BNL-NUREG-34546

CONF-840647--13

BNL-NUREG--34546

DE84 010281

# Parametric Studies on the Load-Deflection Characteristics of Hydraulic Snubbers

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This work was performed under the auspices of the U.S. Nuclear Regulatory Commission, Washington, DC.

## ABSTRACT

Hydraulic snubbers are extensively used in the nuclear power industry for supporting high energy piping systems subjected to dynamic loadings. These devices allow the piping system to displace freely under slowly applied loads, but lock up under sudden excitations. This paper presents the governing differential equations describing the hydro-mechanical mechanisms of a typical snubber. A finite difference computer code, "SNUBER", was developed to solve these equations. Using the code, the load deflection characteristics of the unit were developed for a range of parameters of interest. The parameters included leakage orifice area, initial piston location, eyebolt clearance and reservoir pressures. The results include the load deflection characteristics for various combinations of the controlling parameters and some chamber pressure time history profiles. It is intended that the non-linear characteristic of the snubbers be incorporated into a structural dynamic analysis program to allow prediction of the overall response of nuclear piping supported by these devices and subjected to a variety of loadings.

## NOMENCLATURE

- $t$ = time
- $P$ = pressure
- $B$ = effective bulk modulus
- $V$ = instantaneous chamber volume
- $F$ = volumetric flow
- $A$ = piston area
- $q_r$ = relative displacement of the ends
- $A_D$ = duct area
- $C_D$ = flow coefficient
- $\rho$ = fluid density
- $A_o$ = total orifice area
- $D_R$ = diameter of reservoir duct
- $\mu$ = absolute viscosity
- $D_P$ = piston outside diameter
- $C_r$ = clearance between piston and bore
- $H$ = length
- $Q$ = force resultant

$L_s$ = total snubber length
$H_o$ = initial piston position
$B_L$ = fluid bulk modulus
$B_c$ = container bulk modulus
D = bore diameter
$D_o$ = outer vessel diameter
E = vessel modulus of elasticity
$\nu$ = vessel Poissons ratio
$C_{ri}$ = initial clearance between piston and bore
$A_1$ = area of orifice which remains open
$A_2$ = area of orifice which closes
q = end displacement

Subscripts:

T = tension
C = compression
R = reservoir
P = piston
J = joint
i = i-th end
j = j-th end
r = relative

## INTRODUCTION

For years snubbers have been used in nuclear power plants to limit the adverse response of piping and equipment produced by dynamic loads. Snubbers are generally referred to as shock arrestors and shock suppressors. They are attached to the piping with their major axis in the direction of the maximum thermal growth and dynamic force. During normal thermal cycles these devices extend and retract with the pipe and thus induce very little reactive force on the system. When the pipe is subjected to an earthquake, pipe break, water hammer or relief valve actuation etc., these devices lock up and limit the displacement response.

There are two types of snubbers currently available: hydraulic or mechanical. Hydraulic snubbers function on the principle of the dashpot where piston movement in a hydraulic cylinder is controlled by the flow of fluid through a small orifice. The higher the force the faster the fluid passes through the opening. Since the orifice size limits the flow of the fluid, the piston rod movement is controlled by the orifice size. Other arrangements associated with the snubber design improve the dashpot effect and better control the motion of the piston rod. Due to the compressability of the fluid, these devices absorb energy under dynamic loading. One problem associated with hydraulic snubbers is that they leak and consequentially require constant monitoring and refilling. Mechanical snubbers were designed to circumvent this problem. They have different operating characteristics. However, they are not examined in this report.

This paper presents a mathematical formulation describing the fluid flow in a hydraulic snubber. The flow through valves, orifices, connecting ducts etc. are considered in the study. The chamber pressures in both the tension and compression side of the piston are determined by the numerical integration of the controlling fluid flow equations. These pressures are then used to calculate the resultant forces on the piston which is in turn further transmitted to the piping system. Parametric studies were conducted to determine the effect of leakage orifice area, initial piston position and eyebolt clearance on the load deflection characteristics of this snubber. In addition cases involving variations in reservoir pressure were considered.

It has been found that the behavior of the hydraulic snubber is highly nonlinear and is governed by several factors acting simultaneously. The nonlinearity can be expressed by a general bi-linear curve. The tension and compression chamber pressures govern the slopes of these curves. In addition, a dependence on the static equilibrium position of the piston has also been noted. The leakage orifice size was found to govern the amount of energy dissipated in one cycle of snubber operation. The larger the size, the more energy dissipated by the device. Lastly, eyebolt clearances introduce a gap-element in the force-displacement characteristics about the static equilibrium position.

## MATHEMATICAL FORMULATIONS

The arrangement of a typical hydraulic snubber is shown in Figure 1 where a piston moves within a closed cylinder filled with a viscous fluid. The chambers of the bore are connected to a pressurized accumulator or reservoir through variable valve and orifices. The valves are designed such that they remain open if the pressure in the chamber remains below a prescribed value. When this pressure is exceeded, the valves close and the orifice permits a small but finite amount of leakage to the reservoir and the snubber becomes essentially a hydraulic spring. The analysis then is similar to that of hydraulic valves found in hydromechanical control systems [1].

The analysis of the system is based upon the continuity equation and the fluid equation of state. Internal fluid inertial effects are excluded. Additional equations expressing the fluid flow through the orifices and ducts are used. These are based on laminar and turbulent flow through closed ducts. The development of the equations was first attempted by Hartzman [2] in a technical note at NRC. In that study a generic snubber design was considered. In the present study an actual snubber design was considered and the following equations were developed accordingly. The compressibility effects in the chambers are described by the following differential equations:

Tension Chamber: $$\dot{P}_T = -\frac{B}{V_T}\left[(F_T + F_P) - A_T\dot{q}_r\right] \quad (1)$$

Compression Chamber: $$\dot{P}_C = -\frac{B}{V_C}\left[(F_C - F_P) + A_C\dot{q}_r\right] \quad (2)$$

Reservoir: $$P_R = \frac{B}{V_R}\left[(F_T + F_C)\right] \text{ or Constant} \quad (3)$$

Equations describing the flow of fluid through the orifice valves and connecting ducts:

Flow through tension chamber and orifice valves:

$$F_T = A_{DT}\, C_{DT} \sqrt{\frac{2}{\rho}(P_T - P_1)} \quad (4)$$

where $$\frac{1}{A_{DT}^2 C_{DT}^2} = \frac{1}{A_{HT}^2 C_{HD}^2} + \frac{1}{A_{OT}^2 C_{OD}^2} + \frac{1}{A_{VT}^2 C_{VD}^2}$$

Flow from valve to junction, turbulent flow:

$$F_T = \left(\frac{P_1 - P_J}{f_T}\right)^{\frac{1}{1.75}} \quad (5)$$

where $f_T = .242\, L_T\, \mu^{.25}\, \rho^{.75} D_T^{-4.75}$

Flow through compression chamber and orifice valves:

$$F_C = A_{DC}\, C_{DC} \sqrt{\frac{2(P_C - P_2)}{\rho}} \quad (6)$$

where

$$\frac{1}{A_{DC}^2 C_{DC}^2} = \frac{1}{A_{HC}^2 C_{HD}^2} + \frac{1}{A_{OC}^2 C_{OD}^2} + \frac{1}{A_{VC}^2 C_{VD}^2}$$

Flow from valve to junction, turbulent flow:

$$F_C = \left(\frac{P_2 - P_J}{f_C}\right)^{\frac{1}{1.75}} \qquad (7)$$

where $f_C = .242\ L_C\ \mu^{.25}\ \rho^{.75} D_C^{-4.75}$

Duct from tension-compression junction to reservoir, turbulent flow:

$$(F_T + F_C) = \left(\frac{P_J - P_R}{f_R}\right)^{\frac{1}{1.75}} \qquad (8)$$

where $f_R = .242\ L_R\ \mu^{.25}\ \rho^{.75} D_R^{-4.75}$

Piston leakage between tension and compression chambers

$$F_P = \frac{\Pi D_p C_r^3}{12\mu H_p} (P_T - P_C) \qquad (9)$$

The volume of the contained fluid is affected by the loads acting on it. These loads are applied to the fluid through the motion of the piston within the cylinder. The expansion of the cylindrical structure acts in parallel with the volume changes associated with the bulk modulus of the fluid. This results in an effective bulk modulus which can be expressed as:

$$\frac{1}{B} = \frac{1}{B_L} + \frac{1}{B_c} \qquad (10)$$

where $B_c$ for a thick walled cylinder,

$$\frac{1}{B} = \frac{2}{E} \quad \frac{(1 + \nu)\ D_o^2 + (1 - \nu)\ D_i^2}{2t\ (D_o + D_i)}$$

The instantaneous tension and compression chamber volume and chamber length are calculated as:

$$V_T = A_T H_T \qquad (11)$$

$$V_C = A_C H_C$$

and

$$H_T = L_s - (H_o + q_r) \qquad (12)$$

$$H_C = H_o + q_r$$

where $q_r = q_j - q_i$

The frictional force can be evaluated in terms of a viscous drag force which depends upon a clearance dimension between the cylinder and piston. Under pressurization the clearance itself changes because of the expansion of the cylinder. To include this effect, the friction force acting on the piston is expressed as:

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$$Q_f = \frac{\Pi D_p H_p \mu}{C_r} \dot{q}_r \qquad (13)$$

where $C_r = C_{ri} + \frac{D_i P_{av}}{4B_c}$

and $P_{av}$ = average pressure acting within the cylinder.

The resultant force rate acting on the piston is:

$$\dot{Q} = A_T\dot{P}_T - A_C\dot{P}_C \qquad (14)$$

From equilibrium of the overall system, the force rates acting at the ends which join the structure are:

$$\begin{aligned} \dot{Q}_j &= -\dot{Q} \\ \dot{Q}_i &= \dot{Q} \end{aligned} \qquad (15)$$

The total force acting on the snubber at any time is govern as:

$$\begin{aligned} Q_i &= \int_o^t \dot{Q}_i dt - Q_f \\ Q_j &= \int_o^t \dot{Q}_j dt + Q_f \end{aligned} \qquad (16)$$

The characteristics of the snubber are primarily controlled by the orifice valves. Each orifice valve has two orifice openings: one which closes when the pressure difference across the valve exceeds a predetermined valve ($P_{crit}$), and one which remains open to provide pressure release in the loaded position. The area of each orifice is therefore defined as follows:

Tension Orifice:

$$A_{VT} = A_{1T} + \beta_T A_2 T$$

$$\begin{aligned} \text{where } \beta_T &= 1 \text{ if } (P_1 - P_J) \leq P_{crit} \\ &= 0 \text{ if } (P_1 - P_J) > P_{crit} \end{aligned} \qquad (17)$$

Compression Orifice:

$$A_{VC} = A_{1C} + \beta_C A_{2C}$$

$$\begin{aligned} \text{where } \beta_C &= 1 \text{ if } (P_2 - P_J) \leq P_{crit} \\ &= 0 \text{ if } (P_2 - P_J) > P_{crit} \end{aligned} \qquad (18)$$

## METHOD OF SOLUTION

The computer code SNUBER processes the above equations. The code currently uses a sinusoidal motion with specified amplitude and frequency as input. However, any defined displacement time history could be used as the input forcing function. In addition, the code will accept an input-motion that is applied through a clearance gap. This is accomplished in the code through an engage or disengage command that recognizes when the clearance gap has been transversed.

The force-displacement response of snubber is obtained by numerical integration of the above equations. The program uses the Newton-Raphson numerical scheme to iterate the equations simultaneously while accounting for the state of the orifice valves and the end conditions. This yields the flows and the

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and the incremental pressures and hence, the total internal pressures acting on the piston at any instant, i.e., for any displacement of the piston relative to the snubber body. The total force acting on the snubber is then obtained by applying the equations of equilibrium.

RESULTS

Analyses were performed for various snubber designs. Computer runs for various cases have been obtained. These include cases where one of the design parameters was varied from the original or baseline configuration of a typical snubber. The parameters varied in this study were:

(1) Leakage orifice area
(2) Initial equilibrium piston position
(3) Input forcing frequency
(4) Variable reservoir pressure
(5) Direction of load application
(6) Eyebolt clearance

For the baseline configuration the values of these parameters were:

Leakage orifice area = 0.5% of valve area
Initial equilibrium piston (not in center) = 3.625 in. (9.21 cm)
Input forcing frequency = 30 Hz
Constant reservoir pressure = 20 psig ($137.9 \times 10^3$ N/m$^2$)
Direction of load application = compression chamber initial direction.
Eyebolt clearance = 0.

The force-deflection characteristics for each case are shown in figures 2 through 8. Figure 2 depicts the characteristics for the baseline configuration while the remaining figures depict the characteristics when one or another parameter are varied.

Referring to Figure 2, the following observations, which are common to all cases except where noted, can be made. The initial stroke is in the positive direction causing compression in the compression chamber. This motion is resisted with high apparent stiffness, initial slope of curve. On the return stroke a hysteretic loop is formed indicating energy dissipation within the device, the greater the loop the greater the energy dissipation. After the null position is reached compression of the tension chamber ensues with the motion being resisted with a lower apparent stiffness. The observed initial difference in stiffness to motion in the positive or negative directions is due to the difference in bore length of the compression and tension chambers associated with the initial off center piston position assumed for the baseline configuration. With repeated cycles a shift towards a steady state characteristic is observed.

Figure 3 presents the results for the case where the leakage orifice area is increased to 0.014 in.$^2$ (0.09 CM$^2$). The dominant effect of this change is to markedly increase the size of the hysteretic loop. The energy dissipated by the device is apparently directly proportioned to the leakage orifice area. Some changes in apparent stiffness and maximum force can also be noted.

Figure 4 shows the force-displacement characteristics when the initial position of the piston is aligned at the center of the cylinder initial piston position = 3.25 in. (8.26 cm). The response curves are very different from the other results. The slope and the maximum load are significantly altered becoming symmetric. This steady state characteristic could be represented by a linear spring.

Figure 5 corresponds to the case where the input forcing frequency is 15 Hz instead of 30 Hz. With the slower input the device dissipates a greater amount of energy. An increase in the stiffness of the device, both in tension and compression is also observed.

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Figure 6 shows the response for the case where the reservoir volume is taken small enough to significantly vary chamber pressure with the flow of fluid into and out of the chamber. The pressure changes are governed by the differential Equation (3). The slope and the maximum load valves for this case show significant changes. Unlike the other cases, these curves are symmetrical about the zero displacement point even though the initial position of the piston is not at the center of the cylinder. For this configuration the device could be represented with a bi-linear spring.

Figure 7 shows the results for the case where only the direction of the initial input displacement is reversed. This allows the tension side of the piston to activate first. Unlike the conclusions by Hartzman [2], the general characteristics of this case remain unchanged when compared with Figure 2. Only slight changes in the characteristic curves are evident.

Figure 8 illustrates the last case where an eyebolt clearance affect is considered. An eyebolt clearance of .0015 in. (.0038 cm) was assumed to exist at the pipe and snubber attachment. In this case, the response exhibits a dead zone at the center when the piston is in the clearance range. During this period the pressure in the chambers remain constant. The overall slopes of each side of the plots remain almost unchanged.

Figure 9 and Figure 10 show typical traces of the flow into the reservoir ($F_R$) and the junction pressure ($P_J$) for the case corresponding to Figure 3. It is noted that after an initial time the flow and the pressure converge to a steady state pattern under the cyclic loading. The spikes on these plots correspond to the times when the orifice valves open or close under the flow and pressure conditions.

## CONCLUSIONS

In most cases the response characteristics exhibited by the hydraulic snubber investigated was a bi-linear curve. The bilinearity increases as the static equilibrium position of the piston moves away from the center location of the cylinder. This is due to the compressibility effect of fluid in each chamber within the cylinder. Since this factor defines the stiffness of spring action by the fluid and also depends on the volume of liquid behind each piston side, the bilinearity becomes very distinct when the initial piston position approaches the ends of the cylinder bore. When the initial piston position is near the center of the bore, the bilinear characteristic converges to a mono-linear characteristic.

Leakage through the leakage orifice and leakage past the piston have a similar effect on the characteristics. They both increase the dissipation energy and hence the damping in the system. A similar effect was also noted for the lower input frequency. The higher damping also increases the number of cycles necessary to attain a steady-state characteristic. The overall characteristic of the system however, remains unchanged whether the snubber is initially subjected to a push or pull loading.

Eyebolt clearance in the system introduces a dead zone in the force-displacement characteristic about the static equilibrium position. This strictly introduces more nonlinearity into the snubber behavior, which will be reflected in the overall piping response. Finally, the reservoir pressure condition plays an important role in the snubber response. The snubber acts bi-linear or mono-linear depending on the initial piston position and steady state behavior is reached after a minimum number of cycles.

It can be concluded that hydraulic snubber behavior depends on a number of parameters. The simple assumption of elastic behavior used in current piping analysis is not justifiable. However, in all cases the snubber characteristic could be approximated with a bi-linear spring model. Further studies with earthquake loadings and sudden valve closure loadings should be made in order to understand the actual behavior of the device.

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2. Hartzman, M., "Load-Deflection Characteristics of Hydraulic Snubber Support, USNRC-MEB Technical Note.

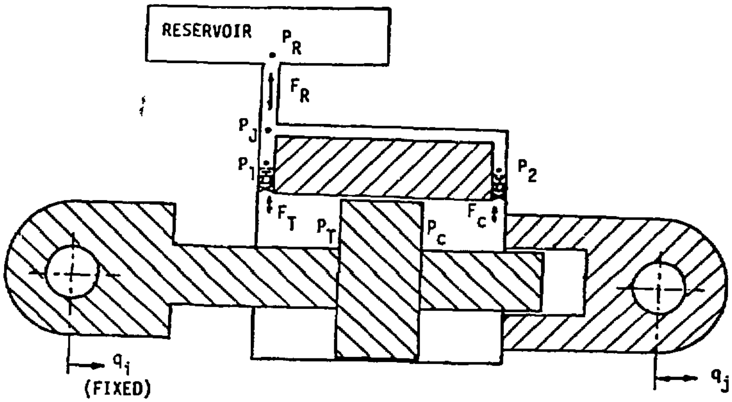


FIG. 1: SCHEMATIC DIAGRAM OF A HYDRAULIC SNUBBER

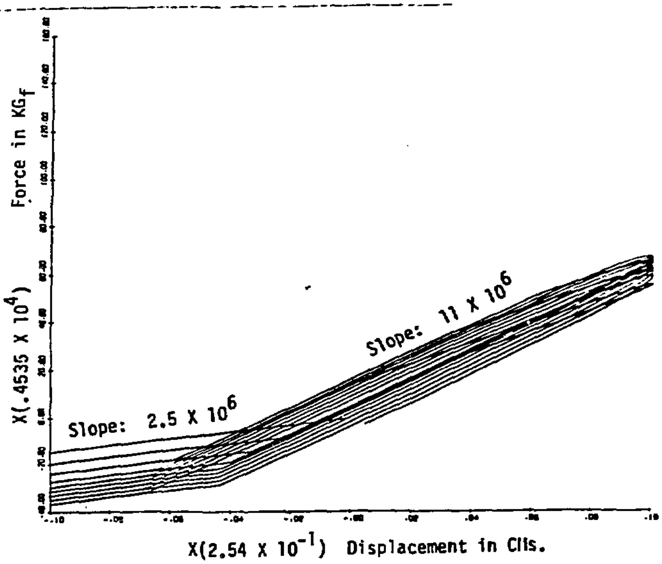


FIG. 2: FORCE-DISPLACEMENT CHARACTERISTICS FOR AS-DESIGNED DATA

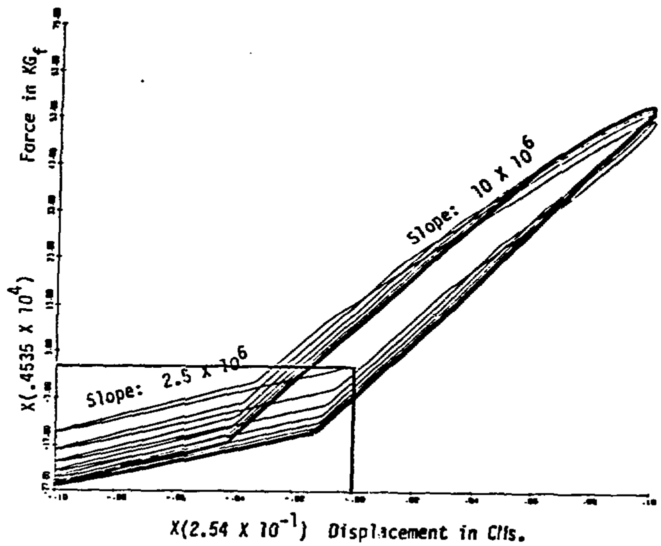


FIG. 3: FORCE-DISPLACEMENT CHARACTERISTICS FOR ORIFICE PARAMETER EQUAL TO 30% OF VALVE AREA

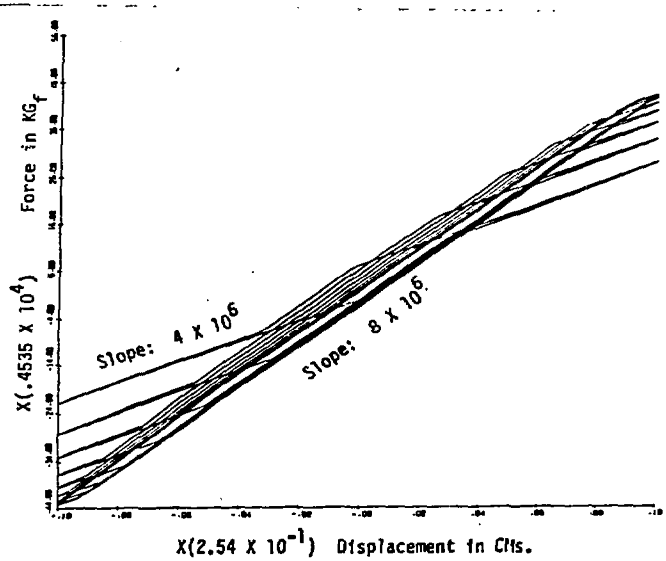


FIG. 4: FORCE-DISPLACEMENT CHARACTERISTICS FOR INITIAL PISTON POSITION AT CENTER

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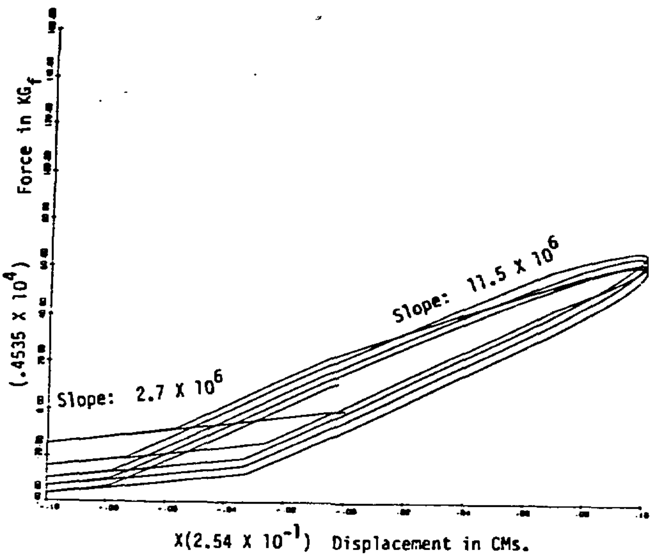


FIG. 5: FORCE-DISPLACEMENT CHARACTERISTICS FOR A FORCING FREQUENCY f = 15 Hz

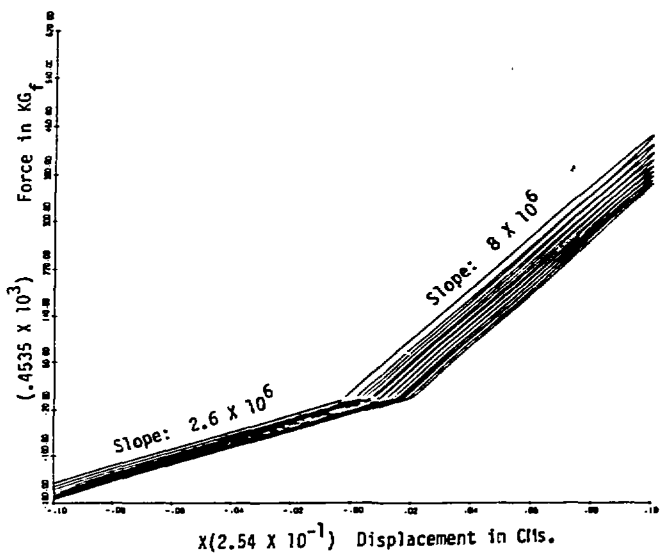


FIG. 6: FORCE-DISPLACEMENT CHARACTERISTICS FOR VARIABLE RESERVOIR PRESSURE

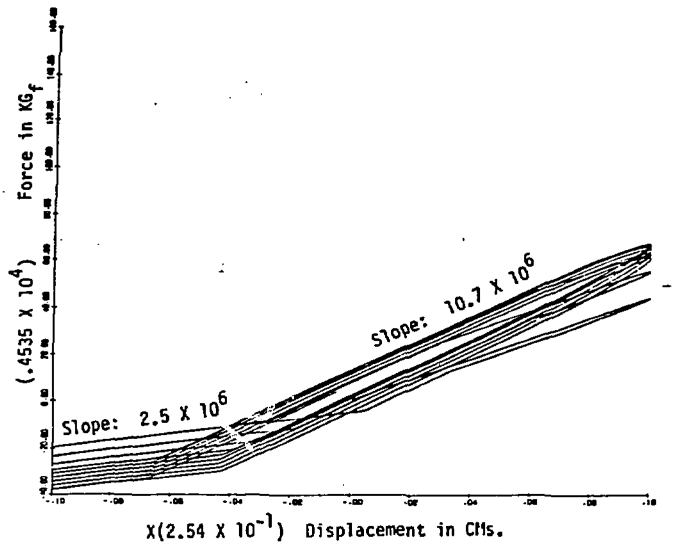


FIG. 7: FORCE-DISPLACEMENT CHARACTERISTICS FOR A REVERSE INITIAL MOVEMENT

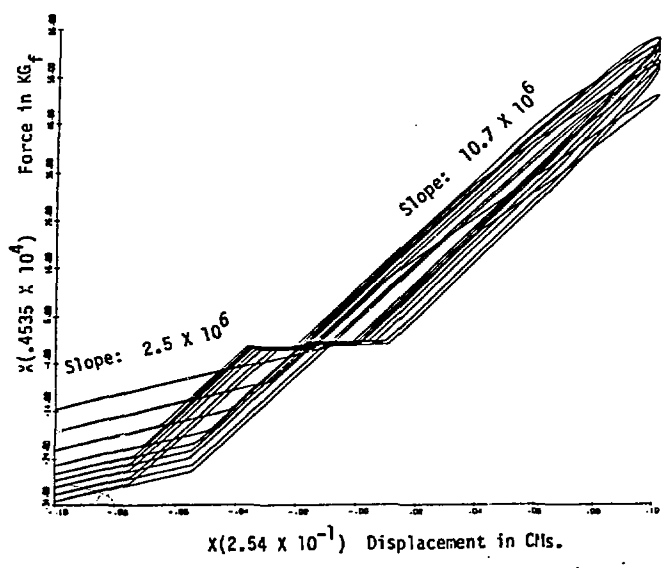


FIG. 8: DISPLACEMENT CHARACTERISTICS WITH AN EYE-BOLT GAP

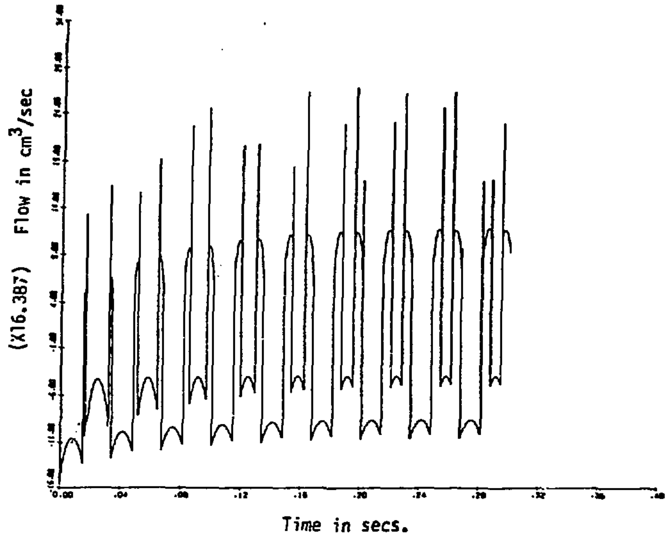


FIG. 9: RESERVOIR FLOW CHARACTERISTICS FOR A TYPICAL SNUBBER AS IN (FIG. 3)

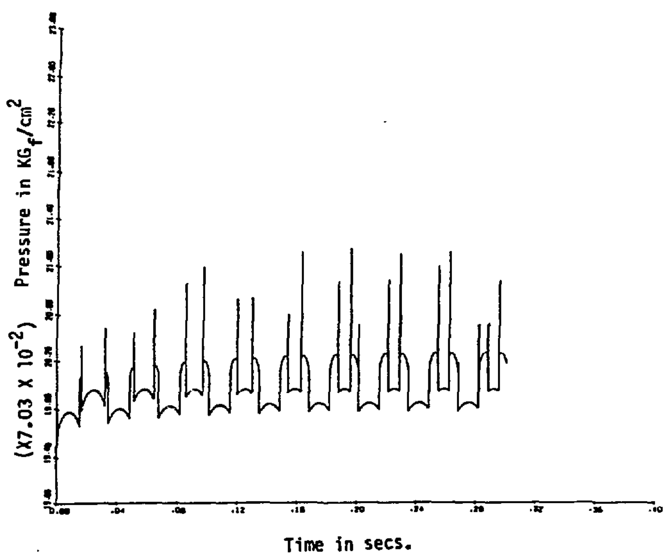


FIG. 10: PRESSURE CHARACTERISTIC AT THE JUNCTION FOR A TYPICAL SNUBBER AS IN (FIG. 3)