10.14311/mecdc.2021.01.01

CRITICAL SHIFTING WINDOW IN SWITCHABLE ROCKER FINGER FOLLOWER

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ABSTRACT
A valvetrain including switchable rocker finger follower is capable of discrete switching between two modes (two cam profiles). The exact moment when switching occurs is called crossover point and this paper reviews the factors that cause the shift of the crossover point from its nominal design position. The range where crossover point can shift is called critical shifting window and its size and factors influencing it will be adressed.
KEYWORDS: CAM, CAM PROFILE, CAM DESIGN, SWITCHABLE ROLLER FINGER FOLLOWER, TOLERANCES, STACK UP, SHIFTING WINDOW, CAE

SHRNUTÍ
Ventilový rozvod s přepínatelným vahadlem s rolnami je schopen přepínat mezi dvěma režimy (přepínání mezi dvěma vačkovými profily). Okamžik, kdy dojde k přepnutí mezi jednotlivými vačkami, se nazývá bod přechodu. V tomto příspěvku budou uvedeny jednotlivé faktory, které způsobují posun bodu přechodu z jeho jmenovité návrhové pozice. Celý rozsah kam se může bod přechodu posunout je označován jako okno bodu přechodu a v příspěvku bude probráno jak jednotlivé faktory ovlivňují jeho velikost.
KLÍČOVÁ SLOVA: VAČKA, PROFIL VAČKY, NÁVRH VAČKY, PŘEPÍNATELNÉ VAHADLO S ROLNAMI, TOLERANCE, TOLERANČNÍ ANALÝZA, OKNO PŘECHODU, CAE

1. INTRODUCTION

Valvetrain mechanism between camshaft and a valve itself allows to transform camshaft rotational movement to the intake and exhaust valve translational movement. The conventional and simplest valvetrain operation allows the fresh air or air-fuel mixture to enter the cylinder during the intake stroke when intake valves are open, participate on combustion and let the combustion products leave the cylinder during exhaust stroke when exhaust valves are open. But as demands on engines increase and fulfilling prescribed emission limits is more and more challenging new technologies and innovation are being used. The valvetrain is no exception and variable valve timing (VVT) and variable valve lift (VVL) are used in vehicles nowadays. Cam phaser is the most common way for VVT implementation. It allows to shift the entire valve lift within the specified range of an engine cycle and it appears in two versions – discrete and continuous timing switching. Switching between different cams is used for the VVL realization. The axial camshaft shifting or switching the cam that controls the valve using advanced finger followers or rocker arms is used by OEMs. Combination of VVT and VVL is commonly called as variable valve actuation (VVA). Different VVA systems used by OEMs are usually called by their marketing name such as VTEC, VANOS, MultiAir, MIVEC etc. Camless valvetrains are the most variable solution but they are used mainly in experimental and research engines so far [1]. More on the topic of VVA can be found in the following publications – [2], [3], [4]. The switchable roller finger follower (SRFF) is one of the ways how to implement discrete variable valve lift. [5] That means it allows to switch between two different valve lifts. The crossover point is the moment when switch is realized, thus the moment when the valve changes cam lobe which prescribes its lift. The principle of SRFF will be explained followed by the thorough description of the critical shifting window, how it is created and influence of the specific factors on the window size.

2. SWITCHABLE ROLLER FINGER FOLLOWER VALVETRAIN

The conventional valvetrain system with a standard roller finger follower shown in Figure 1 is often referred to as Type II valvetrain. It consists of a camshaft that acts on a roller finger follower through its roller. The roller finger follower is in contact with pivot on one side and valve stem on the other side. Improvement of such a system by replacing the roller finger follower by its switchable version (Figure 2) enables to switch between two different lifts on one valve. It allows to switch for example between normal mode and Miller cycle on the intake side.

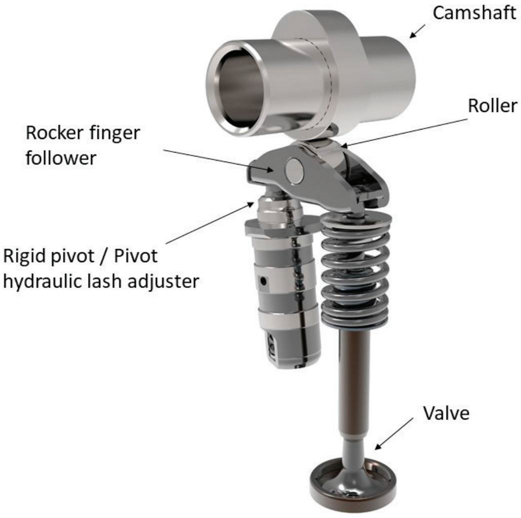

FIGURE 1: Type II valvetrain
OBRÁZEK 1: Ventilový rozvod typ II

The same thing could be applied to the exhaust side where normal exhaust valve lift can be supplemented by small extra lift during the intake stroke, which allows to get some of the exhaust gases entering back to the cylinder and this is often referred to as internal exhaust gas recirculation (iEGR). SRFF can be used for cylinder deactivation or other advanced valve actuation strategies.

Inside the SRFF there is a latch pin (Figure 3) and depending on its position the finger follower responds to the inner roller. When the pin is not latched the inner roller of SRFF makes so called lost motion. On the other hand, when the pin is latched the entire SRFF and thus also the valve reacts on the movement of the inner roller. To be able to perform two different lifts with SRFF valvetrain system a camshaft must have 3 cam lobes per SRFF (Figure 4). Two outer cam lobes are identical and act on outer rollers of the SRFF,

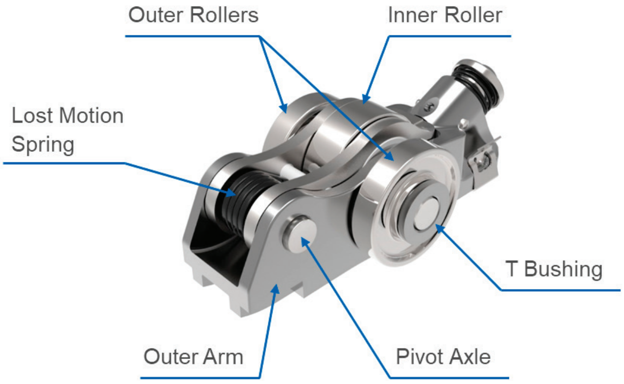

FIGURE 2: Switchable rocker finger follower (SFRR)
OBRÁZEK 2: Přepínatelné vahadlo

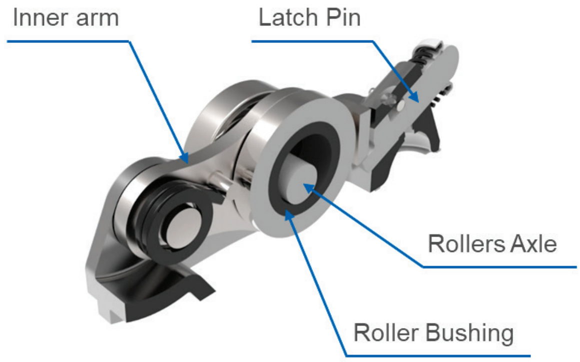

FIGURE 3: SRFF section
OBRÁZEK 3: Řez vahadlem

FIGURE 4: SRFF cam lobes
OBRÁZEK 4: Vačky pro přepínatelné vahadlo

the inner cam lobe acts on inner roller which is connected to the inner arm and can either perform a lost motion or transmit the cam lift into the valve lift. Function of SRFF valvetrain when the pin is not latched is as follows. On the base circle the outer cam lobes are in contact with outer roller (no lash is present because a hydraulic lash adjuster is often used). The lash between the inner cam lobe and the inner roller is present and is called mechanical lash at cam (MLC). As the camshaft rotates the valve lift is influenced only by outer cam lobes. During the camshaft rotation there is a moment when inner cam lobe gets in contact with inner roller and as MLC gets closed the impact on inner roller appears. The lift is not transferred from the inner cam lobe to the valve as pin is not latched and inner arm makes lost

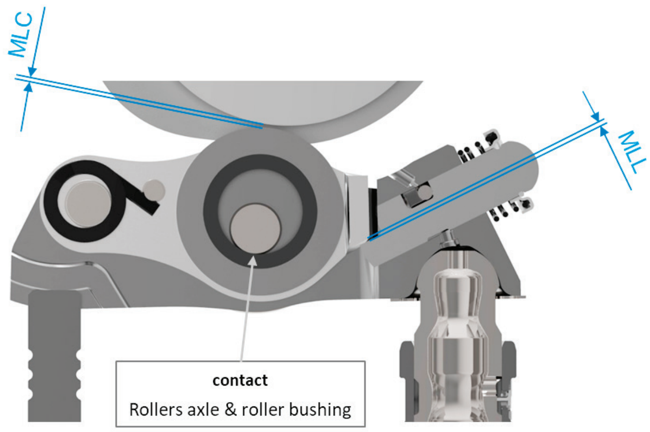

FIGURE 5: Mechanical lashes in the SRFF
OBRÁZEK 5: Vůle v přepínatelném vahadle

motion. When pin is latched the situation in the beginning is similar. Outer rollers are in contact with cam lobes, MLC is present and there is also lash between latch pin shelf and inner arm mating surface which is called mechanical lash at latching pin (MLL). During the camshaft rotation the MLC is closed first, then the inner arm starts to move and MLL is closed. At this moment the valve lift is no more controlled by the outer lobe profiles and starts to be controlled by the inner lobe profile instead. This moment is considered as the crossover point. Very similar conditions and phenomena as during crossover point happen when MLC is closed so further in the article it will be adressed as a crossover point 1 (CP1) and the actual crossover point when MLL is closed as crossover point 2 (CP2). As the lift of the inner cam lobe decreases back to the base circle, the MLL is opened first and outer cam lobes get in contact with outer rollers and valve is again controlled by the them. Further as the inner cam lobe lift goes back to base circle the MLC is opened.

3. APPROACH

GT-Suite is a CAE toolset widely used in industry especially in the automotive as it has many useful features for simulation of the specific parts of the vehicles and engines. It is capable of 1-D flow simulation, kinematics, MBD etc.. Two parts of this complex software package were used in order to examine influence of various factors on width of critical shifting window. The GT-ISE where libraries for valvetrain and multibody dynamics were used and VTDESIGN where cam profiles were designed, and kinematics of the system was examined. In general, when designing cam profile, it is important to control cam velocity and acceleration. Too high velocity during opening and closing ramps results in excessive impacts in the system which result in increased wear or higher failure probability. Acceleration is controlled in order to avoid contact separation in the valvetrain. A separation could happen when inertia forces are higher than force generated by a valve spring. Acceleration has a direct influence on manufacturability as with the high acceleration the concave radius of curvature of the cam decreases. If cam concave radius is smaller than the grinding tool, it will be impossible to grind some areas on the profile. Specific limit values are usually set by internal company guidelines and are often treated as business secret. More about process of developing the cam profile can be found in [6]. The MBD model of the single valve mechanism including SRFF was built in GT-ISE in such a way that position of various components in the valvetrain can be quickly and easily changed which allows to implement manufacturing tolerances and wear of the specific parts in the system. VTDESIGN was used to design cam lobe profiles which are then input in the MBD model. Initially the simulation was performed with all the nominal dimensions and baseline cam profiles thus perfectly fulfilling the moment when CP1 and CP2 were intended to happen based on the specific requirements on the function of the valvetrain and engine. Furthermore, the position of the components was changed in order to simulate influence and sensitivity of moment CP1 and CP2 on manufacturing tolerances and other aspects that will be discussed later in appropriate chapters followed by the interpretation of the results and conclusion. The main motivation for keeping critical shifting window small is because only in this area the crossover point can happen thus only here the velocity difference needs to be controlled. If the CSW is too wide the velocity difference needs to be kept sufficiently small for a long time period and that results in restrictions for cam design of inner and outer cam lobe profile.

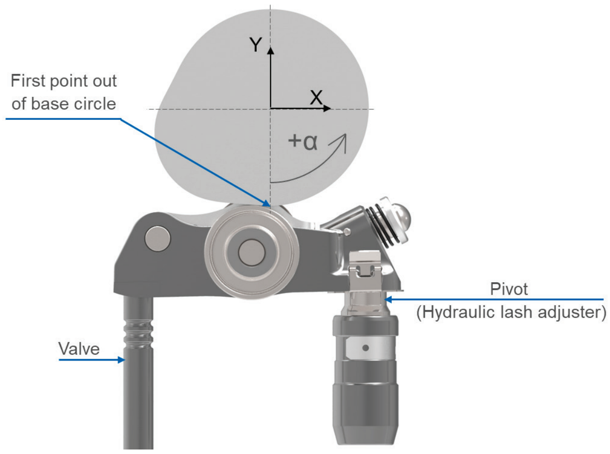

FIGURE 6: Simulation initial state
OBRÁZEK 6: Počáteční stav simulací

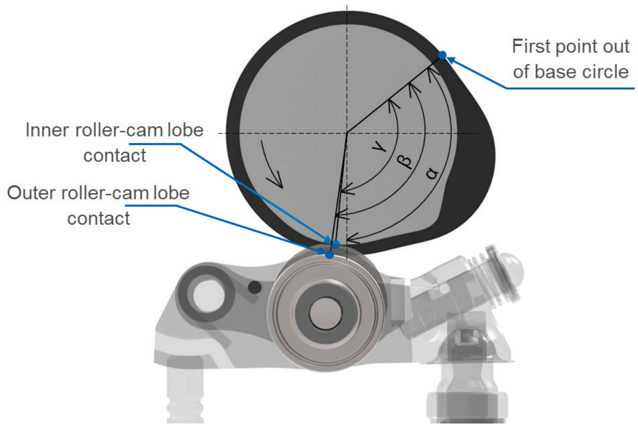

FIGURE 7: Crossover point angles definition
OBRÁZEK 7: Definice úhlů pro bod přechodu

4. BASELINE

The MBD simulation using all the nominal dimensions was performed in order to set the nominal position of the CP1 and CP2. The critical shifting window will be created around those values as different factors will be changed in the following chapters. It is also important to set up the initial position of every simulation and derived angular positions of contacts during the CP. Every simulation performed has the same layout as specified in Figure 6. From the point of view, the valve is on the left and pivot on the right, the camshaft rotates counterclockwise and all the cam lobe first points (first point that is higher than cam base circle) lies on the global negative Y axis. As the simulation time goes forward the camshaft rotates and angles α, β, γ can be observed as in Figure 7. α is the angle between negative Y axis and the cam lobe first point and gives us the information about the timing. It tells when the CP happens. β is the angle between first cam lobe point and the contact point between outer cam and roller. It gives us the information about where on outer cam profile does the CP happen. γ is very similar to β but it goes from the first point of the cam lobe to the contact of inner cam and roller.

All three angles will be used in description of the critical shifting windows. Big deviation in α signs that the function of the inner profile lift can cause not desired influence on the engine cycle as the prescribed CP can happen too early or too late. Angles β and γ gives us the information where is the contact point on the cam when the CP happens. It is important as it gives us the information about the impact that appears in the system during CP. In order to realize CP, the velocity on the inner cam lobe has to be higher than on outer cam lobe so the lashes will get closed. But the velocity difference has to be limited so the strong impacts will not damage and wear the components and cause the system failure.

TABLE 1: Nominal crossover points
TABULKA 1: Nominální přechodové body

Baseline CP1			Baseline CP2		
α [deg]	β [deg]	γ [deg]	α [deg]	β [deg]	γ [deg]
87,8	91,3	91,1	101,9	102,7	101,9

TABLE 2: Baseline relative velocity difference during crossover point
TABULKA 2: Výchozí relativní rozdíl rychlostí na vačkách v přechodových bodech

Baseline CP1 vel. difference [mm/deg]	Baseline CP2 vel. difference [mm/deg]
0,0075	0,0075

Relative velocity difference is calculated as described in equation (1) and (2).

$$v_{diff@CP1} = v_{inner}(\gamma_{CP1}) - v_{outer}(\beta_{CP1}) \tag{1}$$

$$v_{diff@CP2} = v_{inner}(\gamma_{CP2}) - v_{outer}(\beta_{CP2}) \tag{2}$$

Baseline design angles α, β, γ are in the Table 1 and relative velocity difference inTable 2.

CP1 happens when camshaft rotates 87,8° from the initial position and rollers are in contact at 91,3° of outer cam lobe and 91,1° of inner camlobe. CP2 design moment is at 101,9° rotation after initial state. Outer cam lobe is in contact with roller at 102,7° of its profile and inner cam lobe at 101,9° of its profile. Critical shifting window will be created around those nominal values. Same cam lobe profiles will be used if not mentioned otherwise.

5. SRFF TOLERANCE FACTORS

In an ideal case every product going from the same production line would be identical. But in practice even if material goes through same prescribed set of operations there is always some deviation in dimensions or material properties thus the final products have some level of variation. But that does not necessarily mean that the function is affected. Setting the tolerances for manufacturing processes limits the deviation in final products in a way that desired function is assured. But setting the tolerance limits has its other side as well. The tighter are the deviation limits the more accurate thus more costly steps and processes must be utilized. It is always extremely important to find a compromise between the price and tolerance levels. All the component variations are taken in account in so called stack-up analysis to see if the desired function is assured. The stack-up analysis is not the object of interest in this article thus it will not be described in detail what is the cause of position change. Only the stack--up analysis results of parts that affects the critical shifting window will be used. Some of the cases that are discussed are artificially created but it helps to distinguish what is the real factor that moves a crossover point. It can be observed for example in first case where x position of outer rollers is changed. In real scenario the resulting change in position of outer roller would be caused by changed position of the outer roller axis and as this axis is in contact with bushing of the inner roller it would naturally change the initial position of inner roller and size of MLC. For sake of clarity and simplicity let's consider cases where only one specific position is changed and rest stays in its nominal position.

5.1 OUTER ROLLERS POSITION

Influence of roller position tolerance was examined in 9 cases prescribed as in Figure 8– 1 nominal position and then 8 positions of the outer roller axis on the circle with radius of 0.03 mm.
Results are in Table 3 and it can be seen that values for α (the angle describing the timing) go from 84,5° to 91° for CP1

TABLE 3: Results for different outer rollers position
TABULKA 3: Výsledky pro různé pozice vnějších rolen

Outer roller position tolerance		CP1			CP2		
X tol [mm]	Y tol [mm]	α [deg]	β [deg]	γ [deg]	α [deg]	β [deg]	γ [deg]
0	0	87,8	91,3	91,1	101,9	102,7	101,9
-0,03	0	86,4	90,4	90,0	100,4	101,5	100,6
-0,02121	0,02121	88,8	92,1	91,8	102,9	103,7	102,8
0	0,03	90,8	93,6	93,3	105,1	106,0	105,0
0,02121	0,02121	91,0	93,8	93,5	105,1	105,9	105,0
0,03	0	89,3	92,4	91,2	103,4	104,1	103,2
0,02121	-0,02121	86,7	90,3	90,2	100,9	101,8	101,1
0	-0,03	84,7	88,7	88,6	99,2	100,4	99,7
-0,02121	-0,02121	84,5	88,6	88,5	99,0	100,2	99,5

and from 99° to 105,1° for CP2. If it is considered that 1° of cam angle rotation corresponds to 2° of crank angle (CA) rotation the shift of the CP1 in engine cycle can be shown. CP1 can happen 6,6° CA before or 6,4° CA after the designed moment and anywhere in between. CP2 can happen 5,8° CA before or 6,4° CA after the designed moment and anywhere in between. In the next chapters the result description will not be as detailed as here, but only table with results and critical shifting window expressed by the range of α will be mentioned.

5.2 INNER ROLLER POSITION

The same strategy as in the previous chapter was used and 9 cases were simulated including nominal position and 8 axis offset positions on a circle around the nominal position (Figure 9).

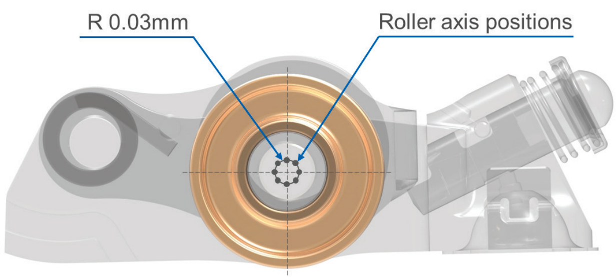

FIGURE 8: Examined outer rolers axis positions
OBRÁZEK 8: Zkoumané pozice osy vnějších rolen

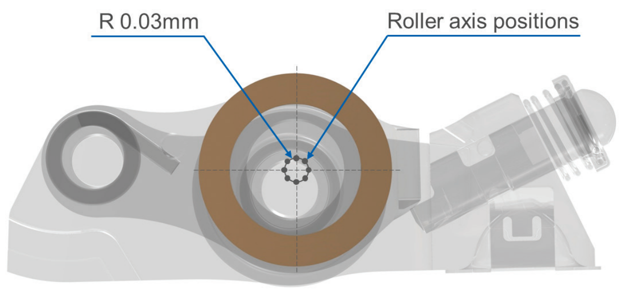

FIGURE 9: Examined inner rolers axis positions
OBRÁZEK 9: Zkoumané pozice osy vnitřních rolen

TABLE 4: Results for different inner rollers position
TABULKA 4: Výsledky pro různé pozice vnitřních rolen

Inner roller position tolerance		CP1			CP2		
X tol [mm]	Y tol [mm]	α [deg]	β [deg]	γ [deg]	α [deg]	β [deg]	γ [deg]
0	0	87,8	91,3	91,1	101,9	102,7	101,9
-0,03	0	89,3	92,5	92,2	103,4	104,1	103,3
-0,02121	0,02121	86,6	90,3	90,2	101,0	101,9	101,1
0	0,03	84,7	88,8	88,7	99,2	100,4	99,7
0,02121	0,02121	84,5	88,6	88,5	99,0	100,2	99,5
0,03	0	86,4	90,1	89,9	100,5	101,5	100,6
0,02121	-0,02121	88,8	92,1	91,8	102,9	103,7	102,7
0	-0,03	90,7	93,6	93,3	105,1	105,9	104,9
-0,02121	-0,02121	91,0	93,8	93,5	105,1	105,9	105,0

Critical shifting window influenced only by the inner roller position goes from 84,5° to 91° in terms of α for CP1 and from 99° to 105,1° for CP2.

5.3 LATCH-PIN SHELF TOLERANCE

When referring to the latch-pin shelf tolerance, the position of surface compared to nominal position as shown in Figure 10 is meant. As this dimension is not anyhow involved during the CP1 its influence only on CP2 will be examined.

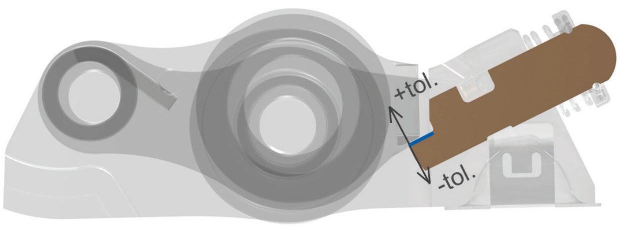

FIGURE 10: Latch-pin shelf tolerance
OBRÁZEK 10: Tolerance obrobení plochy přepínacího čepu

TABLE 5: Reults for latch-pin shelf tolerance
TABULKA 5: Výsledky pro tolerance plochy přepínacího čepu

Pin tolerance	CP2		
[mm]	α [deg]	β [deg]	γ [deg]
0	101,9	102,7	101,9
-0,03	100,0	101,0	100,3
-0,02	100,5	101,5	100,7
-0,01	101,2	102,1	101,3
0,01	102,6	103,4	102,5
0,02	103,4	104,1	103,2
0,03	104,1	104,9	103,9

The critical shifting window for CP2 in term of α can go from 100° to 104,1° due to the latch-pin shelf tolerance.

6. CAM LOBE TOLERANCES

The same as for SRFF is valid for the cam lobe profiles. The tolerances that are taken in account here are cam profile tolerance, wear and cam profile angular tolerance. Profile tolerance is easy to understand as it means that the designed cam profile can be either higher or lower by the specified value. Wear is captured by adding higher value to the negative side of the profile tolerance so the actual cam profile can be lower than the nominal partially because of manufacturing and partially due to wear over the time. Cam angular tolerance means that the cam lobe profile can be shifted relatively to the other cam lobe. In the baseline case, both cam lobes have their first profile point in the direction of negative Y direction but in reality, the profiles can be shifted to each other due to angular position tolerance

6.1 OUTER CAM LOBE PROFILE TOLERANCE

Four cases were tested including again the nominal dimension, then two cases for ±0,03 caused by the manufacturing and then case -0,06 where the half of the value is caused by the manufacturing and half by the wear of the cam lobe. Thus the tested cases and critical shifting window are not symmetrical. Each case results are Table 6.
Critical shifting window influenced only by outer cam profile tolerance and wear goes from 80,3° to 91° in terms of α for CP1 and from 96,4° to 105,0° for CP2.

TABLE 6: Results for outer cam profile tolerance and wear
TABULKA 6: Výsledky pro profilovou toleranci a opotřebení vnějších vaček

Outer cam profile tolerance	CP1			CP2		
[mm]	α [deg]	β [deg]	γ [deg]	α [deg]	β [deg]	γ [deg]
0	87,8	91,3	91,1	101,9	102,7	101,9
-0,06	80,3	85,2	85,2	96,4	98,0	97,5
-0,03	84,5	88,6	88,5	99,0	100,2	99,5
0,03	91,0	93,8	93,5	105,0	105,9	105,0

6.2 INNER CAM LOBE PROFILE TOLERANCE

The same cases as in previous chapter were tested for the inner cam lobe tolerances. Results are in the Table 7.
Critical shifting window influenced only by inner cam profile tolerance and wear goes from 84,5° to 91° in terms of α for CP1 and from 99,3° to 105,1° for CP2. The trend is here opposite to the

TABLE 7: Results for inner cam profile tolerance and wear
TABULKA 7: Výsledky pro profilovou toleranci a opotřebení vnitřní vačky

Inner cam profile tolerance	CP1			CP2		
[mm]	α [deg]	β [deg]	γ [deg]	α [deg]	β [deg]	γ [deg]
0	87,8	91,3	91,1	101,9	102,7	101,9
-0,06	93,9	96,1	95,6	108,5	110,9	110,0
-0,03	91,0	93,8	93,5	105,1	105,9	105,0
0,03	84,5	88,6	88,5	99,3	100,4	99,7

tolerances of outer cam lobe. The higher is the inner cam profile the earlier happen both crossover points while at the outer cam lobe the higher is the profile the later the crossover points occur.

6.3 OUTER CAM LOBE ANGULAR TOLERANCE

Changing the relative angular position of the cams means shifting the profile timing thus changing all the cam lobe characteristics including lift, velocity and other higher derivatives. It is important to check if the relative velocity difference during CP do not exceed the prescribed guideline limits so the impacts in the system are controlled. Results for outer cam lobe angle are in Table 8.
Critical shifting window influenced only by outer cam angular tolerance goes from 83,2° to 93,9° in terms of α for CP1 and from 95,9° to 108,5° for CP2.

TABLE 8: Results for angular tolerance of outer cam lobes
TABULKA 8: Výsedky pro úhlovou toleranci vnějších vaček

Outer cam angular tolerance	CP1			CP2		
[deg]	α [deg]	β [deg]	γ [deg]	α [deg]	β [deg]	γ [deg]
0	87,8	91,3	91,1	101,9	102,7	101,9
-0,5	83,2	88,0	87,5	95,9	98,1	97,1
0,5	93,9	95,7	95,7	108,5	109,5	109,3

TABLE 9: Results for angular tolerance of inner cam lobe
TABULKA 9: Výsledky pro úhlovou toleranci vnitřní vačky

Inner cam angular tolerance	CP1			CP2		
[deg]	α [deg]	β [deg]	γ [deg]	α [deg]	β [deg]	γ [deg]
0	87,8	91,3	91,1	101,9	102,7	101,9
-0,5	93,4	95,7	95,7	108,0	109,5	109,3
0,5	83,7	88,0	87,5	96,4	98,1	97,1

6.4 INNER CAM LOBE ANGLE POSITION TOLERANCE

Same cases as prescribed in previous chapter were tested for inner cam lobe angular tolerance. See the results in the Table 9. Critical shifting window influenced only by inner cam angular tolerance goes from 83,7° to 93,4° in terms of α for CP1 and from 96,4° to 108,0° for CP2. It can be observed that the trends are similar as in cam lobe profile tolerance – shifting outer cam lobe angular position clockwise (+0,5°) cause CPs occur later on the other hand shifting the inner cam lobe same direction causes that CPs occur earlier.

7. WORST CASE SCENARIO

After the examination of each factor influence to the position of CPs the overall impact of all should be added together and see how it can influence the moment of CP1 and CP2. In reality such a case is highly improbable and statistical approach should be applied so the tolerances are not set too strict only for highly improbable combinations. See the results in Table 10.

TABLE 10: Worst case scenario results
TABULKA 10: Výsledky pro kombinaci nejhorších možných tolerancí

Worst case superposition	CP1			CP2		
	α [deg]	β [deg]	γ [deg]	α [deg]	β [deg]	γ [deg]
Beginning of CSW	63,8	72,0	71,6	78,2	83,9	83,1
End of CSW	110,5	112,4	113,3	120,3	126,0	127,5

It can be observed that due to manufacturing tolerances set as prescribed in the previous chapters the CP1 can happen anytime from 63,8 ° to 110,5° and CP2 from 78,2° to 120° in terms of α. Critical shifting window is 46.7° wide for CP1 and 42,1°wide for CP2. Such a width of CSW and level of uncertainty when does the CP happen might not be sufficient for some applications so in the next chapter there will be ways how to influence the the width of CSW.

8. CRITICAL SHIFTING WINDOW ADJUSTMENTS

There are two ways how to adjust the width of CSW. First way is very obvious, and it consists of making tolerances tighter. For our case the tolerances were halved. It can be considered that for roller tolerances the more accurate machine was used to drill the holes in SRFF, and more precise turning was used for rollers. That would result in roller's axis lying in circle of radius 0,015mm around its nominal position. Same applies for cam tolerances,

TABLE 11: Results for worst case scenario with half tolerances
TABULKA 11: Výsledky pro kombinaci nejhorších možných polovičních tolerancí

Worst case with half tolerances	CP1			CP2		
	α [deg]	β [deg]	γ [deg]	α [deg]	β [deg]	γ [deg]
Beginning of CSW	75,0	81,1	80,8	89,3	92,6	91,9
End of CSW	101,2	101,9	101,5	114,7	119,2	119,8

furthermore the better material in terms of wear would be used so the peak wear decreases to 0,015mm thus cam profile tolerance would go from -0,03 mm to +0,015 mm around nominal value and cam angle tolerance ±0,25°. Influence on CSW is in Table 11.
The improvement is significant and CSW for CP1 goes from 75° to 101,2° and for CP2 from 89,3° to 114,7° in terms of α. Then width of the CSW is 26,2° for CP1 and 25,4° for CP2.
Another way how to make CSW tighter is the adjustment of cam design and its velocity specifically. Tolerance deviation is basically increasing or decreasing the initial size of the lashes (MLC, MLL) compared to nominal, which has to be closed. The relative velocity difference tells us how quickly get those lashes closed around CP. Adjusting cam design in a way that position of CP stays the same but relative velocity difference is higher will result in closing the lashes with their deviations faster and

TABLE 11: Results for worst case scenario with half tolerances
TABULKA 11: Výsledky pro kombinaci nejhorších možných polovičních tolerancí

Worst case with half tolerances	CP1			CP2		
	α [deg]	β [deg]	γ [deg]	α [deg]	β [deg]	γ [deg]
Beginning of CSW	75,0	81,1	80,8	89,3	92,6	91,9
End of CSW	101,2	101,9	101,5	114,7	119,2	119,8

TABLE 12: Higher relative velocity difference for new inner cam
TABULKA 12: Vyšší relativní rozdíl rychlostí pro novou vnitřní vačku

Higher CP1 vel. difference [mm/deg]	Higher CP2 vel. difference [mm/deg]
0,0103	0,0132

TABLE 13: Results for new inner cam with higher relative velocity difference
TABULKA 13: Výsledky pro novou vnitřní vačku s vyšší relativní rychlostí

Higher cam velocity difference	CP1			CP2		
	α [deg]	β [deg]	γ [deg]	α [deg]	β [deg]	γ [deg]
Beginning of CSW	68,5	75,9	75,7	84,7	89,2	88,6
End of CSW	102,3	102,6	102,6	116,2	120,9	121,8

so decreasing the influence of tolerances on CSW size. The new inner cam profile was designed wither higher relative velocity difference (Table 12) and its influence on CSW size is in Table 13. The results show the size of CSW can be decreased by proper cam design as well. In this case increasing relative velocity difference for CP1 from 0,0075 mm/deg to 0,0103 mm/deg decreased the size of CSW by 12,9° in terms of α. With velocity difference increase from 0,0075 mm/deg to 0,0132 mm/deg for CP2 the CSW was decreased by 10,6° in terms of α. Increasing relative velocity is not for free as well, since the higher the difference is the higher is the impact that appears in the system during CP. The advantage of making CSW tighter has to be compared with disadvantage of possible higher wear or necessity of using better material.
Last case in this article will be the combination of two adjustments made above. The results for case where tolerances have the half size compared to the worst case and the relative velocity difference is as in Table 12.

TABLE 14: Results for half tolerances and higher relative velocity difference
TABULKA 14: Výsledky pro poloviční tolerance a vyšší relativní rychlost mezi vačkami

Worst case with tighter tolerance and higher velocity difference	CP1			CP2		
	α [deg]	β [deg]	γ [deg]	α [deg]	β [deg]	γ [deg]
Beginning of CSW	79,9	85,1	84,8	92,5	95,1	94,4
End of CSW	96,1	97,6	97,2	110,2	112,3	111,5

Critical shifting window is 16,2° wide for CP1 and 17,7° wide for CP2 in terms of α.

9. CONCLUSION

Concept and principle of critical shifting window was explained and influence of various factors on its size was examined. Detailed study of each factor was performed and based on results the following can be stated. The presence of critical shifting window is inevitable, and its size is prescribed by the manufacturing tolerances and design of a cam lobe profile during CP. Adjustments to the size of CSW can be done either by making manufacturing process more accurate or by increasing the relative velocity difference at cam lobes during the CP. The disadvantage of more accurate manufacturing process is the higher cost. The information about the actual tolerance classes, tolerance-based assembly and the trade-off between cost and CSW width is usually considered as a business secret and it is extremely difficult to reach to such information. It is important to compare the brought advantage for the increased cost. For example, if improving production process of the camshaft would bring the same benefit as improving the accuracy of rollers position but the cost is rapidly higher for the camshaft then focusing on SRFF manufacturing process is the way to go to. The increased relative velocity difference has also its disadvantage because the higher is the velocity difference the higher are the impacts in the system and higher wear can occur. The influence of the tolerances to a valve lift change and to the engine breathing was not the area of interest for this paper but as the values of tolerances are in hundredths of millimetres it is expected to have minor or almost no influence to the engine performance.

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SYMBOLS AND ACRONYMS

CA	crank angle
CAE	computer aided engineering
CP	crossover point
CSW	critical shifting window
iEGR	internal exhaust gas recirculation
MLC	mechanical lash at cam
MLL	mechanical lash at latching pin
OEM	original equipment manufacturer
SRFF	switchable roller finger follower
VVA	variable valve actuation
VVL	variable valve lift
VVT	variable valve timing
$v_{diff@CP1}$	relative velocity difference at CP1
$v_{diff@CP2}$	relative velocity difference at CP2
$v_{inner}(\gamma_{CP1})$	velocity on inner cam lobe at contact point during CP1
$v_{inner}(\gamma_{CP2})$	velocity on inner cam lobe at contact point during CP2
$v_{inner}(\beta_{CP1})$	velocity on outer cam lobe at contact point during CP1
$v_{inner}(\beta_{CP2})$	velocity on outer cam lobe at contact point during CP2
α	rotation angle of camshaft from initial state
β	angle between cam profile first point and contact point at outer cam profile
γ	angle between cam profile first point and contact point at inner cam profile