The Dynamic Analysis of an Automotive Timing Chain System

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ABSTRACT

The purpose of this paper is to describe the development of a complete model for the dynamic analysis of an automotive timing chain system. In order to produce the most accurate results, all of the system components are modeled, including the rolling and silent chains, the chain guide, the chain arm, the sprockets, and the hydraulic tensioner. The completed model has been used to analyze many different characteristics concerning automotive timing chain systems. The nonlinear model is used to simulate the longitudinal and transverse responses of the timing chain, as well as the chain impact dynamics. A detailed model of the hydraulic tensioner is derived and used to predict the plunger and check ball motions. In addition, both the free and forced vibrations of the model are investigated. For the free vibration analyses, the natural frequencies of the system and the associated mode shapes are analyzed. For the forced vibration analyses, the crank motion and the cam torques are prescribed for the model, and then the vibrational amplitudes for each component are examined. For the production engine considered, the experimental measurements of the plunger motion and the natural frequencies are in good agreement with the analytical results.

INTRODUCTION

Chain drives are widely used in a variety of mechanical systems for the transmission of power. Their popularity is rapidly expanding, especially in the automotive industry, because of their numerous advantages. Chain drive systems are easy to assemble and adjust; they are highly efficient; they are durable, reliable, and compact; and they are capable of attaining a wide range of power and speed capacities [1]. Although chain drives provide several advantages, noise, vibration, and harshness are significant design concerns.

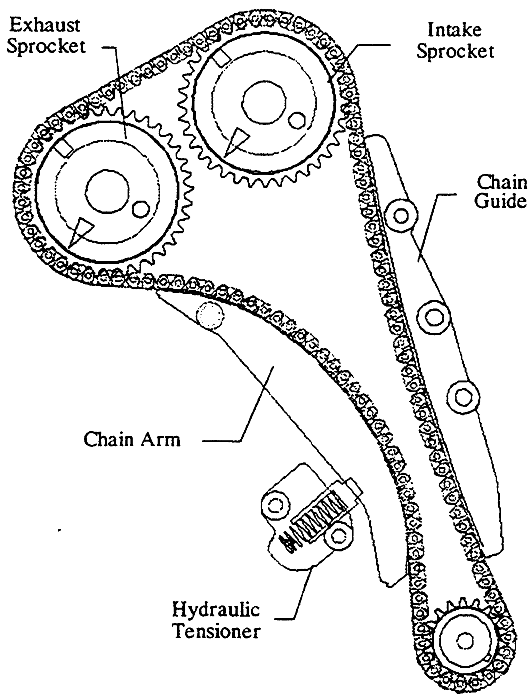

Figure 1: ADAMS timing chain model

With the requirement of lighter weight and higher speed chain drives within the automotive industry, the analysis of timing chain systems is becoming especially important. In order to further optimize their design, a more comprehensive knowledge of the system contact forces and impacts is desired. These aspects have been investigated by a number of researchers but the understanding of the dynamics is still limited.

As with most of the published work concerning timing chains, Fritz and Pfeiffer [2] focused their analysis on the rolling chain type of system. The model described in this paper differs from most others because it places considerable emphasis on the silent chain type of system and it accounts for backlash effects between chain links and sprockets. The model provides a method to accurately calculate the impact dynamics between system components (such as between the chain links and the sprocket or chain guard) and effectively predict the natural frequencies of the complete system.

The entire timing chain system model is created and analyzed using ADAMS dynamic system simulation software. This model consists of four main divisions: the silent chain portion, the rolling chain portion, the chain guide and arm, and the hydraulic tensioner. ADAMS is capable of solving the considerable system of equations that results from such a highly complex system and it enables the model to be easily modified in order to investigate different scenarios. Figure 1 shows one particular configuration. This arrangement includes a silent chain, with a crank shaft and an intake and exhaust sprocket, as well as a tensioner and a chain guide and arm.

CHAIN DRIVE COMPONENTS

Roller Chain

The roller chain subsystem consists of two main components: the chain portion and the sprockets. Figure 2 shows a representative subsection of a roller chain system.

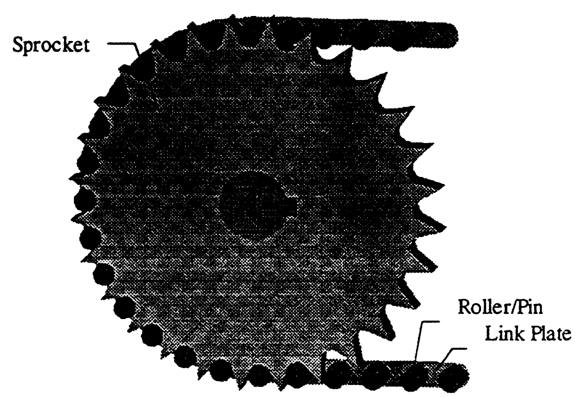

Figure 2: Roller chain configuration

A typical rolling chain is made up of many individual links which consist of rollers connected by link plates with pins. Each of the rollers is modeled as a rigid body part. Modeling the link plates and the pins as separate parts does not substantially affect the dynamics of the system. Therefore their mass and inertia properties are lumped together with those of the rollers.

The flexibility associated with the link plates is represented in the model with bushing (spring and damper) forces which act between adjacent rollers (Figure 3). The stiffness and damping coefficients are determined from the test data. The bushing rates can be modeled as nonlinear or linear.

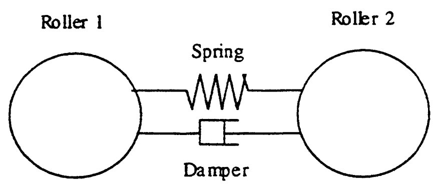

Figure 3: Link flexibility model

Two sprockets are modeled as part of the roller chain subsystem. Each is treated as a separate rigid body. The sprockets are similar to spur gears but they are designed to mesh with the links of the roller chain.

Silent Chain

The silent chain subsystem, like the roller chain subsystem, also includes sprockets and a chain portion. However, these components vary from those that make up the roller chain. This is due to the substantial fundamental differences in the designs of roller and silent chains. A section of a silent chain system is shown in Figure 4.

A silent chain consists of many different links connected with pins. Again, there is no advantage to modeling the pins as separate components so the chain model treats only the links as separate parts. These parts are rigid bodies with mass and inertia properties which take into account the affects of the pins.

An individual silent chain link looks much different than one from a roller chain. The profile, which resembles a tooth, consists of several different lines and curves in a complex arrangement (see Figure 4). As with the roller chain, the connections between these links are modeled with bushings to account for the flexibility.

The sprockets of the silent chain subsystem serve the same purpose as those of the rolling chain subsystem. However, they are designed to engage specifically with the links of the silent chain and thus their tooth contour is different.

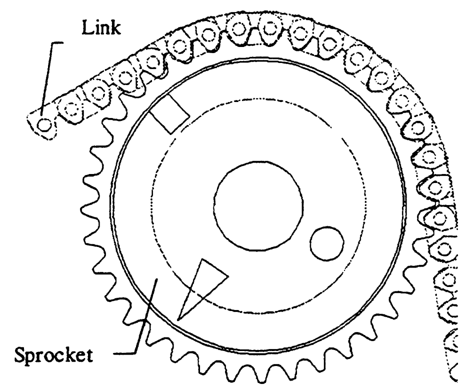

Figure 4: Silent chain configuration

Chain Guide and Arm

In a chain drive system, the chain guide and arm ensure that the chain remains on the path. The chain guide directs the tight chain portion which runs from the camshaft to the crankshaft sprockets. Conversely, the chain arm directs the slack portion of the chain which runs opposite (from the crankshaft to the camshaft). The arm also serves to distribute the force on the chain from the hydraulic tensioner.

The chain guide and the chain arm are both modeled as separate rigid body parts. Their geometric profiles consist of a series arcs with different radii and can be seen in Figure 1.

If desired, the chain guide and chain arm can be modified so that they are modeled as flexible bodies. This allows for calculation of stresses and bending moments for the parts.

Hydraulic Tensioner

The hydraulic tensioner is an integral part of timing chain systems because it provides dynamic control and maintains the chain tension. In addition, hydraulic tensioners increase chain life and prevent excessive vibrations. Figure 5 shows a close up picture of the ADAMS hydraulic tensioner model.

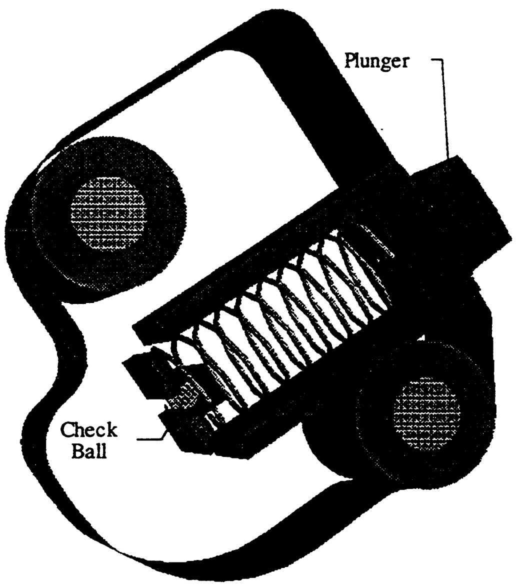

Figure 5: Hydraulic tensioner

The hydraulic tensioner model consists of the housing, a plunger, and a check ball. The plunger is the part of the tensioner which applies the force to the chain arm, which in turn helps to control the tension of the chain. Derivations from fluid dynamic theory are used to prescribe the behavior of the plunger and the check ball. [3, 4].

The fundamental equations used are based largely on the law of conservation of mass. If a control volume with weight flows both in and out is considered, the rate at which mass is stored must equal the difference in the incoming and outgoing mass flow rates:

$$\sum W_{in} - \sum W_{out} = g\frac{dm}{dt} = g\left(\frac{d(\rho V)}{dt}\right)$$

where

W_{in} = weight flow in

W_{out} = weight flow out

g = gravity

ρ = density

V = volume

The above equation is known as the continuity equation and can be expanded further to obtain:

$$\sum W_{in} - \sum W_{out} = g\rho\frac{dV}{dt} + gV\frac{d\rho}{dt}$$

The density of a liquid is a function of both temperature and pressure. This relationship is described by the equation of state. If the equation of state is combined with the continuity equation from above, then a relation can be derived for the change in pressure with respect to time:

$$\frac{dP}{dt} = -\frac{\beta}{V}\left(\frac{dV}{dt} - \sum Q_{in} + \sum Q_{out}\right)$$

where

P = pressure

β = bulk modulus

V = volume

The theoretical incoming flow rate, Q_{in}, occurs at the orifice plate within the hydraulic tensioner, and is represented by:

$$Q_{in} = C_d A_{ori}\sqrt{\frac{2}{\rho}(P_{in} - P)}$$

where

C_d = discharge coefficient

A_{ori} = orifice area

P_{in} = supply pressure

P = chamber pressure

Because the fluid in the tensioner is under high pressure, it often leaks through the annular gap between the piston and the cylinder. To determine this leakage flow rate, Q_{out}, the piston-cylinder arrangement is "unwrapped" and approximated by considering the flow between infinite parallel plates. The derivation yields:

$$Q_{out} = \frac{\pi D_{out} a}{2}\dot{x} - \frac{\pi D_{out} a^3 (P - P_e)}{12\mu L}$$

where

D_{out} = plunger outside diameter

a = gap between plunger and housing

$\dot{x}$ = translational velocity of plunger

P = chamber pressure

P_e = external pressure

μ = viscosity

L = length of plunger

In order to correctly model the hydraulic tensioner, all of the forces acting on the check ball and the plunger should be represented. Figure 6 shows all of the forces which act on the plunger. The net force on the plunger can be expressed as the sum of all of these forces:

$$\sum F_{plunger} = PA_{in} + k_p x_p + \tau_{xy} A_{shear} - F$$

where

P = chamber pressure

A_{in} = internal plunger area

k_p = plunger spring stiffness

x_p = plunger displacement

τ_{xy} = shear stress

A_{shear} = shear area

F = contact force from chain arm

Figure 7 shows all of the forces which act on the check ball. Summation these forces will yield the net force on the ball:

$$\sum F_{ball} = (P_{in} - P)A_{ori} - k_b x_b - F_d$$

where

P_{in} = supply pressure

P = chamber pressure

A_{ori} = orifice area

k_b = check ball spring stiffness

x_b = check ball displacement

F_d = drag force

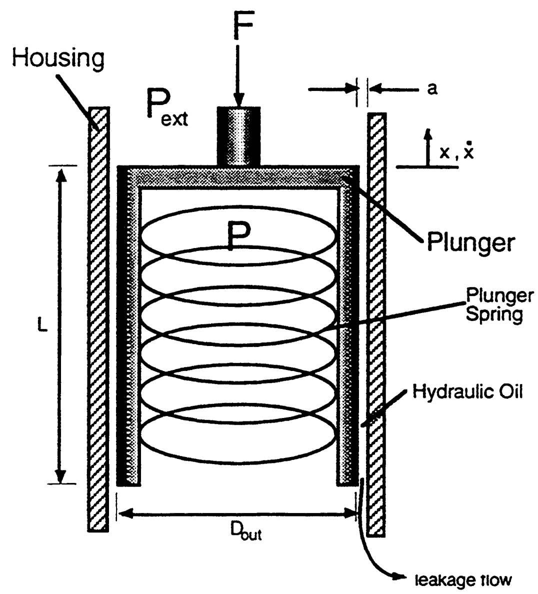

Figure 6: Forces acting on the plunger

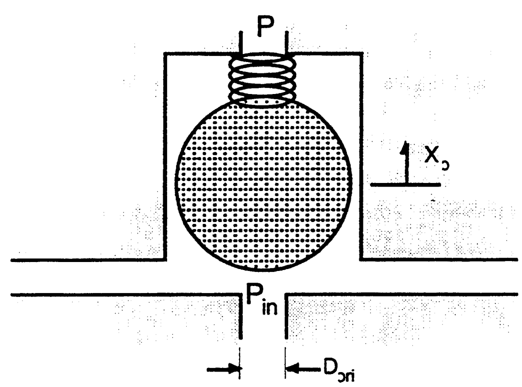

Figure 7: Forces acting on the check ball

CONTACT MODEL

In order to conduct fully dynamic analyses, it was necessary to develop a contact model. Contact occurs at many points throughout the timing chain system. Both the roller and silent chain links were modeled to account for the contact that occurs between the chain guide and arm as well as with the sprockets. In addition, the roller chain and silent chain contact models were created to support multiple contacts per link.

The contact surface profile of the sprockets was constructed of various arcs and line segments. Thus the possible contact types between the roller chain links and the sprockets are circle to circle and circle to line. Because the silent chain teeth and sprocket profiles differ from those of the roller chain, line to line contact is also a possibility for the silent chain subsystem.

To determine the specific points of contact, a geometric search algorithm was needed. An example of one is described in *Finite Element Procedures for Contact-Impact Problems* by Zhi-Hua Zhong [5]. Because the modeled contact was of a chain to a sprocket, determining the specific contact seat on the sprocket immediately helped eliminate much needless searching. From the specific seat, the individual element to element types of contact were then determined.

The contact was modeled using both forces and torques so as to account for all of the impact dynamics. In order to determine their values at each contact point, several variables were utilized. It was necessary to determine the position of the contact, the surface normal at the point of contact, and the inter-penetration distance of the two surfaces. Furthermore, stiffness and damping characteristics were tuned for the impacts.

MODEL VALIDATION

In order to validate the model, simulation data was compared to actual experimental results from a test on a V-6 engine. The engine was instrumented with devices capable of determining the hydraulic tensioner plunger motion and the engine speed. The absolute position of the plunger was monitored with a non-contact laser transducer while an optical tachometer pickup was utilized to monitor the engine speed. Once the engine was brought up to the operating conditions, a slow engine speed sweep was performed and the plunger position was recorded at various RPMs.

The simulation data was obtained from an ADAMS model with a configuration to match the V-6 engine. It included two cam sprockets, a crank sprocket, a hydraulic tensioner, a chain guide, and a chain arm (Figure 1). The model also included the effects of the engine cam torques.

Figure 8 is a comparison plot of the actual and ADAMS simulation plunger displacements versus engine speed. The two overall responses are very similar, and both display resonance behavior at approximately 5000 RPM. The actual and simulation results were in good agreement; thus confirming the validity of the ADAMS model.

CONCLUSIONS

Running dynamic analysis on the aforementioned model provides detailed information concerning:

- Tensioner position
- Hydraulic pressure
- Leakage flow rate
- Chain tension
- Impact forces
- Natural frequencies

Since the results from the ADAMS simulations are in close agreement with actual experimentally obtained data, the model can be effectively used to study the engine performance. In addition the model is easily modified to investigate different design parameters and determine an optimal configuration.

ACKNOWLEDGMENT

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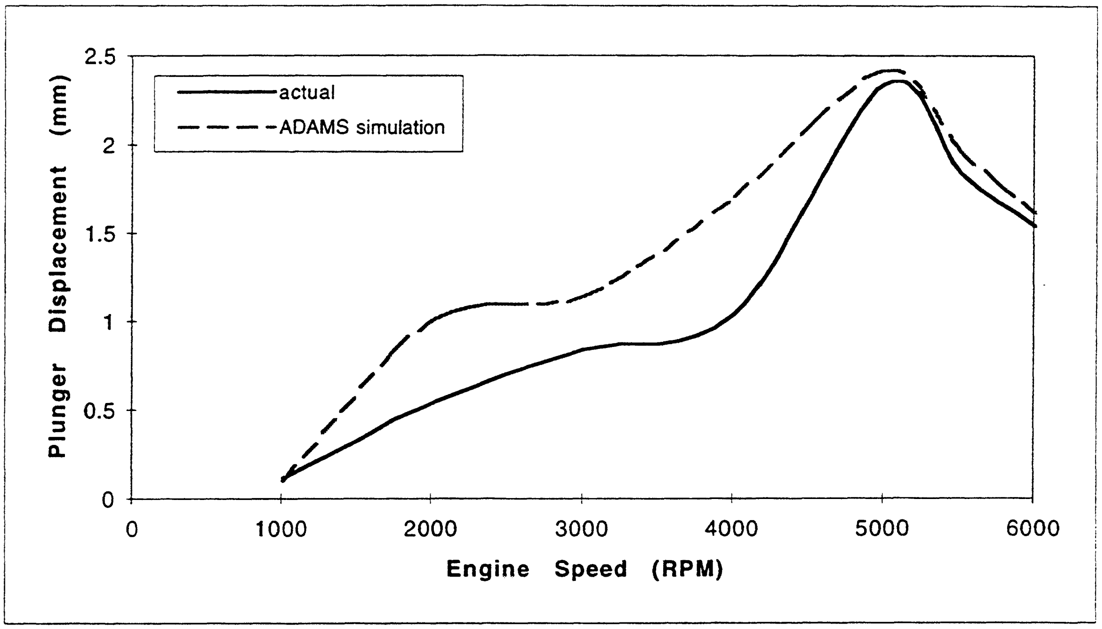

Figure 8: Plunger displacement versus engine speed

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