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Dynamic Stress Analysis of Vehicle Frame Using a Nonlinear Finite Element Method

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Structural integrity of either a passenger car or a light truck is one of the basic requirements for a full vehicle engineering and development program. The results of the vehicle product performance are measured in terms of durability, noise/vibration/harshness (NVH), crashworthiness and passenger safety. The level of performance of a vehicle directly affects the marketability, profitability and, most importantly, the future of the automobile manufacturer. In this study, we used the Virtual Proving Ground (VPG) approach for obtaining the dynamic stress or strain history and distribution. The VPG uses a nonlinear, dynamic, finite element code (LS-DYNA) which expands the application boundary outside classic linear, static assumptions. The VPG approach also uses realistic boundary conditions of tire/road surface interactions. To verify the predicted dynamic stress and fatigue critical region, a single bump run test, road load simulation, and field test have been performed. The prediction results were compared with experimental results, and the feasibility of the integrated life prediction methodology was verified.

Key Words : Ride and Handling, VPG Approach, Dynamic Stress Analysis, Nonlinear Finite Element Method

1. Introduction

The durability test, along with the crashworthiness test, requires a lot of time and expense in the vehicle development process. With the recent rapid developments in CPU capabilities and in applications software, a durability design using CAE tools is now possible even before the prototype vehicle is developed. This reduces time required for both the durability test and actual vehicle production. Existing dynamic stress analysis for the analysis of vehicle fatigue mainly calculates the dynamic stress history and fatigue after performing dynamic analysis and stress analysis with relevant software applications such as DADS and NASTRAN, and then superpositioning the dynamic load history and stress influence coefficient at each joint. This approach is a complex process, when the flexibility of the parts is taken into account. It is, however, incapable of giving accurate consideration to the contacts between components, the non-linearity of materials, and tire-road surface interactions. This approach also requires the analysts to have an expertise in software applications of various kinds or an expert in each area to perform the analysis. This requires a great deal of manpower and time.

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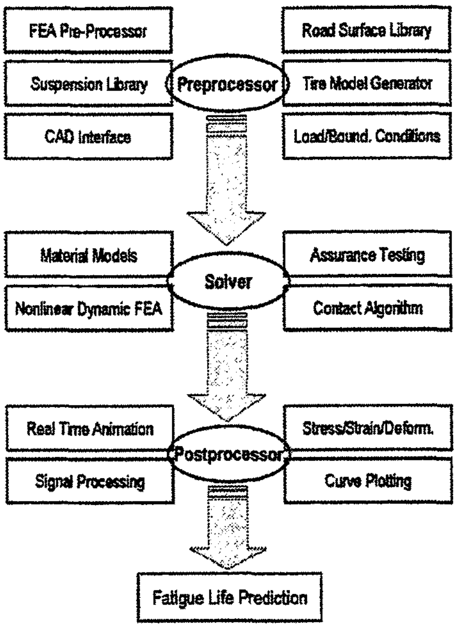


Fig. 1 The routine for fatigue strength evaluation using VPG approach

In order to complement the existing approaches for dynamic stress analysis, this study aims at the followings : (1) to develop the VPG approach which is capable of producing all the necessary results possible with just one model, one program and one process ; (2) to reduce dramatically time and manpower needed for construct a model designed to analyzing dynamics, quasi-static stress, and fatigue ; and (3) to enable an accurate analysis of fatigue by improving the accuracy of dynamic stress. Figure 1 shows a life prediction procedure using the VPG approach.

2. Existing Dynamic Stress Analysis Method

2.1 Vehicle dynamic analysis

The multibody dynamic analysis program which is used for dynamic motion analysis can also be utilized for the dynamic analysis of a vehicle structure. In this study, the research was conducted on multi-purpose vehicles with a frame. The analysis model was constructed using DADS, a commercial CAE tool (DADS). It was assumed that all the parts were rigid bodies. However, the

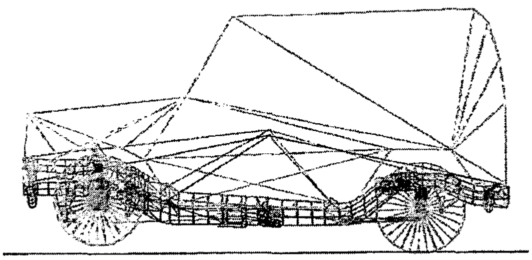


Fig. 2 Graphic animation of a bump run at the time of passing

flexibility of the frame, a major focus of the study, was considered (DADS ; Shin and Yoo, 2000). When parts are assumed to be rigid, it is necessary to check whether the primary vibration frequency of the parts is in a much higher area than that of road load. The definite quantitative criteria, however, have not yet been found.

The front suspension of the vehicles used in this study is an independent suspension of a double wishbone type. The rear suspension is an axle with four links. The simulation of the model was performed on a single bump in which the wheels on both sides pass over the bump simultaneously at a speed of 40 km/h. The purpose is to calculate the reaction force and torque at each joint of the suspension components for the load exerted on the vehicle at the time of passing. It is assumed that the bump was a straight with a width of 160 mm and a height of 85 mm. The vehicles were set in such a way that they passed over the bump after reaching a static equilibrium (eta/VPG ; Papadrakakis, 1981). Figure 2 shows a graphic animation of the bump run at the time of passing. The dynamic load history obtained can be used as input conditions for the quasi-static stress analysis of the frame.

2.2 Quasi-static stress analysis

The dynamic stress history, one of the most popular quasi-static stress analysis methods, could be introduced from the finite element stress analysis which makes use of reaction force and torque at the joint parts where each component movement is restricted. The equation of motion used in the DADS for a multi-body dynamic system which considers the elastic body's flexibility is

$$[M]\begin{bmatrix}\ddot{r}\\ \dot{\omega}'\\ \ddot{a}\end{bmatrix}+S(\dot{a},\ \omega')+U(a)+\Phi_q^T\lambda-Q_{ex}=0 \quad (1)$$

Here, $\ddot{r}$ and $\dot{\omega}$ are an acceleration vector and an anglular acceleration vector, respectively, at the flexible body's arbitrary points. And $\ddot{a}$, $\dot{a}$, a are the acceleration vector, the velocity vector and the displacement vector in regard to flexible body's elastic deformation. In addition, s is a generalized force vector that is dependent on the velocity and U is the internal force vector. Φ_q is a Jacobian of the restriction equation, λ is Lagrange multiplier vector and Q_{ex} is the external force vector. If we ignore the velocity and acceleration in regard to the flexible body's elastic deformation, we can introduce equation (2) from Eq. (1).

$$[M]\{\ddot{R}\}+U(a)+\Phi_q^T\lambda-Q_{ex}=0 \qquad (2)$$

Here, $\ddot{R}$ is the rigid body's acceleration vector. The second term is related to the elastic deformation. If we consider the degree of freedom of the finite element model, Eq. (2) becomes.

$$U(a)=[K]\{u\}=Q_{ex}-[M]\{\ddot{R}\}-\Phi_q^T\lambda \quad (3)$$

Here $\{u\}$ is a displacement vector, $[M]\{\ddot{R}\}$ is inertia force, and $\Phi_q^T\lambda$ is a joint reaction force is calculated from the dynamic analysis.

At a designated time, this equation could be change a to a static equation, where $U(a)$ is equal to an external force, inertial force, and joint reaction force. Eq. (3), therefore, could be solved by quasi-static finite element analysis in a each stage of dynamic analysis (Kuo and Kelkar, 1995). If we express a general load history as $F_i(t)$, the stress history at a specific point of the body could be introduced as a form of linear function which has n load history, as shown in equation (4).

$$\sigma_i(t)=f(F_i(t)) \qquad i=1,\ \cdots,\ n \qquad (4)$$

When $\bar{\sigma}_i$ is the unit force vector of load history $F_i(t)$, the equation could be shown as equation (5).

$$F_i(t)=C_i(t)\,\bar{e}_i \qquad i=1,\ \cdots,\ n \qquad (5)$$

Here, $C_i(t)$ is the magnitude of time varying load.

In the quasi-static stress analysis, if a stress field is constant at a specific time, we can say that linear operator f is independent of time. From the superposition principle and quasi-static assumption, we can introduce equation (6).

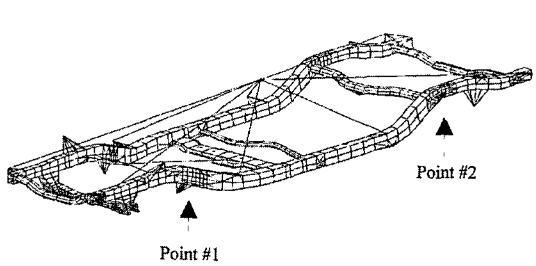


Fig. 3 Quasi-static stress analysis model

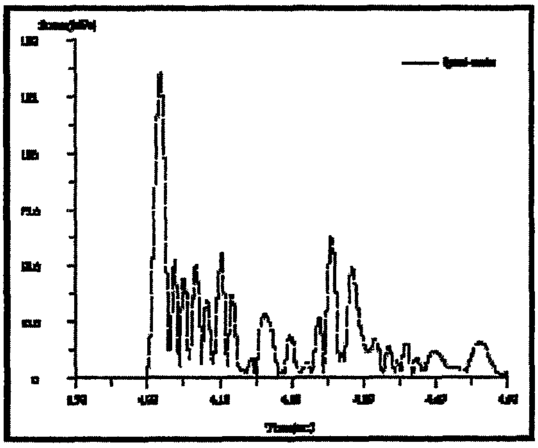


Fig. 4 Dynamic stress history of the point #1

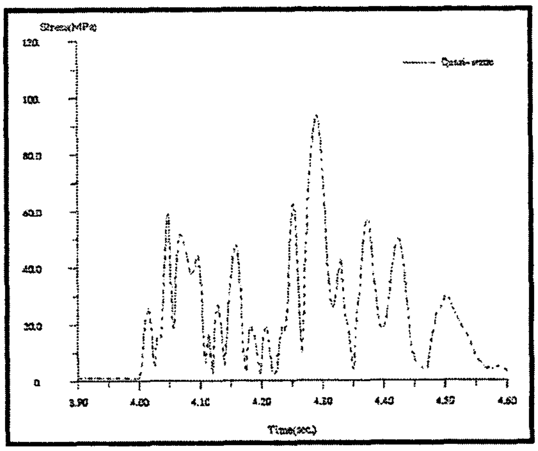


Fig. 5 Dynamic stress history of the point #2

$$f\left[\sum_{i=1}^{n} C_i(t)\,\bar{e}_i\right]=\sum_{i=1}^{n} C_i(t)\,f(\bar{e}_i) \qquad (6)$$

This equation could be very useful in calculating the stress history at a specific time (Beak, 1992).

In this study, we set up a finite element Model of the frame (Fig. 3) and applied quasi-static stress analysis and the superposition principle to this model in order to obtain a dynamic stress history. Among various analysis results, Figures 4 and 5 represent the stress history at point #1 and #2, when the analysis model had run over a bump at the speed of 40 km/h.

3. Dynamic Stress Analysis Through the Virtual Proving Ground Approach

Until this point, as shown in Fig. 6, the existing quasi-static stress analysis method was not available to carry out dynamic and stress analysis using the same analysis software. This is because it cost a great deal of time and money in the processing of the data interfaces between software programs (Zhang and Tang, 1996).

However, by means of the dynamic stress analysis through the virtual proving ground approach, we could delineate contact problems between road surface and vehicle tire, interaction problems between components, the nonlinear behavior of materials, boundary condition, and cut costs as well (Han et al., 2000). Figure 7 illustrates the basic concept of the virtual proving ground approach. In addition, as shown in Fig. 8, many road shapes can be adapted to the simulation analysis.

The successful tire model must be able to support the vehicle weight, provide vehicle control and stability, transfer various forces and torques from road/tire interaction to a vehicle chassis/suspension system. The dynamic effects in terms of tire stiffness and internal damping characteristics in impact loading conditions must also be accounted for in the model. In this research, we made use of a thin shell and solid element for the sake of modeling the most realistic tire and the tire mode validation is carried out with the LS-DYNA quasi-static and transient dynamic analysis capability, simulating several tire

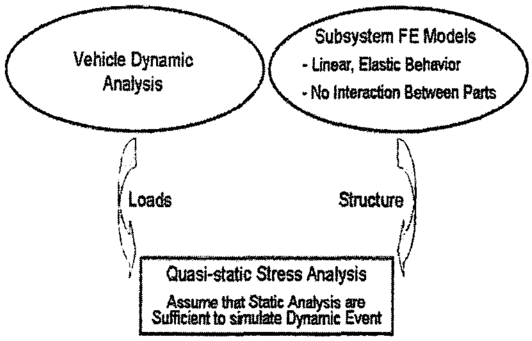


Fig. 6 Current method and assumptions

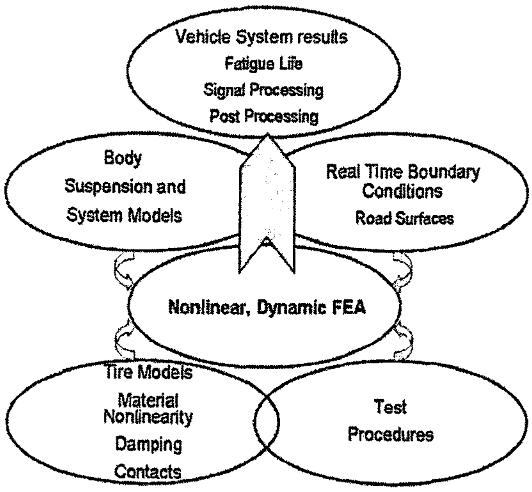
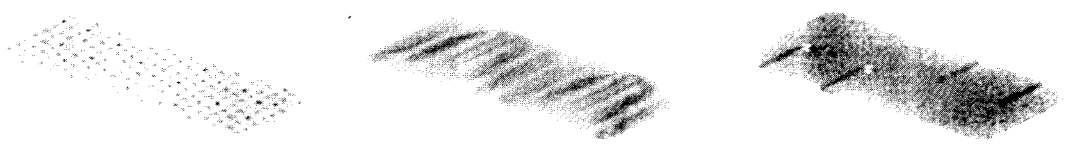
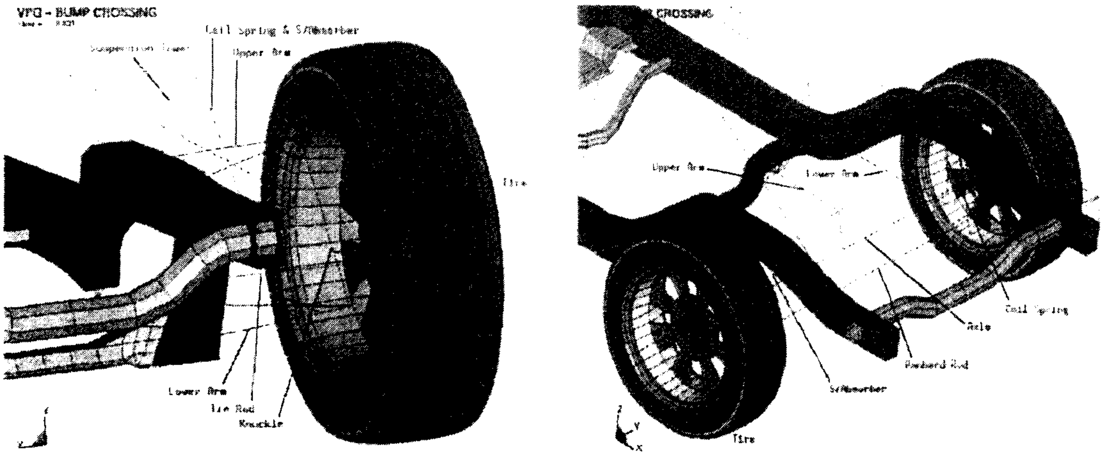


Fig. 7 VPG Concept



(a) Ripple (b) Cobblestone (c) Pothole

Fig. 8 Typical road surfaces



(a) Front suspension part (b) Rear suspension part

Fig. 9 Suspension and tire model

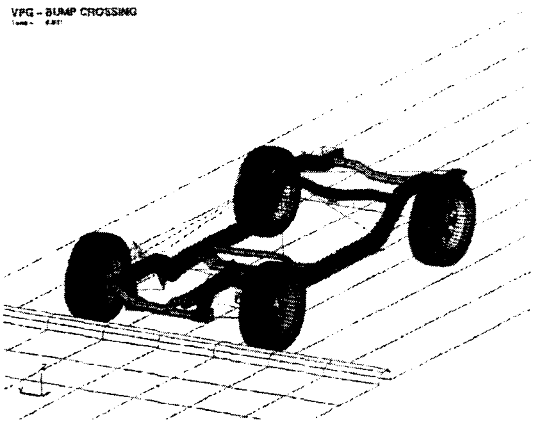


Fig. 10 VPG Analysis model

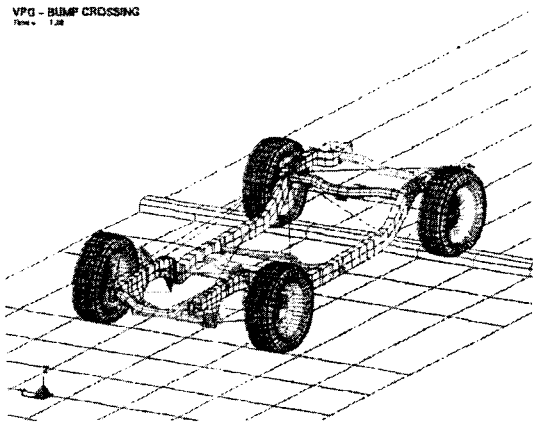


Fig. 11 Graphic animation of bump run test

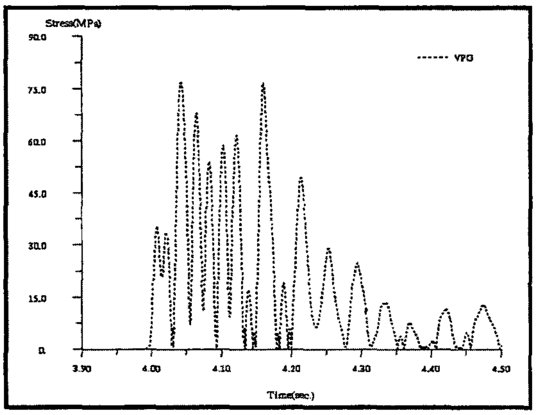


Fig. 12 Dynamic stress history of the Point #1 (VPG)

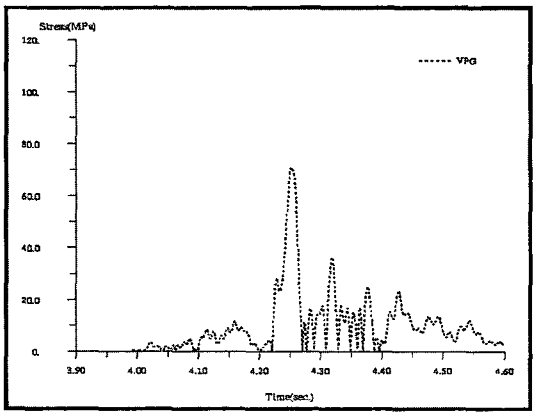


Fig. 13 Dynamic stress history of the Point #2 (VPG)

test procedures (Zhang et al., 1997; eta/VPG). The model of the vehicle was designed by the same dynamic analysis which made use of DADS. Now, Figure 9 shows the front and rear suspension part of the vehicle and a model of the tire.

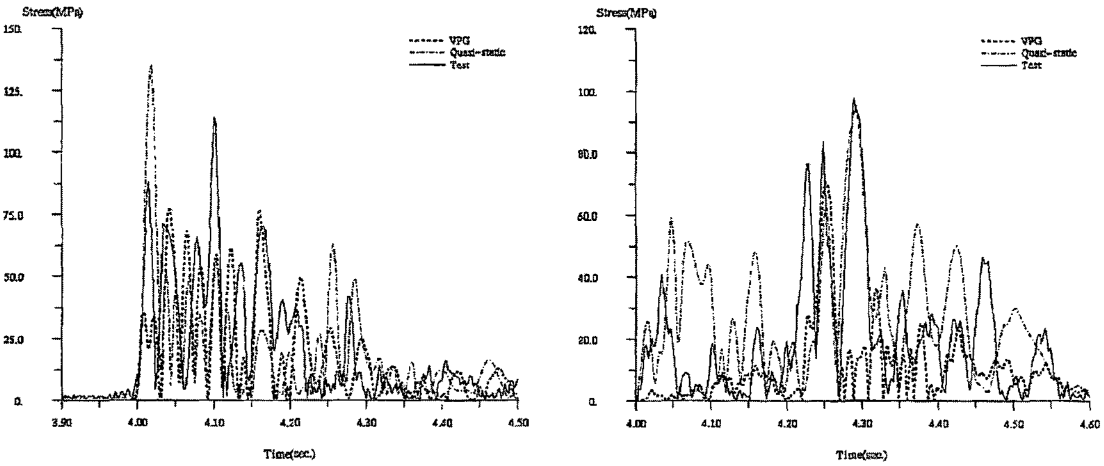
Simulation of the model was carried out by the same method as the quasi-static stress analysis. The figure of the analysis model is shown in Fig. 10. In order to reduce the time consumed analyzing, it was assumed that all of the components and road surface were rigid bodies except the vehicle frame and tire. Figure 11 shows the motion with the vehicle run over a bump and calculated using VPG method. Figures 12 and 13 shows the dynamic stress history at points #1 and #2. In the past, to use the quasi-static stress analysis method, weak points were selected based on the researchers' experience and their own personal judgment. Today, however, with the VPG simulation method, we can continually observe the stress distribution at every timestep and discern the weak points using graphic animation of the stress distribution.

4. Comparison Between Analysis and Experiment

According to the quasi-static analysis method and VPG simulation method, cracks were examined at the same points where they were created from repeated loading with the road surface. Thus, we selected a couple of points to examine the accuracy of the results of dynamic stress history analysis when the vehicle ran over a bump. As shown in Fig. 14, we set up a strain gauge on the vehicle frame and measured the dynamic stress history at 40 km/h speed on the bump run. Then, we compared this result with the dynamic stress history of the point #1 and #2.

Figure 15 compares the dynamic stress history between test, quasi-static analysis, and VPG simulation at points #1 and #2. Although the general tendency of the analysis results were nearly equal, we could notice that the quasi-static analysis showed a somewhat higher stress than that of the experiments and different stress graph. This sort of dynamic stress history can bring in

Fig. 14 Bump run test



(a) Point #1 (b) Point #2

Fig. 15 Comparison of the dynamic stress history

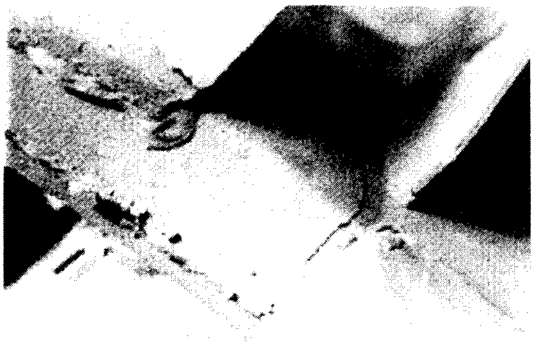


Fig. 16 Fatigue crack at the Point #1

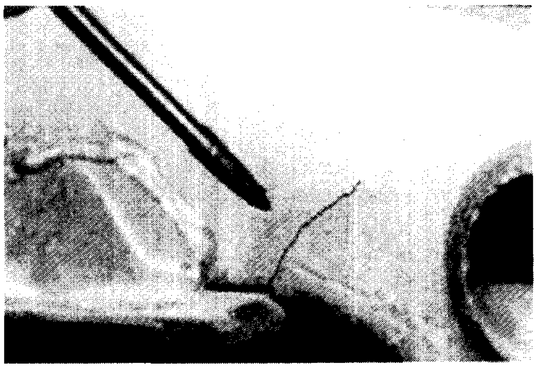


Fig. 17 Fatigue crack at the Point #2

different, or flawed, calculations from reality.

The predicted location of crack points #1 and #2, which were observed from the quasi-static stress analysis and VPG simulation, was the same as those of the road load simulation and actual vehicle durability test. Figures 16 and 17 represent images of cracks that were propagated during the actual vehicle durability test.

It was not possible to compare quantitatively the results of the bump run simulation and a real ground test. However, because normally the damage of a vehicle frame comes from the bending and twisting loading, we can determine the weak points of a vehicle frame by analyzing a single bump run and a zigzag bump run, which contain both bending and twisting loading.

5. Conclusion

Through the VPG simulation method we were able to simulate real driving conditions, and combine the relations between stress-life and strain-life. We were able to set up a standard analysis method to predict the fatigue life of a vehicle frame. This expressed a better analysis capability than the quasi-static analysis method. In particular, the VPG simulation method can simulate contact problems between vehicle components, the relation between stress and strain, as well as various kinds of surface profile. This could be used to save time and costs in developing new vehicle models. In addition, if the analysis method is combined with optimization technology based on its durability, this could be a much better design tool.

The main contents and characteristics of this research were as follows:

(1) To analyze dynamic stress, we developed the Virtual Proving Ground approach which is able to save time and costs better than the existing dynamic stress analysis method.

(2) Because the VPG simulation method could be used without any analysis model change, we could save over 90% of the time normally spent on modeling a new one for each analysis.

(3) Only using the bump run stress analysis of VPG simulation, we could forecast the exact points which were created by the actual vehicle durability test.

(4) This VPG simulation method could also be used for NVH, suspension kinematic analysis and vehicle dynamics analysis.

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