frequency response.

A sample of the experimental results are presented in Figure 5. Notable is the presence of both the rotational natural frequency as well as a "wobble" frequency. This phenomenon has been identified as a bending mode natural frequency of the shaft on which the pulley is mounted.
The "wobble" frequency has been observed to increase with belt tension, possibly due to a stiffening of the pulley in the bearing mount. It has also been observed to decrease as the pulley inertia is increased (mass of the pulley).

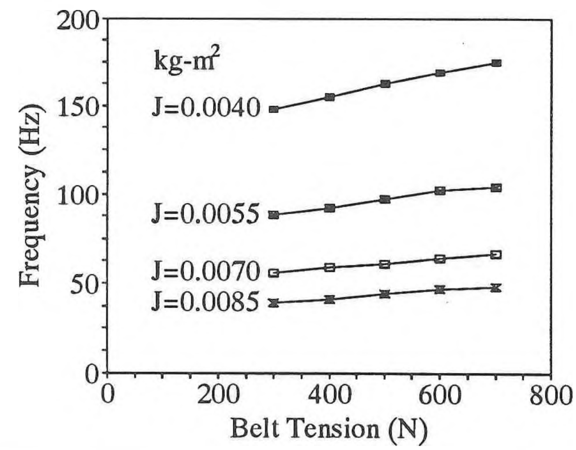


Figure 6 Experimental "wobble" frequencies for the system illustrated in Figure 4.

CONCLUSIONS

Thirteen different systems were investigated, and the model has been shown to provide good agreement with the measured resonance frequencies, both angular and transverse. The systems tested also exhibited a "wobble" frequency in the direction of the resultant belt span tensile force which was associated with the pulley shaft/bearing compliance. Present also was a "wobble" perpendicular to the resultant belt tensile force. This frequency was affected by the same parameters but was lower due to the decreased compliance in the direction of the resultant belt tensile force. For the two-pulley system with a wrap angle of 180 degrees on the pulleys the difference between the two "wobble" frequencies was quite significant. This difference was much less significant for the three-pulley systems because the angle of wrap was only 120 degrees.