International Journal of Advances in Engineering and Management (IJAEM)
Volume 4, Issue 12 Dec. 2022, pp: 572-580 www.ijaem.net ISSN: 2395-5252

Analytical examination of the valve control system of a high-speed internal combustion engine

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Submitted: 10-12-2022 Accepted: 23-12-2022

ABSTRACT: The valve train of internal combustion engines are under continuous development. In order to further improve the efficiency and fuel consumption of the current constructions the limiting factors need to be evaluated. Whether the valve train of the examined engine permitsto achieve better parameters can be defined using parameters derived from the valve lift profile. The examination of the results revealed that in the intake tract already is in need for an improved alternative valve system.
Keywords:internal combustion engine, valve lift profile, fuel consumption, intake valve, engine parameters

I. INTRODUCTION

The gas exchange process of all four-stroke, internal combustion engines is controlled by valves. Today, these engines exclusively use poppet type valves, with disc-shaped heads. They are opened by the cams on the camshaft.A common way to close or keep the valves closed is to arrange one ormore springs around the valve stem. These pull the valve sealing face onto the seat, thus ensuring a gas-tight seal. The system is simple and the sealing force exerted by the spring is assisted by the pressure of the gaseous medium to be sealed. At the same time, it is unfortunate that the valve head, which so practically closes the gas exchange openings of the engine when necessary, significantly limits the gas exchange process itself. In addition to flow problems, each construction has additional limitations. We will examine these areas in detail.

II. THE ENGINE USED IN OUR PROJECT

The engine chosen for the analysis is the power source of a Suzuki SV650 motorcycle manufactured in 2003 (Figure 1), whose technical parameters are listed in Table 1.

Since our later goal is to transform the gas exchange control system, our choice is primarily justified by the fact that it is a widespread, popular type, for which any spare parts can be easily obtained. It has a 2-cylinder layout, that ensures easier accessto engine internals than compared to a car engine. It is also relatively easy to work on, which is quite an important aspect when we are thinking on a conversion. It has one spark plug per cylinder, which allows more general usability of the results. Although it is not a top-of-the-line sports bike, Table (2) contains some interesting data. It also includes the parameters of the Cosworth CA F1 racing car engine. The data of the section highlighted in yellow shows

Table (1):parameters of a 2003 Suzuki SV650 engine

Engine configuration:	4 Stroke,90 degree V2	Valve actuation:	DOHC with bucket type follower
Number of valves per cylinder:	4	Intake / exhaust valves:	2/2
Geometric compression ratio:	11.5:1	Cooling system:	liquid
Fuel supply:	Mikuni port injection	Throttle body inner diameter:	39 mm

International Journal of Advances in Engineering and Management (IJAEM)
Volume 4, Issue 12 Dec. 2022, pp: 529-534 www.ijaem.net ISSN: 2395-5252

That the maximum piston speed of the engine chosen for our analysis is only 3.82% lower than that of the engine representing the technical peak of naturally aspirated, non-hybridized engine of motorsports. This demonstrates that the parts of our motorcycle engine can withstand much higher loads than their current peak performance. Therefore, we can conclude that the SV650 has enough mechanical reserve to withstand a major overhaul of its control system and the resulting increased loads without damage.

Table (2): Comparison of engine parameters of the Cosworth CA Formula 1 engine and the Suzuki SV650 engine

Parameter	Cosworth CA F1	Suzuki SV650
Bore:	98 mm	81 mm
Stroke:	39,77 mm	62,6 mm
Swept Volume:	2,4 liter	0,645liter
Distance between connecting rod small and big end centrelines:	102 mm	120 mm
Specific power:	231.2 kW/liter	79.8 kW/liter
Peak torque engine speed:	16 000 rpm	7500 rpm
Peak torque:	320 Nm	62 Nm
BMEP at peak torque:	16,76 bar	12,08 bar
Mean piston speed at peak torque:	21,21 m/s	15,65 m/s
Peak power engine speed:	17 250 rpm	9000 rpm
Peak power (kW):	564,50 kW	51,46 kW
BMEP at peak power:	16,36 bar	10,64 bar
Mean piston speed at peak power:	22,87 m/s	18,78 m/s
Peak piston speed at peak power:	38,42 m/s	31,55 m/s
Maximum allowed engine speed:	18 000 rpm	11000 rpm
Mean piston speed at peak engine speed:	23,86 m/s	22,95 m/s
Peak piston speed at peak engine speed:	40,08 m/s	38,55 m/s
Difference in peak piston speeds:	2.9%	

III. DETERMINING VALVE LIFT PROFILE AND ANGULAR CROSS SECTION VALUES

A fundamental element of the valve control system is the cam and its lift profile. This can be determined in the traditional way using a dial gaugewitha protractor disc with the camshaft fixed between centres or placed in wedges. The measurement can be performed using the same principle but utilizing electronic equipment to record corresponding lift and camshaft rotation values. In order to further increase accuracy a DEA Mistral - Slant Bridge Technology coordinate measuring devicewas used (Figure 1) to record cam profile. Table (3) contains the data of the equipment used to perform the necessary measuring activities. When designing the valve lift profile, the goal is to completely free the valve flow cross-section in the shortest possible time, keep it fully open for a specified time, and then close just as quickly as it opened (Figure 2). The ideal valve lift "curve" has been best approximated so far by two-stroke engines with rotary disc intake control. On a four-stroke engine this valve lift profile can only be achieved by using alternative gas exchange control systems. Poppet valve systems cannot tolerate such an abrupt opening and closing characteristics. In order to achieve the theoretical maximum valve opening with cam-operated valves, infinitely light components would be needed that can withstand infinite loads without any deformation, while they must not wobble during force changes, and their surface would not be damaged due to Hertzian stress. Obviously, these criteria cannot be met by any structural material or assembly. Even though the cam profileslift the valves seemingly gradually and maximum engine speed is limited by the ECUto 11,000 rpm, forces in the valve train become enormous at this speed. Normally, the motor only operates at this rpm for a very short time, but this speed must be taken into account in the calculations as the maximum loadpossible.The

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recorded lift profile can be divided into several sections: the task of the opening ramp is to eliminate the valve lash, while the closing ramp ensures that the valve closes onto the valve seat at a certain specified speed. If the valve clearance is too large then the valve will rise from its seat too suddenly inducing violent oscillations in the entire mechanism.

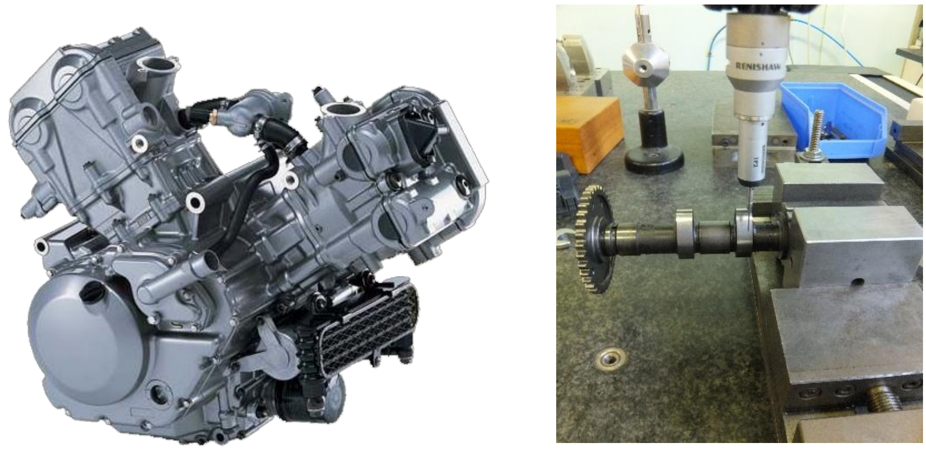

Figure (1): Suzuki SV650 motorcycle engine and its camshaft measurement on a DEA Mistral coordinate measuring machine

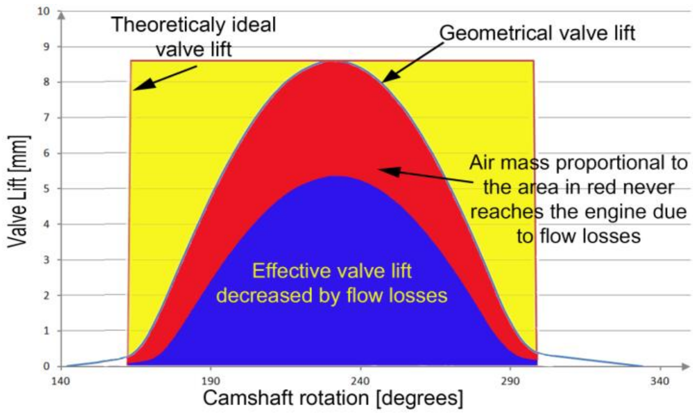

Figure (2): The relationship between the achievable and feasible opening angular cross-section (angle area) in the main opening range

Table 3. Properties of the equipment used to measure the cam and valve springs

Cam measuring device	Name:	DEA Mistral - Slant Bridge Technology
	Individual ID:	707057
	Type of measuring head:	Renishaw PH10 M
	Head diameter	1 mm
	Longitudinal transducer (790 mm):	RSF Elektronik
	Type:	MSA6707
	Resolution:	0,1 µm.
	Individual ID:	3204227601009
Spring testing device	Name:	INSTRON 3367 tensile testing equipment
	Individual ID:	3367K5426

	Force transducer:	INSTRON 500N
	Individual ID:	61638

The exact same too large valve lash causes the valve to raminto the seat when closing, as the valve hits its seat at a higher speed than designed. A small valve gap is no less dangerous: during a slower opening-closing process, the exhaust valve suffers a greater heat load and may burn out, but in the case of the intake valve, the fresh mixture may be contaminated with exhaust gas or "escape" from the cylinder into the intake port. Using the factory-specified valve clearance eliminates these harmful phenomena. Between the ramps is the main opening period during which the majority of gas exchange process actually takes place (Figure 3).

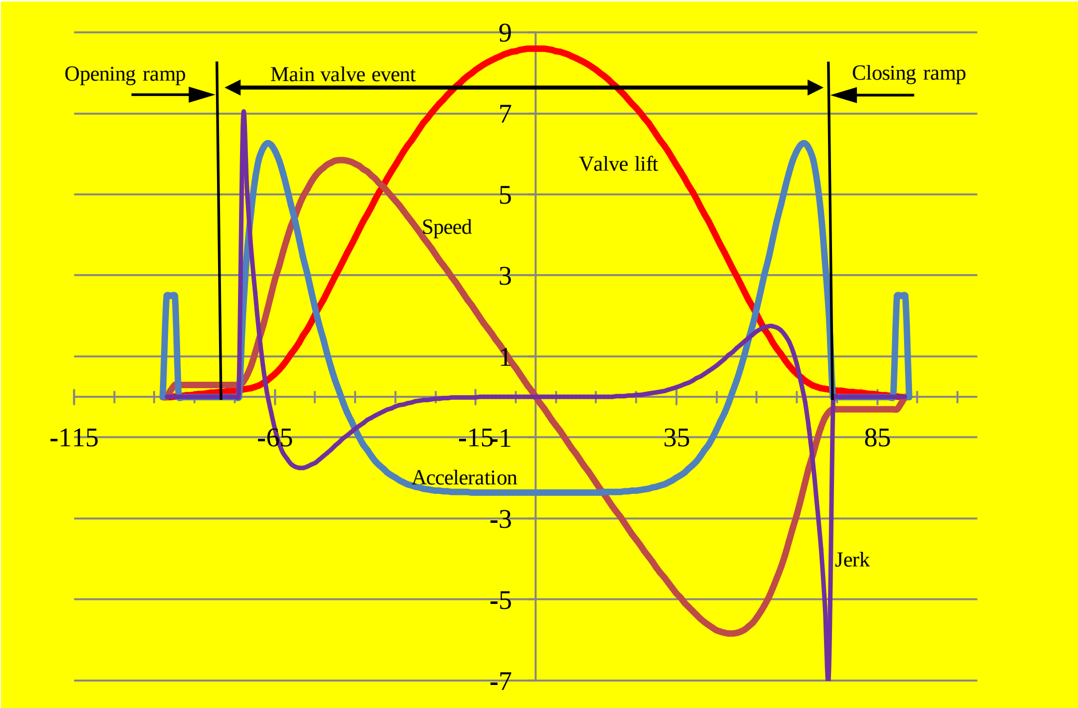

Figure (3): SV650 intake valve opening profile and the derived parameters.

Due to the aforementioned mechanical limitations of the valve control system, a smaller cross-section is available for gas exchange compared to the ideal condition. Since the measurement of the lift profile f(φ) results in a set of discrete points, the area under the curve can be determined by numerical integration. Due to the measuring system and the principle of measurement, the smallest step interval (h) can be 1 degree of camshaft rotation. Inside this interval the other points of the valve lift profile are not known. It follows that the value of the integral can be best approximated by the trapezoidal formula (Figure 4):

$$\int_{\varphi_A}^{\varphi_B} f(\varphi)d\varphi = \sum_{\varphi_A}^{\varphi_B} h_i \frac{f(\varphi_i)+f(\varphi_{i+1})}{2} \qquad (1)$$

$$h_i = \varphi_{i+1} - \varphi_i$$

i=1…N-1,

Where:

–φ_A, φ_B:The full valve opening interval [degrees]

–N: Number of measuring points,

–φ_i: Camshaft rotation degrees in the i-th measuring point ($\varphi_1=\varphi_A$,$\varphi_N=\varphi_B$)[degrees],

–h_i: Camshaft rotation step size, its value is not constant[degrees].

After performing the necessary approximate calculations in the main opening range, the angular cross-section exposed by the intake valve is 721.69 degrees mm, while for the exhaust valve it is 565.72 degrees mm. Thus, the intake valve uses 57.1% of the theoretically available angular cross-section, while the exhaust uses

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55.58% (Blair, 1999). In the case of the intake valve, this is only slightly below the average value of 58-60% used for sports engines, but the difference is quite large on the exhaust side. So there is room for improvement in this area, the question is: can it be implemented?

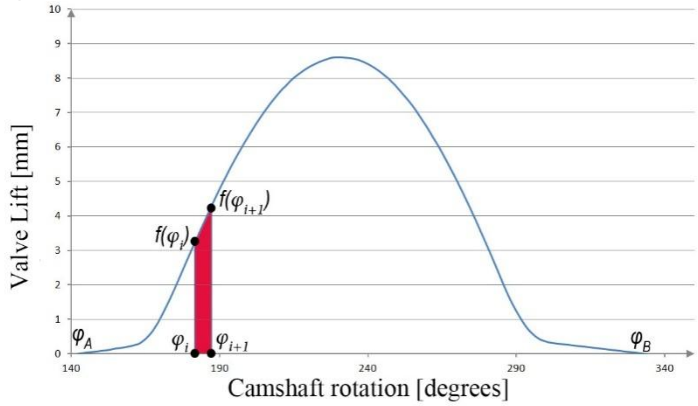

Figure (4): Geometrical definition of angular cross section integral (Angle Area)

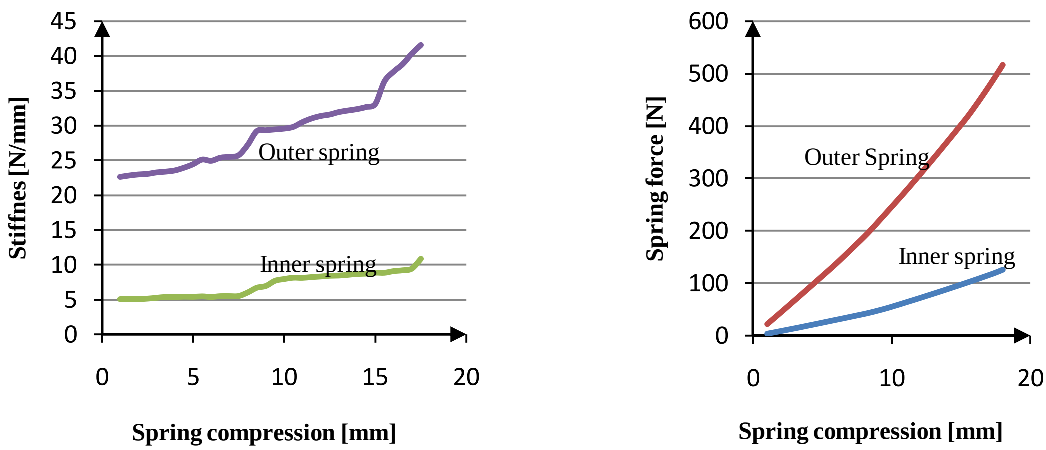

Figure (5): Characteristics of the valve springs of the tested engine

To decide this, it is necessary to check whether it is possible to increase the valve lift to a higher value than the current one. This may be limited primarily by the valve springs, which were tested using an INSTRON 3367 material testing machine with a 500 N test head. The technical data of the equipment are contained in Table (3), while the characteristics determined as a result of the measurement are shown in Figure (5).

There are two springs around the valve stems, their winding direction is opposite. When comparing the data and the physical dimensions of the springs, it was determined that the springs of the intake and exhaust valves can be considered identical. They have progressive characteristics, the inner spring has two stiffness ranges, while the outer one has three. A comparison of the installed length, the size for the largest valve lift and the fully compressed height revealed that a higher valve lift on the intake side cannot be achieved with the vehicle's original springs. At highestlift the inner spring comes close to the fully compressed state by only 1 mm. On the exhaust side, due to the lower maximum valve lift, the internal valve spring leaves 2.4 mm from the fully compressed position. In both cases, the external springs are farther from the fully compressed state: 1.3 mm on the intake side, and 2.7 mm on the exhaust valve. From the above, it follows that the lift height cannot be increased on the intake side, but on the exhaust side it can still be increased by 1.4 mm. Provided that the range of the valve opening cannot be greater than the original, the only way to increase the Angle Area for the given valve is to move it faster during the lifting period. Thus, it reaches the fully open or closed state sooner. In this way, the actual curve can be closer to the ideal

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opening characteristics and the angular cross-section can be increased.

IV. DETERMINING VALVE LIFT SPEED

The method of increasing the angular cross-section described in the previous section can only be implemented if the higher valve speed is made possible by the corresponding structural elements. Since the tested engine is an inverted bucket type lifter construction, it must be determined whether the diameter of the available valve lifter allows faster valve lift than the current one. This is because if the valve movement relative to the diameter of the lifter is too fast, the edge of the valve will grind into the side of the cam.

In the same manner as the angular cross-section has been determined, instead of the continuous profile curve made up of infinite number of points only a set of discrete measuring points was obtained on the displacement profile curve to calculate the valve velocity (v_v). Therefore, the velocity function cannot be calculated with differentiation applied in the case of continuous functions, only an approximate solution could be obtained by using a central difference coefficient with a second-order error term (Kalmár-Baranyi, 2006):

$$v_v = \frac{l_{i+1} - l_{i-1}}{\varphi_{i+1} - \varphi_{i-1}}, \qquad (2)$$

Where:

–v_v: Valve speed in the i-th measuring point[mm/degrees],

–l_i: Valve lift at the i-th measuring point [mm],

–φ_i,:Camshaft position [degrees].

After completing the calculationwith the belonging camshaft position and valve lift point pairs, the maximum speed of the intake valve is ±0.1975 mm/degree, 52 camshaft degrees before and after the highest valve lift. The exhaust valve achieves a maximum speed of ±0.18625 mm/degree, 48 camshaft degrees before and after the highest valve lift. As can be seen, the valve speed is not a real speed, but a displacement per angular rotation measured in degrees.

V. DEFINING ECCENTRICITY

The speed of the valve at any moment is equal to the speed of contact point P in the direction of the valve axis. The discrete values of this were determined in the previous section, but this is not enough to check the necessary diameter of the valve lifter. For this, the distance q of the contact point P located on the cam circumference must be taken into account, measured from the valve axis (Figure 6). This is called eccentricity and is defined as follows. Let du be the circumferential displacement of point P of the cam, which can be calculated as follows from the distance r of point P of the cam from the axis of rotation:

$$du = r\,d\theta. \qquad (3)$$

Its component that points in the direction of the valve stem:

$$dl = du\ \sin(\theta). \qquad (4)$$

Substituting eq. (3) into eq. (4) we get:

$$dl = r\sin(\theta)d\theta. \qquad (5)$$

The eccentricity can be defined as follows:

$$q = r\sin(\theta). \qquad (6)$$

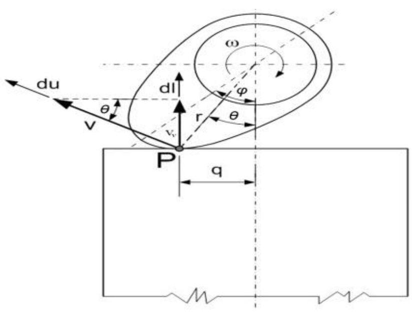

Figure (6): Geometrical representation of instantaneous eccentricity

Substituting eq. (6) into eq. (5) we get the following solution:

$$dl = q d\theta. \qquad (7)$$

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The speed $v_v$of the valve is the displacement relative to the angular rotation of the camshaft φ, so the relationship between the eccentricity q and the valve speed, after matching the measurement units, is described by the following relation, using the change of the angles θ and φ together (dθ=dφ):

$$v_v[mm/deg] = \frac{dl[mm]}{d\varphi\ [deg]}. \qquad (8)$$

After substituting the maximum valve speed, the maximum value of the momentary eccentricity that occurs when the cam profile rotates around is 11.315 mm for the intake cam, while it is 10.67 mm for the exhaust cam.

VI. VERIFYING THE LIFTER DIAMETER

In the tested engine the lifter diameter is 26 mm, the centre lines of the intake cams are offset by 1 mm inwards on their respectivelifters, while the exhaust cams are located 1 mm apart from the centre lines of the lifters. The cam width on both camshafts is 9 mm. From these data, according to equation (9), the safety distance (C_{mg}) that remains between the closest point of the cam and the edge of the lifter at the moment of the highest valve speed (Figure 7) can be determined (Wang, 2007):

$$C_{mg} = \frac{D_{tk}}{2} - \sqrt{\left(O_{fs} + \frac{w}{2}\right)^2 + q_{max}{}^2} \quad (9)$$

Where:
–C_{mg}: Safety distance between the edge of the lifter and the edge of the cam [mm],
–D_{tk}: Lifter diameter [mm],
–O_{fs}: Offset between cam and lifter centre lines [mm],
–w: Width of cam [mm].

After substituting and performing the calculation, it was determined that the intake cams get really close to the edge of the lifters at a distance of 0.62 mm, while a healthier margin of 1.21 mm is left for the exhaust cams. Since the minimum permissible distance is 0.6 mm, it can be seen that the current construction already fully utilizes the available lifter diameter on the intake side, but there are still possibilities to exploit on the exhaust side.

VII. DEFINING VALVE ACCELERATION

After verifying valve lifter diameter, we defined the static valve acceleration. For this, the speed of the valve must be differentiated, using the central difference methodologyas already shown in eq. (2):

$$a_{fvi} = \frac{v_{vi+1} - v_{vi-1}}{\varphi_{i+1} - \varphi_{i-1}}, \qquad (10)$$

Where:
–a_{fvi}: Instantaneous valve acceleration [mm/deg^2].

After differentiating the speed values, it can be observed that acceleration has a unit of mm/deg^2. In order to facilitate the calculation of forces acting upon the valve the variable of camshaft rotation unit of degrees need to be transformed to the time domain using the below equation: (Rothbart, 2004):

Angular speed of the camshaft:

$$\omega\left[\frac{deg}{s}\right] = \frac{d\varphi}{dt}. \qquad (11)$$

The speed of the valve during lift:

$$v_{v,t}\left[\frac{m}{s}\right] = \frac{dl}{dt} = \underbrace{\frac{dl}{d\varphi}}_{v_v\left[\frac{m}{fok}\right]} \cdot \underbrace{\frac{d\varphi}{dt}}_{\omega\left[\frac{fok}{s}\right]} = v_v \cdot \omega. \qquad (12)$$

Utilizing that ω is constant, the speed change of the valve in the time domain:

$$\begin{aligned} a_v\left[\frac{m}{s^2}\right] &= \frac{d^2 l}{dt^2} = \frac{d}{dt}\frac{dl}{dt} = \frac{d}{dt}v_v = \\ &= \frac{d}{dt}\left(\frac{dl}{d\varphi}\omega\right) = \frac{\omega}{d\varphi}\left(\frac{dl}{d\varphi}\omega\right) = \qquad (13) \\ &= \omega^2\frac{d^2 l}{d\varphi^2} = \omega^2 a_{fv}\left[\frac{m}{deg^2}\right]. \end{aligned}$$

The relation between the original a_{fv} measured in units of mm/degand a_v in m/s^2:

$$a_v\left[\frac{m}{s^2}\right] = \frac{1}{1000}\omega^2 \cdot a_{fv}\left[\frac{mm}{deg^2}\right]. \qquad (14)$$

Thus, at the aforementioned 11,000 revolutions per minute, the acceleration of the intake valve is 13,643 m/s^2, and the acceleration of the exhaust valve is 16,790 m/s^2

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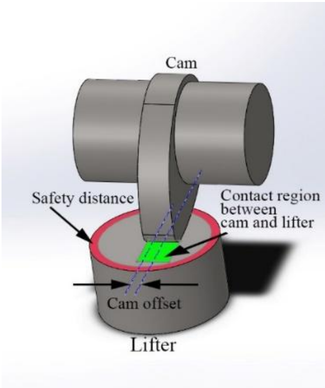

Figure (7): Position of the safety distance on the edge of the inverted bucket type valve lifter

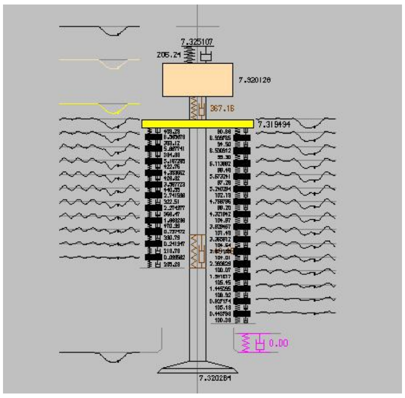

Figure (8): Dynamic model of the valve, the inner (on the right) and the outer (on the left) valve springs and the camshaft

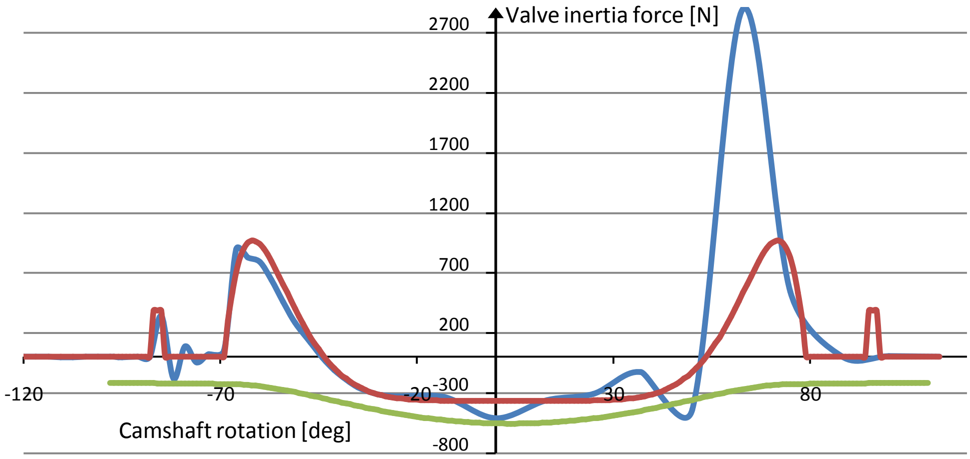

Figure (9): Comparison of the inertia forces of the dynamic valve acceleration (blue), the inertia forces of the static valve acceleration (purple) and the spring force (green) at 11,000 rpm

VIII. DYNAMIC MODEL

Unfortunately, the least rigid elastic element of the valve control system is the valve spring.With the help of the software, we determined the dynamic acceleration characteristic, and then the inertia forces were also calculated from the data obtained in this way. Figure (9) shows the comparison of static and dynamic forces. It is clearly visible that due to the dynamic oscillations the springs are not able to keep the valves in contact with the cam after the largest valve opening, in the camshaft rotation range of 44-51 degrees. Thus, the valves regain contact with the cam surface like a hammer blow, which subjects the components to excessive stress, surface wear and fatigue. Therefore, with the OEM valve springsthe tested power plant cannot be used at engine speeds higherthan the factory predefined rpm.

IX.CONCLUSIONS

As it became clear from the above, the poppet valve design has a general disadvantage: with a certain valve acceleration characteristic and spring natural frequency the valve can no longer follow the trajectory determined by the cam profile. From the calculations and analysis, it can be seen that in the case of the Suzuki SV650, the manufacturer has practically fully exploited the structural possibilities of the traditional DOHC valve control system. It would only be possible to change the valve lift profile on the exhaust side, but this would have yield relatively little benefit. A step forward could be the use of the implementation of an alternative gas exchange system. With this, the desirable, ideal valve opening profile described in Section III. can be approached much better.

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