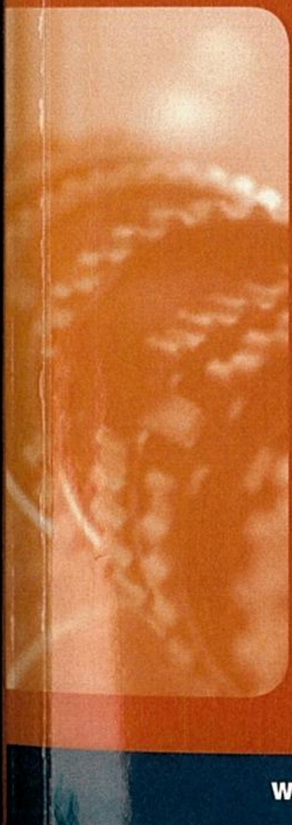
7th International CTi Symposium

Innovative Automotive Transmissions

Volume 2 (3 December 2008)

Gear up for the Transmission Business!



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Grußwort

Sehr geehrte Damen und Herren,

in den letzten zehn Jahren wurden im Bereich der Fahrzeuggetriebe vom Kunden wahrnehmbare und messbare Fortschritte erzielt, die bedeutend größer sind als in den 50 Jahren zuvor. Sind Gespräche über sieben, acht und mehr Gänge heute völlig selbstverständlich, waren diese für Pkw-Stufengetriebe noch vor zehn Jahren so gut wie indiskutabel. Das Getriebe leistet damit einen signifikanten Beitrag zur besseren Ökologie und Ökonomie sowie zum Fahrspaß und zur Sicherheit unserer Fahrzeuge.

Die Elektrifizierung des Getriebes hat im Kontext der Diskussion über nachhaltige Mobilität längst begonnen. Sowohl Fahrzeuge mit Hybrid- oder Elektroantrieb, also auch solche mit konventionellem Antriebstrang, benötigen Getriebe mit idealen Eigenschaften. Wichtig sind hierbei Wirkungsgrad, Leistungsgewicht, Bauraum, Komfort, Qualität, Bedienung und Kosten. Die effiziente und effektive Entwicklung und Produktion dieser Getriebe erfordert optimale Werkzeuge und Prozesse. Diese Themen bilden den Kern des diesjährigen Symposiums.

Erstmals werden im Rahmen des Symposiums einen Einblick in die wichtigen Zukunftsmärkte Asiens gegeben.

Die Podiumsdiskussion wird sich wieder mit den Aussichten auf die künftige Getriebeentwicklung beschäftigen. Diesmal vor dem Hintergrund steigender Energie- und Ölpreise, der unterschiedlichen Fahrzeug- und Antriebskonzepte sowie der Kaufkraftentwicklung in den verschiedenen Märkten.

Mit seinem umfangreichen Programm und der angeschlossenen Transmission Expo bietet das 7. Internationale Getriebesymposium erneut ein wichtiges Forum zum Meinungs- und Erfahrungsaustausch für internationale Getriebe- und Antriebsfachleute.

Ich freue mich über Ihre Teilnahme und wünsche Ihnen viele nutzbringende Gespräche und zahlreiche lohnende Anregungen.

Mit freundlichen Grüßen

Prof. Dr.-Ing. Ferit Küçükay
Geschäftsführender Leiter
Institut für Fahrzeugtechnik
Technische Universität Braunschweig

Welcome Address

Dear Sir or Madam,

Measurable progress has been achieved in the last ten years in the field of automotive transmissions which can be perceived by the customers and which is much more significant than in the 50 years before. Talks about seven, eight or even more gears are possible today, but could not be considered for stepped transmissions in passenger cars ten years ago. The transmissions thus contribute considerably to better ecology and economy as well as to driving fun and automotive safety.

The electrification of the transmission has already begun with regard to the discussion about sustained mobility. Vehicles with hybrid or electric drive as well as those with conventional drive trains need ideal transmissions. Efficiency, weight-to-power ratio, space requirements, comfort, quality, control and costs are important here. The efficient and effective development and production of these transmissions require optimal tools and processes. This year's symposium focuses on these topics.

The Symposium furthermore will give an insight into the important Asian future markets for the first time.

The panel discussion will again deal with the chances of future transmission developments, this time against the background of increasing energy and oil prices, diverse vehicle and drive concepts and the development of purchasing power in different markets.

With the extensive programme and the accompanying Transmission Expo, the 7th International Transmission Symposium is again an important forum for international transmission and drive train experts to exchange opinions and experiences.

I am looking forward to your participation and I wish you a lot of fruitful talks and beneficial motivations.

Best regards

Prof. Dr.-Ing. Ferit Küçükay
Managing Director
Institute of Automotive Engineering
Technical University of Braunschweig

Forum H: Hybrid

Forum I: Transmission Components; Commercial Vehicles

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I1

Das Fliehkraftpendel, die Beruhigung für den Antriebsstrang

The pendulum absorber calms the driveline

by

Dr. Wolfgang Reik
Executive Vice President LuK Group R&D
LuK GmbH & Co. oHG
Germany

Valeo Exhibit 1106, pg. 4

Presenter's Name	Dr.-Ing. Wolfgang Reik	
Company	LuK GmbH & Co. oHG	
Professional Career	1968	Studies of physics at the University of Karlsruhe
	1979	Engineering degree from the Institut für Werkstoffkunde I (Materials Engineering Institute 1) at the University of Karlsruhe
	1979	Dr. Reik joined LuK in Bühl, a supplier to the automotive industry and a specialist for clutches and passenger car drivetrain solutions. Dr. Reik held various entry-level positions at LuK and was later appointed manager of the testing department.
	1989	Executive Vice President for Research & Development
	2004	Dr. Reik has also been responsible for managing the Automotive Advanced Development department for the Schaeffler Group which consists of INA, LuK and FAG
Current position; Responsibilities		Executive Vice President LuK Group Research & Development Executive Vice President Advanced Development Schaeffler Group Automotive

The centrifugal pendulum absorber
Calming down the drivetrain

Dr.-Ing. Wolfgang Reik
LuK GmbH & Co.OHG
Bertrand Pennec
LuK GmbH & C.OHG

Introduction

Rising oil prices, the increasing shortage of crude oil resources and the harmful effects of CO2 on climate warming are forcing vehicle manufacturers to develop drivetrains with significantly better fuel consumption. The conventional drivetrain based on an internal combustion engine is coming under additional pressure by the possibilities of electric-only running that have now come within tangible reach. It is only the fact that battery capacity remains limited that is delaying the introduction of electric cars.
In order to give the internal combustion engine a chance of survival in the long term, the poor efficiency of current drivetrains must be drastically improved.
The fact that barely more than 20% of the inherent fuel energy reaches the wheel and can therefore be used for driving will be unacceptable in future.
Engines and transmissions must therefore be refined to achieve maximum efficiency.
Almost all the measures that are being discussed in this context and are conceivable increase the irregularities in torque and speed affecting the crankshaft.

- Higher power/torque per unit of cylinder capacity
- Higher average pressures
- New, fuel-efficient combustion processes
- Fewer cylinders with identical power
- Generous torque curves, allowing operation at the lowest possible speed

The only way of holding the irregularity at its current level is through considerably greater mass moments of inertia of the flywheel masses. This would, however, come in conflict with the requirements for lightweight construction.
While the greatest potential lies in the engine with its efficiency of barely more than 20%, improvements are also necessary in transmissions, whose efficiency is of the order of 90% depending on the type. The causes of power loss must be optimized or completely removed.

- Low friction in bearings and tooth meshes
- Reduced splashing losses
- Optimized hydraulic components such as pumps
- Replacement of hydraulic actuators by optimum efficiency, electromechanical components

These measures, however, make it more difficult for the drivetrain to damp out the vibrations generated by the engine irregularity and thus avoid rattle and boom noise. Current torsion dampers would no longer appear adequate to this task.
Drivetrain developers are therefore confronted with the situation where urgently required measures to reduce consumption cannot be effectively implemented because the drivetrain vibrations can no longer be overcome.
Many vehicle manufacturers who have clearly recognised this problem have therefore set themselves the task of developing new isolation and damping concepts since the conventional torsion dampers currently used in the dual mass flywheel and converter lockup clutches, are reaching their physical limitations.

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Requirements for vibration isolation in modern drivetrains.

Engine development can look back on considerable progress in the last decade [1]. Engine torque relative to engine capacity has significantly increased and the curves have become more generous, i.e. such high torques are available at speeds of barely more than 1000 rpm that fuel-efficient driving is possible and even promoted in these ranges *(Figure 1)*.

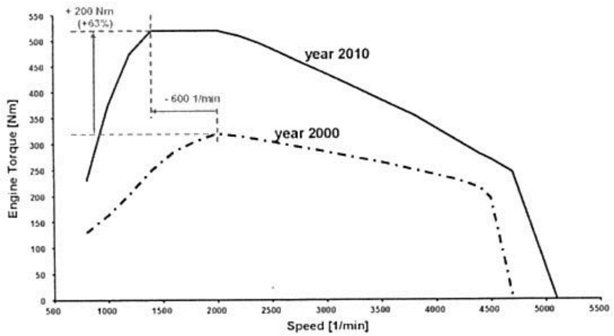


Fig. 1: Increase in Engine Torque from year 2000 to 2010
Based on the Example of a 4 Cylinder, 2.0 litre Diesel Engine

This leads, unfortunately, to correspondingly higher irregularity at the crankshaft *(Figure 2)*.

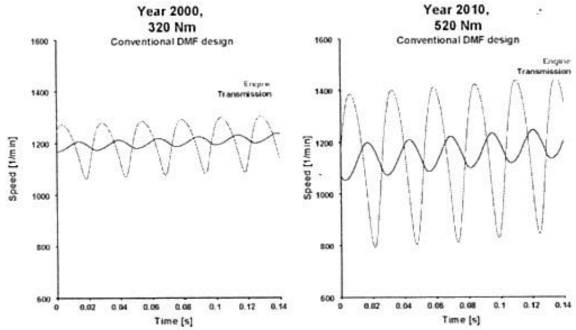


Fig. 2 Increase in Engine Excitation and Resulting Transmission torsional vibrations
Based on the Example of a 4 Cylinder, 2.0 litre Diesel Engine

Since transmissions have been "optimized" over the same period, i.e. designed for lower friction, there is now a situation where even better isolation is required under higher excitation levels in order to fulfil the increased comfort requirements *(Figure 3)*.

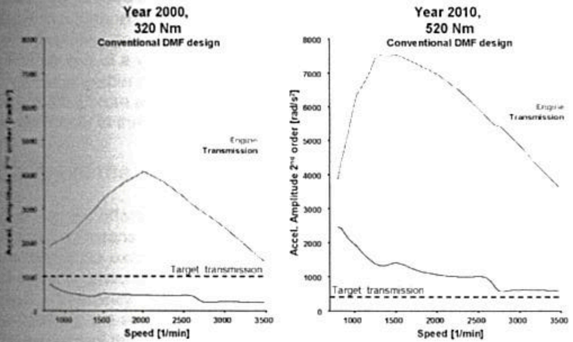


Fig. 3 Increase in Engine Excitation and Resulting Transmission torsional vibrations
Based on the Example of a 4 Cylinder, 2.0 litre Diesel Engine

If this is to be solved with the existing damper technology, which uses torsional elasticity to give decoupling, considerably larger torsion angles must be achieved in order to simultaneously give better isolation at lower speeds.
Figure 4 shows in a qualitative manner how the damper capacity must be increased in order to meet these challenges [2]. This would require considerably greater spring volumes, for which no space is available. As a completely separate issue, it would hardly be possible to achieve comfortable driving with such enormously long curves, since the torsional damper would extremely wind up under full load.

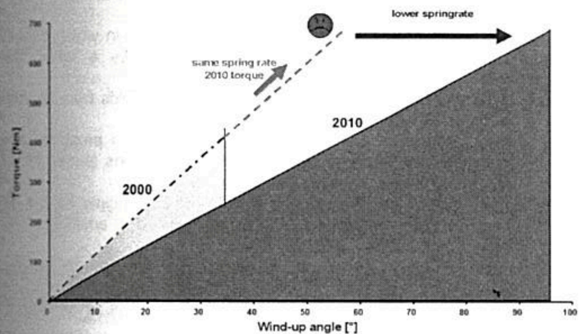


Fig. 4: New Requirements for Damper Design for modern engine

Physical possibilities for vibration isolation

Apart from torsional elasticity-based decoupling using a torsional damper, there are other possibilities for vibration isolation. The fundamental principles are shown in *Figure 5*.

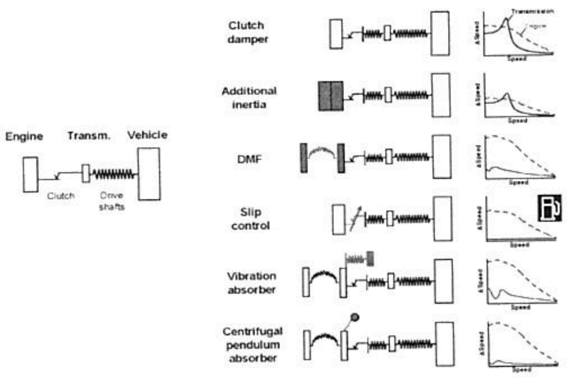


Fig. 5: Different ways to achieve vibration isolation

Prior to the introduction of the dual mass flywheel, only torsional dampers were used in clutch discs and were completely adequate in terms of isolation effect at higher speeds. Since the resonance speed of drivetrain thus equipped is in the range between 1500 and 2000 rpm, isolation cannot be achieved below about 2000 rpm, a speed that is now particularly important for fuel-efficient driving.

Improvements can be achieved by providing an additional mass moment of inertia on the primary side. The irregularity of the engine is reduced in line with this additional mass. In order to fulfill current isolation requirements, however, unrealistically high additional masses would be required.

About 20 years ago, the problem was solved by the dual mass flywheel, in which a well-dimensioned torsional damper was arranged between two masses. As a result, the resonance speed was pushed below the idling speed.

Good isolation of the torsional vibrations was thus possible at lower speeds from approx. 1000 rpm.

Over two decades, the dual mass flywheel was the panacea for vibration problems and drivetrain noise and was successfully implemented in manual transmissions across whole classes of vehicles.

In automated transmissions with converters, in which the dual mass principle cannot be implemented so easily, the specific slip in the lockup clutch was utilised in addition to the conventional torsional dampers in the converter lockup clutches.

According to Coulomb's Law of Friction, the frictional force is not dependent on the slip speed. A slipping transmission element does not transmit any torque variations. The slippage required is directly dependent on the magnitude of the torsional vibrations which lead to relatively high losses at low speeds.

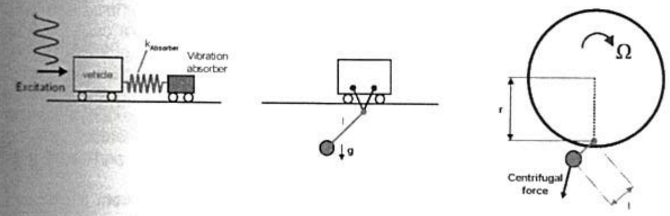
Slip is therefore generally used in conjunction with a torsional damper in order to improve its isolation effect. However, losses due to slip can reach several percent and will presumably no longer be accepted in future.

Vibration absorbers can be considered as a further measure. They are elastically coupled additional masses that lie outside the torque flow path. This type of absorber can, at its own resonance frequency, generate oscillations that directly counteract and thus cancel out the vibration being generated. This effect only occurs, unfortunately, at the absorber frequency and thus only at a very specific speed. In addition, two new resonance points are created, so a simple absorber can only be applied in a very limited way to remove torsional vibrations. There is therefore a need for a speed-adaptive absorber whose resonance frequency adapts automatically to the speed and thus the excitation frequency.

Theory of the centrifugal pendulum absorber

A particularly effective method of reducing vibrations is an absorber that comprises a mass coupled by means of springs. When correctly adjusted, this mass generates vibrations that counteract the excitation and thus act to cancel out the vibrations at the point to which the absorber is attached. However, the absorber only acts at a certain frequency, namely its natural frequency, which is determined by $\omega = \sqrt{k/m}$. At lower or higher excitation frequencies, the absorber does not act as desired and can even act to amplify the problem.

An absorber of this type with a resonance frequency proportional to the speed is very difficult to achieve by means of springs and masses. The example in Figure 6 shows how this can nevertheless be achieved by replacing the springs.



$$\omega = \sqrt{\frac{k_{Absorber}}{m_{Absorber}}} \qquad \omega = \sqrt{\frac{g}{l}} \qquad \omega = \sqrt{\frac{r \cdot \Omega^2}{l}}$$

$$\omega = \Omega \cdot \sqrt{\frac{r}{l}}$$

Fig. 6: From Damper to pendulum absorber

The model of such an absorber arrangement is shown on the left of the picture. The alternating excitation would excite the vehicle to horizontal vibrations. If the absorber is precisely matched to the excitation frequency, it generates vibrations that are precisely in opposition to the excitation frequency. The vehicle itself is brought to rest. Such absorbers are used in many mechanical installations where the requirement is to eliminate a very specific disruptive frequency.

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A mechanically equivalent system can be achieved by converting the spring absorber into a pendulum (Figure 6, centre). In both cases, there is an exchange of kinetic energy and potential energy during the vibrations.
In the normal absorber, the potential energy is held in the springs of the absorber, while in the pendulum it is held in the mass which was raised by a certain amount in the gravitational field of the Earth. The pendulum does not therefore require springs.
However, the pendulum initially has a very defined natural frequency and is therefore equally unsuitable for broadband absorption.
The formula for the natural frequency $\omega = \sqrt{g/l}$ shows that this is dependent only on the length of the pendulum and the force of gravity, better described as the gravitational field of the Earth. On the Moon, the same pendulum would vibrate much more slowly. All mathematical and physical relationships still apply if the gravitational field of the Earth is replaced by any other field. It would be possible to use, for example, electrical or magnetic fields, with which absorbers could be created whose frequency could be adjusted by adjusting the strength of the field.
In drivetrains where the excitation frequency is always proportional to the speed of the internal combustion engine, there is another possibility whose principles were laid down about 80 years ago.
If the pendulum is mounted on a rotating disc, the acceleration due to gravity g is replaced by centrifugal acceleration $a = r\,\Omega^2$ *(Figure 6, right)*.
The natural frequency of such a pendulum is then directly proportional to the speed. It is thus possible to achieve broadband absorbers that can cancel out, or at least significantly attenuate the effect of whole excitation orders over a wide speed range.
This is the simpler part of the theory of the centrifugal pendulum absorber, that is only valid for small pendulum angles, for which $\sin\alpha = \alpha$ is fulfilled. In general, this precondition is not met.
According to the theory behind the absorbers, the amplitude of the absorber increases until a force curve opposing the excitation is established. In the least favourable case, absorber components may be destroyed if the amplitudes become too great. In order to limit the amplitudes, various measures must be implemented.
First, the pendulum mass selected must be large enough that sufficiently large counterexcitations can be created at all. The effect of the centrifugal pendulum absorber becomes smaller and smaller at low speeds since the centrifugal acceleration becomes smaller. At low speeds, the pendulum tries to compensate this through particularly large oscillation amplitudes.
Normally, path curves are therefore selected in preference that deviate from the arc at a larger angle. The aim here is to ensure that the pendulum frequency remains constant up to amplitudes of 45°. If even higher amplitudes occur, the natural frequency is then specifically detuned such that the amplitudes remain limited. This prevents striking noise or even destruction.
In a closer mathematical analysis, further forces must be taken into consideration. In particular, the Coriolis force induces additional forces that, under large vibration angles, lead to deviations in the absorption force curve and, for example, prevent complete absorption or elimination.
Actual centrifugal pendulum absorbers can therefore only be designed by means of simulations taking account of all the forces acting on the pendulum.

History of the centrifugal pendulum absorber

The mathematical principles of the centrifugal pendulum absorber were established around 1930. Even the first design proposals [3] were made at that time. Salomon (1932) envisaged a system of rollers oscillating in circular recesses *(Figure 7)*.

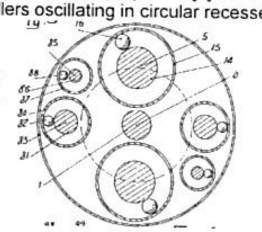

Fig. 7: Centrifugal pendulum absorber with oscillating rollers in a patent by Salomon, 1932

Where pendulum masses must also rotate during oscillation, however, this does not give optimum conditions, as will be shown later.

Sarazin (1937) proposed a bifilar suspension arrangement [4] in which each point of mass describes the same (desired) path *(Figure 8)*.

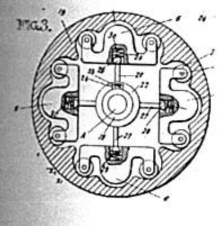

Fig. 8: Bifilar suspension arrangement by Sarazin (1937)

Modern designs of the centrifugal pendulum absorber are a further development of this suspension arrangement. Such centrifugal pendulum absorbers were used in aircraft engines during World War 2 *(Figure 9)*.

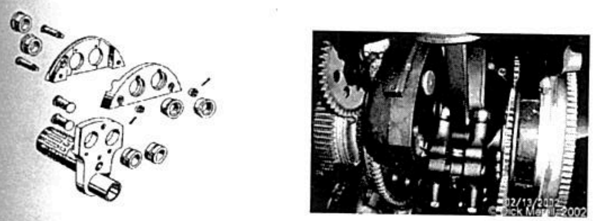

Fig. 9: Centrifugal pendulum absorber with bifilar suspension on the crankshaft of an aircraft engine (R1820 Cyclone) from Pratt & Whitney [6]

In car engines *(Figure 10)*, these pendulums were also generally installed on a crankshaft web.

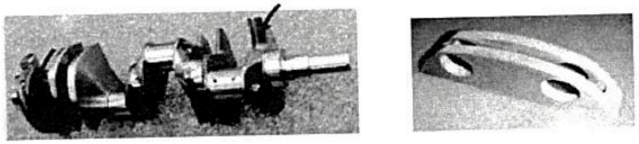

Fig. 10: Crankshaft used in the engine with two pendulum attached (left) and one of the pendulum absorbers used on the crankshaft (right), Ford Motor Co [6]

Following this, no further developments took place on the centrifugal pendulum absorber for a long period. For a time, it was possible to overcome the vibration problem using other simpler solutions.
It is only in recent years that the centrifugal pendulum absorber has come back on the scene, as the performance and thus the irregularity of internal combustion engines has increased while at the same time the requirements for comfort levels have grown.
Several attempts have been made in recent years to revive the centrifugal pendulum absorber. An absorber based on the Salomon rollers was offered under the name "Rattler" by TCI Automotive
(Figure 11).

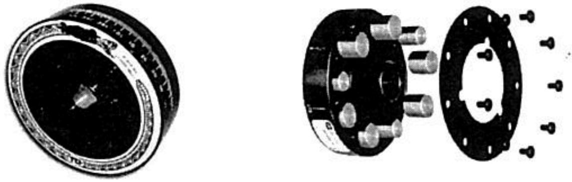

Fig. 11: Absorber with rollers as a centrifugal pendulum absorber [8]

Freudenberg made attempts, by means of pendulums with bifilar suspension in a flywheel, to directly reduce the irregularity of the crankshaft *(Figure 12)*. However, pendulum masses of several kilograms were required since the pendulums had to counteract the total maximum torque peaks during ignition.
While this system was effective in principle, it could not be successfully implemented.
Based on this experience, it became clear that future requirements for vibration isolation cannot be fulfilled with an acceptable outlay by means of centrifugal pendulum absorbers on their own.
Efforts were therefore made to achieve a suitable combination with conventional torsional dampers.

Figure 13 shows, on the left, the model of a centrifugal pendulum absorber linked directly to the flywheel and thus also to the crankshaft. Large pendulum masses are required in order to allow compensation of ignition shocks. *On the right of the picture*, a dual mass flywheel is used as a basis and the pendulum is arranged on the secondary flywheel mass. This allows a division of labour.

Fig 12: Centrifugal pendulum absorber with bifilar suspension in the flywheel [7]

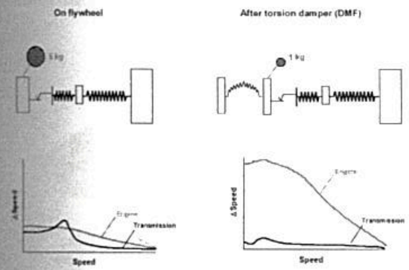

Fig. 13: Essential possibilities for locating a centrifugal pendulum absorber

The dual mass flywheel with its spring coupling gives the first stage of isolation. The secondary flywheel mass is typically subjected only to torque amplitudes below 50 Nm, while the torque acting directly on the crankshaft is many times the mean engine torque and, in diesel engines in particular, may be well over 1000 Nm.
A centrifugal pendulum absorber arranged on the secondary flywheel mass must, in contrast, only compensate the residual irregularity and only requires much smaller masses.

Design features

Since it has been clarified at which point, namely after a torsional damper, the centrifugal pendulum absorber should usefully be arranged, the precise design must now be defined.

Valeo Exhibit 1106, pg. 10

At first, it appears easiest to use rollers in accordance with the design by Salomon (Figure 7). A more precise analysis shows, however, that this is not the most effective way since the rollers rotate during oscillation and part of the kinetic energy is therefore stored as rotational energy (Figure 14),

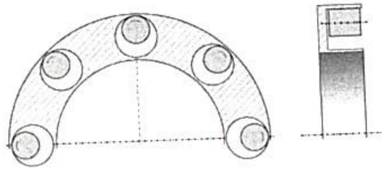

Fig. 14: Centrifugal Pendulum Absorbers with Salomon Rollers.

and the energy for translational motion is thus lacking. It is only the speed along the path of the pendulum that generates centrifugal force and thus contributes to an absorption effect.

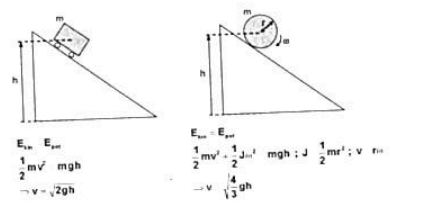

Fig. 15: A carriage acquires a greater speed on an inclined plane than a roller.

In Figure 15 this is shown using the example of an inclined plane. From the same initial height on the ramp, the carriage acquires a greater speed than the roller.
All pendulum arrangements in which rotation of the pendulum itself occurs are therefore not an optimum solution. This also applies to the physical pendulum (Figure 16),

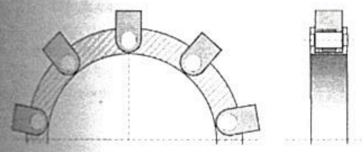

Fig. 16: Physical pendulum

in which it is also significant that relatively large elements of the mass are arranged around the suspension system (bolt) and thus do not contribute to the pendulum effect.
The aim must be to allow effective oscillation of the total available mass. The best way of achieving this, after the mathematical pendulum that cannot be realised in practice, is a bifilar suspension arrangement. In the simplest design, the oscillating mass can oscillate over two bolts that are arranged in kidney-shaped recesses in the mass and in the carrier. The shape of the holes is selected to give an appropriate pendulum path.

Figure 17

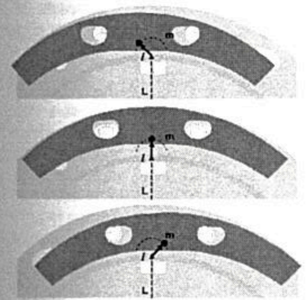

Fig. 17: Movement of bifilar pendulum

shows how the total mass moves without rotation. Each point describes the same path, so the pendulum can be presented as a point type mass at the centre of gravity moving along the curve.
This indirectly imitates the mathematical pendulum and allows the greatest effect for a given mass.
The pendulum masses are advantageously arranged on both sides of a carrier flange and are linked to each other by 2 or 3 rivets such that sufficient axial clearance is present and the pendulums can move without external friction and damping.

Valeo Exhibit 1106, pg. 11

With 4 pendulums each comprising stamped sheet metal parts approx. 5 mm thick, a pendulum mass of approx. 1 kg can be achieved, which should be located on the largest possible diameters.
In the dual mass flywheel, the large diameter area is reserved as before for the bow springs. The pendulums are located on the flange inboard of these springs and thus act on the secondary flywheel mass *(Figure 18)*.

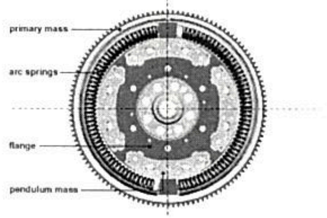

Fig. 18: Dual mass flywheel with pendulum absorbers

In the converter, the centrifugal pendulum absorber can be integrated with the torsional damper in the lockup clutch. Effort is made here too to locate the pendulums with bifilar suspension on the largest possible diameter. Based on the torque flow path, at least 1 torsional damper must be arranged in front of the centrifugal pendulum absorber (Figure 19).

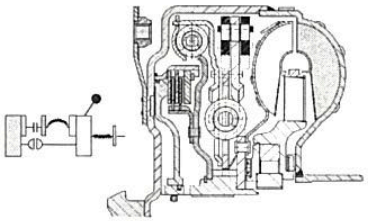

Fig. 19: Torque Converter with Lockup-Clutch with pendulum absorber

Measurements

With a centrifugal pendulum absorber on the secondary flywheel mass of a dual mass flywheel, a further improvement in vibration isolation can be achieved. Figure 20 shows, using the example of a 4 cylinder diesel engine with 400 Nm, the speed fluctuations of the engine and transmission input for various dual mass flywheel types. A zoom of the smaller speed variations is shown on the right.

Compared with the basic design only containing arc springs and the improved variant with inner damper, the centrifugal pendulum absorber replacing the inner damper gives a further significant improvement in isolation. Compared with a normal dual mass flywheel, the irregularity can be further reduced by more than half. It can be seen directly from the curves that, for a specific maximum permissible variation range of the transmission input speed required for running free from rattle and boom noise, the minimum engine speed can be reduced by several hundred revolutions per minute.
Even greater benefits apply to the converter and its torsional damper in the lockup clutch, since the conventional dampers do not yet include the dual mass flywheel effect.

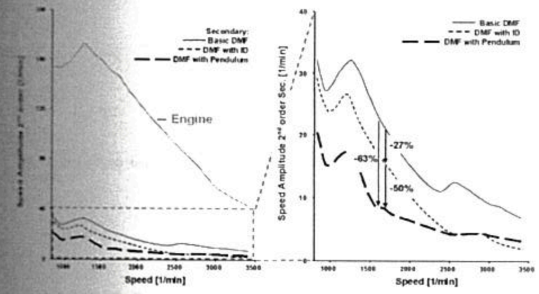

Fig. 20: Full Load Simulation – 4 Cyl. Diesel 400Nm
System comparison basic DMF, DMF with innerdamper and with Pendulum

Figure 21 shows not only the speed amplitude of the engine but also of the transmission input, which is decisive for boom noise.
Even a conventional double torsional damper cannot eliminate the turbine natural mode, at least at speeds around 1300 rpm. A turbine torsional damper, in which the torsional elasticity lies within the torque flow path of the turbine (hence the name) gives a significant improvement that is, nevertheless, not sufficient in various vehicles.
The centrifugal pendulum absorber gives the major benefit here and allows boom-free driving even at speeds down to 1000 rpm.

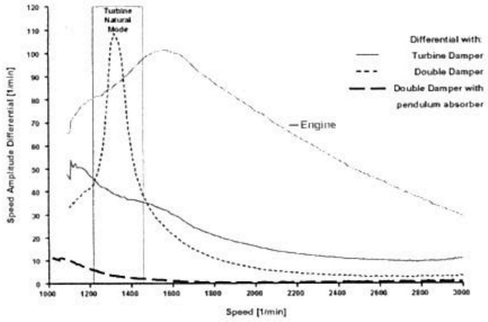

Fig. 21: Full load simulation – 4 cyl. diesel 450 Nm
System comparison basic torque converter, turbine damper, double damper and double damper with pendulum absorber

Conclusion

The requirements for fuel-efficient drivetrains are continually increasing as a result of discussions about climate warming and scarcity of resources.
Many of the fuel-efficient drive concepts increase the irregularity of the engine and the sensitivity of transmissions to these periodical variations of speed and torque. Improved torsional dampers are therefore required. The centrifugal pendulum absorber is an absorber whose resonance speed increases in proportion to the speed. In order to keep the pendulum size within acceptable boundaries, the centrifugal pendulum absorber should be located on the secondary mass of the dual mass flywheel.
The effect of such pendulums in dual mass flywheels and also in converters is described. Significantly improved isolation effects can thus be achieved.

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