A fluid-controlled buffer device in cylinders having a fluid-operated working piston, comprising a cylindrical buffer piston mounted at the end of the working piston and adapted to enter into a cylindrical fluid outlet bore in an end wall of the cylinder, an annular clearance between the cylindrical surfaces of said buffer piston and said bore being sealed by a sealing-ring mounted with radial play in an annular notch located in either of said cylindrical surfaces the other cylindrical surface being provided with passages which when the buffer piston is moved into said bore initially permit escape of the fluid from the cylinder but then are successively closed by the sealing-ring.
The present invention relates to a fluid-controlled buffer device in cylinders having a fluid-operated working piston, comprising a buffer piston concentrically mounted at the end of the working piston and having a substantially less diameter than the working piston, said buffer piston being adapted to enter and gradually throttle a bore communicating with a fluid outlet opening of the cylinder when the working piston is approaching an end position.

A buffer device of this kind is used for protecting the working cylinders and connected machine elements from high stresses caused by the working piston touching the end wall of the cylinder. A certain damping of the working piston motion in its end position can be obtained in a simple way by closing the outlet opening from the cylinder a predetermined distance before the end position of the working piston and passing the remaining fluid to a fixed or adjustable throttle valve. This kind of damping is not particularly effective because the damping pressure decreases when the velocity of the working piston decreases at a constant throttle area.

An improved damping is obtained by means of a conical buffer body entering the outlet opening and gradually throttling this opening. The buffer body is attached to the end of the working cylinder and, thus, a damping is obtained that is dependent on the position of the working piston. Different inconveniences are, however, involved in this solution. Hence the clearance between the buffer body and the outlet opening must be very small to present a proper throttle area which gives rise to a non-turbulent flow characteristic, and thus, the damping is influenced by the viscosity or temperature of the fluid. Due to the small clearance it is necessary to perform a very precise profile grinding of the surface of the buffer body if a theoretically proper area variation should be obtained. Such a profile grinding cannot be used in practice and, thus, the efficiency of the damping will be low.

An essential inconvenience in connection with the known buffer device is that the throttling varies strongly with respect to the eccentricity of the buffer device and the outlet opening due to the non-turbulent flow conditions. Due to the play between the working piston and the cylinder, involved in the mounting means of the piston and the piston rod, it is impossible to avoid eccentricity and for that reason it is necessary to mount the buffer device displaceable with respect to the outlet opening in order to avoid seizing. Consequently a precise positioning of the buffer device in the outlet opening cannot be obtained, and as the flow rate at non-turbulent conditions varies in proportion 3:1 between maximum eccentricity and maximum concentricity at a constant pressure drop it is obvious that the damping efficiency will be very varying.

In order to reduce the pressure drop across the buffer device at the return stroke a non-return valve function must be present.

The object of the present invention is to obtain a buffer device in which the abovementioned inconveniences are eliminated.

This has been achieved by the buffer device according to the invention which is characterized in that said buffer piston consists of a cylindrical body having a cylindrical surface and said bore has a cylindrical surface having a diameter exceeding the diameter of the cylindrical body so that indenter of the usual radial play between the working piston and the cylinder an annular clearance is formed between the cylindrical surfaces of the cylindrical body and the bore when the cylindrical body is positioned in the bore, in which annular clearance a resilient sealing-ring having a non-deformable cross section area is mounted in an annular notch located in either of the cylindrical surfaces of the body and the bore with a radial play between the sealing-ring and the bottom of the notch permitting radial displacement of the body caused by the radial play between the working piston and the cylinder, said sealing-ring slidingly engaging the other of said cylindrical surfaces, and in that the cylindrical surface slidingly engaging said sealing-ring being provided with passages which when the cylindrical body is moved into the bore initially permit the escape of the fluid from the cylinder to the outlet opening but then are successively closed by the sealing-ring, thus gradually reducing the total effective area of the passages, in addition to which the passages are designed to maintain a turbulent flow of the fluid through the passages. Thus, by proper dimensioning of the passages and their positions, which is easily performed, the theoretically proper variation of the throttle area is obtained, i.e. a high damping efficiency. Moreover, a turbulent characteristic of the throttle area is obtained, i.e. no or small influences from temperature and viscosity variations of the fluid, as well as automatic centering effected by small lateral forces acting upon the buffer-chambers, i.e. no or small influences by eccentricity and a reduced risk of seizing. The sealing-ring can easily be displaced also during the damping operation under the influence of small lateral forces because it can be hydraulically balanced. The improved damping is obtained at reduced demands as to shape exactness due to the fact that the sealing-ring is slidingly engaging the cylindrical body and the bore and that the area variations do not depend on the shape of the buffer piston.

The aims and further advantages of the invention will be explained in the following description with reference to the accompanying drawings, wherein

FIG. 1 shows in section an embodiment of the invention in connection with a hydraulic cylinder, and

FIG. 2 shows partly in section a further embodiment of the invention.

FIG. 1 shows an end portion of a hydraulic cylinder 1 having an end wall 2. The end wall 2 is provided with a concentric bore 3 communicating with a fluid outlet opening 4. A working piston 6 attached to a piston rod 5 is slidingly mounted in the cylinder 1 and slidingly engaging the walls of the cylinder 1 by means of an elastic sealing-ring 7. At the end wall of the piston 6 facing the end wall 2 a cup-shaped sleeve 8 is fixedly mounted by means of a bottom pin 9 engaging a recess 10 in the end of the piston rod 5. The outer diameter of the sleeve 8 is smaller than the diameter of the cylindrical bore 3 and the sleeve 8 is perforated by radial holes 11. The clearance between bore 3 and sleeve 8 is adapted with respect to manufacturing tolerances and foreseen or permissible wear of the journaling means of the working piston and its piston rod 5 to prevent touching of the sleeve 8 against the bore 3. This clearance is sealed by a slotted sealing-ring 12 slidingly engaging the sleeve 8 and mounted in an annular notch 13 in bore 3 with a play corresponding to the clearance between sleeve 8 and bore 3. The axial length of the notch 13 exceeds the axial dimension of the sealing-ring 12, and a number of passages 14 connect the bottom of the notch 13 and the cylinder chamber 15 in front of the working piston 6.

When the piston 6 is moved towards the end wall 2 and before the sleeve has reached the bore 3 the fluid in chamber 15 can freely flow out through the bore 3 and the opening 4. When the end of the sleeve 8, which is rounded, reaches the sealing-ring 12 whose edge facing the sleeve is bevelled, the sealing-ring is centered on the end portion of the sleeve 8 after which the sleeve is entering the sealing-ring. The fluid now has to pass through all the passages 11 having a total area adapted so that the maximum permissible pressure is obtained in the cylinder chamber 15.

In order to reduce the pressure acting upon the sealing-ring 12 when initially engaging the end of the sleeve 8 the end portion of the sleeve is provided with recesses 16 and intermediate sections 17. The sections 17 guide the sealing-ring 12 when being centered upon the sleeve at the same time as the fluid can flow through the recesses 16.

When the retarding piston 6 moves towards the wall 2 the area of the passages 11 is gradually decreased until all passages are disconnected or closed, in which case the fluid can pass only through the slot of the sealing-ring.
At the return stroke the sealing-ring 12 is lifted by the sleeve 8 so that the fluid can pass freely through the passages 14 to chamber 15, thus permitting piston 6 to start the return stroke with high velocity.

Because the axial measure of the sealing-ring 12 is small, turbulent flow conditions can be obtained by longitudinal grooves of different lengths in the surface of sleeve 8 replacing the holes 11. The length, width and depth dimensions of the grooves can easily be adapted to a predetermined area variation.

Similar grooves 11 can alternatively be arranged in the surface of the bore 3, as shown in FIG. 2, in which case the sealing-ring 12 is mounted in an annular notch on the buffer piston 8 and also the passages 14 are formed in the surface of the buffer piston 8.

In order to reduce the damping distance as much as possible it is suitable to maintain a constant damping pressure, which can be obtained by a by-pass area through passages 11 defined as below:

$$A_{d_x} = A_{d_{x=0}} \sqrt{1 - \frac{x}{1_{r_{c-o}}}}$$

where
- $x$ = the damping position
- $A_{d_{x=0}}$ = constant
- $1_{r_{c-o}}$ = the total damping distance

Thus, the area variation is defined by a square root which corresponds to a buffer piston of known type having a concave shape instead of the usual conical shape which explains the bad damping characteristics of known buffer devices. The theoretically proper by-pass area can easily be obtained by proper dimensioning of the passages 11 of the buffer device according to the invention in hydraulic as well as pneumatic cylinders. We claim:

1. A fluid-controlled buffer device in cylinders having a fluid-operated working piston, comprising a buffer piston concentrically mounted at one end of the working piston and having a substantially smaller diameter than the diameter of working piston, a bore communicating with a fluid outlet at one end of said cylinder, said buffer piston being adapted to enter and gradually throttle said bore communicating with said fluid outlet upon the working piston approaching said end of the cylinder, said buffer piston comprising a cylindrical body, said bore having a diameter larger than the diameter of the cylindrical body as to form an annular clearance between the peripheral surfaces of the body and the bore upon the cylindrical body being positioned within the bore, a flexible slotted sealing ring having a non-deformable cross-sectional area being mounted in an annular notch located in at least one of said peripheral surfaces and extending into said annular clearance, said notch having a diameter facilitating radial play between the sealing ring and the peripheral surface of the notch to provide for radial displacement of the cylindrical body, said sealing ring slidingly engaging the other of said peripheral surfaces, axial recesses in the peripheral surface slidingly engaging said sealing ring at the end adapted to be initially engaged by the sealing ring for effecting a pressure reduction on the sealing ring until the end portion of said peripheral surface enters the sealing ring, said peripheral surface further including passages adapted to, upon the cylindrical body being moved into the bore, initially permit the flow of fluid from the cylinder to the outlet opening, said passages being adapted to be successively closed by axial movement relative to the sealing ring to thereby gradually reduce the total effective area of the passages while concurrently maintaining a turbulent flow of the fluid through the passages.

2. A buffer device as claimed in claim 1, wherein said annular notch is formed in the peripheral surface of said cylindrical body.

3. A buffer device as claimed in claim 1, wherein said annular notch is formed in the peripheral surface of said bore.

4. A buffer device as claimed in claim 1, wherein the passages comprises longitudinal grooves having varied lengths formed in the surface slidingly engaging the sealing ring, said longitudinal grooves each having length, width and depth dimensions corresponding to a predetermined flow area variation.

5. A buffer device as claimed in claim 1, wherein the cylindrical body is essentially cup-shaped, the bottom surface portion of said body being attached to the end of the working piston, and a plurality of holes extending through the cylindrical wall portion of said body.

6. A buffer device as claimed in claim 1, wherein the interior of the cylinder encompassing said cylindrical body communicates with the annular notch through at least one passage extending into the peripheral surface of the annular notch.

7. A buffer device as claimed in claim 1, wherein the annular notch is dimensioned to provide for axial play between the sealing ring and the annular notch.

8. A buffer device as claimed in claim 6, wherein the annular notch is dimensioned to provide for axial play between the sealing ring and the annular notch.

9. A buffer device as claimed in claim 1, wherein the passages in the surface slidingly engaging the sealing ring form a fluid-flow by-pass area $A_{d_x}$ varying in response to the stroke of the working piston as defined by:

$$A_{d_x} = A_{d_{x=0}} \sqrt{1 - \frac{x}{1_{r_{c-o}}}}$$

where
- $x$ = the damping position
- $A_{d_{x=0}}$ = constant
- $1_{r_{c-o}}$ = the total damping distance