THESIS FOR THE DEGREE OF DOCTOR OF PHILOSOPHY IN
MACHINE AND VEHICLE SYSTEMS

Torsional vibration absorbers in heavy-duty truck powertrains

Simulation and analysis of multiple-mass flywheel concepts

LINA WRAMNER

Department of Mechanics and Maritime Sciences

Division of Dynamics

CHALMERS UNIVERSITY OF TECHNOLOGY

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Department of Mechanics and Maritime Sciences
Division of Dynamics
Chalmers University of Technology
SE-412 96 Göteborg
Sweden
Telephone: +46 (0)31-772 1000

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A picture of a heavy-duty truck powertrain, source: AB Volvo

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ABSTRACT

The heavy-duty vehicle manufacturers face large challenges when it comes to reducing CO_2 emissions from vehicles. The ongoing development of more efficient combustion engines leads to an increase in torsional vibrations. Experience within the industry indicates that the conventional single mass flywheel (SMF) and clutch will not be enough to protect the gearbox and rear driveline from engine induced vibrations in the future; more advanced technology will be needed.

The work presented in this thesis focuses on simulation and analysis of torsional vibration absorbers for heavy-duty truck applications. Different multiple-mass flywheels are analysed, including dual mass flywheels (DMFs), power split vibration absorbers (PSVAs) and DMFs combined with tuned vibration absorbers (TVAs). DMFs have been used in smaller vehicles for many years, but the use in heavy-duty commercial applications is to date very limited. The other two vibration absorbers studied in this work have not yet been industrialised.

The vibrations absorbers are analysed by means of simulations. Methodologies for efficient simulations in time- and frequency-domain have been developed and are presented in the thesis. The frequency-domain methods used include the harmonic response and a harmonic balance method, combined with an arc-length continuation scheme. For models with many gap-activated springs, a time-domain approach is proposed, where the dynamics problem is reformulated as a linear complementary problem (LCP). A detailed DMF model, including internal parts, friction and clearances, is presented for time-domain studies requiring high accuracy. The model is correlated based on test rig measurements.

The torsional vibrations in typical heavy-duty truck powertrains with the different multiple-mass flywheels are simulated in a large engine load and speed range. The results are analysed and compared to corresponding conventional powertrains. It is evaluated how different design parameters affect the torsional vibrations and the feasibility of the concepts for heavy-duty use is studied. The simulations show that the torsional vibration amplitudes are generally significantly lower with a DMF than with an SMF, but under some conditions significant resonance excitation can occur. The PSVA and DMF equipped with a TVA can reduce vibrations further than a corresponding DMF within limited speed ranges, but lead to higher vibration amplitudes outside these ranges.

Keywords: DMF, dual mass flywheel, torsional vibrations, heavy-duty truck, resonances, friction, simulations, powertrain, driveline, PSVA, anti-resonance, power split vibration absorber, tuned vibration absorber, TVA

To my beloved family

PREFACE

The work in this thesis has been carried out from May 2016 to April 2020 at the Department of Mechanics and Maritime Sciences, Division of Dynamics, at Chalmers University of Technology.

The research is part of the project ”Reduced vibration transmissions - reduced energy consumption and environmental impacts together with an increased competitiveness”. It is a cooperation project between AB Volvo, Scania CV AB, Chalmers University of Technology and KTH Royal Institute of Technology. The project is funded by the Swedish Energy Agency (project No. 42100-1), AB Volvo and Scania CV AB. The project includes four work packages with focus on clutches, dual mass flywheels, centrifugal pendulum absorbers and power split absorbers, respectively.

The thesis presents the work within work package 2, Dual mass flywheels and work package 4, Power split absorbers. The theoretical work has been performed by me, in cooperation with my supervisors Professor Viktor Berbyuk, Professor Håkan Johansson (both at Chalmers University of Technology) and Dr. Anders Hedman (AB Volvo). Viktor Berbyuk is the main supervisor and examiner. The experimental part has been performed as a master thesis work by Johan Karlsson under my supervision and with support from Johan Jonsson (AB Volvo), Anders Hedman and other personnel at AB Volvo.

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I would like to express my sincere gratitude to Anders Hedman, without whom I would not have been involved in this project. Anders's continuous support, encouragement and always fast and extensive feedback are highly appreciated.

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Moreover, I wish to acknowledge the support from Håkan Johansson. I am particularly grateful for Håkan's meticulous examinations of formulae and notations. Håkan is also appreciated for his sincerity.

At last, I also want to acknowledge the contributions by Johan Karlsson and Johan Jonsson, regarding the experimental results. Their well performed work is greatly appreciated and valuable for the project.

Göteborg, April 2020

Lina Wramner

THESIS

This thesis consists of an extended summary and the following appended papers:

Paper A	L. Wramner. Torsional vibrations in heavy-duty truck powertrains with dual mass flywheels. Accepted for publication in *International journal of heavy vehicle systems*, 2019.
Paper B	L. Wramner. Sub-harmonic resonance excitation in heavy-duty truck powertrains with piecewise linear dual mass flywheels. To be submitted.
Paper C	L. Wramner. Analysis of power split vibration absorber performance in heavy-duty truck powertrains. Accepted for publication in *Proceedings of the institution of mechanical engineers, part D: Journal of automobile engineering*, 2020.
Paper D	L. Wramner. Dual mass flywheels with tuned vibration absorbers for application in heavy-duty truck powertrains. Accepted for publication in *Proceedings of the institution of mechanical engineers, part D: Journal of automobile engineering*, 2020.
Paper E	L. Wramner. Numerical algorithms for simulation of one-dimensional mechanical systems with clearance-type non-linearities. *ASME Journal of computational and nonlinear dynamics*, 2019, **14**(6), 061008-1/15.
Paper F	L. Wramner, V. Berbyuk, H. Johansson. Vibration dynamics in non-linear dual mass flywheels for heavy-duty trucks. *Proceedings of International Conference on Noise and Vibration Engineering, (ISMA 2018)*, Leuven, Belgium, (2018), 1935-1947.

The appended *Paper F* was prepared in collaboration with the co-authors. The author of this thesis was responsible for the major progress of the work including taking part in the planning of the paper, developing the theories and the numerical implementation, performing the numerical simulations and writing the paper.

CONTENTS

Part I
Extended summary

1 Introduction

1.1 Motivation

Global warming is today considered by most scientists to be a serious threat to our environment. In order to prevent a significant global temperature rise, it is necessary to significantly reduce the emissions of greenhouse gases.

Many of the countries in the world have committed to contribute to the reduction in the emissions according to the Kyoto protocol and the Paris agreement. Within the European Union (EU) there is a target to cut greenhouse gas emissions by at least 40% below 1990 levels by 2030 [10]. Some individual countries have more ambitious targets.

Trucks, buses and coaches produce around a quarter of all CO_2 emissions from road transport in the EU and around 6% of the total CO_2 emissions in the EU [11]. To significantly decrease the CO_2 emissions from the heavy-duty transport sector involves large challenges. Some reduction can be obtained by improved logistics, optimised vehicle use and when possible a shift to more ship and rail transportation. If not followed by an extensive reduction in the amount of transportation, these measures will most certainly not be enough to reach the targets.

Significant improvements in vehicle efficiency will also be needed. On August 2019 the first-ever EU-wide CO_2 emission standards for heavy-duty vehicles were adopted, when regulation (EU) 2019/1242 entered into force [11]. The targets for average CO_2 emissions from new trucks are 15% lower in 2025 than in 2019/2020 and at least 30% (indicative target, subject to review in 2022) lower in 2030 than in 2019/2020.

Electrification and hybridisation are promising measures for reaching the targets of reduced CO_2 emissions for lighter vehicles in city traffic. However, for heavy-duty long-haul applications the efficiency gain from electification and hybridisation is not expected to be as high. Moreover, the required size and weight of the batteries limit the space and weight available for cargo and the capacity requirements on the batteries lead to very high battery costs. It is foreseen that combustion engines (with diesel- or alternative fuels) will be needed for heavy-duty commercial applications in many years to come.

Reduced friction, improved aerodynamics and better engine and gearbox control can contribute to efficiency improvements. It is also necessary to improve the efficiency of the engines. Some ways are to increase the cylinder pressure and to reduce the engine speeds, i.e. down-speeding the engines. If the engine speeds are reduced, a higher torque is required in order to maintain the power. Improved engine efficiency can also be obtained by optimising the engine for a smaller speed range. In that case, the gearbox ratio needs to be adjusted to the vehicle speed more often in order to keep the engine speed within the desired range.

Higher cylinder pressure, higher torque levels and more frequent gear-shifting all lead to an increase in torsional vibrations in the powertrain. Lower engine speeds also result in higher torsional vibrations, due to a resonance in the powertrain that can be excited at low speeds. An increase in torsional vibrations has a negative impact on component life, comfort and emitted noise. Currently, there is also a higher focus on reducing the noise. In 2014, the EU adopted a new regulation with the purpose to reduce traffic noise by around 25%. The new regulation includes a new test method to better reflect current driving behaviour and a decrease of the limit values. For heavy-duty vehicles, the reduction will be 1 dB(A) until 2021 and 2 dB(A) until 2025 [12].

In the future, the vehicle manufacturers need to meet the increasing demand for higher engine efficiency while maintaining or reducing the levels of pollutant emissions and complying with the more strict noise regulations. This is not an easy task. Since engine improvements probably will not go without increased torsional vibrations, good isolation of the engine vibrations will be required. Today, a single mass flywheel (SMF) and clutch are used in most heavy-duty powertrains to protect the gearbox and rear driveline from the engine induced vibrations. In the future, with more efficient engines, it is foreseen that more advanced technology will be needed to limit the torsional vibrations transmitted to the driveline.

Dual mass flywheels (DMFs) have been used in passenger cars for decades and have shown to reduce the torsional vibrations. For heavy-duty applications, DMFs are not common and the performance of DMFs for such applications needs to be further analysed. There are also several other vibration absorbers proposed, where a DMF is combined with additional rotational components to further enhance the vibration reduction capability. More research is needed to understand the potentials of these vibration absorbers.

1.2 Aim of research

The aim of research is to:

1. Develop models and methodology for efficient simulations and analyses of engine-excited torsional vibrations in heavy-duty truck powertrains with various multiple-mass torsional vibration absorbers.

2. Understand the function, potentials and design requirements of multiple-mass torsional vibration absorbers in heavy-duty truck applications with special focus on dual mass flywheels and power split vibration absorbers.

3. Investigate the possibilities to enable significant down-speeding, using multiple-mass torsional vibration absorbers and evaluate the consequences from a driveline torsional vibrations perspective at different engine speeds and operating load levels.

1.3 Scope and limitations

The work deals with engine excited torsional vibrations in the interface between the engine and the gearbox. Vibration frequencies in the range of about 30 Hz - 100 Hz are considered. These frequencies correspond to the main order engine excitation from idle speed up to top engine speed in a normal operating range. The focus is on general conclusions regarding typical heavy-duty truck powertrains. Optimisations concerning specific powertrains are not within the scope of this work. Neither are analyses of strength and life of the different absorbers included.

2 Background

2.1 Heavy-duty truck powertrains

A conventional heavy-duty truck powertrain is typically composed of a 4-stroke, 6-cylinder combustion engine, a single mass flywheel (SMF), a clutch, a gearbox, a propeller shaft, a rear axle and wheels as illustrated in figure 2.1.

Figure 2.1: Truck powertrain, source: AB Volvo (left) and schematic picture of powertrain (right)

Figure 2.2, shows typical maximum mean torque and power curves for a 13-litre engine. Combustion engines generate a highly non-uniform torque. In 4-stroke, 6-cylinder engines, there are three cylinders firing each crankshaft revolution. This results in an oscillating torque from the engine, with three peaks each revolution and hence a frequency that depends on the engine speed. The frequency is referred to as the third order frequency. Figure 2.3 shows an example of how the torques at different powertrain positions vary during two crankshaft revolutions.

The SMF and clutch are designed to reduce the oscillating part of the torque transmitted from the engine to the rest of the powertrain. The SMF is basically a rigid wheel that provides isolation thanks to a large moment of inertia. The clutch also contributes with moment of inertia. In addition, there is a torsional damper in the clutch which provides a low stiffness torsional coupling between the clutch disc and the gearbox input shaft. This combined leads to that only a small part of the torque oscillation from the engine is transmitted to the gearbox. The resulting oscillation

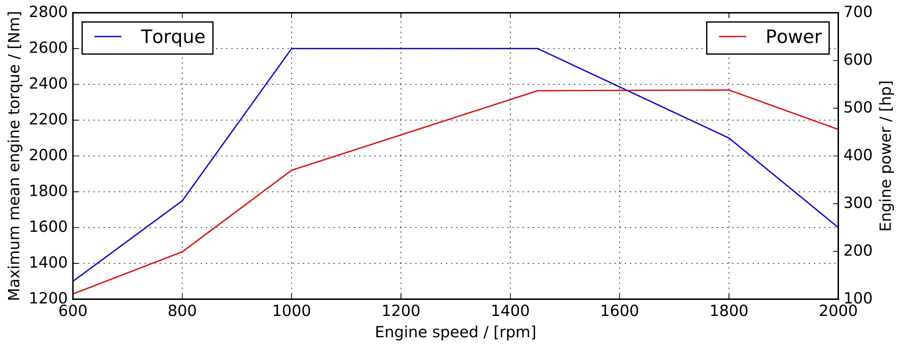


Figure 2.2: Typical heavy-duty engine torque and power (Volvo D13, 540 hp)

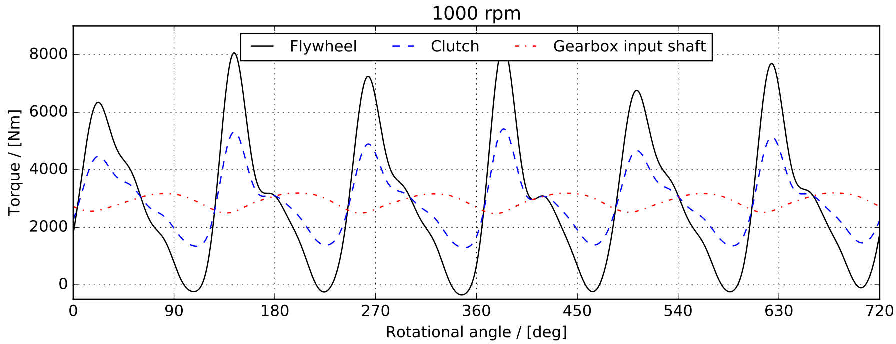


Figure 2.3: Torque at different powertrain positions

is dominated by the harmonic component corresponding to third order frequency. Torsional driveline resonances with frequencies corresponding to third order frequency in the operating range can hence be problematic. Resonances with frequencies corresponding to other multiples of the engine speed, i.e. other engine orders, are generally not a problem. An exception to this is the first powertrain resonance mode, which is associated with driveability and comfort and can be excited when accelerating or decelerating the vehicle or by impulses from towed trailers and road roughness [33, 17].

In typical heavy-duty truck powertrains, with SMFs, there is a torsional resonance with frequency close to the third order frequency at low engine speeds. The resonance frequency depends on the gear selected and on the driveline configuration. Clutch damper design is very much focused on limiting the effect of this resonance. A decrease in clutch damper stiffness reduces the resonance frequency. Design limits on the clutch make it impossible in most cases to reduce the resonance frequency so much that the mode cannot be excited by the third order engine load in the operating speed range for all gears. Therefore the clutch damper also needs to provide significant friction damping in order to limit vibration amplitudes near resonance. The friction damping results in power losses and also leads to increased torsional vibrations at speeds far from resonance. If the operating speed range of the engine is decreased, the resonance is expected to be more problematic.

2.2 Multiple-mass flywheel concepts

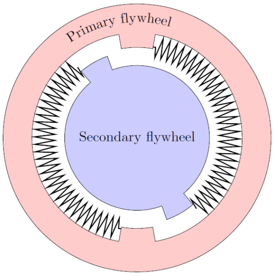
Up to the 1980s, an SMF combined with a clutch damper were sufficient for reducing the torsional vibrations transmitted from the combustion engines to the gearboxes in vehicle applications. In the 1980s, though, gearbox rattle and boom caused by torsional vibrations started to become a major problem in many passenger car applications. As the SMF and clutch damper used at that time were near their physical limits, the development of alternative solutions took off [34]. The introduction of dual mass flywheels (DMFs) was a large break-through that led to a major reduction in torsional vibrations. In the beginning of the 21st century even further reduction could be obtained when the DMFs started to be combined with centrifugal pendulum vibration absorbers (CPVAs) [14, 32]. Currently, the potential for further reduction of vibrations with these components are small in some applications [32, 26] and research interest has turned to new concepts [31, 25]. The power split vibration absorber (PSVA), DMF with tuned vibration absorber (TVA) and the planetary DMF are three concepts that create anti-resonances in a powertrain. In this chapter, these different multi-mass vibration absorbers are briefly described.

2.2.1 Dual mass flywheels

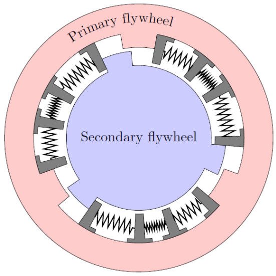
Dual mass flywheels, as a means to reduce torsional vibrations in powertrains, started to emerge on the automotive market in the 1980s. Initially, DMFs were mainly used in top range passenger cars with powerful engines and high requirements on comfort. Today, almost half of all vehicles produced with manual gearshift transmissions are equipped with a DMF [34]. In commercial heavy-duty vehicles, conventional SMFs are still standard and the use of DMFs is mainly restricted to buses and coaches.

A DMF is composed of two flywheels, torsionally connected by springs. By exchanging an SMF for a DMF, the powertrain resonance frequencies are shifted. It is generally possible to design the DMF so that the DMF resonance speed is well below the idle speed of the engine [22]. Thereby, the springs will act as a filter for the high amplitude vibrations from the engine. When there is a peak in the load, the springs will temporarily store energy and this energy is then released when the load is lower. As a result, the torque amplitudes, transmitted to the gearbox, are low.

In figure 2.4, the two most common types of DMF design are illustrated schematically. With the arc spring design in 2.4a, curved springs are located in a toroid-shaped channel in the primary flywheel. When there is relative angular motion between the two flywheels, the springs are compressed and slide against a guide in the channel. With the straight spring design in 2.4b, the two flywheels are instead connected by several straight springs in series, with sliding shoes in between. This type of DMF exists for heavy-duty applications with mean engine torques up to 3500 Nm. The spring channels are filled with grease, which provides lubrication and damping.



(a) Arc spring design



(b) Straight spring design

Figure 2.4: DMF types

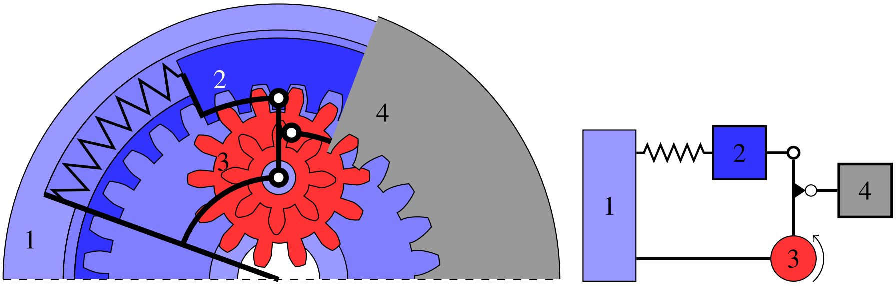
There are many variants of the above two DMF types. The number of springs and stiffness of the springs vary, and the springs generally need to be optimised for different applications. There can be many springs with different stiffness and lengths inside each other. Low stiffness is generally good from a vibrational perspective, but a lower stiffness also reduces the maximum torque that can be transmitted. The DMF is often designed with two or more stiffness steps. At low mean torques, the stiffness is low, leading to low vibration amplitudes. At high mean torques, the stiffness is high, which enables the DMF to transfer high torques without bottoming.

DMFs for heavy-duty truck applications are studied in *Paper A*, *Paper B* and *Paper F*.

2.2.2 Power split vibration absorber

The power split vibration absorber (PSVA) is an evolution of a DMF, where the primary flywheel is connected to a secondary body both by means of springs and planetary gears. This is shown schematically for one planetary gear wheel in figure 2.5a. The primary flywheel is denoted by 1, the secondary body with 2 and the planetary gear wheel by 3. The planetary gearwheels are also connected with a different gear ratio to an output flywheel, denoted by 4. The connections between the different parts are drawn with black solid lines. Figure 2.5b illustrates a corresponding simple torsional lumped mass model. The oscillating torque from the engine, acting on the primary flywheel, is thus divided into two paths. The part of the oscillating torque from the engine that is transmitted through the springs is shifted in phase with approximately 180°. This is due to the engine operating over-critically and is similar to a conventional DMF. The part of the oscillating torque that goes through the planetary gears will, on the contrary, be transmitted with the phase of the oscillation unchanged. Due to the difference in phase, the torque oscillations can cancel when the paths are merged. The torque amplitudes transmitted to the output flywheel will thus be low. The extent to which the oscillation amplitudes are cancelled depends on the oscillation frequency. The PSVA can provide very good vibration isolation near one specific engine speed.

The PSVA has not yet been industrialised, but prototypes have been built and tested [16], verifying the desired functionality. The PSVA is further described and analysed in *Paper C*, with focus on heavy-duty applications.



(a) Schematic figure (b) Simple torsional lumped mass model

Figure 2.5: PSVA

2.2.3 Dual mass flywheel with a tuned vibration absorber

Using a tuned vibration absorber (TVA) is a well-known way to reduce vibrations near a specific tuning frequency, but it also introduces an additional resonance into the system. The frequency of this new resonance is lower than the tuning frequency if the TVA is installed on an intermediate flange between two DMF flywheels. Figure 2.6 illustrates a torsional lumped mass model of the DMF with a TVA on an intermediate flange, m. This design can enable a reduction of vibrations at low engine speeds, without introducing an additional resonance in the normal operating engine speed range. The DMF with TVA, combined with a CPVA has been proposed as a potential option to reduce vibrations further than what would be possible with a conventional DMF and CPVA for some awkward design spaces [24]. The DMF with TVA for heavy-duty applications is further

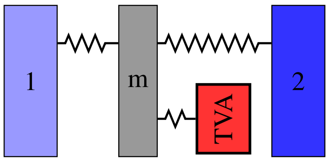


Figure 2.6: Simple torsional lumped mass model of a DMF with TVA

studied in *Paper D*.

2.2.4 Centrifugal pendulum vibration absorber

In the beginning of the 21st century, the DMFs started to be combined with centrifugal pendulum vibration absorbers (CPVAs) [14, 32] in demanding passenger car applications. This led to additional reduction of vibrations. The CPVAs are usually installed on the secondary flywheel of a DMF. A CPVA is a special type of tuned vibration absorber with a tuning frequency that depends on the rotational speed. This makes them suitable for conventional combustion engines, since they can be tuned to reduce oscillations of the main engine order.

The CPVA is not included in the work presented in this thesis, as it forms a separate work package of the overall research project. The associated work is part of ongoing research work at KTH Royal Institute of Technology and Scania CV AB [20, 19]. It is possible to combine any of the multiple-mass flywheel concepts studied in this work with a CPVA. This would provide additional reduction of the vibrations transmitted to the gearbox, but it would also require a larger axial installation space.

2.2.5 Planetary dual mass flywheel

A planetary DMF is a DMF with the two flywheels connected both by springs and planetary gears. It can be considered mathematically as a special case of PSVA with the secondary body (2) and output flywheel (4) merged and with input and output direction reversed. Figure 2.7 shows a simple torsional model of a planetary DMF. The inertia effect of the planetary gear wheels on the secondary flywheel can oppose the torque from the springs, resulting in an anti-resonance at a specific oscillating frequency. This can lead to good isolation of torsional vibrations near a specific engine speed, referred to as the cancellation speed, similar to a PSVA.

A further development of the planetary DMF, that uses a switchable mass moment of inertia of the planetary gear wheels, has also been proposed [31]. The mass moment of inertia can take two different values. As a result there are two different cancellation speeds. Thereby, the speed range where the planetary DMF is efficient can be increased. A prototype based on this concept has been tested and proven successful through rig trials [31].

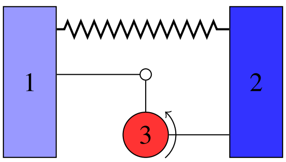


Figure 2.7: Simple torsional lumped mass model of a planetary DMF

The planetary DMF has not been studied in detail in this work, but some conclusions can be drawn from the study of the PSVA for heavy-duty applications in *Paper C*. A major challenge in designing the PSVA is to have the cancellation speed, where the highest efficiency is obtained, at the critical low engine speeds. The quotient of the two gear ratios in the PSVA has a large impact on the cancellation speed. A lower gear ratio quotient results in a lower cancellation speed. The planetary DMF can be seen as a PSVA with gear ratio quotient equal to 1. It can therefore be expected that the difficulties of attaining a low cancellation speed will be more pronounced with the planetary DMF than with the PSVA. To obtain a low cancellation speed in heavy-duty applications, the moment of inertia of the planetary gear wheels would have to be very large.

3 State of the art

Previous work presented in literature related to DMFs comprise experimental and theoretical analyses of torsional properties, studies of modelling and simulation approaches as well as evaluations of the functionality within vehicle powertrains for various load scenarios. In this chapter, the state of the art related to different aspects of a DMF is presented. The previous work, found in literature, related to the PSVA and the DMF with TVA [16, 28, 25, 24], mainly presents descriptions of the basic designs and the expected conceptual functionalities.

3.1 Torsional properties of DMFs

The torsional stiffness and damping of a DMF are non-linear. They depend on the operating conditions, such as rotational speed, mean torque and torque amplitudes. The temperature of the grease in the spring channels also affects the damping [45]. In DMFs with the arc spring design, relative motion between the two flywheels results in friction forces between the guide in the primary flywheel and the springs. The normal forces at the friction surfaces depend on the compression of the springs and on the centrifugal action. With high vibration angles, high mean torques and high rotational speed, the friction is higher [22, 36, 34, 1]. Moreover, at higher speeds and lower torque amplitudes, the springs are compressed inhomogeneously, leading to a shorter effective length of the springs and hence a higher effective stiffness [36, 1]. At very high speeds, the centrifugal action can result in so high friction forces that the springs stick to the guide in the primary flywheel. There will hence be a dynamic lash [36]. The non-linear characteristics of DMFs result in high damping and low stiffness at start-up when the resonance is passed. At normal driving, with low relative vibration angles, there are generally low frictional losses [1, 22]. These non-linear characteristics of arc spring DMFs have been studied and experimentally verified by many authors [2, 36, 6, 7, 18].

In DMFs with straight spring design, there is friction between the sliding shoes and the primary flywheel. A similar dynamic behaviour as with the arc spring DMFs can be expected with straight springs, although the difference in material and shape of the contact surfaces will lead to different levels and proportions of various types of friction. Measurements showing the relationship between the torque and relative displacement angle in a straight spring DMF, have been performed for the static case as well as for rotational speeds of 2000 rpm and 4000 rpm [43]. The torque was measured for angular deflections ranging between about $-60°$ to $60°$, corresponding to the minimum and maximum relative deflection angles. The angular deflection speed was 3 °/s. The results show a significantly higher torque hysteresis at higher rotational speeds. This can be expected since the centrifugal action leads to higher normal contact forces between the sliding shoes and the primary flywheel. No other experimental results showing the dynamic behaviour of straight spring DMFs at different rotational speeds and load levels have been found in literature.

3.2 DMF simulations

The computational models used for simulations of torsional vibrations of DMFs can be classified into different types, dependent on how the springs connecting the two flywheels are modelled.

The first type does not explicitly include the deformation within the springs. The torque from the springs acting on the two flywheels is instead represented by a function of the relative displacement and velocity between the two flywheels and in some cases also the rotational speed. The parameters for the torque functions are typically determined from a fit to experimental data [45, 7, 18]. With this first type of DMF model, high computational efficiency can usually be obtained. If the torque function is not directly derived from physical properties of the DMF, it is difficult to translate the models to DMFs of different design, though. The validity of the models for load cases outside the range for which they are correlated can also be doubted.

The second type of model is characterised by the arc springs being represented by several lumped torsional masses connected by linear mass-less spring elements. This allows for an uneven compression of the springs. Thereby, the model can capture the dynamic stiffness and damping at different operating conditions in a more realistic way. It has been concluded that it is sufficient to model the arc springs with six spring sections [2]. This is also what is used by many authors [36, 30]. The models include friction between the lumped spring masses and the primary flywheel. Some different friction models are used in literature, such as the Karnopp friction model [36] and the LuGre friciton model [23, 30]. The friction modelling is more complicated for arc spring DMFs with a bi-linear stiffness. These DMFs have inner and outer arc springs of different lengths. Between these springs there is friction. Due to the different lengths, the contact points between the springs change as the springs are compressed. A comparison of two different ways to model such inner and outer arc springs shows that the vibration amplitudes near resonance at higher engine rotational speeds are lower if friction between the inner and outer arc springs is included in the model [15].

A third type, which can be seen as a compromise between the above two, has also been presented [29]. It has a high computational efficiency, suitable for simulations on a car's control unit. The model includes one lumped mass representing the arc spring, and uses a quasi-steady-state approximation to determine the position of this lumped mass. The model is validated with test data and is claimed to be capable of describing all dominating physical effects of a DMF.

Most of the simulations described in literature concern DMFs with arc springs. Two articles describing dynamic simulations of straight spring DMFs have been found [38, 43]. In the models used, the sliding shoes are represented by lumped masses. In one of the models, Coloumb friction is included between the sliding shoes and the primary flywheel [38]. The model data are not published. The other model incorporates a Stribeck friction model with parameters correlated to test data [43]. Straight spring DMFs can be modelled similar to arc spring DMFs. Still, there is a need to build experience regarding realistic parameter values and to investigate the validity of the models for operating points valid for heavy-duty applications.

3.3 DMFs in passenger cars

The function of DMFs in passenger car powertrains has been studied and described by many authors [1, 22, 35]. At normal operating conditions, the use of a DMF can significantly reduce the torsional vibrations into the gearbox, compared to an SMF. Thereby, it can alleviate noise issues such as gear rattle and boom [22]. Moreover, due to a smaller mass and moment of inertia being rigidly connected to the crankshaft, bending and torsion of the crankshaft are reduced. There are some operating conditions though, at which a DMF can result in higher vibration amplitudes.

At start-up of the engine, the DMF torsional resonance has to be passed. This can lead to high torsional vibration amplitudes. The starting behaviour is improved if the crankshaft is accelerated fast so that the resonance is passed quickly. High acceleration is accomplished with a strong starter motor. A low DMF resonance frequency and high engine starting speed also helps to improve the situation, as does damping in the DMF [35, 22]. Engine stop does not usually present a problem regarding arc spring and DMF durability but can result in a clatter noise as the DMF resonance is passed just before stand-still [22].

Sudden load changes, such as after clutch engagements, can also be problematic in DMF powertrains [35]. The sudden load can lead to the springs being fully compressed with high peak torques as a result. Countermeasures are longer engagement times, higher DMF bottoming torques and torque limitations. Another consequence of using a DMF is that the crankshaft irregularity increases, i.e. the solid body rotation of the crankshaft [35]. This will lead to higher vibrations in accessory drives.

To prevent acoustic problems at idle, the DMF springs are designed with a lash. This lash should be larger than the engine torsional vibration amplitudes at idle. Thereby, the DMF springs are not activated at idle and the secondary flywheel is to a large extent isolated from the torsional vibrations of the engine. A disturbance in the torque, though, can in some cases lead to amplitudes larger than the lash and hence compression of the springs. As a consequence, sub-harmonic vibrations with oscillating frequencies lower than that of the main engine order can occur, resulting in vibrations and gearbox rattle. Moreover, the engine speed control can be affected since the sub-harmonic vibrations negatively affect the estimate of the engine speed in the engine control unit (ECU). This can result in a faulty control strategy [22, 40].

3.4 DMFs in heavy-duty trucks

A thorough literature search regarding DMFs in heavy-duty powertrains yielded very few results. In one article, it is argued that a DMF is the ideal solution for truck powertrains of which the vibrations cannot be reduced sufficiently by conventional forms of torsional damper [39]. The statement is supported by simulated torsional acceleration levels at two positions in powertrains with an SMF and a DMF. No details about model data are included. In another study, global sensitivity analysis and Pareto optimization are used to show the potential for torsional vibration reduction in heavy-duty trucks with a DMF [3]. Furthermore, the development process for a DMF for heavy-duty applications is described in one paper [9]. It is concluded that the use of a DMF can significantly reduce the torsional vibrations in heavy-duty truck powertrains. Moreover, it is emphasised that the know-how from passenger car applications can only be adopted to heavy-duty applications to a limited degree. In particular, the operating conditions and loads are different, and this must be considered when developing robust and reliable products.

4 Methodology

In this chapter, the powertrain modelling and the simulation methods used in this thesis are described, with particular focus on assumptions and simplifications used.

4.1 Powertrain models

The simulations presented in this work are based on lumped mass models of a typical heavy-duty truck powertrain. Torsional motion, only, is considered. This simplification is common in literature for analyses of powertrain torsional dynamics in smaller vehicles and has shown to give good correlation with measurements [21, 8, 44, 37, 27]. In literature, it is common to model the rotational parts of the engine as one lumped mass [21, 8, 42, 41, 13]. For standard flywheels with large moment of inertia, this is in most cases a good approximation that simplifies implementation and reduces simulation time, without a significant impact on the results. Early in this project, while evaluating different modelling approaches, it was observed that near resonances, the engine model can significantly influence the results rear of the flywheel. Since resonances are studied both in *Paper A* and *Paper B*, an engine model which includes the flexibility of the crankshaft is used. Each crank pin is represented by a lumped mass. The torques from cylinder pressures and inertia forces from the piston and connecting rods are applied at theses masses. In figure 4.1, a schematic figure of the powertrain model used is shown. Various different models of the vibration absorbers are used.

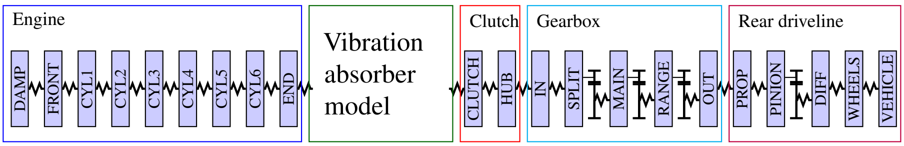


Figure 4.1: Lumped mass torsional model of a powertrain

The model data for the engine, clutch, gearbox and rear driveline have been derived from independent, internal, confidential models of heavy-duty truck powertrains of the same size from two manufacturers. The data are considered to be representative for a typical heavy-duty truck. The lowest resonance modes and frequencies for the frequency range in focus, are for most heavy-duty truck powertrains very similar. Similar resonance excitation occurs in different trucks and with different gears, but at somewhat different engine speeds. It is therefore expected that the general conclusions drawn from simulations with these models are applicable to most heavy-duty trucks.

The clutch and the DMF usually largely affect the torsional resonance modes involved in the engine excited vibrations studied in this work. The damping of these components therefore has a large impact on the vibration amplitudes near resonances. An accurate modelling of the friction is hence necessary to well predict the vibration amplitudes. Moreover, the effective stiffness of the DMF can vary significantly with engine speed and load [36]. This will affect the resonance frequencies. Much work has therefore been put into the modelling of the DMF. Non-linear models are used for the DMF and clutch in the powertrain simulations presented in *Paper A*. The DMF spring packages are modelled in large detail, including a non-linear friction model between sliding shoes and the primary flywheel and clearances between sliding shoes and springs. The DMF model is illustrated in figure 4.2 and is described in *Paper F* and *Paper A*. Test rig measurements

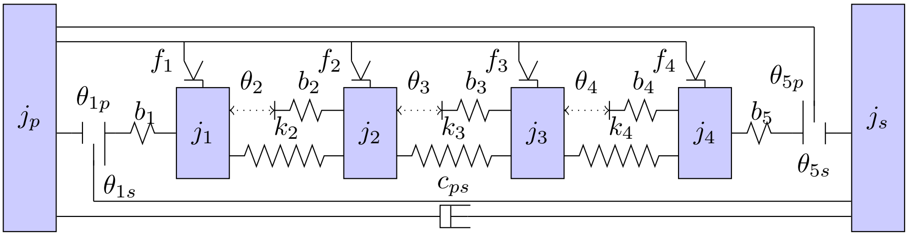


Figure 4.2: Detailed model of a DMF

have also been performed, in order to correlate the model and the friction parameters as described in *Paper F*. The damping in the engine and the driveline components rear of the clutch is usually low or does not have a large impact on the steady-state engine excited torsional vibrations into the gearbox, studied in this work. Therefore, a linear modelling is used for these components.

The purpose with the studies presented in *Paper C* and *Paper D* is to analyse the main functionality of new vibration absorber concepts with focus on how moments of inertia, stiffness and gear ratios affect the performance. For this type of study, linear models are considered suitable, in particular considering that lack of measurement data would anyhow lead to large uncertainties regarding friction parameters.

Sub-harmonic resonance excitation can occur in a DMF with a step in the stiffness. In *Paper B* this is studied, using a piece-wise linear powertrain model with viscous damping between the two flywheels. The purpose is to evaluate the influence of different stiffness values and to obtain a conceptional understanding of the vibrational behaviour expected at different speeds and torque levels near resonance. For this study, the linear damping model is deemed appropriate.

4.2 Simulation methods

Both frequency-domain and time-domain simulation approaches are employed in this work.

The frequency-domain approaches are used to evaluate steady-state vibrations. They are computationally efficient and the results obtained are easy to evaluate. Harmonic response simulations are used for the linear models in *Paper C* and *Paper D*. Harmonic response is also used with linearised models to provide realistic initial conditions for the time-domain simulations described in *Paper A* and *Paper F*. For the piece-wise linear models in *Paper B* the harmonic balance method is used. The harmonic balance method is a frequency-domain approach that can be used to analyse steady-state solutions of non-linear systems. In *Paper B* it is combined with a continuation-scheme used to find nearby solutions at other engine speeds once an initial solution has been found. The process renders a set of steady-state solutions within a range of engine speeds. Both stable and unstable solutions can be obtained with the harmonic balance method and the stability is also assessed in *Paper B*.

Time-domain simulations are considered an adequate approach for the studies in *Paper A* and *Paper F*, due to the complexity of the friction and spring model. With time-domain simulations, time-transient vibrations can be analysed. The evaluation of steady-state is on the other hand more complicated, since it must be verified that steady-state has been reached and a proper time interval must be selected in the evaluation. Some initial difficulties with convergence were encountered in the simulations with the models in *Paper A* and *Paper F*, due to the many stiff impacts. A methodology to facilitate convergence and reduce simulation times in models with many gap-activated linear springs was developed and presented in *Paper E*. An evaluation of the effects of numerical damping and simulation time step on computational accuracy and convergence is also included in *Paper E*.

5 Summary of appended papers

Paper A: Torsional vibrations in heavy-duty truck powertrains with dual mass flywheels. The paper presents torsional models representing typical heavy-duty truck powertrains with an SMF and a DMF. The models are used to evaluate the torsional resonance frequencies and modes. It is described how different powertrain parameters affect the resonances. The impact of replacing an SMF with a DMF is highlighted. Results from simulations of steady-state torsional vibrations in heavy-duty truck powertrains with SMFs and with DMFs in a large speed and load range are shown and evaluated. It is demonstrated how a proper choice of DMF and clutch properties can enable down-speeding and increased engine torque without severe impact on the torsional vibrations into the gearbox.

Paper B: Sub-harmonic resonance excitation in heavy-duty truck powertrains with piece-wise linear dual mass flywheels. DMFs are often designed with a piece-wise linear stiffness. This can result in sub-harmonic resonance excitation. The paper presents harmonic balance simulations of heavy-duty truck powertrains. Sub-harmonic resonance excitation of second order of the torsional resonance mode introduced by the DMF is analysed. It is shown how the excitation varies with engine speed and mean engine torque and the impact of changing the two stiffness values is evaluated.

Paper C: Analysis of power split vibration absorber performance in heavy-duty truck powertrains. The paper presents an evaluation of the concept of power split vibration absorber (PSVA). A mathematical model of the PSVA is proposed and an analytical study shows how different design parameters affect the PSVA performance. Numerical simulations with models representing typical heavy-duty truck powertrains are used in the evaluations. It is concluded that for a low level of damping, the PSVA can provide significantly lower vibration amplitudes than a corresponding DMF within a limited speed range. If the PSVA is optimised for the critical low engine speeds, an overall decrease in the level of vibration can be obtained, but a larger installation space than with a conventional DMF would probably be required.

Paper D: Dual mass flywheels with tuned vibration absorbers for application in heavy-duty truck powertrains. In this work the concept of a DMF combined with a tuned vibration absorber (TVA) is analysed. The TVA efficiently reduces the vibration amplitudes for engine load frequencies near the tuning frequency, but it also introduces an additional resonance into the system. By placing the TVA on an intermediate flange between the two DMF flywheels, the introduced resonance speed will be lower than the tuning speed. Numerical simulations are used to show how the torsional vibration amplitudes in a heavy-duty truck powertrain are affected by the TVA. It is also evaluated how different TVA design parameters affect the vibrations.

Paper E: Numerical algorithms for simulation of one-dimensional mechanical systems with clearance-type non-linearities. Generalisations of an efficient algorithm for simulating systems with gap-activated springs connecting bodies are proposed in this paper. The simulation problem is reformulated as a linear complementary problem (LCP), which can be solved efficiently. The generalisations enable the LCP approach to be used for an arbitrary number of gap-activated springs connecting either different bodies or connecting bodies to ground. The springs can be activated in either compression or expansion or both and a gear ratio can be included between the bodies. The efficiency of the algorithm is verified with an application example representing a DMF. It is also concluded that if numerical damping is used, the time step can be increased significantly when simulating a DMF with many clearances, without large penalties on accuracy.

Paper F: Vibration dynamics in non-linear dual mass flywheels for heavy-duty trucks. In the paper, a non-linear model for simulation of a DMF for heavy-duty applications is proposed. The model includes internal clearances and friction. The LuGre friction model is used, which depends on normal force, relative velocity between the two surfaces and an internal deflection variable. Measurements on the DMF are performed in a test rig and the test rig properties are analysed. The correlation shows that the general behaviour of the DMF is reproduced by the simulation model proposed. The viscous part of the friction is dominant for the cases analysed, with zero mean torque, and a Coulomb friction model would not suffice for this application.

6 Conclusion and outlook

6.1 Contribution of the thesis

To conclude, the aim of research is recaptured and the contributions of the thesis outlined.

1. *Develop models and methodology for efficient simulations and analyses of engine-excited torsional vibrations in heavy-duty truck powertrains with various multiple-mass torsional vibration absorbers.*

The main models and methods developed within this work are listed below.

- Torsional lumped mass models of a DMF for heavy-duty truck application with internal parts, friction and clearances included and correlated to rig measurements (*Paper A*, *Paper F*)
- A torsional lumped mass models the PSVA (*Paper C*)
- Torsional lumped mass powertrain models representative of typical heavy duty-trucks (*Paper A*)
- An efficient time-domain simulation method for piece-wise linear models with many clearances (*Paper E*)
- A harmonic balance approach and arc-length continuation scheme for simulations using lumped-mass torsional powertrain models with non-linear DMFs (*Paper B*)

It is concluded that viscous damping must be included in the friction model in order to accurately predict the dynamics of the DMF for heavy-duty application analysed in this work. At some torque levels, there are stiff impacts occurring in the DMF, which can result in problems with convergence in time-domain simulations. The problems can be overcome with numerical damping. By using numerical damping, the time step used for simulations of a DMF with many clearances can be increased significantly, without a large reduction in accuracy, as shown in *Paper E*. The simulation times for piece-wise linear models can be reduced further if the problem is reformulated as a linear complementary problem (LCP). In powertrains with a DMF with one or more stiffness steps, there can exist several different stable steady-state solutions at the same engine speed and load. The harmonic balance approach proposed in this work leads to efficient simulations for models with many degrees of freedom and a non-linear connection between two degrees of freedom. If combined with an arc-length continuation scheme, this method also provides a structured way to map up solutions within a given speed and torque range.

2. *Understand the function, potentials and design requirements of multiple-mass torsional vibration absorbers in heavy-duty truck applications with special focus on dual mass flywheels and power split vibration absorbers.*

Several different absorbers and different aspects have been analysed. The main contributions are as follows.

- A comprehensive mapping of torsional resonance modes and frequencies in a typical heavy-duty truck powertrain with an SMF and with a DMF, including a study of the influence of different design parameters (*Paper A*)
- Simulations and analysis of torsional vibration amplitudes in a typical heavy-duty truck powertrain with an SMF and a DMF within a large speed and torque range (*Paper A*)
- Evaluation of sub-harmonic resonance excitation in powertrains with a DMF with a step in the stiffness, including a study of the influence of different stiffness values (*Paper B*)
- Evaluation of the PSVA function and the influence of torsional stiffness, moments of inertia, damping and gear ratio values on powertrain torsional vibrations and resonances (*Paper C*)
- Analysis of the design requirements for obtaining a good PSVA design with respect to torsional vibrations in heavy-duty truck applications (*Paper C*)
- Evaluation of the functionality of a DMF with TVA and the influence of torsional stiffness and moments of inertia on vibration amplitudes and resonances (*Paper D*)
- Analysis of the possibilities and design requirements for obtaining a low frequency of the introduced additional resonance with a DMF with TVA, thereby avoiding resonance excitation in operating range in heavy-duty applications (*Paper D*)

When the SMF in a heavy-duty truck powertrain is replaced by a DMF, the torsional resonance modes are affected. A mode, referred to as the DMF mode (Mode B in *Paper A*), with large motion between the two DMF flywheels is introduced. As shown in *Paper A*, a DMF for heavy-duty applications can be designed so that the frequency of this mode coincides with third order frequency well below idle speed. Good vibration isolation can then be obtained, since the engine operates over-critically with respect to this mode. There is also a mode with large motion in the clutch damper. This mode is here referred to as the clutch mode (Mode C in *Paper A*). The clutch mode has a resonance frequency corresponding to the third engine order frequency in the engine operating range for a typical heavy-duty powertrain with a DMF. By having a stiff clutch damper, the resonance frequency can be increased, but it will probably not be possible to push the resonance speed out of the operating range. Having no clutch damper at all would typically result in a similar resonance mode with a node at some other weak part of the driveline, still with resonance speed in the operating range. Severe excitation of the clutch mode can be avoided with a proper level of friction in the clutch damper. Less friction would be needed than what is required to limit excitation of the problematic driveline mode common in conventional heavy-duty truck powertrains with an SMF, as concluded in *Paper A*.

Since the DMF mode has a frequency well below the third engine order frequency at idle, it cannot be directly excited by third engine order in the operating engine speed range. If the DMF is designed with several stiffness steps, sub-harmonic excitation of the DMF mode can occur when the oscillation passes two different stiffness levels and when the third engine order frequency is close to twice the resonance frequency. This means that high vibration amplitudes with frequency corresponding to 1.5 engine order frequency can be observed. For a typical heavy-duty truck powertrain, this can occur at low operating engine speeds, as shown in *Paper B*. It is concluded that the maximum 1.5 engine order amplitudes obtained are almost linearly dependent on the difference in the two stiffness values. The mean torque for which the highest excitation occurs is higher than the transition torque between the two stiffness values. This mean torque also varies with the difference in stiffness values in an almost linear manner. The sub-harmonic excitation is described and analysed more in *Paper B*.

The PSVA provides a means to reduce vibration further than a corresponding DMF within a limited speed range. The evaluations in *Paper C* indicate that one of the main challenges when designing a PSVA for heavy-duty applications is to obtain the high reduction range at the normally critical low engine speeds. Increasing the moments of inertia and reducing the torsional stiffness of the PSVA parts will push down the high reduction range to lower engine speeds, but will also require a larger installation space. A higher quotient of the PSVA gear ratios has the same affect, but leads to higher torsional vibration amplitudes at higher engine speeds instead.

Combining a DMF with a TVA is another way to attain good vibration reduction within a small speed range as shown in *Paper D*. The TVA introduces an additional resonance into the system. By installing the TVA on an intermediate flange between the two DMF flywheels, it is possible to have the resonance speed below the tuning speed of the TVA. It is concluded in *Paper D* that a significantly larger size of the absorber compared to a conventional DMF is necessary in order to have a large separation between the TVA resonance speed and TVA tuning speed in a typical heavy-duty powertrain. Increasing damping reduces resonance excitation but also negatively impairs on the vibration reduction of the TVA. Thus, one of the main challenges with the DMF with TVA design is to obtain a sufficiently large high reduction speed range without having detrimental resonance excitation in operating engine speed range or at start-up of the vehicle.

3. *Investigate the possibilities to enable significant down-speeding, using multiple-mass torsional vibration absorbers and evaluate the consequences from a driveline torsional vibrations perspective at different engine speeds and operating load levels.*

A well matched combination of DMF and clutch can provide very good vibration isolation at low engine speeds in heavy-duty truck powertrains and thereby enable down-speeding, as discussed in *Paper A*. The analysis indicates that down-speeding with 100 rpm is feasible, without a significant increase in torsional vibrations into the gearbox. The properties of the DMF and clutch must be adapted to the powertrain, in order to avoid large resonance excitation in the operating speed range, and all gears need to be considered. A compromise between good properties at low and high mean torques is generally needed.

The PSVA or a DMF with TVA could possibly also contribute to a reduction of torsional vibrations at low engine speeds and thereby pave the way for further down-speeding. This would require a significantly larger installation space.

6.2 Recommendation for future work

The friction model used for simulations of straight spring DMFs in this thesis is correlated with test rig measurements at different rotational speeds and torque amplitudes. The mean torque also affects the friction. Further verification is recommended for different mean load levels, combined with various rotational speeds and torque amplitudes. The models used for simulations of the PSVA and DMF with TVA are well suited for parameter studies in the beginning of a design phase. For good prediction of vibration amplitudes in later development stages, more advanced models are required. Friction and clearances should be modelled in detail.

Much of previous work related to DMFs in vehicles at different operating conditions concerns passenger car applications. However, the operating conditions, design requirements and the expected absorber life are significantly different for a heavy-duty truck than for a passenger car. Future work should address transient scenarios for heavy-duty powertrains with DMF, such as engine start, gear shifting, acceleration, deceleration, stalling and loss of road traction. For these studies, engine, gearbox and clutch control play significant roles and must therefore be accurately modelled in simulations. Vehicle mass and road conditions are also important factors that should be considered.

When engines are down-sized or cylinders deactivated, the torsional vibrations increase. In addition, the frequency content of the engine load changes. The resonance speeds are shifted and this affects the performance of the DMF. DMFs for heavy-duty applications in down-sized engines and when cylinders are deactivated should be subject to future research.

If the PSVA and DMF with TVA are to be industrialised, a lot of future work is needed to verify the concepts for different operating conditions. For a DMF with TVA, vehicle start-off from rest when the resonance is passed would need to be analysed in more detail. Evaluation of the sound emitted from the PSVA gears is recommended for the PSVA. It must also be verified that an acceptable life of these components can be obtained.

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