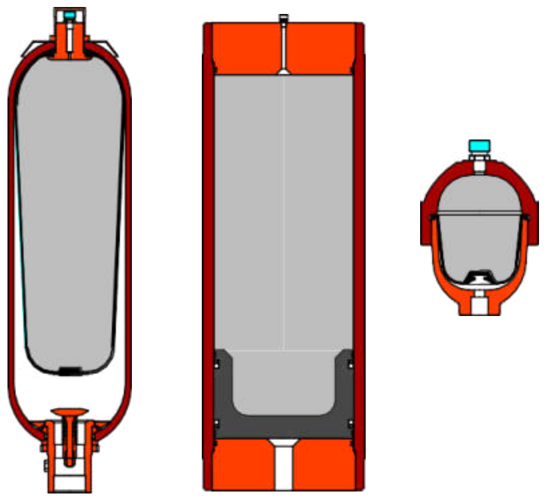
1.1.1 GENERAL

The main task of the hydraulic accumulator is to accumulate fluid under pressure and return it when necessary.
Since the accumulator contains a fluid under pressure, it is treated as a pressure tank and must therefore be sized for the maximum operating pressure according to test regulations in force in the country where it is installed.
To achieve the volume compensation and get the accumulation of energy, the fluid is pre-loaded by a weight, a spring or a compressed gas.



1.1a

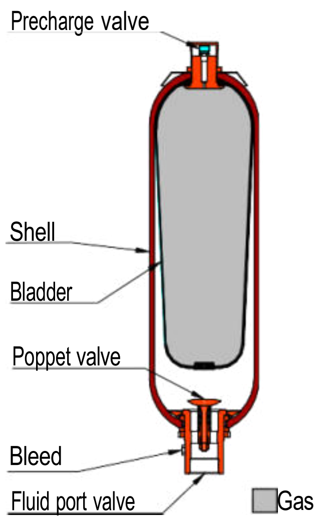
Between the pressure of fluid and the counter-pressure exerted by the weight, the spring or the compressed gas must be in a constant state of equilibrium. Weight and spring accumulators are used in industry only in special cases and thus have a relative importance.
Gas accumulators without a separating element are rarely used in hydraulics due to the absorption of gas by the fluid.
In most of the hydraulic systems are then used the gas accumulators provided with a separating element between gas and fluid.
Depending on the type of separating element, we can distinguish bladder, piston and diaphragm accumulators.

1.1.2 TYPES OF ACCUMULATORS WITH SEPARATING ELEMENT

These accumulators consist of a fluid zone, a gas zone and a separating gas-tight element.
The fluid area is in contact with the circuit. With the pressure increases, a certain volume of fluid enters into the accumulator and compresses the gases.
In the hydraulic systems, are used with the following accumulators with a separating element:
- bladder accumulators (Fig. 1.1b)
- piston accumulators (Fig. 1.1c)
- diaphragm accumulators (Fig. 1.1d)

1.1.2.1 BLADDER ACCUMULATORS

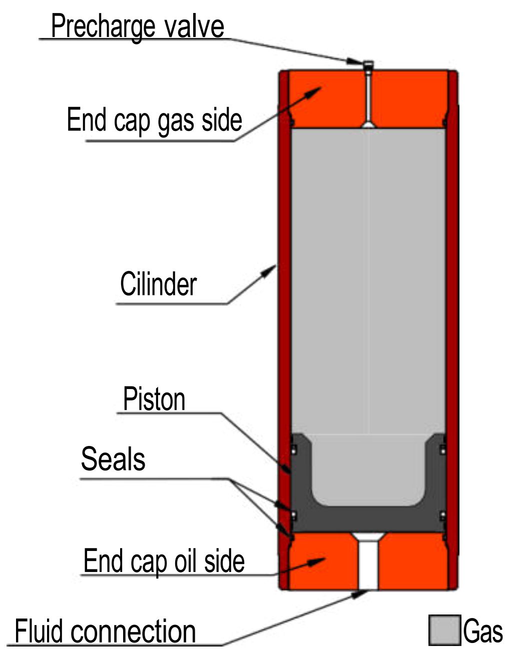
In the bladder accumulators, the fluid area is separated from the gas area by a flexible bladder. The fluid around the bladder is in contact with the circuit, so any increase in pressure causes the entry of the fluid into the accumulator and thereby compresses the gas. Vice versa, every drop of pressure in the circuit causes the expansion of the gas, resulting in delivery of the fluid from the accumulator to the circuit.
Bladder accumulators can be installed in vertical position (preferable), in horizontal one and, under certain operating conditions, also in an inclined one. In the inclined and vertical positions, the valve on the fluid side should face down. The bladder accumulators include a pressure welded or forged vessel, a flexible bladder and the fittings for gas and oil.



1.1b

1.1.2.2 PISTON ACCUMULATORS

In the piston accumulators, the fluid area is separated from the gas area from a metal piston fitted with gas tight seals. The gas area is filled with nitrogen.
The fluid zone is connected to the hydraulic system, so any increase

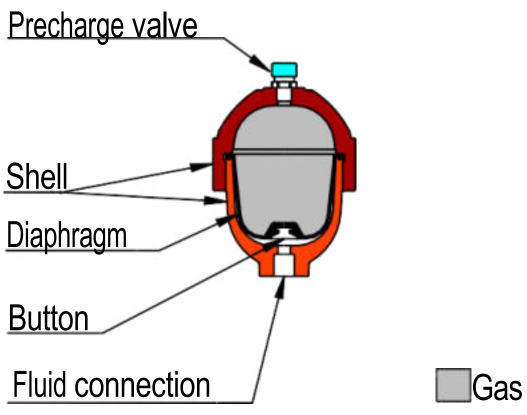


1.1c

in pressure in the circuit causes the entry of fluid in the accumulator resulting in compression of the gas.
Vice versa, at every drop of pressure in the circuit, the compressed gas contained in the accumulator expands and the accumulator delivers the fluid to circuit.
The piston accumulators can operate in any position, but it is preferable to mount them with the gas area upwards in order to prevent that solid contaminants contained in the fluid settle by gravity on the piston seals.
The typical structure of the piston accumulator, represented schematically in Figure 1.1c, includes a cylindrical pipe, a piston with seals, end caps in which there are the fluid side and gas side connections. The pipe serves to resist to the internal pressure and to drive the piston.
To ensure that the pressures of the two chambers are as balanced as possible, during the movement, it's necessary that the friction between the piston and the pipe is minimized.
For this reason, the inner surface of the pipe must be honed. In practice, however, the friction between the piston seals and the pipe creates, between gas area and fluid one, a pressure difference that, however, can be limited to 1 bar with appropriate selection of seals. The position of the piston can be shown continuously through a passing rod. By fixing a cam to the rod, you can also take advantage of the movement of the piston in order to control through limit switches the switching on or switching off of the pump.
For other types of monitoring of the piston position, see Section 4.1.

1.1.2.3 DIAPHRAGM ACCUMULATORS

Diaphragm accumulators are made of a steel pressure-resistant vessel, usually cylindrical or spherical in shape, inside which is mounted a flexible material diaphragm as separating element.
Diaphragm accumulators are manufactured in three versions:
- screwed execution (see Section 5.1.)
- forged execution (see Section 5.2.)
- welded execution (see Section 5.3.)



1.1d

In the screwed version, the diaphragm is blocked by a metal ring fitted between the lower shell and upper shell of the body.
In the welded accumulators, the diaphragm is pressed into the bottom before the welding of two steel shells.
Thanks to appropriate processes such as electron beam welding and also thanks to the special provision of the diaphragm, it's possible to prevent its damage and forging.

1.1.2.4 DERIVATION CONNECTION OF THE GAS BOTTLES

When for a given volume of fluid to provide/absorb the difference between the maximum and minimum pressure in the hydraulic circuit must be of limited size, the volume of the accumulator, obtainable with the calculation, may be very large. Under these conditions, it is preferable to connect the gas side of the accumulator with one or more additional gas bottles (Fig. 1.1l). For the sizing of the accumulator, you should take into account the following parameters:
- the useful volume to provide/absorb
- allowable ratios of pressures and volumes P2/Po = V0/V2
- the expansion of gas volume due to changes in operating temperature.

1.1.3 OPERATING CONDITIONS

Stage A
The accumulator is empty and neither gas nor hydraulic sides are pressurized Po = P = 0 bar

Stage B
The accumulator is pre-charged Po

Stage C
The hydraulic system is pressurized. System pressure exceeds the pre-charge one and the fluid flows into the accumulator Po→P1

Stage D
System pressure peaks. The accumulator is filled with fluid according to its design capacity.
Any further increase in hydraulic pressure would be prevented by a relief valve fitted on the system P1→P2

Stage E
System pressure falls. Pre-charge pressure forces the fluid from the accumulator into the system P2→P1

Stage F
Minimum system pressure is reached. The accumulator has discharged its maximum design volume of fluid back into the system min ΔP (P1min)

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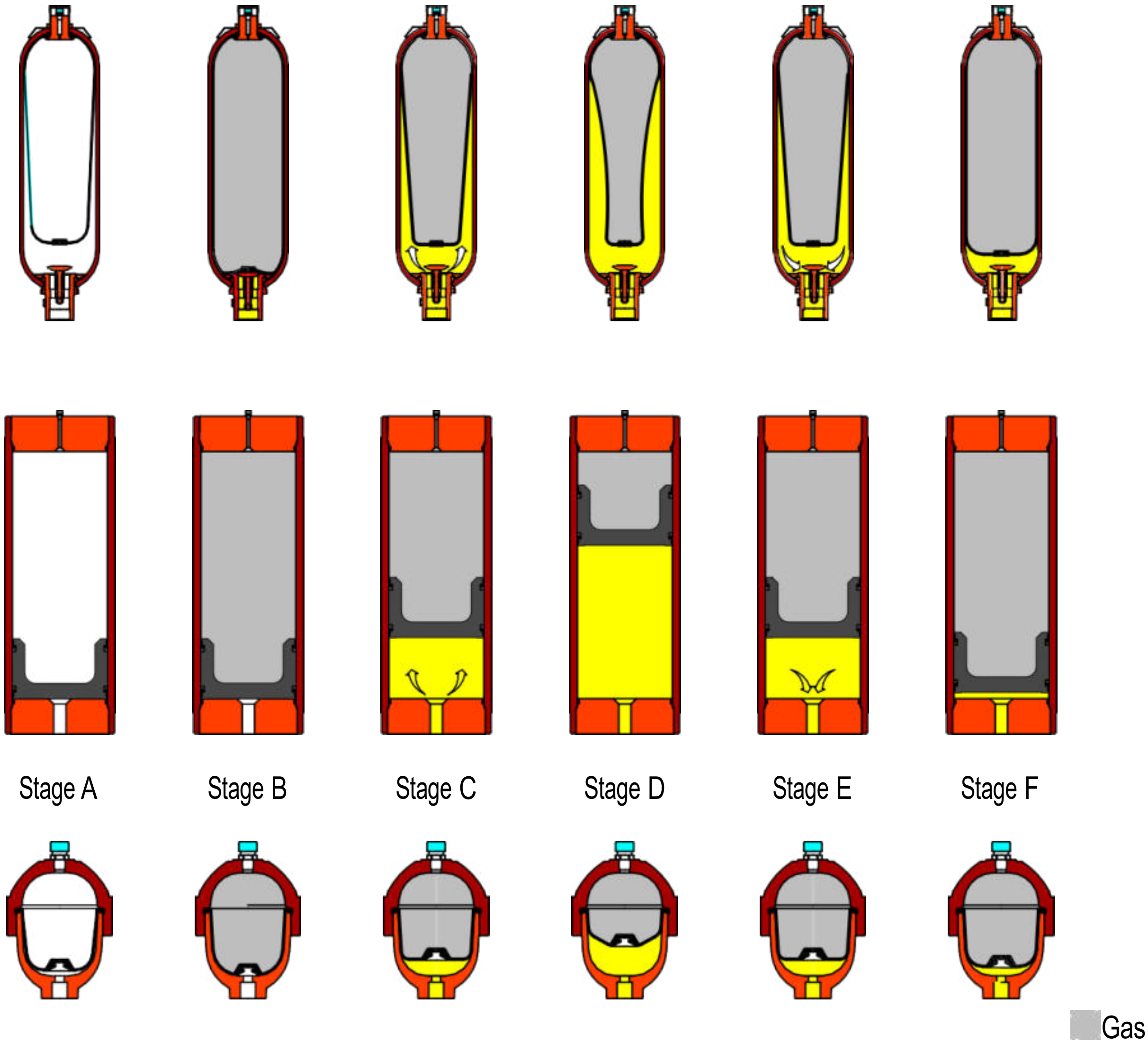


1.1e

1.1.4 ACCUMULATOR SELECTIONS

When selecting an accumulator for a particular application, both system and performance criteria should be taken into account.
To ensure long and satisfactory service life, the following factors should be taken into account.

- failure modes
- flow rate
- response time
- high frequency cycling
- external forces
- output volume
- fluid type
- shock suppression
- sizing information
- temperature effect
- safety
- certification

1.1.4.1 FAILURE MODES

In certain applications, a sudden failure may be preferable than a gradual failure. A high-speed machine, for example, where product quality is a function of hydraulic system pressure.
As sudden failure is detected immediately, scrap is minimized, whereas a gradual failure might mean that production of a large quantity of sub-standard product could occur before the failure becomes apparent.
A bladder/diaphragm accumulator would be most suitable to this application. Vice versa, where continuous operation is paramount and sudden failure could be detrimental as, for example, in a braking or steering circuit on mobile equipment, a progressive failure mode is desirable. In this application, a piston accumulator would be appropriate.

1.1.4.2 FLOW RATE

Fig. 1.1.n shows typical maximum flow rates for Epe's accumulator styles in a range of sizes.
The larger standard bladder designs are limited to 1000 LPM, although this may be increased to 2000 LPM using a high-flow port.
The poppet valve controls the flow rate, with excessive flow causing the

poppet to close prematurely.
Flow rates greater than 2000 LPM may be achieved by mounting several accumulators on a common manifold - see Accumulators station, Section 10.
For a given system pressure, flow rates for piston accumulators generally exceed those of the bladder designs.
Flow is limited by piston velocity, which should not exceed 3 m/sec. to avoid piston seal damage.
In high-speed applications, high seal contact temperatures and rapid decompression of nitrogen, which has permeated the seal itself, can cause blisters, cracks and pits in the seal surface. In this type of application, a bladder style accumulator would be better suited.

1.1.4.3 RESPONSE TIME

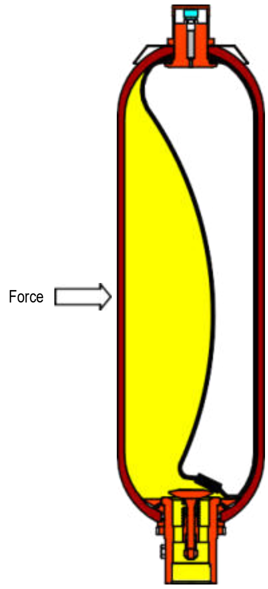
In theory, bladder and diaphragm accumulators should respond more quickly to system pressure variations than piston types.
There is no static friction to be overcome as occurs with a piston seal, and there is no piston mass to be accelerated and decelerated.
In practice, however, the difference in response is not great, and is probably insignificant in most applications.
This applies equally in servo applications, as only a small percentage of servos requires response times of 25 ms or less.
This is the point where the difference in response between piston and bladder accumulators becomes significant.
Generally, a bladder accumulator should be used for applications requiring less than 25 ms response time, and either accumulator type for a response of 25 ms or greater.

1.1.4.4 HIGH FREQUENCY CYCLING

High-frequency system pressure cycling can cause a piston accumulator to "dither", with the piston cycling rapidly back and forth so covering a distance less than its seal width.
Over an extended period, this condition may cause heat build-up under the seal due to lack of lubrication, resulting in seal and bore wear.
For high frequency dampening applications, therefore, a bladder/diaphragm accumulator is generally more suitable.

1.1.4.5 EXTERNAL FORCES

Any application subjecting an accumulator to acceleration, deceleration or centrifugal force may have a detrimental effect on its operation, and could cause damage to the bladder.
Forces along the axis of the pipe or shell normally have little effect on a bladder accumulator but may cause a variation in gas pressure in a piston type accumulator because of the mass of the piston.
Forces perpendicular to an accumulator's axis should not affect a piston model, but fluid in a bladder accumulator may be thrown to one side of the shell (Fig. 1.1f), displacing the bladder and flattening and lengthening it. In this condition, fluid discharge could cause the poppet valve to pinch and cut the bladder.



1.1f

Fig. 1.1f: Perpendicular force causes the mass of the fluid to displace the bladder. Higher pre-charge pressures increase the resistance of the bladder according to the effects of the perpendicular forces.

1.1.4.6 OUTPUT VOLUME

The maximum sizes available of each type of accumulator determine the limits of suitability where large output volumes are required. There are, however, several methods to achieve higher output volumes than standard accumulator capacities suggest - see Accumulators station, Section 10.

Compres-sion ratio	System pressure bar		Recommended Precharge bar		Fluid Output LPM	
	max	min	Bladder	Piston	Bladder	Piston
1,5	210	140	125	130	10,5	11,5
2	210	105	95	98	16	16,5
3	210	70	60	60	21,5	21,5
6	210	35	*	28	*	24

* *Below required minimum operating ratio of 4:1*

1.1g

Fig. 1.1g compares typical fluid outputs for Epe's 35 litres piston and bladder accumulators operating isothermally as auxiliary power sources over a range of minimum system pressures.
The higher pre-charge pressures recommended for piston accumulators result in higher outputs than as occurred in comparable bladder accumulators.

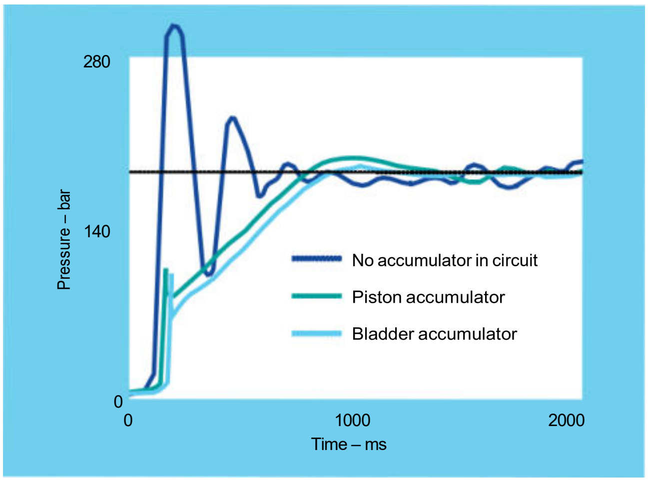
In addition, bladder accumulators are not generally suitable for compression ratios greater than 1:4, as these could result in excessive bladder deformation.
Piston accumulators have an inherently higher output relative to their overall dimensions, which may be critical in locations where space is limited.
Piston accumulators are available in a choice of diameters and lengths for a given capacity, whereas bladder and diaphragm accumulators are frequently offered in only one size per capacity, and fewer sizes are available.
Piston accumulators can also be built to custom lengths for applications in which the available space is critical

1.1.4.7 FLUID TYPE

Bladder/Diaphragm accumulators are more resistant to damage caused by contamination of the hydraulic fluid than piston types.
While some risks exist from contaminants trapped between the bladder and the shell, a higher risk of failure exists from the same contaminants acting on the piston seal.
Bladder accumulators are usually preferred to piston type accumulators for water service applications.
Water systems tend to carry more solid contaminants and lubrication is poor.
Both the piston and bladder type units require some type of preparation to resist to corrosion on the wetted surfaces (example nickel coated) Piston accumulators are preferred for systems using special fluids or where extreme temperatures are experienced as compared to bladders.
Piston seals are more easily moulded in the required special compounds and may be less expensive.

1.1.4.8 SHOCK SUPPRESSION

Shock control does not necessarily demand a bladder/diaphragm accumulator, it is possible to use also a piston accumulator, see example Fig. 1.1h



1.1h

1.1.4.9 MOUNTING POSITION

The optimum mounting position for any accumulator is vertical, with the hydraulic port downwards. Piston models can be mounted horizontally if the fluid is kept clean but, if solid contaminants are present or expected in significant amount; horizontal mounting can result in uneven or accelerated seal wear.
A bladder accumulator may also be mounted horizontally, but uneven wear on the top of the bladder as it rubs against the shell while floating one the fluid can reduce its service life and even cause permanent distortion.
The extent of the damage will depend on the fluid cleanliness, cycle rate, and compression ratio. In extreme cases, fluid can be trapped away from the hydraulic port (Fig. 1.1i),

1.1i

Fig. 1.1i *Ahorizontally-mounted bladder accumulator can trap fluid away from the hydraulic valve reducing output, or the bladder may become elongated, forcing the poppet valve to close prematurely.*

1.1.4.10 SIZING INFORMATION

Accurate sizing of an accumulator is critical if it has to deliver a long and reliable service life. Information and worked examples are shown in Section 2 or accumulator size can be calculated automatically by entering application details into Epe's Sizing software selection program.
Please contact your local Epe distributor for details or contact us at **www.epeitaliana.it**

1.1.4.11 TEMPERATURE EFFECT

Temperature variation can seriously affect the pre-charge pressure of an accumulator. As the temperature increases, the pre-charge pressure increases; Vice versa, decreasing temperature will decrease the precharge pressure. In order to assure the accuracy of your accumulator pre-charge pressure, you need to factor in the temperature variation. The temperature variation is determined by the temperature encountered during the pre-charge versus the operating temperature expected in the system, (see Section 2.2.)

1.1.4.12 SAFETY

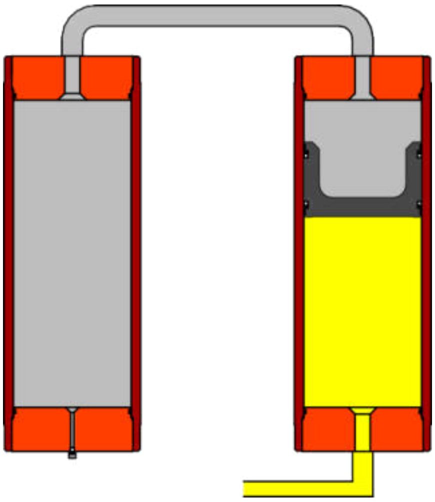
Hydro-pneumatic accumulators should always be used in conjunction with a safety block, to enable the accumulator to be isolated from the circuit in an emergency or for maintenance purposes, (see Section8 e 9).

1.1.4.13 CERTIFICATION

Accumulators are frequently required to conform to national or international certification. These requirements range from simple design factors to elaborate materials testing and inspection procedures carried out by an external agency. Most of the accumulators within Epe's piston, bladder or diaphragm ranges are available with certification PED97/23EC or other on request (see Section 1.4)

1.1.5 GAS BOTTLES INSTALLATION

Remote gas storage offers installation flexibility where the available space or position cannot accommodate an accumulator of the required size. A smaller accumulator may be used in conjunction with an Epe additional gas bottle, which can be located elsewhere (Fig. 1.1l)



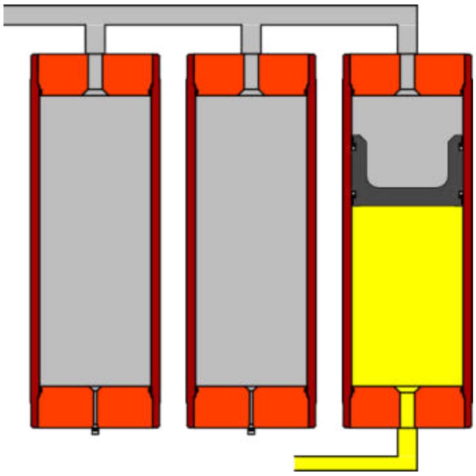
1.1l

Fig. 1.1l Piston accumulator with additional bottles type AB.

The gas cylinder and the accumulator must be sized by Section 2:
Gas bottle installations may use either bladder or piston accumulators, subject to the following considerations.

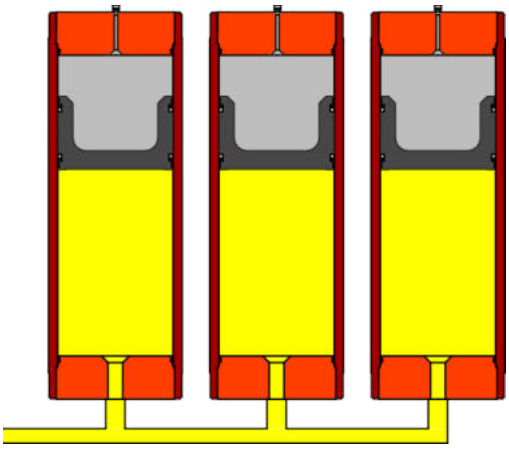
- Any accumulator used with remote gas storage should generally have the same size port of the gas end as at the hydraulic end, to allow an unimpeded flow of gas to and from the gas bottle. The gas bottle will have an equivalent port in one end and a gas charging valve at the other.
- A piston accumulator should be carefully sized to prevent the piston bottoming at the end of the cycle. Bladder designs should be sized to prevent filling of more than 75% full.
- Bladder installations require a special device called transfer barrier at the gas end, to prevent extrusion of the bladder into the gas bottle piping. The flow rate between the bladder transfer barrier and its gas bottle will be restricted by the neck of the transfer barrier tube.
- Because of the above limitations, piston accumulators are generally preferred to bladder types for use in gas bottle installations.
- Diaphragm style accumulators are normally not used in conjunction with gas bottles.

The requirement for an accumulator with an output of more than 200 litres cannot usually be met by a single accumulator, because larger piston designs are relatively rare and expensive, and bladder designs are not generally available in these sizes. The requirement can, however, be met using one of the multiple-component installations shown in Figs. 1.1m and 1.1n.



1.1m

Fig. 1.1m (above) Several gas bottles can supply pre-charge pressure to a single accumulator



1.1n

Fig. 1.1n (above) Multiple accumulators connected together offer high system flow rates

The installation in Fig. 1.1m consists of several gas bottles serving a single piston accumulator through a gas manifold. The accumulator portion may be sized outside of the limitations of the sizing formula on Section 2.2, but should not allow the piston to strike the caps repeatedly while cycling. The larger gas volume available with this configuration allows a relatively greater piston movement – and hence fluid output – than with a conventionally sized single accumulator. Afurther advantage is that, because of the large pre-charge “reservoir”, gas pressure is relatively constant over the full discharge cycle of the accumulator. The major disadvantage of this arrangement is that a single seal failure could drain the whole gas system.
The installation in Fig. 1.1n uses several accumulators, of piston or bladder design, mounted on a hydraulic manifold. Two advantages of multiple accumulators over multiple gas bottles are that higher unit fluid flow rates are permissible, and a single leak will not drain pre-charge pressure from the entire system.
A potential disadvantage is that, where piston accumulators are used, the piston with the least friction will move first and could occasionally bottom on the hydraulic end cap. However, in a slow ore infrequently used system, this would be of little significance.

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1.1.6 FAILURE PREVENTION

Accumulator failure is generally defined as inability to accept and exhaust a specified amount of fluid when operating over a specific system pressure range.
Failure often results from an unwanted loss or gain of pre-charge pressure.
It cannot be too highly stressed that the correct pre-charge pressure is the most important factor in prolonging accumulator life.
If maintenance of the pre-charge pressure and relief valve settings are neglected, and if system pressures are adjusted without making corresponding adjustments to pre-charge pressures, shortened service life will result.

1.1.6.1 FAILURE

Bladder/diaphragm accumulator failure occurs rapidly due to bladder/diaphragm rupture (Fig. 1.1o). Rupture cannot be predicted because the intact bladder or diaphragm is essentially impervious to gas of fluid seepage; no measurable gas or fluid leakage through the bladder or diaphragm precedes failure.

1.1.6.2 PISTON ACCUMULATOR FAILURE

Piston Accumulator failure generally occurs in one of the following gradual modes.

- FLUID LEAKS TO THE GAS SIDE

This failure, sometimes called dynamic transfer, normally takes place during rapid cycling operations after considerable time in service. The worn piston seal carries a small amount of fluid into the gas side during each stroke.
As the gas side slowly fills with fluid, pre-charge pressure rises and the accumulator stores and exhausts decreasing the amounts of fluid. The accumulator will totally fail when pre-charge pressure equals the maximum hydraulic system pressure. At that point, the accumulator will accept no further fluid. As the increase in pre-charge pressure can be measured (Fig. 1.1oa), failure can be predicted and repairs can be carried out before total failure occurs.

- GAS LEAKAGE

Pre-charge may be lost as gas slowly bypasses the damaged piston seals. Seal deterioration occurs due to excessively long service, fluid contamination or a combination of the two. Gas can also vent directly through a defective gas core or an end cap O-ring.
The reducing pre-charge pressure then forces progressively less fluid into the system. As this gradual decrease in pre-charge pressure can be measured (Fig. 1.1ob), repairs can again be carried out before total failure occurs.

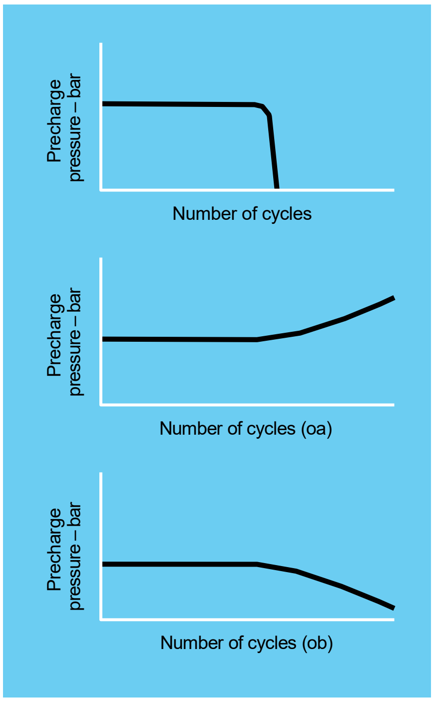


Fig.1.1.o When an accumulator bladder ruptures, precharge pressure immediately falls to zero

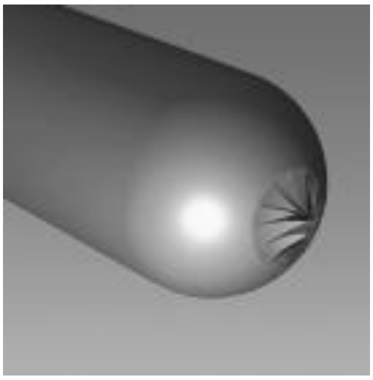
As fluid leaks past an accumulator piston, precharge pressure rises (oa).

Gas leaking past the piston or valve causes precharge pressure to fall (ob)

1.1o

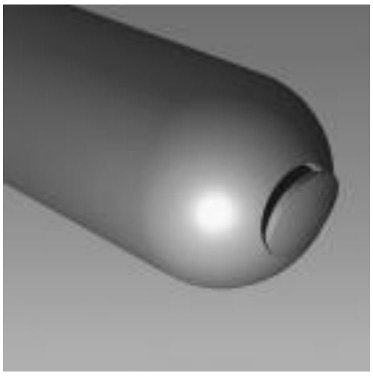
1.1.7 PRE-CHARGING PROCESS

Correct pre-charging involves accurately filling of the gas side of an accumulator with a dry, inert gas such as nitrogen, before admitting fluid to the hydraulic side.
It is important to pre-charge an accumulator under the correct specified pressure. Pre-charge pressure determines the volume of fluid retained in the accumulator at minimum system pressure. In an energy storage application, a bladder/ diaphragm accumulator is typically pre-charged to 90% of the minimum system pressure, and a piston accumulator to 97% of the minimum system pressure at the system operating temperature.
The ability to correctly carry out and maintain pre-charging is an important factor when choosing the type of accumulator for an application.
Bladder accumulators are more susceptible to damage during pre-charging than piston types. Before pre-charging and entering in service, the inside of the shell should be lubricated with system fluid.
This fluid acts as a cushion and lubricates and protects the bladder as it expands. When pre-charging, the first 10 bar of nitrogen should be introduced slowly. Failure to follow this precaution could result in immediate bladder failure: high pressure nitrogen, expanding rapidly and thus cold, could form a channel in the folded bladder, concentrating at the bottom. The chilled expanding rapidly brittle rubber would then inevitably cause the rupture (Fig. 1.1p).
The bladder could also be forced under the poppet, resulting in a cut. (Fig. 1.1q).
Close attention should be paid to operating temperature during pre-charging, as an increase in temperature will cause a corresponding increase in pressure which could then exceed the pre-charge limit.
Little damage can occur when pre-charging or checking the pre-charge on a piston accumulator, but care should be taken to make sure the accumulator is void of all fluid to prevent getting an incorrect reading on the pre-charge.



1.1p

Fig. 1.1p Starburst rupture caused by loss of bladder elasticity



1.1q

Fig. 1.1q C-shaped cut shows that bladder has been trapped under poppet

EXCESSIVELY HIGH PRE-CHARGE

Excessive pre-charge pressure or a decrease in the minimum system pressure without a corresponding reduction in pre-charge pressure may cause operating problems or damage to accumulators.

With excessive pre-charge pressure, a piston accumulator will cycle between stages (e) and (b) of Fig. 1.1e), and the piston will travel too close to the hydraulic end cap. The piston could bottom at minimum system pressure, thus reducing the output and eventually damaging the piston and the piston seal. The piston can often be heard bottoming, warning of impending problems.

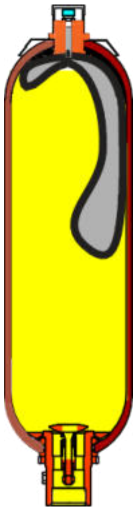
An excessive pre-charge in a bladder accumulator can drive the bladder into the poppet assembly when cycling between stages (e) and (b).This could cause fatigue failure of the poppet spring assembly, or even a pinched and cut bladder, should it become trapped beneath the poppet as it is forced closed (Fig. 1.1q). Excessive pre-charge pressure is the most common cause of bladder failure.

EXCESSIVELY LOW PRE-CHARGE

Excessively low pre-charge pressure or an increase in system pressure without a corresponding increase in pre-charge pressure can also cause operating problems and subsequent accumulator damage. With no pre-charge in a piston accumulator, the piston will be driven into the gas end cap and will often remain there. Usually, a single contact will not cause any damage, but repeated impacts will eventually damage the piston and seal.

Vice versa, for a bladder accumulator, too low or no pre-charge can have rapid and severe consequences. The bladder will be crushed into the top of the shell and can extrude into the gas stem and be punctured (Fig 1.1r). This condition is known as "pick out". One cycle as the one mentioned above is sufficient to destroy a bladder.

Overall, piston accumulators are generally more tolerant with respect to careless pre-charging.



1.1r

2.1.1 DEFINITIONS

Po = nitrogen pre-charge pressure (relative to the atmospheric pressure, namely the "relative pressure"). Measure to be performed when the accumulator is completely oil-free and at a temperature of 20 ± 2°C.

P1 = minimum working pressure of the hydraulic circuit (relative to the atmospheric pressure, namely the "relative pressure"). The minimum pressure must be higher than the pre-charge pressure.

P2 = maximum working pressure of the hydraulic circuit (relative to the atmospheric pressure, namely the "relative pressure").

P3 = calibration pressure of the safety valve (relative to the atmospheric pressure, namely the "relative pressure"). The pressure calibration of the safety valve must be greater than P2 at least of the hysteresis of the safety valve but equal or lower than the PS value.

PS = maximum working pressure of the accumulator (relative to the atmospheric pressure, namely the "relative pressure").

PT = testing pressure of the accumulator (relative to the atmospheric pressure, namely the "relative pressure"). Usually PT = PS x 1.43.

ΔP = is the difference between the maximum and minimum working pressure (P2-P1).

Po/P2 = compression ratio.

Vo = volume of gas under Po pressure

V = volume of fluid when the accumulator is completely full.

VoA = gas volume of the accumulator in case of a transfer bladder or piston accumulator.

V1 = volume of gas under P1 pressure.

V2 = volume of gas under P2 pressure.

V3 = volume of gas under P3 pressure.

ΔV = useful volume. It indicates the difference in volume of the working fluid between V1 and V2. Volume made by the accumulator during the working phase.

TSmin = minimum working temperature.

TSmax = maximum working temperature.

T20 = reference temperature at 20°C.

ts = discharge time of ΔV of the fluid.

tr = recharge time of ΔV of the fluid.

tc = plant cycle time. On a cyclical machine, it's the time between the start of a discharge of ΔV and the start of the next discharge.

N = number of cycles in a time unit.

η = polytrophic exponent.

Q = flow rate by volume.

2.1.2 UNIT OF MEASUREMENT

Pressure - Force/Surface

Pascal	**Pa**	1 Pa = 1 N/m² 1 kPa = 0.01 bar = 0.1 N/cm² = 0.10 mH2O = 7.5 mmHg = 0.0099 atm =0.145 psi = 0.02088 lbf/ft² = 0.334 ftH2O
bar	**bar**	1 bar = 100'000 Pa = 100 kPa = 0.1MPa = 1.0197 kg/cm² = 10.198 mH2O = 750 mmHg = 0.987 atm = 14.5 psi = 33.455 ftH2O
millibar	**mbar**	1 mbar = 100 Pa = 0.010 mH2O = 0.750 mmHg = 0.00102 kg/cm² = 0.0145 psi = 2.088 ldf/ft² = 0.033 ftH2O
millimetres of mercury	**mmHg**	1 mmHg = 133.322 Pa = 0.133 kPa = 0.00133 bar = 0.0136 mH2O = 0.00131 atm = 0.00136 kg/cm² = 0.01934 psi = 2.78 ldf/ft² = 0.045 ftH2O
technical atmosphere = kgf/cm²	**at Kg/cm²**	1 at = 1 kg/cm² = 735.56 mmHg = 10 mH2O = 98066.50 Pa = 98.067 kPa = 0.981 bar = 0.968 atm = 14.22 psi = 2048.16 lbf/ft² = 32.81 ftH2O
metric atmosphere	**atm**	1 atm = 101'325 Pa = 760 mmHg = 1.033 at = 10.33 mH2O = 1.01 bar = 14.696 psi = 2116.22 lbf/ft² = 33.9 ftH2O
water column metres	$\mathbf{m_{H2O}}$	1 mH2O = 9806 Pa = 0.09806 bar = 73.55 mmHg = 0.9806 N/cm² = 0.09678 atm = 0.0999 at = 1.4224 psi = 204.8 lbf/ft² = 3.28 ftH2O
foot of water	$\mathbf{ft_{H2O}}$	1 ftH2O = 2988.87 Pa = 0.0299 kPa = 0.3048 mH2O= 22.419 mmHg = 0.0295 atm = 0.03048 kg/cm² = 0.4335 psi = 62.42 ldf/ft²

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pounds per square inch	**psi**	1 psi = 6.894.76 Pa = 6.894 kPa = 0.069 bar = 0.703 mH2O = 51.715 mmHg = 0.689 N/cm² = 0.068 atm = 0.0703 kg/cm² = 144 lbf/ft² = 2.31 ftH2O
pounds per square foot	**lbf/ft²**	1 lbf/ft² = 2'988.87 Pa = 2.99 kPa = 0.0299 bar = 0.3048 mH2O = 22.418 mmHg = 0.299 N/cm² = 0.0295 atm = 0.0305 at = 0.433 psi = 62.424 lbf/ft²

Volume

cubic meter	**m³**	1 m³ = 1'000 dm³ = 35.3146 ft³ = 61'023.744 in³ = 1.308 yd³ = 264.20 galUS = 219.97 galUK
cubic decimetre; litre	**dm³**	1 dm³ = 1 l = 0.001 m³ = 61.024 in³ = 0.0353 ft³ = 0.00131 yd³ = 0.26417 galUS = 0.21997 galUK
cubic centimetre	**cm³, cc**	1 cm³ = 0.001 dm³ = 0.001 l = 0.061 in³ = 0.000264 galUS = 0.00022gal UK
cubic inch	**in³**	1 in³ = 0.0000164 m³ = 0.0164 dm³ = 0.0005787 ft³ = 0.0043 galUS = 0.0036 galUK
cubic foot	**ft³**	1 ft³ = 0.02832 m³ = 28.32 dm³ = 1'728 in³ = 0.037 yd³ = 7.48 galUS = 6.23 galUK
cubic yard	**yd³**	1 yd³ = 0.764 m³ = 764.55 dm³ = 46.656 in³ = 27 ft³ = 201.97 galUS = 168.18 galUK
gallon US	**galUS**	1 galUS= 0.00378 m³ = 3.785 dm³ = 231 in³ = 0.134 ft³ = 0.0049 yd³ = 0.833 galUK
gallon UK	**galUK**	1 galUK = 0.00455 m³ = 4.546 dm³ = 277.42 in³ = 0.16 ft³ = 0.0059 yd³ = 1.2 galUS

Temperature

kelvin	**K**	K = °C + 273.15 K = 1.8 · °R K = [5/9 · °F] + (459.67/1.8)
degree Centigrade	**°C**	°C = (°F - 32) · 5/9 °C = K - 273.15 °C = (5/9) · °F - (32/1.8)
degree Fahrenheit	**°F**	°F = 9/5 · °C + 32 °F = °R - 459.67 °F = (9/5) · K - 459.67
degree Rankine	**°R**	°R = (5/9) K °R = 491.67 + (9/5) · °C °R = 459.67 + °F

Time

seconds	**s**	s = 0.01666667 min s = 0.00027778 h s = 0.00001157 days
minutes	**min.**	min = 60 s min = 0.01666667 h min = 0.00071428 days
hours	**h**	h = 60 min h = 0.041666667 days h = 3600 s
days	**days**	day = 86400 s day= 1440 min day= 24 h

Flow rate by volume

cubic meters per second	**m³/s**	1 m³/s = 60 m³/min = 3'600 m³/hour = 1'000 l/s = 60'000 l/min = 6'102'374.42 in³/s = 2'118.88 ft³/min = 15'850.32 gpm = 13'198.13 l gpm
cubic meters per minute	m³/min	1 m³/min = 0.0167 m³/s = 60 m³/h = 16.67 l/s = 1'000 l/min = 35.31 ft³/min = 264.17 gpm = 219.97 l gpm
cubic meters per hour	**m³/h**	1 m³/h = 0.000278 m³/s = 0.0167 m³/min = 0.28 l/s = 16.67 l/min = 1017.06 in³/s = 0.588 ft³/min = 4.40 gpm = 3.66 l gpm
litres per second	**l/s**	1 l/s = 0.001 m³/s = 0.06 m³/min = 3.6 m³/h = 60 l/min = 3661.42 in³/min = 2.12 ft³/min = 15.85 gpm = 13.198 l gpm
litres per minute	**l/min**	1 l/min = 0.001 m³/min = 0.06 m³/h = 0.0167 l/s = 61.024 in³/min = 0.035 ft³/min = 0.264 gpm = 0.22 Igpm
cubic inch per minute	**in³/min**	1 in³/min = 0.00027 l/s = 0.016 l/min = 0.00058 ft³/min = 0.0043 gpm = 0.0036 l gpm
cubic foot per minute	**ft³/min**	1 ft³/min = 0.00047 m³/s = 0.028 m³/min = 1.7 m³/h = 0.472 l/s = 28.32 l/min = 1'728 in³/min = 7.48 gpm = 6.23 l gpm
gallon per minute	**gpm**	1 gpm = 0.0038 m³/min = 0.227 m³/h = 0.063 l/s = 3.785 l/min = 231 in³/min = 0.134 ft³/min = 0.833 l gpm
imperial gallon per minute	**l gpm**	1 l gpm = 0.000076 m³/s = 0.00454 m³/min = 0.273 m³/h = 0.076 l/s = 4.55 l/min = 277.42 in³/min = 0.16 ft³/min = 1.2 gpm

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2.2.1 PRINCIPLE OF OPERATION

Gas compression

In hydropneumatic accumulators, oil or other liquids are maintained under pressure by a pre-compressed gas, usually nitrogen. Therefore, we show some principles on the compression of gases, useful then in the calculation of the accumulators. The fundamental characteristics of a gas are: volume, temperature and pressure.

The law governing these functions is the one on the ideal gases of Boyle and Mariotte, which states that in every condition under which we place a certain amount of gas, the product between its pressure (relative to vacuum) and its volume is constant. The law adds that this remains constant even if the passage from one state to another occurs with equal heat exchange with the external environment.

This means that, for a given quantity of gas, if the volume available is halved, the pressure is twice; the product of the volume for the absolute pressure is constant.

$P_1*V_1=P_2*V_2=P_3*V_3=$ .... $=constant$

According to the law of Gay-Lussac: at constant volume, in an ideal gas, the absolute pressure and the temperature are directly proportional. Maintaining a constant pressure in an ideal gas, its volume V varies directly with temperature T:

V1:V2=T1:T2

And maintaining a constant volume, the pressure varies in proportion to temperature changes:

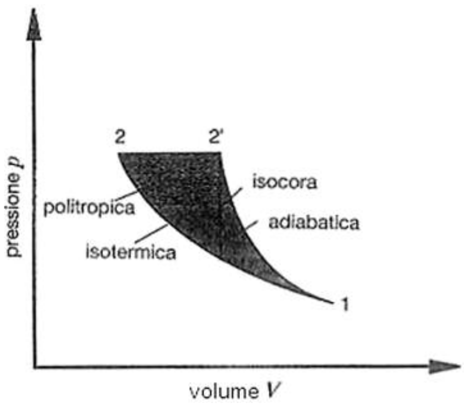
P1:P2=T1:T2

From this it follows that an increase in pressure leads to an increase in temperature and, conversely, a decrease in pressure causes a decrease in temperature. The laws of Boyle and Mariotte and Gay-Lussac are valid exactly only for ideal gases; the nitrogen, being a real gas, is bound to small and influential changes than the laws of the compression of ideal gases. Another crucial factor concerns the change of the aeriform state.

Change in gas state

The state change of the gas may be:
- isochore
- isothermal
- adiabatic
- polytrophic

Diagram 2.2a : change of state in the diagram P - V



2.2a

Changes in isochore

This change of state is characterized by a constant volume of gas. It occurs when the gas area of the accumulator is pre-charged at low temperature and then subjected to a pressure increase at constant volume due to heat exchange with the environment.

Equation of state: $P/T=P_1/T_1=constant$

Isothermal change

This variation, characterized by the constant temperature of the gas, occurs when the charging or discharging of the fluid to / from the accumulator occurs in long times, allowing for the complete heat exchange between the gas and the environment (more than 180 seconds).

Equation of state: $P \times V = P_1 \times V_1 = constant$

Adiabatic change

The adiabatic change occurs when the discharge and charge of the fluid to / from the accumulator is so fast as to prevent any heat exchange between the gas and the environment (less than 60 seconds).

Equation of state: $P \times V^{k} = P_1 \times V^{1k} = constant$

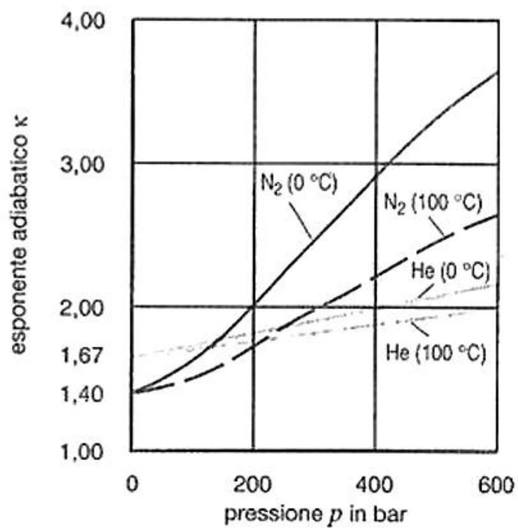
The relationships between temperature and volume and between temperature and pressure are expressed by the thermal equations of state:

$T \times V^{k-1} = T_1 \times V_1^{\,k-1}$

$T \times P^{(1-k)/k} = T_1 \times P_1^{\,(1-k)/k}$

In these equations, “k” is the adiabatic exponent, which for a diatomic gas such as nitrogen under normal conditions, is equal to 1,4.

Diagram 3: evolution of the adiabatic nitrogen exponent depending on the pressure at temperatures of 0°C and 100°C.



2.2b

Polytrophic change

The operation of an accumulator never occur under the theoretical assumptions, namely without heat exchange. In practice, there is an intermediate change of state between the isothermal and adiabatic ones, which takes the name of polytrophic.

The valid relations are similar to those of the adiabatic change, but it has to substitute the adiabatic change adiabatic exponent with the polytrophic exponent N.

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2.2.2 SIZING OF THE ACCUMULATOR

With the sizing of the accumulator, we want to establish the geometric capacity according to the pressures within which it works, the amount of fluid that it has to store and return and the time required.
In light of the above, it follows that the equations to be used for the calculation of an accumulator depends on the actual duration of the process of absorption/delivery of the fluid.
As empiric rule for choosing the appropriate equations, apply the following criteria:
- cycle duration < 1 minute: adiabatic change
- cycle duration < 3 minutes: isothermal change
- cycle duration between 1 and 3 minutes: polytrophic change.

The equations to be used for the calculation of the accumulator are shown in Table 3. It should also be noted that the calculation of the accumulator involves some experimental values, which, on one hand, ensure the optimal exploitation of the accumulator volume and, on the other, allow not to endanger the duration. Table 2 shows the experimental values for the various types of accumulators.

Deviations of the real gases

The equations of state shown in the preceding paragraphs apply only if the gas follows the ideal behaviour. In fact, various gases such as nitrogen, differ (especially at other pressures) by the laws of the ideal gas. This behaviour is called "real".
For real gases, relations between the parameters of state (P, T, and V) can be represented only by approximate equations, whose sufficiently precise use is very laborious and long. We prefer, therefore, to take into account the behaviour of the real gases by introducing appropriate correction factors.
In this case, the real volume for an isothermal change of state is expressed by

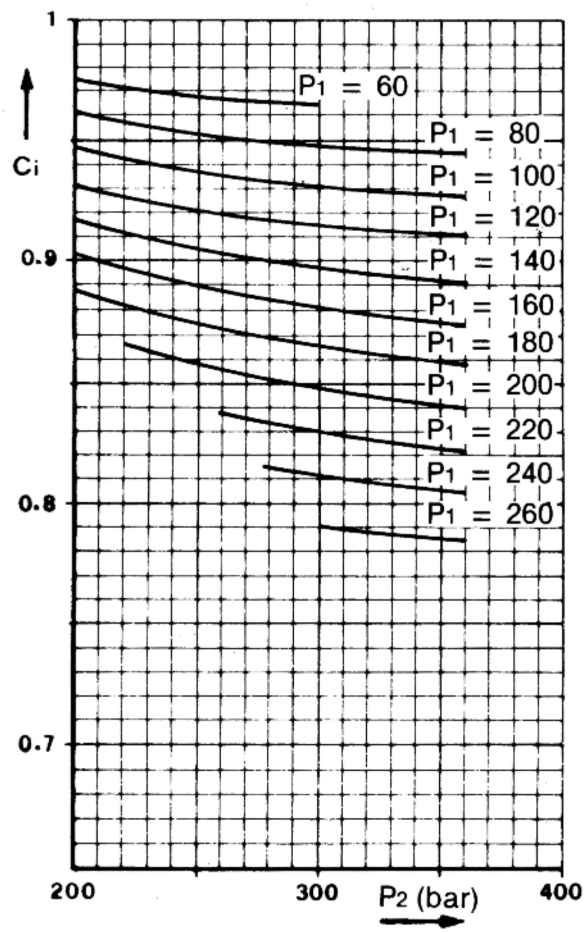
$\mathbf{V_{0\,real} = C_i \times V_{0\,ideal}}$

and for an adiabatic change of state is expressed by

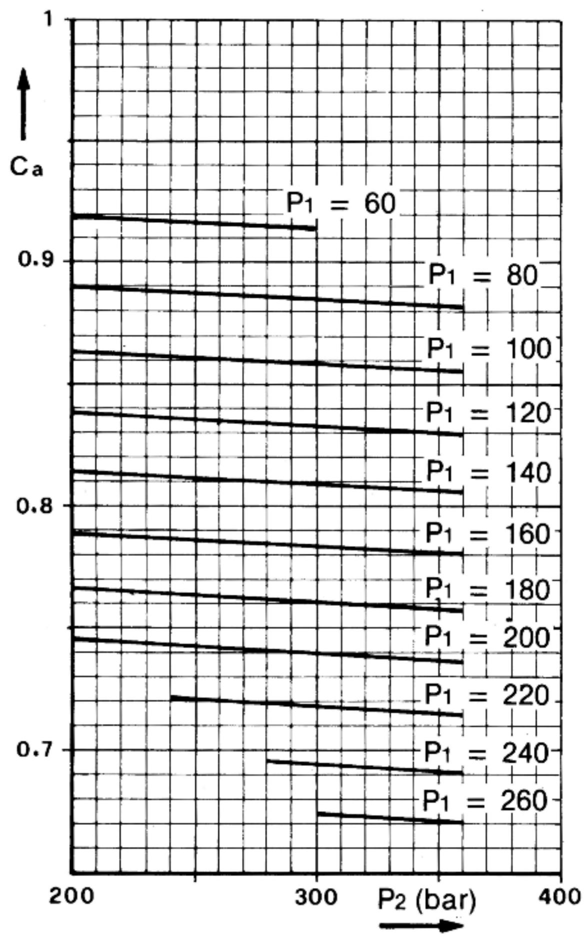
$\mathbf{V_{0\,real} = C_a \times V_{0\,ideal}}$

The correction factors C_i and C_a in the equations can be obtained from the following diagrams

Isothermal correction coefficent C_i



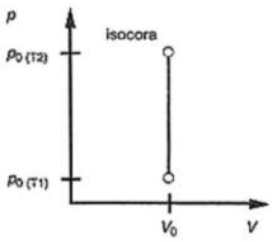
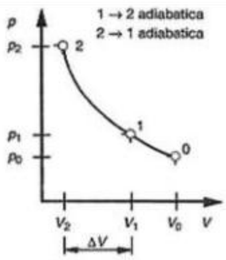
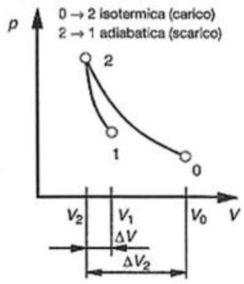
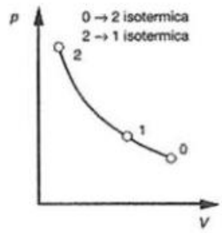
Adiabatic correction coefficient C_a



2.2c

Accumulator	Bladder accumulator High pressure	Bladder accumulator Low pressure	Diaphragm accumulator welded -	Diaphragm accumulator screwed	Piston accumulator with reduced friction
Gas pre-charge pressure P_0 (T_b) (at max. working temperature)	≤ 0.9 $\cdot p_1$ (accumulation of energy) = (0.6-0.9) $\cdot p_m$ (shock absorption)		≤ 0.9 $\cdot p_1$ (accumulation of energy) 0.6 $\cdot p_m$ (pulsations damping)		≤ p_1 - 5 bar < 2 bar (piston with reduced friction) < 10 bar (normal piston)

2.2d

Cycle (state change)	Equation	Notes
	$P_{0\,(T1)} = P_{0\,(T2)} \cdot$ **Ts min/ Ts max**	$P_{0\,(T1)}$ = pre-charge pressure at minimum temperature Ts min (degrees Kelvin) $P_{0\,(T2)}$ = pre-charge pressure at maximum temperature Ts max (degrees Kelvin) **Use** Calculation of the pre-charge pressure when the operating temperature is different from the pre-charge temperature.
	$\Delta V = V_0\,[(p_0/p_1)^{1/n} - (p_0/p_2)^{1/n}]$ $V_0 = \Delta V/\,(p_0/p_1)^{1/n} - (p_0/p_2)^{1/n}$	$\eta = K = 1.4$ for nitrogen (p_0 at temperature Ts min) **Use** Accumulation of energy
	$\Delta V_2 = V_0\; p_0/p_2\; [(p_0/p_1)^n - 1]$ $V_0 = \Delta V \cdot p_2/\,p_0\,/\,(p_2/p_1)^{1/n} - 1$	**Use** Emergency, safety (p_0 at temperature Ts min)
	$\Delta V = V_0\,(p_0/p_1 - p_0/p_2)$ $V_0 = \Delta V\,/\,p_0/p_1 - p_0/p_2$	**Use** Leak and volume compensation (p_0 at temperature Ts min)

2.2e

CALCULATION OF THE ACCUMULATORS

WEBBER HYDRAULICS

Temperature variation

Temperature variation can seriously affect the pre-charge pressure of an accumulator. As the temperature increases, the pre-charge pressure increases; conversely, decreasing temperature will decrease the pre-charge pressure. In order to assure the accuracy of your accumulator pre-charge pressure, you need to factor in the temperature variation. The temperature change is determined by the temperature encountered during the pre-charge versus the operating temperature expected in the system.

NOTE: it is important to wait for the thermal exchange caused by pressure shifts to be stabilized in order to check or adjust the pre-filling pressure. As a safety measure, isolate the nitrogen source during the stabilization period.

Equation used

This equation is used for correction of nitrogen filling pressure Po in relation to the operating temperature.

$$P0\,(Ts) = P0\,(T_{20}) \times \frac{Ts + 273}{T_{20} + 273}$$

P0 (Ts) = filling pressure at checking temperature

P0 (T_{20}) = nitrogen pressure P0 at 20°C

NITROGEN FILLING PRESSURE

200	173	183	186	193	200	207	214	221	227	234	241	248	255	261	268
190	164	171	177	184	190	197	203	210	216	222	229	235	246	248	255
180	155	162	168	174	180	186	192	198	205	211	217	223	229	235	241
170	147	153	158	164	170	176	182	187	193	199	205	211	216	222	228
160	138	144	149	155	160	166	171	176	182	187	193	198	204	209	215
150	130	135	140	145	150	155	160	165	171	176	181	186	191	196	201
140	121	126	130	135	140	145	150	154	159	164	169	173	178	183	188
130	112	117	121	126	130	134	139	143	148	152	157	161	166	170	174
120	104	108	112	116	120	124	128	132	136	141	145	149	153	157	161
110	95	99	103	106	110	114	118	121	125	129	133	136	140	144	148
105	91	94	98	101	105	109	112	116	119	123	127	130	134	137	141
100	**86**	**90**	**93**	**97**	**100**	**103**	**107**	**110**	114	117	120	124	127	131	134
95	82	85	89	92	95	98	102	**105**	108	111	115	118	121	124	127
90	78	81	84	87	90	93	96	**99**	102	105	108	112	115	118	121
85	73	76	79	82	85	88	91	**94**	97	100	102	105	108	111	114
80	69	72	75	77	80	83	86	**88**	91	94	96	99	102	105	107
75	65	67	70	72	75	78	80	**83**	85	88	90	93	96	98	101
70	60	63	65	68	70	72	75	**77**	80	82	84	87	89	92	94
65	56	58	61	63	65	67	69	**72**	74	76	78	81	83	85	87
60	52	54	56	58	60	62	64	**66**	68	70	72	74	76	78	81
55	48	49	51	53	55	57	59	**61**	63	64	66	68	70	72	74
50	43	45	47	48	50	52	53	**55**	57	59	60	62	64	65	67
45	39	40	42	43	45	47	48	**50**	51	53	54	56	57	59	60
40	35	36	37	39	40	41	43	**44**	45	47	48	50	51	52	54
35	30	31	33	34	35	36	37	**39**	40	41	42	43	45	46	47
30	26	27	28	29	30	31	32	**33**	34	35	36	37	38	39	40
25	22	22	23	24	25	26	27	**28**	28	29	30	31	32	33	34
20	17	18	19	19	20	21	21	**22**	23	23	24	25	26	26	27
15	13	14	14	15	15	16	16	**17**	17	18	18	19	19	20	20
10	8.6	9	9.3	9.7	10	10.4	10.8	**11.1**	11.4	11.8	12.2	12.6	13	13.4	13.8
5	4.3	4.5	4.7	4.8	5	5.2	5.3	**55.5**	5.7	5.9	6	6.2	6.4	6.5	6.7
	-20	**-10**	**0**	**10**	**20**	**30**	**40**	**50**	**60**	**70**	**80**	**90**	**100**	**110**	**120**

REFERENCE TEMPERATURE °C

2.2f

2.2.3 EMERGENCY ENERGY RESERVE WITH BLADDER ACCUMULATOR

Typical occasion when storage is slow (isothermal) and discharge is quick (adiabatic).

Volume will be given by:
$V_o = \Delta V / (P_0/P_2)^{1/n} \bullet [(P_2/P_1)^{1/n} - 1]$

And stored volume by:
$\Delta V = V_o (P_0/P_2)^{1/n} \bullet [(P_2/P_1)^{1/n} - 1]$

Where:
n = 1.4 adiabatic coefficient (quick discharge phase)
n_c = 1 ÷ 1.4 polytrophic coefficient (slow storage phase)

$V_o = \Delta V\ P_2/P_0 / (P_2/P_1)^{0,7143} – 1$; $\Delta V = V_o\ P_0\ (P_2/P_0)^{0,7143} – 1/\ P_2$

Example:
An accumulator must discharge 4.6 litres of oil in 3 seconds with a change of pressure from P_2 = 280 bar to P_1 = 220 bar.
The loading time is 4 minutes. Define the capacity keeping in mind that ambient temperature will change from 20°C to 50°C.

$V_o = \Delta V/ (P_0/P_2)^{1/1.1} - [(P_2/P_1)^{1/1.4} - 1]$

$= 4.6 / (199/281)^{0.09091} \bullet [(281/221)^{0.7143} -1] = 33.63$ l

P_1 = 221 abs. bar	n_c = 1.1 (from Fig.2.2a)
P_2 = 281 abs. bar	T_1 = (273+20) = 293 °K
P_0 = 0.9x220 = 198 = 199 bar abs.	T_2 = (273+50) = 323 °K

Considering the correction coefficient for high pressure and the temperature change, we have:

$V_{ot} = V_o/C_m \times T_2/T_1 = 33.63/0.777 \times 323/293 = 47.7$ l

Where:
C_a = 0.72
C_i = 0.834
$C_m = C_a + C_i / 2 = 0.777$

The pre-charge pressure at 20°C will be:

$P_{0(20°C)}$ = 199 x 293/323 = 180.5 bar = 179.5 rel. bar

The accumulator type is **AS55P360**.....

2.2.4 PULSATION COMPENSATOR Q WITH BLADDER ACCUMULATOR

A typical calculation in adiabatic conditions due to high speed storage and discharge.
The fluid amount ΔV to be considered in the calculation depends on the type and capacity of the pump:

$\Delta V = K \bullet q$

Volume becomes:
$V0 = K \bullet q / (P_0/P_1)^{0.7143} - (P_0/P_2)^{0.7143}$

Where:
q = pump displacement (litres)
= A x C (piston surface x stroke)
= Q/n = flow rate (l/min) / strokes/min.
P = average working pressure (bar)
P_1= P-X (bar)
P_2= P+X (bar)
X = α • P/100 (bar) deviation from average pressure
α = remaining pulsation ± (%)
K = coefficient taking into account the number of pistons and if pump is single or double acting.

Pump type	K
1 piston, single acting	0.69
1 piston, double acting	0.29
2 pistons, single acting	0.29
2 pistons, double acting	0.17
3 pistons, single acting	0.12
3 pistons, double acting	0.07
4 pistons, single acting	0.13
4 pistons, double acting	0.07
5 pistons, single acting	0.07
5 pistons, double acting	0.023
6 pistons, double acting	0.07
7 pistons, double acting	0.023

2.2g

Example:
Assume a 3-piston pump, single acting, with a flow rate Q = 8 m3/h and operating pressure of 20 bar. Calculate the volume necessary to limit the remaining pulsation to α = ± 2.5%. Pump RPM 148. Working pressure 40°C.

P = 20 bar	q = 8000/60x148x3 = 0.3 l
P_2 = (20 - 0.5) = 19.5 bar	K = 0.12
P_2 = (20 + 0.5) = 20.5 bar	X = 2.5x20/100 = 0.5 bar
P_0 = (0.7 • 20) = 14 bar	

$V_0 = 0.12 \times 0.3 / (15/20.5)^{0.7143} – (15/21.5)^{0.7143} = 1.345$ l
$P_{0(20°C)}$ = 15 x 293/313 = 14 abs. Bar = 13 bar rel.

The most suitable accumulators is the low pressure type: **AS1.5P80**...

2.2.5 HYDRAULIC LINE SHOCK DAMPER WITH BLADDER ACCUMULATOR

A rapid increase in pressure caused by a high acceleration or deceleration in flow is commonly known as water hammer. The overpressure, **ΔP max**, that takes place in piping, the flow rate, the density of the liquid and the valve shut down time.
This is given by:

ΔP max (bar) = 2 ɣ L v / t x 10^5

The volume of the accumulator, required to reduce shock pressure within predetermined limits AP, is obtained by:

$V_0 = Q/7.2\ (\mathbf{2\ ɣ\ L\ v\ /\ C_0} \times 10^5 - t) / (P_0/P_1)^{0.7143} - (P_0/P_2)^{0.7143}$

Where:
V_0 = accumulator gas capacity (litres)
Q = flow rate in the piping (m^3/h)
L = total length of piping (m)
ɣ = specific gravity of the fluid (kg/m^3)
V = Q/S x 103/3.6 = flow velocity (m/s)
S = Лd 2 / 4 = internal pipe section (mm^2)
d = internal pipe diameter (mm)
ΔP = allowable overpressure (bar)
P_1 = operating pressure by free flow (absolute bar)
P_2 = P + ΔP = max allowable pressure (absolute bar)
t = deceleration time (s) (valve shut down, etc.)

Example:
Assume a water pipe (ɣ = 1000 kg/m^3) with internal diameter d = 80 mm, length L = 450 m, flow rate Q = 17 m^3/h, operating pressure P_1 = 5 bar, allowable overpressure ΔP = 2 bar, valve closure time t = 0.8 s.

ΔP max = 2 x 1000 x 450 x 0.94 / 0.8 • 10^5 = 10.57 bar

The accumulator volume necessary to reduce the ΔP max to 2 bar is:

$V_0 = 17/7.2\ (2 \times 1000 \times 450 \times 0.94 / 2 \times 10^5 - 0.8)/(5.5/6)^{0.7143} - (5.5/8)^{0.7143} = 46.4$ l

Where:
S = Л x 80^2 / 4 = 5026.5 mm^2
V = 17 x 103 / 5026.5 x 3.6 = 0.94 m/s
P_0 = 5 x 0.9 = 4.5 = 5.5 abs. bar
P_1 = 6 abs. bar
P_2 = 5 + 2 = 7 bar = 8 abs. bar

An accumulator of 55 litres low pressure range will be chosen, type **AS55P30**...

2.2.6 PISTON ACCUMULATOR + ADDITIONAL GAS BOTTLES (TRANSFER)

In all case where a considerable amount of fluid must be obtained with a small difference between P1 and P2, the resultant volume V0 is large compared to ΔV.
In these cases, it could be convenient to get the required nitrogen volume by additional bottles. Volume calculation is performed, according to the application, both in isothermal as well as in adiabatic conditions, using the formulas given above always taking temperature into account. To get the maximum of efficiency, it is convenient to fix a quite high pre-charge value. In case of **energy reserve**, it is possible to use:

$\mathbf{P_0 = 0.97\ P_1}$ **or** $\mathbf{P_0 = P_1 - 5}$

Once the required gas volume is calculated, the volume must be allocated between the minimum indispensable portion VA, which represents the volume of additional bottles.

$\mathbf{V_{oT} = V_{oA} + V_{oB}}$

Where:

$V_{oA} \geq \Delta V + (V_{oT} - V_o) / 0.75$

This means that the sum of the required fluid volume plus the volume change due to temperature must be **lower than ¾ of the accumulator capacity**. The bottles volume is given by the difference.

$V_{oB} = V_{oT} - V_{oA}$

Example:
Suppose ΔV = 30 l. to be obtained in 2 seconds, from a pressure P2 = 180 bar to P1 = 160 bar.
Temperature: **q**1 = 20°C; **q**2 = 45°C

$P_{0(45°C)} = 0.97 \times 160 = 155$ bar

$V_0 = \Delta V / (P_0/P_1)^{0.7143} - (P_2/P_1)^{0.7143}$

$= 30 / (156/161)^{0.7143} - (156/181)^{0.7143} = 382.4$ l

$V_{oT} = 382.4 \times 318 / 293 = 415$ l

$V_{oA} = 30 + (415 - 382.4) / 0.75 = 83.5$ l

One accumulator **AP100**... is used with the total V0 = 100 l. plus **6 bottles of 50** l. type **B52P360**... or 4 additional bottles type **B75P360**... of 75 l.

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