Numerical model for simulating cam-shifting systems in automotive engines

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Abstract

One of the central topics in the automotive industry is the reduction of emission and fuel consumption. The improvement of the engine efficiency including the same or even better engine performance established over the past years is a challenging task which can not be faced with standard approaches. A promising concept to solve the requirement is “variability”, i.e. variable engine components which ensure a thermodynamically optimized system for all loads and engine speeds. One of these variability approaches is Schaefflers so-called cam shifting system (CSS) where each engine valve train is driven by two or more cams mounted on a so-called cam piece, see fig.1. The cams are tailored to the customer demands, e.g. for achieving a cylinder deactivation in order to reduce the emission in city traffic, for achieving more engine power in a passing manoeuvre or for cruising on a highway with constant velocity.
An important aspect of the CSS is the switching between the cams which is typically achieved by means of a mechanical pin-groove system, i.e. a pin enters a groove with a prescribed switching contour and shifts the cam piece so that the next cam is located over the finger follower, see fig.1. The contact forces between the pin and the groove contour are crucial for the durability of the pin and the safety of the system. Since the cam piece is rotating very fast, only a small time window is available for the switch. Hence, steep contour shapes may have to be applied which can entail high contact forces. Also the friction is relevant for the functioning of the CSS. If the friction is too high, the cam piece can rest in inconvenient positions or the switch may not be completely finished which results in unnecessary wear of the components.

Due to the dependency of multiple parameters, e.g. the engine speed or the oil pressure in the hydraulic lash adjuster, the system is characterized by a complex behavior. For the design of a proper working CSS numerical simulations are necessary. Therefore, a multidisciplinary dynamical model will be introduced in this paper.
The mechanical part is modeled in a simplified manner. Topologically the system can be regarded as a nonlinear 2-mass oscillator, one mass for the cam piece and one for the pin. In order to prohibit an arbitrary axial motion, the cam piece is attached to the shaft by means of an arrestor, see fig.1. The arrestor is modeled with a ball supported by a spring. The arrestor ball engages with a notch milled in the cam piece fixing the cam piece on the camshaft axially. If the CSS switches, the spring force is overcome and the ball enters another notch, arresting the cam piece again.
Several further contacts act at the cam piece introducing additional friction into the system, e.g. the contact with the roller finger follower, the contact with the shaft spline and others. The contact between the switching groove and the pin is handled in a special manner because it is an unilateral contact.
The equations of motion of the model can be formulated in a compact index-3 DAE system:

$$\begin{aligned}
&\dot{x}_{cam} = v_{cam}\,, \qquad \dot{x}_{pin} = v_{pin},\\
&m_{cam}\ddot{x}_{cam} = f_{ext}(x_{cam}, v_{cam}) + \lambda,\\
&m_{pin}\ddot{x}_{pin} = f_{stiff}\left(x_{pin}\right) + d_{struct}v_{pin} - \lambda,\\
&x_{pin} \leq x_{cam} + x_{contour}\left(\varphi(t)\right),
\end{aligned} \tag{1}$$

where x_{cam} and x_{pin} denote the axial cam-piece and pin position, φ is the rotation angle of the rheonomically driven shaft, f_{ext} summarizes all external forces acting at the cam piece, f_{stiff} and d_{struct} denote the nonlinear stiffness function and the structure-damping value of the pin. The

function $x_{contour}$ holds the shape of the contact contour, locally described in the moving coordinate system of the cam piece. The contact force λ is determined by the unilateral constraint which prohibits interpenetration of the pin and the contour.
In order to achieve an efficient calculation of the equations of motion, the DAE index is reduced algebraically. The resulting explicit index-1 equations are then solved by standard time-integration methods in MATLAB/SIMULINK. The unilateral constraint is implemented by means of a case distinction separating between the non-contact case ($\lambda = 0$) and the contact case where λ results from the constraint. Similar approaches are often used in real-time application where the contacts are identified by means of a collision-detection technique (e.g. using bounding boxes [1]).

A common problem with contact-dominated models is that friction is hard to estimate. Complex friction-model approaches exist where the microscopic surface structure in the contact is taken into account, see e.g. [2,3]. However, such methods suffer from large computational effort. For that reason, in our implementation a simple Coulomb friction approach is applied. The unknown friction values are determined with a global optimization approach matching the model to the results of long-term measurements carried out on a test rig.
Even though the model is rather simple, the comparison of the simulation results with the measurements yields very good results. It can be stated that the main physical effects in the CSS are covered by the model, e.g., resting of the cam piece due to large friction, double contacts at the contour due to rebound or lifting of the cam piece from the contour due to inertia effects.

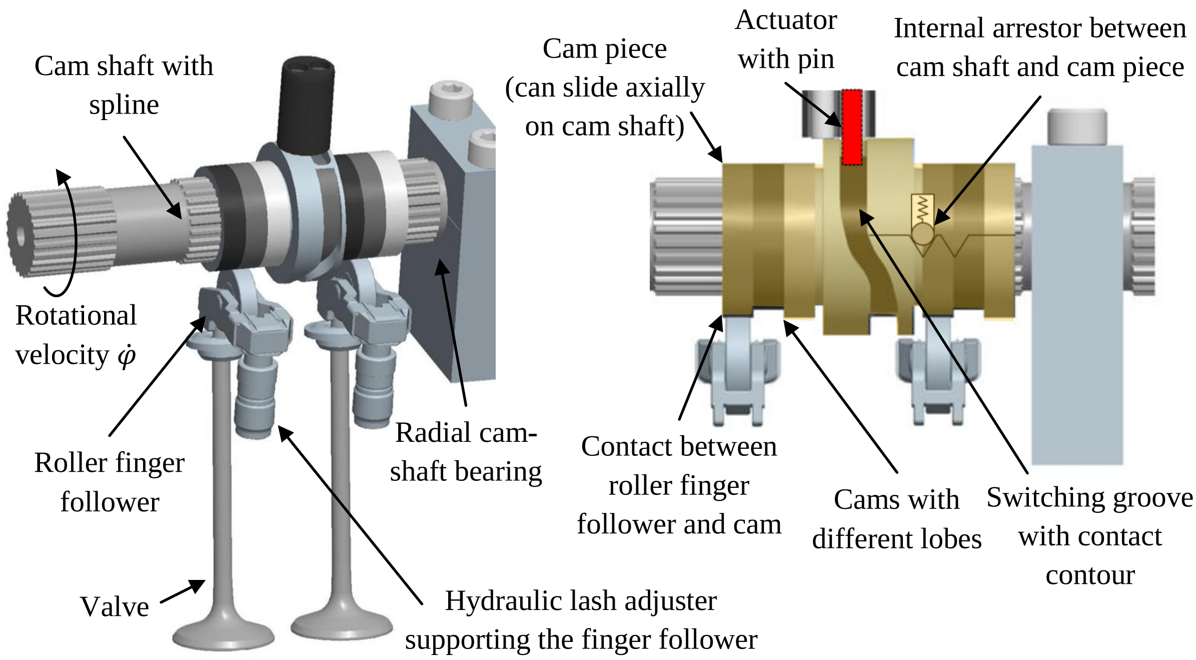

Figure 1: Example of a three-stage cam-shifting system. The cam piece is rotating with the shaft due to a spline. When the pin enters the groove the cam piece is shifted axially and a different cam lobe is used for the valve lift.

References

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