Design and Optimization of 2-stage Variable Valve Actuation Mechanism for Diesel Engines

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Abstract — *The desire for higher fuel economy, improved performance and drivability expectations of customers from engines are gradually increasing along with stringent emission regulations set by the government. There is customer demand for 4 wheelers having good power, torque and better fuel economy throughout the speed range of vehicle and an implied environmental need of improved emission characteristics. Variable Valve Actuation (VVA) has been applied to many engines in order to enhance the engine performance. Many engine manufacturing companies have started the application of variable valve actuation mechanism in their next generation vehicles. The VVA is a generalized term used to describe any mechanism or method that can alter the shape or timing of a valve lift event within an internal combustion engine. There are various ways to improve to the performance of engine some of which are; supercharging, turbocharging, variable compression ratio, variable intake system geometry, variable valve timing and lift etc. In this work we have concentrated on variable valve timing and lift for diesel engines. This work presents a novel two-step VVA mechanism to facilitate variation in valve timing and lift of base engine. Thus this mechanism helps to divide the operating speed range of engine into two zones i.e. low speed and high speed zone and setting a switch over point, thus helping the engine breath effectively.*

Keywords - *Variable Valve Actuation, Valve Lift, Speed Zone, Valve Event.*

I. INTRODUCTION

The reserves of diesel and gasoline fuels are ever decreasing, which plays an important role in the technological development of automobiles. The demands on combustion engines continue to grow. On one hand, customers want more power and torque; while on the other hand, one cannot lose sight of fuel economy and increasingly stringent emissions laws. Another important area in engine research is the implementation of new technologies like Variable Valve Timing (VVT), Variable Compression Ratio (VCR), Variable Intake System, Variable Geometry Compressor, and Exhaust Gas Recirculation (EGR), to improve engine performance by enhancing Combustion efficiency. The multiplicity of types of VVA systems [1][4][5] and their functions in internal combustion engines is well documented. This is particularly so for gasoline engines, with phasing system finding widespread applications [4]. The applications and benefits of these systems are well known and have been thoroughly investigated

However, the application of VVA to diesel engines is not as well understood or documented, although some work has been published on the application of VVA to highly rated engines [1] [2] and [3], concentrating on the control of overlap, and the investigation of briefly opening the intake valve during the exhaust stoke to generate internal EGR[5].

Variable valve actuation systems are significantly, more useful for gasoline engines to improve the overall engine performance. For diesel engines, there are restriction to the wide range application of variable valve actuation because of the clearance between the piston and valves at TDC. This clearance plays a very important role during the functioning of valve closing and opening timing of the engine cycle. The variable actuation systems is useful for gasoline engines to reduce the pumping loses as compared to the diesel engines.

But for the diesel engines it is useful for reduction of exhaust emissions such as NOx by using internal EGR.

The lack of work in this area can probably be attributed to two factors: firstly to meet the requirements, the VVA systems are more complex than current production systems and secondly significant changes have occurred in light duty diesel engine configuration in recent years: turbochargers have become almost universal, the use of intercoolers and EGR has become widespread, and most recently common rail and other fuel injection systems offering very high injection pressure and multiple shot or shaped injection characteristics are becoming the norm.[1]

The variable valve actuation mechanisms provide two lift profiles. These two lift profile system have a set of cam lobe profiles for low-to medium and medium-to high speed range and switch over point is obtained. Arrangement is made for switching between the two cam lobe profiles. The one cam lobe profile is designed for low speed zone. The other cam lobe profile is independently designed for high speed zone. Such two lift profile mechanisms have been used by vehicle manufacturers for many years and these systems have shown fuel economy and improvements in performance and emission.

The discrete two-step VVA systems can be a substitute for various continuously variable systems due to the relative ease of application for a variety of valvetrain types. Overall, the optimization technique yields a balanced system that satisfies vehicle requirements for fuel economy, emissions and performance. The two-step VVA systems are useful to engine manufacturers because they can be utilized for a variety of VVA strategies using common system architecture. Thus there is substantial flexibility to tune engine characteristics for high performance as well as sport/luxury applications because of the ability to reconfigure the VVA system.

II. THE ROLE OF VVT IN GASOLINE AND DIESEL ENGINE

A major goal of engine manufacturers is to minimize specific fuel consumption and emissions from engines. One solution is by independent actuation of the inlet and exhaust valves at any position of the piston.

A major disadvantage of conventional SI engines results from the energy loss due to the inhaling of the sub-atmospheric gases during the suction stroke and expelling of exhaust gases during the exhaust stroke. These pumping losses depend upon the opening and closing position of the throttle valve. The losses are high when the throttle valve tends to close and are low at wide-open throttle. Thus, the pumping losses are inversely proportional with the engine load. Without a throttle valve, control of the air-fuel mixture can be realized by variation of the intake valve-opening period; therefore, the VVT has great potential for reducing pumping losses in gasoline engines.

At low speeds, the pumping losses in gasoline engines are much greater than those of diesel engines because of the throttle intake system. In diesel engines, control of load is obtained by regulating the quantity of fuel injected. Unlike gasoline engines, diesel engines do not have a throttle to control the air-fuel mixture. Thus, due to the absence of the throttle valve, pumping losses at part load are much less. Also, the application of a turbocharger at low speeds reduces the pumping losses by providing air boost.

The compression ratio is a very important parameter in diesel engines. For high speeds as well as at cold starting, the diesel engine needs a high compression ratio. Due to this high compression ratio, the clearance between the piston and valves, at TDC, is very small. Thus, this is one of the mechanical constraints that must be considered in the control of intake and exhaust valves. In the case of medium speed diesel engine, the compression ratio is not as high as in high-speed diesel engines. These engines are used in marine applications, rail transportation, and power generation sets, where exhaust emissions is of main concern.

Several benefits of VVT when applied to diesel engines have been realized in recent years. One of the major benefits is the reduction of NOx by manipulation of exhaust valve timing. Also, improvements in torque and volumetric efficiency could be gained by varying the intake valve timing.

The fixed valve events for conventional cam controlled engines compromises the engine for better performance under all operating conditions. The inlet valve timing is the most important parameter for optimizing the engine volumetric efficiency, whereas the exhaust valve timing controls the residual gas fraction (RGF), which reduces exhaust NOx emission. RGF can be controlled by the valve overlap and can be changed for various speeds and loads by the application of VVT. To operate the engine efficiently and effectively over its entire operating range and conditions, the valve events should be able to vary with speed and load anywhere on the engine map.

III. METHODOLOGY

The base line engine consists of 4-in line cylinder with 2 valves per cylinder. Following are the engine specification used for the modelling of the baseline engine in GT-Power.

TABLE I

ENGINE SPECIFICATION

Engine capacity	3.12 liter
Power	55 KW@2000rpm
Torque	255 Nm@1000rpm
No. of Cylinder	4
Valves/Cylinder	2
Rated speed	2200 rpm
Bore/Stroke ratio	0.8636

The mechanism consists of functional, structural components and linear solenoid. The structural components are, inlet rocker shaft, plunger and fork. The lock pin is functional component which is actuated linear solenoid. The lock pin engages and disengages low and high rocker arms during operation. The force required for pin engagement is estimated. The lock pin which is critical and functional component is designed considering possibility of shear failure and probability of lock pin engagement.

IV. BASE ENGINE SIMULATION MODEL

GT-Power 1D simulation software package is used for the simulation of the baseline engine. The base engine is modelled and simulated in 1D engine simulation software GT V7.4. The performance of the developed engine model is compared with the actual performance of the base engine. Validation is done within 5%. Calibration of the model is done within less than 5% of the base engine values. The engine simulation model is used further to study the effects of varying the inlet valve timing and lift.

Different combination of valve timing and lift is studied on the 1D simulation model to study the effect on the volumetric performance. The intake valve events for low speed and high speed zones are considered to achieve gain in mass of fresh charge inducted in the cylinder. The improvement in the volumetric efficiency was compared with the volumetric performance of the base engine.

V. ANALYSIS AND PERFORMANCE PREDICTION

Figure 1 shows the mathematical model of the base engine used for the thermodynamic analysis.

The model consists of the intercooler, intake manifold, cylinder, intake valve, exhaust valve, exhaust manifold, exhaust gas recirculation (EGR) assembly, compressor and turbine which together form turbocharger. All the parameters required for the modelling of the mathematical model in GT-

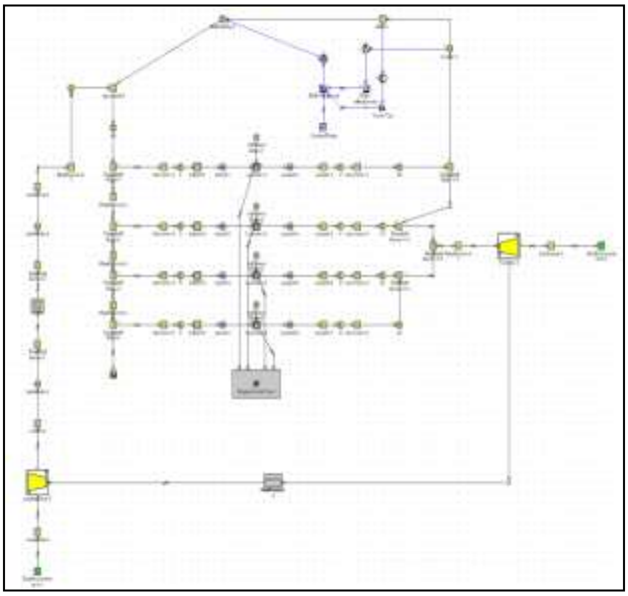
Fig. 1 GT-Power model of 2-stage VVA

Power is given from the baseline engine values. The procedure for the validation is given below:-

1. Firstly base engine GT-model is created which consists of intake manifold, intake port, intake valve, cylinder, exhaust valve, exhaust port and exhaust manifold. The boundary conditions given are compressor inlet conditions at the inlet and turbine outlet conditions at the outlet. The model is validated with the base engine values.
2. In second step intercooler assembly is added to the base GT-model, the boundary conditions given now are compressor inlet conditions at the inlet of the intercooler whereas the exhaust conditions are kept same(turbine conditions). The model is now again validated with that of the base engine values.
   The intercooler used here is modelled and validated separately.
3. In the third step compressor assembly is added to the GT-engine model and all the parameters required are given. The compressor is added before intercooler and the conditions for exhaust are kept same. Validation is again done
4. In fourth step turbine assembly is added with all the parameters required and the validation is done by comparing the mass flow rate through turbine and compressor with the actual engine data.
5. In the last step EGR assembly is added with all the boundary conditions.

Also pressure and temperature is checked at the inlet and outlet of each manifold. Parameters used for validation are power, torque and volumetric efficiency. Figure 2 shows the validation graph of the baseline engine with the GT-Power modelled engine for torque vs rpm.

Similarly figure 3 and 4 shows the validation results of the baseline engine with the GT-Power modelled engine for power vs rpm and volumetric efficiency vs rpm respectively. The simulated results are matching with the experimental values within less than 5%. Calibration of the model is done within less than 5% of the base engine vales for the further analysis

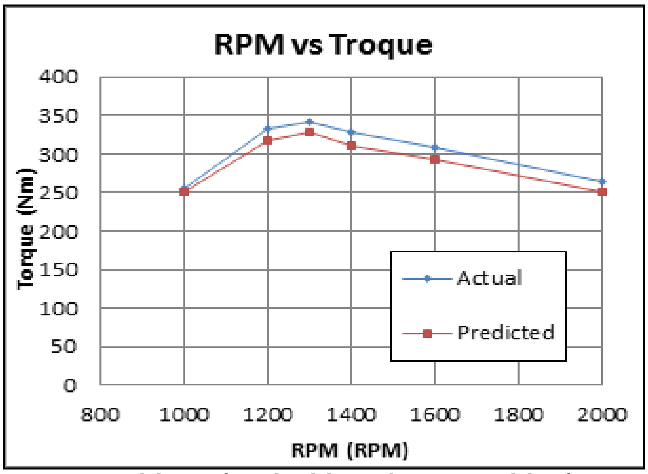

Fig. 2 Validation of simulated data with experimental data for torque vs rpm

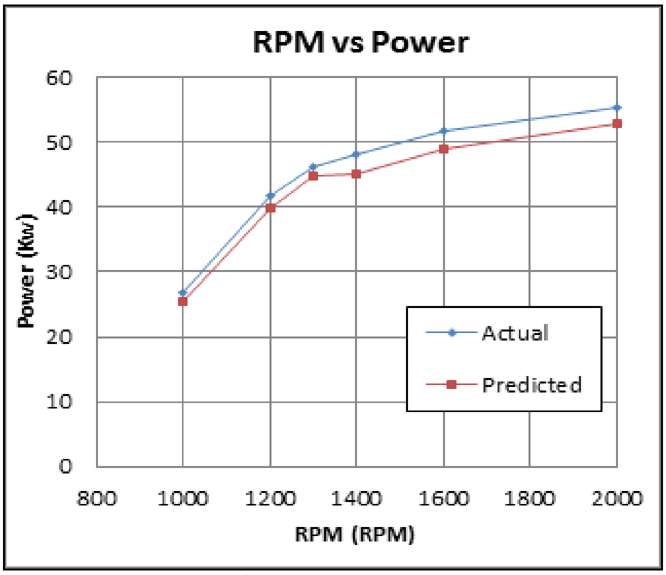

Fig. 3 Validation of simulated data with experimental data for power vs rpm

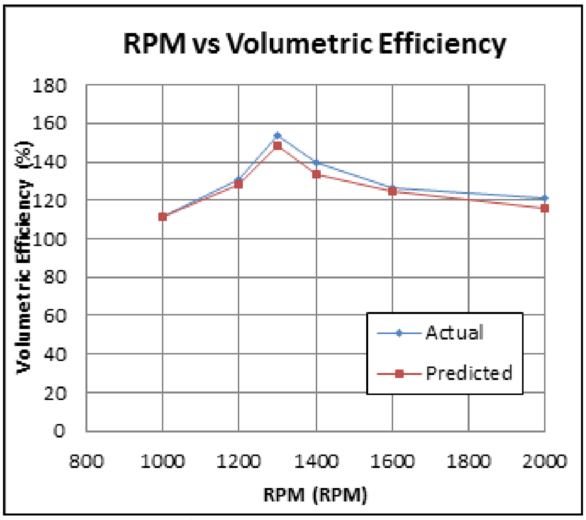

Fig. 4 Validation of simulated data with experimental data for volumetric efficiency vs rpm

VI. DEVELOPMENT OF VVA MECHANISM CONCEPT

The proposed novel two-step VVA system applies to inlet valves but a similar system can be used for the exhaust valves if desired. The inlet valves are operated by two cam lobes while exhaust valve is actuated by single cam lobe. The two-step VVA system incorporates both a Low Lift Cam (LLC) and a High Lift Cam (HLC) for intake valves. The LLC is designed for low speed operation. The HLC is independently designed for peak power at rated engine speed. The switch-over between LLC and HLC is actuated by linear solenoid through lock pin. This can improve the engine breathing during low-speed low-load operation and high-speed full-load operation compared to base engine. A feasible option is presented that is cost effective and simple in operation to vary intake valve timing and lift in two stages of low speed zone and high speed zone keeping the exhaust valve events same.

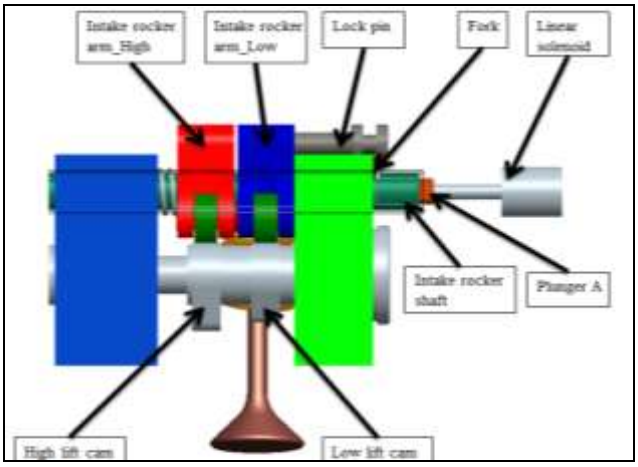

Fig. 5 Two-step VVA system components

VII. TWO STEP VARIABLE VALVE ACTUATION

The difficulties of matching the valve timing and lift to suit the engine speed and load conditions can be partially overcome by having a set of cam lobe profiles for the low to medium speed range. As a result the charge intake speed is kept well up and at predetermined operating conditions in the upper engine speed range, switch to a cam profile which increases valve lift and enlarges the opening period of the valves as a means of maximizing engine power.

The primary issues surrounding two-step VVA system implementation include.

1. The best suitable valve lifts profile and timing to improve fuel economy, performance and emissions over the entire engines operating range.
2. The benefits of two-step VVA systems compared to continuously variable VVA systems.

Here the application of VVA is considered to improve the volumetric efficiency of engine. The effects of IVO/IVC and engine speed on volumetric efficiency, pros and cons of variable inlet valve timing are discussed, and two-step variable valve actuation is considered for possible implementation. The development of novel two-step VVA system is discussed in the below sections.

The valve actuation includes valve timing, valve lift and duration. The valve lift and duration depend solely on the cam shape. However, valve timing is controlled by both the cam shape and relative position of each cam with other. The valve actuation determines the flow dynamics of air/fuel mixture entering the engine and is one of the controlling factors of engine performance, fuel economy and emissions. The valve timing and valve overlap angle affect volumetric efficiency and hence engine performance at low and high speeds.

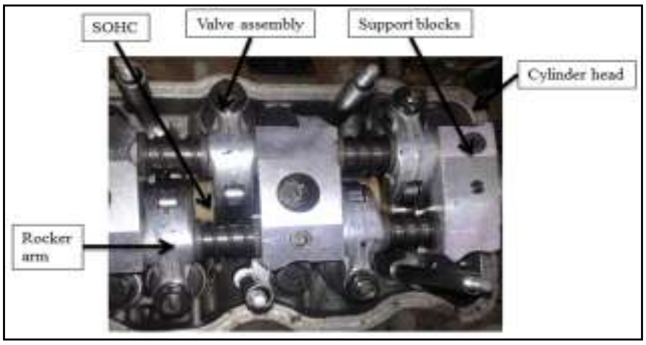

Fig. 6 Base engine valvetrain

VIII. TWO STEP VVA STRATEGY

The low valve lift increases the velocity of charge flow which helps to promote better fuel atomization, reduced HC emissions and better cold start properties at low speeds. During the low speed of engine, it is desired to delay the opening of intake valve to reduce the overlap with exhaust valve for stable combustion and avoid mixing residual gases with fresh gases. Also, at low speeds, the intake valve closing is earlier to retain the maximum compression ratio and reduce back flow. Hence, Late Inlet Valve Opening (LIVO) and Early Inlet Valve Closing (EIVC) with low lift are desirable.

The higher lift helps the engine to breathe properly at higher speeds and take high amount of air in shorter time. During high speed, engine desires to have early opening of intake valve, thereby increasing the valve overlap with exhaust valve to reduce pumping losses. Also, at high speeds, the intake valve closing is delayed to take full advantage of ram charging. Hence, Early Inlet Valve Opening (EIVO) and Late Inlet Valve Closing (LIVC) with high lift are desirable.

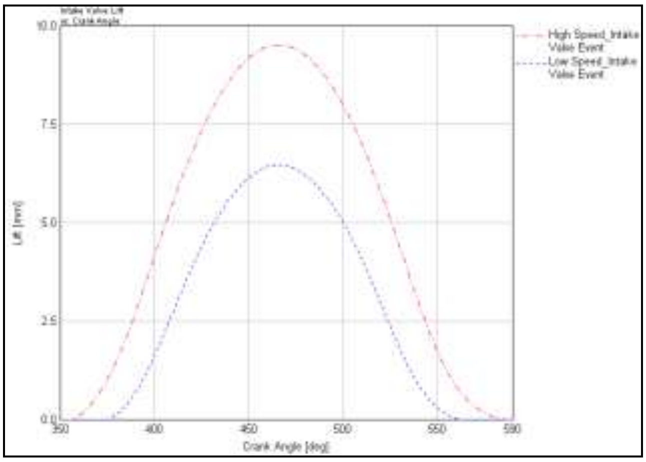

Fig. 6 Two-step VVA strategy

IX. CONSTRUCTION OF VVA MECHANISM

The discussion of two-step VVA system is limited to single cylinder intake valve for simplicity and understanding. Here the construction of the system is discussed in detail.

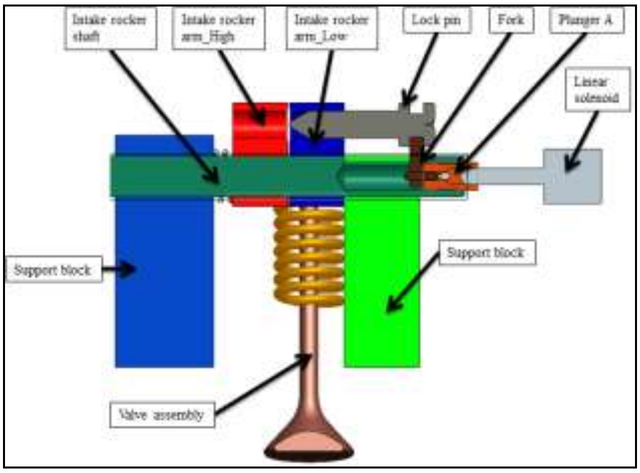

Fig. 7 Two step VVA construction

1. The mechanism consists of 3 rocker arms per cylinder, two for intake valve and one for exhaust valve. The two intake rocker arms are low speed intake rocker arm (Intake rocker arm_Low) and high speed intake rocker arm (Intake rocker arm_High).
2. A single camshaft centrally located operates both the intake and exhaust valve opening periods. There are two intake and one exhaust cam-lobes on the camshaft per cylinder. The rocker arms are constantly in contact with the respective camshaft lobes through rocker arm rollers. An arrangement is made on the intake rocker arms to accommodate lock pin.
3. A lock pin is provided to engage and disengage the Intake rocker arm_Low with Intake rocker arm_High. The position of lock pin is such that it is always inside the Intake rocker arm_Low and disengaged from the Intake rocker arm_High. The lock pin is having flanges that are free to slide around the fork when rocker arms are in rocking motion. The motion to lock pin is provided by fork.
4. The fork is free to slide inside the intake rocker shaft. The fork when sliding inside the rocker shaft moves the lock pin inside the Intake rocker arm_High engaging the two rocker arms and disengages the two when sliding in reverse direction. The fork is connected to plunger A.
5. The linear solenoid is connected to the plunger A. The linear solenoid is used to actuate the switch-over from low speed cam-lobe to high speed cam-lobe, and vice versa. A speed sensor monitors the engine RPM. At a given instant, the sensor signals linear solenoid when the cam-lobe switch-over should take place.

X. WORKING OF VVA MECHANISM

The operation of two-step VVA system is discussed here. The working of mechanism is explained in two stages.

a. Low-to medium engine speed range

The operation of mechanism during low-to medium speed range is discussed here. The linear solenoid coil is de-energized below the set switch over point RPM of the crankshaft. The lock pin remains disengaged from Intake rocker arm_High. So the inlet valve follows the timing and lift from low lift cam-lobe through Intake rocker arm_Low motion. The Intake rocker arm_High keeps rocking but does not transfer motion to intake valve.

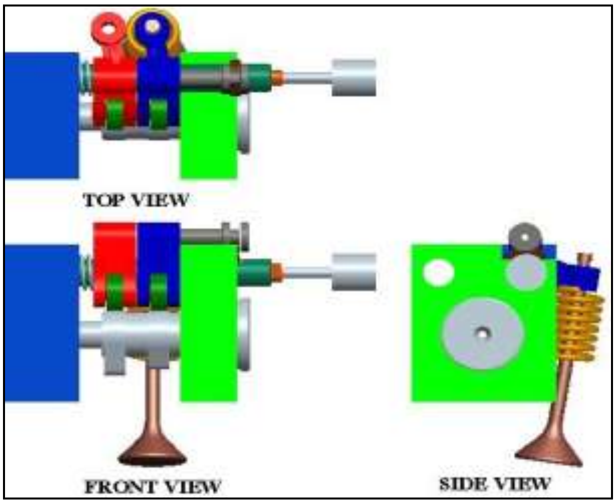

Fig. 8 Two step VVA for Intake valve

The Fig. 9 shows an instance when the lock pin is disengaged from the Intake rocker arm_High.

b. Medium-to high engine speed range

The operation of mechanism during medium-to high speed range is discussed here.

1. As the speed of engine rises above set switch over point RPM, the speed sensor sends trigger pulse which energizes linear solenoid coil. The solenoid is an electromagnetic device that converts electrical energy into a mechanical pushing and/or pulling force or motion.

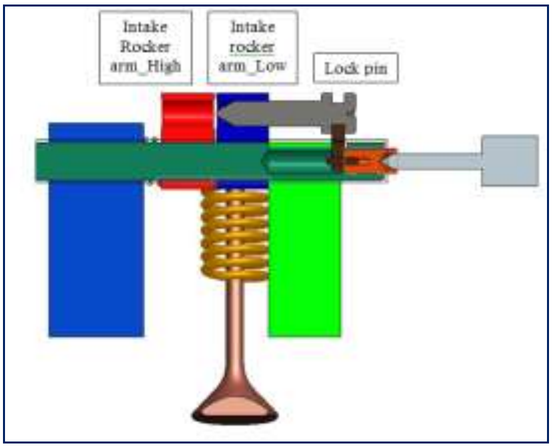

Fig 9. Two-step VVA system disengaged

2. The push stroke of solenoid pushes the plunger A and thereby the fork. At a given instant when the two intake rocker arms bores align, the fork pushes the lock pin ahead engaging the two rocker arms, thereby making the two rocker arms oscillate in unison.
3. When the two rocker arms are engaged, the high lift cam lifts the Intake rocker arm_High early and closes

it late with higher maximum lift as compared to Intake rocker arm_Low. As a result the inlet valve follows the timing and lift from high lift cam-lobe through Intake rocker arm_High motion, lock pin and Intake rocker arm_Low motion. The Intake rocker arm_Low keeps rocking but does not transfer motion to intake valve.

When the engine speed reduces below the set rpm switch over point RPM linear solenoid coil de-energies which pulls the lock pin in reverse operation. This causes the two rocker arms to get disengaged and the intake valve follows the low lift cam lobe timing and lift through Intake rocker arm_Low.

The Fig. 10 shows the instance when the lock pin is engaged with the Intake rocker arm_High.

In this way, variation in valve timing and lift in two speed zones viz, low speed zone and high speed zone can be achieved.

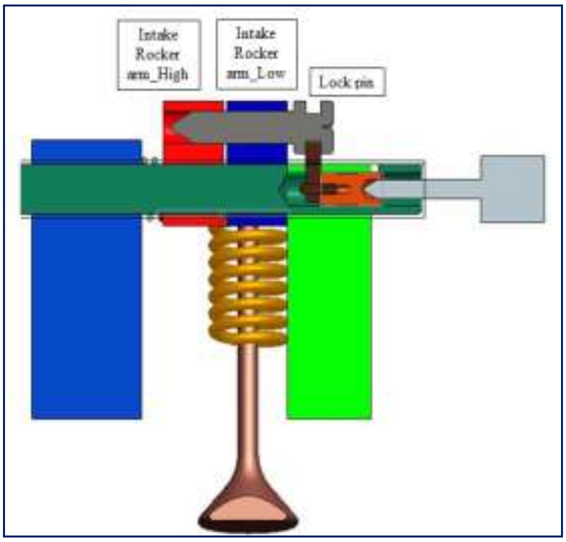

Fig. 10 Two-step VVA system disengaged

XI. OPTIMIZATION OF THE BASELINE VVA SYSTEM

Various valve strategies are established and used for the optimization of the baseline system. This is done by changing the timing of IVO and IVC for different half angle values. With changing IVO and IVC for each case (high speed and low speed), optimum results are obtained at half angle value of 55^0 with valve lift of 10.3mm for low speed zones, i.e. by changing the valve timing from 0^0 to 12^0 from bTDC as IVO for low speed zones. The new timing obtained is 55/229 which means 55^0 half angle value and 229^0 cam angle with the inlet valve opening at 12^0 bTDC. The effect of these conditions on volumetric efficiency can be seen in figure 5.

The above graph shows the changes in volumetric performance for low speed zones i.e. (1000rpm-1300rpm). While the volumetric performance at high speed zone remains the same. Overall 2.36% improvement in the volumetric performance is observed from the baseline.

Similar changes were made in IVO and IVC for high speed zones. Optimum results are obtained at half angle value of 58^0 with valve lift of 10.3mm i.e. by changing the valve timing from 6^0 to 14^0 from bTDC as IVO. The optimised timing are 58/234 i.e. 58^0 half angle value and 234^0 cam angle with valve opening at 13^0bTDC. The effect of this condition on volumetric performance is shown in the figure 11.

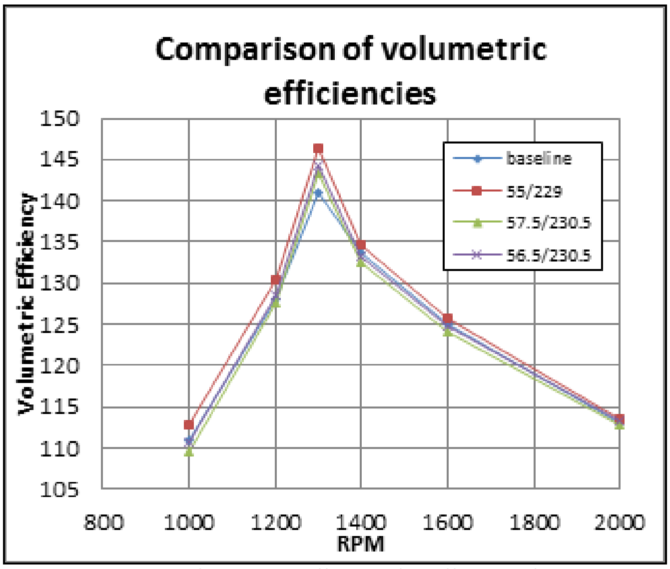

Fig. 11 Comparison of volumetric efficiency for different half angle values for low speeds.

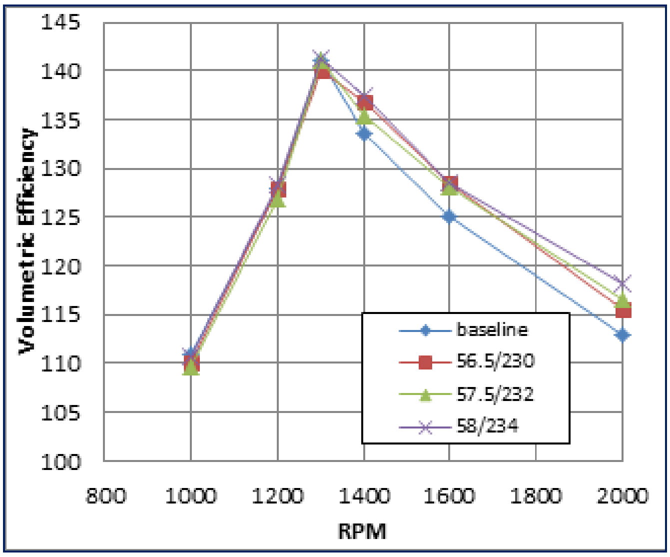

Fig. 12 Comparison of volumetric efficiency for different half angle values for high speeds.

The above graph shows the changes in the volumetric performances for high speed zones i.e. (1400rpm-2000rpm). It can be seen that the volumetric performance at low speed remains the same. There is overall 2.56% improvement in volumetric performance from the base engine for high speed zones.

Figure 12 shows the comparison of volumetric efficiency at low and high speed with baseline.

Figure 13A shows the overall increase in the volumetric efficiency of VVA at both the speed zones (low speed zone and high speed zone) with respect to the baseline engine.

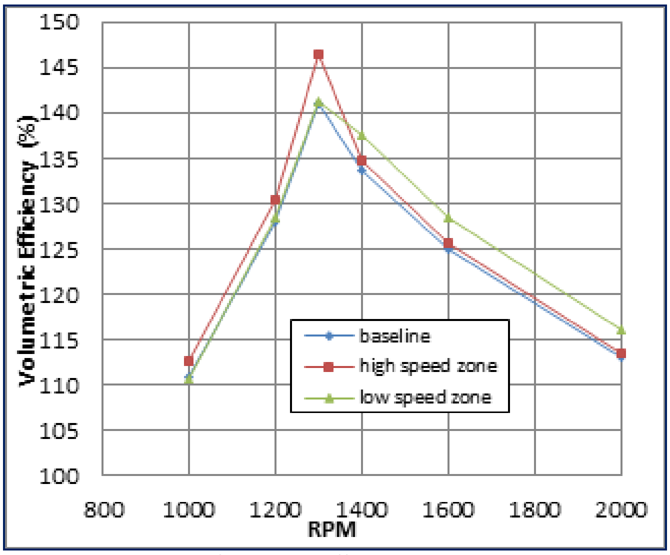

Fig. 13A Comparison of volumetric efficiency at low and high speed with baseline

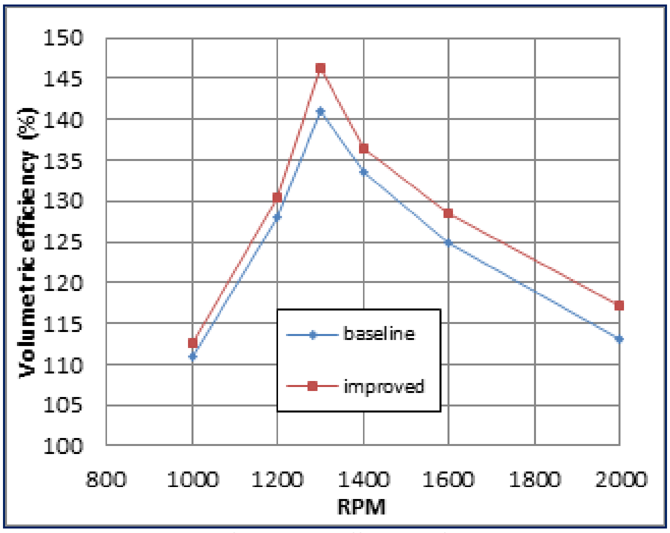

Fig. 13B Comparison of volumetric efficiency of VVA with respect to baseline.

TABLE II
VOLUMETRIC EFFICIENCY IMPROVEMENT DUE TO VVA

Engine Speed (rpm)	Volumetric Efficiency of Air (% improvement)
2000	2.72
1600	2.80
1400	2.18
1300	3.83
1200	1.88
1000	1.53
Average % improvement	2.49

Figure 13A & 13B and table 2 shows the effect of VVA on volumetric efficiency of the engine at the two speed zones. There is no constant value increase in the volumetric efficiency at each speed, but the overall effect is slightly better than the base engine performance. Also if we consider other factors like power, torque, bsfc and emission then certainly there will be huge advantage compare to the base engine.

XII. KINEMATIC AND DYNAMIC ANALYSIS OF CAM

Kinematic and dynamic analysis is done using GT-Valvetrain software package. Following criteria have been used in design of the valve train system & components.

- Air flow and gas exchange requirements (Volumetric efficiency and residual gas contents etc.)
- Kinematic requirements (Controlled acceleration and decelerations, max cam lift etc.)
- Dynamic requirements (Contact forces, valve jump and bounce phenomenon etc.)

In order to analyse the first requirement, it is required to simulate the intake and exhaust gas flow dynamics along with gas exchange process. These processes have been simulated using 'GT-POWER' software (Advance engine cycle simulation).

The schematic of mathematical model for valvetrain analysis generated in GT-VTRAIN is shown in fig 14.

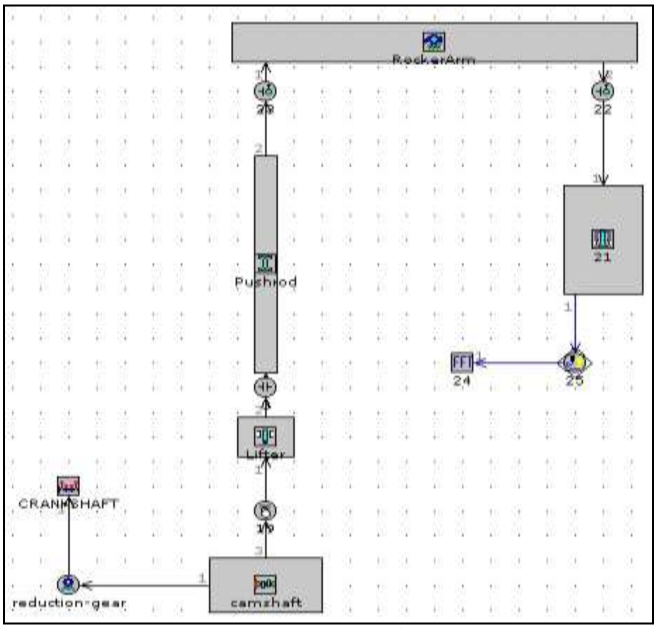

Fig. 14 Mathematical Model of Valvetrain

In order to analyse the first requirement, it is required to simulate the intake and exhaust gas flow dynamics along with gas exchange process. These processes have been simulated using 'GT-POWER' software (Advance engine cycle simulation).

Parametric study is carried out for different valve timing events in order to satisfy the flow requirements at maximum torque and rated power speed points.

Once the optimum valve timing event and maximum valve lift requirements are frozen, POLYDYN cam design approach is used for development of cam profiles. The polydyne cam design approach is followed during design of the proposed cam profiles. This method accounts for the effects of clearances, masses, rigidity and dynamic forces. Attempt is made to achieve linkage stiffness as high as possible with lower weights of valve train components. This improves natural frequency of system/valvetrain. Following design steps are taken during design of cam profiles.

- Estimation of linkage stiffness for both inlet and exhaust by classical methods.
- Estimation of various input parameters such as ramp height, ramp rate, etc. based on the operating requirements. Determination of valve lift, cam lift and tappet lift curves based on polydyne cam design approach and optimisation of polynomial powers and coefficients.
- Optimisation of various parameters like acceleration trends, optimisation of valve spring system, etc. to meet the design and dynamics requirements.
- Determination of cam profile coordinates with flat face follower.

After meeting the air flow and kinematic requirements of valve train design, it is utmost important to give attention towards valve train dynamics. Due to elasticity of valve train linkage and masses of different components, the entire valve train is represented as multi degree mass elastic system.

- **KINEMATIC ANALYSIS OF THE CAM LOBE**

Kinematic analysis is carried out to check the controlled acceleration and decelerations, max cam lift. Following are the results of the analysis

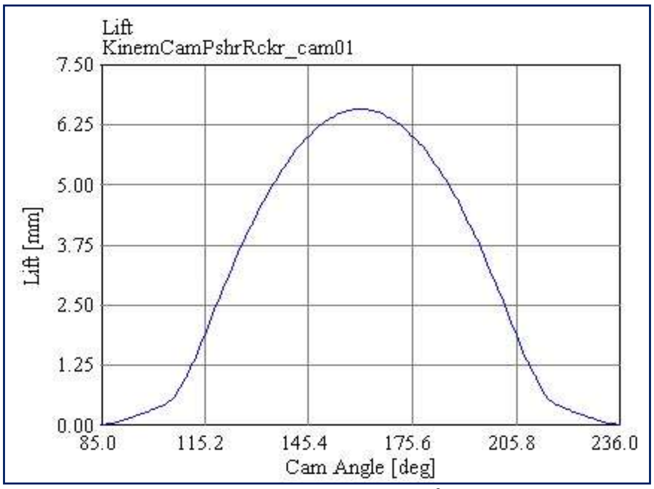

Fig. 15 Kinematic Cam Lift

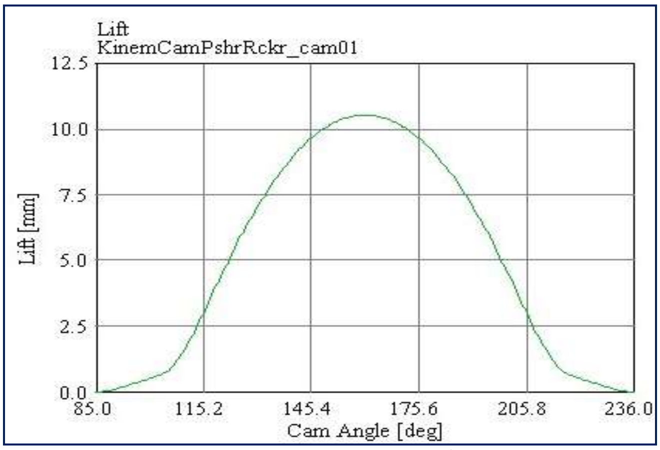

Fig. 16 Equivalent Tappet Lift

From figure 15 it is observed that the graph obtained from the simulation results are smooth and continuous without any irregularity in profile. This shows that the accelerations are controlled. Also the designed cam satisfies the max lift criteria i.e. the parameters which are checked here includes.

- Contact Stresse
- Radius of Curvature
- Nose Radius
- Minimum Tappet Diameter
- Acceleration Ratio.

After performing kinematic analysis of the cam lobe, next step is to perform dynamic analysis. Analysis is carried out at three speeds i.e. at max torque condition, at rated speed condition and at over speed condition.

- **DYNAMIC ANALYSIS OF THE CAM LOBE**

Dynamic analysis is carried out in order check contact forces, valve bounce and valve jump phenomenon. Analysis is carried out at three speeds i.e. at max torque condition, at rated speed condition and at over speed condition.

CASE-I AT MAX TORQUE SPEED OF 1300 RPM.

Following are the results of the analysis.

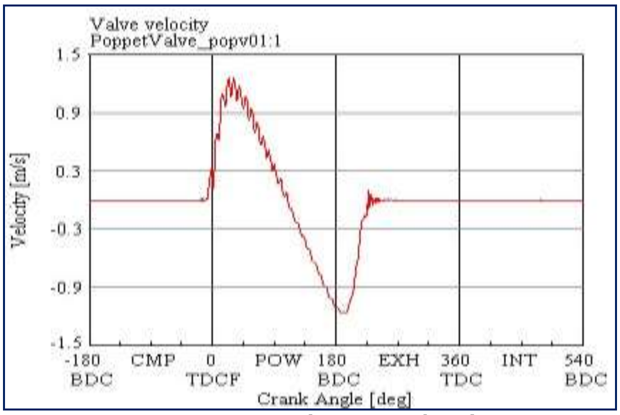

Fig. 17 Dynamic Velocity vs crank angle

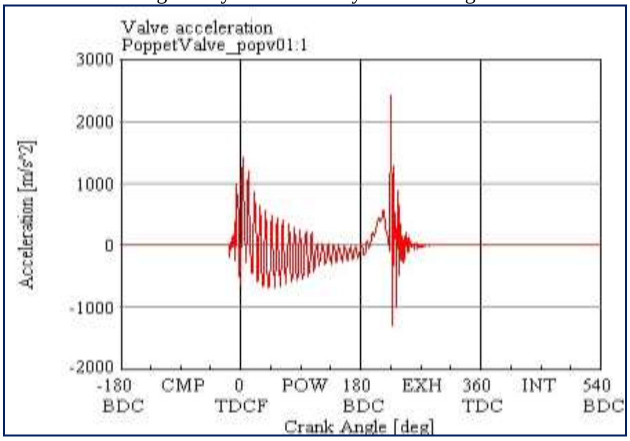

Fig. 18 Dynamic Acceleration vs crank angle

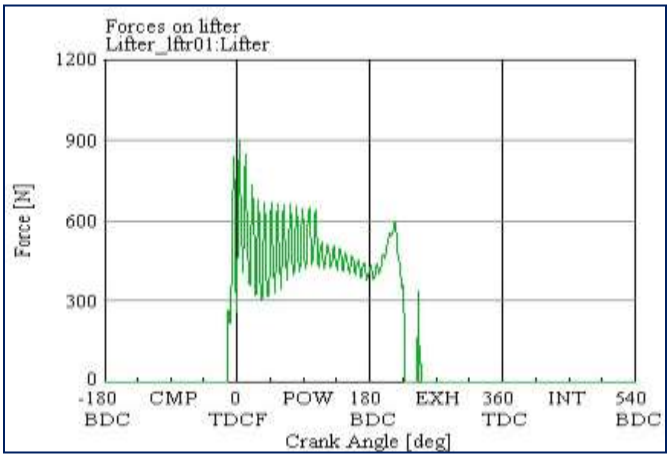

Fig. 19 Dynamic Force vs crank angle

From figure 17 and 18 it can be observed from these graphs of valvetrain response from simulation software that response of system is smooth without any irregularity in profile. Thus there is, no valve bounce phenomenon noticed (i.e. reopening of valve after closing on valve seat)
Figure (19) is the graph of contact force vs Crank angle, the contact force is the resultant force of inertia and spring force. It can be seen from figure that this force is positive throughout the valve event. Therefore, there is no separation of cam and follower.

CASE-II AT RATED SPEED OF 2000 RPM

Following are the results of analysis.

Similarly from figure 20 and 21 it can be observed from these graphs of valvetrain response from simulation software that response of system is smooth without any irregularity in profile. Thus there is, no valve bounce phenomenon noticed. Figure 22 is the graph of contact force vs Crank angle, the contact force is the resultant force of inertia and spring force. It can be seen from figure that this force is positive throughout the valve event. Therefore, there is no separation of cam and follower.

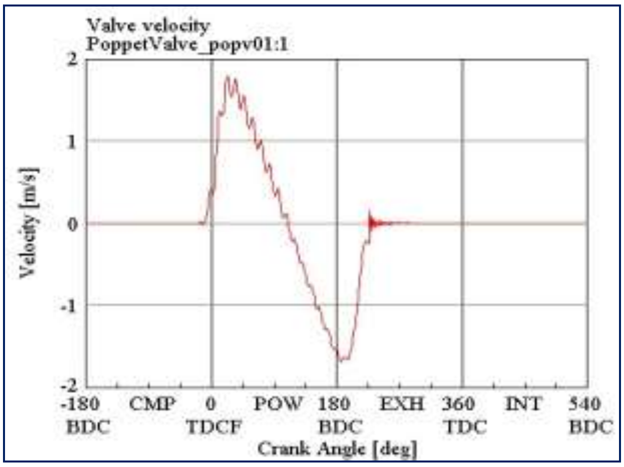

Fig. 20 Dynamic Velocity vs crank angle

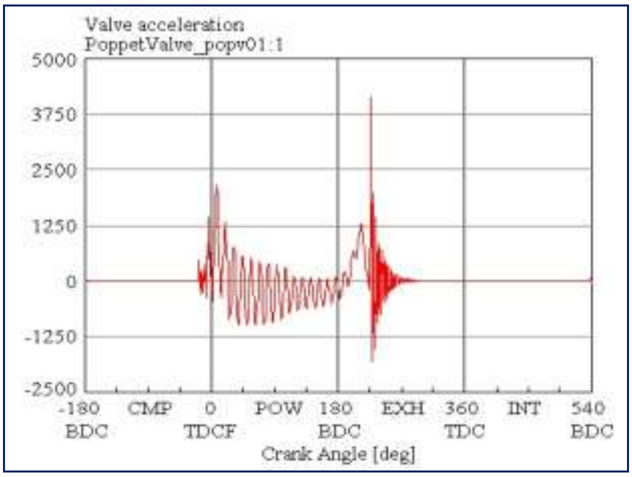

Fig. 21 Dynamic Acceleration vs crank angle

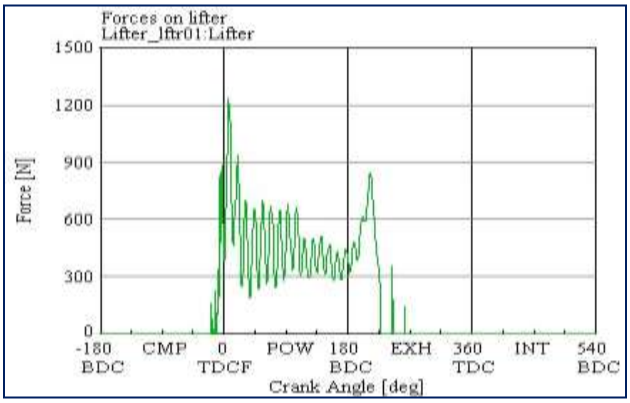

Fig. 22 Dynamic Force vs crank angle

CASE-III AT OVERSPEED OF 2600 RPM

The cam design speed is assessed based on a maximum permissible engine over - speed of 130% of rated speed in order to achieve satisfactory valve train dynamics as regards to valve train separation in case of worst situation of operation.
The rated speed of engine under consideration is 2000 rpm. Maximum engine over speed of 2600 rpm is considered as 130% of rated speed.
The design speed is taken as 10% below the over speed in order to avoid the typical vibration behaviour associated with Polydyne Cams, which comes to be 2340 rpm. The corresponding camshaft speed comes to be 1170 rpm. Therefore, the camshaft design speed is 1170 RPM.

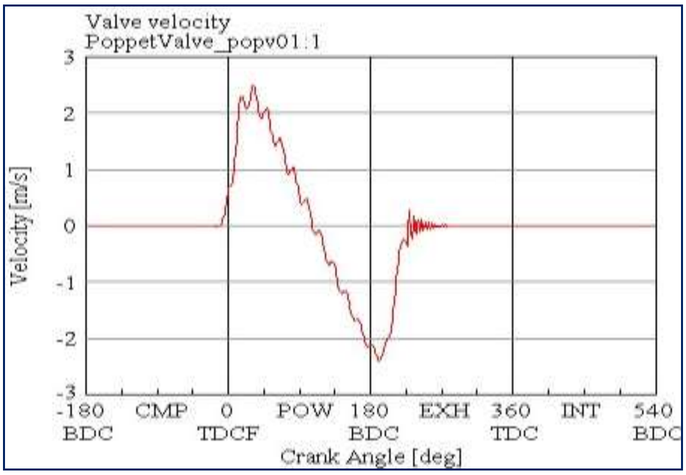

Fig. 23 Dynamic Velocity vs crank angle

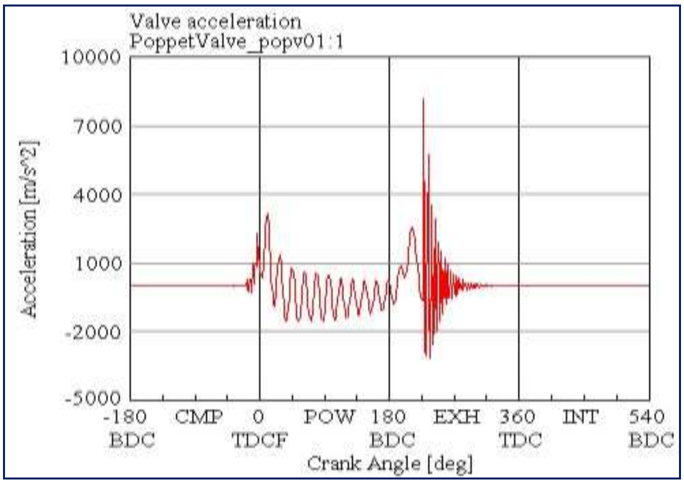

Fig. 24 Dynamic Acceleration vs crank angle

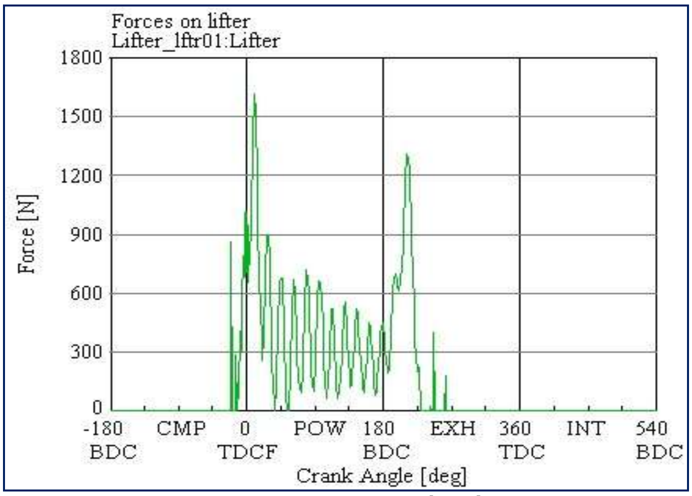

Fig. 25 Dynamic Force vs crank angle

Similarly from figure 23 and 24 it can be observed from these graphs of valvetrain response from simulation software that response of system is smooth without any irregularity in profile. Thus there is, no valve bounce phenomenon noticed (i.e. reopening of valve after closing on valve seat)

Figure 25 is the graph of contact force vs Crank angle, the contact force is the resultant force of inertia and spring force. It can be seen from figure that this force is positive throughout the valve event. Therefore, there is no separation of cam and follower.

For safety purpose the cam is designed at the speed 130% more than that of rated speed. From the above graphs it is observed that at the over speed range of 2600 rpm the simulation results are within the prescribed limits. Which means the design of the cam is safe and can be installed on the existing engine.

XIII. CONCLUSIONS

- The novel 2-stage VVA mechanism divides the operating speed range of engine into two speed zone viz. low speed zone and high speed zone.
- As from the literature review it has been observed that the intake valve timing is the single most parameter to measure the volumetric efficiency at low speed and high speed zone.
- The above designed VVA system is designed for performance point of view in terms of volumetric efficiency. There is improvement of 3.83% for lower speed range and improvement of 2.72% for high speed range in volumetric efficiency and overall improvement of 2.48% is achieved.
- Kinematic and Dynamic analysis is carried out at different speeds (i.e. at Max Torque, at rated speed and at overspeed conditions) in order to satisfy the design criteria. Dynamic analysis is carried- out for inlet Valve Train systems at various engine speed of 1300, 2000, 2600, rpm.
- For Inlet V-Train following are the observations at above rpm range: No valve bounce phenomenon is observed with the new inlet cam profiles. The contact force between cam and follower is positive during the entire valve event. This ensures that there is no cam and follower separation at these speeds. The results obtained are within the prescribed limits of the design criteria.

The designed VVA system can be manufactured and implemented on the baseline engine.

TABLE III
NOMENCLATURE

Abbreviation	Description
VVA	Variable Valve Actuation
TDC	Top Dead Center
BDC	Bottom Dead Center
EIVC	Early Intake Valve Closing
LLC	Low Lift Cam
HLC	High Lift Cam
VVT	Variable Valve Timing
aTDC	After Top Dead Center
bTDC	Before Top Dead Center
aBDC	After Bottom Dead Center
bBDC	Before Bottom Dead Center
RPM	Revolutions Per Minute
IVC	Inlet Valve Closing
IVO	Inlet Valve Opening
EVC	Exhaust Valve Closing
EVO	Exhaust Valve Opening
BSFC	Brake Specific Fuel Consumption

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