The requirements for the design of dual-mass flywheels

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Abstract. In the modern powertrains with new low emission combustion engines, the effort to reduce emissions and weight of the entire system increase the efficiency of vehicle drives but also make the drive much more sensitive to vibration. The new design concepts of dual-mass flywheels offer more options for eliminating vibration in vehicle drive systems. Every design of the dual-mass flywheel must ensure that it has the appropriate characteristics. A suitable characteristic then creates a prerequisite for application in a drive in which the resonance area can be moved to very low operating speed.

1. Introduction
Under the influence of the rapid climate change, both experts and the public have realised that one should be more sensitive to the environment. Pressure is rising for a reduction in emissions from combustion engines installed in both automobiles, and construction, agricultural, and other vehicles and machines. This led to rapid technological advancements in car industry. Emission standards have been introduced to set a limit to the emission content in exhaust gases of cars in traffic. In 1992, the first European emission limits have been set by EURO1 regulation. Because of air pollution due to traffic, the emission limits kept becoming more strict and further regulations were imposed [1]. In order to fulfill these requirements, new strategies and devices, whose role was to reduce air pollution, appeared in cars. Among the early solutions were changes in fuel mixture, filtering and a reduction of certain fuel components by means of chemical reactions [2]. EURO6 regulation demands that the levels of some pollutants be reduced by up to 96 % with respect to the limits from 1992 [3]. The only way to achieve this is by interventions in engine design. It is not unreasonable to presume that the upcoming EURO7 will bring about new design concepts in vehicles that should further reduce their emissions [4, 5].

With the advent of novel technologies in combustion engines, complex issues arise in powertrain dynamics. Even before the arrival of EURO4, engineers attempted to cut down exhaust emission by means of a lower fuel consumption. Multiple techniques exist nowadays that reduce fuel consumption of the combustion engine. Specialised devices which affect the production of exhaust gases serve their purpose, however, they have an adverse effect on some technical characteristics of the vehicle – vibrations perceivable during vehicle operation are an example. There are multiple sources of vibration in a vehicle, and it is desirable to attenuate them [6–8].

Dual mass flywheels (DMFs) are used to diminish vibration in drivetrains. The inertial mass of the DMF is split in two parts. The primary mass is connected to the crankshaft, while the secondary mass
is part of the clutch. Elastic members connect these two masses. The most common design of the DMF at present employs metal compression springs.

2. Problem definition

Power drives can be characterised by the transfer of energy from the driving entity to the entity that is being driven [9–11]. A mechanical system that transmits power from an internal combustion engine to the main drive is shown in figure 1.

![Figure 1. The mechanical system with dual mass flywheel.](image)

According to the theory of mechanical vibrations, this mechanical system can be described a torsional system with \( n \) masses [12, 13]. Such an undamped \( n \)-degree of freedom (DoF) system can be described in a matrix form Eq. (1):

\[
I \ddot{\varphi} + k \varphi = M_k
\]

where: \( I \) – inertia matrix, \( k \) – stiffness matrix, \( M_k \) – vector of torque loads, \( \varphi \) – vector of angles of twist, \( \ddot{\varphi} \) – vector of angular accelerations.

Such torsional mechanical system has \( n-1 \) non-zero natural frequencies (and one zero natural frequency which corresponds to free rotation). When one of these natural frequencies coincides with the frequency of excitation, resonance occurs. As a first estimate of the natural frequency, the \( n \)-degree of freedom system is often reduced to an undamped two- or three-body problem [14, 15].

For the case of an undamped two-degree-of-freedom system (figure 2), Eq. (2) can be derived [16, 17]:

\[
I_1 \ddot{\varphi}_1 + k \cdot (\varphi_1 - \varphi_2) = M_i \sin(i \omega t)
\]

\[
I_2 \ddot{\varphi}_2 - k \cdot (\varphi_1 - \varphi_2) = 0
\]

where: \( I_1, I_2 \) – mass moments of inertia (kg m\(^2\)), \( k \) – stiffness connecting the masses (Nm rad\(^{-1}\)), \( \varphi_1, \varphi_2 \) – angles of twist (°), \( M_i \) – \( i \)-th principal component of harmonic torque (Nm), \( i \) – the order of harmonic component (°), \( \omega \) – angular frequency of the \( i \)-th harmonic component (rad s\(^{-1}\)), \( t \) – time (s).

![Figure 2. Two-degree-of-freedom simplification of a mechanical system.](image)

Solving the equations of motion, the following expression for the angular natural frequency emerges (Eq. (3)):

\[
\Omega_0^2 = k \frac{I_1 + I_2}{I_1 I_2}
\]

where: \( I_1, I_2 \) – mass moments of inertia (kg m\(^2\)), \( k \) – stiffness of the connection between the two bodies (Nm rad\(^{-1}\)), \( \Omega_0 \) – natural frequency (rad s\(^{-1}\)).
Examining Eq. (3), it is evident that the natural frequency depends on both the moments of inertia and torsional stiffness which is a property of the DMF. The moments of inertia are usually constant. If the DMF has a linear characteristic, the torsional stiffness will be constant. If the characteristic of the DMF is non-linear, then the torsional stiffness is, too, non-linear, and hence the natural frequency of the mechanical system will vary.

In dynamic drive control, resonance associated with the principal harmonic component is at the centre of attention. Design precautions are then aimed to avoid an overlap of the principal harmonic component and the natural frequency.

Dynamic ratios in powertrains with internal combustion engines can be conveniently expressed via Campbell diagrams (figure 3). They will be also used to present the trends in low emission engines. Campbell diagrams are used as follows: the chart includes \( i \)-th harmonic components of the load (denoted by numbers 1 to 5). The thick line corresponds to the principal harmonic component \( i \). The dashed line is a plot of the natural frequency as a function of rotational speed. The intersection of the natural frequency and the \( i \)-th harmonic component represents the resonance originating from the \( i \)-th harmonic component – we shall designate it by “\( Ri \)”. The effort to reduce engine emission has an adverse effect on vibrations, whereby resonance arises in the operating range of the engine. Two examples of the effects of torsional stiffness on the natural frequency and associated resonance are shown in figure 3. In the case of figure 3a, the DMF has a linear response and, therefore, its torsional stiffness is linear too. In figure 3b, on the other hand, the DMF has a non-linear characteristic and its torsional stiffness is not constant. In figure 3a, resonance R3 comes from the principal harmonic component and occurs below the engine operating range. As a result, this resonance only occurs when the engine is being started or stopped, and the duration of the resonant vibration is very short. Resonant conditions R2 and R1 originate from subordinate harmonic components and have little effect on the undesirable vibration. A problem arises when, during engine operation, emission reduction measures are applied which change the principal harmonic component (e.g. by a deactivation of cylinders). Such an action could trigger resonance in the operating speed range. A suitable solution would be to use a DMF with variable stiffness (figure 3b). In this case, a carefully designed torsional stiffness of the DMF can avoid the occurrence of resonances from both principal and subordinate harmonic driving components.

![Campbell diagram](image)

**Figure 3.** Campbell diagram.

The design of non-linear DMFs is quite challenging. The non-linear characteristic can be achieved by the use of rubber elastic components. However, the properties of rubber are sensitive to temperature,
and its stiffness degrades above 70 °C, making it less suitable for use in DMFs [18, 19]. For these reasons, metal springs are often used as the elastic members of the DMF – these, however, have linear characteristic. Therefore, two design solutions for the DMF are employed in low emission engines. First, DMFs with constant, yet very low stiffness can be used. This low stiffness will shift the natural frequency to a lower value – ideally approaching zero. Second, DMFs can be designed to have a variable torsional stiffness – i.e. DMFs with controlled torsional stiffness.

Both of these approaches are used in DMF design. Additionally, various other measures are Clutch pre-dampers (CD) [20], and centrifugal pendulum absorbers (CPA) [21], are a few examples. Numerous studies deal with the design and function of the DMF with CPA [22, 23]. Sun et al. examined the non-linear characteristic of DMFs with radial springs in order to optimise key parameters. Further studies include DMFs with magnetorheological chambers [24], variable stiffness [25] and spiral circumference spring [26]. Geometry and operating conditions of the currently used DMFs with wound springs suffer from numerous issues [27, 28]. Moreover, their stiffness and damping changes drastically at higher operating speeds: friction lowers the number of active loops of the spring, increasing stiffness and decreasing damping. The DMF is designed to be made by welding, which should also provide sealing, however, leakage of lubricant often occurs, decreasing the reliability of the DMF [29, 30].

Authors Maffiodo et al. present a novel DMF design in which traditional compression wound springs are replaced by spiral springs [31]. This concept of DMF should exhibit elastic deformation of springs which should be the only factor determining the stiffness of spiral springs, and they will only marginally be affected by torque and speed. Moreover, the system does not require lubrication, hence sealing by welding is redundant.

It has been demonstrated that the topic of DMFs is quite complex, as are the requirements imposed on the designers of DMFs.

3. Solution of the problem

It is desirable to incorporate into the mechanical system such dynamic members that will shift natural frequency to prevent resonance when the engine is running in low emission mode. It is often difficult to achieve this – solutions of complicated dynamics are often put in place. As we mention in Section 2, it is suitable for the drivetrain to include a component that can react to changes in dynamic state – i.e. a component whose natural frequency can be tuned while the drivetrain is in operation [32]. The pneumatic dual mass flywheel (PDMF) shown in figure 4 is a suited example [33]. It consists of the primary mass (1), secondary mass (2) and pneumatic-elastic part (3). It can be seen in figure 4(right), that the torsional stiffness that is achievable spans a large area when plotted against torque.

Figure 4. Pneumatic dual mass flywheel (left) and the space of achievable torsional stiffness (right).
If the PDMF is equipped with a suitable control system (figure 5), it will be possible to continuously change the torsional stiffness during powertrain operation, as needed. As a result, the PDMF can be designed and used in two stages. First, the ratio of the primary and secondary mass can be designed such as to provide a base level of tuning for the mechanical system. Second, the PDMF can be tuned on the run by changing the internal pressure, which in turn shall change the torsional stiffness, and hence the natural frequency. In order to do so, a control system must be used that will manage the PDMF pressure and torsional stiffness. This control system should communicate with the control system of the engine.

![Diagram](image1)

**Figure 5.** Management of pneumatic dual mass flywheel for the mechanical system.

We demonstrate the effect of such tuning using the Campbell diagram in figure 6. The basis is a Campbell diagram which represents the mechanical system with low emission combustion engine. The PDMF is used to react to changes in engine operating conditions. The objective is to achieve a low vibration amplitude by tuning – i.e. continuously changing the torsional stiffness, which shifts the natural frequency of the system [34, 35]. The Campbell diagram in figure 6(left) shows three different functional configurations of natural frequency (denoted A, B, C), which shall be required in various engine operating conditions. It is evident from figure 6(right) that these three configurations can all be achieved by corresponding changes in torsional stiffness. Control systems in PDMFs enable the torsional stiffness to be continuously changed to a desired value. The resulting value can be held constant (configuration A), or variable (B, C). The variable configuration can be linear (B) or non-linear (C). The advantage of the PDMF is the ability to change the torsional stiffness continuously while the engine is running.

![Diagram](image2)

**Figure 6.** Natural frequency in Campbell diagram and controlled torsional stiffness of PDMF.
4. Conclusion
It has been noted in Problem definition that powertrains with low emission engines need to contain components with the ability to react to the various emission reduction strategies in engine management. At present, DMFs are frequently used. However, they suffer from numerous limitations.

An important conclusion has been drawn from Campbell diagrams – changing the natural frequency can avoid the onset of resonance. The change in resonant frequency of a mechanical system can be achieved in two ways emerging from Eq. (3). The first approach is to change the mass moments of inertia. This principle can only be applied when the mechanical system is stationary, whereby rotating masses are added or removed. The second approach is to change the properties of stiffness or damping of one or more powertrain components. If the corresponding component allows for a continuous change of its stiffness, it can be done so when the powertrain is in operation as well.

We have presented the PDMF as a type of the DMF. We have also demonstrated that the PDMF is a component which can fulfil the stringent requirements for vibration reduction in low emission engines.

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Acknowledgments
This paper was written in the framework of Grant Projects: VEGA 1/0473/17, KEGA 041TUKE-4/2017, APVV-16-0259, KEGA 015ŽU-4/2017, VEGA 1/0073/19.