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DESIGN CONSIDERATIONS OF OHV DIESEL ENGINE CAM PROFILE

ABSTRACT

The opening and closing angle of the valves, overlap, and maximum valve lift have a great effect on engine performance at a given speed. These parameters are controlled by the cam which consists of an area with a constant radius (base circle) and an area for valve lift (opening ramp, opening flank, cam tip, closing flank, and closing ramp). The maximum valve lift, opening and closing angles are given by the results of a thermodynamic analysis to achieve the target engine performance. Kinematic and dynamic analysis are required to determine opening and closing ramps, flanks and cam tip shape to obtain a definitive cam profile.

In this paper, general design considerations of cam profile for the OHV diesel engines has been studied and a 4-cylinder and 4-stroke diesel off road engine exhaust cam profile has been designed by using AVL Excite Timing Drive. Following typical dynamic problems have direct influence on durability of the valve train system especially on high cam speeds; excessive valve seating and cam contact forces being occurred due to high velocity and acceleration, cam contact loss between follower and cam, valve bouncing and undesired vibration behavior of the valve springs. These problems can be detected by dynamic analysis results and avoided with an ideal cam profile design.

Keywords: Cam Profile Design, Valvetrain, Dynamic Analysis, Excite Timing Drive

1. INTRODUCTION

Internal combustion engines use valves to control the air and fuel flow in/out of the cylinders. Valves are driven by the camshaft through the rocker arms, pushrods or tappet lifters according to type of the valve train system. Valve train systems are mainly subjected as 2 configurations by considering the location of the camshaft on engine layout which are Overhead Camshaft (OHC) and Overhead Valves (OHV). Figure – 1 shows the general layouts of the OHV and OHC valve train systems. Camshaft places in cylinder head of the engines on OHC configuration thus valves are driven in a more direct manner. Besides on OHV system the camshaft is placed within the cylinder block and uses pushrods or rods to actuate rocker arms through more number of the parts compared to OHC systems. Even though the cost and package advantages of the OHV engines, their valve train have lower stiffness and show much unsteady dynamic behavior due to high number of contact points and long distance for the transmission of the motion from camshaft to valves. Profile of the cam defines the theoretical valve motion however the actual valve motion does not occur in same geometry due to clearances, inertial and external forces and stiffness of the parts which is an actual conclusion of the dynamic behavior of the system. [1]

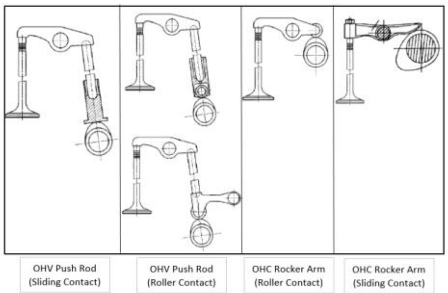

Figure 1: Rocker arm valve train configurations

Cam profile design is initiated with the kinematic calculations in order to provide a rough profile bconsidering given max valve lift from thermodynamic calculations when the right valve train geometry (cam base circle, rocker arm ratio, sufficient stiffness of cams, package constraints, material and min. grinding wheel diameter that confirmed by the supplier) has been defined. As next step, the single valve train model is built for the dynamic analysis.

2. VALVE KINEMATIC MOTION

There must be a clearance between the valve actuating mechanism of the rocker arm and the valve stem end due to heat expansion of the valves and cylinder head and wear between valve head and valve seats during the engine operation. If there is no clearance left, the valves would not close on base circle during the engine operation which can lead performance loss or catastrophic failure of the engine. Initially, when the valves start to move, the clearance between valve stem and rocker arm must be compensated. The valve clearance can be occurred either by turning and adjusting the bolts which is the connection between the pushrod and the rocker arm or with a hydraulic element which is closing the lash automatically. Diesel engines with OHV system are generally uses mechanical lash adjustments.

When the lash is closed, the cam profile initiates a constant velocity at first section of the lift as protect the system from the impact forces which is called as ramp velocity.

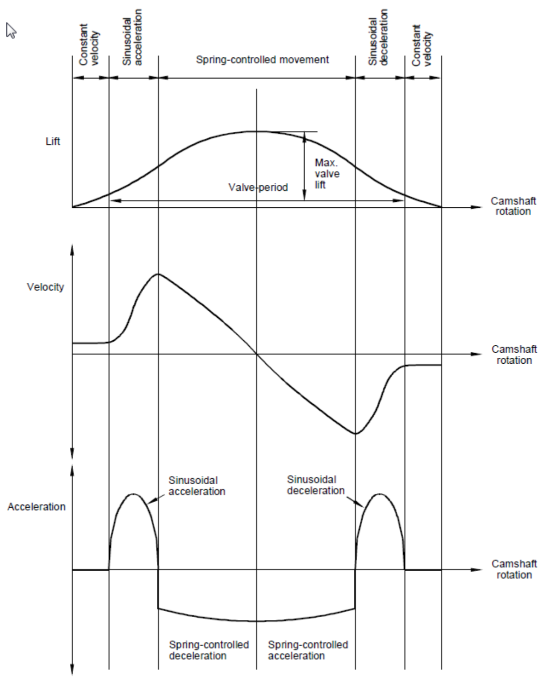

Figure 2. Theoretical valve motion

Under kinematic conditions the opening and closing ramp velocities are usually defined as between 300mm/s and 500mm/s for the diesel OHV engines at the rated engine speed. During the next stage the cam accelerates the valve which happens on the cam opening flank. The opening acceleration generally designs as immediate rise from zero to maximum and back to zero. Deceleration starts due to the spring stiffness is getting increased until the valve approaches the maximum lift. When the valve starts to close, acceleration is also controlled by the valve spring with the spring progressivity. Figure 3 an example of exhaust valve spring progressivity. Last, final deceleration (closing ramp) is controlled by the cam. The cam is designed to give a constant closing velocity in order to limit the impact stresses same as opening ramp phase**.** [1]

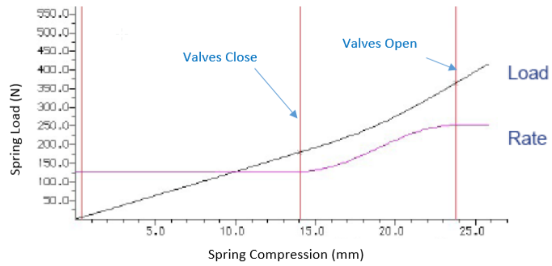

Figure 3. Valve spring progressivity

3. GENERAL DESIGN CONSIDERATIONS

3.1 Thermodynamic Input

The cam lobe must be able to open and close the valve as far, as fast and as smoothly as possible. Thermodynamic side of the cam profile design deals with far and fast behavior of the valve motion in order to get maximum performance from the given engine layout. Flow coefficient versus valve lift and inner valve seat diameter is a main criteria for max. valve lift calculation. Thermodynamic delivers a proposal valve lift curve to design side in order to initiate the cam design.

AVL Boost is a thermodynamic simulation software for the simulation of engine cycle and generation of initial proposal valve lift which simulates a wide variety of engines, 4-stroke or 2-stroke, spark or auto-ignited. Applications range from small capacity engines for motorcycles or industrial purposes up to large engines for marine propulsion. In this study desired valve lift shape has already been defined by engine cycle calculations using AVL BOOST. This valve lift curve and the corresponding timing for the valves will targets for the basic cam design.[2]

3.2 Cam Size Definition

There are three basic limiting factors for the cam size definition. Contact pressure between cam and follower determines the wear of the contact surfaces. There is a small contact area is being created through elastic deformation between cam and follower, thereby limiting the stresses considerably. This stress is called as Hertzian stress (contact stress). Maximum allowable Hertzian stress is limited depending on yield strength of the cam lobe or follower material and type of contact (point or line). Even though the contact stress is related with the velocity, lubrication and surface roughness of the mating parts, general cam size is directly affected by the maximum allowed Hertzian stress. Besides cam width over flank and nose is also depending on the material, follower type (flat, roller) and the contact stress limit. OHV engines generally use chilled cast camshaft and hardened steel flat tappets. This type of pair has typically 1000MPa Hertzian stress limit however detailed evaluation has to be done by simulation in combination with other parameters as lubrication etc.

As the cam size is reduced, another design parameter, radius of curvature is affected. Figure 4. shows a cam shape example with the negative cam contour radius. The cam surface radius of curvature will have a minimum allowable design value that will satisfy the other design factors. Generally, negative cam contour radius is defined by the manufacturer depending on tooling costs.

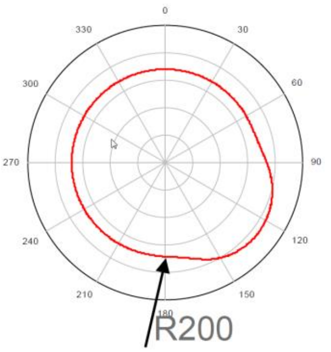

Figure 4. Negative cam contour

Shaft size is another limiting factor in determining the minimum cam size. The camshaft is subject to torsional vibrations. The shaft diameter is determined by the requirement for adequate torsional stiffness. The required shaft outside diameter is depending, whether the shaft is driving other components besides the valves. This is important if it is driving components with high torque fluctuations (such as unit fuel injectors) or components with high inertia.[3]

4. CAM PROFILE DYNAMIC CONSIDERATIONS

Valve train components must satisfy a number of dynamic requirements such as valve closing velocity, valve spring behavior, cam/follower contact stress and there is no loss of contact should be occurred between the frictionally coupled components up continuous over speed in order to keep the target valve train durability and avoid from the failures.

Excessive valve closing velocity which also refers as impact velocity has direct influence on valve/ valve seat wear and thermal fatigue cracks. So that it should be limited under a certain value on lower and higher speeds by the optimization of the cam profile. Typically max. Valve closing velocity is 0.5m/s on rated speeds for OHV diesel engines. It can be allowed bit higher on continues or non-continues over speeds.

Valve bouncing is another important parameter which should be avoided. This problem reduces engine efficiency and performance and potentially increases engine emissions. There are also significant risks to valve spring cracks and piston valve contacts.

Main functions of the valve spring are closing the valves with the potential energy stored during the valve opening phase by the force through the cam lobe and keeping the valves closed. On the other hand valve spring has key role to control the valve train dynamic forces. The spring forces has to be higher than the mass forces of moving parts to ensure permanent contact between cam and follower. Valve spring dynamic response (Forces) has to be checked in the entire speed range in 50rpm steps. It generally occurs on high engine speeds due to high valve closing acceleration and improper valve spring stiffness characteristics.

As mentioned in cam size definition, contact pressure is limited according to material pair of the parts between cam and followers. And maximum contact stress should be under allowable max contact stress limit in every operation speed of the engine. [4]

5. BASIC CAM DESIGN

In this study, a desired valve lift shape has already been defined by engine cycle calculations using AVL BOOST. This valve lift curve and the corresponding timing for the valves are targets for the basic cam design. Thermodynamic exhaust valve lift is used as an input for initial cam design whereas maximum valve lift has been defined as 10mm with considering the mechanical lash.

AVL EXCITE Timing Drive provides a tool for basic cam design and cam modification which is used for the current study. The basic cam design module is included in the timing drive software for preparation of a first cam profile layout at an early design stage. For this example an exhaust cam profile for a valve train with overhead valve is generated. Detailed model can be seen on dynamic analysis section.

Before designing the cam profile, it is necessary to prepare an equivalent system. This system should define the most important geometrical dimensions and some initial values for the following:

- Valve train stiffness which are calculated considering young modulus of the materials and the geometry of the components.

- The masses of the valve train components

- The initial properties of the valve spring which is already defined per max valve lift, exhaust back pressure and valve train geometry. Valve spring design is an iterative study. Spring designer uses the valve train dynamic analysis results again to reach the optimum spring properties.

Table 1. shows individual masses and stiffness's of the valve train components which are used for the cam design and dynamic analysis.

Table 1. Valve train masses

	mass [g]	stiffness [N/mm]
valve_ex_all	120.2	
valve_ex face	92.3	
valve_ex stem	31.1	128629
valve spring retainer	41.6	
valve collet pair	2.5	
valve spring non active coil mass	11.5	
half pushrod UPPER	46.7	744482
half pushrod LOWER	42.4	74448
flat tappet	140.2	4433415
rocker arm	186.8	
adjustment screw	11.1	
adjustment screw nut	3.5	
rocker arm incl. adj. screw, nut	200.5	

For the basic cam design, three principle methods are available:

- Polydyne cams

- Steady acceleration cams

- Section-wise description of the cam profile

The cam design is done interactively by specifying a section to be modified and giving a rule how the cam contour should be modified. Afterwards, modifications can be performed in lift, velocity, and acceleration curves at valve or cam considering the requirements. Polydyne method symmetric for both opening and closing side is used.

Ramps (opening and closing) of the cam profile should be determined accordingly to general cam shape design. Ramps are considered at the cam profile for all types of valve trains, they serve for achieving defined opening and closing behavior of the valve and compensate clearance and the stiffness of the valve train. Ramp definition is independent calculation other than cam design methods. So that, all available ramp description can be combined with all described design methods for main cams.

As this example is to prepare data for a mechanical valve train with a clearance of 0.25 mm at cold engine, a height of the opening ramp of 0.35 mm is considered.

Then according to stiffness, mass, valve train geometry and thermodynamic data, cam design is generated. Figure 5. shows, results of the kinematic valve lift, velocity and acceleration shape.

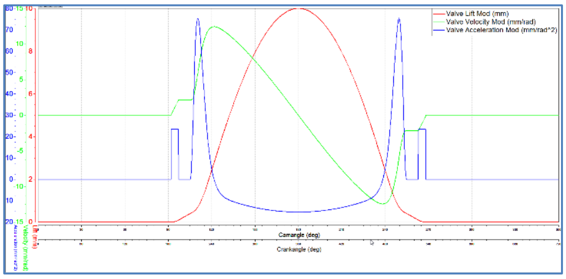

Figure 5. Kinematic valve lift, velocity and acceleration

6. DYNAMIC ANALYSIS

6.1 Model Description

After the basic cam design has been done, the according rough model is extended and refined to a model which can be used for the calculation of the dynamic behavior of the system.

In the current example, simple single OHV exhaust valve train actuated by pushrod and rocker arm with a mechanical tappet of a four cylinder off road diesel engine has been studied as described in figure 6.

In general it consists of:

- Cam

- Mechanical tappet

- Pushrod

- Rocker Arm

- Valve(head and stem)

- Valve spring

- Valve cotter and retainer

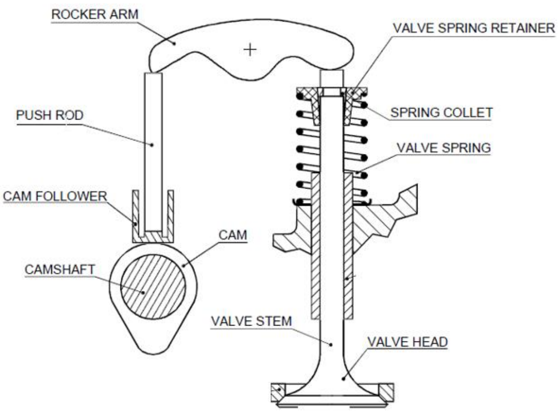

Figure 6. OHV Valve train layout

OHV system of a four cylinder off road diesel engine is modelled as a single valve train lumped mass system using AVL EXCITE Timing Drive as shown in the figure 7.

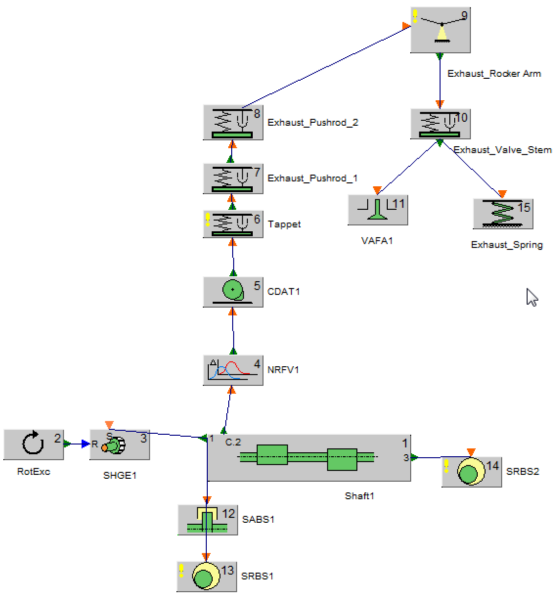

Figure 7. Valve train lumped mass model

The lumped mass model is generated by considering below road map;

- Divide the valve into two parts for the purpose of considering the stiffness and damping of the valve more precisely. The valve is usually “cut” in the middle of the stem. Thus, the following two parts are taken into consideration: Valve Stem and Valve Face
- Masses of the valve cotter, retainer, and the valve-side non-active coils of the spring(s) are also added to the pure stem mass as well when considering the valve stem mass.
- One element (spring model) represents the valve spring.
- Radial and axial bearings describe the connection (stiffness and damping) between a shaft or a gear with surrounding parts. In our example, the radial and axial bearings mainly represents the stiffness of the camshaft.
- Pushrod is considered with two stiffness and damping elements for more precise representation since its long height.
- Directly a rocker arm element is used for the exhaust OHV rocker arm.
- Flat tappet (cam follower) is described with single stiffness and damping element.
- Camshaft is modelled in shaft modeler tool as 2 bearing and single exhaust cam lobe. All the related geometries (cam lobe width, cam lobe base circle diameter, bearing width and diameter and camshaft length) are considered in order to describe the part.

The calculations are conducted for the 1800 rpm engine speed which equals to 900 rpm camshaft speed. Cylinder pressure, friction of the elements, stiffness and damping values derived from the part geometries and masses, dynamic oil viscosity between shaft and bearings, spring properties are used in the model.

The method programmed in the dynamic part of EXCITE Timing Drive calculates the dynamic displacements, velocities, accelerations, and forces in the valve train elements.

6.2 Results

After the calculation is finished the results files are evaluated according to valve train dynamic requirements. Initially, valve lift profile and valve velocity are compared at 1800 rpm. The dynamic valve lift curve can differ more or less from the kinematic curve, depending on the stiffness of the valve train. So that max valve lift has been observed as 9.91mm due to elastic deformation of the valve train parts without considering the valve mechanical lash. On the other hand valve lift is smoothly matching with the thermodynamic lift curve however it’s needed to be double check from the engine performance effect point of view.

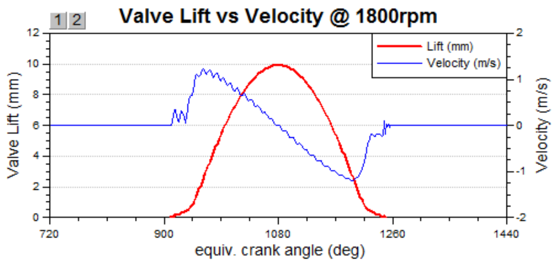

Figure 8. Dynamic valve lift and valve velocity results @ 1800 rpm

By the smooth design of the cam opening and closing ramps, it produced a relatively low valve closing velocity on 1800 rpm which is below than recommended value of 0.5m/s.

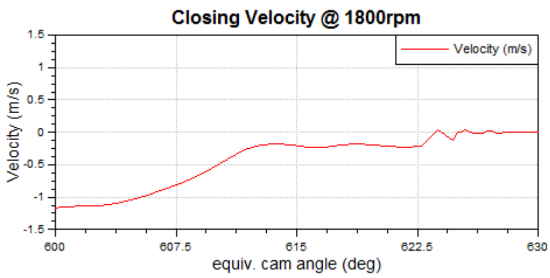

Figure 9. Valve closing velocity @ 1800 rpm

Dynamic spring shows no contact of spring coils according to comparison of the active coil forces.

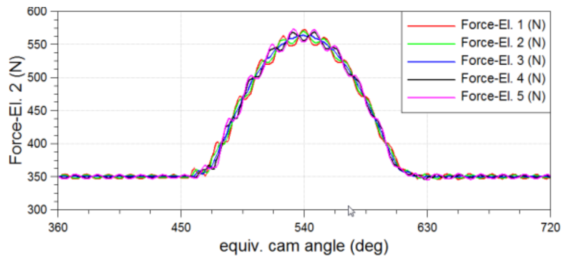

Figure 10. Spring Coil Forces

7. CONCLUSION

The single exhaust valve train of an OHV diesel engine is simulated by means of AVL EXCITE Timing Drive. The study shows that typical dynamic problems can be avoided by reaching to optimum design of the cam profile or other valve train components like rocker and spring.

The results of analysis are:

- Dynamic valve seat velocity and forces
- Dynamic valve lift
- Dynamic behavior of valve springs are satisfy the dynamic requirements of the valve train.

Finally, the analysis is performed on rated speed of the engine which is well below than continuous over speed (2600rpm). Dynamic behavior of the valve train shows severe results on higher engine speeds so that likely some refinements will be needed on cam design on further design studies for higher rpm's.

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