NUMERICAL SIMULATION ON THE VIBRATION OF A VEHICLE DRIVETRAIN WITH DUAL MASS FLYWHEEL

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Abstract: *This paper describes the results of numerical simulations of a vehicle drivetrain model with dual mass flywheel. The differential equations of the model are given. The vehicle inertia, the tire torsional stiffness and the transmission gear ratios are taking into account. Numerical simulations with given mass, elastic and damping parameters are carried out. Natural frequencies of the system are determinate. Vibrational characteristics of the system are shown and analysed.*

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

1. Introduction

The Dual Mass Flywheel (DMF) is widely used in modern diesel or gasoline direct injection engines. Downsizing and downspeeding of engines, lowering friction in gearboxes and lightweighting of transmissions and drivelines cause increased vibration and noise. The periodic combustion cycles of a four-stroke engine create torque and speed fluctuations which cause torsional vibration to be passed down the drivetrain (Fig. 1). The resulting noise and vibration, such as gear rattle, body boom and load change vibration, results in a decrease in comfort [1].

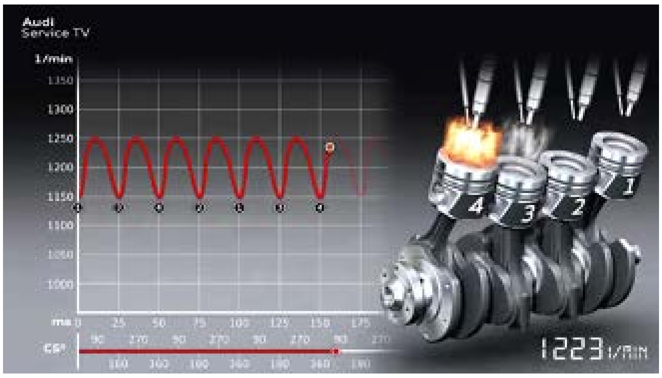

Fig. 1 *Angular speed fluctuation of four-cylinder four-stroke engine [2]*

The dual mass flywheel consist two masses free to rotate independently on a ball bearing. Two long coil springs lower the value of the natural frequency of the system (Fig. 2). Damping device is also present [3]. The secondary mass helps to increase the mass moment of inertia on the gearbox side. The total inertia of the DMF is the same as that of the rigid flywheel.

Fig. 2 *Dual mass flywheel (DMF)[4]*

As the DMF has an integral spring-damper system a rigid clutch driven disc without a torsion damper is normally used [1]. Owing to the additional mass on the transmission input shaft, the vibration torque range, which is normally between 1200 min^{-1} and 2400 min^{-1} with original stiffer torsion dampers in the driven disc, is moved to a lower resonance speed range. This ensures excellent damping of engine vibration even at idle speeds [1]. Driving with less angular speed can reduce fuel consumption and CO_2 emissions [4, 5].

There are many books and research articles on DMF problems [4, 5, 6, 7, 8 and 9]. Their authors examine two or three degree of freedom engine-transmission model. But the system is more complicated because the vehicle mass should be reduced to the engine crankshaft and this is shown in [10, 11] but without mathematical description. The tire torsional stiffness has to be given an account too. Thus, the model whose scheme is given in source [12] most accurately reflects the essence of the real system. Unfortunately, the author does not present the mathematical description of this model, as well as numerical studies with specific values of the parameters. In textbook [13] a modal analysis of a vehicle drivetrain with and without dual mass flywheel is performed. In work [14] a dynamic model of friction self-excited vibration of a vehicle drivetrain with conventional damper in the driven clutch disc is created and numerical simulations are performed. In [15] dynamic model of a vehicle drivetrain with spring and friction damper in driven clutch disk is presented and test results for amplitude-frequency characteristics are shown. In work [16] a linear mechanical model of hybrid electric vehicle drivetrain is considered, a modal analysis is performed and the effect of the control of the drivetrain on the eigenvalues of the vibrational system is studied. In this regard, the purpose of this publication is to draw out the differential equations of dynamic drivetrain model with dual mass flywheel with taking into account of the vehicle inertia, the tire torsional stiffness and the transmission gear ratios and to carry out numerical simulations with given mass, elastic and damping parameters.

2. Dynamic Model

A front wheel drive vehicle with dual mass flywheel drivetrain and its kinematic scheme are shown in Fig. 3 and 4 respectively. The system mainly consists of engine, dual mass flywheel (DMF), gearbox, final drive, drive shafts, drive wheels, driven wheels and car body. The symbols used in kinematic scheme are:

J_c – mass moment of inertia of the crankshaft;

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

J_{f2} – mass moment of inertia of the secondary mass of the flywheel and clutch pressure disc;

J_{g1} – mass moment of inertia of the clutch driven (friction) disc and primary shaft with drive gears of the gearbox;

J_{g2} – mass moment of inertia of the secondary shaft with driven gears of the gearbox;

J_0 – mass moment of inertia of the final gear and the differential;

J_d – mass moment of inertia of the drive shafts;

J_w – mass moment of inertia of the wheels;

m – vehicle mass;

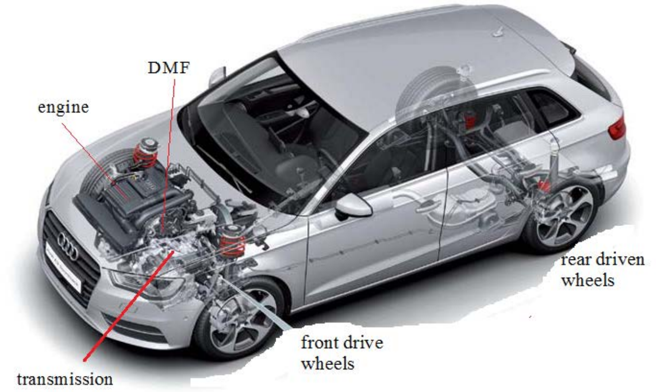

Fig. 3 *Front wheel drive vehicle with DMF [17]*

J_{g2} – mass moment of inertia of the secondary shaft of the gearbox;

J_0 – mass moment of inertia of the final gear and the differential;

J_d – mass moment of inertia of the drive shafts;

J_w – mass moment of inertia of the wheels;

m – vehicle mass;

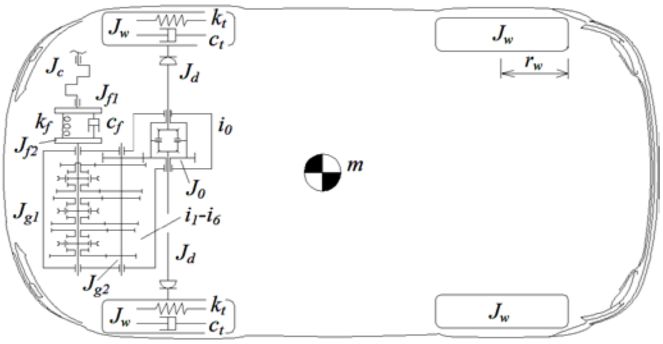

Fig. 4 *Vehicle drivetrain kinematic scheme*

$i_{1\text{-}6}$ – gear ratios of the six speed gearbox;

i_0 – gear ratio of the final drive;

r_w – rolling radius of the wheels;

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

k_t, c_t – tire torsional stiffness and damping of the drive wheels.

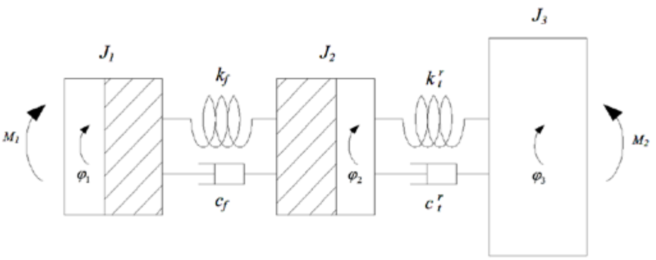

Fig. 5 *Equivalent 3 DOF vehicle drivetrain dynamic model*

The equivalent dynamic model shown in Fig. 5 has three degrees of freedom (3 DOF) and the vector of generalized coordinates is:

$$q = [\varphi_1 \quad \varphi_2 \quad \varphi_3]^T$$

The mass moments of inertia of the dynamic model are:

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

$$J_2 = J_{f2} + J_{g1} + J_{g2}^r + J_0^r + 2J_d^r + 2J_w^r$$

$$J_3 = 2J_w^r + J_v^r$$

J_1 – mass moment of inertia of the crankshaft and primary mass of the flywheel;

J_2 – mass moment of inertia of the secondary mass of the flywheel, clutch, gears and shafts of the gearbox and final drive, drive shafts and drive wheels;

J_3 – mass moment of inertia of the driven wheels and mass of the car body reduced to the engine crankshaft;

The mass moments of inertia reduced (referred) to the crankshaft are [18, 19]:

$$J_{g2}^r = \frac{J_{g2}}{i_{1-6}^2}$$

$$J_0^r = \frac{J_0}{i_{1-6}^2 . i_0^2}$$

$$J_d^r = \frac{J_d}{i_{1-6}^2 . i_0^2}$$

$$J_w^r = \frac{J_w}{i_{1-6}^2 . i_0^2}$$

$$J_v^r = m\left(\frac{r_w}{i_{1-6} . i_0}\right)^2$$

where superscript r means *reduced* or *referred* to the crankshaft;

J_v^r – reduced mass moment of inertia of translating mass of the vehicle to the crankshaft.

For the generalized coordinates can be written:

$$\varphi_1 = \varphi_c = \varphi_{f1}$$

$$\varphi_2 = \varphi_{f2} = \varphi_{g1} = \varphi_{g2} . i_{1-6} = \varphi_0 . i_{1-6} . i_0 = \varphi_w . i_{1-6} . i_0$$

$$\varphi_3 = \varphi_w . i_{1-6} . i_0 = \frac{S . i_{1-6} . i_0}{r_w}$$

where S is the driving distance, *m*.

$$k_t^r = \frac{k_t}{i_{1-6}^2 . i_0^2}; \quad c_t^r = \frac{c_t}{i_{1-6}^2 . i_0^2}$$

where k_t^r, c_t^r – tire torsional stiffness and damping reduced to the engine crankshaft.

The differential equations of the model are:

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

$$J_2 \ddot{\varphi}_2 - k_f(\varphi_1 - \varphi_2) + k_t^r(\varphi_2 - \varphi_3) - c_f(\dot{\varphi}_1 - \dot{\varphi}_2) + c_t^r(\varphi_2 - \varphi_3) = 0$$

$$J_3 \ddot{\varphi}_3 - k_t^r(\varphi_2 - \varphi_3) - c_t^r(\dot{\varphi}_2 - \dot{\varphi}_3) = -M_2$$

The excited moment from the engine is:

$$M_1 = M_o + M \sin \omega t$$

And

$$M_2 = \frac{M_R}{i_{1-6} . i_0}$$

where M_R is resistive moment acted on the vehicle. It includes rolling resistance, road inclines and aerodynamic drag.

3. Numerical Simulation

The numerical simulations are performed in program field of MATLAB. The parameters of the model are taken from different literature sources [13, 14, 19 and 20]. The actual values of the parameters are given in Table 1. Some of the parameters are reduced (referred) according to the formulas given above to the engine crankshaft for the accurate conduction of the simulations.

Table 1: *Parameters of the dynamic model*

Parameter	Value	Dimension
J_c	0,045	kg.m^2
J_{f1}	0,115	kg.m^2
J_{f2}	0,115	kg.m^2
J_{g1}	0,004	kg.m^2
J_{g2}	0,004	kg.m^2
J_0	0,003	kg.m^2
J_d	0,015	kg.m^2
J_w	0,850	kg.m^2
m	1765	kg
$i_{1\text{-}6}$	3,615; 1,947; 1,281; 0,973; 0,778; 0,646	-
i_0	4,056	-
r_w	0,316	m
k_f	600	Nm/rad
c_f	100	Nms/rad
k_t	35000	Nm/rad
c_t	50	Nms/rad

Natural frequencies of the system when 1st gear is engaged:

$$\omega_1 = 0 \text{ - rotation of the system}$$

$$\omega_2 = 15{,}3 \ Hz$$

$$\omega_3 = 4{,}2 \ Hz$$

Natural frequencies of the system when 6th gear is engaged:

$$\omega_1 = 0 \text{ - rotation of the system}$$

$$\omega_2 = 19{,}9 \ Hz$$

$$\omega_3 = 9{,}12 \ Hz$$

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 and the excitation frequency from the engine is [21]:

$$\omega_{ex} = \frac{z}{2}.\frac{n}{60}$$

where z is the number of cylinders;

n – engine speed, min^{-1}.

For 4-cylinder engine on idle speed - 900 min^{-1}:

$$\omega_{ex} = \frac{4}{2}.\frac{900}{60} = 30Hz$$

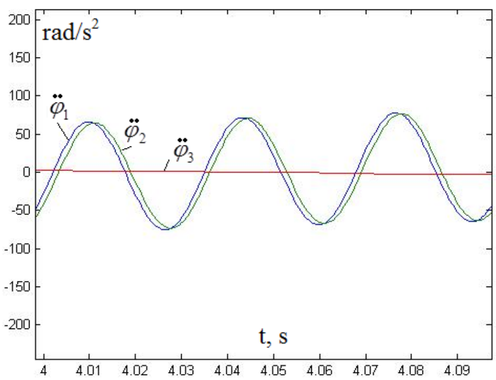

Fig. 7 *Angular acceleration of the first (blue), second (green) and third (red line) mass of the model on idle speed - 900 min^{-1} on 1st gear*

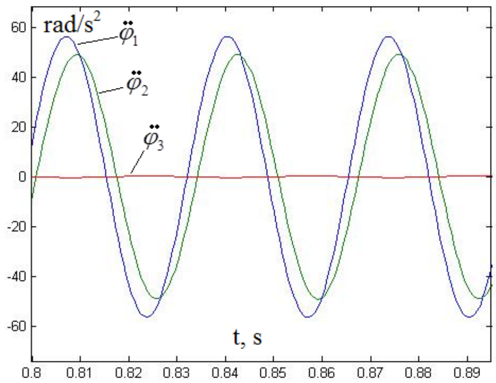

Fig. 8 *Angular acceleration of the first (blue), second (green) and third (red line) mass of the model on idle speed - 900 min^{-1} on 6th gear*

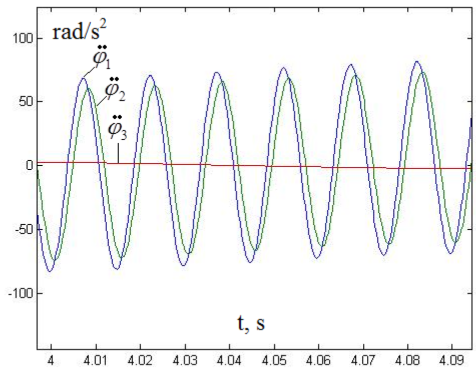

Fig. 9 *Angular acceleration of the first (blue), second (green) and third (red line) mass of the model, 2000 min^{-1} on 1st gear*

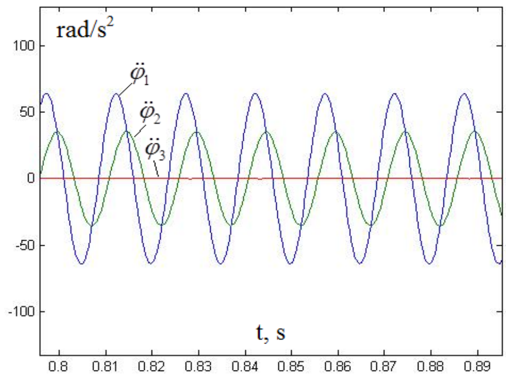

Fig. 10 *Angular acceleration of the first (blue), second (green) and third (red line) mass of the model, 2000 min^{-1} on 6th gear*

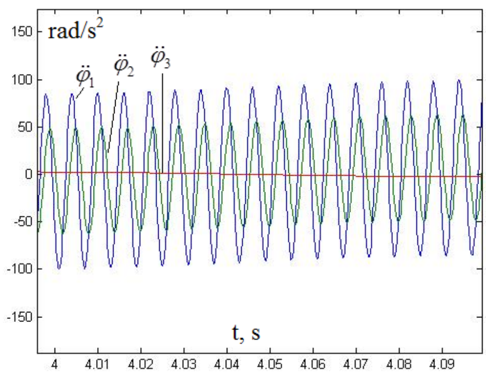

Fig. 11 *Angular acceleration of the first (blue), second (green) and third (red line) mass of the model, 5000 min^{-1} on 1st gear*

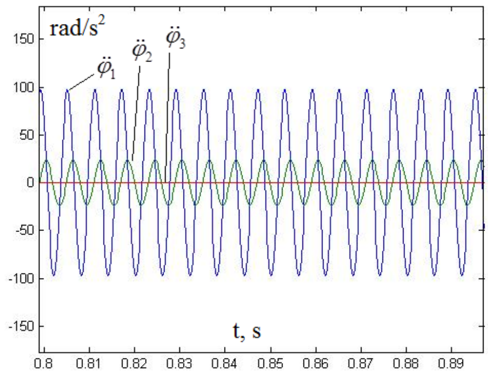

Fig. 12 *Angular acceleration of the first (blue), second (green) and third (red line) mass of the model, 5000 min^{-1} on 6th gear*

Analysis of the results for excited vibration shown in Fig. 7 to Fig. 12 indicates that the system effectively lowers the acceleration of the vehicle body (third mass of the model) and allows driving the vehicle on low engine angular speed. The acceleration of the second mass is less when the higher gear is engaged. Acceleration decrease when increase the angular speed of the engine.

4. Conclusion

The considered model enable to study the vibration in vehicle drivetrain and fluctuation in linear movement of the vehicles with dual mass flywheel. The shown consideration for reduction of all parameters to the engine crankshaft allows to be studied the vibration on various gears in vehicle gearbox. The numerical experiments with given parameters show good damping of acceleration of the third mass (vehicle) in all speed range of the engine and for all gears of the gearbox. The acceleration of the second mass (transmission) decreased with increase the angular speed and with engagement of the higher gear in the gearbox.

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