

Modeling and Simulation of Torsional Vibration of the Compliant Sprocket in Balance Chain Drive Systems

H.Y. Isaac Du, Xiangming Fang and Jia-Shiun Chen
General Motors Corporation

Keith D. Moss and William C. Prescott
LMS International

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ABSTRACT

The work presented in this paper outlines the development of a simulation model to aid in the design and development of a compliant sprocket for balancer drives. A design with dual-mass flywheel and a crank-mounted compliant chain sprocket greatly reduces interior noise levels due to chain meshing. However, experimental observations showed the compliant sprocket can enter into resonance and generate excessive vibration energy during startup. Special features are incorporated into the compliant sprocket design to absorb and dissipate this energy. Additional damper spring rate, high hysteresis and large motion angle that overlap the driving range may solve the problem during engine start-up period.

This work develops a simulation model to help interpret the measured data and rank the effectiveness of the design alternatives. A Multibody dynamics system (MBS) model of the balancer chain drive has been developed, validated, and used to investigate the chain noise. This model includes a cranktrain system coupled to a full balancer drive system that includes the chain drive, balance shafts, and water pump. The nonlinear stiffness and damping effects of the baseline compliant crank chain sprocket and its design iterations are added to the model to understand the basic characteristics and find an optimal setup for the engine start-up period. New MBS solver methodologies were implemented and the subsequent productivity increases are quantified. For subjective noise assessment, simulations of a solid sprocket versus compliant sprocket balancer drive were carried out and compared for certain operating engine speed.

INTRODUCTION

Chain driven balance shaft assemblies are commonly used to counteract the natural imbalance inherent to various engine architectures. This particular application focuses on a 2.0L Inline 4 cylinder turbo SIDI engine, with dual counter-rotating balance shafts driven by a chain that also drives the water pump. The overall layout is shown in Figure 1. One challenge for balance chain design is to address chain whine noise, analyzed on this same system in [1], where it was shown how a compliant sprocket design (CSD) was shown to be effective at reducing balance chain whine noise when a dual mass flywheel (DMF) was also employed. Despite this, the use of a Compliant Sprocket introduces a different noise source that occurs during engine start-up condition.

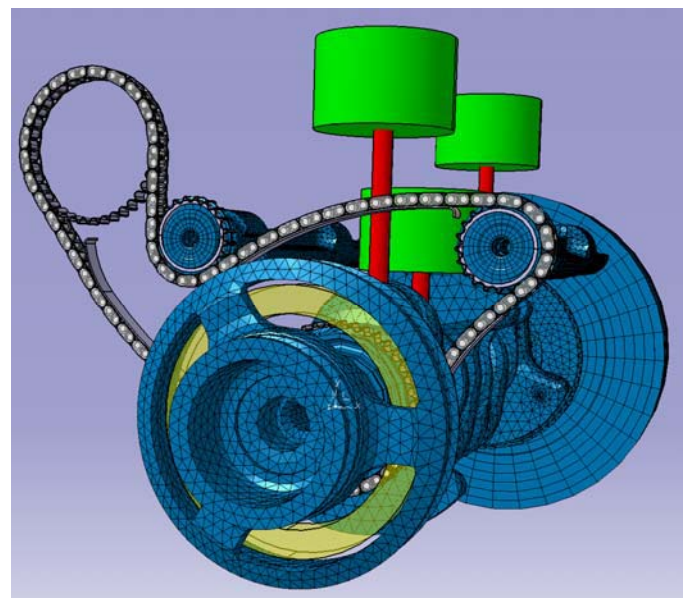


Figure 1. Crank train & Balance shaft layout

The engine starting process begins with a starter motor applying torque to the crankshaft. After the crankshaft breaks free from rest, the starter motor torque drives the crankshaft against resistive frictional and pumping torques until it reaches the speed that combustion can successfully initiate. At this point the starter motor torque is removed and the combustion loads drive the engine up to idle speed.

There have been various publications on engine startup simulation, which focus on challenges faced by Diesel engines in cold temperature conditions, which introduce additional resistive torques to overcome. These challenges can also be compounded by Integrated Starter Generators with lower torque ratios [5, 6]. This paper does not look at engine startup simulation from this perspective, but instead on the effect that start-up transients have on the balance chain system. Specifically, the loads during start-up are strong enough that a compliant sprocket design can wind up to its maximum limit of angular deflection, resulting in audible noise from impact against its dead-stops.

In this paper, the first section will discuss the system layout and summarize new MBS solver enhancements relevant for modeling chain systems with LMS Virtual.Lab Motion. A series of design variants will then be described and results explored to assess their ability to affect the start-up impact noise.

MULTI-BODY DYNAMICS MODEL

The MBS model contains the entire balancer drive system coupled with a crank train system as shown in Figure 2. New solver procedures, model construction techniques, and loading functions will be discussed.

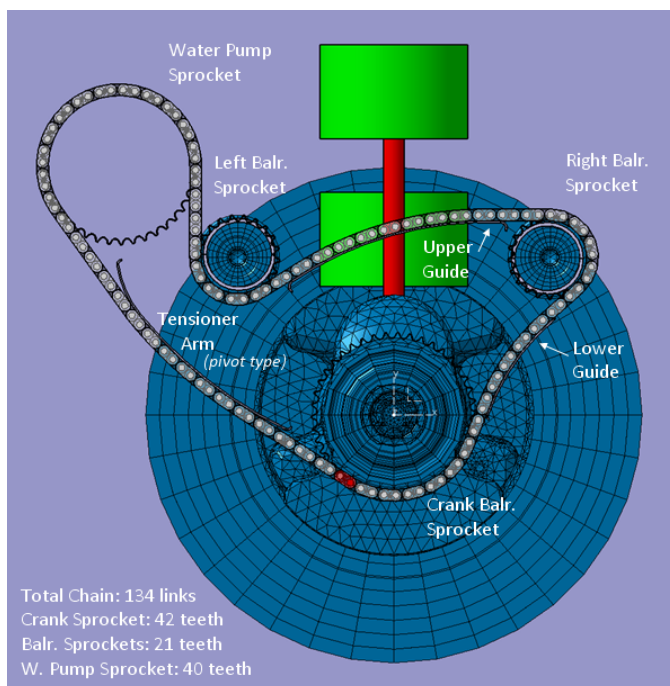


Figure 2. Model Layout

CHAIN MODELING

To efficiently construct the Virtual.Lab Motion model, a special-purpose pre-processor called Powertrain Dynamic Simulator (PDS) is used. A looping algorithm automatically generates all the chain link bodies to a precise location, sets the initial position of the tensioning device for proper initial static tension, and prepares initial velocity terms for all links to minimize solution transients regardless of the rpm the simulation is started from.

Each chain link pair is kinematically constrained to its neighbor using a compound Revolute-Translational joint, which allows one rotational DOF and one translational DOF. A bushing element is used to apply a rotational damping term to the free rotational DOF, whereas a TSDA element (Translational Spring Damper Actuator) is used to represent the axial stretching stiffness and damping terms along the translational DOF. An alternative topology could have been used if 3d misalignment effects were of interest, however in this case it was a suitable assumption to constrain the chain links to a plane.

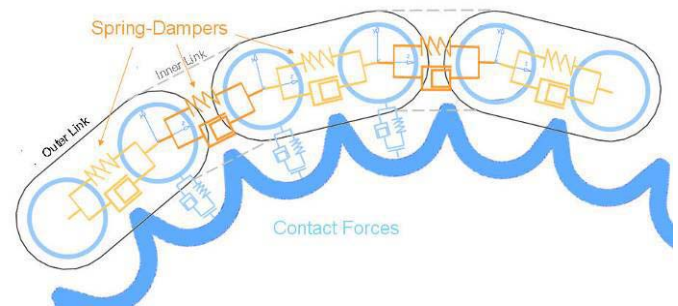


Figure 3. Graphic Showing the Chain Link Constraint Topology

Aside from the above-described constraints which control the relative DOF between neighboring chain link bodies, all the remaining interactions are governed strictly by contact elements. Individual contact profiles are generated to handle the intermittent impacts between each roller pin and each of the four ISO 606 sprockets, as well as for the contact interactions between the outer side of each side link-plate and each of the three guide surfaces. Additional contact elements are generated between each link side-plate and the outer diameters of each of the two metallic cushion rings on the balance shafts. Finally, contact elements are also setup between circular flanges on the balance shafts and inner diameters of the cushion rings. Metallic cushion rings are used to reduce the vibrations due to chordal meshing effect; a more thorough description can be found in [1]. In total there are 1208 contact elements in the model.

The tensioning device is based on a hydraulic check valve assembly with amplitude and frequency dependency. Virtual.Lab Motion has this capability but for this model a representative model with linear stiffness and damping proved adequate. Before

continuing with the description of the remainder of the model, we'll now enter the discussion of the MBS solution procedure, as it the solver enhancements described will relate directly to the chain portion of the model

MBS SOLUTION PROCEDURE

LMS Virtual.Lab is a general purpose multibody dynamics solver. The equations of motion are modeled in descriptor form using a maximal set of coordinates. The generalized coordinates $[x, y, z, e_0, e_1, e_2, e_3]$ consist of the Cartesian positions of the center of gravity and the quaternion of Euler parameters. Since a maximal set of coordinates is used, a library of constraints and joints are employed to reduce the number of degrees of freedom. For the model in Figure 1, flexible bodies are employed. A modal superposition method with floating reference frame approach is used to incorporate linear-elastic flexible motion. This method models the flexible body as a set of representative mode shapes originating either from Experimental Modal Analysis or from a Finite Element model. For this work, Finite Element models were used to generate a Craig-Bampton set of modes [3]. The final set of flexible equations of motion in explicit form is then

$$(1) \quad \begin{bmatrix} M_{rr} & M_{rm} & \Phi_{qr}^T \\ M_{mr} & M^m & \Phi_{qm} \\ \Phi_{qr} & \Phi_{qm} & 0 \end{bmatrix} \begin{bmatrix} \ddot{q}_r \\ \ddot{q}_m \\ \lambda \end{bmatrix} = \begin{bmatrix} Q^r \\ Q^m \\ \gamma \end{bmatrix}$$

Here q is the vector of generalized coordinates, where r refers to rigid body coordinates, and m denotes modal coordinates. Q^r is the vector of generalized forces acting upon rigid body coordinates, and Q^m are the modal forces. Φ_q represents the constraint Jacobian, which is the first partial derivative of the constraint equations with respect to the generalized coordinates.

The algebraic equations are coupled to the differential ones through a set of LaGrange Multipliers λ that relate to the constraint reaction forces between bodies. Eq. (1) is then solved for the accelerations of both the rigid bodies and modal participation factors. These accelerations are then integrated with an explicit Adams-Bashforth-Moulton based Predictor - Corrector scheme (PECE). Direct integration of the accelerations can result in drift from the constraint manifold; therefore a generalized coordinate partitioning method is used to project the solution on the constraints [14].

Inclusion of flexible bodies in a multibody model can often result in a set of numerically stiff equations which become better suited to an implicit integrator. An implicit integrator requires the equations of motion be formulated as an implicit set of equations instead of explicitly as in Eq. (1).

$$(2) \quad F(x) = \begin{cases} \dot{q} - v = 0 \\ \dot{q}_m - v_m = 0 \\ M\dot{v} + \Phi_{qr}^T \dot{\Psi} - Q^r = 0 \\ \Phi_q \dot{v} - \Phi_t = 0 \\ M_m \dot{v}_m + C v_m + K_r q_m + \Phi_{qm}^T \dot{\Psi} - Q^m = 0 \end{cases}$$

In Eq. (2) the matrix K represents the modal stiffness matrix and C the modal mass matrix. The acceleration constraint equation of motion has been replaced by the velocity equations. This set of equations forms an index one set of DAEs (Differential-Algebraic Equations) to which an implicit type integrator can be applied. For all the work in this paper, Virtual.Lab Motion's implicit Backwards Difference Formulation (BDF) integrator is used, which is a derivative of the DASSL integrator [15].

Solution of the Eq. (2) by an implicit integrator uses a Newton-Raphson set of iterations to find the set of generalized coordinates that satisfy $F(q) = 0$. A Newton solution requires the Jacobian, or the first partial derivative of F with respect to the state variables $x = [q_r, q_m, v_r, v_m, \Psi]$.

For chain models like this one, the solution time is commonly split nearly evenly between resolving the contact forces and the linear algebra required to formulate and factorize the Jacobian. Computation of the Jacobian can be an expensive operation; therefore an analytical or "hard-coded" Jacobian has been derived for most elements, which removes the need for finite-differencing across all the elements and all state variables in the model.

Aside from efficient generation of the Jacobian, another large computational burden lies in the linear algebra solution process where the (large) Jacobian is factorized. Its sparsity pattern is derived and a sparse matrix solver is employed. One way to reduce the factorization time of the solution is to condense the Jacobian matrix by reducing the number of generalized coordinates, and it is here where the constraint topologies inherent to chain models can be exploited. The chain portion of this model consists of rigid bodies connected by compound revolute-translational joints that allow one relative rotation and one translation between two links. A recursive or relative coordinate approach is employed to reduce the number of generalized coordinates introduced by each chain link, thus ultimately reducing the overall dimension of the Jacobian matrix. The sparsity pattern of the reduced matrix is still known, which still allows the application of a sparse matrix solver on a smaller matrix that is factorized more efficiently.

The topology of the non-chain portion of the model, without long closed kinematic loops, does not lend itself

to a recursive approach, so this portion of the model is left in the standard maximal coordinate formulation.

Finally, a revised contact search algorithm that exploits the axisymmetric nature of sprocket-based contact profiles was implemented to perform faster contact detection. All of these enhancements produce significant efficiencies for chain models compared to DADS, which was the legacy MBS solver used in the previous work [1].

For startup transient runs, the Virtual.Lab Motion solver including the additional solver enhancements introduced in this paper (Analytic Jacobian, Recursive Formulation, and new contact sorting algorithm) reduces the run times from 12.3 cpu hours to 3.3 cpu hours, a 73% improvement over DADS.

Further testing was done on models running at steady state. At 6000RPM WOT, the legacy DADS environment would solve at a rate of 52 minutes per crank rotation. The Virtual.Lab Motion Solver using the new solver enhancements solved at a rate of 14 minutes per rotation, a 72% improvement over DADS.

All tests were performed on a quad core Intel Core2Extreme QX6700 clocked to 2.89GHz running Windows XP x64.

CRANKTRAIN MODEL

The crank train system contains the crankshaft, torsional vibration damper, connecting rods, pistons, and a dual-mass flywheel (DMF). Flexibility was added for the crankshaft, the DMF, and each of the balance shafts. The modes were generated by Finite Element models as previously described. The overall approach effectively captures the gyroscopic effects inherent to high speed rotating shafts experiencing bending deformations [2].

For the DMF, only the secondary mass is brought in as a separate FE model; the primary mass is incorporated into the Crankshaft FE model. The crank & secondary DMF components are coupled through a 6 dof bushing element. The DMF is then connected with a series of constraints and force elements to represent the prop-shaft & dynamometer dynamic reactions using an approach that allows the model to be run at a controlled rpm while still allowing the crankshaft to deform realistically.

The modeling technique for the connecting rods and pistons are similar to the approach described in previous work [2], with the main journals modeled using an oil film algorithm based on the impedance method, a mathematically equivalent approach to the Mobility method [4].

LOADING

The inertial effects due to reciprocation of the piston & connecting rod assembly are naturally accounted for in

the MBS formulation. The cylinder combustion pressure loads shown in Figure 4 are applied to the model based on WOT condition.

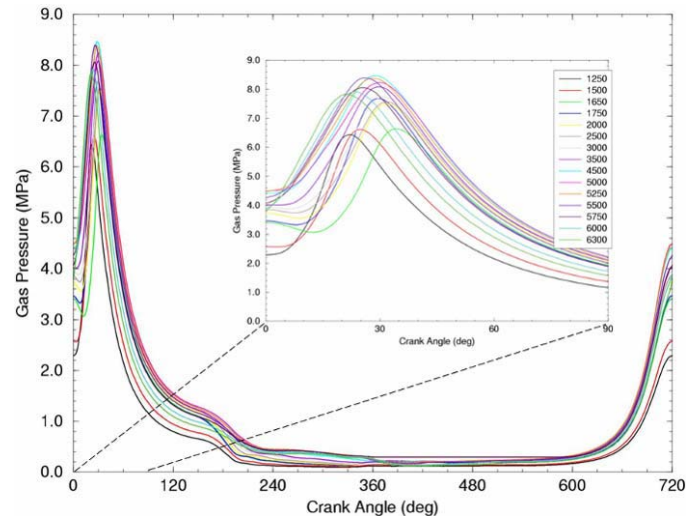


Figure 4. Combustion Data for Turbo SIDI Engine

The balance shafts introduce a small amount of drag torque on the system mostly due to rotational inertia. The water pump also introduces a resistive torque applied to its sprocket which transmits through the balancer chain to the crank balancer sprocket. The water pump torque is applied as a function of crank speed, from an experimentally generated look up table.

COMPLIANT SPROCKET

The compliant sprocket, shown in Figure 5, introduces a torsional DOF between the crank hub and the balance drive sprocket teeth. It incorporates a frictional surface and a pair of helical springs resting in a pair of curved pockets, as shown in Figure 5. Torsional oscillations from crankshaft resonances thus have their effect upon the chain dynamic tensions minimized, which ultimately results in a reduced amount of chain whine noise [1].

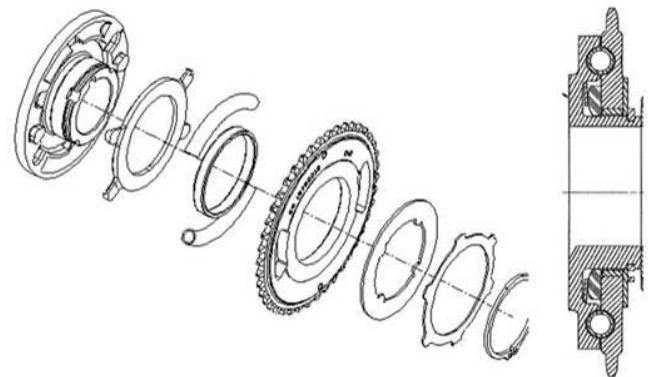


Figure 5. Compliant Sprocket Design

The torsional stiffness of the compliant sprocket is shown in Figure 6, where it can be seen that the total angular deflection is limited by a pair of asymmetric dead stops. The response of the compliant sprocket in torque and angle deflection along the driving direction is