

Elementary Body Structure Analysis

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ABSTRACT

Recently vehicle development timeline is becoming shorter, so there is an urgent need to be able to develop vehicles with limited resources. This means the efficiency of the body structure development process must be improved. Specifically it is important to reduce the amount of design re-work required to meet performance targets as this can have a large influence on the body development time.

In order to reduce the afore mentioned design re-work, we developed simple calculation models to apply a “V-Flow Development Process” to the preliminary stage design of the automobile body structure.

The “V-Flow” advantages are as follows: (1) simple and easy to use, (2) defects are found at early stage, (3) avoids the downward flow of the defects.

The advantage of preliminary stage design is that there is design flexibility since not many specifications have been determined yet.

The advantage of simple calculation models is their lightness as to input data and calculation time which makes them easy to implement for preliminary stage analysis. It is easy to analyze many changes quickly using these types of simple models.

This paper proposes a V-flow development workflow suitable for development of a body at the preliminary stage and the required methods used at each phase of the workflow.

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INTRODUCTION

Usually, we treat the body-in-white (BIW) as a single component, but it is too big to apply a V-process to body design, as shown in [Figure 1](#). This means we are not able to analyze during each part of design, and re-work will be caused as shown in [Figure 2](#). Therefore, we need to introduce a new method to break down performance to smaller parts so that we can understand BIW performance based on component level performance. In this way we can realize the advantages of the V-Flow process as shown in [Figure 3](#). However, when the BIW is divided into components, there is not yet sufficient understanding of how to combine the stiffness values of each part to calculate the stiffness value and natural frequency of the entire BIW. This makes it difficult to carry out development following a flow whereby targets accounting for the stiffness contribution of each part are set and then structures are designed that satisfy those targets.

FOA (First Order Analysis) [1] has been proposed as a CAE technique that can easily be applied in the concept design stage, but this approach examines the validity of an assumed design, and is not primarily intended for setting targets for the stiffness of each part. In addition, although it is a high-functionality method of analysis that

uses beam elements and panel elements, the method is based on numerical analysis, which makes it difficult to intuitively understand the relationship between the individual parts and the body as a whole.

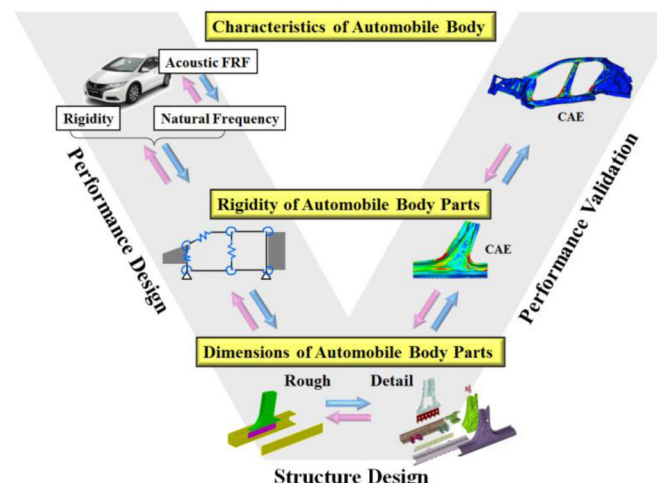


Figure 1. Development Workflow of Automobile Body

With this background, we developed the spring-mass model which relates stiffness of parts to the stiffness value and natural frequency of the entire BIW and, therefore, enables setting the part-level targets quickly. We also developed a statistical simulation model to give rough estimation of the structure required for each part to reach the part-level targets.

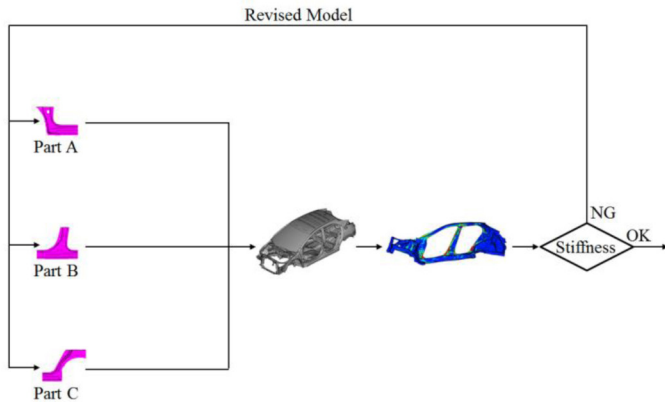


Figure 2. Workflow without Analysis during each part of design

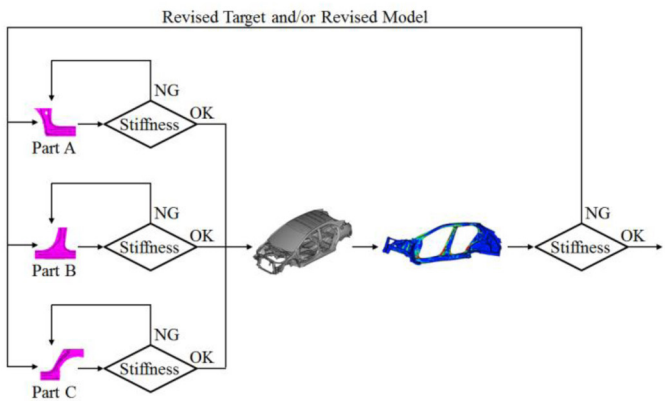


Figure 3. Workflow with Analysis during each part of design

1. ASSIGNMENT OF BODY MODE FREQUENCY

After a target for acoustic frequency-response-function (FRF) has been defined by investigating transfer path analysis (TPA) results of a previous model, we start to estimate the targets of body mode natural frequency. Hence, we developed a simple calculation method [2] of acoustic FRF to be used at the preliminary stage.

Acoustic FRF P_{Resp}/f_s is given in equation (1), (2), (3)

$$P_{\text{Resp}} = \Phi_{A,\text{Resp}} \xi_A \quad (1)$$

$$\left(\begin{bmatrix} \Lambda_S & \mathbf{a}_{SA} \\ \mathbf{0} & \Lambda_A \end{bmatrix} - \lambda \begin{bmatrix} \mathbf{I} & \mathbf{0} \\ -\mathbf{a}_{SA}^T & \mathbf{I} \end{bmatrix} \right) \begin{Bmatrix} \xi_S \\ \xi_A \end{Bmatrix} = \begin{Bmatrix} \Phi_S^T \mathbf{f}_S \\ 0 \end{Bmatrix} \quad (2)$$

$$\mathbf{a}_{SA} = \Phi_S^T \Lambda_{SA} \Phi_A \quad (3)$$

where,

P_{Resp} : Sound pressure at the response point

Φ_A : Acoustic cavity modes consist of 0th constant and first harmonic mode

Φ_S : Structural modes consist of second-order bending of body, first front glass, and first tailgate mode.

Λ_A, Λ_S : Eigen value matrix of acoustic modes and structural modes respectively

ξ_A, ξ_S : Modal participation factor of acoustic modes and structural modes.

$\mathbf{a}_{SA}$: Modal coupling matrix between acoustic modes and structural modes.

$\Phi_{A,\text{Resp}}$: Contribution of the acoustic cavity mode to the response point

Λ_{SA} : Coupling matrix that indicates surface area between acoustic and structure

Acoustic cavity modes Φ_A are computed in two dimension FE model. The FE model is automatically generated based on input geometry data such as body length, height, seat position, etc. as shown in Figure 4.

Structural modes Φ_S are estimated from the regression curves. The curves that show the relationship between mass and mode vector are shown in Figure 5.

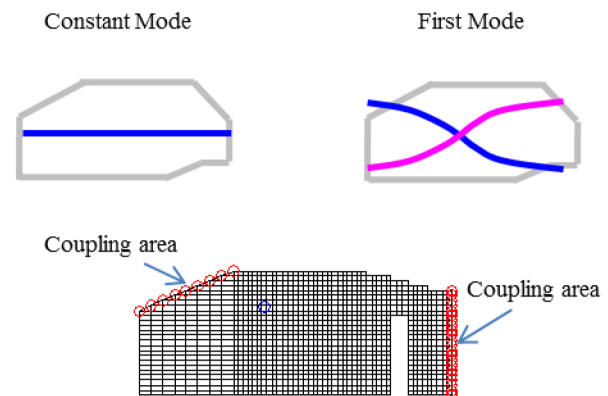


Figure 4. Two Dimension FE Model to Calculate Acoustic Cavity Modes

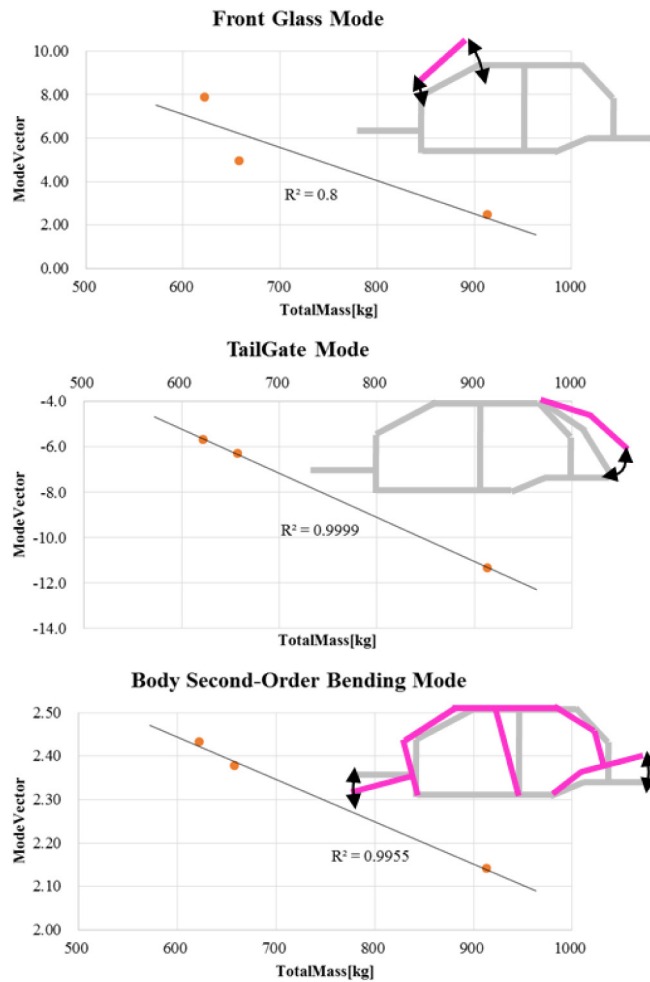


Figure 5. Regression Curves Showing Relationship between Mass and Mode Vector

Figure 6 shows iterated acoustic FRFs using structural mode natural frequency as parameters. We are able to get a new acoustic FRF meeting the target and the structural mode natural frequency required from these results.

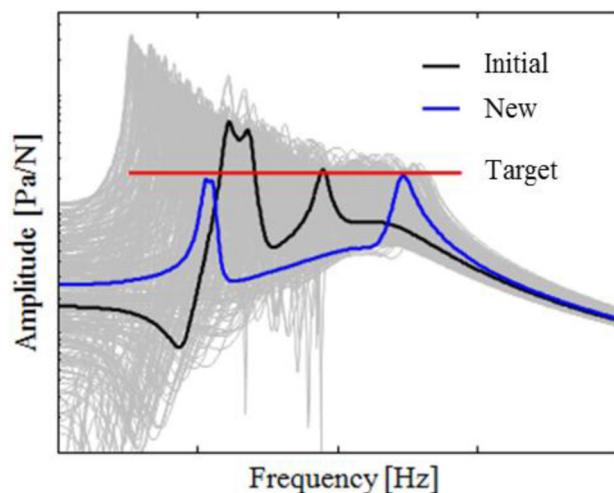


Figure 6. Acoustic FRFs

2. ASSIGNMENT OF PARTS STIFFNESS

2.1. Second-Order Bending Mode of Body

2.1.1. Natural Frequency

If the typical body is approximated as a uniform beam, it is possible to calculate the natural frequency of the body bending mode using basic beam equations [3,4,5]. Figure 7 illustrates the second-order bending mode of the automobile body, which obeys the dynamic bending constraints shown in Figure 8 and the static bending and load constraints shown in Figure 9. On this basis, the basic equation for f_2 , the natural frequency of second-order bending of the body, is given in equation (4).

$$f_2 = \frac{\pi}{L} \sqrt{\frac{M}{3\theta m}} \quad (4)$$

$$\frac{M}{\theta} = \frac{F_A h_A L}{\frac{\ell_A \delta_A}{h_A} + \frac{\ell_B \delta_B}{h_B}} \quad (5)$$

Where, m is the mass. In addition, second-order bending deformation of the body, as shown in Figure 10a, is expressed by moment M and bending angle θ . Figure 10a can be converted to Figure 10b using forces F_A and F_B and bending angles θ_A , θ_B , and θ , with the stiffness of the second-order bending of the body given in equation (5), and the natural frequency f_2 given in equation (6).

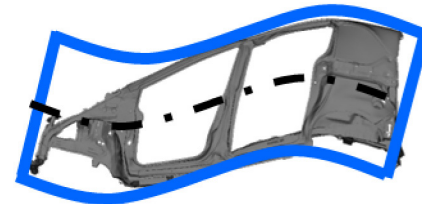


Figure 7. Second-Order Bending Mode of Body

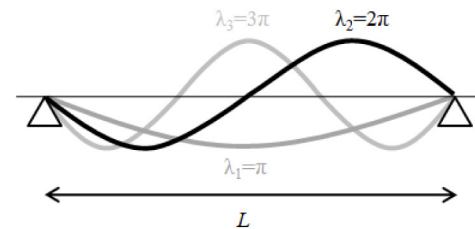


Figure 8. Bending Modes of Uniform Beam

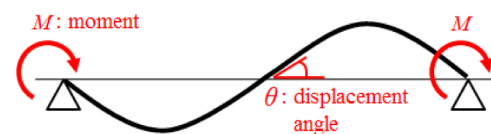


Figure 9. Static Bending of Uniform Beam