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Analysis of Overhead Valve Push Rod Type Valve Train for off Road Diesel Engine

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ABSTRACT

Push rod type valve train is still widely used for low and medium speed engines. This paper describes the methodology followed for design of valve train for new 6 cylinder turbocharged diesel engine for generating set and industrial applications. Advanced prediction tools like GT power and Matlab program were used to optimize the design. Air requirement was calculated considering air fuel ratio and thermal efficiency. Inlet valve size was finalized based on gulp factor for optimum volumetric efficiency. Valve timing was optimized to get best fuel economy and optimum emission levels using GT power model. Polydyne cam was used for minimizing vibrations and valve jump. Ramp height is calculated using cold valve clearance and static deflection of pushrod, rocker lever, rocker shaft, rocker support and cylinder head. Valve spring was designed to take care of valve jump. Finally camshaft was designed considering torque and bending load from valve train.

Keywords: *Valve Train, Polydyne cam, Valve timing, Camshaft*

Notations

D	:	cylinder bore diameter, mm
L	:	engine Stroke, mm

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η_v	:	volumetric efficiency
N	:	engine Speed, RPM
mf	:	air mass flow, m^3
S_v	:	cylinder volume, m^3
d	:	throat diameter, mm
C_{Fmean}	:	mean flow coefficient
a	:	peed of sound at valve inlet condition, m/s
M_{ve}	:	valve end effective mass, kg
M_{ce}	:	cam end effective mass, kg
m_v	:	mass of valve, kg
m_{sr}	:	mass of spring retainer, kg
m_{sp}	:	equivalent mass of valve spring, kg
m_{re}	:	equivalent mass of rocker arm, kg
m_p	:	mass of push rod, kg
m_t	:	mass of tappet, kg
rr	:	rocker ratio
δ	:	deflection of valve train, mm
K_0	:	valve train stiffness, N/mm
fn	:	natural frequency of valve train, cycles/min
y_x	:	valve lift, mm
y_t	:	tappet lift, mm
yc	:	cam lift, mm
R_a	:	ramp height, mm
k_r	:	equivalent spring rate ratio
K_v	:	valve spring stiffness, N/mm
Nc	:	cam design speed, RPM
c	:	dynamic constant
TDC	:	top dead centre
BDC	:	bottom dead centre
$y_{t_{actual}}$	:	actual Tappet lift, mm
θ_{main}	:	half main event length, degree
θ	:	angle of cam from maximum lift point, degree
b	:	Contact width between cam and tappet, mm
N_{max}	:	Maximum engine overspeed, RPM
J	:	Moment of inertia of the rocker arm about its centre of rotation
l_v	:	Valve side length of the locker arm, mm
A_p	:	Length of positive acceleration pulse, degree
S_c	:	Contact stress, N/mm^2
$S_{f_{max}}$	:	Maximum contact force, N
d_c	:	diameter of camshaft, mm
$C_2, C_4, C_p, C_q, C_r, C_s,$	:	Coefficient of polynomial equation
$p, q, r, s,$	:	polydyne Exponents

Introduction

Valve train mechanism uses intake and exhaust valves to time entry of air and exit of exhaust gases in each cylinder following each cycle of engine operation. Optimized valve train can reduce pumping losses to great extent. Intake and exhaust valves, seals the cylinder combustion pressure and temperature. Valve train decides the volumetric efficiency of engine and hence the performance. Designing optimum valve train for an engine involves number of trade off's. Valves must be opened and closed as soon as possible. However, dynamic forces induced during operation puts limit on how fast the valves are opened and closed. To keep inertia forces to minimum, valve train components must be light in weight at the same time they must have sufficient stiffness to keep the deflection to minimum.

Out of the several types of valve train, Over head valve push rod type valve train is widely used for medium and slow speed engines. This is due to simplicity and compactness of this design. [1]

At Mahindra & Mahindra, new series of engines with increased stroke was developed for power generation and industrial applications that is off road stationary applications. This paper explains design analysis of push rod type valve train for 6 cylinder engine of this series. Prior to valve train design, engine objective have already been defined and bore stroke determined.

Engine Parameters

Off road stationary applications requires medium speed engines which operates on 70 to100 percent load for most of the operating time. Hence it is essential that these engines have minimum fuel consumption in this range. At the same time they should meet the emission norms. Following rating was developed for meeting the requirement for power generation application.

Engine Specifications

Engine configuration – 6 cylinder, turbocharged, water cooled diesel engine
Engine rating – 104 HP @ 1500RPM
Engine capacity – 5859 cc

Air Flow Requirement

Air flow requirement was calculated based on air fuel ratio. Air fuel ratio was matched inline with current design. The value has been established by past experience.

Valve Train Design

Valve Size and Lift

Two valves per cylinder were used for this engine. Valve sizes were considered in proportion to bore diameter as mentioned in Table 1. Proportions are derived based on proven designs and past experience.

Table 1: Valve Proportions with Respect to Bore Diameter

	Intake Min - Max	Exhaust Min - Max
Valve throat diameter	0.39-0.42	0.37-0.39
Valve head outer diameter	0.41-0.43	0.38-0.4
Valve gauge diameter	0.39-0.41	0.37-0.40

Size of inlet throat was optimized such that Gulp factor was not exceeding 0.6 at rated speed and not below 0.25 at all other operating speed. With value of Gulp factor above 0.6 at rated speed, volumetric efficiency starts to fall off sharply [2]. Gulp factor was calculated by following formula

$$G = \frac{2LN}{aC_{Fmean}} \left(\frac{D}{d} \right)^2 \tag{2}$$

Exhaust throat size was kept 0.85 times inlet throat size. This is because slight pressure drop across exhaust valve is acceptable as it does not affect the performance. Considering bore size, injector location and cylinder head design, intake and exhaust valve locations were decided. Figure 1 shows location of intake, exhaust and injector.

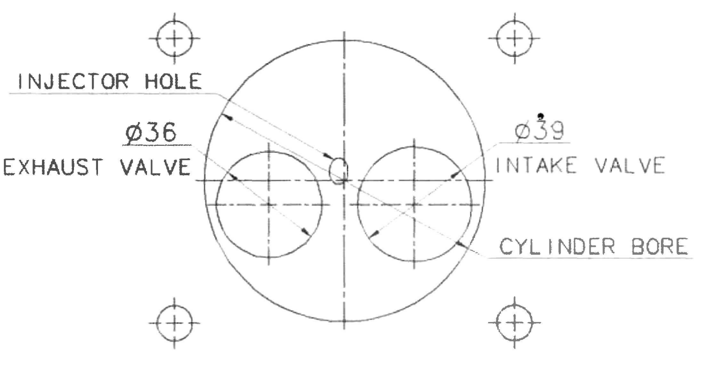


Figure 1: Exhaust and Intake Valve Configuration

Maximum valve lift is determined such that valve area at max lift is equal to throat area. With 45° valve seat angle maximum valve lift was calculated to be 0.25 times port diameter for inlet valve and 0.29 times port diameter for exhaust valve.

Valve Timing

Valve timing for turbocharged diesel engine is different from naturally aspirated engine. Intake valve opening and exhaust valve closing was optimized to get proper scavenging of combustion space and cooling of exhaust components. The earlier intake valve opening was facilitated by providing valve recess in piston top. Exhaust valve opening was optimized for getting required boost pressure from turbocharger and loss of expansion work. Intake valve closing was optimized for time available for air to fill cylinder and loss of compression stroke. Proper valve timing enables the intake pressure to be more than or equal to exhaust pressure at rated point. Due to this pumping losses are zero or negative that is gain in power from compressed air from turbocharger. This is reflected in fuel consumption of engine.

Valve timing was optimized using GT power module of GT suite software. Model of engine was built and analyzed for effect of valve timing on fuel consumption and emission. Optimum valve timing was selected and used for cam design. Figure 2 and 3 shows effect of inlet and exhaust valve opening on fuel consumption and emission.

From Figure 2 and 3 it is clear that inlet valve opening before TDC is advantageous for fuel consumption. Inlet valve opening of 10 degree before TDC is optimum for both fuel consumption and NOx emission. However exhaust valve opening of 50 degree BTDC best for NOx but worst for fuel consumption. The mean value of exhaust valve opening that is 40 degree BDC is chosen as the deterioration in fuel consumption and NOx is not significant compared to best values.

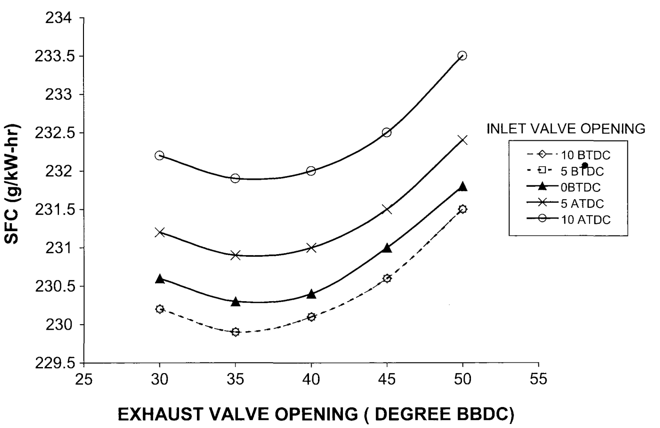


Figure 2: Effect of Inlet and Exhaust Valve Opening on Fuel Consumption

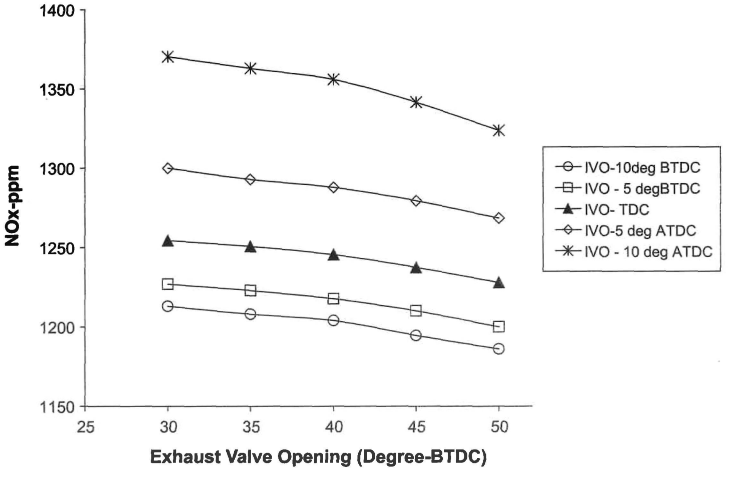


Figure 3: Effect of Inlet and Exhaust Valve Opening on NOx Emission

Valve Train Geometry

A first cut work out of geometry of valve train components was done considering previous design. Rocker arm, tappet and valve retainer was used from earlier design. Valve spring, push rod and cam were designed for satisfying performance and dynamic requirement. Figure 4 shows the over all valve train configuration.

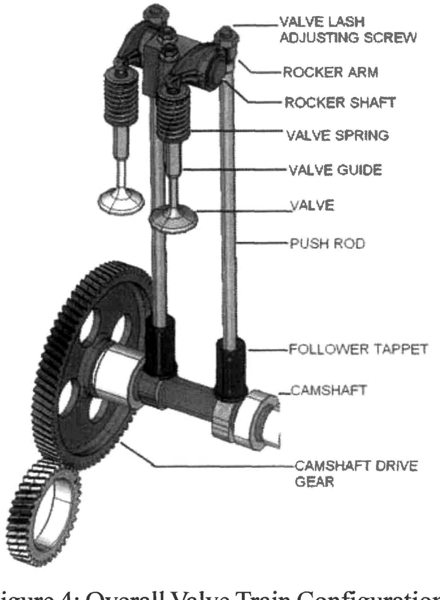


Figure 4: Overall Valve Train Configuration

Valve Train Analysis

Analysis was done by considering equivalent two mass system as shown in Figure 5. Friction in valve guide and rocker lever bearing was neglected.

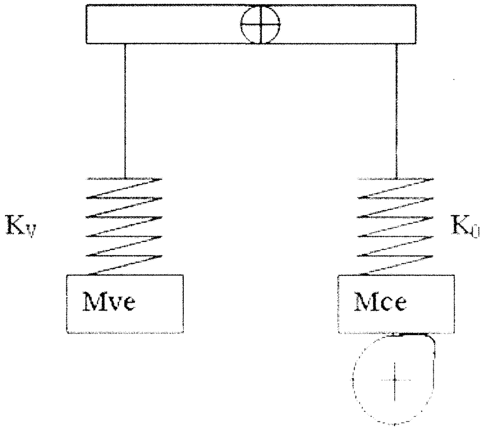


Figure 5: Two Mass System

Two masses are connected by spring representing valve train stiffness K_0. Masses are held in contact with this spring by valve spring of stiffness K_v.

Equivalent mass at valve end and cam end are determined by equations 3 and 4 [3]

$$M_{ve} = m_v + m_{sr} + m_{sp} + m_{re} + \frac{m_p + m_t}{rr^2} \tag{3}$$

$$M_{ce} = m_t + \frac{m_p}{2} \tag{4}$$

$$m_{sp} = \frac{m_s}{3} \text{ (for natural frequency calculation)} \tag{5}$$

$$= \frac{m_s}{2} \text{ (for inertia calculation)} \tag{6}$$

$$m_{re} = \frac{J}{l_v^2} \tag{7}$$

Flat face follower was used for this design as roller follower required major modification in cylinder block which was used from current design.

Stiffness of valve train was determined by Finite element method using Ideas solver. Some load F was applied at valve end and deflection δ was measured at valve end of rocker arm.

$$K_0 = \frac{F}{\delta} \tag{8}$$

Stiffness was calculated to be 5565 N/mm

Natural frequency of valve train was calculated using equation 9

$$fn = \frac{30}{\pi}\sqrt{\frac{1000K_0}{Mve}} \tag{9}$$

Practically, rocker arm design yield variable rocker ratio over the lift period. This was considered in cam design. A rocker ratio variation with reference to lift is shown in Figure 6.

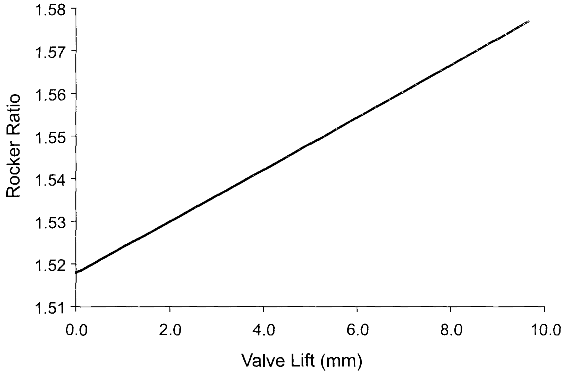


Figure 6: Valve Lift versus Rocker Ratio

Cam Design

Symmetrical Polydyne cam was used for the engine. This type of cam results in theoretically complete compensation of valve train vibrations at design speed. Figure 7 shows geometry of cam. Angle of main event length was decided based on valve timing. Ramp was used on rise and fall sides of cam. It provides margin to accommodate valve clearance, static deflection and change in length due to thermal affects[4]. Ramp consist of constant velocity section adjacent to main event section, followed by transition to base circle by constant acceleration section.

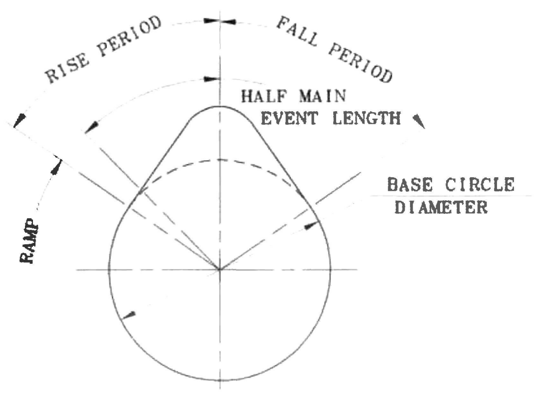


Figure 7: Cam Geometry

Valve lift y_x is determined by considering polynomial in equation 10

$$y_x = L(1 + C2x^2 + C4x^4 + C_p x^p + C_q x^q + C_r x^r + C_s x^s) \tag{10}$$

Where $x = \dfrac{\theta}{\theta_{main}}$

$C2$, C_p, C_q, C_r and C_s are determined by using boundary conditions given in equations 11 -15

For zero valve lift, at cam end $x = 1$

$$y_{x_{x=1}} = 0, \tag{11}$$

$$\frac{dy}{dx}_{x=1} = 0, \tag{12}$$

$$\frac{d^2y}{dx^2}_{x=1} = 0, \tag{13}$$

$$\frac{d^3y}{dx^3}_{x=1} = -\frac{R_h}{c}, \tag{14}$$

$$\frac{d^4y}{dx^4}_{x=1} = 0, \tag{15}$$

$C4$ is shape parameter which decides mainly the transition point. Value of $C4, p, q, r, s$ can be adjusted to meet dynamics and gas exchange requirement. These values are taken by past experience and based on similar kind of engine details.

The equivalent tappet lift is given by equation 16 [4]

$$y_t = R_a + k_r y + c\frac{d^2 y}{dx^2} \tag{16}$$

$c = \dfrac{36N_c M_{ve}}{1000K_0}$ - is dynamic constant.

Actual tappet lift was calculated by dividing rocker ratio with equivalent tappet lift. Radii of cam contour were calculated by using equation 17. The detail derivation of equation 17 can be found in [5]

$$R = R_b + y_{t_{actual}} + \frac{d^2 y_{t_{actual}}}{dx^2} \tag{17}$$

Figure 8 shows variation of intake and exhaust valve lift and actual tappet lift for one complete engine cycle.

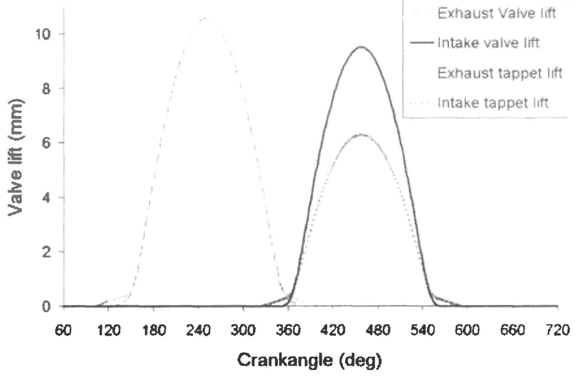


Figure 8: Intake and Exhaust Valve and Actual Tappet Lift

Camshaft Design Speed

Design speed is decided such that maximum over speed is not more than 1.1-1.2 times design speed and no operating point fall at 70% of design speed. N

$$N_c \geq \frac{N_{max}}{1.1} \tag{18}$$

Design of Valve Spring

Tappet tends to bounce of the surface of cam during maximum negative acceleration period . This phenomenon is called as 'jump' or 'bounce'. Function of spring is to keep tappet in contact with cam surface during entire operating speed range.

Valve spring was designed considering following design criteria:

1. To avoid jump spring force must be 1.3-1.5 times the maximum negative acceleration force.
2. Valve spring natural frequency must be sufficiently high in order to avoid excessive valve spring vibration. Recommended range is 12-14 times rated camshaft speed.
3. Spring preload should not be more that half the maximum spring force.
4. Spring stresses must be below permissible valve.

Figure 9 shows valve spring force and Inertia force variation with cam angle.

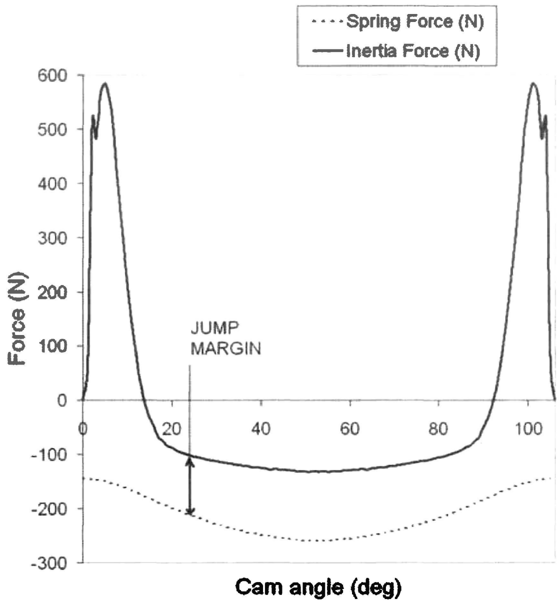


Figure 9: Variation of Inertia Force and Valve Spring Force with Cam Angle

Length of positive acceleration period was optimized to avoid jump phenomenon. For low vibration operation of the valve train at speeds, the length of positive acceleration pulse should be at least 1.2-1.3 times the angle of camshaft rotation occurring within the time of one valve train vibration period at design speed.

$$A_p = 1.25 \frac{360 N_c}{f_n} \tag{19}$$

Where f_n is calculated by equation no 6.

Cam Tappet Contact Stresses

Hertz contact stress was calculated using equation 20 [7]. Poisson's ratio is assumed as 0.3 for both cam and tappet material5.

$$S_c = 0.59\sqrt{\frac{F_c E_1 E_2}{R(E_1 + E_2)b}} \tag{20}$$

With increase in speed, contact force and contact stress increases within acceleration period of cam. This is because spring force and inertia force superimpose within acceleration period. Whereas, within deceleration period of cam, inertia force counteract spring force and contact stress and contact force is higher at lower speed. Therefore for design purpose, contact stresses were determined at cam design speed for acceleration period and zero speed condition was considered for determining contact stress within deceleration period. Figure 10 shows cam-follower contact stress variation at cam design speed. The maximum contact stress, out of the mentioned two cases was found less than permissible value of stress for combination of steel cam and steel follower.

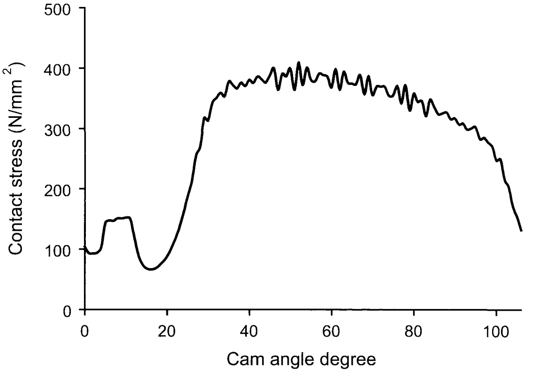


Figure 10: Cam – Tappet Contact Stress

Camshaft Design

Camshaft was considered as simply supported beam with two bearing supports at ends. Contact force is exerted by valve train on cam lobes. Camshaft was designed for torsion and bending load acting from valve train. Variation of drive torque with cam angle for main event length is shown in Figure 11. Considering factor of safety diameter of camshaft was derived.

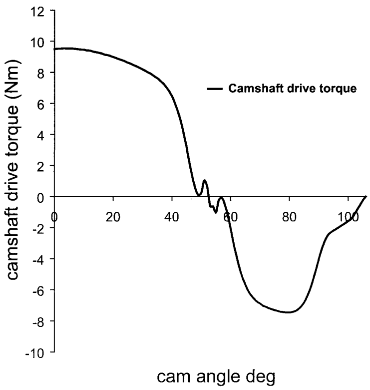


Figure 11: Variation of Camshaft Drive Torque with Respect to Cam Angle

The deflection was calculated by equation 21 [3]. Deflection should be minimum to increase stiffness of valve train. Calculated deflection was found with in acceptable limit.

$$y = \frac{0.8 S_{f_{\max}} i^2 j^2}{d_C^4 E l} \tag{21}$$

Journal diameter was chosen considering the crankcase geometry. Copper lead bearing were used to support the journals of camshaft. Bearing Clearance was maintained 0.001 times journal diameter.[8]. Width of journal and bearing was optimized to get Minimum oil film thickness greater than minimum acceptable value of 0.001 mm.

Performance Simulation

Performance of engine was simulated using GT Power model. The valve lift profile, valve timing and valve train component dimensions were given as input to the model and simulation was carried out. Figure 12 shows variation of fuel consumption with power out of engine. the fuel consumption curve is almost flat from 50% load to 100% load. The best value of fuel consumption is predicted at 75% load. Thus the design satisfies major requirement of off-road stationary application.

Figure 13 shows that value of NOx emission increases with increase in load. This is because as load increases turbine speed increases and more air is supplied to engine resulting in higher NOx emission. The value of NOx in ppm is lower than that predicted in valve timing optimization results. This is due to retarded fuel injection timing considered for predicting the final performance of engine.

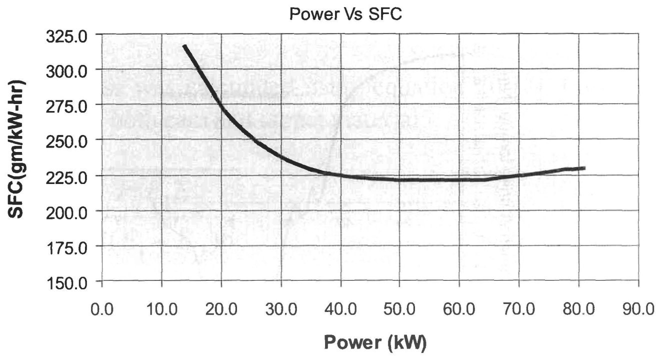


Figure 12: Predicted Fuel Consumption

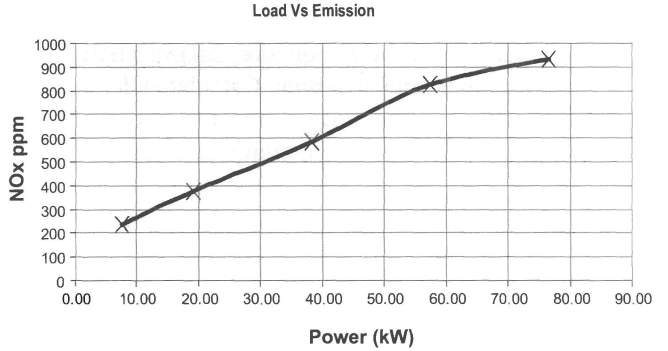
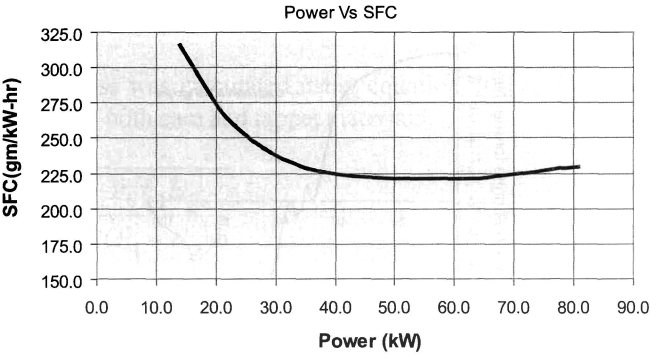


Figure 13: Predicted Nitrogen Oxide Emission

Conclusion

Valve train satisfying the gas exchange and dynamic behavior requirement was designed. The gas exchange requirement were achieved by optimizing valve timing. The engine simulated performance is satisfying the target performance. Fuel consumption is minimum over the range of 50 to 100 percent load which is the major requirement of the application for which engine is designed. The dynamic behavior of valve train was considered in terms of vibrations and contact stresses. Vibrations of valve train was considered in terms of two parameters namely length of positive acceleration pulse and spring force during negative acceleration period. The variations of inertia force and spring force due to vibration is not studied in detail however the margin of spring force over inertia is kept sufficiently higher to take care of these variations and avoid valve jump. Contact stress at point of contact between tappet and cam were kept below



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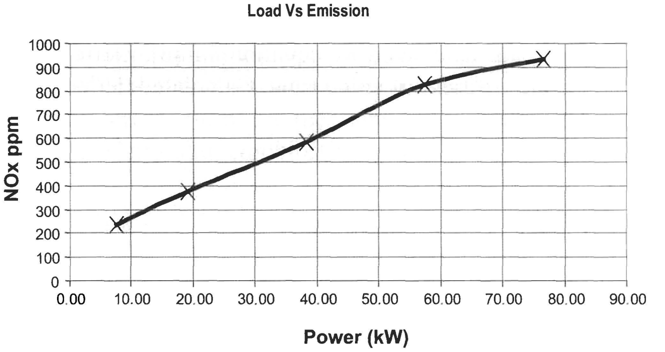

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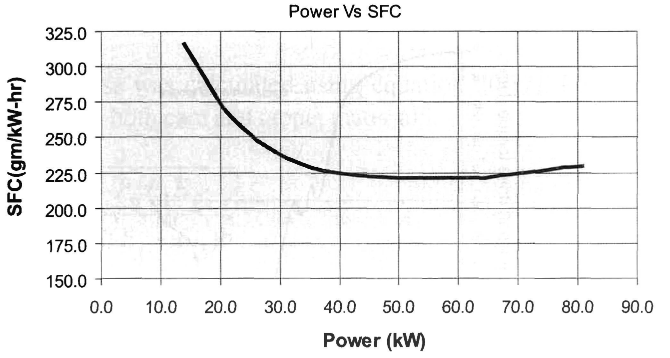

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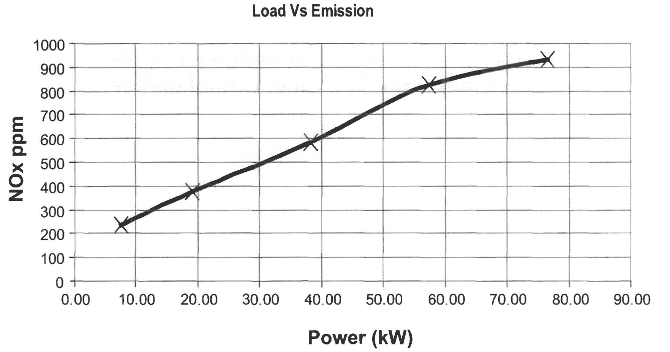

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maximum permissible level. All other parameters were optimized to satisfy the constraints which were established on previous design.

Acknowledgement

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