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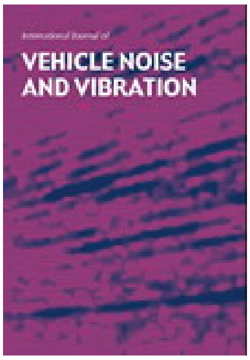

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Fluid structure interaction study of shock absorber for clicking noise refinement

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Abstract: Clicking noise refinement in car shock absorber is studied using fluid structure interaction (FSI) and design of experiment (DOE). FSI helps to refine clicking due to rapid deformation of orifice. Individual effects of rebound cushion hardness (H), cushion height (h), percentage change in oil pressure (P_j) and their mutual interactions on rebound cushion impact noise are investigated using full factorial design. Combination of factor levels for the minimum targeted clicking noise is found as H = 70 shore A, h = 14 mm, P_j = 20% from full factorial design, it is found that H and h have 72.42% and 6.25% contribution on cushion impact noise $(SPL – A)_f$. Interaction between h and P_j is found significant as 20.21% contribution. However, individual P_j obtained from FSI result has shown 0.25% contribution which is insignificant. Correlation is established between FSI and experimental oil pressure (P_j) during nonlinear deformation of orifice.

Keywords: clicking noise; fluid structure interaction; FSI; a-weighted SPL at frequency; oil pressure; full factorial design.

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1 Introduction

Clicking noise consists of loud and sharp component which annoys the passenger when such shock absorbers are assembled in the electric cars. Clicking noise is the structure borne noise source and studies are considered with two different types according to the mechanism of sound generation, which are

1 impact noise of rebound cushion when acted as rigid

2 metallic noise of orifices assembled in piston when acted in rapid deformation.

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 utilises complete working stroke of shock absorber assembly. The clicking 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. Generally, it is not advisable to reduce the working stroke of the shock absorber to avoid impact phenomenon. The provision of pressured oil in working stroke of shock absorber can damp the cushion vibration and reduce impact noise. The pressurised oil can provide more restriction to the upward motion of shock absorber while piston assembled with different orifices. In case of physical tuning 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) models generate the desirable oil pressures with most optimised orifice stack tried in piston in much shorter time. Moreover, the FSI models mainly contribute to fatigue life prediction and metallic noise refinement of orifice stack when acted in rapid deformation.

It is observed from the literature, many experimental and FSI studies are adopted to identify or refine the noise sources. Table 1 shows the various research works carried out in the shock absorber for better refinements/reduction of different noise sources. Recently in electric cars, the clicking noise of shock absorber known to generate disturbance in vehicle cabin. This entire work provides the information which is the basis for clicking noise refinement in shock absorber assembly.

Table 1 Studies on noise refinements of shock absorber

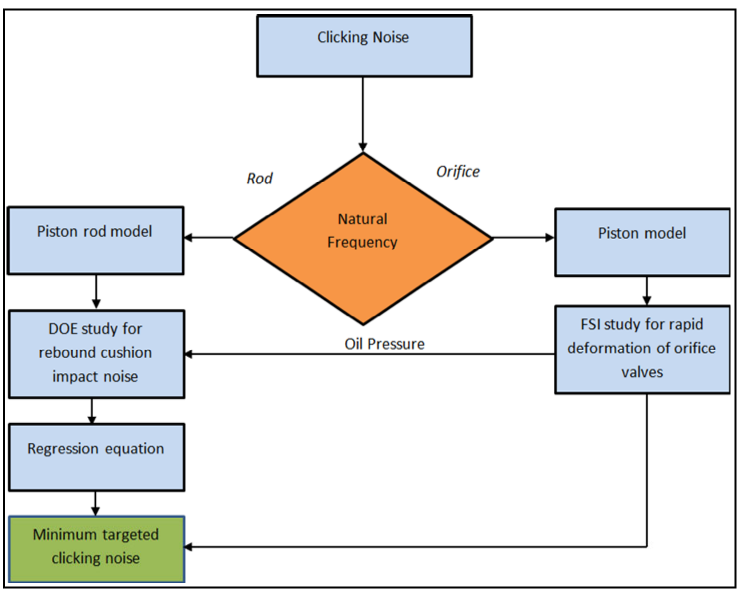
Year of publication	*Various noise sources*	
	Experimental study	*FSI study*
1999	Air borne (Cherng et al., 1999)	-
2002	Chuckle (Kruse, 2002)	-
2010	Rattle (Park et al., 2010)	-
2013	-	Valve acceleration (Qi-Ping et al., 2013)
2015	Rattle (Benaziz et al., 2015)	-
2016	Knock (rod vibration) (Bartolini et al., 2016)	
2019	Squeak (Huang et al., 2019)	-
2020	Friction induced (Kulkarni et al., 2020)	Swish and rattle (Kulkarni et al., 2019)

Nonlinear dynamic model of displacement sensitive shock absorber is proposed (Lee and Moon, 2005). Damping force is computed by force acted on both sides of piston. Effect of displacement sensitive orifice length on damping force is determined. Pressure variations in chambers due to excitation velocities are computed. Variation in flow rate is studied in this work. Sound source generated due to cavitation phenomenon present in simple thin orifice is studied (Testud et al., 2007). Experimentation is conducted on orifice which consists of single and multiple holes. The cavitation present in two regimes is distinguished and corresponding spectrum of broadband noise is observed at downstream of orifice. Cavitation regimes are generated during the fluid flow. Acoustic spectrum collapses in cavitation and tonal component is presented with periodicity in vortex shedding. Researchers mentioned that multiple hole orifice is much quieter than the orifice of single hole with considerably same opening area. The nonlinear dynamic analysis of coil spring valve is studied in which constant orifice section is analysed (Benaziz et al., 2013, 2012). 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. Structure borne noise occurred because of sudden opening and closing of orifice valves. The orifice valve deformation and force relation using finite element analysis (FEA) is developed (Shams et al., 2007). Force exerted on valve as a function of temperature for three piston velocities are also identified. Turbulence intensities are quantified at 0.5 and 0.3 mm deformations of valve. The modelling of oil flow through cavity inside shock absorber orifice valve using computational fluid dynamics (CFDs) is proposed (Sławik et al., 2010). Simulations are performed at two characteristics points where first point corresponds to minimum opening of orifice valve and second point corresponds to maximum opening of orifice valve. The flow induced vibration phenomenon of piston rod assembly is studied experimentally (Sikora, 2014). This study was to develop the model which accounts for effect of oil compressibility, orifice valve stiction and inertia. Piston rod acceleration values are presented in time and frequency domains. The correlation between 1/8th and complete 360° finite element FSI model of valve assembly is achieved (Czop et al., 2012) Von-Misses stresses of valves are evaluated for total valve deformation of 0.44 mm and well correlated. The throttle deformation of orifice slice is modelled (Zhou et al., 2008). Researchers have studied differential equation for deformation of orifice valve and deformation coefficient was obtained. The noise generated due to impact of orifice valves is presented (Gauduin et al., 2008). From the experiment, the impact is occurred due to clearance between the orifices assembled in piston setting. Rattling noise is investigated using experimental approach and characterised in noise operating frequency band of 100–400 Hz. In spectrum, Pseudo Margenau-Hill representation is used to show the pressure signals. The fluid flows through orifices of different shapes are studied (Kadam and Chaudhari, 2018). Acoustic spectrum is analysed. In case of divergent geometries of orifice, tonal noise is generated in low flow rates and broadband noise is generated for high flow rates. Most of the results presented on sound source are of circular orifices with sharp edges (Kadam and Chaudhari, 2018; Testud et al., 2007). Vehicle level noise perceived due to loose components presented in shock absorber assemblies has been analysed (David et al., 2009). 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 1,000 Hz.

It is observed from the above literature, the research works are mainly focused on hydraulic and acoustic behaviours of orifice valves. Limited studies on bending modes of orifice valves are presented in experimental and FSI approaches. Individual contribution of orifice bending modes on damping force variation, rattle noise and fatigue life are investigated (Kulkarni et al., 2020, 2019, 2014, 2013a, 2013b). Three point even and two-point uneven bending modes are obtained using FSI approach in which uneven bending mode has increased oil pressure abruptly. Experimental study is concentrated for rattle noise in the frequency band of 20–500 Hz and flow induced noise in the frequency band of 500–3,000 Hz. Infinite fatigue life (>1 × 10^6 cycles) of orifice stack was obtained from FSI approach. Fatigue failure of thin orifice stack is also one of the sound radiation mechanisms while assessing the clicking noise. In the present study, noise radiation due to rebound cushion impact and rapid deformation of the orifice valves are investigated. Finally, the work directed towards experimental characterisation of impact noise phenomenon.

Figure 1 Flowchart of clicking noise generation mechanism (see online version for colours)



2 Clicking noise generation mechanism

Figure 1 shows the flowchart of clicking noise generation inside the shock absorber assembly. At first, the natural frequency of the experimental setup is maintained far away from the natural frequency of shock absorber assembly. The results should not be disturbed due to assembly looseness and improper sealing of shock absorber. Looseness in shock absorber assembly resulted in rattling noise (Kulkarni et al., 2020). Further, clicking noise generation mechanism is distinguished in two categories. In first category, impact noise of rebound cushion is evaluated. In order to identify the noise enablers, study is engaged by means of design of experiment (DOE) approach. In second category, metallic noise due to rapid deformation of orifice is studied. In order to control the noise enablers, study is engaged by means of FSI approach. Results of orifice velocity and contact forces are studied. In case of non-refined shock absorber assembly, the rebound cushion impact and metallic noise due to rapid deformation of orifice both are the moving noise sources. Both noise sources are much closer to each other. FSI approach ensures that metallic noise is avoided by even bending mode and lower acceleration of orifice. Eventually, it reduces the piston vibrations when orifices oscillation is well controlled (Kulkarni et al., 2019). Oil pressure value obtained from this FSI result is taken as one of the factors for impact noise refinement and perhaps it helps in damping the cushion vibrations. Minimum targeted clicking noise level is estimated at the frequency band of interest using full factorial design type of DOE approach.

3 Experimental setup

The servo-hydraulic test rig is used for noise evaluation for shock absorbers as shown in Figure 2. 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 1,000 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 clicking noise when complete height (stroke) of rebound cushion is utilised. Noise radiation of cushion is recorded using noise data acquisition system and analysed 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 band is identified by,

$$\text{Middle frequency}\left(f_o\right) \tag{1}$$

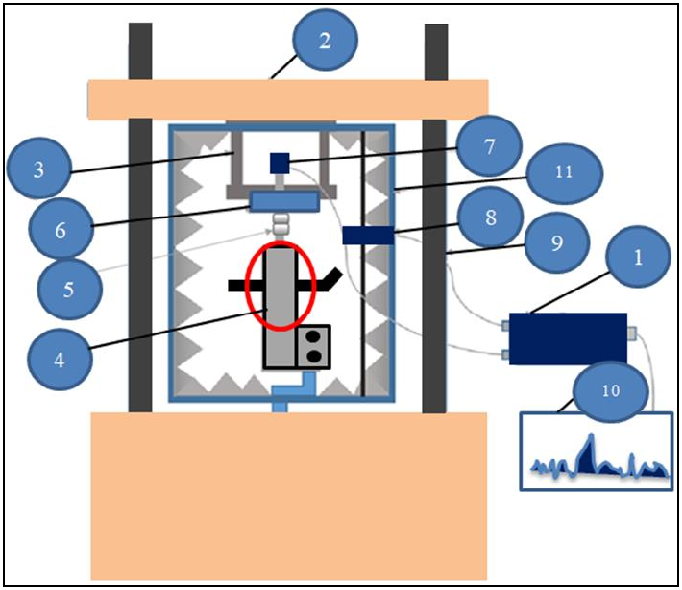
$$\text{Lower frequency}\left(f_o \big/ 2^{1/6}\right) \tag{2}$$

$$\text{Upper frequency}\left(f_o \times 2^{1/6}\right) \tag{3}$$

Pre and post impact of rebound cushion operations are shown in Figure 3. Oil is damped in the vicinity of rebound cushion and rod guide before the impact of cushion on guide. Rod guide is fitted to guide the reciprocating motion of piston rod and avoid further displacement of rebound cushion when comes in contact with cushion. 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.

Figure 2 Schematic diagram of servo-hydraulic test rig (see online version for colours)



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

Impact noise characteristics is evaluated in 1/3rd octave band and it is shown in Figure 4. Listening study is performed to understand the effect of change in frequency components on audible noise levels for human ear. It is observed that difference in noise radiation of 2,000 Hz component is critical as it has attained the well correlation with the human noise perception.

4 FSI method

A pure fluid model is always analysed using Eulerian coordinate system. However, FSI model must be based on arbitrary Lagrangian-Eulerian coordinate system or coupled Eulerian-Lagrangian (CEL coupling) since fluid structure interface is deformable.

Therefore, the solution variables of fluid flow include usual fluid variables, such as, pressure, velocity as well as displacements. Large deformation and low strain formation for Lagrangian structure mesh is generally adopted as orifices are quite thin.

Figure 3 Piston rod model in rebound operation (see online version for colours)

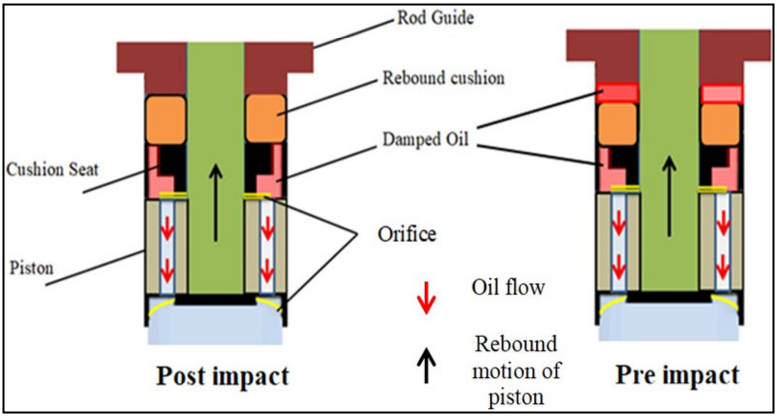
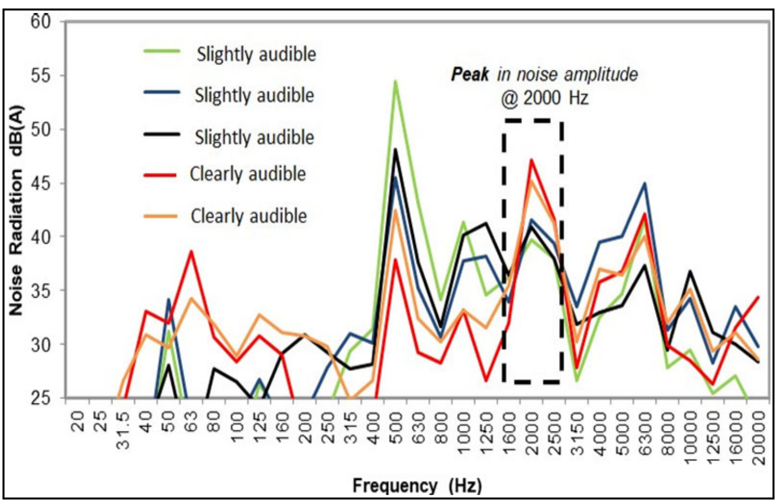


Figure 4 Octave band of noise during rebound cushion impact (see online version for colours)



Following assumptions are made while preparing FSI model (Kulkarni et al., 2019, 2013a, 2013b)

- Contact irregularities of shim valves are ignored.
- Shim structure is assumed as isotropic and homogeneous.
- Thickness of shim valve is assumed as completely uniform.
- Surface defects and material irregularities are also ignored.
- Thermal effects have been ignored as simulation is conducted at room temperature.

Components which are responsible for generation of metallic noise are identified by FSI method. Nonlinear explicit FSI 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 (Qi-Ping et al., 2013). Computational methods are applied by the researcher for investigating the oil flow characteristics of shock absorber assemblies (Benaziz et al., 2013; Kulkarni et al., 2014, 2013a, 2013b; Hou et al., 2011; Bhiungade et al., 2015). 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 2 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 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 ~55°.

Figure 5 Geometry of piston assembly (see online version for colours)

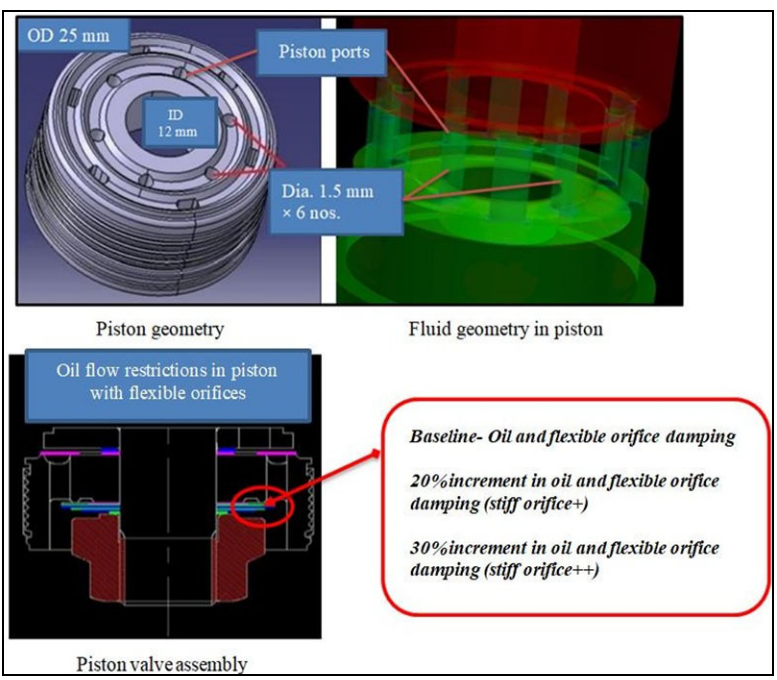


Figure 6 Mesh model of piston assembly (see online version for colours)

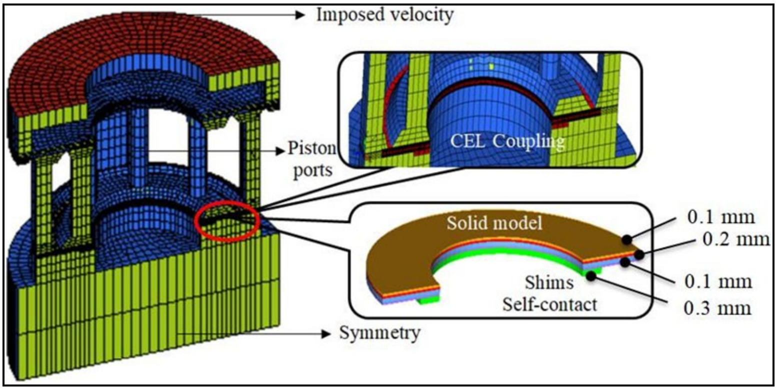


Table 2 Meshing strategy

Mesh description	*Fluid*	*Structural*
Mesh type	Hexahedral	Hexahedral
Approach	Eulerian	Lagrangian
Material	HPCL oil	Steel
Element	50,841	8,058
Nodes	59,677	6,068

Table 3 Material properties

Material property	*Fluid*	*Structural*
Dynamic viscosity (kg/mm-sec)	9×10^{-6}	-
Young's modulus (N/mm^2)	-	2×10^5
Poisson ratio	-	0.3
Density (kg/mm^3)	850×10^{-9}	8×10^{-6}

Table 3 provided the oil and orifice structure properties for the present numerical analysis (Kulkarni et al., 2019). 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 (Kulkarni et al., 2013a, 2013b). Orifice velocities are mapped in order to control the oscillations of the different orifices. Pressure difference across the orifice vicinity has ensured the damping characteristics as expected from the particular orifice assembly. Orifices of different thicknesses are taken to control the oscillations. It can be seen from Table 4 that the imposed velocity of piston (V_{piston}) is 0.1 m/s in which predicted pressure is slightly higher than the experimental results due to higher contact and geometric (large deformation) nonlinearities are associated in real life deformation of orifices. More refinement of mesh with increase in fluid and structure nodes resulted in the negligible change in oil pressure values. Grid independent study is performed with the use of 2D quad element and 3D hexahedral mesh is generated based on it. Quad element sizes of 2, 2.5 mm have been used for grid independent study to optimise the overall computational time.

Table 4 Grid independent study

Total 3D hexahedral nodes	*2D quad element size*	*FSI results (MPa) nodal pressure (P_j)*	*Experimental result (MPa) damping force/rebound area (P_j)*
(Fluid and structure)		*Piston speed (V_{piston}) = 0.1 m/s*	
65,745	2.5 mm	0.23	0.20
68,781	2 mm	0.25	

Figure 7 Correlation between FSI and experimental results (see online version for colours)

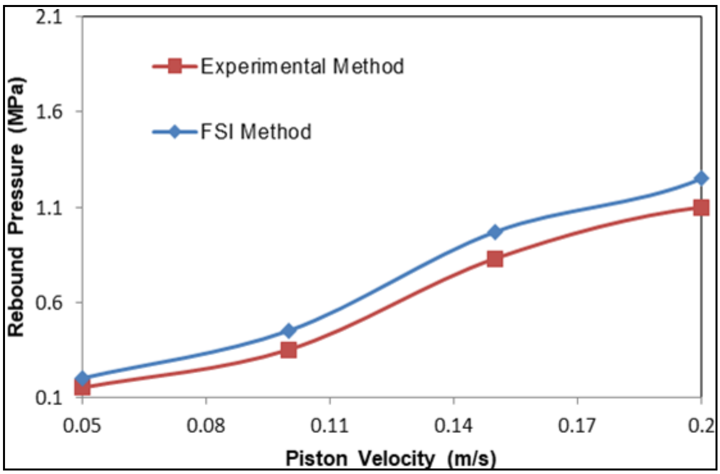
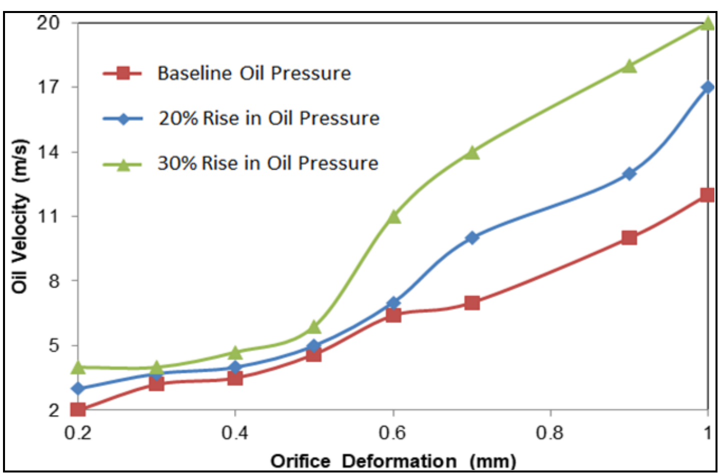


Figure 8 Relationship between orifice deformation and oil velocity (see online version for colours)



The comparison of FSI study has provided the close correlation with experimental method as shown in Figure 7. Geometric (large deformation) and contact nonlinearity defined at orifice interface has achieved the desirable correlation. FSI model has resulted in comparatively lower orifice deformation than the experimental result although both methods have followed similar trend of increase in the rebound pressure with respect to piston velocity.

Figure 8 shows the orifice deformation and oil velocity plot. Pressure values corresponding to orifice deformation values obtained from FSI 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. Oil velocity increased with increase in the oil pressure which also increased the flow induced noise potential (Kulkarni et al., 2019). Increase in oil velocity may help in masking of the impact noise of rebound cushion over the flow induced noise which can be seen further using Full Factorial DOE.

Table 5 Oil pressures (P_j) at orifice valve

Orifice stack	*Fluid pressure at orifice MPa*
Baseline	0.25
Stack 1 (orifice stiffness+) (20% increment in damping)	0.30
Stack 2 (orifice stiffness+) (30% increment in damping)	0.32

Comparatively higher oil pressures (P_j) are generated in piston assembly with respect to baseline configuration. Different stiffness (thickness) of orifice assembly is responsible for more pressure generation as shown in Table 5. Baseline has shown the orifice assembly with contact force value of 236 N. Internal forces are developed between the orifice interfaces and positive force value indicated that appropriate contact interfaces are defined while reviewing the FSI results (Kulkarni et al., 2019, 2014, 2013a, 2013b).

5 Full factorial DOE approach

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 (Gauduin et al., 2008). 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 CFDs (Bhiungade et al., 2015). In this work, noise radiation due to rebound cushion impact is considered as response variable. Three factors are considered in this study which are rebound cushion hardness, height and percentage change in oil pressure. Two levels are considered for analysing the contribution of three factors on this response variable. Cushion impact has occurred gradually with the mating surface. L8 array is prepared after evaluating the results of eight experiments. Oil pressure ensured that the contact forces are appropriate and cushion impact is the only source of noise radiation. Based on the FSI results, shock absorbers are prepared with two more oil pressure values along with baseline to see the effect P_j on this type of noise radiation.

These three factors are involved in cushion impact and 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 (Kulkarni et al., 2013a, 2013b; Bhiungade et al., 2015). Refer to the study on rebound cushion impact in case of standard shock absorber (Niculescu, 2009). 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.

Table 6 L8 array for DOE

Factors		*Noise response @ 2,000 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

5.1 Contribution of individual factors on cushion impact noise

Individual contributions of three factors H, h and P_j are studied using full factorial DOE. As Table 7 shows the P-values less than 0.01 for H, it shows the 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 is real and not by chance. Figure 9 shows the histogram of residual points obtained from the observations. Residual analysis is conducted to determine the normal distribution probability. Histogram has not formed the bell shaped pattern. Since the graph has not shown the clear trend. It signifies the random variation in dataset. Regression was performed and lower R-square value was obtained. The R-square adjusted value is 78.95 %. Difference of 15.79% was obtained between R-square and R-square adjusted which is higher. It is not the whole representation of clicking noise due to rigid body radiation. Provision of oil pressure in working stroke of shock absorber alone has not helped in reduction of cushion vibration and clicking noise. Due to this large deviation, the present study considers the mutual interaction between the selected parameters which is explained in the next section.

Table 7 Full factorial DOE with individual contribution for $(SPL – A)_f$

Source	*Degree of freedom*	*Adj. (SS)*	*F-value*	*P-value*	*Contribution %*
H	*1*	*36.12*	*289*	*0.0*	*72.42*
P_j	1	0.12	1	0.391	0.25
h	1	3.12	25	0.015	6.25
Error	3	10.50	-	-	*21.05*
Total	7	49.87	-	-	*100*

Figure 9 Histogram residual plot for $(SPL - A)_f$ (see online version for colours)

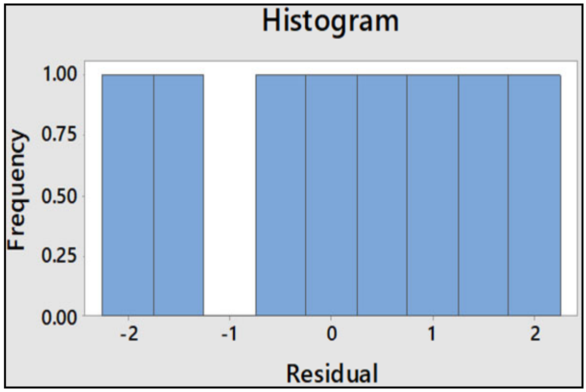
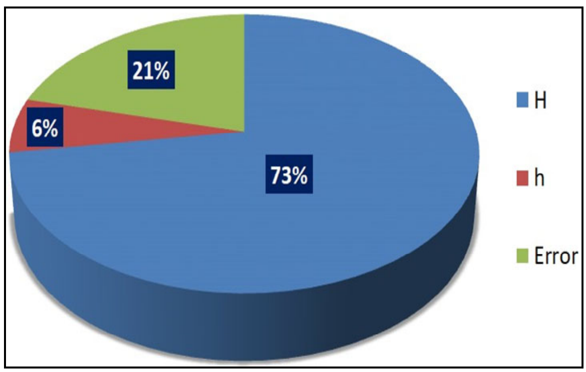


Figure 10 Contribution of parameters on $(SPL - A)_f$ (see online version for colours)



5.2 Contribution of mutual interaction between factors on cushion impact noise

In order to obtain the most significant factors which are responsible for clicking noise refinement. The work is conducted towards the direction of reducing the error percentage and reducing the difference between the R-square and R-square adjusted values obtained in previous section. Mutual interaction between the three factors H, h and P_j are studied for obtaining the accuracy in the model. As Table 8 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. Figure 11 shows the Histogram of residual points obtained from the observations. It has formed the bell shaped graph. It is one of the indicators for goodness of fit (Fernandes et al., 2017).

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 clicking noise due to rigid body radiation. In other words, provision of oil quantity above cushion and cushion height together helped in reduction of cushion vibration and clicking noise. Regression equation is formed to control the impact noise of rebound cushion for the range of selected parameters in the present study:

$$(SPL-A)_f = 52 + 0.21\,H - 1.1\,P_j - 1.2\,h + 0.04\,P_j \times h \tag{4}$$

Table 8 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*

Figure 11 Histogram residual plot for $(SPL – A)_f$ (see online version for colours)

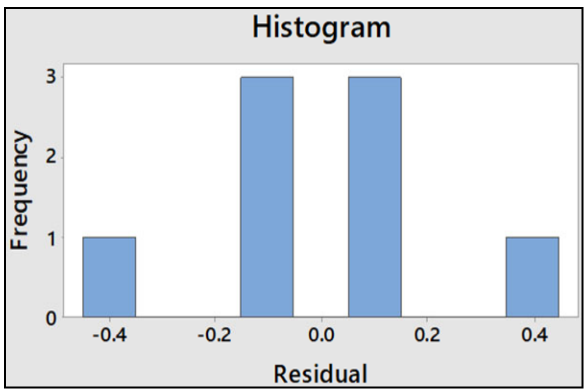
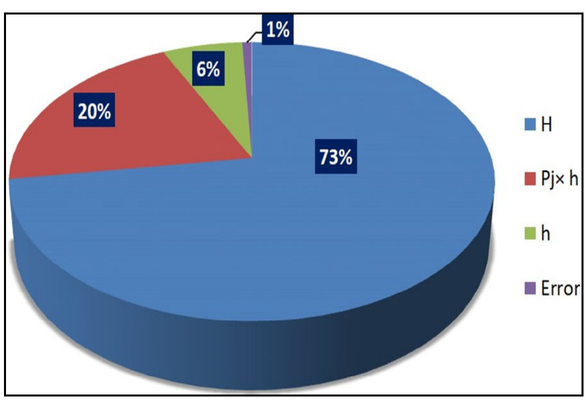


Figure 12 Contribution of parameters and their interactions on $(SPL – A)_f$ (see online version for colours)



6 Conclusions

Clicking noise refinement in a shock absorber is studied using FSI and DOE. The optimisation of significant factors and their interactions involved in impact noise of rebound cushion assembly is discussed. Initially, the research was performed by analysing FSI model of piston and orifice valve assembly to determine their effect on cushion impact noise. CEL formulation is used for simulating shock absorber oil and nonlinear orifices. Contact forces, velocities and deformations of orifices were investigated using FSI with contact nonlinearity approach. Also this work helped to avoid the metallic noise of orifices during rapid deformation. The following conclusions are made from the research work,

- From the full factorial DOE, it is found that combination of factor level for refinement of impact noise is H = 70 Shore A, h = 14 mm, P_j = 20%.
- H and $P_j \times h$ are the significant factors by considering a 95% significance level to refine the impact noise. It avoids the oscillation at the rebound cushion collision and reduces the noise.
- Individual factor P_j has shown negligible contribution to impact noise refinement. Although oil flow/swish noise increased with the increment in oil pressure and velocity, it has not masked the impact noise potential.

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