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THEORETICAL AND EXPERIMENTAL INVESTIGATION OF A KINEMATICALLY DRIVEN FLYWHEEL FOR REDUCING ROTATIONAL VIBRATIONS

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Abstract. *Modern turbocharged internal combustion engines induce high fluctuating torques at the crankshaft. They result in rotational crankshaft vibrations that are transferred both to the gearbox and the auxiliary engine systems. To reduce the rotational crankshaft vibrations, a passive mechanical device for compensating fluctuating engine torques has been developed. It comprises a flywheel that is coupled to the crankshaft by means of a non-uniformly transmitting mechanism. The kinematical transfer behavior of the mechanism is synthesized in such a manner that the inertia torque of the flywheel compensates at least one harmonic of the fluctuating engine torque. The degree of non-uniformity of the mechanism has to be adapted to the actual load and rotational speed of the engine. As a solution, a double-crank mechanism with cycloidal-crank input and adjustable crank length is proposed and analyzed. Parameter synthesis is achieved by means of a simplified mechanical model that calculates the required transfer function for a given engine torque. To analyze the overall dynamic behavior, the device is modeled in a multibody domain. Simulation results are validated using an electrically driven test rig. Comparisons between simulation and experimental results demonstrate the potential of the device.*

1 Introduction

The strong demand for more efficient automobiles forces the development of so-called downsized combustion engines with high specific power. The combination of a small number of cylinders and high cylinder pressures achieved by supercharging leads to fluctuating torques with high amplitudes at the crankshaft. The change in full-load speed-torque characteristics of a typical four-cylinder diesel engine (see Fig. 1) shows clearly that considerably higher fluctuating torques at the crankshaft occur at lower angular speeds and, by this, at lower excitation frequencies. To fulfill the likewise increasing noise comfort requirements of the power train, new measures to reduce the engine-induced angular oscillations are needed.

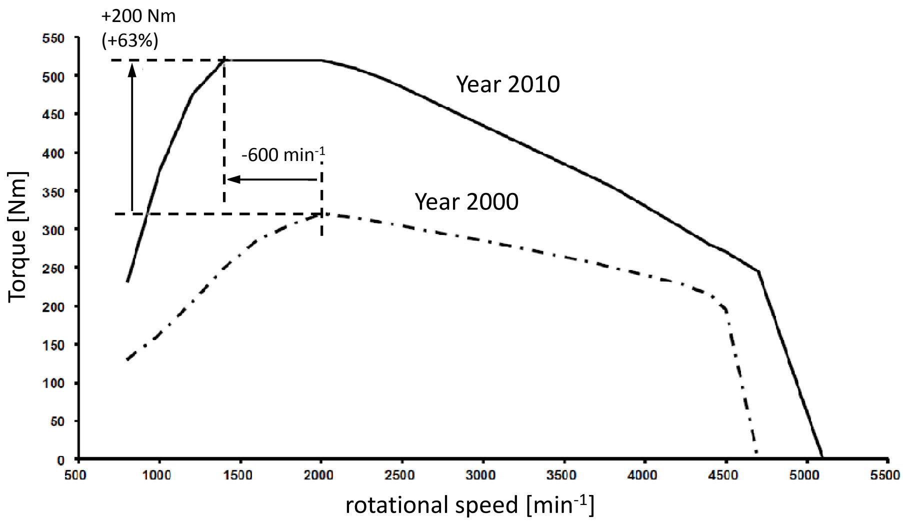


Figure 1: Change in full-load characteristics of a combustion engine from 2000 - 2010 [7]

State of the art to decouple the crankshaft oscillation from the driving shaft of the gearbox is a Dual Mass Flywheel (DMF). It contains a primary flywheel rigidly connected to the crankshaft and a secondary flywheel bolted to the input shaft of the gearbox. Both are coupled using a spring-damper system that is attuned to have its eigenfrequency below the excitation frequency at idle speed. Decreasing idle speeds require softer springs that lead to large relative angular displacements between both flywheel masses. Even with nonlinear spring characteristics and new innovative concepts like the combination of DMF with Centrifugal Pendulum Vibration Absorbers it gets more and more difficult to find appropriate system parameters to fulfill the desired comfort and efficiency requirements over the complete operating range. Therefore, research is undertaken to find new measures to reduce crankshaft oscillation or its transmission to the drivetrain.

Patents are published for non-uniformly transmitting mechanisms that change in a kinematically prescribed manner the effective inertia of the flywheel in dependence of the crankshaft rotation angle. A solution is to change the mass moment of inertia of a flywheel that is rigidly coupled to the crankshaft by means of movable, kinematically coupled auxiliary masses [1, 2]. Another solution is to couple a flywheel to the crankshaft using a non-uniformly transmitting mechanism leading to a difference in angular speeds of the flywheel and the crankshaft [3, 5]. This paper describes a mechanism with constant inertia of the flywheel and an appropriate mechanism to adjust the difference in angular speed. By designing such a system, several requirements have to be taken into account.

In order to use the system for all combustion engines, the topology of the mechanism must enable the design for different compensating torques depending on the number of cylinders. For example, for a four-cylinder four-stroke engine the compensation torque has to be set to the second order, while it is for a three-cylinder four-stroke engine the 1.5th order. Further, the compensation torque has to be continuously adjustable with respect to the engine state as the amplitude and phase of the excitation torque changes with angular speed and load. In addition, a direct force flow through the mechanism is preferable because of the limited design space and the high amplitudes of the required compensation torque.

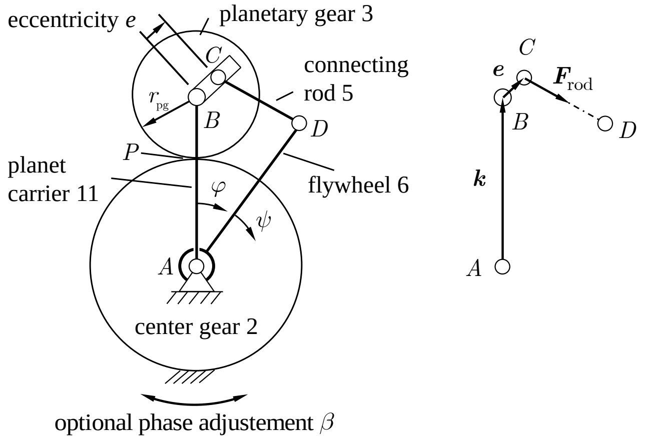
The paper is organized as follows. In section 2 a double-crank mechanism with cycloidal-crank input and adjustable crank length is proposed that fulfills the aforementioned requirements. In section 3 the transfer function from the crankshaft rotation angle to the flywheel rotation angle needed for a desired compensation torque is calculated by using a simplified dynamic model. The proposed double-crank mechanism is parametrically synthesized in section 4 by means of a numerical optimization procedure based on a multibody model of the overall crankshaft-flywheel system. Here, it has to be taken into account that neither the fluctuating torque produced by the flywheel mass nor the non-uniform torque produced by the combustion engine correspond to a single harmonic. Therefore, it is appropriate to consider the higher-order vibrations during parameter synthesis as well. Section 5 describes the experimental setup and the numerical model of the test rig, and in section 6 some model validations are shown.

2 Design of a double-crank mechanism with cycloidal-crank input

Comparing several designs of transmitting mechanisms for coupling a flywheel to the crankshaft shows that a double-crank mechanism with cycloidal-crank input according to Fig. 2(a) appropriately fulfills the requirements formulated above [4, 6]. The input crank BC of the double-crank mechanism $ABCD$ is part of a planetary gear, driven by the planet carrier AB that is rigidly connected with the crankshaft. Thus, the input crank BC carries out a cycloidal motion around the fixed center gear. By means of the connecting rod CD, the input crank BC is coupled with the output crank AD that is formed by the flywheel. Thus, the crankshaft rotation angle φ is transmitted into the flywheel rotation angle ψ. The degree of non-uniformity of the transfer function $\psi(\varphi)$ is controlled by the effective length (eccentricity) e of the input crank BC. For $e = 0$ the angular speeds of crankshaft and flywheel coincide, corresponding to a conventional single mass flywheel. For $e > 0$ the flywheel angular speed oscillates around the angular speed of the crankshaft. The main order of this oscillation depends on the gear ratio between center and planetary gear. If necessary, the phase angle of the flywheel oscillation can be adapted, by turning the center gear by an angle β. By this, the system can be used for a broad variety of combustion engines.

Another advantage of this system is the achievable high relative acceleration $\Delta\ddot{\varphi} = \ddot{\psi} - \ddot{\varphi}$ between crankshaft and flywheel mass that can be exploited to generate high amplitudes of the compensating torque. This becomes evident by looking at the velocity field of the planetary gear with its instantaneous center of rotation P at the tooth engagement. Considering the extreme case of the eccentricity e coinciding with the planetary gear radius r_{pg}, the speed of point C oscillates between $v_C = 0$ and $v_C = 2k\dot{\varphi}$, with the length k of the planet carrier AB, during one rotation of the planetary gear. Consequently, the angular velocity of the flywheel $\dot{\psi}$ oscillates between zero and approximately twice the angular velocity of the crankshaft $\dot{\varphi}$.

In a mechanical design of the system, eccentricities e will be small with respect to the planetary gear radius r_{pg} to be constructively realizable. Therefore, the moment arm for the torque at the planetary wheel is small and will reduce the load in the tooth engagement in point P.



(a) (b)

Figure 2: (a) Double-crank mechanism with cycloidal-crank input. (b) vector definition

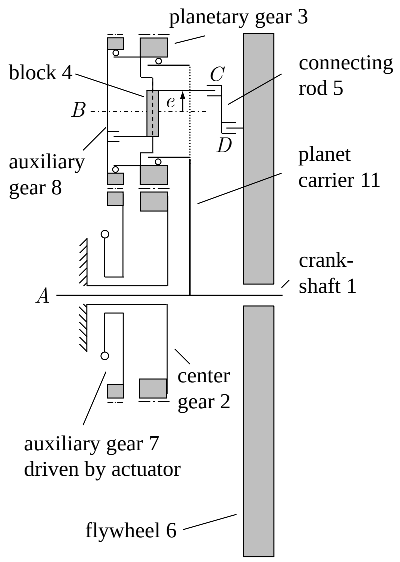


Figure 3: Mechanism to adjust the eccentricity e

Considering the load on the mechanical components, it should be noted that a connecting rod force $\boldsymbol{F}_{\mathrm{rod}}$ causes a torque at the planetary gear of $\boldsymbol{M}_{\mathrm{pg}} = \boldsymbol{e} \times \boldsymbol{F}_{\mathrm{rod}}$, while the compensation torque on the crankshaft $\boldsymbol{M}$ is considerably higher (see Fig. 2(b)). Assuming $e << r_{\mathrm{pg}}$ it can be approximated by $\boldsymbol{M} \approx (\boldsymbol{k} + \boldsymbol{e}) \times \boldsymbol{F}_{\mathrm{rod}}$.

A mechanical scheme for an adjustable eccentricity e in the double rotating system of the planetary gear (rotation around the crankshaft axis and around the planet wheel axis) is shown in Fig. 3. The corresponding mechanical design is illustrated in Fig. 4. The components seen in Figs. 3 and 4 are the planet carrier 11 rigidly connected with the crankshaft 1, the fixed center gear 2, the planetary gear 3, the connecting rod 5, and the flywheel 6. The block 4 is movable in a radial slot of the planetary gear 3 and bears the pivot point C. The task is to kinematically move block 4 in order to adjust the eccentricity e.

This is achieved by a pair of auxiliary gears 7 and 8 that are arranged in-parallel to the load-bearing gears 2 and 3 and have the same radii. The auxiliary center gear 7 rotates around the axis of the load-bearing center gear 2. The auxiliary planetary gear 8 rotates around the axis of the load-bearing planetary gear 3. A pin 9 on the block 4 is engaged into a sliding track 10 milled into the auxiliary planetary gear 8 in such a manner that a relative rotation of the planetary gears 3 and 8 forces the radial movement of the block 4 in its slot. The relative rotation of the planetary gears 3 and 8 is controlled by turning the auxiliary center gear 7 by means of an appropriate actuator in the resting system. If the auxiliary center gear 7 is kept fixed, the auxiliary planetary gear 8 rotates together with the load-bearing planetary gear 3 keeping the eccentricity e constant. Altogether, the eccentricity e can be altered from zero (no compensation torque) up to a maximum value e_{max} (compensation torque with maximal amplitude) by turning the auxiliary center gear 7 relative to the fixed load-bearing center gear 2. Additionally, it is possible to adjust the phase angle of the compensating torque by turning both center gears 2 and 7 through the same angle using an additional actuator. For force-balancing of the mechanism, the counterweight 12 is provided.

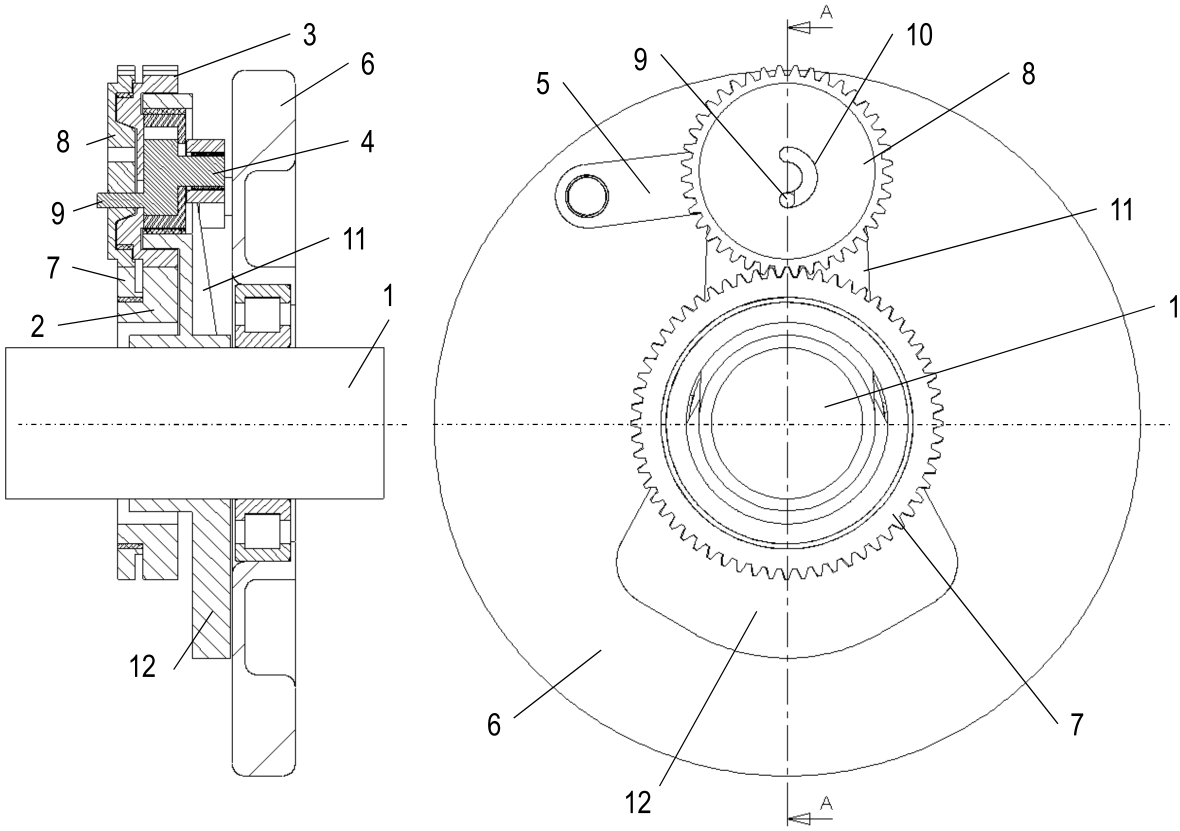


Figure 4: Design for the geared four-bar mechanism with adjustable eccentricity [4].

3 Transmission function for a desired compensation torque

The schematic arrangement of the compensation system with a flywheel coupled to the crankshaft by means of a general transmitting mechanism is shown in Fig. 5(a). The mechanism converts the rotation angle of the crankshaft φ non-uniformly into the rotation angle of the flywheel ψ. The transfer function $\psi(\varphi)$ needed to realize an ideal compensating torque can be derived independently from the design of the mechanism. For this purpose, the compensation system is cut free at the input of the mechanism according to Fig. 5(b). With the inertia of the flywheel I, the compensation torque $M(\varphi)$ is obtained for a given crankshaft rotation $\varphi(t)$ by

$$M(\varphi) = I\ddot{\psi}\psi' \qquad \text{with} \qquad \psi' = \frac{\mathrm{d}\psi}{\mathrm{d}\varphi}, \quad \ddot{\psi} = \frac{\mathrm{d}^2\psi}{\mathrm{d}t^2}. \tag{1}$$

Exploiting the second-order time derivative $\ddot{\psi} = \psi'\ddot{\varphi} + \psi''\dot{\varphi}^2$ leads to

$$M(\varphi) = I\,\psi'^2\,\ddot{\varphi} + I\,\psi'\,\psi''\,\dot{\varphi}^2. \tag{2}$$

In the assumed case of ideal compensation the crankshaft rotates uniformly, thus $\dot{\varphi} = \Omega = \text{const.}$ and $\ddot{\varphi} = 0$. Then, the compensating torque

$$M_c(\varphi) = I\,\psi'\,\psi''\,\Omega^2 \tag{3}$$

must completely compensate the fluctuating part of the torque generated by the combustion engine. By means of (3), the transfer function $\psi(\varphi)$ required for a desired compensation torque

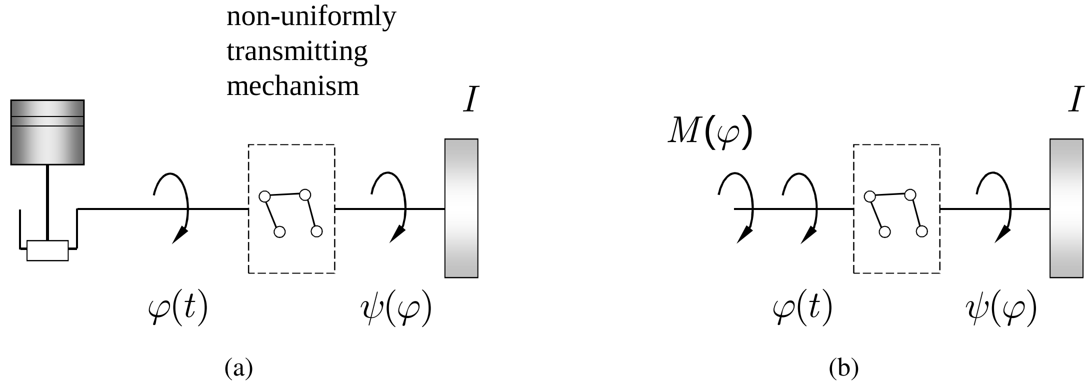


Figure 5: Kinematically driven flywheel for a combustion engine
(a) Schematic arrangement, (b) Free-cut compensation system

$M_c(\varphi)$ can be calculated. For this purpose, the variables φ and ψ in (3) are separated under consideration of $\psi'' = \frac{\mathrm{d}\psi'}{\mathrm{d}\varphi}$, and both sides of the equation are integrated,

$$\int_0^{\psi'} \psi' \,\mathrm{d}\psi' = \int_0^{\varphi} \frac{M_c(\varphi)}{I\Omega^2} \,\mathrm{d}\varphi \,. \tag{4}$$

Integrating (4) yields with the integration constant C_1

$$\frac{\psi'^2}{2} = \frac{1}{I\Omega^2} \int_0^{\varphi} M_c(\varphi)\,\mathrm{d}\varphi + C_1 \quad \rightarrow \quad \frac{\mathrm{d}\psi}{\mathrm{d}\varphi} = \sqrt{\frac{2}{I\Omega^2} \int_0^{\varphi} M_c(\varphi)\,\mathrm{d}\varphi + C_1} \,. \tag{5}$$

Another integration with the integration constant C_2 leads to the desired transfer function $\psi(\varphi)$,

$$\psi(\varphi) = \int\limits_0^{\varphi} \sqrt{\frac{2}{I\Omega^2} \int_0^{\varphi} M_c(\varphi)\,\mathrm{d}\varphi + C_1} \;\mathrm{d}\varphi + C_2 \,. \tag{6}$$

The constant C_2 represents the angle ψ for $\varphi = 0$. It can be set to zero without loss of generality. To calculate the constant C_1, the given angular period of the desired compensation torque $\Phi = m\pi$ is taken into account. For example, the angular periods of four-stroke engines with three and four cylinders are $\Phi = 3\pi$ and $\Phi = 4\pi$, respectively. The constant C_1 is then obtained from the condition that $\psi(\varphi, C_1)$ must have the same periodicity, thus $\psi(\varphi + \Phi, C_1) = \psi(\varphi, C_1) + \Phi$, or with $\psi(\varphi = 0) = 0$ without loss of generality

$$\psi(\varphi = \Phi, C_1) = \Phi. \tag{7}$$

This nonlinear equation can be solved for C_1 iteratively, using numerical methods.

In general, the transfer function (6) cannot be exactly realized by means of a mechanism with given topology like that presented in section 2. Hence, an optimal approximation of the system parameter set $\boldsymbol{p}$ of the mechanism has to be found.

4 Multibody simulation

To evaluate the potential of the system, a multibody model of the double-crank mechanism shown in Fig. 2a is build up. The model can be used to calculate an appropriate parameter set $\boldsymbol{p}$ as well. For this purpose, the quadratic deviations of the compensation torque $M(\varphi_k, \boldsymbol{p})$

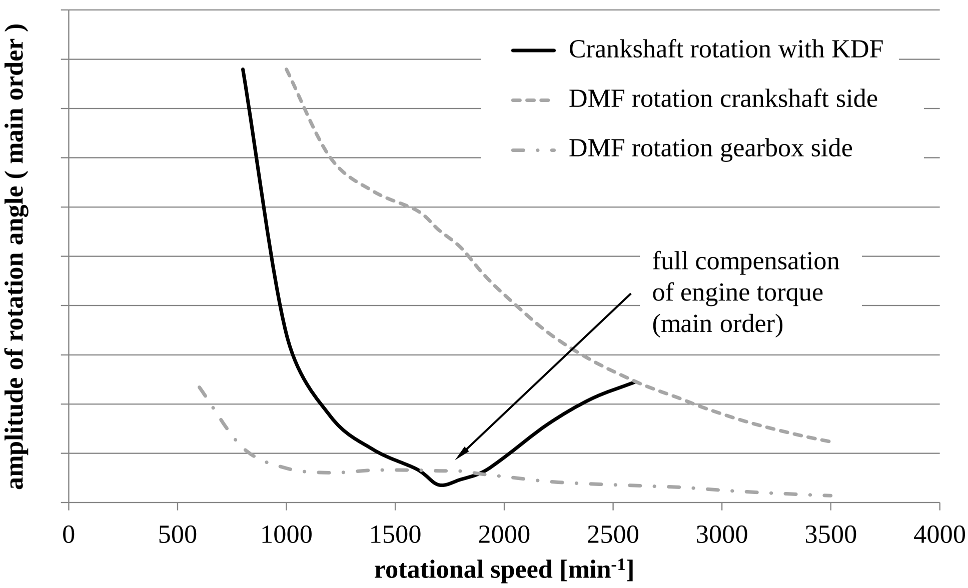


Figure 6: Comparison of vibration amplitudes between Dual Mass Flywheel (DMF) and kinematically driven flywheel (KDF)

calculated by means of the multibody model and the desired compensation torque $M^{\mathrm{des}}(\varphi_k)$ needs to be minimized. Considering N numerical values φ_k of the crankshaft angle over the angular period Φ, the objective function is

$$Z(\boldsymbol{p}) \equiv \sum_{k=1}^{N} \Big[M(\varphi_k, \boldsymbol{p}) - M^{\mathrm{des}}(\varphi_k) \Big]^2 = \min_{\boldsymbol{p}} . \tag{8}$$

The capability of the proposed mechanism is tested using a model of a modern combustion engine combined with the multibody model of the mechanism with a set of precalculated system parameters. The system is analyzed under full load conditions. For comparison, the same engine model is simulated with the standard DMF of the engine.

Figure 6 compares the main order vibration of the crankshaft using the proposed mechanism with the main order vibration at the crankshaft and at the input shaft of the gearbox using the DMF. Looking at the crankshaft oscillations of the DMF model, it is clear that they are considerably higher than the oscillations on the gearbox side of the DMF. The crankshaft oscillation using the kinematically driven flywheel is high at very low rotational speeds of the crankshaft, where the compensation system is not yet much effective. With increasing rotational speed, the oscillation amplitude decreases until it reaches its minimum at 1700 rpm. At this speed, the mechanism compensates the main order engine torque completely, resulting in no rotational vibration of the main order. The following increase in rotational oscillation at higher rotational speeds results from an overcompensation. Hence, by reducing the eccentricity and, therefore, the amplitude of the compensation torque, the main order crankshaft vibration can be vanished in this example for all speeds higher than 1700 rpm. The minimal speed for which full compensation is possible can be lowered by increasing the inertia of the flywheel or by increasing the eccentricity, respectively, whereby the later raises the relative acceleration of the flywheel. When half load conditions are applied, the minimal speed with complete compensation decreases to about 1000 rpm.

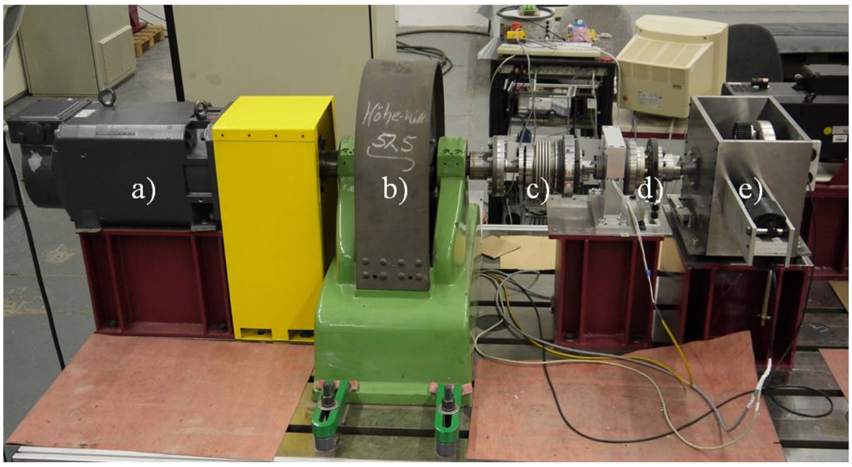


Figure 7: Test rig: a) driving motor, b) big flywheel, c) safety clutch, d) torque sensor, e) proposed mechanism in housing

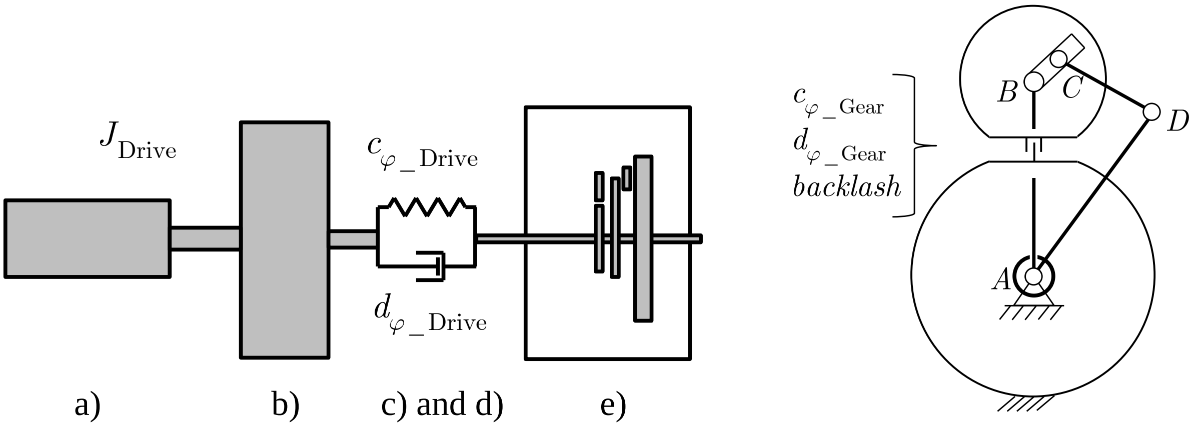


Figure 8: Simulation model of the test rig with elastic drive train and elastic tooth engagement with backlash

5 Experimental setup

To verify the results from the multibody simulation, a test rig was built up and the simulation model is adapted to represent the test rig. The mechanism is tuned to compensate the 1.5th order as main compensation order. The test rig shown in Fig. 7 contains a test stand flywheel with a high inertia of 10.2 kgm^2, that is driven by an asynchronous motor. The proposed mechanism is coupled to the stand flywheel using a safety clutch and a torque sensor. The motor runs with a given constant speed and drives the test stand flywheel and the mechanism. Depending on the eccentricity of the mechanism, a fluctuating torque is generated that acts on the test stand flywheel, resulting in a small oscillation of the rotational speed. To avoid damage to the speed controlled motor and its converter, a low-pass filter is used in the angular speed feedback loop.

The torque between mechanism and test stand flywheel, the force at the connecting rod, the rotation angle of the planet carrier, and the rotational acceleration of the test stand flywheel are measured. The setup allows an almost constant rotational speed of the planet carrier, which is the ideal operating state in a combustion engine, meaning the generated torque compensates the fluctuating torque of the engine completely.

The adapted simulation model of the test rig is shown in figure 8. The inertia of the driving motor and the inertia of the test stand flywheel are reduced to J_{Drive}. Damping and elasticity

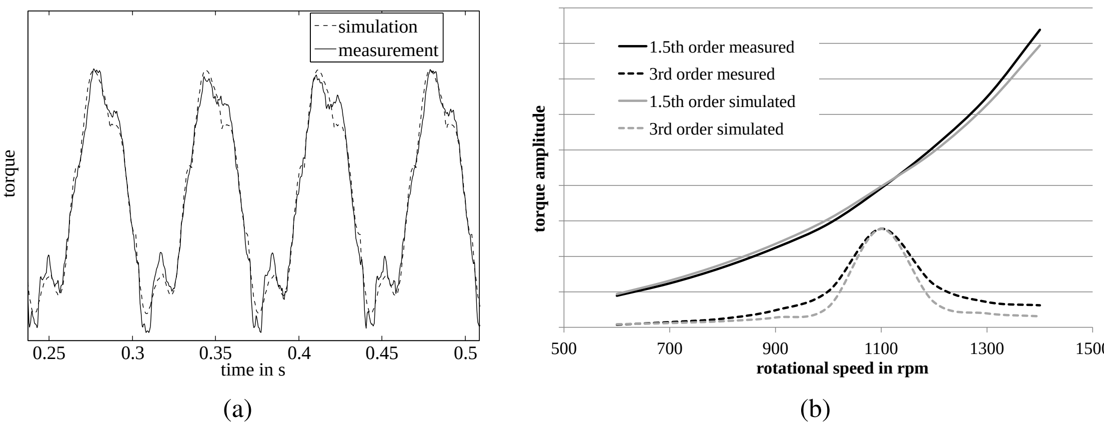


Figure 9: Comparison between simulation and measurement
(a) torque over time, (b) 1.5th and 3rd torque order over rotational speed

are taken into account between the test stand flywheel and the mechanism representing the overall stiffness of the clutch and torque sensor. For the mechanism with all its joints and gears, backlash has been modeled as well. This is done with an elastic tooth engagement with play. System parameters have been optimized within physically realistic boundaries to achieve correlation between simulated and measured results. For that purpose, the parameter estimation tool in SimMechanics has been used.

6 Model validation

A comparison between measured torques and simulated torques over time at 600 rpm and 4 mm eccentricity is shown in Fig. 9(a). It can be seen that the simulation data correlate to the measurement in good agreement. To validate the simulation data for different speeds, the order tracking of the 1.5th and third orders from simulation and measurement are compared in Fig. 9(b). Again, a good correlation can be found. Even the first eigenfrequency of the test rig that gets excited by the third torque order at 1100 rpm matches between simulation and measurement. Further correlations between measurement and simulation have been made but are not be presented here.

7 Conclusion and outlook

A non-uniformly transmitting mechanism is proposed that couples a flywheel kinematically to the crankshaft of a combustion engine in order to compensate fluctuating engine torques. An advantage compared to other countermeasures against driveline vibrations like a dual mass flywheel is that the oscillations are compensated directly at the crankshaft where they arise. The system can be designed to generate compensation torques of an arbitrary main order. The amplitude and optionally the phase angle of the compensation torque are to be adapted to the actual angular speed and load of the combustion engine by external actuators. Appropriate settings for different engine conditions can be found by numerical optimization. They can be stored in lookup tables within the motor controller. The mechanism has been built and tested on an electrically driven test stand. The results showed good correlation between measurements and simulations. Thus, the principle of the mechanism works well, although the mechanical design has to be improved with respect to fatigue strength.

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