NUMERICAL SIMULATION ON THE VIBRATION OF A TEST BED WITH ENGINE WITH DUAL MASS FLYWHEEL

Assist. Prof. Eng. Pavlov N. PhD
Faculty of Transport, Technical University of Sofia, Bulgaria

npavlov@tu-sofia.bg

Abstract: *This paper describes the results of numerical simulations of a vehicle engine with dual mass flywheel coupled with dynamometer. The differential equations of the model are given. The mass moments of inertia and the torsional stiffness are taking into account. Numerical simulations are carried out. Natural frequencies of the system are determinate. Bode diagrams of the system are shown and analysed. Recommendations to reduce vibration by using an additional flexible coupling are given.*

Keywords: DUAL MASS FLYWHEEL, TRANSMISSION, DRIVELINE, DYNAMOMETER, DYNAMICS, VIBRATION

1. Introduction

The Dual Mass Flywheel (DMF) is widely used in modern diesel or gasoline direct injection engines. Obtaining the performance characteristics of the engines is important part of the engine research and development process. For testing the Internal Combustion Engines (ICE) the dynamometer test beds are widely used.

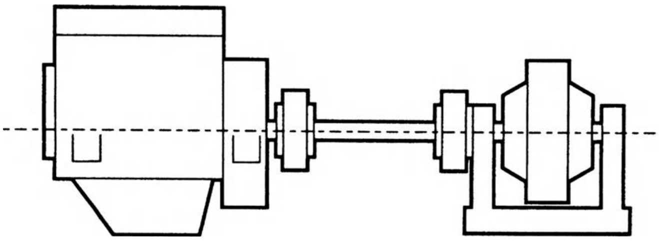

Fig. 1 *Engine test bed [1]*

The mechanical part of test bed shown in Fig. 1 consists of Internal Combustion Engine, propeller shaft (cardan), dynamometer and mountings.

The dynamometers are used, also when the transmission tests are performed. The test bed setup shown in Fig. 2, a) is intended for testing the internal combustion engine and transmission only [2]. In this case, an appropriately suited dynamometer is used to apply a load to the unit under test's cardan shaft.

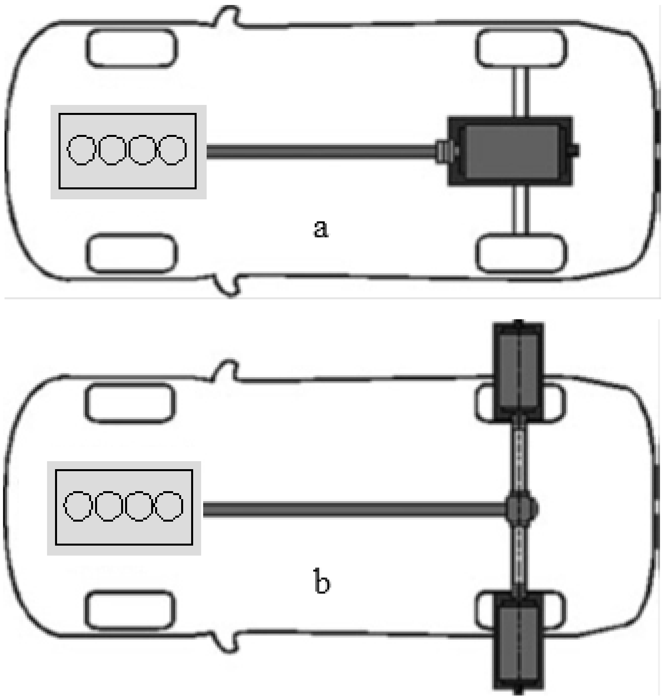

Fig. 2 *Powerpack test bed for standard drives [2]*

The setup shown in Fig. 2, b) allows testing of the transmission and the rear-axle final drive and differential.

Figure 3 presents the typical test bed setup for vehicles with front-wheel drive [2].

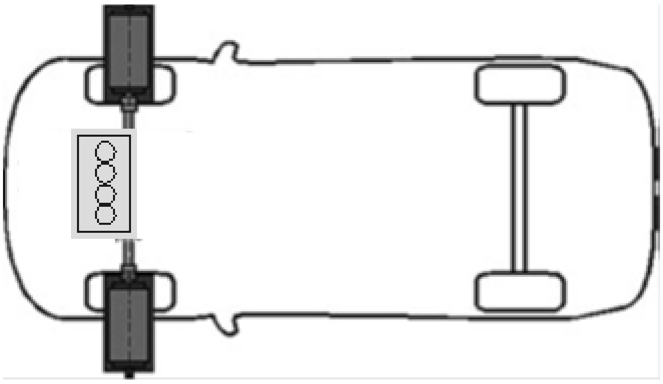

Fig. 3 *Powertrain test bed for front-wheel drives [2]*

The test bed setup with four dynamometers shown in Fig. 4 is used for testing four-wheel drive vehicle's powertrains.

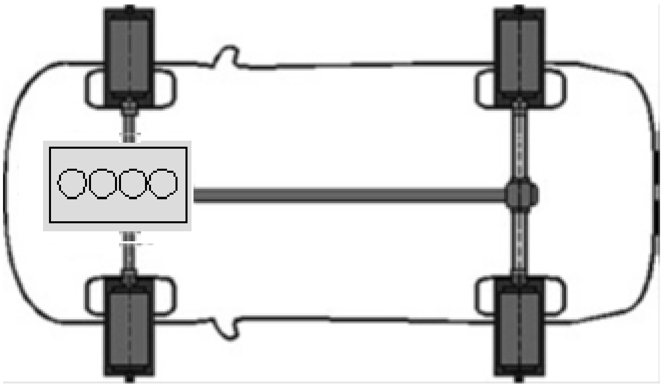

Fig. 4 *Powertrain test bed for four-wheel drives [2]*

In any case of the test bed layout concepts shown above, the inertia of the moved vehicle, reduced to the crankshaft is missed. The mass moment of inertia of the dynamometer rotors is much smaller than the vehicle mass inertia. Thus, the values of the natural frequencies of the dynamical system are much higher than the natural frequencies of the system with moved vehicle. As result the vibrations of the test bed increases. Some test mechanics and engineers welded the primary and secondary mass of the dual mass flywheel, or they used substituting single mass flywheel. This does not lead to good results with modern direct injection engines with high power received from a less number of cylinders.

In this regard, the purpose of this publication is to draw out the differential equations of an engine-dynamometer dynamic model with dual mass flywheel and to carry out numerical simulations. To gives recommendations and technical solution for reducing vibration in test beds when engine with DMF is used.

2. Dynamic Model

A test bed dynamical model is shown in Fig. 5. The model consists of two elements of inertia, spring and damper. They are:

$$J_1 = J_c + J_{f1}$$

J_c – mass moment of inertia of the engine crankshaft;

J_{f1} – mass moment of inertia of the primary mass of the flywheel.

$$J_2 = J_{f2} + J_{ps}$$

J_{f2} – mass moment of inertia of the secondary mass of the flywheel;

J_{ps} – mass moment of inertia of the propeller (cardan) shafts;

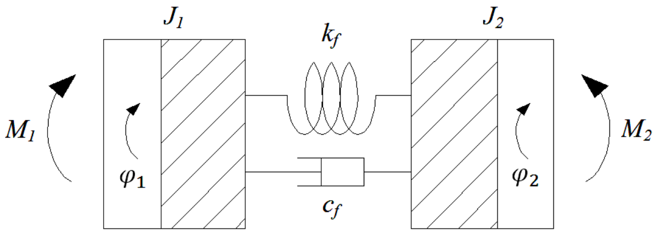

Fig. 5 *Equivalent 2 DOF engine-dynamometer dynamic model*

k_f, c_f – spring stiffness and damping of the dual mass flywheel;

M_1 – exciting moment from the engine;

M_2 – the dynamometer resistant moment.

Some companies offer rubber elastic couplings [3] that are mounted between the flywheel and the dynamometer. The spring stiffness and damping of the additional coupling are k_c and c_c.

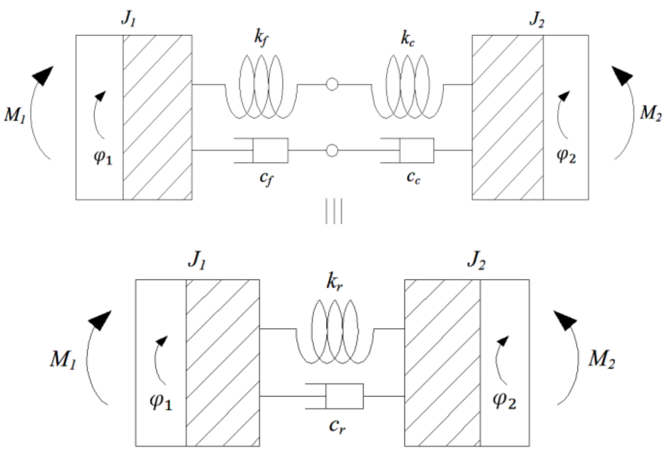

Fig. 6 *Equivalent 2 DOF engine-dynamometer dynamic model with additional elastic coupling*

Then according to Fig. 6 the reduced stiffness of the spring will be:

$$k_r = \frac{k_f . k_c}{k_f + k_c}$$

The same holds for the dampers:

$$c_r = \frac{c_f . c_c}{c_f + c_c}$$

The differential equations of the model in Fig. 5 are:

$$J_1\ddot{\varphi}_1 + k_f(\varphi_1 - \varphi_2) + c_f(\dot{\varphi}_1 - \dot{\varphi}_2) = M_1$$

$$J_2\ddot{\varphi}_2 - k_f(\varphi_1 - \varphi_2) - c_f(\dot{\varphi}_1 - \dot{\varphi}_2) = -M_2$$

And the differential equations of the model in Fig. 6 are:

$$J_1\ddot{\varphi}_1 + k_r(\varphi_1 - \varphi_2) + c_r(\dot{\varphi}_1 - \dot{\varphi}_2) = M_1$$

$$J_2\ddot{\varphi}_2 - k_r(\varphi_1 - \varphi_2) - c_r(\dot{\varphi}_1 - \dot{\varphi}_2) = -M_2$$

3. Numerical Simulation

Natural frequency of the system is:

$$\omega_{nat} = \sqrt{\frac{k(J_1 + J_2)}{J_1 . J_2}}\text{, rad/s}$$

In overcritical operating conditions ($\omega_{ex} > \omega_{nat}$), it must be ensured that the minimum excitation frequency will in all operating points will remain to a sufficient degree above the natural frequency [3]. The exciting frequencies (ω_{ex}) are, hence, the basic frequency (number of work cycles per unit time) and their integral multiples. They are proportional to the crankshaft speed. All of these exciting frequencies can resonate with one of the natural frequencies (Fig.7).

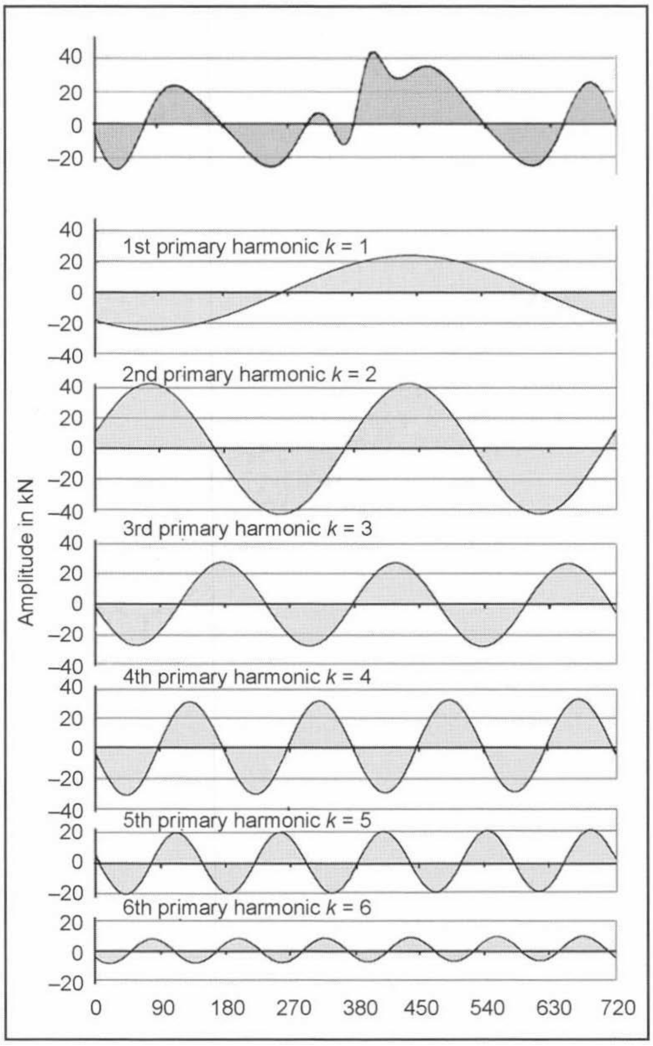

Fig. 7 *Fourier analysis of a tangential force diagram: The tangential force curve is composed of the first six harmonics [4]*

The numerical simulations are performed in program field of MATLAB. The Bode diagram in logarithmic scale is shown in Fig. 8. It can be seen that when the crankshaft is coupled with the dynamometer, using only a dual mass flywheel, the maximum (ω_{nat}=95,9 rad/s) of the amplitude response (blue line) is within the engine operating range (between 94,2 and 628 rad/s). This can lead to undesirable vibrations in the test bench. When using a single mass flywheel without any elastic coupling, the maximum offsets to the higher frequencies again in the engine operating range (red line - ω_{nat}=372 rad/s).

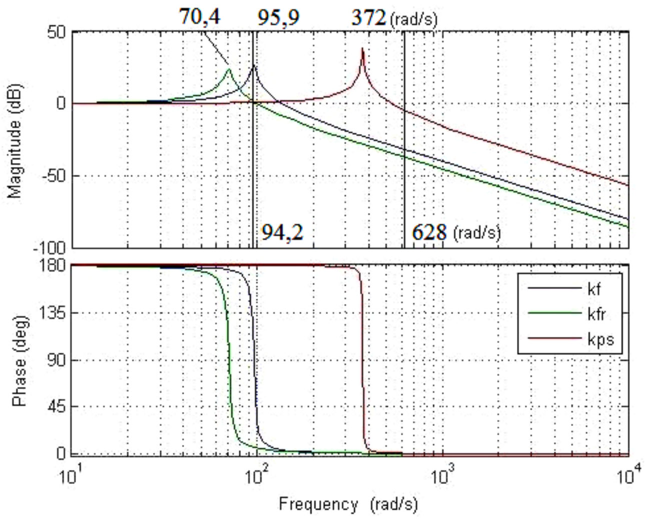

Fig. 8 *Bode diagram of the system with DMF (blue line), DMF and high flexible coupling (green) and single mass flywheel without any elastic coupling (red).* k_f *=600;* k_r*=323;* k_{ps}*=9000 N.m/rad*

Only when the crankshaft is coupled with a double mass flywheel and highly flexibility coupling to the dynamometer rotor, the maximum of the amplitude-frequency characteristic, offsets to the left, below the minimum engine speed (green line - ω_{nat}=70,4 rad/s). In this way there is good damping of the vibrations of the first and higher order in the whole operating range of the engine.

4. Conclusion

The considered model enables to study the vibration in test bench with engine with dual mass flywheel. The numerical experiments with different variants of coupling the engine to the dynamometer are considered. The simulation shows that it is necessary to use a flexible coupling to reduce the total stiffness in the system. Thereby reducing the vibration transmitted from the engine to the test bed.

References:

[1] Martyr, A., M. Plint. Engine Testing - The Design, Building, Modification and Use of Powertrain Test Facilities. Elsevier, 2012.

[2] Paulweber, M., K. Lebert. Powertrain Instrumentation and Test Systems. Development – Hybridization – Electrification. Springer, 2016.

[3] Technical Information - Highly Flexible Couplings. Voith Turbo.

[4] Basshuysen, R., F. Schäfer. Internal Combustion Engine Handbook. Basics, Components, Systems, and Perspectives. SAE, 2004.