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Foreword

Innovations are shaping our future. Experts predict that there will be more changes in the fields of transmission, electronics and safety of vehicles over the next 15 years than there have been throughout the past 50 years. This drive for innovation is continually providing manufacturers and suppliers with new challenges and is set to significantly alter our world of mobility.

LuK is embracing these challenges. With a wealth of vision and engineering performance, our engineers are once again proving their innovative power.

This volume comprises papers from the 7th LuK Symposium and illustrates our view of technical developments.

We look forward to some interesting discussions with you.

Bühl, in April 2002

Helmut Beier

Helmut Beier

President
of the LuK Group

Content

DMFW – Nothing New?

Ad Kooy
Achim Gillmann
Johann Jäckel
Michael Bosse

A Critical Look at the Arc Spring DMFW

Things have become tense in the DMFW market. Alternative designs to the arc spring DMFW have recently gone into production. It is well known that competition brings life to business, and many innovative solutions have started at LuK as well.

We will not go into an exhaustive discussion of the general principle of the arc spring DMFW (figure 1) at this point. This can be found in previous colloquia [1], [2], [3].

- The wide distribution of the DMFW is due not least to the arc spring, which is in many respects very functional and cost-efficient. Its strengths include:
- High friction at large angles of rotation (start-up) and low friction at small angles of rotation (drive)
- Low spring rate due to good, flexible use of space

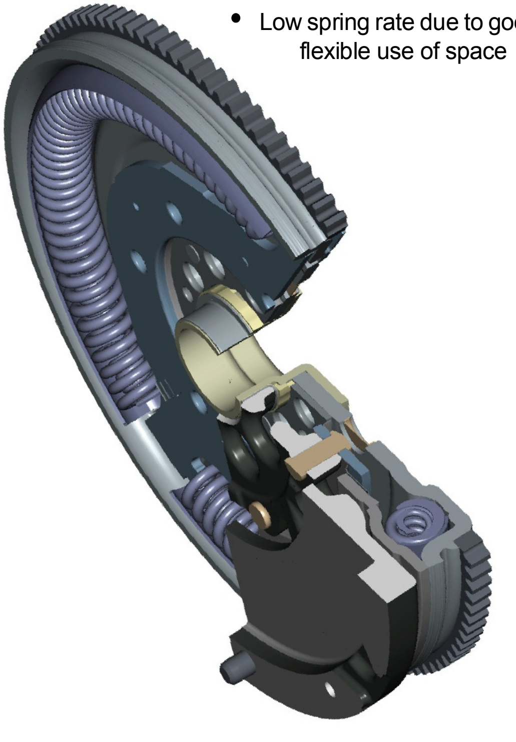


Fig. 1: *Typical DMFW*

Integration of a soft stop (damping spring) is possible.

But of course, every design also has its drawbacks. Additional friction is created between the arc spring and shell due to the centrifugal force. The isolation of the transmission in drive, which is the main reason why the DMFW is used ,would be better without this additional friction. In addition, this friction can cause the arc spring to remain preloaded in the channel. If a flange strikes a preloaded arc spring, it acts as a rigid wall. The vibration isolation is then lost.

In the course of this presentation, we will focus mainly on measures that improve several operating conditions. The discussion will be based on the following two operating conditions:

- Isolation in drive, the main function of the DMFW
- Back-out, a typical example of a condition that is affected by the preload of an arc spring in the channel.

Physical Context

For an easier understanding of the factors that influence arc springs behaviour under centrifugal force the physical context is used.

Centrifugal force = $m \cdot \omega^2 \cdot r$
Torque to move arc springs = $\mu \cdot m \cdot \omega^2 \cdot r^2$
Torque to move arc springs at 3000 min^{-1}
$0.15 \cdot 1$ kg $(300\ rad/s)^2 \cdot (0.12\ m)^2 = 200$ Nm

At 3000 min^{-1}, the torque required to move the arc spring is around 200 Nm. If such an arc spring is loaded and pressed against the stop, even after unloading its preload remains 200 Nm. As one can imagine, typical coast torques of around 40 Nm can no longer resiliently actuate the arc springs. (Countermeasures can, however, be taken by changing the flange geometry or the inside spring!).

But even at, for example, 1500 min^{-1}, an especially interesting range for isolation in drive, this friction is 50 Nm, and cannot be ignored.

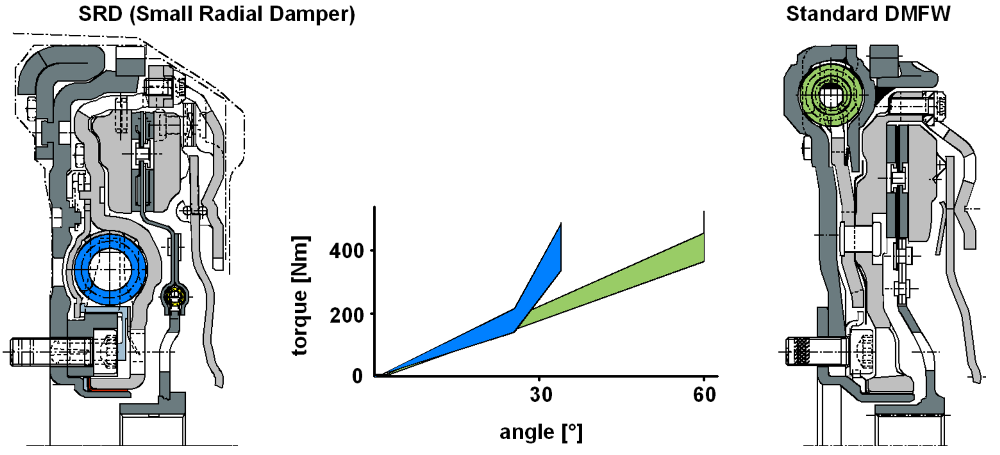


Fig. 2: Design Comparison between SRD and DMFW

The contact radius between the arc spring and arc spring shell has the greatest effect. If the damper is shifted radially inward, as has been done, for example, in the DFCII [2] or in its more modern version, the SRD (Small Radial Damper, figure 2), the friction decreases quadratically. Such concepts can largely resolve problems that are due to centrifugal force, but have not as yet been implemented. This is because the spring capacity lost limits the engine torque that can be covered.

There is not much potential for improvement of the weight of the arc springs, since lower weight necessarily means less effective volume and thus a poorer degree of coupling. Lower weights are easiest achieved with a higher allowable stress. The potential therefore is most likely almost exhausted. It is also possible to achieve a 20% weight reduction by eliminating the inside spring. To maintain the stop torque, the wire thickness of the outside spring must be increased. The resulting stiffness is higher than the original spring combination. The isolation in drive, however, remains the same, because the lower weight causes lower friction, which compensates for the increased stiffness. In any case, this can only be implemented if the engine is not critical for start-up.

Reducing Friction

It is tempting to achieve the functional improvements by reducing the friction between the arc spring and the shell. Then, of course, the friction necessary for start-up is reduced, but this can be compensated for by using friction control plates or by using two-stage arc springs with their favourable low first rate .

The coefficient of friction can be reduced in theory by changing each of the three components of the tribosystem: the arc spring shell, the grease and the arc spring.

Coatings can be used to influence the behaviour of the arc spring shell. Trials with very different coatings showed, however, sufficient wear resistance cannot yet be achieved. These results will not be discussed here.

In addition to the standard grease a friction-optimised type of grease is now available. Further improvements appear to be difficult. Interestingly, it was shown that increasing the amount of grease does bring an improvement in the static to sliding friction ratio.

We will discuss the effect of guide shoes, which change the friction contact from steel/steel to polyamide/steel, for the back-out. Figure 3 shows a measurement of back out.

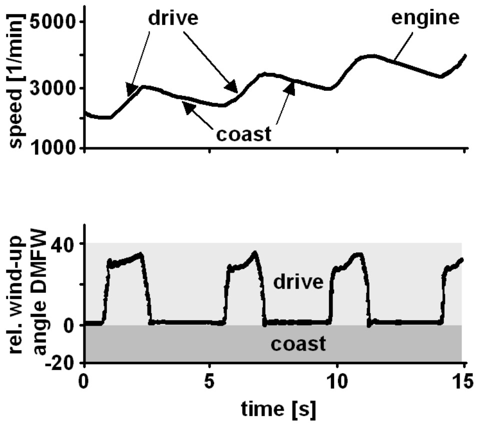


Fig. 3: *Tip-in/Back-out for an Arc Spring DMFW 2nd Gear*

The behaviour of the arc springs during the alternating drive and coast phases, can be interpreted with the relative angle of rotation . It shows, the arc spring is moved in the drive phase, but hits a 'wall' in the coast phase. In the vehicle under study here, this impact was perceptible and was cause for complaint.

In order to understand the behaviour of the DMFW, the tip-in/back-out behaviour was simulated under speed on the test stand (figure 4). For visualization, figure 5 shows the position and the preload distribution (width of the arrows) when the flange strikes the arc spring during back-out.

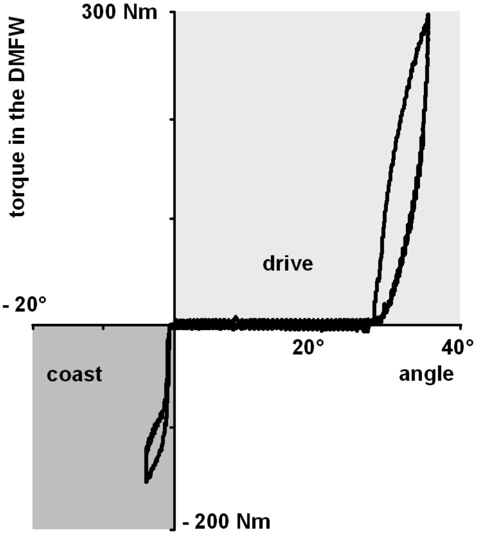
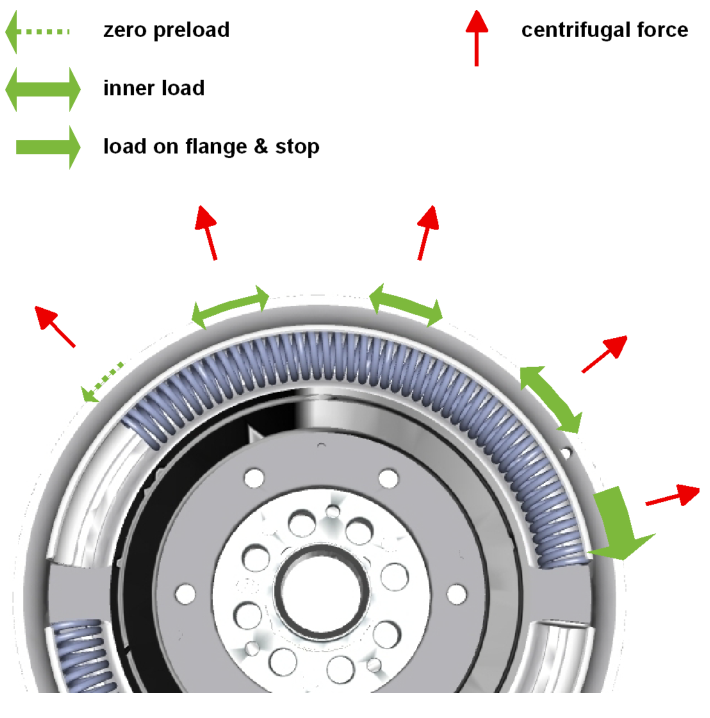


Fig. 4: *Characteristic at 3000 min^{-1} for an Arc Spring DMFW*



After loading to 300 Nm in drive and subsequent unloading, the arc spring is held at 28° .It is then preloaded in the channel. Before hitting the 'wall' of 120 Nm in coast, it must traverse a free travel of 28°. During this travel, the engine, together with the primary flywheel, would gain momentum before hitting the 'wall'. The low hysteresis in coast causes a nearly complete elastic pulse exchange with the 'wall'.

A clear reduction of the coefficient of friction would help in many ways: First, it causes a smaller free angle, so that the engine can build less momentum. Secondly, when the arc spring is not preloaded as high, the 'wall' is lower and thus easier to move. In fact, the use of slide shoes provides a clear improvement.

Fig. 5: *Position and Load of the Arc Spring when the Flange Strikes the Arc Spring during Back-out*

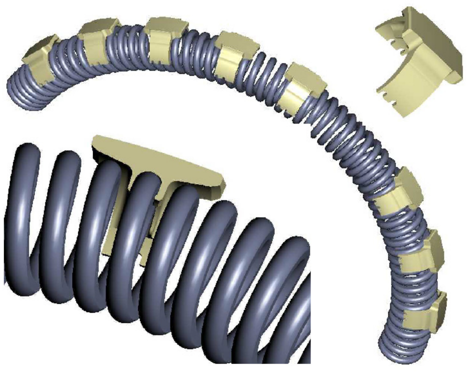


Fig. 6: *Arc Spring with Plastic Slide Shoes*

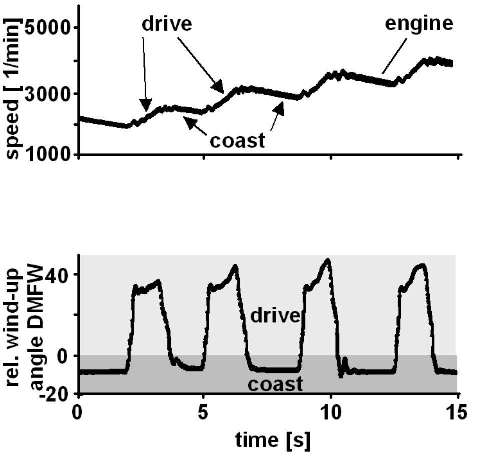


Fig. 7: *Tip-in/Back-out for a Slide Shoe DMFW in 2nd Gear*

The slide shoes are simply clipped to the coils of the arc spring (figure 6), allowing all coils to remain active. When the arc spring is compressed to block, there remains enough space between the coils such that impact loads are not transmitted directly via the shoes.

The improvement brought about in this way is astounding (figure 7). Now the preload can be overcome in coast. The arc springs are moved by approximately 10°.

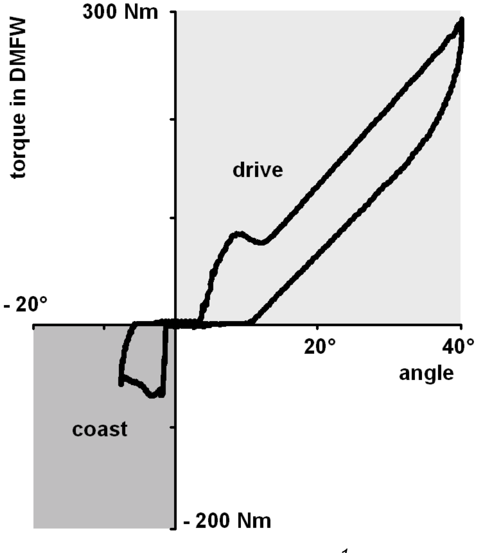


Fig. 8: *Characteristic at 3000 min^{-1} for a Slide Shoe DMFW in 2nd Gear*

This is also reflected in the characteristic curve measured on the test stand (figure 8). The arc spring has a reduced free angle of 12° (less momentum), the 'wall' is clearly reduced(easier to move) and the hysteresis is high (good impact damping).

The noticeable improvements in the tip-in/back-out load cycle have unfortunately not led to a clear improvement in drive isolation (figure 9).

This is also reflected in the partial loops, which represent drive. No reduction in friction was achieved (figure 10).

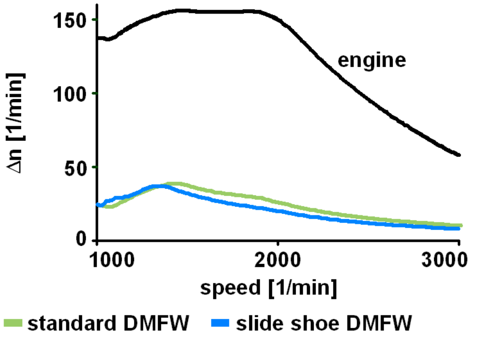


Fig. 9: *Slide Shoe DMFW, Measured Isolation in Drive*

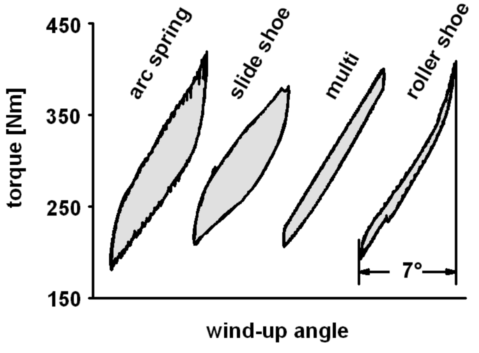


Fig. 10: Partial Loop Comparison at 1000 min^{-1}

The reasons for this unexpected behaviour are not yet entirely understood: It is certainly not just the coefficient of friction which plays a significant role, but also the static to sliding friction ratio.

The Search for Better Isolation in Drive

The main function of the DMFW is, as mentioned before, to provide the best possible decoupling for the transmission. In order to cover the ever-increasing engine torques within the same bell housing, the characteristic curves must be made steeper. This naturally leads to a deterioration of the isolation in drive. Since higher engine torques are also accompanied by higher irregularities, the damper needs to provide even greater performance to keep the irregularities, and thus the noise level in the transmission low.

This topic comes up again and again. As early as 1990 [1], it was shown how a frictionless inner damper can improve isolation in drive. Such DMFWs have been in production for many years. The following design allows an even lower spring rate.

The Multi-DMFW

The multi-purpose DMFW (multi-DMFW) uses a new inner damper (figure 12).

Two compression springs are located behind one another actuated serially. The tabs of a surrounding special ring, whose job it is to take up the unavoidable centrifugal forces without friction, is located in between these.

This damper differs little in its cross-section from a conventional inner damper. It requires similar space (figure 13). It provides a considerable angle of rotation of 18° at a stop torque of 400 Nm. This can cause a marked improvement to the drive isolation (figure 11).

Since this inner damper takes a considerable part of the work away from the outer arc spring damper during the start-up phase, the latter can also be designed stiffer for critical 4-cylinder diesel applications. This allows a new arc spring arrangement, the ‘four-part division’. The standard arc springs are divided, and the flange is placed between the two parts of the arc spring instead of above the stop pins (figure 12). One arc spring is now only active in drive and the other only in coast.

The flange can then no longer strike a preloaded arc spring, the impact in coast during backout is eliminated and the isolation during coast is excellent.

Since in the standard arrangement, only a part of the arc spring is active in drive, the drive isolation is affected little by the ‘four-part division’. Different layouts of the arc springs for drive and coast provide further potential for optimisation.

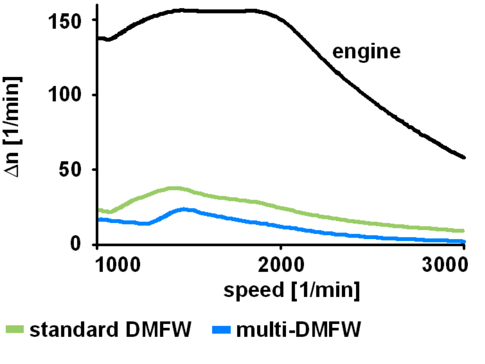


Fig. 11: Multi-DMFW (Simulation of Drive Isolation)

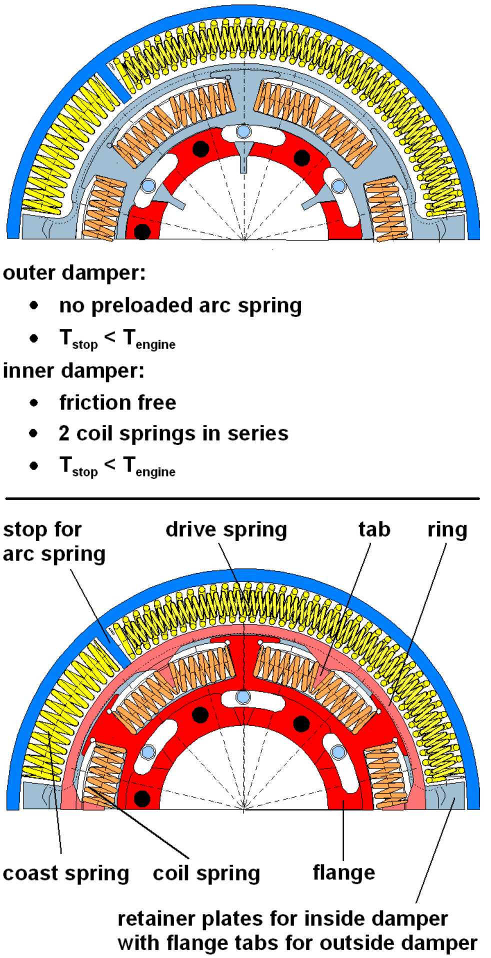


Fig. 12: *Multi-DMFW, with Closed Inner Damper shown above, Opened Inner Damper shown below*

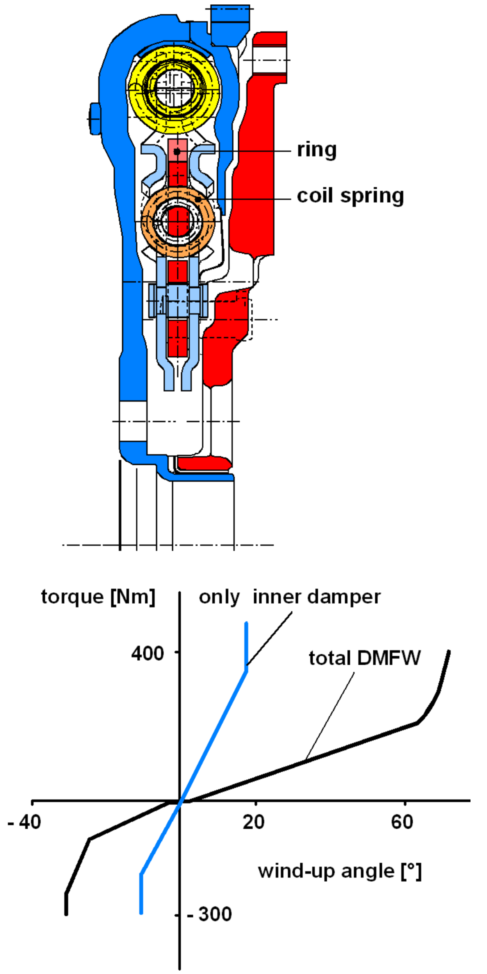


Fig. 13: *Multi-DMFW, Cut and Characteristic*

DMFW with Centrifugal Pendulum-Type Absorber

Attempts have been made time and time again to use a centrifugal pendulum-type absorber [4]. In theory, this could eliminate the primary excitation forces, the 2nd order for 4-cylinder engines. Intensive analysis has shown, however, that few such pendulums could be accommodated in the existing space in the clutch bell housing. But how would it be if the pendulum were combined with the DMFW? There are two conceivable designs (figure 14). If the centrifugal pendulum-type absorber is arranged on the primary side, it also works for the accessory drive, but requires a relatively large mass (3-5 kg).

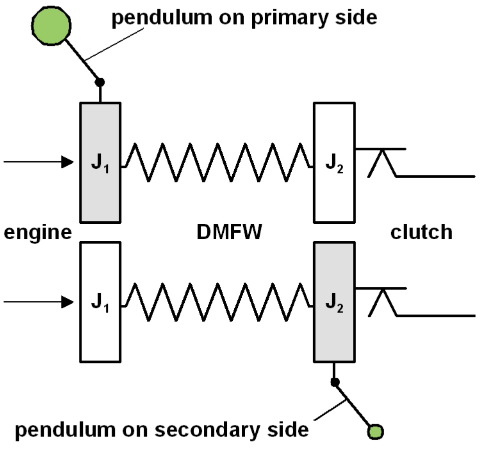


Fig. 14: Options for Combining the DMFW with a Centrifugal Pendulum-Type Absorber

This solution is being investigated in a co-operative development between LuK and Vibracoustic. This will be reported on later.

If the centrifugal pendulum-type absorber is arranged on the secondary side, it does not work for the accessory drive, but the mass can be reduced to around 1 kg. The additional space requirements are comparable to those of a conventional inner damper. If the pendulum is placed inside the primary flywheel, the DMFW grease can provide the required lubrication without on-costs (figure 15).

To make the large pendulum angle manageable below idle speed, the frequency of the centrifugal pendulum-type absorber is tuned slightly above the 2nd order. This also makes it distinctly less tolerance-sensitive (figure 16).

The gain in drive isolation is slightly less, but it still represents a 20% improvement (figure 17).

In a 4-cylinder with very high irregularities and higher resonance speeds, the effect is less (figure 18), if the pendulum mass cannot be raised accordingly. It remains to be seen whether and for what applications the centrifugal pendulum-type absorber can be used.

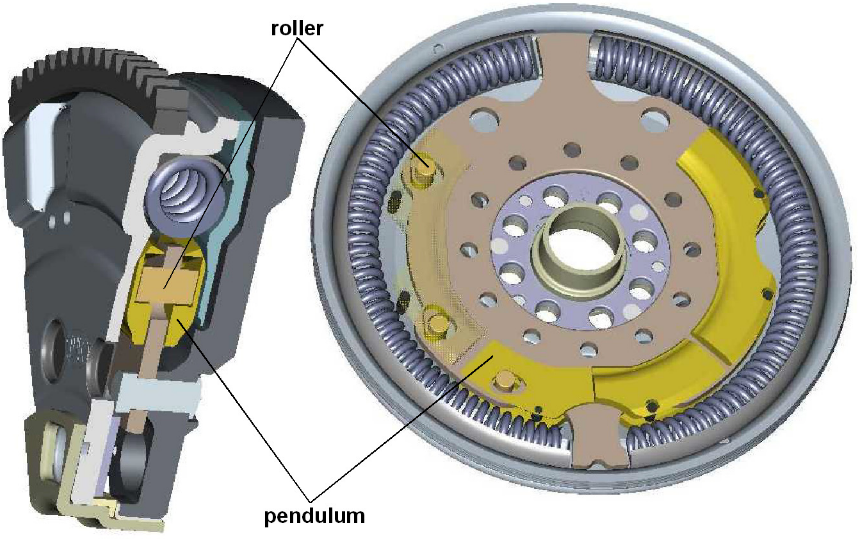


Fig. 15: DMFW with Secondary Side Centrifugal Pendulum-Type Absorber

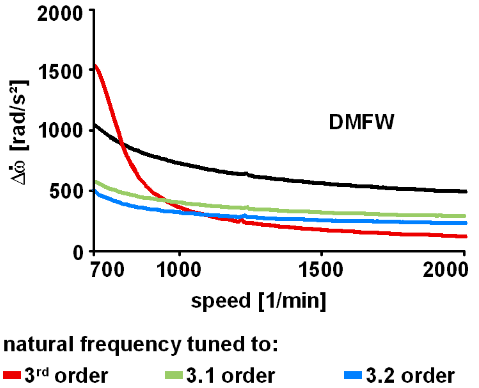


Fig. 16: Drive Isolation for Secondary Side Centrifugal Pendulum-Type Absorber (6-cylinder)

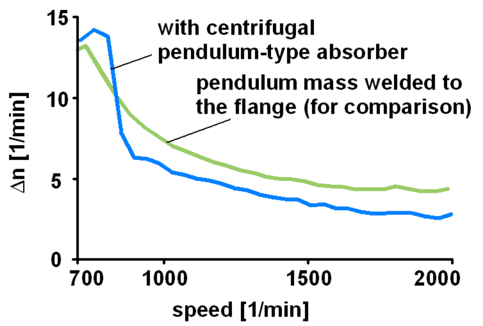


Fig. 17: Measured Drive Isolation for Secondary Side Centrifugal Pendulum-Type Absorber (6-cylinder)

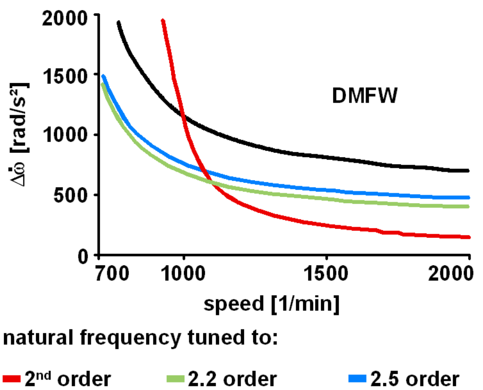


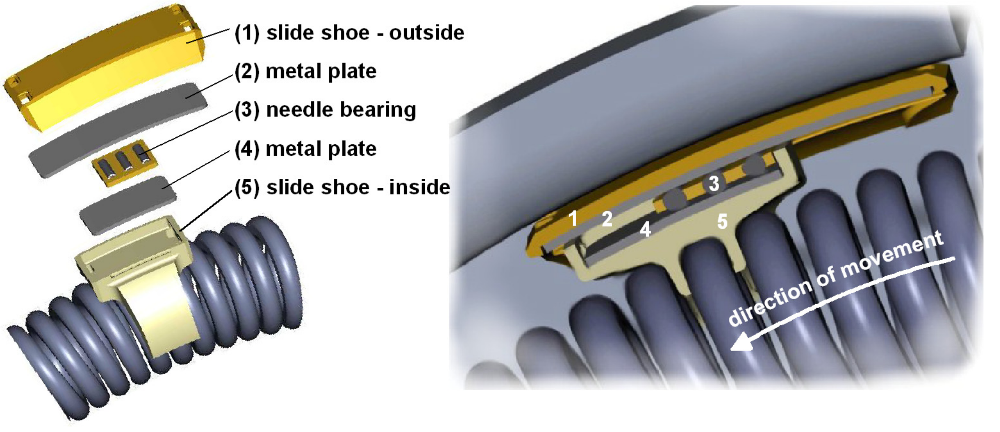
Fig. 18: Drive Isolation for Secondary Side Centrifugal Pendulum-Type Absorber (4-cylinder)

The Roller Shoe DMFW

There are also innovative ways to improve the drive isolation over the guide shoe DMFW: the roller shoe DMFW. The core of this idea is to eliminate the friction between the primary flywheel and the shoes by using a needle bearing in the shoes.

The design is depicted in figure 19. While the outer guide shoe provides the usual friction contact with the primary flywheel, the needle bearing, which runs on hardened strips of sheet metal, decouples the inner guide shoe. Since this type of linear bearing can only be designed for a restricted angle of a few degrees, the outer guide shoe takes on greater angular movements. This causes a separation of functions between the drive, characterised by small angular movements, for which friction is undesirable, and start-up, characterised by large angular movements, for which friction is desirable.

The space for the arc spring is actually somewhat reduced, but this is by far compensated for by the friction-reducing needle bearing. The angle within which the needle bearing is active can be ±3.5°, so that the drive range for a 4-cylinder diesel from about 1500 min^{-1} can be covered. This principle can bring about a 40% improvement in the drive isolation (figure 20). Below 1500 min^{-1}, where the vibration angle is over ±3.5°, the rollers are only effective to a limited extent. The improvement in isolation is therefore limited. The required space naturally reduces the diameter of the arc spring somewhat. The average spring rate, which is important for start-up, only increases by about 10%.



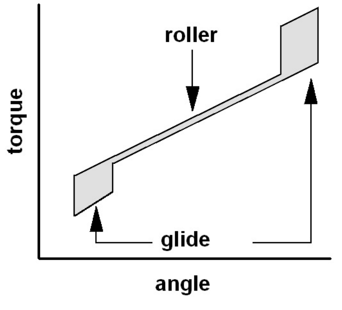


Fig. 19: DMFW with Roller Shoes

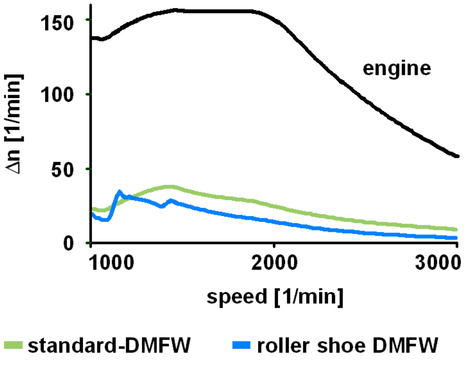


Fig. 20: Drive Isolation Increase with a Roller Shoe DMFW (Simulation)

Summary

DMFW – Nothing New? The answer to this question is clear. Significant improvements are still being made even 16 years after its introduction. Thus, the DMFW will be in a position to meet increased customer requirements in the future, even at higher engine torques.

Together with the automobile manufacturers, suitable solutions for all applications can be developed.

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