

Dynamic Analysis of the Audi Valvelift System

2010-01-1195

Published
04/12/2010

Robert Huber
Technische Univ. Muenchen

Peter Klumpp
AUDI AG

Heinz Ulbrich
Technische Univ. Muenchen

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ABSTRACT

Fully variable valve trains provide comprehensive means of adjustment in terms of variable valve timing and valve lift. The efficiency of the engine is improved in the operating range and in return, an increasing complexity of the mechanical design and control engineering must be handled.

For optimization and design of these kinds of complex systems, detailed simulation models covering different physical domains, i.e. mechanics, hydraulics, electrodynamics and control are needed. Topic of this work is the variable valve train named Audi valvelift system (AVS) e.g. used in the Audi 2.8l V6 FSI engine. The idea of AVS is to use different cam lobes at different operating points. Each intake valve can be actuated by a large and a small cam. For full load, the two inlet valves are opened by the large cam profile - ideal for high charge volumes and flow speeds in the combustion chamber. Under partial load, the small cam profiles are used. As a result, the gas exchange improves, throttling losses are minimized and fuel consumption is reduced.

To investigate the dynamical behavior and interactions between all subsystems, e.g. between chain vibrations and shifting events, an overall simulation model comprising the timing chain drive, chain tensioner, hydraulic cam phasing system, valve train and the AVS system including the electromagnetic actuator has been derived.

For modeling and simulation of the different subsystems, several highly specialized and efficient programs are used. To couple these subsystems co-simulation techniques are applied with the advantage that all subsystems can be computed in parallel on multi-core architectures to speed up integration time making optimization possible.

In this paper a dynamical analysis of the entire AVS valve train including the electromagnetic actuators combining experiments and simulation is presented. The simulation model is described and validated with experimental results using different test rigs. Furthermore, interactions between the subsystems and optimization results are presented.

INTRODUCTION

New emissions legislation, tougher CO₂ standards and increasing customer requirements concerning fuel consumption make new solutions in valve train technology necessary. Whereas in the past increased engine power, torque and dynamics have been the major development objectives, nowadays fuel efficiency is the most important factor in engine technology.

Due to the nature of fuel injection and gas exchange of internal combustion engines some conflicts of goals arise designing such systems. Engineers have to compromise between engine characteristics in terms of fuel consumption, torque-progress, maximum power, idle stability and emissions. To meet the demands, additional degrees of freedom are needed.

Fully variable valve trains provide comprehensive means of adjustment in terms of variable valve timing and valve lift. The efficiency of the engine is improved in the operating range and in return increasing complex mechanical and control design must be handled. For optimization and design of such kinds of complex systems, detailed simulation models covering different physical domains, i.e. mechanics, hydraulics, electrodynamics and control are needed.

Topic of this work is the variable valve train Audi valvelift system (AVS) e.g. used in the Audi 2.8l V6 FSI engine [1], [2]. To investigate the dynamical behavior and interactions between all subsystems e.g. between chain vibrations and shifting events an overall simulation model comprising the timing chain drive, chain tensioner, hydraulic cam phasing system, valve train and the AVS system including the electromagnetic actuator has been derived.

For modeling and simulation of the different subsystems, several highly specialized and efficient programs developed at the Institute of Applied Mechanics, Technische Universität München, are used: The dynamics of the valve train, the AVS and the cam phaser are analyzed with the multi-body simulation software MBSim [3], extended with modules for hydraulics, electromagnetic actuators and special contact kinematics. The timing chain drive is simulated with the program KETSIM. The subsystems are coupled using parallel co-simulation techniques. To save computing time and make optimization possible all subsystems are computed in parallel on multi-core architectures.

AUDI VALVELIFT SYSTEM

The Audi valvelift system manages the valve timing in a gasoline engine in a way that provides more power and torque and at the same time leads to lower fuel consumption. The aim is to open and close the valves in such a way, that the correct charge of air is drawn into the cylinders with minimum losses at the right time.

To adjust the valve timing, the Audi 2.8l V6 FSI engine comes with a hydraulic cam phasing system at both inlet and exhaust camshaft. The hydraulic cam phaser operates like a slew motor. In combination with a proportional 4/3-way flow control valve the relative angle between the camshaft and crankshaft can be controlled continuously.

Based on the described cam phasing system, which permits the valve opening and closing times to be varied, the Audi valvelift system goes a step further by varying the valve lift of the intake valves in two stages.

The idea of AVS is to use different cam lobes at different operating points. Each intake valve can be actuated by a large and a small cam. For full load, the two inlet valves are opened by the large cam profile (shown in red in [Figure 1](#)) -

ideal for high charge volumes and flow speeds in the combustion chamber. Under partial load, the small cam (green) profiles are used.

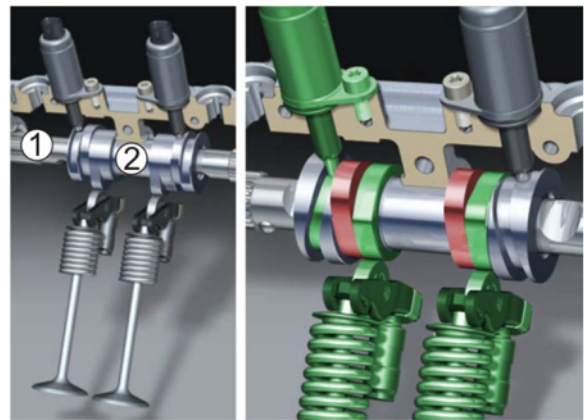


Figure 1. Audi valvelift system

The four cam lobes for one cylinder are mounted on a cam element (1), which can be moved sideways on the teathed camshaft (2). Which cam acts on the roller cam follower depends on the longitudinal position of the cam element. The shifting motion is induced by a pair of metal pins, which are extended by electromagnetic actuators. The pins are located above a spiral groove milled into the cam element. When the metal pin is lowered, it engages the spiral groove on the cam element and pushes the cam piece in longitudinal direction. A spring-loaded ball stop locks the cam element in the new position. In this way, the operating cams are changed from one set to another set. The second actuator is used to reverse the cam element. The changeover takes place in the range between 700 and 4000 rpm and is completed within one revolution of the camshaft. An axial movement of the cam element is only possible between two valve actuations.

As a result, the gas exchange improves, throttling losses are minimized and the fuel consumption drops by up to seven percent. Although Audi valvelift System is an easy to manufacture and robust system, it delivers about 80% of the results for about 20% of the cost compared to continuously variable valve train systems.

MATHEMATICAL MODEL

To describe the dynamical behavior of mechatronic systems like modern timing-assemblies in an accurate way, different physical domains - mechanics, hydraulics, electrodynamics and control - must be taken into account and integrated into one consistent mathematical model. In order to give the theoretical framework for modeling the presented AVS and subsystems, the governing equations are presented in this section. The description of the system dynamics is based on

the equations of motion for multi-body systems extended by hydraulic equations.

MULTI-BODY SYSTEM DYNAMICS

In this section a brief overview for the equations of multi-body system with bi- and unilateral constraints e.g. contacts, friction and impacts due to closing contacts is given. A main focus is laid upon the treatment of contact problems. Two different ways describing contacts are used within this work:

Classically, contacts in multi-body dynamics are described using spring and damper elements minimizing penetration of the bodies coming into contact. This obvious approach results in stiff differential equations with unintentional high eigenfrequencies and therefore high computational effort as well as uncertainties in the parameters for contact stiffness and damping. To avoid stiff differential equations rigid contact models are used. Within this non-smooth approach the stiffness in the contact zone is assumed to be infinite. Therefore impacts due to closing contacts lead to jumps within the velocities of the system. For rigid contacts, the dissipation of energy due to collisions is described by the coefficient of restitution. Whereas for the integration of smooth systems standard ODE and DAE integration schemes are used, special integration schemes and constraint solvers are needed for rigid models.

A general mathematical description of multi-body systems with uni- and bilateral contacts can be found in [4], especially for non-smooth systems see [5] and [6].

Equations of motion

The basic equations of motion for constraint multi-body systems write

$$\mathbf{M} \dot{\mathbf{u}} = \mathbf{h} + \mathbf{W} \boldsymbol{\lambda} . \quad (1)$$

The mass matrix $\mathbf{M}(\mathbf{q})$ depends on the vector of generalized positions $\mathbf{q}$. $\mathbf{h}(\mathbf{u}, \mathbf{q}, t)$ denotes the vector containing all smooth external and gyroscopical forces depending on $\mathbf{q}$, generalized velocities $\mathbf{u}$ and time t . To allow for constraint forces the Lagrange multiplier method is used with the vector $\boldsymbol{\lambda} = [\lambda_N^T \lambda_T^T]^T$ combining the magnitudes of the normal and tangential contact forces. The constraint 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 direction of the contact forces.

Equation (1) expresses the dynamics of a constrained multi-body system during an impact free phase at acceleration level. For impacts, a more appropriate formulation can be

achieved by replacing (1) by a finite dimensional measure differential equation [7]:

$$\mathbf{M} d\mathbf{u} = \mathbf{h} dt + \mathbf{W} d\boldsymbol{\Lambda} . \quad (2)$$

The measure for the velocities $d\mathbf{u} = \dot{\mathbf{u}} dt + (\mathbf{u}^+ - \mathbf{u}^-) d\eta$ is split in a continuous part $\dot{\mathbf{u}} dt$ for smooth dynamics and a discrete part $(\mathbf{u}^+ - \mathbf{u}^-) d\eta$ at discontinuities e.g. impacts. The velocities $\mathbf{u}^+$ and $\mathbf{u}^-$ before and after the impact are weighted by the sum of the *DIRAC* delta functions $d\delta$ at the discontinuities t_i :

$$d\eta = \sum_i d\delta(t - t_i)$$

$$\int_T f(t) d\delta = \begin{cases} f(t_i) & t_i \in T \\ 0 & \text{else.} \end{cases} \quad (3)$$

In analogy to the velocity measure, the reaction measure $d\boldsymbol{\Lambda}$ contains smooth forces $\boldsymbol{\lambda}$ and discrete impulses $\boldsymbol{\Lambda}$ due to collisions of bodies at impact times t_i :

$$d\boldsymbol{\Lambda} = \boldsymbol{\lambda} dt + \boldsymbol{\Lambda} d\eta . \quad (4)$$

Integrating equation (2) under consideration of the *DIRAC* delta function (3) yields the classical equation of motion for a constrained system (1) as well as the impact equations

$$\mathbf{M}_i(\mathbf{u}_i^+ - \mathbf{u}_i^-) = \mathbf{W}_i \boldsymbol{\Lambda}_i \quad (6)$$

being valid at times t_i of impacts.

Contact Kinematics

For the description of the contacts between two bodies, the contact geometry needs to be analyzed. The potential contact point $\mathbf{r}_c$, as well as the gap distance g_N and relative velocities $\dot{g}_N$ and $\dot{g}_T$ in normal and tangential direction have to be determined. These information is used to calculate the contact forces and generalized force directions $\mathbf{W}$.

Contours are described by the position vector $\mathbf{r} = \mathbf{r}(\mathbf{q}, s)$, the contour normal $\mathbf{n} = \mathbf{n}(\mathbf{q}, s)$ and the associated tangent $\mathbf{t} = \mathbf{t}(\mathbf{q}, s)$, all depending on the generalized position of the associated body and on the contour parameter s . Potential contact points can be identified by the necessary contact

condition, that the difference vector $\mathbf{r}_D = \mathbf{r}_{C1} - \mathbf{r}_{C2}$ between potential contact points on both bodies is perpendicular to both tangents. As an example the evaluation of the contact kinematics between the spiral groove (index 1 in Figure 2) of the cam element and the pin (index 2) is pointed out briefly.

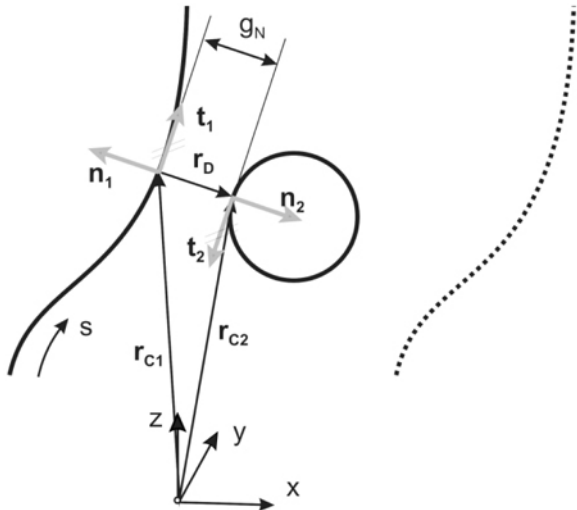


Figure 2. kinematics of contact between pin and groove

The contour of the spiral groove is interpolated with a cubic spline for each coordinate. Hence, a single point on the contour can be referenced by the scalar contour parameter s . The normal and tangential vectors $\mathbf{n}$ and $\mathbf{t}$ are obtained using standard geometry formulations. The contour parameter s for potential contact points can be calculated by solving

$$f(q, s) = \mathbf{r}_D^T \mathbf{t}_1 = 0. \quad (7)$$

Equation (7) is solved in two steps: First, using cylindrical coordinates, the groove contour is approximated by a helical curve with constant slope. Thus the contour can be described using analytical functions and an analytical solution s_0 for Eq. (7) exists. Second, this approximate solution is used as a starting value for a fix point iteration to calculate the exact value of the contour parameter. The computational procedure of the fixed point iteration is obtained by an orthogonal projection of the center of the pin on the tangent of the contour :

$$\mathbf{r}_{C1}^{i+1} = \mathbf{r}_{\text{pin}} - \mathbf{n} \mathbf{n}^T (\mathbf{r}_{\text{pin}} - \mathbf{r}_{C1}^i). \quad (8)$$

The distance between the two bodies results from $g_N = \mathbf{r}_D^T \mathbf{n}_2$. Based on the contact points the $\mathbf{J} = (\partial \dot{\mathbf{r}}_{Ci} / \partial \mathbf{u})^T$ between the system space and the

generalized coordinates are developed. The generalized force directions write :

$$\mathbf{w}_N^T = \mathbf{n}_1^T \mathbf{J}_{C1} + \mathbf{n}_2^T \mathbf{J}_{C2} \quad (9)$$

$$\mathbf{w}_T^T = \mathbf{t}_1^T \mathbf{J}_{C1} + \mathbf{t}_2^T \mathbf{J}_{C2}. \quad (10)$$

HYDRAULIC SYSTEMS

Modern valve trains and timing assemblies include hydraulic components like chain tensioners, cam phasing systems or hydraulic lash adjusters. To represent the dynamics and interactions between the subsystems the equations of motion are extended by hydraulic equations.

Modeling Hydraulic Systems

Hydraulic systems are considered to be a network of basic components like valves, hydraulic lines or hydraulic cylinders connected by nodes and links. The behavior of links is described by their volume flow Q . The characteristic states of nodes are their pressure p and their volume V . Considering compressibility of the fluid results in nonlinear differential equations for the pressure p and the Volume V of a node:

$$\dot{p} = \frac{E}{V} \dot{V} \quad (11)$$

$$\dot{V} = \sum_i Q_i + A_{hmi} u. \quad (12)$$

Therein Q_i denotes the volume flow into the node and A_{hmi} the areas of mechanical interfaces to the hydraulic volume projected into the direction of the generalized velocity u of a connected multi-body system. The bulk modulus E depends on the air content and the pressure describing the compressibility of the fluid. The nodes are connected by links. Using momentum conservation a differential equation for the volume flow Q is obtained:

$$\dot{Q} = \frac{A}{\rho l} (p_{\text{in}} - p_{\text{out}} - \Delta p) \quad (13)$$

$$Q \propto \Delta p^e \quad (14)$$

Neglecting the inertia of the moving fluid mass the flow is given by the algebraic Eq. (14) wherein the exponent e is 0.5 for conservative systems and 1.0 for laminar flow. The