Torsional Vibration Isolation in the Drive Train An Evaluative Study

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The previous presentations have described the vibration isolating effect of conventional torsion dampers, dual mass flywheels and slipping clutches. Other vibration isolating systems are also employed in motor vehicles. Prop shaft dampers mounted at the transmission output shift resonance frequencies. Hydrodynamic couplings and torque converters have been used successfully in automatic transmissions for decades. And other systems are in development, such as the viscous coupling.

The following presentation provides an overview of the physical options available for reducing torsional vibrations in the drive train. All known coupling elements used between the engine and the transmission will be discussed. However, this article will not deal with the reduction of engine irregularity by structural modification of the engine design, such as the addition of more cylinders. These kinds of comparative studies are known in the literature [1, 2], but each is based on a specific vehicle.

In contrast, this presentation assumes a simple, generally applicable drive train. There will be no attempt to discuss special, isolated drive train problems. This limitation is necessary in order to communicate an overview of existing and conceivable systems for vibration isolation.

Vibration Model

The proven three-inertia vibration model of the drive train will be used for comparative evaluation of vibration isolation options (Figure 1). The mass moments of inertia and torsional spring rates chosen correspond to those for a compact passenger car operating in 3rd gear.

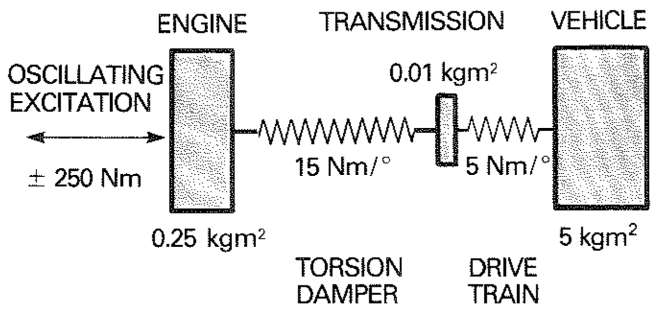


Figure 1:
Vibration model with harmonic excitation

In contrast to the previous studies, vibrations affecting this vibration model will not be derived primarily from actual engine excitation. Engine excitation involves, in addition to the basic excitation, components that are multiples of the basic frequency. In fact, any abrupt change in gas pedal load will even result in broad-band excitation.

Harmonic excitation proves more appropriate for a basic comparison of different variations in the drive train. Therefore, a sinusoidal excitation torque $M = M_0 \cdot \sin \omega t$ with the amplitude $M_0 = 250$ Nm is introduced into all the following calculations.

The vibration model reacts to this excitation with vibrations, which will be the subject of discussion. The reaction of the transmission and the vehicle mass are critical for vehicle comfort. Vibration and control engineering designates this as a system response. In the higher frequency range, the acceleration amplitudes in the transmission are closely associated with gear rattle. Typical rattle ranges occur in association with excitation frequencies of 30 to 200 Hz. The oscillating acceleration of the vehicle generally causes few or no noises. However, in some cases the driver perceives this condition as objectionable surging and even clunk.

Conventional Torsion Dampers

Normally, in a manual transmission automobile, the engine and the transmission are connected using an elastic connection, the clutch disc torsion damper. Engine irregularity is not supposed to pass through to the transmission, at least not in certain speed ranges. As discussed in a number of previous presentations, this effort is not totally successful.

Figure 2 shows this relationship for a sample torsion damper with an ordinary spring rate of 15 Nm/°. The angular acceleration of the transmission and the vehicle has been standardized in order to provide a better overview. For this purpose, the amplitude at the transmission has been referenced to the angular acceleration that would result using a rigid connection between the engine and the transmission. Therefore, values below 1 indicate vibration isolation, and those above 1, amplification. The acceleration amplitude of the vehicle was also standardized in similar fashion. It was referenced to the acceleration that would result from a totally rigid drive train. The center graph shows the standardized amplitude of the vehicle acceleration, and the bottom graph, the transmission acceleration as a function of the harmonic excitation frequency. Since both accelerations and frequencies extend over a wide range, a logarithmic scale was used.

The vehicle acceleration shown in Figure 2 would lead us to expect strong surging at about 4 Hz for a conventional torsion damper if an appropriate excitation were present. Significantly higher frequencies are no longer capable of exciting the large, inert vehicle mass to vibration.

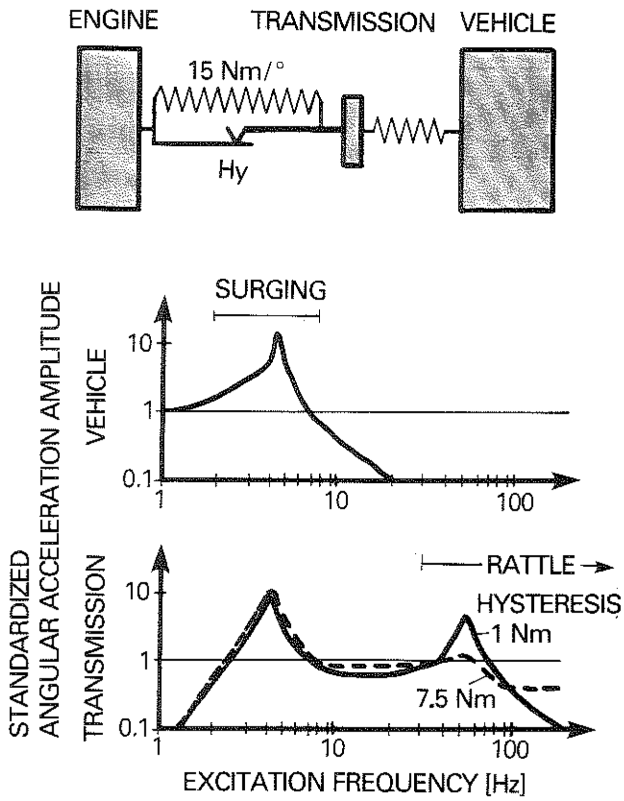


Figure 2:

System response with conventional clutch disc

Of course, the transmission excitation also exhibits relatively high acceleration amplitudes in conjunction with the surging frequency, but no objectionable gear rattle is generated. Surging and clunk are usually the only objectionable response.

Gear rattle can be anticipated at frequencies over 30 Hz, and the resonance point is located in precisely this range. This point is significantly influenced by the torsion damper characteristic.

Figure 2 also shows the influence of the damper hysteresis on transmission acceleration. A high hysteresis prevents any significant resonance. However, it diminishes vibration isolation at higher frequencies. The hysteresis has virtually no influence on vehicle acceleration or surging.

The resonance range with amplification is typical for spring-coupled systems. To combat resonance amplification, high damping is necessary, which will cause the acceleration amplitude to approach the value of 1 at high frequencies, thus approximating rigid performance.

Figure 3 shows the influence of the spring rate on the vibration performance of the drive train for a linear torsion damper rate. In the rattle range, a lower rate has a favorable effect. It shifts the resonance to lower frequencies and reduces the amplitude at the transmission. It has hardly any effect on vehicle vibration. Only the resonance frequency is shifted, and it is scarcely perceptible from a subjective standpoint.

This changes, however, if we use multi-stage characteristics such as those used to combat gear rattle in idle mode. The extremely low rate for the idle stage acts similar to lash in the drive train and can cause tip-in/back-out performance to deteriorate.

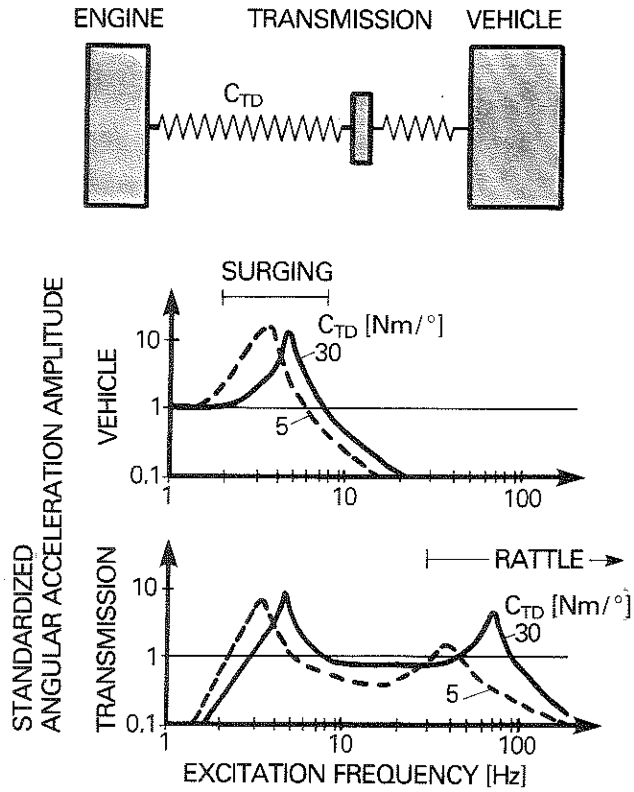


Figure 3:

System response with conventional clutch disc

Dual Mass Flywheel

If we take the flywheel inertia, which is normally attached to the engine, and divide it into a primary and a secondary inertia, we can achieve a marked shift in the resonance at the transmission input (Figure 4). The secondary inertia increases the effective inertia of the transmission. This is particularly effective – as demonstrated in one of the previous presentations – if a very low spring rate is used between the two flywheel inertias. This is a primary feature of modern dual mass flywheels.

The transmission features excellent vibration isolation at all critical rattle frequencies above 30 Hz. For lower excitation frequencies, for instance those that occur during engine start-up and shut-off, the low spring rate of 2 Nm/° is clearly superior to the spring rate of 10 Nm/°. Vehicle vibration can also be reduced by a low spring rate and the appropriate damping value. As a result, modern DMFWs promise improvements with respect to surging as well.

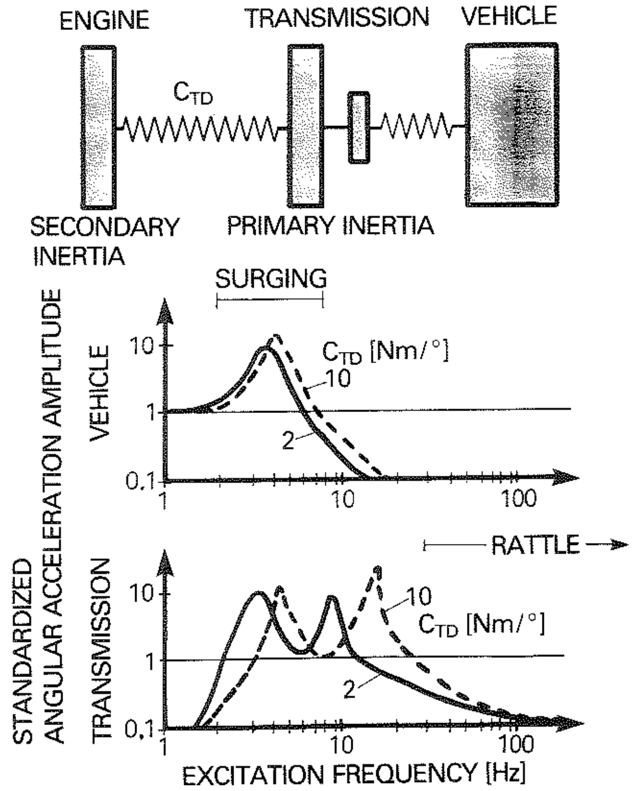


Figure 4:

System response with spring-coupled dual mass flywheel

Prop Shaft Dampers

Prop shaft dampers represent a totally different option for eliminating torsional vibrations. These dampers are occasionally attached to the transmission output for rear-wheel drive vehicles. They are capable of decreasing the amplitude of torsional vibrations in the transmission at excitation frequencies corresponding to the frequency of the prop shaft damper (Figure 5).

In operation, the prop shaft damper generates a vibration opposing that of the engine in the effective speed range and can thus compensate for at least part of the irregularity transmitted to the transmission via the torsion damper. In favorable cases, the torque generated by the prop shaft damper is precisely the opposite and equal to the oscillating torque transmitted by the torsion damper. The resulting oscillating torque at the transmission is completely neutralized and the oscillating accelerations disappear.

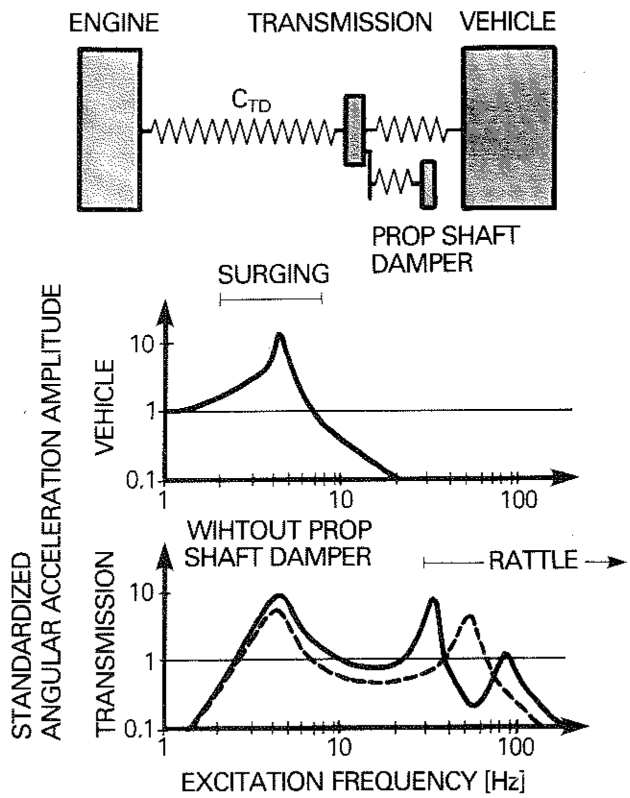


Figure 5:
System response with prop shaft damper

Unfortunately, this damping effect is limited to a narrow speed range. On either side of this range an equiphase vibration can even occur in the prop shaft damper itself. This vibration is associated with an amplification that can only be diminished somewhat by appropriate damping in the prop shaft damper, which reduces its effect at the tuned frequency. The prop shaft damper has no effect on surging.

Centrifugal Pendulum

The ideal prop shaft damper ought to possess a normal frequency that would change with the excitation frequency in order to overcome the disadvantage of its narrowly limited range of effectiveness. Theoretically, the prop shaft damper frequency would have to adjust to the excitation frequency. This would require that the prop shaft damper have a torsional spring rate that would increase with the square of the excitation frequency. This is impossible with the springs that are commonly used for this purpose.

However, there are ways to create a vibrating system without using springs. Instead of briefly storing energy in springs, it is possible to transform it into potential energy. The simple mathematical pendulum of length l is an example of this principle. The resonance frequency is represented by

$$f = 2\ \pi \sqrt{g/l}$$

where g is the acceleration due to gravity.

If this kind of pendulum is attached to a rotating disc, the acceleration due to gravity must be replaced by the centrifugal acceleration $r\Omega^2$ (see Figure 6, top), where r is the radius on which the pendulum mass is located and Ω is the angular acceleration of the disc. The natural frequency of this kind of rotating pendulum is

$$f = 2\pi\ \Omega\ \sqrt{r/l}$$

and is thus proportional to the speed. Values of r and l can be selected to achieve a precisely defined order. To be able to damp the main excitation in four cylinder engines, we select

$$\sqrt{r/l = 2}.$$

With this kind of properly tuned centrifugal pendulum attached to the clutch disc torsion damper, it is possible to completely damp excitation frequencies with two full vibration phases per revolution (Figure 6). All other orders remain virtually unaffected. Unfortunately, it is impossible to effect surging tendencies with this kind of centrifugal pendulum because surging excitation is usually broad-band and is not proportional to speed. Hence it cannot be assigned to a single order.

Many studies have examined the option of using a centrifugal pendulum (sometimes called a Taylor or Sarazin pendulum) [see 3 – 8]. Obviously, many developers were attracted by the possibility of completely eliminating an entire order for a broad frequency range, but no one has arrived at any practically applicable solution. LuK has also studied this option closely [9]. Numerous difficulties, for instance tolerance problems, overly large pendulum mass and very high vibration amplitudes at low speeds, hampered any success.

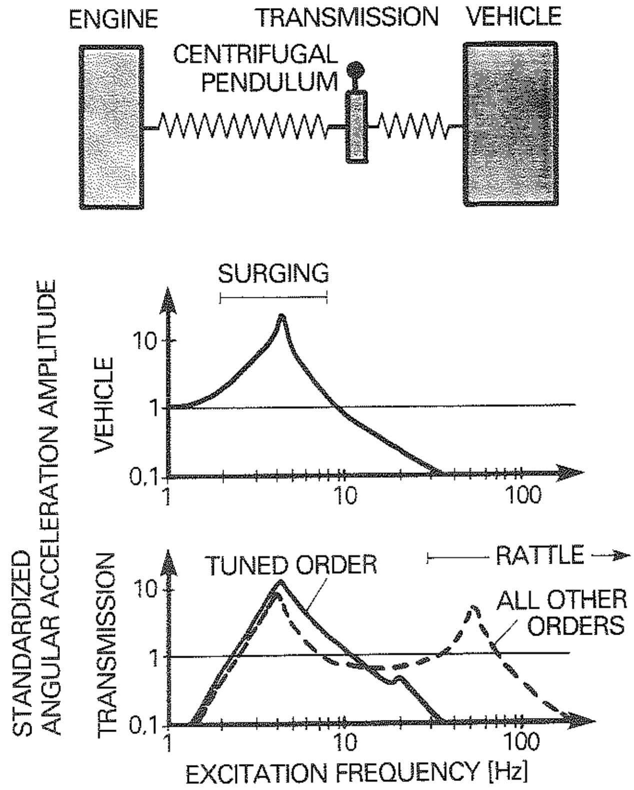


Figure 6:
System response with centrifugal pendulum

Hydrodynamic Torque Transmission

The hydrodynamic coupling – sometimes called the Föttinger coupling - and torque converters represent a completely different kind of coupling between the engine and the transmission. No spring mechanism is present to serve as a short-term energy store, a requirement for vibrations. Consequently, resonance between the engine and the transmission is impossible. However, since there is no torsional elasticity whatsoever, with which vibration isolation might be achieved, the transmission component must permit slip between the engine and the transmission.

Automatic transmissions use torque converters or hydrodynamic couplings to isolate vibration. Their behavior with respect to vibration excitation has been described in the literature [10 – 13]. Because the torque converter becomes increasingly stiffer with increased speed, vibration isolation is strongly dependent on speed. Figure 7 shows a typical example. Low impeller speeds produce particularly good vibration isolation. However, this solution exacts a trade-off in the form of a high slip, which results in high energy loss.

Figure 7 also reveals that not only is the high rattle frequency isolated, low-frequency surging is also significantly reduced.

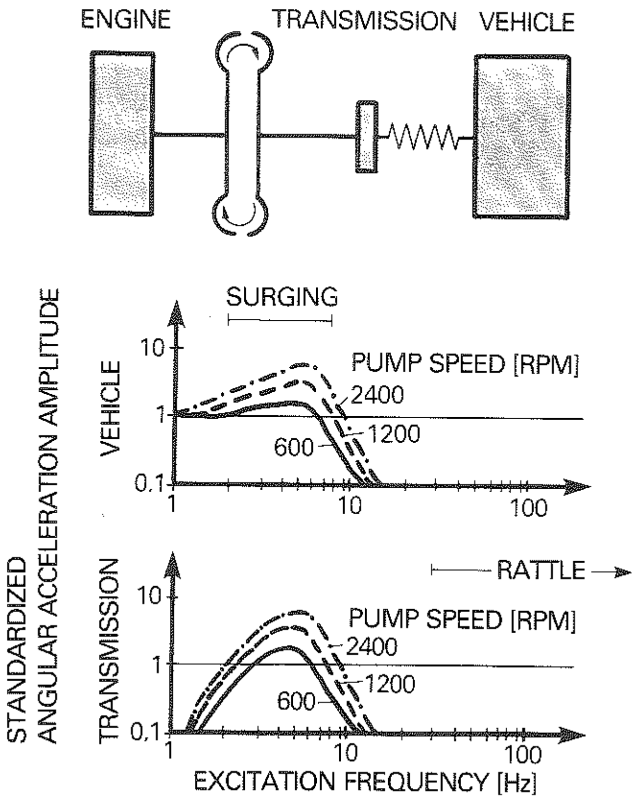


Figure 7:
System response with hydrodynamic coupling

Hydrostatic Torque Transmission

Instead of being hydrodynamic, the coupling between the engine and the transmission can also be hydrostatic [14]. This system uses a pump featuring a hydraulic flow control valve that transmits the torque from the pump unit to the drive shaft. In the LuK design shown in Figure 8, several parallel gear pumps are situated between the primary and secondary flywheel inertias. Slip can be varied using flow control valves that are controlled by centrifugal force. These valves are not shown in this drawing. Figure 9 shows the acceleration amplitudes of the transmission and the vehicle. Here as well, vibration isolation improves at higher slip speeds, that is, at wider valve openings.

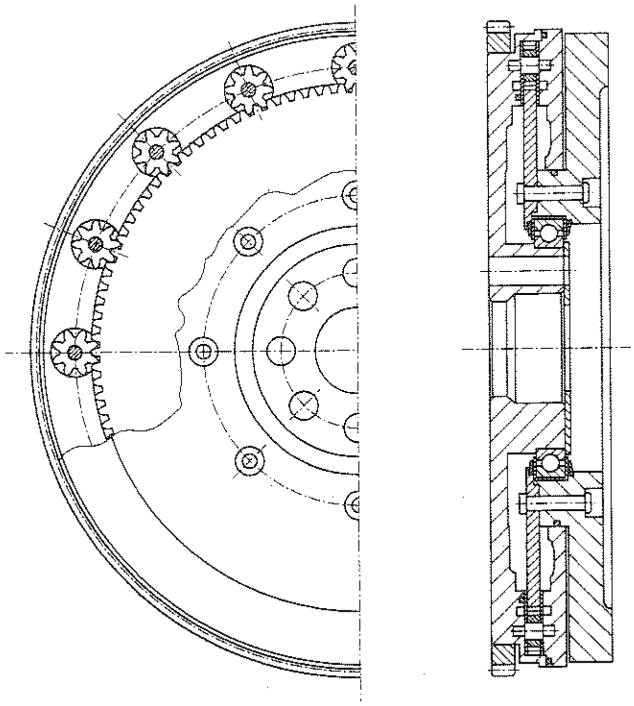


Figure 8:
Hydrostatic dual mass flywheel

In comparison to torque converters (Figure 7), it is obvious that the isolating effect on the transmission is still relatively poor at high frequencies. Furthermore, vehicle surging vibrations are only eliminated at the cost of very high slip. Tip-in/back-out measurements taken using a hydrostatic dual mass flywheel like the one shown in Figure 8 confirm these observations (Figure 10). The speed curves for the engine and the transmission after a tip-in are plotted as a function of time. The pump is blocked in the top graph in Figure 10. Consequently, the engine and the transmission speed have an identical curve. After the tip-in, there is a strong surging vibration that then decays gradually.

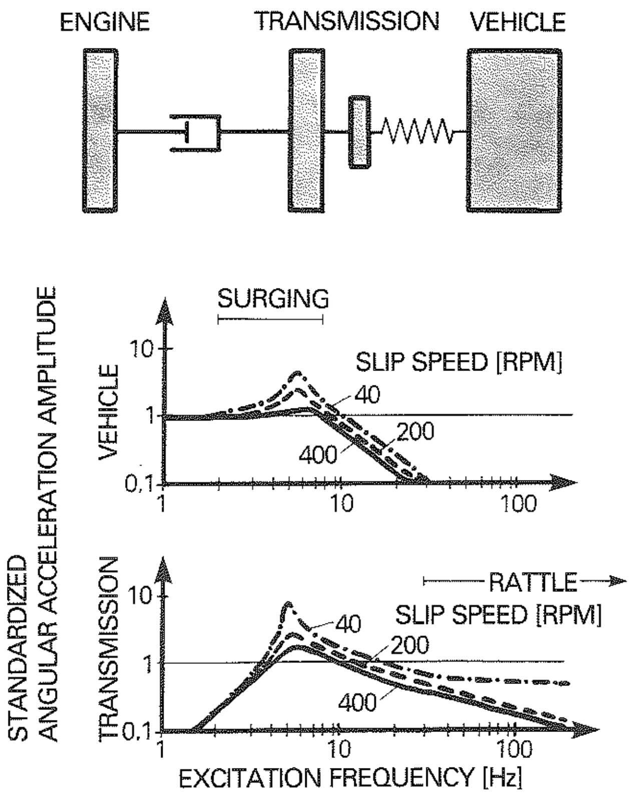


Figure 9:

System response with hydrostatic dual mass flywheel

In contrast, the bottom graph in Figure 10 reflects the same process using a relatively high slip of 200 rpm. To be sure, the surging vibration is significantly damped, but falls far short of being eliminated. Nor was this to be anticipated based on Figure 9.

The high slip factor required for efficient operation causes thermal problems that are made even worse as a result of the frictional heat generated by the clutch. As we all know, at high temperatures, for instance under extreme driving conditions, the viscosity of all oils decreases drastically. This increases slip even more and can lead to thermal destruction. As a result, LuK has discontinued any efforts to develop a purely hydrostatic DMFW.

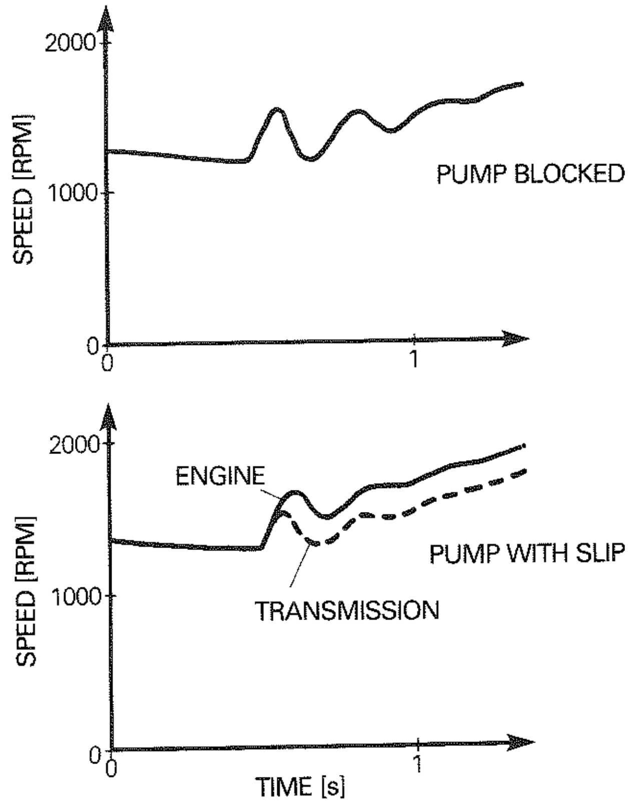


Figure 10:

Tip-in for hydrostatic dual mass flywheel

Viscous Coupling

In addition to the hydrostatic pump, a viscous coupling can be used in a dual mass flywheel. At first glance, the mathematical derivation of vibration performance reveals surprising results. We observe exactly the same performance as with hydrostatic load transmission. The reason for this similarity is that in both cases load is transmitted using fluid shear. This is not readily apparent in the case of the pump because the actual shear occurs in the hydraulic flow control valve. The pump itself just converts torsional movement into hydraulic flow. In the case of the viscous coupling, on the other hand, the fluid transmission medium is sheared directly between rotating plates.

If we don't use the viscous coupling or the hydrostatic pump in a DMFW and employ a clutch disc instead – that is, if we do not use a secondary inertia between the engine and the transmission – vibration isolation is considerably worse (Figure 11). It is scarcely possible to eliminate gear rattle with an acceptable amount of slip. However, similar to the situation with the hydrostatic DMFW, high slip improves vehicle surging.

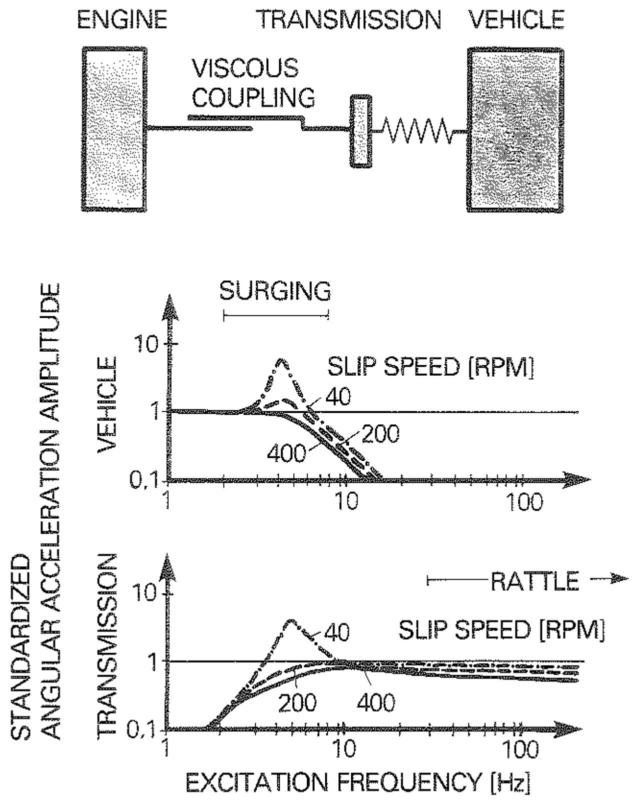


Figure 11:
System response with viscous coupling

Torque Control Isolation

Torque control isolation, which is the topic of one of the presentations in this series, offers totally new options. An electronic control loop allows us to change drive train vibration performance over wide ranges. With optimum tuning – see Figure 12 – isolation is achieved at excitation frequencies as low as 5 Hz. Both rattle and surging can be completely eliminated.

However, it would be inappropriate to demand isolation at even lower excitation frequencies because this would introduce a delayed vehicle response to drivers' acceleration needs. Therefore, at very low excitation frequencies, the standardized amplitude of the vehicle angular excitation must approach 1 in order to transmit constant torque. The limiting frequency, at which the steep decrease in the transmitting function occurs, should therefore lie between approximately 2 to 5 Hz.

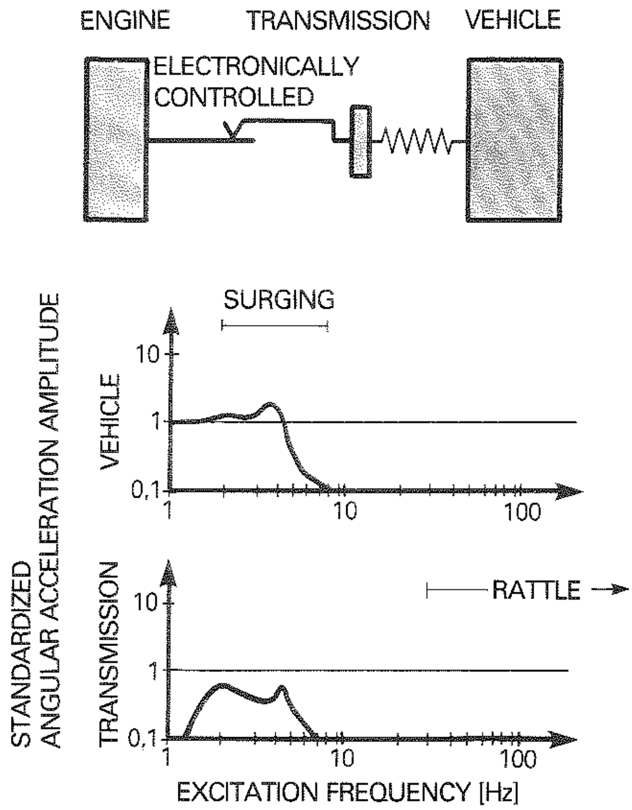
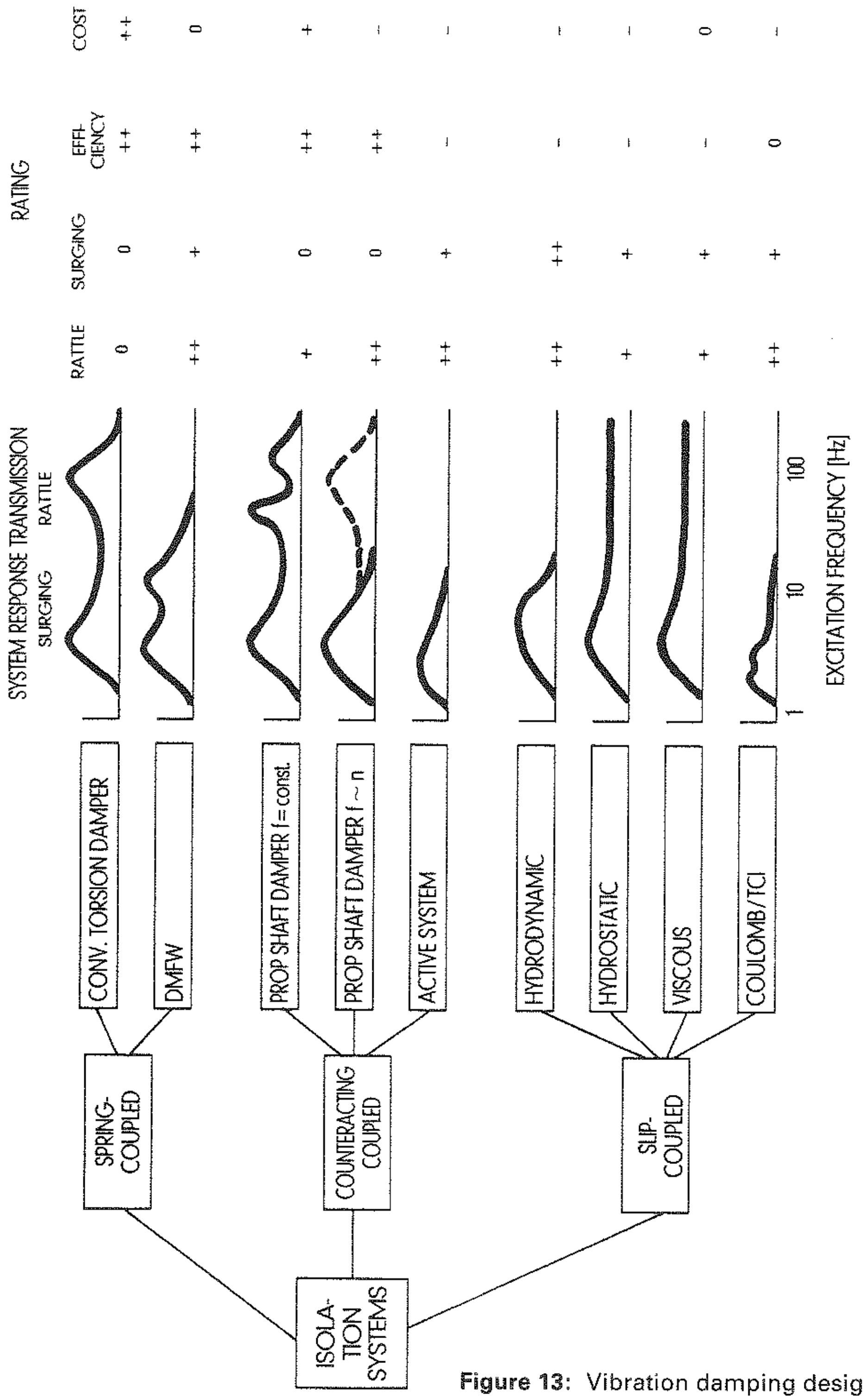


Figure 12:
System response with torque control isolation

Vibration Damping Procedures: A Comparison

The current options for producing vibration isolation between the engine and the transmission can be divided into three major groups. With the first group, the engine and the transmission are connected by an elastic coupling. As shown in Figure 13, this group can be further subdivided into conventional torsion dampers and spring-coupled dual mass flywheels. Several strong peak resonance points are characteristic. Vibration isolation is not clearly evident until we reach a range above the highest resonance frequency.



	RATTLE	SURGING	EFFI-CIENCY	COST
CONV. TORSION DAMPER	0	0	++	++
DMFW	++	+	++	0
PROP SHAFT DAMPER f = const.	+	0	++	+
PROP SHAFT DAMPER f ~ n	++	0	++	-
ACTIVE SYSTEM	++	+	-	-
HYDRODYNAMIC	++	++	-	-
HYDROSTATIC	+	+	-	-
VISCOUS	+	+	-	0
COULOMB/TCI	++	+	0	-

Figure 13: Vibration damping designs

The advantage of the dual mass flywheel lies in the fact that transmission isolation starts at a much lower speed. Moreover, it is possible to improve surging with very flat spring rates and appropriate damping values. All spring-coupled systems have a high degree of efficiency. Minimal losses resulting from damping are insignificant.

For the next major group, the counteracting coupled systems, vibration isolation between the engine and the transmission is not of primary importance. With these systems, a counteracting torque is introduced at the transmission. This torque is designed to neutralize the oscillating torques generated by the engine. Generally speaking, however, prop shaft dampers are attached to the transmission output shaft. They are capable of neutralizing vibration and eliminating rattle, but only for a fixed frequency.

The centrifugal pendulum would provide some improvement in this regard. With this design, the damper frequency is proportional to speed, so an entire order of excitation can be completely canceled out. This solution provides an almost ideal transmitting function. However, other orders are not affected. At the moment, this design poses unsolved problems and is associated with high costs.

Both options, the prop shaft damper and the centrifugal pendulum, are actually passive systems. But truly active, counteracting coupled systems are also conceivable. In such systems, the counteracting torque would not just be an attached, elastic mass, but would instead be generated by an actuator or supplemental motor. An electronic control would generate a counter-phase torque in opposition to the measured transmission acceleration and would then be able to neutralize torsion vibrations completely over a wide frequency range. This design would entail high technical expenditure and would exhibit poor efficiency because of the energy required to operate the extra motor, both of which factors represent definite disadvantages.

To our knowledge, this kind of expensive, active compensation system has not yet been developed for a motor vehicle drive train. Isolated designs have been implemented for suspension systems, machine tools, and structures [15 – 17].

In the case of the third group, the slip-controlled systems, no additional resonance occurs. This is an exceptional advantage, but it must be bought with energy loss.

A comparison shows the advantages of the hydrodynamic torque converter and the electronically controlled friction clutch, particularly at the higher rattle frequencies. Low surging frequencies can be damped with high slip values in all systems.

As already noted with respect to torque control isolation, slip and the associated efficiency loss can be minimized using appropriate characteristic fields. However, the costs for such a system are high.

Combined Systems

None of the systems illustrated in Figure 13 is able to rack up plus points in all categories. However, a skillful combination of systems can be used to counteract weak points. Such combinations can feature either serial or parallel designs. In some cases, individual systems are activated in certain speed ranges.

An example of this kind of hybrid design is the torque converter with a lock-up clutch. At low speeds, the system utilizes the vibration isolating capability of the torque converter. At high speeds, a lock-up clutch with a conventional torsion damper acts parallel to the torque converter to provide complete torque transmission without any slip, while at the same time filtering out vibrations at high excitation frequencies. This arrangement permits the elimination of energy-consuming slip, at least in the high speed range.

For the same reasons, it is advisable to use a standard, serial torsion damper design with torque control isolation. Low excitation frequencies are filtered out via TCI, and the torsion damper takes care of the high frequency components. In this range, the clutch can be locked up to avoid slip.

Figure 14 shows an additional combination option involving elements of both a spring and a slip control system. A hydrostatic pump in a DMFW is situated in series with an elastic component. At first glance this doesn't appear to make much sense, because neither system provides adequate protection against low-frequency surging vibrations, at least not with an acceptable degree of slip. However, we can use the following trick. The pump flow control valve is closed during normal operation and the slip factor is reduced to a minimum of approximately 10 rpm. The elastic torsion damper provides vibration isolation to completely absorb engine irregularity. During tip-ins, for instance when the driver suddenly steps on the gas, pressure builds up in the throttled pump system. This pressure triggers the flow control valve to open briefly, thus providing for increased slip in order to prevent surging vibrations. Because this valve recloses automatically within a few seconds, the total energy loss is limited.

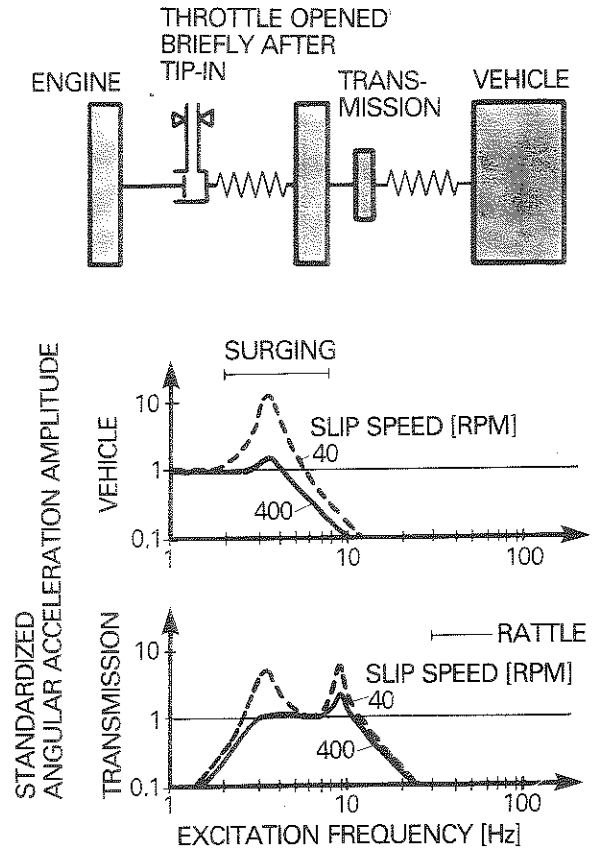


Figure 14:

System response with hydrostatic pump and spring

Figure 15 shows this kind of design. The gear pump is located on the outer diameter, and the serial springs are arranged around the inner diameter. However, this kind of combination is hardly ready for production and may prove to be very expensive to build.

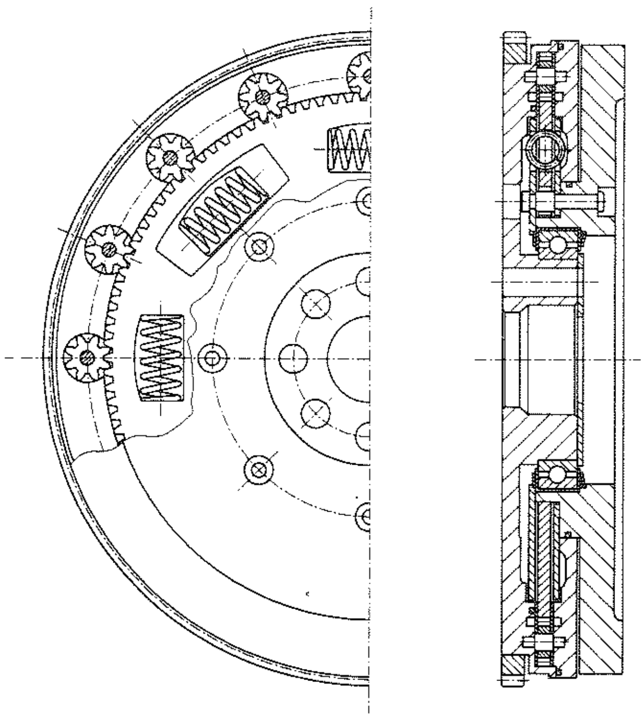


Figure 15:
Dual mass flywheel with serial design featuring a hydrostatic pump and springs

Summary

Systems for isolating torsional vibrations in motor vehicle drive trains can be divided into three groups:

- spring-coupled systems, for instance conventional torsion dampers and spring-coupled dual mass flywheels
- counteracting systems, for instance prop shaft dampers or centrifugal pendulums
- slip-coupled systems, such as torque converters, torque control isolation or viscous couplings.

In the case of spring-coupled systems, engine excitation is absorbed by torsional elasticity. There is virtually no energy loss involved because the energy stored in the springs is later returned to the system. The additional resonance associated with the energy store can lead to gear rattle. By optimizing the inertial relationships and introducing an extremely reduced spring rate in association with a modern, long-travel DMFW, we can shift these additional resonance points out of the driving range, thus eliminating gear rattle.

Counteracting systems generate a counter-torque at the transmission. Technically practical prop shaft dampers have a narrow, limited efficiency range and are only capable of improving gear rattle in individual cases.

In the case of slip-coupled systems, engine irregularity is compensated with slip. No resonance can occur between the engine and the transmission. High slip can be used to damp low-frequency surging vibrations effectively. The unavoidable disadvantages include energy loss and high cost.

Consequently, in the future appropriate combinations will be developed joining slip- and spring-coupled systems in which the slip system is only engaged as long as necessary to combat low-frequency vibrations.

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