Towards the Limits of Vibration Attenuation in Drivetrain System by Torsional Dynamics Absorber

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Abstract. Automotive industry drives development towards down-sized and down-speeded engines and higher cylinder pressure. This leads to increased torsional vibrations and therefore puts higher demands on the drivetrain vibration capabilities. The paper presents the results on the study of the limits of torsional dynamics absorbers for vibration attenuation in drivetrain systems obtained by using global sensitivity analysis and multiobjective optimization. The global sensitivity analysis comes with a mapping between the total sensitivity indices of the vibration attenuation measures of a drivetrain system and mass-inertia, stiffness and damping parameters of a torsional dynamics absorber. The multiobjective optimization is resulted in Pareto fronts showing the trade-off between the measures of vibration attenuation and energy losses making possible to identify the limits of the quality of performance of a torsional vibration absorber for a drivetrain system operating on a set of engine input loads. Detailed numerical results are presented on study of application of a dual mass flywheel for heavy-duty truck drivetrain systems in operating engine speed range up to 2000 rpm. The third engine order vibration harmonic is in focus of analysis as one of the most significant contribution to the oscillatory response in drivetrains of heavy-duty trucks.

Keywords: Vibration · Drivetrain system · Torsional dynamics absorber · Dual mass flywheel · Global sensitivity analysis · Pareto optimization

1 Introduction

Ground vehicles and many other engineering systems comprise drivetrains as their important subsystems. The increasing demand for higher efficiency of engineering systems requires improvement of existing and development of novel drivetrain functional components. For instance, automotive industry drives development towards down-sized and down-speeded engines and higher cylinder pressure. It requires advancing the available solutions for noise and vibration attenuation making the design of efficient torsional vibration absorbers for drivetrain systems of ground vehicles very important and challenging problem.

Many researches have been already done both in academy as well as in industry dealing with modeling, simulation and analysis of torsional vibration dynamics, design

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and optimization of vibration absorbers for drivetrains of different engineering systems (see, e.g. [1] and references there in). Since 1985 a dual mass flywheel (DMF), a well-known concept of torsional dynamics absorber (TDA), is already used for torsional vibration attenuation in drivetrain systems of passenger cars [2]. In heavy-duty trucks the conventional single mass flywheel is still the most common and application of a DMF for heavy-duty trucks is an important research topic [3–5].

In the present paper the results of analysis of dual mass flywheels and the study of their limits of vibration attenuation in drivetrain systems of heavy-duty trucks in operating engine speed range up to 2000 rpm are presented. The third engine order vibration harmonic is in focus of analysis as one of the most significant contribution to the oscillatory response in drivetrains of heavy-duty trucks.

2 Modelling and Problem Formulations

2.1 Mathematical Model of a Generic Drivetrain System with a TDA

A generic drivetrain system (DTS) of a road vehicle is considered. The DTS comprises an engine, a TDA and a load transmission system. The engine generates the torque $T_e(t, \mathbf{d}_e)$ with oscillations at the input shaft of the TDA. Here $\mathbf{d}_e = [d_{e1}, \ldots, d_{ene}]^T$ is the vector of parameters of the input load. The design of a TDA may consist of several bodies connected by different stiffness and damping elements. The load transmission system consists of a set of machine components (a clutch, a gearbox, propeller shaft, rear axle and wheels) that all together use the torque at the transmission input shaft $T_g(t, \mathbf{d}_g)$ for goal-directed operation of a road vehicle. The torque $T_g(t, \mathbf{d}_g)$ has the vector of parameters of the load transmission system $\mathbf{d}_g = [d_{g1}, \ldots, d_{gng}]^T$.

In Fig. 1, the generic DTS with a TDA is shown. The TDA consists of three rotating rigid bodies, the primary flywheel (PFW), the secondary flywheel (SFW) and a tuned mass damper (TMD), that are connected by different stiffness and damping elements. This dynamics absorber will be called a DMF having TMD.

The equations of torsional vibration dynamics of the DTS with DMF having TMD can be written as follows

$$J_p\ddot{\varphi}_p = T_e(t, \mathbf{d}_e) - k_1(\varphi_p - \varphi_s) - c_1(\dot{\varphi}_p - \dot{\varphi}_s) \tag{1}$$

$$J_s\ddot{\varphi}_s = k_1(\varphi_p - \varphi_s) + c_1(\dot{\varphi}_p - \dot{\varphi}_s) + k_0(\varphi_0 - \varphi_s) + c_0(\dot{\varphi}_0 - \dot{\varphi}_s) - T_g(t, \mathbf{d}_g) \tag{2}$$

$$J_0\ddot{\varphi}_0 + k_0(\varphi_0 - \varphi_s) + c_0(\dot{\varphi}_0 - \dot{\varphi}_s) = 0 \tag{3}$$

In Eqs. (1)–(3) $\varphi_p, \varphi_s, \varphi_0$ are absolute angles of rotation of the PFW, the SFW, and the TMD, respectively; J_p, J_s, J_0 are the moments of inertia of the PFW, the SFW and the TMD, respectively; k_1, c_1, k_0, c_0 are the coefficients of torsional stiffness and torsional damping.

The Eqs. (1)–(3) together with the initial conditions

$$\varphi_p(t_0) = \varphi_p^0,\ \varphi_s(t_0) = \varphi_s^0, \varphi_0(t_0) = \varphi_0^0, \dot{\varphi}_p(t_0) = \dot{\varphi}_p^0, \dot{\varphi}_s(t_0) = \dot{\varphi}_s^0, \dot{\varphi}_0(t_0) = \dot{\varphi}_0^0 \quad (4)$$

constitute the mathematical model of the DTS with DMF having TMD. When the TMD is not presented in the TDA and $k_0 = c_0 = 0$, the Eqs. (1)–(2) constitute the mathematical model of the DTS with a DMF.

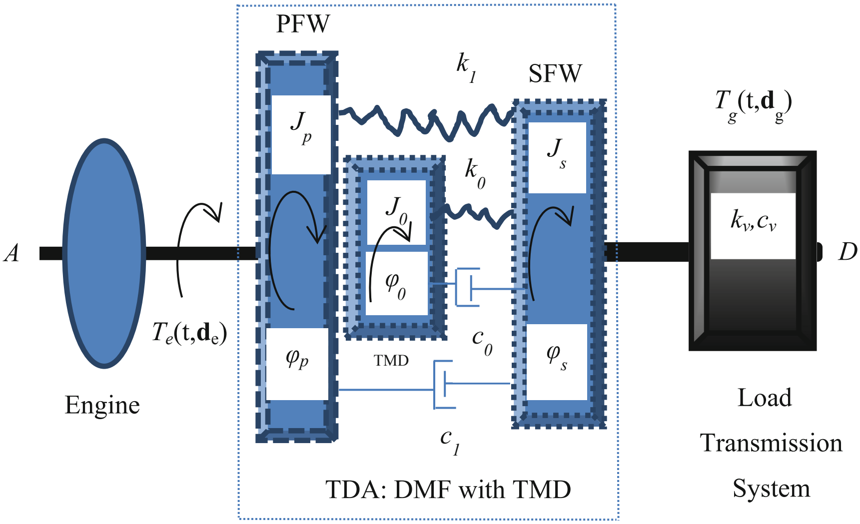

Fig. 1. A generic drivetrain system with torsional dynamics absorber comprising a dual mass flywheel having a tuned mass damper.

2.2 Problem Formulations

For the considered model of a generic drivetrain system the following problems of global sensitivity analysis (GSA) and multiobjective optimization (MOO) are stated.

Problem GSA of DTS. For given operational scenario (OS) of a DTS

$$OS = \{T_e(t,\mathbf{d}_e), \quad T_g(t,\mathbf{d}_g), \quad t \in [t_0, t_f], \quad \mathbf{d}_e \in \Omega_e \ , \quad \mathbf{d}_g \in \Omega_g\} \quad (5)$$

and chosen quality measures of the design of a TDA, $F_j, j = 1, 2, \ldots, n_F$, it is required to determine the total sensitivity indices, $S_i^T(F_j), i = 1, \ldots, m, j = 1, \ldots, n_F$, for all random varying design parameters, d_i, $i = 1, \ldots, m$, subject to differential equations of torsional vibration dynamics (1)–(3) and given restrictions on the design parameters, $\mathbf{d} = [d_1, d_2, \ldots, d_m]^T \in \Omega$.

Solution to the Problem GSA of DTS gives mapping between the values of the total sensitivity indices $S_i^T(F_j)$ of the measures F_j and the design parameters d_i of the absorber. Then, the vector of the most important design parameters, $\mathbf{d}_s = [d_{s1}, d_{s2}, \ldots, d_{sk}]^T \in \Omega$,, $1 \leq sk \leq m$, as well as the most sensitive measures (functionals), F_{jk}, $1 \leq jk \leq n_{F1} \leq n_F$, are identified and the following multiobjective optimization problem for the drivetrain system in question can be stated.

Problem MOO of DTS. For a given operational scenario (5) of a DTS it is required to determine the vector of design parameters, $\mathbf{d}_s^* = [d_{s1}^*, d_{s2}^*, \ldots d_{sk}^*]^T$, $sk \in [1, \ldots, m]$, and torsional vibration dynamics, $\mathbf{q}(t) = \mathbf{q}^*(t)$, that all together satisfy the system of variational equations

$$\min_{\mathbf{d}_s \in \Omega}(F_{jk}[\mathbf{q}(t), \mathbf{d}_s]) = F_{jk}[\mathbf{q}^*(t), \mathbf{d}_s^*] = F_{jk}^*,\ jk = 1, \ldots, n_{F1}, \quad n_{F1} \leq n_F \tag{6}$$

subject to differential Eqs. (1)–(3) and restrictions on design parameters $\mathbf{d}_s$.

Solution to the Problem MOO of DTS is resulted in Pareto fronts and corresponding Pareto sets of design parameters which all together make it possible to identify the limits of vibration attenuation of the dynamics absorber in question.

3 Methodology

The formulated problems have been solved by using the computer code for **s**ensitivity **a**nalysis and **m**ultiobjective **o**ptimization (SAMO) of engineering systems developed at the division of Dynamics Chalmers University of Technology [6]. The SAMO solves the global sensitivity analysis and multiobjective optimization problems based on multiplicative dimensional reduction method and genetic algorithm [7–9].

To evaluate the quality of the design of a TDA the following functionals (measures) are chosen:

$$\begin{aligned} &F_1(\mathbf{d}) = std(T_g[\mathbf{q}(t), \mathbf{d}]), \quad F_2(\mathbf{d}) = std(T_f[\mathbf{q}(t), \mathbf{d}]) \\ &F_3(\mathbf{d}) = std[\varphi_p(t) - \varphi_s(t)], \quad F_4(\mathbf{d}) = peak\_peak(T_g[\mathbf{q}(t), \mathbf{d}]) \\ &F_5(\mathbf{d}) = peak\_peak(T_f[\mathbf{q}(t), \mathbf{d}]), \quad F_6(\mathbf{d}) = peak\_peak[\varphi_p(t) - \varphi_s(t)] \end{aligned} \tag{7}$$

The functionals (7) measure standard deviation and peak-to-peak value of the torque at the transmission input shaft $T_g(t, \mathbf{d})$, the friction torque $T_f(t, \mathbf{d})$, and the torsional vibration $\varphi_p(t) - \varphi_s(t)$ of a DMF.

For both the global sensitivity analysis and multiobjective optimization problems, the mass-inertia, stiffness and damping characteristics of a TDA are chosen as the design parameters: the vector $\mathbf{d} = [d_1, d_2, d_3, d_4]^T = [k_1, c_1, J_p, J_s]^T \in \Omega$ in case of a DMF, and the vector $\mathbf{d} = [d_1, d_2, d_3, d_4, d_5, d_6, d_7]^T = [k_1, c_1, J_p, J_s, k_0, J_0, c_0]^T \in \Omega$ for the DMF having TMD. The total sensitivity indices, $S_i^T(F_j)$, $j = 1, \ldots, 6$, are determined by using dimensional reduction method [6, 7].

The global sensitivity analysis and multiobjective optimization problems are considered on a set of input torques describing the excitation of a combustion engine of a heavy-duty truck in the operating speed range up to 2000 rpm. The third engine order vibration harmonic is in focus of analysis as one of the most significant contribution to the oscillatory response. Sensitivity analysis and design optimization problems are considered for a DMF within the range of its structural parameters values feasible for application in heavy-duty truck drivetrain systems [3, 4].

In all simulations, the operational scenario (5) is defined by the following

$$\begin{aligned} &T_e(t,\mathbf{d}_e) = T_m + a_e \sin(\omega_{n_0} t + \alpha_e), \quad \omega_{n_0} = n_0\omega\,, \quad \omega = 2\pi n_e/60 \\ &T_g(t,\mathbf{d}_g) = k_v(\varphi_s - \varphi_v) + c_v(\dot{\varphi}_s - \dot{\varphi}_v), \quad \varphi_v(t) = \omega_v t + a_g \sin(\omega_g t + \alpha_g) \end{aligned} \tag{8}$$

In expressions (8), T_m is the mean value of engine input torque, ω_{n_0} is the n_0 - engine order vibration frequency, that is n_0 times the angular velocity ω, and n_e is the engine speed in rpm. k_v, c_v are equivalent torsional stiffness and damping coefficients of the load transmission system, φ_v, ω_v are absolute angle of rotation and angular velocity of the transmission input shaft. The parameters a_g, ω_g, α_g define the vibration at the transmission input shaft. The following values $T_m = 300\,\text{Nm}$, $n_0 = 3$, $a_e = 500\,\text{Nm}$, $k_v = 100000\,\text{Nm/rad}$, $c_v = 0{,}1\,\text{Nms/rad}$, $\omega_v = \omega_{n_0}/3$, $a_g = \alpha_g = \alpha_e = 0$ are used in the operational scenarios (5) and (8). The engine speed n_e was chosen in the range of 600 rpm–2000 rpm. The nominal values of the design parameters of a DMF that were used for global sensitivity analysis and Pareto optimization are the following

$$\mathbf{d}_{DMF}^{Nom} = [k_1, c_1, J_p, J_s]^T = [12732\,\text{Nm/rad}, 30\,\text{Nms/rad}, 1.8\,\text{kgm}^2, 0.9\,\text{kgm}^2]^T \tag{9}$$

4 Results

Different feasible input loads for the range of engine speed 600 rpm–2000 rpm have been chosen to study the limits of the quality of performance of a TDA in a generic DTS. Two conceptual designs of a TDA are considered for detailed study: a DMF and the DMF having torsional TMD. The obtained results of global sensitivity analysis (solutions of the Problem GSA of DTS) show the following. All design parameters of a TDA can significantly affect the level of attenuation of the oscillation of the torque at the transmission input shaft as well as the torsional vibration of a DMF and the friction torque. As example, the solution of the Problem GSA of DTS for operational scenario defined by expressions (5), (8) with engine speed n_e = 1600 rpm is presented in Fig. 2. As it follows from Fig. 2, the stiffness and mass-inertia characteristics are the most important design parameters of the DMF. Even damping characteristic has effect on torsional vibration of the DMF.

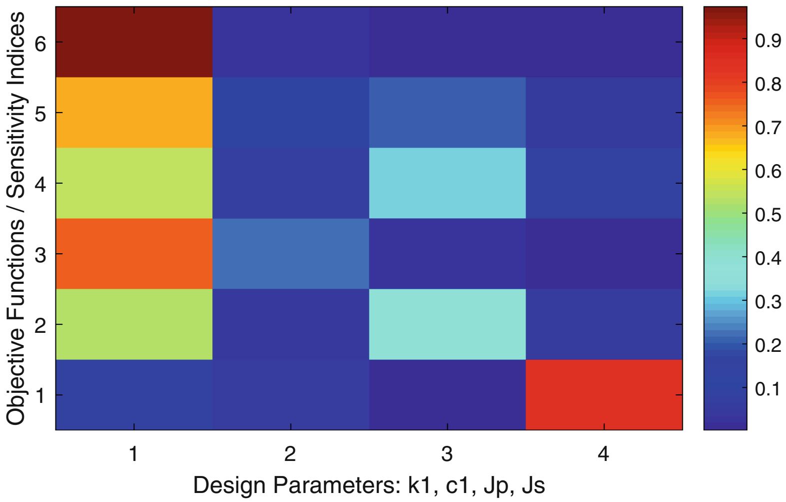

Fig. 2. Sensitivity indices of $F_1(\mathbf{d}),\ldots,F_6(\mathbf{d})$ for a DMF with engine speed $n_e = 1600$ rpm.

The Pareto fronts obtained by solution of the Problem MOO of DTS for the prescribed engine speeds in the range of 800 rpm–2000 rpm are depicted in Fig. 3. Here the DTS is equipped by the DMF. The analysis of the plots shows that there exists clear trade-off between efficiency of vibration attenuation by the DMF and its energy efficiency. The limiting possibilities of the vibration attenuation at the drivetrain system in question are determined by the design parameters of the DMF corresponding to the lower points of the curves $std(T_f(\mathbf{q}^*(t),\mathbf{d}^*)$ versus $std(T_g(\mathbf{q}^*(t),\mathbf{d}^*)$ in Fig. 3.

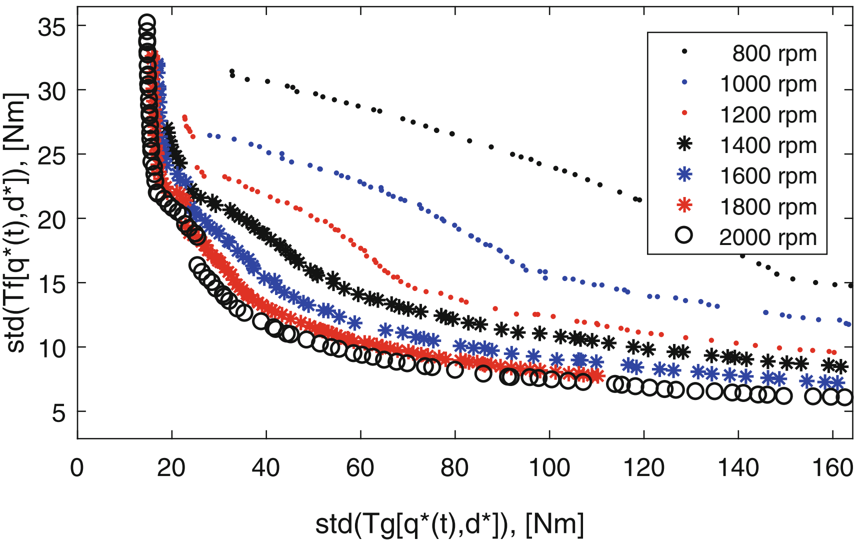

Fig. 3. Pareto fronts for the DTS with a DMF for engine speeds $n_e \in [800\ \text{rpm}–2000\ \text{rpm}]$.

For instance, the obtained optimized design parameters of the DMF for engine speeds n_e = 800 rpm and n_e = 1600 rpm are given by the vectors (10) and (11), respectively.

$$\mathbf{d}^*_{800\,\text{rpm}} = [k_1^*, c_1^*, J_p^*, J_s^*]^T = [10501\,\text{Nm/rad}, 51\,\text{Nms/rad}, 3.6\,\text{kgm}^2, 0.1\,\text{kgm}^2]^T \quad (10)$$

$$\mathbf{d}^*_{1600\,\text{rpm}} = [k_1^*, c_1^*, J_p^*, J_s^*]^T = [10867\,\text{Nm/rad}, 88\,\text{Nms/rad}, 2.3\,\text{kgm}^2, 2.0\,\text{kgm}^2]^T \quad (11)$$

In Fig. 4, the engine speed history of standard deviation of the torques at the transmission input shaft are presented for the DMF with nominal (black curve) and with optimized design parameters obtained for prescribed engine speed n_e = 800 rpm (red curve) and for n_e = 1600 rpm (green curve). Analysis of Fig. 4 shows that in comparison with the DMF having nominal values of the design parameters (9), the DMF optimized for prescribed engine speed n_e = 800 rpm (parameters (10)) significantly attenuates oscillation of the torque at the transmission input shaft.

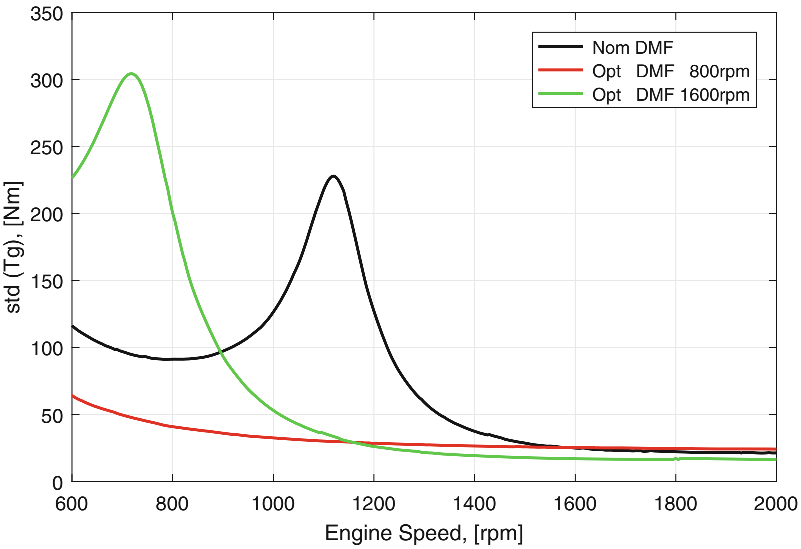

Fig. 4. Engine speed history of standard deviation of the torques at the transmission input shaft for the DMF with nominal (black curve) and optimized design parameters for engine speed 800 rpm (red curve) and speed 1600 rpm (green curve).

As can be seen in Fig. 4, the DMF optimized for prescribed engine speed n_e = 1600 rpm (parameters (11)) does not reduce high resonance peak of oscillation of the torque at the transmission input shaft. Additionally, it should also be mentioned that the obtained vectors of the optimized design parameters (10)–(11) are characterized much higher moments of inertia of flywheels in comparison to their nominal values (9). The above mention is the reason why it is important to consider optimization of vibration absorbers not only for the prescribed value of the engine speed but for the engine operating speed range and to impose more sharp restriction on mass-inertial characteristics of the flywheels suitable for implementation in heavy-duty truck powertrains.

Below, multiobjective optimization methodology is applied to design vibration absorbers for the best attenuation of oscillation of the torques at the transmission input shaft in the engine operating speed range up to 2000 rpm. With intention to take care both with resonances of oscillation of the torque at the transmission input shaft as well as to enhance the vibration attenuation in the operating engine speed range it is proposed to use the following functionals for optimization of design of vibration absorbers

$$F_7(\mathbf{d}) = \int_{600}^{2000} std(T_g[\mathbf{q}(t), \mathbf{d}, n_e])dn_e, \quad F_8(\mathbf{d}) = \int_{600}^{2000} std(T_f[\mathbf{q}(t), \mathbf{d}, n_e])dn_e \qquad (12)$$

The measures $F_7(\mathbf{d}), F_8(\mathbf{d})$ characterize the energy of oscillations of the torque at the transmission input shaft and the energy dissipating in a DMF in the operating engine speed range $600\,\text{rpm} \leq n_e \leq 2000\,\text{rpm}$.

The Problem MOO of DTS was solved in the operating engine speed by determining the vector of design parameters of the DMF, $\mathbf{d} = [k_1^*, c_1^*, J_p^*, J_s^*]^T = \mathbf{d}^*_{gDMFenergy} \in \Omega$, and the torsional vibration dynamics, $\mathbf{q}(t) = \mathbf{q}^*(t)$, that minimize the objective functions (12) subject to the differential equations of motion of the absorber.

The obtained values of the design parameters of the DMF which minimize the objective function $F_7(\mathbf{d}) = \int_{600}^{2000} std(T_g[\mathbf{q}(t), \mathbf{d}, n_e])dn_e$ are the following

$$\mathbf{d}^*_{gDMFenergy} = [k_1^*, c_1^*, J_p^*, J_s^*]^T = [10966\,\text{Nm/rad}, 41\,\text{Nms/rad}, 2.7\,\text{kgm}^2, 0.45\,\text{kgm}^2]^T \qquad (13)$$

The same Problem MOO of DTS was also solved in the operating engine speed $600\,\text{rpm} \leq n_e \leq 2000\,\text{rpm}$ by determining the vector of design parameters of the DMF having TMD, $\mathbf{d} = [k_1^*, c_1^*, J_p^*, J_s^*, k_0^*, J_0^*, c_0^*]^T = \mathbf{d}^*_{TMDenergy} \in \Omega$, and the torsional vibration dynamics, $\mathbf{q}(t) = \mathbf{q}^*(t)$, that minimize the objective functions (12) subject to the differential equations of motion of the DTS with DMF having TMD.

The obtained values of the design parameters of the DMF having TMD which minimize the objective function $F_7(\mathbf{d}) = \int_{600}^{2000} std(T_g[\mathbf{q}(t), \mathbf{d}, n_e])dn_e$ are the following

$$\mathbf{d}^*_{TMDenergy} = [k_1^*, c_1^*, J_p^*, J_s^*, k_0^*, J_0^*, c_0^*]^T = [10551, 43, 2.7, 0.5, 9948, 0.28, 0.18]^T \qquad (14)$$

In Fig. 5, standard deviation of the torques at the transmission input shaft as function of engine speed are depicted for the DMF with nominal design parameters (9) (black curve), for the DMF with optimized design parameters defined by the vector (13) (red curve), and for the DMF having TMD with optimized design parameters (14) (blue curve). Evaluation of the objective function $F_7(\mathbf{d}) = \int_{600}^{2000} std(T_g[\mathbf{q}(t), \mathbf{d}, n_e])dn_e$ for nominal (the parameters (9)) and for optimized values of the design parameters of a DMF, (the parameters (13)), makes it possible to compare quantitatively the obtained engine speed history of standard deviation of the torques at the transmission input shaft depicted in Fig. 5. It is found that the optimized DMF increases up to 40% efficiency of attenuation of the oscillations of the torque at the transmission input shaft in comparison to the performance of the DMF with nominal design parameters. Analysis of Fig. 5 shows also that incorporation of the TMD into the DMF can enhance attenuation of the oscillation of the torque at the transmission input shaft in comparison to optimized DMF.

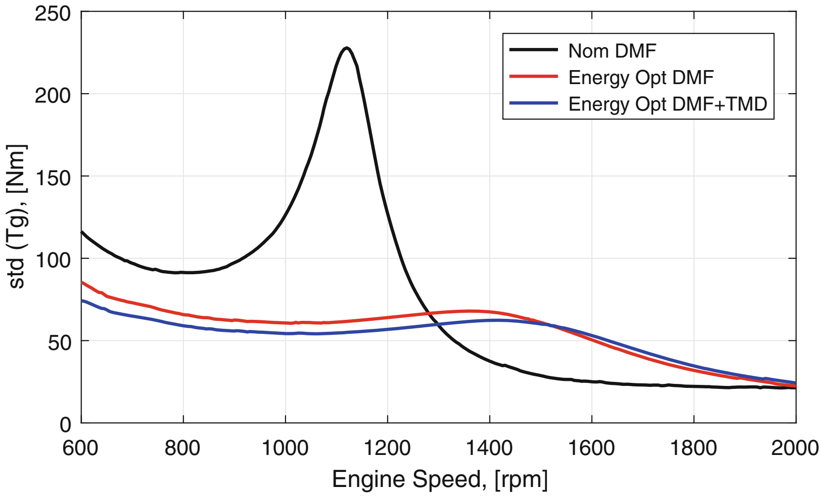

Fig. 5. Standard deviation of the torques at the transmission input shaft in the operating engine speed range 600 rpm $\leq n_e \leq$ 2000 rpm for the DMF with nominal design parameters (black curve) and with optimized parameter (red curve), as well as with optimized parameters for the DMF having TMD (blue curve).

5 Conclusion and Outlook

The methodology based on global sensitivity analysis and Pareto optimization has been proven to be efficient to study the limits of vibration attenuation and for designing of torsional dynamics absorbers for drivetrain systems. The results show that the most sensitive design parameters of a torsional dynamics absorber are the mass-inertia and stiffness characteristics that enable to mitigate efficiently the engine torque oscillations transmitted to a drivetrain system. There exists a clear trade-off between the measures of vibration attenuation and energy efficiency in design of torsional dynamics absorbers for generic drivetrain systems. The obtained Pareto fronts and corresponding Pareto sets define the limits of vibration attenuation in a drivetrain system by the vibration absorber in question. Incorporating torsional tuned mass damper into the design of dual mass flywheel with appropriate optimization of the obtained vibration absorber can enhance its vibration absorption capability.

The study of the limiting possibilities of vibration absorption in drivetrain systems by using the vibration absorbers within the frame of non-linear models can be the focus of future research. Verification and validation of the results obtained by using complete model of a drivetrain system of a heavy-duty truck as well as experimental data are also important next steps of the study.

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