DMF simulation techniques –

Finding the needle in the haystack

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Introduction

With the development time for new vehicle models continuously falling (Figure 1) and the demand by the automotive industry for cost savings during the development phase increasing, the supplier chain must also rethink its product development.

In concrete terms, this means that fewer and fewer experimental vehicles are being built and made available to the supplier industry. In the development and optimization of torsional vibration dampers, the practical opportunities for vehicle testing are continuously shrinking, which has resulted in simulation technology gaining in importance.

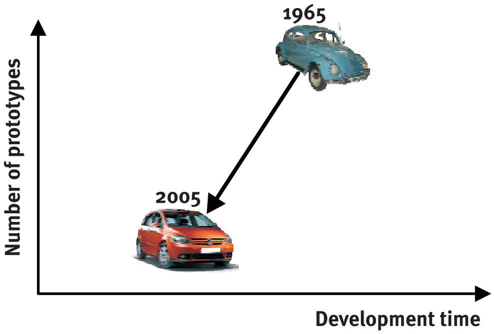

Figure 1 Changes in the development process

Rising comfort demands (Figure 2) alongside ever stronger engines pose ever more difficult challenges for torsional vibration damper development. In order to transmit increasing engine torques within the same package space, the characteristic curves must be made steeper. Since higher engine torques also result in greater irregularities, the damper must do ever more to minimise vibrations in the drivetrain.

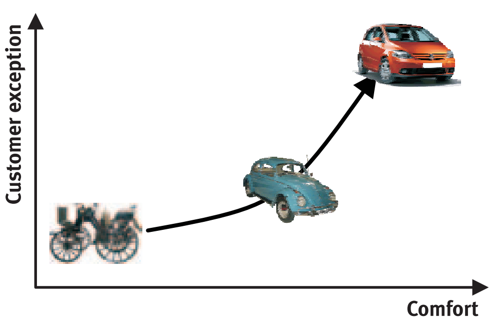

Figure 2 Expectations of increasing comfort

The boundaries of what is physically feasible can, in principle, only be reached with the use of simulation technology and then expanded through the application of innovative technical ideas.

Although simulation technology already plays a significant role in the development and optimization of torsional vibration dampers, its use to date has been mainly problem-oriented. With the design concept already chosen, there is often very little freedom to fully implement the improvements identified.

The goal is thus, to integrate simulation technology into both the pre-development stage and into the very early phase of product development, using simulation to optimize the products from the beginning.

The DMF as an element in improving vibration comfort

During the 1980s, LuK made a decisive contribution in the area of driving comfort. The dual mass flywheel (DMF) made it possible to insolate the torsional vibrations of the engine from the rest of the drivetrain (Figure 3). Annoying gearbox rattling noises were eliminated and body boom considerably reduced. It became possible to drive at very low engine speeds, therefore saving fuel.

Driving along with a situation dependent and thus varying engine speed is the most important and dominant vehicle operating point. Nevertheless, there are many other operating points which must also be considered. Firstly, the engine must be started and later stopped at the end of the journey and perhaps also at traffic lights. The drive itself begins with the vehicle launch. Changes in the accelerator pedal position as well as gear changes cause load changes in the drivetrain, or the vehicle coasts without load. These are only a few of the additional operating points in which there is a high demand for comfort. Figure 4 shows the engine speed and engine speed irregularity curves for the main driving states.

In order to meet ever-increasing comfort demands, LuK developed a new method for optimizing the torsional vibration damper. The primary metric is to optimize the DMF, giving appropriate consideration to all operating points and all relevant vehicle parameters. To take advantage of the full potential of the available parameters,

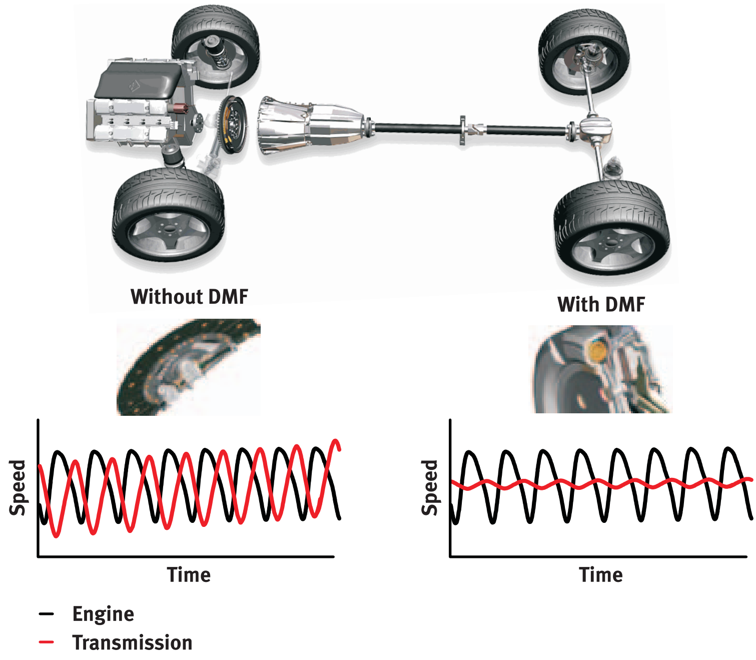

Figure 3 The effect of the DMF on comfort

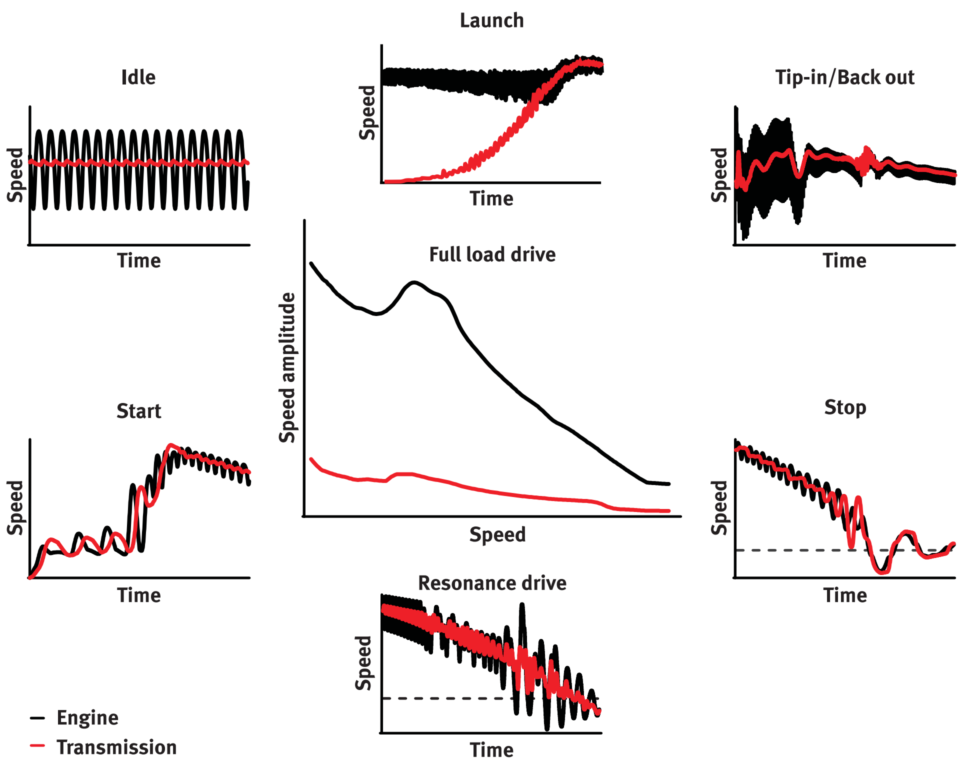

Figure 4 Important vehicle operating points

optimization must be implemented as early as possible in the concept phase. The method of optimization will be covered in greater detail below.

In achieving this metric, additional requirements for the simulation technology arise, which are described below.

Simulation technology prerequisites

Relevant operating points

First, all potentially problematic operating points must be defined. The overview in Figure 5 lists the most important drivetrain problems.

Operating point	Problem
Idle	Gear rattle
Drive	Gear rattle, boom
Coast	Gear rattle, boom
Engine stop	Gear rattle, clatter
Tip-in and back-out	Surging
Vehicle launch	Judder, rattle, surging
Engine start	Durability, comfort
Sub-idle speed resonance drive	Durability

Figure 5 Drivetrain problems

These problems can be largely divided into 3 groups.

- Acoustic problems (gear rattle and body boom)
- Driveability problems (surging and judder)
- Durability (component strength)

Metrics, subjective evaluation and sensitivity graphs

Suitable metrics must be found in order to objectively evaluate the design quality. This means determining which physical values must be measured and establishing the extent to which these measured values correlate with subjective evaluation.

Figure 6 is a schematic representation of the determination of the metrics for the operating state “full load in drive.” Possible problems here are gear rattle and body boom. To evaluate the quality of the drive, the vehicle is accelerated with maximum engine torque. Speeds are measured with high resolution at the transmission input, the differential input (for rear-wheel-drive vehicles) and at the wheel. Depending on the metric, the signals are differentiated or integrated over time and their amplitudes graphed as a function of engine speed. To evaluate the gear rattle, the average amplitudes of the angular fluctuations of the transmission signal are taken as the metric and evaluated over the critical engine speed range. Body boom correlates well enough with the over-accelerations at the wheel and at the differential input. In the range below 1500 rpm, the maximum acceleration of the wheel serves as the metric. Especially in rear-wheel or all-wheel-drive vehicles, drivetrain resonance can bring about boom noises. As the metric for this, the peak acceleration is evaluated at the differential input.

Ideally these measurements are performed with different DMF designs and damper systems. In this way, the largest possible range of measured metrics is captured. The various systems are evaluated subjectively when the measurements are taken. When the subjective evaluations are plotted against the respective metrics, the result is known as the sensitivity diagram (Figure 7).

The use of systems that exceed the comfort achievable with a DMF alone is optimal. For example, torsional vibrations can be reduced to a minimum by using an add-on clutch slip controller. In this way, it can be seen, for example,

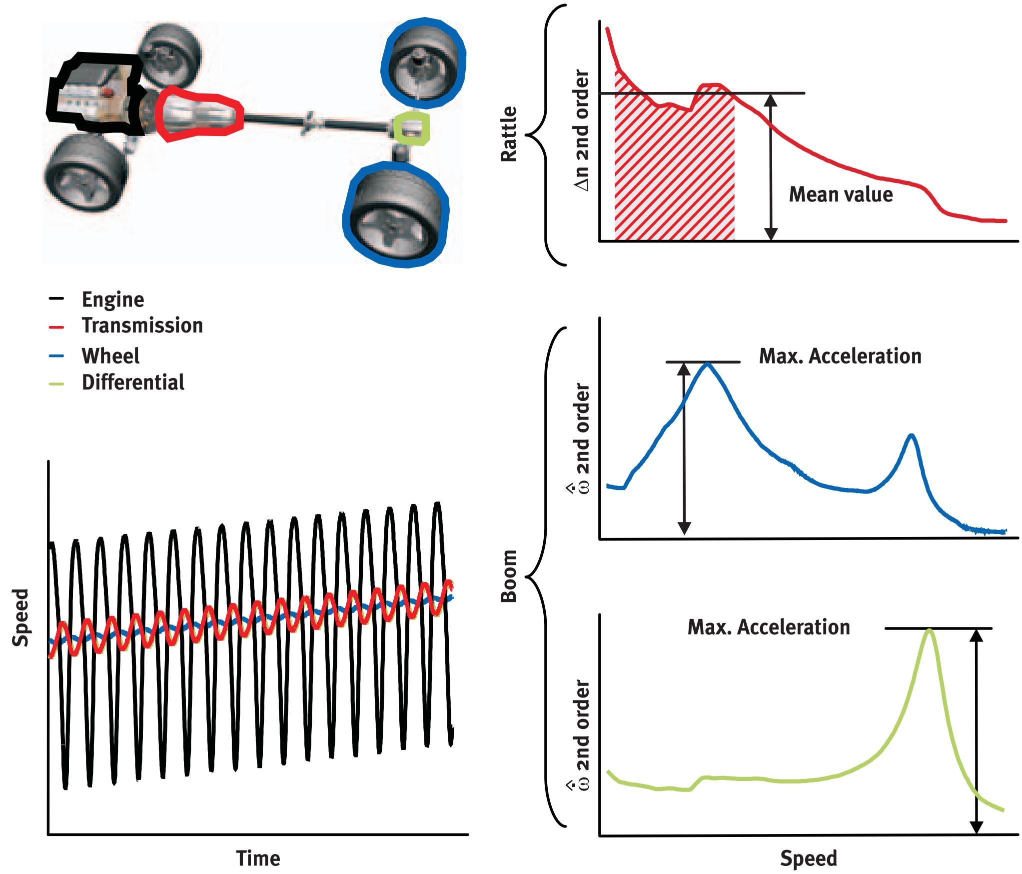

Figure 6 Full load in drive mode: Analysis of the metrics for rattle and boom

to what extent a noise can be reduced by minimising torsional vibration. The above mentioned add-on slip controller was specifically developed to produce the most comprehensive and continuous sensitivity diagram possible. This add-on slip controller is integrated directly in the release system, similar to a clutch-by-wire system.

Figure 7 shows the subjective evaluations plotted against the metric using the example of body boom at low engine speed. A rating of 10 corresponds to noiseless driving, a rating of 1 to intolerable body boom. For most customers, the target rating is 8, but a rating of 6 is the absolute minimum.

With such measurements, on many vehicles of different types, LuK expands its know-how on a continuous basis.

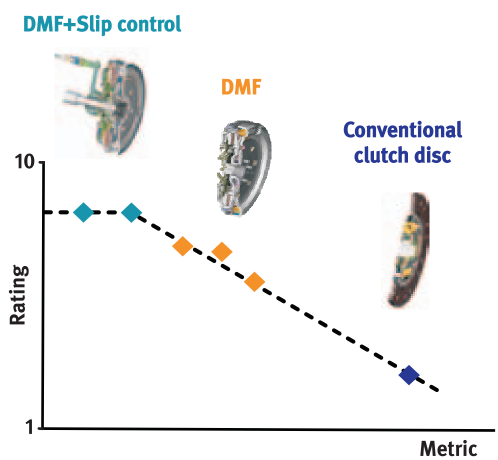

Figure 7 Sensitivity diagram for boom at low engine speeds

Efficient models

Limitations due to computational capacity, demand the use of efficient simulation models with adequate accuracy.

A model with 9 rotating inertias is used to represent the vehicle in the drive mode (Figure 8). When possible, only the relevant natural modes and drivetrain parameters are considered.

In contrast, all power-transmitting elements and exciting objects are modelled in greater detail. The torque fluctuations of the combustion

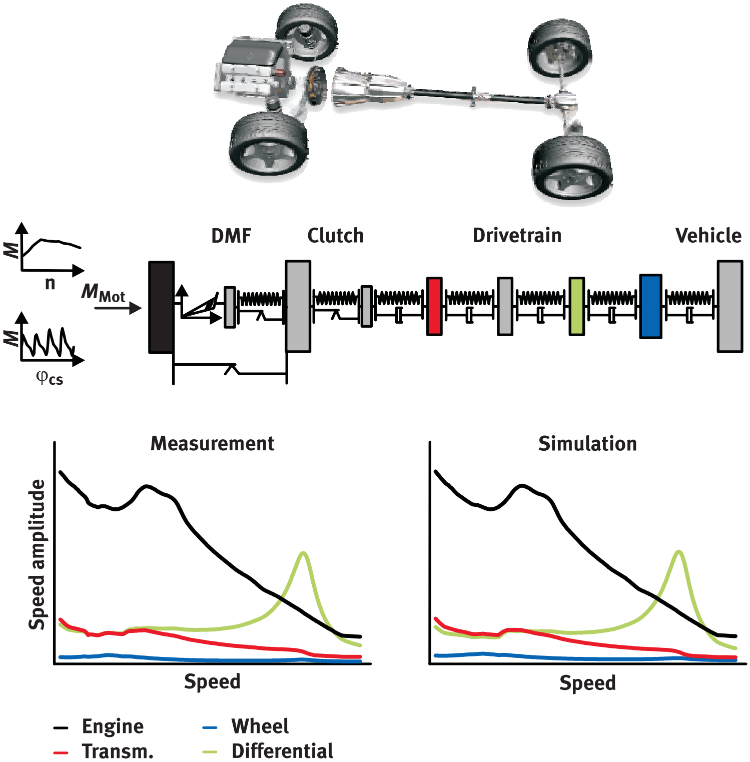

Figure 8 Full load in drive mode: analytical vibration model, measurement and simulation

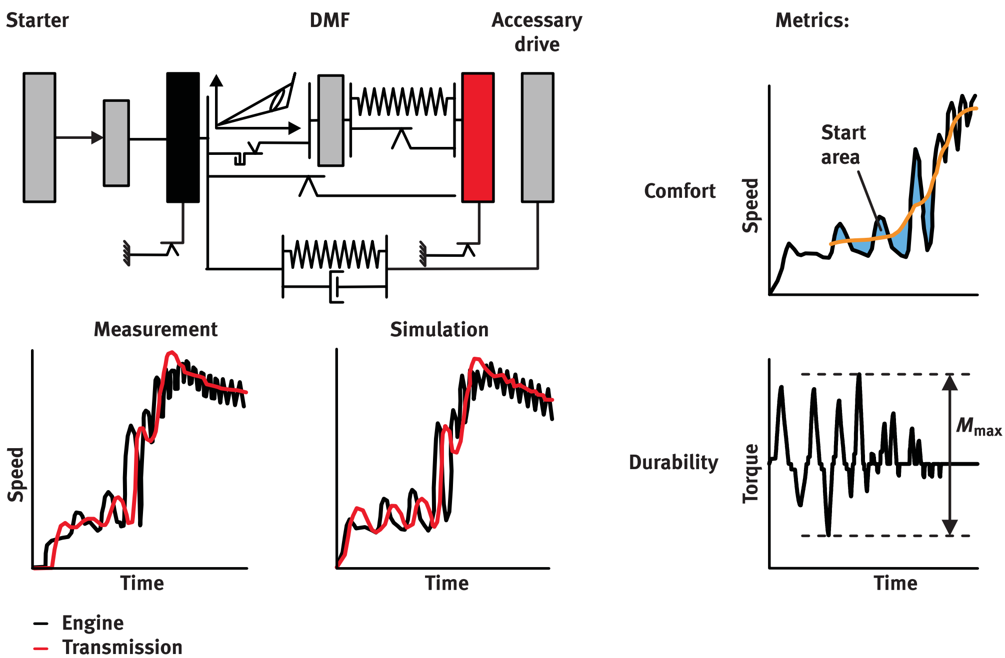

Figure 9 Engine start: analytical vibration model, measurement. simulation and metrics

engine are represented as a function of the crankshaft angle and the static torque. The transmission behaviour of the DMF is considered with all of the complex and nonlinear properties of the arc spring. The stiffness, damping and lash of the drivetrain are determined experimentally with special measurements.

Such a model is able to describe the operating point drive (Figure 9) with sufficient accuracy.

Figure 9 shows a comparable analytical vibration model, measurement-simulation comparison and metrics for the engine start operating point, which can also be very well described.

As was the case with the drive mode, the engine start as a whole is considered. In addition to the DMF parameters, the engine, engine management system, starter motor and accessory drive are considered. The metrics here are the starting time and the maximum torque in the DMF. Whilst the starting time metric is a comfort variable, the maximum damper torque metric is a durability variable. Similar procedures have been developed for all of the other operating points.

Consideration of the complete vehicle

A total system consideration is another important feature in performing simulation calculations. Not only LuK products such as the DMF and the complete clutch system are considered, but also the entire drivetrain, the engine and the engine management system.

This makes it possible to better assess the full potential and limitations of the various LuK products so that other areas of the drivetrain can also be included in the problem solving process when needed.

The total system consideration also makes it possible to detect potential problems with interactions between the torsional vibration damper and other components in the vehicle. In particular, the engine management system, with idle regulator, cylinder balancing controller, anti-jerk controller, etc., is very often in complex interaction with the DMF. Here too it is very desirable to be able to affect an improvement early in the development phase.

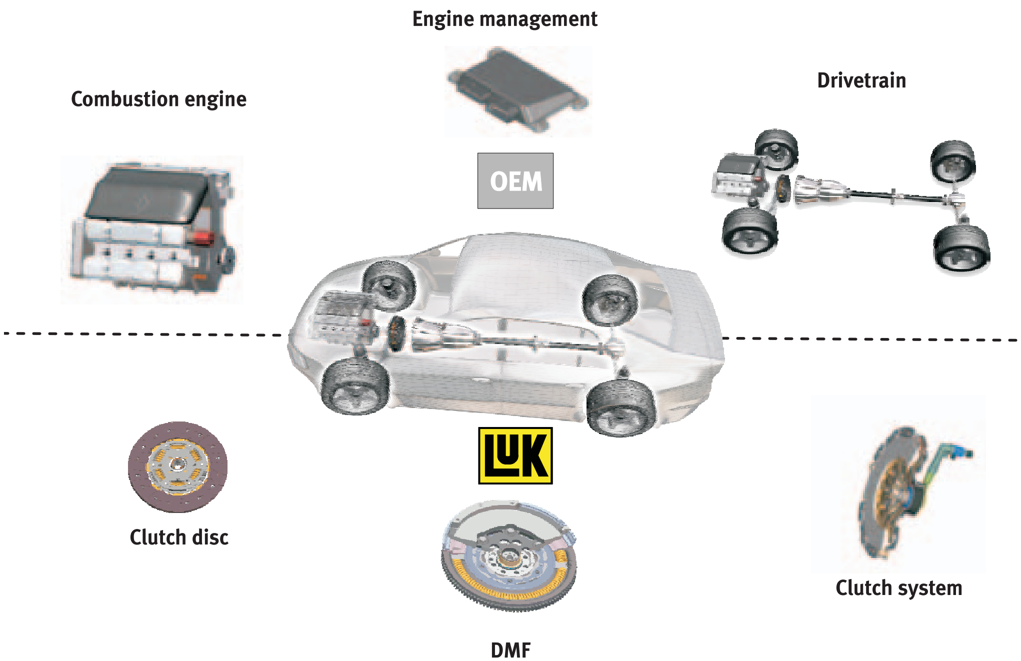

Figure 10 Total system consideration

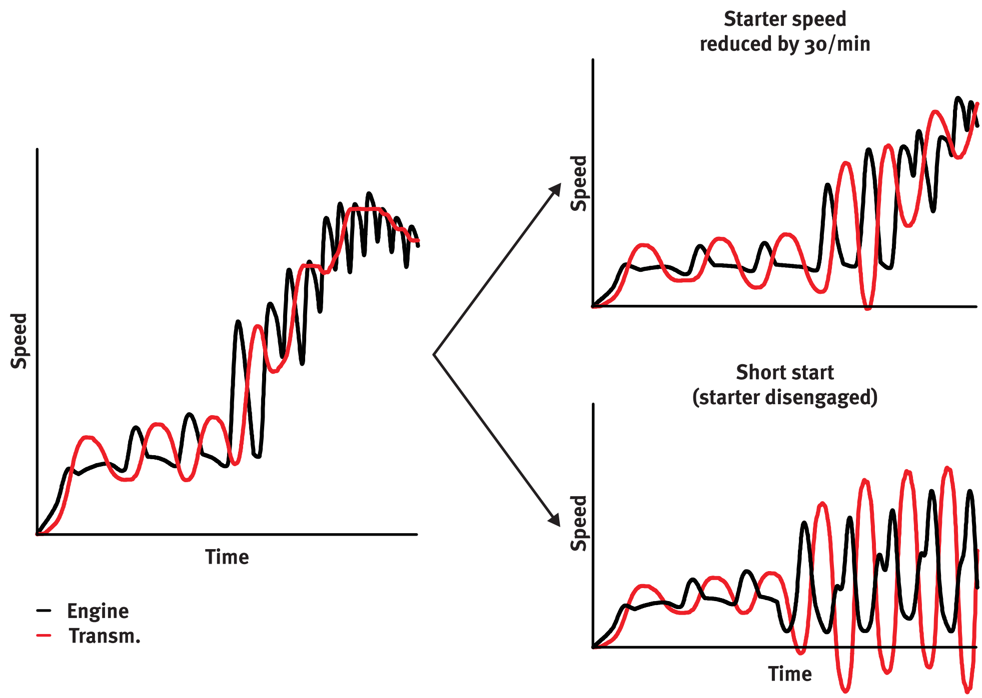

Figure 11 Engine start: Interaction of the DMF with the starter and engine management system

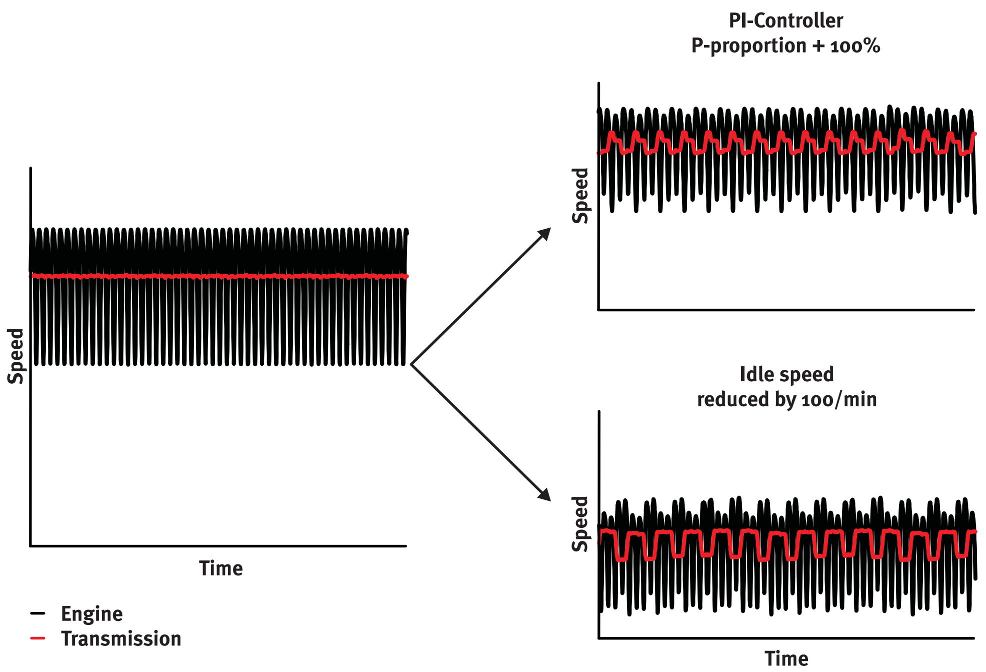

Figure 12 Idle: interaction between the DMF and engine management system

Figure 11 shows the effect of engine management and starter speed on the engine start. The 3 graphs show the speed signals of the engine and transmission. First is a phase in which the unfired engine is propelled by the starter motor alone. After a few cycles, the engine is started. Then the engine speed runs through the DMF resonance, which frequently causes problems during vehicle tuning.

A drop of just 30 rpm in the starter speed has noticeable negative effect on the engine start-up. (Figure 11, top right graph). The situation becomes critical if the starter disengages too soon, e.g. before the first combustion event. The consequence is unacceptable resonance vibrations, which in an extreme case could cause a resonance hang-up (Figure 11, bottom right graph).

Interactions between the complex torque transmission behaviour of the DMF and the engine management frequently have a negative effect on driving comfort in idle mode as well. A lash is intentionally introduced in order to achieve perfect isolation of the firering frequency between the primary and secondary masses of the DMF. Annoying gear rattle being completely eliminated (Figure 12, left graph). This lash and the combined effects of the engine speed and regulator system can also excite low-frequency oscillations, causing unpleasant vibrations (Figure 12, right graphs).

For both examples, it is clearly necessary to include engine related parameters early in the design. This naturally requires that the customer be included early in the development process. The accuracy of the vehicle data has a decisive effect on the quality of the simulation results and thus on the quality of the product.

Automated calculation sequence

Thousands of simulations must be performed for the calculation, depending on the number of influencing parameters and operating points to be calculated. This requires a simulation environment with the greatest possible degree of automation. Appropriate damper and vehicle models are generated from ProE drawings and vehicle data. The calculation plan is created using the influencing parameters. The simulations in the time domain provide raw data for analysis. From the metrics thus determined, an optimal torsional vibration damper is then designed. LuK uses programs developed in-house to perform the simulations and determine the metrics.

Integration in the early development process

Up to now, vehicle tunings have occurred very close to the start of production, i.e. the design concept is already fixed and there remains little freedom in which to make improvements. However, to take advantage of the full potential of the available parameters, it is necessary to include the damper optimization very early in the development process. It is much easier to implement suggestions for optimization in this phase.

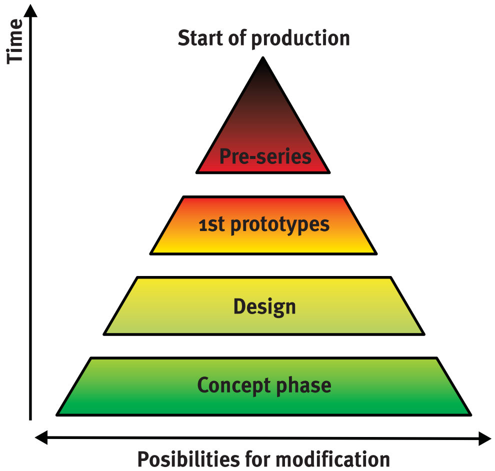

Figure 13 Opportunities for change throughout the development process

Optimization procedure

The process that has been developed is oriented around three essential requirements that are included by definition in product optimization.

- The function of the LuK products depends on many design parameters. Specifically for the dual mass flywheel, there are normally between 18 and 24 parameters (depending on the complexity).
- The products are not optimized in relation to any individual metric, rather they must function well in all customer-relevant driving situations.
- The products are mass-produced. Therefore the production tolerances must be taken into consideration when developing an optimal product.

These requirements essentially determine the optimization strategy. Figure 14 shows the parts that are contained in a simple DMF.

Naturally some of these components and their geometry can be reduced to a limited number of parameters in the evaluation of the function and

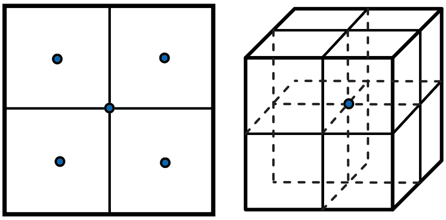

Figure 15 Example: parameter spectrum divided into two- and three-dimensional cases

strength of the DMF. However, since a description of, for example, a coil spring must consider not just its characteristic curve, but also the wire geometry to calculate the stress, the number of parameters becomes relatively large. It becomes larger the more complex the specific structure is (simple DMF, DMF with inner damper, DMF with inner damper and torsion-damped clutch disc, etc.).

This gives the above mentioned number of parameters, which determines the dimension of the parameter spectrum in which the optimization is carried out. This dimension raises exponentially the number of points that would be necessary to scan the spectrum evenly. Figure 15 illustrates this fact. If we were to subdivide each parameter axis in the two-dimensional case into two sections, we would have $2^2 = 4$ regions (squares). In the three-dimensional case, there are $2^3 = 8$ regions (cubes). In a realistic twenty-dimensional case, there would be $2^{20} = 1,048,576$ regions.

Unfortunately, it is normally completely insufficient to vary the parameters on just two levels. If we subdivide each axis into several sections, this produces an astronomical number of possible parameter combinations, which could not be managed even with the most modern simulation technology.

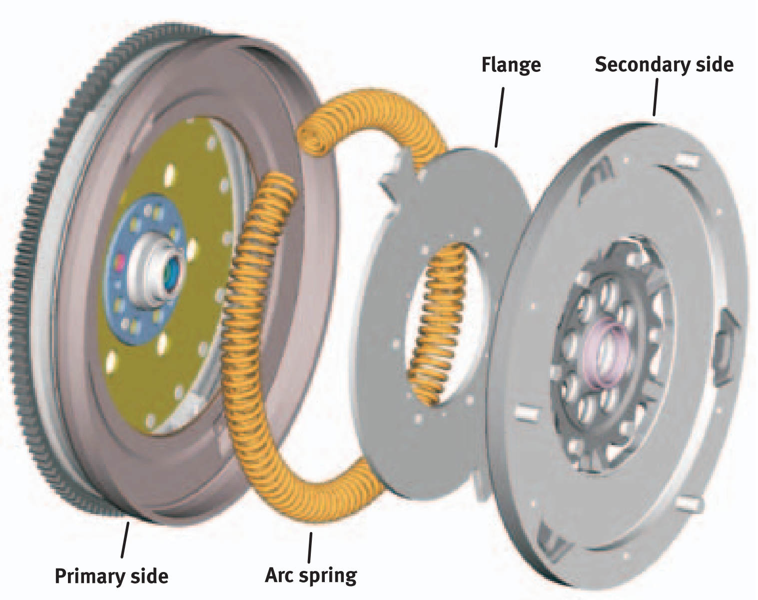

Figure 14 Components of a DMF

The solution is found in the systematic application of statistical methods, which were originally developed for test planning. These methods make it possible to limit the number of parameter combinations and reduce them to a size that can more easily be handled. One factor in favour of this method is that the variation limits of individual parameters must be related to the tolerance variations in series production. This idea is illustrated in Figure 16.

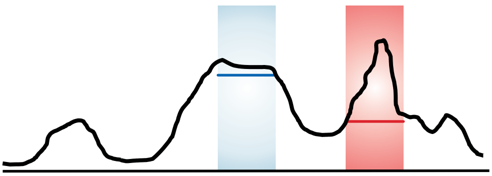

Figure 16 Relation between narrow peaks and wide plateaus during optimization for mass production

Assuming that the curve in this diagram represents the optimization target, the best result would then be achievable at the narrow peak on the right in the diagram. If, however, we place the width of this peak in relation to production tolerances (red rectangle), it can be seen that a clearly worse result is obtained within the tolerances than if the wide plateau in the middle of the diagram were selected as the optimization target (this variance is indicated by the blue rectangle).

In other words, it must be ensured that no part in mass production is worse than a certain limit.

Another and entirely more important aspect is the diverse number of driving situations mentioned above that can be influenced by the DMF. This is illustrated schematically in Figure 17 in the form of a radar graph. Only the most important driving situations are listed. Each of these terms includes several, sometimes very different phenomena, which should be kept separate from one another when evaluating a vehicle. Under engine start, for example, starting time is distinguished from comfort, and under stop, rattling is distinguished from clatter.

All of these initially subjectively evaluated phenomena are represented using the sensitivity diagrams mentioned above.

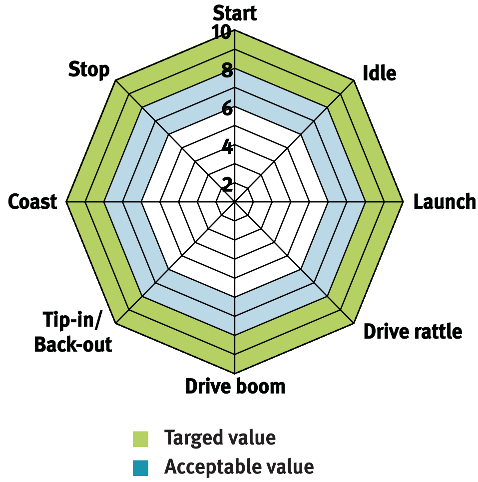

Figure 17 Diversity of driving situations and individual evaluations

In every product development or product optimization process, the question naturally arises as to how the various contradictory requirements of a system can be combined. In other words, how can we find an acceptable trade-off? The usual procedure is to combine individual metrics using a weighted average into one overall metric.

$$Z = \frac{1}{N}(k_1 z_1 + k_2 z_2 + \dots + k_N z_N)$$

$$k_1 + k_2 + \dots + k_N = 1$$

Z is the overall metric, $z_1, z_2, \dots, z_N$ has the individual metrics or driving situations (total of N), $k_1, k_2, \dots, k_N$ shows the weighting of the driving situations. This method also has a basic disadvantage. If a parameter combination is found that results in very poor ratings in certain driving situations, but looks good in all others, it could be viewed as optimum by the automated optimizer. However, this is contrary to real world requirements. A product must not perform below a certain limit in any driving situation. This is called the acceptance boundary. In addition to comfort-related metrics, there are also durability targets (e.g. stress). These boundaries must be met in all cases.

In order to take this into account, LuK does not use the arithmetic mean, but the geometric mean:

$$Z = \sqrt[N]{(\max\{z_1-g_1, 0\})^{k_1} \cdot \ldots \cdot (\max\{z_N-g_N, 0\})^{k_N}}$$

$$k_1 + k_2 + \ldots + k_N = 1$$

$g_1, g_2, \ldots, g_N$ shows the respective acceptance boundaries for the relevant metrics. This form ensures that as soon as one metric falls below the acceptance boundary, the entire parameter combination is also automatically rated unacceptable.

The acceptance boundary for comfort-relevant metrics generally corresponds to a subjective rating of 6. For durability related metrics, it corresponds to the minimum permissible safety factor (e.g. 1.2). Every effort is made, however, to achieve a subjective rating of 8 for the comfort-relevant metrics.

The weightings $k_1, k_2, \ldots, k_N$ make it possible to prioritize the targets specifically for each customer and/or vehicle model. Naturally, the driving characteristics and the resulting comfort demands are different for a limousine than for a sports car. When establishing the weightings, intensive discussions between OEM and supplier are essential.

In summary, we can say that the developed method enables a comprehensive and holistic analysis and optimization of a product within a broad field of variations. The procedures described and the conclusions to be drawn from them are illustrated in the next section by way of an example.

Optimization example using a DMF

The described procedure is illustrated in the optimization of a conventional DMF. The selected object of the optimization was a compact vehicle with a modern 4-cylinder diesel engine with rear wheel drive. This selection poses particularly stringent requirements of the torsional damper and is therefore representative for the conditions in which such systems will have to prove themselves in the near future.

Before the optimization can be carried out, a few steps must be taken. First, a preliminary study is made of various pre-selected concepts-, whereby a few concepts – featuring modular design – are compared with one another. For example, the following components are available for selection:

- A one- or two-stage arc spring
- The DMF can be equipped with a one- or two-stage inner damper or have no inner damper
- The clutch disc can be rigid or have a torsion damper

Once a basic concept has been decided upon, the necessary parameters and their variation limits can be established. The first consideration here is the available installation space.

Every optimization starts with a basic design drawing. The quality of the basic drawing is highly dependent on the experience of the designer. However, by using the optimization method, the influence of the designer's experience is greatly reduced. The quality of the basic drawing does not determine the optimization result, only the time taken until the optimum is reached.

In our case, a DMF with a two-stage arc spring was selected for the optimization (see Figure 14). Its function and durability can be

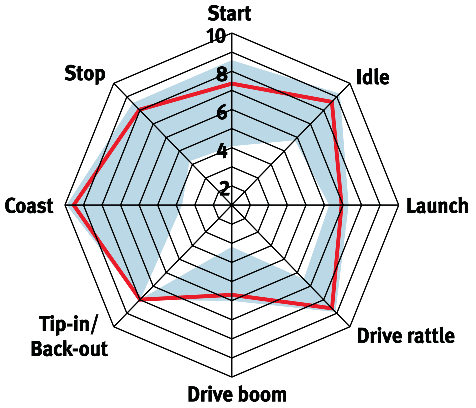

Figure 18 Optimization results for DMF with two-stage arc spring

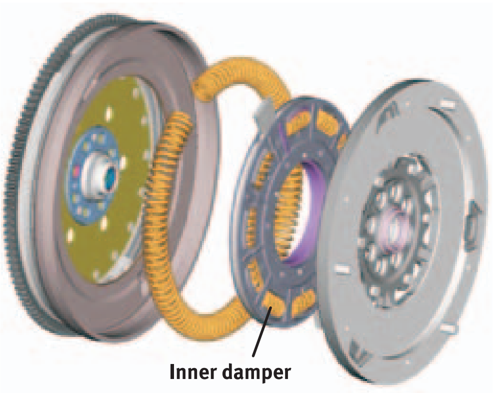

Figure 19 Components of a DMF with inner damper

described with 16 parameters. The function was evaluated in seven driving situations. Normal driving was described with two different phenomena: drive-rattle and drive-boom. This takes the special importance of these phenomena into account. Around 5600 simulation calculations are performed in the optimization. The optimization results are shown in Figure 18.

The blue coloured area shows the range of variation of the evaluations made during the optimization. There is no DMF that would correspond to the inside or outside edge of this range. A DMF assigned a rating of 10 in coast, for example, was only given a 7 in stop. The boundaries show, however, what is possible in general with the concept under study in the various driving situations.

The red line indicates the optimization result. This DMF is actually acceptable in nearly all driving situations. Only for booming in drive mode did it receive a rating just under 6, and so it has

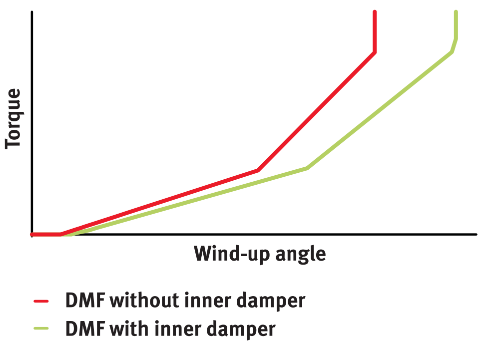

Figure 20 Comparison of spring characteristics of DMF with and without inner damper

delivered the best possible rating for the concept under study.

Since the optimal evaluation still lies within the unacceptable range, the introduction of additional components in the DMF must be considered. The next obvious step is to add an inner damper to the two-stage arc spring. This design is shown in Figure 19.

This raises the number of parameters to 22 and results in a “softer” torsional characteristic curve, with reduced hysteresis, which benefits the overall vibration isolation (see Figure 20).

The optimization result for this concept is shown in Figure 21. Figure 22 shows the comparison between these two concepts.

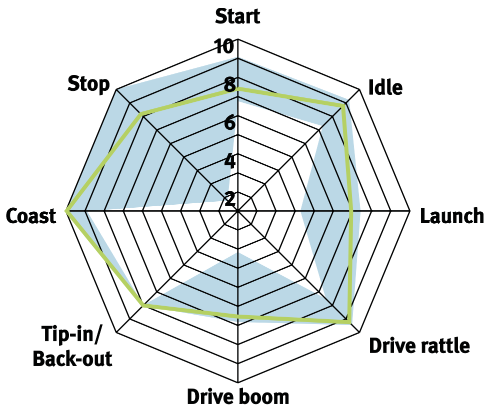

Figure 21 Optimization results for DMF with two-stage arc spring and inner damper

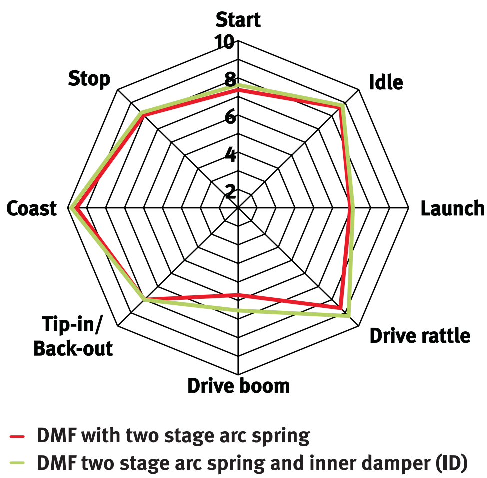

Figure 22 Comparison of DMF with and without inner damper

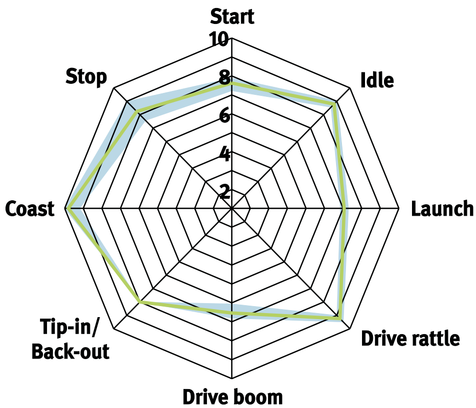

Figure 23 Results of the tolerance study

With considerable additional expense, we are able here to raise the evaluation of the drive-boom at its optimal point to a rating of 6.5. This relates to the nominal design, in which mass production tolerances are entirely ignored. However, using the previously described simulation tool, we can also undertake a tolerance study. Here, all parameters are varied within production tolerances. The variation of individual parameters was determined from actual production data. The result (Figure 23) shows a clear dispersion of the expected evaluations, with a range up to the equivalent of an entire rating.

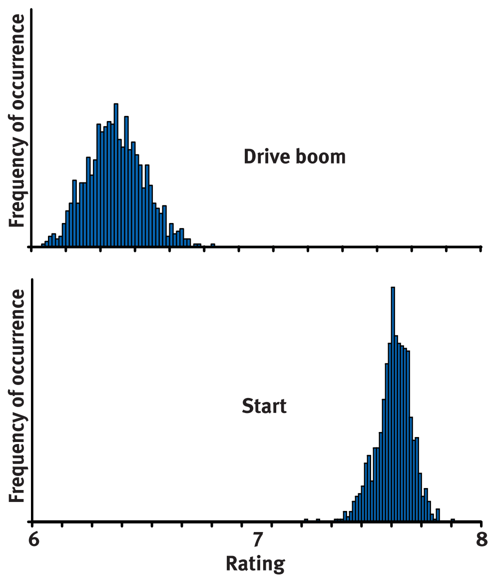

Figure 24 Distributions of the ratings for start and drive mode boom

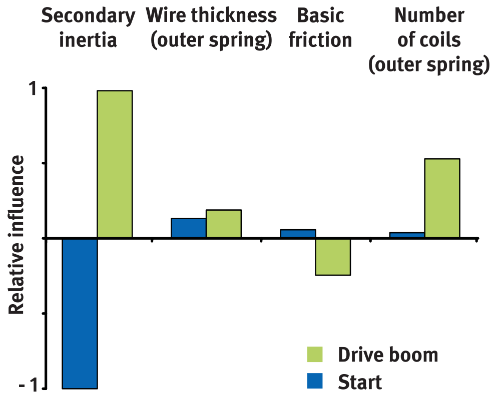

Figure 25 The most important influencing factors and their effects in relation to boom in drive mode and start behaviour

It can be seen, however, that under mass production conditions, a rating above 6 can be ensured in all driving situations. This method also makes it possible to estimate the probabilities of individual evaluations. The corresponding frequency distributions are shown in Figure 24.

The analysis of the most important influencing factors also makes the conflicting goals between start and boom in the drive mode apparent. In order to improve starting behaviour, for example, the secondary mass had to be reduced, and the basic friction increased. Both changes worsen boom in the drive mode, however (Figure 25).

As can be seen from Figure 21, further improvement with regard to booming was not possible with the concept under study. Alternative techniques with new influencing factors are required to realise significant further improvements.

Solution through innovation

The problem described in the last section can only be efficiently solved by changing the structure of the torsion damper. If we stay with the conventional spring-mass systems, we must not only consider the described conflict between starting and booming, but also the perpetual conflict between the transmission of engine torque and the quality of vibration isolation. If the available installation space remains

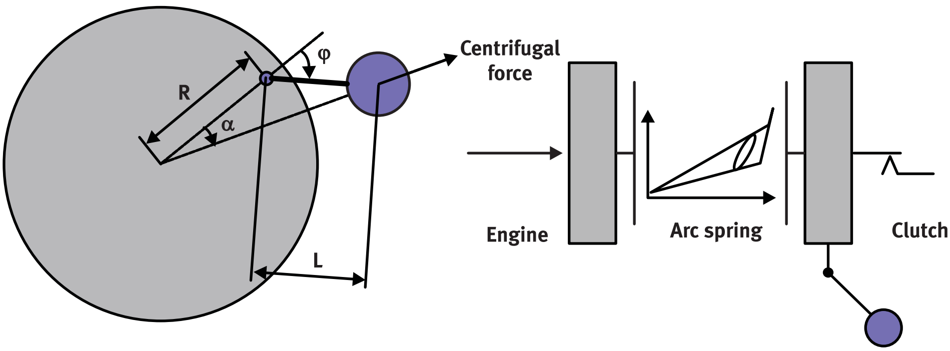

Figure 26 Functional arrangement of the centrifugal pendulum-type absorber

unchanged, increasing torques require ever stiffer spring rates, which degrade the vibration isolation.

One alternative comes from an idea known for many decades – but as yet never successfully applied in the automotive industry – of a centrifugal pendulum-type absorber. The functional principle of a secondary side centrifugal pendulum-type absorber is illustrated in Figure 26.

The centrifugal pendulum-type absorber works like a damper, whose effective stiffness is generated by centrifugal force. The corresponding correcting torque is

$$M = -mRL\varphi'^2 \sin\varphi \, .$$

This means that the natural frequency of the vibration absorber is proportional to the speed. This makes it possible to combat the selected orders of excitation (e.g. the main excitation) very efficiently with the right tuning of the vibration absorber. Figure 27 shows the transfer function of an ideal centrifugal pendulum-type absorber tuned to the second engine order.

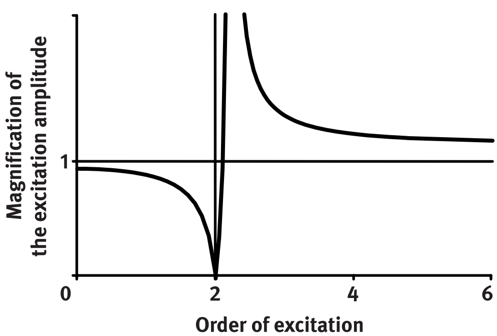

Figure 27 Theoretical transfer function of a centrifugal pendulum-type absorber

The effect of the centrifugal pendulum-type absorber is of course limited by its mass and the vibration angle. For that reason, the centrifugal pendulum-type absorber can only be used in addition to a spring-mass damper in the drive-train (see Figure 26). The basic isolation still comes from a spring-mass system. The remaining vibrations are counteracted by absorption at the engine firing frequency.

One design realization of the DMF with a secondary-side centrifugal pendulum-type absorber is shown in Figure 28.

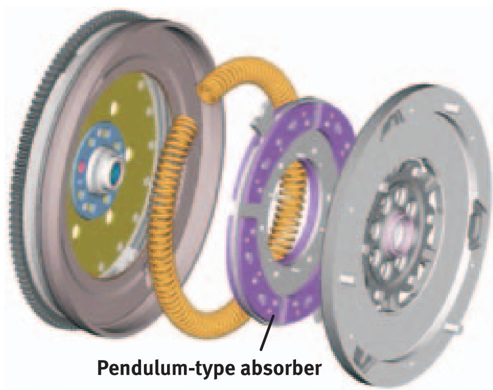

Figure 28 Design realization of a DMF with secondary-side centrifugal pendulum-type absorber

The space in a conventional DMF where the inner damper would be located is instead used to accommodate the centrifugal pendulum-type absorber.

The optimization method already described is an ideal tool for designing such a system. With the use of the centrifugal pendulum-type absorber to reduce torsional irregularities in drive, it

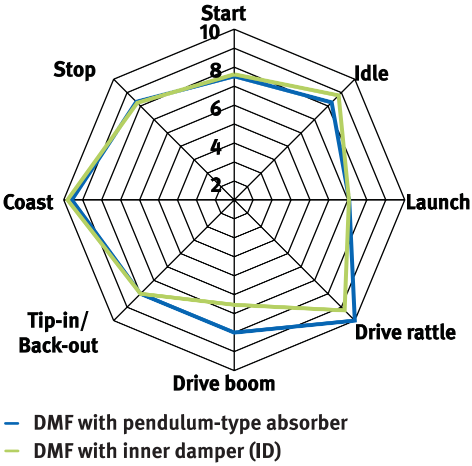

Figure 29 Comparison of optimization results of a conventional DMF with inner damper and a DMF with secondary-side centrifugal pendulum-type absorber

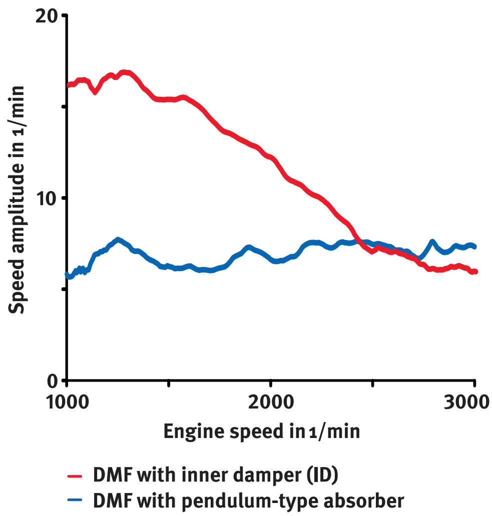

Figure 30 Comparison of test results between a conventional DMF with inner damper and a DMF with centrifugal pendulum-type absorber

becomes possible to tune the spring components differently and to significantly improve the noise level in drive (both rattling and booming). The optimization result is shown in Figure 29.

Due to the separation of functions, the vibration isolation in drive could be improved by approximately one rating. This improvement in the subjective evaluation means in practical terms halving the vibration amplitude in the lower speed range, as illustrated in Figure 30.

This comparison affirms on one hand to the effectiveness of the new system, but on the other hand points to possible further developments. A certain worsening of the isolation in the higher speed range is not perceived subjectively

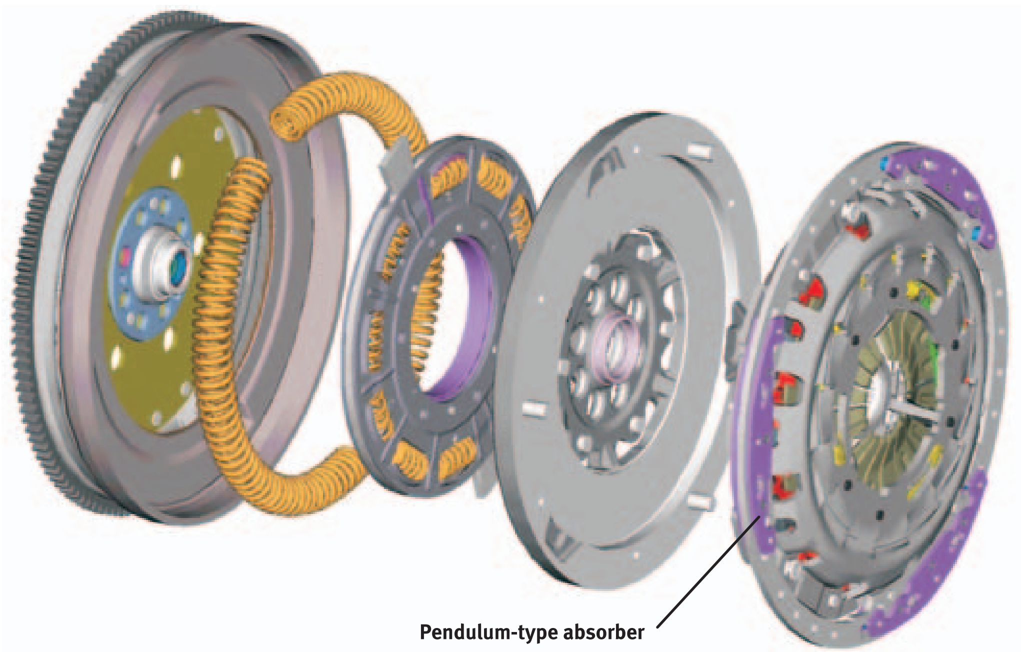

Figure 31 DMF with cover-side centrifugal pendulum-type absorber

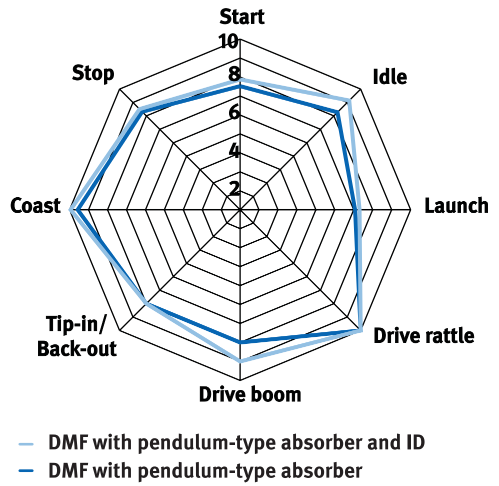

Figure 32 Comparison between centrifugal pendulum-type absorbers, inner and outer with additional ID

as critical in the tested vehicle, but must be overcome so that the new system can be used in the broadest possible spectrum of applications. This worsening has two causes. Firstly, the spring-mass system now becomes stiffer without the inner damper (see Figure 20) and secondly, because the friction in the suspension of the centrifugal pendulum-type absorber itself increases with speed. These reduce the degree of isolation.

In order to overcome at least the first problem, LuK has developed the concept of the cover-side centrifugal pendulum-type absorber (Figure 31). This new concept makes it possible to recover the space needed for the inner damper. In our example, the drive-boom evaluation is clearly one rating better. The expected rating of this system is given in Figure 32.

Summary

We have presented a new technique to support the development of torsional vibration dampers. The aim of this method is to integrate simulation technology into the early concept phase of product development. Here, the proposed concepts are tested using simulation technology, and the influencing parameters on dynamic behaviour optimized with respect to the entire drivetrain.

What is particularly new about this method is the fact that all operating points and production tolerances are considered in optimizing the torsional vibration damper. This provides for continuous quality and process assurance between design, testing and production. Another important factor in the assurance of product quality is the availability of very detailed vehicle and engine data. This requires the close integration of the customer, from the very beginning, within the product development process. Potential problems, such as interactions of the dual mass flywheel and the engine management system can be recognised early on and solved in their entirety.

Since all operating points as well as a great variety of parameters and variations must be considered, a large number of simulation calculations are required, necessitating a high degree of automation. The models and the resulting simulation results must achieve a high level of accuracy. Metrics that best reflect subjective evaluations are defined to evaluate the simulation results.

This method was developed for the optimization of dual mass flywheels, but is in principle also applicable to all torsional vibration dampers, as well as to the optimization of the entire clutch system. One example can be found in the article “The Clutch and the Release System” [1]. The simulation tools developed can also be used for design quality control.

In addition to product optimization and quality assurance, the systematic application of this method reveals conflicting design targets and tests the limitations of existing concepts. Prompting the search for new and innovative ideas that can be comprehensively tested, evaluated and developed virtually, right up to the prototype phase using the methods described.

Literature

[1] Zink, M,; Hausner, M.; Welter, R.; Shead, R.: The Clutch and the Release System – An engaging topic 8. LuK Kolloquium 2006