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# AN ADAPTIVE DYNAMIC CRANKTRAIN MODEL

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***Summary:*** *The internal combustion engine development process requires CAE models which deliver results for the concept phase at very early stage and which can be further detailed on the same program platform as the development process progresses. The vibratory and acoustic behaviour of the powertrain is a highly complex one, consisting of many components that are subject to loads that vary greatly in magnitude and which operate at a wide range of speeds. The interaction of the crankshaft and engine block is a major problem for the powertrain designer when optimising the vibration and noise characteristics of the powertrain. Finite Element Method (FEM) and Multi-Body Systems (MBS) are suitable for creation of 3-D calculation models. Non-contact measurements with laser vibration meters enable to verify the calculation models.*

**Key words:** internal combustion engine, crankshaft, engine block, bearing, FEM, BEM, torsional vibration meter, laser vibration meter

## 1. Introduction

In the internal combustion engine development process several numerical simulation techniques have continuously gained importance. In the case of the simulation of cranktrain dynamics (see **Fig. 1**) hydrodynamic effects are very important [1].

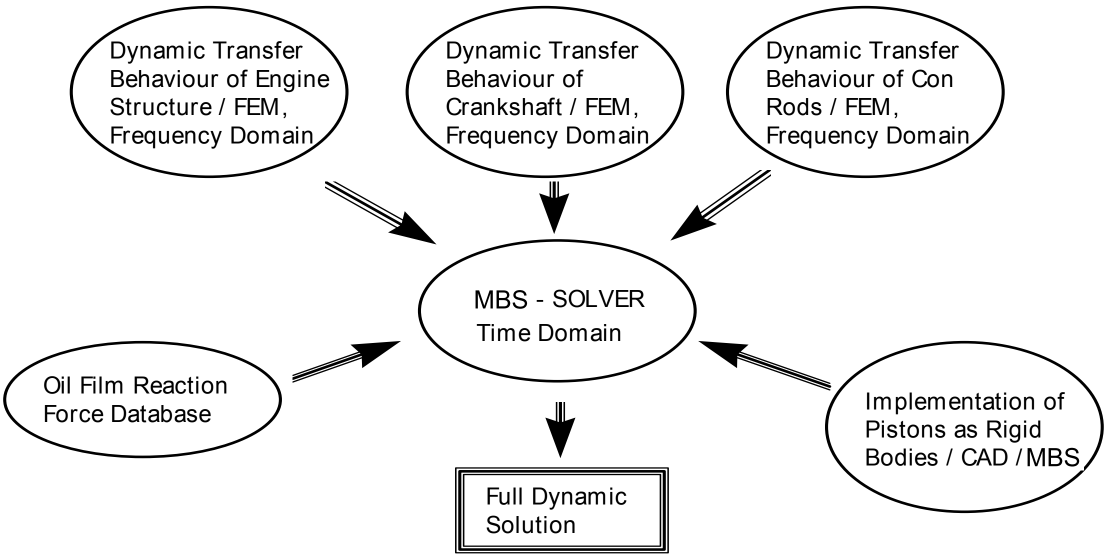


**Fig. 1** Interaction among FEM, MBS and hydrodynamic bearing models

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## 2. Cranktrain models

The engine development process starts with a kinematic model made up of rigid bodies specified with mass and moment of inertia, approximation of which can be derived from the first design sketches for a new engine. The model is excited with its inertia forces under uniform rotation and with a cylinder pressure signal. The simulation supplies important specification criteria for the concept phase, e.g. free engine forces and torques, approximate bearing forces, loads on connecting rods and individual cranks.

In the case of dynamic structural calculations the Finite Element Method (FEM) is state of the art. The calculation of the structural transfer behaviour of single components is efficiently possible in frequency domain, considering a large number of degrees of freedom. Finite element models of engine block and crankshaft of a six-cylinder in-line Diesel engine are shown in **Figures 7** to **10**.

Multi-Body Systems (MBS) are suitable for the calculation of the dynamic relations between several parts. Solving the equations of motion in time domain, the MBS-solvers are useful for the consideration of non-linear effects as well. The reduction of the engine block and crankshaft finite element model to derive a reduced model for ADAMS was performed using ANSYS. The degrees of freedom selected comprised all the critical freedoms which represent the application of loading to the model. Unrestrained modes of vibration of engine block and crankshaft of the six-cylinder in-line Diesel engine are shown in **Figures 11** to **16**.

In the case of the simulation of powertrain dynamics hydrodynamic effects are very important. For the calculation of the radial plain bearing several hydrodynamic simulation codes, mostly based on the solution of the Reynolds differential equation, have been developed for the calculation of the oil pressure distribution in the bearing.

Elasto-hydrodynamic techniques additionally consider the backward influence of local structural deflections of the bearing shell on the oil pressure.

## 3. Calculations and measurements

Comparison of calculations and measurements is still a decisive quality criterion in the CAE environment. Non-contact measurement with laser Doppler vibration meters enable to verify the complex calculation models [2]. The torsional vibration meter B & K Type 2523 (see **Fig. 2**) was designed for making torsional vibration measurements where it is not feasible to mount a transducer onto a rotating object.


**Fig. 2** Torsional vibration meter B & K Type 2523

A schematic of the electronic and optical components in the torsional vibration meter is shown in **Fig. 3**. The central module of the system is a low power Ga-Al-As laser. The laser beam is split into two equal-intensity parallel beams separated by a distance

$$d = R_A \cos \alpha_A + R_A \cos \alpha_B$$

The beams strike the shaft surface at points A and B, where the velocity is $v_A$ and $v_B$, respectively.

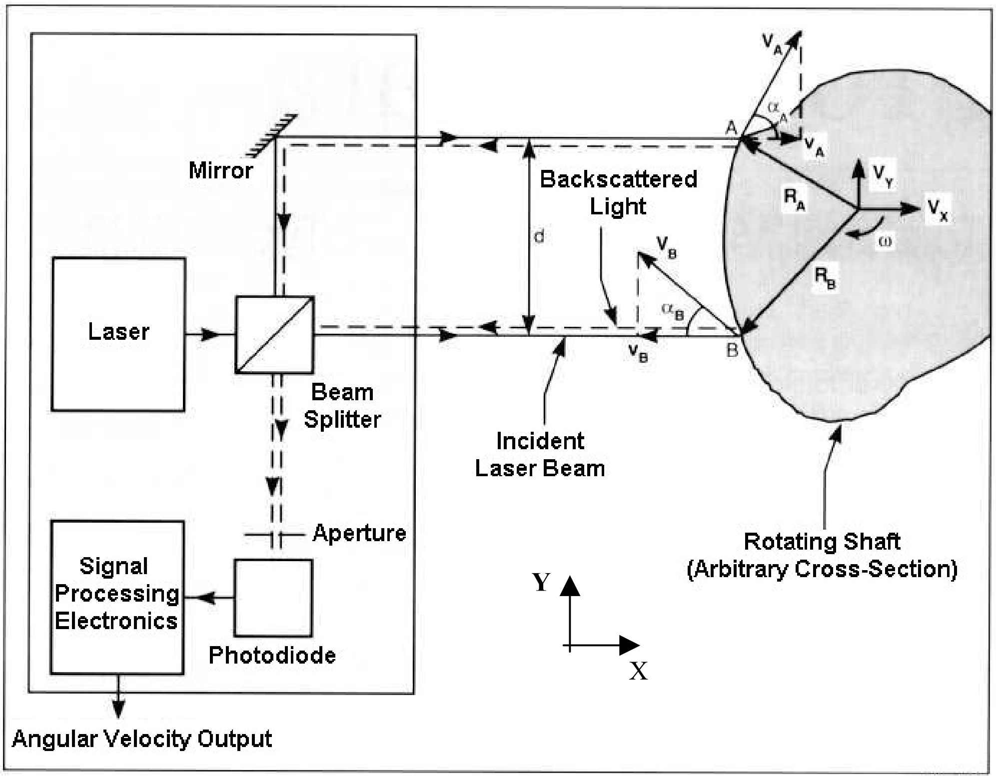


**Fig. 3** The arrangement of the electronic and optical components within the B & K Type 2523

Each beam sees only the velocity in the x-direction

$$v_A = -V_A \cos\alpha_A - V_x = -\omega R_A \cos\alpha_A - V_x$$
$$v_B = V_B \cos\alpha_B - V_x = \omega R_B \cos\alpha_B - V_x$$

and is thus frequency-shifted

$$f_A = \frac{2v_A}{\lambda} \quad \text{and} \quad f_B = \frac{2v_B}{\lambda}$$

The returning beams then heterodyne, giving an output current modulated at the beat frequency, $f_D$, that is, the difference between the frequencies of the Doppler-shifted beams

$$f_D = f_B - f_A = \frac{2}{\lambda}(v_B - v_A) = \frac{2\omega d}{\lambda}$$

Thus, the beat frequency is directly proportional to the shaft speed $\omega$ and is independent of any solid body motion ($v_X + v_Y$) of the shaft. If the plane of the laser beams is not perpendicular to the shaft axis, then $f_D$ is also a function of $\cos\Phi$, where $\Phi$ is the angle between the plane of the laser beams and the plane perpendicular to the axis of shaft rotation.

Torsional vibrations of the six-cylinder in-line Diesel engine have been measured and evaluated by in-house software.

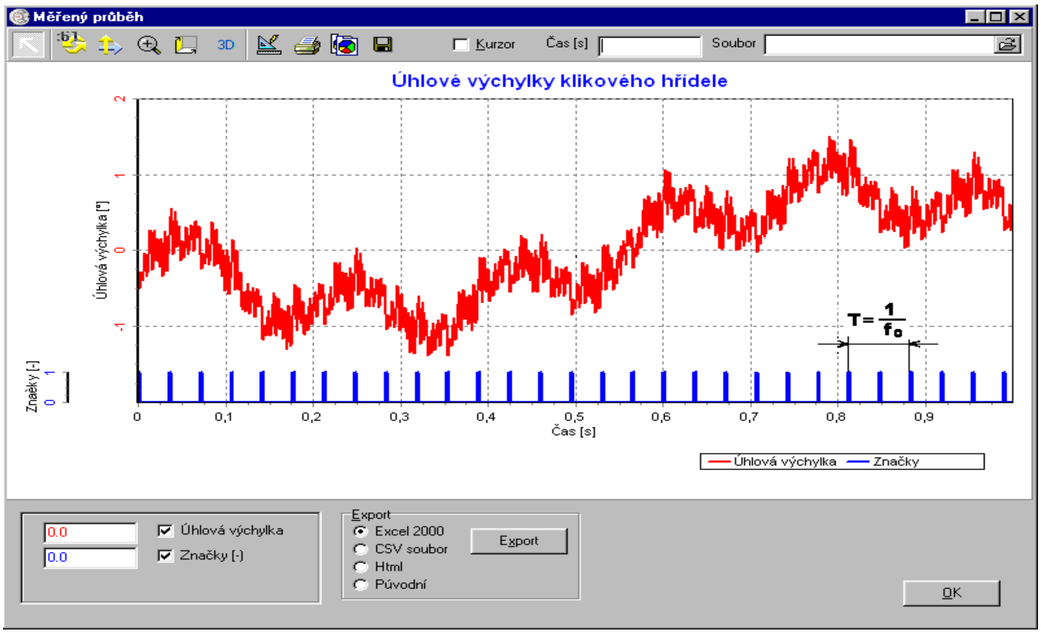


**Fig. 4** Pulley torsional vibration measurement ($n = 1700\ \text{min}^{-1}$)

The measured signal (see **Fig. 4**) has been processed by the averaging filter

$$y(t) = \frac{1}{K}\sum_{k=0}^{K-1} x(t - kT)$$

whose characteristic

$$G\left(\frac{f}{f_0} + i\right) = G\left(\frac{f}{f_0}\right) = \frac{1}{K}\left|\frac{\sin\left(K\pi\dfrac{f}{f_0}\right)}{\sin\left(\pi\dfrac{f}{f_0}\right)}\right| , \qquad i = 1,2,\ldots$$

is periodical (see **Fig. 5**).

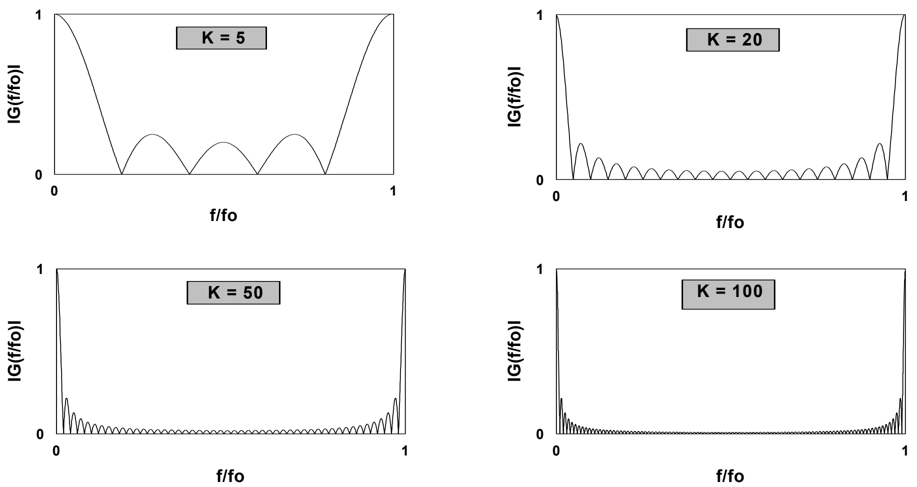


**Fig. 5** Filter characteristics

The torsional vibration of the pulley after averaging is shown in **Fig. 6**.

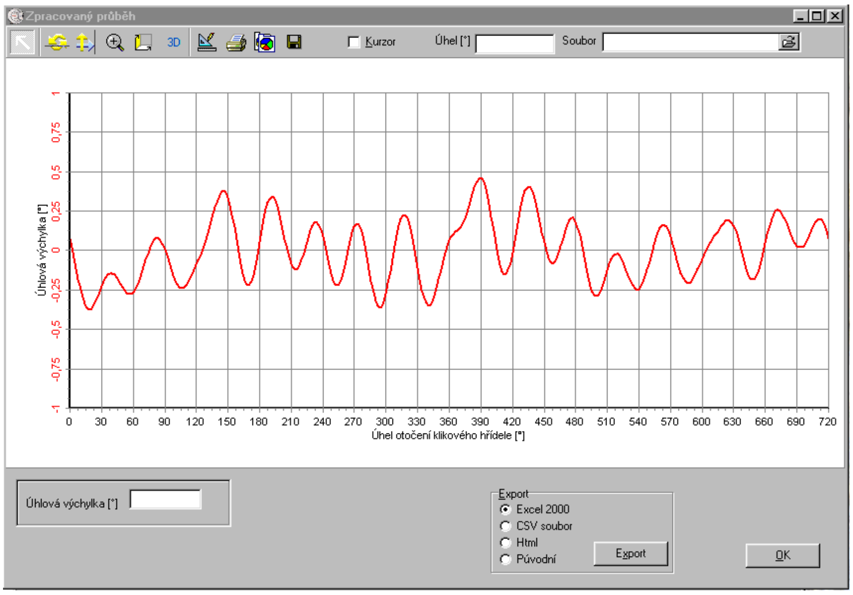


**Fig. 6** Pulley torsional vibrations after averaging (K = 100)

## 4. Conclusion

The simulation of the cranktrain dynamics is a central module of the virtual engine. It will be developed into a tool that enables to calculate the total engine mechanics and to support the design in the development concept phase to make sure that already the first prototype will be on a high level of technical development. Laser measuring technique in combination with the special in-house software is an efficient means to verify complex powertrain dynamic models.

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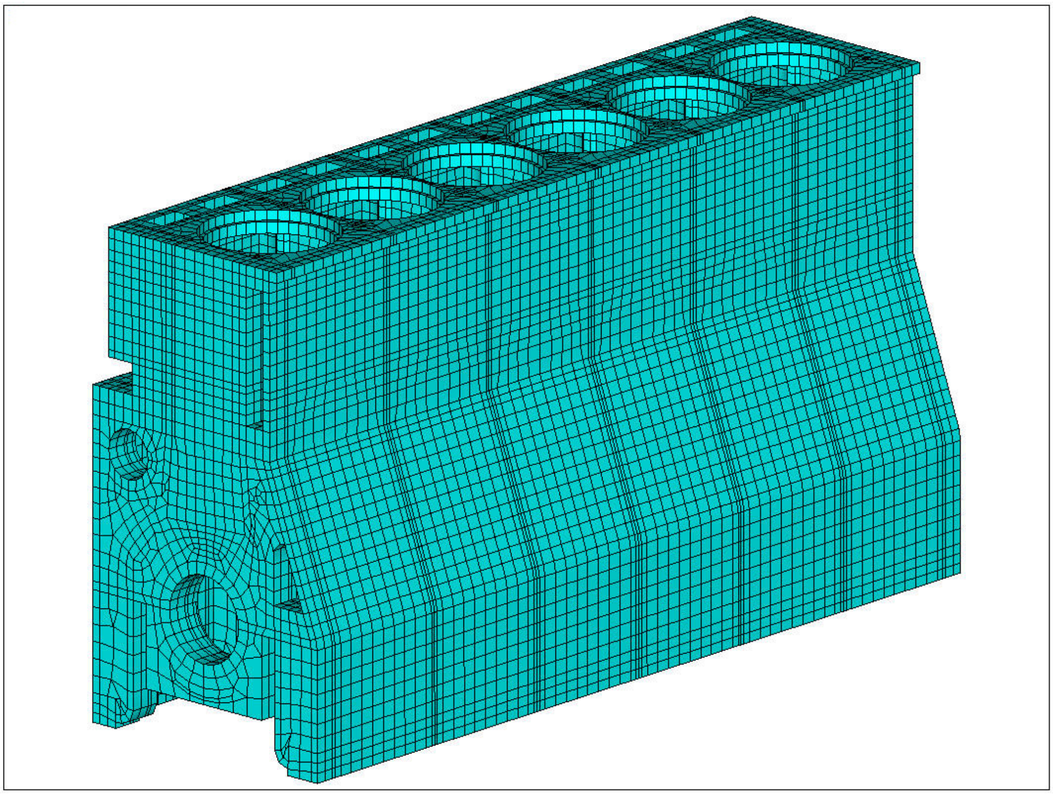

**Fig. 7** Finite element model of engine block

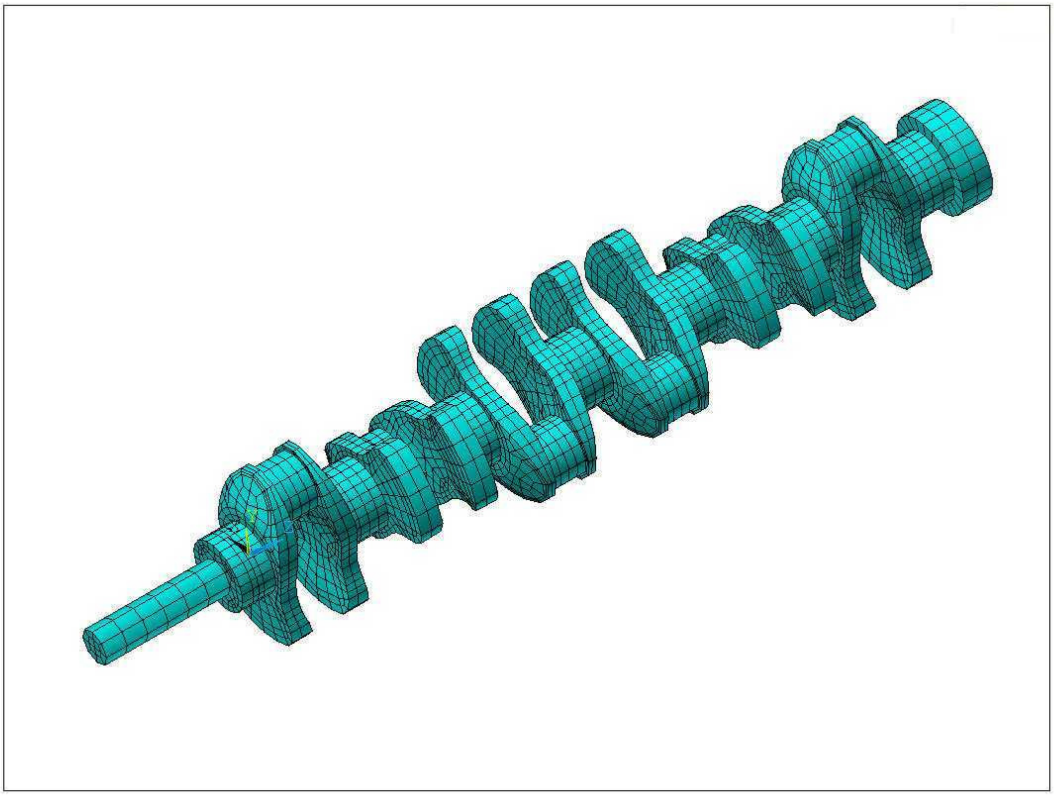

**Fig. 8** Finite element model of crankshaft

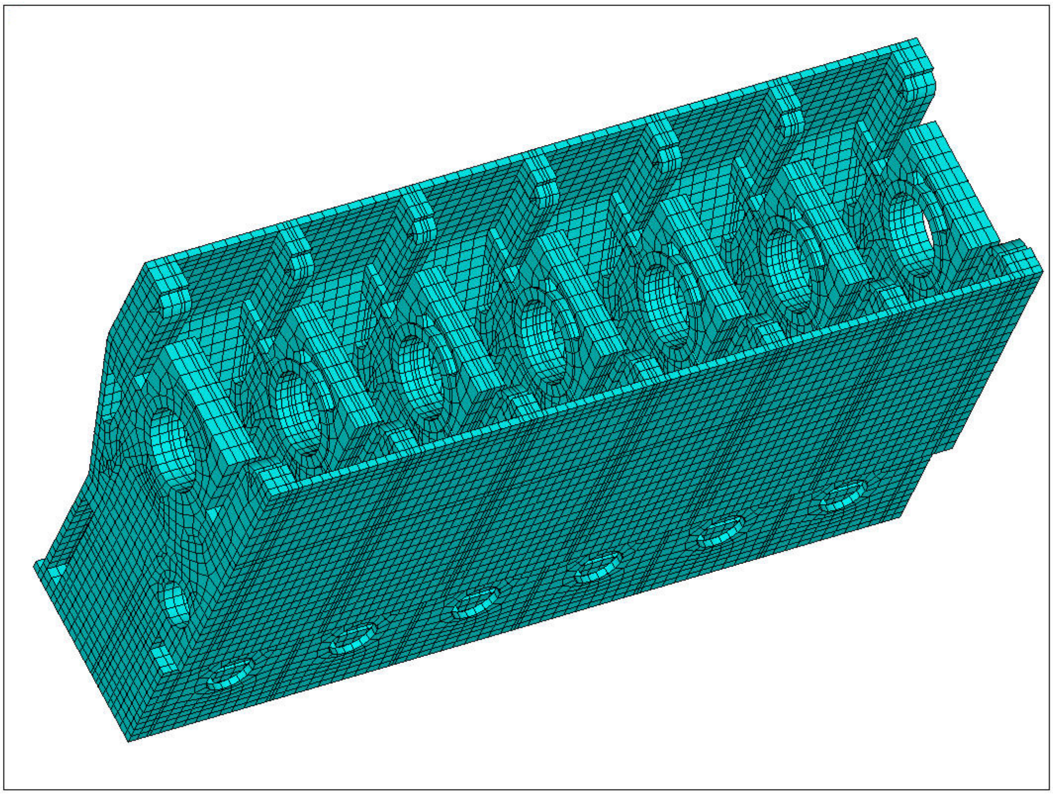

**Fig. 9** Finite element model of engine block

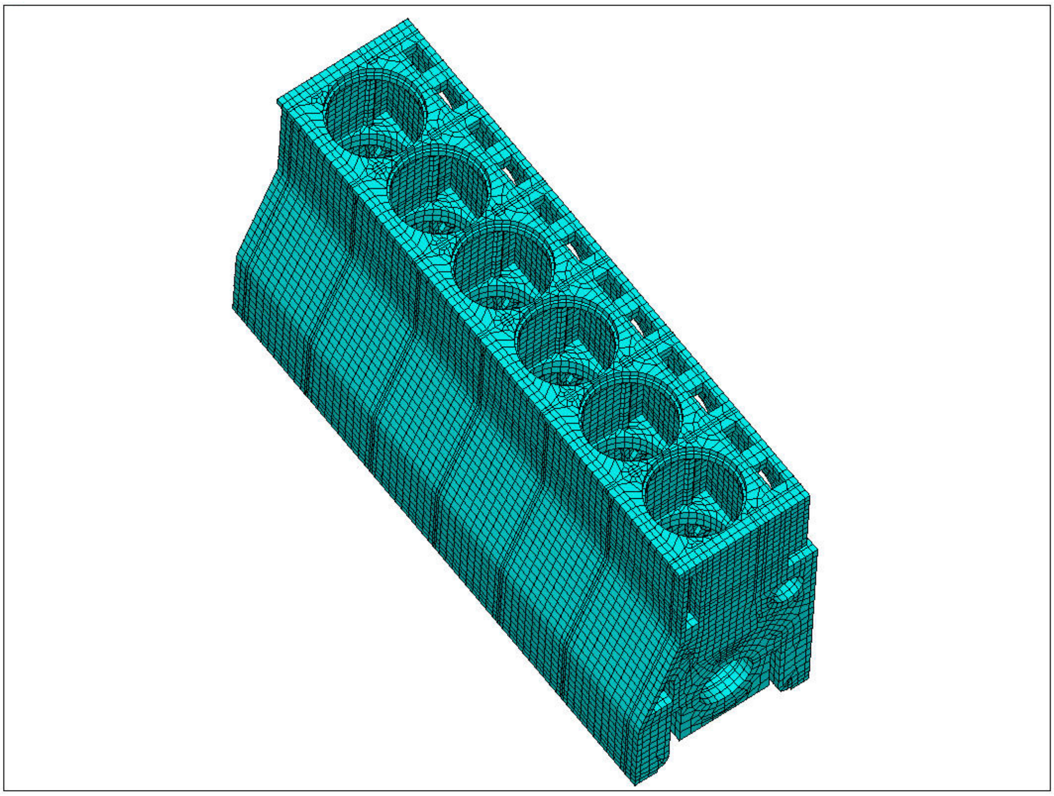

**Fig. 10** Finite element model of engine block

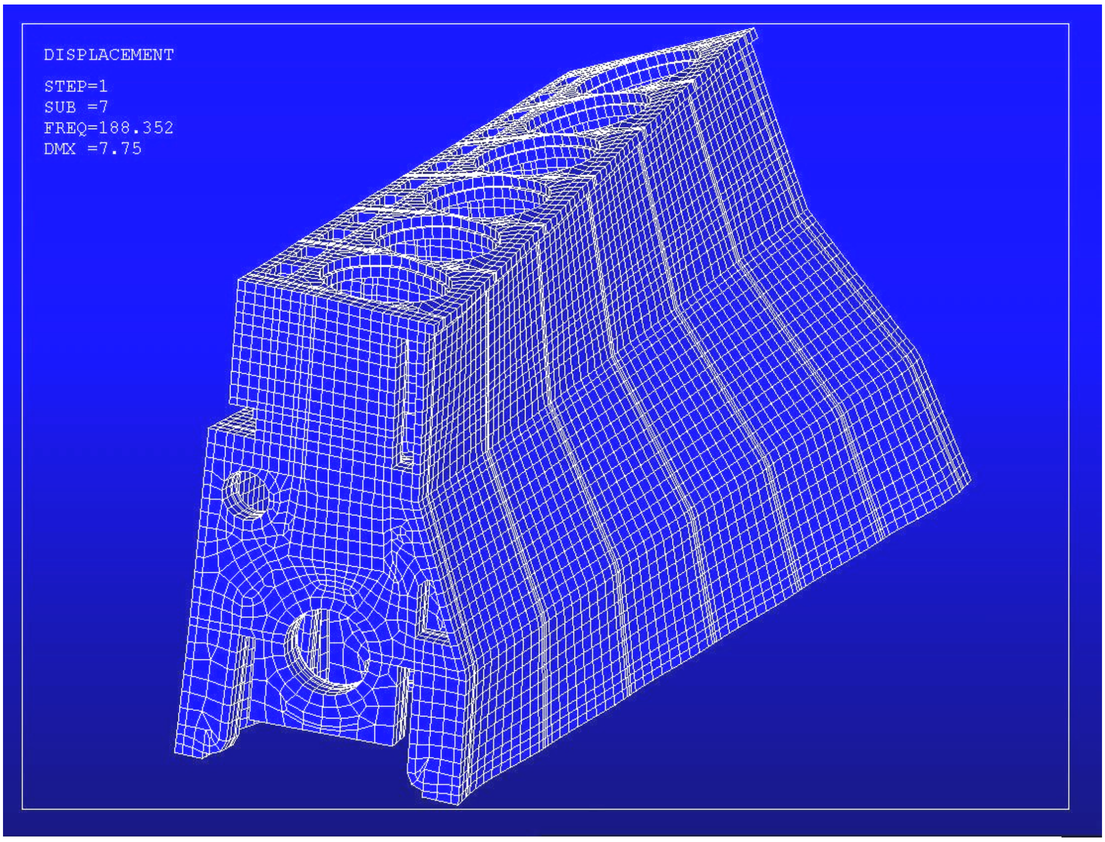


**Fig. 11** The first vibration mode of engine block

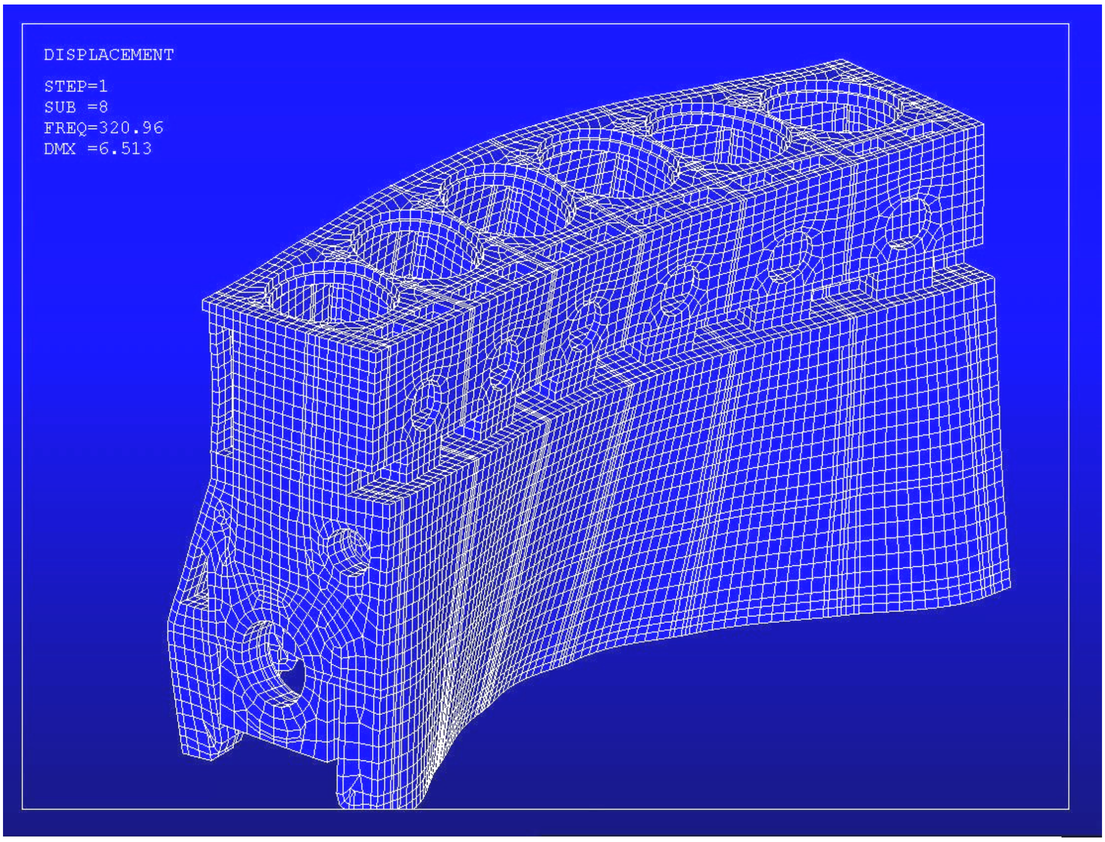


**Fig. 12** The second vibration mode of engine block

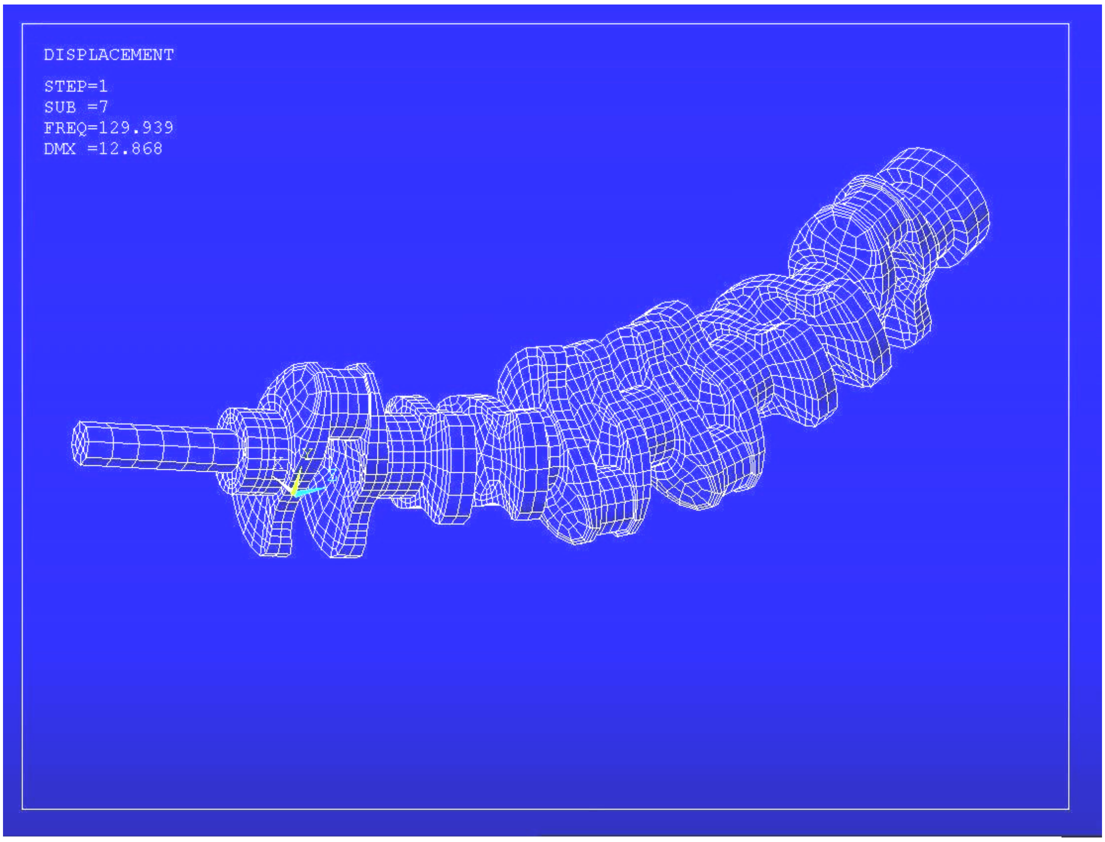


**Fig. 13** The first vibration mode of crankshaft

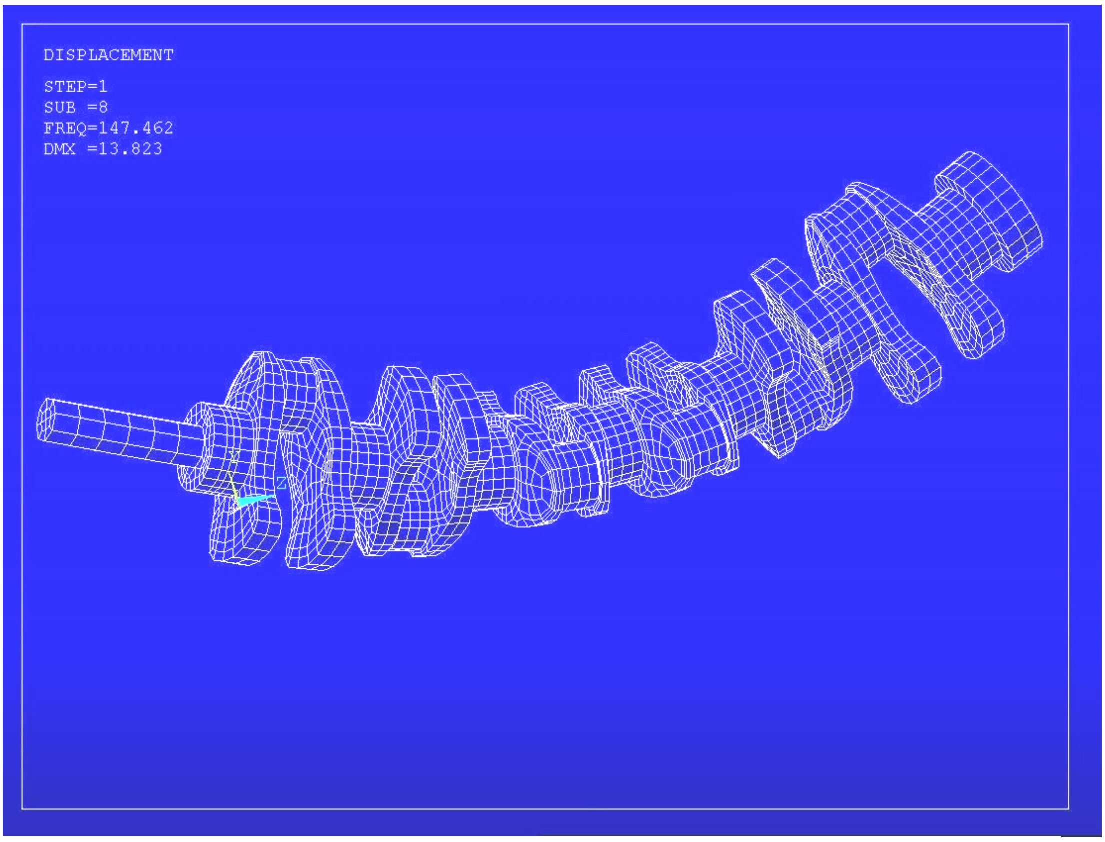


**Fig. 14** The second vibration mode of crankshaft

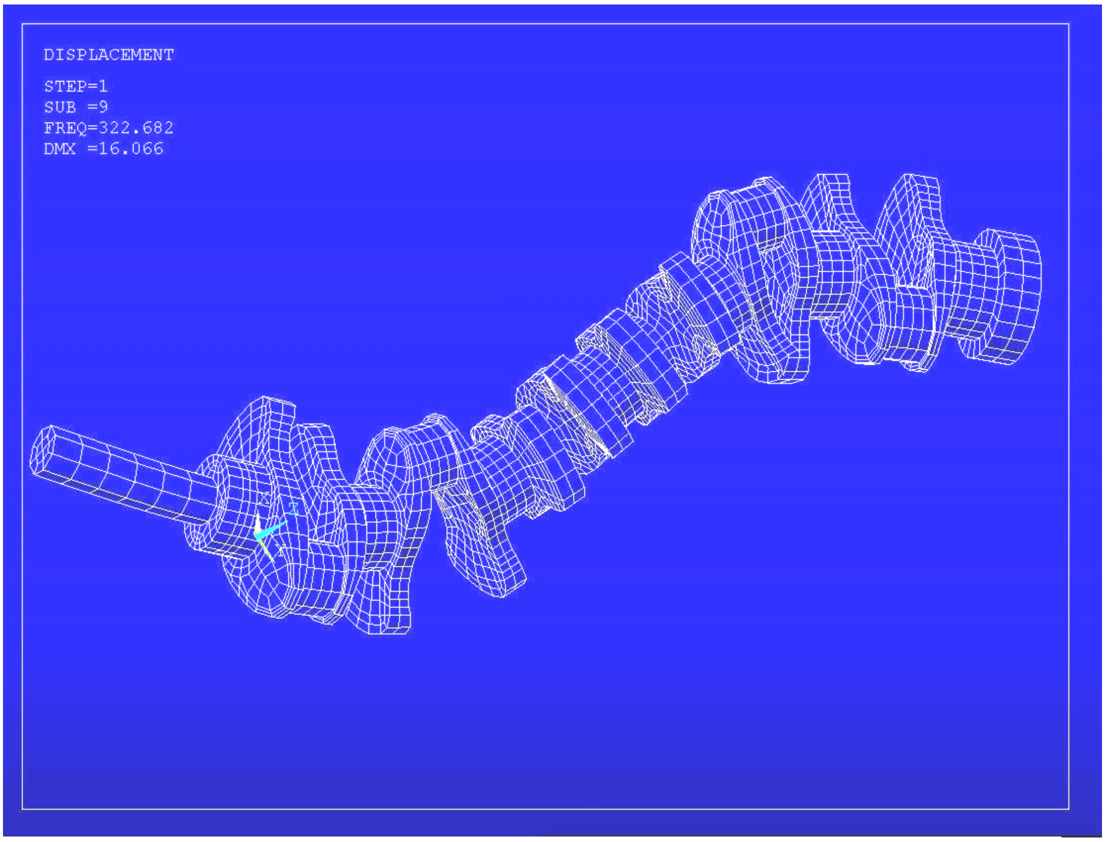


**Fig. 15** The third vibration mode of crankshaft

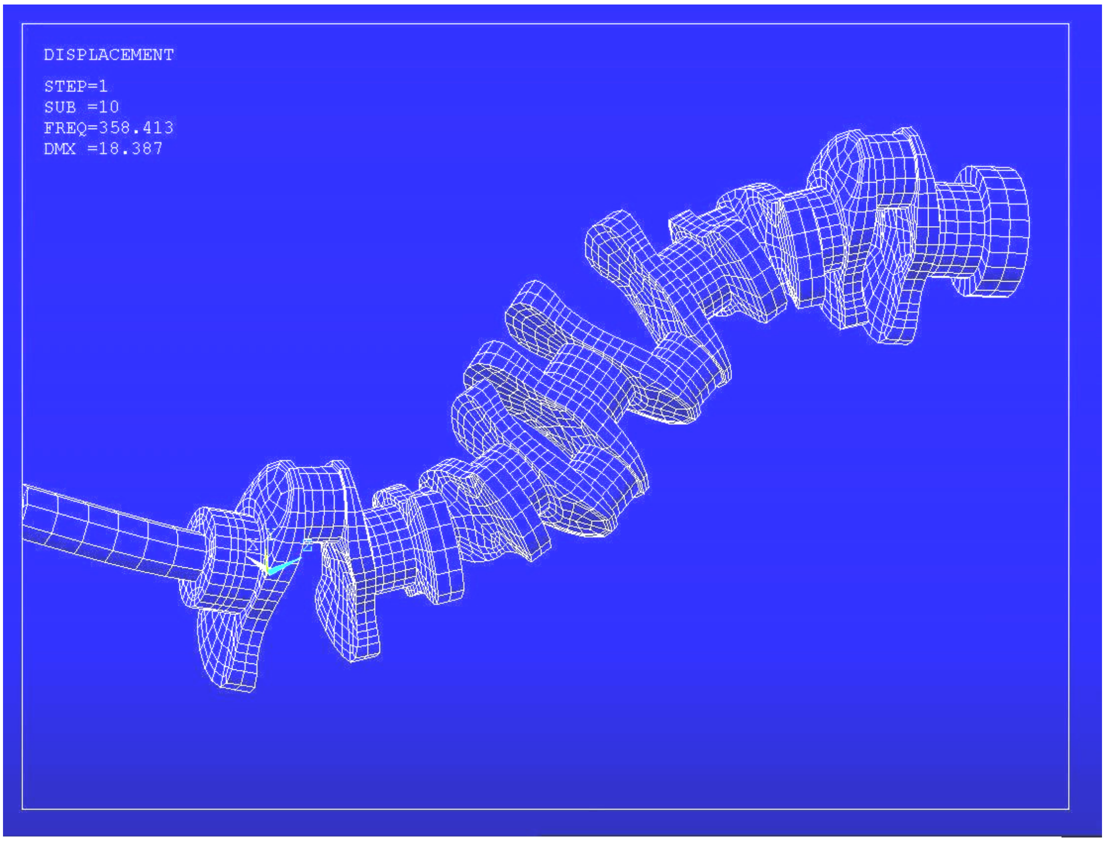


**Fig. 16** The fourth vibration mode of crankshaft