Powertrain Analysis Applications Using

ADAMS/Engine *powered by* FEV

Part I: Valve Spring

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Abstract

ADAMS/Engine, powered by FEV, has been developed with the goal to more easily incorporate virtual prototyping methods into the engine-development process. It fills the gap between specialized tools commonly used in the industry and multi-purpose software such as ADAMS, by combining the engine-specific knowledge with the generic simulation environment in a process-tailored interface and architecture.
A collection of engine specific elements has been introduced, which take into consideration best modeling practices. These elements include, but are not limited to, chains, gears, belts, hydraulic lash adjusters, push rods, gas force, dynos, and valves. The list of elements is growing in accordance to the development plan of ADAMS/Engine. Depending on the stage in the engine-development process, these elements can be used in different levels of refinement. The ADAMS/Engine architecture makes those adjustments very easy, such that the level of refinement can be changed without recreating the model depending on data availability or in accordance with the task.
The valve spring is a particularly important element when considering the dynamics in a valve train. This paper reviews several existing modeling methods and introduces a new method to model valve spring. The model reflects the general aim for both high efficiency and high accuracy. This flexible-body-based model expands the number of existing valve-spring modeling methods within ADAMS/Engine, thus extending the list of possible levels of refinement.
The results with respect to accuracy and performance are in line with expectations.

Mechanical Dynamics

General Considerations for Valve Springs

Application

When designing a valve train, there is a general conflict between the demands for fast opening and closing of the valve at specific points, which means large valve accelerations, and the requirement not to exceed load limits, which means small accelerations. To optimize the design within those constraints, one has to consider several aspects of the valve train: weight of the components, kinematic characteristic, and dynamic characteristic of components, such as the hydraulic lash adjuster or the valve-spring. The kinematics is usually optimized in consideration of the acceleration requirements of the valve train by adding “ramps” to the lift curve.

Valve-springs have to provide sufficient force to balance the forces resulting from the masses and accelerations in the system. To refine the effects of the spring, the spring has to have a certain stiffness and frequency characteristic. The stiffness characteristic can be linear or nonlinear, depending on the shape and pitch of the spring. Therefore, by changing the spring wire path, one can design the stiffness characteristic to the desired specification. Of course, in addition to the wire path, the cross section and material properties of the wire have direct influence on the stiffness.

The valve-spring is usually the softest component with the lowest frequency in the valve train, such that its influence on the overall dynamics of the system is significant and can not be ignored in a numerical model. The dynamic response of the spring is substantially different from the static response due to the internal dynamics of the spring as shown in [5]. The internal oscillations in the spring are commonly referred to as surge modes.

The contact between the windings, are responsible for a large portion of the non-linearity of a spring, since it reduces the number of active coils during the compression of the spring. Another phenomena in conjunction with this contact is called coil clash, which happens when external excitation cause waves going through the spring with amplitudes high enough for adjacent windings to touch. This causes undesired force responses and has negative impact on the durability of the spring [6].

By combining two or more springs in a package, a designer has more parameters available to satisfy the overall design requirements stated above. In race applications, we find that coil-to-coil-interaction is utilized to avoid negative effects caused by the internal dynamics of each individual spring.

Dynamic Model

Depending on the task at hand, a numerical model of the valve spring should satisfy certain requirements balancing the need for high numerical performance with the level of model refinement. For instance, when investigating the overall mass/inertia effects of the valve train on the engine, a simple spring model including two masses, connected via a force with a function representing the static stiffness characteristic, is sufficient. On the other hand, when optimizing the performance of the valve train itself, a more detailed model is needed. Such a model should be able to capture the nonlinear characteristic in terms of stiffness and frequency. This means the model has to consider the internal dynamics. In some cases, when in-depth understanding of the spring itself is required, a model, which also considers the buckling and the coil interaction in great detail, is necessary. Models should be based on physical parameters, or be derived from test data.

Existing Methods to Model Valve Springs

Dual-Mass Spring Approach

The dual-mass spring approach represents the lowest level of refinement. For instance, it is appropriate to use this model, when the valve train is used as the excitation mechanism in a larger assembly that includes a timing mechanism.

To consider some of the mass effects on a larger system, the spring mass is distributed into two masses. The distribution ratio may be adjusted according to experience. The masses are connected via a force with a function representing the nominal (static) stiffness characteristic of the spring. Figure1 shows the schematic representation of this model in ADAMS/Engine.

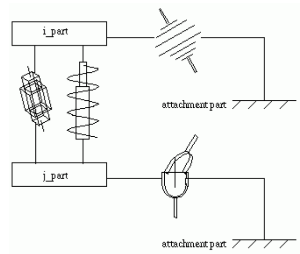

Figure 1

This model does not consider any internal dynamics of the spring and is therefore not suited for the dynamic investigation of a valve train.

Surge-Mode Approach

The surge-mode approach takes advantage of the fact that an elastic medium fastened at both ends can be described in terms of normal modes. For instance, the first longitudinal mode captures the in-phase motion of the spring mass. This mode has the lowest frequency and has direct impact on the overall performance of the valve train, as stated under the general considerations for the valve spring. The model only considers longitudinal effects.

The model, as described in [1], assumes uniform mass and stiffness distribution, such that the wave speed traveling through the spring is constant. With ϕ being the displacement of each element in the spring, and x being the displacement along the length of the spring, the displacement of each spring element is governed by the wave equation:

$$\frac{\delta^2\phi}{\delta t^2} = c^2(x)\frac{\delta^2\phi}{\delta x^2}$$

With the boundary conditions that one end of the spring remains at a fixed position, and that the mass and stiffness distribution is uniform, the force at the moving end can be obtained through a summation of elastic modes:

$$F = -K(S + h(t)) + \pi K \sum G_n(t)$$

Where S is the pre-compression length, h is the length of the spring, K is the stiffness associated with each mode and G represents the modes.

Usually, it is sufficient to consider the first five surge modes of a spring, which means that in the ADAMS implementation, 10 ODE's are required.

This model does not consider any non-linearity caused by the interaction between windings, thus coil clash is not included. The model is relatively accurate, as long as the spring is well within its operating range. This is very often not sufficient, since it is desired to optimize the valve train to its dynamic limits where some interaction between the windings may occur or in some cases may even be desired. Due to this limitation, this model has not been included into the ADAMS/Engine component library.

Multi-Mass-Spring Approach

The multi-mass-spring approach, which is used to obtain the forces under dynamic conditions, is based on the discretization of the spring. The model parameters are derived from physical properties, such as wire dimensions and material properties.

To capture the internal dynamic effect often referred to as spring surge, the valve spring is modeled via a series of masses coupled with spring dampers. All masses represent a segment of the spring wire. Every winding is usually split in 4 – 8 segments. The torsional stiffness of each wire segment is used to determine the equivalent translational spring stiffness, such that the stiffness may vary depending on the diameter or change in cross section. The damping represents the material damping.

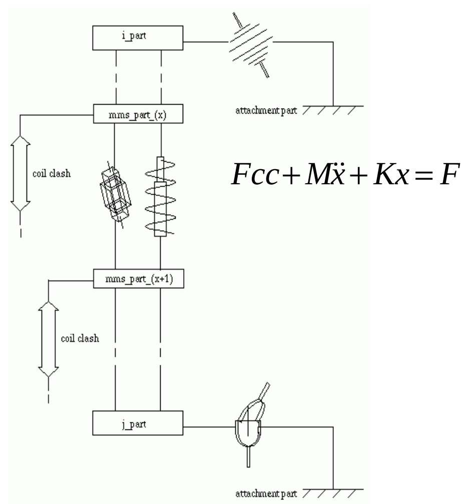

Figure 2

To include the nonlinear behavior of the spring caused by the changing number of active coils during a compression, it is necessary to consider the interaction between the coils. This is done via additional nonlinear contact forces, which are dependent on the approximation of the appropriate parts. Due to the presence of nonlinear contact forces, it is also possible to consider coil clash.

Figure 2 shows the schematic representation of the model, as implemented in ADAMS/Engine [3].

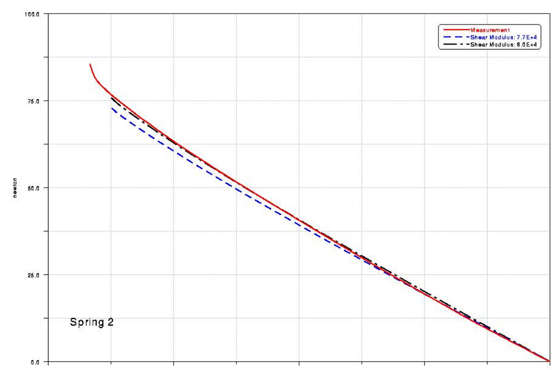

Figure 3

Figure 3 shows the correlation between test data and simulation, with respect to the stiffness of the spring. This model is also well suited to capture the change in frequency depending on the compression of the spring, as shown in Figure 4.

The high number of parts and forces, as well as the nonlinear characteristic of the contact force represent a significant solution effort. This model has been used in project work and has been validated as shown in [4].

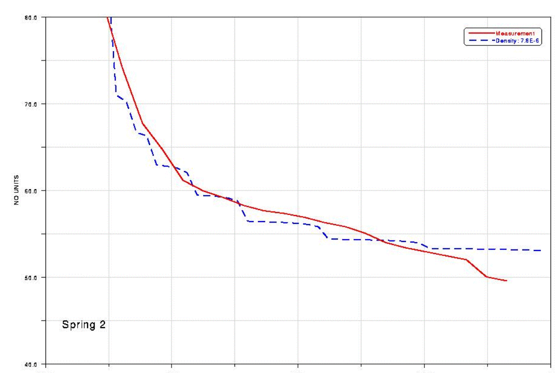

Figure 4

The New Flexible Body Based Approach

Overview

Nonlinear effects, such as coil-clash and surge, exclude the usage of a plain modal representation as described under the surge-mode approach. Nonetheless, the modal approach has some merits. The number of equations to be solved is minimal, thus the computational effort is small. The introduction of non-linearity into the model is required for this model to be valuable for valve-train simulations. The effect of the spring on the valve train is the main concern, therefore performance has a high priority.
On the other hand, there is the requirement for predicting internal static and dynamic loads in the spring to a great level of detail, to be able to make durability assessments and to optimize the spring design itself. In this case, model accuracy is of higher priority then performance.
This has led to the development of the two-stage flexible body based approach introduced in this paper. The first stage is referred to as the detailed model, and the second stage is referred to as the equivalent model.
The detailed model, which consists of three to five flexible bodies per winding, considers the geometric non-linearity, as well as the interaction between windings, thus governing the non-linearity in stiffness and frequency. The model also captures buckling and all other three-dimensional effects. The flexible bodies are auto-generated, as described in more detail below.
The equivalent model consists of one flexible body, which only represents the linear, longitudinal portion of the spring model. Forces between certain nodes are superimposed to introduce the non-linearity determined by the detailed model. To generate the equivalent model, the results of a quasi-static compression of the detailed spring are required.
The detailed model can be used for quasi-static compression tests in a virtual test rig or in dynamic mode in a single valve-train simulation. The equivalent model should be used for simulations of single or complete valve-trains.
The models are auto-generated, based on design parameters such as wire dimensions, wire centerline path, and material properties.

Flexible Body Modeling (detailed model)

The detailed flex spring uses flexible bodies for a very accurate spring model. The flexibility enables the internal surge of the spring, and the coil interaction gives the non-linearity of the total spring. As for the multi-mass spring, the flex spring is modeled by means of physical spring properties. The spring wire is split up into several flexible bodies (3 – 5 per winding), so that the nonlinear deformation of the spring wire can be captured. The flexible bodies are created using the AFI-file interface in ADAMS.

Each cross section of the spring has one center node and 12 circumference nodes. The elements used for the wire are linear (8 node) brick elements with one collapsed side. Additionally, there is a spider web element (RBE2) at the ends of each spring section, which is used to connect the sections. Because the rotation of the center node is needed, and the brick elements

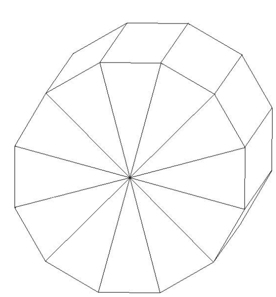

Figure 5: FE-model

cannot handle this, a spider web element (RBE3) has been added to all internal cross sections. Figure 6 shows one segment of a wire section.

Each section of the spring wire consists of six center nodes (five segments) with circumference nodes. The modes calculated for a wire section are six Craig-Bampton modes for the center nodes at each end, and one (vertical) mode for the center nodes in between.

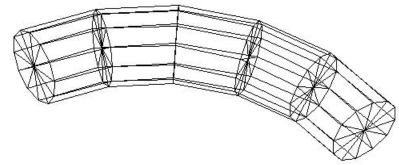

Figure 6: Spring section

The flexible body at each end of the spring has an active part and an inactive part. The active part consists of five segments with the same topology as the wire section parts. The inactive part has 15 segments where the wire is, causing the center node on the centerline, to which the RBE3 elements are attached, to be no longer coincident with the center node of the brick elements. The center node of the RBE3 elements is connected to the end center node of the spring by means of RBE2 elements. Six Craig-Bampton modes are calculated for the center node at the active end, and for the main center node (center of the big spider web). One vertical mode is calculated for the remaining four center nodes of the active part.

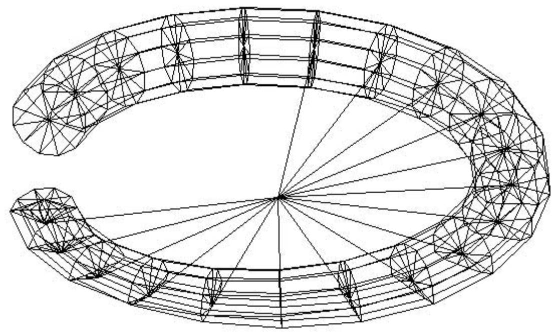

Figure 7: End section

In the detailed spring all flexible bodies are connected with fixed joints. The ends of the spring are also attached to the I-part and the J-preload part with fixed joints. Because the spring is assembled in free-length condition, two rigid parts are found in the J-end of the spring – the J-part and the J-preload part. A motion between these parts provides the preload of the spring during the first steps of an analysis.

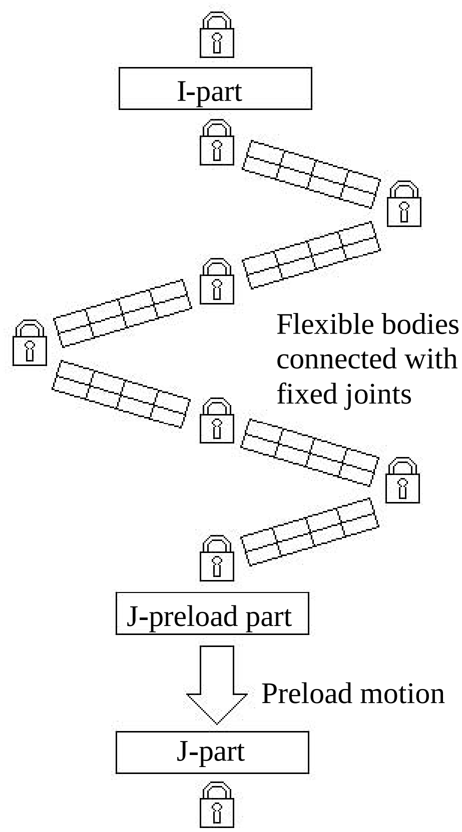

Figure 8: Schematic ADAMS model

Detailed Spring Analysis

A request monitors the overall force in the spring, as well as the displacement of every fixed joint. For each time step this information is written to an analysis output file for later usage (in the equivalent spring model). This request is only active when executing a component analysis of the spring, and not when using the spring in a valve train.

ADAMS/Solver Implementation

The interaction between the windings of the valve spring is modeled via GFORCE's applied at the nodes on the centerline. The reaction force is on ground, such that special attention has to be paid to the balancing of the forces. The contact forces are dependent on the shape of the spring wire, thus the node locations are monitored in the GFOSUB that is used to compute the forces.

This means all GFORCE's have the same dependencies on states in the system. The total number of dependencies is determined by the total number of modes in the spring model and by the directions that are considered. The result is a large number of calls to the GFOSUB routine. To conserve time, the number of force evaluations has to be minimized. A method, which is described in [2] is taking advantage of the fact that the set of dependencies is identical for all GFORCE's. In that method, ADAMS computes all forces during the path for the first GFORCE through the GFOSUB routine and stores them under consideration of an index. This index is based on the GFORCE ID, as well as the partial call.

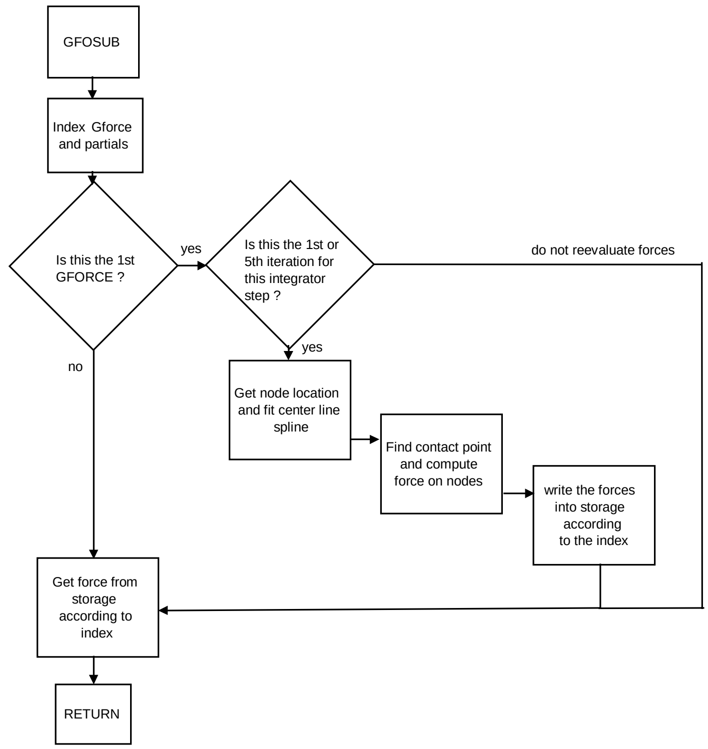

Figure 9: GFOSUB flow chart

For all subsequent calls to compute the other GFORCE's, the force is retrieved directly from the storage, thus significantly reducing the amount of computations. Additionally, computing the forces only during the first and fifth iteration of each integrator step saves time.

The implemented approach is assuming that the GFORCE's are grouped together in the adm file. Special consideration has to be given to the indexing. Also, the fact that ADAMS only evaluates the forces for partials, if the actual evaluation was non-zero, should be carefully considered.

The Equivalent Spring

The equivalent spring consists of one single flexible body with as many elements as there are flexible bodies in the detailed spring. Each element's stiffness is reflecting the stiffness of the flexible body it "replaces". The non-linearity that comes from the coil interaction in the detailed spring is then added by superimposing a nonlinear force on each element. The equivalent spring is constrained at the ends by a hooke joint and an at-point joint primitive.

The basis for the creation of the equivalent spring is the analysis output file written when running a component analysis on the detailed flex spring. For each time step of the analysis, the forces and torques acting on the I-reference of the spring, as well as the distance between the I- and J-reference, are written. Additionally, the current length in the axial direction of the spring is stored for each section.

```
$--- Results from detailed Model ---
$-----------------------------------
STEP_0
[STEP_0]
(FORCE_0)
{DZ  FX  FY  FZ  TX  TY  TZ}
 40.36  0.0  0.0  0.0  0.0  0.0  0.0
(DISP_0)
 SECTION_1  =          2.78697
 SECTION_2  =          1.85370
 SECTION_3  =          2.40642
 SECTION_4  =          2.40927
 SECTION_5  =          2.39948
 SECTION_6  =          2.40927
 SECTION_7  =          2.40492
 SECTION_8  =          2.40383
...
```

Figure 10: Detailed model output file

Before using this information for the equivalent spring, the data is reformatted such that a force-deformation curve is generated for each section of the spring (solid curve in Figure 11).

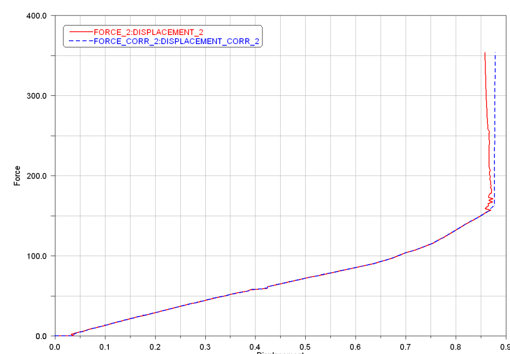

Figure 11: Force deflection of a particular joint

To be able to use this curve in a nonlinear force, the data has to be modified slightly to get a unique force-deformation relationship (dotted curve). This introduces an error in the spring characteristic as the spring approaches block length. Further investigation on how to correct this in the model is required.

The curve is now split into a linear and a nonlinear part. The linear stiffness is determined through a user-entered linearization force. Any difference to this linear force-curve is put into the nonlinear curve. The curve in Figure 12 shows the nonlinear part of the curve in Figure 11, when the linearization force equals 120 N.

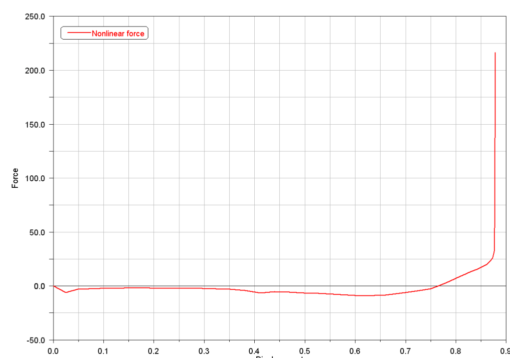

Figure 12: Nonlinear force deflection law

These force-deflection curves are imported into the spring as splines and are used in the superimposed nonlinear forces.

The flexible body for the equivalent spring is based on the stiffness derived from the linear part of the force-deformation curves. The elements used are linear beam elements. The stiffness in the longitudinal direction for each element is described as follows:

$$\overline{\overline{K}} = \begin{bmatrix} k & -k \\ -k & k \end{bmatrix}$$

With n nodes and (n-1) elements, the complete stiffness matrix results in:

$$\overline{\overline{K}}_{spring} = \begin{bmatrix} k_1 & -k_1 & 0 & & \cdots & 0 \\ -k_1 & k_1 + k_2 & -k_2 & & & \vdots \\ 0 & -k_2 & k_2 + k_3 & & & \\ & 0 & & \ddots & -k_{n-3} & 0 \\ \vdots & & & -k_{n-3} & k_{n-2} + k_{n-1} & -k_{n-2} \\ 0 & \cdots & & 0 & -k_{n-2} & k_{n-1} \end{bmatrix}$$

The mass of the spring is collected into n nodal masses:

$$\overline{\overline{M}}_{spring} = \begin{bmatrix} m_1 & 0 & \cdots & 0 \\ 0 & m_2 & & \vdots \\ \vdots & & \ddots & 0 \\ 0 & \cdots & 0 & m_n \end{bmatrix}$$

This system can now be put into the eigenvalue problem:

$$\left(\overline{\overline{K}}_{spring} - \lambda \overline{\overline{M}}_{spring}\right)\{D\} = \{0\}$$

where λ is the eigenvalues of the system, and {D} contains the eigen modes.

Solving this system gives the free-free modes for the equivalent spring in the longitudinal direction. The modes in all other directions are set to zero. This results in a one-dimensional flexible body with n nodes and (n-1) free-free modes (and one additional rigid body mode).

Calculating the wire path the same way as for the detailed spring creates the graphics of the equivalent spring. Massless nodes are then created on the surface of the spring wire. The longitudinal mode of the nodes is calculated by linear scaling of the modes for the main center nodes. For a given wire node, the main center node right above and below is used for the mode scaling.

The resulting topology and graphics of the equivalent spring is shown below:

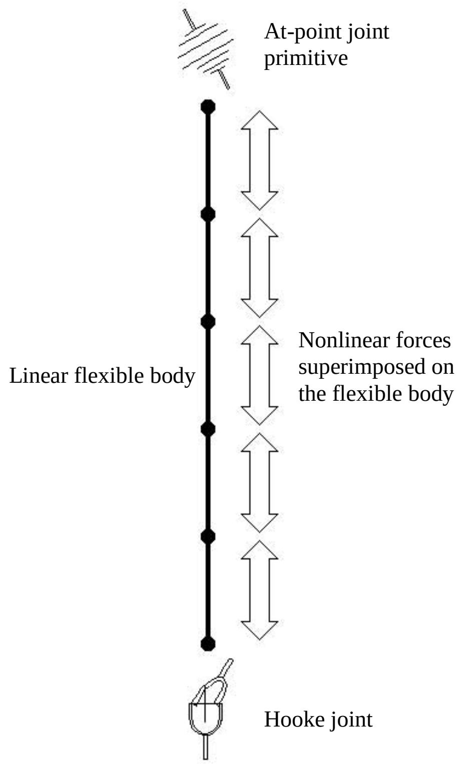

Figure 13: Schematic ADAMS model

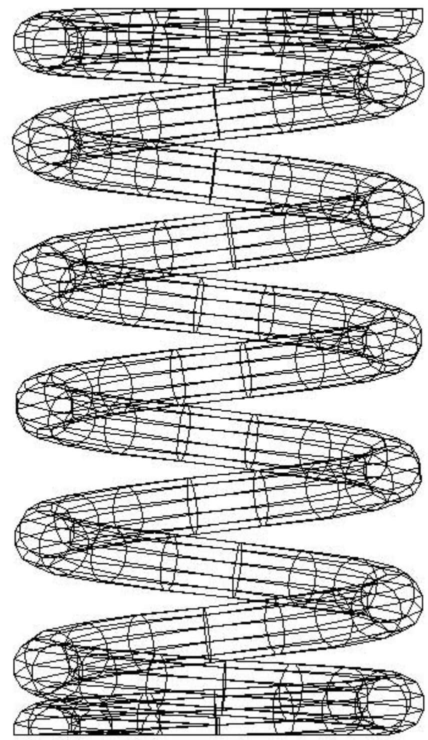

Figure 14: Equivalent model flexible body

Results

The compression of the detailed spring model took approximately 20 minutes on a 500 MHz NT. Figure 15 shows example results of a cylindrical spring with variable pitch. The results include but are not limited to the forces, torques, and displacement at the end of the spring. The solid curve represents the force versus displacement, the dotted curve represents the torque about the longitudinal axis of the spring versus displacement.

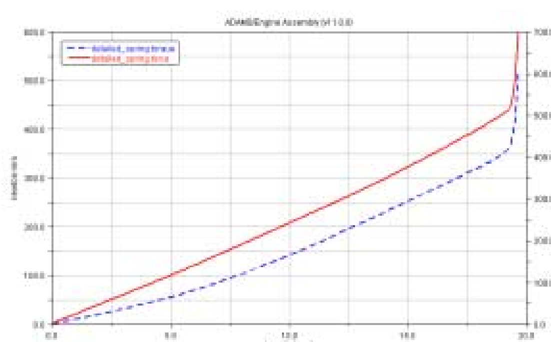

Figure 15: Force and Torque vs. Displacement

The same analysis using the equivalent spring took about 20 seconds on the same machine. Figure 16 shows force versus displacement in the equivalent spring.

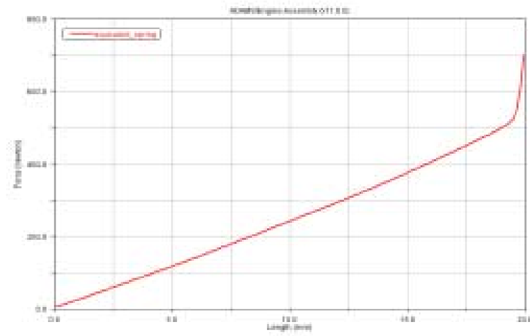

Figure 16: Force vs. Displacement

When comparing the results of the compression analysis of the detailed and the equivalent spring model it becomes apparent, how good the quasi-static characteristics of both models match. Figure 17 shows the results of both models, the solid curve represents the detailed model and the dotted curve represents the equivalent model. The difference at block length is due to the force-law correction described under *The Equivalent Spring*.

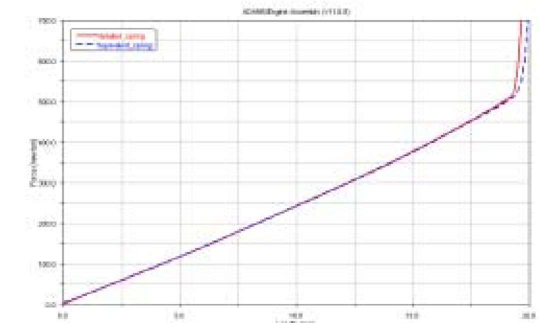

Figure 17: Force vs. Displacement

A single-valve-train assembly as shown in Figure 19 has been used to compare the models under dynamic conditions. In both cases the model was exercised at 3000 engine rpm. The plot in Figure 18 shows the acceleration of the valve versus time. The solid curve represents the detailed model and the dotted curve represents the equivalent model.

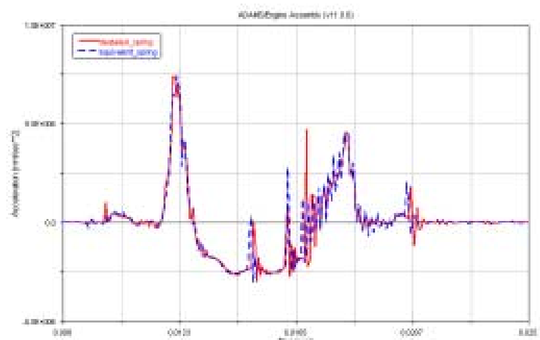

Figure 18: Valve Acceleration vs. Time

Summary

In this paper a new method to analyze valve springs has been introduced. The detailed model captures internal dynamic effects such as surge and coil clash as well as all three dimensional effects such as buckling and shear at the spring ends very well. This method is well suited for spring design. The equivalent model matches the results of the detailed model with respect to stiffness. It is a very efficient model, which captures all longitudinal dynamic effects of the spring. The equivalent model is therefor well suited for valve train analysis. The only drawback of this method is that results of a detailed spring analysis are a prerequisite. The new flexible-body based approach was introduced into the ADAMS/Engine environment to complete the selection of spring models available to the user.

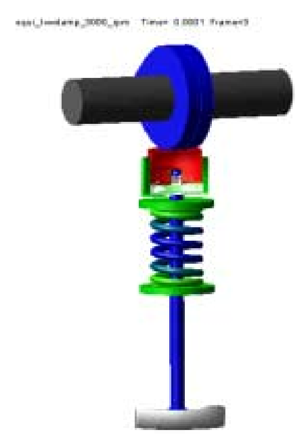

Figure 19

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