2

Land in sight?

Torsional vibration damping for future engines

Jürgen Kroll
Ad Kooy
Roland Seebacher

Introduction

Based on public discussions in recent months, one could easily gain the impression as a consumer that the age of combustion engines is already at an end, and that electric power will soon edge out conventional engines altogether. To the experts, however, it is clear that this is a fallacy. While an increasing number of hybrid applications – and for city traffic, even purely electric cars – will certainly come to market over the next several years, the combustion engine will continue to bear the majority of the load in mobility for the foreseeable future.

This fact makes developing more efficient engines not only a primary objective for all automakers, but a matter of their very survival. Driven by stricter laws on CO_2 reduction, the operating point shift moves squarely to the foreground for the combustion engine. This shift can be achieved in principle both by downsizing, i.e., reducing the piston displacement or reducing the number of cylinders, and by downspeeding, i.e., adjusting the transmission ratios towards lower engine speeds (Figure 1).

It must also be considered for a large group of customers that, with all economy, the engine performance must be maintained. An apparent conflict of aims results, the solution to which has been aptly described with the term "efficient dynamics." So in the future there will be high-powered, highly charged engines with lower numbers of cylinders, which, compared to today's engines, will again introduce increased torsional vibrations in the drive train.

In middle class applications, 4-cylinder engines will edge out the classic 6-cylinders, while today's 4-cylinders will be replaced by 3-cylinder models. Such engines, belying the implications of the term "downsizing," will be highly challenging technically in order to fully live up to their potential with regard to consumption and driving performance. This applies particularly to high-quality vehicles, for which the customer will accept no curtailment of drivability or quiet performance. As a result, the reduction of torsional vibrations, not always the engine developer's primary focus in the past, becomes a decisive development objective. This is where most of the damping systems available today will meet their limits, especially when one considers the even greater demand for torsional vibration isolation caused by the transmissions with reduced friction (Figure 2).

Another trend, which can already be seen now, and which will only intensify in the present economic climate, is the consumer's desire for smaller, low-cost vehicles. In this customer group, functionality and price are the primary deciding factors when purchasing a vehicle. Comfort demands must often take a back seat. Whether, in the end, 3- or even 2-cylinders will make up the majority of engines here as well or, for cost reasons, 4-cylinder engines of relatively simple design will dominate, at least for a transitional period, remains to be seen. For such vehicles, when it comes to reducing torsional vibrations, the task is generally to find the right compromise between technical efficiency and cost.

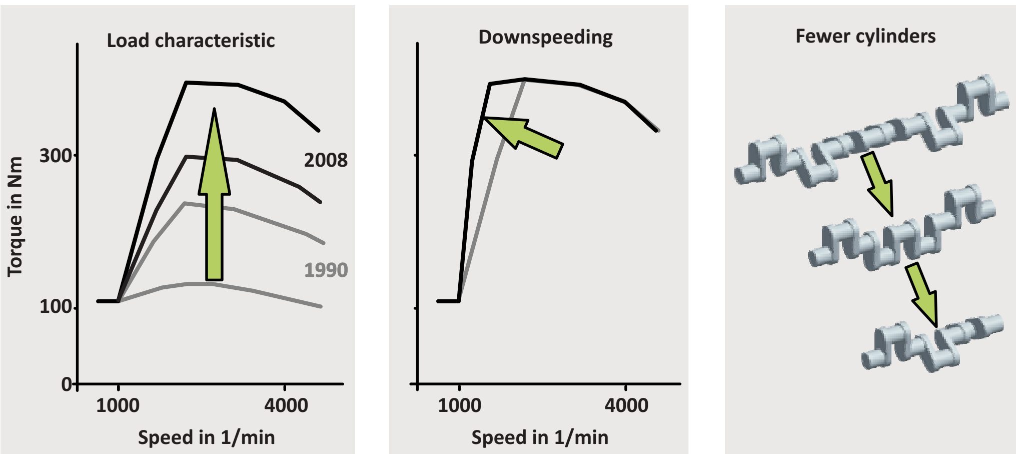

Figure 1 Development trends in engines

Torsional dampers for engines of the future 2

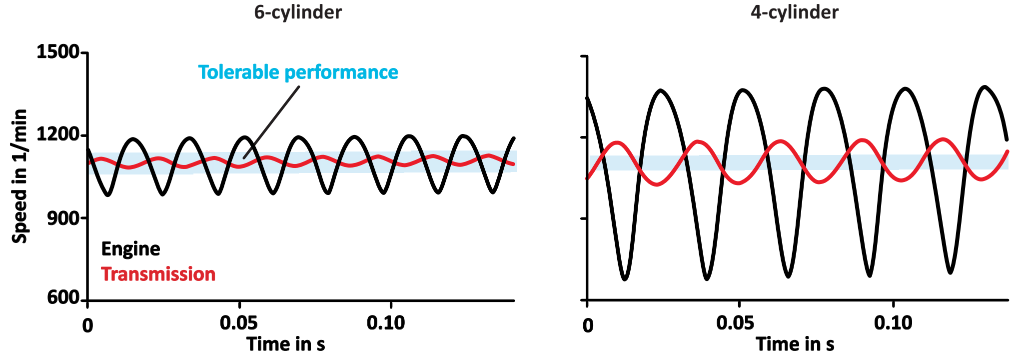

Figure 2 Rising demand for vibration isolation with downsizing

Here we will explain the basic process for selecting the optimal damper and show which systems are best suited for the different requirements.

Damper System Design

In order to minimize development times and risks when designing damper systems, an optimization ideally comes in the early stages of development, with a system simulation using one-dimensional vibration models within the time range. As already presented in Symposium 2006, excellent mathematical methods for optimization have been developed, which contain all relevant operating points and, in combination with sensitivity diagrams, enable a comprehensive evaluation [1].

Since then, the process has been refined and tightened and can now be used in any new development. In the conception phase, different damper systems, including arc springs and inner dampers, but also drive train parameters, such

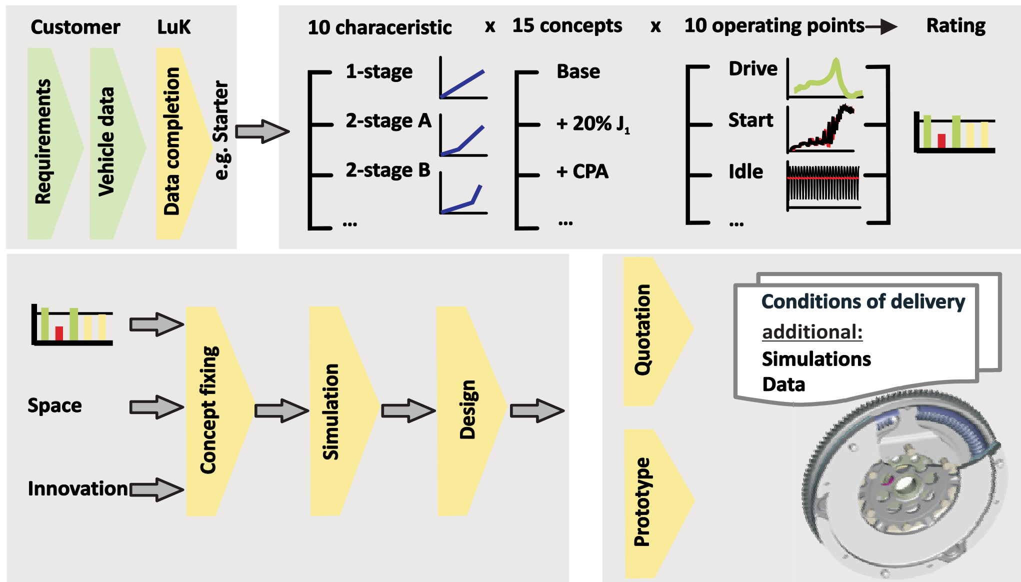

Figure 3 Simulation variations for DMF design

as the starter characteristic, can be thoroughly simulated and evaluated (Figure 3). This makes it possible to evaluate a vibration-optimized complete drive train concept through simulation.

While extensive automation means that running the programs themselves presents no major problems, the drive train parameters, motor regulation data and vehicle sensitivities necessary for the data input are often of insufficient quality during the conception phase. In some cases, up to 90 % of the simulation-relevant parameters must be estimated. Of course, the great store of experience built up at LuK over many years helps in such estimations. However, especially with new concepts, it is not possible to foresee all unwanted effects in this way and correct them where necessary.

An example of this phenomenon is shown in Figure 4, which depicts various starter characteristics that are known from the past. The bold dotted line is used where we have no data available from the OEM for the characteristic curve. In order to ensure an adequate engine start with this conservatively selected characteristic, a two-stage damper characteristic with a flat first stage would have to be selected here. For the drive isolation, however, a one-stage damper characteristic would be more advantageous. This could be implemented if the starter ensured a higher starting speed, as it would with the red characteristic curve, for example. As a result, possible isolation potential is wasted in this case, only because the actual starter curve in the simulation calculation is not known and must be estimated.

Gaps in the data can only be closed through intensive cooperation with the automobile manufacturer. Precise forecasting considerably reduces development risks and expensive secondary measures – otherwise in borderline cases the vehicle concepts themselves have to be modified by enlarging the space.

Optimized DMF with Centrifugal Pendulum-Type Absorber

Today, for the reasons given in the introduction, 6-cylinder systems are increasingly being replaced with highly charged 4-cylinders. This

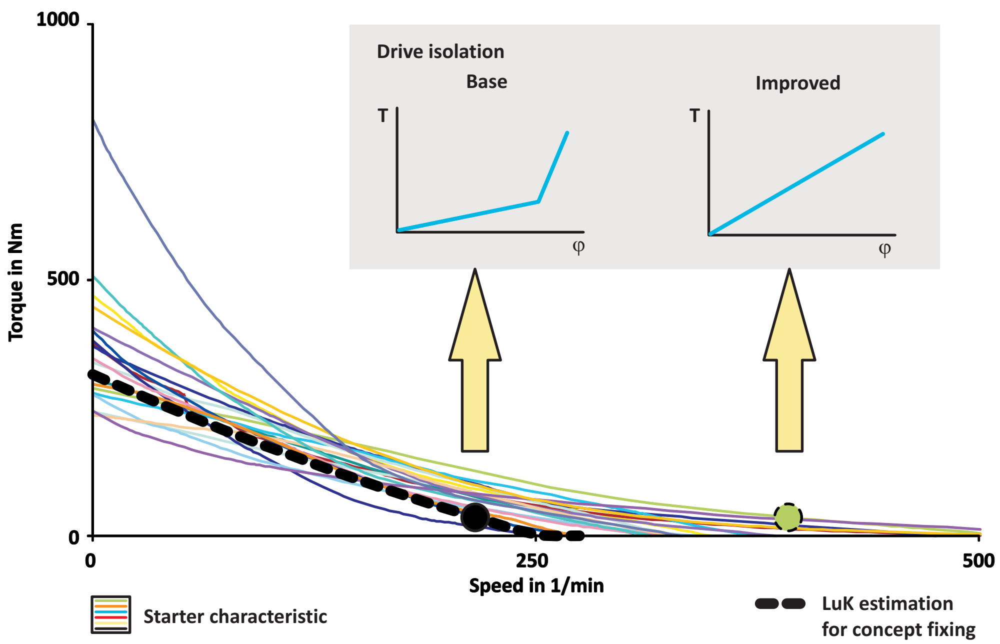

Figure 4 Influence of data quality on the performance of the damper

leads, solely because of the resulting shift in resonance speed, to a significant rise in torsional vibrations in the lower speed range, the range in which driving is particularly economical. And, as the cylinder pressures continue to rise, the irregularities also increase considerably (Figure 5).

But since the driver is kept from driving in the lower speed range by the booming and rattling noises this generates, the consumption advantage of low-speed driving gets insufficient use, despite fully adequate acceleration. Noise and vibration behavior become the Achilles heel of engine development. Damper concepts that allow good isolation even at low speeds are in demand.

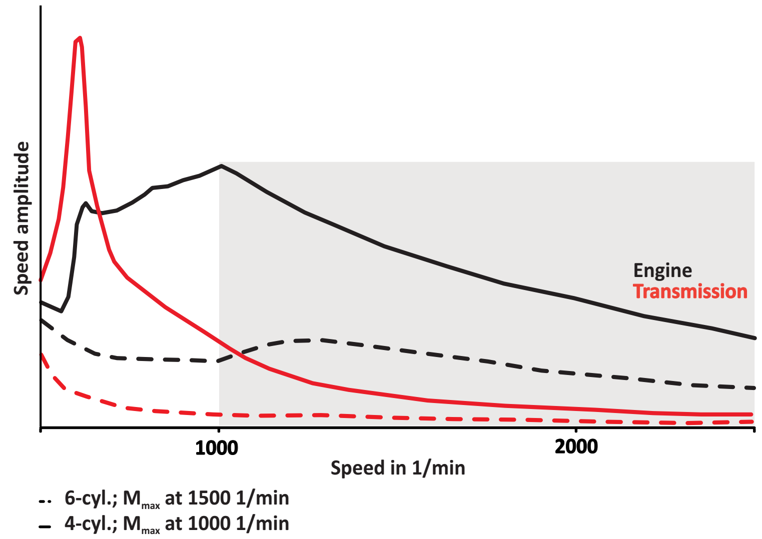

Figure 5 Comparison of engine and transmission vibrations in 4- and 6-cylinder diesel engines (same maximum power, same damper)

The dual mass flywheel (DMF) is the most effective system today for reducing torsion vibrations in the drive train, but its performance has to be improved in order to meet these new demands. Due to their lack of friction, DMFs with inner dampers enable good isolation in the speed range above 1500 1/min, but cannot achieve it at lower speeds because of the relatively high spring rate.

The centrifugal pendulum-type absorber (CPA) is an excellent way to improve DMF performance. This is an absorber damper with a secondary spring mass system outside the flow of power, which, with its resonance frequency excited, moves opposite the exciting vibrations and thus ideally cancels them out. In a classical prop shaft damper connected by means of a spring (steel or elastomer); this effect unfortunately occurs only at one resonance frequency, the described resonance frequency of the prop shaft damper. Two resonance points above and below the prop shaft damper resonance frequency occur so that a classical prop shaft damper is unsuitable for the applications described here.

It is different with the CPA. The reset load of the spring mass system is primarily determined by the dominant centrifugal force acting on it and not by the negligible gravitational force. But since this centrifugal force, unlike the constant gravitational force, is speed-dependent, a speed-adaptive prop shaft damper is created, i.e., a prop shaft damper whose natural frequency shifts as the speed changes. In this way it is possible to cancel out a fixed order of excitation and not a fixed frequency. This makes the CPA an ideal component for a piston engine since the pendulum is tuned to the primary order of excitation and can, at least theoretically, cancel it out. In practice, this cannot be achieved with a CPA arranged on the crankshaft or the rigid flywheel. The engine irregularities are too great and the vibration angle and the size of the CPA too small, because of space restrictions, to achieve this effect.

So how can we use the CPA effectively for the drive train? By combining it with a DMF! The CPA is coupled with the secondary side of the DMF, with the element for which the level of torsion vibrations has already been reduced to 10 % to 20 % of the starting value on the engine. Since now only these remaining irregularities must be compensated for, significantly lower pendulum masses and vibration angles are necessary [2]. In this way a CPA can be inte-

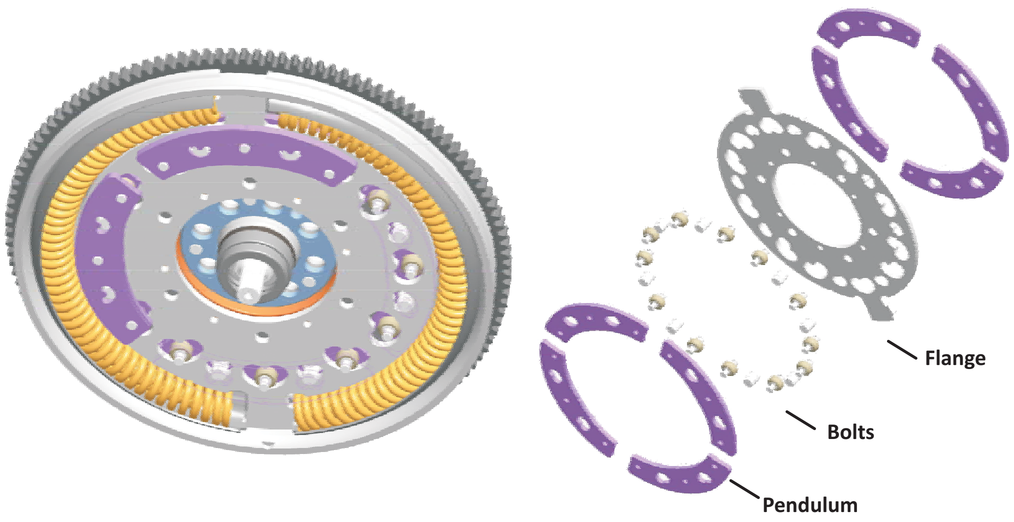

Figure 6 CPA design in DMF

grated in the available space. It is easy to see that more space means better performance here too.

The principle of a DMFW with CPA was already presented at LuK Symposium 2002 [3]. The exact design of the first CPA used in production is shown in Figure 6. It is a bifilar pendulum, or a pendulum with two suspension points. The pendulum masses are suspended by bolts, which move in kidney-shaped tracks in the prop shaft damper masses and in the DMF flange. The CPA order is determined by the shape of these tracks. The masses do not rotate relative to the flange; all points of the pendulum describe the same curve. In design and production, free vibration of the pendulum must always be ensured. Friction between pendulum and carrier reduces the isolation of the torsional vibrations.

The further the pendulum's center of gravity lies from the center, the more effective the pendulum is at the same mass. Depending on the space available, this then results in different designs (Figure 7).

Figure 7 CPA variants

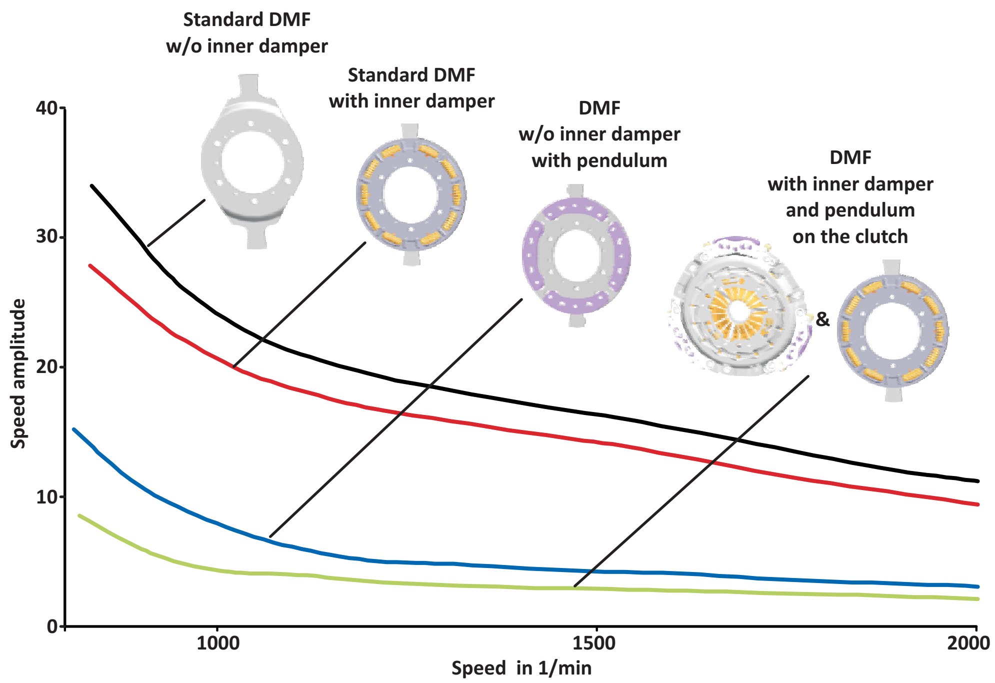

Figure 8 Performance of the DMF with CPA

In small spaces, it can be housed protectively radial below the arc spring. If more space is available, it should be placed next to the primary flywheel or on the clutch cover because of the larger effective radius.

Figure 8 shows the performance of the DMF with CPA. In some cases, measurements in the vehicle showed an isolation of over 99 % based on the engine irregularity. This corresponds to an improvement of 2.5 ratings compared to a DMF without CPA!

With such outstanding results, it is no wonder that the DMF with **CPA** has been so positively received by the industry press. Auto, Motor & Sport, AutoFigure and Autozeitung all praise the excellent driving comfort of the vehicles at low speeds: "Even at 1000 rpms, the 320de doesn't grumble – good job."

How can we improve this system even more for the future? The work is focused on further reducing the costs for such a **CPA** in order to open up many more applications. But the efficiency can also be further increased. New pendulum concepts are currently being evaluated, and here again a great step forward in improving isolation is emerging.

For another thing, the basic function of the DMF, vibration isolation, can and must be optimized. The connection is made clear in Figure 9 with a schematic energy view. The vibration energy of the engine, which is approximately constant over speed, is already greatly reduced by the arc spring damper on the secondary side. It is the job of the CPA to store this residual energy temporarily and then release it again. The energy that the CPA can take up at the maximum vibration angle depends strongly on the speed, however. Therefore in the lower speed range, the secondary side energy can only be partly stored. There remains some residual energy which can excite booming noises in particular. Both increasing the pendulum mass and improving the arc spring damper reduce this residual energy and optimize the isolation. Any improvement to the arc spring damper also facilitates the work of the **CPA**.

One such measure is to reduce the arc spring friction by using a better lubricant.

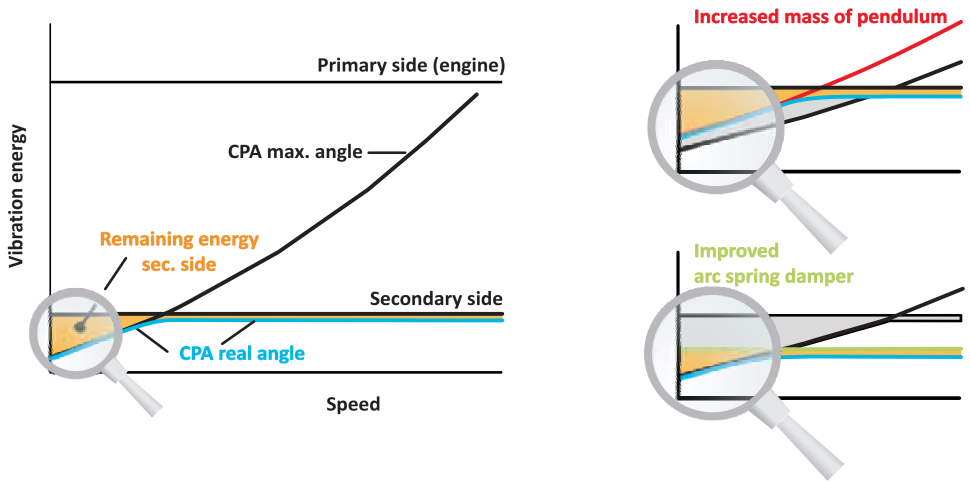

Figure 9 Improving the isolation of the DMF with CPA by optimizing the arc spring damper

Another possibility is a reduction in the spring rate and a related reduction in the spring weight by raising the design stress. To do this, it is necessary to have a precise knowledge of the stress in the vehicle for the relevant driving cycles. With a combination of speed measurement in the vehicle followed by drive simulations of the entire distance, it is possible, with the help of the latest methods such as rainflow analysis and miner rule to depict the total strain in a Wöhler diagram. Driver, vehicle and route influence can be evaluated and design guidelines optimized (Figure 10).

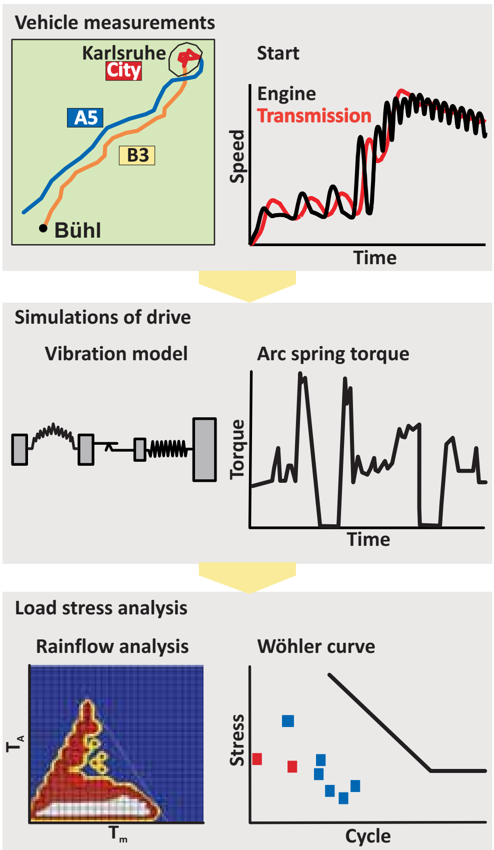

Figure 10 Load stress analysis for DMF optimization

In sum, it can be said that the DMF remains the first choice, especially with the **CPA**. Through its function, the potential offered by future engines can for the first time be expanded to a broader customer base. The consumption simulation in Figure 11 shows just how great this potential is in the NEDC.

Allowing the same excitation in the transmission, a DMF shifts the driving range by 500 revolutions to 1300 1/min, thus resulting in a 14 % fuel savings compared with a system with a rigid flywheel. With a DMF with CPA, an additional 7 % fuel can be saved by a further 300 1/min decrease in the engine speed after shifting.

Economical Alternatives

As already mentioned, in addition to the supercharged 4-cylinder diesel engines, 4-cylinder

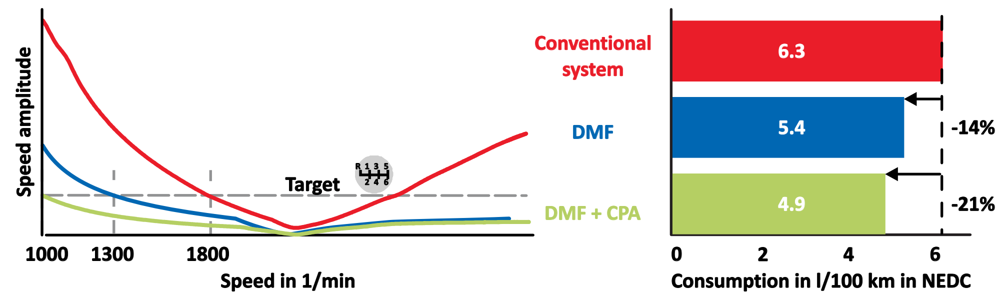

Figure 11 Fuel savings from DMF and DMF with CPA

diesel and gasoline engines with conventional torques, which are economical to manufacture, will continue to play at least a transitional role. These are used in vehicles whose customers strongly emphasize low price. For them, comfort often plays a reduced or secondary role.

The task is to offer economical alternatives to the DMF for such engine and vehicle combinations. Various systems from LuK can be used here depending on the comfort demand. A conventional system with a rigid flywheel and torsion-damped clutch disc is absolutely sufficient for many applications. If a flywheel of drawn steel is used instead of the usual cast flywheel there is a further cost advantage (Figure 12).

Should this concept inherently provide inadequate isolation, then we can turn to the Zmart Damp (ZMD) (Figure 13). Here, costs are lower

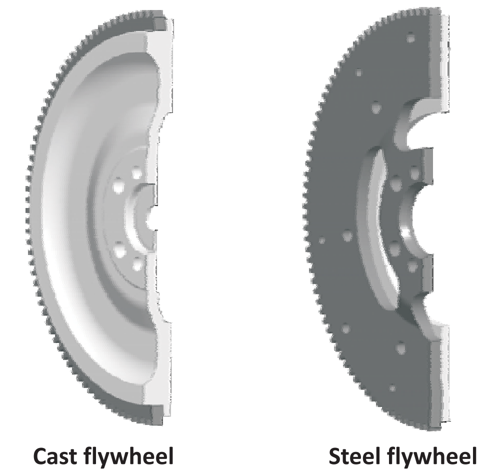

Figure 12 Cast and steel flywheels in composite design

compared to the traditional DMF due to strict optimization or the elimination of individual components. Of particular interest are the single-stage arc spring placed closer to the center and the secondary flywheel as a stamping or formed part. Such a ZMD, which can be used with a torque range up to 240 Nm, provides a significantly better vibration isolation compared to the damped clutch disc, but does not achieve the isolation of a DMF. The costs for the ZMD are also between the two designs.

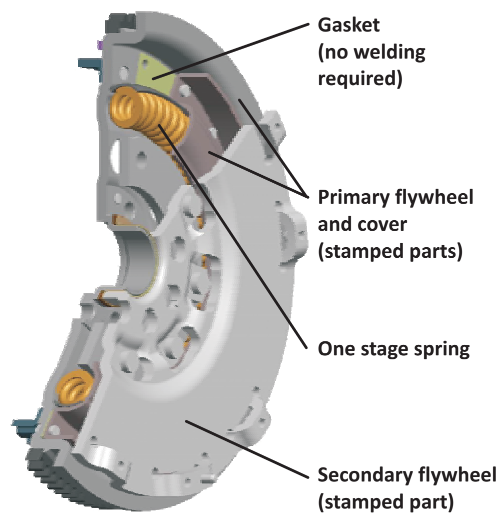

Figure 13 Zmart Damp (ZMD)

Because of the shorter characteristic curve of the ZMD, a sufficient engine torque and a sufficiently high starter speed must be ensured for starting. Early simulations with jointly verified data thus play a decisive role for the design of this concept as well.

3- and 2-Cylinder Engines

While the 4-cylinder engine is increasingly supplanting the 6-cylinder engine in the higher power class, the 3-cylinder engine is certainly gaining traction in the power class from 70 KW to 100 kW.

These engines too have high torques at low engine speeds. But what makes things more difficult with the 3-cylinder engine is that the DMF resonance narrows the drivable "desired" speed range due to the shrinking order of excitation when using a DMF.

In the discussion of ideal damper concepts, the advantage of an overall view of the torsional vibration analysis becomes noticeable. Thus in addition to the DMF parameters, the stiffness of the drive train and here particularly of the axle shafts has a considerable influence on the position of the DMF resonance.

Stiff drive shafts, as they tend to be found in front-wheel drive vehicles, shift the DMF resonance towards higher speeds. By contrast, a somewhat flexible drive train design is favorable for a DMF design, the DMF resonance then being below the driving range (Figure 14 left).

At the same time, the drive train stiffness also has a great influence on the excitation of the natural frequencies. Thus, stiffer axle shafts shift the natural frequencies towards higher speeds, but at the same time, the excitation of these natural frequencies is reduced. This becomes noticeable in an advantageous way with conventional clutch discs in the lower speed range and in higher gears. While "flexible" drive trains have a beneficial effect on DMF designs, "stiff" drive trains are particularly suited for conventional systems.

If the boundary conditions (space, stiffness, drive train) permit, viewed with regard to the vibration amplitude decoupling in the drive train, the DMF is still the suitable torsional vibration damper. As far as reducing the engine vibrations and thus the excitation of the accessories, a conventional system has a certain advantage due to the greater rotating inertia directly on the engine. With a long travel damper in combination with a drive train with stiff axle shafts, a considerable torsional vibration reduction in the drive train can absolutely be achieved, at least in the higher gears (Figure 14 right).

Another option would be a switchable DMF with torsion-damped clutch disc. In the lower speed range, the benefits of a conventional system would come into play with a "rigidly" coupled DMF, in the higher speed range, the benefits of the hypercritical decoupling of a DMF. However, such a solution would be very challenging and expensive, and difficult to implement in the vehicle segment described here.

With 2-cylinder engines, a DMF can be ruled out due to the even smaller order of excitation compared to the 3-cylinder engines. The DMF resonance would always be within the driving range here. Also, for cost reasons, only solutions with a rigid flywheel and damped clutch disc come into use with these engines.

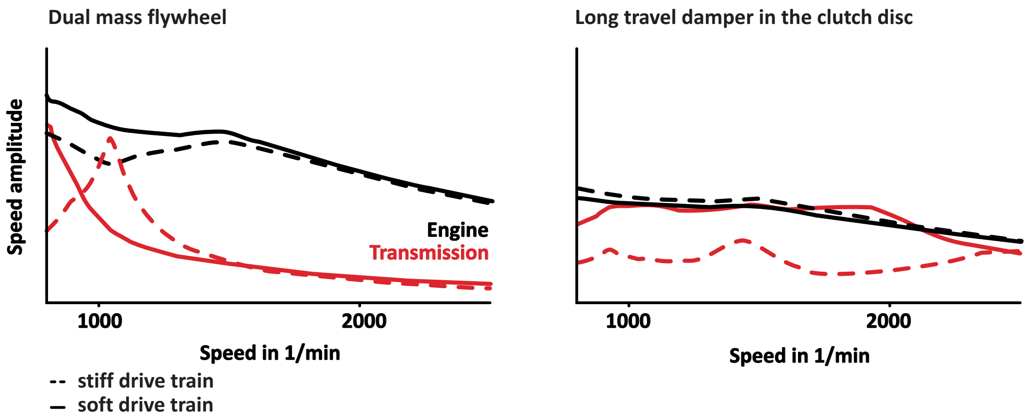

Figure 14 Effect of drive train stiffness on the damper concept in the 3-cylinder engine

Summary

How can we now answer the question posed at the beginning? Are solutions for future engines in sight?

In the range with the highest comfort demands, the DMF with CPA will soon be a standard solution. Further innovations in the area of absorber dampers will enable even better isolation in the medium term.

Where lower-cost solutions are required, there are various options to choose from. Whether conventional clutch disc dampers, long travel dampers or ZMD – the desired compromise between cost and performance can be reached.

For 3-cylinder engines, the optimal damping system must be selected depending on the drive train stiffness. Both DMF and in the clutch disc long travel dampers can be used. With 2-cylinder engines, conventional solutions are used exclusively.

With all applications, it is of utmost importance to evaluate the various options with drive train simulations early in the development phase and finally to select the best. It is also important to determine the space requirements for the system and reserve it in future vehicles. Close cooperation between the automaker and supplier is required in this early development phase in order to be certain of finding the best solution.

Literature

[1] Fidlin, A.; Seebacher, R.: DMF Simulation Techniques. 8th LuK Symposium, 2006

[2] Reik, W.; Fidlin, A.; Seebacher, R.: Good Vibrations – Bad Vibrations, VDI Conference Vibrations in Drives, 2009

[3] Kooy, A.; Gillmann, A.; Jäckel, J.; Bosse, M.: DMF – Nothing New? 7th LuK Symposium, 2002