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Methodology for the geometric layout of a mechanically fully variable valve train with two synchronously rotating cam disks

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Abstract

New engine concepts such as Miller, HCCI or highly diluted combustion offer great potential for further optimization of ICEs in terms of fuel economy and pollutant emissions. However, the development of such concepts requires a high degree of variability in the control of gas exchange, characterized by variability in valve spread, maximum valve lift and – ideally independent of these two variables – in valve opening time. In current series variable valvetrains, valve lift and opening duration are usually directly dependent one from the other. In the ideal case, however, engine concepts such as Miller require a fully flexible variation of the closing time of the intake valve while still maintaining the same intake opening time. Here, a methodology for the geometric layout of fully variable valve trains with significantly extended functionalities is presented. In this concept, the control of the valve opening and closing events is distributed to two synchronously rotating cam disks. This geometric separation allows to vary the valve opening duration at constant maximum valve lift by varying the phase offset between the two disks. On the other hand, the geometric properties of the system can be used to vary the maximum valve lift at the constant valve opening and/or valve closing (depending on the layout), as well as for switching additional valve events on or off.

The methodology presented here includes the computer-aided and partially automated generation of the characteristic geometric features of the system and the kinematic simulation and evaluation of the concept. By kinematic simulation, various possible resulting valve lift curves can be evaluated and optimized by adapting the geometry and the motion rules. The subsequent investigations on a component test bench serve to assess the newly developed concept with respect to functionality, required drive torque, stiffness and speed capability, thus proving its technical feasibility.

Introduction

The use of new combustion processes continues to be a promising way of improving the efficiency of internal combustion engines. Approaches such as homogeneous compression ignition (HCCI) or charge dilution with a particularly high residual gas concentration (from internal or external EGR) bring about advantages in fuel economy as well as in pollutant emissions, but require special boundary conditions to ensure robust operation without misfire. In order to enable these processes, particularly the gas exchange of the engine needs to be controlled very precisely. The main objectives in this regard are to exactly dose the amount of fresh air and residual gas, to precisely control the in-cylinder flow and thereby to produce the required boundary conditions for combustion. [1,6,8]

The novel system studied in this investigation will be referred to as HVVT (**H**ighly **V**ariable **V**alve **T**rain) in the following. It is based on the mechanically fully variable valve train "UniValve" [2]. Like UniValve, most of the mechanically fully variable valve trains such as "VALVETRONIC" from BMW [3] or "UpValve" from Rheinmetall Automotive [4] use a rotating camshaft as well as a stationary actuator shaft. In contrast, the system under consideration here uses two synchronously rotating camshafts. Starting from the UniValve system, the eccentric shaft is replaced by another camshaft. An intermediate lever contacts both cam discs with a separate roller, producing an overlay of both cam profiles. This motion of the intermediate lever is transmitted via a working curve (intermediate lever support) to a roller cam follower and then to the valve. By decoupling the opening and closing mechanisms, the corresponding portions of the valve lift can be designed and implemented independently of each other.

HVVT offers very high flexibility and can be used for different applications by adapting the design strategy. On the inlet side, for example, HVVT can be used to vary the valve opening duration. This allows the amount of fresh air supplied to the cylinder to be controlled. In contrast to the systems commonly used today, defined closing times with consistently high closing accelerations are possible. With this system, early as well as late inlet closing can be implemented, while the opening duration can be varied independently at the same time. For the exhaust valve, on the other hand, it is possible to vary the valve lift and opening duration of a second exhaust lift event (e.g. parallel to the intake process) without affecting the main exhaust lift. This functionality can be used to adjust the amount of recycled residual gas with cycle-to-cycle accuracy. [9]

In order to develop this novel valve train system, a specially adapted design methodology is required. In particular, the mathematic-geometrical design of the HVVT cannot be carried out simply by defining two motion functions, as this was still the case with the predecessor system UniValve, since the second rotating camshaft adds a further degree of freedom. Additionally, the number of required optimization loops between the concept idea and the application in a fired research engine needs to be minimized in order to save cost and time in development. This is realized by extending the kinematic simulation by an analysis of the curvature, as well as by the early integration of gas exchange simulation.

Background

In mechanically fully variable valve trains, the rocker arm (and thus the valve) is not actuated directly by the camshaft. Instead, the cam actuates an intermediate lever which then transmits the movement to the rocker arm via a guiding profile. The position of the intermediate lever relative to the rocker arm can be changed using a rotatable eccentric disk. This changes the transmission behavior and thus the effective lift curve transmitted to the valve. The eccentric disk is mounted on the control shaft, which is typically actuated either continuously or in defined steps by an electric motor. The development history of several mechanically fully variable valve trains is shown in Figure 1.

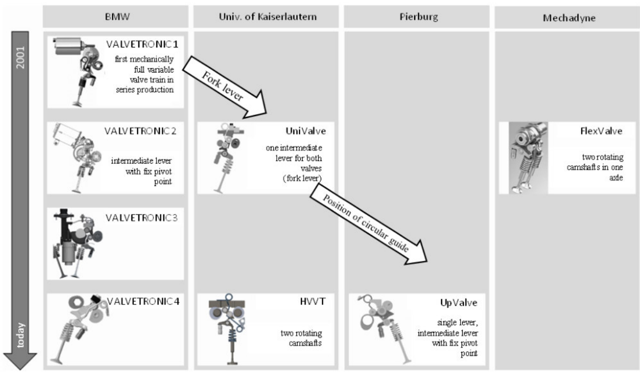

Figure 1. Development chronology of the discussed valve trains [2] [3] [4] [5]

With VALVETRONIC, BMW has implemented the first mechanically fully variable valve train in series production. Since then, it has been optimized in four generations. In [3], it is demonstrated how the required installation space, speed capability and cost have been improved through these four generations. In large-scale production, the system has so far been used mainly for dosing the aspirated air mass and thus for load and residual gas control in gasoline engines. In addition, a so-called "phasing" (differential lift) between the two intake valves of a cylinder is used to increase in-cylinder charge motion at small valve lift by inducing swirl.

At the University of Kaiserslautern, a fully variable valve train has been derived from VALVETRONIC, with similar functionality, but a different geometric layout. This concept was called UniValve. In [2], such a system was integrated in a gasoline engine, and load and residual gas control were studied subsequently. [7]

Based on the UniValve system, Rheinmetall Automotive (Pierburg) introduced a further development called "UpValve" [4]. In this concept, both the circular guideline roller and the circular guide were moved from the cam roller axle to the eccentric roller axle. This provides the intermediate lever with a fixed pivot point, meaning that it no longer rolls on the eccentric roller. However, the basic behavior remains the same.

Another mechanically fully variable valve train which is different from the previously mentioned systems is "FlexValve", which is also offered by Rheinmetall Automotive. As shown in [5], the system has very similar functionality to the HVVT under consideration here. The intermediate lever also contacts two cam disks. However, these are supported by two shafts which are located one inside the other. One cam is connected to the inner shaft by a bolt and can therefore be adjusted relative to the cam on the outer shaft. The phase adjustment is thus limited by the feasible size of the recess in the outer shaft. Since in HVVT the two shafts are separated, this restriction is not valid here, resulting in a significantly enlarged control range.

In the "FlexValve" system, the intermediate lever is connected to the roller cam follower by an axle. HVVT has an additional degree of freedom of motion in this regard when compared to FlexValve, meaning that the intermediate lever of HVVT needs to be kept in position by two springs instead of just one.

The aim of realizing the HVVT concept is to further expand the experimental capabilities available at the institute beyond the functionality of UniValve, in order to provide a functionality similar to FlexValve, but with an extended adjustment range. HVVT is intended for use in a single-cylinder research engine for combustion process development, and not for series applications. Therefore, it is focused especially on realizing the maximum possible variability in valve actuation for research purposes.

HVVT is designed as a purely mechanical fully variable valve train and requires a classic roller cam follower. However, the camshaft does not actuate the roller cam follower itself, but an intermediate lever. This lever is synchronously actuated by a second camshaft, which will be called “eccentric shaft” in the following. The intermediate lever is guided in vertical direction by the circular guide. The superposition of the cam lift and the eccentric lift is transmitted by the intermediate lever to the roller cam follower via the intermediate lever contour, which will be called “working curve” here. The working curve represents a transfer function, converting the pivoting movement of the intermediate lever into the desired valve lift.

In this context, cam lift means the movement of the cam follower roller ① during one revolution of the camshaft. The eccentric lift also describes the movement of the eccentric shaft follower roller ② (see figure 2).

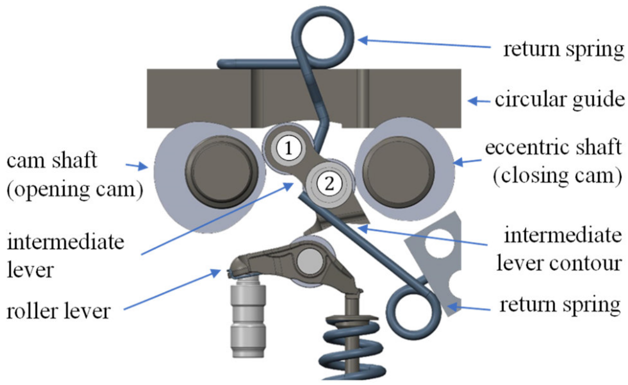

Figure 2. CAD Model HVVT

As with other mechanically fully variable valve trains, the cam lift is transferred completely to the valve when the eccentric roller is fully actuated. With HVVT, the eccentric reaches its maximum and minimum position with each revolution of the camshaft and thus behaves in the same way as a camshaft. The superposition of the cam lift and eccentric lift can be regarded as a kind of multiplication. If one of the cams is at 0 mm lift, the rocker arm roller is not actuated and no lift is transferred to the valve. On the contrary, only when both cams are in their respective maximum lift positions, the maximum valve lift will be carried out by the valve.

Methodology

The development approach of a fully variable valve train, which is to be used for the optimization and further development of modern combustion processes, comprises a series of development steps from the concept idea to a fully functional system in the research engine in fired operation. Figure 3 shows the iterative process for design optimization of the fully variable valve train.

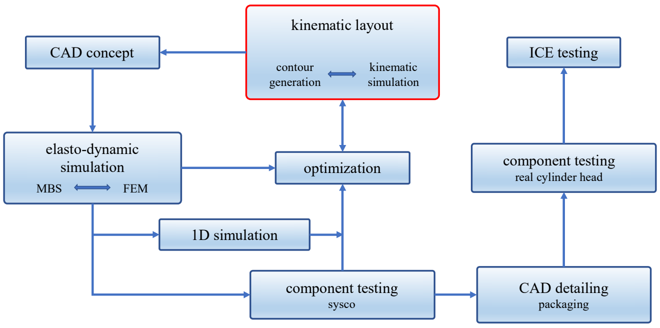

Figure 3. Methodology for designing the mechanically fully variable valve train

The first step is the kinematic layout, i.e. the mathematical design of the characteristic valve train components, which is then followed by a kinematic simulation. This kinematic simulation provides initial information about the adjustment characteristics of the calculated lift curves and their curvatures. In the next step, a first concept design is developed from this valve train data. This CAD model is then used for multi-body simulation (MBS). The multi-body simulation serves to verify the contours developed and simulates the dynamic behavior of the valve train. The results are again used to optimize the valve train. The MBS simulation is usually complemented by a coupled FEM simulation, through which individual components can be analyzed with regard to their load. For the evaluation of the theoretically possible valve actuation functionalities and their influence in fired engine operation, a 1D gas exchange simulation can be carried out using the calculated resulting valve lift curves. This is followed by prototype manufacturing of the components, which will then be used for testing in a so-called system cylinder head. During component testing, the influence of the individual components on friction is identified, the speed stability is analyzed, and the effective valve lift curves in real operation are measured. Subsequently, further optimization loops are carried out in order to define the design for the integration into a real engine cylinder head in the next step. This system is analyzed again on the component test rig and then finally installed in the engine intended for fired operation.

Kinematic Layout

The kinematic layout describes the numerical methodology for generating the characteristic curves in the valve train. In the case of HVVT, these are the cam, eccentric and working curve contours.

The contour generation of cams in classic (non-variable) valve trains is realized by defining a lift function. The lift function describes the valve lift as a function of the camshaft angle. For curve generation in mechanically fully variable systems such as UniValve, a second motion function is usually required. While the previously mentioned lift function still defines the movement of the valve, in addition a swivel function is necessary to define the cam lift (through the swivel movement of the intermediate lever). From these two specifications and the geometry, the working curve results from the stepwise superposition of the movements (degree by degree). The design of HVVT requires a different approach here than previously used, since the additional continuously rotating shaft provides a further degree of freedom of movement.

As a start of the concept layout, the zero lift circle and maximum lift circle diameters of the cam and of the eccentric as well as the positions of the rocker arm roller at the maximum and the minimum position of the valve are defined in advance. The working curve results from these end positions and the geometric boundary conditions of the system. As already known from the predecessor system UniValve, the working curve requires an area in which the movement is transmitted to the valve, and another zone which is supposed to compensate for the additional movement of the cam which would go beyond the end positions of the valve. These two areas are called “lift” and “zero lift” area, respectively (figure 4).

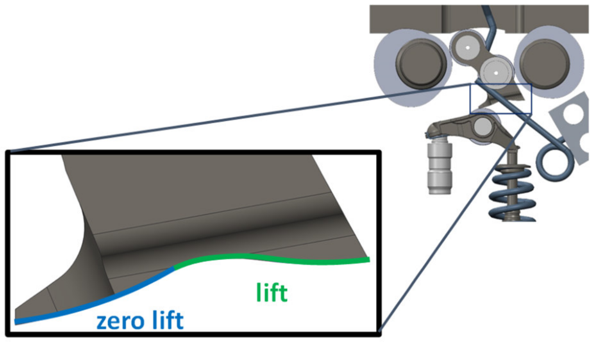

Figure 4. Lift and zero lift area of intermediate lever contour

The main functionality of HVVT is achieved by the geometric separation of the control of the opening and closing portions of the resulting valve lift curve to the two rotating cams. In order to fully transmit the opening characteristic of the valve as defined by the camshaft (i.e. the opening cam) without any additional influence from the working curve, a linear rise in the lift range of the working curve is introduced, taking into account the rotary motion of the intermediate lever. Therefore, the lift function transmitted to the intermediate lever by the opening cam is finally transferred to the valve by the working curve by a pure multiplication. The same correlations apply for the closing characteristic.

Direct control (opening or closing) of the valve without any influence of the second shaft is only possible if the second shaft transfers its maximum lift to the rocker arm. This results in an adapted basic shape for the cam geometry in the HVVT valve train. A complete separation of the opening and closing events of the valve can only be achieved by cam shapes with a significant lift plateau.

Figure 5 shows the cam and eccentric lifts applied to the intermediate lever and the resulting valve lift curve. At the beginning of valve opening, the eccentric (“closing cam”) has already reached its maximum lift ①. Because the eccentric now stays at maximum lift, the opening lift of the cam ("opening cam") is transmitted directly to the valve until the latter is finally opened at its maximum lift. The opening cam then maintains its maximum lift, and the eccentric in turn initiates the closing of the valve ②. As soon as the closing movement of the eccentric (and thus of the valve) is finished ③, the cam reaches the zero lift area of the working curve. This means that no further valve lift is carried out, and the necessary resetting movements of the cam and eccentric for the next cycle can be performed.

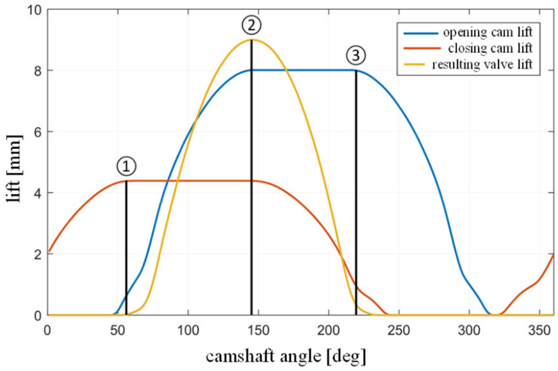

Figure 5. Movement of opening cam (camshaft), closing cam (eccentric shaft) and resulting valve lift curve. Here the maximum lift plateau in the cam movements can be seen.

Creation of the cam and eccentric contours

The approach for cam profile generation is based on the definition of a desired valve lift curve as a first step, which is defined as an input for the calculation of the cam contour. During this so-called "cam synthesis", the other camshaft (or eccentric shaft, respectively) is permanently kept at its maximum lift. This ensures that the maximum achievable valve lift only results when both shafts are at maximum lift, and that a "zero lift" of the valve can also be produced when the second shaft is at maximum lift.

The input valve lift curve is applied to the valve step by step depending on the cam angle. By means of an optimized numerical intersection detection method, the resulting lift position of the rocker arm is transferred to the intermediate lever and on to the cam follower roller. This results in the cam contour being generated as a function of the cam angle. For the second shaft, this process is carried out identically. By varying the input valve lift curves, different opening and closing characteristics can be generated in the cam contour. Since the characteristic of a cam lift plateau is essential for the desired functionality - especially with respect to the achievable adjustment range -, the input valve lift curves are optimized in advance, with the aim to ensure that the longest possible lift plateau is available. After carrying out the cam synthesis for both shafts, all contact contours are calculated in the system in the next step. However, no prediction is yet possible at this point with regard to the potential motion characteristics of the system. For this purpose, a kinematic simulation is carried out subsequently.

Kinematic simulation

The kinematic simulation includes a complete contact point calculation of the valve train as a two-dimensional model. It determines the geometrically resulting valve lift curves, taking into account the adjustment of the valve train, i.e. including the phasing of the two control shafts relative to each other. In addition to the resulting valve lift curves, a first evaluation can already be carried out concerning valve lift speed and acceleration. Besides the necessity of continuous differentiability, the major evaluation criteria are the maximum positive and, particularly, the minimum negative accelerations. By superposition of the cam and eccentric lift motion, extreme acceleration peaks can be produced. The input valve lift curves for generating the actuator shaft contours can be specifically adapted to optimize the behavior. In addition to the maximum and minimum accelerations, the kinematic simulation also analyzes the overall resulting valve accelerations and velocities as well as the curvature of the cam lobes and of the working curve (Figure 6), which may then be optimized. In this way, excessive curvature can be avoided, and the dynamic behavior of the valve train can be influenced in a positive way even in advance of the multi-body simulation.

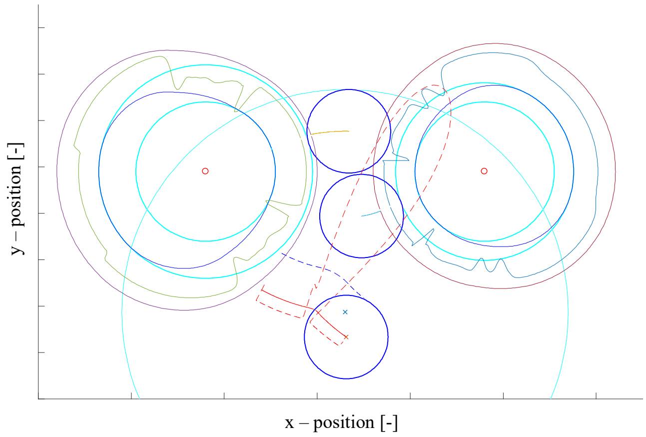

Figure 6. Schematic representation of cams, eccentric and working curve curvature

The high flexibility of the valve train – and also of the design methodology – allows the implementation of a large number of different valve train functionalities with HVVT. Each design strategy produces individual cam contours, while the working curve remains unchanged. The following paragraphs will provide an overview of various functionalities.

1) Opening duration with lift plateau

As described above, the valve train system can be used to achieve complete decoupling of the opening and closing portion of the valve lift. This functionality is used in the design described here for a fully variable adjustment of the valve opening duration. The length of the resulting lift plateau can be adjusted by rotating the cam and the eccentric relative to each other. For this design variant, very long lift plateaus on the cam discs are required, as clearly visible in Figure 7.

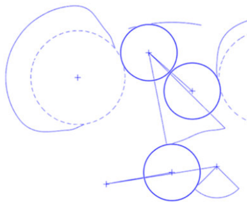

Figure 7. Profiles of “opening duration control”

The direction of rotation of the two cams in the figure is counter-clockwise. First, the right shaft (eccentric) sets its maximum lift. The working curve is travelled through in its zero lift area. At maximum eccentric lift and minimum cam lift, the transition point between the zero-lift range and the lift range of the working curve is in contact with the cam follower roller. The eccentric shaft initially remains in its maximum position, and the camshaft enters into its lift event. Since the lift area of the working curve is now in contact, the movement of the cam disk (i.e. the left cam disk in Fig. 7) is directly converted into the opening motion of the valve.

Then, the tasks of the shafts are interchanged. The cam disc now keeps its cam lift constant by a plateau, and the eccentric shaft closes the valve. The working curve is then again at the transition between the lift and zero lift area. The cam lift is subsequently reduced to zero in order to reset the system for the next revolution.

As shown in Figure 8, the accelerations of cam and eccentric lift are not influenced mutually, because at the moment of opening and closing, the acceleration of one of the cams is always zero. By changing the phase between both shafts, the valve opening duration can be adjusted by extending the valve lift plateau.

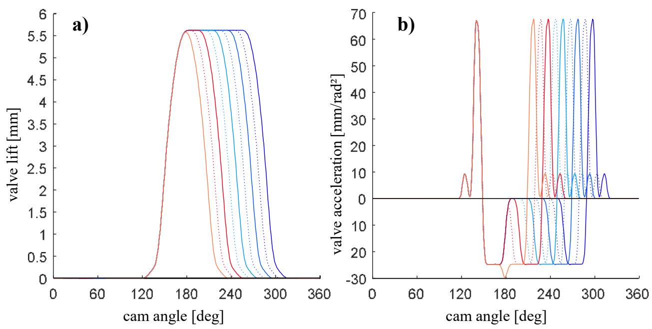

Figure 8. Valve lift (a) and acceleration (b) with variable valve opening duration and constant maximum lift

2) Second Event

Another design strategy follows the principle that the desired lift is completely represented by the cam. The eccentric disk (i.e. the right disk in Fig. 9) takes on the adjusting role here. The eccentric is designed in such a way that the maximum lift circle is approached quickly from the base circle (zero lift circle) and maintained at a plateau for a long duration. This is comparable to a conventional mechanically fully variable valve train, in which the eccentric is at full lift only for a limited period of time, and completely reset during the remaining time. By shifting the closing section of the eccentric shaft into the valve lift duration, the valve event can be reduced (in lift) or shortened (in opening duration).

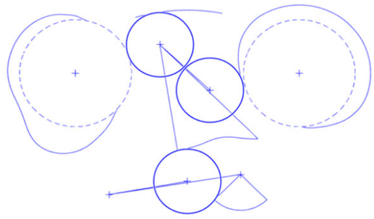

Figure 9. Profiles of “second event control”

This function may also be used to vary a separate additional valve lift event (so-called "second event") without affecting the main event. During the period in which the main lift is performed by the cam, the eccentric is at its lift plateau. The closing profile of the eccentric shaft then determines the timing at which the second event closes. Using this method, a series of lift profiles was generated, as shown in Figure 10. The main lift remains unchanged for all variants, while the second event was varied from zero to maximum lift.

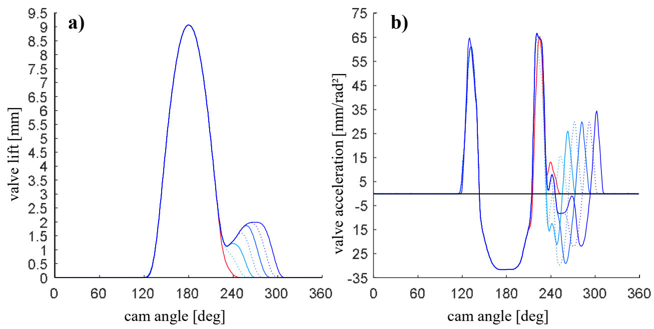

Figure 10. Valve lift (a) and acceleration (b) with variable second event

3) Opening duration and lift

The combination of the two design variants mentioned so far produces a superposition the motion functions of cam and eccentric lift. This makes it possible to adjust not only the opening duration, but also the maximum valve lift. An exemplary geometry is shown in Figure 11.

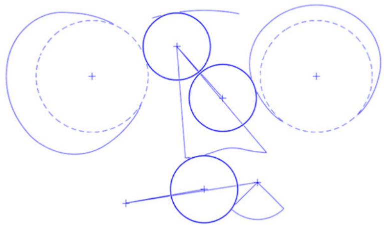

Figure 11. Profiles of “lift reduction control”

Figure 12 shows a family of lift profiles with variable valve opening duration and variable maximum valve lift. The cam and eccentric functions of HVVT are overlaid, similar to the variable opening duration with a lift plateau. This results in very slim valve lift curves, whose closing accelerations remain constantly high for a long time and only drop off to small lifts. However, there is a superposition of the negative acceleration ranges, which must be considered in the design. In this exemplary design (Figure 12) the negative acceleration is superimposed to a value of -26 mm/rad² and thus remains sufficiently moderate. This limit was carried over from the UniValve system to check the feasibility with HVVT, and to be able to compare both systems directly.

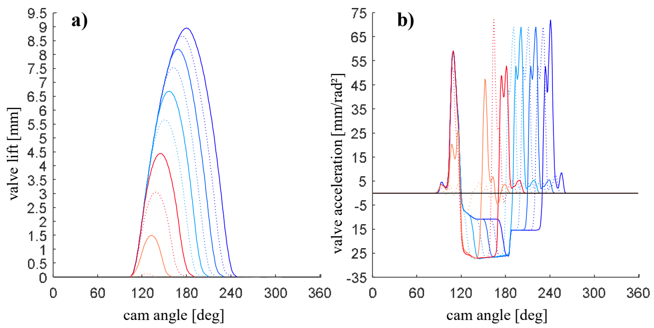

Figure 12. Valve lift (a) and acceleration (b) with variable valve opening duration and maximum lift

The closing accelerations of mechanically fully variable valve trains with a static adjustment device (static eccentric shaft) like UniValve drop much more rapidly. UniValve leads to a longer opening duration for the same lift compared to HVVT for partial lifts. This correlation is shown in Figure 13. The maximum valve lift is plotted over the corresponding valve opening duration of the partial lift. The valve opening duration is evaluated here at a valve lift of 0.3 mm. With the same partial lift, HVVT produces a shorter opening duration than a UniValve system. For example, the opening duration for a 2 mm lift is 160 degCA for the UniValve system and 104 degCA for the HVVT system.

Figure 13 b) shows the closing acceleration related to the maximum lift for the respective partial lift. The lift curve of HVVT is capable of exploiting the maximum closing acceleration of 70 mm/rad² already at 3 mm.

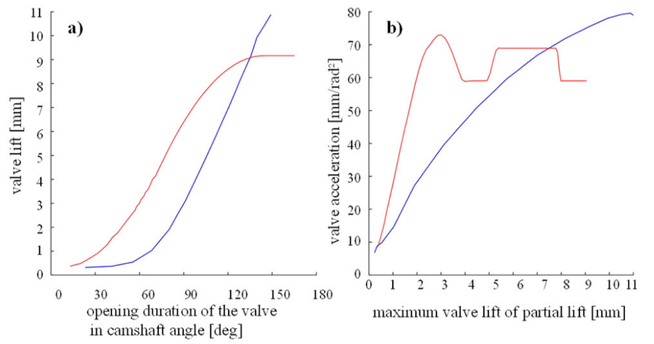

Figure 13. Maximum valve lift related to valve opening duration of the respective partial lift (a) / Maximum closing acceleration related to maximum lift (b) - red: HVVT, blue: UniValve

Dynamic Layout

After the characteristic curves (cam, eccentric and intermediate lever profiles) have been derived and optimized with regard to curvature and control characteristics, the real components are designed using CAD. In order to quickly evaluate different approaches, the shape-defining parameters for the levers are transferred. Then, the next step in the development process is the mechanical validation based on a co-simulation of a multi-body simulation (MBS) and a structural-mechanical analysis based on the finite element method (FEM). In the MBS, all contacts and bearings are modelled on the basis of previous measurements and sensitivity analyses [10,11]. The rapid generation of CAD models for the levers allows to systematically adjust the inertia properties of the components within a short time frame. On this basis, the components experiencing significant deformation are modelled more accurately. With the help of FEM, the deformations as well as the effects due to vibrations in the corresponding natural modes can be considered. An exemplary time step of such a simulation during the maximum lift is shown in Figure 14. The deformation of the levers is shown in a color scale on the components and runs from dark blue (minimum) to red (maximum).

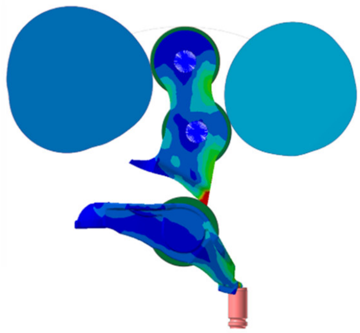

Figure 14. Elastic deformation of intermediate lever and rocker arm from MBS-FEM co-simulation at maximum lift (red: max. deformation, dark blue: min. deformation)

This simulation includes mechanisms that lead to a deviation of the valve lift from the numerical design. These mechanisms include component deformation, bearing stiffness, contact stiffness, inertia and spring models including the calculation of natural frequencies [10, 11]. From these calculations, additional information is gained on the stiffness and on the high-speed stability of the system. This method allows the desired valve lift to be quickly converted into hardware and reduces the number of necessary optimization loops on the test bench.

Layout of Gas Exchange

In the following, the previously calculated valve lift curves (including adjustment characteristics) are used as input variables for gas exchange simulations using GT-Suite as a software tool to estimate the influence of the extended variability of the valve train in fired engine operation. In the present case, a turbocharged gasoline engine with direct injection is used as a basis for the modelling. This adjusted engine model integrates a detailed cooling circuit model with a water jacket. The heat transfer, flow and tumble coefficients as well as the turbulence generation (from tumble break-up) are derived from CFD calculations carried out with AVL FIRE. In addition to the basic measurement of the series engine, extensive data from experiments with various modifications of the valve train of this engine were available for the adjustment of the engine model. In this way, the UniValve system has been integrated step by step on the intake and exhaust side, including functions such as cylinder deactivation and second event on the exhaust side [7]. With the help of the experimental data, a physical model of a predictive combustion could also be implemented, which is controlled by residual gas mass fraction, charge dilution and flow boundary conditions (p, T, TKE) at ignition timing. This makes it possible to reproduce the basic measurement with correlation coefficients of R > 95%.

The "second event" of the exhaust valve can be used to control the residual gas in the cylinder. During the intake stroke, fresh air is supplied to the cylinder, and as the exhaust valve opens for its second event, exhaust gas is rebreathed from the outlet additionally. The amount of residual gas in the cylinder can be controlled by continuously adjusting the second exhaust lift.

This engine model uses the HVVT in different versions on the intake and on the exhaust side. The original intake lift and residual gas control by a second event on the exhaust side were modeled. As obvious from Figure 15 (load point BMEP = 2 bar at n = 2000 min^{-1}), a residual gas mass fraction from 12% to almost 50% can be adjusted within a range of 46° phase angle between both shafts.

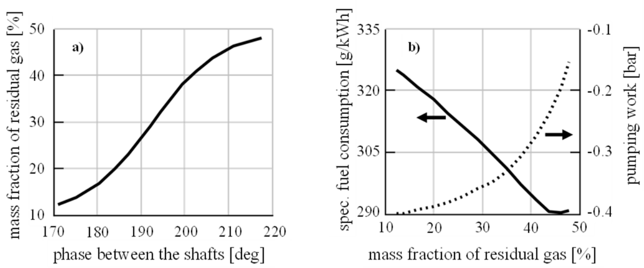

Figure 15. Mass fraction of residual gas as a function of the phase between cam and eccentric shaft (a) / fuel consumption and pumping work vs. mass fraction of residual gas (b)

Due to residual gas recirculation, intake throttling is reduced significantly, as can be seen in Figure 16 from the steadily decreasing pumping work. However, this is only noticeable in specific fuel consumption up to a certain amount of residual gas. As the amount of residual gas increases further, efficiency is no longer determined primarily by gas exchange, but combustion takes over the more decisive role. Particularly in the case of cycle-by-cycle residual gas control, e.g. by an additional lift on the exhaust side, the colliding intake and exhaust flows lead to a partial elimination of the tumble motion, thus reducing the in-cylinder charge motion and ultimately turbulence intensity. This reduces the advantage due to throttling significantly at high residual gas content, which is also obvious in Figure 15. At extremely high amounts of residual gas, specific fuel consumption is not reduced any further. Here, the effects of the reduced pumping work and the deterioration of combustion are counter-balancing one another.

Figure 16 shows another way of influencing the in-cylinder flow by reducing the pressure in the intake manifold. With reduced intake pressure, smaller second event lifts are necessary to achieve a certain amount of residual gas. This leads to improved boundary conditions for combustion. This control lever for the in-cylinder flow can be used in HCCI operation to control the homogenization of the in-cylinder charge and, as a result, the start and the progress of combustion.

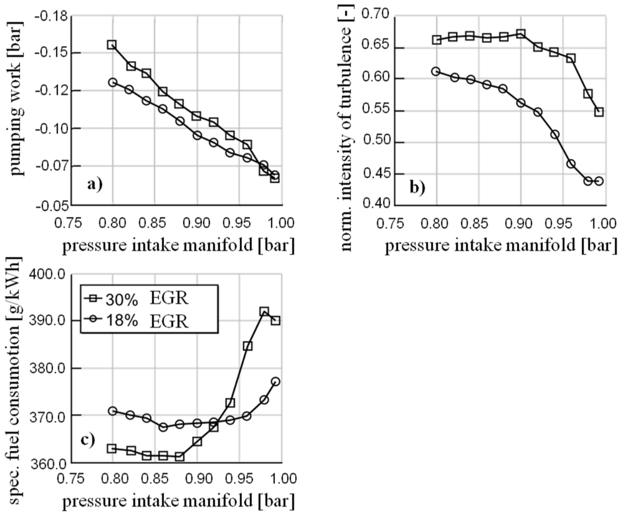

Figure 16. Influence of intake manifold pressure on pumping work (a), turbulence intensity (b) and specific fuel consumption (c) at n = 2000 rpm, BMEP = 2 bar, EGR = 18% vs. EGR = 30%

Experimental Validation of Valvetrain Dynamics

In the next step, the valve train is mounted on a component test bench in order to check the validity of the simulation results and to demonstrate the feasibility of the concept. On the component test bench, a so-called “system cylinder head unit” contains the valve train of a single cylinder, but without the intake and exhaust ports or the water jacket (Figure 17). Both the camshaft bearings and the eccentric shaft bearings are supplied with pressurized oil, identical to the situation in a real engine cylinder head. Additionally, the upper rollers of the intermediate lever are lubricated by an oil spray. The phase adjustment between the camshaft and the eccentric shaft can only be carried out manually in this development stage. For the current layout, the camshaft and the eccentric shaft are coupled by two gearwheels. By disassembling these gears, the shafts can be rotated against each other. In this constellation, the cam and eccentric shafts rotate in opposite direction. When integrated into the real engine head of the research engine in a future step, phase control will be realized by automatic camphasers, both for the relative adjustment of the camshafts and for the control of intake and exhaust cam phasing relative to the crankshaft.

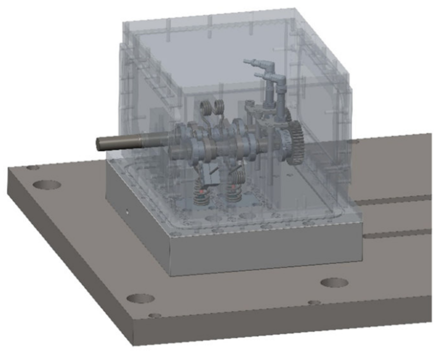

Figure 17. Assembly: cylinder head unit with valve train

Both the camshaft and the eccentric shaft have triple hydrodynamic bearings. These bearing bushings can be exchanged within a few minutes by bearing bushings with other bearing positions. Furthermore, the cylinder head unit uses a so-called "assembled" (modular) camshaft. The cam lobes are fixed to the shaft by means of a feather key and locknut (Figure 18).

Figure 18. Modular camshaft with eroded cams

The cam lobes can be wire EDM-ed from a disc within one day and are then used directly. In combination with the cam, the intermediate lever can also be exchanged. The intermediate lever is manufactured in large quantities as a pre-milled and hardened blank. For a new variant, only the working cam and the contact surface to the eccentric are high-precision wire EDM-ed from the solid in a single mounting setup. If the cam maximum lift is to be modified, cam and intermediate lever always have to be exchanged in pairs, as otherwise this would result in a different valve lift maximum.

Figure 19 shows the basic setup of a system cylinder head unit on the component test bench. Oil flows through the cylinder head unit at a defined pressure and temperature in a closed cycle. The camshaft is driven directly by an electric motor, and the friction power on the camshaft is measured. Below the baseplate, a laser vibrometer is mounted which records the lift, speed and acceleration of the valve. Furthermore, due to the excellent accessibility of the valve spring, it is possible to measure the spring contact forces using strain gauges or suitable force sensors. Additionally, the circular guide as well as the roller and the intermediate lever can be equipped with strain gauges. The entire measurement data is recorded simultaneously as a function of the cam angle and can be visualized online on the test bench.

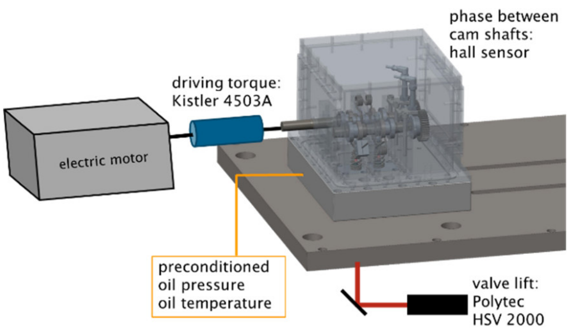

Figure 19. component test bench with mounted cylinder head unit

The valve lift curves of the previously developed valve train can be measured under realistic boundary conditions in this system cylinder head unit. By this, reduced valve lift resulting from component deformation, manufacturing tolerances or valve clearance can be determined and integrated directly into the early development steps by comparison with the MBS simulation model. Furthermore, the system component test bench is used for friction analysis of the system. In particular, the influence of the adjustment characteristics can be evaluated (Figure 20). Due to the additional rotating shaft, the driving torque of HVVT is approximately 15% higher compared to the predecessor UniValve.

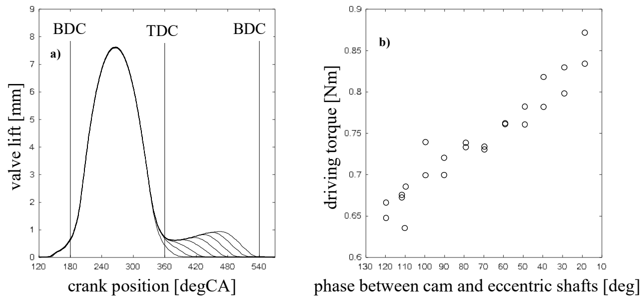

Figure 20. Measured second event valve lift (a) / friction torque vs. phasing between the shafts (b)

The results achieved demonstrate on the one hand the feasibility of the innovative valve train, and on the other hand, they provide further information for optimizing the valve train in order to proceed to the next step, the integration into a real engine cylinder head.

Summary

The development of future highly variable valve train systems requires a new systematic approach for the entire development process. From the basic idea to the functional realization on an engine in fired operation, numerous development and optimization steps are required. The work presented here introduces the development methodology for the design of the novel highly variable valve train (HVVT). This valve train uses two synchronously rotating cam disks instead of a camshaft and a control shaft. On the one hand, this makes it possible to vary valve lift and opening duration at the same time, identical to current variable valve trains, and therefore to realize functionalities like load control without additional throttling, or cylinder deactivation. Furthermore, a second lift event of the exhaust valve, which can be varied independently of the main lift, can be used to control the amount of residual gas rebreathed into the cylinder during the intake process. In addition to the control of the amount of fresh air delivered to the cylinder, it is also possible to influence the in-cylinder charge motion by varying the valve opening duration via a lift plateau. By means of engine process simulation, it could be demonstrated that with the residual gas adjustment through a second event, the residual gas quantity can be dosed very exactly. This provides a valuable tool for the efficient development of new combustion processes with high residual gas rates (e.g. HCCI or EGR-diluted combustion).

The methodology for putting such a complex valvetrain into reality has been developed for this specific purpose. It is structured into several subsequent steps and iteration loops for achieving the optimum solution:

The **kinematic layout** of the system provides the characteristic curves of the valve train as well as the geometric positions of the individual components. By means of a curvature analysis and an optimization of the characteristic curves which has already been integrated at this point, the dynamic behavior of the valve train can be improved even before the multi-body simulation. By this, the basic geometry of the valve train is defined.

In the **multi-body simulation**, the mechanical layout of the components is carried out in combination with an FEM simulation in order to evaluate the dynamic and elastic behavior. The advanced simulation model allows an estimation of the expected valve lift loss due to limited component stiffness, manufacturing tolerances and valve clearance compensation, which leads to a further reduction of optimization steps in the development process. In addition, the early integration of **gas exchange simulations** allows the evaluation of additional influencing factors in fired engine operation. With this knowledge, the valve lift curves can be optimized for real engine operation already at a very early stage.

Component testing is then carried out in the next step based on the so-called system cylinder head concept. By this approach, it is possible to determine the optimum geometry, the cam contours and the working curve of the valve train for a new cylinder head in advance and at low cost. Also, the installation space limitations can be considered at an early stage of development. In addition, the loads on the cylinder head can be determined and the cylinder head can already be dimensioned appropriately, and the engine speed limit can be detected without high risk of damages. Similarly, the system cylinder head allows to carry out fatigue tests or endurance runs with particularly low risk, since the modular design allows all components to be replaced quickly, and no expensive cylinder head prototype has to be exposed to potential damage.

By using a pressurized oil supply and an external drive of the valve train by an electric motor, close-to-reality operation can be achieved. Forces, drive torques and resulting valve lift curves are measured during operation and can thus be used to check the plausibility of the results from the kinematic simulation and the multibody simulation. As an example, the influences from real component deformation, manufacturing tolerances or valve clearance can be compared with the simulation models.

The simulation results from the kinematic simulation and the MBS can be plausibility-checked and compared with regard to valve lift losses resulting from real component deformation, manufacturing tolerances or valve clearance, for example. These experimental results can be used for the further optimization of the valve train.

Conclusions and Outlook

With the help of the above methodology, it is possible to determine the optimum geometry, cam contours and working curve of the valve train for a new cylinder head in advance and at low cost. The time required for the design and modification loops of a prototype valve train is significantly reduced with this approach. The results reached so far have shown that the knowledge gained from system cylinder head testing can be almost completely transferred to a complete engine cylinder head. This allows a highly realistic, adaptable, cost-effective and time-efficient design and testing of valve trains, and particularly of highly variable valve trains. The newly developed "HVVT" concept was successfully optimized using this new methodology. For the future, it is expected that it will become possible to use the experience gained from the system cylinder head tests in the multi-body simulation to finally carry out the valve train design almost completely by simulation.

Based on the methodology developed here, the HVVT concept for the fired engine will undergo further steps of development. Starting from the simplified system cylinder head valve train, this concept will be complemented by a unit for the automatic adjustment of the valve train. In addition to this, it is planned to integrate camphasers on the intake and exhaust side of the engine. This will give the valve train even more variability, which can be used perfectly well for detailed combustion process development on a single-cylinder research engine.

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Definitions/Abbreviations

BDC	Bottom Dead Center
CAD	Computer Aided Design
CFD	Computational Fluid Dynamics
degCA	Crank Angle in [°]
DI	Direct Injection
EDM	Electrical Discharge Machining
FEM	Finite Element Method
HCCI	Homogeneous Charge Compression Ignition
ICE	Internal Combustion Engine
MBS	Multi Body Simulation
TDC	Top Dead Center
TKE	Turbulent Kinetic Energy
sysco	System Cylinder Head Concept