EVALUATION OF DYNAMIC STIFFNESS AND DAMPING FACTOR OF A HYDRAULIC DAMPER

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**Abstract**: The responses of a structure to earthquake or any other dynamic excitation can be brought down by using a suitable damper such as a hydraulic damper. This report presents an analysis for evaluating the dynamic characteristics i.e. stiffness and damping for such a damper. An analytical model has been developed for turbulent flow type damper, which consists of a cylinder and piston arrangement with a bypass pipeline. The stiffness of the system is primarily due to the compressibility of the fluid and the damping is largely due to the pressure drop in the bypass line. The dynamic response of the hydraulic damper has been evaluated for an assumed sinusoidal motion of the piston and varying the frequency of the piston displacement, initial pressure of the working fluid, bypass pipe diameter and amplitude of the piston displacement. The report presents detailed results of the study. It was seen that a system with certain specified size of components and initial pressure can operate only within certain limits of amplitude of motion and frequency. The characteristics of the damper, thus obtained, will be useful in determining the dynamic response of the whole system to which this damper will be attached.

**Keywords/Descriptors**: EARTHQUAKES; HYDRAULIC EQUIPMENT; DAMPING; EVALUATION; PHWR TYPE REACTORS; SEISMIC EFFECTS; FLUID FLOW; CYLINDERS; STRESSES; NUMERICAL ANALYSIS; FLOWSHEETS; PIPES; FUEL MANAGEMENT
ABSTRACT

The responses of a structure to earthquake or any other dynamic excitation can be brought down by using a suitable damper such as a hydraulic damper. This report presents an analysis for evaluating the dynamic characteristics i.e. stiffness and damping for such a damper. An analytical model has been developed for turbulent flow type damper, which consists of a cylinder and piston arrangement with a bypass pipeline. The stiffness of the system is primarily due to the compressibility of the fluid and the damping is largely due to the pressure drop in the bypass line.

The dynamic response of the hydraulic damper has been evaluated for an assumed sinusoidal motion of the piston and varying the frequency of the piston displacement, initial pressure of the working fluid, bypass pipe diameter and amplitude of the piston displacement. The report presents detailed results of the study. It was seen that a system with certain specified size of components and initial pressure can operate only within certain limits of amplitude of motion and frequency. The characteristics of the damper, thus obtained, will be useful in determining the dynamic response of the whole system to which this damper will be attached.
1. INTRODUCTION

In case of online refuelling in an Indian Pressurised Heavy Water Reactor (PHWR) two fuelling machines are engaged, one on each side of coolant channel. Seismic forces will be generated on the coolant channel due to earthquake and it will be more during online refuelling. These higher seismic forces are generated due to the motion of fuelling machine head in the event of an earthquake (see figure-1). Based on an earlier study by Reddy et al. [1] of the seismic force transferred by each fuelling machine head to the coolant channel, it was felt that it may be necessary to provide a suitable energy absorber for this purpose. For example, hydraulic dampers on each side of fuelling machine may be considered.

A hydraulic damper may be used between the cradle and the gimbal of the fuelling machine as shown in figure-2. The fuelling machine head is clamped with the gimbal and the head moves horizontally due to the movement of gimbal.

Hydraulic dampers using oil can be classified into two types.

1. Viscosity resistance damper utilising high viscosity oil in a laminar flow. The damping force of this type is proportional to the velocity of the vibrating object.
2. Constant orifice damper which utilise oil in a turbulent flow. The damping force of this type is proportional to the square of the velocity of the vibrating object.

Various types of dampers have been proposed in the literature [2-4, for example]. However detailed mathematical model for the dynamic characteristics of the system could not be found in the literature.

An analytical model has been developed for turbulent flow type damper. A schematic diagram of proposed damper is shown in figure 3. It consists of a cylinder and piston arrangement with a bypass pipe. Piston of the damper will be directly connected to the gimbal.

This study aims to determine the stiffness and damping of the proposed damper. These parameters will be useful in determining the dynamic response of the fuelling machine and the coolant channel system coupled with the damper.
2. ANALYSIS

The present analysis considers the response of the damper to an assumed motion imparted to the piston. Initially, the pressure is the same in the two halves of the cylinder on either side of the piston. As the piston starts moving to the right (say) from its position of rest (see fig. 3) the pressure in volume-1 increases due to compression while the pressure in volume-2 reduces due to rarefaction. Due to this difference in pressure, the fluid flows from volume-1 to volume-2 through the bypass line. Consequently, the pressure in volume-1 reduces and that in volume-2 increases. As long as the pressure in volume-1 is more than that in volume-2 the direction of flow remains the same. As the piston reverses its motion, there is rarefaction in volume-1 and compression in volume-2. The flow in the bypass line will reverse when the pressure in volume-2 exceeds that in volume-1. It is assumed that there is no leakage of fluid from the side of the piston and that there is no friction between the piston and the cylinder.

2.1 Evaluation of Parameters of Fluid Flow

Mass flow rate in the bypass pipe due to piston movement, considering the conservation of mass in volume-1, can be given as

\[
\frac{dM_{32}}{dt} = A_{cl} \rho_1 \frac{dX}{dt} - A_{cl} \left( L - X \right) \frac{dp_1}{dt}
\]  

(1)

where \( \frac{dM_{32}}{dt} \) is the mass flow rate through the bypass line.

Considering the continuity equation for volume-2 it can also be written as,

\[
\frac{dM_{32}}{dt} = A_{cl} \rho_2 \frac{dX}{dt} + A_{cl} X \frac{dp_2}{dt}
\]  

(2)

In liquids, for large pressure change, density variation is significant, which can be given as

\[
(p_1 + \alpha)(\rho_1)^n = (p_2 + \alpha)(\rho_2)^n = const.
\]  

(3a) and (3b)

The constants \( \alpha \) and \( n \) are the properties of the fluid.

Considering the total pressure drop in the bypass pipe, which consists of entry and skin friction pressure drop, the pressure in the two halves of the cylinder are related as follows.

\[
\frac{p_1 - p_2}{\rho} = \frac{1}{2} \left[ 1 + \frac{f L_{sp}}{d_{sp}} \right] V_{sp}^2 + K_1 \frac{V_{pp}^2}{2}
\]  

(4a)
Where \( \rho \) is the density, which is assumed to be equal to density of fluid in the volume driving the flow, i.e. it is equal to \( \rho_1 \) in case of flow from volume-1 to volume-2 and \( \rho_2 \) in case the flow is from volume-2. \( K_1 \) is the loss coefficient due to entry pressure loss. Effect of compressibility is already considered in the mass flow rate equations 1 & 2 and the constitutive equation (3). The effect of compressibility on the pressure drop in the bypass line will be very small and has been ignored.

\[
\frac{dM_{32}}{dt} = V_{pp} A_{pp} \rho \tag{4b}
\]

The maximum velocity in the pipe, \( V_{max} \) is obtained when friction factor \( f \) is equal to zero. Friction factor is given by the Blassius equation:

\[
f = \lambda \frac{V_{pp}^{-0.25}}{\sqrt{A}} \text{ where } \lambda \text{ given as }
\tag{4c}
\]

\[
\lambda = 0.079 \left( \frac{\rho D_x}{\mu} \right)^{-0.25}
\]

Since \( V_{pp} \) depends upon \( f \) (friction factor), iterations are required for evaluating these parameters.

Differentiating equation (3a) with respect to time, one obtains

\[
\frac{dp_1}{dt} \left( \rho_1 \right)^n - n \left( \rho_1 + \alpha \right) \left( \rho_1 \right)^{n-1} \frac{d\rho_1}{dt} = 0 \tag{5}
\]

From the above equations one obtains,

\[
\frac{dp_1}{dt} = \frac{n \left( \rho_1 + \alpha \right)}{A_{ci} \left( L - X \right) \left( \rho_1 \right)} \left[ A_{ci} \rho_1 \frac{dX}{dt} - \frac{dM_{32}}{dt} \right] \tag{6}
\]

\[
\frac{dp_2}{dt} = \frac{n \left( \rho_2 + \alpha \right)}{A_{ci} \left( \rho_2 \right) X} \left[ A_{ci} \rho_2 \frac{dX}{dt} - \frac{dM_{32}}{dt} \right] \tag{7}
\]

From equations 1 and 2, one obtains

\[
\frac{d\rho_1}{dt} = \frac{1}{\left( L - X \right)} \left[ \rho_1 \frac{dX}{dt} - \frac{1}{A_{ci}} \frac{dM_{32}}{dt} \right] \tag{1a}
\]

\[
\frac{d\rho_2}{dt} = -\frac{1}{X} \left[ \rho_2 \frac{dX}{dt} - \frac{1}{A_{ci}} \frac{dM_{32}}{dt} \right] \tag{2a}
\]
The four differential equations 6, 7, 1a and 2a are solved for the four unknowns: \(p_1, p_2, \rho_1\) and \(\rho_2\) using fourth order Runge Kutta scheme.

2.2 Evaluation of Damping Factor & Stiffness

Proposed damper system can be as viewed a combination of a spring, mass with a dashpot. The governing equation can be given as

\[ M\ddot{X} + C\dot{X} + K_sX = \text{Applied Force} \]

Where \(M\) is the total mass of piston, piston rod and the fluid. \(C\) is the damping coefficient and \(K_s\) is the stiffness of the system.

Stiffness of the system can be given as\[5\]

\[ K_s = \frac{(p_1 - p_2)A_e}{\Delta x} \quad (8) \]

where \(\Delta x\) is the piston displacement during the interval of time under consideration.

The damping force can be given as [5]

\[ F_d = \Delta p_{\text{loss}} A_e \quad (9) \]

where \(\Delta p_{\text{loss}}\) is the pressure loss in the line 3. From equation (4a)

\[ F_d = \left[ K_s \rho \frac{V_{pp}^2}{2} + \Delta P_f \right] A_e \quad (9a) \]

Where \(\Delta P_f\) is the friction pressure drop in the bypass pipe and can be given as

\[ \Delta P_f = \frac{f}{2} \frac{L \rho V_{pp}^2}{d_{pp}} \]

\[ F_d = \left[ \frac{K_s \rho A_e^3}{2 A_{pp}^2} + \frac{f}{2} \frac{L_{pp}}{d_{pp}} \rho A_e^3 \right] X^2 \quad (9b) \]
If the above equation is compared with the standard equation of vibration, damping coefficient can be given as

\[ C = \left[ \frac{\rho A^2_f}{2 A^2_p} \right] \left[ K_1 + K_f \right] \dot{X} \]  \hspace{1cm} (10)

where \( K_f = \frac{f l_{pp}}{d_{pp}} \)

Damping factor can be given as

\[ \xi = \frac{C}{2 \sqrt{K_p M}} \]  \hspace{1cm} (11)

where \( K_p \) is the stiffness of the system.

### 2.3 Stresses in the Cylinder

While evaluating the pressure in the fluid, the effect of the boundary of the cylinder has not been considered. In reality, the pressure surge in the fluid due to compression will propagate as a pressure wave. This wave will be reflected from the fixed end of the cylinder as a tension wave and the process will continue. The pressure will further increase due to sudden stoppage of the fluid at the wall.

The total pressure, \( P_u \), can be given as

\[ P_u = P_{\text{max}} + \Delta P_s \]  \hspace{1cm} (12)

Where \( P_{\text{max}} \) is the maximum fluid pressure and \( \Delta P_s \) is the pressure rise due to sudden acceleration or stoppage of fluid, which can be given as [6]

\[ \Delta P_s = \rho V_s \frac{dx}{dt} \]  \hspace{1cm} (13)

Where \( V_s \) is the velocity of sound in the working fluid.

Hoop stresses was calculated using equation

\[ \sigma_{\text{hoop}} = \frac{P_{\text{at}} D_{\text{cl}}}{2 W_t} \]  \hspace{1cm} (14)

where \( W_t \) and \( D_{\text{cl}} \) are the wall thickness and diameter of cylinder respectively.
3. NUMERICAL ANALYSIS

In order to get an insight of the dynamic behavior of such systems prior to applications in design, numerical analysis has been carried out using some assumed values of the component dimensions and piston motion. The geometrical parameters are shown in Table-1. Flow diagram of the computer program is shown in figure 4. Water has been considered as the working fluid.

The analysis has been carried out by considering various values of the time step (Δt) for numerical integration to ensure convergence.

A sinusoidal motion of the form \( X = X_0 \sin \omega t \) is applied on the piston. However, the computer program can read digitised displacement data or velocity data and can compute, by interpolation, the necessary input.

It is important to maintain the pressure of the fluid within the system above its saturation pressure at the operating temperature. For the pressure in the rarefaction zone to be above the saturation pressure, either there has to be a limit on the piston movement or else the initial pressure has to be sufficiently high so that the piston can complete the intended motion without causing cavitation in the cylinder. Further, the flow from the volume at a higher compression also has to be regulated so that in that compartment the compression of fluid due to piston movement is not outweighed by the outflow through the bypass line. The pressure drop in the bypass line has also to be kept within limits to prevent cavitation. Table-2a presents the minimum value of the diameter of the bypass line for operation with certain chosen values of the input parameters. The complete range of permissible values of the diameter of the bypass line is presented in Table-2b for frequencies of 5 Hz and 10 Hz, which shows that the range of operability increases with pressure.

In the parametric study presented here, frequency of input motion, amplitude, diameter of the bypass line (d_{pp}) and the initial pressure (p_0) have been varied.

Detailed transients are presented for \( f = 5 \) Hz, initial pressure, \( p_0 \) of 0.5 MPa and \( d_{pp} = 0.035 \) m, the lowest permissible diameter of the bypass line that can sustain the given sinusoidal motion with amplitude of 0.014 m. Pressure transients were obtained for both the volumes, which are shown in figures 5 & 6. Piston displacement input is shown in figure 7. As pressure increases in volume-1 due to reduction in volume caused by the piston displacement, the pressure in volume-2 starts decreasing. Variation of mass flow rate through bypass pipe is shown in figure 8. The variation of stiffness of the system is shown in figure 9 and the average stiffness is \( 6.95 \times 10^7 \) N/m. Variation of damping factor of the system is shown in figure 10. The average damping is found to be about 20% of critical. Other responses of the system such as total pressure, maximum fluid pressure, hoop stress etc. are shown in Table-3.

4. CONCLUSION

Thus, it will be seen that a system with certain specified size of components and initial pressure can operate only within certain limits of amplitude of motion and frequency.
Increasing the initial pressure in the fluid enhances the limit of operability. But this increase in pressure will have to be restricted from structural considerations. The characteristics of the damper, thus obtained will be useful in determining the dynamic response of the whole system to which this damper will be attached.

**NOMENCLATURE**

- A - area
- C - damping coefficient
- f - friction factor
- $K_s$ - stiffness
- L - length of cylinder
- M - mass
- $M_{32}$ - mass flow through pipe
- $p$ - pressure
- $p_0$ - initial pressure
- t - time
- $V$ - velocity
- $W_t$ - wall thickness
- X - piston displacement

**Sub Script**

- 1 - refer to volume-1
- 2 - refer to volume-2
- pp - bypass pipe
- cl - cylinder
- s - sound
- tt - total
- max - maximum
- min - minimum

**Greek Symbol**

- $\sigma$ - stress
- $\rho$ - density
- $\alpha$ - properties of fluid
- $\xi$ - damping factor
- $\mu$ - viscosity

**REFERENCES**


2. K. Sunakoda, H. Sodeyama and K. Suzuki, Development of the Large Capacity High Damping Oil Damper, Transactions of the 13th International Conference on


### TABLE-1 (INPUT)

**Geometrical Parameters**

<table>
<thead>
<tr>
<th></th>
<th>Cylinder</th>
<th>Pipe</th>
<th>Piston &amp; piston rod mass with working fluid</th>
</tr>
</thead>
<tbody>
<tr>
<td>Length (m)</td>
<td>0.5</td>
<td>0.7</td>
<td></td>
</tr>
<tr>
<td>Diameter (m)</td>
<td>0.3</td>
<td>0.005 - 0.10</td>
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</tr>
<tr>
<td>Thickness (m)</td>
<td>0.01</td>
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</tr>
</tbody>
</table>

**Properties of Fluid**
(see equation 3)

<p>| | |</p>
<table>
<thead>
<tr>
<th></th>
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</tr>
</thead>
<tbody>
<tr>
<td>Alpha ((\alpha))</td>
<td>3.0E+07</td>
</tr>
<tr>
<td>Density (\rho) at 0.1 Mpa (kg/m(^3))</td>
<td>1.0E+03</td>
</tr>
<tr>
<td>Dynamic viscosity (\mu) (N-s/m(^2))</td>
<td>1.0E-03</td>
</tr>
</tbody>
</table>

**Parameters of Input Motion**

<p>| | | |</p>
<table>
<thead>
<tr>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Amplitude (m)</td>
<td>0.014</td>
<td>1.0 - 10.0</td>
</tr>
<tr>
<td>Angular velocity (Hz)</td>
<td>1.0E-05</td>
<td></td>
</tr>
</tbody>
</table>
### TABLE - 2A

**Lower Bound Values of Diameter of the Bypass Line**

SPECIFIED $X_0 = 0.014$ m

<table>
<thead>
<tr>
<th>Pressure (0.1 MPa)</th>
<th>0.5 MPa</th>
<th>1.0 MPa</th>
<th>0.1 MPa</th>
<th>0.5 MPa</th>
<th>1.0 MPa</th>
<th>0.1 MPa</th>
<th>0.5 MPa</th>
<th>1.0 MPa</th>
</tr>
</thead>
<tbody>
<tr>
<td>$D_{pp}$ (m)</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>0.01</td>
<td>N;0.00135</td>
<td>N;0.000871</td>
<td>N;0.002303</td>
<td>N;0.000114</td>
<td>N;0.000624</td>
<td>N;0.001282</td>
<td>N;0.000114</td>
<td>N;0.000598</td>
</tr>
<tr>
<td>0.015</td>
<td>N;0.000171</td>
<td>Y</td>
<td>Y</td>
<td>N;0.000119</td>
<td>N;0.000686</td>
<td>N;0.001484</td>
<td>N;0.000114</td>
<td>N;0.000624</td>
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<td>N;0.000273</td>
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<td>N;0.000449</td>
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</tbody>
</table>
**TABLE - 2B**

Lower and Upper Bound Values of Diameter of the Bypass Line

SPECIFIED $X_0 = 0.014$ m

Can the motion be sustained for long time without cavitation? (Y/N)

Piston Displacement (m) if there is cavitation before reaching $X_0$

<table>
<thead>
<tr>
<th>$D_{pp}$ (m)</th>
<th>Frequency ($f$) = 5 Hz</th>
<th>Pressure = 0.1 MPa</th>
<th>0.5 MPa</th>
<th>1.0 MPa</th>
<th>Frequency ($f$) = 10 Hz</th>
<th>0.1 MPa</th>
<th>0.5 MPa</th>
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For Sl. No. 8 of the above table

Total pressure = 1.47 MPa
Hoop stress = 224.9 kg/cm²
FIGURE 1. FUELLING MACHINE ENGAGED WITH THE COOLANT CHANNEL

FIGURE 2. POSITION OF PROPOSED DAMPER

FIGURE 3. SCHEMATIC DIAGRAM OF PROPOSED HYDRAULIC DAMPER
FLOW DIAGRAM

Read input: Dimensions, Material Properties, Initial Condition, Imax, Time step

T = 0

Calculate x, dx/dt

Runge Kutta Integration for new pressure and density for both the volumes

Is Pressure < Psat?

Yes

Print: Pressure < Psat

Get Friction Factor for Velocity in the bypass pipe satisfying (4a) and (4b)

Calculate dm/dt, Stiffness, Damping factor

T = T + Δt

I < Imax?

Yes

no

Print

Stop

Fig 4 Simplified Flow Diagram of the Computer Program
FIG. 5 VARIATION OF PRESSURE IN VOLUME-1

$P_0 = 0.5 \text{ MPa, } f = 5 \text{ Hz}$
FIG. 6 VARIATION OF PRESSURE IN VOLUME-2
Fig. 7 Variation of Piston Displacement With Time

\[ P_0 = 0.5 \text{ MPa}, f = 5 \text{ Hz} \]
FIG. 8 VARIATION OF MASS FLOW RATE IN THE BYPASS PIPE

$P_0 = 0.5 \text{ MPa}, f = 5 \text{ Hz}$
$P_0 = 0.5 \text{ MPa}, f = 5 \text{ Hz}$

**FIG. 9** VARIATION OF STIFFNESS WITH TIME FOR A GIVEN PISTON DISPLACEMENT
FIG. 10 VARIATION OF DAMPING FACTOR

$P_0 = 0.5 \text{ MPa, } f = 5 \text{ Hz}$
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