# Pneumatic buffer: An engineering approach

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**Abstract**

This article shows the calculation of simple pneumatic buffers. It is specially made for pulling cylinders with pneumatic buffers, used in heave compensation systems. This article shows a modeling approach and the main parameters which should be taken into account while designing the buffer. This article shows a case study, including a dynamic simulation, and the important parameters which are of most importance while designing the buffer and cylinder. The several parameters during designing are linked with the case study, to show the choices for the case study.

Cylinders are used for various applications. The cylinders which are part of this study, are a special kind: The bottom side is filled with nitrogen, while the rod side is filled with oil. In this configuration the cylinder is a pulling cylinder: The working pressure at rod side is much larger than the pressure at bottom side. The reaction force of the wire pulls on the cylinder.

The bottom side of the cylinder is connected to a bottom bottle. The bottom bottle is prepressurized to a low pressure when the cylinder is retracted, to make sure there is no vacuum when the cylinder is extended and makes sure that the piston, rod and sheavebox do not drop when unsupported by the wire when the cylinder is mounted rod upwards. The pressure also prevents air, and thus oxygen, from entering the system, which would cause oxidation in the bottle and cylinder. The larger the bottom bottle, the less the bottom pressure variates over the total stroke of the cylinder.

The rod side is connected to a piston accumulator through a valve. The valve can be closed, to hydraulically lock the cylinder. This is used for instance during maintenance or when the cylinder is out of operation. The piston accumulator separates the oil of the cylinder from the gas in the working bottles. During operation, the working bottles are set to a certain pressure, which equals a force on the cylinder. Due to the (large) volume of working bottles, the pressure variation at rod side is kept within boundaries: The larger the volume of the working bottles, the lower the pressure variation. An overview of the system is shown in figure 1.

These cylinders are for instance used for heave compensation. At the end of the cylinder is a sheave box, to reeve a wire. These heave compensation cylinders, with their low pressure at bottom side, can obtain high velocities due to operator error and in a wire break event: The wire connected to the cylinder breaks and the reaction force is suddenly gone. The pressure in the piston accumulator accelerates the piston of the cylinder, which collides with the bottom end cap of the cylinder. To prevent damage to the cylinder and lower the (dynamic) forces, a buffer is included at the bottom of the cylinder. This buffer works as an air cushion, or an air spring as you like.

## 1 The pneumatic buffer

The buffer which is of interest, is a relative new design at Huisman. A cross-section of the cylinder is shown in figure 2. The bottom of the cylinder has a gap, to connect the cylinder to the bottom bottle. Once the piston is too close to the end-of-stroke position, the buffer seals the gap. A captive seal is used, because the seal is only supported at both sides once the buffer is in the gap.

Once the buffer is in the gap and isolates the

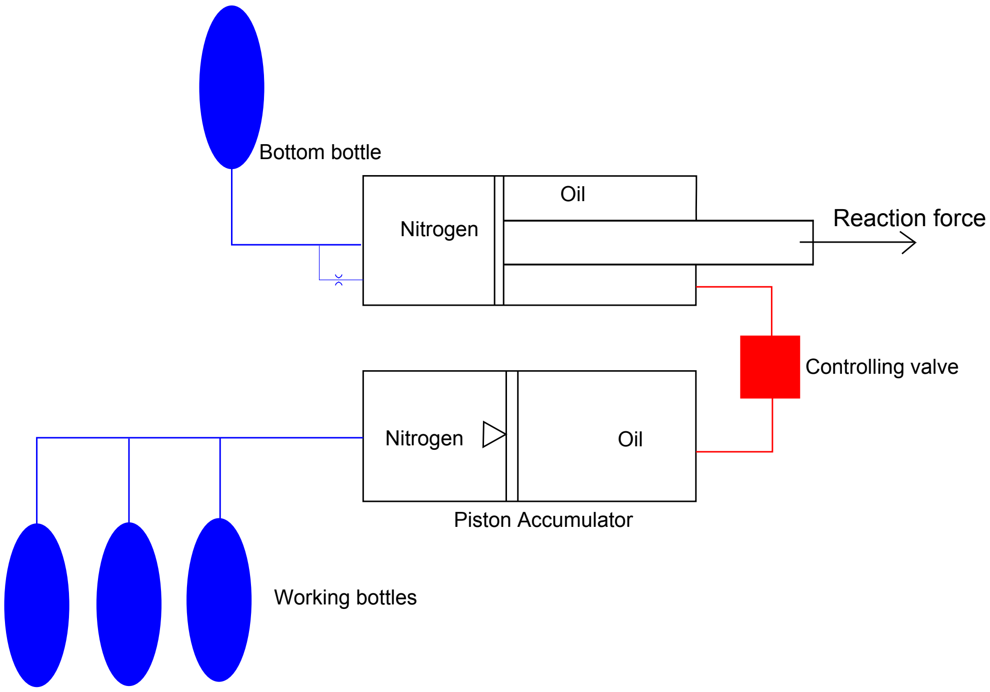



Figure 1: An overview of the system as intended for this buffer

bottle from the bottom of the cylinder, the nitrogen in the bottom of the cylinder is compressed when the piston moves further. Due to the relative small volume in the buffer, the pressure rises quickly and slows down the piston. Important for this design are the tolerances at the seals and that there is a small orifice from the cylinder bottom to the bottom bottle, to make sure that over a longer period of time the pressure will drop and the piston will come to the end of stroke position.

# 2 Simple modeling the buffer

At first only a few factors are taken into account. These factors are:

- Mass of the piston, rod and sheave box;
- Pressure at the bottom side;
- Pressure at the rod side;
- Pressure drop over the valve;
- Reaction force of the wire.

## 2.1 Overview of the simple model

The mechanical part is obtained by using Newton's second law on the cylinder's piston, rod and sheave box. The forces acting on the piston are the pressure at rod side, the pressure at bottom side and the reaction force. Damping is added for simulation stability, but will be small or zero.

$$\begin{aligned} -\left(p_{work}+\Delta p_{valve}\right)\tfrac{\pi}{4}\left(D_{bore}^2-D_{rod}^2\right) \\ +p_{bottom}A_{bottom}+F_{rod} \\ =m\ddot{x}+d\dot{x} \end{aligned} \tag{1}$$

On the next page is a table where all symbols are explained. All terms are considered separately in the following subsections. The only exception is the mass ($m$), which als needs to be estimated.

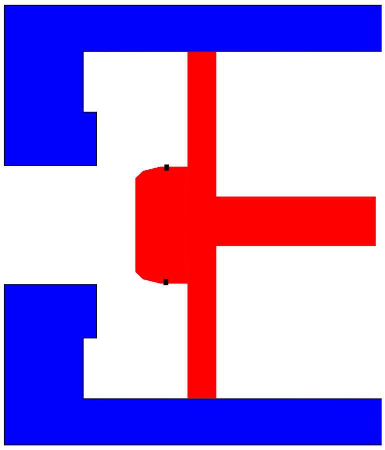


Figure 2: An abstract sketch of the cross-section of the cylinder: The buffer is attached to the piston (both red). Once it enters the gap at the bottom of the cylinder (blue), the gap is sealed. A captive seal (black) on the buffer is used to properly seal the cylinder.

| **Explanation of the variables** | |
|---|---|
| $p_{work}$ | Working pressure in the piston accumulator |
| $\Delta p_{valve}$ | Pressure difference over the controlling valve |
| $D_{bore}$ | Diameter of the bore of the cylinder (bottom side) |
| $D_{rod}$ | Diameter of the rod of the cylinder |
| $p_{bottom}$ | Pressure at bottom side of the cylinder |
| $A_{bottom}$ | Bottom area, dependent on the whether the piston is in- or outside the buffering zone. |
| $F_{rod}$ | Reaction force |
| $m$ | Mass of the piston, rod and sheave box of the cylinder |
| $d$ | Damping of the system |
| $\ddot{x}$ | Acceleration of the piston, rod and sheavebox of the cylinder |
| $\dot{x}$ | Velocity of the piston, rod and sheavebox of the cylinder |

### 2.1.1 Rod pressure

The pressure at rod side is dependent on the pressure in the piston accumulator and the pressure drop over the controlling valve. The pressure in the piston accumulator can be calculated by estimating the pressure at half the stroke, which is approximately the reaction force divided by the rod area, and the gas pressure differences. The change in this pressure is dependent on the position of the piston of the piston accumulator. The piston of the accumulator and the cylinder are linked with (non-compressible) fluid, so this is also directly dependent on the position of the piston of the cylinder. The difference in gas volume determines the pressure. This is calculated using the ideal gas law, assuming an adiabatic process:

$$p_{work} = p_{middle} \left( \frac{V_{work} + \frac{1}{2} s A_{annular}}{V_{work} + x A_{annular}} \right)^k \tag{2}$$

With:

| **Explanation of the variables** | |
|---|---|
| $p_{middle}$ | Pressure in middle position |
| $V_{work}$ | Volume of working bottles |
| $s$ | Stroke of the cylinder |
| $A_{annular}$ | Rod area of the cylinder |
| $x$ | Position of the cylinder piston, 0 is fully retracted |
| $k$ | Adiabatic constant ($k = 1.6$) |

Where $A_{annular} = \frac{\pi}{4}(D_{bore}^2 - Drod^2)$.

The pressure drop over the control valve is caused by friction. From resistance laws (flow friction), it is known that: $\Delta p = \zeta \frac{1}{2} \rho v^2$ (please note that the cross area is constant, and $Q = Av$). This means that the pressure drop over the valve is dependent on the flow and that this relation is quadratic. In the valve documentation graphs are found where two different points will be used to fit a quadratic function $\Delta p = aQ^2$. Note that a fixed point is (0,0), so no flow is no pressure drop over the valve. The other point will be determined by the user of the model using the graphs, (A,B), where $A$ is the flow and $B$ is the pressure drop. The constant $a$ is found by:

$$a = \frac{B}{A^2} \tag{3}$$

Now the pressure drop over the valve can be estimated using $\Delta p = aQ^2$ and the flow can be calculated by the velocity of the piston of the cylinder multiplied with the annular area $\left(\dot{x} \left(\frac{\pi}{4}\right) \left(D_{bore}^2 - D_{rod}^2\right)\right)$.

In equation 1 the pressure drop is possitive, but for a negative velocity the pressure drop should be negative. This is incorparated in the model by substracting the pressure drop in that case. For no velocity the sign does not matter, because the pressure drop is zero.

### 2.1.2 Reaction force

During the simulation the reaction force is constant until a wire break event happens. When heave compensation is simulated, this value will have a sinusoidal signal on top of the constant value, but this is not required for only buffer simulation. In this paper this will be neglected and the force will be constant and keep the piston in equilibrium:

$$F_{rod} = p_{work} A_{annular} - p_{bottom} A_{bottom} \tag{4}$$

At a wire break event the reaction force is suddenly zero.

### 2.1.3 Bottom pressure

The pressure at bottom side is determined by the pre-filling (filled when cylinder is fully retracted) and the position of the piston of the cylinder. A large difference is when the piston is in the buffering zone: Outside this zone the buffer is not closed and the bottom bottle is connected to the bottom side of the cylinder. Inside the buffering zone the bottom bottle is not coupled with the bottom side, resulting in a much smaller volume which is compressed. This results in a much larger pressure difference for the same displacement of the piston of the cylinder. Because this compression is done in a relative short time, the flow through the orifice is neglected.

There are different volumes which have influence on the bottom pressure. These are listed below. Note that the volume of the piping is neglected, because it is normally small in comparison with the bottom bottle. If this is not the case, it should be added to the volume of the bottom bottle.

| **Explanation of the bottom volumes** | |
|---|---|
| $V_{bottle}$ | Bottom bottle volume |
| $V_{dead}$ | Dead volume (the volume at the bottom of the cylinder when the cylinder is fully retracted |
| $A_{stroke}$ | Stroke area, which multiplied with the actual stroke ($x$) gives the stroke volume: $V_{stroke} = \frac{\pi}{4} D_{bore}^2 \, x$ |
| $k$ | Adiabatic constant |

**If $x > L_{buffer}$**

When the position of the cylinder ($x$) is larger as the length of the buffer ($L_{buffer}$), the bottom bottle is connected to the bottom of the cylinder. In this case the volume of the buffer is neglected, because it is considered small with respect to the total volume and dependent on the actual design of the buffer (hollow versus solid design, wall thicknesses, etcetera). The pressure at bottom side is calculated as follows:

$$P_{bottom} = P_0 \left( \frac{V_{bottle} + V_{dead}}{V_{bottle} + V_{dead} + A_{stroke} x} \right)^k \tag{5}$$

In the first part of the simulation, the cylinder is set at half stroke and the gas has all the time to get to ambient temperature, so this is considered an isothermic process ($k = 1$). When the wire breaks, the gas does not have the time to transfer heat with the environment, which is considered adiabatic ($k = 1.4$ in this case, due to the low pressure).

**If $x < L_{buffer}$**

Once the cylinder is in the buffering zone ($x \leq L_{buffer}$), the bottom bottle is no longer connected with the bottom of the cylinder. This means that the volume is drastically decreased, which gives a larger pressure rise for the same displacement of the piston of the cylinder. The bottom pressure is calculated as follows:

$$A_{buffer} = \frac{\pi}{4}(D_{bore}^2 - D_{buffer}^2) \quad (6)$$

$$P_1 = P_0 \left( \frac{V_{bottle} + V_{dead}}{V_{bottle} + V_{dead} + A_{stroke}x} \right)^1 \quad (7)$$

$$P_2 = P_1 \left( \frac{V_{bottle} + V_{dead} + A_{stroke}x}{V_{bottle} + V_{dead} + A_{buffer}L_{buffer}} \right)^{1.4} \quad (8)$$

$$P_{bottom} = P_2 \left( \frac{A_{buffer}L_{buffer} + V_{dead}}{A_{buffer}x + V_{dead}} \right)^{1.6} \quad (9)$$

In these equations the adiabatic constant ($k$) is already filled in. The idea behind it is that it first calculates the pressure at half stroke (isothermic process, $k = 1$), then adiabatic compression towards the buffer (adiabatic process, low pressure, $k = 1.4$), and last it compresses the buffer (adiabatic process, high pressure, $k = 1.6$).

#### 2.1.4 Damping

The damping which is incorporated in the model compensates for the friction which is not included in this model. The amount of damping should be estimated if required. The damping due to the valve friction is incorporated in the model. All other kinds of friction are considered small, so the result from this damping coefficient should also be small. Neglecting this parameter gives a good idea of how the buffer will work, although it might give too harsh conditions (like too large velocities) for the design of the buffer.

### 2.2 The simulation of the simple model

For this simulation, the differential equation is rewritten to two first order differential equations:

$$\underline{f} = \begin{bmatrix} \dot{x} \\ x \end{bmatrix} \quad (10)$$

This means that the first derivative is:

$$\underline{\dot{f}} = \begin{bmatrix} \ddot{x} \\ \dot{x} \end{bmatrix} \quad (11)$$

With equation 1 the following first order differential equation is obtained:

$$\underline{\dot{f}} = \begin{bmatrix} \frac{-(P_{work}+\Delta P_{valve})A_{annular}+P_{bottom}A_{bottom}+F_{rod}-d\dot{x}}{m} \\ \dot{x} \end{bmatrix} \quad (12)$$

The parameters used for the simulation, are:

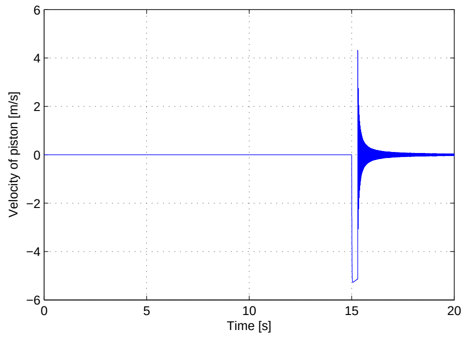
| **Used parameters** | | |
|---|---|---|
| $m$ | 1000 | $[kg]$ |
| $D_{bore}$ | 0.19 | $[m]$ |
| $D_{rod}$ | 0.10 | $[m]$ |
| $D_{buffer}$ | 0.07 | $[m]$ |
| $s$ | 3 | $[m]$ |
| $L_{buffer}$ | 0.025 | $[m]$ |
| $P_0$ | 20e5 | $[Pa]$ |
| $V_{dead}$ | 0.01e-3 | $[m^3]$ |
| $V_{bottle}$ | 500e-3 | $[m^3]$ |
| $V_{work}$ | 800e-3 | $[m^3]$ |
| $P_{middle}$ | 210e5 | $[Pa]$ |
| $d$ | 0 | $[\frac{kg}{s}]$ |

The estimation of the pressure drop over the valve ($\Delta P_{valve}$), which is a NG32 valve of Moog in this case, was done by the flow ($Q = 700[\frac{L}{min}]$) and pressure ($p = 2.1[bar]$).

During this simulation, the first five seconds are used to calculate the bottom gas pressure with a isothermal process, as explained in section 2.1.3. Up to 15 seconds nothing is changed and all parameters are steady. At $t = 15[s]$ the wire breaks, meaning that the reaction force is suddenly zero.

The result is shown in figure 3. Damping is kept zero here, but due to the low dead volume, the piston does not hit the the bottom of the cylinder (if it is doubled, so $0.02[m^3]$, it would hit according the simulation). The lowest clearance between end cap and piston is $0.3[mm]$, which is very small.

What is also interesting to see, is that without extra damping (so neglecting all other frictions), the cylinder already has a maximum speed of about 5.3 $\left[\frac{m}{s}\right]$. This means that the pressure drop over the controlling valve is already sufficient to limit the maximum velocity. In reality it will be even lower due to other frictions.



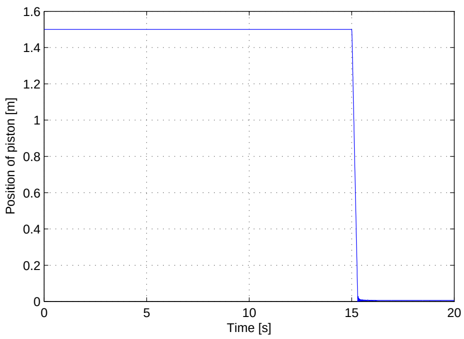




Figure 3: Result of the simple simulation

When zoomed in for the position where the buffer is active, the result is as shown in figure 4. Due to the lack of damping, the vibration is well visible. The end position is just above $6[mm]$ from the end of stroke position. In this position an equilibrium is reached between the force of the bottom pressure in the buffer and the force of the rod pressure.

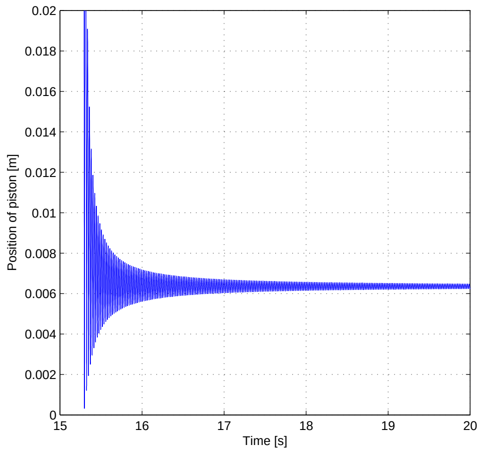



Figure 4: Zoomed in on the position at the area where the buffer is active

# 3 Engineering of a pneumatic buffer

During the design of a pneumatic buffer, the buffer is not simulated every time. The engineer must know what rules of thumb to follow to design a pneumatic buffer, to make sure it will provide sufficient damping and it will not reach the end stop. In this section, the most important design parameters are shown, to help the engineer to design the buffer properly.

The important parameters which will be covered, are:

- Equilibrium pressure;
- Energy to be dissipated;
- Volume of the buffer;
- Maximum pressure during buffering;
- Dead volume.

In this section the required volume is determined at two different approaches due to two different demands on the design: The maximum allowable pressure and the volume due to the energy dissipation. The largest volume should be chosen for the buffer to fulfill both demands.

## 3.1 Equilibrium pressure

The first thing to start with, which always requires to be checked, is the equilibrium pressure in the buffer. The most conveniant way of checking this static situation, is by using the cylinder ratio and an estimation of the rod pressure when the cylinder is retracted. Using this, the following is found:

$$p_{bottom} = p_{rod} \left( \frac{\frac{\pi}{4}(D_{bore}^2 - D_{rod}^2)}{\frac{\pi}{4} D_{bore}^2} \right) \tag{13}$$

For the example in section 2.2 the pressure at rod side is $198[bar]$ in the static situation after the wire break event, which means a pressure of $p_2 = 104[bar]$ at the bottom side of the cylinder.

The temperature of the gas at this state can be calculated using the ideal gas law:

$$\frac{pV}{T} = CONSTANT \tag{14}$$

$$T_2 = T_1 \left( \frac{p_2 V_2}{p_1 V_1} \right) \tag{15}$$

When this is done for the case in section 2.2 and $T_1 = 293[K]$ is assumed, $T_2 = 444[K]$ is found, which is $171[°C]$.

## 3.2 Energy to be dissipated

The energy which needs to be dissipated, or actually stored in the buffer, can be calculated using the kinetic energy of the cylinder's piston, rod and sheavebox. This leads to the following equation:

$$E_{kin} = \frac{1}{2} m v_{max}^2 \tag{16}$$

The mass ($m$) is often known: the mass of the piston, rod and sheave box which is attached to the rod. If the rod and piston mass is not known, a (very) rough estimate of half the cylinder weight can be used. The maximum velocity ($v$) is normally also already known, otherwise a simulation as done in section 2.2 can be used. Please note that in this case the buffer speed should be taken into account, not the normal operation speed.

For the case of section 2.2 this means about $14[kJ]$.

## 3.3 Dead volume

The dead volume is the volume on the bottom side of the cylinder which is left when the cylinder is fully retracted, so when the piston is touching the cylinder head. This volume is important, as will be shown in section 3.4. This should always be checked, because if this dead volume is too large, the buffer will not work properly. Of course: Most frictions are neglegted here, but note that the friction will be low compared to all other forces. On the other hand, the maximum pressure during buffering becomes lower with a larger dead volume, which means that the maximum pressure can be kept below the design pressure, although it will mean that the buffer length must be increased as well (see also the following subsections).

## 3.4 Volume of the buffer

As the nitrogen in the buffer is quickly compressed, it can be seen as an adiabatic process (no transfer of heat). For an adiabatic process Poisson's relations are used:

$$pV^k = C \tag{17}$$

where C is a constant, which can be determined from each known state.

From the first law of thermodynamics ($Q = \Delta U + W$) the nitrogen is compressed (raise of $\Delta U$) due to the work ($W$) the cylinder has put in the nitrogen (note that $Q = 0$ due to adiabatic process). The work can be described as the integral of the pressure over the volume:

$$W = \int_{V_A}^{V_B} p dV \tag{18}$$

Using these two relations, the following is obtained:

$$W = \int_{V_A}^{V_B} \frac{C}{V^k} dV \tag{19}$$

$$W = C \int_{V_A}^{V_B} V^{-k} dV \tag{20}$$

$$W = C \left( \frac{V_2^{1-k} - V_1^{1-k}}{1-k} \right) \tag{21}$$

From the kinetic energy and the work, the volume can be calculated:

$$-E_{kin} = C\left(\frac{V_2^{1-k} - V_1^{1-k}}{1-k}\right) \quad (22)$$

$$V_2^{1-k} - V_1^{1-k} = \frac{-E_{kin}(1-k)}{C} \quad (23)$$

$$V_1^{1-k} = V_2^{1-k} - \frac{-E_{kin}(1-k)}{C} \quad (24)$$

With this expression, where $V_2$ is the volume at which the cylinder is stopped and $V_1$ is the volume at which buffering starts, the required difference in volume can be calculated. Please note that the volume $V_2$ needs to be larger as the dead volume, to prevent the piston from hitting the cylinder head. For the maximum velocity it is wise to have some stroke left, so the extra stroke together with the dead volume determines the volume $V_2$.

In the case of section 2.2, the constant $C \approx 14,85$ and is the volume at which buffering starts $V_1 = \frac{\pi}{4}(D_{bore}^2 - D_{buffer}^2)L_{buffer} \approx 6.23 \cdot 10^{-4}[m^3]$. The volume $V_2$ can now be calculated and is $V_2 = 2 \cdot 10^{-5}[m^3]$. During the simulation it was already said that if the dead volume ($V_{dead}$) is $0.02[ltr]$ the piston touched the cylinder, which is actually calculated at this section.

## 3.5 Maximum pressure during buffering

The maximum pressure during buffering is an important design parameter. The maximum pressure should never exceed the design pressure of the cylinder. This should prevent a too large pressure, which can cause damage to the cylinder. Note that exchanging these cylinders is very hard, time consuming and costly.

The maximum pressure defines the buffer volume. To calculate the volume of the buffer, the design pressure ($p_2$), pressure at the point where the buffer is closed ($p_1$) and the volume at the moment the cylinder is stopped ($V_2$) need to be known or chosen. Please note that $V_2$ also includes the dead volume, as shown in section 3.4. The starting volume of the buffer $V_1$ can be determined as follows:

$$V_1 = V_2\left(\frac{p_2}{p_1}\right)^k \quad (25)$$

The adiabatic constant should be chosen at 1.6 due to the large pressures in the buffer.

With the volume $V_1$ of this subsection and the previous subsection, the minimum required buffer volume is determined. To fulfill both requirements, choose the largest required volume.