

An Efficient Spring Model Based on a Curved Beam with Non-Smooth Contact Mechanics for Valve Train Simulations

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ABSTRACT

The valve train plays a huge role in the performance of internal combustion engines by controlling the combustion process and is therefore one starting point to increase the efficiency of combustion engines. Considering the dynamics, the valve spring is the component with the lowest natural frequency in the motor and therefore plays a crucial role in the overall dynamics of the valve train. The spring force must be high enough to close the valve reliably and prevent the valves from bouncing of the seating due to surge modes after they have closed. Conversely, the spring force affect the friction level in the engine and therefore fuel consumption. For this reason the spring forces should be kept as low as possible.

Modelling valve springs it has to be taken into account, that the dynamic response of the spring is substantially different from the static response. The internal dynamics of the spring like the moving masses of the coils, contacts between the windings leading to a non-linear spring characteristic and coil clash must be considered. Consequently, for the design and optimization of modern valve trains sophisticated valve spring models considering the internal dynamics are needed.

In this paper a new valve spring model for efficient multi-body simulations of vale trains is presented. The approach is to approximate the spring wire as a curved beam and derive the equations of motion yielding hyperbolic partial differential equations. These equations are discretized using finite element methods and integrated in a multi-body simulation approach. The interactions between the windings are included using the Augmented Lagrangian Method and non-smooth contact mechanics.

Compared to a multi-mass-model approach the presented method allows reducing the degrees of freedom and the stiffness of the resulting differential equations. Therefore the computational effort can be minimized at slightly increased accuracy, yielding an appropriate model for multi-body simulation. Using a test rig, the model is validated in the frequency and time domain with experimental data. Furthermore, the presented new model is compared to existing models like multi-mass-models.

INTRODUCTION

Over the past years there have been increasing efforts to reduce CO_2 emission. Especially the car manufacturers are demanded to reduce emissions. On the one hand, there is a public demand for cars with lower specific fuel consumption and one the other hand, the willingness of consumers to pay much more for this reason is not very high. Therefore improvements are needed in all domains of the cars.

A greatly significant part of the car that affects the fuel consumption is the engine itself. Beside other devices the valve train plays an important role concerning performance and efficiency questions. It is responsible for opening and closing the valves that let fresh air fuel mixture into the cylinder and exhaust gases out. The valves are kept closed by a coil spring that pushes the valve head into its seat. The coil spring is generally the most flexible part in the entire valve train. The other parts can nearly be considered rigid in comparison. Thus, the valve-spring is the component with the lowest elastic natural frequency and dominant in the behavior of the valve train.

The tendency of the last years has been to develop valve trains that allow changing parameters like valve lift or valve

timing during operation. This is one of the reasons why the valve spring needs to be simulated in an increasingly detailed way. An efficient spring model would allow trying various parameter configurations without a lot of time consuming experiments. In the majority of cases, valve springs of current engines have a progressive behavior. This behavior is mainly caused by a non-constant pitch between adjacent coils. The coils with a smaller pitch come earlier into contact as the coils with a bigger pitch, which results in a stiffer behavior of the spring the more the spring is compressed.

An operating point near the eigenfrequency can result in excessive vibrations and forces in the spring. This characteristic is called *surging* and has a negative effect on the durability of the spring and causes the valves to bounce of the seating after they have closed. Moreover, if the forces are powerful enough adjacent coils may come into interaction which is called *coil clash*.

In the present development mainly three dynamic spring-models are used: **Modal-Model**, **Multi-Mass-Model** and **Multi-Beam-Model**.

The motion of a flexible body can be described by superimposing its different eigenmodes [1]. The eigenmodes are weighted with time-dependent functions. In most cases it is sufficient to consider the first few eigenmodes, because these modes represent the main part of the vibrational energy. The key benefit of this spring-model is that the number of degrees of freedom is small and therefore the computational time is relatively short. But the serious disadvantage is that no contact modeling is possible. It is for this reason that this model doesn't have enough potential for detailed simulations.

The Multi-Mass-Model is used for example in FEV Virtual Engine (former MSC ADAMS/Engine) [2] and is presented by Engelhardt [3].

The valve-spring is modeled by a series of masses coupled with spring-damper-elements. More precisely each coil is split into 4 to 8 masses. The overall stiffness of the spring can be calculated from a static experiment. The individual stiffness of the spring i can be calculated from the overall stiffness according to Eq. (1):

$$\frac{1}{k_{sum}} = \sum_{i=1}^n \frac{1}{k_i} \quad (1)$$

The advantages of the Multi-Mass-Model are that a progressive behavior can be modeled and that coil interactions are possible. But on the other hand, the model has two meaningful drawbacks: The amount of degrees of freedom is high and the equations of motion will mostly

become stiffer, which may result in a higher computational effort.

In the Multi-Beam-Model straight beams are used to approximate the spring wire. As straight beams are applied, a large amount of beams are needed to approximate the curvature nature of the wire geometry. Tichánek et. al. [5] uses 24 elements per coil, resulting in significant higher computational effort like with the Multi-Mass-Model.

Because of these facts, the aim of the spring-model presented in this paper is to present a model that has few degrees of freedom, produces less stiff equations of motion and allows contact-modeling. The model is implemented in the multibody simulation software MBSim (<http://mbsim.berlios.de>), which is developed at the Institute of Applied Mechanics, Technical University of Munich.

The first section of this work gives a short outline about multibody-dynamics. The following part describes the spring-model by a partial differential equation which is discretized with the Finite-Element-Method. The continuous model is verified with an ANSYS-model and the discretized model is verified with the continuous model.

Furthermore, the test rig for the experimental validation of the simulation model is described. The spring-model is validated in the static and dynamic case. To round the paper up, the main advantages and methods are recapitulated.

NON-SMOOTH MULTI-BODY DYNAMICS

A short overview is given for the equations describing the general dynamic of multibody-systems. More information can be found in [6]. An emphasis is laid upon how to deal with contacts, e.g. between clashing coils.

There are mainly two possibilities for modeling contacts and connections: smooth and non-smooth mechanics. In most cases smooth mechanics are used, which means that spring-damper-elements are used to model contacts and connections.

In contrast to smooth contacts and connections, non-smooth or rigid contacts and connections can be seen as contacts with infinite stiffness of the spring. It is for this reason that rigid links result in jumps in the velocities of the system. Therefore, special integration schemes are used for such systems. For solving the non-smooth equations of motion a lot of different approaches like time-stepping-methods [4] or event-driven-methods exist.

EQUATIONS OF MOTION

The basic equations of motion for a smooth multi-body-system can be described by Eq. (2):

$$\mathbf{M}(\mathbf{q})\ddot{\mathbf{q}} = \mathbf{h}(\dot{\mathbf{q}}, \mathbf{q}, t) \quad (2)$$

The vector $\mathbf{h}$ contains all smooth external and gyroscopical forces depending on the system-states $\mathbf{q}$, system velocities $\dot{\mathbf{q}}$ and the time t . The mass matrix $\mathbf{M}$ only depends on the system-states $\mathbf{q}$. The implementation of the valve spring model allows using smooth and non-smooth contact mechanics. Therefore, the formulation of the basic equations of motion needs to be enhanced with the equation for non-smooth contact mechanics:

$$\mathbf{M}(\mathbf{q})\ddot{\mathbf{q}} = \mathbf{h}(\dot{\mathbf{q}}, \mathbf{q}, t) + \mathbf{W}\lambda \quad (3)$$

The force-direction-matrix $\mathbf{W} = [\mathbf{W}_N, \mathbf{W}_T]$ is a transformation from the space of constraints to the configuration space and can be interpreted as the normal and tangential directions of the contact forces in the configuration space. The magnitudes of the forces can be written in the vector $\lambda^T = [\lambda_N^T, \lambda_T^T]$.

FORCE LAWS

Smooth Force-Laws

Smooth contacts and connections are realized by applying a spring-damper-element between two bodies. The main advantage of this type of constraint is that the corresponding forces can be calculated directly from the system states. Therefore, standard ODE-solvers can be used to integrate the system.

But on the other hand, significant drawbacks have to be mentioned: The stiffness of the spring is finite and it is for this reason that two linked bodies can penetrate. To reduce penetration, the stiffness can be increased which results in stiffer equations of motion. Moreover, this may lead to longer integration times. Finally, it is difficult to determine adequate contact stiffness and contact damping.

Non-Smooth Force-Laws

In the previous section smooth force laws were described which are based on spring-damper-elements with a finite stiffness.

Because of the above-mentioned disadvantages, a second approach is taken into account. This is called rigid approach. The stiffness in the contact zone is assumed to be infinite resulting in a unilateral constraint.

This means that an explicit function between the system states and the constraint forces can not be stated any more. The contact reactions are described by set valued force laws like the *Signorini-Fichera*-condition (Eq. 4).

$$g_N \geq 0, \lambda_N \geq 0, g_N \lambda_N = 0 \quad (4)$$

In the case of a rigid contact only two system states can be distinguished. Either the bodies are separated leading to $\lambda = 0$ or the bodies come into contact which results in $\lambda > 0$. The exact value of λ must be calculated with special numerical methods. More information can be obtained in *Förg* [4] and *Glocker* [7].

Figure (1) shows a flexible and a rigid contact (unilateral constraint) in comparison neglecting damping. In the case of the flexible contact a spring-damper-element is activated as soon as the contact-gap g_N gets smaller than zero. The more negative the gap, the higher the contact force. The contact force consists of a part resulting from the spring and a part resulting from the damper. Equation (5) describes the contact situation mathematically with contact stiffness c and damping d . To avoid discontinuous damping forces the coefficient d is a function of the gap distance g_N . For $g_N = 0$ the coefficient is equal to zero rising to a constant end value with increasing penetration.

$$\lambda_N = \begin{cases} -c \cdot g_N - d(g_N) \cdot \dot{g}_N & \text{if } g_i < 0 \\ 0 & \text{else} \end{cases} \quad (5)$$

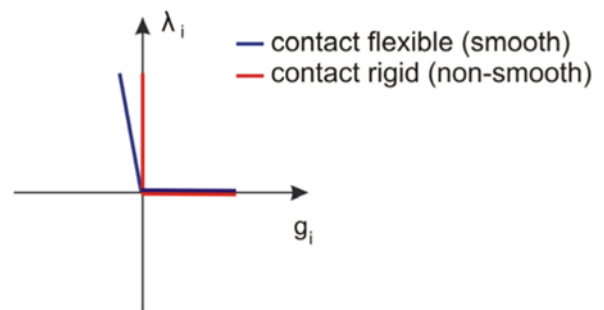


Figure 1. Comparison between flexible and rigid unilateral contact

In Figure 2 the situation for a bilateral constraint is illustrated. Using a rigid connection penetration of the involved bodies is not allowed.

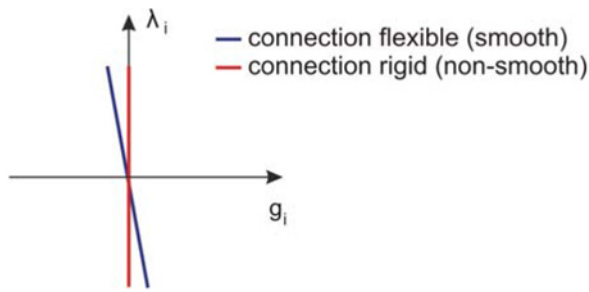


Figure 2. Comparison between flexible and rigid bilateral connections

COIL SPRING MODEL

In the following section the spring model is described. At first, the necessary coordinate systems are introduced. One step further the governing equations are mentioned. Finally, the constitutive equations are outlined leading to the full and reduced equations of motion.

COORDINATE SYSTEMS

For the description of the spring model two coordinate systems are introduced (Figure 3).

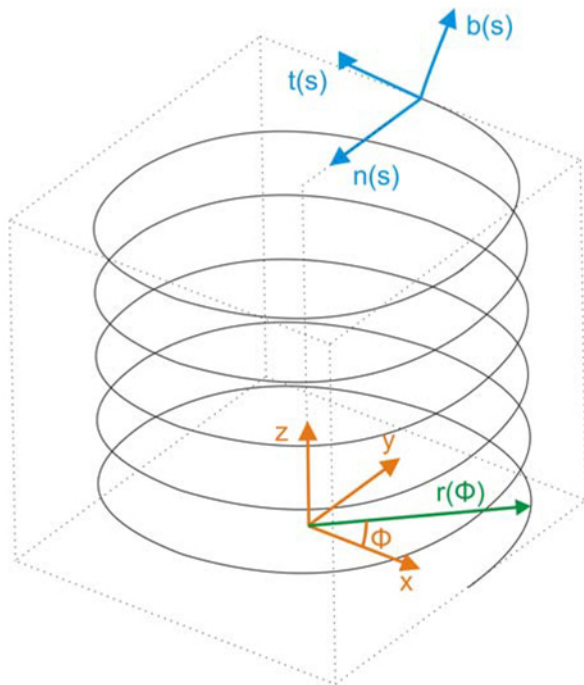


Figure 3. Spring with cartesian and Frenet-Serret-Frame coordinate system

Consider the coil spring wound about a central axis as in Figure (3). A cartesian coordinate system (x, y, z) is introduced. The cartesian coordinate system is placed in such a way that the z -axis coincides with the central axis of the

coil spring. One end of the spring lies in the x, y -plane of the coordinate-system at $z = 0$.

The second coordinate system is called *Frenet-Serret-Frame* [8]. It consists of the tangent, normal and binormal vector and follows the spring wire.

The spring-wire can be parameterized by two different variables: the spring wire length s and the angle ϕ .

GOVERNING EQUATIONS

In order to describe its kinematics, the spring wire is approximated as a curved beam. In accordance with regular beam theory the motion of an individual cross section is parameterized. Based on this, the displacement and strain of an arbitrary point of the wire can be determined.

In the following $r(s)$ denotes the undeformed state, u the displacement of the center line and θ the rotation of the cross section.

An arbitrary point Q can be referenced from its corresponding point on the centre line P like in Fig. 4.

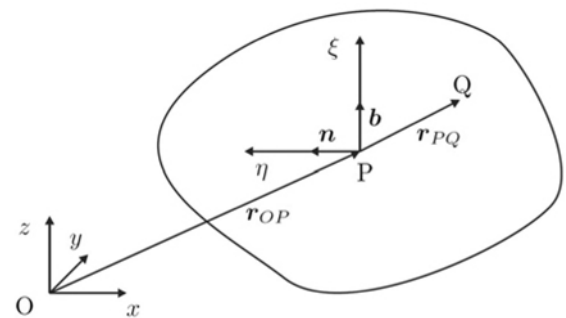


Figure 4. Describing arbitrary point from corresponding point P

For the governing equations it is assumed that the total motion of a cross section can be described by three components:

- a translation
- a small rotation
- warping

Using these assumptions, expressions for the displacement and the strain can be derived. More information can be found in [9].

CONSTITUTIVE EQUATIONS

After deriving the governing equations a material law is defined.

The spring is modeled as an isotropic hookean material with Young's modulus E and lateral contraction coefficient ν . Hooke's generalized law gives the relationship between strains and stresses in accordance with Eq. (6).

$$\begin{aligned}\sigma_{tt} &= \frac{E}{1+\nu} \left(\varepsilon_{tt} + \frac{\nu}{1-2\nu} (\varepsilon_{tt} + \varepsilon_{nn} + \varepsilon_{bb}) \right) \\ \sigma_{nn} &= \frac{E}{1+\nu} \left(\varepsilon_{nn} + \frac{\nu}{1-2\nu} (\varepsilon_{tt} + \varepsilon_{nn} + \varepsilon_{bb}) \right) \\ \sigma_{bb} &= \frac{E}{1+\nu} \left(\varepsilon_{bb} + \frac{\nu}{1-2\nu} (\varepsilon_{tt} + \varepsilon_{nn} + \varepsilon_{bb}) \right)\end{aligned}\quad (6)$$

$$\begin{aligned}\tau_{tm} &= \frac{E}{(1+\nu)} \varepsilon_{tm} \\ \tau_{tb} &= \frac{E}{(1+\nu)} \varepsilon_{tb} \\ \tau_{nb} &= \frac{E}{(1+\nu)} \varepsilon_{nb}\end{aligned}$$

The cross section is assumed to be elliptical with the major and minor axis a and b (Fig. 5).

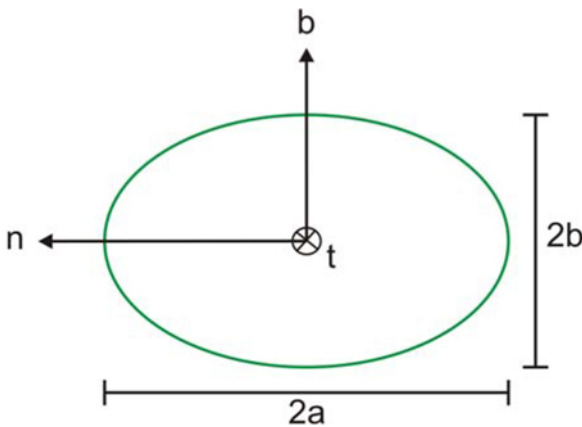


Figure 5. Elliptical cross section with major and minor axis a and b

With the torsional constant J (7), the shear modulus G (8), the curvature κ and the torsion τ the area integral of the stress functions over the elliptical cross section yields (9).

$$J = \left(1 + \frac{b^2 - a^2}{a^2 + b^2} \right) I_\xi + \left(1 - \frac{b^2 - a^2}{a^2 + b^2} \right) I_\eta = \frac{\pi a^3 b^3}{a^2 + b^2} \quad (7)$$

$$G = \frac{E}{2(1+\nu)} \quad (8)$$

$$\begin{aligned}T_t &= EA \left(\frac{du_t}{ds} - \kappa u_n \right) \\ T_n &= GA \left(\frac{du_n}{ds} + \kappa u_t - \tau u_b - \theta_b \right) \\ T_b &= GA \left(\frac{du_b}{ds} + \tau u_n + \theta_n \right)\end{aligned}\quad (9)$$

$$\begin{aligned}M_t &= GJ \left(\frac{d\theta_t}{ds} - \kappa \theta_n \right) \\ M_n &= EI_\eta \left(\frac{d\theta_n}{ds} + \kappa \theta_t - \tau \theta_b \right) \\ M_b &= EI_\xi \left(\frac{d\theta_b}{ds} + \tau \theta_n \right)\end{aligned}$$

EQUATIONS OF MOTION

Finally, the equations of motion are derived to describe the behavior of the spring.

Six main assumptions are made:

- the effect of curvature is negligible for the leverage of the resultants in the equilibrium of moments
- distributed loads can be approximated as constant in the infinitesimal part
- the motion due to warping is negligible
- the cross section is small in comparison to the radius
- pitch is small
- torsional warping of the cross section is equal to that of a straight rod

Making use of these assumptions, the equation of motion result in (Eq. 10).