Panel Contribution Analysis
By using Acoustic Reciprocity

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ABSTRACT

Vehicle Vibration & Noise is evaluated by Sound Pressure Level in cavity. In order to reducing Sound Pressure Level, Vehicle vibration is to reduce.

Vehicle Vibration is affected by engine force, road profile and Structure of Vehicle etc. Engine force and road profile is impossible to change, so to reduce SPL(Sound Pressure Profile) is to lower level of panel vibration. After running normal mode analysis, Mode shapes showed that which modes were global or local. Modal participation to SPL(Sound Pressure Level) got from Modal Participation factor analysis. But the contribution of panel vibration to SPL was not found.

This paper focuses on how to get contribution of panel vibration. Sensitive position of panel was found by applying acoustic reciprocity. To verify results, mass was added to sensitive panel position. SPL was reduced after mass was added. Also similar result got from result of modal participation factor analysis.

This method is applicable to predict position of sensitive panel vibration. All calculation was performed using **MSC.Nastran.**

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INTRODUCTION

Vehicle is excited by engine force, road profile and machinery rotation. Also flow induces vehicle vibration which comes from pressure difference between inner and outer.

Input forces to the body structure are transmitted to the panels that are around the passenger compartment vibrates and excites the acoustic mode of the passenger compartment acoustic modes.

To reduce noise level of the passenger compartment is essential to find noise source and vibration magnitude of panels which is around the passenger compartment.

Two methods are used to find source identification. The one is TPA(Transfer Path Analysis). The other is NTF(Noise Transfer Function Analysis). The former is to find path of vibration, which affect the SPL(sound pressure level). Generally this method is efficient to find mounting vibration affectation and is restricted to design vehicle which is built on because this method need real vehicle. The latter is focused on the effect frequency finding. This method is to find which frequency is correlated with vibration and sound pressure level. Therefore it is useful to find specific frequency. But to find panel contribution is restricted. Another approach is mechanical-acoustic reciprocity method[6)] is fulfilled. That benefit is to find the dominant component.

Each experimental TPA(Transfer Path Analysis) and mechanical acoustic-reciprocity is not to adapt the vehicle that is built on before.

Numerical NTF is possible to find affective vibration point but improper to find contribution of panels. Because to calculate numerical NTF(Noise Transfer function) is needed many times.

To find panel contribution, numerical acoustic reciprocity method is used.

This objective of this paper is to compare the result of modal participation factor, panel participation factor, with acoustic reciprocity and evaluate the method

THEORY

(1) Fluid-Structure Coupling

Fluid-structure motion equation is as follows.

$$\mathbf{M}\ddot{\mathbf{u}} + \mathbf{K}\mathbf{u} = \mathbf{F} \qquad \text{------(1)}$$

$$\mathbf{M} = \begin{bmatrix} \mathbf{M}_s & \mathbf{0} \\ \mathbf{A} & \mathbf{M}_f \end{bmatrix} \qquad \text{------(2)}$$

$$\mathbf{K} = \begin{bmatrix} \mathbf{K}_s & -\mathbf{A}^T \\ \mathbf{0} & \mathbf{K}_f \end{bmatrix} \qquad \text{------(3)}$$

$$\mathbf{F} = \begin{bmatrix} \mathbf{F} \\ \mathbf{0} \end{bmatrix} \qquad \text{------(4)}$$

In case of structure in cavity, Dynamic Reduction is needed. Then

$$\mathbf{u} = \phi_s \xi_s \qquad \text{------(5)}$$

$$\left[\mathbf{m}_s\right] = \phi_s^{\mathbf{T}} \mathbf{M} \phi_s \qquad \text{------(6)}$$

$$\left[\mathbf{k}_s\right] = \phi_s^{\mathbf{T}} \mathbf{K} \phi_s \qquad \text{------(7)}$$

Substitute equation (6) and equation (7) with equation (5), and put into variable

$$\mathbf{M} = \begin{bmatrix} \mathbf{m}_s & \mathbf{0} \\ \mathbf{A}\phi_s & \mathbf{M}_f \end{bmatrix} \qquad \text{------(8)}$$

$$\mathbf{K} = \begin{bmatrix} \mathbf{k}_s & -\phi_s^{\mathbf{T}} \mathbf{A}^{\mathbf{T}} \\ \mathbf{0} & \mathbf{k}_f \end{bmatrix} \qquad \text{------(9)}$$

$$\mathbf{F} = \begin{bmatrix} \phi_s \mathbf{F} \\ \mathbf{0} \end{bmatrix} \qquad \text{------(10)}$$

In case of fluid in structure, the procedure is same as case of structure in cavity.

$$\mathbf{p} = \phi_f \xi_f \qquad \text{-------(11)}$$

$$\left[\mathbf{m}_f\right] = \phi_f^{\mathbf{T}} \mathbf{M} \phi_f \qquad \text{-------(12)}$$

$$\left[\mathbf{k}_f\right] = \phi_f^{\mathbf{T}} \mathbf{K} \phi_f \qquad \text{-------(13)}$$

$$\mathbf{M} = \begin{bmatrix} \mathbf{M}_s & \mathbf{0} \\ \phi_f^{\mathbf{T}} & \mathbf{m}_f \end{bmatrix} \qquad \text{-------(14)}$$

$$\mathbf{K} = \begin{bmatrix} \mathbf{K}_s & -\mathbf{A}^T \phi_f \\ \mathbf{0} & \mathbf{k}_f \end{bmatrix} \qquad \text{-------(15)}$$

After arranged equations, the motion equation is as follows.

$$\begin{bmatrix} \mathbf{m}_s & \mathbf{0} \\ \phi_f^T \mathbf{A} \phi_s & \mathbf{m}_f \end{bmatrix} \begin{Bmatrix} \ddot{\xi}_s \\ \ddot{\xi}_f \end{Bmatrix} + \begin{bmatrix} \mathbf{k}_s & -\phi_s^T \mathbf{A}^T \phi_f \\ \mathbf{0} & \mathbf{k}_f \end{bmatrix} \begin{Bmatrix} \xi_s \\ \xi_f \end{Bmatrix} = \begin{bmatrix} \phi_s^T \mathbf{F} \\ \mathbf{0} \end{bmatrix} \qquad \text{------(16)}$$

s : Structure f : Fluid p : Pressure

M : Mass Matrix K : Stiffness Matrix A : couple Matrix

? : Scaling Factor ? : Modal vector

(2) Modal Participation Factor

The displacement of structure is summation of each mode displacement. Modal participation factor represents participation of mode.

Acoustic Modal Participation Factor

$$\{\mathbf{p}\} = [\phi_f]\{\xi_f\}$$

$$= \sum_{i=1}^{L} \{\mathbf{p}_i\} (= \xi_{fi} \phi_{fi}) \qquad \text{-----(17)}$$

L : the total number of acoustic modes

Structural Modal Participation Factor

$$\{\xi_f\} = [\mathbf{Z2}](\omega^2 [\mathbf{a}]\{\xi_s\})$$

$$[\mathbf{a}] = [\phi_f]^T [\mathbf{A}][\phi_s]$$

$$[\mathbf{Z2}] = (-\omega^2 [\mathbf{m}_f] + \mathbf{k}_f)^{-1}$$

Therefore

$$
\begin{aligned}
\{\mathbf{p}\} &= \omega^{2}[\phi_{\mathbf{f}}][\mathbf{Z2}][\boldsymbol{\alpha}]\{\xi_{\mathbf{s}}\} \\
&= \sum_{\mathbf{j}=1}^{\mathbf{sm}}\{\mathbf{p}_{\mathbf{j}}\}
\end{aligned}
$$
---(18) sm : total number of structural modes

(3) Mechanical-Acoustic Reciprocity

Total structure borne sound at driver ear location is linear summation of partial pressure, which is transmitted by panel vibration

$$\mathbf{P}_{\mathbf{k}} = \sum_{\mathbf{ij}} \mathbf{P}_{\mathbf{ijk}}$$ ---(19)

Partial sound pressure Pijk is calculated as follows

$$\mathbf{P}_{\mathbf{ijk}} = \mathbf{H}_{\mathbf{ijk}}\, \boldsymbol{f}_{\mathbf{ij}}$$ --(20) H :mechanical-acoustical transfer function f : force(operational)

From equation (19) and equation (20)

$$\mathbf{P}_{\mathbf{k}} = \sum_{\mathbf{ij}} \mathbf{H}_{\mathbf{ijk}}\, \boldsymbol{f}_{\mathbf{ij}}$$ ---(21)

Equation (21) is expressed by mounting stiffness and displacement.

$$\boldsymbol{f} = \mathbf{K}(\boldsymbol{x}_{\mathbf{body}} - \boldsymbol{x}_{\mathbf{chassis}})$$ --- (22) K : Mounting dynamic stiffness

If mounting dynamic stiffness is not used, the force at body side is expressed as equation (23).

$$\boldsymbol{f} = [\mathbf{H}]^{-1}\ddot{\boldsymbol{x}}$$ ---(23) [H] : accelerance matrix

From equation (23), transfer function of acceleration is as follows.

$$[\boldsymbol{H}] = \frac{\ddot{\boldsymbol{x}}}{\boldsymbol{f}}$$ ---(24)

Transfer function in equation (24) is expressed as follows.

$$[\boldsymbol{H}] = \frac{\boldsymbol{P}_{k}}{\boldsymbol{f}_{i}}$$ --(25)

From equation (24), (25),

$$[\boldsymbol{H}] = \frac{\boldsymbol{P}_{k}}{\boldsymbol{f}_{i}} = \frac{\ddot{\boldsymbol{x}}}{\boldsymbol{q}_{k}}$$ --(26)

ANAYSIS MODEL

BODY

The detailed finite element model of BIW was constructed by using the linear thin elements that were represented each panel.

The windshield model was constructed by using the linear thin elements and connected with BIW(Body In White) by using spring element which to simulate adhesive binding.

The trimmed body was modeled by adding windshield glass and trims to the BIW. The trim parts were modeled using lumped elements, rigid elements, constraint element, and non-structural element. The closures, seat, heater, blower, radiator, and fan were modeled using the lumped and constraint element to prevent stiffening the vehicle. The bumpers, speakers, lamps, and washer tank were modeled using the lumped and rigid element. The mass of insulation and anti vibration pad was included in the body panels by using the non-structural mass.

CHASSIS SYSTEM

The steering system is composed of several parts which are tie-bar assembly, steering column, steering column jackets, wheel rim, mounting brackets, and spoke etc. Steering shaft, wheel rim, and spoke were modeled by linear beam element, which have general section property. The tie bar and mounting brackets were presented by the shell elements. The connection between steering column and steering jacket was modeled by linear spring elements which value was determined by the test.

The chassis system is consist of several subsystems such as powertrain, front suspension, subframe, rear suspension, and exhaust pipe system. The front subframe, knuckle, lower control arm, and torsion beam were constructed by using the shell elements. The stabilizer bar, exhaust pipe, tie rod, and strut were represented by beam element.

The suspension spring, engine mount, tire, bellows, hangers and all of bushes were modeled by spring elements and lumped mass element. The joints were modeled by using the rigid element and the end release code of the rigid element modified. The mechanical motions in area of power-unit to drive shaft and steering rack to pinion are modeled by using the MPC(Multi-point constraint equation)

TOTAL VEHICLE

The total vehicle model was constructed by connecting Trimmed Body and Chassis System. The connection between trimmed body and chassis system was modeled by rigid elements.

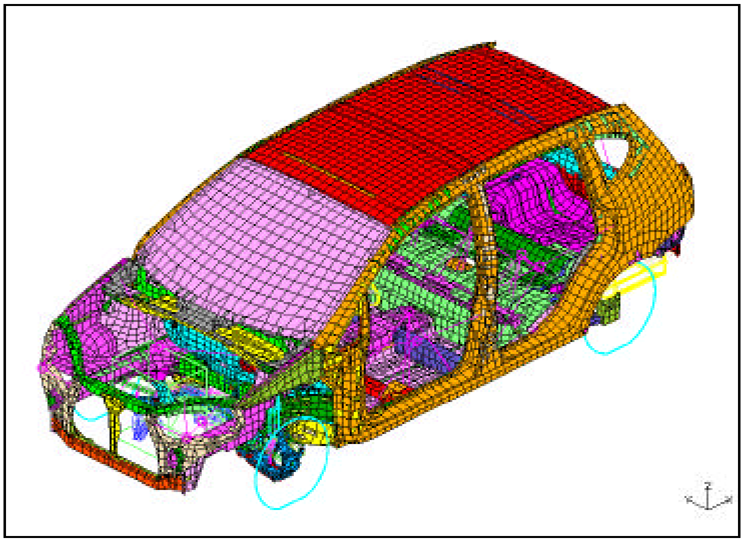


Fig. 1 FE-VEHICLE MODEL

CAVITY

The Cavity was modeled by using solid elements, which represented the cavity volume. Element length is sufficient to present wave until 200Hz. The air density is defined 1.225kg/m^3 and seats were simulated in this model by assigning a higher density.

Fig. 2 shows the cavity model

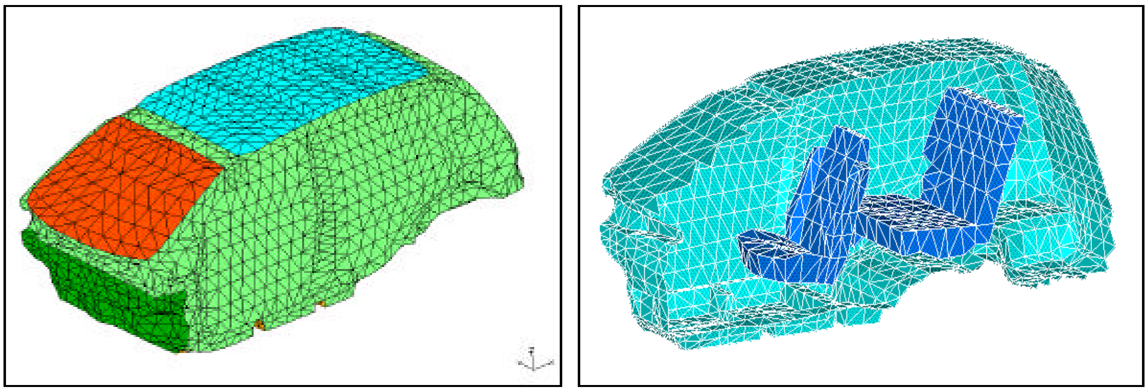


Fig. 2 FE-CAVITY

LOADS

For engine noise, Engine force was calculated. That has two forces, which one is vertical force unbalance and the other is torque oscillation. Vertical force unbalance is associated with the reciprocating motion of pistons. Torque oscillations related to crank slider mechanism are resulting from both reciprocating inertia force and combustion force.

For acoustic reciprocity, acoustic source was used. Acoustic force's magnitude was unit power and was applied to driver's right ear position.

RESULT

(1) Powertain Noise

When engine force excites body through engine mount and drive line, the sound pressure level at front driver's ear location was same as Fig. 3.

There were 4 pesks in Fig 4. Those peaks had higher probability occurring "BOOM"

To reduce the Sound pressure level, the identification of effective mode to SPL was needed. To evaluate the effect of mode, Modal participation factor calculation and Panel Participation factor analysis was implemented.

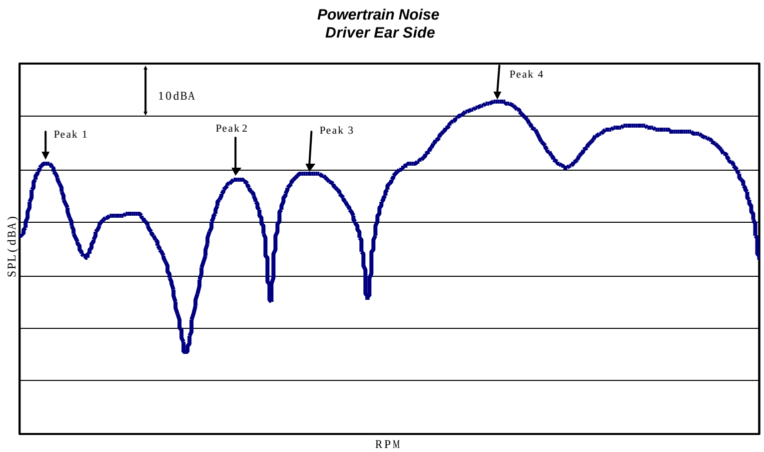
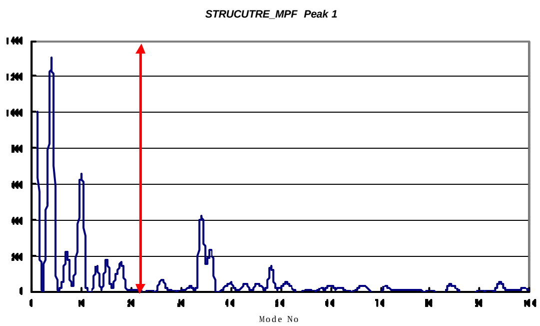


Fig. 3 Powetrain Noise .

(2) Modal Participation Factor Analysis

Fig. 4 to Fig 7 showed modal participation factor at each peak



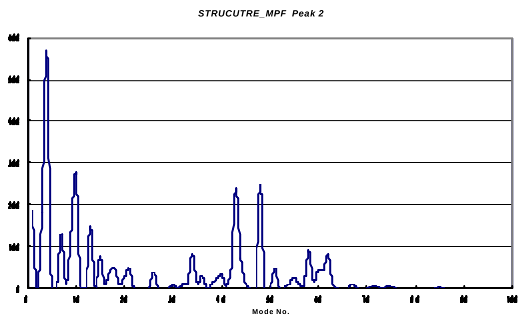
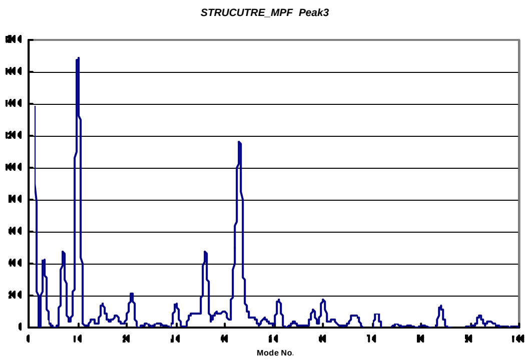


Fig. 4 Peak 1 MPF

Fig. 5 Peak 2 MPF



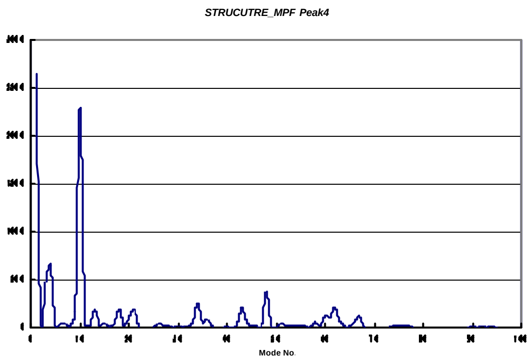


Fig. 6 Peak 3 MPF **Fig. 7 Peak 4 MPF**

Modes were separated 2 parts. The one was rigid body mode, which represented powertrain and suspension mode in low frequency range. Low frequency range was from 0 to 20Hz. The other was flexible mode, which represented body flexibility over 20Hz.

Each peak modal participation factor showed that Lower mode was more effective than higher mode. According to vehicle normal mode analysis, Flexible modes showed over 20th modes. The modes over 20th modes were focused on.

At peak 1, Fig 4 showed that the 34th mode affected the sound pressure level at driver's ear position and higher mode effect was low.

Fig 5 showed that the sound pressure level was mainly affected by 43rd and 48th modes. The 58th and 62nd modes effect was lower than the 43rd and 48th modes but these modes were important.

At peak 3, the 43rd mode was dominant and Fig 8 showed that effective mode was not found. At each peak, the 43rd mode appears significantly.

To lower the level of peak, it was required to assess 43rd mode and development of body

(3) Panel Participation Factor Analysis

Fig. 8 showed panel sets, which were used for panel participation factor analysis. From Fig. 9 to Fig 12 showed the result of panel participation factor. At peak1, the windshield factor was dominant. That means that windshield motion affected the sound pressure level.

At peak 2, panel contribution of roof was dominant in lower frequency but main part was windshield. Fig 11 showed that front floor panel contribution was dominant. Roof and front floor contribution were more dominant at Peak 4. Globally windshield panel contribution appeared.

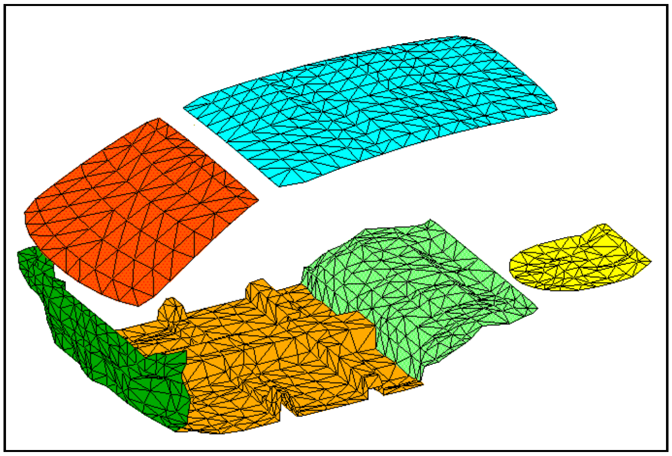


Fig. 8 Panel sets for Panel Participation factor analysis

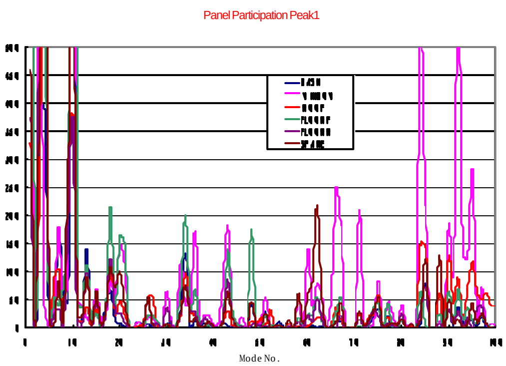


Fig. 9 Peak 1 Panel Participation

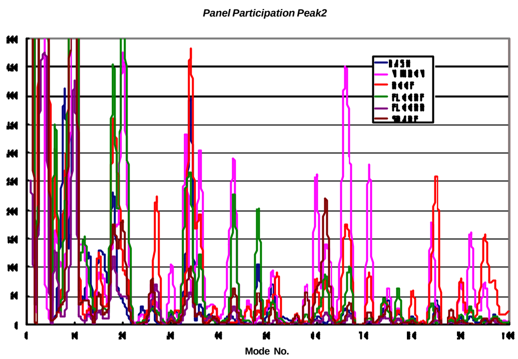


Fig. 10 Peak 2 Panel Participation

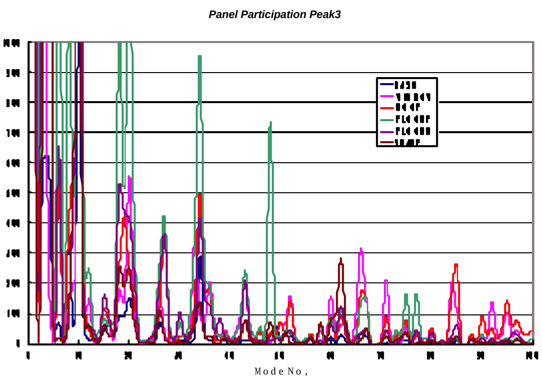


Fig. 11 Peak 3 Panel Participation

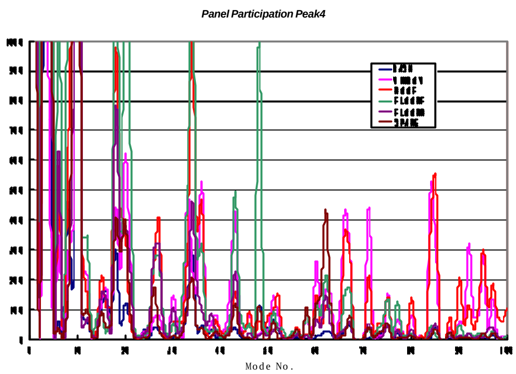


Fig. 12 Peak 4 Panel Participation

(4) Acoustic Reciprocity Analysis

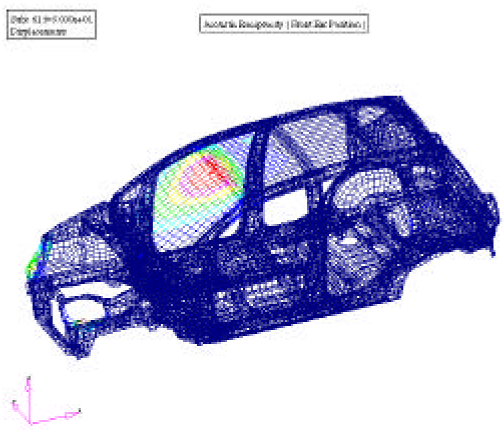


Fig. 13 Peak1

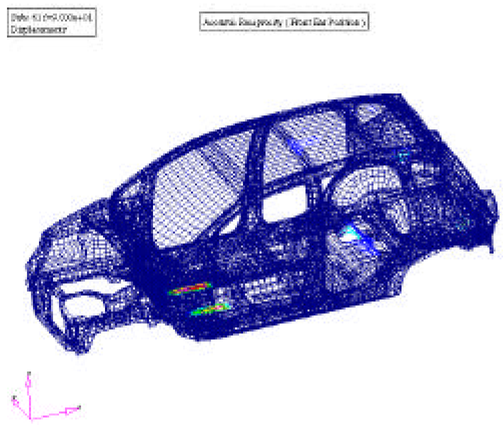


Fig. 14 Peak 2

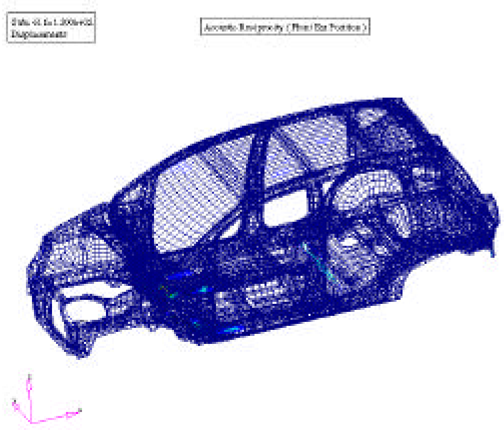


Fig. 15 Peak 3

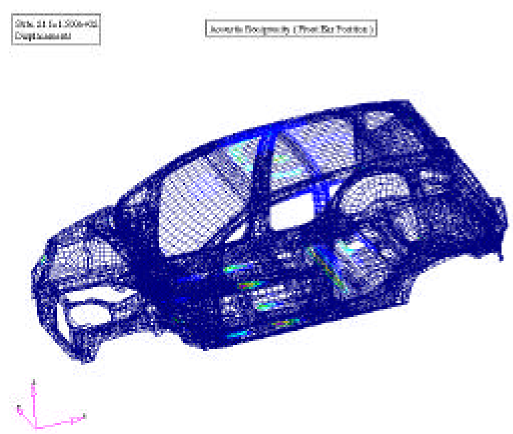


Fig. 16 Peak 4

Fig 13 showed the largest displacement position was windshield. That result was agreed with panel participation factor. Fig 14 showed that sensitive panel position was front floor panel. That position was more precise than panel participation factor. Also the other result fig.s were same as panel participation factor. At peak 4, the result showed that the sensitive panels were roof and front floor panel. At identical frequency between Panel contribution and acoustic reciprocity result was same.

(5) Analysis Verify

To verify acoustic reciprocity method, constant mass was added to largest displacement position.

Fig. 16 showed the sound pressure level was lowered by added mass. This meant acoustic reciprocity method is applicable

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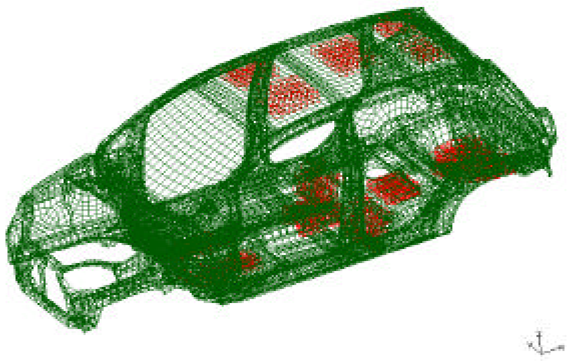

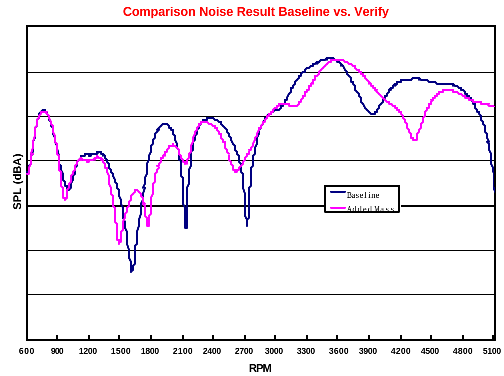

Fig. 17 Position of Mass **Fig 18 Verify Effect of Mass**

CONCLUSIONS

To identify the affective mode was used modal participation factor and panel participation factor. But this method could not find panel position. Therefore this method was not proper. Also NTF(Noise Transfer Function) was not proper to find affective panels.

To find affective panel to sound pressure Peak, Acoustic reciprocity method was used.

After Comparing Modal participation factor, The result of Panel participation factor was agreed with acoustic reciprocity method,

It was proven to agree with two results. Therefore the acoustic reciprocity method was applicable to find sensitive panel position to sound pressure peak.

This method benefit was to decide position of deadning material and development of Body.

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