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# Study on Shock Absorber Assembly for Rebound Cushion Impact Noise Refinement

**[1] Swapnil S. Kulkarni, [2] Dr. A. Satheesh**

[1] Research Scholar, Department of Thermal and Energy Engineering, School of Mechanical Engineering, Vellore Institute of Technology

[2]* Professor, Department of Thermal and Energy Engineering, School of Mechanical Engineering, Vellore Institute of Technology

* Corresponding author: satheesh.a@vit.ac.in

***Abstract***

The research presents the most optimized noise performance of shock absorber assembly for impact noise refinements. Individual effects of rebound cushion hardness (H), cushion height (h), percentage change in oil pressure ($P_j$) and their mutual interactions on cushion impact noise is investigated using full factorial design approach. Fluid structure interaction simulation is performed to evaluate the percentage change in oil pressure. Cushion impact noise has been identified as noise response via servo-hydraulic test rig. Combination of factor levels for the minimum targeted cushion impact noise is found as H = 70 Shore A, h = 14 mm, $P_j$ = 20 %, H and h have 72.42% and 6.25% contribution on impact noise $(SPL - A)_f$ . Interaction between h and $P_j$ are found significant as 20.21% contribution. However, individual $P_j$ has shown 0.25% contribution which is insignificant. Research provides guideline for selection of conventional cushion mechanism using fluid structure interaction simulation and design of experiment (DOE) studies.

**Index Terms**: Cushion impact noise, fluid structure interaction simulation, A-weighted SPL at frequency, oil pressure, full factorial design.

## 1. INTRODUCTION

Shock absorber noise is usually among primary noise source attributes in electric car development. It is also an issue with recent vehicles as powertrains (engines and hybrid powertrains) become comparatively quieter [1]. It affects initial buying decision of customers when car is driven at low speed on slightly rough road. In engineering standpoint, flow induced vibrations of shock absorber are related to non-linear damping force-velocity characteristics [2]. In the view of damping force requirement, electric car requires comparatively higher shock absorber stiffness and energy dissipation than conventional IC Engine car. Although being mature technology, operating principle (internal process) has not changed in recent years [3]. Scholarly research on shock absorber noise has been on-going since early 1999s, with much of early work under taken by University of Michigan on prediction of shock absorber noise using experimentation and numerical simulation [4]. In context of increase in suspension induced noise of electric car mainly in rural parts of India, large hump, potholes and speed breakers on road surface utilizes complete working stroke of shock absorber assembly. The impact noise of shock absorber arises due to rigid body impact/topping of rebound cushion. The mechanism that was designed in upper chamber of shock absorber to avoid jerk due to upward motion of shock absorber body is called rebound cushion [5]. Generally speaking, it is not advisable to reduce the working stroke of the shock absorber to avoid impact phenomenon. Constant damping force along the stroke being unable to protect shock absorbers, body and axles on high unevenness road [6]. Conventional rebound cushion is made up of natural rubber, synthetic rubber or synthetic resin [7]. Hydraulic rebound cushion is developed in recent years but it is considered as costly solution based on industrial experience of problem solving.

Studies are considered with two categories according to target setting for shock absorber noise, which are (i) Failure modes (ii) Contribution of internal process. It is observed from the literature, experimental and numerical studies are adopted to identify different failure modes. Table 1 represented the various research works carried on failure modes and contribution of internal process. Failure modes or abnormal noise sources are controlled by working on hardware damage. Internal processes or operational noise sources are controlled by working on factor levels which are responsible for noise generation.

Table 1. Studies on abnormal and operational noise

| Year | Failure modes | Internal process |
|---|---|---|
| | Abnormal (Structure Borne) | Operational (Air Borne) |
| 1999 | (Below1.5 kHz) [4] | Swish (1000-30kHz) [4] |
| 2002 | Impact (0.5- 2 kHz) [8] | Swish (2-8 kHz) [8] |
| 2010 | Rattle (200-240 Hz) [9] | - |
| 2013 | Valve acceleration (6000 m/s2) [10] | - |
| 2015 | Rattle (200-800 Hz) [11] | - |
| 2016 | Knock (50-700 Hz) [12] | |
| 2019 | Squeak (100-600 Hz) [13] | - |
| 2020 | Friction induced (1-5 kHz) [14] | Swish (500-8 kHz) [14] |

Significant work has been carried out over the last 20 years to identify failure modes and their frequency range of interest. The contribution of internal processes in shock absorbers are studied by defining the targeted performance in noise context.

Benaziz et al. [15, 16] studied the nonlinear dynamic analysis of coil spring valve in which constant orifice section is analyzed. High frequency vibration in the band of 200 Hz to 1 kHz was considered during investigations. Oscillation frequency for three different values of constant orifices section is considered. Oscillation frequency was decreased with increased in orifice section. Oscillation frequency is around 500 Hz where natural frequency of valve is 142 Hz in vacuum. Contribution of internal process (orifice valve operation) is responsible for noise generation. Noise generated due to sudden opening and closing of orifice valves. Sikora [17] experimentally studied the flow induced vibration phenomenon of piston rod assembly. This study was to develop the model which accounts for effect of oil compressibility, orifice valve stiction and inertia as a part of internal process for noise generation. Piston rod acceleration values are presented in time and frequency domains and dominant frequency component is observed at 890 Hz. Gauduin et al. [18] presented the noise generated due to impact of orifice valves. From the experiment, the impact is occurred due to clearance between the orifices assembled in piston setting which is the failure mode and considered as hardware damage. Rattling noise is investigated using experimental approach and characterized in noise operating frequency band of 100-400 Hz. Another failure mode is studied by David et al. [19]. Authors' mentioned that the vehicle level noise perceived due to loose component present in shock absorber assemblies. The noise occurs due to soft damping of shock absorber. Particularly, it is perceived when vehicle is driven at low speed and experimentally related to shock absorber amplitude (± 5 mm) and input excitation frequency of 10-15 Hz. Such transient noise phenomenon is observed in the noise operating frequency band of 250 to 1000 Hz. Clarke and Krazewski et al [20,21] worked on controlling the extreme rebound limit of shock absorber. Authors' mentioned that the conventional rebound cushion mechanism presents the undesirable harshness when shock absorber extends into rebound limits of travel. Authors' have introduced the concept called switching harshness. Switching harshness results when shock absorber changes over the higher damping rate to assist sudden impact at rebound limit. To the best of authors' knowledge, no such extensive root cause analysis work is attempted in conventional rebound cushion impact context. Deferme et al. [22] mentioned that swish noise is perceived at higher velocity strokes. Andreas et al. [23] studied the another failure mode which is classified under the airborne noise. Cavitation is generated and characterized as high frequency hissing sound. Cavitation normally occurs in valves. Shock absorber noise can also be generated by cavitation. This cavitation phenomenon can be presented at normal ambient temperature which is mainly due to improper balancing of valve assembly. High frequency component can be presented due to cavitation in shock absorber sub-assembly. Wang et al. [24] mentioned that orifice opening and closing has to be smoothened to avoid knock and chuckle noise. Also high frequency noise is generated by oil flow when passed through orifices. Noise should be kept below specific limit. This type of high frequency airborne noise called as swish noise. Air borne noise (swish or hiss) problem generated due to cavitation and turbulence in oil flow are also studied by few researchers but its controlling factors are still under the scope of research. Kadu et al. [25] adopted the DOE approach and controlling factors are studied for maximization of damping force. However, maximization of damping force resulted in swish noise tendency which is beyond the scope of earlier research works.

Noise problems, approaches, factor levels and acceptance limits are summarized in Figure.1. Briefly, the authors proceed with numerical simulation approach for selecting the factor levels for change in oil pressure (Pj) in impact noise problem. Fluid structure interaction (FSI) technique is applied in numerical simulation section. In case of physical tuning and testing of different orifices tried in piston, generation of desirable oil pressure is time consuming process as it involves proto build time, cost and many iterations. Fluid structure interaction (FSI) technique generate the desirable oil pressures with most optimized orifice stack tried in piston in much shorter time. In case of non-refined shock absorber assembly, the rebound cushion impact and valve swish noise both are the moving noise sources. Both noise sources are much closer to each other. Here, the target definition and subjective correlation both are crucial aspects.

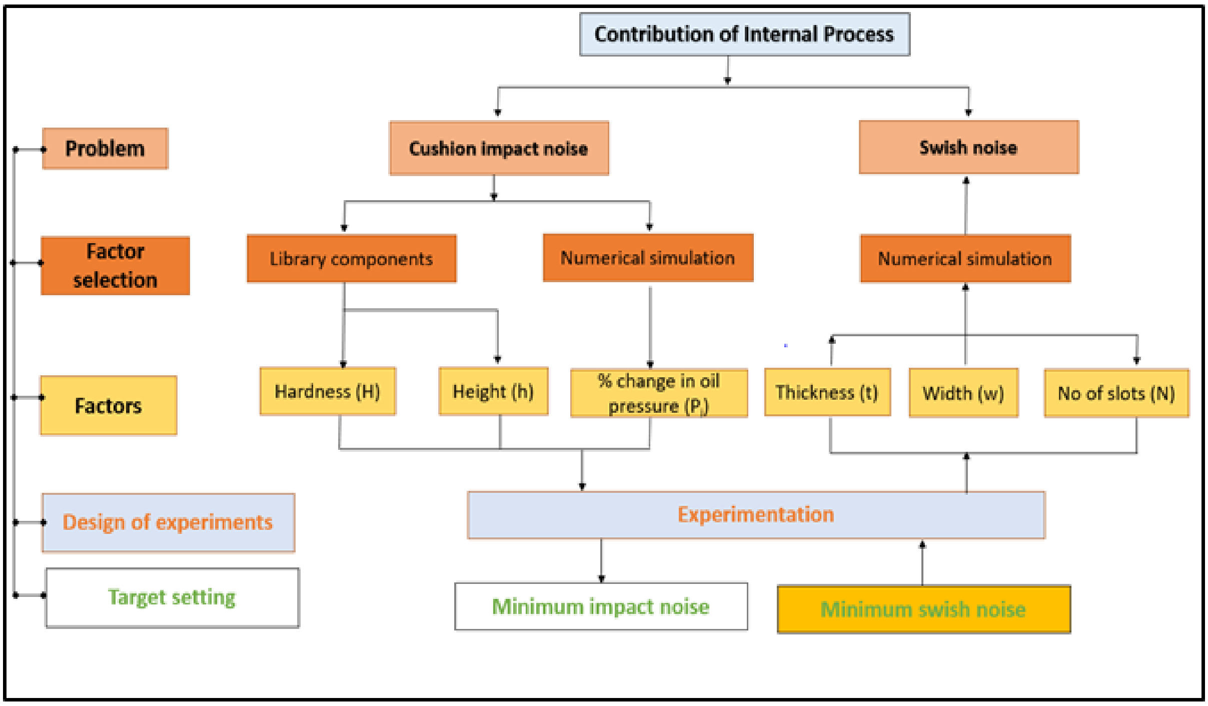



Fig. 1. Target setting procedure for impact and swish noise refinement

Based on the location of components in shock absorber assembly and study of prior art, rebound cushion stiffness and oil pressure (damping) is studied for impact noise refinement.

Following factors were chosen: Rebound cushion impact noise refinement: 1- Hardness (H), 2- Height (h), 3- Oil pressure ($P_j$).

Reduction of oscillation during rebound cushion impact is

The equation of motion of rebound cushion:

$$m_1\ddot{x}_1 + c_1(x_1 - x_2)^2 + k_1(x_1 - x_2) = F_{e\ rebound\ cushtion} \qquad (1)$$

Forced vibration of rebound cushion can be controlled by changing the damping coefficient, stiffness of rebound cushion. Damping coefficient is changed by increasing the percentage of oil pressure ($P_j$) inside the rebound chamber of shock absorber. Stiffness of cushion is changed by increasing the thickness of cushion material. Hardness of cushion material is changed to targeting the reduction in its oscillation.

Based on the domain experience, potential factors which are responsible for impact and swish noise sources are categorized in Figure 2. Oil is damped in upper portion of chamber before the impact of cushion at the end of stroke. Provision of oil quantity above cushion can damp the cushion vibrations and helpful for noise refinement.

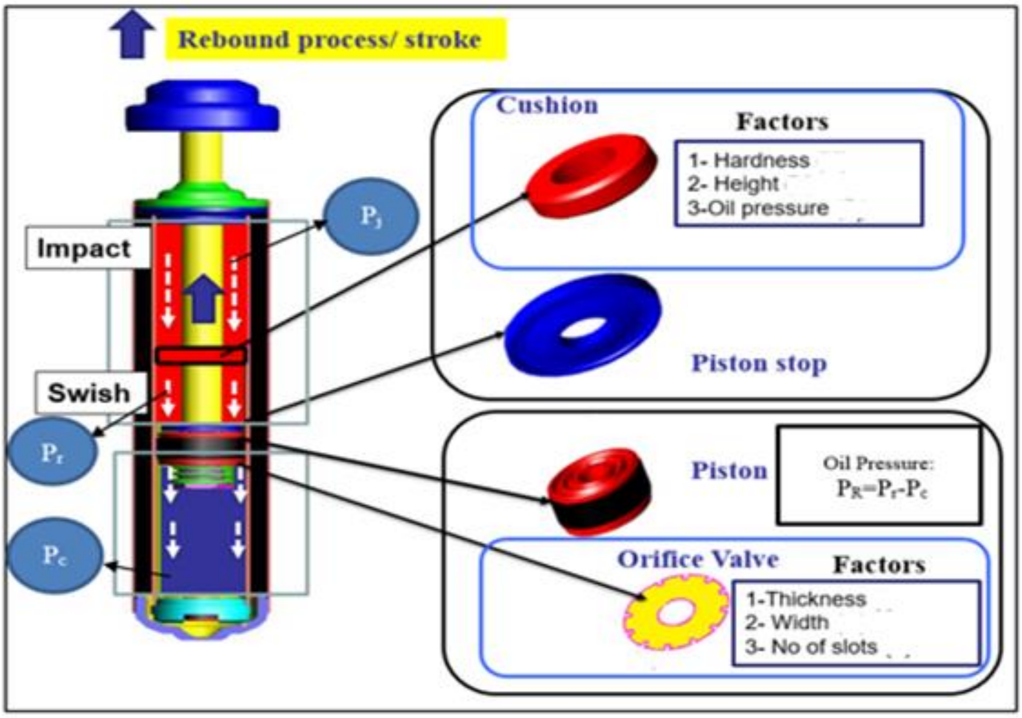
This hypothesis has to be confirmed by full factorial DOE approach. Comparatively higher oil quantity can be attained by maintaining the more oil pressure in rebound chamber using piston and orifice valve assembly. In other words, it can provide the high damping coefficient along the upward motion of piston which assures the high level energy absorption and smooth collision of rebound cushion, avoiding jerk to passengers. Refer to Micro process model of hydraulic shock absorber with abnormal structural noise [26]. By the theory of vibration, the differential equation of vibration of the piston head when oil passes through fixed orifice can be written as

$$\{m\ddot{x} + c\dot{x} + kx = \mathrm{sign}(\dot{x}_r - \dot{x})f_d + c_f(\dot{x}_r - \dot{x})\} \qquad (2)$$

$$c_f(\dot{x}_r - \dot{x}) \leq F_{cf}^{up} \qquad (3)$$

where, $F_{cf}^{up}$where, $F_{cf}^{up}$represents critical damping force between orifice and piston. $c_f$ $c_f$ represents damping coefficient as fluid passes the orifice. Damping force (oil pressure) is reduced by changing the stiffness and flow area of orifice valve to targeting the reduction in swish noise.

Fig. 2. Contribution of internal process in shock absorber noise




## 2. EXPERIMENTAL SETUP

The servo-hydraulic test rig is used for noise evaluation for shock absorbers as shown in Figure 3. Noise chamber is used to cover the shock absorber, so that it isolates the background noise during the operation. Machine frame is designed such that the natural frequency of frame is much ahead than frequency response of shock absorber which is greater than 1000 Hz. Microphone is connected near to the rebound cushion position in the noise chamber to evaluate the noise radiated from the rebound cushion during impact with mating (rod guide) surface.

Integrated circuit piezoelectric (ICP) type microphone is used (sensitivity: 46.4 mV/Pa) to measure the pressure fluctuations. Twin tube shock absorber consists of pressure and reservoir tubes filled with oil which is also the acoustic medium. Piston assembly is moving inside the pressure tube which is the moving noise source. Piston moves up and down at the velocity of 0.1 m/s and shock absorber position is set as the rebound cushion comes in contact with the mating (rod guide) surface which generates the impact noise. The complete height (stroke) of rebound cushion is utilized. Piston moves up and down at velocity of 0.3 m/s. The piston position is set at the middle portion of the shock absorber. It generates swish noise when oil is pressurized between the chambers. Noise radiation of cushion is recorded using noise data acquisition system and analyzed in 1/3rd octave band frequency. Noise signals are usually represented in 1/3rd octave band frequency as it linked to the perception of noise by human ear and permits the compression of noise information.

Each 1/3rd octave bands are identified by,

Middle frequency $(\mathbf{f_o})$ $(\mathbf{f_o})$ (4)

Lower frequency $\left(\mathbf{f_o}/\mathbf{2^{1/6}}\right)\left(\mathbf{f_o}/\mathbf{2^{1/6}}\right)$ (5)

Upper frequency $\left(\mathbf{f_o}\times\mathbf{2^{1/6}}\right)\left(\mathbf{f_o}\times\mathbf{2^{1/6}}\right)$ (6)

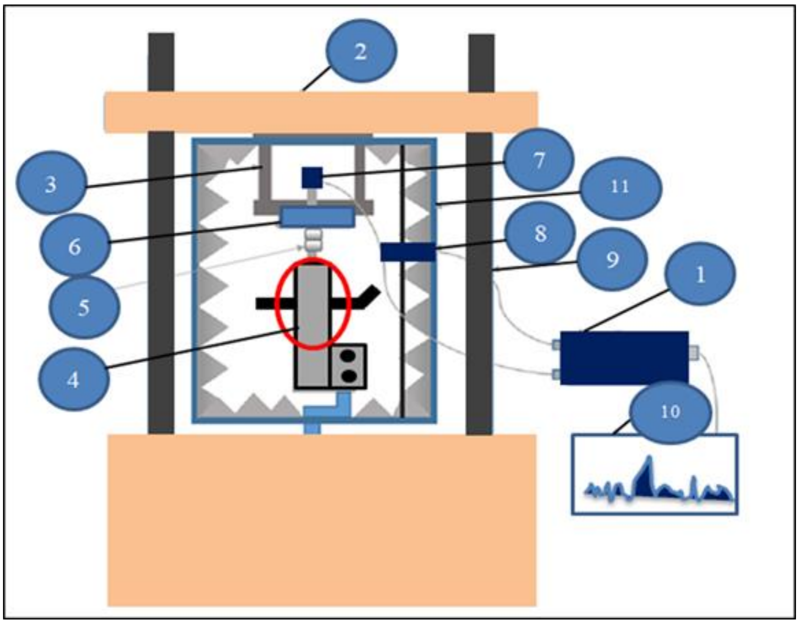



Fig. 3. Schematic diagram of servo-hydraulic test rig

1: Servo hydraulic test rig, 2: Frame of the machine, 3: Noise chamber, 4: NVH data acquisition instrument, 5: Piston (rebound valve), 6: Pressure (inner) tube, 7: Base (compression) valve,8: Reservoir (outer) tube, 9: Microphone, 10: NVH data post-processing, 11: Noise chamber

Impact noise characteristics of rebound cushion is evaluated in 1/3rd octave band and it is shown in Figure 4. Listening study is performed and restricted to five different noise samples to understand the effect of change in frequency components on audible noise levels for human ear. These recordings were replayed in sound quality studio with precise microphone. Subjective assessment is conducted by fifteen juries which are from product design and quality backgrounds. The words used in the semantic differential method and ratings are defined. 5 rating is given for extremely annoyance and 1 rating is given for no annoyance. Pearson correlation coefficient (r) is used to measure how strong a relationship between two variables i.e. annoyance and frequency and defined as

Table 2. Subjective- Objective Correlation

| **Noise sample number** | **Frequency** | | | **Annoyance** |
|---|---|---|---|---|
| | 525 Hz | 2000 Hz | 6300 Hz | **Average** |
| 1 | 54.3 | 39.7 | 42.2 | 2.42 |
| 2 | 48.1 | 41.6 | 45 | 2.80 |
| 3 | 45.5 | 40.9 | 37.3 | 2.23 |
| 4 | 42.4 | 47.1 | 42.5 | 4.65 |
| 5 | 37.9 | 45.2 | 40 | 4.21 |
| **r** | **-0.74** | **0.97** | **0.17** | **-** |

Subjective-objective correlation is shown in Table.2. It is observed that difference in noise radiation at frequency component of 2000 Hz is critical as it has attained the well correlation with the human noise perception. Here, the Pearson correlation coefficient has attained the value of 0.97 at frequency component of 2000 Hz.

$$r = \frac{\sum(x_i - \bar{x})(y_i - \bar{y})}{\sqrt{\sum(x_i - \bar{x})^2 \sum(y_i - \bar{y})^2}} \qquad (7)$$

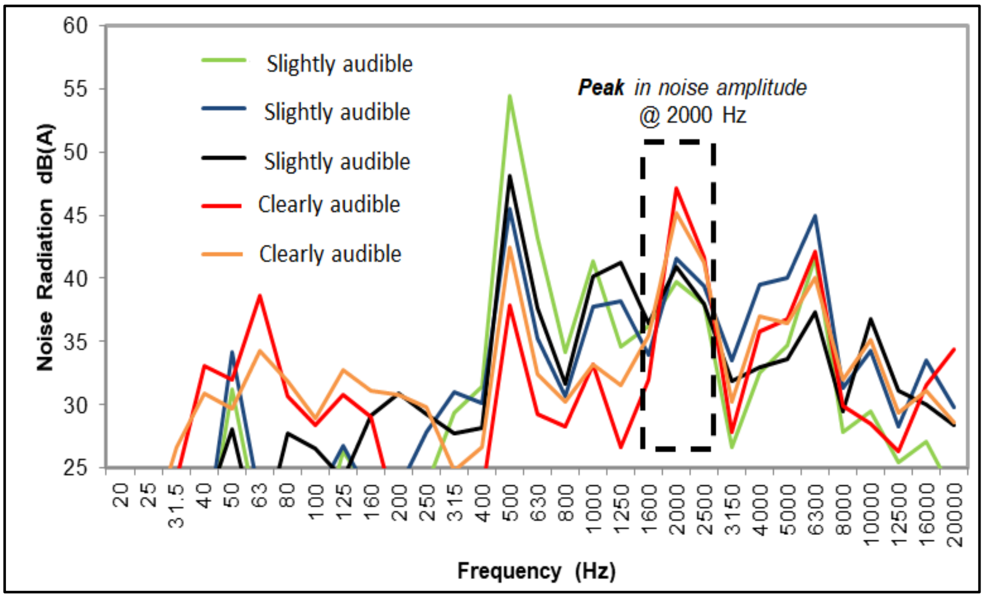



Fig. 4. Octave band of noise during rebound cushion impact

## 3. NUMERICAL SIMULATION

Two ways coupling between shock absorber fluid and thin structure is considered. Fluid flow through shock absorber is governed by thin structure. Fluid and structure mesh is prepared in single deck. [14, 27, 28,29]. Fluid and solid mechanics can be studied independently. Low speed rebound and compression: Orifice valve is not deflected. Fluid flows through bleed passages. High speed rebound and compression: Orifice valve is gradually deflected. This allows the fluid to flow through unobstructed piston and base valve ports. Various imposed velocities or displacement vs. time can be given as input on piston rod; the force velocity curve is generated. Refer to Adina theory manual for fluid structure interaction simulation [30].

For ALE system, non- conservative continuity equation becomes,

$$\frac{\partial\rho}{\partial\tau} + (v - w)\cdot\nabla\rho + \rho\nabla\cdot v = 0 \qquad (8)$$

Only difference between Eulerian and ALE formulation is that relative velocity replaces the convective velocity. Furthermore, $w$=0 corresponds to a purely Eulerian description while $w$=$v$ corresponds to a purely Lagrangian description. A recommended stiffness factor of the interface for fluid structure interaction problems can be obtained by:

$$St_{fac} = \frac{\rho * v^2 * S_{el}}{Gap} \qquad (9)$$

$$gap = 1.5 \cdot L_c \qquad (10)$$

Nodal pressure values which are contributed cushion impact and valve swish noise are identified by fluid structure interaction method. Nonlinear explicit fluid structure interaction technique has ensured that the contact force is acting between the orifice valves. It has simulated the realistic behaviour of the orifice valve subjected to oil loading. It can be considered as the initial condition to avoid the orifice flutter due to rapid deformation. Valve acceleration values can be computed using this approach [10]. Computational methods are applied by the researcher for investigating the oil flow characteristics of shock absorber assemblies [4,8,14]. Oil flow characteristics in piston and orifice valves are investigated using FSI simulations. Oil chamber is modelled on top and bottom of the piston to ensure the fully developed flow regimes. Orifices of different thicknesses are configured in piston to get desired increments in fluid pressures at orifices. Oil flow regions can be extracted from the three dimensional geometry as shown in Figure 5. Geometry representing the space occupied by oil was considered. Half symmetry finite element mesh model was prepared as depicted in Figure 6, to simulate the flow behaviour through piston valve at rebound motion. Table 3 provided the details of mesh used in the present FSI numerical study. Non conformal mesh is prepared at fluid and structural interface. Large deformation formulation of orifices can be possible by this FSI method. Hexahedral elements are preferred for coupled Eulerian Lagrangian (CEL) coupling. Two-way coupling is considered for oil and orifice structure to evaluate the orifice velocities. Element quality check is performed to ensure aspect ratio < 10 and warpage aroung55°.

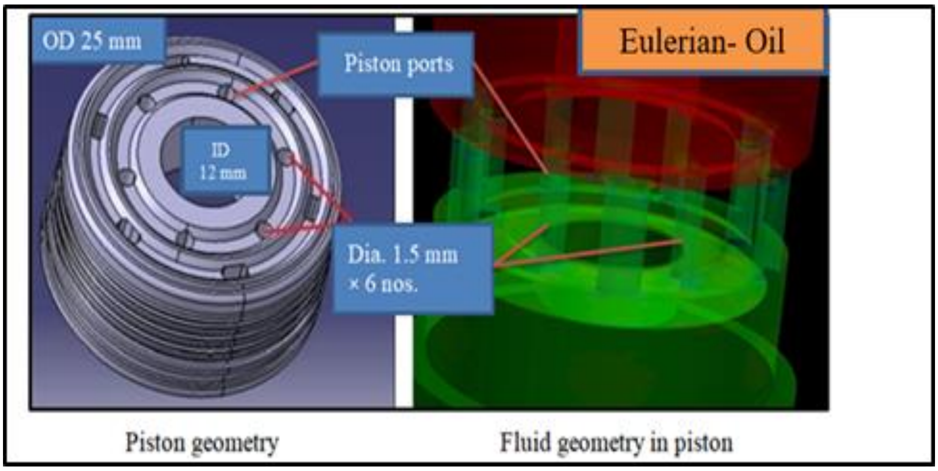



Fig. 5. Geometry of piston assembly

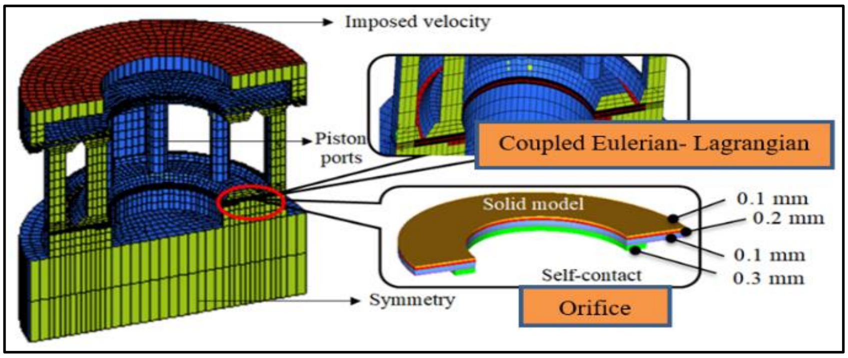



Fig.6. Mesh model of piston assembly

Table 4 provided the oil and orifice structure properties for the present numerical analysis [14]. Fluid geometry of piston valve setting is extracted from three dimensional computer aided design model. Flow domain is extracted in accordance with oil flows through the piston valve setting. Solid type contacts between the orifices are defined [28].

Table 3. Meshing strategy

| Mesh description | Fluid | Structural |
|---|---|---|
| Mesh Type | Hexahedral | Hexahedral |
| Approach | Eulerian | Lagrangian |
| Material | HPCL Oil | Steel |
| Element | 70841 | 8058 |
| Nodes | 79677 | 6068 |

Table 4. Material properties

| Material Property | Fluid | Structural |
|---|---|---|
| Dynamic Viscosity (kg/mm-Sec) | $9 \times 10^{-6}$ | - |
| Young's Modulus (N/mm$^2$) | - | $2\times10^5$ |
| Poisson Ratio | - | 0.3 |
| Density (kg/mm$^3$) | $850 \times 10^{-9}$ | $8\times10^{-6}$ |

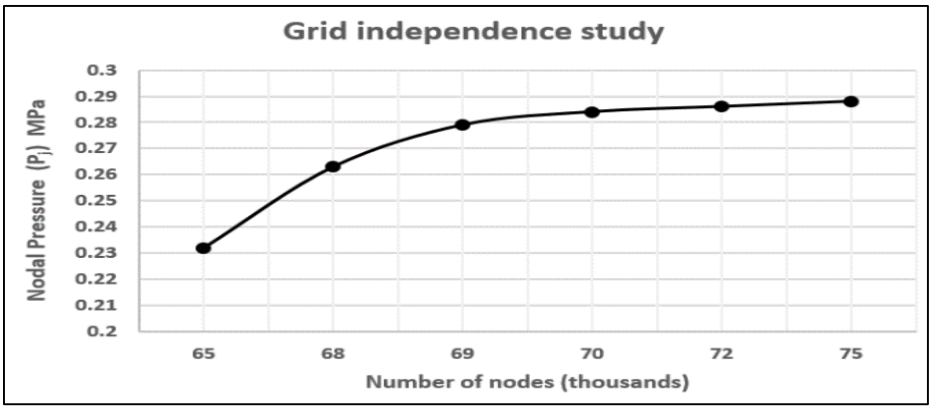



Fig. 7. Grid Independence study

More refinement of mesh with increase in fluid and structure nodes resulted in the negligible change in oil pressure values. Grid independence study is performed with generation of fine mesh and increase in number of nodes as shown in Figure 7. Nodal pressure output is stabilized beyond 70000 nodes.

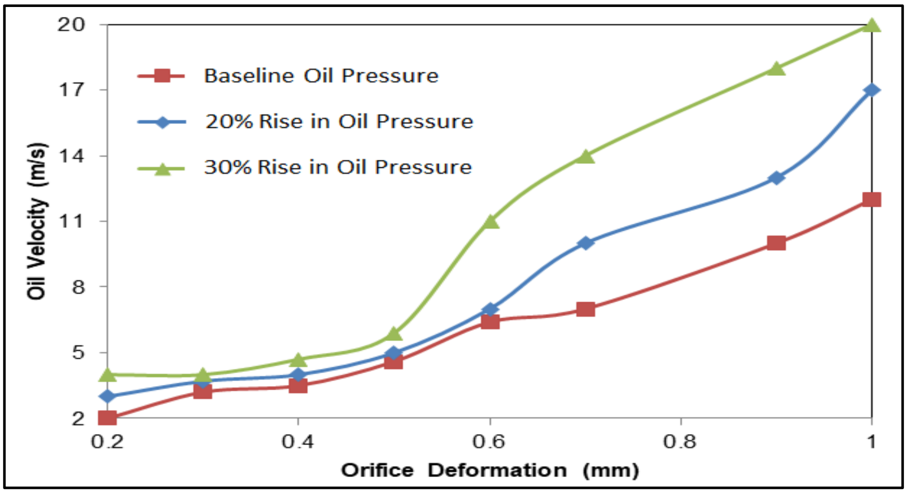



Fig. 8. Relationship between orifice deformation and oil velocity

Figure 8 represented the orifice deformation and oil velocity plot. Nodal pressure values corresponding to orifice deformation values obtained from fluid structure interaction method. Here, 20% and 30 % rise in oil pressure values are obtained due to two different configurations of orifice thickness/stiffness used in piston assembly than baseline configuration.

Initial deformation of orifice occurred below 0.4 mm which resulted in lower rise in oil velocity. Bending mode of orifices increased beyond 0.4 mm deformation which resulted in higher rise in oil velocity and pressure. Uneven bending mode resulted in cavitation and noise tendency in valve assembly [14]. Even bending modes are obtained using numerical simulation while generating these 20 and 30 % increments in oil pressure values. Oil cavitation is absent in damping force- piston displacement characteristics [2,14]. However, rise in oil pressure in rebound chamber can be acted as damping medium for reduction in rebound cushion oscillations. This hypothesis is studied using design of experiments.

## 4. DESIGN OF EXPERIMENT (DOE) STUDY

Table 5. L8 Array for DOE

| Factors | | | Noise Response @ 2000 Hz |
|---|---|---|---|
| *H* (Shore A) | $P_j$ (%) | *h* (mm) | $(SPL - A)_f$ |
| **70** | 20 | 8 | 38 |
| **70** | 30 | 8 | 36 |
| **70** | 20 | 14 | 34 |
| **70** | 30 | 14 | 37 |
| **90** | 20 | 8 | 42 |
| **90** | 30 | 8 | 40 |
| **90** | 20 | 14 | 39 |
| **90** | 30 | 14 | 41 |

DOE is defined as systematic procedure carried out under controlled conditions to discover the contribution of individual factors on response variable and furthermore provides the effect of interactions between the factors on response variable. Root cause of the decrease in the damping force is identified using DOE approach [24]. Individual contribution of orifice valves and piston ports on variation in oil pressure is studied using full factorial DOE. Major contribution of orifice valve in low and mid piston velocities is identified using experimentation approach and computational fluid dynamics [31].

These three factors are involved in cushion impact as shown in Table.5. Both levels come from industrial specification of cushion hardness and height available for experimentation. Percentage change in oil pressure $P_j$ is selected such that it should not contribute to significant change in hydraulic performance. Hydraulic performance means the desirable damping performance due to significant change oil pressures [27,28]. Refer to the study on rebound cushion impact in case of standard shock absorber [32]. Load on rebound cushion increases linearly from 0 to -500 daN beginning at the touch point of rebound cushion up to cushion height (h) which is called as the stroke of the rebound cushion under the load -500 daN.

Mutual interaction between the three factors H, h and $P_j$ are studied for obtaining the accuracy in the model. As Table 6 shows the P-values less than 0.01 for H and $P_j \times h$, it shows that individual factor H and the interaction between factors $P_j \times h$ has a real impact on cushion impact noise. The larger the F value, the more likely it is that the variation caused by H and $P_j \times h$ are real and not by chance.

Table 6. Full factorial DOE with mutual interactions for $(SPL - A)_f$

| Source | Degree of freedom | Adj (SS) | F-Value | P Value | Contribution % |
|---|---|---|---|---|---|
| ***H*** | **1** | **36.12** | **289** | **0.000** | **72.42** |
| ***$P_j$*** | 1 | 0.12 | 1 | 0.391 | 0.25 |
| ***h*** | 1 | 3.12 | 25 | 0.015 | 6.25 |
| ***$P_j \times h$*** | 1 | 10.12 | 81 | **0.003** | **20.29** |
| ***$H \times h$*** | 1 | 0.125 | 1 | 0.5 | 0.25 |
| ***$P_j \times H$*** | 1 | 0.125 | 1 | 0.5 | 0.25 |
| **Error** | 3 | 0.37 | - | - | **0.75** |
| **Total** | 7 | 49.87 | - | - | **100** |

Regression was performed and higher R-Square value is obtained as it gives better fit. The R-Square adjusted value is 98.25%. Smaller difference between R square and R Square adjusted is obtained from this model. This model has truly represented the tendency of impact noise due to rigid body radiation. Regression equation is formed to control the impact noise of rebound cushion for the range of selected parameters in the present study.

$$(SPL\text{-}A)_f = 52 + 0.21\,H - 1.1P_j - 1.2h + 0.04\,P_j \times h \quad (11)$$

## 4. CONCLUSIONS

Cushion impact noise refinement is studied using fluid structure interaction (FSI) and design of experiment (DOE). The optimization of significant factors and their interactions involved in assemblies are discussed. Coupled Eulerian- Lagrangian formulation is used for simulating shock absorber oil and nonlinear orifices. The following conclusions are made from the research work,

H and $P_j \times h$ are the significant factors by considering a 95% significance level to refine the cushion impact noise. It avoids the oscillation at the rebound cushion collision and reduces the impact noise.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## NOMENCLATURE

| | |
|---|---|
| $m_1$ | Mass of cushion [kg] |
| $\ddot{x}_1$ | Velocity of cushion [m/s] |
| $c_1$ | Damping of oil [N.sec/m]$c_1$ Damping of oil [N.sec/m] |
| $x_1$ | Displacement of cushion [mm] |
| $x_2$ | Displacement of rod guide [mm] |
| $k_1$ | Stiffness of cushion [N/mm] |
| $F_{e\ rebound\ cushtion}$ | Rebound force of cushion [N] |
| $F_{cf}{}^{up}$ | Force between orfice and piston [N] |
| $c_f$ | Damping coefficient as oil passes orifice [N.sec/m] |
| $f_o$ | Frequency of noise [Hz] |
| r | Pearson correlation coefficient |
| $x_i$ | Values of X variable in a sample |
| $y_i$ | Values of y variable in a sample |
| $\bar{x}$ | Mean of the values of the X variable |
| $\bar{x}$ | Mean of the values of the X variable |
| $\bar{y}$ | Mean of the values of the y variable |
| $\bar{y}$ | Mean of the values of the y variable |
| $L_c$ | Length of fluid element [mm] |
| $\rho$ | Oil density [kg/m$^3$] |
| $S_{el}$ | Surface of Lagrangian element |
| v | Speed of sound [m/s] |
| $\tau$ | Oil stress [Pa] |
| $\mathbf{v}$ | Velocity vector [m/s] |
| $\mathbf{w}$ | Mesh velocity vector [m/s] |

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