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ANALYSIS PROCESS APPLIED TO A HIGH STIFFNESS BODY FOR IMPROVED VEHICLE HANDLING PROPERTIES

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ABSTRACT−This paper describes the process of analyzing vehicle stiffness in terms of frequency band in order to improve vehicle handling. Vehicle handling and ride comfort are highly related to the systems such as suspension, seat, steering, and the car body design. In existing analytical processes, the resonance frequency of a car body is designed to be greater than 25 Hz in order to increase the stiffness of the body against idle vibration. This paper introduces a method for using a band with a frequency lower than 20 Hz to analyze how stiffness affects vehicle handling. Accordingly, static stiffness analysis of a 1g cornering force was conducted to minimize the deformation of vehicle components derived from a load on parts attached to the suspension. In addition, this technology is capable of achieving better performance than older technology. Analysis of how body attachment stiffness affects the dynamic stiffness of a bushing in the attachment parts of the suspension is expected to lead to improvements with respect to vehicle handling and road noise. The process of developing a car body with a high degree of stiffness, which was accomplished in the preliminary stage of this study, confirms the possibility of improving the stability performance and of designing a lightweight prototype car. These improvements can reduce the time needed to develop better vehicles.

KEY WORDS : High stiffness body, Load transfer path, Body attachment stiffness, Cornering force

1. INTRODUCTION

Development of a car body with a high degree of stiffness is needed to improve vehicle handling, crashworthiness, and durability, as well as to reduce noise, vibration, and harshness (NVH), all of which determine the market quality of a vehicle. Such development, however, may lead to an increase in weight or cost due to reinforcement of materials or thicker paneling. Additionally, because the increase in weight conflicts with the desirable objective of reducing the fuel expenses of a vehicle, a design for minimization is needed to satisfy both objectives. (Kim *et al.*, 2005)

This paper describes a process for analyzing vehicle stiffness in terms of frequency bands for the purpose of improving vehicle handling. A technical trend in the literature involves the use of a new simulation method with experimental modal analysis for the prediction of body deformation during the operation of a vehicle. Studies have highlighted the relationship between the vehicular body mechanism and the vehicle handling performance. The existing analytical processes were developed with a focus on NVH performance; hence, they tend to not thoroughly examine the effect of these factors on the car body, particularly in the band where the frequency is lower than 20 Hz. However, to analyze how a high degree of stiffness in a low-frequency band affects vehicle handling, this paper described a static stiffness analysis for a 1g cornering force. Our results compare favorably with other results in the literature. With older technology, the suspension performance is evaluated on the premise that a car has a rigid body; however, this premise ignores the fact that cars experience torsion of the body or local deformation in the suspension attachment parts. As a result, the estimation of performance is often inaccurate.

To determine how static stiffness affects vehicle handling, this study analyzed the relationship between body displacements and the lateral movement of loads in suspension attachment parts. In particular, this paper explained the loads caused by the roll delay in the front and rear suspension, and its findings are used to determine the potential for improving the yaw rate or response to vertical-direction speed.

For an effective evaluation of durability, stiffness, and the NVH of vehicle parts designed in the early stage of development of a new car, this study conducted the

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following analytical processes (Michael *et al.*, 1997).

First, an extreme situation that can occur during vehicle operation is established and the maximum load that a vehicle can receive is calculated. Next, the load at the wheels, which is the load transfer path from the road surface to the vehicle, is measured.

Following that, this study obtained the loading on the suspension members, joints, and the bushing area of the designed suspension system, and the load was added to the wheels. After using finite elements to model the suspension members, the load on the joints and the bushing was entered to evaluate the stiffness of the individual members.

The suspension system was integrated into a reliable body model, and the static and dynamic stiffness analysis makes it possible to examine the effect on the body, chassis, and bushing attachment parts.

In the early stage of design, this study used a mother model and described the design guidelines and analysis of the static stiffness of the car body and the suspension system for the purpose of improving vehicle handling. By studying the correlation between the dynamic stiffness of the bushing in the suspension attachment parts and the stiffness of the body attachments, a modified body structure was proposed to improve vehicle handling and road noise as well as market quality.

2. CORNERING FORCE

The market quality of vehicle handling is important not only for comfortable driving but also for preventing accidents at curves. The turning properties of a vehicle at a curve are as follows: when a driver turns the steering wheel to make a turn, the vehicle tries to move outwardly due to centrifugal force at the center of the vehicle weight. To make a stable turn, the vehicle needs a force, called cornering force, to counterbalance the centrifugal force.

The cornering force mostly occurs at sideslip of tires when there is no sideways slope on the road surface. This phenomenon is explained by the fact that while the body is pushed outwardly due to centrifugal force, the contact surface of the tires moves in different direction. The cornering force is affected by the load and the slip angle of the tires. A car can turn at a fixed speed because the centrifugal force is counterbalanced by the cornering force on the tires (Yoshitaka and Toshiya, 2001).

The cornering force that affects the front wheels in a left turn generates a movement toward the center point of the vehicle; in a right turn, the rear wheels are affected. Therefore, as shown in Figure 1, when the moments of the cornering forces in the front and rear wheels achieve a balance, a turn can be made within a certain turning radius.

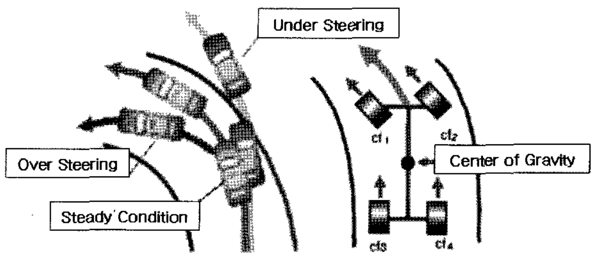


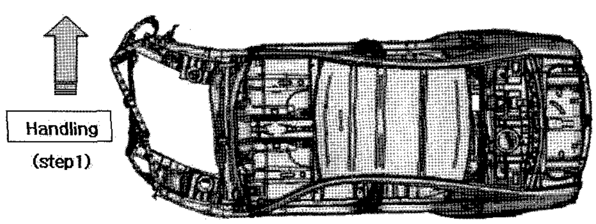
Figure 1. Driving condition of a curve road.

When the moment of cornering force is greater at the rear wheels than at the front wheels, the vehicle is pushed outwardly to a larger tuning radius even if the steering angle is constant; this phenomenon is called under steering.

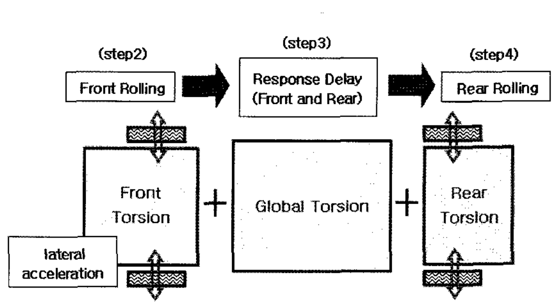
In contrast, when the moment of cornering force is greater at the front wheels for some reason, the front of the vehicle is drawn inward even with a constant steering angle. This inward force reduces the turning radius; as a result, the vehicle may cross the centerline or turn around, sometimes causing an accident or a deviation from the driving course. This phenomenon is called over steering.

2.1. Mechanism of Cornering Force

Figure 2 shows the load transfer path of a vehicle for a 1g cornering load. To define the load transfer path and evaluate the stiffness of each system for the purpose of analyzing vehicle behavior, a vehicle was examined in



(a) The lateral mode of body structure



(b) The mechanism of load transfer path of a vehicle

Figure 2. Load transfer path for 1 g cornering load.

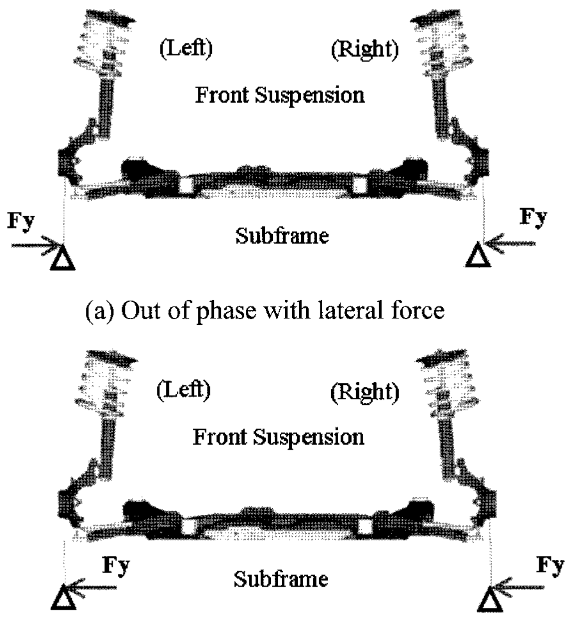
three parts: the front end, the back end, and the vehicle as a whole.

Body properties in terms of vehicle handling are estimated as follows: when a vehicle is cornering, a roll delay occurs between the front suspension and the rear suspension. While the front suspension is rolling, the load on the lateral moves; after a while, when the rear suspension rolls, the load moves laterally. Consequently, depending on the overall stiffness of the car body and the stiffness of the suspension attachments, there is a variation in the yaw rate and the lateral acceleration response. Due to these effects, it is required to systematically analyze how the vehicle handling is affected by the deformation of the attachment parts which occurs as a result of the loading on the suspension attachment parts and the vehicle torsional stiffness.

2.2. Front Lateral Compliance

When a load is obtained from a tire patch or a wheel center, a suspension system is constructed for each vehicle type to determine the load operating on each suspension part. After constructing a basic template for the suspension system, an ADAMS/Car simulation is used to faciliate the creation of a new suspension system by changing the location of the hard point by vehicle type. However, when the load operating on each suspension part is obtained by using the ADAMS/CAR software, the data is organized as a load at the joint or bushing.

When the load at the joint or bushing of a suspension system is obtained, the next step is to determine the volume of stress on each part by means of analysis and finite element modeling of the suspension parts. Even if a large load is applied to the joint or bushing, each force is applied in an X, Y, and Z direction in a complex way; thus, it is difficult to explain the behavioral effect of the suspension parts without finite element analysis. With finite element analysis, the maximum stress and weak points in the suspension parts are clarified for the purposes of selecting appropriate design stiffness specifications to counteract the stresses.



(a) Out of phase with lateral force

(b) In phase with lateral force

Figure 3. Load condition of a chassis model.

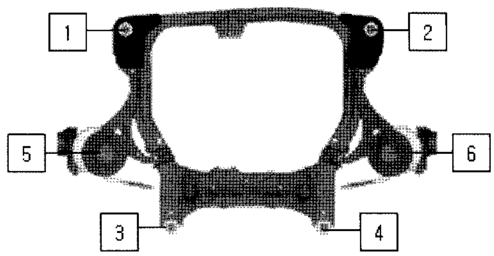


Figure 4. Attachment points of the front suspension.

Figure 3 shows the results of the front lateral compliance analysis of the front suspension for out-of-phase and in-phase conditions.

Figure 4 and Table 1 show the analytical results regarding the input forces of the body attachment points of the front suspension for a rigid body.

Table 1. Input forces of the body attachment points.

No	Body attachment points (unit:Kgf=9.8N)	Out of phase			In phase		
		Fx	Fy	Fz	Fx	**Fy**	Fz
1	Subframe front left	–13.0	27.4	11.7	21.3	**–95.8**	–5.3
2	Subframe front right	–14.3	–25.7	11.0	–19.6	**–102.5**	5.2
3	Subframe rear left	13.9	–6.2	12.3	32.9	**–202.8**	–15.8
4	Subframe rear right	12.0	4.5	13.5	–34.5	**–204.9**	15.9
5	Front strut top left	0.5	71.3	5.5	–0.5	**71.2**	–5.2
6	Front strut top right	0.5	71.3	5.3	0.5	**71.2**	5.3

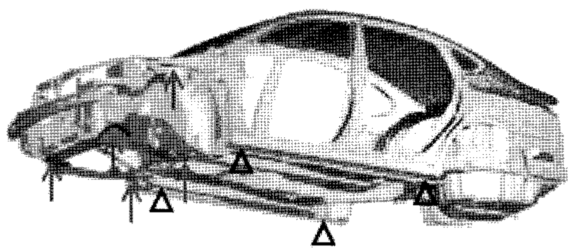


Figure 5. Load condition of the body and subframe.

Figure 5 shows the analytical model of the 1 g cornering force with input forces (shown in Table 1) for the body attachment points.

3. ANALYTICAL PROCESS

In the initial stage of design, a conceptual analysis was performed with the mother car to examine how a static stiffness of less than 20 Hz affects the ride comfort and handling performance. First, with the aid of an analytical model with a subframe installed into a car body, this study used multi-body dynamics to calculate the suspension input load. Next we calculated the displacement of the suspension attachment parts and used this value as a basis for determining the level of torsional stiffness in the low-frequency band.

In older technology, the body resonance frequency was generally more than 25 Hz and there was no clear understanding of how the body properties in the band with less than 10 Hz affected ride comfort and handling. To identify the body properties in the band with a frequency lower than 20 Hz, this study assigned a load transfer path and analyzed the deformation of each system in terms of vehicle torsion for a 1g cornering load (Nozaki, 2006).

The body vibration properties derived from excitation of an actual car were analyzed with the aid of a road simulator under road surface conditions. The conditions were recreated with a hydraulic vibrating exciter. The results of the analysis confirm several tendencies. First, in the band around about 10 Hz, where body resonance does not exist, the whole vehicle is deformed due to torsion; furthermore, the smaller the torsional deformity, the larger the suspension stroke. When the local stiffness in the suspension attachment parts is weak, vehicle deformity occurs before the stroke, bushing isolation becomes difficult, and the damping force of the shock absorbers becomes adverse. Thus, when body torsional stiffness increases in the low-frequency band and the body attachment stiffness of the suspension attachment parts becomes as stable as the dynamic stiffness of the rubber bushing, the volume of the suspension stroke increases, and the damping force of the shock absorbers increases, leading to a reduction in the load that enters through the suspension attachment parts.

As a means of improving vehicle handling, this paper presents design guidelines in Table 2 with respect to the relationship between body stiffness and frequency band.

The previous process examines a design for increasing lateral bending stiffness for a mode in the lateral directions, and it does so by analyzing the resonance frequency of a car body in frequency band higher than 25 Hz. However, for better vehicle handling, this study describes the analytical process of static stiffness to secure a high degree of stiffness in a band with a frequency lower than 20 Hz. Furthermore, the correlation between the body attachment stiffness and the dynamic stiffness of the rubber bushing in the suspension attachment parts is analyzed to secure local stiffness in the high-frequency band. By analyzing the suspension attachment parts, it is expected that vehicle handling and road noise performance can be improved.

3.1. 1 g Cornering Static Stiffness

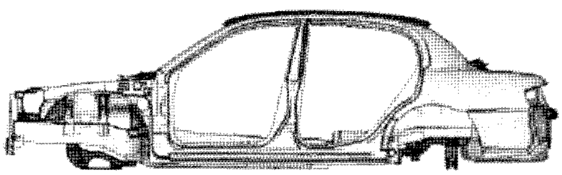
Figure 6 shows a full model and a half model of a body structure attached to a subframe and a rear cross member. The front half model was constructed by considering the effect of boundary conditions and the analysis time.

Six degrees of freedom were used for the cutting planes in the front part of the vehicle, and the front torsion stiffness was tested by applying a cornering load to the front suspension.

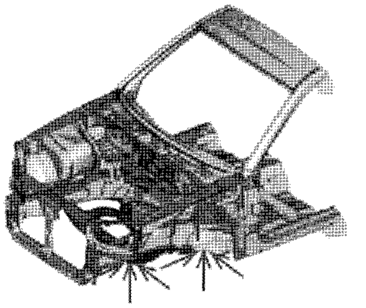
Following the same principle, the rear torsion stiffness is tested at the rear of the vehicle, and the torsional dynamic stiffness in the full model of the body is evaluated to analyze stiffness for each system.

Table 2. Body structure specifications.

Mode	Frequency range (Hz)	Support	Specification
1g cornering static stiffness	0~20	Handling	Benchmarking target
Torsional stiffness	20~50	Vibration	Separation from engine idle rpm and suspension mode
Bending stiffness			
Lateral stiffness			
Body attachment stiffness	100~500	Noise	1000 kgf/mm ↑

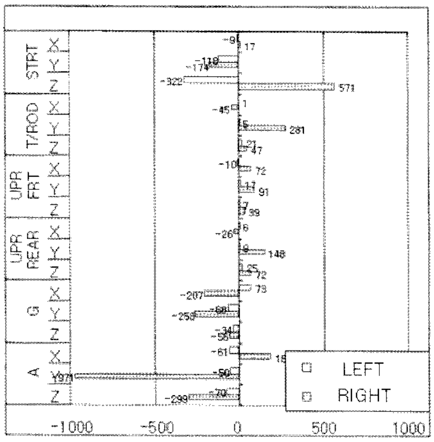


(a) Full model of the body with frame parts attached

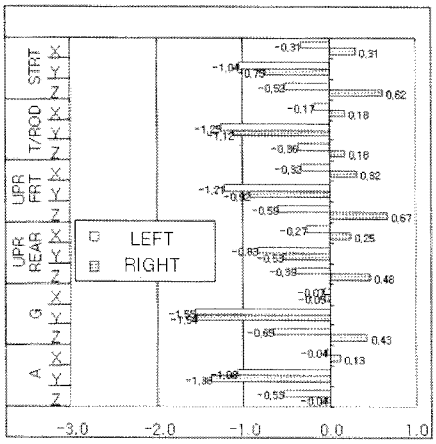


(b) Half model of the body with frame parts attached

Figure 6. Application of cornering force at the front.

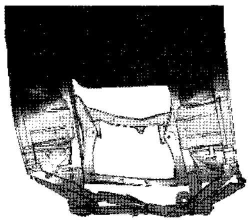


(a) Dynamic load of the suspension attachment points

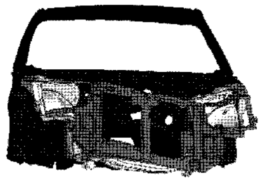


(b) Displacement of the suspension attachment points

Figure 7. Application of cornering force at the front.



(a) Top view



(b) Front view

Figure 8. Operating deform shape of the engine room.

Figure 7(a) shows the load at each arm attachment part of the front suspension. This load is analyzed to determine the entering direction and magnitude of the main load. Figure 7(b) shows the displacements for the case of Figure 7(a). This displacement is analyzed to identify parts with relatively weak stiffness and then present design guidelines for structural improvement.

Figure 8 shows the operating deformed shape of the analytical results for the front part of the vehicle under a given front torsion load. Our analysis of the deformation behaviors and strain energy of the whole vehicle provides sufficient data, as shown in Figures 7 and 8, for analysis of parts with weak stiffness and for improvement analysis.

This analytical process enabled us to draw up design guidelines for the initial stage, particularly with respect to analyzing the effect of the chassis, the body, and the bushing attachment parts. It also enabled us to ensure adequate body stiffness for improved handling performance in an actual vehicle.

3.2. Dynamic Stiffness

While an engine is idling, the vibration caused by torque changes can cause body-bending vibration. Drivers often think that the vibration, which comes through the wheels and car floor to the seat, adversely affects the ride comfort. This type of vibration is called idle vibration.

In relation to idle vibration, the longitudinal bending in the frequency band less than 50 Hz is in the hop mode of

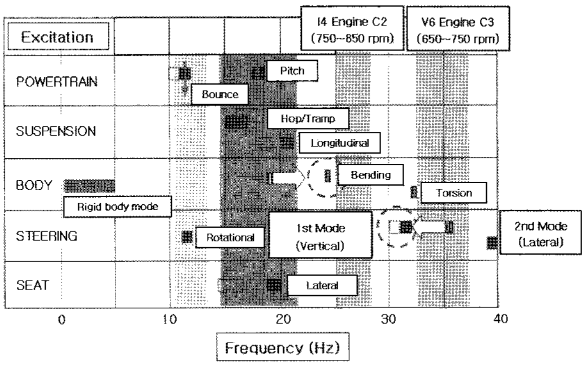


Figure 9. Example of a vibration mode map.

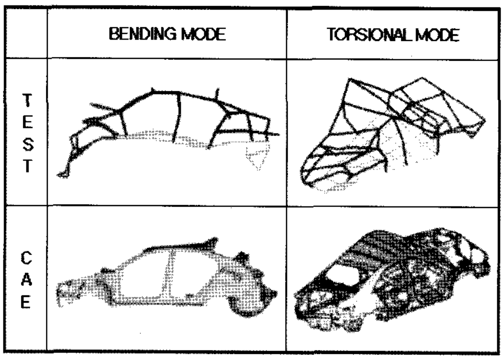


Figure 10. Body global mode shapes.

the suspension and in the range of resonance isolation; furthermore, the torsion mode and the lateral mode are developed within the range of resonance isolation and in the tramp mode of the suspension. These factors are managed in the initial stage of design through the vibration mode map shown in Figure 9; in this way, they are not superimposed with the mode of the engine idle RPM and the steering bending mode.

As shown in Figure 10, longitudinal bending is a mode of vibration in the direction of vehicle length. Moreover, because a driver feels this vibration most closely through the steering wheel and the floor, stiffness has a large effect on the idle vibration of an actual vehicle. Torsion, which is a mode of diagonal twisting and vibration, softens the lateral mode of the lateral directions in an actual car; thus, it has a large effect on interior noise due to changes in the interior sound field.

When a vehicle stops suddenly or corners sharply, a handling problem with the suspension can occur in the short term; in particular, the car tends to lean towards one side. The application of a front tower bar and a dash partition panel is examined to decrease the roll angle and to minimize the difference in deformation between the left and the right in the attachment parts of the front suspension.

Three types of tower bars (shown in Figure 11) were analyzed. This paper describes that type A, which

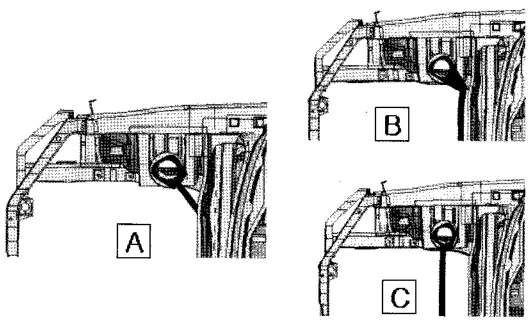


Figure 11. Three types of front tower bars.

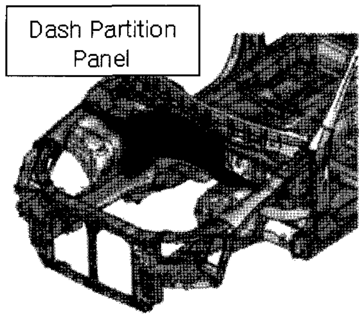


Figure 12. The dash partition panel of the engine room.

connects the front strut and the cowl top, is the most advantageous. It has suitable static stiffness for the front strut input load; furthermore, aside from its light weight, it facilitates the handling of lateral force and has dynamic stiffness and good crash performance.

Figure 12 shows a structure that connects the dash panel and a front side member with a partition. This structure is expected to increase the thermal performance as well as the stiffness in the lateral direction of the car body. The structure, which we applied in consideration of the layout in the initial stage of design, has the effect of increasing the lateral bending and torsional stiffness.

3.3. Body Attachment Stiffness

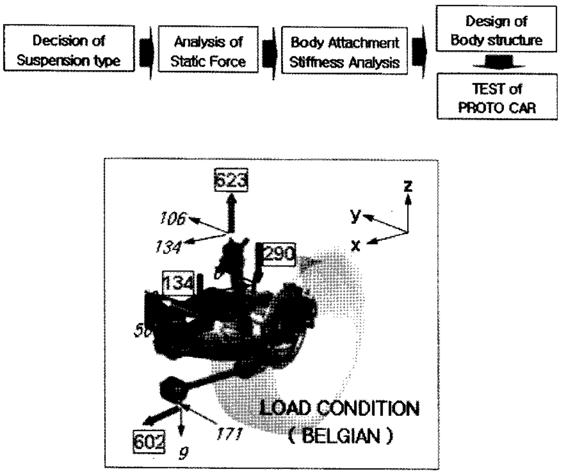
To find ways of reducing noise, this paper discussed the processes by which structural optimization increases body attachment stiffness in the vibration path, as well as the way noise is transferred to the body from such sources as the engine, the suspension system, and the road surface.

Body attachment stiffness has significance in relation to the path design because it isolates the vibration that is transferred into the interior against the excitation force of the engine or road surface. Generally a design guide to develop stiffness is greater by 5~10 multiples than the bush dynamic stiffness based on the analysis standard (LMS and HMC Report, 2001).

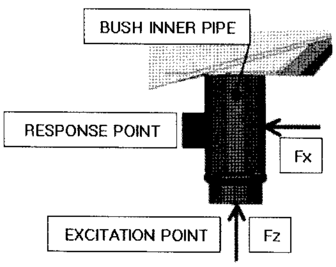
Another significant factor is the need to increase local stiffness in the attachment parts as a means of reducing the body leaning phenomenon caused by a delay between the suspension and the car body under situations of sudden stopping or braking.

An evaluation of the effect of body attachment stiffness in low-noise vehicles confirms that body attachment stiffness reduces road noise by 30% and idle noise by 40%; moreover, it has the same effect as the suspension input force and the drive shaft.

Figure 13 shows the body design process of the rear suspension attachment points for road noise performance. After deciding on the suspension type, this study obtained the static force in order to determine the main



(a) The static force of the load condition



(b) The analysis condition (excitation and response)

Figure 13. The process of body attachment stiffness.

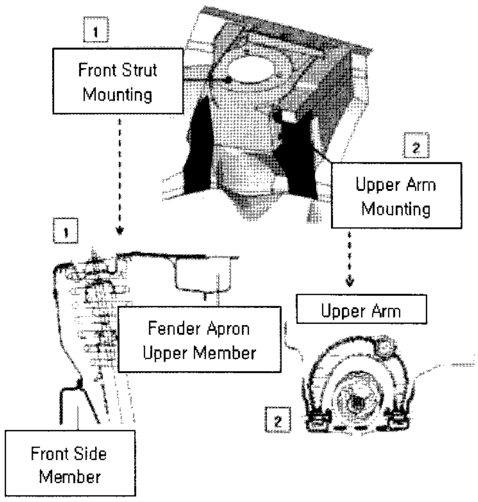


Figure 14. The structure of the front suspension mounting.

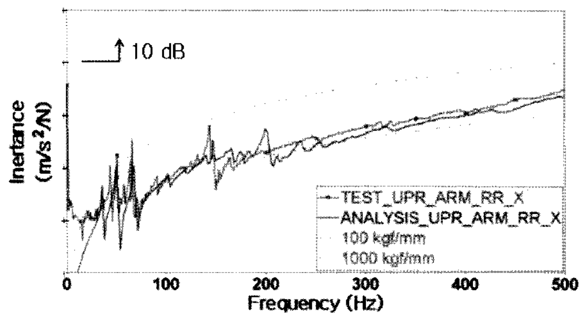


Figure 15. Comparison of the test and analysis results.

loading direction from the chassis group. The data results confirm that the main load came in an up-and-down direction from the rear strut and the rear cross member attachment parts, whereas the trailing arm load was applied in a front-to-rear direction. For the analysis, the excitation points were at the center of the bushing inner pipe and at the bottom of the bolt; the response point was at the center of the bushing inner pipe (Kim and Choi, 2003).

3.3.1. Front suspension attachment parts

As shown in Figure 14, we designed a structure for the front suspension attachment parts in the shape of a double wishbone; the design is based on a new body structure for the purpose of achieving a high degree of stiffness and a light weight. The package tray pass was constructed through the longitudinal and lateral members.

The housing cover and upper-arm mounting bracket are combined, the upper cover is connected with an upper member of the fender apron, and the lower bracket is expanded to front-side member. Figure 15 shows the analysis of the upper-arm attachment part as well as the test data, and the target frequency band is between 100 and 500 Hz; it was shown to have a stable level of stiffness and sensitivity.

3.3.2. Rear suspension attachment parts

To improve the rear seat noise in the initial stage of design, as shown in Figure 16, this study applied a package tray pass-through structure, which is connected from the rear floor through the inner panel of the wheel house to the side of the package tray. A chassis cross-member in the section is mounted where the rear floor side member (in the direction of the car length) meets the cross member in the lateral directions; furthermore, a strut attachment part makes up the side of a box inside the housing inner panel, securing local stiffness (Kim and Kim, 2007).

When the rear cross member attachment parts were evaluated, we found that the stiffness level of the current vehicle is very good comparing with the competing cars as shown in Figure 17.

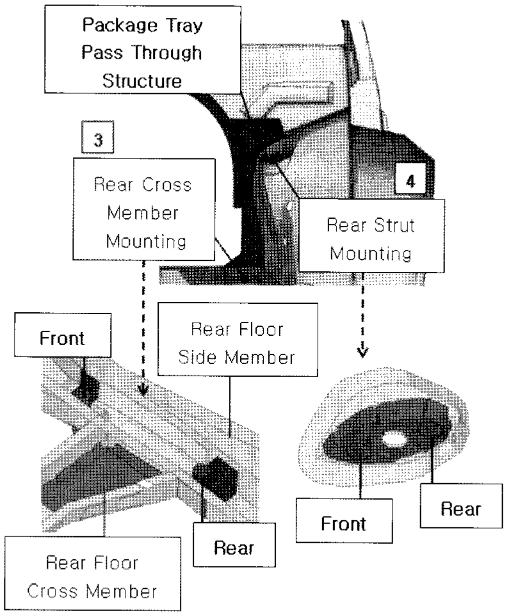


Figure 16. The structure of the rear suspension mounting.

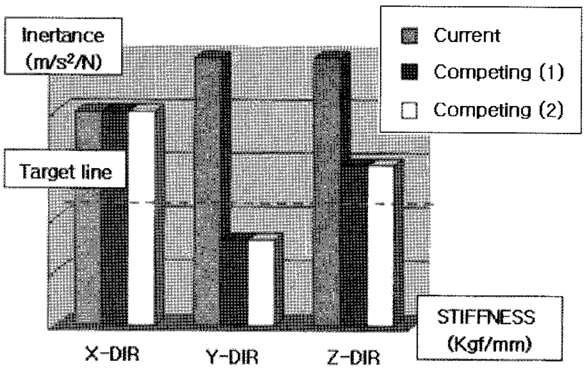


Figure 17. Comparison with competitive vehicles.

4. CONCLUSIONS

We have suggested research directions for design regulations and analysis of body stiffness for different frequency bands in order to improve vehicle handling. Our conclusions are summarized as follows: in the preceding stage, design guidelines recommended the use of a mother car. However, we have developed a method of analyzing static stiffness for a 1 g cornering load to ensure a stable performance. This analytical process enables us to analyze how body stiffness in an band with a frequency lower than 20 Hz affects vehicle handling.

We can improve vehicle handling by preventing the body frame mode from being superimposed with that of the suspension system, the steering mode, and the RPM of an idling engine, while securing static stiffness for a 1 g cornering force. To achieve this objective, we need to ensure that the definition of the mode map and the resonance isolation design are completed in the initial stage of design. When the package tray pass-through member is applied to increase the body attachment stiffness in the suspension attachment parts, it is expected to decrease the road noise as a result of resonance isolation in the bushing attachment parts; furthermore, with the increased suspension stroke and the increased damping force of the shock absorbers, it is a decrease in the load that enters through the suspension attachment parts is expected.

In the design stage, this study demonstrated various processes which aimed to achieve a high degree of stiffness and a light weight for the vehicle. Overall, the results show improvement in the suspension performance and fuel efficiency, as well as in indirect effects such as cost reduction and a shortened development period.

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