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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-wheeler vehicle 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.**

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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, one cannot lose sight of fuel economy and increasingly stringent emissions laws. Another important area in combustion of engine research is the implementation of new technologies like Variable Valve Timing (VVT),Variable Compression Ratio (VCR), Variable Intake System, Variable Geometry Compressor, 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 engine overall 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 of necessity 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 forvarious 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 a common system architecture. Thus there is substantial flexibility to tune engine characteristics for high performance as well as sport/luxury applicationsbecause of the ability to reconfigure the VVA system.

II. METHODOLOGY

Thebase 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 NO. 1
ENGINE SPECIFICATION

Engine capacity	3.12Litr
Power	55Kw @2000rpm
Torque	255Nm@1000rpm
No. of Cylinder	4
Valves/Cylinder	2
Rated RPM	2000rpm
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

III. 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.

IV. ANALYSIS AND PERFORMANCE PREDICTION

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

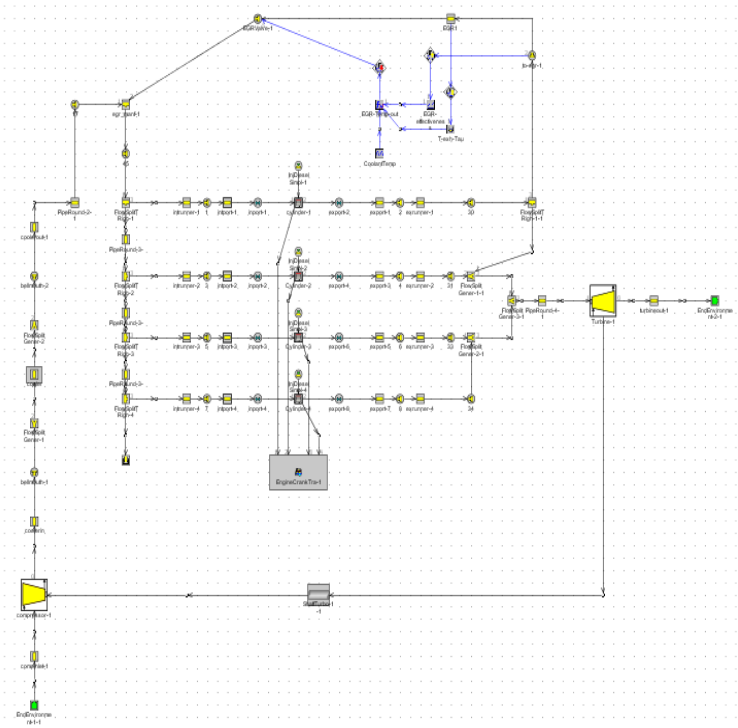

Figure 1 GT-Power model of 2-stage VVA

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 modellingof the mathematical model in GT-Power are 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. At last 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.

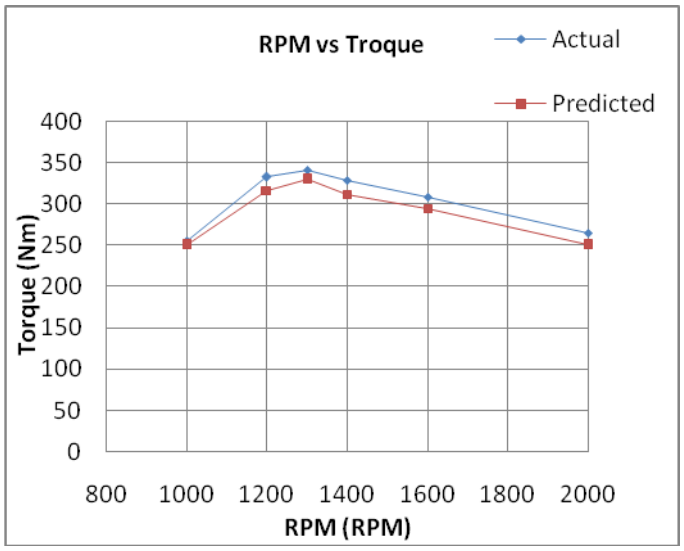

Figure 2 Validation of simulated data with experimental data for
torquevs rpm

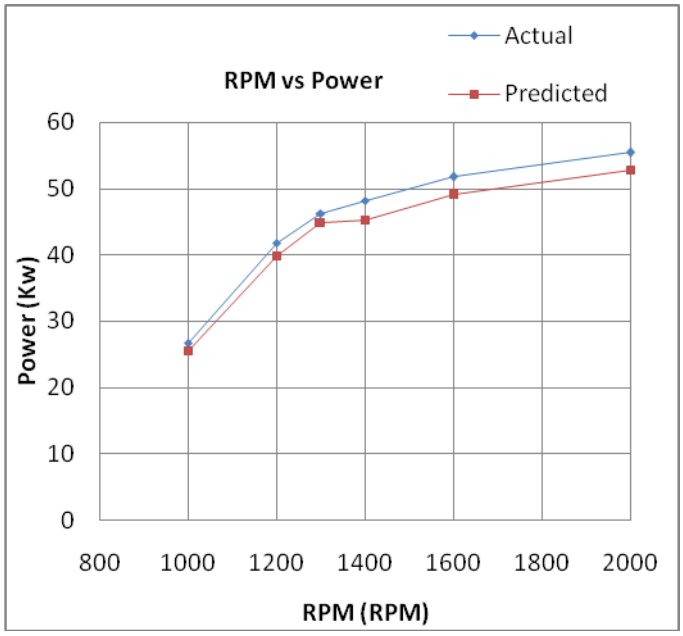

Figure 3 Validation of simulated data with experimental data
for power vs rpm

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.

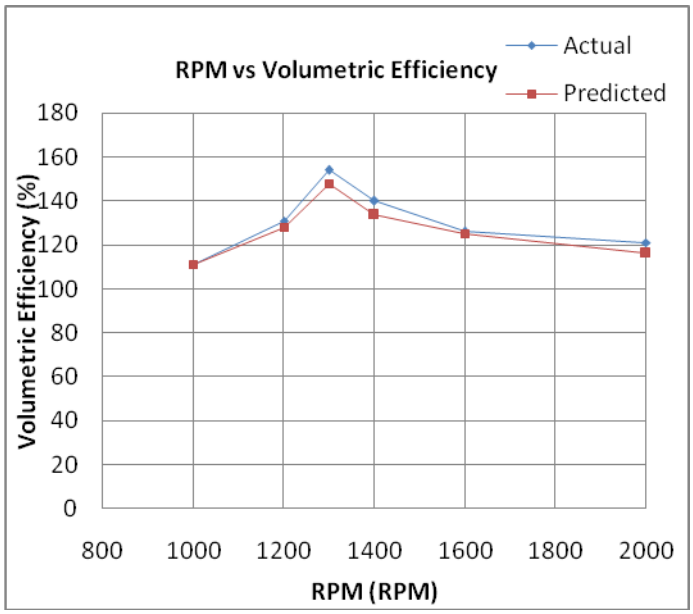

Figure 4Validation of simulated data with experimental data for volumetric efficiency vs rpm

V. 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 5 with valve lift of 10.3mm for low speed

zones, i.e. by changing the valve timing from to 1 from bTDC as IVO for low speed zones. The new timing obtained is 55/229 which means 5 half angle value and 22 cam angle with the inlet valve opening at 1 bTDC. The effect of these conditions on volumetric efficiency can be seen in figure 5.

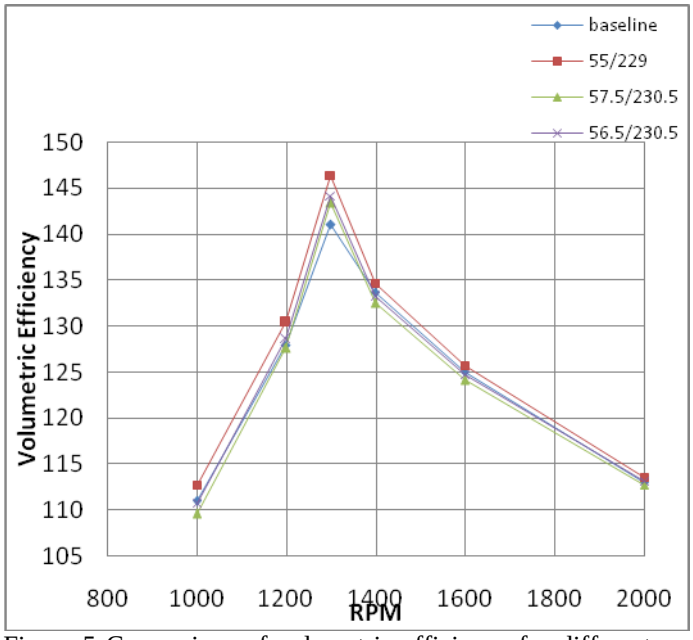

Figure 5 Comparison of volumetric efficiency for different half angle values for low speeds.

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 5 with valve lift of 10.3mm i.e. by changing the valve timing from to 1 from bTDC as IVO. The optimised timing are 58/234 i.e. 5 half angle value and 23 cam angle with valve opening at 1 bTDC.The effect of this condition on volumetric performance is shown in the figure 6.

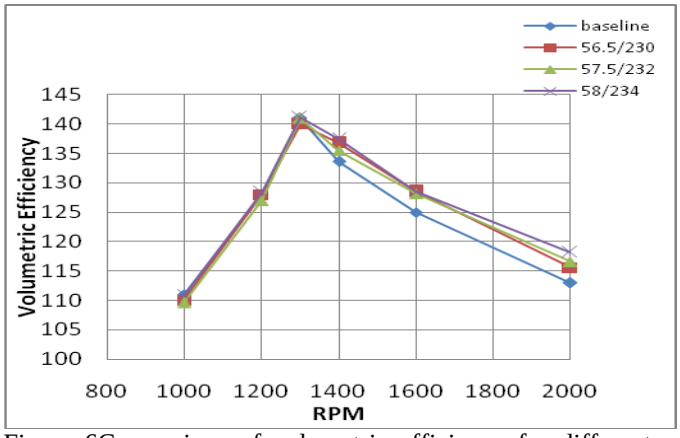

Figure 6Comparison of volumetric efficiency for different half angle values for high speeds.

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

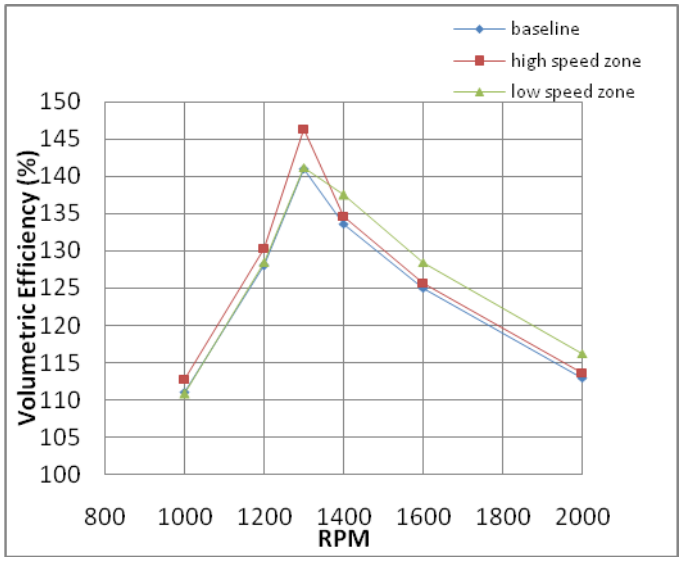

Figure 7 Comparison of volumetric efficiency at low and high speed with baseline

Figure 8A 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.

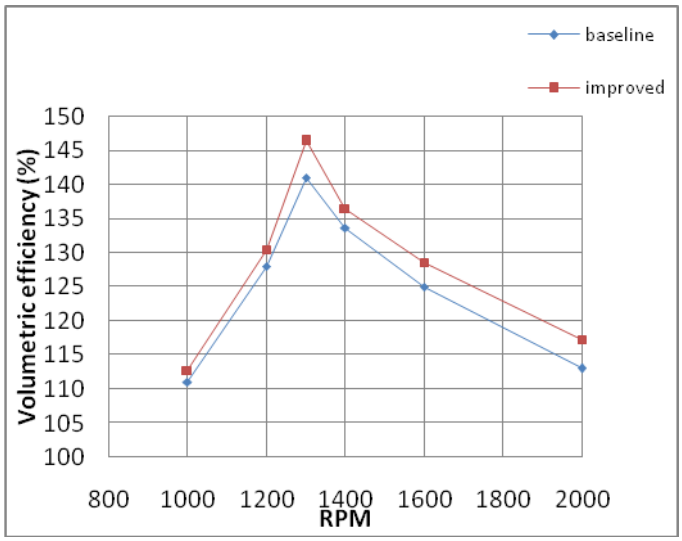

Figure 8 Comparison of volumetric efficiency of VVA with respect to baseline.

TABLE 2
VOLUMETRIC EFFICIENCY IMPROVEMENT DUE TO VVA

Engine Speed (rpm)	Volumetric Efficiency of Air (% improvement)
2000	2.716814
1600	2.8

1400	2.170658
1300	3.829787
1200	1.875
1000	1.531531
Average % improvement	2.48729

Figure 8 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.

VI. CONCLUSION

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 2.36% for lower speed range and improvement of 2.56% for high speed range in volumetric efficiency and overall improvement of 2.48% is achieved.

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

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TABLE 3
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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