Analysis and design of cushioning in hydraulic cylinders

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Abstract

Hydraulic cylinders, when used to drive heavy loads at high velocities, need to be decelerated or cushioned at the ends of the stroke in order to prevent damage. In this paper, one of the methods of cushioning, namely, the annular clearance method of cushioning is analysed and based on the results of the analysis, a suitable procedure for the design of the cylinder is suggested. An example is worked out to illustrate the design procedure.

Key words : Hydraulic cylinders, cushioning.

1. Introduction

Hydraulic cylinders are hydrostatic devices which work from a pressurized oil supply and utilize the potential energy of the oil for exerting mechanical force and causing linear motion. Hydraulic cylinders provide excellent control over velocity (from zero to maximum), displacement of the load and smooth and instant reversals without damage to the equipment. Hydraulic cylinders are very versatile because they can be designed to exert a gentle force of a few grams to a brute force of several thousand kilograms. Although cylinders produce only a linear motion, they have numerous applications where operations such as pushing, pulling, raising, lowering, etc., are required. Further, the application of cylinders can be extended[1] by the use of various kinematic linkages which convert linear motion into limited rotary or oscillatory motion.

A typical hydraulic cylinder consists of a piston with a rod fitted inside a cylindrical bore as shown in Fig. 1. The piston rod is connected to the load to be moved. The load can be moved in either direction by supplying the cylinder with pressurized oil either at port A or at port B shown in Fig. 1.

2. Need for cushioning

Hydraulic cylinders are quite satisfactory when used to drive small loads at low speeds. But, when an ordinary cylinder is used to drive a heavy load at a high velocity, there

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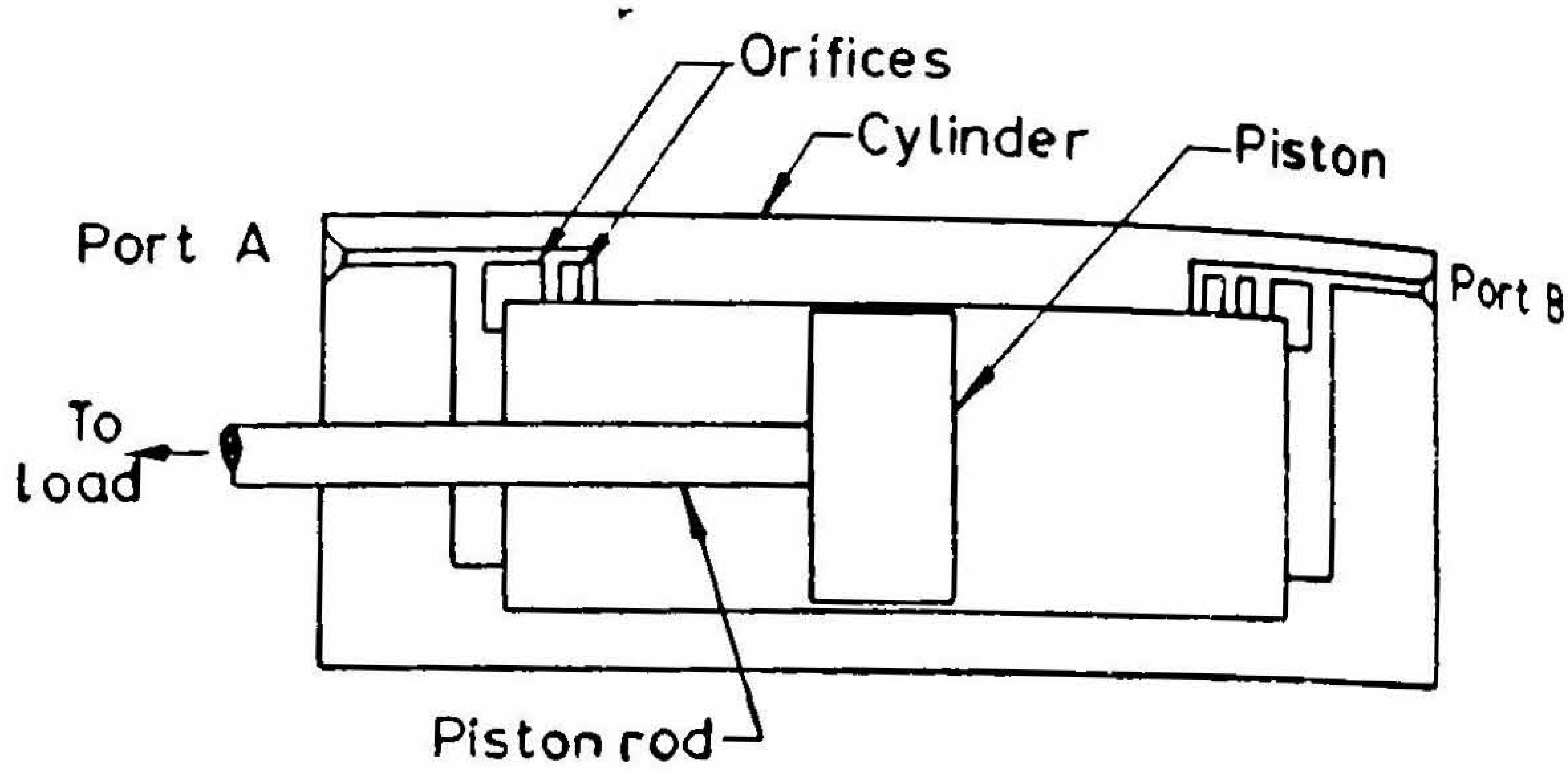

FIG. 1. Hydraulic cylinder with orifice method of cushioning.

is a danger of the piston hitting the cylinder heads at the ends of the stroke with a large force thereby inducing high stresses in the cylinder body and causing noise. In some extreme situations, the kinetic energy of the load may be large enough to pull the piston off the cylinder thereby destroying the cylinder. In order to prevent such an event, it is necessary to decelerate the load and the piston towards the ends of the stroke. The process of achieving controlled deceleration of the load towards the ends of the stroke is called cushioning. By proper cushioning, it is possible to have a fast power stroke with controlled deceleration at the ends of the stroke and to reduce the end-of-stroke forces that are damaging to the cylinder.

3. Methods of cushioning

The basic idea underlying different methods of cushioning is to restrict the oil flow out of the cylinder towards the ends of the stroke thereby inducing a back pressure and consequently producing a retarding force which results in decelerating the load. The three important methods of cushioning are: (*a*) Deceleration valve method, (*b*) Orifice method, and (*c*) Annular clearance method. Of these, the first method achieves cushioning by restricting the oil flow out of the cylinder by means of a valve called the deceleration valve which is operated towards the end of the stroke by a cam from the piston rod. The other two methods decelerate the load towards the ends of the stroke by making use of the piston movement itself to create a restriction to the flow out of the cylinder. In the orifice method[2] of cushioning, the cylinder ports are designed as shown in Fig. 1 with a number of orifices which can be covered and uncovered by the movement of the piston. It is clear from this figure that the piston as it approaches the ends of the stroke closes the orifices one after another thereby decreasing the flow out of the cylinder and consequently producing a cushioning action.

In the annular clearance method[2] of cushioning, the cylinder heads and the piston are designed as shown in Fig. 2. As the piston approaches either end of the stroke, the protruding cylindrical cushioning nose on the piston enters the matching cylindrical cavity in the head thereby metering the flow through the annular clearance between the nose and the cavity and consequently providing a braking force. The deceleration of the load at the ends of the stroke can be controlled by adjusting the by-pass flow through the needle valve shown in Fig. 2. The check valves, which permit unobstructed flow into the cylinder, allow the oil to enter the cylinder freely for the start of the return stroke.

In this paper, equations pertaining to the annular clearance method of cushioning are obtained, and suitably nondimensionalized and analyzed so that the results are universally applicable to this type of cushioning. Based on the results of this analysis, a design procedure applicable to the annular clearance method of cushioning is proposed. The suggested design procedure is illustrated with an example.

4. Analysis of the annular clearance method of cushioning

Consider a cylinder with the annular clearance method of cushioning driving a heavy inertia load. Assume that the load is being driven at a constant velocity before the cushioning nose enters the cavity. It is necessary to derive the equation of motion of the load after the nose enters the cavity. A situation when the nose is partially inside the cavity is shown in Fig. 3 wherein it is assumed that the nose and the cavity are concentric and that the by-pass needle valve shown in Fig. 2 is completely closed for maximum cushioning effect. Then the flow out of the cylinder is through the annular clearance between the nose and the cavity. Regarding the annular flow, it is assumed[3] that (i) the flow is parallel to the axis of the cylinder, (ii) the peripheral component of the flow is negligible, (iii) fully developed laminar flow is present throughout the length of the flow path, (iv) pressure drop at entry is negligible and (v) compressibility effects and leakage past the piston are negligible. With these assumptions, the flow through the

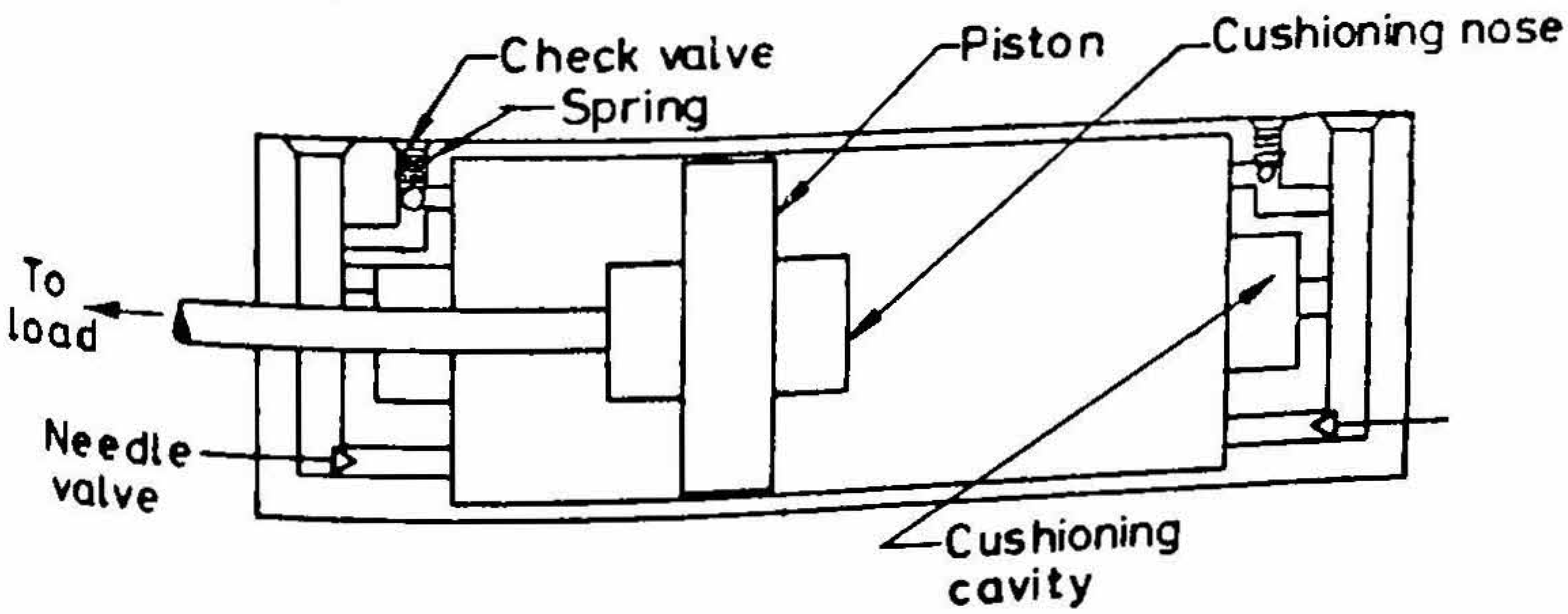

FIG. 2. Hydraulic cylinder with annular clearance method of cushioning.

annular clearance can be considered as being laminar and two-dimensional. Further, in a realistic situation, it is reasonable to assume that the clearance c between the nose and the cavity is small compared with the diameter d of the nose. Hence the annular flow can be essentially regarded as that between two parallel plates, one of which is moving and the other stationary. Therefore, the volumetric annular flow rate Q can be given[3] by

$$Q = \pi d \left[-\frac{c^3 (P_1 - P_2)}{12 \mu Z} + \frac{c}{2} \dot{Z} \right] \tag{1}$$

where P_1 and P_2 represent the oil pressures as shown in Fig. 3, μ is the viscosity of oil, $Z(t)$ represents the extent by which the leading edge of the nose has advanced into the cavity at any time t and the dot over the letter Z indicates differentiation with respect to time t.

It is also easy to see that the volume of oil displaced per second by the piston must be equal to the outflow per second from the cylinder. Thus

$$A_1 \dot{Z} = Q \tag{2}$$

where A_1 is the difference between the area of the piston and that of the nose. Considering the motion of the load, force balance requires

$$M\ddot{Z} + B\dot{Z} = (A_2 P_s - A_1 P_1) \tag{3}$$

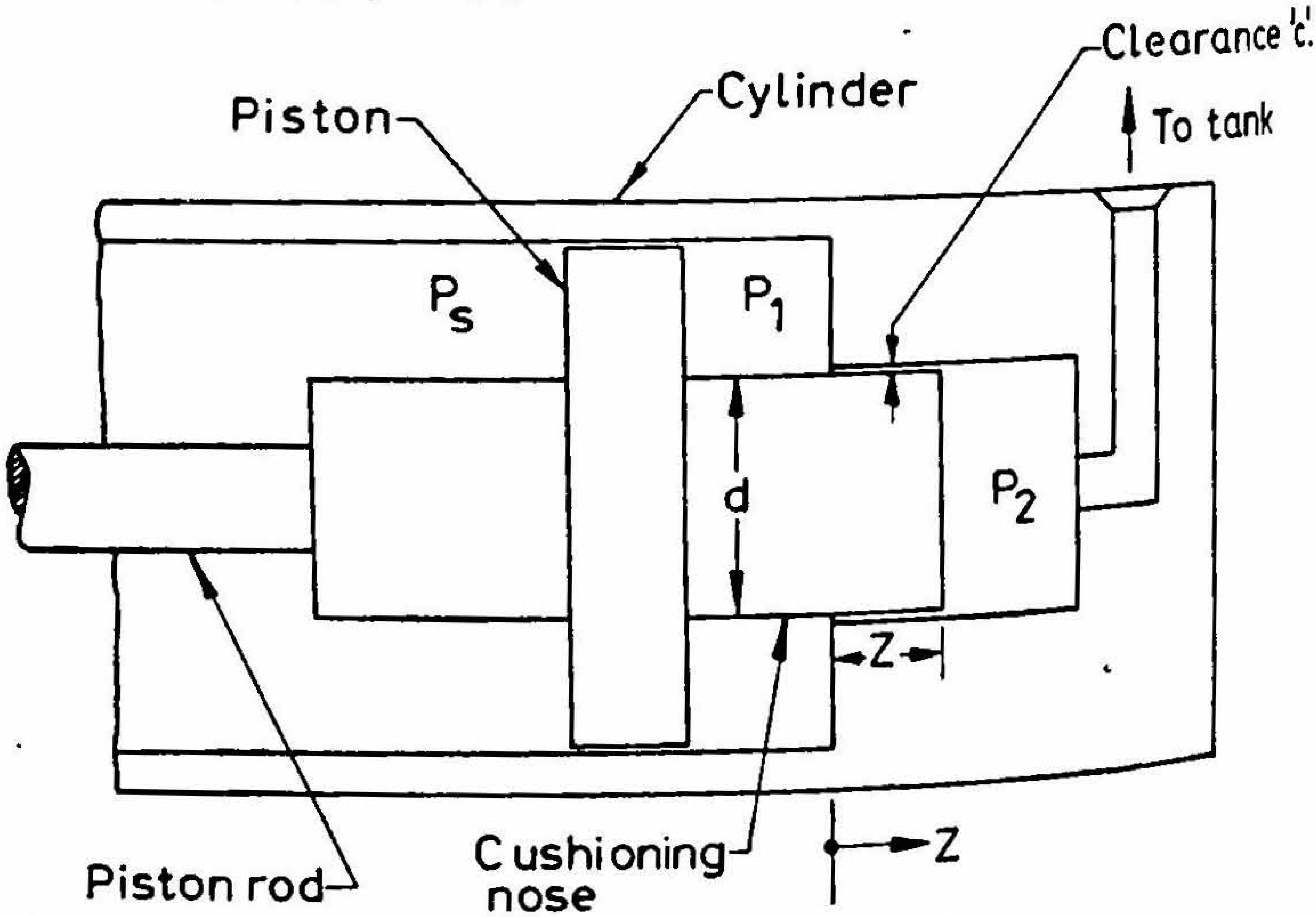

FIG. 3. Cushioning arrangement in the annular clearance method.

where M is the load mass, B the viscous damping coefficient, A_2 the difference between the areas of the piston and the piston rod, and P_s the supply pressure acting on the left side of the piston in Fig. 3. In a practical situation, the coefficient B is difficult to determine because it varies with the cushioning distance. However, in order to reduce the complexity of the analysis, it is assumed here that a mean value of B estimated from experience is used in eqn. (3).

Assuming $P_2 = 0$ and the supply pressure P_s to be a constant, eqns. (1), (2) and (3) can be solved to give the equation of motion of the load as

$$M\ddot{Z} + B\dot{Z} + (12\mu A_1/\pi dc^3)\ (A_1 - 0{\cdot}5\pi dc)\, Z\dot{Z} = A_2 P_s.$$

In the above equation, the term $0{\cdot}5\pi dc$ is very nearly equal to the annular area through which flow occurs and is very small compared with the area A_1. Hence, neglecting this term, the equation of motion becomes

$$M\ddot{Z} + B\dot{Z} + (12\,\mu A^2{}_1/\pi dc^3)\, Z\dot{Z} = A_2 P_s. \tag{4}$$

The above equation is nonlinear and no closed form solution appears possible though the equation can be integrated once.

In order to make a complete analysis of the problem, it is necessary to solve eqn. (4) for various combinations of values of the parameters. This is clearly a formidable task because there are six parameters. In order to reduce the complexity of the analysis and also to make the results of the analysis universally applicable to the situations of the type considered, it is necessary to properly group the parameters into some nondimensional quantities. Identification of the relevant nondimensional quantities can be achieved by making a dimensional analysis of eqn. (4). Such an analysis suggests the introduction of nondimensional time τ and nondimensional distance X defined by

$$\tau = (B/M)\,t, \tag{5}$$

$$X = (\mu/B\sigma)\,Z \tag{6}$$

where

$$\sigma = (dc^3/A_1^2). \tag{7}$$

Nondimensionalization of eqn. (4) using the above quantities yields

$$X'' + X' + (12/\pi)\,XX' = U \tag{8}$$

where the primes represent differentiation with respect to τ and the quantity U is given by

$$U = (\mu A_2 P_s M/B^3\sigma).$$

Thus it is clear that nondimensionalization of eqn. (4) has resulted in a much simpler eqn. (8) containing only one nondimensional parameter U.

Earlier, it was assumed that the piston was moving with a constant velocity before the entry of the cushioning nose into the cavity. The magnitude of this velocity $\dot{Z}(0)$ of the piston when the nose just enters the cavity is given by (A_2P_s/B). Equivalently, the nondimensional velocity $X'(0)$ when the nose just enters the cavity is given by

$$X'(0) = (\mu M/B^2\sigma)\,\dot{Z}(0)$$

$$= (\mu M/B^2\sigma)\ (A_2P_S/B) = U. \tag{9}$$

Thus, it is seen that the parameter U is equal to the nondimensional entry velocity of the nose into the cavity.

An equation giving the variation of the retarding force A_1P_1 can be obtained by nondimensionalizing eqn. (3) to yield

$$F = (A_1P_1/A_2P_S) = 1 - (X' + X'')\,U^{-1}. \tag{10}$$

The above equation also gives an idea about the variation of the pressure P_1.

Equation (8), being nonlinear, was solved on a computer for various values of U and the results graphically shown in Fig. 4 by curves of X' vs. X and X'' vs. X. This form of presentation of results was chosen because it directly shows the variation of the load velocity and acceleration as the cushioning nose advances into the cavity and also readily gives the nondimensional length of the cushioning nose required to bring down the load velocity to any given fraction of the initial velocity.

The variation of the normalized retarding force F given in eqn. (10) as the cushioning nose advances into the cavity is shown in Fig. 5. This plot gives an idea about the peak transient pressure during cushioning for a given ratio of A_2 to A_1.

Observation of Fig. 4 reveals that the rate of fall of load velocity is very small after it has reduced to about 10% of its initial value and any further drop in velocity is necessarily accompanied by a larger distance of travel of the load.

Define

$$X_1 = (\mu/B\sigma)\,Z_1 \tag{11}$$

and

$$X''_{\min} = (\mu M^2/B^3\sigma)\,\ddot{Z}_{\min} \tag{12}$$

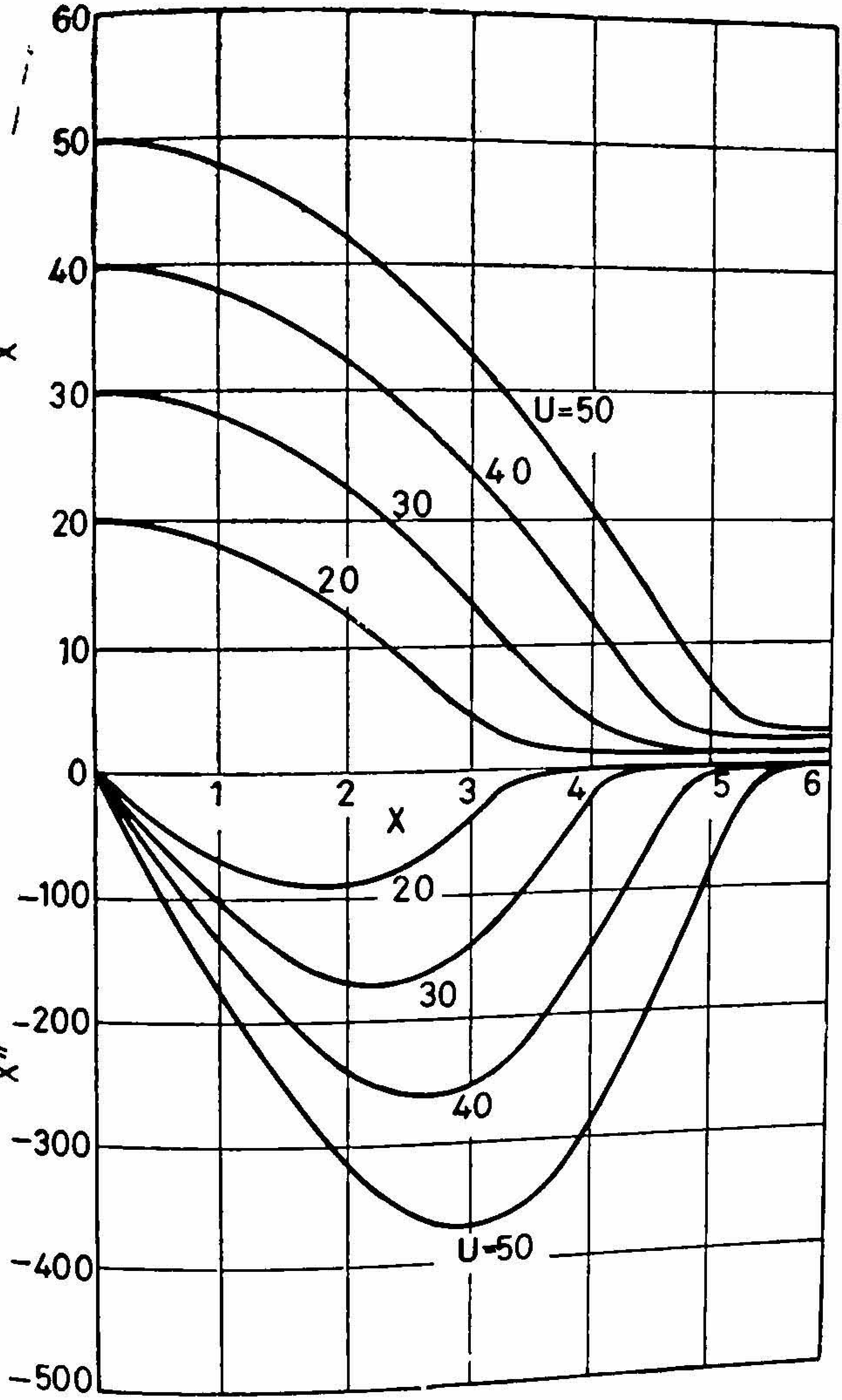

FIG. 4. Variation of load velocity and acceleration as the cushioning nose advances into the cavity.

where Z_1 is the distance at which the velocity $\dot{Z}$ is equal to one-tenth of the actual entry velocity $\dot{Z}$ (0) and $\ddot{Z}_{min}$ is the actual minimum acceleration (maximum deceleration) experienced by the load. Let F_{max} be the maximum value attained by the normalized retarding force F during cushioning. The values of X_1 and X''_{min} corresponding

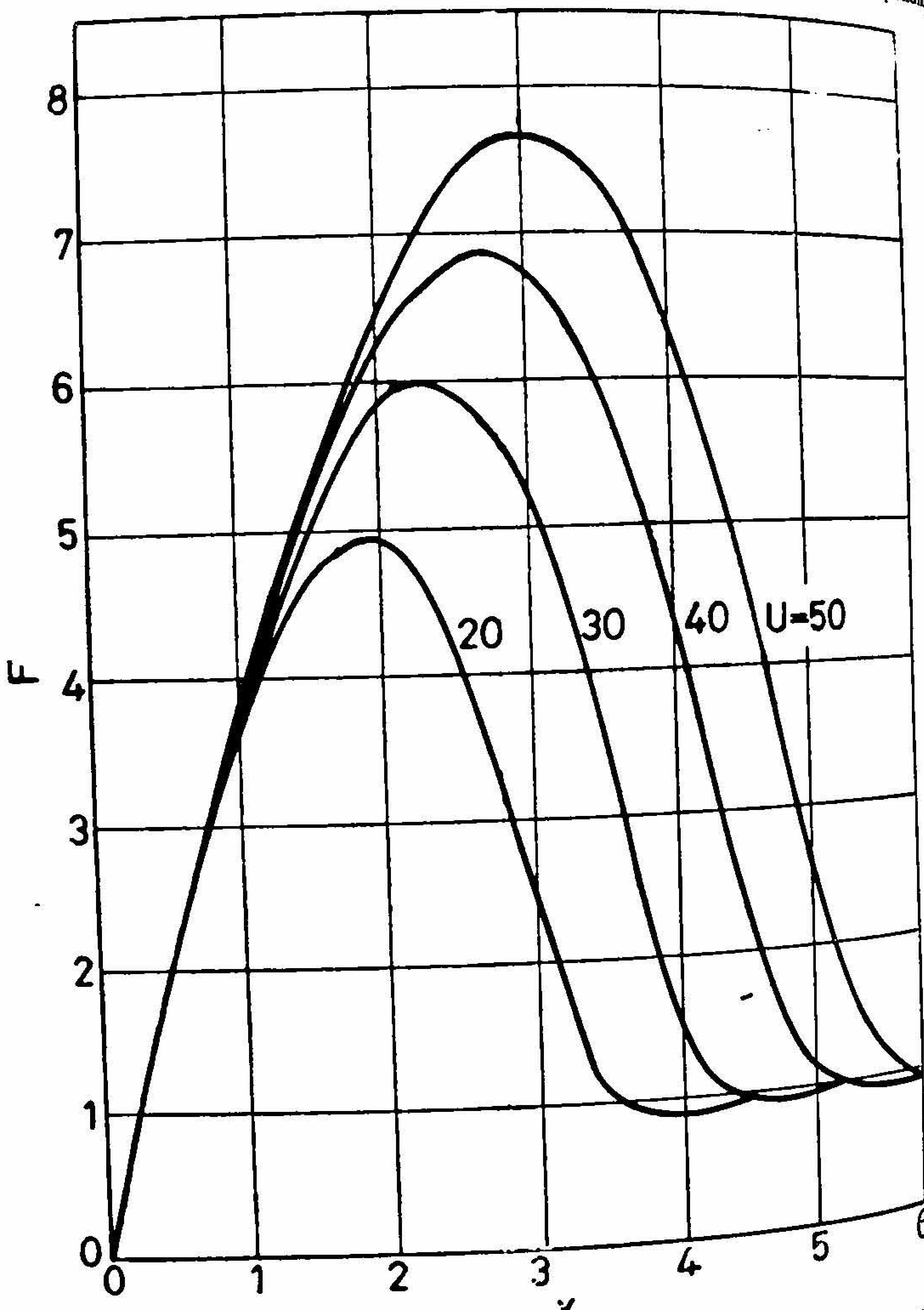

FIG. 5. Variation of the normalized retarding force as the cushioning nose advances into the cavity.

to different values of U read off from Fig. 4 and those of F_{max} read off from Fig. 5 are graphically depicted in Fig. 6 against the entry velocity U. This figure will be made use of in the design procedure.

5. Design problem

It is assumed that the following data are either given or estimated: (*a*) load mass M, (*b*) viscous frictional coefficient B, (*c*) viscosity of the oil μ, (*d*) supply pressure P_s or flow rate Q_S and (*e*) the steady load velocity $\dot{Z}(0)$ from which the load has to be decelerated and reversed. For reasons mentioned earlier, the coefficient B cannot be accurately specified. However, it is assumed here that an estimated mean value of B is given. It is also possible that the situation may put an upper bound on the permissible load deceleration and/or on the peak transient pressure during cushioning.

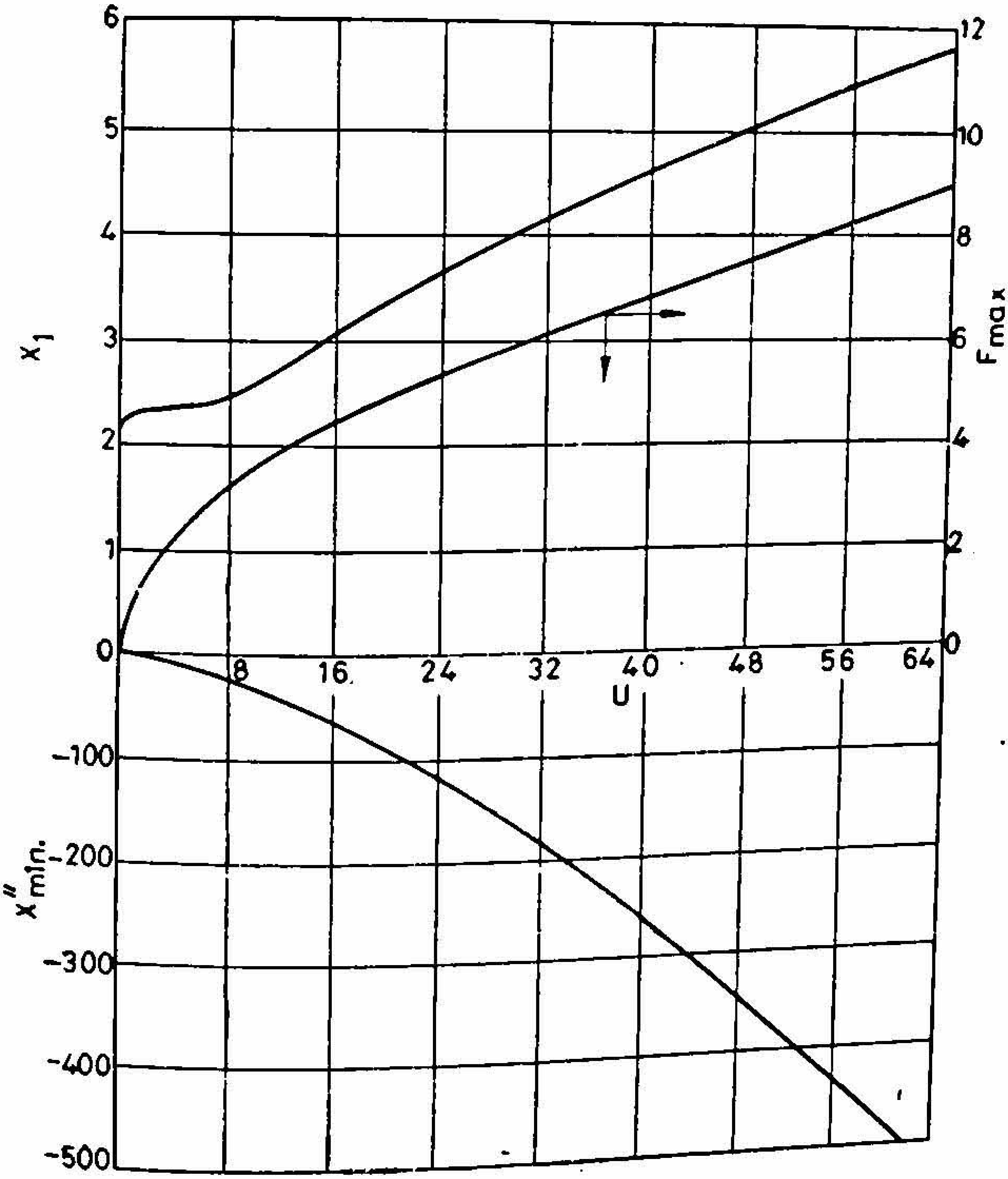

FIG. 6. Graphs of X_1, X''_{min} and F_{max} against U.

The design problem then involves determination of the dimensions of the cylinder including the cushioning arrangement.

It can be observed from Fig. 4 that it takes infinite time and infinite length of the cushioning nose to bring the load absolutely to rest. It is therefore necessary to specify the tolerable impact at the end of the stroke. Since the load deceleration has been observed to be very small after the load velocity reaches about 10% of its initial value, it appears that a reasonable specification in this respect would be to require that the impact at the end of the stroke be not more than that corresponding to a load velocity equal to one-tenth of that at entry. In other words, it is equivalent to requiring that the cushioning nose be long enough so that by the time it reaches the end of the cavity, the load velocity be reduced to one-tenth of the entry velocity. This is also equivalent to requiring that 99% of the kinetic energy of the load at entry be dissipated by the time the cushioning nose reaches the end of the cavity. The remaining 1% of the kinetic energy can be expected to be absorbed on impact at the end of the stroke bringing the load to rest before reversal. Though any limit other than 10% of the entry velocity may be specified, the 10% limit is assumed in this paper.

Putting a bound on peak transient pressure during cushioning is equivalent to specifying the maximum permissible value of the normalized transient pressure (P_1/P_s). Let the maximum permissible value of (P_1/P_S) be α. This bound on (P_1/P_s), through eqn. (10), requires that

$$(A_2/A_1) \leqslant (a/F_{\max}). \tag{13}$$

The above bound on (A_2/A_1) is meaningful only when $F_{\max} < a$ since $A_1 < A_2$. Therefore, in the design procedure, it is only necessary to consider values of U for which $F_{\max} < a$.

6. Design procedure

Substituting the given data in eqn. (9), obtain the product σU. For an assumed value of σ, calculate U, read the corresponding values of X_1, $X''_{\min}$ and $F_{\max}$ from Fig. 6 and determine Z_1, $Z_{\min}$ and the bound on (A_2/A_1) using eqns. (11), (12) and (13). Repeat this process for several assumed values of σ and arrange the results in a table. Having regard to the bounds on permissible load deceleration and/or normalized transient pressure during cushioning that might have been imposed, select a suitable value of σ from this table. The required length of the cushioning nose is then given by the value of Z_1 corresponding to the chosen value of σ. Calculate the dimensions of the cylinder including the cushioning arrangement using the equation

$$\dot{Z}(0) = (A_2 P_S/B) = Q_S/A_2$$

and eqn. (7) while observing the bound on (A_2/A_1) for the selected value of σ. The preceding step requires suitable choices to be made for two of the parameters because there are only two equations and one inequality to determine four parameters. Hence the method still gives freedom to select two of the parameters based on other considerations (like the selection of piston rod diameter based on strength consideration). It may be mentioned here that the clearance c is of the order of a small fraction of a millimeter, the nose diameter d is of the order of a few centimeters. Thus the approximate range of σ is from 10^{-16} to 10^{-8}. This design procedure is illustrated with an example in the following section.

7. Design example

Let the data be : $M = 10^5$ kg, $B = 2 \times 10^5$ kg/sec., $\mu = 93 \times 10^{-4}$ N. sec/m^2, $Q_s = 750$ cm^3/sec and $\dot{Z}(0) = 50$ cm/sec. Let the bounds on permissible load deceleration and peak transient pressure be respectively 350 cm/sec^2 and 6 times the supply pressure. Then

$$\sigma U = \mu M \dot{Z}(0)/B^2$$
$$= 1 \cdot 1625 \times 10^{-8}.$$

The accompanying table can be prepared as given in the preceding section.

Since the load deceleration is limited to 350 cm/sec^2 and since the ratio (A_2/A_1) should necessarily be greater than unity, any one of the combinations of σ and Z_1 given in the last three rows of Table I can be selected. But the combination corresponding to

Table I

Sl. No.	σ	U	X_1	$\lvert X''_{min} \rvert$	F_{max}	Z_1 (cm)	$\lvert \ddot{Z}_{min} \rvert$ (cm/sec^2)	Bound on (A_2/A_1) $= (a/F_{max})$
1.	$2 \cdot 076 \times 10^{-10}$	56	5·45	430	8·2	2·43	770	0·73
2.	$3 \cdot 632 \times 10^{-10}$	32	4·15	190	6·2	3·25	592	0·97
3.	10^{-9}	11·625	2·70	40	3·9	5·8	344	1·54
4.	$1 \cdot 453 \times 10^{-9}$	8	2·45	22·5	3·2	7·65	281	1·88
5.	$2 \cdot 906 \times 10^{-9}$	4	2·35	10	2·3	14·7	250	2·6

the third row of the table will be very nearly the optimum because any higher value of σ gives a longer cushioning nose while any lower value of σ results in exceeding the bound on load deceleration. Thus selecting $\sigma = 10^{-9}$, a possible choice for the dimensions of the cylinder is:

(*a*) Piston diameter = 5 cm

(*b*) Piston rod diameter = 2·42 cm

(*c*) Cushioning nose diameter = 3·5 cm

(*d*) Clearance = 0·003057 cm

(*e*) Length of the cushioning nose = 5·8 cm.

The above set of dimensions for the cylinder results in $(A_2/A_1) = 1{\cdot}5$.

8. Conclusions

This paper has analyzed the annular clearance method of cushioning employed in hydraulic cylinders and evolved a suitable design procedure. An example has been worked out to illustrate the proposed design procedure.

Analysis of a cylinder with a tapered or parabolic cushioning nose can be suggested as an extension to the work presented in this paper. Understandably, the analysis of such a situation can be expected to be more complicated.

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