

Modeling and Analysis of Valve Train, Part I - Conventional Systems

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ABSTRACT

In recent years, computer simulations gained an increased role in the design, development, optimization, and calibration of the valve train systems. With the development of non-conventional systems and actuation mechanisms, computer modeling became even more important. Part I of this article presents an overview of the current modeling and simulation methods of conventional valve trains at component and system level. First, the modeling of the valve train kinematics, including cam shape design and optimization, is summarized. Mathematical modeling of the valve spring, hydraulic lash adjuster, oil aeration, bulk modulus, contact stiffness and contact damping in multibody systems are discussed. The benefits and limitations of the different modeling approaches of the valve train dynamics are pointed out. Another important aspect is the valve train tribology. Elastohydrodynamic contact modeling, lubricant film thickness prediction, lubricant rheology, starvation, and asperity contact are discussed extensively. Current prediction capabilities of friction and wear in the main tribological contacts are presented and new research directions are pointed out. Part II of this article will discuss the modeling and simulation of variable cam timing systems and variable valve actuation mechanisms. Nontraditional and conceptual solutions including mechanical, electromechanical, electrohydraulic, electromagnetic, pneumatic, piezoelectric, and magnetorheologic actuation as well as control theories will be presented.

1. INTRODUCTION

Analytical predictions of the valve train performance are currently an integral part of the design and optimization process. Newly developed discrete [1,2,3,4,5] and fully variable valve train systems [5,6,7,8] led to an increased role of simulation tools and also to additional challenges. Often,

these systems involve interaction among multi-physics domains such as structural, fluidic, electrostatic, and electromagnetic. Current simulation methods have limited capabilities in handling simultaneously multi-domain interactions with sufficient level of detail, accuracy, efficiency, and solver stability.

Optimum valve train design involves a compromise among contradictory requirements. The cam must ensure fast opening and closing events for an optimum gas exchange process. Generally, dynamics of the valve train and its effect on durability and NVH are the limiting factors of most mechanical valve train systems. Valve train dynamics is typically carried out by using models of low complexity. Coupled multibody dynamic simulation of the full engine (valve train, timing drive, and crank train) with sufficient level of detail remains a challenging task. The modeling of individual components (hydraulic lash adjuster, oil aeration, bulk modulus, contact damping) is generally simplified and relies heavily on empirical equations. The capabilities of modeling detailed elastohydrodynamic (EHD) contact and lubrication (e.g. cam-follower) integrated into large scale multibody dynamic systems are minimal.

The concentrated contact conjunction between cam and follower is exposed to one of the most severe tribological conditions within internal combustion engines. Because of the high loads and small contact area, contact pressure is high and the elastic deformations are often several orders of magnitude higher than the minimum film thickness. In addition, at such high contact pressures, the viscosity of the lubricant can be ten orders of magnitude higher than the viscosity at atmospheric pressure [9]. The mineral oil loses its liquid character and experiences transition to glassy state (semi-solid) as shown experimentally. In this case the lubricant becomes non-Newtonian. Although the Newtonian assumption of the lubricant is sufficient to predict film

thickness, it cannot predict accurately friction which is usually one order of magnitude higher than in measurements [10, 11].

Valve train friction and wear is a complex process depending on the component geometry, loads, lubrication, sliding speed, materials, coatings, etc. In conventional systems, except for bucket tappet (type I) valve train, sliding is replaced by rolling contact leading to a significant reduction in friction. However, sliding contact is encountered in modern 2-step [4] and 3-steps [5] discrete cam profile switching systems as well as in the majority of mechanical continuously variable valve actuation mechanisms [7]. The main causes for wear in cam-follower and cam-sliding pad contacts are high stresses combined with mixed regime lubrication and variations of lubricant properties with pressure, temperature, and shear rate.

This paper presents a summary of current modeling and simulation methods of conventional valve train systems. First, the cam design and optimization methodologies are discussed. Then, the benefit and limitation of current valve train dynamic simulation methods are presented. The mathematical modeling considerations involving the valve spring, hydraulic lash adjuster, oil aeration, bulk modulus, and dry contact are detailed. The characteristics of the elastohydrodynamic cam to follower contact and implications of various theories on the accuracy of the lubricant film predictions are pointed out. Finally, friction and wear of main contact interfaces are presented.

2. VALVE TRAIN KINEMATICS

Figure 1 shows typical conventional valve train configurations. A discussion of the main advantages and disadvantages of each configuration is found in [12]. Variations of these standard configurations can be found in production engines (bridge, dual lifter, etc). A typical kinematic model of the valve train uses lumped masses which move under geometrical constraints. A dynamic model includes elasticity of the bodies and component separation. In both cases, motion is represented by an imposed angular speed at cam.

CAM DESIGN

The basic requirement of the curve describing the cam shape is that it must possess continuous first (velocity) and second (acceleration) derivatives [13]. Various functions have been proposed for the definition of the cam profile: polynomial [14], trigonometric series [15], trigonometric splines [16], trapezoidal, B-spline [17, 18], Bernstein-Bezier splines [19], cubic splines [20,21,22]. Polynomial cams use polynomial functions whose coefficients are determined from boundary conditions; trigonometric cams use sine and cosine functions while trapezoidal curves use pairs of isosceles trapezoids to

approximate the valve acceleration. Cam curves can be described by a combination of these functions which satisfy boundary and continuity conditions to assure smoothness. Spline cams rely on curve fitting algorithms to specify acceleration curve and have better flexibility for shape definition and computer optimization.

Optimum cam design involves a compromise among contradictory thermo-dynamical, mechanical, and manufacturing requirements. The cam must ensure a maximum flow area under the valve curve which translates into fast opening and closing events for an optimum gas exchange process. On the other hand, an aggressive cam profile leads to high accelerations, high forces, and undesirable dynamic behavior of the valve train. In addition, the cam design must satisfy manufacturing limitations with respect to maximum concave radius and durability requirements. Therefore, the cam shape generation is an integrated process which requires simultaneous evaluation of valve train performances rather than a standalone process of curve generation. Design considerations and performance parameters used in the cam design process are summarized in Table 1.

<figure 1 here>

For a given valve train layout and thermodynamically optimized valve lift duration, the cam shape and, therefore, effective valve lift is normally restricted by the mechanical requirements and manufacturing capabilities. Ideally, the effective valve lift will match perfectly the thermodynamic optimized curve. However, the effective valve lift can differ from the thermodynamic curve without a significant loss of gas exchange efficiency depending on the valve port flow coefficients. The flow coefficient can increase significantly in the first section of the valve event as shown in Figure 2 after that being almost constant. This suggests that the first section (e.g. 1mm valve lift) of the opening area is the most important to be considered in the cam design. Opening and closing durations should be small even if this may limit the maximum valve lift. In general, the minimum cam radius of curvature and maximum valve acceleration are the limiting factors in designing aggressive opening and closing events.

Figure 3 shows the valve acceleration for three different valve train designs with significantly different dynamic and thermodynamic performances. The reference curve A is characterized by a superior fullness of the valve lift (area under valve lift curve), a borderline radius of curvature, good spring margin (surplus), and good dynamic behavior. The more aggressive valve acceleration curve B has the advantage of increasing thermodynamic performances, however, with an unacceptable dynamic behavior. The less aggressive curve C features the best dynamic behavior by sacrificing thermodynamic performances.

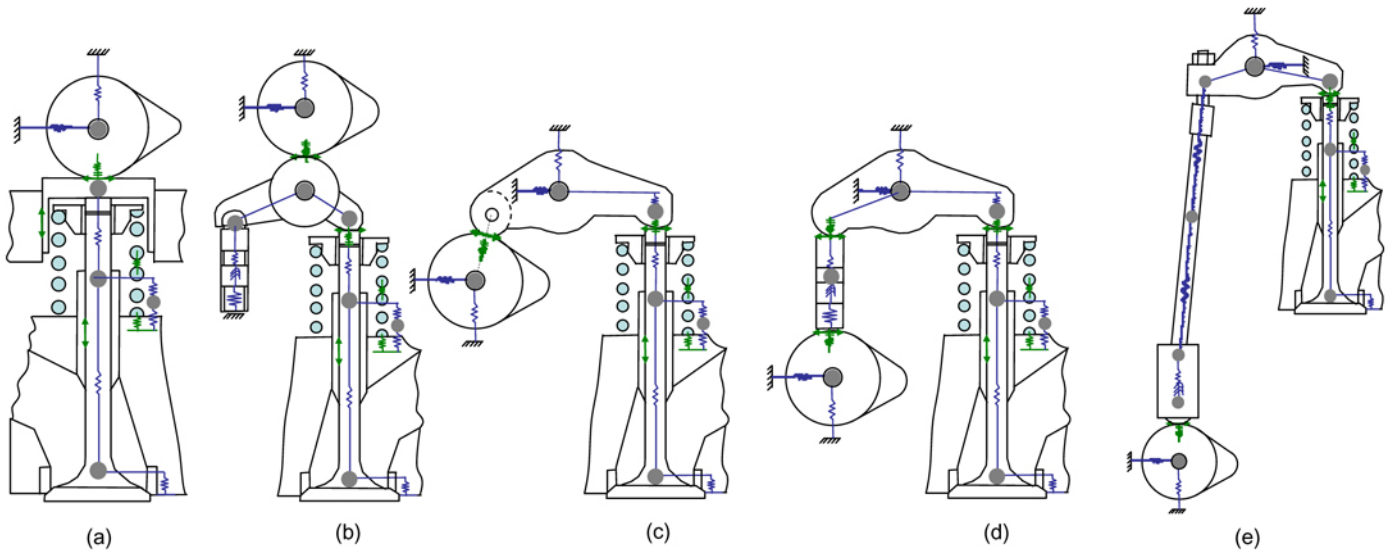


Figure 1. Typical valve train configurations: (a) type I - bucket tappet, (b) type II - finger follower, (c) type III - rocker arm, (d) type IV, (e) type V - pushrod.

Figure 4 shows typical kinematic results of a type II valve train design. A more comprehensive list of parameters evaluated is shown in Table 2. To ensure acceptable dynamic valve train behavior and avoid cam-follower contact loss, the valve acceleration trace and the valve spring margin should be balanced. Cam-follower contact stress at low and high speed is the main criteria used for cam material selection. The valve to piston clearance is calculated to prevent collision between valve and piston. This is even more important for engines with variable cam timing.

<table 1 here>

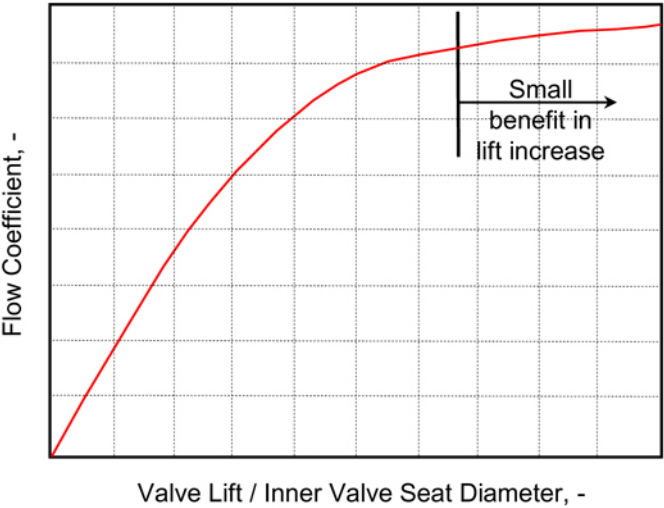
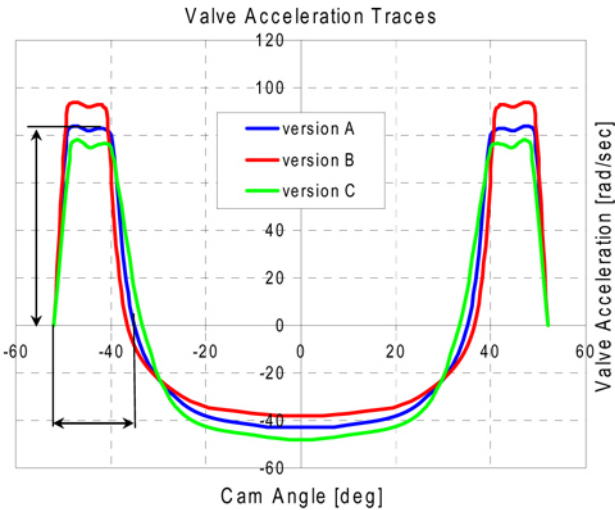


Figure 2. Dependence of valve flow coefficients on normalized valve lift.



	Characteristics of Valve Acceleration Curve
Fullness of valve lift curve	Version A: good Version B: best Version C: bad
Cam concavity (radius of curvature)	Version A: good Version B: bad Version C: good (no concavity)
Spring margin	Version A: good Version B: bad Version C: best
General dynamic behaviour	Version A: good Version B: bad Version C: best

Figure 3. Characteristics of valve acceleration in its implications on valve train performances

Table 1. Design considerations and performance parameters used in the cam design and evaluation of the valve train performance.

Design/ Analysis	Requirement (target)	Performance Parameter
Cam design	Thermodynamics	Maximum valve lift, valve opening/closing duration, duration at low (e.g. 1 mm) effective valve lift.
	Mechanical	Maximum valve acceleration, maximum valve velocity, ramp velocity and ramp height (or duration and height).
	Manufacturing	Minimum radius of curvature.
Kinematics /Kinetics	Valve spring	Preload, stiffness, spring stress, distance between coils at maximum lift.
	Durability / NVH	Safety factors against contact loss (spring margin), Hertzian stress at cam-follower, Hertzian stress at valve tip, Hertzian tip pressure vs. sliding velocity, minimum valve tip contact margin, lubrication number (type I), minimum axial force on the lash adjuster, force angle acting on HLA pivot (type II), minimum valve to piston clearance.
Dynamics	Valve spring	Ratio of natural frequency to camshaft speed, spring stress, distance between coils at maximum lift, spring coil clash.
	Durability / NVH	Valve seating (closing) velocity, cam-follower contact stress, valve tip contact stress, cam-follower contact loss (maximum and accumulated values), pushrod loads and buckling, valve bounce angle and height, HLA pumping up, lift loss (opening, maximum, closing), dynamic factor, system resonances, valve train natural frequency.
	NVH at low speed	Rate of change of cam-roller force, rate of change of valve-seat force.

<figure 4 here>

CAM OPTIMIZATION

Cam optimization can be carried out by using a standalone cam definition without considering the valve train system [14, 21, 22] or by simultaneously evaluating valve train kinematics and dynamics [17, 19, 23, 24]. Various optimization algorithms were proposed such as non-linear programming [14], genetic algorithm [17], or optimal control [25] to optimize acceleration [19, 24] or cam lift curves [17].

Yu and Lee [14] used non-linear programming techniques to optimize polynomial cams. The optimization of cam profile referred to a translating follower without considering system kinematics and dynamics. The objective functions for the optimization were related to the maximum velocity and maximum acceleration. Spline functions were also used for standalone cam optimization [18, 20]. Lampinen [17]

employed a genetic algorithm for cam shape optimization by taking into account system dynamics. The cam was defined using B-spline functions characterized by 40 control points. The objective function was to minimize the value of maximum pushrod force with constraint on minimum radius of curvature and valve seating velocity. Scheucher and Hellinger [19] employed Bezier splines description of cam acceleration. The objective was to maximize the area below the valve lift with kinematic and dynamic constraints related to the cam radius of curvature, contact stress, coil clash, contact loss, and valve bounce. Jeon et al [23] optimized the cam profile by modifying the Fourier coefficients of the cam lift curve. The optimization process was completed in two steps. In the first step, the objective function was maximization of the total flow area during the cam event. In the second step minor changes were made to reduce maximum valve acceleration with minimum changes of the flow area established in first step.

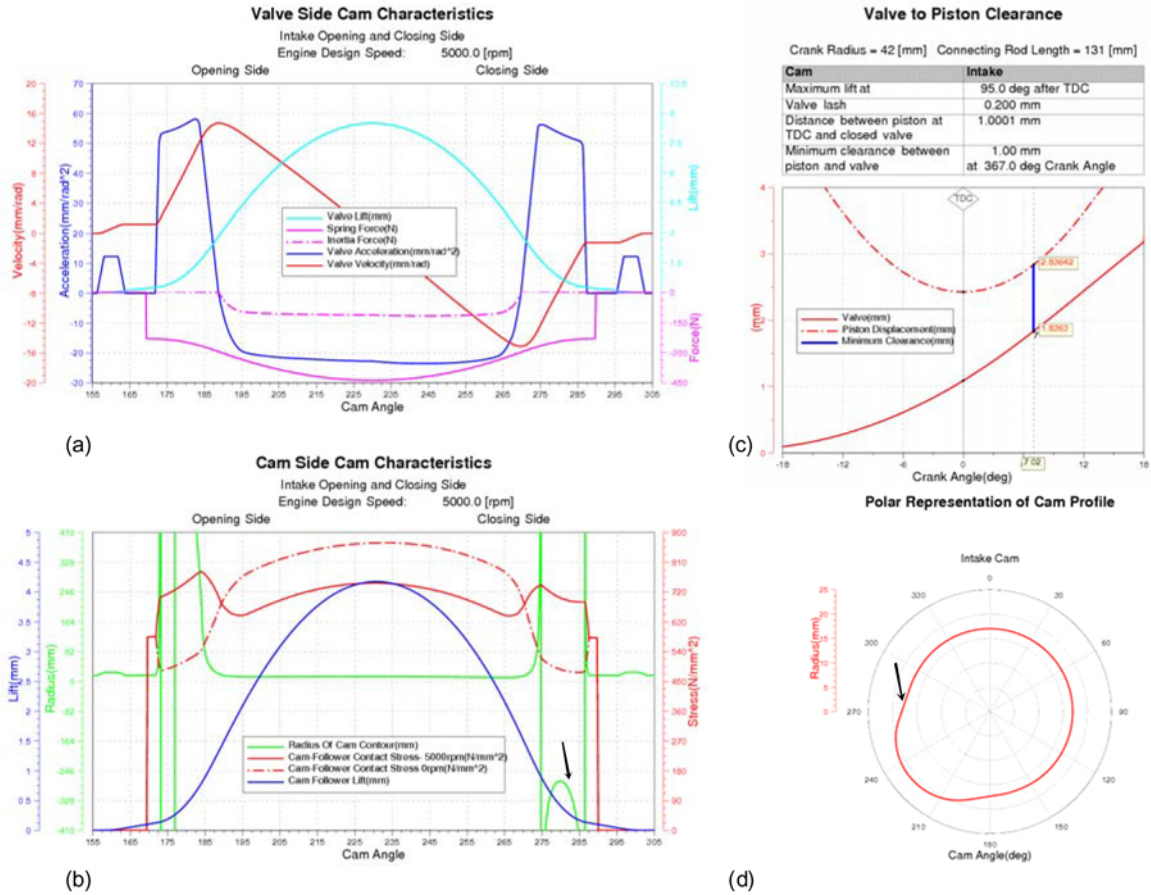


Figure 4. Valve train kinematic results (a) valve lift, valve velocity, valve acceleration, spring force, and inertia force; (b) cam follower lift, cam radius of curvature, and kinematic cam-follower contact stress; (c) valve to piston clearance; (d) cam profile in polar coordinates.

The general constraint optimization problem of the cam profile can be written as [19]

$$\min_p f(y, z, p, t) \text{ or } \max_p f(y, z, p, t) \quad (1)$$

subjected to constraints $g_k(y, z, p, t) \leq 0$, $k = 1, 2, \dots, q$, where f is the objective function, p represents design variables, y and z are the state variables.

Objective Functions

The maximization of area under valve lift curve is defined as [19, 23, 24, 26]

$$\max \int_{\alpha_0}^{\alpha_c} l(\alpha) d\alpha. \quad (2)$$

where $l(\alpha)$ is the valve lift, α_0 and α_c are valve opening and closing angles, respectively. Other possible objective

functions include minimization of the maximum pushrod force [17, 23], minimization of cam-follower contact stress and residual vibrations [26].

Constraints

Constraints functions imposed to the cam optimization problem are summarized below:

(c1). Valve velocity and lift at beginning, $v_o(t_{start})$, and end, $v_o(t_{end})$, of the cam profile are defined in advance and maintained unchanged during the optimization process [19]. These values are determined based on the definition of the ramps.

$$g_1 = v_o(t_{start}) = V_{start}, \quad g_2 = l_o(t_{start}) = l_{start}, \dots$$

$$g_3 = v_o(t_{end}) = V_{end}, \quad g_4 = l_o(t_{end}) = l_{end}.$$

(3)

(c2). Negative radius of curvature, ρ_c , is greater than limit value, ρ_{Limit} , because of manufacturing constraints and cost [19, 17, 14]

$$g = \min(0, \rho_c) + \rho_{Lim} \leq 0 \quad (4)$$

where $\rho_c = \frac{(s^2 + S')^{3/2}}{s^2 + 2S'^2 - S''s}$, $s = R_b + l$, R_b is the radius of cam base circle, l is the valve lift.

(c3). The contact stress between cam and follower, σ_c , is lower than the limit value, σ_{min} [19, 24]. This constraint is dictated by the cam material strength.

$$g = \sigma_c - \sigma_{Lim} \leq 0. \quad (5)$$

(c4). The force acting on coil springs, F_s , is lower than the limit value, F_{SLim} [19]. This constraint limits coil clash.

$$g = F_s - F_{SLim} \leq 0. \quad (6)$$

(c5). The cam-follower contact loss is lower than the limit value. There are two limits related to the single contact loss, α_{SL} , and accumulated contact loss, α_{AL} ,

$$g = \alpha_{SL} - \alpha_{SLLim} \leq 0 \text{ and } g = \alpha_{AL} - \alpha_{ALLim} \leq 0. \quad (7)$$

(c6). The valve bounce amplitude is lower than the limit value. Valve bounce amplitude is defined as difference of dynamic valve lift, l_b , and kinematic valve, l_b^{kin} , at closing ramps

$$g = l_b - l_b^{kin} \leq 0. \quad (8)$$

Other possible constraint functions refer to the valve seating velocity [17, 23], maximum valve acceleration [14, 23, 24], maximum valve velocity [14], maximum valve lift, valve opening and closing angles, and valve to piston clearance [24]. A performance indexes which combines two or more objective functions were defined in [23, 17]. The relative importance of each objective must be defined by the cam designer.

3. VALVE TRAIN DYNAMICS

The main parameters used to evaluate the dynamic performance of the valve train are summarized in Table 1.

Valve train dynamics is typically carried out by using models of varying complexities. A coupled multibody dynamic model of the full engine (cranktrain, valve train, and timing drive) with a high level of details is a challenging task for any commercial software. Large scale models generate solver instabilities and non-practical simulation time. Another limitation consists of inadequate capabilities of modeling elasto-hydrodynamic contact and lubrication. Often, valve trains are modeled with limited degrees of freedom e.g. neglecting tilting of the lifter or valve and the 3D motion of components such as chain and sprockets. Damping and friction are formulated arbitrarily or based on experience with little or no mathematical foundation. The hydraulic lash adjuster is generally simplified and modeling of oil aeration is difficult. Another common simplification is the representation of components as rigid bodies connected by springs with constant stiffness, although stiffness of the rocker arm varies continuously during the valve event. Additional challenges emerge when modeling coupled interactions of the valve train with variable cam timing systems, or modeling electrohydraulic, electromechanical, and electromagnetic variable valve actuation systems which involve coupled simulation of multi-physics domains.

The most common approach is to evaluate the dynamic performances of the valve train at the subsystem level by using a single valve train model [27,28,29,30]. This approach neglects the interactions among cylinders as well as the effect of timing drive and crank speed irregularities. If dynamic, durability, and NVH performance targets are satisfied at the subsystem level then it is assumed that the full system will also meet the performance requirements. The disadvantage of this method is that it cannot capture the torsional and bending vibrations of the camshaft, and the interactions with timing chain, belt, gears, or VCT systems.

Another approach is to consider the complete valve train by including the camshaft and valve trains corresponding to each cylinder without timing drive [31,32,33]. In this case interactions among individual valve trains as well as the torsional and bending effect of the camshaft are taken into account. The benefit of this approach is, however, limited because it neglects the bending of the camshaft induced by the axial force in the timing chain, belt, or gears. In addition, the torsional vibrations of the cranktrain and timing drive are neglected. A possible improvement of this method is to incorporate chain forces and speed irregularities obtained from a separate simulation of the chain drive or from a test.

A third approach has a higher degree of complexity and consists in coupled simulation of the valve train, camshaft, and timing drive. Bending and torsional vibrations of the camshaft as well as the effect of timing drive forces and vibrations are taken into consideration. The addition of VCT would further improve the reliability of the results and offer capability of multi-system investigation. Finally, an enhanced

analysis consists of coupled simulation of valve train, timing drive, and crank train [34].

Du and Chen [33] developed a complete dynamic model of type II valve train for a V6 engine considering the interaction between valve train and crankshaft. The torsion vibrations of the crankshaft resulted in speed fluctuations affecting the dynamics of the valve train components. The amplitude of angular displacement of 0.2-0.3 degrees resulted in a decrease of the minimum axial force on HLA which can lead to pumping up. Rob and Arnold [31] represented the camshaft by discrete masses connected by springs with bending and torsional stiffness. They pointed out that camshaft bending has a major influence on the dynamic performances of the valve train. Takagishi et al. [34] employed a large scale model including the crankshaft, timing chain, valve train, fuel injection pump, oil pump, and accessories belt drive for a supercharged direct injection diesel engine. The fuel injection pump is driven by the camshaft resulting in large fluctuation torque and causing large loads on the timing chain.

Representation of the valve train components as flexible body rocker arm [33, 035, 36] and camshaft [33] instead of rigid bodies connected by spring and dampers capture more accurately stiffness and inertia distribution of the components. Also, removing constraints and increasing the number of degrees of freedom allow redistribution of forces among contacting bodies. Ito [37] investigated both analytically and experimentally the skew motion of the roller tappet for a push rod valve train. The author showed that, for accurate results, a mathematical model should include the elastic deformation of the cam and roller and elasticity of the tappet. The skew amplitude was affected by the tappet-guide misalignment (clearance), roller profile, and moment of inertia of the tappet. Roth [38] characterized empirically the viscous damping force between valve stem and valve guide as a function of relative velocity. At low relative velocities, damping (drag) force was high enough so that the system was overdamped. Damping force decreased exponentially as relative velocity increased although the author did not give an explanation of this phenomenon. High valve stem damping is beneficial since it reduces the valve bounce and the amplitude of the response after the impact event.

The next paragraphs discuss modeling of the valve spring, hydraulic lash adjuster (HLA) as well as modeling of the contact in multibody dynamic systems.

VALVE SPRING

First, the valve spring has to provide sufficient preload to keep the valve closed on the base circle. This requirement becomes more important at GTDI and diesel engines with high valve back pressures. Second, the spring has to provide sufficient force to balance dynamic loads and avoid component separations or large residual vibrations. The

spring properties are in general defined by stiffness and frequency characteristics. During compression, the number of active coils decreases resulting in change of stiffness, effective mass of the active coils, and frequency. The nonlinearity of the spring is not related to intrinsic characteristic of the spring wire but to the reduction of the number of active coils. A nonlinear stiffness characteristic is usually employed to change the spring frequency during valve event or avoid excessive oscillations, surge modes, and coil clash at high engine speeds. Accurate modeling involves definition of centerline path (pitch profile), coil cross section, and mean diameter profile. A spring model should capture non-linear stiffness, frequency, and spring surge.

A linear spring surge model was proposed by Philips et al. [39] under the assumption that each individual infinitely small element is governed by the wave equation

$$\frac{\partial^2 \psi(x,t)}{\partial t^2} = c^2 \frac{\partial^2 \psi(x,t)}{\partial x^2} \quad (9)$$

where $\psi(x,t)$ is the displacement of each element of the spring, $c = \sqrt{L^2 k / m}$ is the speed of wave propagation, L , k and m are spring free length, stiffness, and mass, respectively. Assuming that the spring is fixed at both ends and mass and stiffness are distributed evenly along the spring, the resulting force is given by

$$F = -k(S + h(t)) + \pi k \sum_n n G_n(t) \quad (10)$$

where S is the difference between free length and installed length, $G_n(t)$ is the n-th mode shape.

The model gives satisfactory results for springs with linear stiffness characteristic, constant pitch profile, constant coil diameter, and no coil interactions.

Schamel et al. [40] proposed a discrete valve spring (multi-mass) model consisting of series of masses connected with spring and damper elements. The contact between coils during the coil clash is modeled using a nonlinear (elastic and damping) contact force. For an accurate representation, each active coil should be discretized in at least one mass. Mass and stiffness distribution of each discrete element is defined as

$$m_i = \rho A 2 \pi \Delta N_i r_i \text{ and } \kappa_i = I_i G_i \cos(\theta) / 2 \pi r_i^3 \Delta N_i \quad (11)$$

where ρ , A , G , I_t are density, cross section area, shear modulus, and torsional moment of inertia, respectively, ΔN is discretized coil number, r is radius of discretized coil, and θ is pitch angle.

Schamel's model was adopted in [41, 42]. An equivalent spring model consisting of one linear flexible body and nonlinear contact forces to simulate interactions among coils was proposed by Ortoman and Skovbjerg [41]. Iritani et al. [29] developed a spring model by representing the active coils as beam elements including axial and shearing forces, as well as twisting and bending moments. Takashima et al. [43] developed a spring model used for detail analysis of the stress state and shape effect on spring durability. An optimum spring shape was determined by parametric studies.

The natural frequency of the valve train is dominated by the most flexible component. The first mode is primarily due to valve spring [27]. However, other components such as HLA oil with high aeration, finger follower/rocker arm or the pushrod can have a significant influence on stiffness. Increasing valve spring natural frequency might be difficult and costly because of increasing coil stress which demands more expensive materials. Another option is to include a damper coil within the valve spring. This, however, leads to increased friction and reduced spring fatigue life because of spring heating. Alternatively, multiple springs per valve can be used. Seidlit [27] mentioned that lowest valve spring frequency should be higher than the frequency corresponding to the 15th engine order of crank rotation. However, valve spring in modern engines have lower frequency in the range of 11th-12th engine order.

HYDRAULIC LASH ADJUSTER

A typical hydraulic lash adjuster (HLA) contains a high pressure chamber, supply chamber, check valve, plunger, return spring and outer housing. The check valve is normally closed and opens to refill the high pressure chamber after a certain amount of oil is lost because of leakage or to compensate for the change in the thermal valve lash. The hydraulic tensioner in chain drives has a similar construction and working principle. An alternative design involves a normally open check valve (NOLA) which has the advantage of avoiding the HLA "pumping up" but has the disadvantage of non-uniformity of valve timing at low engine speeds and larger valve lift loss.

The HLA is required to compensate for the thermal valve lash. The disadvantage is low stiffness due to oil compressibility which is further reduced by the presence of un-dissolved air. In addition, the oil leakage leads to a reduction of the HLA length which translates into valve lift loss. Lower valve train stiffness limits the dynamic performances of the valve train leading to NVH and

durability issues. Modeling of the hydraulic lash adjuster (HLA) involves modeling of leakage between plunger and outer housing, flow through the check valve during the refilling period, oil aeration, and bulk modulus.

The oil pressure in the high pressure chamber (HPC) is governed by the following first order differential equation

$$\frac{dP(t)}{dt} = \frac{\beta}{V(t)} \left(-\frac{dV(t)}{dt} + Q_{cv}(t) + Q_{leak}(t) \right) \quad (12)$$

where β is the bulk modulus which includes the effect of oil aeration, V is the HPC volume, Q_{cv} is the volumetric flow through check valve, and Q_{leak} is the leakage through the annular path between plunger and outer housing.

Check Valve Flow

The flow through the check valve is related to the pressure differential and depends on the flow area and ball position (lift)

$$Q_{cv} = \pm C_d A_h \sqrt{\frac{2|\Delta p|}{\rho}} \quad (13)$$

where C_d is discharge coefficient, A_h is oil flow cross section, and ρ is oil density. Depending on the pressure differential, the check valve flow can have positive or negative values.

Leakage

The leakage flow has a component due to pressure difference along the path (Poiseuille flow) and a component due to relative motion of the plunger (Couette flow). The leakage flow in an HLA is generally modeled by assuming the idealized formula for annular laminar flow

$$Q_{leak} = \pm \frac{\pi d_p \delta^3}{12\mu_{oil}} \frac{|\Delta p|}{L} \quad (14)$$

where d_p is the plunger diameter, δ is the radial clearance between plunger and housing, μ_{oil} is dynamic viscosity of oil, Δp is the pressure difference between HPC and supply chamber, and L is the leakage length.

Although widely used, idealized leakage formula (14) does not allow a good correlation with data from a pulsation test

especially at high frequency excitations. Equation (14) is derived by assuming continuous, laminar, and perfectly annular flow. In practice, these assumptions are not completely valid. A more flexible expression of the leakage flow is

$$Q_{leak} = \pm \frac{\pi d_p^a \delta^b}{12\mu} \frac{|\Delta p|^c}{L^d} \quad (15)$$

where a, b, c and d are empirically determined exponents to fit experimental data.

Oil Aeration

The high pressure chamber (HPC) of the hydraulic lash adjuster (HLA) contains both dissolved and undissolved air. The amount of air dissolved depends on the pressure in the HPC. If the pressure is higher than saturation pressure (P_{sat}) then, at equilibrium, the air is completely dissolved. Below saturation pressure, the fraction of dissolved air is given by Henry's law. The process in which air is dissolved or separated from the oil is not instantaneous and occurs over a range of pressures. In the HPC, the pressure is generally higher than the saturation pressure which would suggest that the air is totally dissolved in the oil. This is not the case because in the highly transient events specific to valve trains, the air does not have enough time to dissolve. Transition process from undissolved to dissolved state involves a large time constant while the reverse process is faster but not instantaneous. The oil pressure in HPC is different from the oil circuit pressure. The measurement of oil aeration in HPC is difficult therefore measurements for the oil circuit are usually performed instead. Indirect measurement techniques to determine oil aeration by measuring the collapse of the plunger and subtracting the effect of leakage were proposed by Zhao et al. [44].

Oil aeration increases at higher engine speed [45]. Increasing the oil pressure at the supply chamber can reduce aeration [46]. Figure 5 shows the influence of oil aeration on valve seating velocity. At high aeration levels, valve seating velocity exceeds the target NVH limit of 1 m/s. Oil aeration has a significant influence on the valve lift and valve velocity closing angle as shown in Figure 6.

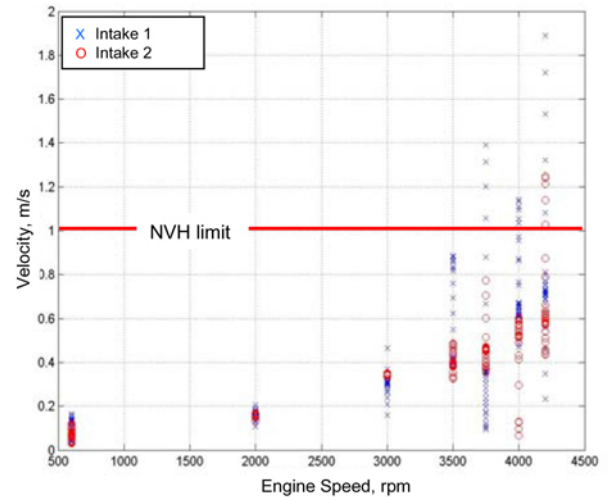


Figure 5. Dependence of valve seating velocity on engine speed for various values of oil aeration (0%-30%).