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Less is more!

Using unconventional means to design new products

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The previous articles have introduced technologies that have already achieved a high level of sophistication. They are either in volume production with the first few customers or will achieve this status over the next few years. Basic development is nearly complete. Engineers now know what the product will look like and the exact characteristics it will have. Most likely, there will be no more surprises.

Companies would be well advised to look at the more distant future. Some of the more distant goals may appear blurred or distorted, including the ongoing discussion in the automotive industry regarding how to ensure mobility in the future.

This article attempts to describe such distant goals relating to engines and transmissions. Of course, while the foundations are only now being laid, the exact goal remains to be defined. Obstacles – which are not going to be easy to overcome – are already visible on the horizon.

With this in mind, readers should view this report as a preview of potential topics for the next Schaeffler symposium. We hope that a few of these topics will still be relevant and can be added to those projects that are closer to volume production. Your opinion matters because you can tell us whether you consider these goals worth the effort.

Dual mass flywheel without springs

The previous articles have treated the effects of centrifugal pendulum-type absorbers on dual mass flywheels, torque converters and other torsional vibration dampers, allowing entire exciter orders to be eliminated. This is an absolute necessity for modern drive trains that achieve their consumption advantages by reducing engine speed. These down-speeding designs can be implemented because new loader generations ensure nearly full engine torque even at speeds of just over 1000 1/min (Figure 1). Damping systems with centrifugal pendulum-type absorbers are especially suitable because they provide excellent isolation at speeds of around 1500 1/min and extend the range for lower speeds.

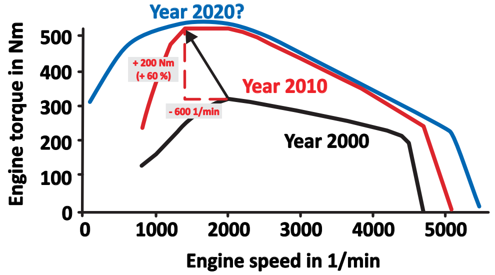

Figure 1 Development of engine torque from 2000 to 2010 and a forecast for 2020 using a 2.0 four-cylinder diesel engine as an example

But what happens if engine development progresses at the same rate as it has over the past few years? Will this soon lead to a demand for running at full load at under 1000 1/min? Could the torsional dampers then thwart further engine development? For this reason, new approaches are being investigated.

The essential principle of a dual mass flywheel is the division of a flywheel into two separate masses (the primary and the secondary flywheel mass) that are linked via torsional elasticity. This connection, which is usually carried out by compression springs because they have a very high energy density, contains a large part of the engineering in dual mass flywheels. A skillful selection of curve slopes with the relevant overlapping of damping friction, multi-stage curves, etc. generally allows all operating ranges to reach the torsional vibration isolation required for comfortable driving without rattling and humming noises. This is why the torsion springs have always been regarded as the heart of the dual mass flywheel. Continuous optimization efforts have improved these springs to represent optimal curves.

For the reasons mentioned above, it may be surprising to witness attempts at doing without springs in a dual mass flywheel. The springs have two major tasks. First, the medium torques of the engine and the transmission have to be transmitted. Second, irregularities must also be filtered out as much as possible, which is the typical task of a mechanical low-pass filter. For the isolation effect, an excitation frequency load must also be applied to and removed from the springs. During this process, the spring absorbs potential energy that is then immediately released. The spring thus functions as a buffer. If there is too much energy available because the internal combustion engine rotates faster after an ignition, this surplus is buffered in the spring. During the next compression, when the engine rotates more slowly, the spring releases this energy again.

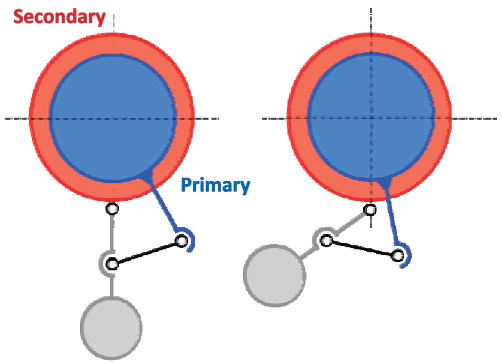

Figure 2 In dual mass flywheels without springs, the torsion curve is generated via masses in the centrifugal force field

When considering the general effect of a torsional vibration damper, it becomes clear that intermediate energy storage is required that does not necessarily have to be carried out by a spring. Potential energy may also be buffered in other ways. In the simplest case, a mass is increased against a force. This force may be caused by a gravitational field or any other kind of field, such as magnetic or electrical fields. In rotating systems, the centrifugal force field, or more simply, the centrifugal force comes to mind.

In the diagram shown in Figure 2, the primary side of the dual mass flywheel is connected to the secondary side by a stabilizer link that engages in a pendulum on the secondary side. If torque is applied, the pendulum is displaced. However, centrifugal force causes the pendulum to return to its original position. As in a spring coupling, this generates a torque that increases with an increasing torsion angle. This curve though is no longer constant but depends on the speed, resulting in a group of curves with the speed as its parameter. The energy used to smooth engine irregularity is also buffered through changes in the potential energy of the oscillating mass.

The isolation values of such a springless dual mass flywheel closely correspond to those of a conventional dual mass flywheel (Figure 3).

This requires large pendulum masses of approx. 5 kg that perform relatively space-filling motions. This will make it difficult to find a design solution. The isolation behavior of this springless dual mass flywheel can even be improved considerably if the

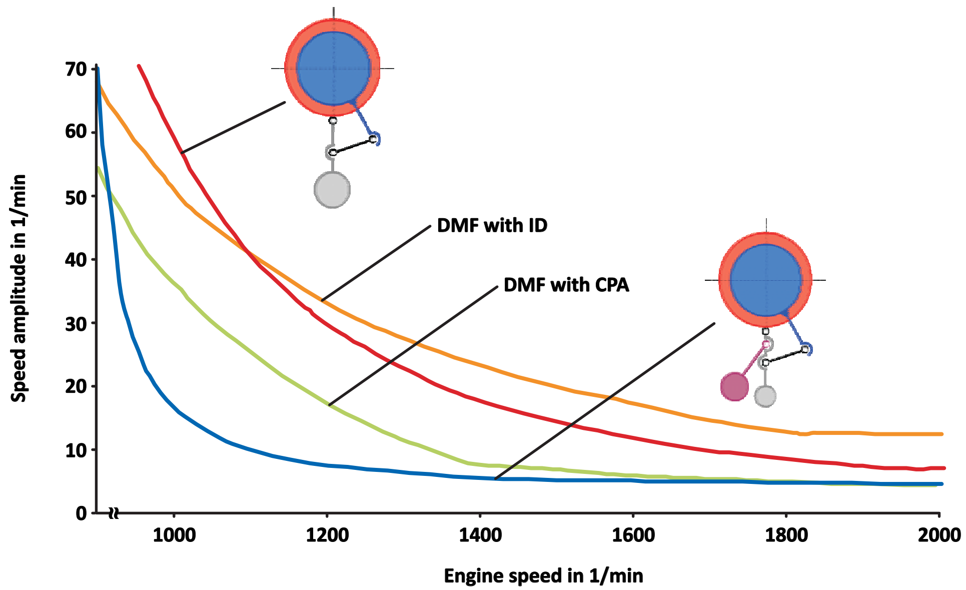

Figure 3 Comparison of the isolation effect of springless and conventional dual mass flywheels

pendulum is separated into a double pendulum, as shown in Figure 4. This heavier pendulum, which has the effect of a regular centrifugal pendulum-type absorber, hangs from the (lighter weight) coupled pendulum.

Figure 4 shows that this arrangement can even exceed the isolation effect of a regular dual mass flywheel with a centrifugal pendulum-type absorber. With double pendulums, the coupled pendulum attempts to return to its original position under centrifugal force. Both pendulums contribute to this, resulting in curves that become more rigid and have a higher maximum final torque (Figure 5) as the speed increases. In this case, the pendulum masses should be so great that the engine torque present at lower speeds can be covered. At higher speeds, the curve would be able to transmit much more torque than the engine can supply. This is not necessary, but would not do any harm either.

It is also true for the double pendulum that the pendulums that are relatively heavy overall (approx. 5 kg) glide over large areas, and a design would require a great deal of design space. In addition, completely springless dual mass flywheels do not generate a torsion curve unless centrifugal force is applied. During standstill, the pendulum masses would not apply any resistance to the torsion of the two flywheel masses against each other. A fully springless dual-mass flywheel would thus be more interesting in theory because it shows that potential energy is not only buffered in springs. For a practical design, a conventional but very weak spring would be used in the case of extremely low speeds and standstill.

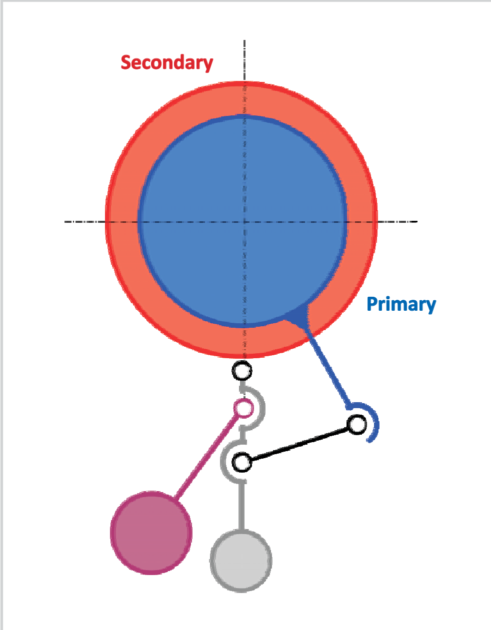

Figure 4 Springless dual mass flywheel with additional, freely swinging centrifugal pendulum-type absorber for optimum vibration isolation

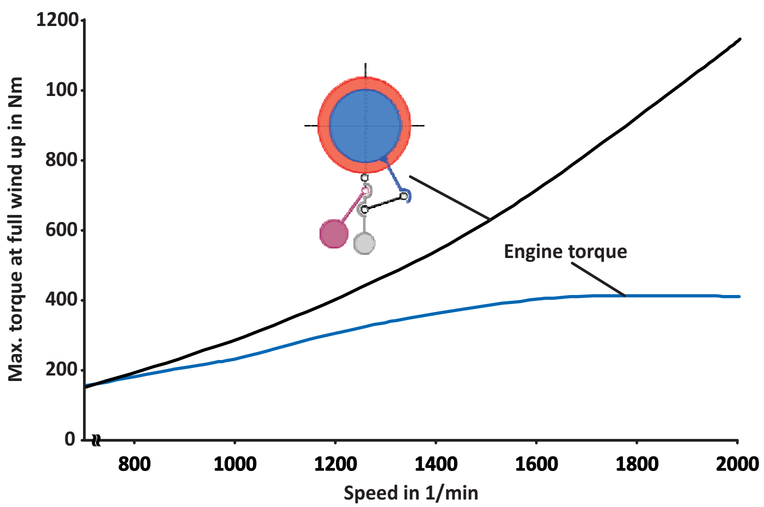

Figure 5 Increase of the maximum torque of the torsion curve in relation to the speed

Clutch with hardly any actuation force

There is an obvious global trend towards automated drive trains, even though there appears to be resistance to this change in some parts of the world. If this trend continues, will we still need clutches? This is a question that traditional clutch manufacturers will have to address. Fortunately, the answer is very complex. The relative share of manual transmissions is declining. Due to the growing new markets, however, absolute volumes are even increasing [1]. In addition, there are new drive train designs that require a clutch. The table below shows today's trends, including exclusively electrical vehicles.

Drive train designs	Number of shift system elements (clutches + brakes)
Manual shifting	1
Double clutch transmission	2
Automatic transmission	~ 5
Hybrid drives	Up to 3
Electric vehicles	2 (if a two-speed transmission is required)

It is interesting to note here that many of the new designs also require clutches and some, as is the case with the double clutch transmission or hybrids, even need more than one. It is only all-electric vehicles that seem feasible without any clutches. But even here a two-speed transmission would more likely be used to achieve an optimal balance between maximum speed and hill start. In this case, two shifting elements or clutches would be required.

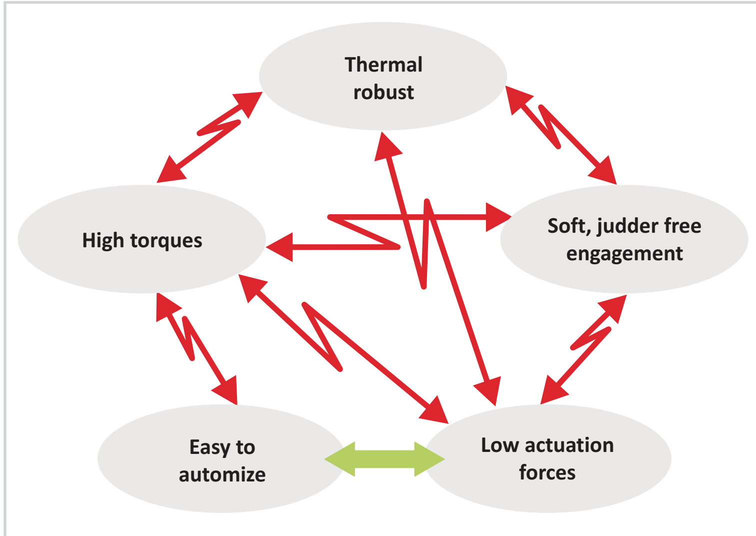

Figure 6 Most important requirements for clutch systems

It is to be expected, however, that the clutches will have to be adapted to future requirements, resulting in designs and design types that are very different from today's manual gear shift systems. It must still be assumed that the essential function, namely to transmit torque through friction to adjust speeds, will be maintained. This will essentially keep the basic requirements for clutch systems the same, which also means that typical clutch issues such as heat balance and judder, etc. will continue to be a problem even in new drive train designs.

Figure 6 shows the typical conflicts encountered in clutch development. Nearly every requirement conflicts with several others. Only the desire for good automation options and low actuation force seem to complement each other well. This is why the actuation force, or to be more exact, little actuation work, is probably of particular importance. This will be considered below.

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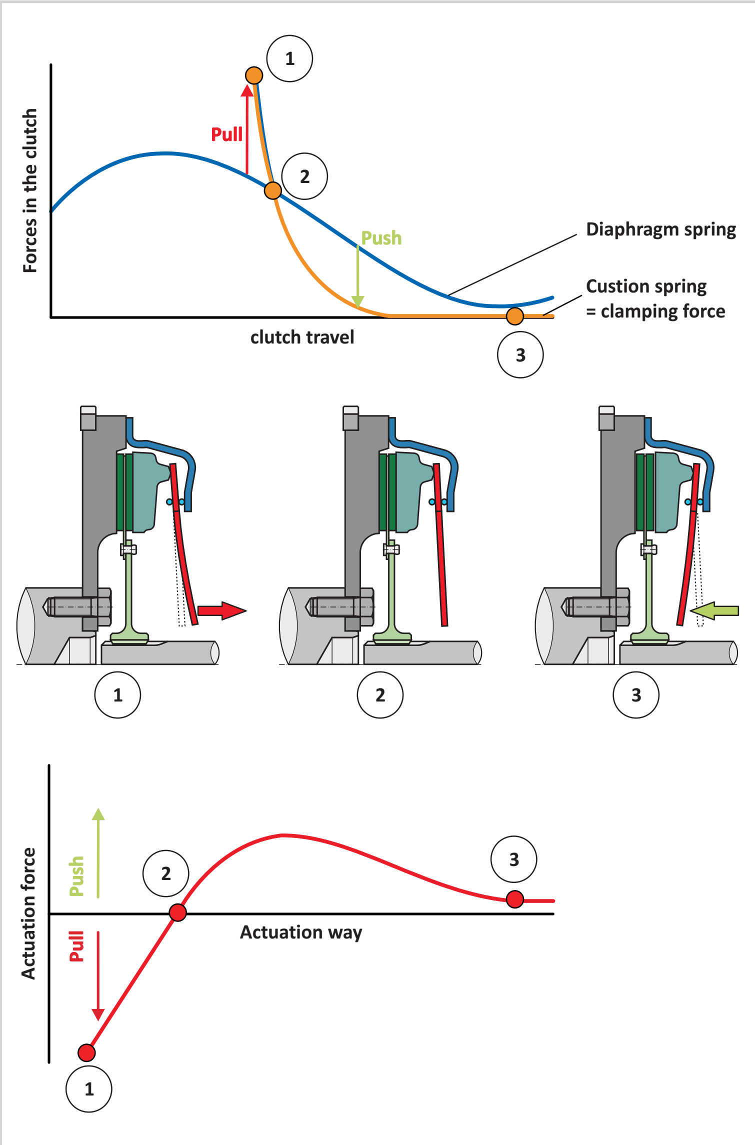

Figure 7 Conditions present in a clutch. The Figure in the center (2) shows a conventional manual transmission clutch in a closed position, the Figure on the right (3) shows the open condition and the Figure on the left (1) shows closing with increased clamping force.

The actual task of a clutch is shown in Figure 7. A (clutch) disk is braced or clamped between two pressure plates to connect a torque flow. Torque can then be transmitted through friction. The clamping force must be eliminated to open the clutch and interrupt the torque flow. The disk can rotate freely. In manual shift systems, the clamping force on a clutch is applied via a diaphragm spring preload. Without any external influence, the clutch is closed. To open the clutch, the diaphragm spring force is suspended via the diaphragm spring fingers that function as two-armed levers and the clutch ventilates.

The reverse principle also applies. With many wet running clutches and dry double clutches, an open clutch is desirable in the event of a fault. In these cases, the external actuator is used to close the clutch. The clamping force is applied externally. New electronics and software designs can also handle faults in clutches closed by spring force [2]. This allows for more and even entirely new options when selecting the clutch design. The following considerations are based on a clutch closed by spring force because this makes the principle of reducing the actuation force easier to explain.

It is a very common error to assume that the clamping force is directly related to the actuation required to suspend the clamping force. This is not the case, since the clamping force consists of the internal distortion of the diaphragm spring, signaled by its torque, and the external actuation force. The best example that can be used to explain this is a cable car (Figure 8). The internal force, i.e. the tension of the cable, is proportional to the weight of the vehicles. The torque that the engine has to generate for the cable on the pulley, however, depends to a much greater extent on how well the weights of the two cars are balanced. At best, the cars could be set in motion with any number of small torques while neglecting friction.

The situation is similar for clutches. The clamping force is determined by the force (or more precisely by the force curve) of the diaphragm spring. As with the cable car, actuation only requires the difference of the forces (diaphragm spring and cushion spring in a quasi balance). An effort must be made to design a degressive diaphragm spring force that is as close as possible to the cushion spring force (Figure 7, top). This in itself is not a new idea [2] and this was the essential driving force behind the development of the self-adjusting clutch. In actuator-operated systems, it has considerable potential for expansion. While it is important in foot-operated clutches to ensure that a pedal force curve is generated that is pleasant for the driver and allows easy clutch modulation, for the most part this requirement can be eliminated in automated systems, resulting in a much greater actuation force reduction.

In the past, the curve tolerances and their displacement towards each other due to facing wear always hindered the implementation of this simple principle. For this reason, a compensation of the lining wear and the tolerances is essential. Self-adjusting clutches have become widely accepted for higher engine torques. However, since wear adjustments are not entirely precise, a truly consistent implementation of this idea has not been possible, and designers have been able to reduce the actuation force by only a little over 25 %.

As mentioned above, there are clutches in which the clamping force is generated by an external actuation force. The question here is whether these two principles can be combined. To achieve this, the actuation system is mounted on the diaphragm spring fingers in a way that not only exerts pressure on the diaphragm spring fingers but that also allows motion by means of tension (Figure 7, Figure on left, position (1)). By applying tension, the clamping force of the clutch and the torque to be transmitted can be increased further. For a given torque, the actuation force could be reduced (roughly by half) if the actuator is designed for pushing and pulling. In general, electro-mechanical actuators can do this. An electric motor is capable of generating the same torque in both rotating directions.

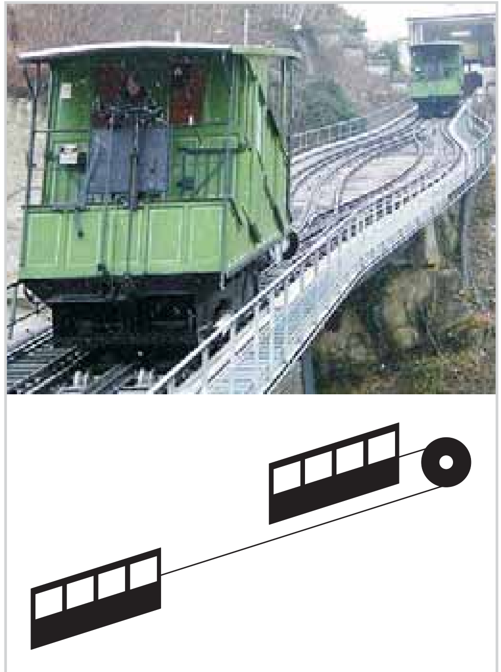

Figure 8 Balance of forces in a cable car

This new clutch design is able to generate roughly half the clutch torque without external actuation (position (2)). To fully open the clutch, pressure is exerted on the diaphragm spring finger, similar to a manual transmission clutch (position (3)). To close the clutch completely, the fingers can be pulled, resulting in an additional external clamping force that is applied, similar to a clutch opened without force (position (1)). This has thus resulted in the development of a new clutch type – the clutch half closed without force – and is coupled with a significant reduction in actuation force.

There is also another advantage. Until now, the balancing of the diaphragm spring and cushion spring curves has been insufficient. There has been too great a risk that the curves will overlap due to the tolerances and the fact that the clutch involuntarily actuates itself because it suddenly attempts to snap over. This can be prevented if the actuator is connected to the diaphragm spring fingers in a way that permits both pushing and pulling. This design then permits the force to change its algebraic signs during actuation. The clamping condition is then controlled by the actuation travel and not by the actuation force. This becomes the smaller the better the adjustment is performed, allowing further reductions in actuation forces and thus smaller actuators.

Actuation can be performed all the more quickly, the smaller the design of the actuator. This allows solutions to be found for clutch problems that are otherwise difficult to manage. Friction linings, regardless whether they are wet or dry, can generate judder vibration during the slip phase. The cause are torque fluctuations produced by small geometric errors or by a sliding friction value that decreases with the differential speed. The associated vibration in the drive train – approximately 10 Hz when starting – represents a serious comfort problem. Fast actuation can practically eliminate such friction vibration if the clutch is adequately counter-modulated in time with the judder frequency [3]. This is shown in Figure 9. This type of anti-judder control will result in significantly increased comfort. Moreover, it will also allow the use of linings with higher friction values. This is not possible today since an unwritten "friction law" states that judder tends to increase with increasing friction. Although there is no physical explanation for this, there is a great deal of practical experience. So once a functioning and fast anti-judder control is available, linings with high friction values can be used to further reduce the press-on force and actuation work.

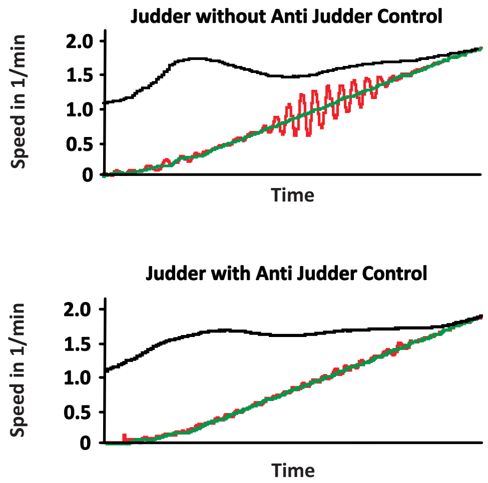

Figure 9 Starting process for lining with severe judder: without (top) and with anti-judder control (bottom)

All measures taken to reduce force and work, to control the way a clutch closes or opens, become relevant when the actuation system is to be integrated in the clutch. One potential solution is an electronic clutch, in which the actuator rotates along in the clutch. The energy for the actuator as well as the adjustment information are to be transmitted via an electrical torque transmitter. Figure 10 provides a general idea of what this might look like using a double clutch as an example. Experts hope to be able to develop compact clutches with a simple and space-saving actuation system that fit well into applications such as hybrid drive trains.

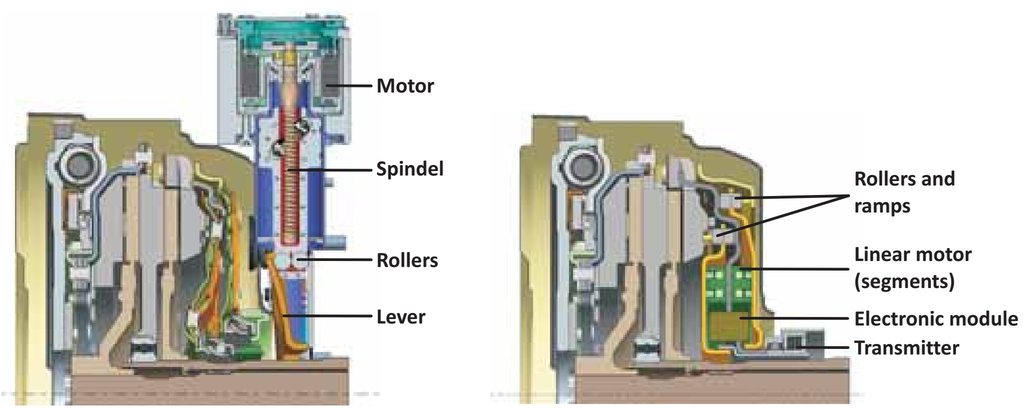

Figure 10 Double clutch with external actuator (left) and an envisioned integrated actuator (right)

Engine without camshaft

In the past, there have been a number of developments whose goal it was to simplify the valve train of an internal combustion engine and to replace the camshaft as a drive for the valves. Most of these efforts have been abandoned because insurmountable difficulties prevent them from being successful. There have been various reasons for this. A collision of the valve with the piston could not be reliably prevented, the energy required for actuating the valves was too high or the costs exceeded permissible limits. That is why it may be surprising to see another attempt being made in this area. The reason for this lies in the development of the UniAir system [4] which is ready for volume production. It is described in this symposium volume and offers new opportunities.

The essential feature of the UniAir system is the hydraulic actuation of the valves through a slave cylinder. The required hydraulic pressure and the oil volume are applied through camshaft-controlled hydraulic pumps that could also be called master cylinders. This hydraulic separation of the cam lift and the valve lift results in ideal opportunities for creating impact when the hydraulic feed line is opened or closed by a solenoid valve. Lift and time changes can then be made with wide limits to the valve lift curves. Even a second opening is possible under certain conditions. This design also permits other variants, including even an engine without a camshaft.

The first step is to combine the individual cam drives with the master cylinders to form one unit. To this end, a hydraulic central pump has been developed that is based on a vane type pump (Figure 14). Unlike regular pumps of this design, the rotor consists of a displacer that is not completely round and has a base circle with a cam, similar to the cam on a camshaft. The vanes are arranged in the stator to be radially movable form chambers in pairs that are connected to the slave cylinder of an intake valve. Every time the cam slides over such a chamber, oil is displaced, which opens the intake valve and then closes it again. The cam shape, the vane distance and the vane width are selected in such a way to obtain the desired maximum valve lift curve. As soon as the cam leaves a chamber, it glides across the next one, generating hydraulic displacement for all cylinders one by one.

One of the development goals was to design the hydraulic feed lines to allow the central pump to be installed at a sufficiently large distance from the cylinders. A feed line length of 40 cm was achieved with good mapping of the cam shape on the valve lift curve.

After successful completion of the first step, the complete UniAir functionality can now be inte-

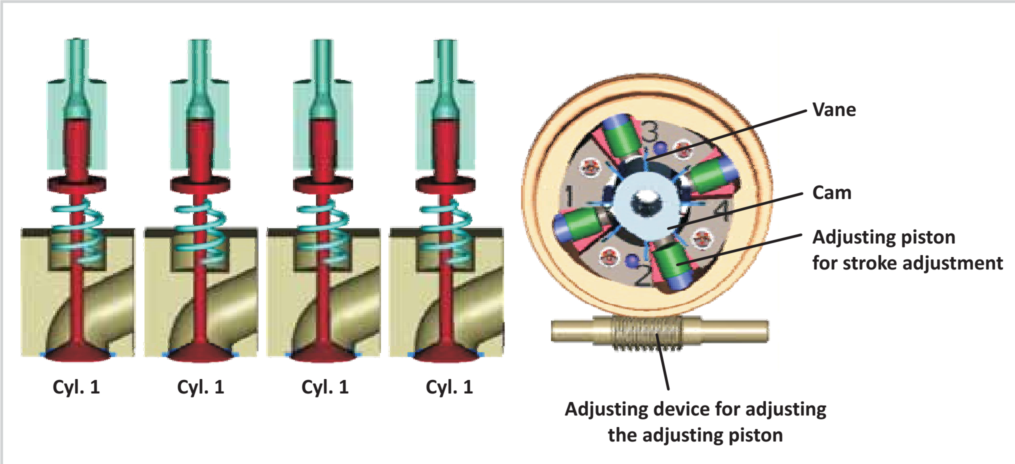

Figure 14 Vane type central pump whose chambers are connected with the individual valves via hydraulic feed lines

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grated by also integrating the hydraulic shift valves (including the electronic system) that are necessary for interrupting pressure and early closing of the intake valves into the hydraulic central unit (Figures 15 and 16). The central unit thus includes everything required for controlling the valves, such as the hydraulic cam, electric shift valves for controlling volume flow, the electronic system and the required hydraulic wiring. A hydraulic feed line is used to connect the intake and exhaust valves of the cylinders with this central unit. The hydraulic unit can be installed in almost any position, such as on the side of the engine. What is important is that the central pump can be operated with half the crankshaft speed. For this purpose, a toothed chain, a toothed belt or face spline can be used.

The need for a conventional camshaft is eliminated, creating space in the cylinder head. The height of the engine can be designed to be lower. The requirements for pedestrian protection are easier to meet. The freed-up design height can also be used to integrate a special trans-

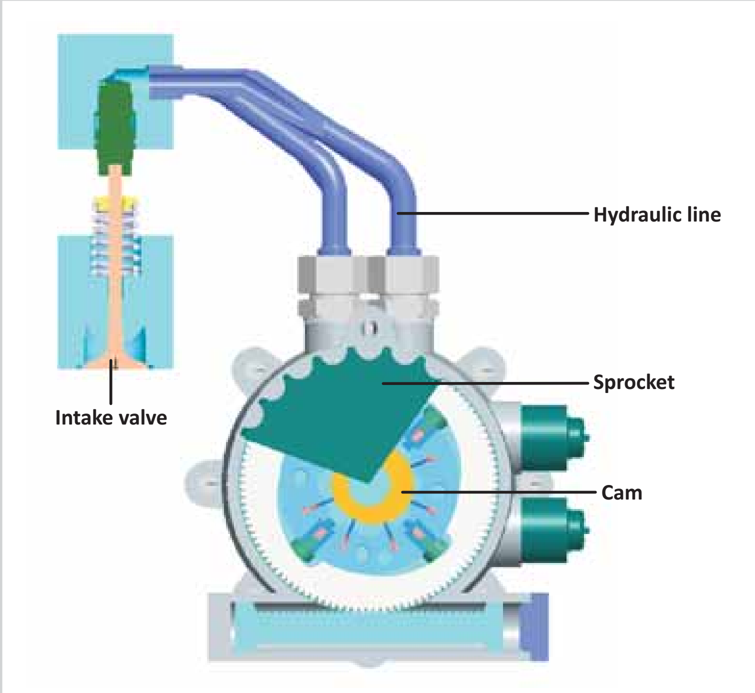

Figure 15 Central pump with flanged on electronic system and solenoid valves for influencing valve lift curves

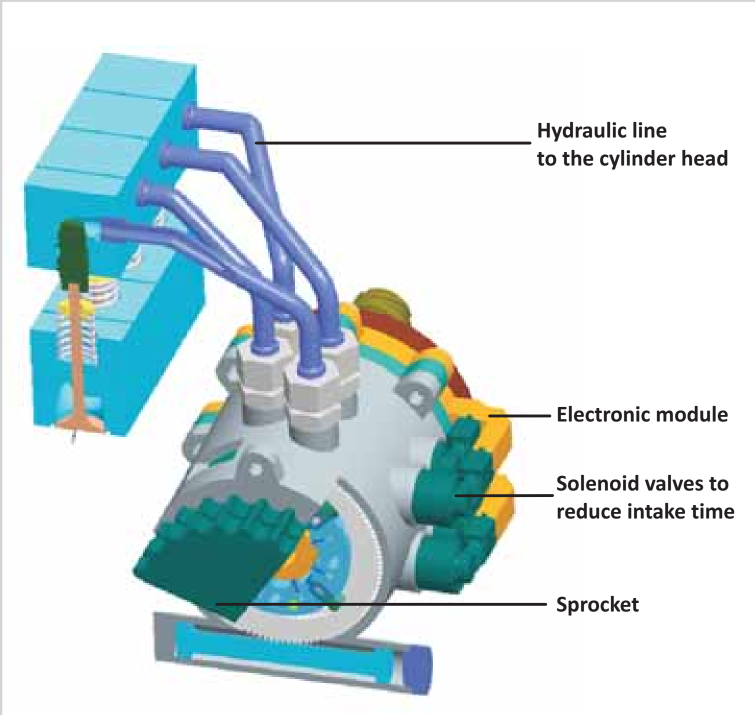

Figure 16 Side view of Figure 15

mission into the engine housing, as shown in the section below. Unlike other camshaft-less valve trains, the cam in the hydraulic central pump continues to determine the maximum valve lift curve that is designed as a reliable means of preventing the valve from colliding with the piston. As with the UniAir system, nearly all smaller valve lifts can be performed with reduced duration within this maximum curve.

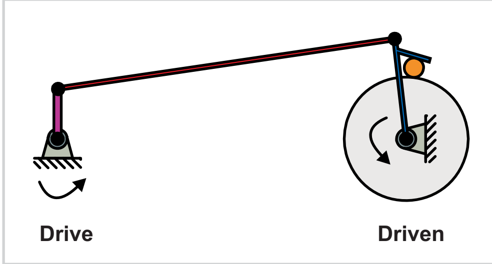

Figure 17 Crank gear allowing a continuously variable transmission from the start position

Transmission without housing

If there are any clearly defined interfaces in a vehicle, they are between the engine and the transmission. The usual structure automobile manufacturers use reflects this. Engine and transmission development are sovereign departments that are not consolidated until at the very top of the hierarchy. The interface looks like this: a flange the transmission is bolted on to and a crankshaft hole pattern that the flywheel is attached to. Especially in cases where an interface is not touched for decades, it might be worth considering to do away with this rigid limit and increase the degree of integration between the engine and the transmission more strongly. The following proposals should be regarded as ideas rather than specific design instructions.

The following considerations are based on the crank gear presented at the 2002 LuK Symposium [5]. At the time, when everyone was talking about continuously variable transmissions, a proposal was made to generate a back and forth motion impacting a one-way clutch via an adjustable crank mechanism (Figure 17). When changing the crank length settings on the drive, the starting speed is continuously variable from zero to maximum speed. The greater the amplitude of this driving motion, the stronger, i.e. faster, the one-way clutch is driven. Several overrunning alternator pulleys are driven at different times using different cranks to achieve an even output. This means that at least one one-way clutch is engaged at all times and the output speed is sufficiently smooth.

Things have been quiet for years now with regard to this proposal. New considerations are

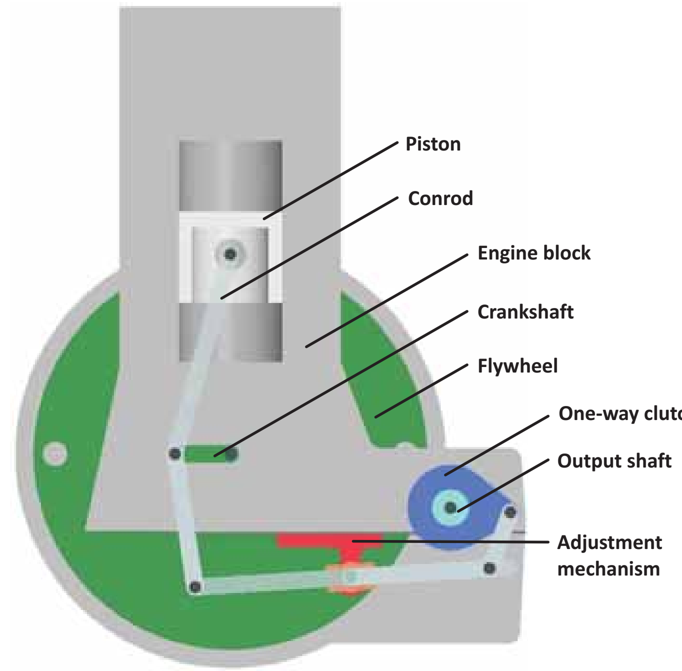

Figure 18 Design principle of a crank gear integrated in the engine crank housing

based on the fact that the crank gear requires a back and forth motion that, to be precise, is already present in the engine's cylinders. That is why, as shown in Figure 18, additional connecting rods have been introduced that operate a compensator whose contact point can be moved. This generates an up and down motion at the other end of the compensator that is continuously variable and drives a one-way clutch. The starting speed is continuously variable from zero to a maximum speed. This proposal would not only integrate most of the transmission in the crank housing but would also have an additional benefit. The piston force is almost directly transmitted to the compensator. Ideally, the crankshaft transmits only small torques and is essentially required to synchronize the individual cylinders and cranks.

Since it would be ideal to drive eight one-way clutches at different times to achieve an even torque transmission, two one-way clutch connecting rods are allocated to one engine connecting rod to activate the compensator in a four-cylinder engine. Figure 19 shows how the angle-displaced one-way clutch connecting rod bearings are arranged to the left and right of each engine connecting rod.

It should also be mentioned that a number of unresolved problems could stand in the way of integrating the engine and the transmission in this way. It is not certain whether one-way clutches can work with contaminated engine oil. A simple design of the one-way clutch mechanism cannot transmit an overrun torque. In the past, proposals have been made for using switchable one-way clutches, but this would increase the mechanical work required. That is why a combination with a hybrid drive could be an ideal solution. The electric motor handles the overrun torque, regaining braking energy and replacing the reverse gear.

Summary

This article treats some new design ideas associated with engines and transmissions. As mentioned above, these are not fully developed products or products ready for volume production but rather general development trends.

These efforts have been initiated by the fast changes that CO_2 problems and finite resources have brought about. The question whether and how fast e-mobility can replace today's drive methods will essentially depend on the changes today's technologies are capable of. And it may not only be small steps that bring sufficiently quick progress but larger steps. And to this end, unconventional methods must be found. This paper attempts to provide a small selection of methods that should be regarded as some initial suggestions.

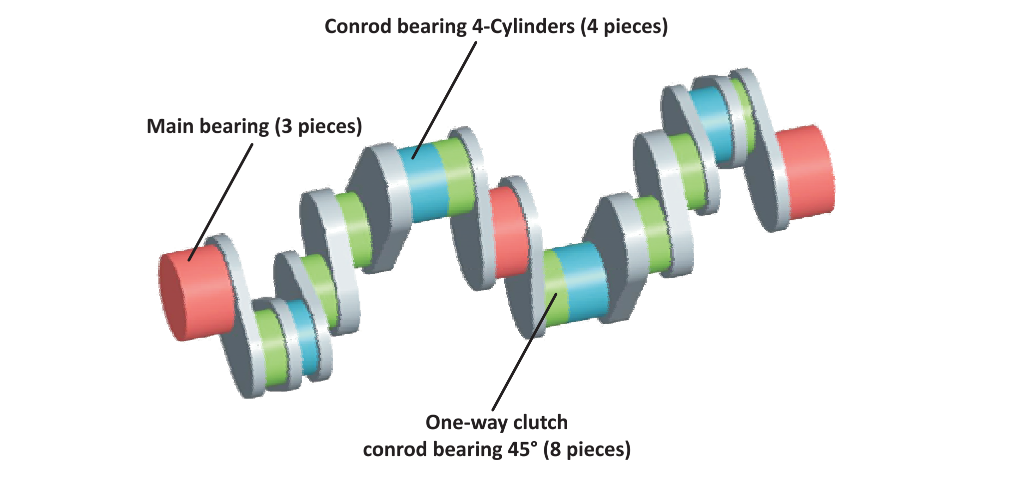

Figure 19 Crankshaft with additional bearings for transmission connecting rods

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