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Quo vadis hydraulic variable camshaft phasing unit?

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Electrification of motor vehicles

With the continuous development of electric components, and promoted to a great extent by the growing need for reducing fuel consumption, the electrification of motor vehicles has gained much importance during the last few years.

Hydraulic power steering systems have been replaced by electromechanical systems that must only be supplied with energy as the need arises. Electric water pumps are now being used in internal combustion engines to set the engine cooling to meet the requirements of the relevant operating point in the best way possible.

The question now arises as to whether a similar change in technology will take place in the case of variable camshaft phasing (VCP) systems from hydraulic camshaft phasing (HCP) units to electromechanical camshaft phasing (ECP) systems.

Requirements

In order to answer this question, it is necessary to investigate which functions are required and which solutions can be used to provide the customer with as little investment as possible. The speed under load characteristic diagram of the internal combustion engine is suitable for describing the most important functions of variable camshaft phasing systems (Figure 1). At a constant operating point, the accuracy of the setting of the required camshaft timing (represented by the timing angle a) is very important. The combustion methods of the future in particular, such as HCCI, place very high requirements on accuracy. In transient operation, when changing to another operating point in the characteristic diagram, the shifting velocity for setting the new camshaft timing is important. This influences, for example, the speed and the level of the increase in engine torque. Insufficient shifting velocities must be compensated by adjusting the ignition and fuel injection in engine applications. This can lead to disadvantages in terms of fuel consumption. A further important function is the degree of freedom in selecting the camshaft timing for the start of the internal combustion engine. During engine operation, camshaft timing might be set to a position that is unsuitable for engine start. Therefore, tight control of engine timing would be required during engine start. In the future, multiple camshaft positions might be required for different engine start-up conditions (e.g. hot and cold).

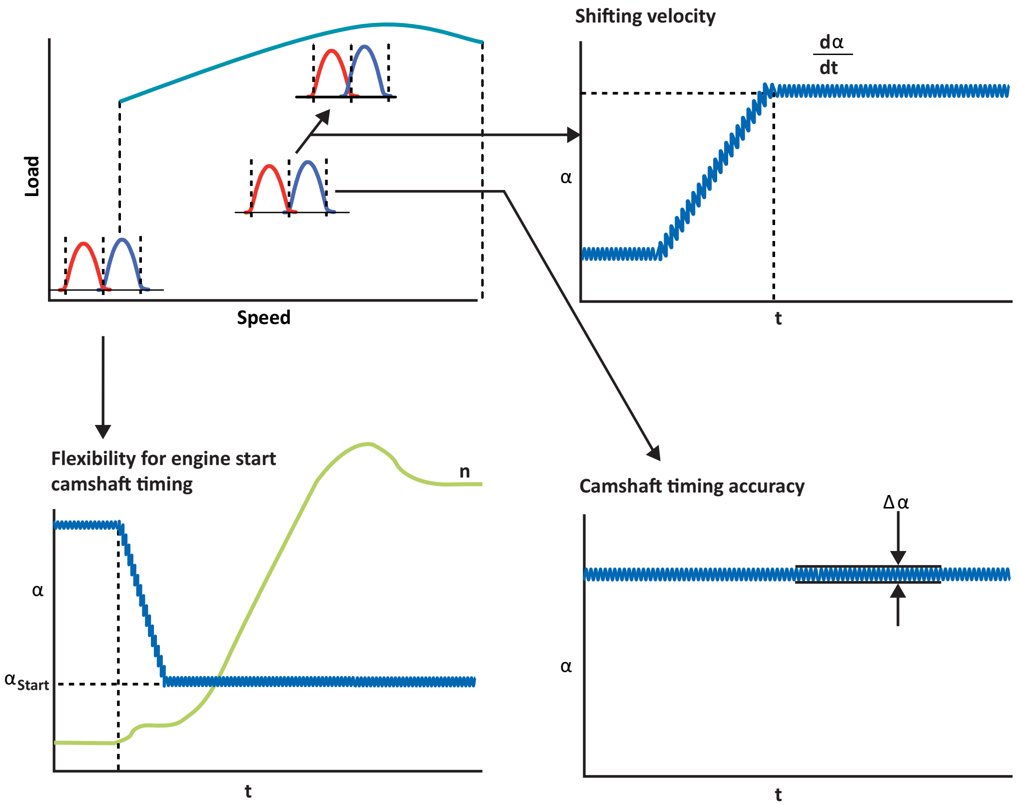

Figure 1 Function requirements

ECP and HCP system design

Description of the HCP system

The VCP system with a remote solenoid valve, a so-called cartridge valve, is comprised of a vane type variable camshaft phasing unit and an INA proportional solenoid valve (Figure 2). This type represents the standard design.

The solenoid valve is controlled by the engine control unit with pulse width modulation and is connected to the engine oil circuit. The system facilitates the continuous control of the variable camshaft phasing unit.

The unit is locked in a specified base position in advanced or retarded camshaft timing. In this case, the solenoid valve does not receive a signal from the engine control unit and remains in the de-energized state. Reaching the base position in advanced camshaft timing can be supported by a spring designed to meet the requirements of the application.

Figure 2 HCP with cartridge

Figure 3 HCP with central valve

The phasing unit can be driven by a timing chain drive or timing belt drive.

The VCP system with a central valve (ZVEN) is comprised of a vane type variable camshaft phasing unit, a central valve and a central solenoid (Figure 3).

The solenoid is controlled by the engine control unit with pulse width modulation. The central valve is connected to the engine oil circuit and simultaneously serves to locate the phasing unit on the camshaft. The system facilitates the continuous control of the variable camshaft phasing unit.

The unit is locked in the specified base position in advanced or retarded camshaft timing In this case, the solenoid valve does not receive a signal from the engine control unit, and remains in the de-energized state. Reaching the base position in the advanced camshaft timing can be supported by a spring matched to meet the requirements of the application.

The phasing unit can be driven by a timing chain drive or timing belt drive running in oil.

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Figure 4 Phasing unit with ECP

Description of ECP system

The ECP system is comprised of a three-shaft transmission that is mounted on the camshaft in the same way as a hydraulic variable camshaft phasing unit. The transmission output shaft is connected to the camshaft to be adjusted. The input shaft of the three-shaft transmission is connected to a compact 12 V electric motor. This adjusts the phase angle between the crankshaft and the camshaft. The third shaft, the transmission housing, is connected to the pulley or sprocket wheel of the primary drive. Figure 4 shows the arrangement.

$$\dot{\varphi} = 2\pi * \left(\frac{2\, n_{e\_motor} - n_{engine}}{\text{Transmission gear ratio}} \right) / 60$$

When a constant timing angle is desired, the drive shaft of the electric motor rotates at the same speed as the camshaft and transmission housing. If the phase angle is to be adjusted, the drive shaft of the electric motor must rotate more quickly or more slowly than the transmission housing in accordance with the required adjustment direction.

The difference in speed between the drive shaft of the electric motor or input shaft of the three-shaft transmission (n_{e_motor}) and transmission housing (n_{engine}), can be used to calculate the adjustment speed according to this equation, whereby the unit of n_{e_motor} and n_{engine} is rpm. Typical transmission ratios are between 40:1 and 100:1.

Comparison of systems

In Figure 5, HCP systems with cartridge or central valves and ECP units are compared in terms of functional and cost requirements.

The shifting velocity of HCP systems is determined to a considerable extent by the performance capability of the engine oil system. Compared to a cartridge valve solution, the central valve solution offers reduced pressure losses and therefore a greater potential for achieving higher shifting velocities at slightly increased efforts. On the other hand, ECP systems facilitate very high shifting velocities with greater flexibility over the RPM and temperature range; however this is associated with higher efforts.

In the case of HCP systems, the oil volume is confined within the chambers of the phasing unit in order to set the camshaft timing via the control valve. The accuracy of the camshaft timing control is essentially dependent on the compressibility of the oil and on any leakage points. The central valve solution offers a slight advantage, since the transfer of oil in the control lines between camshaft and cylinder head that is prone to leaks is omitted. ECP systems offer a stiffer connection between the drive wheel and camshaft.

HCP systems are usually equipped with locking mechanisms. This enables advanced or retarded camshaft timing to be used for engine start-up. Another camshaft timing can be selected after the build up of engine oil pressure. Electric VCP systems enable any camshaft timing to be set during engine start-up. ECP systems offer the highest degree of freedom when selecting the camshaft timing during start-up.

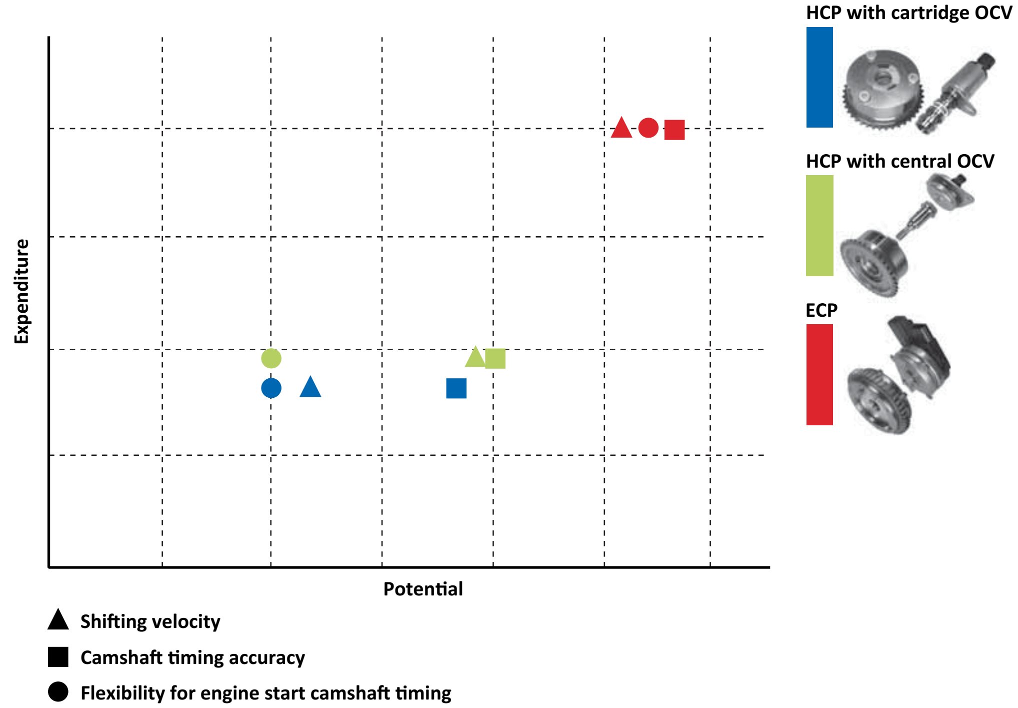

Figure 5 Comparison of systems

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Hydraulic accumulators as a potential solution

The gap that exists between hydraulic and electric variable camshaft phasing systems regarding the shifting velocity can be reduced using a hydraulic accumulator.

As part of a conventional hydraulic system with cartridge valves or central valves, the accumulator stores energy in the form of oil pressure during engine operation. This energy is returned to the phasing system during the phasing process. To fulfill this task, the pressure accumulator shown in Figure 6 is located before the hydraulic control valve and is connected with the oil supply line. A one-way valve is located between the accumulator and control valve that prevents the engine oil from the phasing system from flowing back into the engine and accumulator. This increases the operating range of the phasing system beyond the current limits.

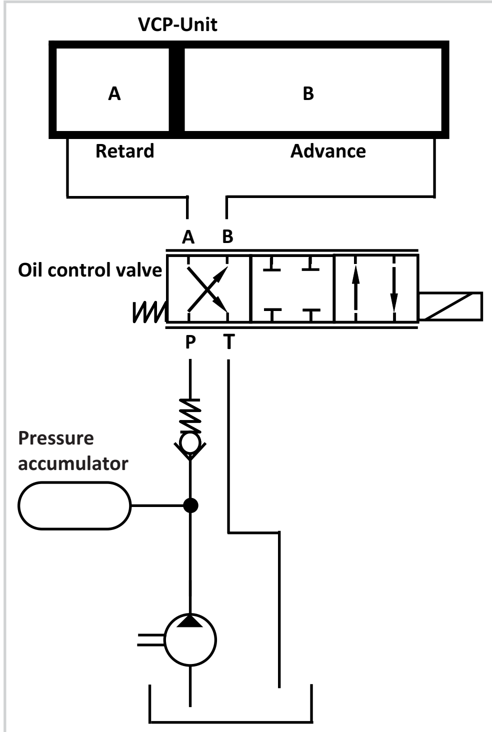

Figure 6 Hydraulic accumulators as a potential solution

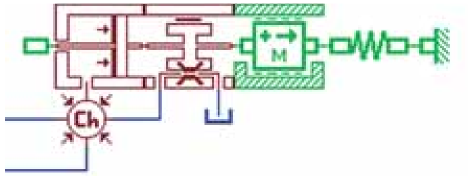

Figure 7 Simulation of the effect of an accumulator in an oil system

A simulation model was constructed to estimate the shifting velocity potential and to optimize the pressure accumulator. The pressure accumulator that is part of this model is shown in Figure 7. In simple terms, the pressure accumulator can be described as a spring-mass system which is subjected to oil pressure. The system reaches equilibrium when the compressive force is equal to the spring force. A series of input parameters were varied in order to determine the optimum design criteria of the pressure accumulator. These were primarily the piston mass, the preload and the spring rate of the compression spring. The secondary influencing parameters such as leakage and friction were determined in accompanying component tests and were included in the simulation as constant values.

The result of the simulation is shown in Figure 8. It shows the shifting velocity of the variable camshaft phasing system in relation to the existing oil pressure in the cylinder head. In the upper part of the diagram, the system shifts from advanced to retarded camshaft timing. In contrast, the lower section of the diagram shows the adjustment from retarded to advanced camshaft timing.

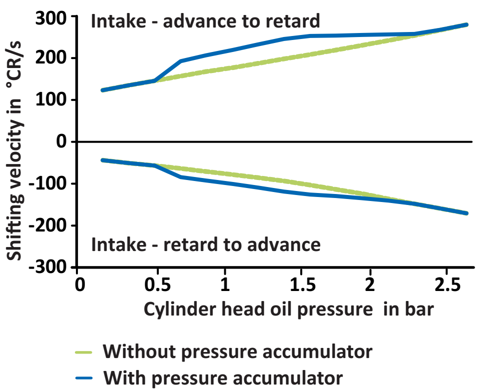

Figure 8 Improvement in timing velocity at 90 °C

Figure 8 also shows the potential for improvement by using an accumulator over a wide oil pressure range. The characteristics of the compression spring influence at which pressure range and at which level this potential exists. In this example, the compression spring was adjusted so that the charging process of the pressure accumulator begins approximately at 0.6 bar and the piston reaches its final position approximately at 1.5 bar relative oil pressure. The simulation results were verified by measuring the shifting velocity of the variable camshaft phasing system on a fired 2.0 l 4-cylinder gasoline engine. Figure 9 shows an example of a measurement of the intake camshaft phasing unit that was taken at 910 1/min engine speed, 90 °C oil temperature in the cylinder head and at zero load.

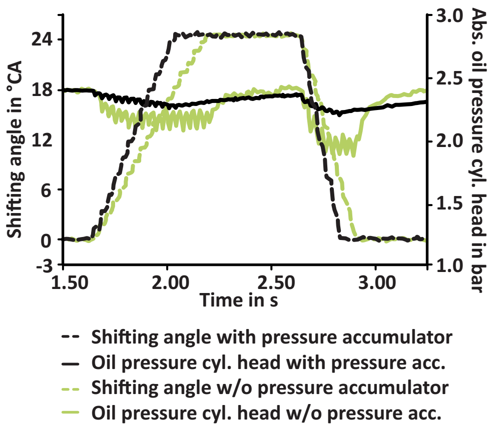

Figure 9 Test results of passive accumulator at idle, 90 °C and zero load

The measurement shows the angular position of the variable camshaft phasing system with and without an accumulator and the associated oil pressure in the cylinder head in relation to time. The shifting velocity can be derived from the angular position and the time. Comparing both systems shows that the pressure accumulator system (black curve) reaches the end stop in the stator earlier than the system without a pressure accumulator (green curve). This advantage in shifting velocity does not depend on whether the shift occurs away from the base position (0° camshaft) or towards the base position. The frictional torque of the camshaft alone causes the shift to be unsymmetrical in both directions.

The answer to the question as to why the variable camshaft phasing system with an accumulator facilitates faster adjustment can be found when comparing the oil pressure of both systems. In the case of the system with the pressure accumulator (black curve), the oil pressure decreases more slowly during adjustment than in the system without a pressure accumulator (green curve). This is due to the fact that the majority of the required oil volume is provided by the pressure accumulator and therefore more energy is made available to the phasing system for the phasing operation. The reduction in oil pressure that occurs here is primarily determined by the design of the compression spring. The greater the oil volume that can be pushed out of the accumulator during a difference in pressure, the lower the decrease in oil pressure in the engine.

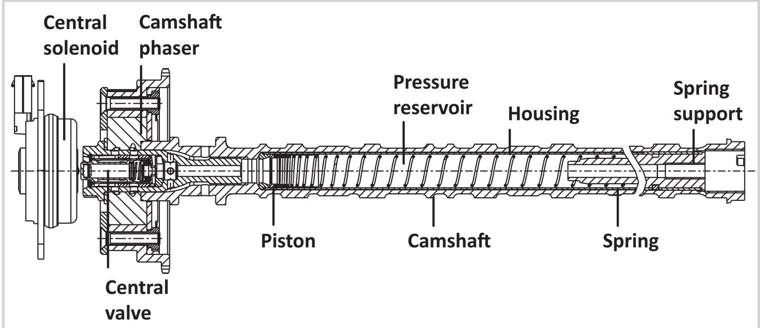

Figure 10 Design of the passive pressure accumulator (switching position 1)

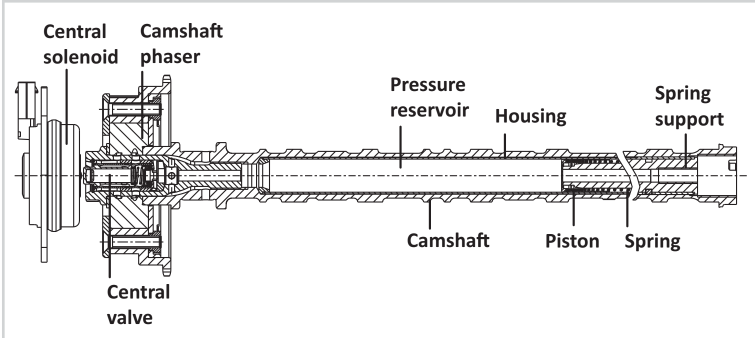

Figure 11 Design of the passive pressure accumulator (switching position 2)

Description of the "passive" pressure accumulator system in the camshaft

The passive pressure accumulator shown in Figures 10 and 11 is comprised of a cup-shaped piston, compression spring, guidance element and a thin-walled housing with a closing plug mounted on the end face. The piston is guided inside the housing. It converts the oil pressure provided by the oil pump during filling into potential energy that is stored in the compression spring. The movement of the piston is limited by two stops. In the released base position, the piston strikes the inside of the closing plug and in the end position, it strikes the guidance element.

The compression spring forces are characterized by the preload in the base position and the spring rate that defines the increase in force via the travel of the piston up to the end position. The guidance element at the back of the housing guides the compression spring. The guidance element has a central bore. This bore facilitates the ventilation of the space in which the compression spring operates. At the same time, the bore allows oil leakage to drain into the tank.

The assembly is pushed into the camshaft on the side not facing the phasing unit, so that it seats against the stop and is located axially with a screw plug. The screw plug is hollow so that the connection between the central bore in the guidance element and the engine compartment is not interrupted. Both ends of the pressure accumulator unit housing are supported by conical interfaces. The circumferential gap generated by centering prevents any deviations in the cylindrical shape of the camshaft inside diameter from being transferred to the piston running surface. This ensures reliable operation in the internal combustion engine.

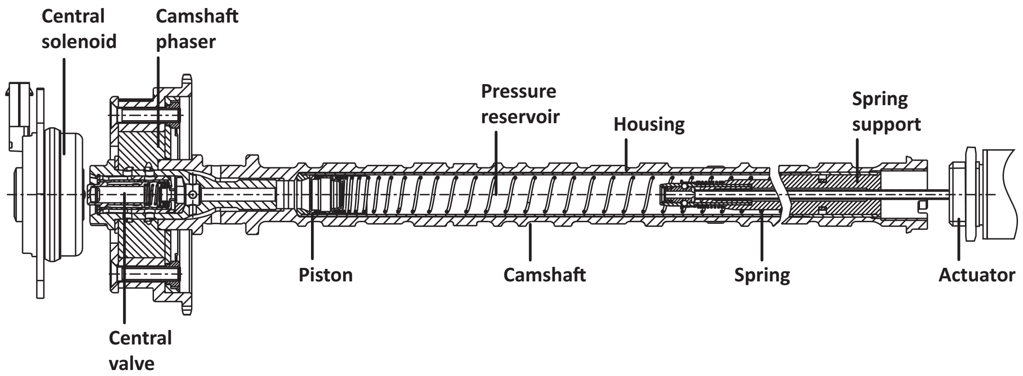

Figure 12 Design of the active pressure accumulator (switching position 1)

Description of the "active" pressure accumulator system

The active pressure accumulator is comprised of a housing, compression spring and a cylindrical piston also used in the passive system. It also includes a switchable coupling mechanism securely located on the camshaft that creates a detachable lock for the piston when the oil reservoir is full. This is located at the rear of the accumulator as shown in Figure 12.

When the engine is switched off and at zero engine oil pressure, the engine oil remains in the oil reservoir, and is not, as in the case of the passive pressure accumulator system, immediately squeezed out via the leakage points on the engine. Since the oil pump requires a certain amount of time after the engine is started to produce the oil pressure required for the phasing of the HCP unit, discharging the pressure accumulator can immediately facilitate a phasing operation from the base position. The active pressure accumulator is, for example, suitable for engines with stop-start systems. The discharge procedure when the engine is started is decisive for the dimensioning of the working pressures of the active accumulator. The required working pressure level is higher than the optimum pressure level of the passive pressure accumulator that would be necessary to improve the shifting velocity during hot idling (Figure 13).

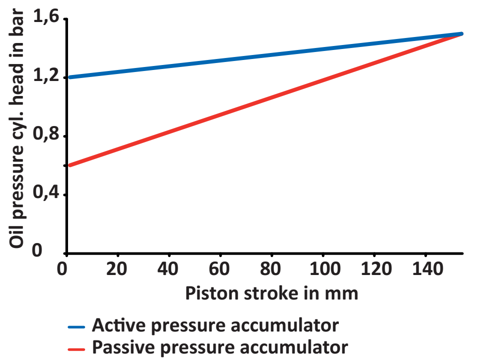

Figure 13 Comparison of the working ranges of active and passive accumulators

The locked condition shown in Figure 14 is used to describe the function. To unlock the piston, an electromagnetic actuator located on the cylinder head pushes a rod against a return spring on the switching pin with a circumferential groove.

As soon as the balls can move in a radial direction into the circumferential groove they are pushed inward and radially by means of the compression spring force which releases the piston. The compression spring force acts on the balls via the piston and a plate pressed into the piston. In order to prevent the balls from falling out through the bores, another compression spring (sliding plate spring) slides the sliding plate over the bores after the piston is released as shown in Figure 15.

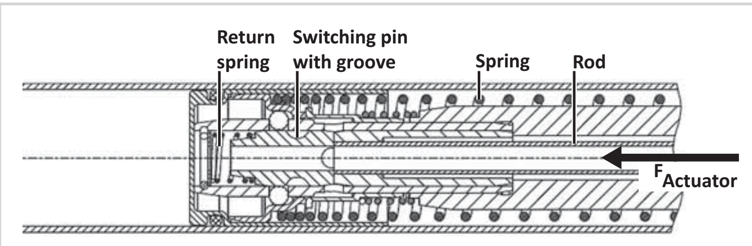

Figure 14 Detailed view of coupling mechanism in locked condition

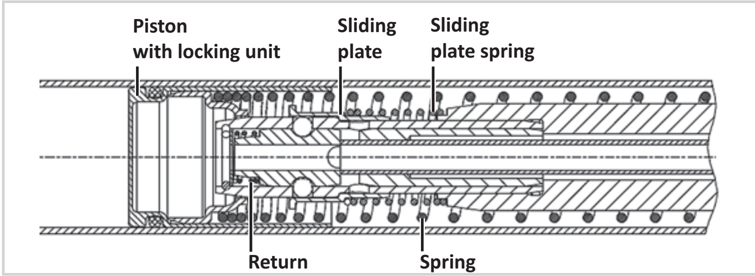

Figure 15 Detailed view of coupling mechanism in unlocked condition

If the accumulator fills with oil, the piston automatically snaps into the coupling mechanism. During this process, the piston locking unit pushes the sliding plate back against the sliding plate spring until the base of the piston mates to the coupling mechanism. In this position, the switching pin is moved in an axial direction via the return spring and the balls are pushed outwards from the groove in a radial direction, i.e. the piston is secured. During this process, the rod and the actuator are moved back to their original position. The piston can be unlocked again by briefly feeding the actuator with current.

Test results for the active accumulator system

The measurements on the fired test engine show an immediate increase in engine oil pressure in the camshaft when the engine is started, due to the discharging of the pressure accumulator. This means that the shift from the base position can occur earlier than without a pressure accumulator. In the case of short engine downtimes, e.g. when waiting at traffic lights, the pressure is built up in the camshaft with the accumulator. In the case of long downtimes, for example, when the vehicle is parked overnight, the oil reservoir empties to a great extent via the leakage points of the camshaft bearings and the variable camshaft phasing unit. This means that in this case, no support can be provided when starting the engine. Figure 16 shows an example of an engine start at an engine oil temperature of 40 °C after 10 minutes downtime with and without support from a pressure accumulator. When comparing the pressures in the camshaft (red curves) the rapid increase in pressure can be seen in the system with a pressure accumulator. The timing angle is shown in black. The phasing process from the base position begins earlier with the accumulator. In the system without the accumulator and actuated control valve, the timing angle oscillates from the beginning of the phasing process due to poorer hydraulic clamping.

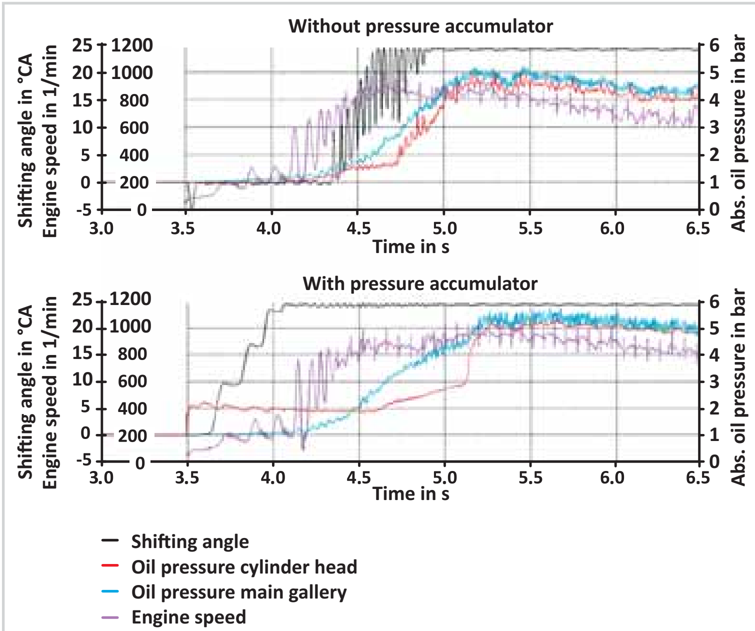

Figure 16 Results of test with active accumulator when starting engine

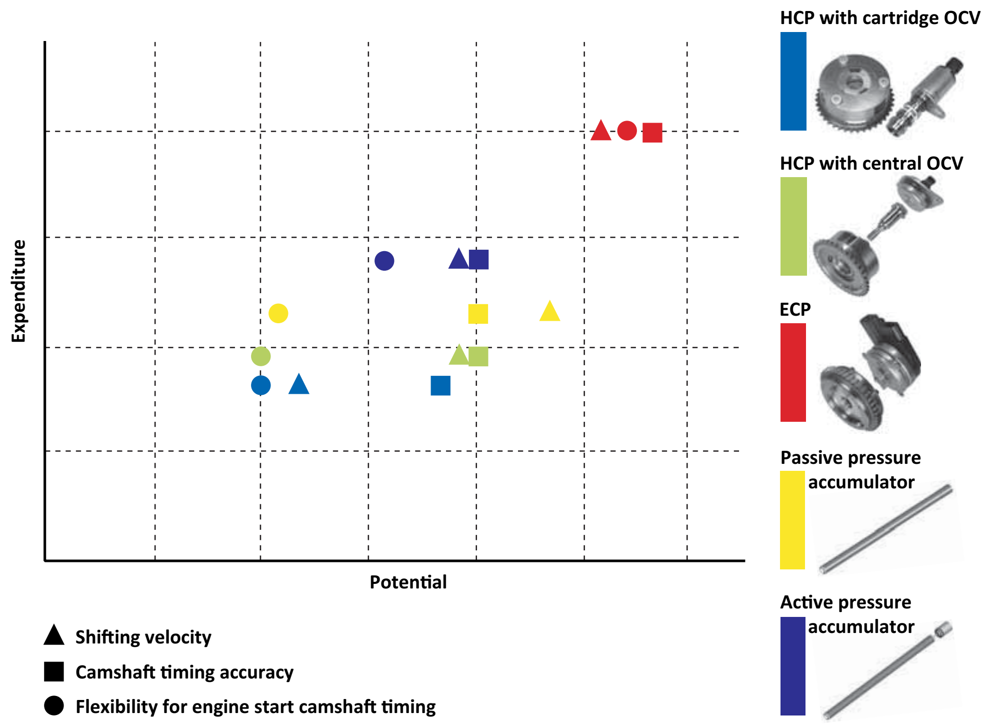

Figure 17 Positioning of the accumulator in a system comparison

Positioning of the accumulator in a system comparison

If HCP systems are upgraded to include a passive accumulator, the gap between these systems and ECP systems, in terms of the achieved shifting velocity, can be reduced to a great extent (Figure 17). The additional investment involved is moderate. The accuracy of the camshaft timing remains unaffected. Since the decrease in oil pressure is delayed when switching off the engine due to the accumulator releasing pressure, reaching the “desired camshaft timing” is supported during engine shut off. Therefore, the potential for selecting the starting camshaft timing is improved in comparison with conventional VCP systems.

Using an active accumulator enables the selection of the starting camshaft timing to be improved further. However, this is associated with higher costs. The shifting velocity and the accuracy of the camshaft timing control of hydraulic systems are also influenced.

Conclusion

The transition from HCP to ECP systems is determined by the function requirements of internal combustion engines. ECP systems will replace HCP systems as soon as the overall costs for providing the required functions with hydraulic solutions exceed the costs associated with electric solutions. The development of combustion processes in the future will have a considerable influence on when this transition occurs.

Literature

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[2] Strauss, A.; Busse, M.; Schäfer, J.: Late Intake Valve Closing Concepts – A Challenge for Innovative Stop-Start Functions, Haus der Technik 2009, Conference Variable Valve Control