Development of the Super-Long-Travel Dual Mass Flywheel

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Whenever familiar technology reaches its limits, new approaches and basic improvements are necessary in order to overcome the barriers that arise. In the middle of the 80s, after decades of development on the classic torsion damper, we reached a limit that prevented us from making further improvements in automotive noise comfort. This situation required that we look for new ways to meet rising expectations.

At LuK we took up this challenge. Extensive development work was required, but we found a solution.

The Dual Mass Flywheel – DMFW

The dual mass flywheel has set a new standard for fighting torsional vibrations.

As is the case with all trail-blazing developments, the DMFW is conceptually simple (Figure 1).

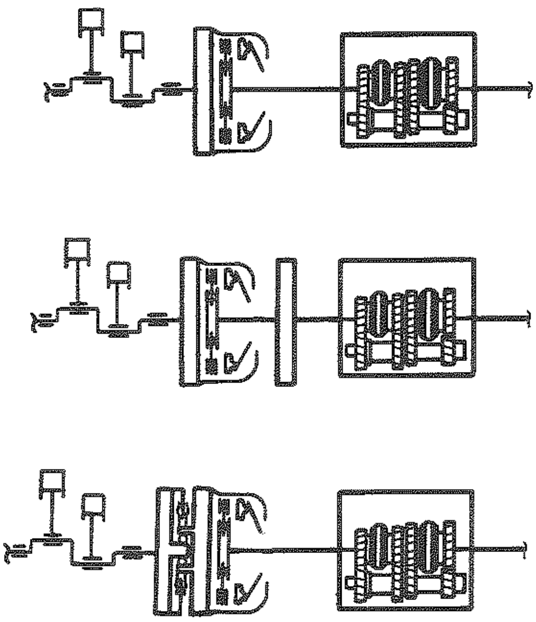

Figure 1:
Basic solution

In conventional designs, the resonance point lies between 700 and 2000 rpm. We have added a supplemental inertia on the transmission input shaft in order to shift this resonance point into the low-speed range. This change produces a vibration isolation range beginning with idle speed.

Shifting the clutch to a position behind the divided flywheel ensures that the transmission can still be synchronized.

Torsional vibration measurements (Figure 2) for the clutch disc and the DMFW show two totally different conditions [1]. In the speed range around 1000 rpm, the DMFW already provides perfect isolation, reducing vibration amplitudes on the transmission side. At this point, the damped clutch disc is as yet unable to filter out vibrations.

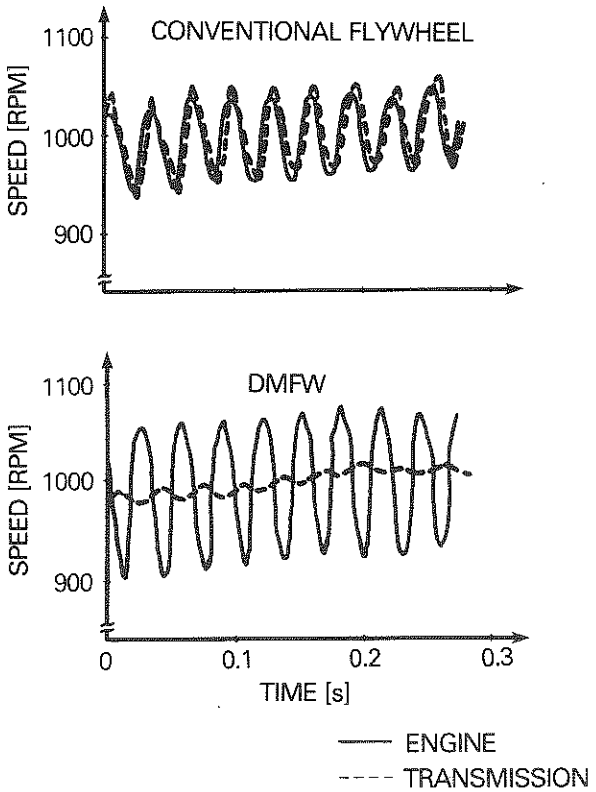

Figure 2:
Comparison: conventional flywheel – DMFW

When we install the DMFW, engine irregularity usually increases, as can be seen in the measurements. This is because the effective mass moment of inertia on the engine side is generally lower. However, this is usually permissible with respect to gear rattle because the isolation effect of the DMFW provides adequate reserves.

The first generation DMFW was developed for six cylinder vehicles [2,3]. It solved noise problems in these vehicles by achieving the necessary degree of isolation described above.

However, four cylinder vehicles exhibit a higher degree of irregularity, accompanied by higher resonance speeds. Consequently, this DMFW could not be used for these applications. We had to consider other solutions. First of all, we have to draw upon a vibration model in order to make some basic observations.

Vibration Model

As has already been noted in other presentations, complete description of a drive train would require a complicated vibration model [4]. However, if we limit ourselves to the resonance performance of the DMFW, a three-inertia vibration system like that shown in Figure 3 is adequate for explaining the effects of various corrective measures.

The elements of the model are identified as follows:

J_1 = Engine inertia, including the primary flywheel inertia and its damper components
J_2 = Secondary flywheel inertia with clutch, clutch disc and transmission component
J_3 = Vehicle inertia
C_{TD} = Torsion damper spring rate
C_A = Drive train spring rate

The three-inertia vibration system has two resonance frequencies with their accompanying natural modes.

During surging, the vehicle J_3 vibrates in opposition to the engine J_1. The torsion damper and drive train spring elements operate serially between these two inertias. The secondary DMFW inertia lies in the vicinity of a vibration node, which means it has virtually no influence on the system. The related natural frequency of 2 – 5 Hz is so low that any excitation due to the ignition frequency is ruled out. However, this vibration mode can appear after a rapid torque change, such as a tip-in. It takes the form of objectionable surging.

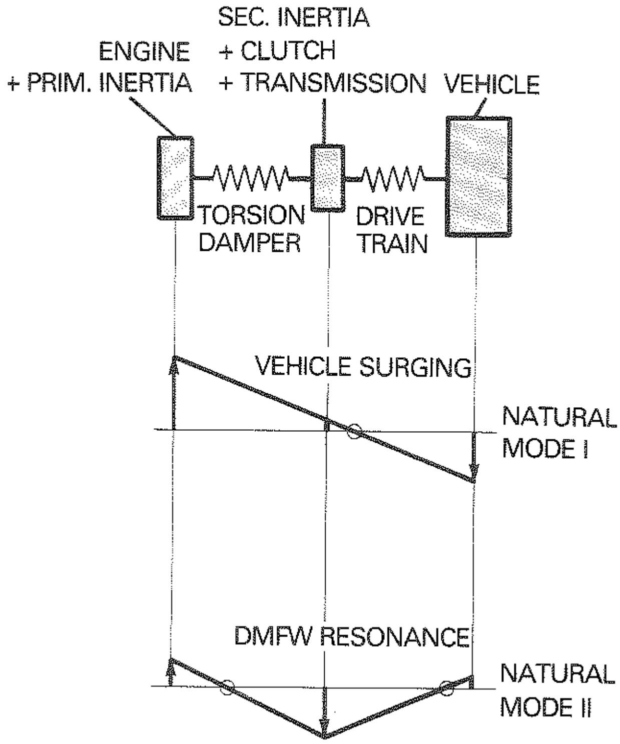

Figure 3:
Vibration model and natural modes

In the case of the second natural mode, a resonance occurs between the primary and the secondary DMFW inertias. At high amplitudes, the two DMFW inertias, which are connected together via springs, vibrate in opposition to one another. The vehicle itself is hardly involved at all. Consequently, the following discussion refers to this second natural frequency as "DMFW resonance." This phenomenon represents an important physical characteristic of the DMFW system, which must be given careful attention when making design calculations.

Both natural modes must also be taken into consideration when designing traditional clutch discs. However, the frequencies and resonance speeds involved are different.

Resonance Speed Field

As noted during the first presentation for this symposium, engine excitation is not harmonic. Therefore, in addition to the ignition frequency, we have to deal with additional excitation frequencies. The whole-number multiples of the ignition frequency are especially important, since they produce higher order excitations. It follows that there are many conceivable speeds at which the model shown in Figure 3 can pass through resonance. However, these higher order excitations have no real effect, both because of their low energy content and because of the damping capacity inherent in the DMFW.

For this reason, the following discussion will only deal with the primary excitation. Depending on the number of ignition cycles per revolution, this is the 2nd order for a four cylinder engine, the 2.5th order for a five cylinder engine and the 3rd order for a six cylinder engine.

Figure 4 shows the calculated resonance speeds for these kinds of engines plotted as a function of the spring rate realized by the DMFW. In creating this graph, we chose to use system parameters that are typical for a compact vehicle. Resonance speeds are defined as those speeds at which the primary excitation – the ignition frequency – excites the two possible resonances in the drive train. The fact that resonance speeds decrease as the number of cylinders increases is attributable to higher ignition frequency at the same engine speeds.

Transmission ratios change the vibration model. Spring rates and inertias located after the gear set must be adapted using the square of the speed ratio. Consequently, each gear produces a different resonance curve. Therefore, the lower boundary of the various shaded areas represents 1st gear and the upper boundary, 5th gear.

Even in 5th gear, the first natural mode, surging, is positioned below 250 rpm, and exhibits virtually no remaining dependence on the torsion damper spring rate above 10 Nm/°. DMFW resonance, defined as the second natural mode, clearly reflects the influence of the number of cylinders and of the spring rate. However, it is not enough just to know the resonance speed in order to completely describe DMFW performance. We also have to take a look at the vibration amplitudes that occur.

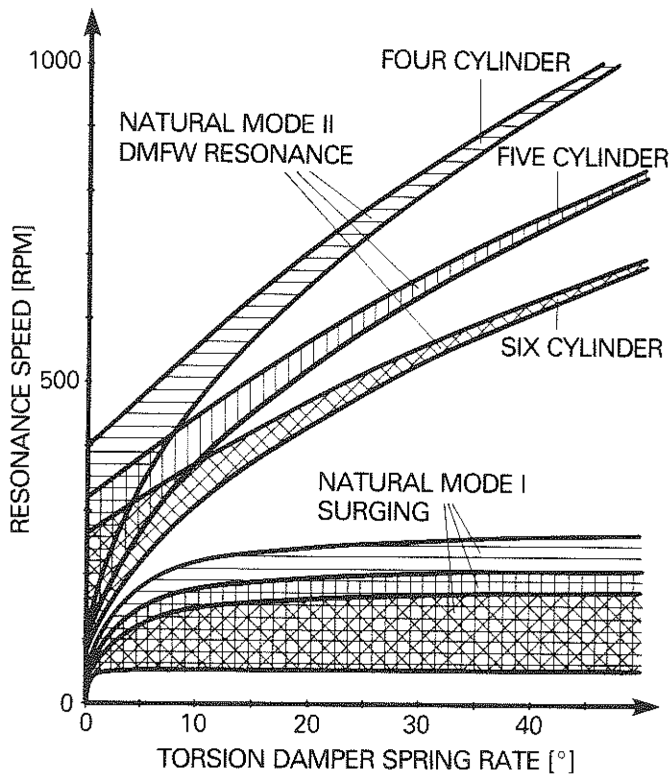

Figure 4:
Resonance speed ranges

Amplitude Curve

The vibration model shown in Figure 3 allows us to calculate the amplitudes at the transmission input shaft that are critical for transmission rattle.

Diverging from a conventional design, Figure 5 shows increased secondary inertias. This demonstrates one option used with 1st generation DMFWs. All other parameters for the model of a four cylinder application remain constant. Increasing the inertia shifts the resonance point to lower speeds. However, the vibration amplitudes increase considerably.

Although it is possible to reduce the peak amplitude, this kind of reduction requires design features such as friction or damping elements that impair the function of the DMFW in the vibration isolation range.

High peaks in transmission amplitude are typical for vibration systems that employ a large secondary inertia to shift the resonance speed. During start-up and shut-off, the vibration system can pass through resonance. This problem can require additional complicated damping mechanisms.

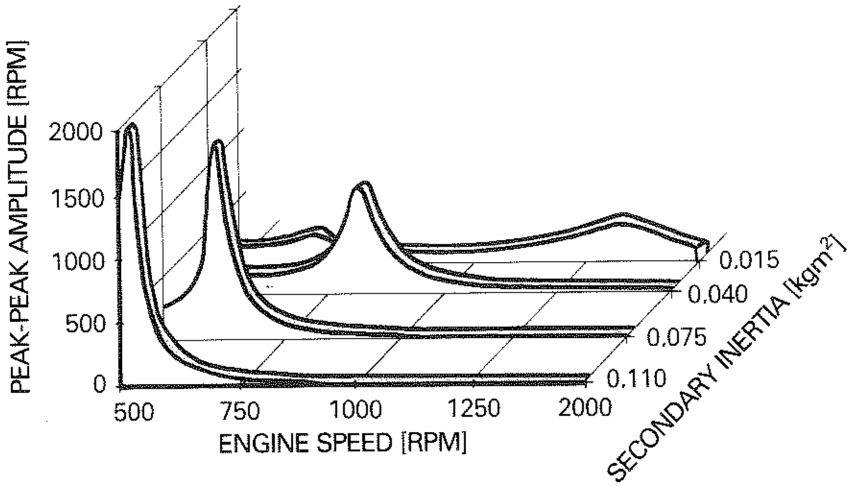

Figure 5: Variation in the secondary inertia

A higher secondary inertia on the transmission side, coupled with a lower spring rate, can significantly shift the resonance speed in a favorable direction. Figure 6 shows resonance curves for a DMFW whose secondary inertia – 0.110 kgm^2 – corresponds to the first curve in Figure 5. In this case, the torsion damper spring rate has been varied.

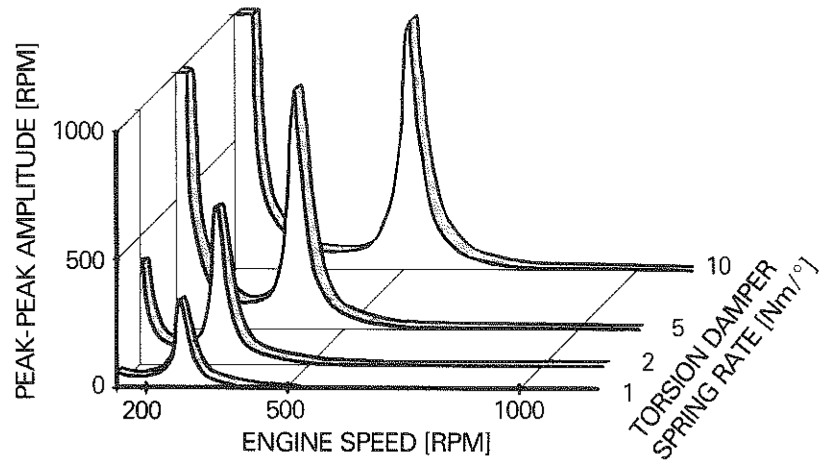

Figure 6: Variation in the spring rate C_{TD}

Note in particular in this case that using softer springs not only shifts the resonance range to lower speeds, but also reduces vibration amplitudes considerably without introducing any additional corrective features, that is, without adding supplemental damping devices.

Start-up Measurements

The graph in Figure 7 shows the measured curve for engine and transmission input speeds during engine start-up. These curves illustrate the undesirable consequences that can result when the system passes through resonance in conjunction with a high spring rate. This is an example where the system gets stuck in the resonance range. This can occur primarily in diesel vehicles after a very brief activation of the starter motor.

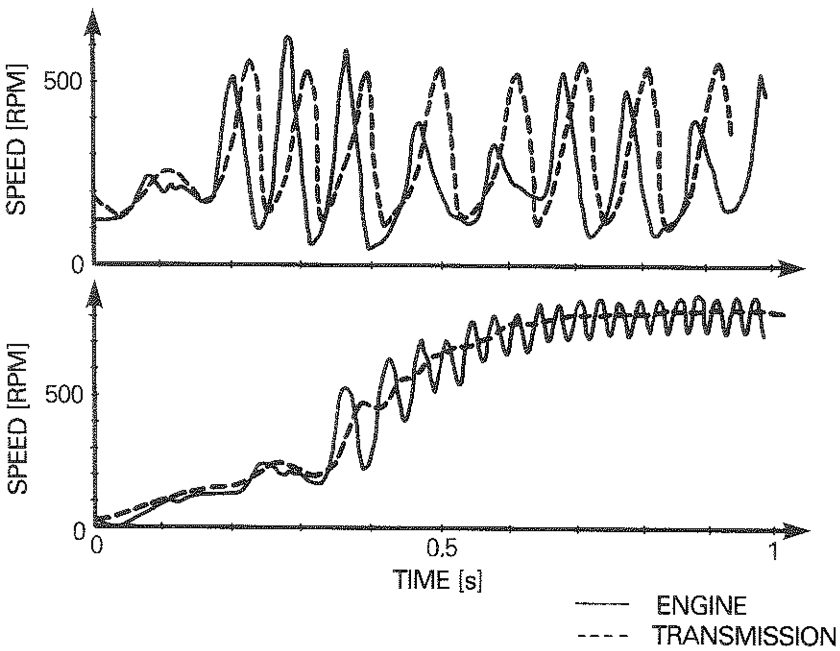

Figure 7: Measured start-up curve (diesel)

Because the diesel fuel injection pump is already injecting fuel at the maximum rate during this phase, it is even impossible to accelerate up to idle speed by stepping on the gas. In this kind of extreme circumstance, very severe dynamic stresses can occur, which vastly exceed the maximum engine torque and can destroy the dual mass flywheel in a very short time.

The graph shown in the bottom of the illustration demonstrates how we can completely eliminate this tendency to "get stuck" in the resonance mode. This is accomplished by significantly decreasing the spring rate, which in turn reduces the associated resonance speed to about 200 rpm. In this case, we achieve good vibration isolation above starter speed. This eliminates the overloading of the DMFW during resonance operation. This advantage is associated with additional improvements in noise comfort when the system does pass through resonance, such as during engine start-up and stop.

In the case of gasoline engines, there is no danger of start-up resonance. However, a low spring rate does produce clear-cut improvements during the start-up phase (Figure 8). Lower spring rates enable us to achieve lower transmission amplitudes during start-up, which yields improved noise performance.

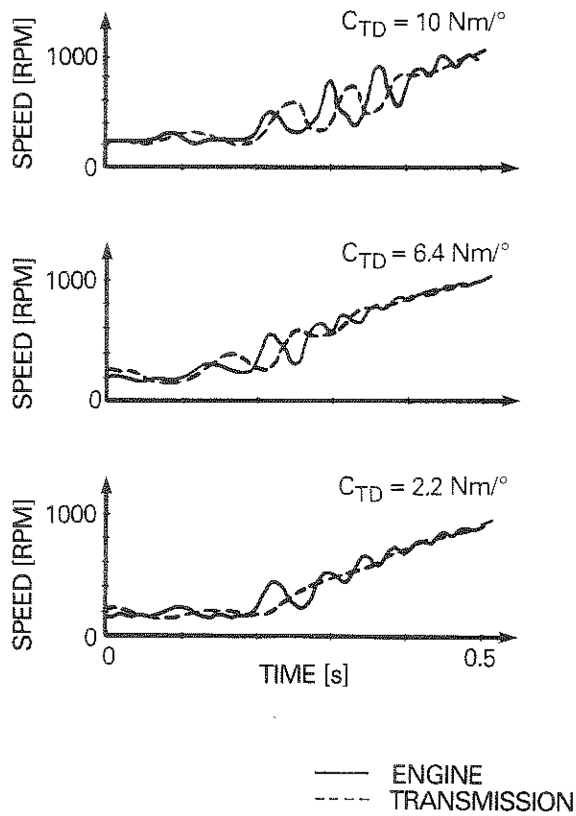

Figure 8:

Measured start-up curve (gasoline engine)

Damper Capacity

Up to this point, the discussion has clearly indicated the decisive significance of a low spring rate for ensuring reliable DMFW operation. Vibration isolation can be extended to significantly lower speeds, and the start-up resonance, which was much-feared in the early years of DMFW development, can be avoided without employing additional, expensive corrective measures.

Hence the spring rate or damper capacity has become the deciding factor for achieving optimum DMFW performance today. The elastic energy Q stored in the springs of a torsion damper is described by the area under the wind-up characteristic, which can be calculated for a linear spring rate using the equation

$$Q = \tfrac{1}{2} \cdot M \cdot \varphi \qquad (1)$$

where M = torque capacity [Nm]
and φ = maximum wind-up angle [°]

The maximum amount of energy that can be stored in the damper – in short the damper capacity – constitutes the primary design variable. Figure 9 provides a graphic illustration of the historical development of damper capacity in dual mass flywheels.

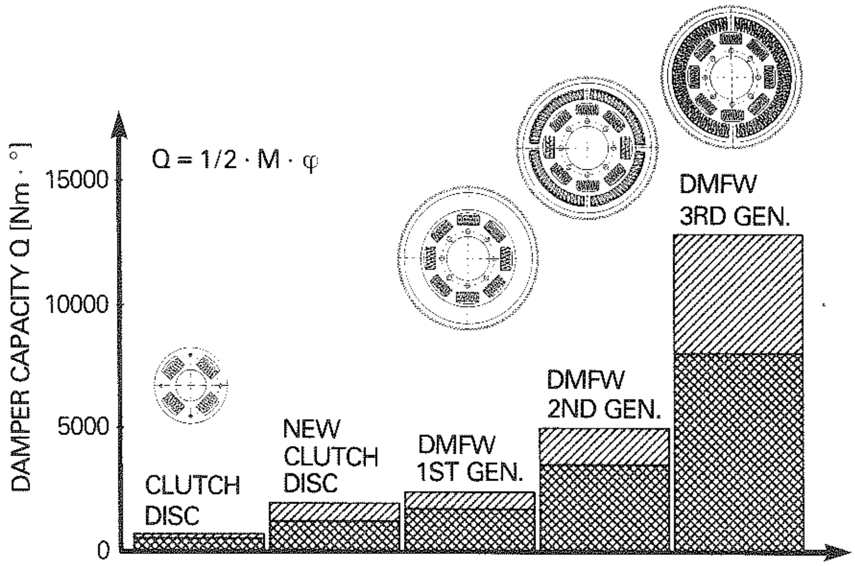

Figure 9: Torsion damper capacity

As the basis for our comparison, we chose a conventional torsion damper in a standard design that is still widely used. This damper is shown on the left of the chart in Figure 9, followed by a modern LuK long-travel clutch disc.

The first generation DMFW contained spring configurations like those used in conventional torsion dampers, with the coil springs arranged far to the inside. This design provided only a minimal space for coil springs, accompanied by modest damper capacity.

Without increasing the installation space required by the dual mass flywheel, we have been able to achieve a five-fold increase in damper capacity by:

- shifting the springs to the outside
- using a serial instead of parallel spring arrangement wherever possible
- using large-diameter coil springs.

Implemented Spring Rates

Equation (1) shown above indicates that damper capacity is proportional to the product of the max. torque and the max. wind-up angle. Hence, the lowest achievable spring rate depends on the available installation space as well as the maximum engine torque. Figure 10 shows the spring rates achieved for different DMFWs in different vehicles, plotted as a function of engine torque.

The differences between the three DMFW generations are very clear. The first generation of DMFWs was based on mass as the primary design consideration. These were special solutions that could not be employed for a wide range of applications because of their high spring rates. With the newly developed DMFWs of the third generation, we can solve the noise problems in virtually all vehicles.

In spite of variations in installation space and DMFW-diameter, spring rates lie very close to the line of best fit drawn on the graph. This graph also allows us to make a rough estimation of the slopes that we can expect to use in future vehicles.

Figure 10 shows the achievable spring rate as a function of engine torque. However, it is impossible to read the resonance speed from this graph. As noted above, this value depends on the number of cylinders and hence on the order of the primary excitation.

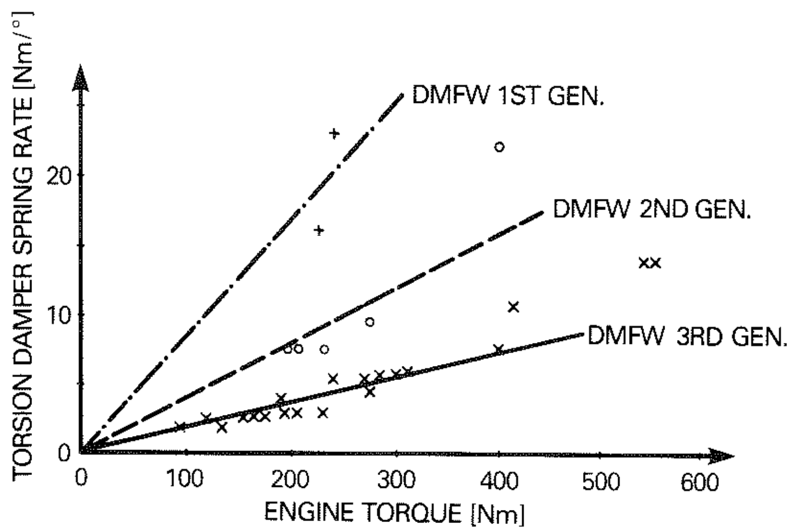

Figure 10: Spring-rate – engine torque

Figure 11 shows the resonance speeds actually obtained, plotted as a function of the maximum engine torque and the number of cylinders.

The vertical bars represent the critical speed ranges in the various gears for a given vehicle. The top of each bar corresponds to fifth gear and the bottom to first gear. This speed value can also be used for idle mode, provided that there is no separate torsion damper stage.

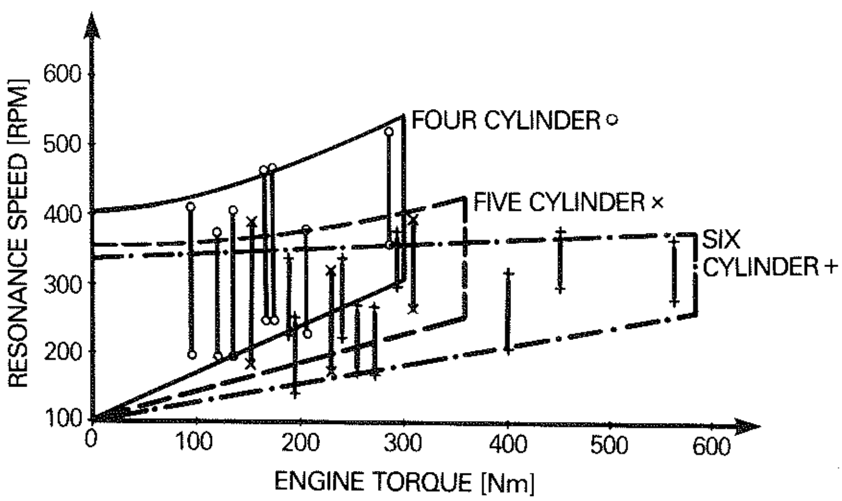

Figure 11: Resonance speed ranges for four, five and six cylinder engines

Distribution of Mass

In order to obtain the lowest possible resonance speed for a given total DMFW mass, theoretical considerations [5, 6] indicate a mass distribution of:

$$\lambda = J_1/J_2 \lesssim 1$$

Where J_1 is the sum of the torsional inertias associated with the primary flywheel ($\approx$ 60 %), the crankshaft plus the rotating components of the crank drive ($\approx$ 35 %) and the damper components, and where J_2 includes the torsional inertias for the secondary flywheel ($\approx$ 50 %), the clutch ($\approx$ 40 %), the clutch disc, damper components, and the transmission.

However, there are frequently a number of reasons why we have to deviate from this ideal value ($J_1 \lesssim J_2$). For instance, we must provide an acceptable thermal capacity for the clutch system on the secondary side of the DMFW. In addition, engine irregularity is determined by the inertia of the primary side of the DMFW. Maximum permissible engine irregularity is determined by the accessory drives attached to it. Figure 12 shows values that are currently realistic.

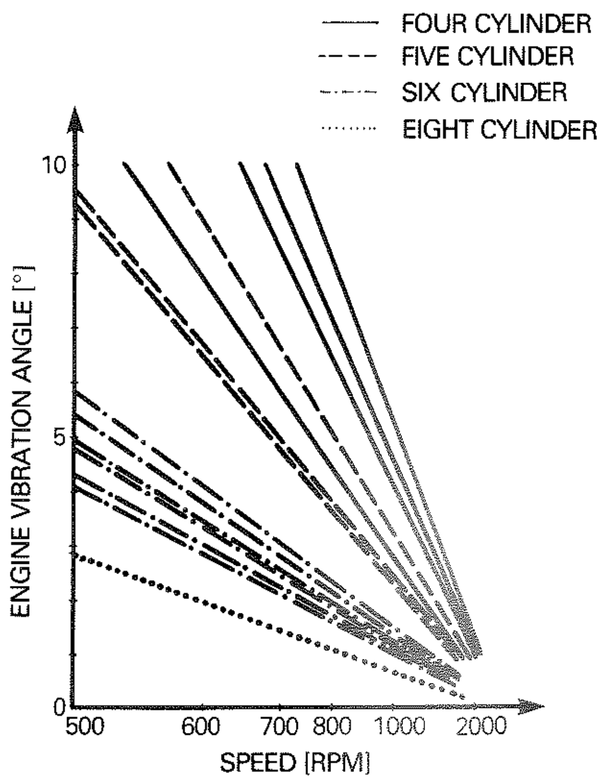

Figure 12:
Comparison of engine vibration angles

The engine vibration angle – measured at full load – is plotted as a function of engine speed for a wide variety of vehicles.

The speed scale is plotted quadratically in order to approximate straight lines. As anticipated, engine irregularity is higher for lower cylinder numbers.

It is impossible to make general statements about max. permissible values because these values are determined individually in each vehicle based on marginal conditions. In isolated cases, it is necessary to employ an additional ring-shaped inertia on the primary side in order to reduce irregularity. Even in such special cases, LuK has proven solutions available.

DMFW with Torque Limiter

Figure 13 shows the simplest modern DMFW design.

The primary flywheel is made of formed sheet metal parts. The main functional component of the DMFW consists of two arc springs, each featuring a second inside spring. These spring assemblies are arranged in two 170° semi-circles. The spring guide races are situated far to the outside of the flywheel in order to provide the most favorable effective diameter. A second important functional component, the torque limiter, is located further in toward the center of the flywheel. This device features a carefully designed slip coupling, which absorbs any torque peaks that occur. The maximum torque that can be transmitted by this component determines the stress limit for all other components.

The inside of the primary side of the DMFW is filled with grease. A membrane that requires very little axial installation space provides an effective seal to prevent this grease from contaminating surrounding components.

If the primary and secondary sides of the DMFW are rotated in opposition to one another, the lubricated arc-shaped coil springs glide in the spring guide races.

Because the springs are arced, the actuating forces on the two springs are not oriented in the same direction. As a result, the arc springs load the spring guide races with a radial force component. Centrifugal force increases this radial force of the arc springs. This generates friction between the spring coils and the spring races, which has a critical influence on the characteristics of the DMFW.

In order to study this important effect and to account for it with respect to the DMFW design, LuK designed a new kind of test stand for dynamic measurement of DMFW and conventional torsion damper characteristics at different speeds. The test stand can achieve wind-up angles of up to $\pm$ 50° with frequencies of 0 – 30 Hz and speeds up to 6,000 rpm.

This test stand has provided the impetus for DMFW development.

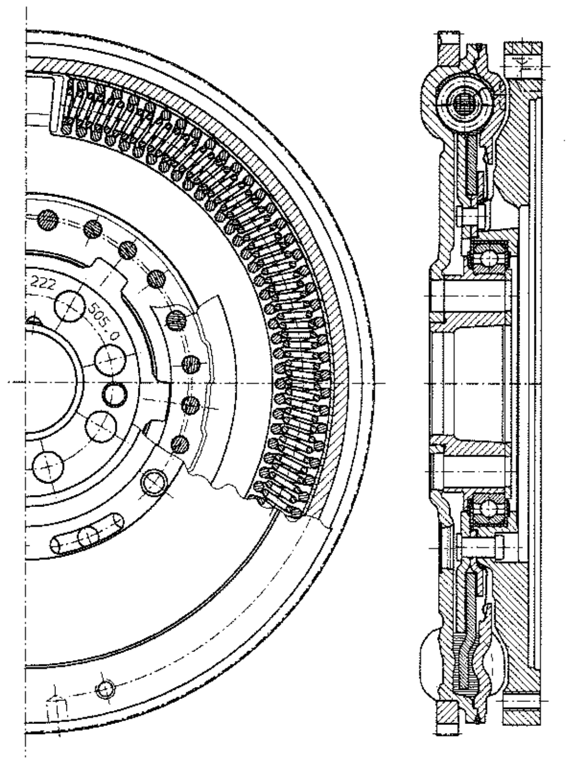

Figure 13:
DMFW with torque limiter

The measurements in Figure 14 show effects that are only obtainable with the dynamic torsion test stand.

We measured so-called partial loops. The test was run with an angle of ± 1°, working around an operating point of 150 Nm in drive mode at a frequency of 1 Hz. The superimposed speed was varied in stages from 0 to 2,000 to 6,000 rpm. The results indicate a massive change in the characteristic for the partial loops. The frictional hysteresis increases with the speed. At the same time, the characteristic becomes stiffer because part of the arc spring coils are blocked by static friction.

The stiffer spring rate increases the resonance speeds, and the increased frictional hysteresis increases damping. Both factors diminish the isolation effect of the DMFW. In most vehicles, this does not cause any problems because engine irregularity decreases at higher speeds, so trade-offs can be made with respect to the filtering function.

However, in individual cases, boom and rattle noises are still audible at higher speeds, thus requiring improved vibration isolation. LuK offers the DMFW with a decoupled inner damper as a solution for these vehicles.

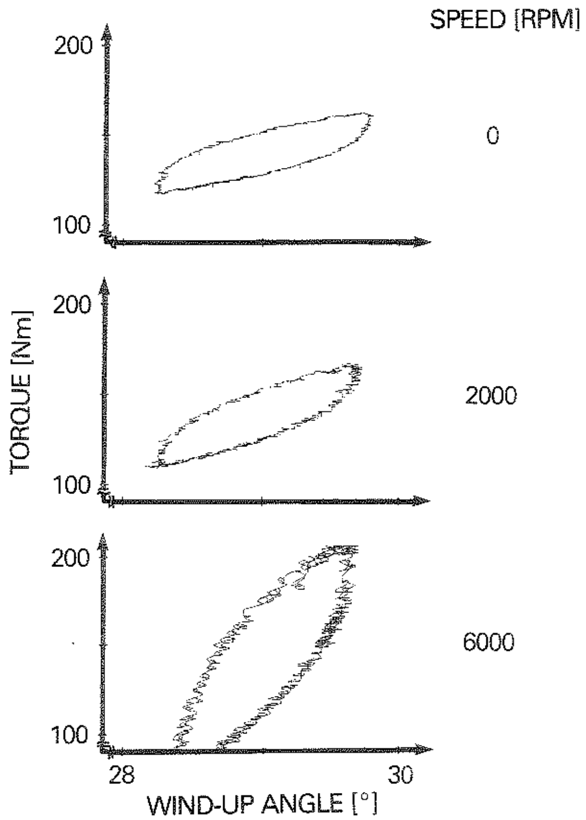

Figure 14:
DMFW with torque limiter partial loop in the drive range

DMFW with De-coupled Inner Damper

Once we had recognized the problem posed by centrifugal forces in certain vehicles, we attempted to decrease friction at high speeds. These efforts resulted in a DMFW like the one shown in Figure 15. This design uses two serial dampers, which, to a certain extent, serve two different functions.

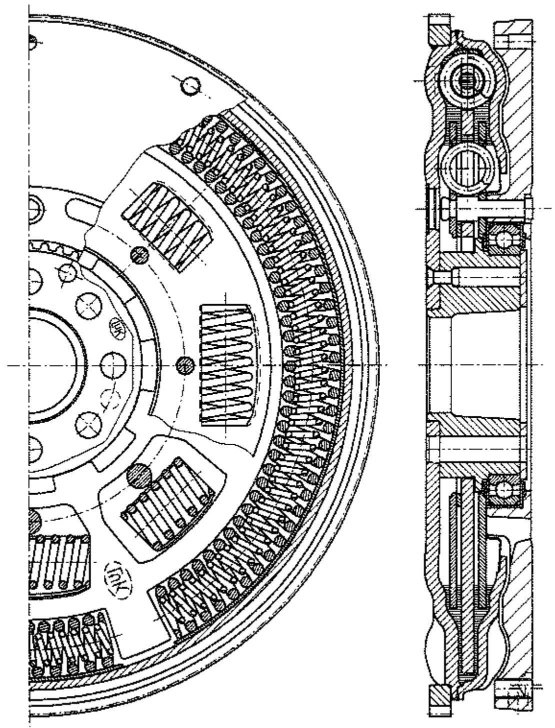

Figure 15:
DMFW with de-coupled inner damper

The outside damper design corresponds essentially to the large arc springs described in Figure 13, with their favorable effective diameter and their low spring rate. The spring rate and hysteresis of the arc spring damper are speed-dependent, as shown above. A second serial damper with short straight coil springs in conventional spring windows is installed on the inside of the dual mass flywheel in place of a torque limiter. Because only the end coils of the springs make contact and the actuating forces at the spring ends are oriented in the same direction, the internal friction of this damper is virtually independent of load and speed. Hence we can speak of it as being decoupled. Consequently, as shown in Figure 16, the frictional hysteresis and the effective spring rate no longer increase with rising speed. The good vibration isolation of the DMFW is maintained up to the highest speeds. As a result, this design is called a DMFW with de-coupled inner damper.

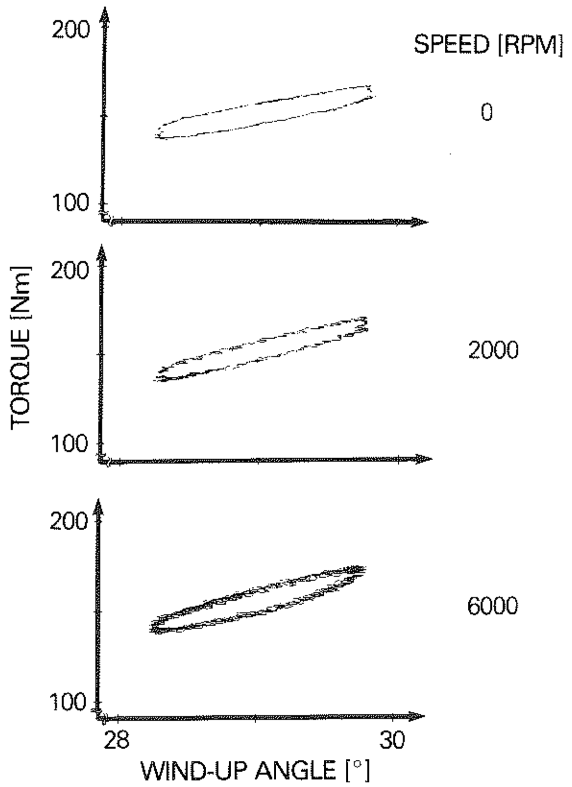

Figure 16:
DMFW with de-coupled inner damper; partial loop in the drive range

The static spring rate for the inner damper is naturally higher than that for the outer arc spring damper because of the lower achievable damper capacity. When combined with the secondary dual mass flywheel inertia attached to the transmission, this somewhat stiffer characteristic curve is quite sufficient to cope with higher speeds (see Figure 5). The very soft serial arc spring damper becomes effective at higher vibration amplitudes, for instance as a result of increased engine irregularity at lower speeds or also because of tip-in/back-out.

The models shown in Figure 17 summarize both basic systems used in current DMFW designs. The speed-dependent arc spring damper is combined with either a torque limiter or an inner damper and arranged between the two DMFW inertias.

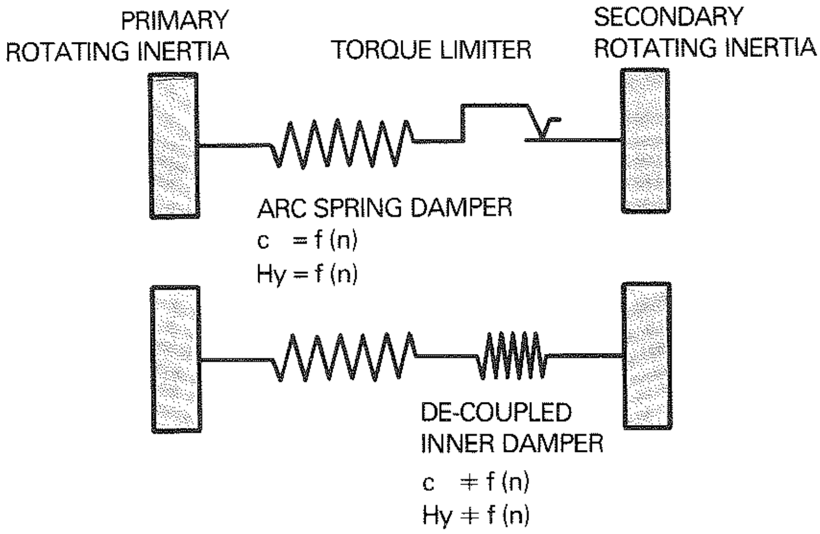

Figure 17: DMFW vibration model

One further closing comment on DMFW design must be noted here. The foregoing discussion of DMFW design options has presented the two subsystems – the torque limiter and the decoupled inner damper – as alternatives. It is important to note that the inner damper cannot fulfill the role of the torque limiter, which is to say, it cannot reduce excessive torques. It can be necessary to combine the two systems.

Actually, in many cases it is also possible to do without the torque limiter. The serial arrangement of the inner damper achieves additional reduction in the DMFW spring rate, accompanied by low resonance speeds and vibration amplitudes, as shown in Figure 6. The incidence of resonance operation, for instance when a vehicle stalls in gear at a traffic signal or during start-up, is diminished or even fully eliminated. In these cases, excessive torques are prevented entirely, rendering the torque limiter unnecessary.

LuK's experience, plus a few tests on our engine tests stands and in special vehicles, provide the necessary information for determining whether a torque limiter is required. These tests can be limited to the resonance range, that is to start-stop operation and driving at extremely low engine speeds.

Tip-in / Back-out Performance

Up until now, this discussion has dealt with the 2nd natural mode of the vibration model shown in Figure 3, which is responsible for resonance conditions in the DMFW. The following section of this presentation will deal with the influence of the new DMFW generation on surging and tip-in/ back-out performance. These operating conditions are described by the 1st natural form of the basic model.

As shown in Figure 3, the central inertia, that is the secondary flywheel mass, is located very close to a vibration node and is therefore hardly involved in the vibration. The total effective spring system situated between the engine and the vehicle can be viewed as a serial arrangement consisting of the torsion damper and the drive train. This observation reveals that the only way to affect tip-in/back-out performance is to decrease the torsion damper spring rate until it is in the order of magnitude of the drive train spring rate.

It is impossible to achieve the required spring rate using a conventional torsion damper. By introducing the super-long-travel DMFW, we have been able to obtain values below the drive train spring rate.

This applies particularly for 3rd, 4th and 5th gears because the spring rate of the drive train is proportional to the square of the drive line ratio and thereby significantly higher in higher gears. Hence it is that much simpler to remain below this value and thus to improve surging.

We have to consider tip-in and back-out in the lower gears to be the critical operating conditions. We have plotted measurements and calculated values for second gear in order to illustrate these influences.

Let us look again at Figure 4. The lower section of the graph shows the resonance ranges for "surging vibrations" in the first natural mode as a function of the torsion damper spring rate. The resonance speed depends greatly on the gear selected. This factor is reflected in the very broad fields for the different cylinder numbers. It is impossible to achieve any significant influence on the surging resonance speed until we get below a 10 Nm/° rate in the torsion damper. This point comes at the beginning of the upper limiting curve for the 5th gear.

Until now, only the influence of the spring rate on the resonance speed has been viewed with respect to surging natural mode. If we use a correspondingly low spring rate, it is possible to shift this speed, which means we can change the frequency at which surging becomes perceptible in the vehicle during tip-ins and back-outs. However, this factor alone will not produce any improvement in tip-in/back-out performance. In fact, theoretically speaking, the lower spring rates can even result in deteriorated tip-in/back-out performance. Consequently, in addition to the spring rate, the damping performance of the super-long-travel DMFW is very important.

The computer simulation shown in Figure 18 is designed to clarify the relationships involved here. These calculations were based on a typical compact vehicle in which vehicle surging – as described by the engine and transmission speeds – is calculated based on a sudden tip-in/back-out. At point 1, the driver steps down full on the gas, and at point 2 he lets up on it suddenly. The graphs show variations in the hysteresis, which acts as a damping factor in the DMFW.

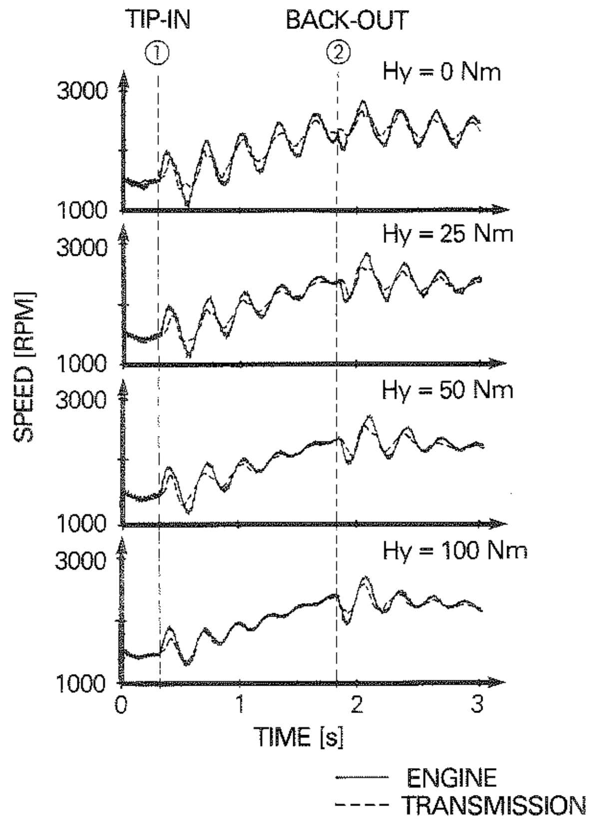

Figure 18:

Tip-in/back-out calculations with hysteresis variation for C_{TD} = 2 Nm/°

The top section of the graph shows the situation for a DMFW that displays virtually no friction. Damping in the drive train is limited. The vibration excited by the tip-in and the later back-out only decays slowly.

We do not achieve good damping performance until we superimpose an appropriate frictional hysteresis on the very long torsion characteristic. Consequently, we can't allow the damping factor to approach zero for the kind of wide wind-up angles that occur in the DMFW during tip-in and back-out. This consideration runs counter to the requirement for the best possible isolation of high-frequency engine excitation in the range above the natural frequency.

However, the following measurements shown in Figure 19 indicate that LuK's super-long-travel DMFW exactly meets these requirements, even though they appear at-first-glance to be contradictory. Partial loop measurements using small wind-up angles such as those that occur during constant load condition show a low frictional hysteresis. This indicates low damping accompanied by correspondingly good isolation. However, when we measure the DMFW characteristic over the entire wide wind-up angle, we discover a high hysteresis. This very high DMFW friction is effective over a very wide wind-up angle and is capable of effectively damping objectionable surging vibrations during tip-in/back-out.

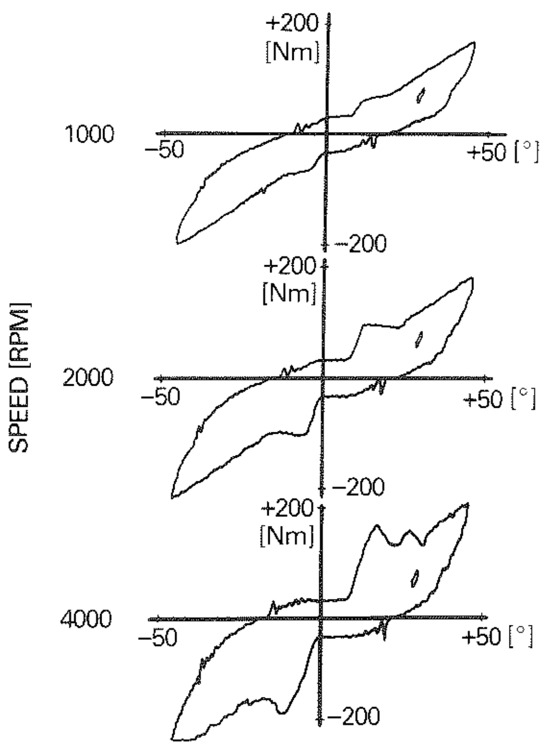

Figure 19:
Dynamic measurements tip-in/back-out and partial loop

This damping performance is generated primarily by the fact that the arc springs are radially supported against the spring guide races. Furthermore, we can also introduce an additional controlled frictional hysteresis by adding a friction control plate in order to tailor the DMFW characteristic to the requirements of the vehicle in question. Due to these design features, the friction generated in this system is not significantly dependent on centrifugal force.

The interaction of a DMFW with a very low spring rate and an appropriately designed frictional hysteresis for damping purposes improves vehicle performance during tip-in and back out. The tip-in/back-out calculations shown in Figure 20 again clearly illustrate the influence of the torsion damper spring rate in this context.

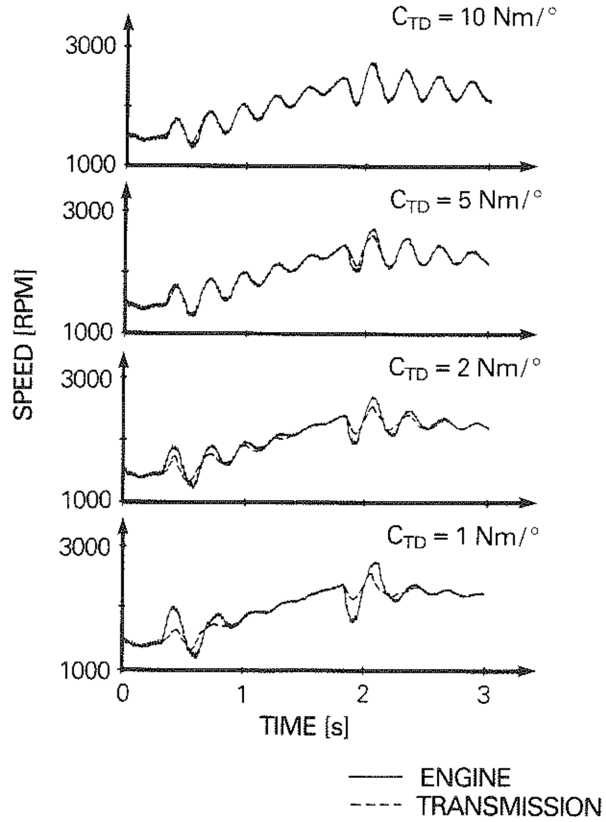

Figure 20:
Tip-in/back-out calculations with spring rate variation for Hy = 75 Nm

Damping accelerates as the torsion damper spring rate decreases, although the first peak engine amplitude increases as a result of good decoupling.

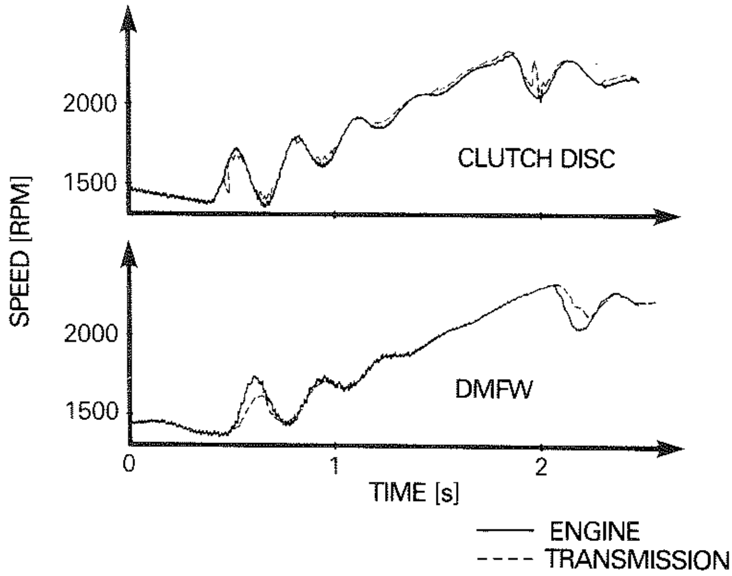

Figure 21: Tip-in/back-out measurements 2nd gear with clutch disc and DMFW

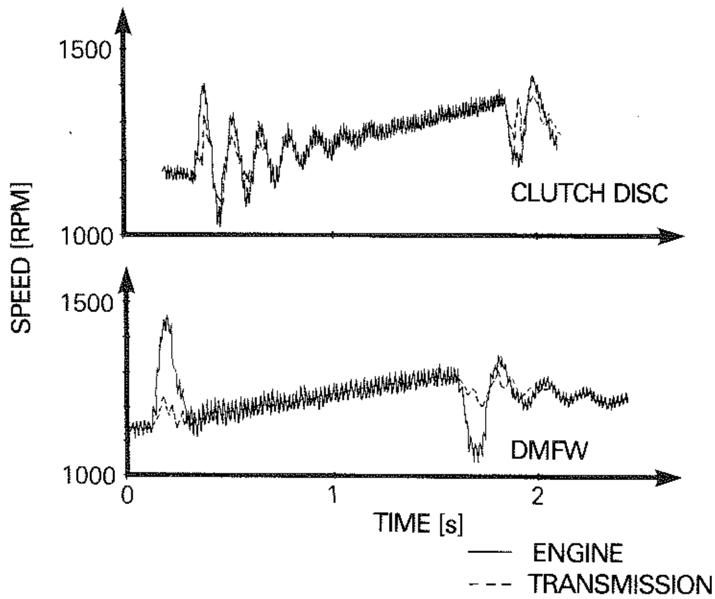

Figure 22: Tip-in/back-out measurements 4th gear with clutch disc and DMFW

In the case shown (Figure 21), the 2nd gear torsion damper spring rate C_{TD} and the drive train spring rate C_A are of the same magnitude, at 2.5 Nm/° each. Surging amplitudes decay rapidly with the DMFW.

In 4th gear (Figure 22) with a drive train spring rate of 7 Nm/°, we register no subsequent bounces, while several bounces were still evident when we conducted the same measurement for the clutch disc.

Summary

Thanks to on-going, consistent development of the DMFW, LuK can now offer a dual mass flywheel solution to improve noise comfort and driving performance for almost all vehicle models and for a wide variety of engines, including four cylinder engines.

LuK's newest DMFW designs demonstrate that resonances can be shifted far out of the driving range and that transmission rattle and boom can be effectively eliminated. We have been able to prove this trend on many vehicles, using measurements, theoretical calculations, and subjective evaluations. We are even able to achieve positive results combatting vehicle surging.

Increased production and unbroken interest confirm that the dual mass flywheel is an ideal solution for increasing noise comfort and improving the future marketability of modern vehicles. LuK is in a position to commit its entire DMFW know-how to an individual customer's needs with respect to development, production and quality assurance in order to ensure a mutually profitable future through improved technology.

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