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CRANKTRAIN DYNAMICS SIMULATION

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Summary: This paper presents advanced and efficient computational approaches to an analysis of cranktrain dynamics. The approaches are represented by complex computational model of a powertrain (virtual engine) which is a powerful tool for the solution of structural, thermal and fatigue problems. The computational model of dynamics includes flexible parts and hydrodynamic approach for slide bearing solution. Presented virtual engine is assembled as well as numerically solved as a Multi-Body System and it is focused on cranktrain vibrations identification. Virtual engine results are validated by measurements on a compression-ignition in-line four-cylinder engine.

Key words: cranktrain, dynamics, vibrations, virtual engine.

INTRODUCTION

Contemporary powertrains are complex thermomechanical systems improved by large and gradual development. Vehicle producers are still increasing the engine parameters such as the engine performance, together with significantly decreasing the fuel consumption. In addition, low levels of noise, vibrations and engine emissions are required by national legislature of the European, US or Japanese markets.

However, an increase in the engine performance often leads to an increase in the powertrain noise and vibrations. These problems can be solved by experimental or computational approaches. Experimental methods are time-consuming and quite expensive therefore computational methods are in use increasingly. In the sphere of powertrain dynamics, experimental methods are often applied for complex computational model validations because computational models can provide very exact results but only on the condition that exact inputs are included.

All computational and measurement methods reported here are applied to four-stroke in-line four-cylinder compression-ignition engine of these parameters:

- Bore: 105 mm,
- Stroke: 120 mm,
- Displacement: 4.15 l,
- Compression ratio: 17.8 : 1,
- Rated output: 96 kW @ 2200 rpm.

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1. COMPUTATIONAL METHODS

1.1 Virtual Engine

A complex computational model of the engine (in other words a virtual engine) is solved in the time domain. This enables different physical problems including various non-linearities to be incorporated. The virtual engine is assembled as well as numerically solved in MBS ADAMS. ADAMS is a general code and enables integration of user-defined models to be made directly using ADAMS commands or user-written FORTRAN or C++ subroutines (2).

In general, the virtual engine includes all significant components necessary for noise, vibrations and harshness (NVH) or fatigue analyses. The included modules are represented in particular by a cranktrain, a valvetrain, a gear timing drive with an fuel injection pump and a rubber damper. Following analyses just deal with a cranktrain as the main module of the virtual engine.

A cranktrain module consists of solid model bodies, linearly elastic model bodies and constraints between them.

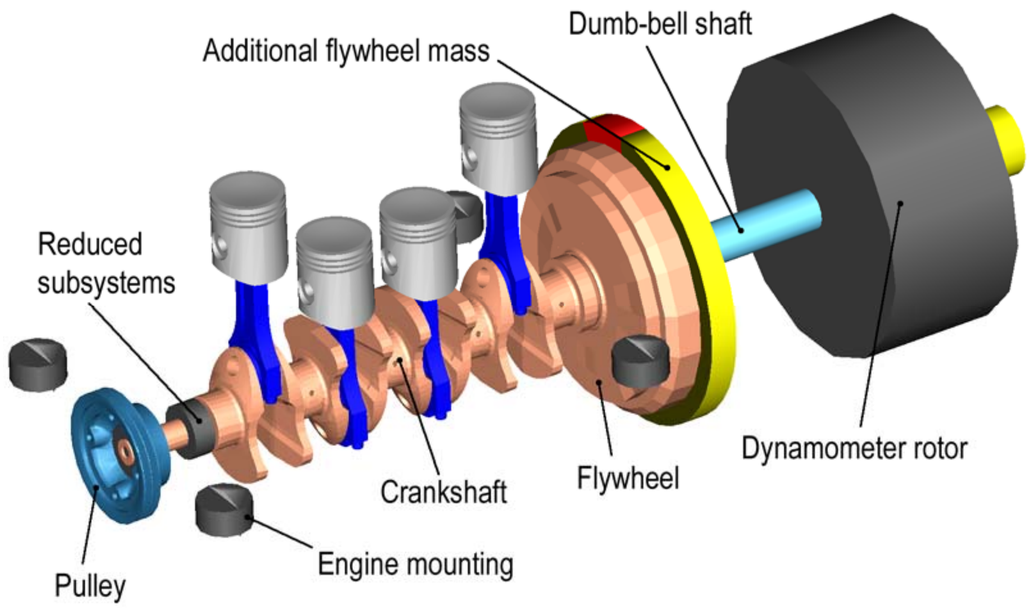

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Fig. 1 – Virtual engine cranktrain with dynamometer model

The solid model bodies are:

- Piston assembly,
- Connecting rod assembly,
- Crankshaft pulley,
- Additional flywheel mass,
- Dynamometer rotor.

The linearly elastic model bodies are modally reduced Finite Element (FE) models suitable for dynamic simulation. These are:

- Crankshaft,
- Major part of used flywheel.

A dumb-bell shaft connecting a flywheel with a dynamometer rotor is represented by a body with defined torsional stiffness and damping. These characteristics are adjusted on account of torsional vibrations measurement.

Some subsystems having indispensable influence on cranktrain vibrations are modelled by their inertia characteristics or power requirement with corresponding speed ratio (valvetrain, balancing shafts, and an injection unit).

The interaction between the crankshaft and the engine block is ensured via a non-linear hydrodynamic journal bearing model, where pre-calculated force databases obtained when solving separate hydrodynamic problem are used.

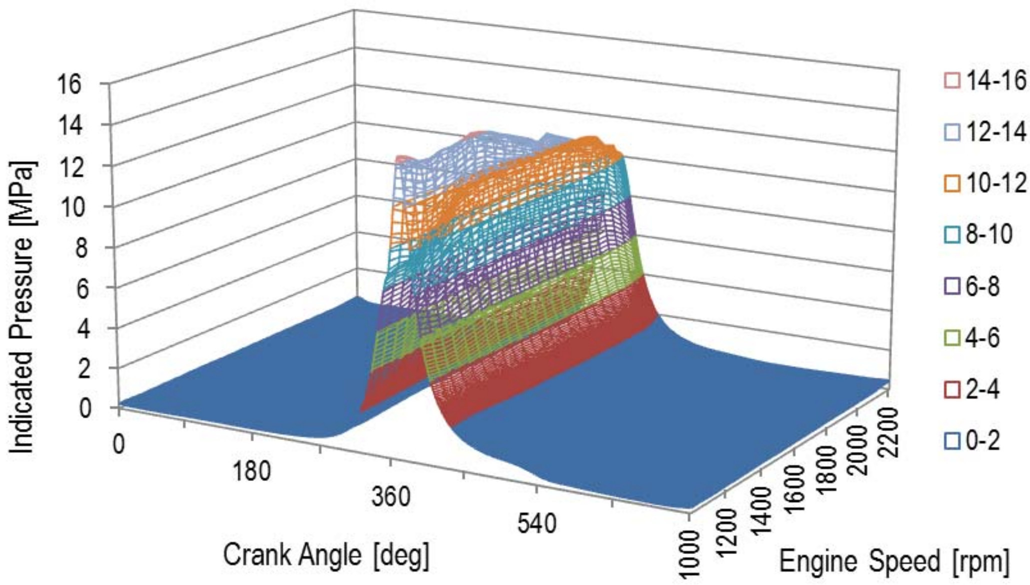

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Fig. 2 – Indicated pressure for virtual engine excitation

The virtual engine is excited by means of cylinder pressure defined by a high-pressure measurement and via inertial forces from moving parts. Simulation starts from 1100 rpm and is carried out to 2400 rpm (pressures between 2200–2400 rpm are extrapolated).

1.2 Validation of Computational Model

The determination of cranktrain torsional vibrations represents a fundamental step in the powertrain development. These vibrations can significantly influence fatigue of crankshaft. Load of each subsystem coupled with the cranktrain is also affected.

An experimental determination of torsional vibrations using, for example, laser vibration tools is an advance and it can help to validate computational models.

A summary of a cranktrain torsional behaviour can provide a harmonic analysis of torsional vibrations determined from crankshaft pulley angular velocity.

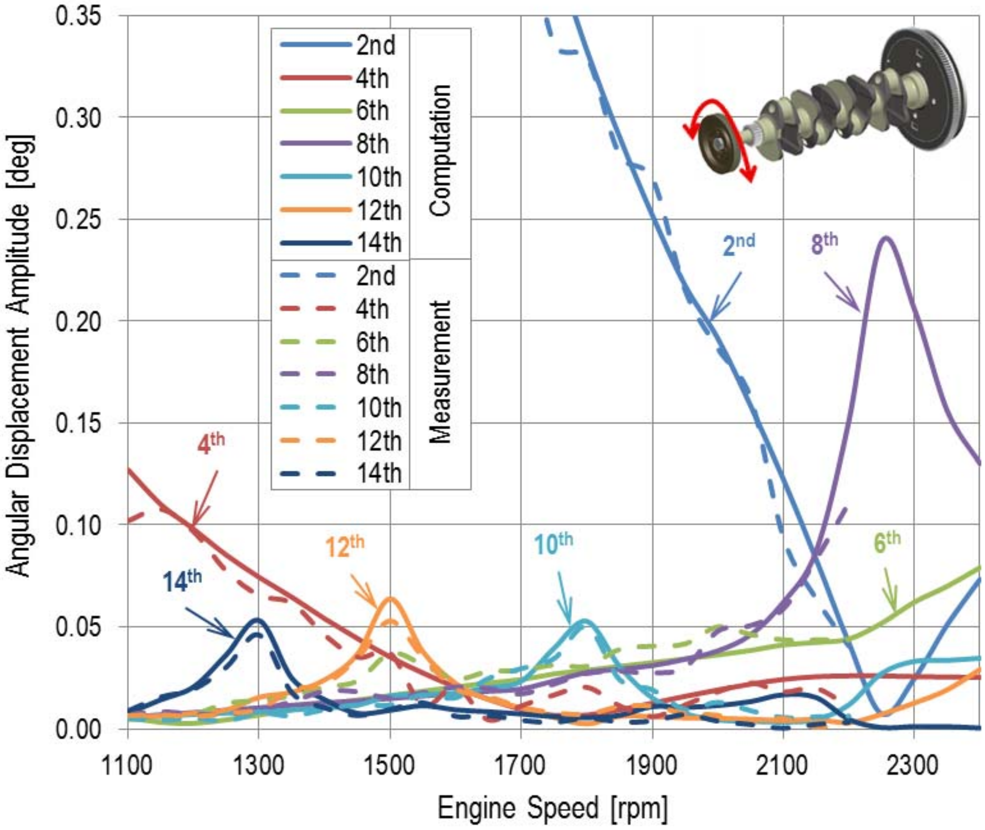

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Fig. 3 – Harmonic analysis of crankshaft pulley torsional vibrations without torsional damper

Figure 3 shows harmonic analysis results of computed and measured cranktrain torsional vibrations originated in the powertrain without a torsional damper. Brüel & Kjær Rotational Laser Vibrometer Type 2523 and POLYTEC 4000 Series Laser Vibrometer are used for model verifications. These experimental tools enable measurements of angular velocities of arbitrary rotating parts. For in-line four-cylinder four-stroke internal combustion engine, there are defined the main harmonic orders (which have generally the biggest amplitudes) as integral multiples of two. This fact is confirmed by computation as well as measurement whereas relatively very good conformity of the computation and the measurement is found. Possible variations are caused by not simultaneous performance of torsional vibrations measurement and cylinder pressure measurement due to the substantial volume of data captured per time unit.

2. CRANKSHAFT DYNAMICS SOLUTION RESULTS

The virtual engine can provide a large number of results which are necessary for different stages of the new powertrain development. The results presented in this paper do not provide complete information about the target powertrain because the aim of the presented results is to give insight into some cranktrain influences, mainly those which can be validated by measurements.

2.1 Crankshaft Torsional Vibrations

Vibrations are common to internal-combustion engine crankshafts nevertheless torsional vibrations in fully embedded shafts (each crank pin is next to two main pins) which is caused by shaft flexibility is the most dangerous one. The crankshaft mechanism generates alternating torque due to time varied combustion pressure in conjunction with the alternating effect of reciprocating parts inertia. This torque brings the elastic crankshaft in vibrations about the axis of rotation which is known as forced torsional vibrations. Such vibrations are superimposed on the crankshaft oscillation that appears due to the engine run irregularity and the static torsion from tangential forces. This type of vibrations can cause cracking and crankshaft failure, and therefore it is very dangerous crankshaft effort (3).

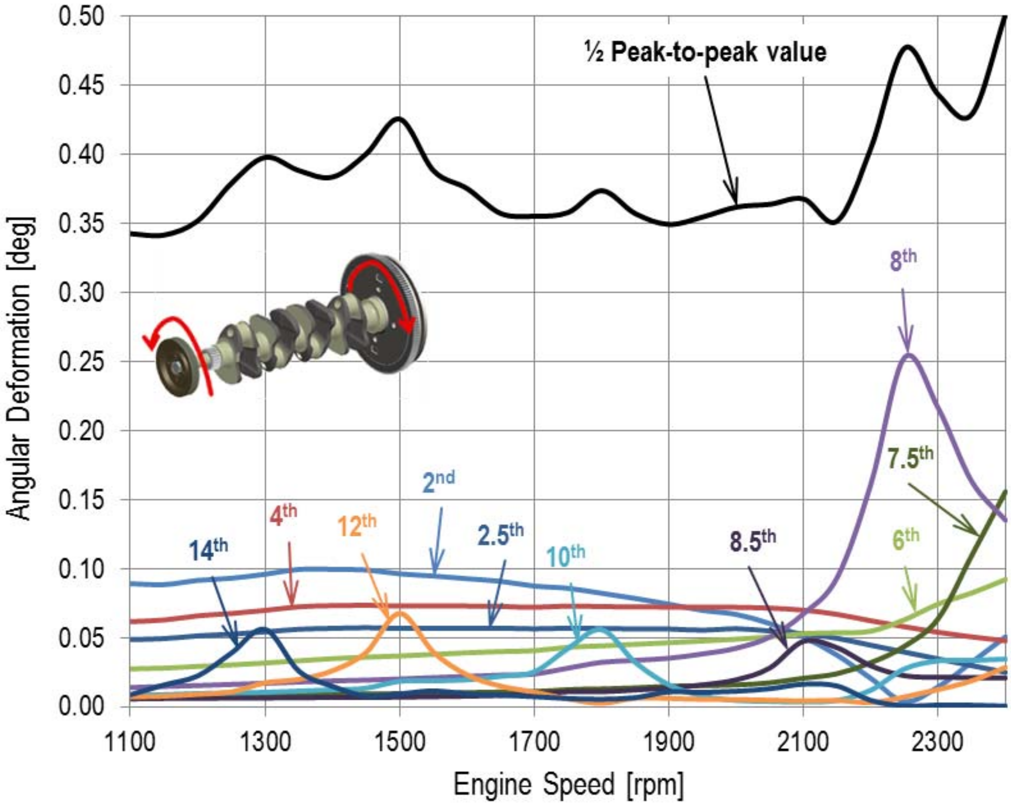

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Fig. 4 – Harmonic analysis of crankshaft torsional vibrations without torsional damper

Figure 4 presents harmonic analysis results of computed crankshaft torsional vibrations of the powertrain without any torsional damper. In this case, by crankshaft torsional vibrations is meant angular deformation between a crankshaft pulley and a flywheel flange. Peak-to-peak value represents a difference between maximal and minimal values for one engine cycle. The eighth harmonic order is the most dominant and resonance of this order occurs at engine speed 2250 rpm. The nominal engine speed is very close to this critical engine speed, therefore, the torsional damper application seems to be appropriate to decrease torsional vibrations and increase cranktrain component fatigue.

2.2 Crankshaft Axial Vibrations

Crankshaft axial vibrations are a mode of vibrations resulting from the crankshaft expanding and compressing along its axis of rotation due to cranktrain elasticity, alternating acting forces and journal damping. This mode of vibrations affects the crankshaft web, its pins and the other related components.

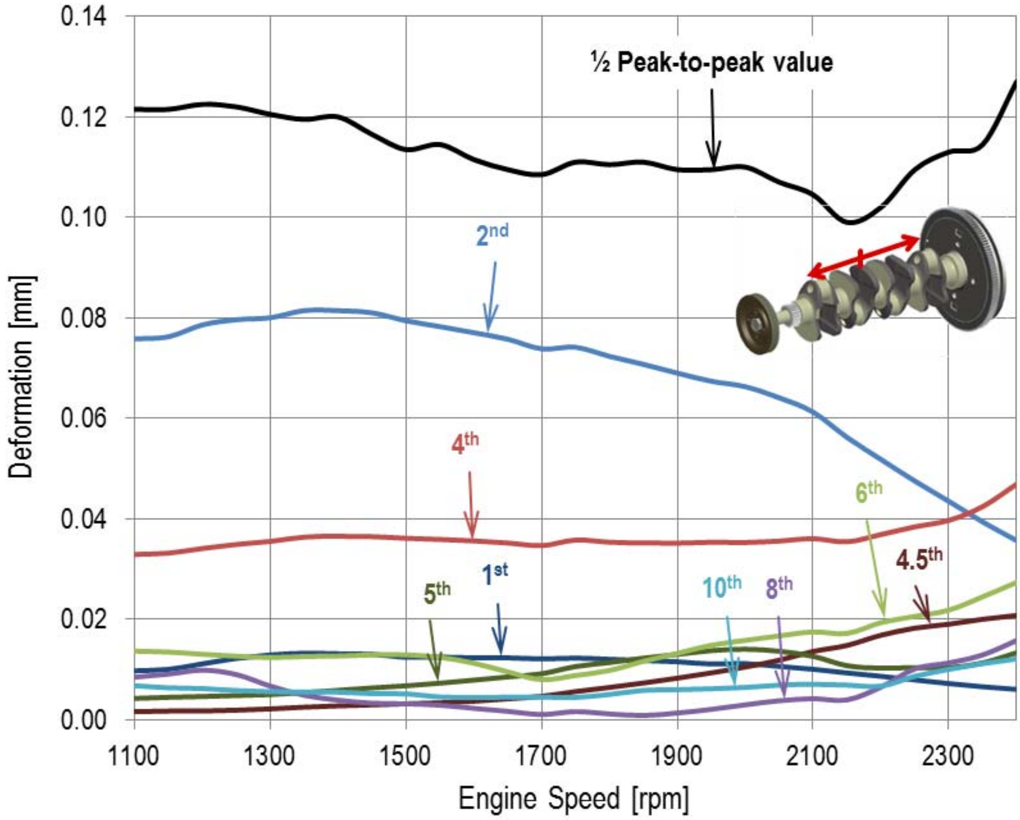

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Fig. 5 – Harmonic analysis of crankshaft axial vibrations without torsional damper

Figure 5 shows harmonic analysis results of crankshaft axial vibrations of the powertrain obtained by dynamic simulations. The dominant orders are the second one and the fourth one. The level of peak-to-peak half value is not very critical for whole operating speed nevertheless it can be useful for design and location of the thrust bearing especially with reference to a timing-drive mechanism.

2.3 Main Bearing Load

The virtual engine is able to give various bearing load data. These can be represented by bearing forces and moments, dimensionless pin tilting angle in the plane of narrowest oil film gap, dimensionless tilting angle in the plane perpendicular to the plane of the narrowest oil film gap, independent bearing clearance and by relative eccentricity.

Relative eccentricity is defined as

$$\varepsilon = \frac{2e}{D-d} \qquad (1)$$

where e [mm] is an absolute eccentricity of the pin centre against the centre of the bearing shell, D [mm] is the bearing shell diameter and d [mm] stands for the pin diameter. All is based on the assumption of an absolutely stiff pin and shell. If the centres of the pin and shell are identical, $\varepsilon = 0$. If the pin and shell surfaces hit, the relative eccentricity reaches $\varepsilon = 1$ (3).

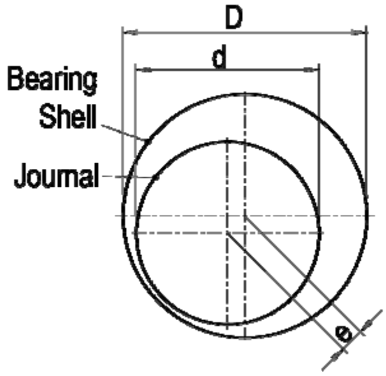

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Fig. 6 – Journal bearing eccentricity

Regarding the bearing life, the highest relative eccentricity in the course of the working cycle seems to be the most important. This is also, among others, due to the fact that the bearing response (represented by decrease in the oil film thickness) to the drive pulse is not immediate (3).

Figure 7 presents relative eccentricity maxima of all main bearings through the entire engine speed range. Even in the worst case, relative eccentricity amounts to approximately $\varepsilon = 0.92$, and that is why each main bearing can be regarded as acceptable in respect to bearing load. However, obtained bearing load data can be processed by specialized computational software to get detail information such as bearing pressure distribution. Those data may be then used for design of bearings, bearing clearances and shape and dimensions of oil distribution grooves.

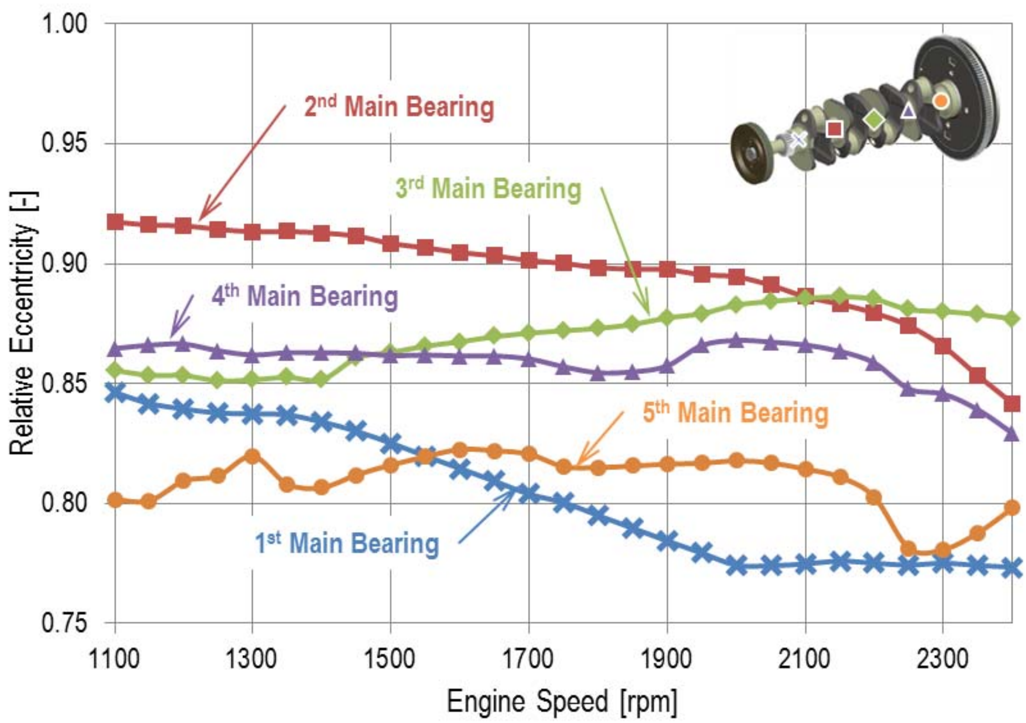

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Fig. 7 – Relative eccentricity of main bearings

CONCLUSION

The introduced virtual engine presents a powerful tool for powertrain vibrations analyses. It is able to incorporate not only linear but also non-linear behaviour of powertrain components and subsystems.

The complex computational model should be verified by some suitable experimental methods. The results of vibrations analyses are shown at crankshaft torsional and axial vibrations and main bearing load that can be further used for vibrations eliminating steps such as application and design of a torsional damper and main and thrust bearing layout. The influence of all the modifications and new designs can be simulated and proved also by this virtual engine. The results of simulations are also very valuable sources for more detail analyses based on specialized software.

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