676. Nonlinear vibrational analysis of diesel valve gear

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Abstract. A valve gear system is currently more or less a classical mechanical system in a majority of diesel engines. In our case, diesel engine durability test was carried out incorporating a conventional valve gear. After the 3000-hour engine test, the manifestations of an intensive wear detected on the exhaust side of the valve gear. The wear of the exhaust cam was particularly intensive right under the top. As evident from dynamical analysis and experimental data of the valve gear the loads are excessive mainly on the exhaust cam. The assessment of the minimum oil film thickness at the top of the exhaust cam does not provide any favorable results. As the largest loads are exerted at the top of the cam, where the highest wear was measured, it is necessary to reduce the normal stresses and improve the lubrication properties. Dynamic valve gear analysis was performed in order to estimate cam wear intensity and to compare the new cam designs with the existing ones. The model of the actual valve gear design can be represented by means of the partial differential equations. We used a vibrational system with three degrees of freedom. Partial differential equations were solved by means of finite differences and Runge-Kutta methods. Vibrational analysis allows studying kinematic and dynamic behavior of the nonlinear spring and nonlinearity of the damping forces in the valve gear system. The presented theory will be applied in the future research works for determining the conditions for chaotic behavior of the valve gear system. The dynamic model of the valve gear was used to analyze the causes of the excessive wear. Since the cam, designed in accordance with the polysine curve, offered too few possibilities for an optimum cam profile, we wanted to manufacture a completely new type of cam with more possibilities for an optimum adjustment. At the same time, we attempted to summarize some findings on the dependence of the cam design on the fuel consumption and valve gear noise. Thereby a new MULTICAM cam was developed. Contrary to the conventional theory of polysine cam, the motion in the MULTICAM cam can be written by means of seven curves. The analysis of Hertz pressures provides more favorable results for the MULTICAM cam profile. By using the new cam profile the Hertz pressures were substantially reduced. The top of the cam is subjected to the lowest loads with the MULTICAM curve shape. In spite of a higher contact force the normal stresses are lower mainly due to the higher radius of cam curvature. Dynamic analysis demonstrates that both newly designed cams exhibit lower stresses at the top of the cam and better lubrication properties, whereas the flow geometrical cross-sections and the other control values remain similar for all three cam versions.

Keywords: diesel valve gear vibration, nonlinearity, cam design, valve gear dynamics.

Introduction

Vibrational analysis of diesel valve gear system is of vital importance. Gear system in modern truck represents one of the most important sectors of the diesel engine. This paper describes the nonlinear vibrational analysis of a new type of valve gear. Dynamic valve gear analysis was performed in order to estimate cam wear intensity and to compare the new cam designs with the existing ones. The model of the actual valve gear design can be represented by means of the partial differential equations, which are difficult to solve. We used vibrational systems with three degrees of freedom. The differential equations were solved by means of finite differences and Runge-Kutta methods. The proposed models allow analysis of the kinematic and dynamic behavior of the nonlinear valve spring. On the basis of presented theory we will find in the future research the conditions for chaotic behavior of the valve gear system.

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Existing Valve Gear

In our case, the engine durability test was carried out in diesel engine, incorporating a conventional valve gear (Fig. 1).

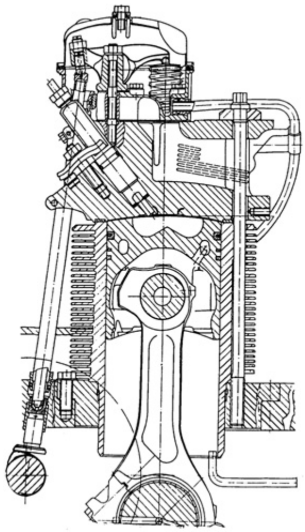

Fig. 1. Diesel valve gear design

After the 3000-hour engine test the manifestations of an intensive wear were detected on the exhaust side of the valve gear. The wear of the exhaust cam was especially intensive right under the top. This type of wear is referred to as the surface wear and it occurs at low peripheral speeds, when the lubrication properties are aggravated and high friction is induced between the cam and the follower when the side pressure is high. Both the inlet and exhaust cams in the test engine were designed in accordance with the polysine curve theory [2].

Some basic data on cams are indicated in Table 1.

Table 1. Basic data on cam

	Inlet cam	**Exhaust cam**
Cam width (b_N)	23 mm	23 mm
Base circle radius (r_0)	24 mm	20.8 mm
Highest lift (h)	7.9 mm	9.0 mm
Ramp angle (Φ_1)	14°	14°
Acceler. angle (Φ_2)	18°	23°
Angle at highest lift (Φ)	67°	76.5°
Valve clearance in cold	0.2 mm	0.3 mm

New MULTICAM cam design

By using the dynamic model of the valve gear we analyzed the causes of excessive wear. Since the cam (NI1), designed in accordance with the polysine curve, offered too few possibilities for an optimum cam profile, we wanted to manufacture a completely new type of cam with more possibilities for an optimum adjustment. At the same time, we attempted to summarize some findings of authors on the dependence of the cam design on the fuel consumption and valve gear noise. Thereby the new MULTICAM cam (NI2) was created.

By designing the new cam profile we aimed at reduction of stresses primarily at the top of the cam without having to substantially change the other parameters. The following factors had to be taken into account:

1- We did not want to considerably change the basic characteristics of the cam (lift, angle of opening).
2- Excessive accelerations and changes in the speed of accelerations by cam follower lifting and lowering have to be avoided.
3- Geometrical cross-sections of valves do not have to be substantially changed.
4- The position between the valve travel and piston around TDC has to be checked to prevent any contact in this area.
5- We did not want to change the layout and design solution of the valve gear.
6- We had to verify the intensity of the inertia and spring force due to the potential interruption of the contact.

Contrary to the conventional theory of polysine cam the motion in MULTICAM cam can be written by means of seven curves. Equations (1 to 7) present the lift of the cam in the vertical direction.

Zone $0<\varphi_0<\Phi_0$: $$h_0 = -\frac{C_1}{2}\left(\frac{\Phi_0}{\pi}\right)^2 cos\left(\frac{\varphi_0\pi}{\Phi_0}\right) + \frac{C_1}{4}\varphi_0^{\,2} + k_1\varphi_0 + k_2 \quad (1)$$

Zone $0<\varphi_1<\Phi_1$: $$h_1 = -\frac{C_2}{2}\left(\frac{\Phi_1}{\pi}\right)^2 sin\left(\frac{\varphi_1\pi}{\Phi_1} - \frac{\pi}{2}\right) + \frac{C_2}{4}\varphi_1^{\,2} + k_3\varphi_1 + k_4.. \quad (2)$$

Zone $0<\varphi_2<\Phi_2$: $$h_2 = \frac{C_2}{2}\varphi_2^{\,2} + k_5\varphi_2 + k_6 \quad (3)$$

Zone $0<\varphi_3<\Phi_3$: $$h_3 = \frac{C_2}{2}\varphi_3^{\,2} - \frac{C_2}{12}\cdot\frac{\varphi_3^{\,4}}{\Phi_3^{\,2}} + k_7\varphi_3 + k_8 \quad (4)$$

Zone $0<\varphi_4<\Phi_4$: $$h_4 = \frac{C_3}{12}(\Phi_4 - \varphi_4)^4 - C_5\frac{\varphi_4^{\,2}}{2} + k_9\varphi_4 + k_{10} \quad (5)$$

Zone $0<\varphi_5<\Phi_5$: $$h_5 = -\frac{C_5}{2}\left(\frac{\varphi_5}{\pi}\right)^2 sin\left(\frac{\pi\varphi_5}{\Phi_5} - \frac{\pi}{2}\right) - \frac{C_5}{4}\varphi_5^{\,2} + k_{11}\varphi_5 + k_{12} \quad (6)$$

Zone $0<\varphi_6<\Phi_6$: $$h_6 = C_6 + k_{13}\varphi_6 + k_{14} \quad (7)$$

The constants in equations (1-7) were obtained by means of boundary conditions.

Vibrational model based on finite differences

Dynamic valve gear analysis was performed in order to estimate cam wear intensity and to compare the new cam designs with the existing ones. In the references [3] and [4] we have developed the mathematical model for the rigid system and linear vibrational system.

The real valve gear system [5] (shown in Fig. 1) is unsuitable for the formulation of an equation of movement, mainly because it is described by means of partial differential equations. Consequently, the real system is replaced by an equivalent three-mass system (shown in Fig. 5), which may be described through three ordinary second-order differential equations. In this reaction, every element of the system is represented by two concentrated masses, connected by a weightless spring, having the stiffness of this element. In addition, rotational movement of a rocker arm is replaced by linear movement. Portions of masses of two adjacent elements are integrated into a single mass (e. g. a portion of the mass of a push rod and a portion of the mass of a rocker arm form a single mass in the equivalent system), resulting in a system of three concentrated masses, interconnected with springs. Reduced masses and reduced stiffnesses of the equivalent system are determined on the basis of equality of kinetic and potential energy of the real and equivalent system.

Table 2. Optimum cam angles

New cam profile	MULTICAM (NI2)
Full lift (H)	8.65 mm
Ramp lift (H_o)	0.25 mm
Half angle ϕ	80°
Angle ϕ_0	10°
Angle ϕ_1	4.5°
Angle ϕ_2	0.3°
Angle ϕ_3	15.5°
Angle ϕ_4	42°
Angle ϕ_5	7.4°
Angle ϕ_6	0.3°

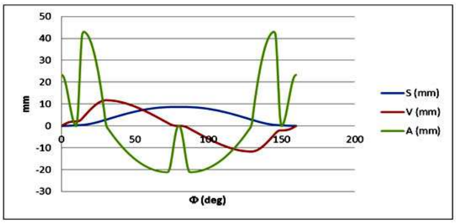

Fig. 2. Lift, velocity and acceleration profile for MULTICAM cam

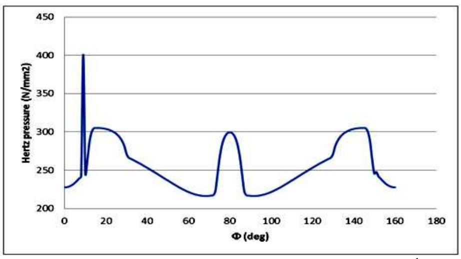

Fig. 3. Hertz pressure for n=2100 min^{-1}

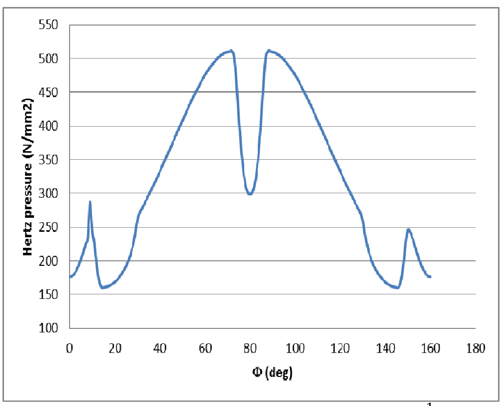

Fig. 4. Hertz pressure for n=500 min^{-1}

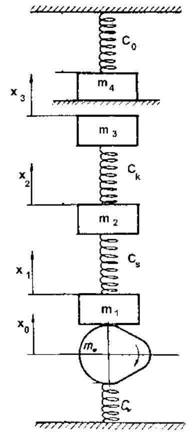

Fig. 5. The mathematical model

The concentrated masses shown in the equivalent system model of the engine valve gear (Fig. 5) are identified in the following way:

- mass m_o represents a mass of a part of the camshaft related to the corresponding valve gear system:

$$m_0 = m_{bv} \tag{8}$$

The valve gear equivalent system model (Fig. 5) serves for the formulation of the movement equations of individual masses. The movement of masses m_1 and m_o is defined by the equations:

$$m_1\ddot{x}_1 + m_0 x_0 + c_s\left(x_1 - x_2\right) + c_{bv} x_0 = 0 \tag{9}$$

$$x_0 = x_1 - h_p \tag{10}$$

where x_1 is the cam follower movement or its actual lift, h_p is cam follower movement, defined by cam profile. The movement of mass m_2 is determined by the equation:

$$m_2\ddot{x}_2 + c_k\left(x_2 - x_3\right) + c_s\left(x_2 - x_1\right) = 0 \tag{11}$$

At the beginning of the movement, when $h_p < h_z$, the movement equation for mass m_3 has the form:

$$m_3\ddot{x}_3 + c_k\left(x_3 - x_2\right) = 0 \tag{12}$$

After the compensation of clearance ($h_p \geq h_z$) mass m_4 is also involved in the movement, opposed by the valve spring force F_0 and the force of gases F_g in the cylinder, acting on the valve:

$$\left(m_3 + m_4\right)\ddot{x}_3 + \left(x_3 - x_2\right)b_k + \left(x_3 - x_2\right)c_k + c_0 x_3 = -F_0 - F_g \tag{13}$$

By rearranging the equations (2) and (3) we obtain:

$$\left(m_1 + m_0\right)\ddot{x}_1 + c_s\left(x_1 - x_2\right) + c_v x_1 = m_0\ddot{h}_p + c_{bv} h_p \tag{14}$$

In this case, equations (8) and (9) may be written in the form:

$$m_3^*\ddot{x}_3 + c_k\left(x_3 - x_2\right) + c_0^* x_3 = F^* \tag{15}$$

$$\text{for } h_p < h_z,\ m_3^* = m_3,\ c_0^* = 0,\ F^* = 0 \tag{16}$$

$$\text{for } h_p \geq h_z,\ m_3^* = m_3 + m_4,\ c_0^* = c_0,\ F^* = -F_0 - F_g \tag{17}$$

where h_z is valve clearance. Movement x_o represents the movement or deflection of the camshaft to be found in a functional relation, resulting from the cam follower movement, described by means of equation (10). New variables may be introduced into the system of equations (9-17), namely:

$z_1 = x_1$ - actual cam follower lift,

z_2 - cam follower push rod deflection, (18)
z_3 - rocker arm deflection.

On the basis of Eqs. (17) we can write the next expression:

$$x_1 = z_1,\ x_2 = x_1 - z_2 = z_1 - z_2,\ x_3 = x_2 - x_3 = z_1 - z_2 - z_3 \tag{19}$$

By differentiating the preceding expressions we obtain:

$$\dot{x}_1 = \dot{z}_1,\ \dot{x}_2 = \dot{z}_1 - \dot{z}_2 = \dot{z}_1 - \dot{z}_2,\ \dot{x}_3 = \dot{x}_2 - \dot{x}_3 = \dot{z}_1 - \dot{z}_2 - \dot{z}_3$$
$$\ddot{x}_1 = \ddot{z}_1,\ \ddot{x}_2 = \ddot{z}_1 - \ddot{z}_2 = \ddot{z}_1 - \ddot{z}_2,\ \ddot{x}_3 = \ddot{x}_2 - \ddot{x}_3 = \ddot{z}_1 - \ddot{z}_2 - \ddot{z}_3$$

By introducing these variables into the system of equations (13-18) a final form of differential equations is obtained, describing the movement of the equivalent model of the valve gear system:

$$\begin{bmatrix} m_1 + m_0 & 0 & 0 \\ m_2 & -m_2 & 0 \\ m_3^* & -m_3^* & -m_3^* \end{bmatrix} \begin{Bmatrix} \ddot{z}_1 \\ \ddot{z}_2 \\ \ddot{z}_3 \end{Bmatrix} + \begin{bmatrix} C_{bv} & C_s & 0 \\ 0 & -C_s & C_k \\ C_0^* & -C_0^* & -\left(C_0^* + C_k\right) \end{bmatrix} \begin{Bmatrix} z_1 \\ z_2 \\ z_3 \end{Bmatrix} = \begin{Bmatrix} m_0 \ddot{h}_p + C_{bv} h_p \\ 0 \\ F^* \end{Bmatrix} \tag{20}$$

or in the final form:

$$[M]\{\ddot{z}\} + [K]\{z\} = \{F\}_t \tag{21}$$

$$\{z\}_{t+\Delta t} = [A]^{-1}\left(\{F\}_t - [B]\{z\}_t - [G]\{z\}_{t-\Delta t}\right) \tag{22}$$

$$[A] = \frac{1}{\Delta t^2}[M],\ [B] = [K] - \frac{2}{\Delta t}[C],\ [G] = \frac{1}{\Delta t^2}[M] \tag{23}$$

From this equation the value of the unknown variable z may be obtained in the next moment, if its values are known in two previous moments. This means that for the initial moment the values of the unknown variable z must be known, while subsequently the value of the unknown variable z may be found for an arbitrary moment of time through a predetermined sufficiently small time interval.

Runge-Kutta vibrational model

We can transform Eq. (20) to solve the problem on the basis of Runge-Kutta numerical solution:

$$\dot{z}_1 = z_4, \dot{z}_2 = z_5, \dot{z}_3 = z_6 \tag{24}$$

$$\begin{Bmatrix} \dot{z}_4 \\ \dot{z}_5 \\ \dot{z}_6 \end{Bmatrix} = [M]^{-1} \left[\{F\} - [C] \begin{Bmatrix} \dot{z}_1 \\ \dot{z}_2 \\ \dot{z}_3 \end{Bmatrix} - [K] \begin{Bmatrix} z_1 \\ z_2 \\ z_3 \end{Bmatrix} \right] \tag{25}$$

Runge-Kutta nonlinear model

In the presented paper we have taken into account the nonlinearity of valve spring. The spring force in the model is represented with the next equation:

$$F_{spring} = C_{01} x + C_{02} x^n \tag{26}$$

The first part on the right side shows linear part and the second part presents the nonlinear contribution of n-th order:

$$\dot{z}_1 = z_4, \dot{z}_2 = z_5, \dot{z}_3 = z_6 \tag{27}$$

$$\begin{Bmatrix} \dot{z}_4 \\ \dot{z}_5 \\ \dot{z}_6 \end{Bmatrix} = [M]^{-1} \left[\{F\} - [C] \begin{Bmatrix} \dot{z}_1 \\ \dot{z}_2 \\ \dot{z}_3 \end{Bmatrix} - [K] \begin{Bmatrix} z_1 \\ z_2 \\ z_3 \end{Bmatrix} - \begin{Bmatrix} 0 \\ 0 \\ C_{02}(z_1 - z_2 - z_3)^n \end{Bmatrix} \right] \tag{28}$$

Results and conclusion

Presented mathematical models enable determination and analysis of some very important parameters. By designing and analyzing the valve gear, we seek to reduce the stresses primarily at the top of cam without having to substantially change other parameters. Due to the presented problems we have used MULTICAM cam [3], designed by means of seven curves. At the same time we wanted to analyze the influence of valve spring nonlinearity on valve and cam follower lift and to analyze the resulting deflections.

Fig. 6 shows, on the basis of the finite differences method, the cam follower push rod deflection (z_2), rocker arm deflection (z_3), the deflection of camshaft (z_{bv}) and the complete deflection of the valve gear system (z_{sum}). Fig. 7 shows the deflections obtained by Runge-Kutta 4 linear method. Figures 6 and 7 show the cam lift (z_1) and the deflections obtained by Runge-Kutta 4 nonlinear model with the second order of nonlinearity. Figures (4, 6-7) are obtained for n=500 rpm.

It is evident from Figs. 6-8 that the results obtained with finite differences and Runge-Kutta methods are in good agreement with respect to the amplitude and shape of the frequency spectrum. Figs. 7 and 8 reveal that the linearity has a slight influence on the magnitude of the amplitude and on the shape of the frequency spectrum.

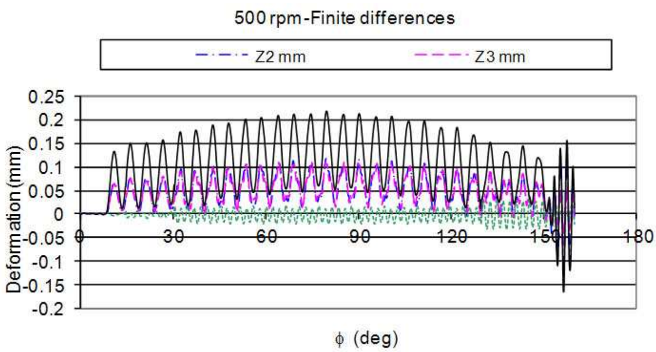

Fig. 6. Numerical results obtained with finite differences method

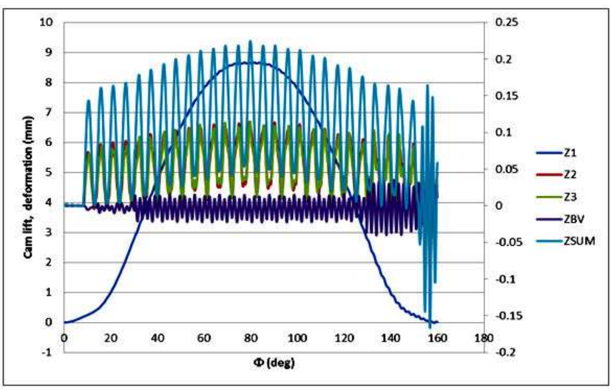

Fig. 7. Deformation with nonlinearity of the second order at n=500 min^{-1}

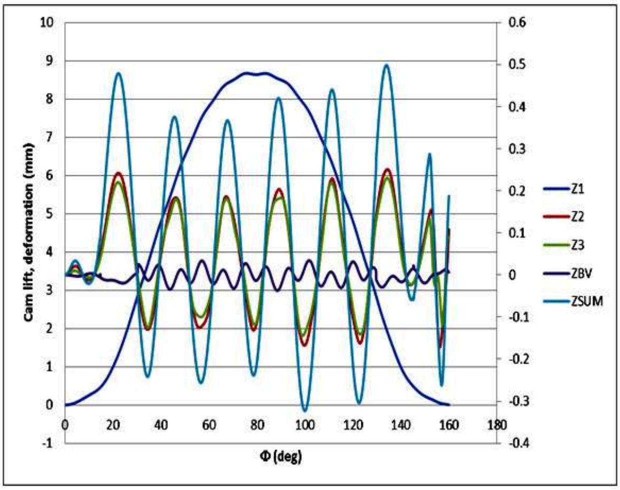

Fig. 8. Deformation with nonlinearity of the second order at n=2100 min^{-1}

Conclusions

The design of the diesel engine valve gear is often accompanied by a large number of complex problems, which have to be solved. In our case the problem was the intensive wear at the top of the exhaust cam profile after the 3000-hour test on the engine. The paper considered vibration systems having three degrees of freedom. We have also analyzed the nonlinearity of the valve spring with the MULTICAM cam design.

The presented article shows the vibrational analysis of the new version of MULTICAM cam and the influence of nonlinearity on MULTICAM cam.

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