Improvement of Heat-Regenerative Hydraulic accumulators

Aleksander STROGANOV, Leonid SHESHIN

Abstract: The paper presents an improvement of heat-regenerative accumulators with piston and elastic separators aimed at expansion of the operational conditions range as well as long service life and durability.

A metal compressible regenerator embedded into a piston accumulator provides nearly isothermal character of gas compression and expansion. This increases accumulator efficiency at various operational modes including long storage.

For accumulators with elastic separators (membrane, bladder) there is a known method of efficiency increase by filling the gas reservoir of the accumulator with elastomeric foam. The paper presents a solution for foamed filler protection providing long service life and foam resistance to fast irregular separator movements.

An alternative method of approaching isothermality of gas compression/expansion is also presented. The new method is based on forced turbulent gas heat exchange and can be applied to any type of accumulator.

Presented experimental data demonstrate a noticeable increase of the energy recuperation efficiency in all tested regimes when using the improved accumulators.

Keywords: Energy recuperation efficiency, hydropneumatic accumulator, thermal losses, heat regeneration

1 Introduction

For fluid power recuperation applications the major losses in accumulator are caused by the very nature of the processes in the gas reservoir of accumulator [1]. Since gas compression/expansion processes are polytrophic and therefore irreversible, gas always returns back less energy at expansion than it was stored at compression.

Thermal losses strongly depend on compression/expansion rate and ratio and can reach one-third of the stored energy at commonly used compression ratio of 2 to 3.

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They also increase when required energy storage period is long, for example for hydraulic hybrid delivery or utility trucks with long load/unload stops and for hybrid vehicles driving in traffic jams with short start/stop impulses and long idle intervals.

For reduction of thermal losses it was suggested [2] to place elastomeric foam into the gas reservoir of the accumulator, so that the foam is to perform the function of heat regenerator and insulator. When the gas is being compressed, the compressed elastomeric foam takes away some heat from the gas and reduces its heating, and, when the gas is being expanded, the expanded due to its intrinsic elasticity foam returns back heat to the gas and reduces its cooling. The small size of the foam cells (about 1 mm) provides decreased (compare to the ones without foam) temperature gradients in the gas reservoir during heat exchange between the gas and foam. Thus heat exchange reversibility during gas compression and expansion considerably increases. The porous structure of the elastomeric foam reduces convective gas flows and thus heat transfer from the gas to the gas reservoir walls decreasing respective thermal losses. Therefore, at short storage intervals, the major part of the heat transferred from the gas to the foam during compression is returned to the gas during expansion, while the recuperation efficiency increases considerably [2].

Major disadvantage of the described solution is low durability of elastomeric foam. At continuous operation, fatigue degradation of the elastomeric foam leads to deterioration of its elastic properties and development of residual deformation of the foam. As a result, the foam loses its ability to reshape and to fill the entire volume of the gas reservoir, while the recuperation efficiency decreases. In the experiments [3], the accumulated residual deformation reaches one quarter of the initial volume of the foam. Growing losses of the fluid power in the piston accumulator already within 36000 cycles (400 hours) of slow (0,025 Hz) compression and expansion was observed. Foam degradation strengthens considerably in real hydraulic systems, in which the piston moves non-uniformly with jerks. Exploitation at increased or low temperatures, typical for mobile applications, also accelerates the processes of foam degradation. An improved solution for bladder accumulators is presented in [8]. There the porous elastic body of foamed synthetic resinous material is intimately joined to the wall of the bladder.

In both described solutions the boundary layers of the foam body adjacent to the separator (piston or bladder) are not protected from fast separator movements, vibrations and jerks. With vibrating impact of the jerking separator, the boundary layer of the elastomeric foam adjacent to the separator is exposed to the highest load and destruction. Its springiness is not sufficient to transmit acceleration from the separator to the entire mass of the foam. If the amplitude of the separator vibration is commensurate with the cell size, the boundary layer is crushed and destroyed. This is followed by the destruction of the next layer and so on. In addition, no reliability is ensured during working gas charging and discharging. The cleavage stress of the existing elastomeric foams is low, about 0.1 – 1.0 MPa. During fast processes of gas charging and discharging, considerably larger local pressure drops in the foam may arise, especially near the gas port, where the gas flow density is the highest. This may cause foam destruction. The danger of the foam being entrained into the gas port of the accumulator during fast gas exchange processes also restricts the use of gas receivers in combination with the described accumulators.

The paper presents: a solution for elastomeric foam protection, a regenerative solution for thermal losses reduction in piston accumulators and a solution for all types of accumulator based on alternative approach to heat regeneration.

2 Foam solution for membrane and bladder accumulators

Cheap and easy to implement solution with elastomeric foam [2] is well suitable for the use in membrane and bladder accumulators. To make the foamed accumulator durable the foam degradation problem described in [3] must be solved.

2.1 Foam protection means

Foam degradation can be substantially reduced by attaching the foam (porous material) to the separator (membrane or bladder) and the walls of the gas reservoir as described in [8] in combination with special means for protecting the boundary layer of the porous material. The protection means prevent development of local deformations exceeding the limit of reversible deformations of the porous material at maximum jerks of the separator. This limit can be specified as relative elongation at which the initial size of the pores of the undeformed porous material is restored. Thus the protection means reduce local deformations of the foam boundary layer in case of fast irregular movements of the separator reducing residual deformations which contribute to fatigue degradation and rupture of the foam boundary layer adjacent to the separator.

2.2 Protection means embodiments for different types of accumulator

Pneumatic and elastic protection means can be implemented alone or in combination with each other. *Figure 1* presents different embodiments of the protection means. (Some elements numbered in the figures are not referred to for the sake of brevity). Sketch in Figure 1.a shows the pneumatic protection means for piston accumulator made as a set of membranes 11 inserted into the porous material 7 near the piston 6 transversally to its movement direction. The gas permeability of these membranes along the piston movement is smaller than the average gas permeability of the porous material. As the piston movement become faster, the growing pressure drop at the membrane 11 provides higher acceleration of the membrane and the adjacent foam layers, thus reducing the load on them and decreasing their local deformations. The membranes 11 are made with holes 12.

Sketches in Figure 1.b and 1.c present the elastic protection means made as a set of elastic elements 9 connecting the separator 6 (membrane and bladder correspondingly) with remote from the separator layers of the porous material 7. These elastic elements can be made as elastically extensible polymeric bands or metal springs. Possible is also distributed embodiment of elastic elements in the form of reinforced webs between the pores in the foam boundary layer wherein the springiness of the reinforced webs is as higher as they are closer to the separator. The bladder accumulator shown in Figure 1 also contains membranes 11 to protect the porous material located in area of maximal speed of the bladder 6 movements.

For preventing porous material losses during gas discharging the gas port 5 is separated from the porous material 7 by filter 18 transmitting gas and entrapping the porous material. The filter can be made in the form of a membrane with the holes of the average size not exceeding the average thickness of the webs between the foam pores and the average distance between the holes being less than the average cross

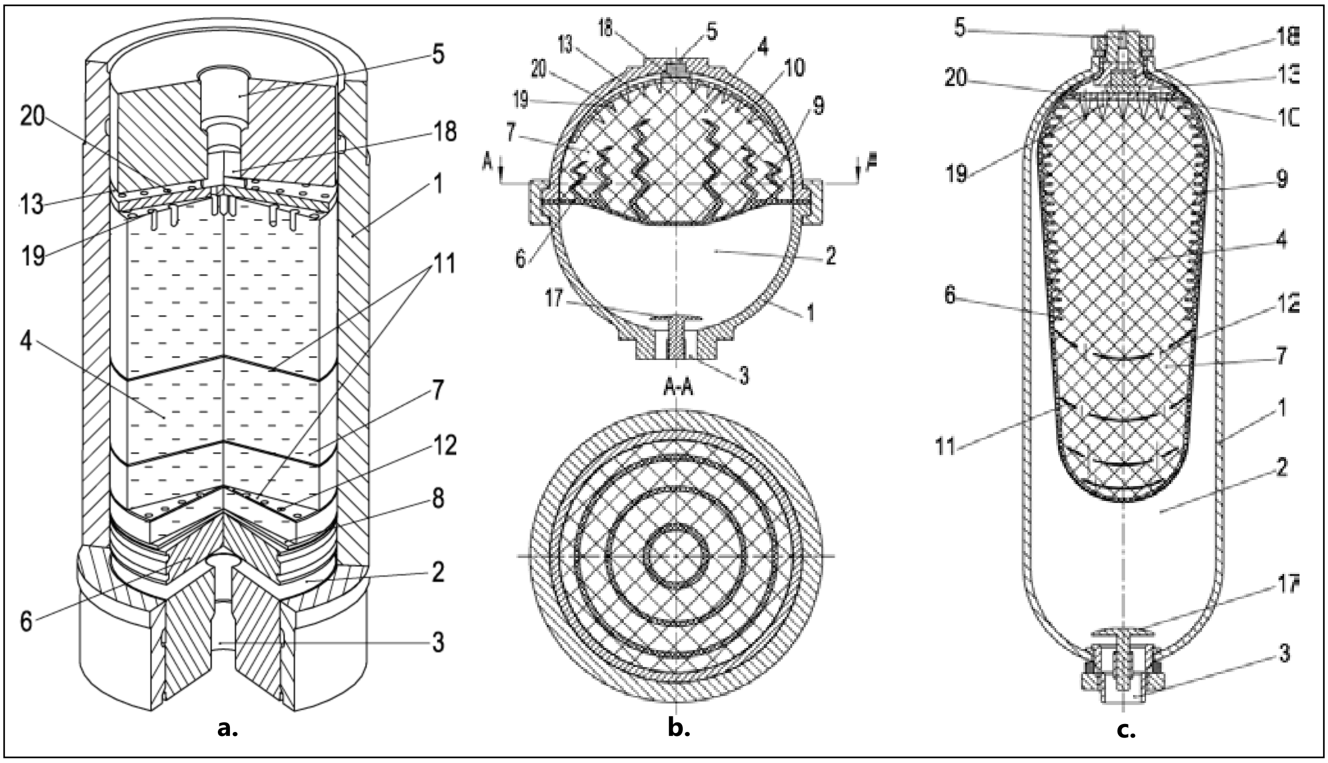


Figure 1. *a)Piston accumulator with pneumatic protection means, b)membrane accumulator with elastic protection means and c) bladder accumulator with elastic and pneumatic protection means*

dimension of the canals between the foam pores. For preventing porous material from damage during gas charging and discharging the filter reduces the gas flow through the gas port so as the pressure drop in case of open gas port exceeds (preferably more than 10 times) the maximum pressure difference between different areas of the porous material. A separate flow restrictor in the form of a throttle separated by a filter from the foam can be used as well as an integral embodiment where the filter itself restricts the gas flow as described above. The filter can be made for example, in the form of a three-dimensional solid porous structure with increased gas-dynamic resistance.

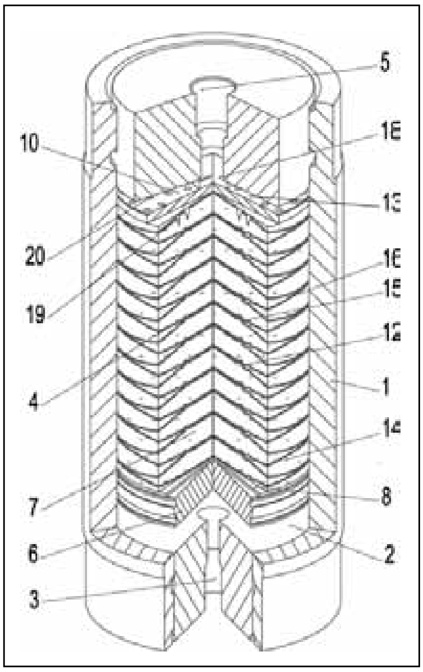


Figure 2. *Piston accumulator with pneumatic protection means*

When fast gas charging/discharging is operation condition (for example in accumulator + gas bottle systems) the porous material close to the gas port must be made with increased gas permeability exceeding the average permeability of the rest of the porous material.

In this case the porous material can either have separate drainage canals or be made from the material with increased sections of canals between the pores near the gas port. Another solution is to make the porous material with increased springiness close to the gas port, for example, from a denser porous material but with increased pore size and sections of canals between them. *Figure 2* presents a piston accumulator with porous material in its gas reservoir. In this design the plates 14 integrate two functions: heat regenerator and pneumatic protection means for the porous material which serves as additional heat regenerator between plates 14 and as a heat insulator preventing gas convection along the walls of the gas reservoir.

3 Metal compressible regenerator for piston accumulators

Substantial reduction of thermal losses in piston accumulators can be achieved by inserting a compressible metal regenerator (Heat Spring) into the gas reservoir of the accumulator.

3.1 Operating principle

Heat Spring is made of round metal plates located transversally to the piston motion direction with gaps between each other. The plates split the accumulator gas reservoir into intercommunicating gas layers. *Figure 3* presents a version of HS with spacers 2 providing gaps between the plates 1. The splitting of the gas

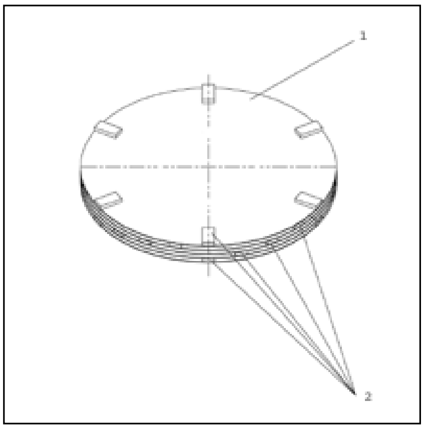


Figure 3. *Fragment of Heat Spring with spacers*

reservoir into thin gas layers reduces the average distances to the metal heat-exchange surfaces which facilitates heat transfer between the gas and metal and reduces the gas temperature gradients. This increases the reversibility of the gas compression and expansion processes and, hence, the efficiency of the accumulator. The calculations of Heat Spring parameters, material considerations and more detailed principle of operation were presented in [4] and [5].

3.2 Review of HS design and storage system packaging

There are various possible versions of HS design and packaging of the piston accumulator with HS. Depending on application HS can have compressible or stretchable design [7]. Compressible design of HS with spacers might be easier to implement and scale to the size of accumulator. Its disadvantage is compression ratio limitation due to the spacers. This type of HS can not be compressed to zero gap between the plates, thus this design has limited application field. Stretchable design does not suffer from this limitation but is less reliable in joints between the plates. Compressible design without spacers (*Figure 4*) seems to be optimal solution for wide range of applications. With appropriate measures taken against scratching the walls of accumulator by the plates this design combines simplicity of manufacturing and assembling with wide range of operating parameters.

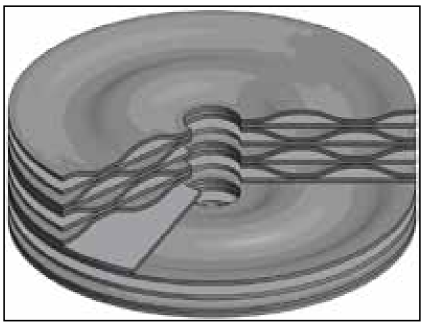


Figure 4. *Fragment of Heat Spring without spacers, sectional view*

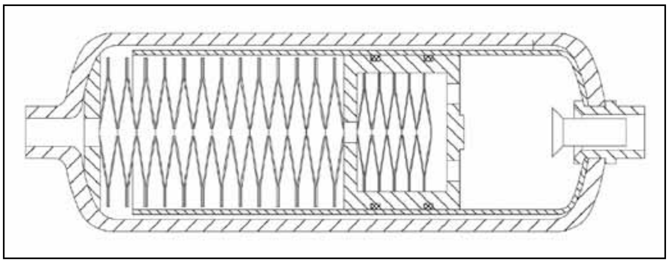


Figure 5. *Lightweight piston-in-sleeve accumulator with HS and pressure ripple damper*

One of the most promising solutions of the HS accumulator especially for mobile applications is based on piston-in-sleeve design of the accumulator [6]. *Figure 5* shows schematic view of possible embodiment based on this design. The accumulator shell is made of lightweight composite material. The piston moves inside of a thin metal sleeve balanced relative to the pressure forces acting on its surfaces. The gas reservoir contains HS of any appropriate design. For prolonged service life the piston is made hollow with metal leaf bellows installed inside its cavity. The bellows functions both as an additional heat regenerator and as a pressure ripple damper. The lightweight bellows takes the high-frequency ripples of the flow and pressure while the more massive piston moves uniformly or does not move. This ensures integrity of the seals of the piston and high degree of ripple smoothing.

For storage systems aggregated with gas bottles the same regenerative approach can be implemented both to the accumulator and gas bottle. The latter is easier since gas bottle doesn't have movable parts inside and can contain a fixed regenerating heat exchanger in the form of some metal porous structure. The aggregate volume of this heat exchanger material must be in the range from 10 to 50% of the internal gas bottle volume and the aggregate area of the heat exchange surfaces of the heat exchanger reduced to the aggregate internal gas bottle volume must exceed (preferably) 10 000 cm^2/liter. Thus, at gas compression or expansion in the gas bottle the heat exchange between the gas and the regenerating heat exchanger occurs at small average distances between the gas and the heat exchange surfaces and on a large heat exchange area and, therefore, with smaller temperature gradients, which increases reversibility of the heat exchange processes in the gas bottle its and recuperation efficiency. For effective heat exchange the heat exchanger must be made with the average pore size below 5 mm. On the other hand to reduce gas-dynamic resistance of the porous structure the average pore size must be no less than 0.05 mm. With 10 to 50% share of the gas bottle volume occupied by the metal of the heat exchanger the thermal capacity of the latter exceeds the thermal capacity of the gas in the bottle. For example, the specific thermal capacity of the heat exchanger made from steel will be from 400 to 2000 kJ/K/m3 while the thermal capacity of nitrogen at working pressures of 100 - 300 bar (and ambient temperature) is from 120 to 360 kJ/K/m3. The higher the gas working pressure, the higher the thermal capacity of the heat exchanger must be, i.e. the larger the share of the gas bottle volume occupied by the material of the heat exchanger. The steel or aluminum porous structure can be made from ready metal items to be disposed or from

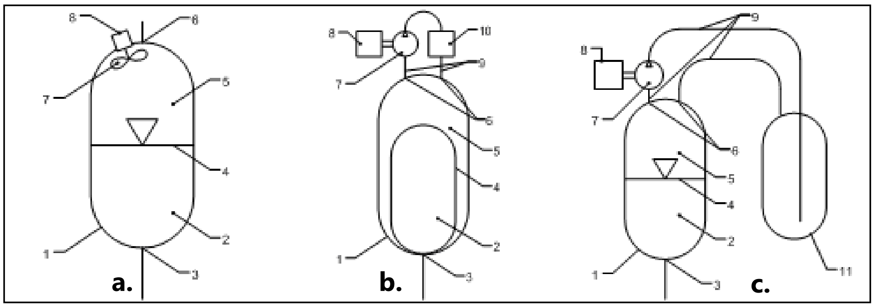


Figure 6. *a)Accumulator with internal gas blower, b) bladder accumulator with external gas blower and heat exchanger and c) system with gas bottle and external gas blower*

metal-working production wastes. It can also be formed from turnings or metal cuttings resulting from another process of metal machine work (drilling, milling cut operation, etc.). To increase thermal capacity and vibration resistance these metal turnings or other cuttings inside the bottle can be additionally compacted. To prevent penetration of particles of the heat exchanger from the gas bottle through its gas port the latter must be equipped with some blocking element. To reduce the gas-dynamic resistance the axial length of this blocking element must exceed 20% of the axial length of the gas bottle.

3.3 Tested prototype of a piston accumulator with Heat Spring

Figure 3 shows a fragment of the tested Heat Spring. The prototype was made of a regular piston accumulator (2 liters Hydac SK350-2/2212A6) with HS introduced into its gas reservoir. The HS was made of stainless steel flat round plates 1 fastened together by steel spacers 2 glued to the plates 1 with the angular offset of 60 degrees. On the other side of each plate there were also 6 spacers 2 with the same offset relative one another. In this case the whole configuration of the spacers 2 on one side of the plate was shifted relative to the configuration of the spacers on the other side of the same plate by 15 degrees. Thus, the configuration of the spacers in every successive gas layer between the plates was turned by 15 degrees relative to the previous one while the configurations with the similar angular position repeated with the period 4 and was separated by 3 gaps between the plates. The angular size of the spacers 2 was considerably less than 15 degrees, which allowed of HS compression with relatively small bending strains of the plates. At full compression the average depth of one gas layer was one-fourth of the spacer thickness. So this HS design provided for volumetric compression ratio of 4. HS itself occupied 28% of the gas reservoir volume. Its total mass was 4,45 kg and total heat capacity was 2,25 kJ/K that was about 12 times higher than total heat capacity of the gas (0,19 kJ/K) in the gas reservoir at initial pressure of 1,05 MPa. The same non-modified Hydac accumulator was taken as the reference for testing.

4 Alternative solution for all types of accumulator

Heat Spring solution based on gas heat regeneration has shown the best thermal losses reduction (see below). But it can be implemented into piston accumulators only. Since it's not possible to introduce movable regenerator in accumulators with elastic separator, the idea is to circulate gas through a regenerator during compression and expansion. Below is described a solution for all storage system configurations based on any type of accumulator.

4.1 Principle of operation

So the alternative approach is to maximize convection flows in the gas reservoir and to use the walls of the reservoir as heat regenerating means with or without a special fixed regenerator. This forced convection can be created in the accumulator or gas bottle as alternative to the honeycomb structure of receiver described in [5].

The forced convection is provided by a gas blower and intensifies multiply heat exchange between the gas and the surfaces blown by the gas including the walls of the gas reservoirs (accumulator and/or gas bottle). Thus heat is transferred to these surfaces more intensively from the gas at compression and returns back to the gas more intensively at expansion. This in turn reduces temperature gradients in the gas and it's heating and cooling in recuperation cycle. At slow and medium rate recuperation cycles the gas blower provides near isothermal gas compression/expansion processes at relatively low power being consumed by the gas blower. At fast cycles the gas blower can be switched off thus allowing for adiabatic processes which are more efficient at fast gas cycles.

The gas blower can be driven electrically or hydraulically. In latter case the hydromotor driving the gas blower is convenient to supply with the same liquid flow circulating through the liquid reservoir of the accumulator.

4.2 Various installations of the forced convection unit

Figures 6, 7 present few variants of the convection unit location in a recuperation system (in schematic representation). *Figure 6.a* presents the simplest embodiment with one electrically driven gas blower 7 introduced into gas reservoir 5 of accumulator 1. The system does not include a gas bottle.

Sketch in *Figure 6.b* presents the embodiment with external electrically driven gas blower 7 installed outside of gas reservoir 5 of bladder accumulator 1. External heat exchanger 10 and gas blower 7 are connected to the gas reservoir 5 by gas lines 9. The system in the sketch

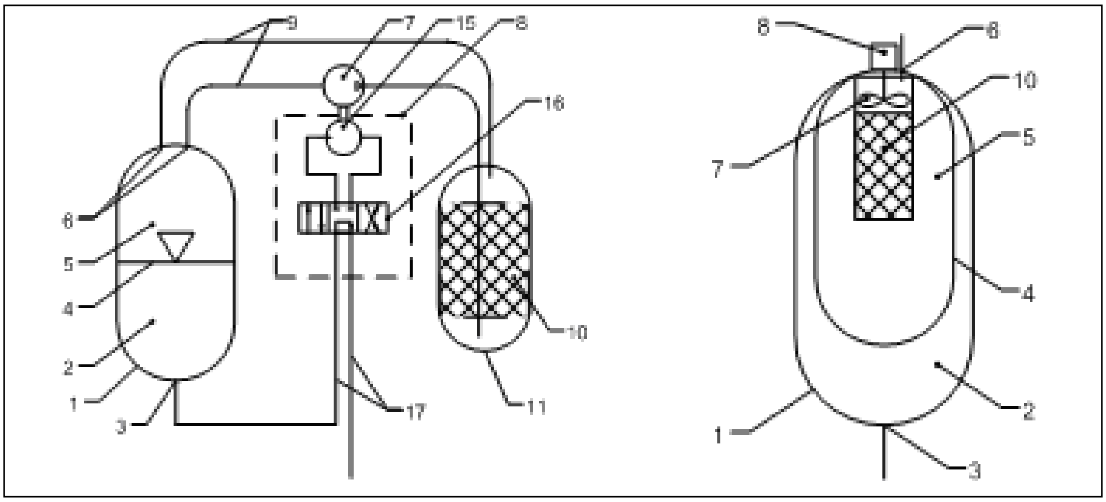


Figure 7. *a) System with gas bottle, hydraulically driven external gas blower and heat exchanger inside the gas bottle and b) bladder accumulator with internal gas blower and regenerative heat exchanger*

of Figure 6.c includes accumulator 1, gas bottle 11, external electrically driven gas blower 7 and gas lines 9. Here convection flows are created both in gas reservoir 5 of the accumulator and in the gas bottle. One of gas lines 9 is inserted deeper into the gas bottle than the other one thus providing fanning of its walls. The system presented in the sketch of Figure 7.a includes accumulator 1, gas bottle 11 with internal regenerative heat exchanger 10 and external hydraulically driven gas blower 7. Hydromotor 15 is supplied with the liquid charging the liquid reservoir 2 of the accumulator. Switching valve 16 provides unidirectional rotation of the gas blower both at charging and discharging of the accumulator.

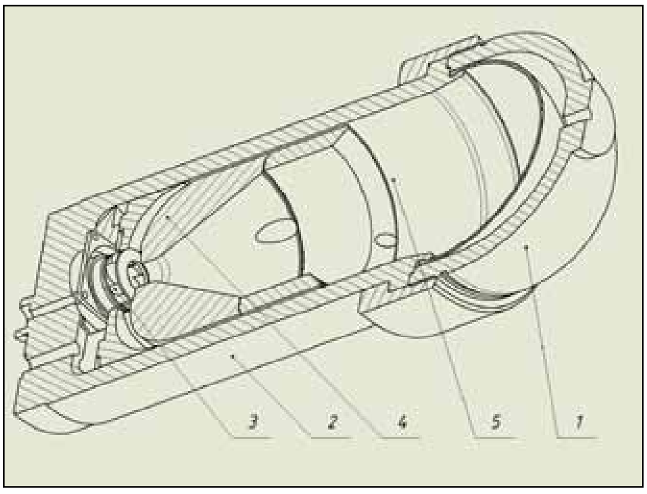


Figure 8. *Prototype of membrane accumulator with internal gas blower and regenerative heat exchanger*

Sketch in Figure 7.b demonstrates bladder accumulator 1 with internal regenerative heat exchanger 10 and internal electrically driven gas blower 7 installed inside gas reservoir 5.

4.3 Tested prototype of a membrane accumulator with forced convection unit

Figure 8 presents the prototype of a membrane accumulator with internal gas blower and internal regenerative heat exchanger. The oil part 1 of a regular membrane accumulator (2 liters Hydac SBO250-2A6/112A6--250AK) has been used as it is. The gas part 2 was redesigned so that to contain the gas blower 3 and the regenerative heat exchanger 4. Its volume was 3 liters. A miniature high speed radial gas blower (Micronel U51DL--012KK-5) was used for creating forced convection. The blower was installed inside the aluminum heat exchanger 4 of parabolic--like form which in its turn was installed into the gas reservoir 5. Electric wires for the blower drive motor and a thermocouple were inserted into the gas reservoir through the high pressure seals. The mass of the heat exchanger was 3 kg which provided 2,79 kJ/K of heat capacity for regeneration heat of the gas. The gas blower provided 400 l/min at atmospheric pressure. When tested as a reference the prototype operated with the gas blower switched off.

5 Simple solution for gas bottles

When the system equipped with a gas bottle there is an additional option of gas bottle improvement. Since there are no moving parts in the gas bottle a simple regenerator in the form of metal porous structure can be installed inside it. This structure has to have enough heat exchange surface and thermal capacity. The cheapest solution for gas bottles would be the use of metal turnings or cuttings resulting from a process of metal machine work (like drilling, turning, shaping, and so on). This might be for example the turnings coming from rotary machining used for manufacturing accumulators.

5.1 Tested prototype of a gas bottle with regenerative heat exchanger

This solution was tested. The prototype (*Figure 9*) was made out of 2,5 liter piston accumulator filled with steel cutting. The mass of the steel cutting was 2 kg which provided 0,92 kJ/K of heat capacity for regeneration heat of the gas. The steel cutting occupied 10% of the gas bottle volume. Another 2 liter piston accumulator was used to build complete storage system.

Figure 9. *Prototype of gas bottle with regenerative heat exchanger made out of steel cutting*

6 Power efficiency test setup

6.1 Test rig for efficiency measurement

Figure 10 presents the test rig scheme for power efficiency measurements. This circuit was used for testing the HS prototype at the Institute of Fluid Technology in Dresden (IFD) Dresden. Tested accumulator 1 is charged by pumping station 11. The fluid flow from the pumping station goes through proportional valve 6 to first port 3 of double acting cylinder 2 while its second port 4 is connected to the accumulator. The position of piston 5 of cylinder 2 is measured to obtain the oil (and consequently gas) volume variation in the accumulator. Proportional valve 6 is used as a discharge load. It switches the first port 3 of the cylinder 2 between the pressure line of pumping station 11 and its return line thus providing three modes of the accumulator operation: charge, hold and discharge. The oil and gas pressures in the accumulator, the gas temperature and piston 5 position of cylinder 2 are recorded against the time by pressure transducers 7, 8 and position transducer 9. (Some auxiliary elements of the test rig are not shown). Pressure relief valve 10 protects the tested accumulator from out-of-range pressure.

The test rig for testing the membrane prototype with forced convection unit was equipped with an adjustable throttle used as a load. Instead of proportional valve one switching valve was used to direct oil flow. Since the schemes of both rigs are almost the same there is no need to give a picture of the second one.

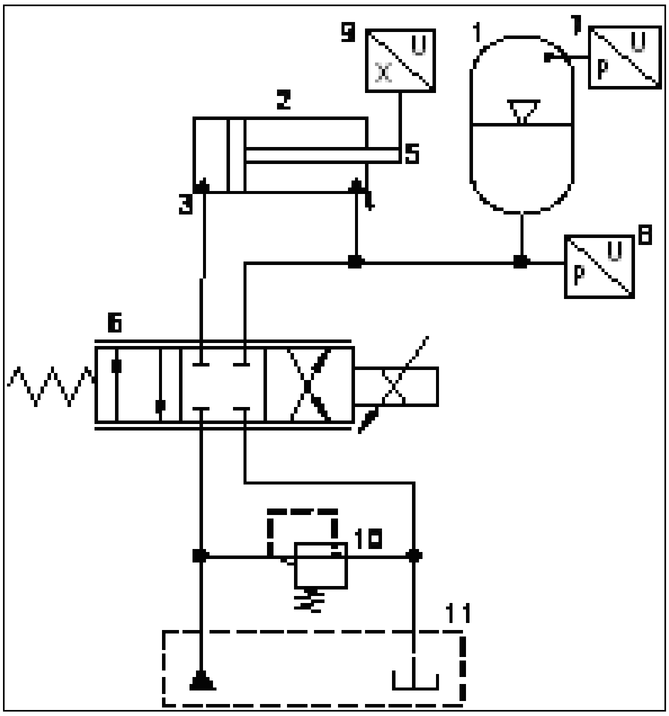


Figure 10. *Simplified test rig scheme*

6.2 Test procedure

For both tested piston accumulators (with and without HS) the initial gas pressure was 105 ± 2% bar with compression ratio in the cycle of 2.1 ± 2%. The six cycles were chosen close to real charge-hold-discharge cycles of the accumulator as if it were installed on a hydraulic hybrid vehicle and tested at different driving regimes. Parts of the urban drive cycle (ECE-15) were taken as representative cycles for mobile applications. Four different cycle shapes were taken as representatives of industrial cycles according to Rupprecht classification: triangle, impulse, sawtooth, and rectangle [9]. Their shapes timing and measured efficiencies are given in *Table 2*.

For the tested membrane accumulator the initial pressure was 100 ± 1% bar with compression ratio in the cycle of 2.03 ± 2%. Feasibility of the forced convection solution was demonstrated in one cycle chosen from the six used for testing HS solution. The temperature in the gas reservoir and the gas blower power consumption were measured together in addition to hydraulic parameters. The gas blower was powered during charge and discharge of the accumulator and switched off during holding period. The cycle with the gas blower switched off was taken as a reference.

For the tested system with a gas bottle the initial pressure was 70 ± 4% bar with compression ratio in the cycle of 2.1 ± 7%. The same cycle as for the forced convection solution was chosen for testing. The same system without cutting in the gas bottle was taken as a reference.

7 Power efficiency test results

7.1 Piston accumulator with the embedded compressible regenerator (HS)

The prototype was tested for two types of applications: mobile and industrial.

The data presented in [5] for mobile applications demonstrate significant difference between the reference accumulator and accumulator with Heat Spring, with at least a ten-fold reduction of thermal losses

Table 1. *Recuperation efficiency of the piston accumulator for different driving cycles*

Cycle shape			Efficiency of accumulator, %	
charge time, sec	hold time, sec	discharge time, sec	regular, gas / oil	with HS, gas / oil
Fragments of the urban drive cycle (ECE-15)				
17	24	4	80.7 / 80.5	97.3 / 96.6
5	22	11	81.6 / 81.4	97.8 / 96.9
11	22	24	82.8 / 82.7	98.6 / 97.8
Delivery truck cycles with load/unload				
5	60	11	79.1 / 78.7	97.5 / 96.9
5	180	11	78.3 / 78.2	97.1 / 96.5
5	900	11	78.1 / 78.0	96.7 / 95.8

in the HS prototype. *Table 1* (taken from [5]) gives the comparative test results for 3 parts of the ECE-15 cycle and 3 cycles with long holding time (up to 15 minutes), which correspond to hydraulic hybrid delivery truck operation. The efficiency values are given for the gas side and oil sides. The latter corresponds to the total efficiency of the accumulator. As expected the better result compare to the reference accumulator was obtained for long holding time where irreversibility of gas compression/expansion processes gives maximal contribution to the thermal losses.

Table 2 presents the results of the test for industrial applications repeated for two cycle frequencies differing by one order of magnitude. The HS prototype has shown high efficiency for all shapes and frequencies. At the same time, the reference accumulator showed higher efficiency at fast steep slopes and short holding time. The dependence of the efficiency of the cycle on its frequency was examined using the reference accumulator and HS accumulator. The results of this comparative testing are presented in Figure 11 which indicates that better efficiency of the Heat Spring as compared to the reference accumulator was obtained for slower pulse fronts or longer storage time.

7.2 Membrane accumulator with forced convection unit

The P-V diagrams given in *Figure 12* were recorded at the gas side of the membrane prototype for the chosen cycle (a part of ECE-15). The thin curve corresponds to the cycle for reference accumulator (with the gas blower switched off). The thick curve – to the cycle with forced convection.

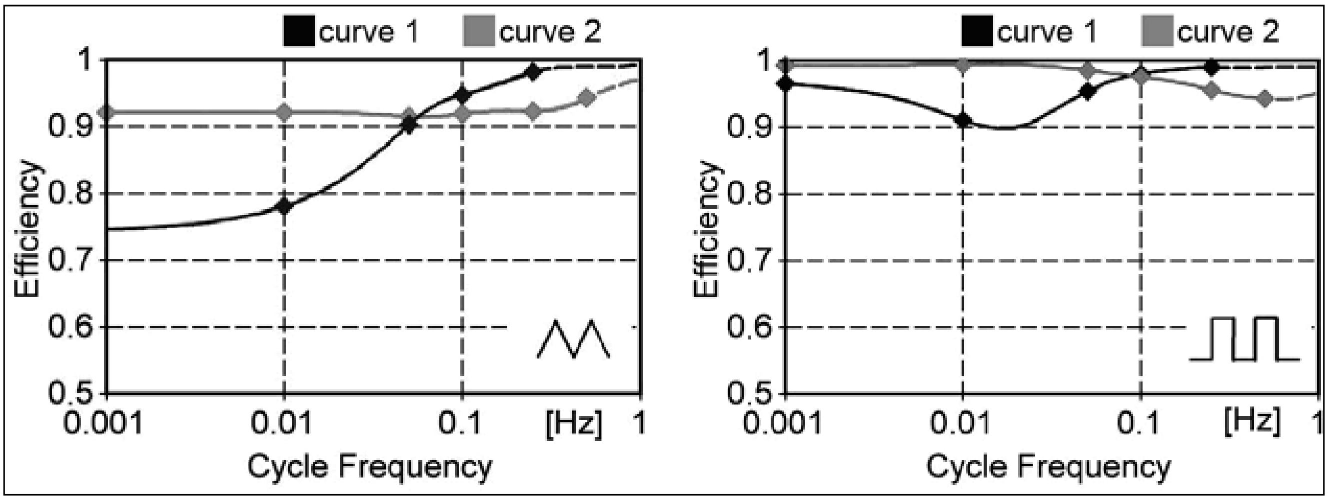
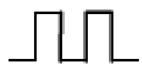


Figure 11. *Frequency dependence of recuperation efficiency for triangle a) and rectangle b) cycles for the reference (dark curve 1) and HS accumulator (gray curve 2)*

Table 2. *Recuperation efficiency for different industrial cycles*

	Cycle shape			
				
Freq [Hz]	Cycle in progress with time [sec]			
0.1	5-0-5-0	1-8-1-0	7-2-1-0	1-4-1-4
0.01	50-0-50-0	1-98-1-0	70-29-1-0	1-49-1-49
	Power efficiency (gas) Reference / HS [%]			
0.1	97.8 / 97.6	98.0 / 92.3	97.3 / 95.2	94.7 / 92.2
0.01	91.0 / 99.4	97.2 / 91.8	84.8 / 95.0	78.0 / 92.1

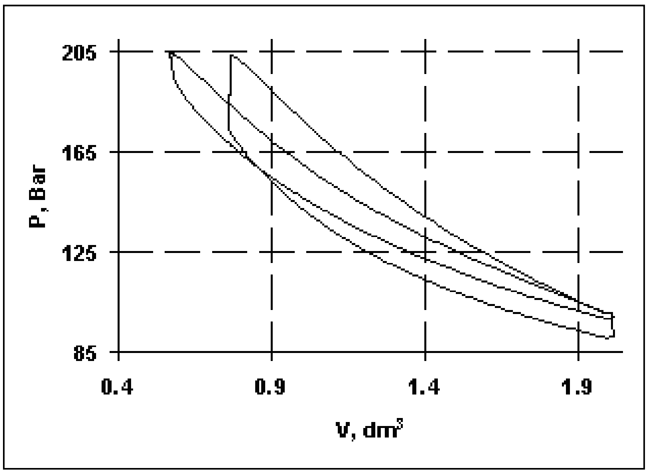


Figure 12. *P-V diagrams for the accumulator with the gas blower on and off for the cycle: 11 s - charge, 22 s - hold, 24 s – discharge*

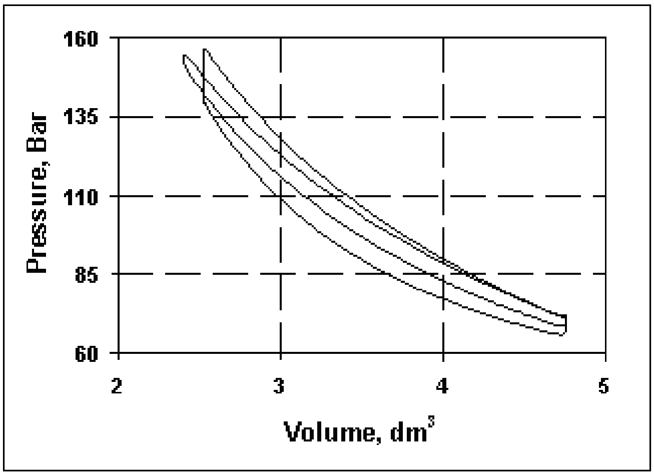


Figure 13. *Gas P-V diagram for the accumulator + gas bottle with and without cutting*

The loop area of the thin curve corresponds to 82.2 % of fluid power recuperation efficiency for the reference accumulator while the loop of the thick curve represents 92.8 % of fluid power recuperation efficiency for the cycle with forced convection. To evaluate total efficiency of the prototype the power consumed by the gas blower during the forced convection cycle was considered. These data are given in *Table 3* for several values of gas blower power consumption.

Table 3 indicates that the more power is spent on the gas blower the more intensive convection is and thus the higher is recuperation efficiency. However there is a range of the gas blower power consumption where the total efficiency of the accumulator is maximal. This optimal range is 20 ÷ 40 % of the maximal power applied to the gas blower during this series of testing.

The used gas blower was designed for air at atmospheric pressure and obviously its efficiency was not optimized for the gas at 100 ÷ 200 Bar. Even though the total efficiency of the membrane prototype with forced convection could reach 90 % which is close to the results for the foamed accumulator tested at the same cycle with the same initial pressure [4].

7.3 System with a gas bottle

The P-V diagrams given in Figure 13 were recorded for the reference and prototype storage systems at the chosen cycle (the same part of ECE-15). The thin curve corresponds to the cycle for the reference system (without cutting in the gas bottle). The thick curve – to the cycle for the prototype.

The loop area of the thin curve corresponds to 84.7 % of fluid power recuperation efficiency for the reference system while the loop of the thick curve represents 94.2 % of fluid power recuperation efficiency for the system with cuttings in the gas bottle. Thus the loops show about 10% of efficiency increase in the prototype compare to the same gas volume reference system without cutting in the gas bottle.

Table 3. *Total efficiency of the membrane accumulator with and without forced convection for the cycle: 11 s - charge, 22 s - hold, 24 s – discharge*

Power consumed by gas blower, J	Recuperation efficiency of accumulator, %	Total efficiency of accumulator, %
0	82.2	82.2
89	89.0	88.6
303	91.5	90.0
665	92.0	88.7
938	92.8	88.3

8 Conclusion

Experimental results for the prototype of a piston accumulator with HS presented in [4] and [5] show that the use of HS substantially improves accumulator efficiency in recuperation cycles of relatively low frequency or with long hold time (efficiency $\varepsilon > 0.95$ compared to $\varepsilon \approx 0.8$ for the regular accumulator of the same volume). This can be beneficial in applications, which require prolonged hold period (like hydraulic hybrid vehicles). As compared to the reference accumulator, the efficiency increase reaches almost 17%. In high frequency cycles with fast charge/discharge and short hold (industrial applications); the HS accumulator has also shown increase in efficiency of more than 90%. The reference accumulator in high frequency cycles showed equal or even better performance as compared to the HS prototype. This observation agrees with the modeling of the HS and findings on relationship between the gap, pulse frequency, and storage time [5]. Since 1 mm gap between the plates was used in the tested prototype, better results for slower pulse fronts or longer storage time were expected. This means that for fast industrial applications the HS must have thinner plates and smaller gaps between them. It was found that the gap value is proportional to the square root of the compression/expansion time. For faster cycles of 0.1 - 1 Hz, the gap must be 3 times smaller than that chosen for the tested prototype. The completed calculations and modeling of the Heat Spring created the basis for its design. Incorporation of the HS leads to unavoidable gas volume reduction (about 28% in the tested prototype) and thus the amount of energy that can be stored. However, the fact that the HS accumulator returns back more energy then the regular accumulator can store, this gas volume reduction can be disregarded.

In applications where the use of a piston accumulator is inappropriate due to its relatively bigger mass and cost the foam solution might be applicable provided with the foam protection means. A durability test (not described here) has shown foam resistance to fast (1.2 sec) cycles of compression/expansion at ambient and 1000C temperatures. After about 1 000 000 cycles (50% at 1000C) the tested foam did not show any signs of rupture or fatigue degradation. The gas port protection measures enable creating foamed accumulators for fluid power recuperation systems with reasonable service life at regular operation conditions. Accordingly to [4] the efficiency increase for the prototype of a piston accumulator filled with foam ranges between that for the accumulator without foam and the same accumulator with described above Heat Spring.

For wide range of applications the forced convection solution seems to be a good alternative. It is capable of providing recuperation efficiency close to the foam solution at any recuperation cycle parameters. This solution requires competent gas-dynamic calculations of the embedded regenerator and appropriate gas blower in order to maximize total power efficiency of a storage system. The forced convection solution is applicable to any type of accumulator and any type of a storage system assembly (with or without gas bottles). For slower cycles the gas blower should be on to achieve close to isothermal regime of gas compression/expansion. For fast cycles the blower should be off to leave the gas compression/expansion processes close to adiabatic regime. The storage system becomes versatile. Obvious disadvantages of this solution are cost (compare to the foam solution) and relative structural complexity. In case of electrically powered gas blower it requires special high pressure seals to deliver electricity inside the gas reservoir.

For storage systems with gas bottles metal turnings or cuttings in the gas bottle provides up to 11% of power efficiency increase. This solution is cheap and easy-to-implement. It's capable of working for systems based on any type of accumulator. Due to its configurability lower weight and price it's rather attractive for mobile applications including hydraulic hybrid vehicles.

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Izboljšava toplotne regeneracije hidravličnih akumulatorjev

Razširjeni povzetek

Prispevek predstavlja izboljšave toplotne regeneracije hidravličnih akumulatorjev (HA) z batom in elastičnimi ločevalniki, namenjenimi razširitvi delovnega področja kot tudi podaljšanju uporabne dobe.

Pri vsakem HA se pojavljajo izgube zaradi hitrih sprememb stanja plina (kompresija/ekspanzija), ki povzročajo segrevanje/ohlajanje plina. Zaradi tega HA vrne manj hidravlične energije, kot jo je prejel. Razlika med vstopno in izstopno hidravlično energijo se pretvori v toploto. Izkoristki običajnih HA so zato boljši pri počasnejših spremembah plina. Za HA z elastičnimi ločilnimi elementi (membrana, meh) je poznana metoda izboljšanja izkoristkov s polnjenjem prostora, kjer je plin, z elastomerno peno. Nalogi pene sta torej izolacija in regeneracija toplote. Ko se plin v HA stiska (kompresija), se generira toplota, ki jo prevzame penasto polnilo. Pri ekspanziji plina znotraj HA se plin ohlaja, pena pa mu odda pri kompresiji generirano toploto in s tem zmanjšuje učinek hlajenja plina. Tako vstavljena elastična pena izboljša izkoristek HA. Glavni problem pri uporabi pene kot polnila znotraj plinske komore HA je kratka uporabna doba. Pena začne običajno že po 400 delovnih urah najprej razpadati na stičnih ploskvah z ločevalnikom (meh, membrana). Razpadanje je pospešeno tudi pri nižjih delovnih temperaturah. Prispevek predstavlja rešitev za zaščito penastega polnila zaradi podaljšanja uporabne dobe in povečanja odpornosti pene na nenadne hitre pomike (stisnitve/sprostitve) polnila.

Dodane ojačitvene elastične membrane (slika 1a, poz. 11), oblite z elastično peno (sl. 1a, poz. 7), vstavljene v batni HA, preprečujejo hiter razpad pene in obenem omogočajo skoraj izotermne preobrazbe plina pri kompresiji in ekspanziji. S tem se vidno izboljšuje izkoristek HA pri različnih delovnih operacijah, vključno z daljšim časom shranjevanja hidravlične energije. HA z membrano (slika 1b) in HA z mehom (slika 1c) imata na elastični ločevalnik (membrana, meh) pritrjene vzmetne polimerne ali kovinske obroče (slika 1b in slika 1c, poz. 9), ki boljše in trajneje povezujejo elastični ločilni element z občutljivo elastično peno, ki obliva vzmetne polimerne ali kovinske obroče. Zaradi nevarnosti izločanja poroznega materiala elastične pene iz HA pri spuščanju plina je na plinskem priključku nameščen filter (slika 1b in slika 1c, poz. 18). Na sliki 2 je prikazan batni HA s porozno elastično peno in vzmetnimi kovinskimi ploščami na plinski strani HA.

Občutno znižanje temperaturnih izgub lahko dosežemo tudi z vstavitvijo kovinskih kompresijskih regeneratorjev ("toplotnih vzmeti") v plinsko komoro HA. "Toplotne vzmeti" so izdelane iz okroglih kovinskih plošč, ločenih z distančniki (slika 3), ki vzdržujejo režo med ploščami. Tako sestavljena "toplotna vzmet" razdeli plinsko komoro HA na manjše volumne, kar pripomore k hitrejšemu prenosu toplote in posledično k boljšemu izkoristku HA. Boljša konstrukcijska rešitev za "toplotno vzmet" so oblikovane upognjene vzmetne plošče brez dodatnih distančnikov (slika 4).

Prispevek prikazuje tudi različne izvedbe HA s prisilno konvekcijsko enoto (slika 6 in slika 7). Predstavljena je nova alternativna metoda približevanja izotermičnim spremembam plina (kompresija/ekspanzija), ki sloni na prisilni turbulentni toplotni izmenjavi in se lahko uporabi pri vsakem tipu HA. Prikazana je enostavna rešitev za boljšo izmenjavo in shranjevanje toplote v tlačni plinski posodi. Ker tlačne posode nimajo gibajočih se delov, lahko vanje vstavimo kovinske ostružke oz. kovinske odpadke od mehanske obdelave. Ti omogočajo dober prenos in "shranjevanje" toplote.

Predstavljeni rezultati meritev (preglednica 1, 2 in 3, slika 11 in slika 12 ter slika 13) prikazujejo znatno povečanje izkoristka izboljšanega HA pri vseh testiranih režimih. Rezultati meritev prototipa batnega HA s "toplotno vzmetjo" prikazujejo visok izkoristek izboljšanega HA tako pri srednje hitrih kot tudi pri počasnih spremembah stanja plina, in sicer so izkoristki izboljšanega HA večji od 95 %, izkoristki običajnega HA pa so okoli 80 % pri podobnih pogojih. Celo pri visokofrekvenčnih ciklih obratovanja oz. sprememb stanja plina znotraj HA je izkoristek izboljšanega HA večji od 90 %. Pri uporabi sistema HA z dodatno tlačno posodo, napolnjeno s kovinskimi ostružki, se izkoristek izboljša za 11 %. Ta rešitev je poceni in enostavna za izvedbo in jo lahko uporabimo s katerimkoli HA.

Ključne besede: izkoristek pri pretvarjanju energije, hidravlično-pnevmatični akumulator, toplotne izgube, regeneracija toplote