HYDAC INTERNATIONAL

Double Piston Accumulator
Innovative hydraulic accumulator for hydraulic hybrid drives

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Abstract

Hydraulic hybrid systems frequently use motor-pump units which can be ‘swiveled through zero’ for sensitive and efficient proportioning of the input or output torques. High-pressure accumulators are used in these systems to store the energy. Low-pressure accumulators are frequently used on the suction side of hydrostats in order to consistently guarantee suitable suction conditions when in pump operation. However, this configuration has the disadvantage of bringing along an increase in pressure in the low pressure accumulator as it is being charged, which has negative effects on the usable energy and the performance of the overall system.

The HYDAC double piston accumulator represents an accumulator solution which keeps the pressure level constant on the suction side of the hydrostats. This allows the full performance potential of the high pressure accumulator to be exploited. There are also further advantages resulting from the special system design in comparison to conventional systems with reference to the energy balance, the required installation space and the weight.

Based on actual load cycles from operational experience, the effects on the energy and power balances as well as the system advantages for potential users of hybrid applications are demonstrated. Finally, a preview of possibilities for lightweight designs to be found in the compact double piston accumulator design is given.

1 Introduction

The increasing sensibility for energy efficiency in the mobile and stationary drive technology is leading to the increased usage of hybrid solutions in addition to energy optimisations. However, it has become clear that the hybrid concept must be adapted extremely close to the specific load profile of very different machines. For this reason, there is a series of different hybrid concepts, which can differ substantially from one another with reference to the form of energy (electrical, hydraulic, mechanical), the structure (parallel, series, power split) or the arrangement (primary / secondary side).

The strength of hydraulic hybrid systems therefore lies in the enormous power density, meaning the characteristic of being able to implement quantities of energy within very short intervals of time [1]. This means that hydraulics are predestined for applications in the mobile and stationary drive technology where highly-dynamic working cycles occur. This is a robust, tried and tested drive technology which is consistently represented and which can be relatively easily extended by using hydraulic accumulators.

Hydraulic accumulators are designed according to the relevant regulations, such as for example the Pressure Equipment Directive [2], which guarantees a maximum of security. Using miscellaneous technical devices (pressure relief valves, burst discs, fuses etc.), hydraulic accumulators can be relaxed reliably and therefore switched in a de-energised condition. This, too, makes hydraulic accumulators extremely safe energy storage devices.

Sophisticated hybrid solutions use hydrostatic variable displacement units for the sensitive application of the drive torque or for power regulation. In order to avoid pressure losses through additional reverse valves, hydrostats are frequently used to switch between operation of the motor and of the pump, which allows "swivel through zero". Here, for example, displacement units which are applied in a closed circle are suitable. In order to guarantee that they function correctly (good degree of efficiency, prevention of cavitation, low sound emissions etc.) in hybrid operation as well, the suction side of these units is preloaded. This results in a configuration as represented in Fig 1. Here, the hybrid unit is equipped with high pressure (HP) and low pressure (LP) accumulators.

However, the additional use of a separate LP accumulator has disadvantages in the implementation of energies and powers as well as required installation space and weight. Finally, the management of oil volumes, meaning the monitoring of the filling levels via pressure and temperature measurement, or filling level measurement using displacement measurement technology, means a substantial amount of extra effort and increased complexity.

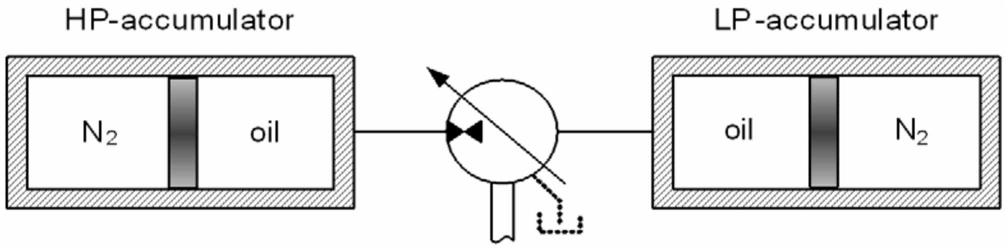

Fig. 1: Sketch of the principles of a conventional HP- and LP-accumulator system

The HYDAC double piston accumulator system (DPA system) represents an innovative technological solution which compensates for the disadvantages of the separate LP-accumulator and at the same time guarantees ideal suction conditions on the displacement unit. The DPA concept has already been successfully tested on an IFAS demonstrator car at the RWTH Aachen, and presented within the scope of the 7th IFK [3]. In addition, HYDAC has decades of experience in the construction, development, production and utilisation of double piston accumulators on other (non-hybrid) applications, such as for example deep sea applications.

The structural design and the method of function of the double piston accumulator is explained below. Based on actual load cycles from operational experience, the effects on the energy and power balance as well as the system advantages for potential users of hybrid applications are shown.

2 Structural design / method of function of double piston accumulators

The characteristic properties of the double piston accumulator consist of the rigid coupling of the pistons from the HP and LP sides (see Fig. 2). Here the pistons are fixed to a rod. The resulting four working chambers are directly coupled to each other with reference to their volumes. The energy accumulation takes place as on a conventional system through the compression of gas (here nitrogen N_2). Separated from this area by the HP piston, the high pressure oil (HP oil) is stored in a chamber.

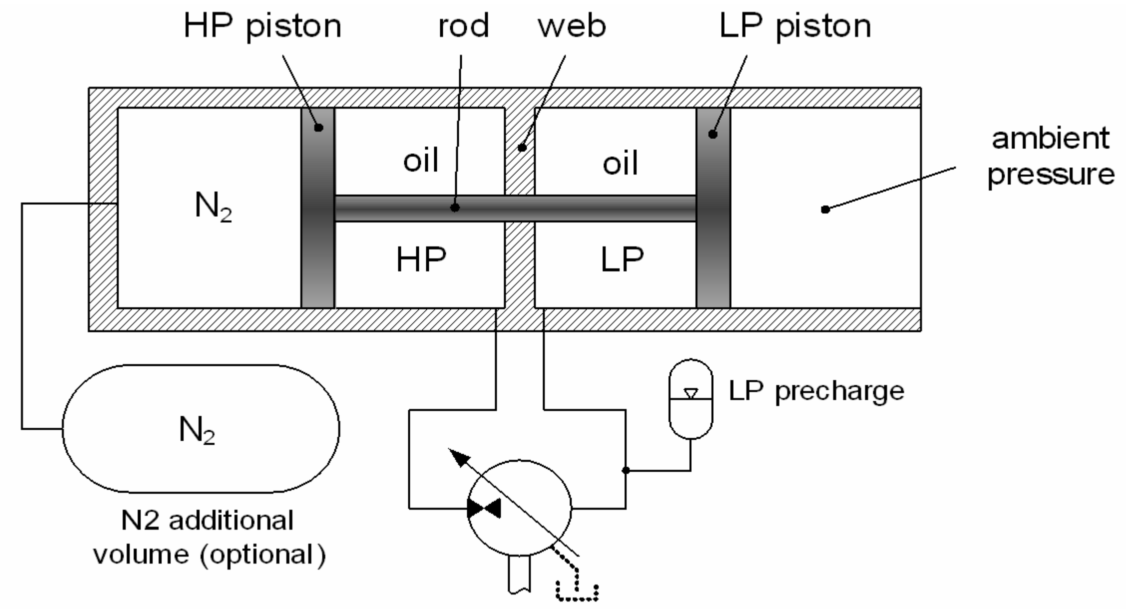

Fig. 2: Sketch of the principles of a double piston accumulator

The web separates the high pressure side from the low pressure side. The latter is then divided by the LP piston into the LP oil chamber and a "depressurised" chamber (ambient pressure). The allocation of the chambers described here represents the preferred design for this application. Other distributions are also possible.

When the motor is operating, the HP oil chamber is emptied and the LP oil chamber is filled. The piston rod unit then moves to the right. During pump operation, the process runs in the opposite direction. As in the HP oil chamber, movement of the piston rod unit in the LP oil chamber is equalised via the inflow or outflow of oil. By preloading it with the aid of a small diaphragm accumulator, the pressure on the suction side of the pump is kept constant at the required level (preload LP). The diaphragm accumulator only serves to compensate for temperature effects.

Leakage effects on the motor-pump unit are to be ignored during this consideration; they can be subsequently compensated for by using a suitable feed device.

The system-related advantages resulting from the special design are explained based on the balance of forces in the double piston accumulator (see Fig. 3).

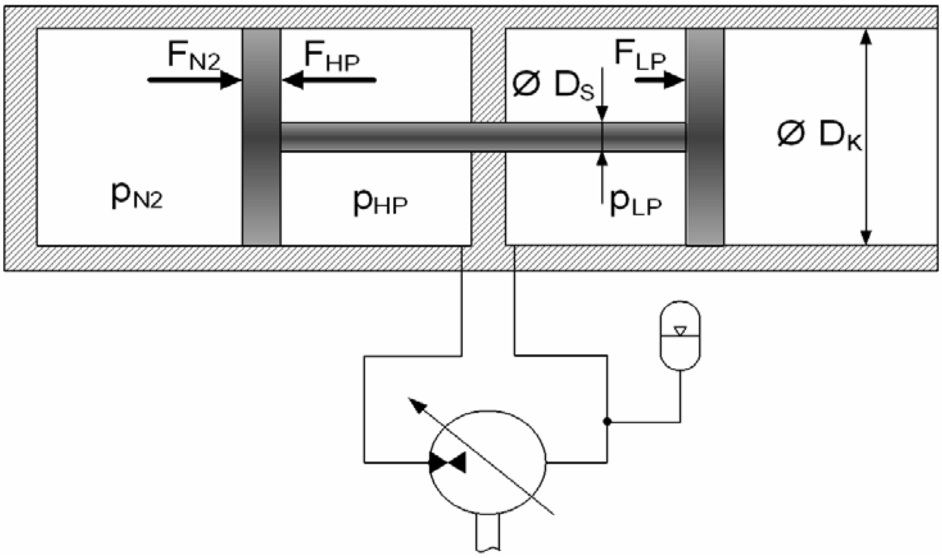

Fig. 3: Balance of forces on the double piston accumulator

The following applies:

$$F_{LP} + F_{N_2} - F_{HP} = 0 \tag{1}$$

It follows result for the oil pressure on the HP side (p_{HP}), with the pressure in the LP side p_{LP}, the piston diameter D_K and the rod diameter D_S and the pressure in the nitrogen chamber p_{N2}.

$$p_{HP} = \frac{{D_K}^2}{{D_K}^2 - {D_S}^2} \cdot p_{N_2} + p_{LP} \tag{2}$$

Equation (2) shows that the sum of the oil pressure on the HP side is compiled of two summands. The first summand represents the pressure transfer between the N_2 chamber (p_{N2}) and the HP chamber (p_{HP}). The factor before p_{N2} is always larger than 1, but only exceeds this value slightly due to the typical ratio between D_S and D_K (D_S<<D_K). The second summand is the constant pressure in the LP oil space, which superimposes the high pressure.

The typical pressure profile for a DPA system is qualitatively represented in Fig. 4. p_{HP} and p_{LP} are applied over the associated oil exchange volume (ΔV_{Oil}) and p_{N2} is applied over the appropriate gas volume (ΔV_{N2}). These reference volumes are slightly different due to their different piston surfaces. As explained above, p_{LP} is constant and freely selectable with reference to the suction side requirements of the pump. p_{N2} follows the appropriate pressure profile for the real gas behaviour and heat transfer processes to the surroundings. p_{HP} runs nearly parallel to this in the relationship shown in equation (2).

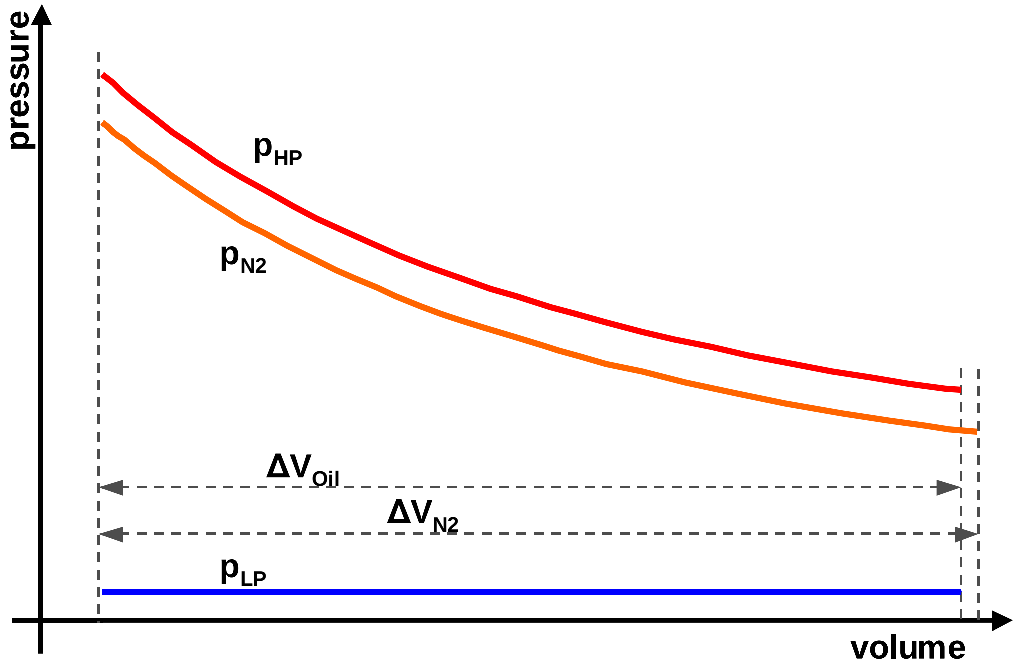

Fig. 4: Typical pressure profile for a double piston accumulator system

3 System comparison

The following simulation system comparison is intended to compare the energy and power balance of the conventional HP-LP system with that of the double piston accumulator. In order to simulate the various accumulator systems, the ***A**ccumulator **S**imulation **P**rogram (ASP), which is free for download* [4] and an appropriate HYDAC accumulator model for Matlab/Simulink were used. Knowhow from numerous scientific studies was integrated into both simulation models, in which the storage behaviour, taking the real gas behaviour and the thermodynamic processes into account, was precisely mapped. The comparison of the energy and power balance for both systems was carried out based on actual simulation examples.

3.1 Energy and power balance of a partial cycle

In order to indicate the principle differences between the two accumulator systems, a partial cycle (discharging process) is considered below. A summary of the simulation parameters used is shown in Table 1.

Parameters	V_0	ΔV	p_0-HP	p_0-LP	p_1-LP	$p_{Start\text{-}oil}$	$P_{Start\text{-}N2}$	$\Delta t_{discharge}$	$Q_{discharge}$
Standard	50 L	15 L	170 bar	22.5 bar	25 bar	360 bar	360 bar	12 s	75 L/min
DPA				-	25 bar = const.		329 bar		

Tab. 1: Simulation parameters

With reference to the accumulator parameters, a configuration was selected based on practical experience, whereby the HP nitrogen volume totalled 50 L. The preload pressure p_0 on the high pressure sides was assumed to be 170 bar each. Considering a discharge process (e.g. for the acceleration of a vehicle or slewing gear), the start pressure totals an exemplary sum of 360 bar.

On the LP side, the start pressure, which is constant at all operating points on the DPA system, is 25 bar. The preload pressure in the LP accumulator is 22.5 bar in this example. 15 L of oil are removed within 12 seconds from the high pressure section in the accumulator systems.

The simulation results can be found in Fig. 5. Here, in both cases the pressure in the HP and LP accumulator is applied over the offset *oil volume* (ΔV_{oil}).

Evaluation of the pressure profiles on the LP sides show that the low pressure of the DPA is constant. A natural increase in pressure can be observed on the standard system. The maximum pressure in the standard LP accumulator at the end of the loading procedure totals 43.5 bar. The end pressures in the high pressure chambers also differ substantially.

The cause of the steep drop in oil pressure on the HP side of the standard system in comparison to the DPA lies in the lower start pressure level on the nitrogen side of the DPA (329 bar) in comparison to the HP accumulator (360 bar). This means that the DPA works along a flatter characteristic curve than the conventional high pressure accumulator with the same oil volume exchange.

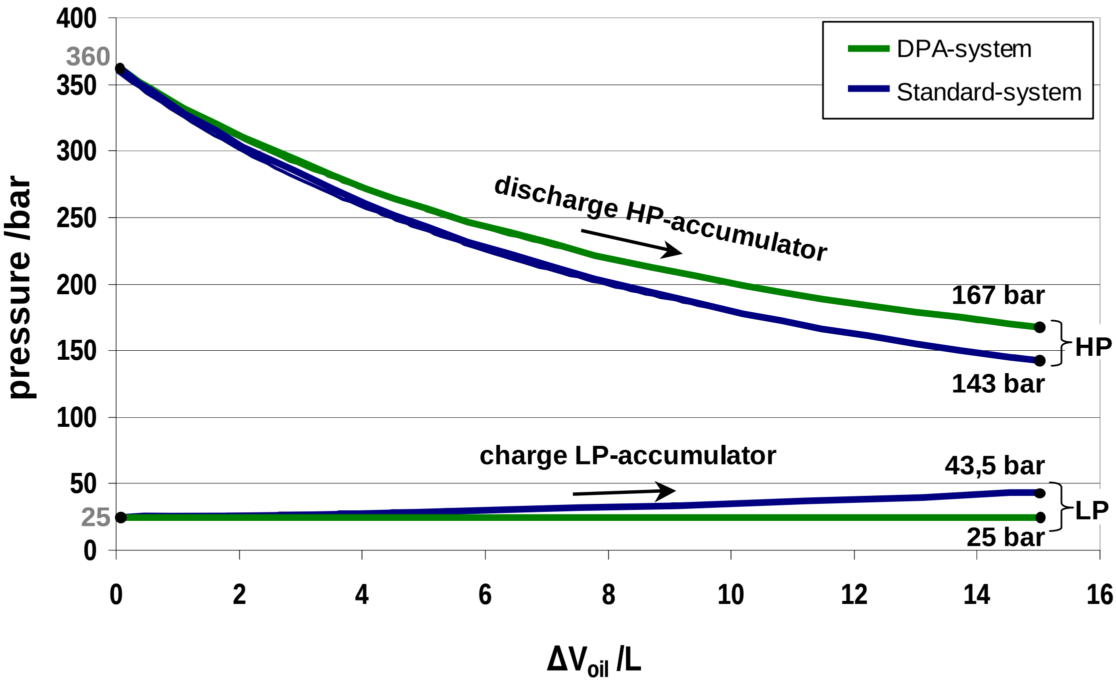

Fig. 5: Oil pressure profiles of the standard and of the DPA system (discharge procedure)

The differential pressure Δp (HP – LP) on the motor-pump unit is decisive for the power and energy output or intake of the accumulator systems. Fig. 6 shows the profile of the differential pressures derived from Fig. 5. The differential pressure on the DPA system is **42 %** higher at the end of the discharging process in comparison to the standard system. Accordingly, the torque or power output (e.g. for aiding the acceleration of a vehicle or slewing gear) on the DPA is also 42 % higher.

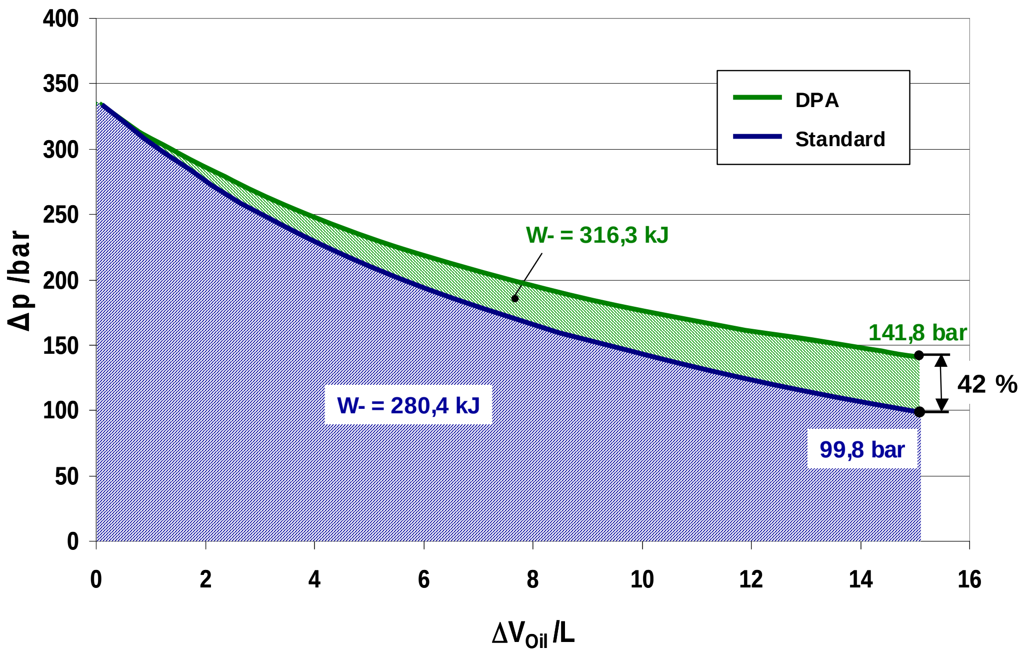

Fig. 6: Differential pressure profiles with energies and end pressures

The area below the pressure differential curve, which is defined according to equation (3), can be used in order to evaluate the energy balance:

$$W = \int p \, dV \tag{3}$$

According to this, the energy extraction from the DPA at 316 kJ is approximately **13%** higher than from the conventional system with the HP and LP accumulators.

3.2 Evaluation of the DPA in a practical application

To be able to evaluate the DPA as practically as possible, an actual load cycle from an exemplary application is applied. Here, a city bus with a hydraulic parallel hybrid is considered. In Fig. 7, the associated drive configuration is represented with a conventional HP-LP system (with a 50 L volume each).

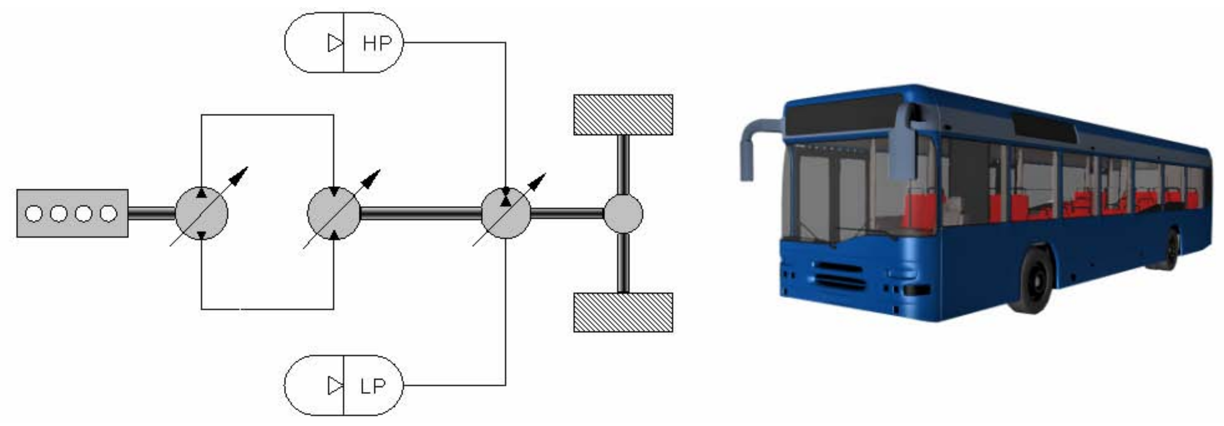

Fig. 7: Function diagram, city bus

Fig. 8 shows an associated exemplary measuring printout. This is a record of the travelling speed and the pressure in the high pressure accumulator during a scheduled drive over a distance of approx. 2.5 km and a time of approx. 10 minutes.

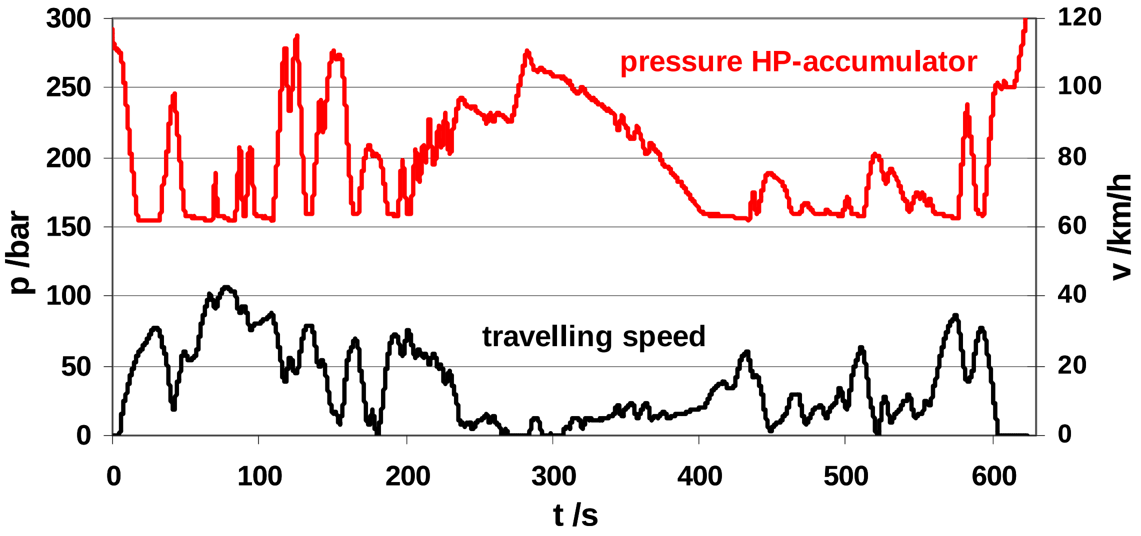

Fig. 8: Pressure / speed profile

The appropriate volume flow rates were first calculated from the HP and LP pressures measured on the city bus hydraulic standard hybrid system using a pressure-controlled hydraulic accumulator model made by HYDAC in Matlab/Simulink. From these volume flow rate profiles, the energy and power balances for both systems could be simulated and compared using the volume flow rate-controlled hydroaccumulator model. Fig. 9 shows the resulting pressure difference resulting from this and available on the motor-pump unit in the p-V-diagram. Because these measurements result from an actual road traffic situation, where no thermodynamic state of equilibrium can take place in the accumulator, each of them is an array of curves with appropriate hysteresis.

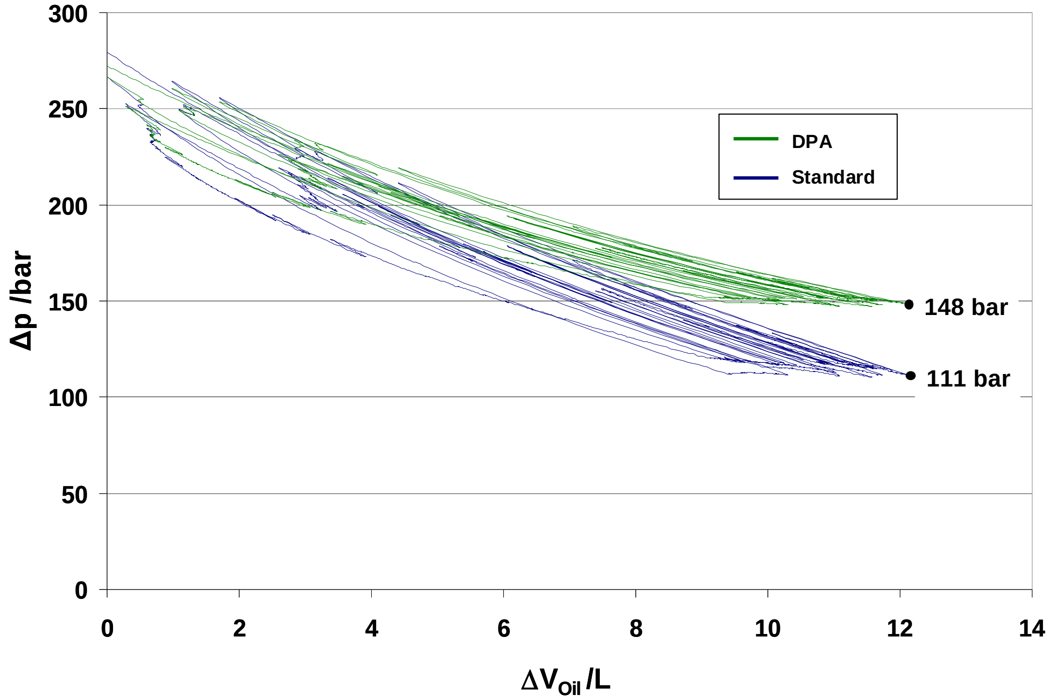

Fig. 9: Pressure difference profiles on the city bus cycle

Comparable with the simulation results from Fig. 6, the pressure differences on the standard system remain at a lower level compared to the DPA system. This becomes more and more apparent as the HP accumulator empties. When the HP accumulator is discharged, the DPA provides a system advantage of 37 bar with a pressure difference of 148 bar.

When considering the energy capacities, it has to be differentiated between the absorbed and the transferred amounts of energy in the driving cycle. W_+ represents the sum of all the energies absorbed into the accumulator systems during the cycle (loading processes). Accordingly, W_- represents the sum of all transferred energies (discharging processes). If we carry out this balance for the cycle presented in Fig. 9, we can see that the double piston accumulator is able to absorb or transfer **14 %** more energy in comparison to the standard system. With reference to fuel savings on the hybrid systems, which currently lie at approx. 20 % in comparison to the conventional drive train, this means a renewed optimisation of 3 %.

In addition to the inspections of energy capacities, the degree of effectiveness on the two hydraulic accumulator systems was also considered. These each lay at approx. 98 % due to the dynamics of the cycles, which is not atypical for the city bus application under consideration.

3.3 Comparison of construction space and weight

As can be seen from the structural design in Fig. 2, in comparison to the standard HP-LP solution, part of the LP accumulator is no longer required on the DPA, which reduces the weight and possibly also construction space. Due to the coupled arrangement, the oil side lids on the HP and LP chambers are joined into a common separating web. This allows the integration of further hydraulic components such as, for example, pressure relief valves and shut-off valves, as well as filters for guaranteeing the required oil purity. The short, direct connection between the HP and the LP chambers provides safety-relevant system advantages, as no external piping is required. In addition, pressure and/or temperature sensors as well as length measurement systems for monitoring the accumulator filling level can easily be integrated. An example of this is described in [3]. The floor side lid on the low pressure side is not required. Finally, alone the removal of the low pressure side nitrogen (50 L with 20 bar) means a weight reduction of approx. 1.2 kg. Set against this is only the additional weight of the piston rod, which simply has to transfer the tension forces between the pistons.

In order to optimise the weight even further, the external diameters of the DPA can each be adapted precisely to the max. occurring working pressures on the high and low pressure chambers, whereby a lightweight construction design with circumferential wrapping (e.g. carbon fibre) can be realised.

The DPA solution is particularly interesting for applications with relatively low oil offset volumes and large nitrogen volumes, as the low pressure space could hereby be made relatively short. Packaging advantages, on the other hand, could be achieved by using a downstream nitrogen bottle, whereby on the DPA only the active oil volume is exchanged. This is predefined on many hybrid applications with parallel drive structure through the kinematic forced coupling between the output and the displacement volumes on the hydrostats.

4 Summary and prospectus

Double piston accumulators guarantee optimum suction conditions on hydraulic hybrid system motor-pump units as, in contrast to conventional HP-LP systems, the pressure on the suction side remains constant. This means that no additional feed pumps are required. DPA increases the energy capacity and the implementable performance in comparison to conventional HP-LP accumulator systems, as - in particular as the HP accumulator empties - higher differential pressures can be made available to the motor-pump units. This has been demonstrated on a discharge procedure and an actual simulation example based on real measurement data for a city bus.

The DPA may also have advantages over a conventional HP-LP accumulator solution in terms of weight and possibly construction space. Isolating the gas volume from the oil volume provides further system advantages with reference to the packaging, and makes integration into the machine easier.

Due to the permanent coupling of the HP and LP chambers, only one side of the accumulator needs to be monitored with reference to the filling level or the piston stroke. This makes the oil management or feed after volumetric losses far easier.

Due to the impressive advantages in efficiency, the double piston accumulator will soon be used in several hybrid applications.

5 Bibliography

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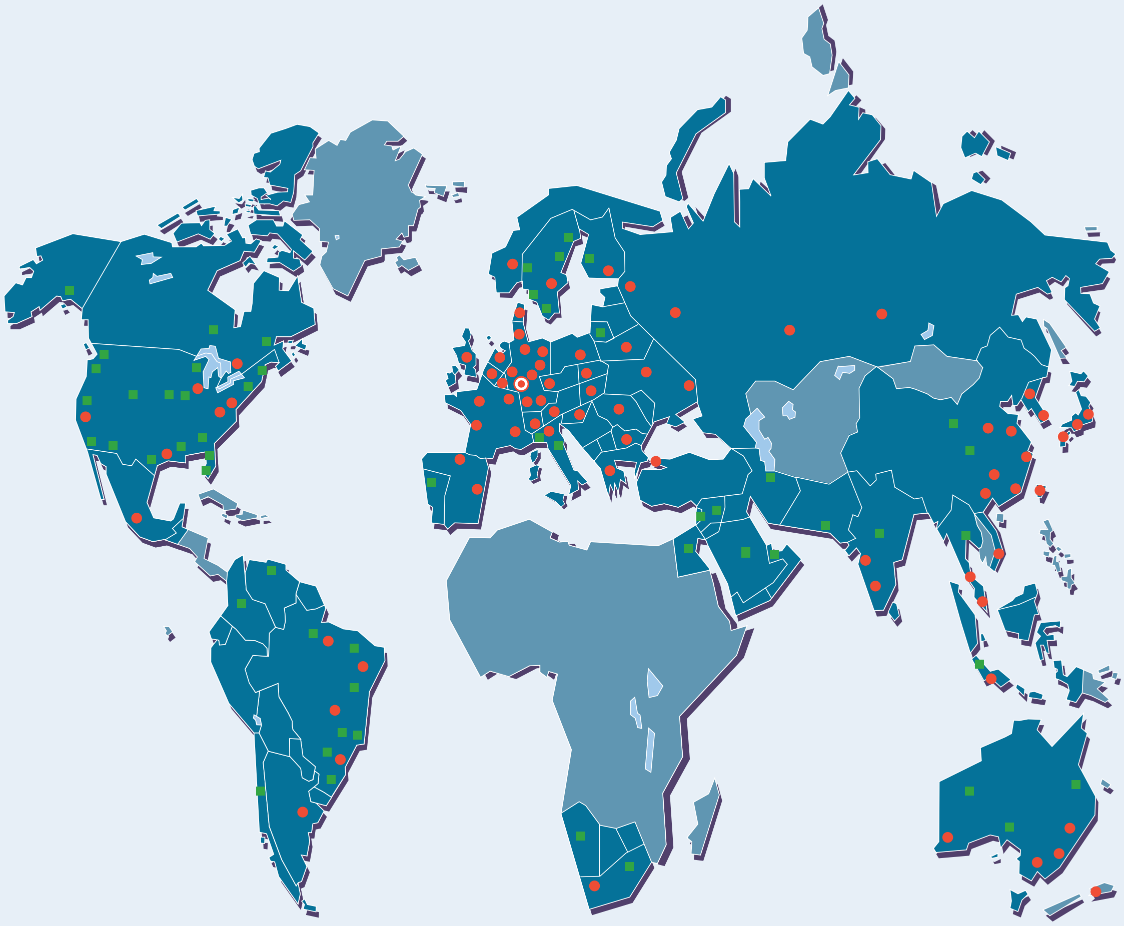

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