

Reduction of Engine Noise in the New Accord Using Hybrid CAE Technology

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ABSTRACT

Hybrid CAE models were constructed for the 2003 Accord to reduce cabin noise induced from engine order vibration. The Hybrid CAE method utilizes a combination of two different approaches: numerical and experimental analysis. Targets for each system's contribution to cabin noise were established on the basis of tests of the prototype during the initial stage of development. The achievement of these targets has significantly reduced engine noise in the new Accord.

1. Introduction

Reductions in product development periods and in the number of prototypes have been constant matters of concern in recent years. Even if the number of experimental models is cut back and development periods are shortened, there will be no benefit if the end product does not perform well. Recently, dramatic improvements over previous models and various special features have been required to maintain a competitive edge and meet the demands of the evolutionary trends in the market.

A broad definition of 'product performance' includes many factors, from handling, acceleration performance, noise and vibration, ride comfort, to collision safety, cost, and weight. Many of these performance indicators stand in contradiction to each other. Performance goals are set by the product concept, but when elements conflict with each other it is essential to decide how to balance them. Those decisions have the potential to sidetrack development and could even result in a product with little resemblance to the initial concept.

The vibration and noise performance of a vehicle is determined by the interrelationship between various systems including the engine, subframe, body, bushings, and the suspension. Efforts to improve them can all too easily lead into the situation mentioned above. Therefore, throughout development, it is necessary to bear in mind which system is

most efficient for handling issues and how to minimize the effects of elements that detract from other performance parameters.

Over a decade ago, the investigation of product performance was centered on a process of creation and analysis of prototypes; multiple prototypes were made in succession until the final one matched the product concept. Today, when the product must fulfill high performance requirements within a short period and research must be carried out to prevent features that compete with other desired performance parameters that may alter the product concept, it is vitally important to assure vehicle performance with a high degree of precision before any prototypes are made.

The object of this report is cabin noise caused by engine vibration, mainly that above 100 Hz. At present, it is not realistic to create numerical models of all the components affected by this region and attempt to analyze the aggregate vehicle model for highly precise predictions of performance. Consequently, a hybrid CAE technique tying numerical models to physical models has been employed; its superiority has been previously reported^{(1), (2)}. In the developmental process presented here, good results were obtained when the technique was applied to the new 2003 model Accord (in-line 4-cylinder engine). The experimental results with this CAE technique are included in this report.

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2. Determining Contributions to Cabin Noise and Setting System Targets

The targets for cabin noise in the new Accord and those assigned for each system are described below.

2.1. Targets for 2003 Models

The cabin noise levels and the reduced noise target for the basic mass-produced model (2.4 liter displacement, for U.S. market) are shown in Fig. 1. The aggressive goal of a reduction of about 5 dB under the previous model was set for the 2003 Accord. This was mandated in the concept so that a majority of users would notice the improvement.

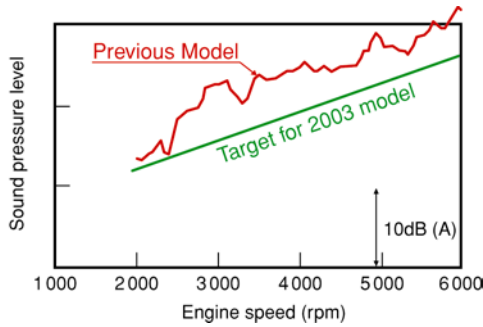


Fig. 1 Interior noise target for 2003 Accord compared with previous model

2.2. Determining Cabin Noise Contribution Levels and System Targets

The targets were studied for each system by analyzing the transfer paths for engine vibrations. Orders second through eighth were analyzed to include all the input in the structure borne noise frequency range (up to 800 Hz).

Equation (1) shows the components of noise S_i on each path due to the operational engine vibrations X_{Ei} , body vibrations X_{Bi} , engine mounting dynamic characteristics k_i , and body noise transfer function H_i , of engine mounting points i . Equation (2) shows the summation of the transfer path components to the total cabin noise S_{all} .

$$S_i = (X_{Ei} - X_{Bi}) \times k_i \times H_i \quad (1)$$

$$S_{all} = \sum_{i=1}^{All} S_i \quad (2)$$

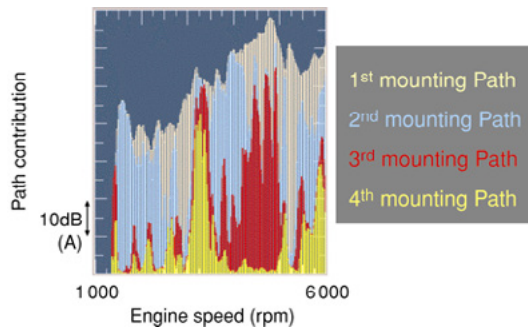


Fig. 2 Transfer path contribution in 1998 model

The transfer path contribution for the 6th order component of the cabin noise in the mass-production model (1998) is shown in Fig. 2. The paths with the highest contributions at the various engine RPM speeds are clear in this figure.

The second through the eighth orders were investigated in all 18 transfer paths (6 engine mounting points $\times$ 3 translational axes) prior to allocating the targets for the systems. Targets were set for the engine vibration inputs and for the body noise transfer function for each path. When calculating engine vibration inputs, body vibrations were assumed to be 20 dB lower than the engine vibrations.

Parameters for bushings such as those at the engine mounts and the subframe floating mounts were taken from component measurements. These known results were used as constraint conditions for establishing the hybrid model.

3. Verification of Engine Vibration

Engine vibration is expressed in Eq. (3) by the product of the forces F released inside the engine (engine interior forces) and the frequency response functions H^{EB} describing the transfer characteristics from the interior of the engine to the tips of the engine mounting brackets. These frequency response functions H^{EB} are again shown in Eq. (4) as being composed of distinct frequency response functions of the engine and brackets.

$$\{X_E\} = \{H^{EB}\} \cdot \{F_i\} \quad (3)$$

$$H^{EB} = H_{ts}^{Brkt} (H_{ss}^{Eng} + H_{ss}^{Brkt} + k)^{-1} H_{si}^{Eng} \quad (4)$$

H^{Eng} :Frequency response function matrix of the engine (i, interior points; s, connection points)

H^{Brkt} :Frequency response function matrix of the brackets (t, bracket tips; s, connection points)

k :Stiffness matrix of connection structures

H^{EB} :Frequency response function matrix of the combination of the engine and brackets

F :Engine interior forces

The frequency response function matrix of the engine was constructed as an experimental model using the measured data, and the bracket frequency response matrices were made using finite element (FE) models. The engine interior forces were calculated using cylinder interior pressures and engine vibrations⁽¹⁾.

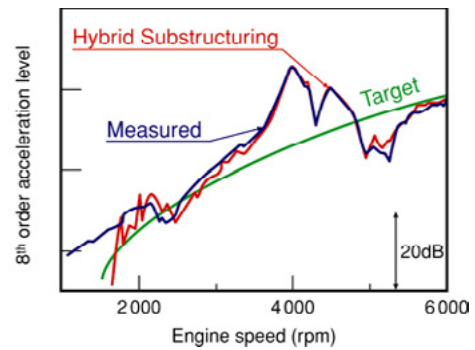


Fig. 3 Estimated 8th order vibration at 3rd engine mounting bracket

The test results at the 3rd engine mount for 8th order in the initial prototype are shown in Fig. 3. It is clear that the results of the hybrid calculations matched well with the measured data.

This was shown to be a good model, as nearly the same levels of precision were found for the second, fourth, sixth and eighth orders at the other mounting points. It was also clear that the prototype did not attain the targets set by allocating the transfer path contributions to cabin noise.

3.1. Experimental Model of Engine

An analysis of the frequency response matrix of the initial prototype revealed that the mating portions of the cylinder lower block and the transmission block were large factors in the failure to meet the targets in the region below 500 Hz. While the engine mount brackets connecting to the rear wall of the cylinder block were factors in the failure to meet the targets at frequencies above 500 Hz. Figure 4 shows the portions of the experimental engine model and of the blocks needing shape improvement, and Fig. 5 shows the corresponding portions on the rear wall of the cylinder block and the transmission case.

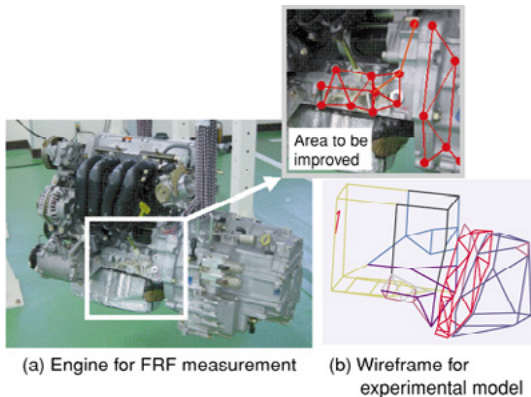


Fig. 4 Experimental engine model and improve section of powertrain block

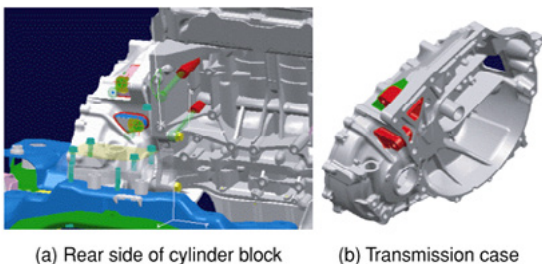


Fig. 5 Improved area for connecting points of engine mounting brackets

In addition to the component shape improvements just described, stiffeners were added in two patterns to see if they would improve the frequency response function matrix as well. Figure 6 shows the results of the proposed modifications to the cylinder block response function (excitation at the 5th journal). One of the stiffener designs proved effective for meeting the goal and was used.

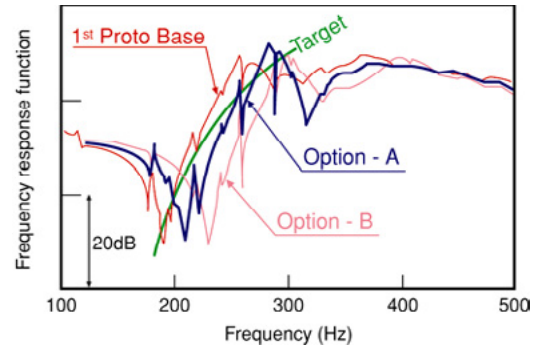


Fig. 6 Comparison of powertrain stiffness with two optional countermeasures

3.2. Finite element Model of Bracket

Figure 7 shows all of the engine mountings. All of the engine mounting brackets were updated as described in an earlier report by the authors⁽¹⁾; this provided improved precision in calculation of the eigenmodes.

The number of fasteners in the first and second mounting brackets was changed from three in the previous model to four in the new model. The shapes of the tip and the attachment region were changed as suggested by an analysis of the transfer function and components were thinned to reduce weight. Other mounting brackets were modified accordingly and the material of four of the six brackets was changed to aluminum.

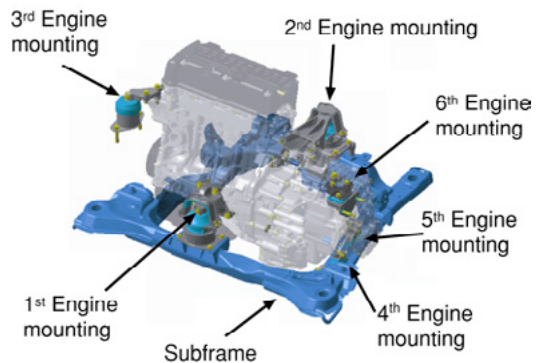


Fig. 7 Newly developed engine mounting system

3.3. Results on Engine Vibration

The engine vibration levels accomplished in the 2003 Accord are shown in Fig. 8. The goals were reached due to the measures taken in the various locations: the stiffening of

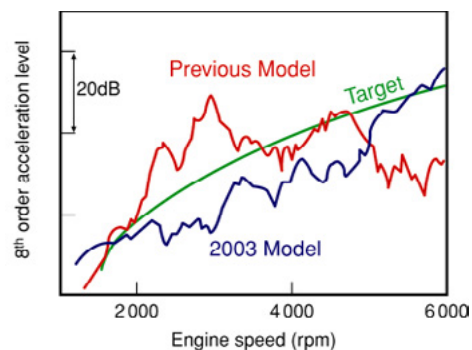


Fig. 8 Improvement of operational engine vibration-8th order at 2nd engine mounting bracket

the power plant, of the block walls, and of the brackets. A large reduction in engine vibration of 5 – 20 dB was accomplished in relation to the previous model. Similar reductions in engine vibration (second through eighth orders) were seen in the other mounting locations.

4. Verification of Body Noise Transfer Function

The same process of modeling was carried out as for engine vibrations. Four vibration transfer paths through the subframe and two paths directly to the body were investigated.

4.1. Noise Transfer Path in Subframe

A frequency response matrix of the subframe was constructed from the FE model, and a frequency response function matrix of the body was constructed from experimental data. Figure 9 presents the model of the subframe. As in the brackets, the model was updated in order to improve the correlation with the experimental models.

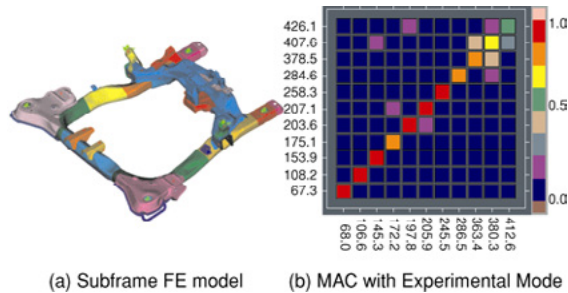


Fig. 9 Subframe FE Model

Table 1 shows the errors of the calculated eigenvalues in relation to the experimentally determined figures and the MAC values. The error stayed within about 5% up to the ninth mode. The MAC value remained above 0.8 or more, indicating a good correlation between the models.

The results obtained by combining the analytical subframe model with the experimental body model in the initial prototype are shown in Fig. 10. The Hybrid model showed a high correlation with the experimental model until

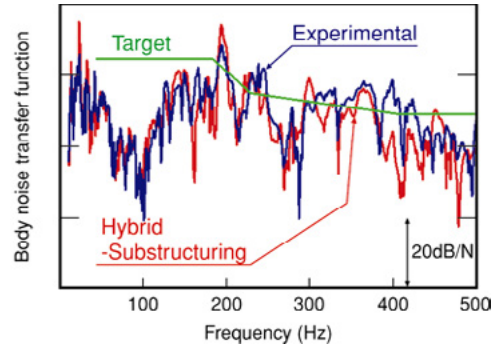


Fig. 10 Body noise transfer function of 2nd engine mounting point on subframe

the 380 Hz region, indicating that high precision had been maintained. In the vicinity of the tenth and eleventh modes, where the MAC value fell below 0.7, the hybrid calculations significantly underestimated the test results. This was noted for reference during later experiments.

It became clear when investigating the frequency response matrices for both the subframe and the body models that improvements in the subframe members, use of floating structures, and improvements to the attachment structures of the body would be necessary in order to attain the targets.

For the subframe, major structural modifications based on the use of a hydro form cradle allowed improvements of 84% in the bending mode and 29% in the torsional mode (compared to the previous model).

4.2. Noise Transfer Path in Body

The body frequency response matrix was found using a measurement setup similar to the one shown in Fig. 11. Figure 12 shows the results with the hybrid model composed of the physical model of the initial prototype and the FE model of the mounts; this indicates a good match with the prediction. It was evident that the frequency response matrices of both the body and the mounts would have to be improved in order for noise transmission through this path to meet the target.

The overall height of the 3rd engine mount was greatly reduced and it was changed to a low-CG (CG 40% lower than in the previous mount), high-stiffness hydraulic design, integrated with the body attachment bracket. The 6th mount was reconfigured to resin-steel design, which offered high dynamic stiffness. Both are shown in Fig. 13.

Table 1 Correlation of experimental subframe mode

No.	Experimental Mode Hz	FE Mode Hz	Difference		MAC
			Hz	%	
1	67.3	68.0	0.8	1.1	0.97
2	108.2	106.6	-1.7	-1.5	0.96
3	153.9	145.3	-8.6	-5.6	0.95
4	175.1	172.2	-2.9	-1.6	0.79
5	203.6	197.8	-5.8	-2.9	0.87
6	207.1	205.9	-1.3	-0.6	0.86
7	258.3	245.5	-12.9	-5.0	0.96
8	284.6	286.5	1.9	0.7	0.80
9	378.5	363.4	-15.1	-4.0	0.81
10	407.6	380.3	-27.3	-6.7	0.62
11	426.1	412.6	-13.5	-3.2	0.58



Fig. 11 Experimental trimmed body

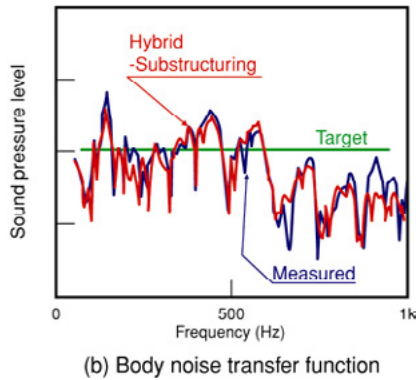
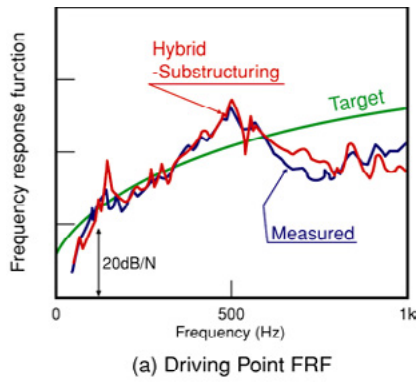


Fig. 12 Validation of sub-structuring for 3rd engine mounting FE model with experimental body



Fig. 14 FE model of trimmed body

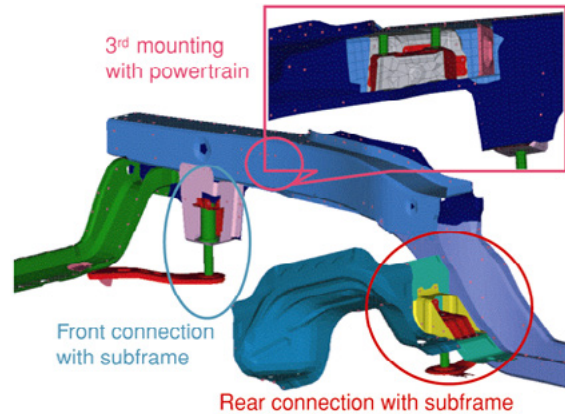


Fig. 15 Area for connection with subframe

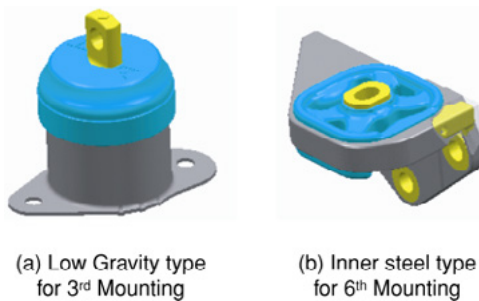


Fig. 13 3rd and 6th engine mounting

It was noted that, within this body frequency response function matrix, the dynamic stiffnesses at the mounting points had to be increased by factors of 6 to 10.

Experimental data was used in the frequency response function matrix, but FE models of the attachment structures were built and verified as valid for investigation of these structures. The trimmed body model is shown in Fig. 14 and the FE models of the attachment structures are shown in Fig. 15.

Since it was difficult to assure high accuracy in this frequency region at the body attachment points in the calculations required for this report, the flexibility of the hybrid model in permitting deletion of structure before performing experiments proved to be very useful.

4.3. Results on Body Noise Transfer Function

The body dynamic stiffness target was attained by shape improvements of both the collar structure at the body attachment bolts and the interior cross section of the bulkhead in addition to incorporation of the new engine mounts. Figure 16 shows the resulting dynamic stiffness levels of the low-CG 3rd engine mount body connection point.

The target for the body noise transfer function at the engine mounting points was achieved by structural improvements of the subframe, possible through hydro forming manufacturing, increasing isolation by making the subframe floating, and improvement of the dynamic stiffness at the body mounting points. The body noise transfer function at the 2nd engine mount on subframe is shown in Fig. 17.

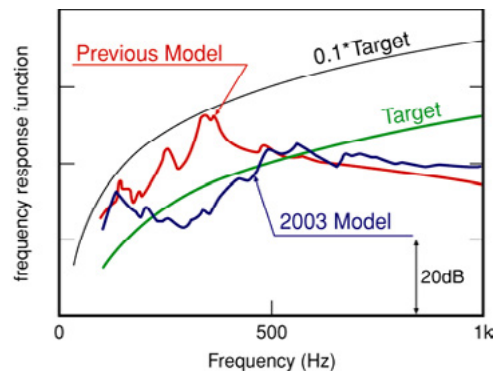


Fig. 16 Improvement of dynamic stiffness for 3rd engine mounting point on an actual body

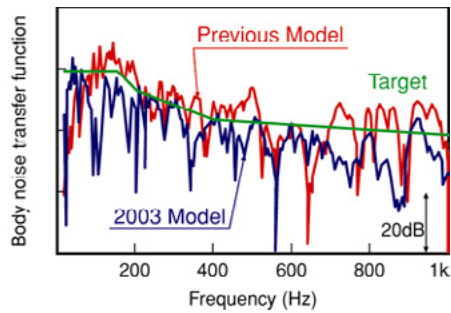


Fig. 17 Improvement of body noise transfer function at 2nd engine mounting point on a subframe

5. Cabin Noise Levels

As shown in Fig. 18, the goal for the 2003 Accord cabin noise was reached through this effort to achieve the targets for each system. A significant noise reduction compared to the previous model was accomplished.

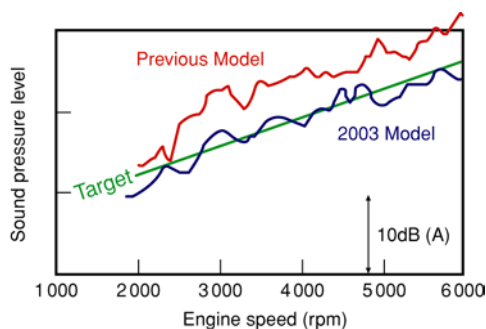


Fig. 18 Improvement of cabin noise

6. Summary

A hybrid CAE model was constructed for use in the development of the 2003 Accord. The target for each system of the model was set after its contribution had been quantitatively associated with the noise reduction target. As a result of the steps taken to achieve the targets set for each system, a large reduction in engine noise transmitted to the cabin was accomplished. The new Accord met the original vehicle performance objectives set at initiation of development.

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