Exploiting the Capacity of Industrial Hydraulic Buffers

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Abstract. Industrial hydraulic buffers are standard equipment for industrial machinery. They are used for the reduction of impact loads on structures during processes of kinetic energy reduction. This is realized by a more or less constant buffer force acting along the stroke of the buffer. The product of buffer force and stroke results in the energy being dissipated during a buffering process. The typical constant, restricted buffer force makes sense in order to protect the surrounding structure. On the other hand, a non-constant, optimized buffer force may be useful to exploit the energy potentially dissipated by a buffer. The article considers the basics of hydraulic buffer technology, the question of a non-constant buffer force and the potentials arising with this option.

1 BUFFER DESIGN AND FUNCTION

Industrial hydraulic buffers as shown in Fig. 1 work according to a simple principle. The moving piston of the buffer presses hydraulic fluid through a throttle. The pressure drop at the throttle $\Delta p_{Throttle}$ produces a fluid pressure and an according piston force F_p.

Figure 1 Crane track limit buffer

The piston force Fp is used to decelerate a mass with its initial kinetic energy, a crane mass at the crane track limit e.g. For a first approach the piston force may be assumed to be constant against the piston rod position. As the product of constant piston force $F_{b,const}$ and stroke s_{stroke} in this case equals the energy dissipated W_{diss}, there are different design options for the buffer. A higher buffer force in combination with a smaller stroke gives the same energy dissipated as a lower buffer force in combination with a larger stroke. The choice of combination has a significant impact on the buffer force, the buffer stroke and the buffer geometric properties as the total buffer length in a relaxed condition (Fig. 2).

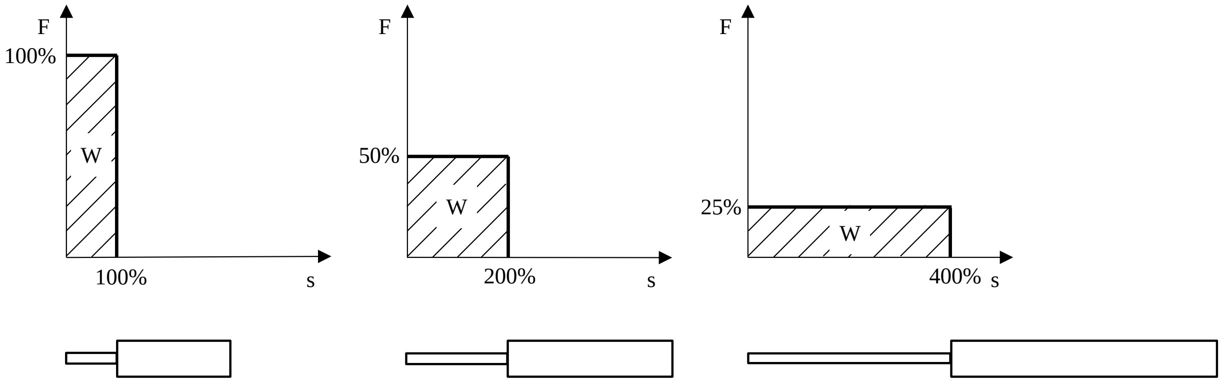


Figure 2 Buffer forces and buffer lengths at constant energy dissipated

In practice the buffer curve is not rectangular (Fig. 3). The slopes of the buffer curve at the beginning and the end of the buffer curve are not infinite e.g. This results in a maximum buffer force $F_{b,max}$ well above the constant buffer force $F_{b,const}$. The relation of the maximum buffer force $F_{b,max}$ to the constant buffer force $F_{b,const}$ is indicated by parameter k.

$$W = \int_0^{s_{stroke}} F \, ds = F_{b,const} \cdot s_{stroke} \tag{1}$$

$$F_{b,max} = k \frac{W}{s_{stroke}} = k \cdot F_{b,const} \tag{2}$$

with

$$k > 1 \tag{3}$$

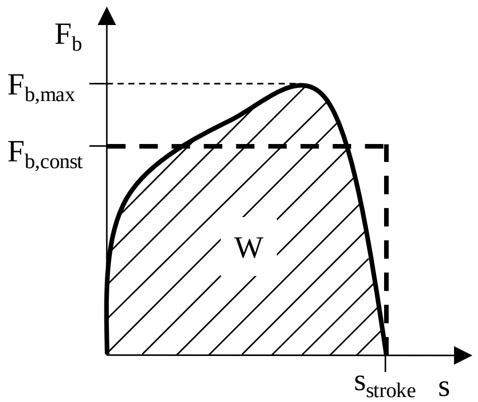


Figure 3 Theoretical vs. practical buffer characteristics

For several simple buffer types, a non-constant buffer force is typical. Metallic springs or cellulose buffers show a parameter k well above one. This leads to a high maximum buffer force $F_{b,max}$. Hydraulic buffers are designed to reach a parameter k of about one. For a certain stroke and a certain maximum buffer force $F_{b,max}$ they absorb the maximum energy. Hydraulic buffers absorb certain energy at a minimum buffer force $F_{b,min}$. This leads to a positive impact on static system requirements.

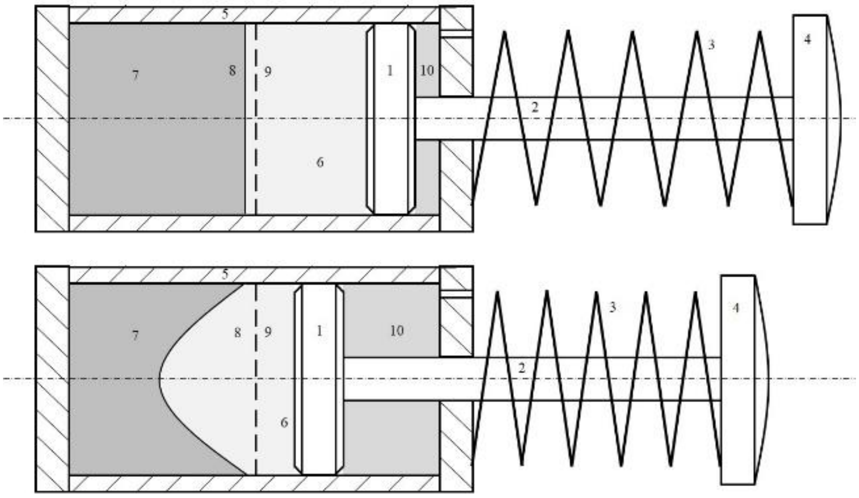


Figure 4 Hydraulic buffer in relaxed (above) and articulated (below) position

Fig. 4 shows a design principle of a hydraulic buffer. Shown above is the hydraulic buffer in its relaxed position. A hydraulic buffer consists of a piston (1) running in a cylinder (5). As the piston is loaded by forces onto the head (4) via a piston rod (2), it moves to the left-hand side. The closed space left of the piston is separated by a membrane (8). The space next to the piston is filled with oil (6), and the space behind the membrane is filled with gas (7).

With the piston moving to the left, the oil volume is pressed to the left against the deformed membrane and the increasing gas pressure behind the membrane (Fig. 4, below). The characteristics of the buffer during this movement are realized by the characteristics of the throttle (9) the oil is pressed through. On the right-hand side of the piston, all time exists outside air pressure (10). As the head is not loaded, the piston is reset to the original position, caused by the resetting spring (3) and the gas pressure.

There are further parameters besides the stroke influencing the buffer force. The oil pressed through the throttle (9) has to find new space. This space is delivered by the adjustable membrane (8), working against the gas pressure. This gas pressure has some initial value, to move the relieved piston rod against friction back to its initial position, even after small displacements. This task may be supported by a pre-stressed spring (3). The gas pressure varies according to the piston rod position and reaches its maximum at the maximum stroke. Additionally, the piston force does not only have to decelerate the mass. It also has to equalize an external driving force acting on the piston rod. External driving forces occur at movements in a gravitational field typically.

The shown principle of a hydraulic buffer may be modified by alternative designs of the gas containment, the moving border of the gas containment, the throttle, and the mechanism for withdrawal of the released piston e.g. The membrane may be substituted by a second piston system. The throttle may comprise an adjustment of characteristics depending on the piston position. This adjustability may be realized by throttle holes positioned along the stroke (Fig. 5). The moving piston covers a certain part of the holes and this causes a variable throttle cross-section and characteristics. A hydraulic buffer may allow refillability of gas volume in order to replace diffused gas share.

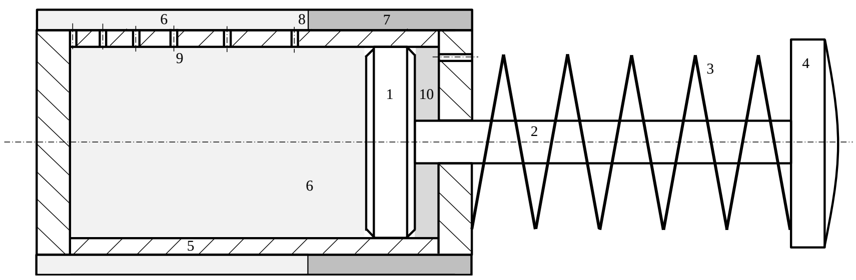


Figure 5 Hydraulic buffer with adjustable characteristics

The buffer comprises different limit states, restricting the performance:

- Cylinder strength
- Cylinder base fastening strength
- Throttle strength
- Piston rod buckling force strength
- Sealing mechanical strength
- Sealing heat strength
- Sealing wear lifetime
- Oil heat strength
- Oil lifetime e.g.

Exceeding a limit state probably means the limit energy to be absorbed was exceeded:

- Energy excess in the sense of excess mass
- Energy excess in the sense of excess speed
- Energy excess in the sense of excess driving force
- Energy excess in the sense of excess frequency.

2 REFERENCE BUFFER

A reference buffer situation with the following parameters is considered:

Throttle hole diameter	$d = 1\cdot 10^{-3}$ m
Final throttle diameter	$d_{final} = 1\cdot 10^{-3}$ m
Piston diameter	$d_p = 0.1$m
Driving force	$F_{Drive} = 0$ N
Load mass	$m = 1{,}280{,}000$ kg
Initial gas pressure	$p_0 = 500{,}000$ Pa
Maximum stroke	$s_{max} = 0.4$m
Rest stroke	$s_{rest} = 0.05$
Initial gas volume	$V_0 = 6\cdot 10^{-3}$ m^3
Impact speed	$v_i = 0.5$ m/s
Initial speed of piston rod	$v_0 = 0.5$ m/s
Oil density	$\rho = 890$ kg/m^3
Isentropic exponent	$\kappa = 1.35$
Efficiency	$\eta = 1.0$

As the force braking the load F_{Brake} is assumed to be constant, the deceleration of the mass a_0 is constant as well. This results in a linear decrease of the mass speed over time and a quadratic increase of the mass displacement over time (Fig. 6). With these assumptions the speed gets very slow at the end of the stroke. As the position over time shows an asymptotic behaviour the piston seems not to move at the end of the stroke anymore (Fig. 6).

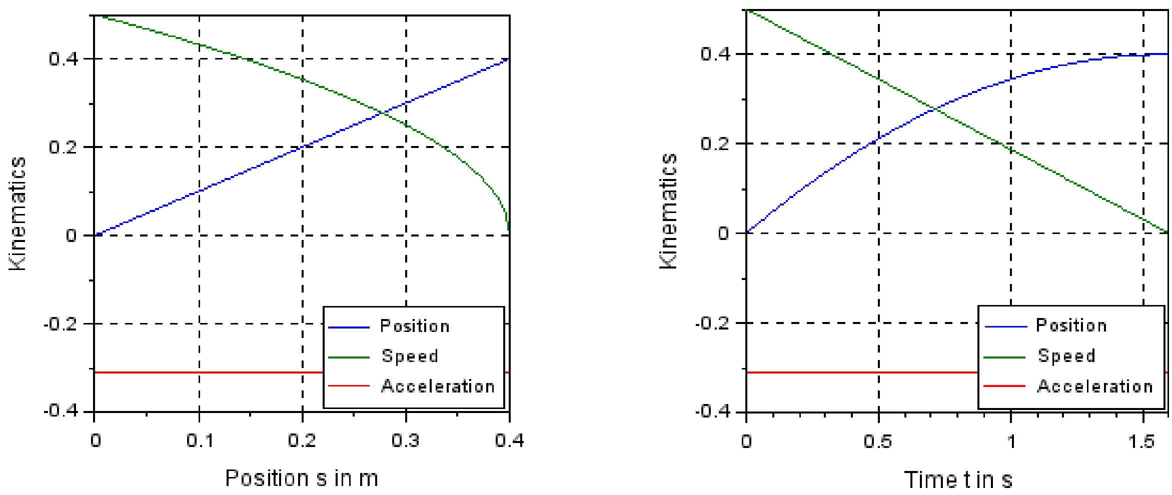


Figure 6 Kinematic values over position and over time

The initial speed of the piston rod v_0 is considered equal to the impact speed of the mass v_i. The impact process is neglected due to the relation of load mass m to piston rod mass. The constant braking force F_{Brake} required results out of the constant deceleration:

$$F_{Brake} = m|a_0| \quad (4)$$

The braking force F_{Brake} available constitutes out of the spring force F_{Spring}, the gas force F_{Gas}, the throttle force $F_{Throttle}$ and the driving force F_{Drive}:

$$F_{Brake} = F_{Spring} + F_{Gas} + F_{Throttle} - F_{Drive} \quad (5)$$

Frictional losses are neglected in this consideration (Efficiency $\eta=1$). A spring force is assumed not to be present here:

$$F_{Spring} = 0 \quad (6)$$

$$F_{Brake} = F_{Gas} + F_{Throttle} - F_{Drive} \quad (7)$$

The compression of the gas is assumed to take place as an isentropic compression with isentropic exponent κ. The pressure drop at the throttle is assumed to be proportional to the square of fluid speed with loss coefficient ξ.

The gas force F_{Gas} typically is quite small and of low influence. At a constant braking force F_{Brake} and a constant driving force F_{Drive} the throttle cross-sectional area A_T is about proportional to the piston speed v. Assuming all other sizes are constant or neglectable, the throttle cross-sectional area shows about the same characteristics as speed over position.

The throttle cross-sectional area A_{T0} constitutes out of the throttle cross-sectional area distributed over the stroke A_{dist} and of the throttle cross-sectional area located at the end of the stroke A_{final}. The throttle cross-sectional area located at the end of the stroke A_{final} is implemented to give the piston the opportunity to reach the dead point with finite speed.

Considering the final throttle cross-sectional area A_{final} this regime leads to the following characteristics for the throttle cross-sectional area A_T:

$$A_T = \sqrt{1 - \frac{s}{(s_{max} - s_{rest})}}\left(A_{T0} - A_{final}\right) \tag{8}$$

In practice, the throttle cross-sectional area may not distributed continuously. The throttle may be realized by discrete holes implemented at certain distances from each other. The real throttle cross-sectional area A_{Treal} depends on the hole diameter d and the number of holes N.

So far oil was assumed not to be compressible. Thus, oil pressure immediately depends on gas pressure and throttle pressure. In fact, oil is compressible. A certain stroke is required to build up oil pressure. At the very beginning of the buffering process, the stroke is realized at a lower buffer force level. It may be an option to implement this aspect to a more detailed model.

The discretized throttle cross-sectional area curve implements the decrease as singular oil outlet geometries. They have no extension with regard to length in the direction of the stroke. In fact, already simple holes have an extension with the hole diameter as length and certain characteristics. It may be an option to implement this in more detailed models. It may also be an option to consider more intrinsic geometries than round holes.

The reference buffer comprises a throttle suitable to create a constant buffer force against stroke. Four variations of this throttle are considered (Fig. 7):

- Continuous throttle, without final throttle.
- Continuous throttle, with final throttle.
- Discontinuous throttle, hole as step, with final throttle.
- Discontinuous throttle, hole as a ramp, with final throttle.

The continuous throttle without final throttle shows a throttle cross-sectional area converging against zero at the end of the stroke. All other variations show the final cross-sectional area up to the end of the stroke. The discrete throttle characteristics show a stepped change of throttle cross-sectional area. At the real throttle, the steps are represented by holes in the throttle bushing along the stroke. According to the models, the steps take place in certain hole centre positions (step) or along a certain distance corresponding with the hole diameter (ramp) (Fig. 7).

The initial cross-sectional area of the throttles differs for the discontinuous characteristics from the continuous characteristics. The continuous characteristics meet the theoretical value A_{T0}. As the discontinuous characteristics depend on a sum of holes, their initial cross-sectional area only can be close to the theoretical value A_{T0}.

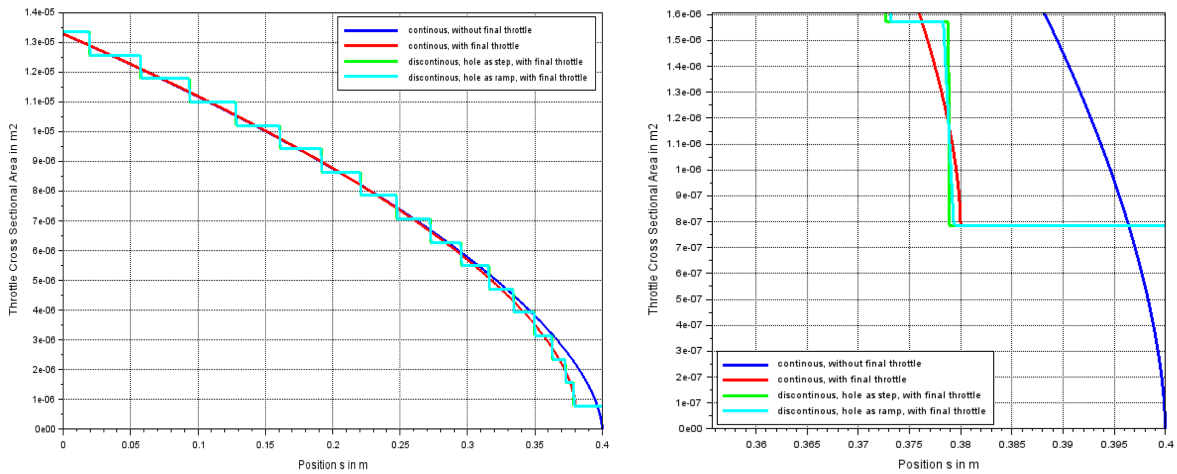


Figure 7 Throttle characteristics along the stroke

3 BUFFER CHARACTERISTICS

The throttle cross-sectional area with continuous distribution starts at $A_{T0} = 1.428 \cdot 10^{-5}$ m^2 and finishes with $A_{final} = 0$ m^2. The buffering process shows a deceleration close to being constant during the first part of the stroke. In the second part of the stroke, the acceleration decreases significantly. Speed over time at first is decreasing quite fast. Towards the end of the stroke, the speed is small. It takes quite a time to drive across the final distance. As a speed limit is taken as one termination condition for simulation, the simulation is terminated before the piston reaches maximum stroke. The total oil pressure consists of a high pressure drop in the throttle and a low gas pressure. This allows a high force at the working stroke of the buffer and a low force at the buffer back stroke afterwards. This allows high energy to be dissipated by the buffer. Finally, the total kinetic energy of E = 160,000 J is dissipated.

The throttle cross-sectional area with discontinuous distribution starts at $A_{T0} = 1.428 \cdot 10^{-5}$ m^2 and finishes with $A_{final} = 7.85 \cdot 10^{-7}$ m^2 (Fig. 8). At the end of the stroke, the throttle cross-sectional area is constant for some distance. The buffer with final throttle shows a behaviour quite similar to the buffer without final throttle. The piston reaches the region of the final throttle, but also not maximum stroke (Fig. 8). The main difference in comparison to a buffer without final throttle is that when the buffer reaches the throttle, there occurs a peak in throttle pressure and all associated conditions (Fig. 8, Fig. 9, Fig. 10). This is due to the non-continuous deviation of the throttle cross-section characteristics. The height of the peak in correlation with throttle characteristics in detail and in practice may be of interest.

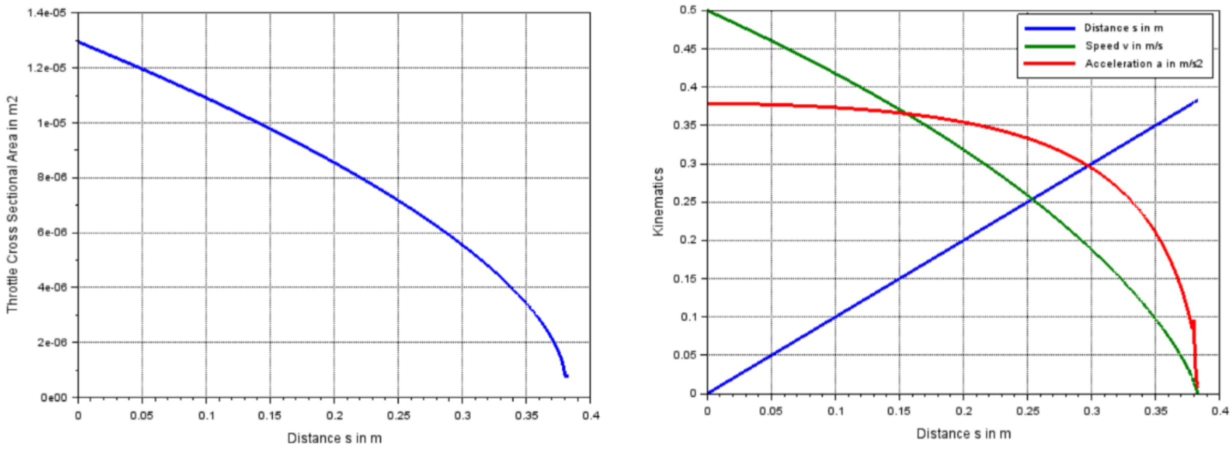


Figure 8 Throttle cross-sectional area, Kinematic values over position (Reference buffer)

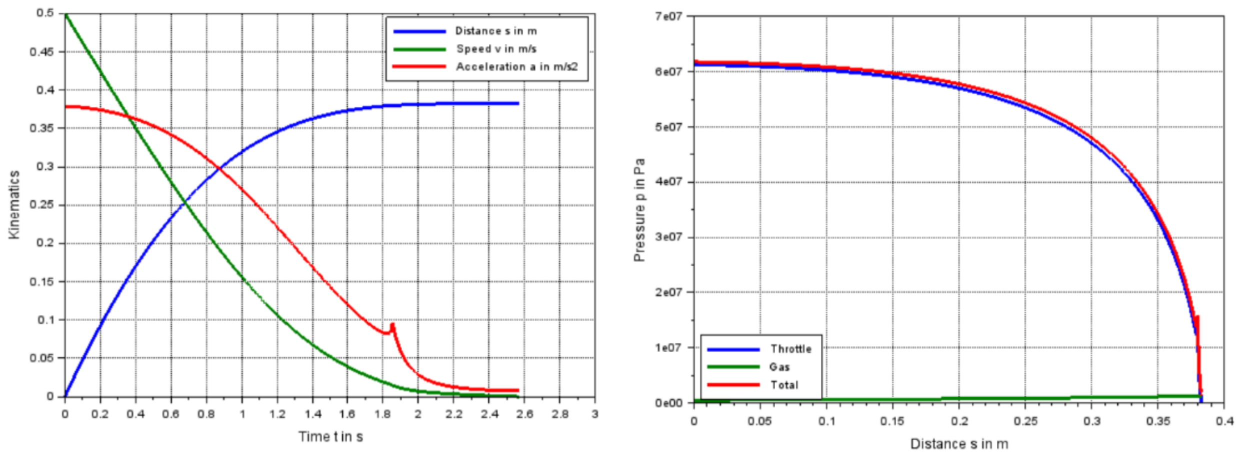


Figure 9 Kinematic values over time, Gas pressure, throttle pressure and total pressure (Reference buffer)

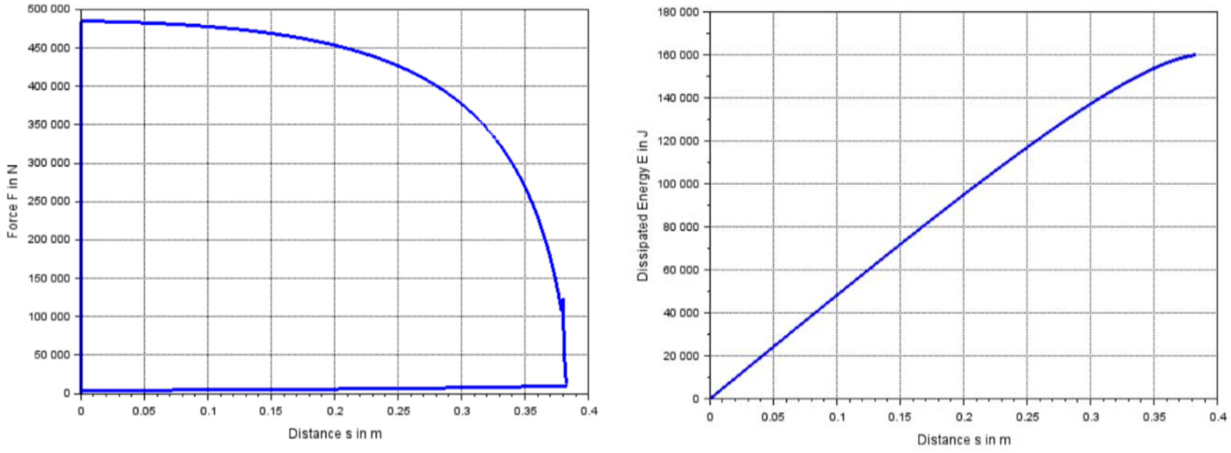


Figure 10 Buffer force, Energy dissipated (Reference buffer)

4 OPTIMIZATION OF BUFFER CHARACTERISTICS

Now two limit states of the hydraulic buffer are considered: the permissible oil pressure due to the sealings installed and the permissible piston rod force due to buckling of the piston rod. A certain maximum oil pressure leads to a constant limit force to the buffer. A maximum piston rod force leads to an inverse quadratic limit force to the buffer. The characteristics of different maximum oil pressures and certain piston rod diameters are shown in Fig. 11 (left).

As it absorbs the same energy as the reference buffer, the total absorbed energy of E = 160,000 J, a buffer with a maximum oil pressure of 400 bar and a piston rod diameter of 30mm is in focus now (Fig. 11, right). Up to piston positions of about x = 260mm, the maximum oil pressure defines the deciding limit state. Above piston positions of about x = 260mm, the maximum piston rod force with regard to buckling defines the deciding limit state. Based on these limit states for the certain buffer design, the maximum energy exploitance can be evaluated. This maximum energy exploitance might be of interest if a small buffer design is required rather than a constant buffer force. This represents a different optimization approach.

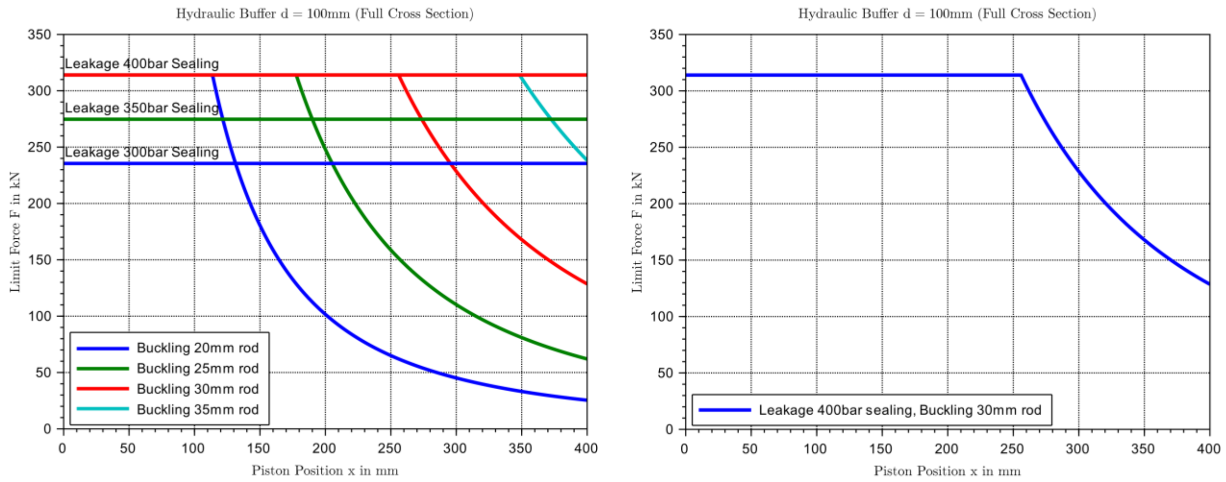
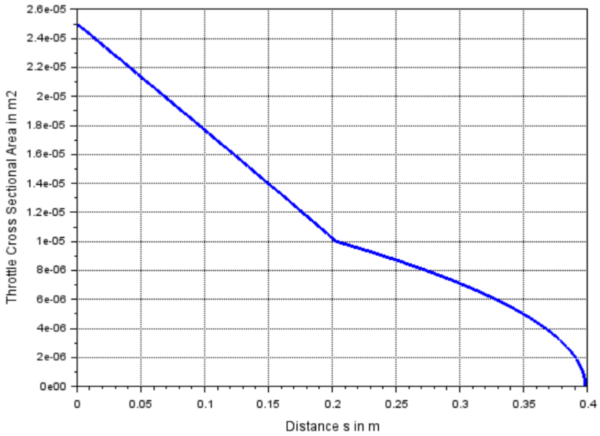


Figure 11 Buffer force limits, Buffer force characteristics intended

The continuous throttle cross section is now modified in order to meet the buffer force characteristics intended. The modified throttle cross-sectional area curve shows a starting value well above the value for the reference buffer. This is caused by the low buffer force limit at the beginning of the buffering process, caused by the buffer buckling force limit. At the passage from buffer force growth to constant buffer force, the throttle cross-sectional area shows a kink (Fig. 12, left).



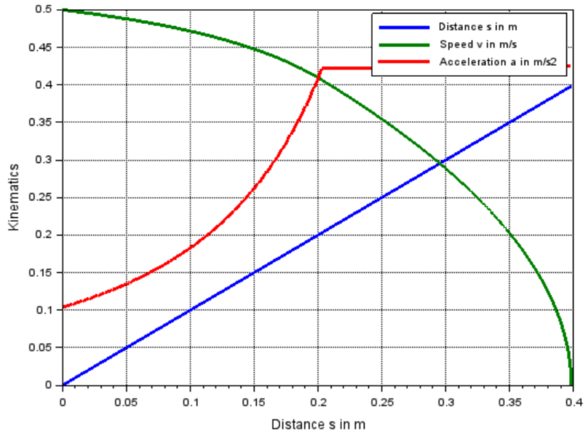
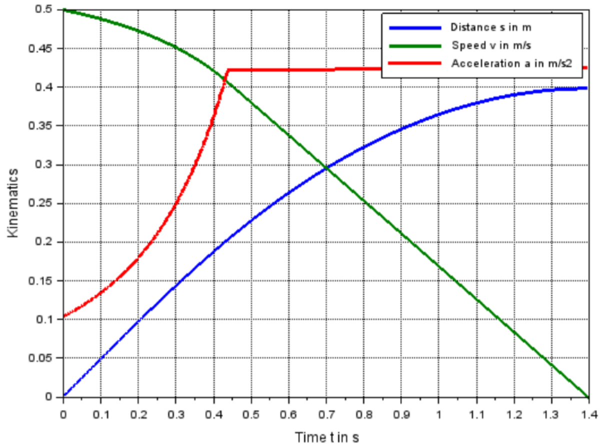


Figure 12 Throttle cross-sectional area, Kinematic values over position (Optimized buffer)



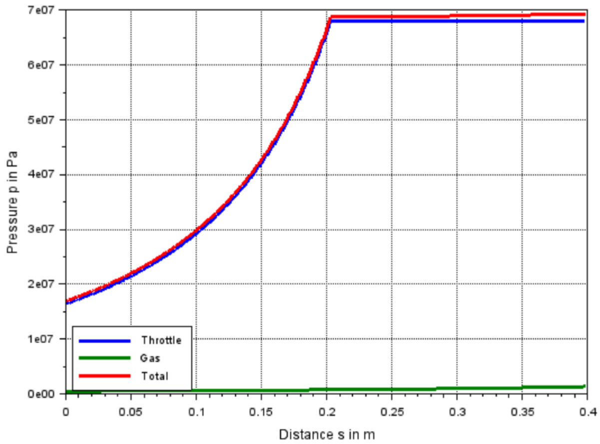
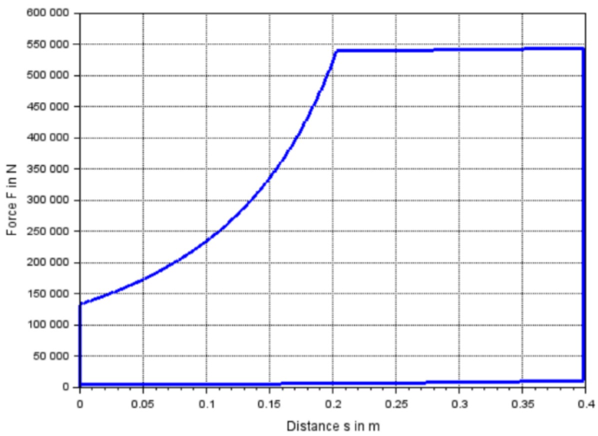


Figure 13 Kinematic values over time, Gas pressure, throttle pressure and total pressure (Optimized buffer)



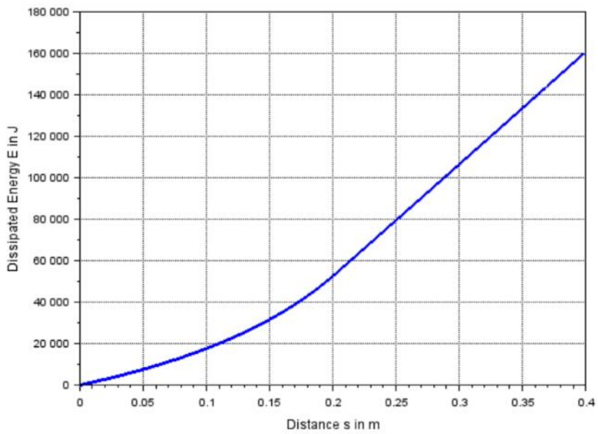


Figure 14 Buffer force, Energy dissipated (Optimized buffer)

The optimized buffer comprises a design suitable to absorb the same energy as the reference buffer (Fig. 14). As the initial throttle cross-sectional area for the optimized buffer is larger than for the reference buffer, the optimized buffer dissipates less energy at the beginning of the stroke and more energy at the end of the stroke (Fig. 15, right).

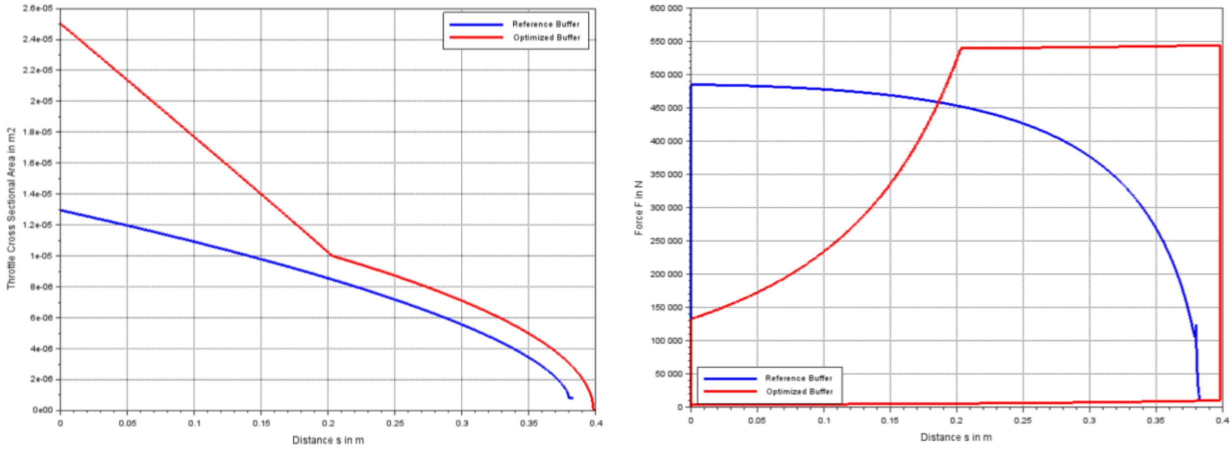


Figure 15 Throttle cross-sectional area, Buffer force (Reference buffer vs. optimized buffer)

As piston forces are quite low, the optimized characteristics give the potential to prevent buckling of the piston rod at the beginning of the stroke. The loss of energy absorbed may be compensated at the end of the stroke. There the buffer force may be larger than at the reference buffer. The buffer force is restricted by the sealings pressure limit e.g., maybe the maximum buffer force can even be lifted at the end of the buffer stroke. This gives the potential for a higher amount of energy to be absorbed by the reference buffer, just equipped with an optimized throttle (Fig. 15).

5 CONCLUSION

Means of calculation give tools to adapt buffer characteristics exactly to given conditions. This provides the possibility to adapt a drive force F_{Drive} changing over stroke; another possibility is the adaption of the different limit states of a buffer. In the first analysis, the optimization of the buffer force against the stroke was shown. These options are suitable to reach an improved exploitation of the buffer capacity. The adaption to further limit states may be found accordingly. An upcoming project will analyze an optimized buffer exploitation under consideration of various buffer limit states as listed in the article.

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