Hydraulic Accumulators

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1 Introduction

Accumulators have a wide variety of applications in hydraulic systems because of their very useful properties.

Their main applications are for:

- Energy storage
- Emergency operation
- Leakage fluid make-up
- Volume compensation
- Shock absorption
- Pulsation damping

As its name suggests, a hydraulic accumulator is a pressure vessel that is able to store a specific volume of fluid sufficient to perform its intended purpose. When necessary, the fluid that the accumulator has taken from the system is returned to it without the need for any external supply of energy. The storage of pressure energy in the volume of fluid can be effected by either weight, spring or gas (i.e. hydro-pneumatically). Since hydro-pneumatic accumulators are the most popular the following chapter will concentrate on them.

2 Types of hydro-pneumatic accumulator

As *Fig. 47* shows, hydraulic accumulators can be classified according to two features:

- the energy carrier and
- the separating element.

The purpose of all hydraulic accumulators is to store pressure energy. In the mechanical types (i.e. weight-loaded and spring-loaded) it is performed by a change in potential energy. In contrast, with the gas-loaded accumulator it is the internal energy of a gas that is changed. For this type of accumulator, classification according to the separating element is ideal because they can be classified as either with or without a separating element.

Hydraulic accumulators with a separating element can be divided into:

- bladder-type accumulators
- diaphragm-type accumulators
- piston-type accumulators

The mode of operation of these accumulators utilizes the compressibility of a gas for storing a fluid. Nitrogen is often the energy carrier. Basically, a hydro-pneumatic accumulator comprises a fluid part and a gas part and a gas-tight separating element. The fluid part of the accumulator is connected to the hydraulic circuit so that, as the pressure rises, the gas in the gas part of the accumulator is compressed. Then, as the pressure in the system falls, the compressed gas expands and forces the stored fluid back into the system.

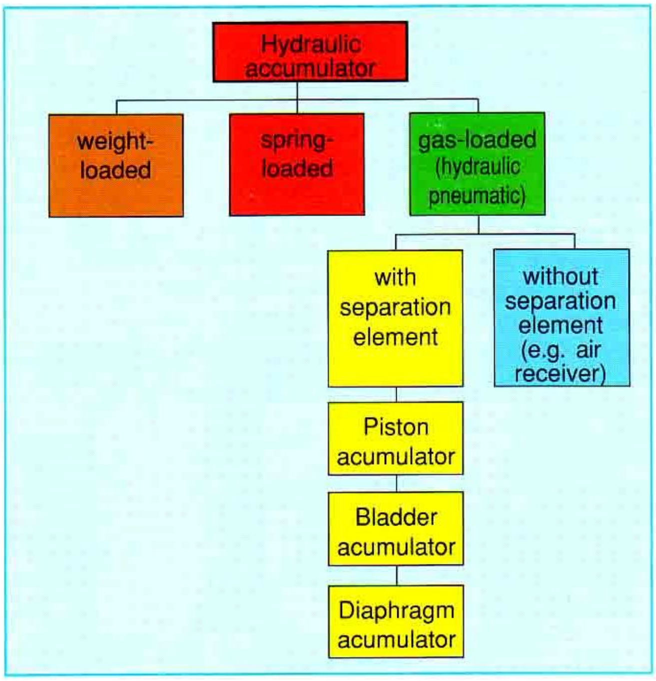


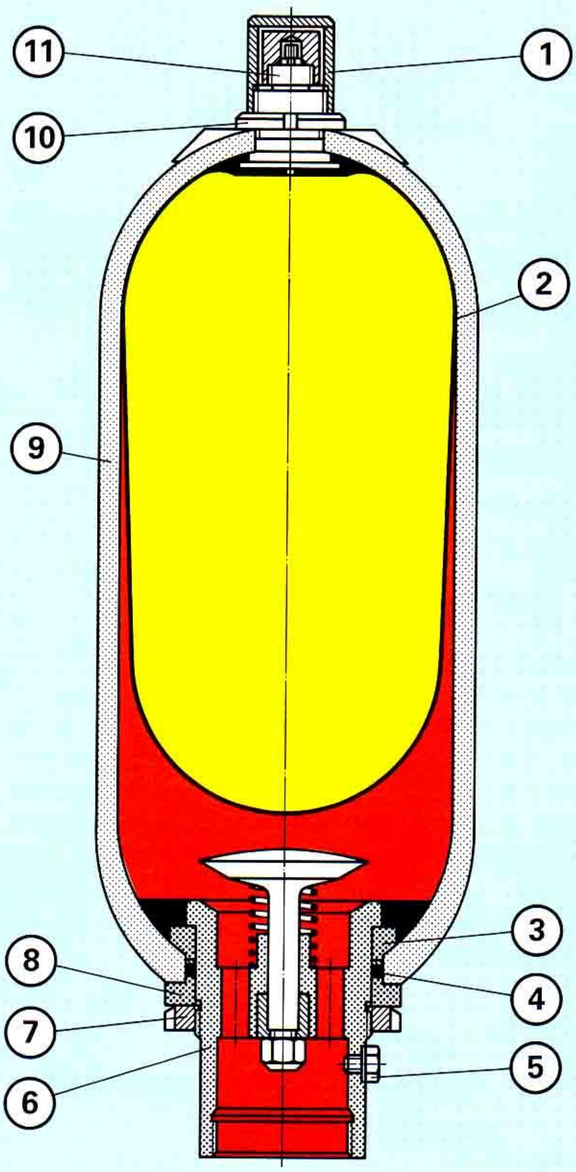
Fig. 47: *Classification of hydraulic accumulators*

2.1 Bladder-type accumulators

The bladder-type accumulator shown in *Fig. 48* comprises a strong pressure vessel, able to withstand the full system pressure, and an internal bladder containing the gas and made of a rubber-like material (elastomer). The bladder is filled through the gas valve at the top. The main purpose of the fluid valve at the bottom of the pressure vessel is to prevent the bladder being ejected with the out-flowing fluid.

For this purpose theopening area of the valve is sized so that a maximum volumetric flow depending on the size of the accumulator (approximately 120 L/s) cannot be exceeded. Volumetric flows of up to 140 L/s are possible with certain special designs called high-flow accumulators *(Fig. 49)*. The special feature of this design is a perforated disc in the connection fitting with the greater necessary opening area for the higher volumetric flow. An alternative version of the high-flow design is shown in *Fig. 50*. In this case the accumulator can be used at operating pressures up to 290 bar and the connection fitting contains a pre-loaded check valve which again prevents the bladder being ejected if there is a sudden drop in system pressure or the accumulator is emptied completely. The stem of the valve also incorporates a damping device so that the valve itself is not harmed by the high velocity flow during opening and closing.

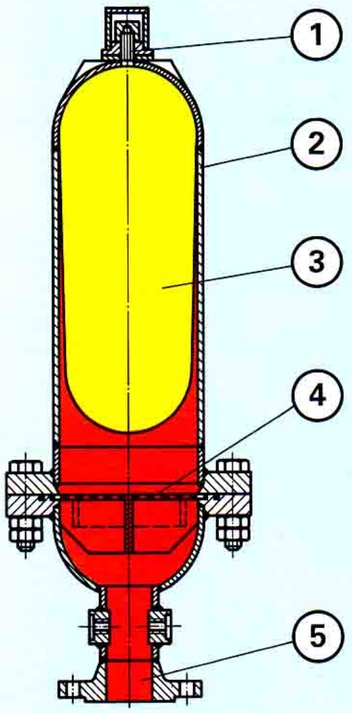
Generally speaking, it is necessary for the hole in the pressure vessel for fitting the fluid valve to be of larger diameter than that for the gas valve. Consequently, it is normal to insert and remove the bladder from the fluid end. In a few exceptional cases when removing the accumulator in order to change the bladder would involve a large amount of dismantling, or fast bladder changing is essential, it is also possible to insert and remove the bladder from the gas end (the "top repairable" type, *see Fig. 51*). The mode of operation of the bladder-type accumulator is as follows, referring to *Fig. 52*: The bladder is filled with nitrogen to a certain pressure specified by the manufacturer according to the operating regime. At this point the fluid valve is closed. If, now, the charging pressure of the accumulator is exceeded in the system, the valve opens and the hydraulic fluid flows into the accumulator. As the pressure increases further the gas is compressed up to the maximum operating pressure p_2. The change in gas volume in the bladder between minimum and maximum operating pressure represents the useful fluid capacity.



1 Sealing cap
2 Bladder
3 Split ring
4 Sealing ring
5 Vent screw
6 Fluid valve
7 Clamp nut
8 Spacer ring
9 Pressure vessel
10 Retaining nut
11 Gas valve

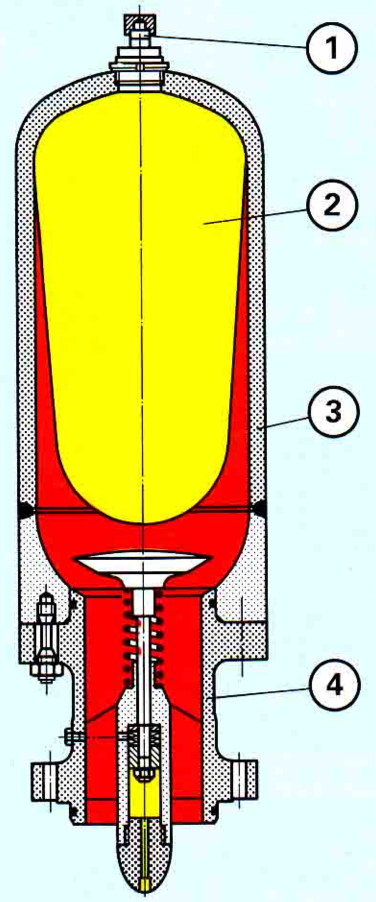
Fig. 48: *Hydro-pneumatic bladder-type accumulator*

Although bladder-type accumulators can technically be mounted in any position, the vertical is preferred. In vertically mounted or inclined accumulators, the fluid valve must always be at the bottom.



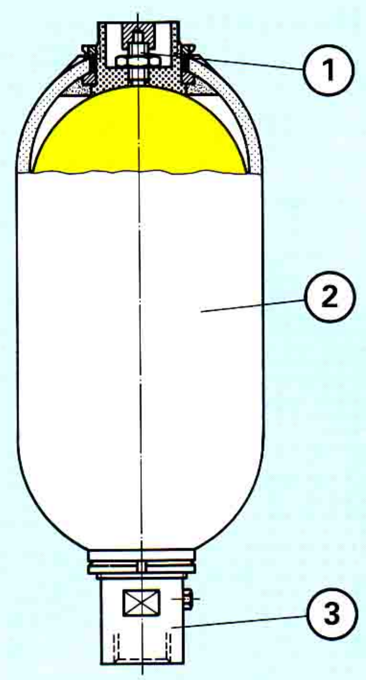
1 Gas valve
2 Pressure vessel
3 Bladder
4 Perforated disc
5 Fluid connection

Fig. 49: *High-flow low-pressure bladder-type accumulator*



1 Gas valve
2 Pressure vessel
3 Bladder
4 Fluid valve

Fig. 50: *High-flow high-pressure bladder-type accumulator*



1 Gas valve
2 Pressure vessel
3 Fluid valve

Fig. 51: *Hydro-pneumatic bladder-type accumulator, "top repairable" design*

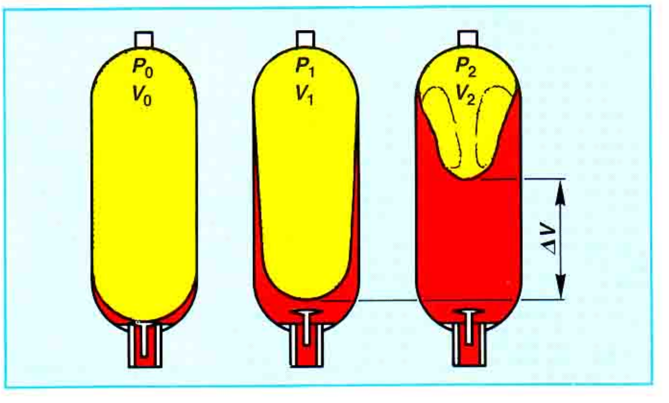


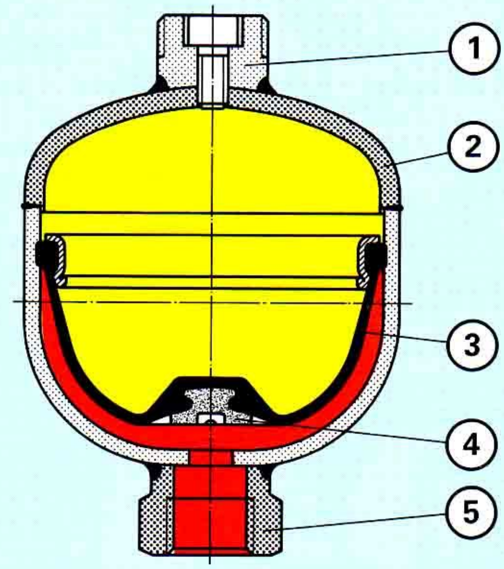
Fig. 52: *Schematic illustration of the operation of a bladder-type accumulator*

2.2 Diaphragm-type accumulator

The diaphragm-type accumulator shown in *Fig. 53* comprises a strong steel pressure vessel which is usually spherical or cylindrical. Inside the vessel is a separating element in the form of a diaphragm made of an elastic, malleable material such as an elastomer. For certain applications where high demands are made on the life of the elastometric material, as when corrosive fluids are present, it is an advantage to change the diaphragm at regular intervals. Consequently, there are two different designs:

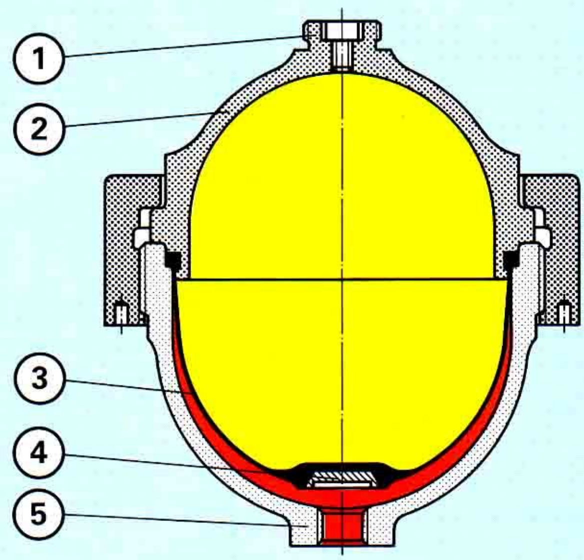
- welded *(see Fig. 53)* and
- threaded body *(see Fig. 54)*.

With the welded design the diaphragm is pressed into the bottom half of the vessel before the seam is welded. A suitable type of welding, such as electron beam, and the special diaphragm arrangement ensure that the elastometric material does not suffer damage during welding. In the case of the threaded body design the diaphragm is held between the top and bottom halves which are held together by a nut. In both designs there is a valve plate at the bottom to prevent the diaphragm being ejected through the fluid connection. There is a danger of this, when the accumulator is emptied completely. The principle of the diaphragm-type accumulator can be described best by referring to *Fig. 55*. At the start, the gas side of the diaphragm is connected to nitrogen at the appropriate charging pressure p_0. This causes the diaphragm to mould itself to the internal contour of the vessel and the valve plate seals off the fluid connection. In the same way as the bladder-type accumulator, the valve plate lifts when the minimum operating pressure p_1 is reached and hydraulic fluid can flow into the accumulator. The difference between the two gas volumes at minimum and maximum operating pressure represents the useful capacity for fluid. Although diaphragm-type accumulators can technically be mounted in any position the vertical is preferred.



1 Gas valve
2 Pressure vessel
3 Diaphragm
4 Valve plate
5 Fluid connection

Fig. 53: *Diaphragm-type accumulator, welded version*



1 Gas valve
2 Pressure vessel
3 Diaphragm
4 Valve plate
5 Fluid connection

Fig. 54: *Diaphragm-type accumulator, threaded body version*

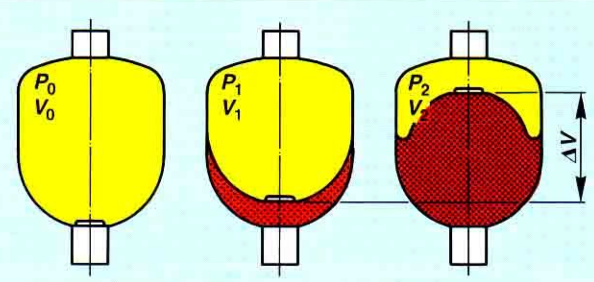
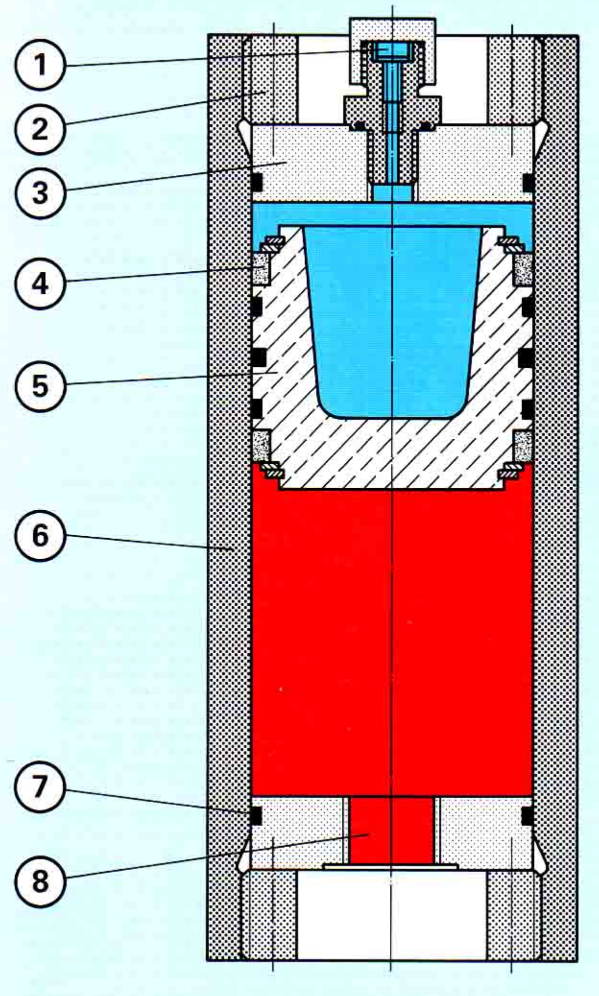


Fig. 55: *Mode of operation of a diaphragm-type accumulator*

2.3 Piston-type accumulator

The construction of a typical piston-type accumulator is illustrated in *Fig. 56*. Its principal components are the outer cylinder tube, the piston with its sealing system and the end covers which also contain the fluid and gas connections. The cylinder performs two functions - it stores the fluid pressure and also guides the piston which forms the separating element between the gas and fluid parts. The mode of operation of this type of accumulator is as follows, referring to *Fig. 57*.



1 Gas valve
2 Threaded retainer ring
3 Cover
4 Seal system
5 Piston
6 Cylinder
7 External seal
8 Fluid connection

Fig. 56: *Piston-type accumulator*

Filling the gas space with nitrogen to the appropriate charging pressure forces the piston on to the end cover at the fluid end, so covering the fluid inlet connection. As the fluid pressure in the system rises and passes the minimum operating value, the piston is forced towards the gas end and compresses the gas in the cylinder. The compressed gas volume between V_1 and V_2 represents the useful volume available V. In order for there to be balanced pressures between the two pressure spaces it is essential for the friction between the piston seals and the inner wall of the cylinder to be very low as the piston moves. Therefore, the internal surface of the cylinder tube must have a very fine finish. Due to the unavoidable presence of some friction at this point, however, it is impossible for a difference in pressure between the gas and fluid spaces to be eliminated completely.

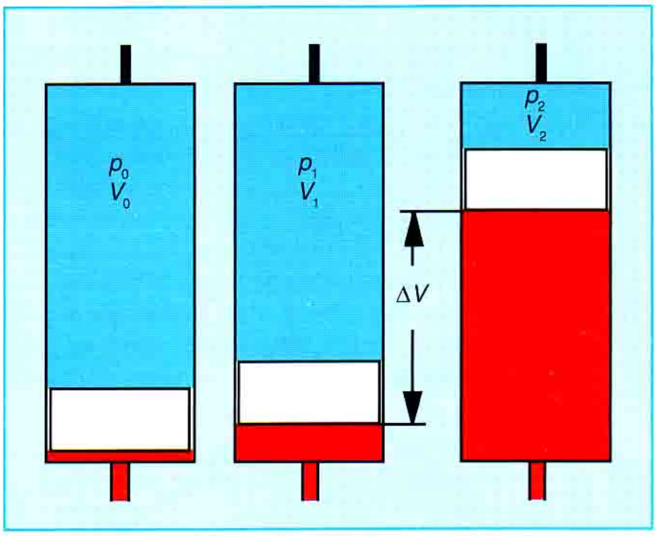


Fig. 57: *Mode of operation of a piston-type accumulator*

Diagram 18 clarifies this frictional effect. It shows graphs of the fluid and gas pressures against time for a single accumulator cycle with two different types of sealing system. It shows clearly that a low-friction sealing system causes less differential between the two pressures and so provides better operating characteristics. However, the frictional resistance is not constant; it increases with the operating pressure. At lower operating pressures the frictional resistance predominates over the piston motion so use of the accumulator at low pressure levels is not normally sensible.

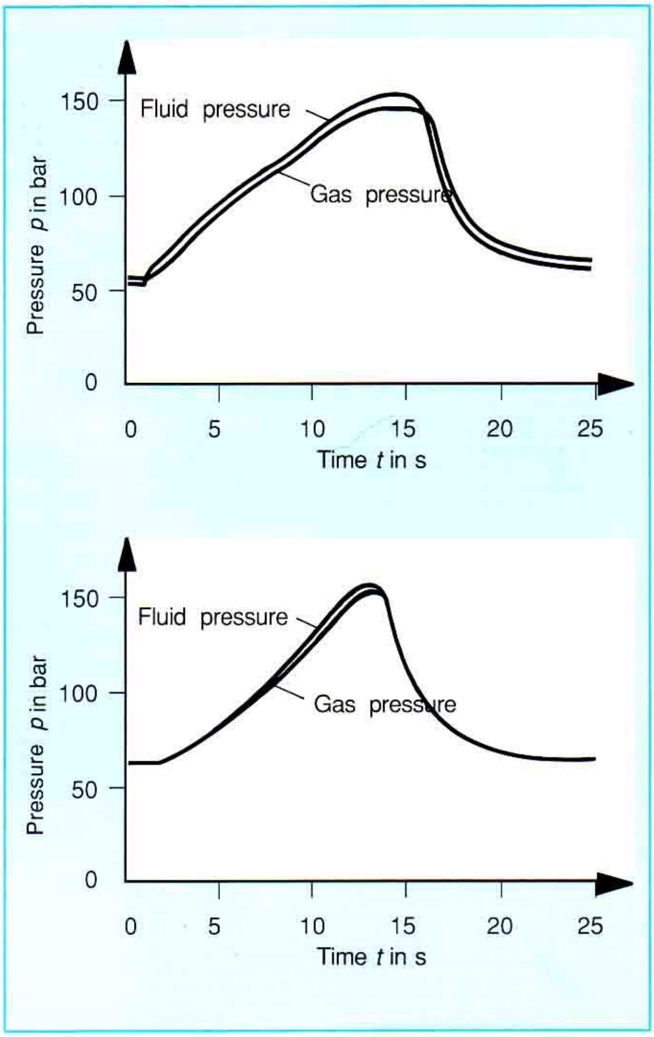
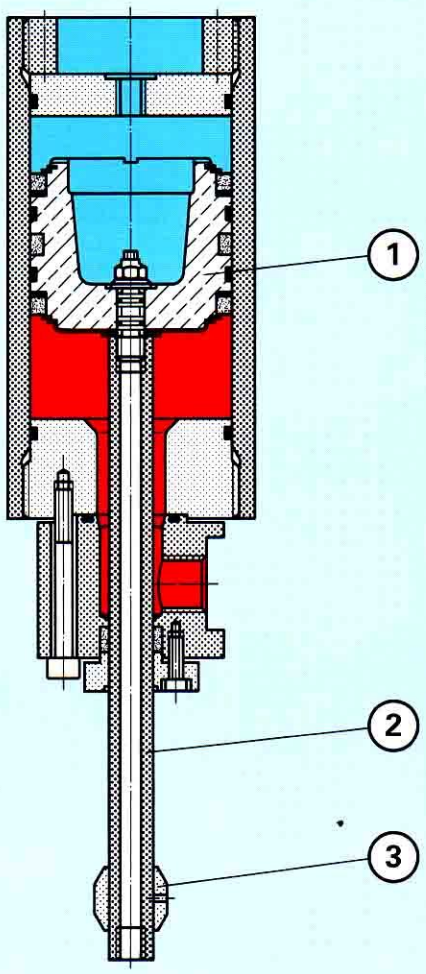


Diagram 18: *Graphs of pressure versus time for a piston-type accumulator with a normal sealing system (top) and a low-friction sealing system (bottom)*

Certain functions of the hydraulic system, such as stopping of the supply pump or monitoring of the charging of the accumulator, can be achieved directly by modifying the piston. As shown in Fig. 58 a piston rod is attached to the piston and extended through one of the end covers. This allows a variety of control functions to be achieved in a number of different ways, either:

- mechanically with switching cams or
- electrically with permanent magnets or inductive proximity switches.

Yet another method of determining the position of the piston is to use an ultrasonic measuring system which employs a microprocessor making direct use of data such as piston position and simultaneous measurement of gas pressure for the different control functions.

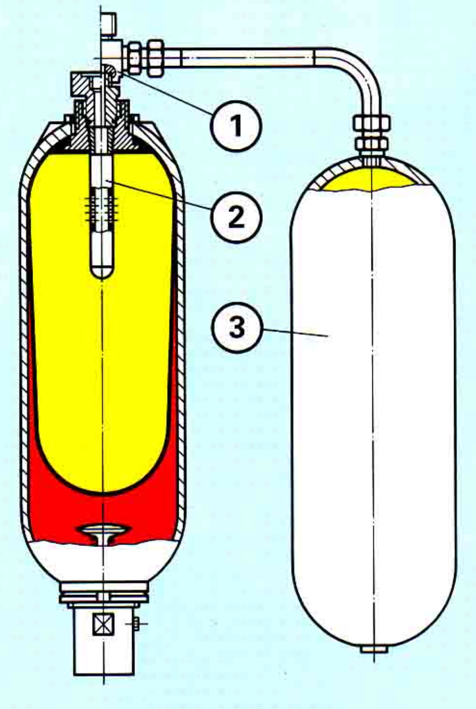


1 Piston
2 Piston rod
3 Switching cam

Fig. 58: *Piston-type accumulator with external piston rod*

2.4 Adding nitrogen bottles

For certain applications it can be an advantage to increase the gas volume by adding extra nitrogen bottles. One typical such application is when the difference between the minimum and maximum operating pressures is small. The volume of nitrogen in the accumulator is then only compressed slightly and the useful portion of the storage volume is insufficient for the purpose. Depending on the operating conditions the gas volume can be doubled by adding extra nitrogen bottles. *Fig. 59* shows such an arrangement with a bladder-type accumulator. The gas end of the accumulator has a modified connection for the nitrogen bottle and, so that the bladder does not suffer any damage when the accumulator is being charged, there is also an additional Crepin tube inside the bladder. The nitrogen volume can be increased in the same way with piston-type accumulators. In both cases, bladder-type and piston-type, the designer must match the maximum connected gas volume accurately to the operating temperature and ambient temperature.



1 Adapter
2 Crepin tube
3 Nitrogen bottle

Fig. 59: *Bladder-type accumulator with added nitrogen bottle*

2.5 Hydraulic dampers

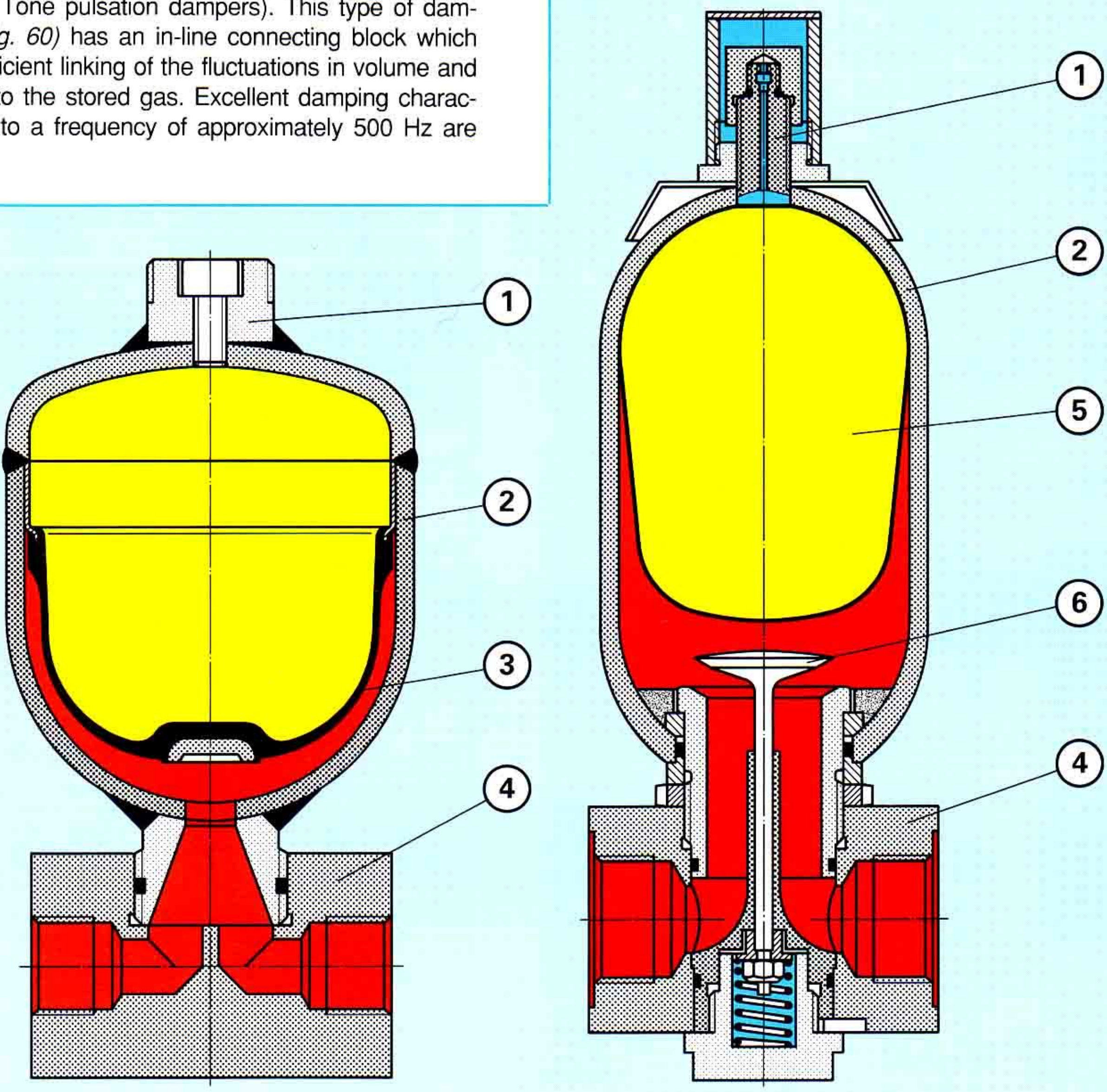
Fluctuations in pressure can occur in hydraulic systems as a result of changes in the flow of fluid due to a variety of circumstances in the system such as:

- irregularities in the displacement pump
- spring-mass systems (pressure compensators in valves)
- sudden linking of spaces at different pressures
- operation of fast opening and closing
- starting and stopping of displacement pumps.

Associated with these circumstances are fluctuations in volumetric flow and pressure which have a negative effect on the service life of all components in the system. Depending on the origin, fluctuations in pressure vary between pressure shocks and more regular pulsations. In order to ensure that they do not affect the functioning of the system it is necessary to ascertain the magnitude of the fluctuations at the planning stage and to incorporate suitable measures to provide damping. There are numerous ways of damping out pressure fluctuations but the hydraulic damper has shown itself to be particularly suitable for hydraulic systems. The basic specification for such dampers can be divided into physical, structural and operating sections. The physical parameters are primarily related to good damping characteristics over a large frequency range with a minimum pressure drop. The structural side involves as simple a construction as possible with easy installation and suitability for the relevant temperature, fluid and pressure. The operating aspects concentrate on minimal maintenance so that the operating reliability of the installation is not affected.

2.5.1 Construction and mode of operation

Depending on the mode of operation, hydraulic dampers employ the principle of hydro-pneumatic bladder-type and diaphragm-type accumulators or a fluid silencer. In the case of the hydro-pneumatic dampers the compressibility of a gas (usually nitrogen) is utilized for the damping. With a bladder-type accumulator, for example, the bladder is compressed or expanded according to the magnitude of the pressure fluctuations. The diaphragm-type accumulator behaves similarly. Since using normal bladder accumulators or diaphragm accumulators does not always produce good damping due to the imperfect link between the hydraulic fluid and the gas volume, special hydro-pneumatic dampers have been developed (e.g. Pulse Tone pulsation dampers). This type of damper *(see Fig. 60)* has an in-line connecting block which provides efficient linking of the fluctuations in volume and pressure into the stored gas. Excellent damping characteristics up to a frequency of approximately 500 Hz are achieved.



1 Gas valve
2 Pressure vessel
3 Diaphragm
4 "Pulse Tone" connector block
5 Bladder
6 Check valve

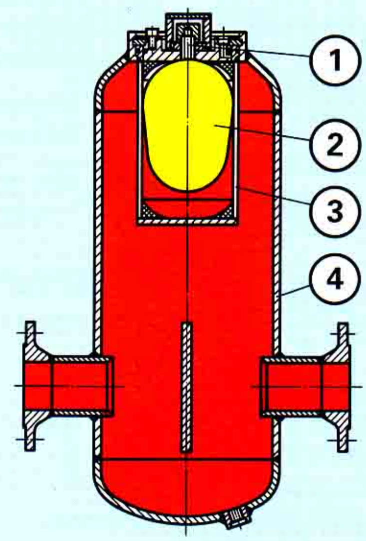
Fig. 60: *Hydro-pneumatic dampers, Pulse Tone diaphragm accumulator (left) Pulse Tone bladder accumulator (right)*

Another form of pulsation damping, specifically for reducing fluctuations in pressure on the suction side, can be effected with the suction stabilizer shown in *Fig. 61.* Basically, it comprises a small volume of gas surrounded by a volume of fluid many times larger. It performs the function of a storage vessel and considerably reduces the acceleration effects of the flow.

Special hydraulic dampers, called shock absorbers *(see Fig. 62)*, have been developed for damping out the pressure shocks that can be associated with the fast opening and closing of valves and the starting and stopping of pumps.

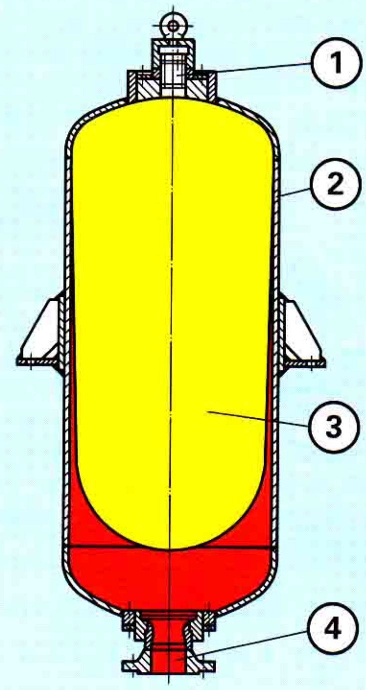
In construction, the shock absorber is like a bladder-type accumulator and stops further propagation of the pressure shock through the conversion of potential energy into kinetic energy and vice versa.

Damping can also be achieved without an additional volume of gas by means of a fluid silencer. The design of such a silencer is shown in *Fig. 63* and is totally different from the hydro-pneumatic dampers. In this case the fluctuations in volume and pressure are reduced by careful flow design employing such features as a resonator or an expansion chamber.



1 Gas valve
2 Bladder
3 Strainer basket
4 Pressure vessel

Fig. 61: *Suction stabilizer*



1 Gas valve
2 Vessel
3 Bladder
4 Fluid connection

Fig. 62: *Shock absorber*

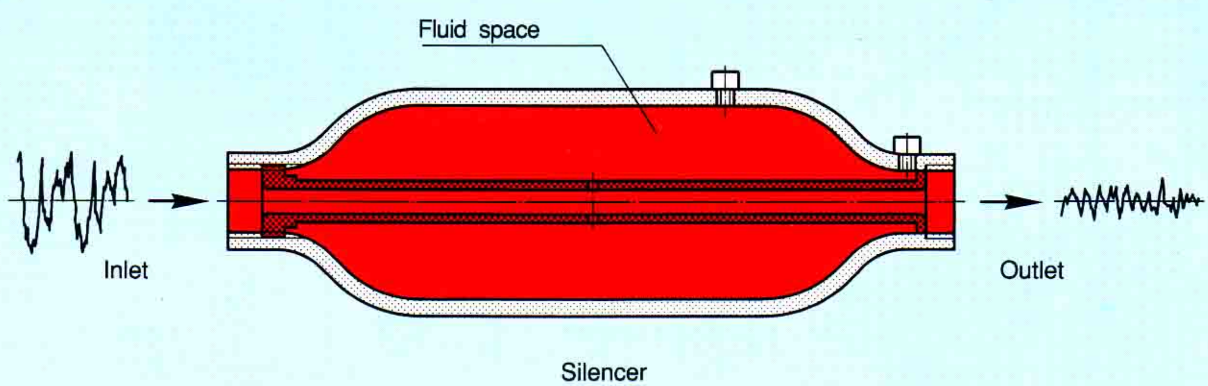


Fig. 63: *Fluid noise transmission damper(silencer)*

3 Design of hydro-pneumatic accumulators

The performance required of a hydro-pneumatic accumulator varies according to the particular application. In the design the initial factors of interest are only the requirements regarding useful volume and pressure energy. In addition there are certain secondary factors to be taken into account, mostly specific to the type of installation, e.g. mobile equipment needs a maximum ratio of energy capacity to weight. Once the size of accumulator necessary for the demands has been established, component details can be laid down, such as the quality of elastomer for the seals and separating elements.

3.1 Definitions of operating parameters

The parameters needed for designing a hydro-pneumatic accumulator are best clarified with a diagram of a piston-type accumulator *(see Fig. 64)*. Of course, the same relationships hold good for the other types of hydro-pneumatic accumulator. The parameters for describing the state of the gas - also called state variables - are pressure, temperature and volume. The following variables are defined for the various states which arise during operation of an accumulator

Pressures

p_0 Charging pressure of the gas space with the fluid space depressurized.

p_1 Minimum pressure required to open the valve. With bladder and diaphragm accumulators this pressure is normally about 10% higher than the charging pressure. The charging pressure can be made lower with piston accumulators.

p_2 Maximum operating pressure of the hydraulic system with bladder and diaphragm accumulators

p_0/p_2 Maximum permitted pressure ratio for operating conditions

Temperatures

T_i Gas temperature corresponding to the various states (i = 0, 1, 2). The temperature of the fluid affects the heat exchange with the compressed gas and so is only needed indirectly for designing the accumulator.

Volumes

V_0 Effective gas volume at charging pressure

V_1 Gas volume at minimum pressure

V_2 Gas volume at maximum operating pressure

ΔV Useful volume

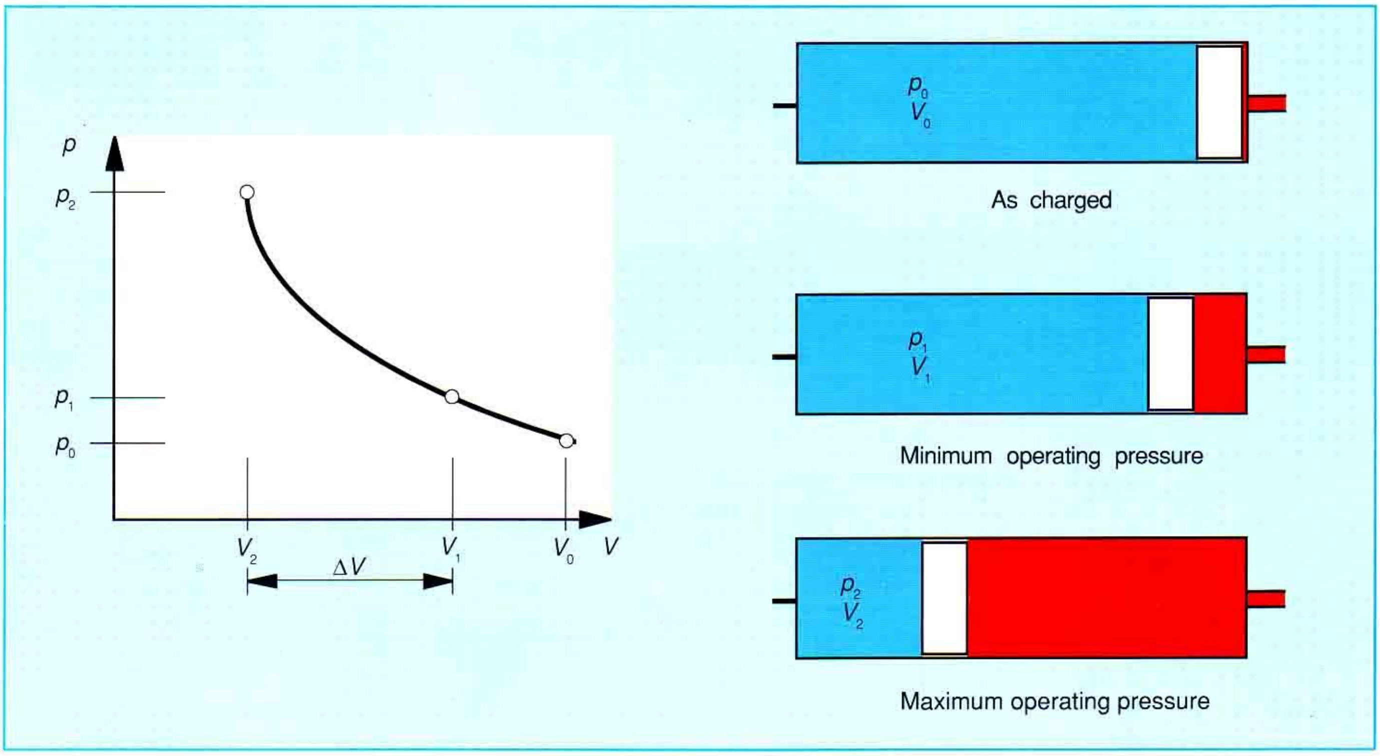


Fig. 64: *Diagram of operating state showing variables applicable to a piston-type accumulator*

3.2 The fundamental physics

In the thermodynamic sense the volume of gas in a hydraulic accumulator can be regarded as a homogeneous closed system with appropriate state variables. Although the basic physical equations for accumulator design will be explained by referring to a piston-type accumulator it does not restrict the universal nature of the equations in any way (the friction between piston and cylinder wall is neglected in this case). The flow of hydraulic fluid into or out of the accumulator has a direct relation to the change of state of the volume of gas inside. Firstly, the hydraulic fluid causes an exchange of work with the gas and, secondly, an exchange of heat between the surroundings and the gas if the gas temperature is not equal to the ambient temperature. The "surroundings" mean the separating element, the accumulator vessel and the hydraulic fluid.

In *Fig. 65*, moving the piston an infinitely small distance *ds* to change the volume by an amount *dV* requires the following amount of work

$$dW_V = -p \cdot A \cdot ds = -p \cdot dV \qquad (1)$$

The change in volume also involves a change in state of the gas.

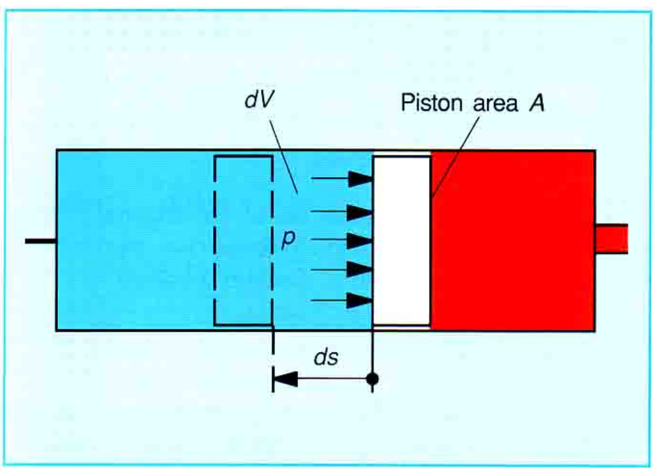


Fig. 65: *Volumetric relationship*

Note on Equation 1:
There is a recognized mathematical sign rule to establish whether the system is absorbing work (+) or expending work (-). According to this rule, the work is considered to be positive with compression ($dV<0$).

Assuming that it is an ideal gas, the relation between pressure, temperature and volume can be described with the equation of state

$$p \cdot V = m \cdot R \cdot T \qquad (2)$$

Where R is a constant dependent only on the type of gas. For nitrogen (N_2) it is:

$$R = 297 \frac{J}{kg\,K}$$

In order to continue, it is important to know about the individual processes which take place in the accumulator in order to understand the change in state of the gas. The states and their changes are as follows:

a) Charging of the gas space at low temperature with subsequent change in the charging pressure through heat exchange with the surroundings.

b) The charging or discharging cycle of the accumulator by the hydraulic fluid takes place over a time span sufficient for a complete exchange of heat with the surroundings to occur.

c) The charging or discharging cycle takes place so rapidly that no exchange of heat with the surroundings is possible.

In the change of state described in **a)** no work of volume change is expended, i.e. no change in volume takes place. This change of state is called isochore and can be described by the following simplified equation

$$\frac{p}{T} = \frac{p_1}{T_1} = \text{const.} \qquad (3)$$

The change of state described in **b)** is called isothermal and takes place without change in temperature if complete exchange of heat with the surroundings is assumed. The mathematical relation between the state variables can be derived from the thermal equation of state and, for an isothermal change, is

$$p \cdot V = p_1 \cdot V_1 = \text{const.} \qquad (4)$$

The change of state described in **c)** is called adiabatic. In this case there is only an exchange of work between the hydraulic fluid and the gas and the relevant equation is

$$p \cdot V^{\kappa} = p_1 \cdot V_1^{\kappa} = \text{const.} \qquad (5)$$

The relation between temperature, volume and pressure can also be obtained from the thermal equation of state

$$T \cdot V^{\kappa -1} = T_1 \cdot V_1^{\kappa -1} \quad \text{and} \qquad (6)$$

$$T \cdot p^{(1-\kappa)/\kappa} = T_1 \cdot p_1^{(1-\kappa)/\kappa} \qquad (7)$$

In these equations κ represents the adiabatic index which can be taken as 1.4 for a diatomic gas such as nitrogen under normal conditions *(see Diagram 19).*

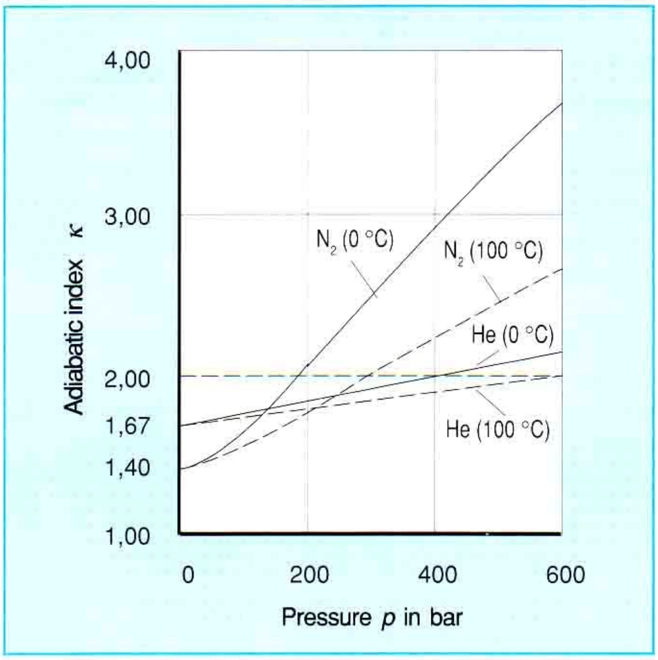


Diagram 19: *Adiabatic index of nitrogen and helium in relation to pressure at 0 and 100°C*

Since an accumulator never functions precisely according to theory with no exchange of heat, the change of state will lie somewhere between the isothermal and the adiabatic. This type of state change is called polytropic. The mathematical relationships are similar to those for the adiabatic change of state but with the adiabatic index replaced by the polytropic index *n*. The *p-V* diagram *(Diagram 20)* shows the different changes in state and it is obvious that the isothermal and adiabatic are extremes of the polytropic.

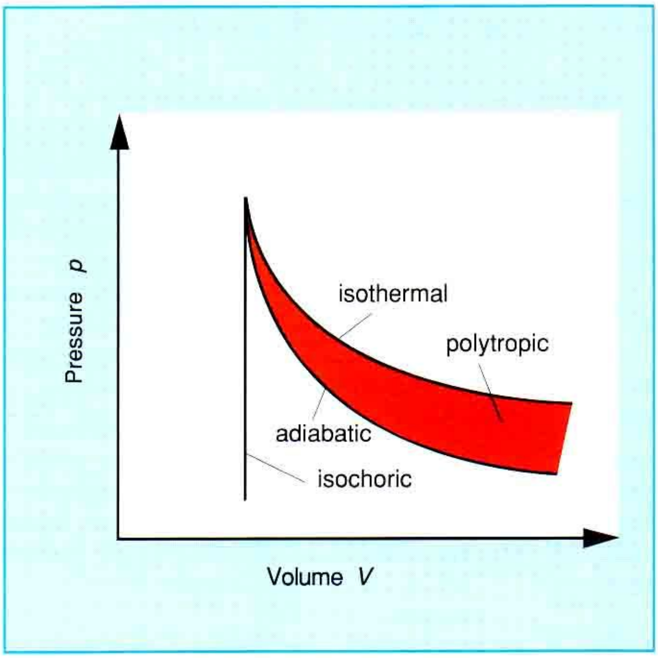


Diagram 20: *p-V diagram of change in state*

Accordingly, the equations to be used for designing a hydraulic accumulator depend on the time effect of the charging or discharging process. The following limits can serve as a useful rule-of-thumb for using the appropriate equation:

Cycle time < 1 minute
→ adiabatic change in state

Cycle time > 3 minutes
→ isothermal change in state

Cycle time between 1 and 3 minutes
→ polytropic change in state.

In order to be more precise about the change of state taking place it is necessary to know the thermal time constant that is dealt with in *Section 3.2.2.*

For design purposes it is an advantage to rearrange the equations so that the required variables can be calculated. Primarily these are the effective gas volume (V_o) corresponding to the pressure conditions and the charging pressure p_o. *Table 15* lists the basic equations for accumulator design.

Also when designing an accumulator, there are certain empirical values to be adhered to which ensure, firstly, optimum utilization of the accumulator volume and, secondly, long service life. *Table 16* lists the relevant empirical values for different types of accumulator.

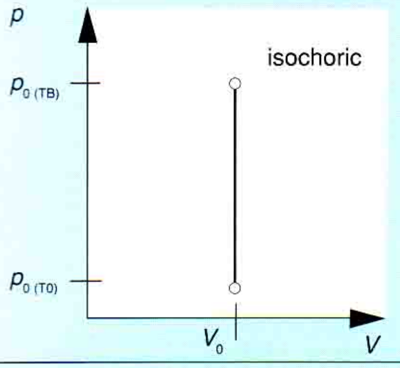
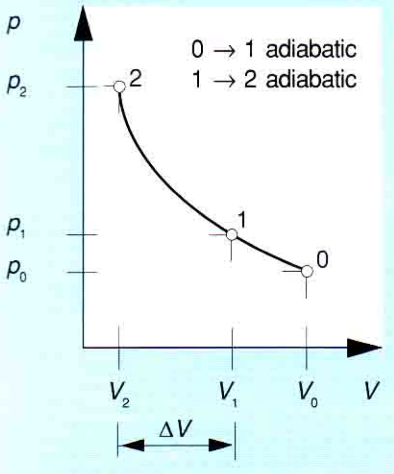
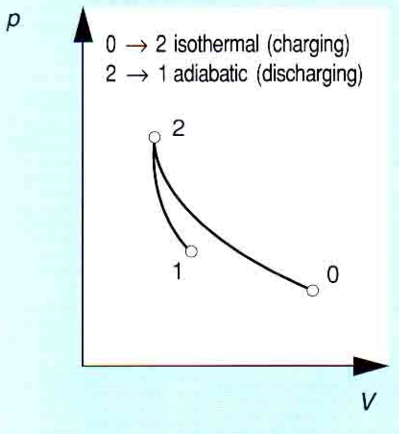
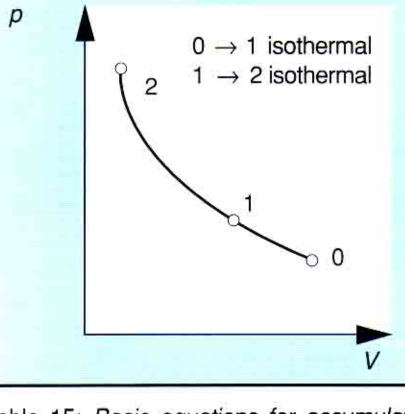
Cycle (change of state)	Equation	Remarks
	$p_{0(T0)} = p_{0(TB)} \cdot \frac{T_0}{T_B}$	$p_{0\,(T0)}$ = Charging pressure at charging temperature T_0 $p_{0\,(TB)}$ = Charging pressure at operating temperature T_B **Application:** Calculation of the charging pressure when the operating temperature is different from the charging temperature
	$\Delta V = V_0\left[\left(\frac{p_0}{p_1}\right)^{\frac{1}{n}} - \left(\frac{p_0}{p_2}\right)^{\frac{1}{n}}\right]$	$n = \kappa = 1{,}4$ for nitrogen **Application:** Energy storage
	$V_0 = \dfrac{\Delta V}{\left(\frac{p_0}{p_1}\right)^{\frac{1}{n}} - \left(\frac{p_0}{p_2}\right)^{\frac{1}{n}}}$	
	$\Delta V = V_0 \frac{p_0}{p_2}\left[\left(\frac{p_2}{p_1}\right)^{n} - 1\right]$	**Application:** Emergency functions, safety functions
	$V_0 = \dfrac{\Delta V \cdot \frac{p_2}{p_0}}{\left(\frac{p_2}{p_1}\right)^{\frac{1}{n}} - 1}$	
	$\Delta V = V_0\left(\frac{p_0}{p_1} - \frac{p_0}{p_2}\right)$	**Application:** Leakage fluid make-up
	$V_0 = \dfrac{\Delta V}{\frac{p_0}{p_1} - \frac{p_0}{p_2}}$	

Table 15: *Basic equations for accumulator design*

Condition	Bladder accumulators	Diaphragm accumulators	Piston accumulators
Gas charging pressure p_0	$\leq 0{,}9 \cdot p_1$ $= 0{,}6$ to $0{,}9 \cdot p_m$ (shock absorption) $= 0{,}6 \cdot p_m$ (pulsation damping)	$\leq 0{,}9 \cdot p_1$	$\leq p_1 - 5$ bar ≥ 2 bar (low-frictionpiston) ≥ 10 bar (Normal piston)
Max. permitted pressure ratio p_2/p_0	$\leq 4:1$	$\leq$ 6:1 to 8:1 (welded version) $\leq$ 10:1 (screwed version)	No restrictions
Max. fluid flow	to 15 L/s according to size to 140 L/s at High-Flow version	to 6 L/s	Max. piston velocity = 3,5 m/s (low-friction version) = 2 m/s (normal version)

Table 16: *Conditions of application for hydro-pneumatic accumulators*

In the case of the arrangement with added nitrogen bottles the designer must also examine the useful volume of the accumulator. He should begin with an isothermal change from charging pressure to maximum operating pressure. The increased useful volume $\Delta V'$ can be calculated from:

$$\Delta V' = V_{0G}\left(1 - \frac{p_0}{p_2}\right). \qquad (8)$$

For bladder accumulators with added nitrogen bottles $\Delta V'' = 0.75 \cdot V_{OG}$ should not be exceeded because of excessive distortion of the bladder. V_{OG} is the total effective gas volume (accumulator plus nitrogen bottles). The increased useful fluid volume $\Delta V'$ must always be less than the effective gas volume of the accumulator. The value of gas volume must be chosen so that these conditions are fulfilled.

3.2.1 Deviations from the ideal gas

The equations of state described in the previous section are only applicable to an ideal gas. Practical gases such as nitrogen *(see Diagram 21)*, however, do not follow the ideal gas laws, particularly at higher pressures. This behaviour is called "real" or "imperfect". The mathematical relation between the state variables (p, T and V) for a real gas can only be given by an approximate equation. The use of such an equation with sufficient accuracy is very tiresome in practice and requires a large amount of computing time which can only be provided by a mainframe computer. For this reason it is advisable to introduce correction factors that allow for the behaviour of the real gas.

Consequently, the volume with an isothermal change of state becomes

$$V_{real} = C_i \cdot V_{ideal} \qquad (9)$$

and with an adiabatic change of state

$$V_{real} = C_a \cdot V_{ideal}\,. \qquad (10)$$

The correction factors C_i and C_a in *Equations (9) and (10)* can be taken directly from *Diagrams 22 and 23* according to the pressure ratio p_2/p_1 and the maximum operating pressure.

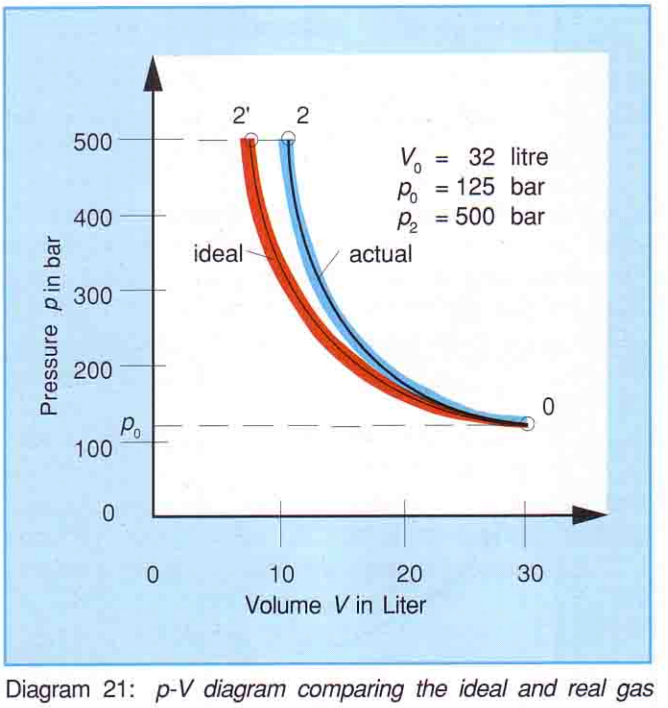


Diagram 21: *p-V diagram comparing the ideal and real gas behaviour of nitrogen in compression*

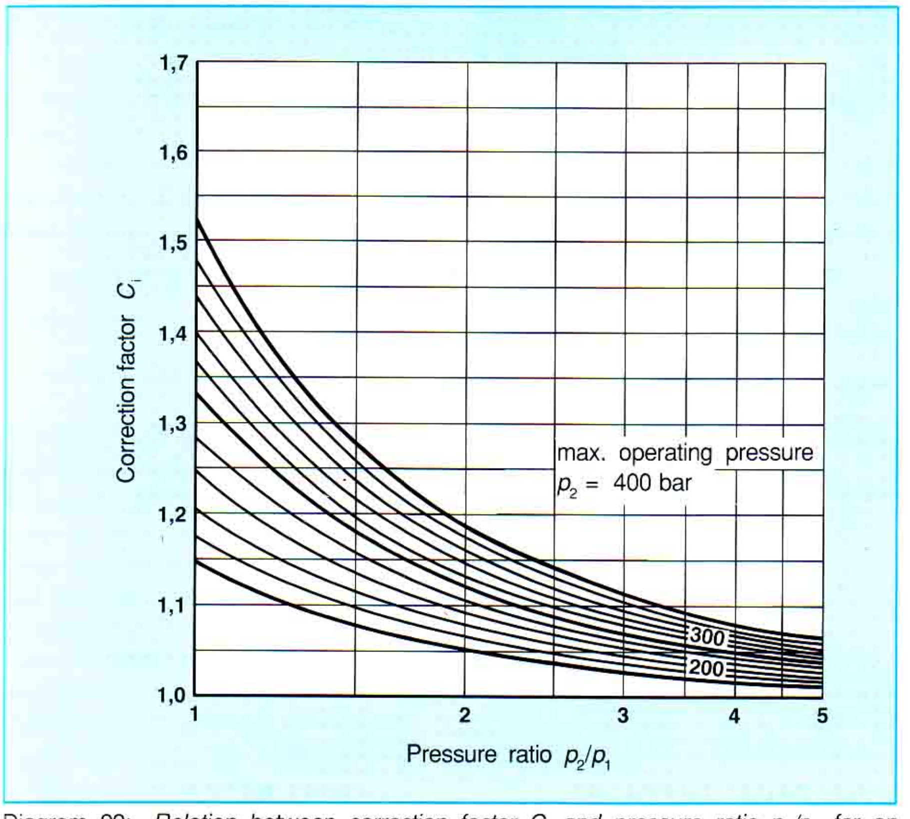


Diagram 22: *Relation between correction factor C_i and pressure ratio p_2/p_1 for an isothermal change of state*

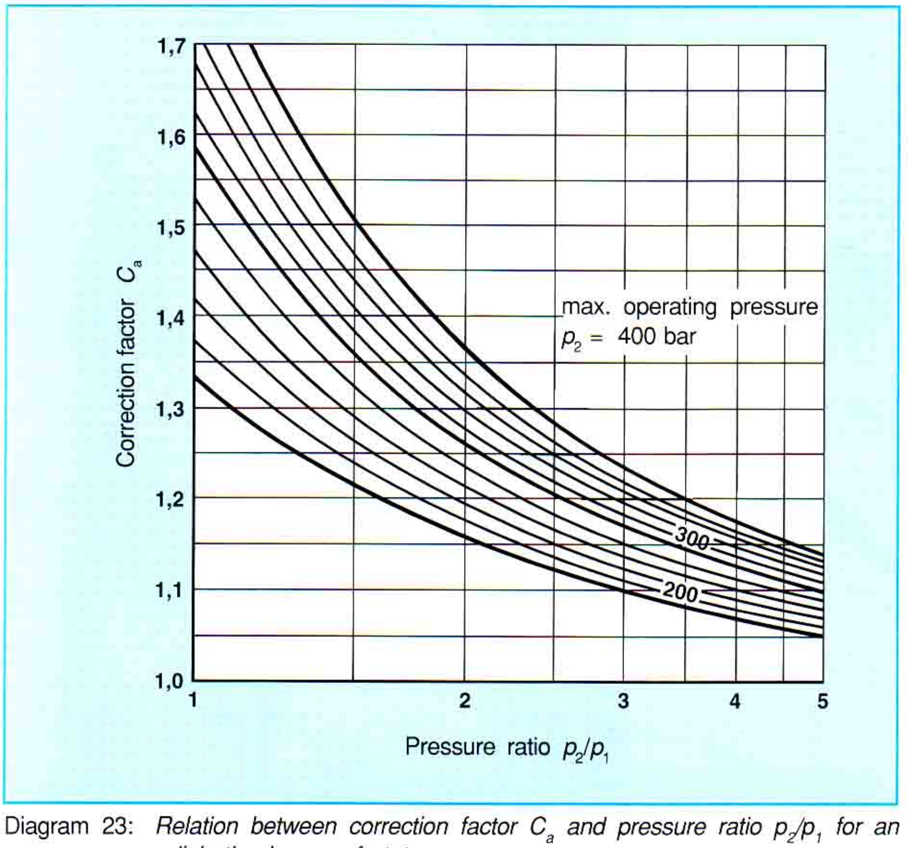


Diagram 23: *Relation between correction factor C_a and pressure ratio p_2/p_1 for an adiabatic change of state*

The correction factors are referred to a temperature of 50°C. Any deviations resulting from changes in temperature can be neglected in the permitted temperature range (-10 to +80°C).

3.2.2 Thermal time relationships

Approximate time limits were quoted in *Section 3.2* in order to establish the type of change of state. In order to arrive at a more accurate design for an accumulator it will be necessary to analyze the processes of thermodynamic exchange. In the case of intermittent operation, in particular when there is a rapid sequence of changes, the processes are determined from the intensity of the heat transfer. Describing and analyzing the thermal time response of hydraulic accumulators makes use of the thermal time constant

$$\tau = \frac{c_V \cdot m}{\alpha \cdot A} \qquad (11)$$

Where c_v is the specific thermal capacity at constant volume, m the mass of the gas, α the heat transfer coefficient and A the total area of heat transfer.

The time constant can be determined very easily by experiment. Since it depends on the charging pressure and the type and size of accumulator, it must be determined by test for each different type of accumulator. *Diagrams 24, 25 and 26* show the results of tests according to |1| for the different types. The thermal time constants are plotted in relation to the charging pressure for various nominal volumes of the different sizes of accumulator. Using these time constants and a suitable simulation program an accumulator can be designed for a given cycle of operation.

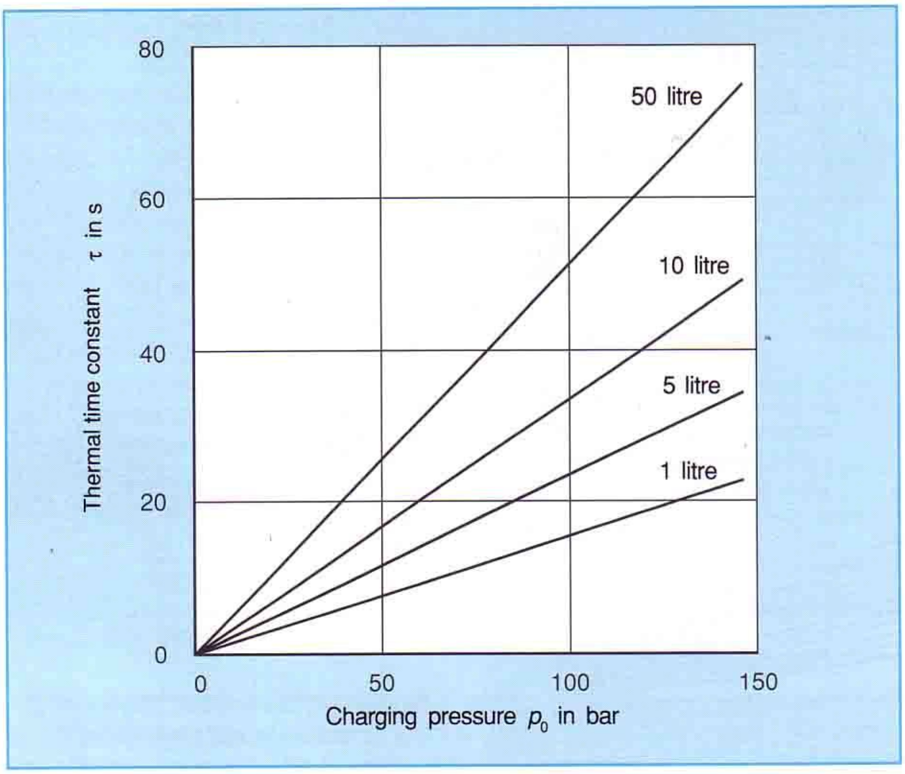


Diagram 24:
Thermal time constant for bladder-type accumulators

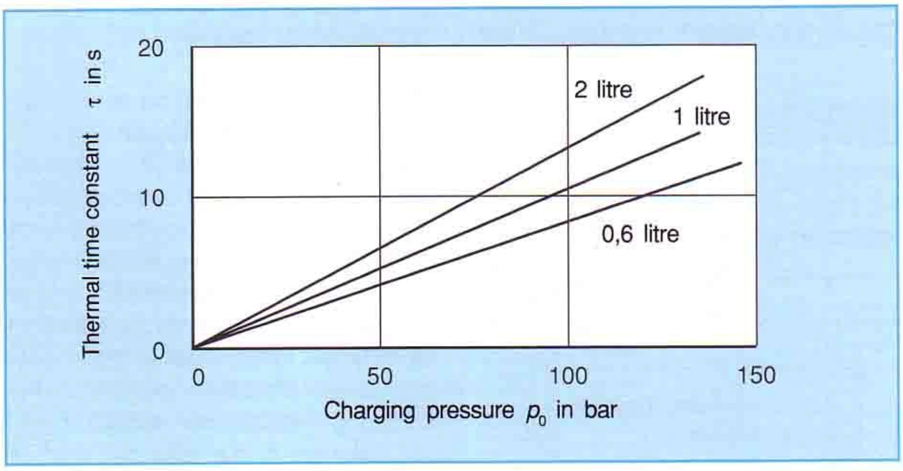


Diagram 25:
Thermal time constant for diaphragm-type accumulators

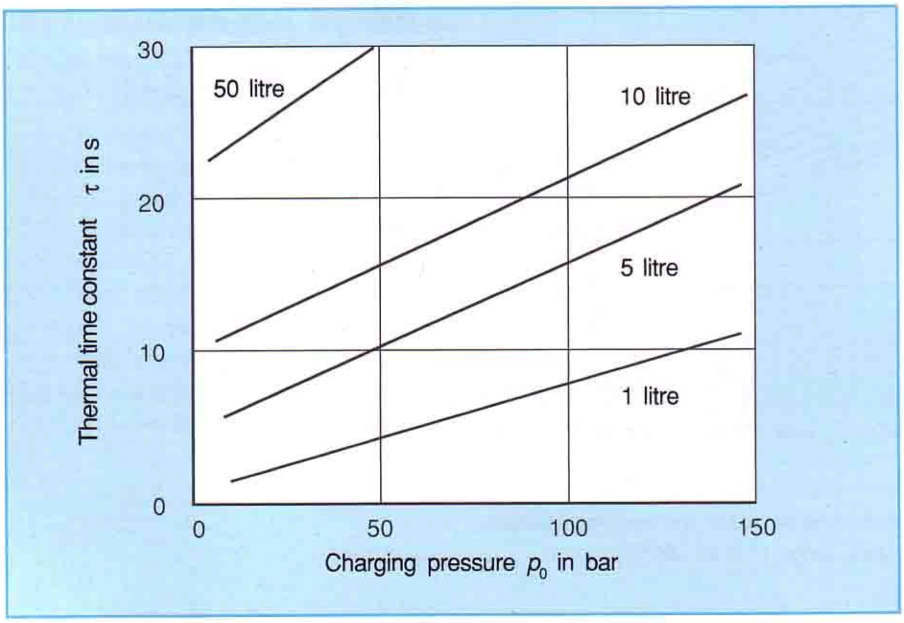


Diagram 26:
Thermal time constant for piston-type accumulators

3.3 The design procedure

In calculating and establishing the appropriate size of accumulator it can be assumed that the necessary volume of fluid V or the necessary energy W is available for the demand. Taking account of certain peripheral factors, such as

- maximum operating pressure
- maximum and minimum operating temperature
- working pressure differential

The first step in the design procedure is to assume initially that the change of state between the working pressure p_1 and p_2 is adiabatic. This limiting assumption is permissible since it means that the other possible changes of state are always fulfilled. The design can then be corrected through a subsequent check of the calculations in terms of the time response and the associated deviation from the assumed adiabatic change of state. Since the changes of state of the gas are related to the operating temperature, the capacity must be sufficient for each state. This gives rise to various restrictions on the design which are examined as follows:

– Restriction 1a and b

The required volume of fluid V_{req} or the required energy W_{req} must still be available from the accumulator at a maximum attainable operating temperature.

– Restriction 2

At the minimum operating temperature the permitted value of working pressure difference $p_{permiss}$ must not be exceeded. The various restrictions can be explained by referring to a diagram of a polytropic change of state *(see Diagram 27).*

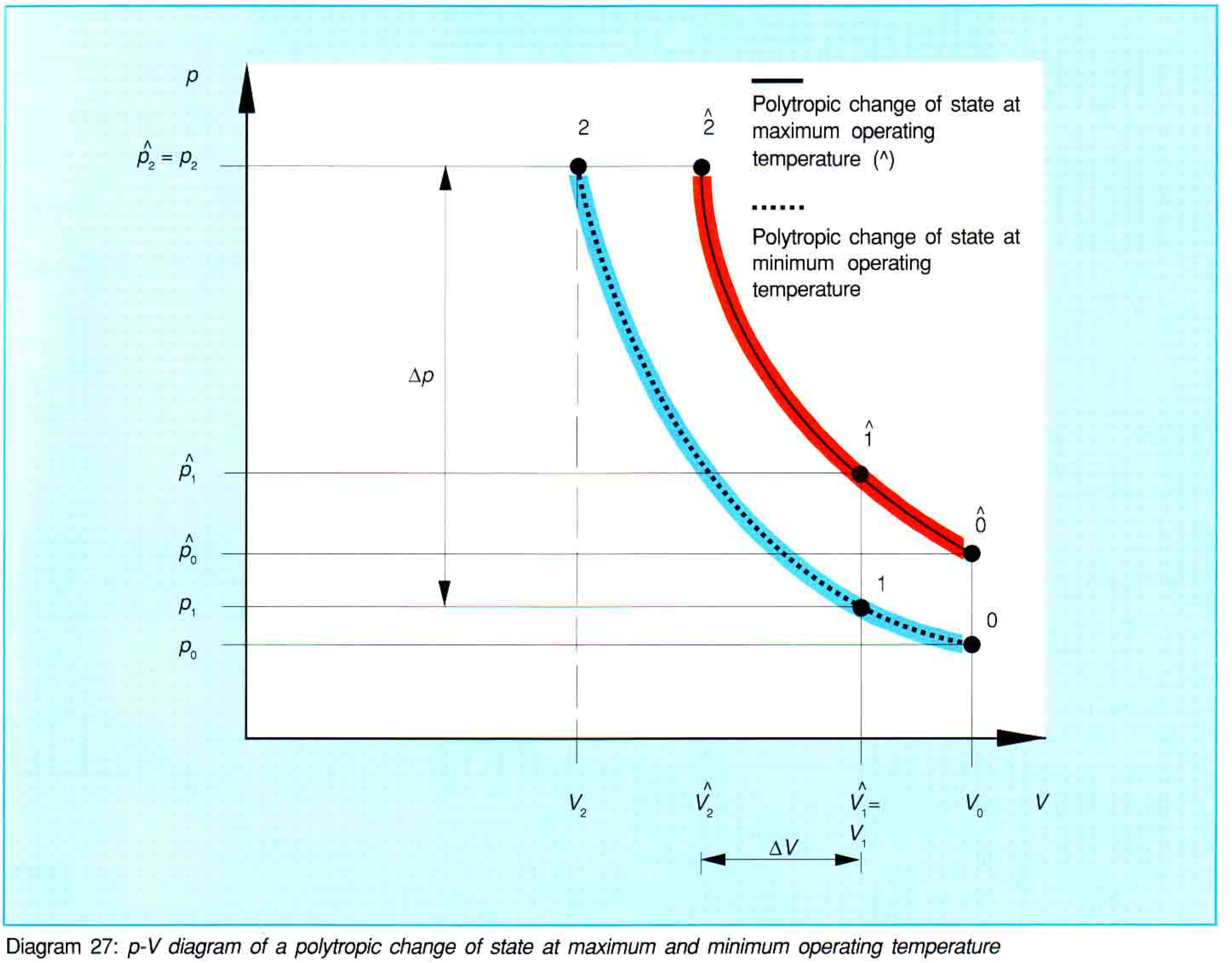


Diagram 27: *p-V diagram of a polytropic change of state at maximum and minimum operating temperature*

– **Restriction 1a**

The volume of fluid is calculated from:

$$\Delta V = \hat{V}_1 - \hat{V}_2 \geq \Delta V_{erf}. \quad (12)$$

The changes of state are:

Filling:

isochoric $$\frac{p_0}{T_0} = \frac{\hat{p}_0}{T_{max}}, \quad (13)$$

Charging to minimum operating pressure:

isothermal $$\hat{p}_1 \cdot \hat{V}_1 = \hat{p}_0 \cdot V_0, \quad (14)$$

Compression to maximum operating pressure

adiabatic $$\hat{p}_2 \cdot \hat{V}_2^{\kappa} = \hat{p}_1 \cdot \hat{V}_1^{\kappa}. \quad (15)$$

Substituting the three Equations (13), (14) and (15) in Equation (12) gives

$$\Delta V = \frac{\hat{p}_0}{\hat{p}_1} V_0 \left(1 - \left(\frac{\frac{T_{max}}{T_0} p_0}{\frac{\hat{p}_0}{\hat{p}_1} \hat{p}_2}\right)^{\frac{1}{\kappa}}\right) \geq \Delta V_{req} \quad (16)$$

or

$$\frac{\Delta V_{erf}}{V_0} \leq \frac{\hat{p}_0}{\hat{p}_1} \left(1 - \left(\frac{\frac{T_{max}}{T_0}}{\frac{\hat{p}_0}{\hat{p}_1}} \cdot \frac{p_0}{\hat{p}_2}\right)^{\frac{1}{\kappa}}\right). \quad (17)$$

Equation (17) describes the restricted relationship for the required volume of fluid at the maximum attainable operating temperature T_{max}.

– **Restriction 1b**

The energy stored by the accumulator must be equal to or greater than the required energy W_{req} at the maximum operating temperature. Work is done when the gas is compressed from point $\hat{1}$ to point $\hat{2}$. The associated change in internal energy is then

$$W_{\hat{1}\hat{2}} = -\int_{\hat{1}}^{\hat{2}} p \, dV \geq W_{12 \; req} \quad (18)$$

With the equation for an adiabatic change of state, Equation (18) becomes

$$W_{\hat{1}\hat{2}} = \frac{\hat{p}_1 \cdot \hat{V}_1}{\kappa - 1} \left(\left(\frac{\hat{p}_1}{\hat{p}_2}\right)^{\frac{1-\kappa}{\kappa}} - 1\right). \quad (19)$$

The restricted relationship for the energy at maximum operating temperature is obtained by substituting Equations (13) and (14) in Equations (18) and (19)

$$\frac{W_{\hat{1}\hat{2}erf}}{\hat{p}_2 \cdot V_0} \leq \frac{p_0 \cdot T_{max}}{\hat{p}_2 \cdot T_0 (\kappa - 1)} \left(\left(\frac{\frac{T_{max}}{T_0}}{\frac{\hat{p}_0}{\hat{p}_1}} \cdot \frac{p_0}{\hat{p}_2}\right)^{\frac{1-\kappa}{\kappa}} - 1\right). \quad (20)$$

– **Restriction 2**

The difference in pressure at minimum operating temperature between operating states 1 and 2 is:

$$\Delta p = p_2 - p_1 \leq \Delta p_{permit}. \quad (21)$$

With equations

$$\frac{p_0}{T_0} = \frac{p_0}{T_{min}} \quad (22)$$

and $$p_1 \cdot V_1 = p_0 \cdot V_0 \quad (23)$$

it is possible to rearrange Equation (21) to give the following relationship for pressure difference

$$\Delta p = p_2 - \frac{\frac{T_{min}}{T_0}}{\frac{p_0}{p_1}} \cdot p_0 \quad (24)$$

Thus, the second restriction relationship for the pressure ratio p_0/p_2, which is applicable to both fluid volume and energy at minimum operating temperature, can be given as:

$$\frac{p_0}{p_2} \geq \frac{\frac{p_1}{p_0}}{\frac{T_{min}}{T_0}} \left(1 - \frac{\Delta p_{zul}}{p_2}\right) \quad (25)$$

Graphical representations of the restriction relationships of Equations (17), (20) and (25) are given in *Diagrams 28 and 29*. They show clearly the range of validity within which a design is permitted under the given conditions. The point of intersection of the restriction curves characterizes the optimum design. However, this is not always attainable in practice because of the steps in vessel sizes, and therefore gas volumes. Nevertheless, economic efficiency demands that every attempt be made to get the design as close as possible to this point.

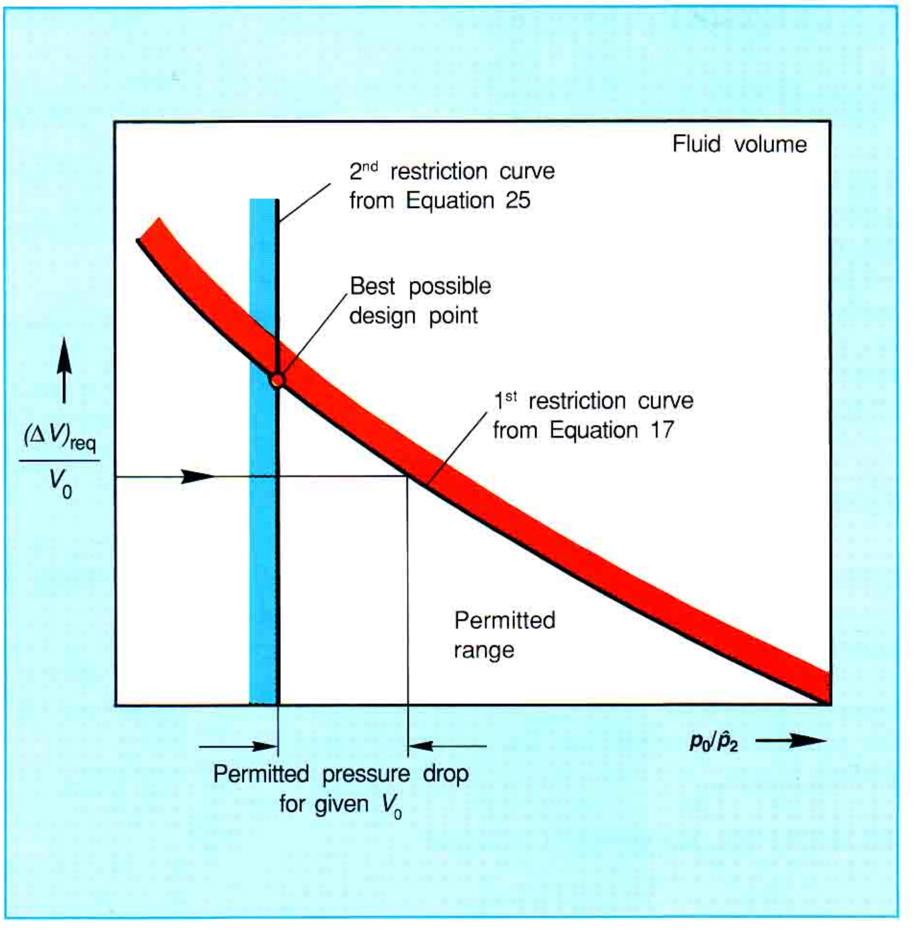


Diagram 28:
Graphic representation of the restriction relationship for fluid volume

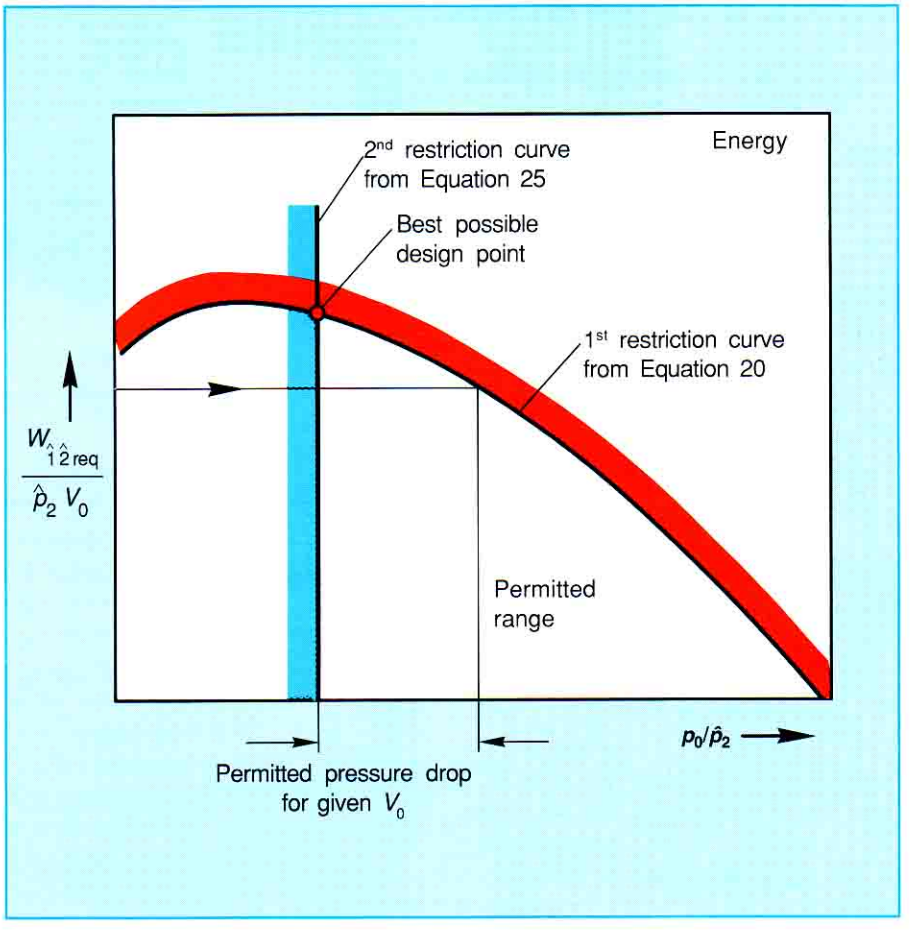


Diagram 29:
Graphic representation of the restriction relationship for energy

4 Typical Calculations

Example 1

In an injection moulding machine 5 litres of fluid are to be made available in 2.5 s. The maximum operating pressure is 200 bar and the minimum working pressure must not fall below 100 bar. The charging time is 8 s and the operating temperature is given as 45 °C. Calculate the size of accumulator required and the gas charging pressure at 22 °C using a real gas. Finally, check the results using the restriction relationships.

Solution

As it is a high-speed process (the outflow time is less than 1 minute) the change in state of the gas can be regarded as adiabatic.

Note:
The pressures substituted in the equations must be absolute values.

Calculating the gas charging pressure

$$p_0 = 0{,}9 \cdot p_1 = 0{,}9 \cdot 101 \approx 91 \text{ bar}$$

Calculating the volume of gas required

Assuming that nitrogen will be used as the gas:

$$V_{0\,ideal} = \frac{\Delta V}{\left(\frac{p_0}{p_1}\right)^{0{,}714} - \left(\frac{p_0}{p_2}\right)^{0{,}714}} = \frac{5}{0{,}9^{0{,}714} - \left(\frac{91}{201}\right)^{0{,}714}} = 13{,}87 \text{ L}$$

Calculating the correction factor from Diagram 23

$$C_a = 1.16 \text{ where } \frac{p_2}{p_1} \approx 2{,}0$$

$$V_{0\,real} = C_a \cdot V_{0\,ideal}$$

$$= 1.16 \cdot 13.87 = 16.09 \text{ L}$$

A 20 Litre bladder-type accumulator with an effective gas volume of 17.4 L is selected.

Calculating the gas charging pressure at 20°C

Note:
The temperatures used in the equations must be in Kelvin.

$$p_{0(T0)} = p_{0(TB)} \cdot \frac{T_0}{T_B}$$

$$p_{0(20°C)} = 91 \cdot \frac{20 + 273}{45 + 273} = 83{,}8 \text{ bar}$$

So that a gas charging pressure of 91 bar is available at an operating temperature of 45°C, the accumulator may only be charged to 83.8 bar at 20°C .

Checking the results with the restriction relationships for fluid volume

In *Diagram 30* the restriction curve to Equation (17) is drawn with the governing variables for the example. Also plotted are the volume ratios

$$\frac{\Delta V_{req}}{V_0}$$

for three bladder accumulators of sizes 10 L , 20 L and 32 L (effective gas volumes of 9 L , 17.4 L and 32.5 L).

The 2nd restriction curve to Equation (25) provides a value of 0.452 for the given variables.

From *Diagram 30* it can be seen that the smallest accumulator of 10 L nominal volume does not intersect the design area and so does not satisfy the requirements. The 20 L accumulator is at the optimum point of the design area so it is the correct choice. Although the 32 L accumulator also satisfies the requirements it is oversized.

Example 2

In a hydraulic system ,various cylinders are operated by directional control valves. The installation is for emergency operation and is to be driven by stored energy. The accumulator also has to make up the leakage losses at the directional control valves. For this, a low-capacity pump is to be started every 5 minutes. The pressure switches keep the pressure between 180 bar and 200 bar. When emergency operation is initiated, 8 L of fluid is needed to maintain certain functions and the pressure must not fall below 110 bar. The system incorporates 5 directional control valves with a leakage fluid loss of 30 cm^3/min each and 2 directional control valves with a leakage fluid loss of 140 cm^3/min each. Calculate the required size of accumulator and the required charging pressure.

Solution

Calculating the total volumetric flow of leakage fluid

$$\dot{Q}_L = 5 \cdot 30\ \text{cm}^3/\text{min} + 2 \cdot 140\text{cm}^3/\text{min} = 430\ \text{cm}^3/\text{min}.$$

The useful volume of the accumulator required to make up the leakage fluid is therefore

$$\Delta V = \dot{Q}_L \cdot t = 430\ \text{cm}^3/\text{min} \cdot 5\ \text{min} = 2{,}15\ \text{L}.$$

The charging pressure can be calculated from

$$p_0 = 0{,}9 \cdot p_1 = 0{,}9 \cdot 111\ \text{bar} = 100\ \text{bar}.$$

Calculating the gas volume in order to make-up leakage fluid

As it is a slow process (the delivery time is more than 3 minutes) it can be assumed to be an isothermal change of state

$$V_0 = \frac{\Delta V}{\frac{p_0}{p_1} - \frac{p_0}{p_2}} = \frac{2{,}15}{\frac{100}{181} - \frac{100}{201}} = 39{,}1\ \text{L}$$

Calculating the gas volume for emergency operation

In this case there is a slow accumulator charging process (isothermal) and a fast discharging process (adiabatic).

Since there is the possibility during emergency operation of the accumulator being charged to a pressure of only 181 bar, this must also be taken as the maximum pressure for the design.

$$V_0 = \frac{\Delta V \cdot \frac{p_2}{p_0}}{\left(\frac{p_2}{p_1}\right)^{\frac{1}{n}} - 1} = \frac{8 \cdot \frac{181}{100}}{\left(\frac{181}{111}\right)^{\frac{1}{1{,}4}} - 1} = 34{,}6\ \text{L}$$

In selecting the accumulator the larger gas volume is the governing factor and a 50 L accumulator with an effective gas volume of 47.5 L is chosen.

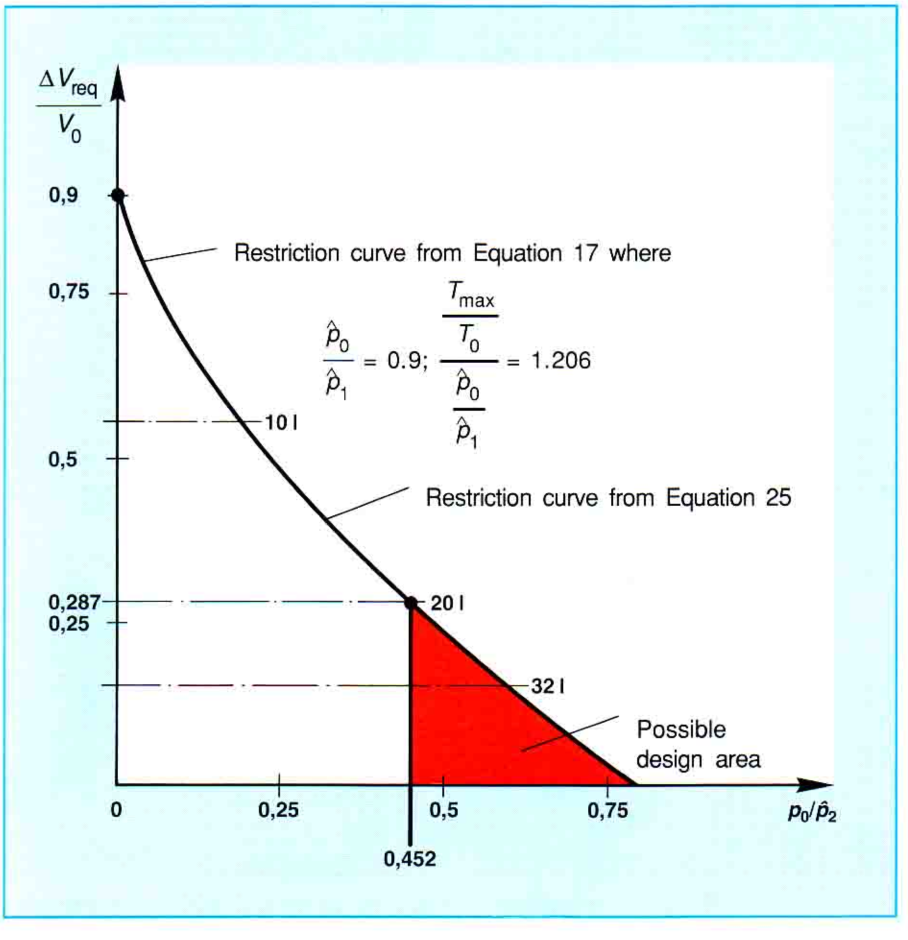


Diagram 30:
Graph of the restriction relationship for Example 1

Example 3

Between a maximum operating pressure of 180 bar and a minimum working pressure of 120 bar, a piston-type accumulator is to supply 35 L of fluid in 2 seconds. The accumulator must be refilled in 4 minutes.

Solution

Calculating the gas charging pressure

$$p_0 = p_1 - 5\text{ bar} = 121\text{ bar} - 5\text{ bar} = 116\text{ bar}$$

Calculating the gas volume required

The calculation is performed for an adiabatic change of state (the delivery time is less than 1 minute).

$$V_0 = \frac{\Delta V}{\left(\frac{p_0}{p_1}\right)^{0{,}714} - \left(\frac{p_0}{p_2}\right)^{0{,}714}} = \frac{35}{\left(\frac{116}{121}\right)^{0{,}714} - \left(\frac{116}{181}\right)^{0{,}714}} = 144{,}3\text{ L}$$

This volume of gas can be provided by a piston-type accumulator with an effective gas volume of 150 L . For economic reasons, however, it is advisable to employ an accumulator with a smaller gas volume and to add separate nitrogen bottles in an arrangement such as one 50 L piston-type accumulator (with an effective gas volume of 52.5 l) and two 50 L nitrogen bottles.

Checking the useful storage volume

According to Equation (8) it is necessary for the enlarged useful volume V' to be less than the effective gas volume V_0 of the accumulator. The total effective volume for the arrangement chosen is

$$V_{0G} = 1 \cdot 52{,}5\text{ L} + 2 \cdot 50\text{ L} = 152{,}5\text{ L}.$$

Therefore:

$$\Delta V' = V_{0G}\left(1 - \frac{p_0}{p_2}\right) = 152{,}5\left(1 - \frac{116}{181}\right) = 54{,}8\text{ L}.$$

Thus $\Delta V' > V_0$.

This shows that the volume of the accumulator selected is too small. When another calculation is performed for an arrangement with one 60 L accumulator (with an effective gas volume of 62.5 l) and two 50 L nitrogen bottles the total effective volume becomes

$$V_{0G} = 1 \cdot 62{,}5\text{ L} + 2 \cdot 50\text{ L} = 162{,}5\text{ L}$$

and the increased useful volume

$$\Delta V' = 162{,}5\left(1 - \frac{116}{181}\right) = 58{,}4\text{ L}.$$

This arrangement satisfies the required operating conditions.

5 Typical applications

5.1 Energy storage

From the pattern of power demand for a plastics injection moulding machine, shown in *Diagram 31*, it can be seen that maximum power is only needed for a short time during the high injection velocity into the mould. For economic reasons it would not be sensible to cover this peak demand entirely with pump power. However, it is sensible to rate the pump for a medium power demand and to cover the shortfall with an accumulator. The arrangement of the accumulator for this application is shown in *Fig. 66*.

Benefits

Smaller hydraulic pump, lower power, less heat, simpler maintenance and installation, lower operating costs. In addition there is the extra shock and pulsation damping which allows a longer service life to be anticipated.

Typical applications

Bladder-type and piston-type accumulators for energy storage in injection-moulding and blow-moulding machines, transfer lines, steelworks machinery, rolling mills, construction machinery, machine tools, hydraulic presses and shears, transport systems, marine engineering and power stations, trip-out systems on steam turbines and nuclear power plants.
Diaphragm-type accumulators are used for energy storage in pilot control circuits, braking systems, machine tools and jig and tool making.

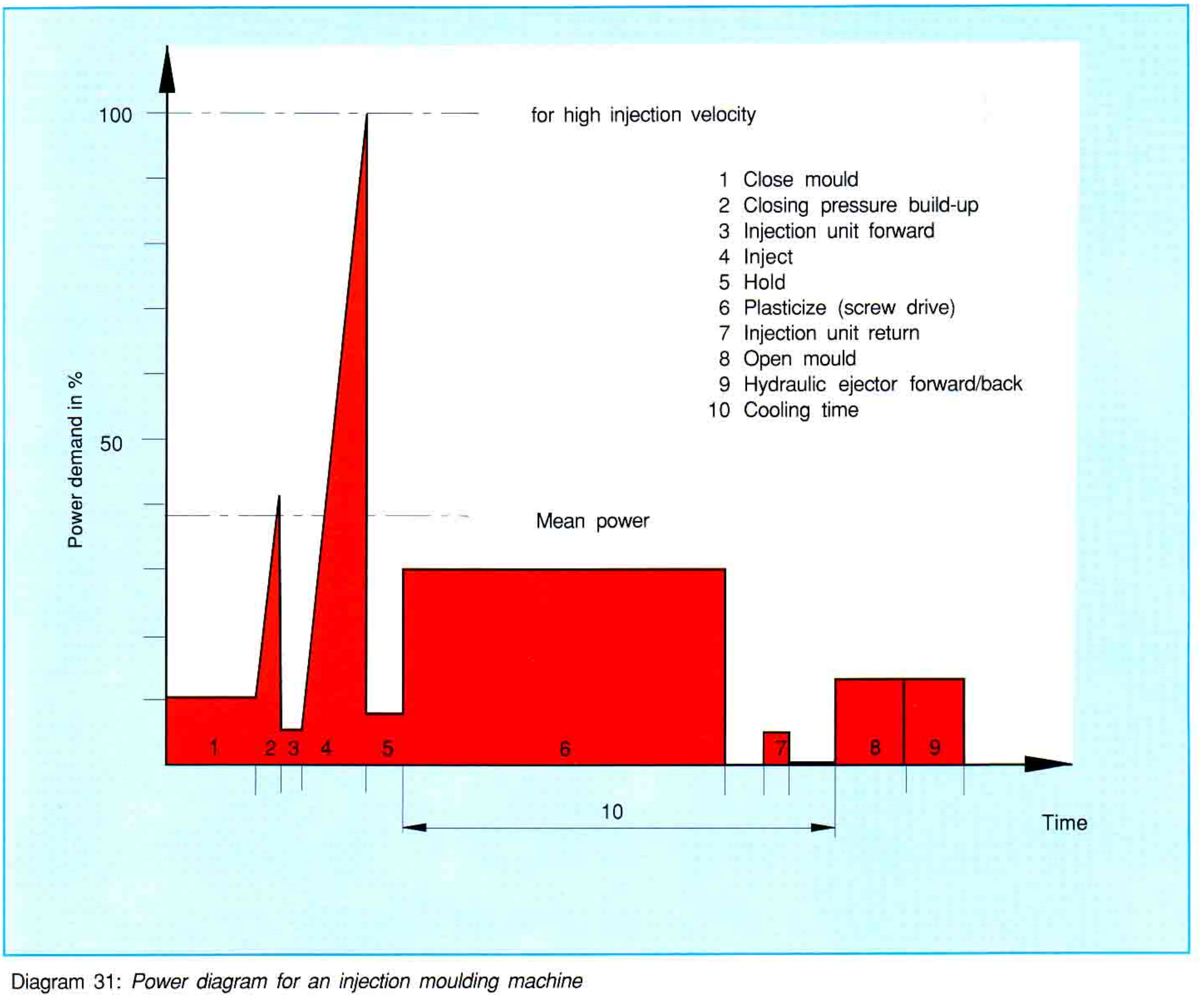


Diagram 31: *Power diagram for an injection moulding machine*

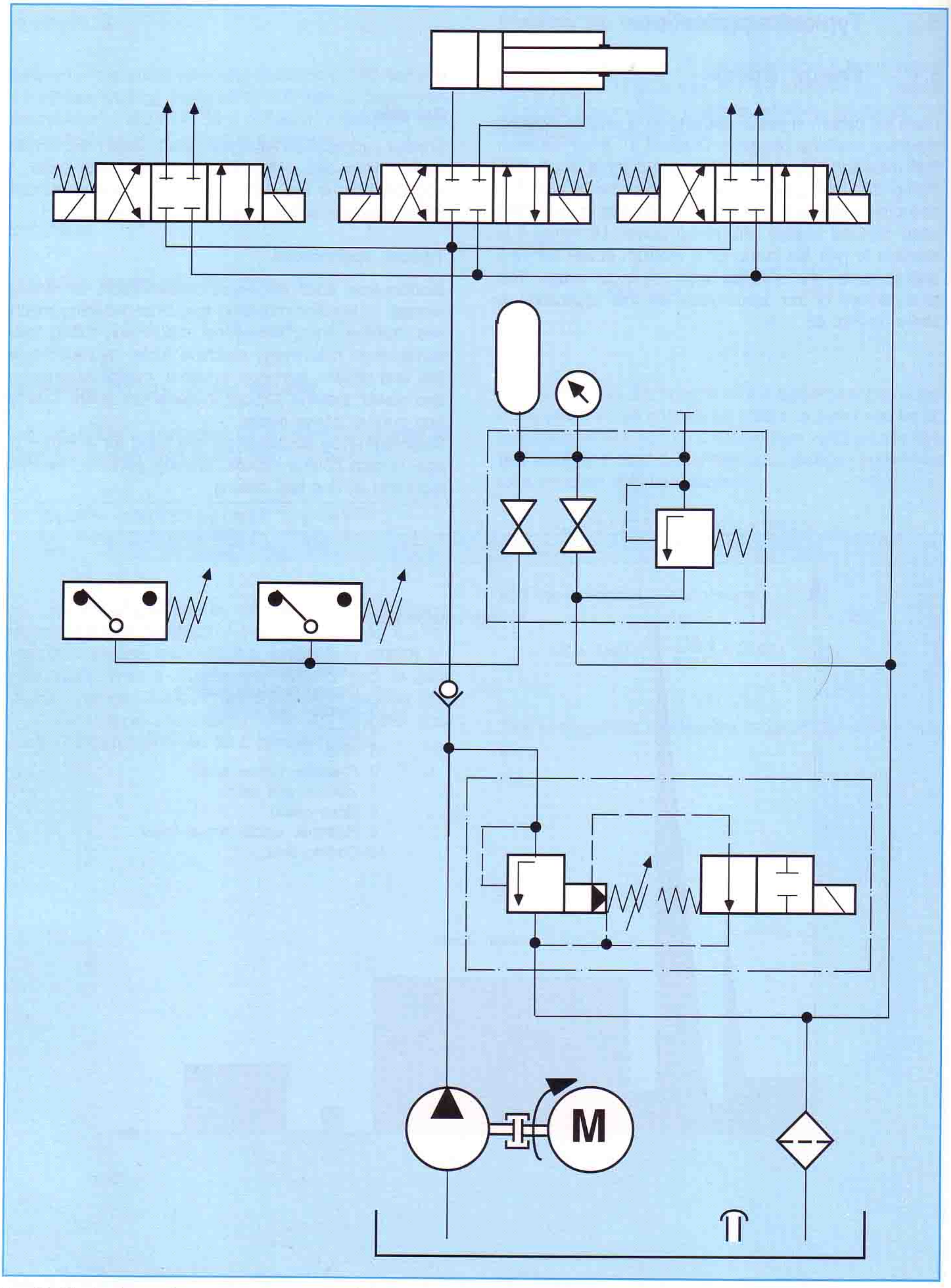


Fig. 66 *Circuit diagram for energy storage on an injection moulding machine*

5.2 Emergency operation

In an emergency such as a power failure, it is possible to use stored energy to operate a device such as a cylinder for a working stroke or closing stroke. *Fig. 67* shows a circuit diagram for emergency operation with a solenoid valve which, when triggered by the emergency, opens and allows the fluid stored under pressure in the accumulator to flow to the piston rod end of the cylinder causing it to retract.

Benefits

The energy stored is available immediately, it can be stored indefinitely, there is no fatigue or inertia, safety is at a maximum and maintenance at a minimum.

Typical applications

Bladder-type and diaphragm-type accumulators for closing bulkhead doors, dampers, sluices, bunker valves, silos and transport devices in the event of a power failure.

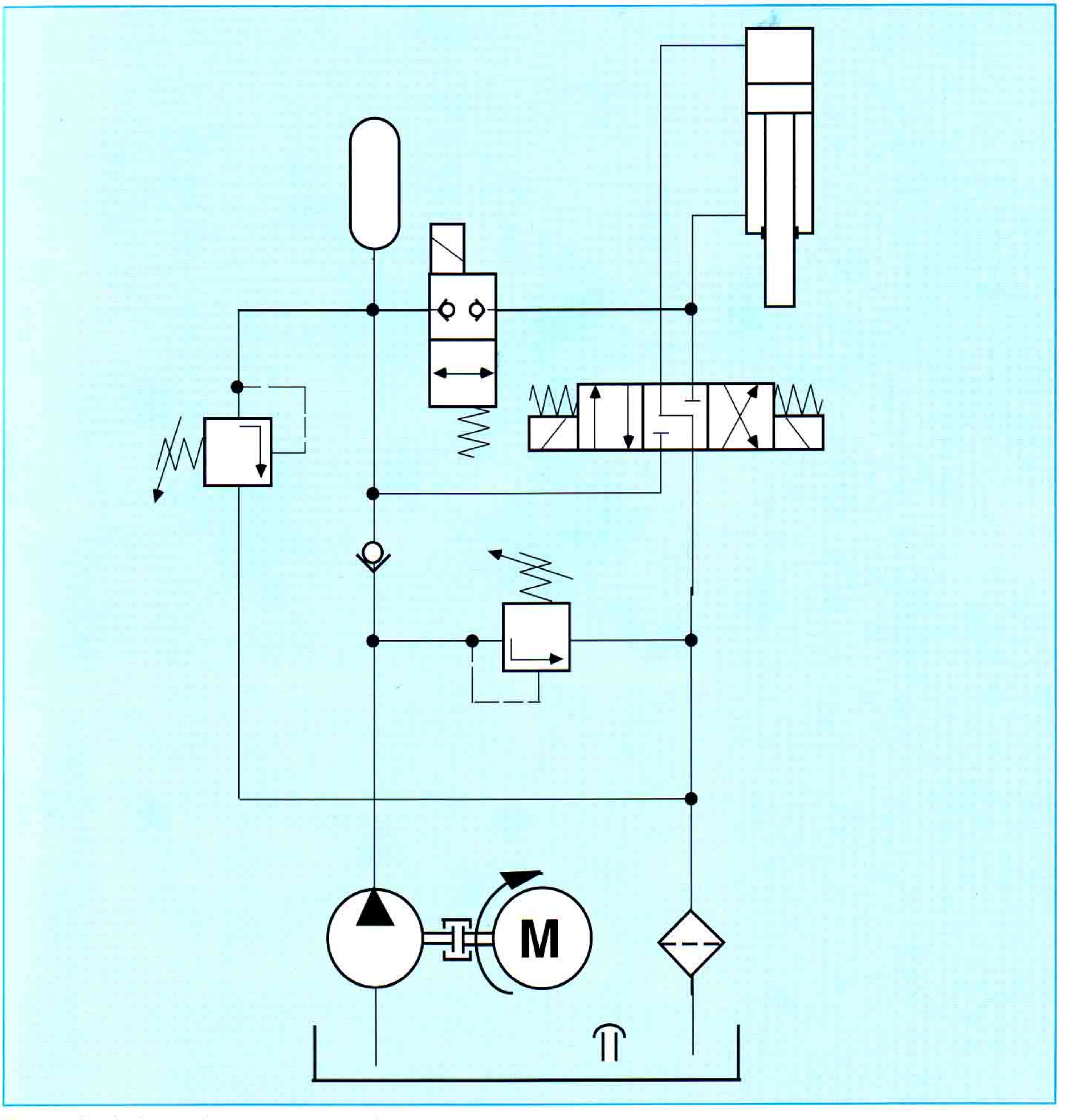


Fig. 67: *Circuit diagram for emergency operation*

5.3 Force balancing

Hydraulic accumulators can be used for force or stroke balancing. This can be necessary, for example, in a continuous production process such as steel rolling when uneven loads from the material being rolled could cause the rolls to adopt an off-centre position. Balancing the rolls produces a uniform wall thickness. *Fig. 68* shows a circuit diagram for a roll balancing system incorporating an accumulator and a direct-mounted safety and shut-off block.

Benefits

Ensures uniform forming, smooth force balancing and therefore less load on foundations and roll stand, elimination of counterweights and therefore a reduction in total weight.

Typical applications

Bladder-type, diaphragm-type and piston-type accumulators for machine tools, roll stands and crane jibs.

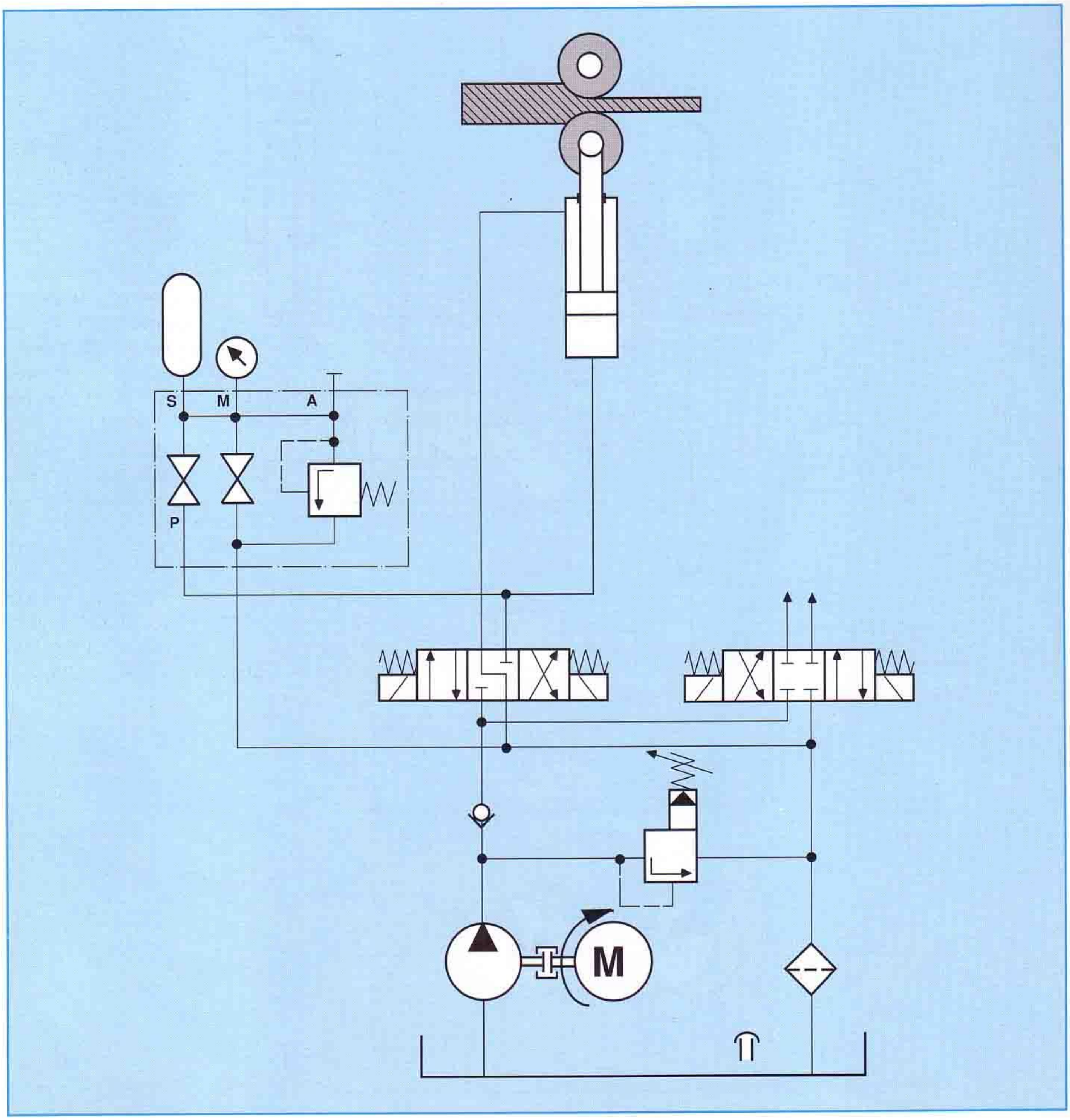


Fig. 68: *Circuit diagram for a roll balancing system*

5.4 Leakage fluid make-up

The pre-tensioning force exerted by a hydraulic cylinder can only be maintained if the leakage fluid losses in the system are made up. An accumulator is ideal for this task and Fig. 69 shows how it is used. It can be seen that the leakage fluid make-up from the accumulator is discharged into the cylinder below the piston and the pump is only started again when the pressure falls below a preset value.

Benefits

The pump does not run continuously, there is less heat wastage and service life is longer.

Typical applications

Bladder-type and diaphragm-type accumulators for leakage fluid make-up in jig and tool making, presses, lifting platforms, clamps and fixtures for machine tools, conveyor belts, roll stands, etc.

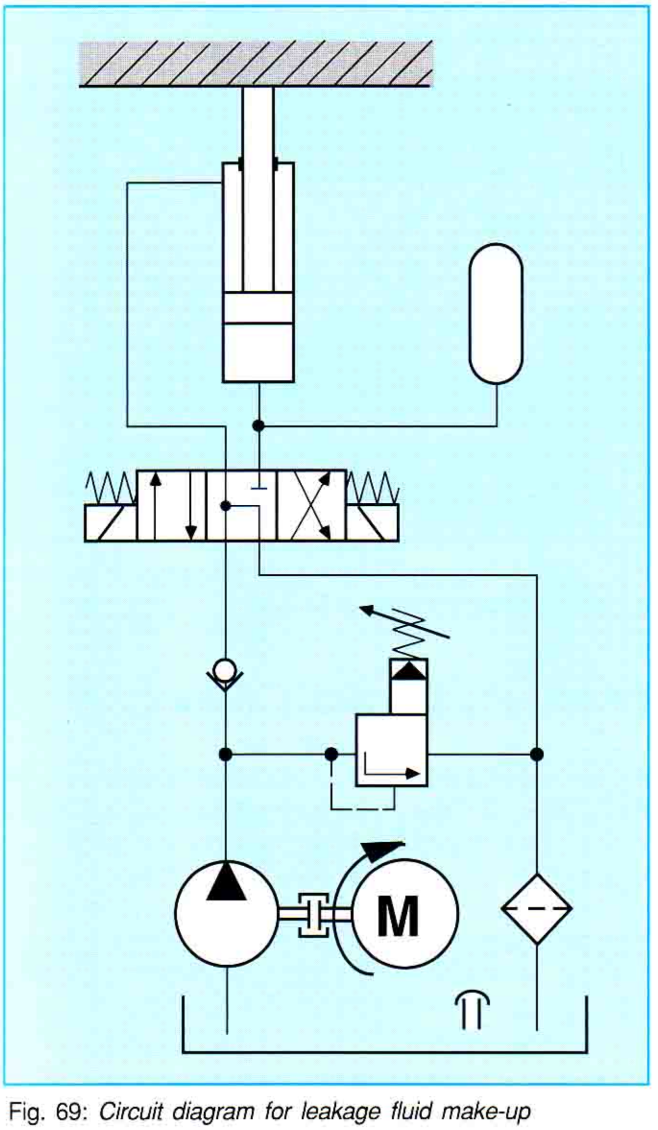


Fig. 69: *Circuit diagram for leakage fluid make-up*

6 Safety regulations

As pressure vessels, hydraulic accumulators are subject to the regulations governing pressure vessels which came into force in West Germany on July 1, 1980. Their design, installation and operation are covered by the TRB regulations. The pressure vessels of the accumulators are classified as follows according to the maximum permitted operating pressure p in bar, the capacity l in litres and the product of pressure and capacity ($p \cdot l$):

Class II: $p > 1$ bar and $p \cdot l < 200$

Class III: $p > 1$ bar and $200 < p \cdot l < 1000$

Class IV: $p > 1$ bar and $p \cdot l > 1000$

Hydraulic accumulators of Classes III and IV may only be put into service after an authorized inspector (i.e. TÜV) has inspected them (initial inspection, design inspection and pressure test) and certified that they are in satisfactory condition. This inspection by an authorized inspector can be waived if type-test approval has been obtained.

In the case of Class II accumulators the manufacturer himself certifies satisfactory manufacture and successful pressure testing. The operator's authorized inspector performs an acceptance test and provides appropriate certification. In-service tests are carried out by authorized inspectors on Class IV accumulators. The interval between internal inspections is 10 years when non-corrosive fluids are used, otherwise it is every 5 years. A pressure test is performed by the inspector every 10 years. Class II and III accumulators are inspected at intervals determined by the operator according to experience with the mode of operation and operating fluid.

Only inert gases such as nitrogen may be used in accumulators. No work may be carried out on accumulator pressure vessels until the fluid has been drained. No work may be carried out on the gas side of an accumulator until it has been depressurized.

7 Accessories for hydro-pneumatic accumulators

7.1 Safety and shut-off block

The safety and shut-off block is a device for protecting, isolating and depressurizing hydro-pneumatic accumulators. It conforms to the relevant safety regulations, especially those for pressure vessels (TRB 403 and 404 in West Germany) which require suitable devices for:

- measurement of pressure
- protection against over-pressure
- isolation.

With a simple arrangement of the various components in a compact unit, each one of these points is covered and additional benefits incorporated such as:

- minimum space requirement
- fast installation
- connections for all types of accumulator with imperial and metric fluid ports and flanged and welding connections
- extra valves such as pilot operated check valves, relief valves, flow valves, combined flow and check valves in add-on or integral form.

The construction of a safety and shut-off block is illustrated by the circuit diagram in *Fig. 70*. Basically, the unit comprises the valve block, the main shut-off cock and the depressurizing valve. It also incorporates the necessary connections to the tank, pressure gauge, accumulator and supply.

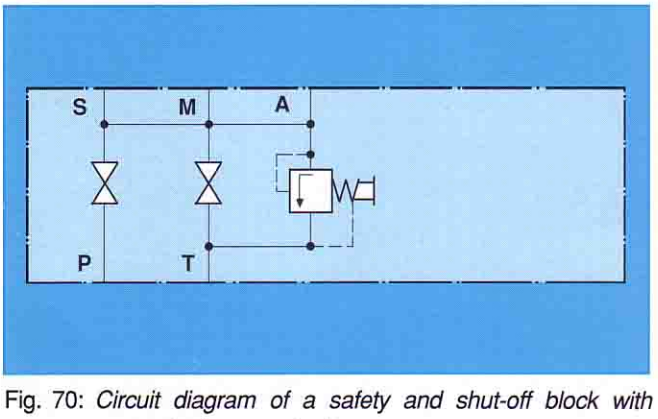


Fig. 70: *Circuit diagram of a safety and shut-off block with manual depressurization*

By fitting an extra solenoid operated two-way valve *(see Fig. 71)* it is also possible to depressurize the accumulator automatically. Another device that can be fitted is a pilot-operated pressure relief valve *(see Fig. 72)* which allows the accumulator to be depressurized quickly or in a controlled manner.

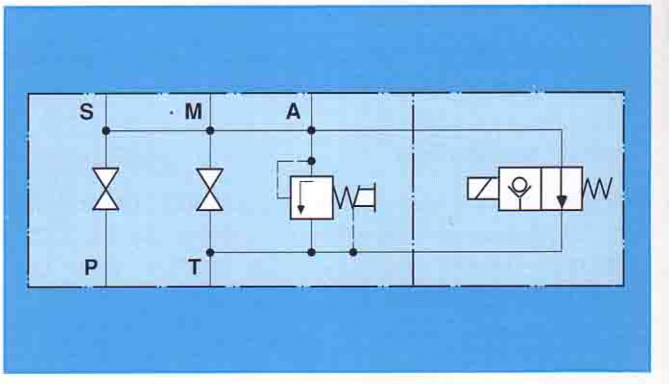


Fig. 71: *Circuit diagram of a safety and shut-off block with electro-magnetically-controlled depressurization*

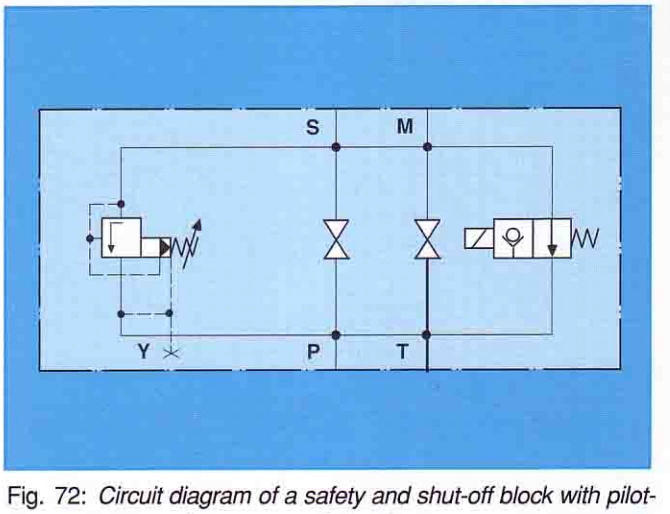


Fig. 72: *Circuit diagram of a safety and shut-off block with pilot-operated pressure relief valve*

7.2 Charging and testing device

A major loss of nitrogen is an unusual occurrence with hydro-pneumatic accumulators. However, it is still advisable to check the gas charging pressure at regular intervals so that the separating element cannot be damaged by a drop in the charging pressure and stop the accumulator working properly. The charging and testing device is of great assistance in charging accumulators, checking the gas pressure and adjusting it if necessary.

In order to charge an accumulator with the appropriate gas, the charging and testing device must be connected up as shown in *Figs. 73 and 74* with a screw connection to the gas valve of the accumulator and a flexible hose to a standard nitrogen bottle. If the gas pressure is only being checked or reduced, the charging hose will not be required. The recommended intervals between tests of the value of gas charging pressure stated on the rating plate or accumulator body are - at least once during the first week after installation, once more after a further 4 months and then annually.

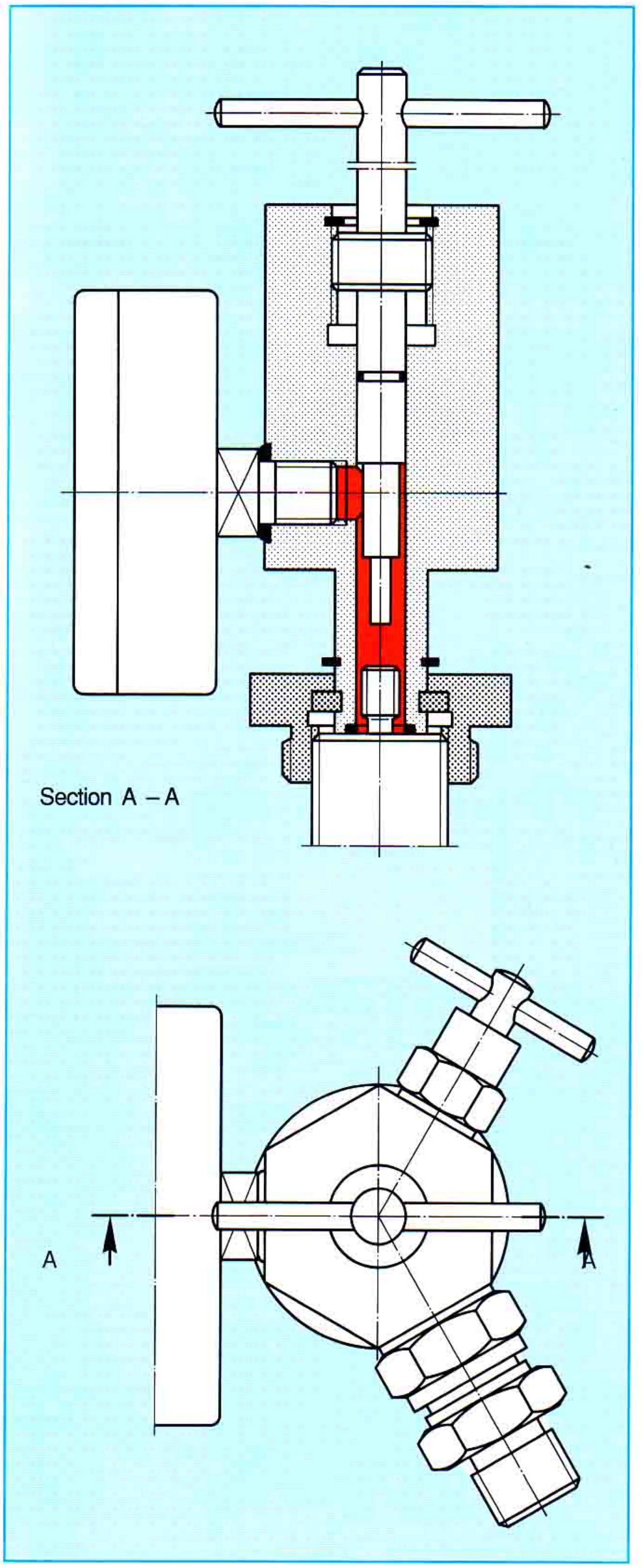


Fig. 73: *Charging and testing device for bladder-type accumulators*

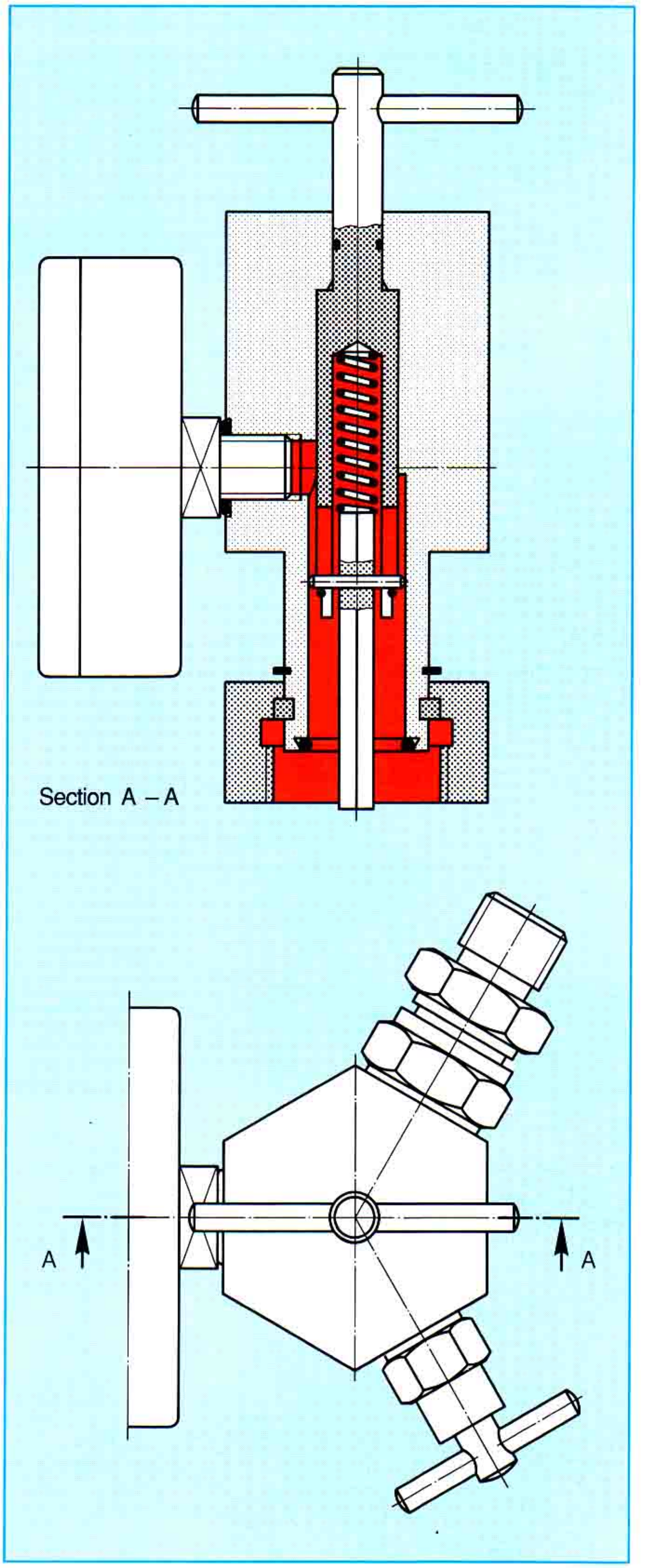


Fig. 74: *Charging and testing device for piston-type and diaphragm-type accumulators*

7.3 Nitrogen charging unit

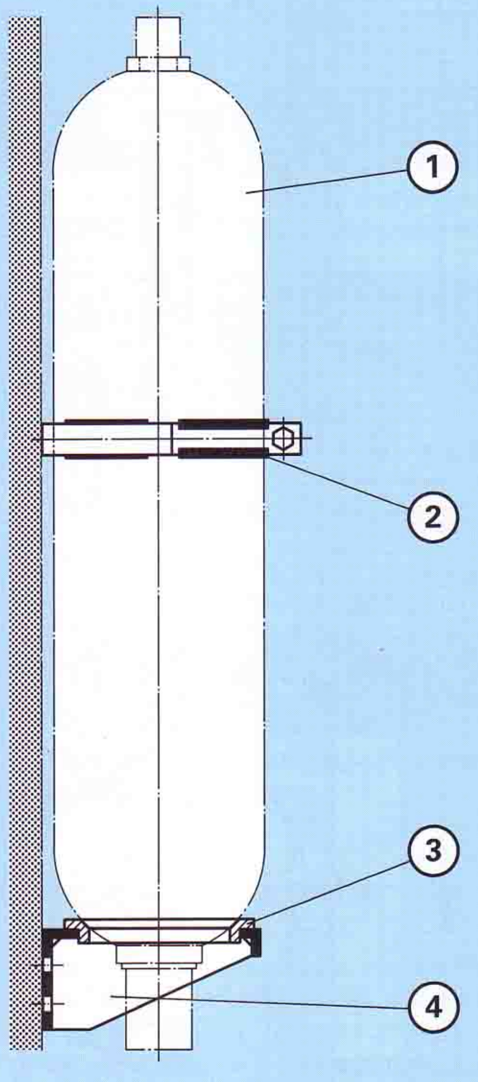
The nitrogen charging unit shown in *Fig. 75* is suitable for charging small accumulators or for topping up the charging pressure of large multiple accumulator systems. The accumulator charging pressure can be increased using the unit; this is mostly necessary when the pressure of a commercially available nitrogen bottle is insufficient for the charging operation.

Fig. 75: *Mobile nitrogen charging unit*

7.4 Mounting accessories

Hydro-pneumatic accumulators are heavy and also subjected to acceleration forces due to the flow of fluid so they must be firmly supported and secured. The mounting arrangements must be such that no stresses or strains are transmitted to the pipework from the accumulator.

Fig. 76 shows a typical mounting arrangement for a bladder-type accumulator employing a bracket and clip. Similar mounting arrangements can also be used for diaphragm-type and piston-type accumulators.



1 Bladder accumulator
2 Clip
3 Rubber seat
4 Bracket

Fig. 76: *Mounting arrangement for a bladder-type accumulator*

8 Symbols and subscripts

Symbols

Symbol	Units	Quantity
A	m², cm², mm²	Area
c_v	J/(kg K)	Specific thermal capacity at constant volume
m	kg	Mass
p	bar	Pressure
$\dot{Q}$	cm³/min, L/s	Flow
R	J/(kg K)	Gas constant
s	m, cm, mm	Travel, stroke
T	°C, K	Temperature
t	s, min	Time
V	L	Volume
W	J	Work
α	W/(m² K)	Heat transfer coefficient
τ	s	Thermal time constant

Dimensionless symbols

Symbol	Quantity
C	Correction factor
n	Polytropic index
κ	Adiabatic index

Prefixes

Symbol	Quantity
Δ	Difference
d	Differential
$\int$	Integral

Head marks

Symbol	Quantity
$\dot{}$	Referred to time
$\hat{}$	Maximum
$'$	Deviation from initial value

Subscripts

Symbol	Quantity
0, 1, 2	Changed state
a	Adiabatic
B	Operating status
req	Required
tot	Total
i	Isothermal
ideal	Ideal gas
L	Leakage fluid
max	Maximum
actual	Actual gas
TB	Operating temperature
V	Referred to volume
permit	Permitted