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OPTIMIZATION OF VALVETRAIN MECHANISM FOR THREE CYLINDER TURBOCHARGED INTERCOOLED GASOLINE ENGINE

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ABSTRACT

Present work describes the effect of in-cylinder pressure in turbocharged gasoline engine, on valvetrain analysis. Consideration of cylinder pressure gives corrected stresses and also it gives scope for reduction in moving masses in valvetrain. Reduction in moving mass and valve spring stiffness has significant impact in valvetrain kinematics & dynamics. These parameters are analysed in detail in this work.

Initially, kinematic and dynamic analysis on the benchmark three cylinder naturally aspirated engine valve train is performed. The impacts of in cylinder pressure (P-Ө) on kinematic parameters are analysed. The boundary conditions are modified suitable to TCIC engine and the impact on valvetrain kinematic parameters such as valve lift, velocity, acceleration, contact stress between cam and follower at idle speed and high speed, inertia and spring force safety factor, dynamic valve lift and valve bouncing are analysed & tabulated.

These analyses revealed that there is the scope to reduce the contact stresses by 5% at least. Further optimization of the valvetrain masses and stiffness to bring down the stress levels of turbocharged engine is performed. Valvetrain kinematic and dynamic analyses are being simulated in 1D analysis software AVL-EXCITE.

KEYWORDS: Design Guidelines, Dynamic analysis, Kinematic analysis, Valvetrain.

INTRODUCTION

Engine downsizing is a recent trend in new generation gasoline engines due to its many advantages such as increased fuel efficiency, high power to weight ratio, good low end torque etc. Turbocharger is key to the engine downsizing. Compared with naturally aspirated engine of identical power output, the fuel consumption of a turbocharger engine is lower, as some of the wasted exhaust gas energy contributes to the engine's efficiency.

High combustion pressure developed in downsized turbocharged engine leads to higher loads on mechanical systems such as valvetrain, crank-train of an engine. Mechanical parts need to be evaluated for functionality for this high combustion pressure while developing turbocharged gasoline engine from naturally aspirated engine. The strategy need to be defined to redesign optimally the mechanical systems for turbocharged engine with minimal changes as compared to naturally aspirated engine. Engine valvetrain is a mechanical system which controls in a timely way the entry of charge and exit of exhaust gas in each individual cylinder following each cycle of the engine operation through intake and exhaust valves. The valves must respond quickly to the valvetrain actions, and they must seal against the combustion pressures and temperatures. The valvetrain system consists of a camshaft, one or

more transfer elements between the camshaft and valves, valve springs, retainers, keepers, valve stem seals, and the actual valves.
Criteria of valvetrain system design include minimum contact pressure between the cam and follower, maximum rigidity of the camshaft bearing mounting, minimum valvetrain inertia, maximum valvetrain stiffness, minimum acceleration and deceleration during valve opening and closing, minimum valve seating velocity, minimum spring load without causing the valve to jump at high engine speed. Effect of increased combustion pressure in turbocharged gasoline engine when compared to naturally aspirated engine needs to be analyzed against the valvetrain system design criteria.

LITERATURE REVIEW

Valvetrain being a very integral part of IC engine a much research has been carried out on this subject. Technology and designing processes has got refined to large extent in due course of time. New technology and software have taken over the use of conventional design methodology, so recent research work and relevant technology is considered for review in this project.

Yeongching Lin *et al* [1] predicted valve bounce speed, valve displacement, hydraulic lash adjuster motion and strain in the rocker arm using ADAMS software. In his analysis, he validated measured strain of rocker arm, valve displacement with simulation result. He used ADAMS to model the valvetrain system and calculated static and dynamic responses. ADAMS Engine is a template-based tool, it is easy to analyze components such as rocker arms, valve spring, Hydraulic Lash adjusters and engine subsystems, as well as the whole system. Valve is close to its bounce speed because the displacement at the closing ramp is higher than 0.1 mm and valve closing velocity is higher than 750 mm/sec which is mentioned in design criteria.

An-Hsuan Liu *et al* [2] carried out comprehensive experiments concerning the dynamic characteristics of high speed valvetrain. In this research he used 125cc two valves one cylinder engine which is four-stroke with OHC valvetrain system. A nonlinear model which combined a modelling technique and a system decoupling technique was developed to simulate the valvetrain response up to and above 9000 engine rpm. At low engine speeds the valve motion follows the kinematic curve of camshaft very close. As the engine speed increases the deviation of the valve motion from the kinematic curve will increase due to the inertia of the valvetrain components. The experimental and simulated valve displacements at higher engine speed and also the occurrence of the valve bounce as predicted by the model is in good agreement with the experiment. At high speeds the maximum strain of the rocker arm will be occurred close to the maximum acceleration points of the camshaft because of the increasing of the inertia force.

H. Y. Isaac *et al* [3] developed multibody 3D dynamic model in DADS software for the dynamic response of finger follower, V6 engine with overhead twin cam system by considering the interaction between valvetrain and camshaft. A high-fidelity flexible model of valve spring captured the surge mode and the coil clash of the valve spring at high speed. Coupled with camshafts shows adjuster force has a decrease of residual force after 1st peak due to torsional vibration of camshaft and it leads to a loss of contact period at high speed and cause valve bouncing. The vibrations transmit to cylinder head structure via the valve seat, spring seat and journal bearings. This dynamic model provides valve seating impact loads, spring seating loads and cam earing loads. Valvetrain designers can include considerations of camshaft torsional vibrations to minimize noise and valve bounce.

Kishiro Akiba *et al* [4] introduced the impulse force at the valve closing moment which is adversely affects the wear of the valve force. For determining this dynamic impulse force the five mass model has developed to simulate the valve mechanism. The following points obtained from five mass model.

- The impulse force does not always increase as cam speed rises.
- The impulse force does not always increase as the velocity of the cam ramp becomes higher.
- The five-mass model predicts accurately the complicated impulse force along with the valve lift and the pushrod force.
- This five-mass model is an effective tool which gives closely results with measured data.

A. S. More *et al* [5] developed flexible multi-body valvetrain model using Mat Lab simulation software. This developed model contains the geometry of valvetrain system, mass, stiffness and damping of each part. Both kinematic and dynamic simulation method are more effective way to depict the characteristics of valvetrain mechanism. At higher engine speed the performance of the valvetrain becomes critical at the dynamic valve lift, velocity and acceleration. Thus the dynamic valve lift can be increased by reduction in the energy losses in the valvetrain or by increasing the forces acting on the valve to the limit of the durability.

Tushar Kiran *et al* [6] carried out kinematic and dynamic analyses of cam follower mechanism with polynomial cam profiles. The kinematic analysis presents follower displacement, velocity, and acceleration driven by a cam rotating at a uniform angular velocity. Dynamic analysis presents static and inertial forces developed in the mechanism. He analysed 2-3 polynomial cam profile, 3-4-5 polynomial cam profile and 4-5-6-7 polynomial cam profile. He observed the degree of the polynomial increases the slope of the displacement curve, peak velocity and acceleration also increases for rise stroke. A 2-3 polynomial cam profile shows discontinuous follower acceleration at the ends of the rise and return stroke making it unsuitable at higher speeds. The simulation of cam follower mechanism with 3-4-5 polynomial cam profile shows higher follower response at high cam speeds mainly due to the low inertial forces; hence, 3-4-5 polynomial cam profile preferred at higher speeds.

Prof. H.D.Desai *et al* [7] developed the program for kinematic and dynamic analysis of the cam and follower and critical angular speed is determined for each design to predict when the follower jumps off the cam. Analytical method is used for more accurate results and programmed for the complete solution. The kinematic analysis involves the calculations of displacements, velocity and acceleration of the follower whereas the dynamic analysis includes the static and inertia force analysis of the follower. The analytical results computed for 10° rotation of cam from the program developed for kinematic and dynamic analysis of the follower.

Roland Ernst *et al* [8] used parameter optimization which is applied on the cam lobe and valve spring for reducing oscillation amplitudes and improving control of the valvetrain over a broad speed range. Oscillation amplitudes of optimized profiles are smaller and reduced peaks at the end of the event which indicates a smaller impact force for the valve seating process and also reduced the amount of contact loss. It has been used to maximize the spring force and to increase the speed range where no loss of contact occurs. . The comparison with measurements for an optimized valve spring design which shows significant improvement against base line and proves its validity.

Mahesh R. Mali *et al* [9] converted the line contact of flat face follower to point contact curved face follower for minimising the frictional losses. First is pre-processing which involves modeling, geometric clean up, element property definition and meshing. Next step includes solution of problem, which involves imposing boundary conditions on the model and then solution runs. Next in sequence is post processing, which involves analyzing the results plotting different parameters like stress, strain, natural frequency. From the analysis it is observed that change of the flat face of roller follower to a curved face roller follower mechanism results in low frictional losses due point contact which results in improved in mechanical efficiency of internal combustion engine by 65% to 70%.

Liviu Jelenschi *et al* [10] made seven rigid bodies direct acting valvetrain model using 3D virtual Lab motion software. He performed Kinematic and Dynamic analyses at different engine speeds. The kinematic analysis is used to determine the valve motion with respect to the manufacturer specifications like valve lift, velocity and acceleration. He found that on increase in engine speed the inertial forces rise and the behaviour of the valve worsens. Also an unwanted valve bouncing appears which causes the valve and valve seat wear. The valve bounce phenomenon influences the shapes of the velocities and the acceleration curves.

G. SANDU *et al* [11] created a method for designing a cam profile for an oscillating roller follower valvetrain. In this study, he used Virtual Lab LMS and Auto CAD for creating the cam profile and the kinematic analysis of the new designed cam profile. He used input as the Cartesian coordinates of the cam profile in Virtual Lab environment. He found valve lift and velocity curves are same as input data and error maximum 0.1mm which is acceptable between the original valve lift and the one obtained from simulation.

Tsiafis *et al* [12] applied genetic algorithm to find the optimal design solution of a cam mechanism with translating flat face follower. The design parameters, like the cam base circle radius, the follower face width and the follower offset can be determined considering as optimization criteria the minimization of the cam size, input torque and the contact stress. The optimization is achieved by the development of programs using the high level computing language MATLAB with GA (genetic algorithm) toolbox application. He determined the kinematic requirements displacement, velocity and acceleration of the follower. The dynamic force analysis is done to find out the friction force between follower and its guide and the friction force between cam and flat face follower.

Aan A. *et al* [13] developed method for designing of a radial cam on the worksheet of Mathcad including the calculation of follower's displacements, values of its velocities and accelerations, the optimization of the contour of a cam by choosing the eccentricity of follower and the radius of base circle of cam and the simulation of the working process of the cam mechanism. The optimal coordinates the contour of the cam are calculated. Mathcad environment is used that data format necessary for CNC part program generation. Mathcad allows the simulation of motion of virtual models of cam mechanisms. Mathcad can be used to compute cam contour coordinates, which can be used for CNC part program.

Outcome from Literature Survey:

a) The effect of in-cylinder pressure (Pressure-theta diagram) is not considered during the kinematic and dynamic analysis work of valvetrain.

b) There is no direct transformation for developing the valvetrain from Naturally Aspirated to Turbocharged Intercooled Engine configuration.

c) There is no comparative study of developing valvetrain from one engine configuration to other.

d) They have performed the Kinematic and dynamic analysis for individual engine configuration and find out the analysis results accordingly.

MATHEMATICAL MODEL

In the first level of computations, the valve lift curve is determined by a higher order polynomial. The mathematical law defining the lift during the main event is a 4-power polynomial. The number of polynomial powers depends on the conditions to be specified at start and max. lift, for eg. lift, velocity acceleration, and jerk. The number of polynomial powers is kept to a minimum in order to reduce the many values, which can be assumed for them. This aids the selection process whilst maintaining adequate control of the event [6].

The following equation is the general polynomial used for the evaluation of the valve lift curve.

$$y = L(1 + C2x^2 + C4x^4 + Cpx^p + Cqx^q + Crx^r + Csx^s) \qquad (1)$$

The co-efficients C2, C4, Cp, Cq, Cr, Cs are determined by the boundary conditions. C4 is a free eligible coefficient by which in particular the shape of the negative acceleration curve may be varied for an optimization of the cam profile.

Valve lift, velocity, acceleration and jerk

The displacement (y) of the follower is given by

$$y = f(\theta), \text{ mm} \qquad (2)$$

Where θ is the cam angle rotation in radians.

Or, because the cam rotates at constant angular velocity, the displacement can be expressed as

$$y = g(t) \text{ and } \theta = \omega t \qquad (3)$$

Where t is the time for the cam to rotate through angle θ, and ω is the cam angular velocity.

The velocity (v) is established as the instantaneous time rate of change of displacement or slope of the displacement curve at angle θ or time t,

$$v = \frac{dy}{dt} \text{ , mm/s} \qquad (4)$$

$$v = \frac{dy}{dt} = \left(\frac{d\theta}{dt}\right)\left(\frac{dy}{d\theta}\right) = \omega\left(\frac{dy}{d\theta}\right) \qquad (5)$$

The acceleration is the instantaneous time rate of change of velocity or slope of the velocity curve at angle θ or time t,

$$a = \frac{d^2y}{dt^2} = \left(\frac{d^2\theta}{dt^2}\right)\left(\frac{d^2y}{d\theta^2}\right) = \omega\left(\frac{d^2y}{d\theta^2}\right) \qquad (6)$$

The shape and values of acceleration curves are of critical concern for moderate to high speed engines. From it, analysis can be made regarding the shock, noise, wear, vibration and general performance of a cam follower system. It will be shown that for the best action, the acceleration curve should be smooth and have the smallest maximum values possible. Another term, Pulse (commonly called "jerk"), is used to define the instantaneous time rate of change of acceleration or the slope of the acceleration curve at angle θ or time t,

$$p = \frac{d^3y}{dt^3} = \frac{da}{dt}\text{, mm/s3} \qquad (7)$$

For high-speed engines, it will be shown that the maximum values of the pulse should not be too large.

Valve Spring

Valve springs are helical compression springs. They are used to follow precisely the corresponding cam profile motion during each cycle to keep the cam and transfer elements in contact at all times. During the acceleration period, cam exerts a positive accelerating force on the follower. The magnitude of ineria force is equal to the product of the mass of the valvetrain (m) and the acceleration (a) at the speed of operation, that is,

$$f = ma \qquad (8)$$

The actual force on the camshaft will exceed this value by the amount of the spring load and the gas pressure on the head of the valve at the beginning of the lift period. When the follower is on the deceleration portion of the cam contour, it must be forced to follow the prescribed path by the use of a spring [14].

Spring Force = Preload + (Valve Lift x Stiffness) (9)

Safety Factor is the ratio of summation of spring force and cylinder gas force to inertia force.

Safety Factor = (Spring Force + Cylinder Gas force) / Inertia Force (10)

Valve springs must operate the engine valves at different speeds (i.e., 600 to more than 7000 rpm crankshaft speed). The movement is restricted in the axial direction of the valve, and the ends must be ground for proper seating. Also, the I.D. is restricted by the cylinder head boss diameter. The loads also change from valve closed to valve open. The environment temperature ranges from ambient to 150°C. The valve spring is restricted on one end by a boss on the cylinder head and on the other end by a retainer. The wire hardness must be compatible with the retainer. If aluminum is used for the cylinder head, a spring seat washer must be used between the spring and the head.

The wire in a helical compression spring is stressed in torsion. The torsional stress is expressed as,

$$\sigma = \frac{8FC^3}{\pi D^2}(Y) = \frac{\Delta}{N}\frac{G}{\pi dC^2}(Y) \qquad (11)$$

$$Y = \frac{4C-1}{4C-4} + \frac{0.615}{C} \qquad (12)$$

$$\delta = \frac{\Delta}{N} = \frac{8PC^4}{GD} = \frac{\pi dC^2}{GY}\sigma \qquad (13)$$

$$k = \frac{P}{\Delta} = \frac{GD}{8C^4N} \qquad (14)$$

Under elastic conditions, torsional stress is not uniform around wire cross section due to coil curvature and a direct shear load. The maximum stress occurs at the inner surface of the springs and is computed using a stress correction factor (Y).

Contact Stress

The force at the cam-tappet interface is responsible for the compressive stresses being developed at the point of contact called as Hertz or Point contact stress.
Mean modulus of elasticity between the cam and the follower defined as:

$$\frac{1}{Em} = \frac{(1-Vf^2)}{Ef} + \frac{(1-Vc^2)}{Ec} \quad (15)$$

Hertz contact stress for a Valvetrain employing a flat-faced follower:

$$\sigma = \sqrt{\left(\frac{Em\,F}{\pi WRc}\right)} \quad (16)$$

SIMULATION MODEL

Valvetrain analysis procedures are carried out in two stages kinematic and dynamic analysis. Kinematic analysis is mostly used for design of a valve lift profile and Hertzian stresses etc. Dynamic analysis is used to determine the dynamic movement of valvetrain component considering the effect of inertia and stiffness.
In this project, naturally aspirated 3-Cylinder having Single Overhead Camshaft (SOHC) valvetrain is used as input. Figure 1 shows direct acting single overhead cam valvetrain is used in the existing naturally aspirated engine. Direct acting valvetrain consist of camshaft, tappet, valve retainer, spring seat, valve, valve spring, valve guide, valve stem seal, valve seat and valve lock half.

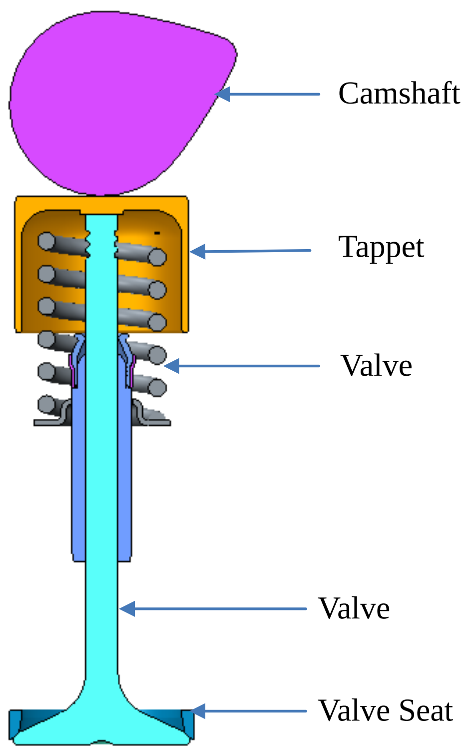

Fig. 1 Direct acting valve train

All the valvetrain components are modelled using Pro-E Wildfire 5.0 and Creo Parametric 2.0. Kinematic and dynamic analysis of valvetrain is to be analysed by using multi-body approach analysis using AVL EXCITE Timing drive software for 3-Cylinder Naturally Aspirated (NA)and Turbocharged Intercooled (TCIC) Engine Valvetrain. These results are to be verified with design guidelines [14]. If the results are within the limit of design guidelines, then same valvetrain components can be used for further turbocharged engine valvetrain for analysis. If the results are not within the limit of design guidelines [14], then same valvetrain components needs to be modified until the results are within the limit of guidelines. Next step of the project is to take the verified valvetrain components of existing naturally aspirated engine and input the P-θ (Pressure – crank degree) values in that model. Simulation is to be done on that P- θ input model. Check the results with design guideline. If the results are within the limit of design guidelines then same valvetrain components are to be finalized for turbocharged engine valvetrain. If the results are not within the limit of design guidelines then same valvetrain components needs to be modified until the results are within limit of design guidelines. Compare with the finalised AVL EXCITE simulation model results and the numerical calculation for validation. Simulation results should be closer to the numerical calculation. Finally conclude the project by considering all the results and validation.

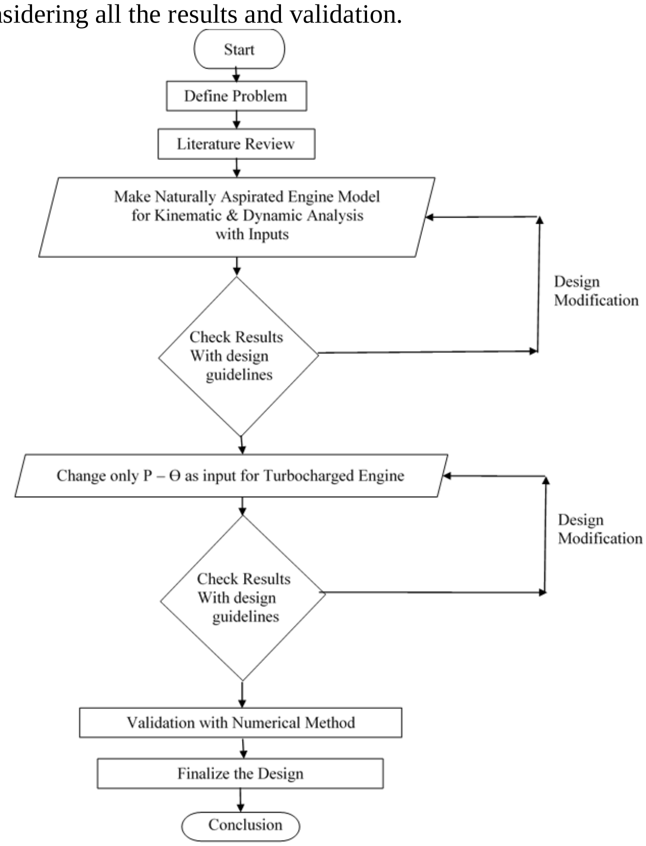

Fig.2 Methodology Flowchart

Model Preparation For Analysis

Table 1 shows input as geometrical dimensions of Cam profile, valve, valve spring and their masses, stiffness. In the pre-processing stage of simulation, all the inputs are to be given for valvetrain element connected model in AVL EXCITE Timing drive. There are four elements in simulation model of AVL EXCITE Timing drive.

Simulation Model Procedure

Figure 3 shows four parts are required for the cam design model of the single over-head Camshaft (SOHC) valve train for FEA:

1. The cam profile element: To define basic geometrical information and to visualize the full system.
2. One mechanical element to cover the masses of tappet, a part of the valve stem the upper non-active coils of the spring and the parts connecting the valve spring with the valve stem.
3. The valve face element: To define the mass and stiffness of valve seat and Valve face element.

4. The valve spring element: To define the spring characteristics such as active coil mass, pre-load and stiffness.

	Parameter	Current Specification
Cam shaft	Cam Profile	Grinding Co-ordinates
	Cam base circle	20mm
	Cam width	17.5mm
	Axial offset	2mm
	Modulus of elasticity	205GPa
	Poisson's ratio	0.3
Moved masses	Intake valve	50g
	Tappet	45g
	Retainer	12g
	Cotters	1.5g
	Valve Spring	15g
Valve Spring	Pre load	200N
	Spring rate	25 N/m
	Max spring force	450N
Engine Speed	Min. Speed	800 RPM
	Max. Speed	6000 RPM

Table 1: Inputs for Simulation

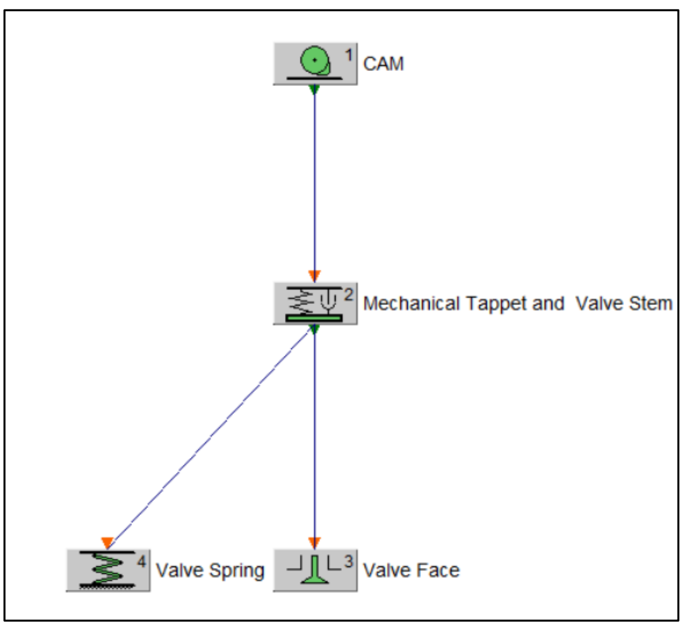

Fig. 3 Valvetrain connected element Model- AVL EXCITE Simulation model

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RESULTS & DISCUSSIONS

Figure 4 and 5 shows valve lift, velocity and acceleration versus degree of crank angle. Camshaft parameter kept same for both NA & TCIC so plot shows same pattern of lift velocity and acceleration. Valve seating velocity is kept constant at opening & closing ramp region for avoiding valve bouncing. After, in flank region velocity goes on increasing and in nose region (at maximum valve lift) velocity decreases to zero.

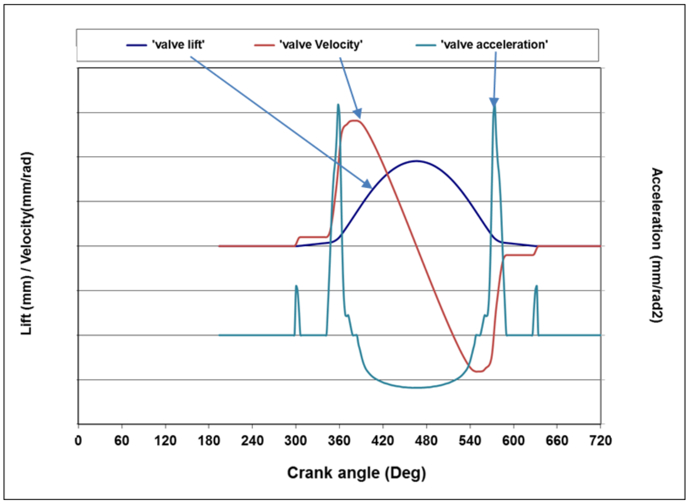

Fig. 4 Lift, Velocity and Acceleration plot for NA Engine

Acceleration shows constant at the start of cycle after it gives maximum peak value which is within the limit of 120 mm/rad2. Negative acceleration found in maximum valve lift area.

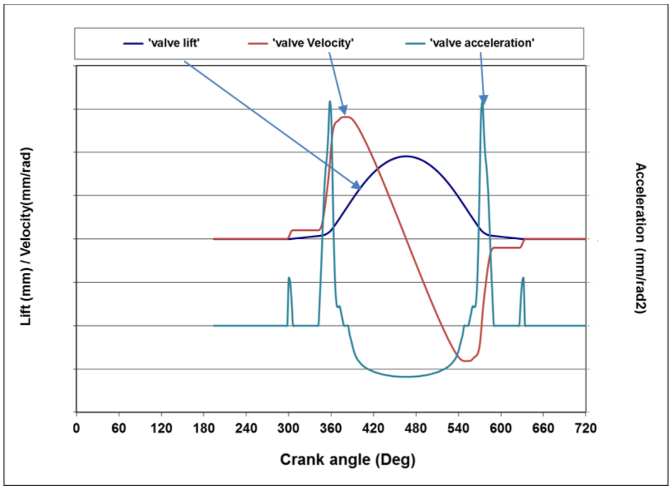

Fig. 5 Lift, Velocity and Acceleration plot for TCIC Engine

Figure 6 shows contact stress for Naturally Aspirated engine. The contact stress is normalised as 100% at maximum value. Maximum contact stresses found at idle rpm.

Figure 7 shows contact stress for TCIC engine. As compare with Naturally Aspirated engine, contact stresses are increased by 3.8% . Stresses are increased due to increase in-cylinder pressure of TCIC engine.

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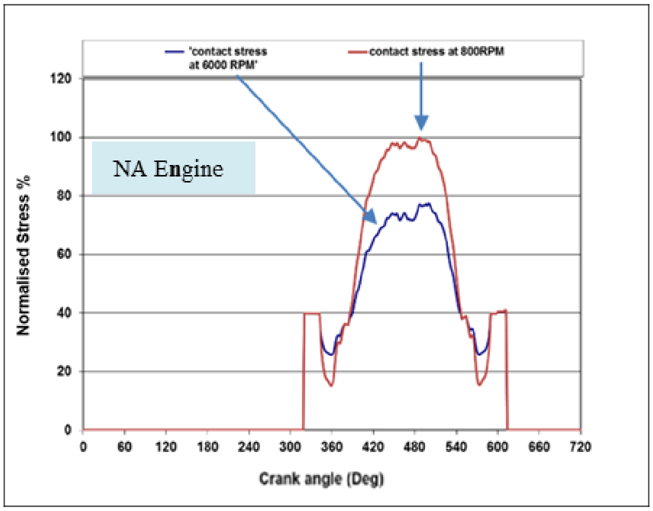

Fig. 6 Contact Stress plot for NA Engine

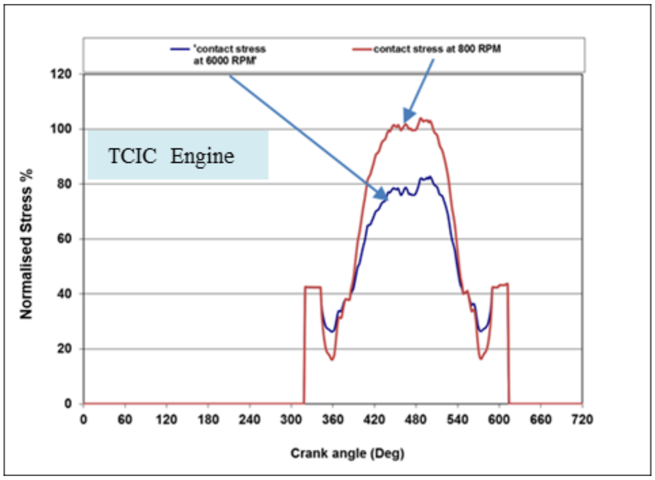

Fig. 7 Contact Stress plot for TCIC Engine

Figure 8 shows contact stress for optimized TCIC engine. As compare with TCIC engine, contact stresses are reduced by 6%. Stresses are reduced by optimizing the valve spring stiffness and mass.

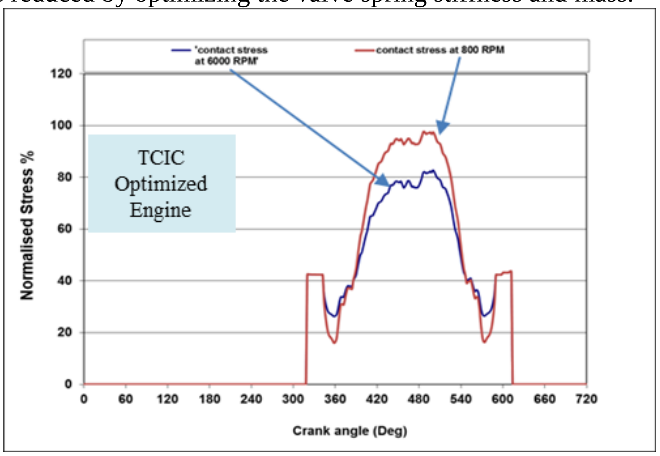

Fig. 8 Contact Stress plot for optimized TCIC Engine

Figure 9 shows spring force, inertia force and safety factor plot for Naturally Aspirated engine. Safety Factor is the ratio of summation of spring force and cylinder gas force to inertia force (10). Safety factor of NA engine is 1.76 which is above the acceptance limit of 1.2. It avoids the loss of contact between tappet and cam.

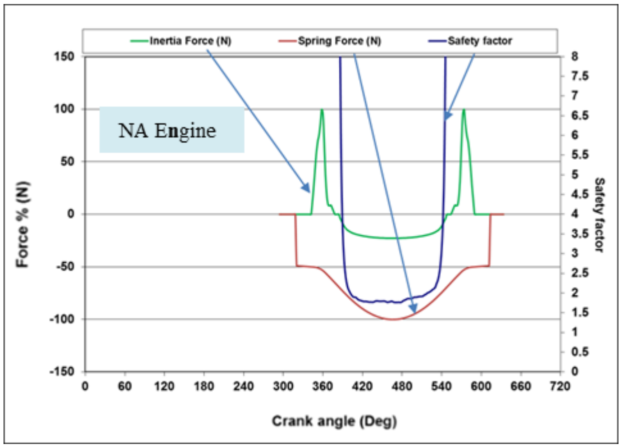

Fig. 9 Safety factor plot for NA Engine

Figure 10 shows spring force, inertia force and safety factor plot for TCIC engine. Safety factor of TCIC engine is 1.82 which is above the acceptance limit of 1.2. Safety Factor increases due to increase in-cylinder pressure of TCIC engine.

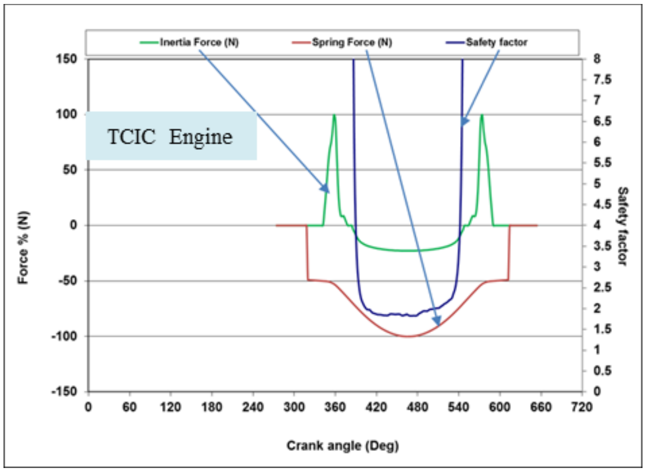

Fig. 10 Safety factor plot for TCIC Engine

Figure 11 shows spring force, inertia force and safety factor plot for optimized TCIC engine. Safety factor of optimized TCIC engine is 1.56 which is above the acceptance limit of 1.2. Turbocharger engine has high pressure compare to naturally aspirated engine. This high pressure is added the benefits to optimize the valvetrain stiffness and masses. So it is modified by changing valve spring stiffness and mass.

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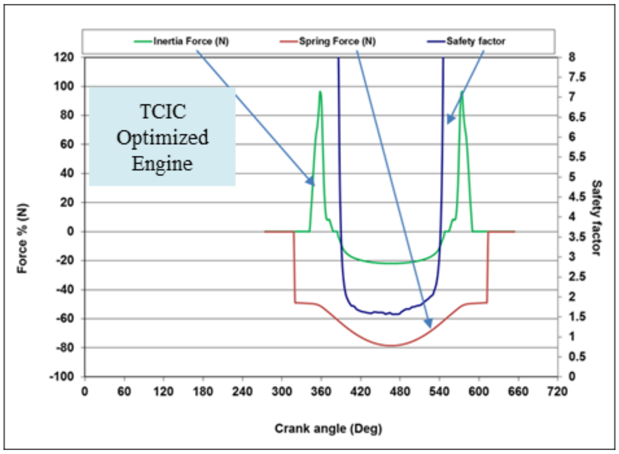

Fig. 11 Safety factor plot for TCIC Engine

Table 2 shows simulation summary results of NA, TCIC and optimized TCIC Engine. Optimized turbocharged engine shows 6% less contact stresses compare to turbocharged engine. It is because of optimizing the valve spring stiffness and mass of valvetrain.

	Naturally Aspirated Engine	Turbocharged Engine	Optimised Turbocharged Engine
Normalised Stress %	100	103.8	97.54
Safety Factor	1.76	1.82	1.56

Table 2: Simulation Summary Results

Table 3 shows design of existing and modified valve spring. All the parameters are normalized with existing values of valve spring parameters. Valve spring mass and stiffness is optimized.

	Existing Spring	Modified Spring
Mass (m)	x	0.7x
Wire diameter(d)	d	0.8d
Inner Diameter (ID)	d1	d1
Outer Diameter (OD)	d2	0.95d2
Spring Index	C	1.16C
Stiffness	K	0.6K

Table 3: Valve spring design parameters

Table 4 shows testing parameters and procedure of valvetrain. In testing of valvetrain, engine runs at different speeds overspeeding, idle and rated rpm. Overspeeding is carried out for checking the cam and follower are in contact or not. Endurance test is run for 400 Hours for checking the components performance. It runs on high speed, low speed and dual speed. After the endurance test all the parts are dismantled and check all the components thoroughly. In visual inspection wear pattern check for valve, valve seat, valveguide, tappet clearance.

1	Speed [rpm]
	a) Overspeeding
	b) Idle RPM
	c) Rated RPM
2	Endurance Test (400 HRS)
	a) High speed
	b) Dual speed
	c) Low speed
3	Wear Pattern
	a) Valve to valve seat insert wear
	b) Valve stem to Tappet
	c) Valve to valve guide
	d) Tappet clearances

Table 4: Testing parameters

CONCLUSION

- It presents the optimisation of Turbocharged Intercooled (TCIC) engine valvetrain. In this paper comparison of kinematic analysis is done for base Naturally aspirated (NA) engine & Turbocharged Intercooled (TCIC) engine valvetrain.
- In-cylinder pressure has significant effect on valvetrain kinematics.
- Consideration of the same has helped in optimising valve spring stiffness & valvetrain moved masses.
- This gives 6% reduction in contact stresses and valve spring mass reduces by 30%.

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NOMENCLATURE

Notation	Meaning	units
σ	Maximum shear stress in the spring	N/mm2
δ	Axial deflection of the spring, per coil	mm
Δ	Total deflection	mm
k	Spring stiffness	N/mm
P	Applied load	N
C	Spring index	
N	number of active coils	
G	Shear modulus of the material	N/mm2
D	Pitch diameter of the coil	mm
d	Wire diameter	mm
Y	Stress correction factor for the helical spring	
L	Maximum lift above the ramps	mm
Y	Instantaneous lift at any point in the main event	mm

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V	Velocity	mm/s
A	Acceleration	mm/s2
P	Pulse	mm/s3
σ'	Contact stress (maximum compressive stress)	N/mm2
F	Normal force (perpendicular to the surface at the point of contact)	N
Vf, Vc	Poisson’s ration for follower and cam materials	
Ef, Ec	Young’s modulus for follower and cam material	N/mm2
W	Width of contact zone	mm
Rc	Radius of curvature of cam surface.	mm