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Fatigue Analysis of Vehicle Chassis Component Considering Resonance Frequency

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공진 주파수를 고려한 차량 섀시 부품의 피로해석

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Abstract

The purpose of this paper is to assess the benefits of frequency domain fatigue analysis and compare it with more conventional time domain techniques. The multi-body dynamic analysis, FE analysis and fatigue life prediction technique are applied for the frequency domain fatigue analysis. To obtain the dynamic load history used in the frequency domain fatigue analysis, the computer simulations running over typical road profiles are carried out by utilizing vehicle dynamic model. The fatigue life estimation for the rear suspension system of small-sized passenger car is performed by using resonance durability analysis technique, and the estimation results are compared with the conventional quasi-static durability analysis results. For the pothole simulation, the percent changes of the fatigue life between the two durability analysis techniques don't exceed 10%. But, for the Belgian road simulation, because of the resonance effect, the fatigue life using the resonance durability analysis technique are much smaller estimated than the quasi-static durability analysis results.

Key Words : Belgian Road(벨지언 로드), Pothole(포트홀), Quasi-Static Durability Analysis(준정적 내구해석), Resonance Durability Analysis(공진내구해석), Suspension System(현가장치)

1. Introduction

In the initial design stage of a vehicle system, it is necessary that engineers predict accurately the fatigue life of the components consisting of the vehicle chassis and structural systems. Also, it is essential that they design the vehicle components satisfying the required durability performances. In general, for the durability assessment of the vehicle components, the field test through urban and/or rural areas, the proving ground

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test running over a series of specially constructed sections of road, and the full-scale simulated road fatigue test, etc. are performed. However, such conventional durability tests using prototype vehicle need excessive cost and time in applying the test results to design change.

The number of experiments required in the initial stage of vehicle development can be effectively reduced by using the virtual durability analysis techniques. Therefore, to take such advantages, many automotive companies have used the virtual durability analysis technique[1~4] together with the conventional durability tests.

Generally, the virtual durability analysis techniques can be classified into two major types: 1) the quasi-static durability analysis technique and 2) the dynamic durability analysis technique.

The quasi-static durability analysis technique performs the fatigue life assessment by using the dynamic stress history, which is generated by combining the dynamic load time history with the static stress of mechanical or structural system. Such a durability analysis technique is mainly used under the assumption that the dynamic effect of the mechanical or structural system rarely affects the fatigue life of the system, therefore, this technique is useful for very stiff systems[5].

However, when a system has a large mass and/or when the dynamic loads acting on the system get near the natural frequencies of the system, the vibration characteristics of the system seriously affect the fatigue life[6]. For example, when a vehicle system runs over a rough road, if the natural frequencies of vehicle suspension system approach the excitation frequencies acted on the suspension system by a rough road surface, the vibration characteristics of the suspension system will have serious influence on the fatigue life of each suspension component. In this problem, the fatigue life of the suspension components can be predicted exactly by considering the vibration effect of the structural system. Recently, some researches on the frequency domain life estimation considering dynamic effect have been made[7].

In this paper, a resonance durability analysis technique is proposed for the fatigue life estimation of a chassis component considering the vibration effect of a vehicle structural system. To obtain the dynamic load data acting on the vehicle system, the computer-aided dynamic simulations running over typical road profiles are carried out. The frequency response data for the given structural system are obtained through the finite element analysis. And also, the fatigue life estimation of chassis component is performed with the dynamic stress, which is obtained by combining the dynamic load data in the frequency domain with the frequency response data obtained from the normal mode analysis. In this study, commercial multi-body dynamic analysis software, ADAMS is used to produce the dynamic load data, and also the finite element analysis softwares, MSC/NASTRAN and MSC/FATIGUE, are used to perform vibration analysis and fatigue life analysis, respectively. Using the developed technique, the durability analyses for the rear suspension system of small-sized passenger car are carried out.

2. Durability Analysis Technique

2.1 Quasi-Static Durability Analysis

In the quasi-static durability analysis, the fatigue life of a vehicle structural component is computed by using the dynamic stress, which is obtained by combining the dynamic load data generated from the dynamic simulation or the prototyping test with the static stress obtained from the structural analysis of the component. This durability analysis technique is useful when the dynamic effect of the mechanical or structural component composing the vehicle system is small.

In the quasi-static durability analysis, the static stress of the vehicle component is obtained with the linear static finite element analysis in which the unit load is acted on the component in the direction that the actual dynamic load is transmitted. To transform the computed static stress into the dynamic stress, the static stress is combined with the dynamic load time history obtained through the experiment or the multi-body dynamic simulation. And the obtained irregular dynamic stress history is decomposed into equivalent sets of block loading by the rainflow cycle counting[8]. The numbers of cycles in each block are recorded in stress range histogram.

In general, because a ground vehicle is subjected to randomly variable load while the vehicle is running, the possibility, which a series of same load history occurs to specific vehicle component within a given period of time, is rare. To investigate the fatigue life under the variable amplitude loading, the analytical function describing the dynamic characteristics of randomly changing load has to be formulized. Generally, a probability density function (PDF) is used to characterize the variable amplitude loading.

Finally, the fatigue damages are calculated with the Minor's rule, which can predict the fatigue life under the variable amplitude loading. Minor's rule evaluates the fatigue life by summating the damage fractions for each stress level, and the fatigue fracture is occurred when the summation of the damage fractions is greater than or equals to 1[9].

$$D=\sum D_i=\sum\left(\frac{n_i}{N_i}\right)\geq 1 \tag{1}$$

where D_i, n_i and N_i are the damage fraction, the number of cycles and the fatigue life respectively at a particular stress level.

2.2 Resonance Durability Analysis

Since the dynamic behavior of a vehicle system is affected not only by the excitation frequencies transmitted from the road surface but also by the structural natural frequencies of the vehicle components, the resonance effect that is caused by the excitation frequencies and natural frequencies has to be considered for the accurate durability assessment.

In the resonance durability analysis, the fatigue life estimation of a vehicle component is performed with the dynamic stress, which is obtained by combining the dynamic load data of the frequency domain with the frequency response data obtained from the normal mode analysis of the vehicle component. In this durability analysis process, the dynamic stress history is expressed in the form of a power spectral density (PSD). Minor's rule that is used to calculate the fatigue damage in the quasi-static durability analysis can be replaced with the equation including the PDF for the PSD of stress[10~12].

Eq. (2) shows the fatigue damage equation used in this resonance durability analysis. The probability density function, $P(S)$, simply represents the characteristics of the loading, and the material data are defined by using the stress intensity factor, K, and crack growth exponent, m. And, the total number of cycles in time, T is defined as the number of peaks per second, $E(P)$. If the damage caused in the time, T is greater than or equal to 1.0 then the vehicle component is assumed to have failed. Or alternatively the fatigue life can be obtained by setting T=1.0 and then finding the fatigue life in seconds from the equation.

$$\begin{aligned} D &= \sum \frac{n_i}{N(S_i)} \\ &= \frac{S_t}{K}\int S^m \\ &= \frac{E(P)T}{K}\int S^m P(S)\,dS \end{aligned} \tag{2}$$

Many researches have been done so far to produce the probability density function, $P(S)$. In 1985, Dirlik[7] produced an empirical closed form expression for the PDF of rainflow ranges, which was obtained using extensive computer simulations to model the signals using the Monte Carlo technique.

The PDF used in this study is expressed in Eq. (3)[13].

$$P(S)=\frac{\frac{D_1}{Q}e^{\frac{-Z}{Q}}+\frac{D_2 Z}{R^2}e^{\frac{-Z^2}{2R^2}}+D_3 Z e^{\frac{-Z^2}{2}}}{2\sqrt{m_0}} \tag{3}$$

where

$$D_1=\frac{2(x_m-\gamma^2)}{1+\gamma^2},\qquad D_2=\frac{1-\gamma-D_1+D_1^2}{1-R}$$

$$D_3=1-D_1-D_2,\qquad Z=\frac{S}{2\sqrt{m_0}}$$

$$Q=\frac{1.25(\gamma-D_3-D_2 R)}{D_1},\qquad R=\frac{\gamma-x_m-D_1^2}{1-\gamma-D_1+D_1^2}$$

$$\gamma=\frac{m_2}{\sqrt{m_0 m_4}}, \qquad x_m=\frac{m_1}{m_0}\sqrt{\frac{m_2}{m_4}}$$

In the previous equations, m_0, m_1, m_2, m_3 and m_4 represent each spectral moment.

3. Application

Both the quasi-static durability analysis technique and the resonance durability analysis technique are applied in the durability assessment for the rear suspension system of a small-sized passenger car.

3.1 Vibration Analysis

The finite element model of the rear suspension system used in this study consisted of 7,000 nodes and 6,000 elements. Free vibration analysis without boundary conditions is performed by using the commercial finite element analysis software, MSC/NASTRAN. Fig. 1 shows the 1st vibration mode of the rear suspension system, and the natural frequency of the system is 56Hz. From the analysis result, if the rear suspension system is excited over 56Hz, it can be predicted that the vibration characteristics of the rear suspension system will considerably affect the fatigue life of suspension components.

3.2 Linear Static Analysis

To generate the static stress data used in the quasi-static durability analysis, the linear static analysis is carried out for the rear suspension system. As boundary conditions for the analysis, the rear suspension system is constrained at 7 connecting points with the spring elements. Fig. 2 indicates the static load points, and the stress influence coefficients are obtained by acting in turn the unit load at each load point. Fig. 3 shows the stress distribution due to the longitudinal unit load acting at the hub-rear axle connecting point. As shown in this figure for such load case, the maximum stress is generated near the damper-bracket connecting point.

3.3 Modal Analysis

The modal analysis to obtain the frequency response data of the rear suspension system is performed. The boundary conditions for the modal analysis are equal to the linear static analysis stated in the previous section, and the rear suspension system is excited from 0Hz to

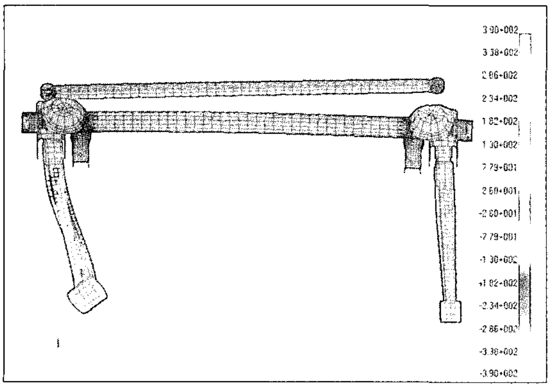


Fig. 1 1st vibration mode of rear suspension system

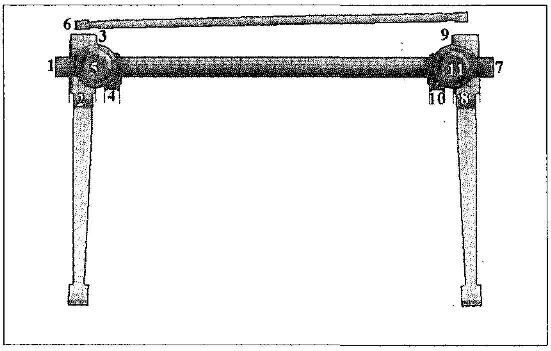


Fig. 2 Unit load positions of rear suspension system

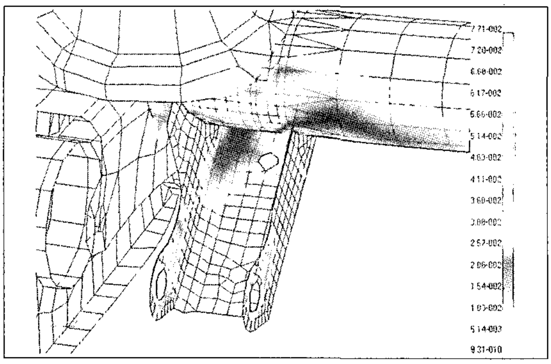


Fig. 3 Stress distribution due to longitudinal unit load acting at hub-rear axle connecting point

450Hz at intervals of 2Hz.

3.4 Dynamic Analysis

The multi-body dynamic model of the small-sized passenger car used in this study consisted of 24 rigid bodies, kinematic joints and force elements. Fig. 4 shows the vehicle dynamic model used in this study. Through the dynamic simulation using the commercial dynamic analysis software, ADAMS, the dynamic load histories for the vehicle system are generated.

3.4.1 Pothole Simulation

Virtual proving ground(VPG) simulation is carried out running over the potholes by using the small-sized vehicle dynamic model. The dynamic load data are extracted for 4 seconds, and then the extracted data are converted into the frequency domain data by using the fast Fourier transformation(FFT). The dynamic load time histories extracted are used in the quasi-static durability analysis, and the dynamic load data converted into the frequency domain are used in the resonance durability analysis.

Fig. 5 shows a portion of the dynamic load data taken from the pothole simulation. As shown in these figures, the vehicle system is excited under 50Hz when the vehicle system runs over the potholes at the velocity of 40km/h, therefore, the dynamic characteristics of the vehicle system due to the vibration behavior of the rear suspension system will not affect the durability performance.

3.4.2 Belgian Road Simulation

To perform the Belgian road simulation, 4 drivers are set up at the front lower control arm(LCA)-knuckle joints and at the rear axle-wheel hub joints of the vehicle dynamic model. The dynamic load data are extracted for 8 seconds by using the Belgian road simulation dynamic model.

Fig. 6 shows a portion of dynamic load data taken from the Belgian road simulation. As shown in these figures, because the vehicle system is excited up to 400Hz when the vehicle system runs over the Belgian road, the excitation frequencies are overlapped with the natural frequency

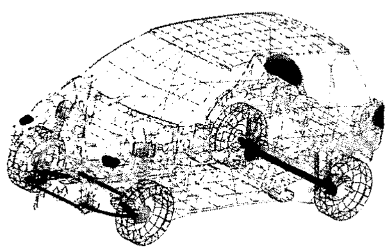
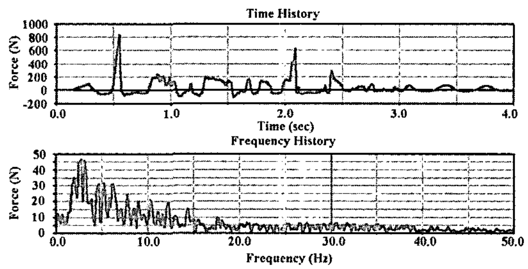
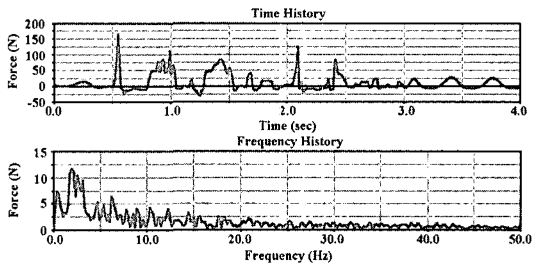


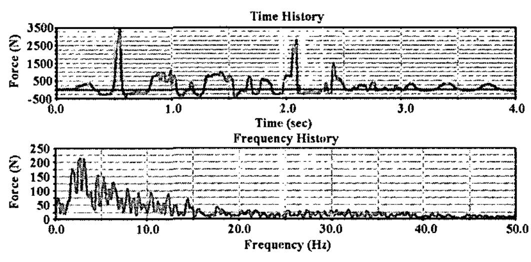
Fig. 4 Vehicle Dynamic Model



(a) Longitudinal load at rear damper-bracket connecting point

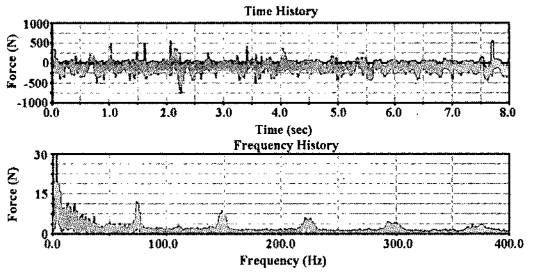


(b) Lateral load at rear damper-bracket connecting point

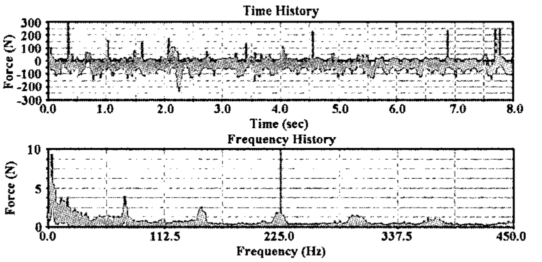


(c) Vertical load at rear damper-bracket connecting point

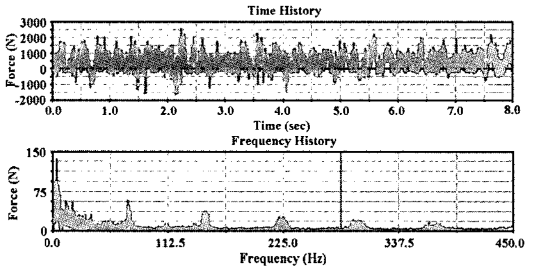
Fig. 5 Dynamic loads due to pothole simulation



(a) Longitudinal load at rear damper-bracket connecting point



(b) Lateral load at rear damper-bracket connecting point



(c) Vertical load at rear damper-bracket connecting point

Fig. 6 Dynamic loads due to Belgian road simulation

range of the rear suspension system.

3.5 Durability Analysis

In this study, the durability analysis is performed by using commercial software, MSC/FATIGUE. And SAE 1035-169-CON material is used in this durability analysis. Table 1 lists the analysis conditions for the fatigue life estimation.

Table 1 Analysis conditions for fatigue life estimation

SAE 1035-169-CON	
Yield strength	410MPa
Ultimate tensile strength	550MPa
Elastic modulus	210GPa
Stress range intercept	2.137GPa
Fatigue transition life	1E6 cycle
First fatigue strength exponent	-0.0872
Second fatigue strength exponent	-0.0872
R-ratio of test	-1

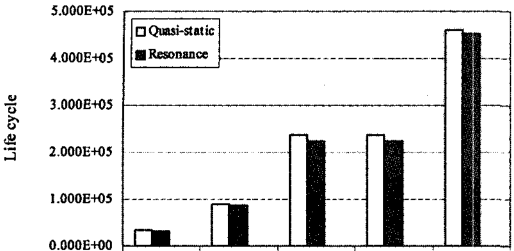
3.6 Durability Assessment

3.6.1 Durability Assessment for Pothole Simulation

The 5 nodes, in which small fatigue lives are calculated for the pothole simulation, are illustrated in Fig. 7. All nodes shown in this figure are located near the damper-bracket connecting part of the rear suspension system. As shown in this figure, the percent changes of the fatigue life between the two durability analysis techniques don't exceed 10%. As shown in the previous section, because the excitation frequencies are occurred under 50Hz when the vehicle system runs over potholes, the excitation frequencies don't occur the resonances with the natural frequencies of the rear suspension system. Therefore, there are no large differences for the fatigue lives between the two durability analysis techniques.

3.6.2 Durability Assessment for Belgian Road Simulation

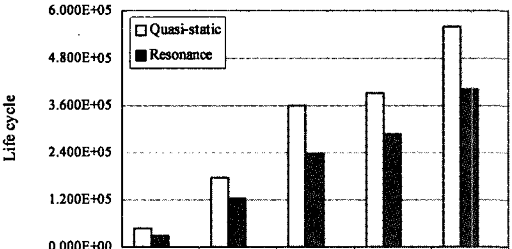
For the Belgian road simulation, the 5 nodes, in which small fatigue lives are calculated, are illustrated in Fig. 8. In this simulation using the Belgian road, there are remarkable differences in the fatigue lives between the two durability analysis techniques. As shown in the modal analysis, because the excitation frequencies are generated up to 450Hz when the vehicle system runs over the Belgian road, the resonances are occurred when the excitation frequencies approach the natural frequencies of



	200239	200512	200252	200229	200238
Quasi-static	3.505E+04	8.995E+04	2.380E+05	2.380E+05	4.611E+05
Resonance	3.189E+04	8.568E+04	2.227E+05	2.234E+05	4.529E+05
Percent change	9.00%	4.70%	6.40%	6.10%	1.80%

Node number

Fig. 7 Comparison of fatigue life for pothole simulation



	200239	200512	200252	200229	200238
Quasi-static	4.783E+04	1.761E+05	3.596E+05	3.931E+05	5.595E+05
Resonance	2.701E+04	1.216E+05	2.380E+05	2.865E+05	4.003E+05
Percent change	43.50%	30.90%	33.80%	27.30%	28.40%

Node number

Fig. 8 Comparison of fatigue life for Belgian road simulation

the rear suspension system.

In result, because of the resonance effect, the fatigue lives using the resonance durability analysis technique were smaller estimated than the quasi-static durability analysis results. Therefore, in the case that the natural frequencies of the system and the excitation frequencies are gotten near each other, if the durability assessment is carried out with the quasi-static analysis technique, the fatigue life is larger estimated than the case of considering the vibration effect. So, to obtain more accurate fatigue damage or fatigue life, it is desirable to apply the resonance durability analysis technique to the vehicle system including the flexible components such as the rear suspension system.

Fig. 9 illustrates the fatigue damage contour plot for the resonance durability analysis using the Belgian road simulation.

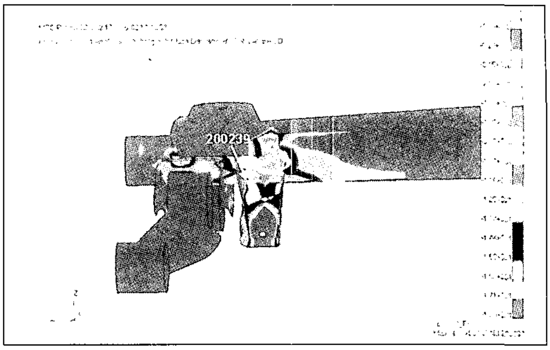


Fig. 9 Fatigue damage contour plot for resonance durability analysis using Belgian road simulation

4. Conclusion

In this paper, the resonance durability analysis technique is developed for the fatigue life assessment considering the vibration characteristics of a vehicle chassis system. When the resonance durability is analyzed, the frequency response data and the dynamic load data of the frequency domain are used. Multi-body dynamic analysis, finite element analysis, and fatigue life prediction method are applied for the virtual durability assessment. To obtain the dynamic load histories, the computer simulations running over the typical pothole and the Belgian road respectively are carried out with the vehicle dynamic model.

The durability estimations for the rear suspension system of the small-sized passenger car are performed by using the proposed resonance durability analysis technique and then compared with the quasi-static durability analysis results. For the pothole simulation, the percent changes of the fatigue life between the two durability analysis techniques don't exceed 10%. But, for the Belgian road simulation, because of the resonance effect, the fatigue lives using the resonance durability analysis technique are smaller estimated than the quasi-static durability analysis results.

The study shows that the fatigue life considering the resonance frequency of the vehicle system can be effectively estimated in early design stage.

Acknowledgement

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