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A Study on the Methodology for Determining Dynamic Loadings of Automotive Suspension System Using Measurement and Modeling

실차측정과 모델링을 이용한 자동차 현가시스템의 동적하중 결정 방법론에 관한 연구

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ABSTRACT

To design suspension system and estimate its durability, the loading history of each suspension part exposed to various operation conditions should be known from either measurements or computations. Based on these results, stress analysis is carried out to obtain the optimal shape and to reduce the production cost through the proper selection of manufacturing processes.

In this paper, first the measurements of 3-directional accelerations of wheel center using an accelerometer are undertaken from a vehicle running on Belgian road. Then the data measured from experiments are pre-processed with filtering. Based on the pre-processed data the methodology for determining the dynamic loadings to each suspension part is developed by simply modeling the suspension system with ADAMS software. Eventually, it is expected that dynamic loadings can be used for the dynamic stress and fatigue analyses.

Key words : Proving Ground Test, Acceleration Measurement, Suspension Dynamic Load, Suspension Modeling, Fatigue Analysis

1. Introduction

The critical parameters in the phase of designing an automotive vehicle are performance, fuel economy, production cost, safety, durability, and so on. Among these parameters, fuel economy which is strongly related with operating cost and global warming can be mainly achieved by making vehicles light-weight. [1] *One of the feasible methods to* manufacture lightweight vehicles is to reduce the

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weight of suspension component, weighing about 70Kg. Meanwhile, since suspension components are irregularly and repeatedly loaded over the entire vehicle life span, their strength and fatigue life must be guaranteed prior to design and manufacturing stages. [1] If light-weight suspension parts under such loading conditions do not have sufficient strength and durability, the functions of parts will gradually degrade. Also initiation and growth of cracks in suspension parts can lead to a catastrophic failure i.e., fatigue failure in service. Therefore, the appraisal technology of the fatigue strength is vital to successfully accomplish reduction of weight while maintaining optimal performance and without jeopardizing the safety as well as durability of a vehicle. [2), 3)]

Car makers have pursued to acquire design technology for their own suspension system realizing that suspension is of paramount importance affecting drivability and safety. As the current trend of suspension system becomes multi-link type, entirely different from the conventional types, the loading characteristics of each suspension part are completely remote from those of a conventional one. To design suspension system and estimate its durability, the loading history of each suspension part exposed to various operation conditions should be known from either measurements or computations. Based on these results, stress analysis is carried out to obtain the optimal shape and to reduce the production cost through the proper selection of manufacturing processes.

The magnitude of dynamic loadings to suspension components are generally known to be varied: typically a combination of braking 1G, cornering 1G and bumping 3G, or unidirectional 6G under gross vehicle weight. For example, vehicles running on Belgian road are most severely loaded in both lateral and longitudinal directions. To estimate the durability and strength of each suspension part, the most severe loading should be selected. Then, it is applied to stress analysis of each suspension part for design. Usually the magnitude of loadings can be determined either by calculation based on vehicle data, or by an experiment using suspension equipped with a sensor or a measuring wheel. [4] However, the experimental approach is often favored for more reliable load determination. [1), 5)] Since the direct measurement of dynamic loadings to each suspension part cannot be easily achieved, an indirect method has been preferred. The indirect method utilizes the acceleration of wheel center, from which the equivalent loadings to each suspension part are computed through the modeling of the suspension system. [6] Based on these dynamic loading data, the optimal design of each suspension part and the prediction of its fatigue life are possible. In addition, a durability test method simulating the actual vehicle in-use condition and a durability testing machine need to be developed to assure the durability after suspension parts are designed and made.

In this study, first the measurements of 3-directional accelerations of wheel center using an accelerometer are undertaken from a vehicle running on Belgian road. Then the data measured from experiments are pre-processed with filtering. Based on the pre-processed data the methodology for determining the dynamic loadings to each suspension part is developed by simply modeling the suspension system with ADAMS software. In the future, it is expected that dynamic loadings can be used for the dynamic stress and fatigue analyses.

2. Measurement procedures

In order to estimate the durability of automotive parts, road loading history during their operations must be known in advance. Life expectancy can be predicted using loading history, and thus optimization and reduction of their weight can also be achieved. To test the suspension effectively and reliably, testing conditions are most important. Proving ground, generally used for a durability test of vehicles, includes a variety of particular purpose roads as well as the high speed way. To acquire high reliability and reproducibility of tests, test courses in the proving ground are built according to their test purposes. In reality, vehicles may be driven on all types of roads. [1) However, since repetitive loadings from the most severe road can accelerate the durability testing of each suspension part, [7) Belgian road in the proving ground is used.

In this study, 3-directional accelerations of wheel center are measured from a vehicle running on Belgian road. Its speed is 30 km/h, sampling rate 400 Hz, and sampling time 2 seconds. Measurements of 3-directional accelerations have been made using an accelerometer mounted on wheel center. Signals from sensors are amplified by a charge amplifier. These signals are recorded on a tape recorder and processed in a laboratory with a digital filter. A schematic diagram for measuring data is depicted in Figure 1. The mounting locations of the accelerometer and measured data are shown in Figure 2 and 3, respectively. x-, y- and z-directions in Figure 3 designate longitudinal, lateral and vertical directions, respectively. It is known that the magnitudes of accelerations of wheel center are large in longitudinal and vertical directions while vehicles are running on Belgian road. [8) The same pattern is also observed in Figure 3. Although loadings from Belgian road are irregular, these loadings have a certain statistical meaning. The dynamic behavior of a wheel for 2 seconds imparts dynamic loadings to each suspension part. And these loadings represent a circumstance under which the fatigue of a suspension component is accelerated.

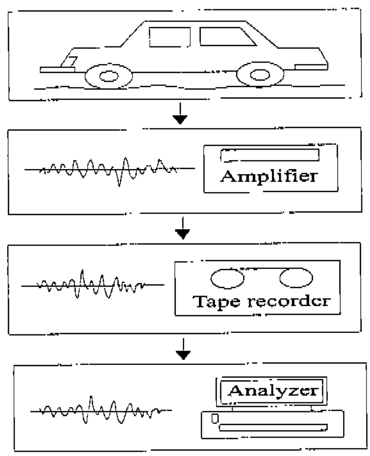


Fig. 1 A schematic diagram of data measurement

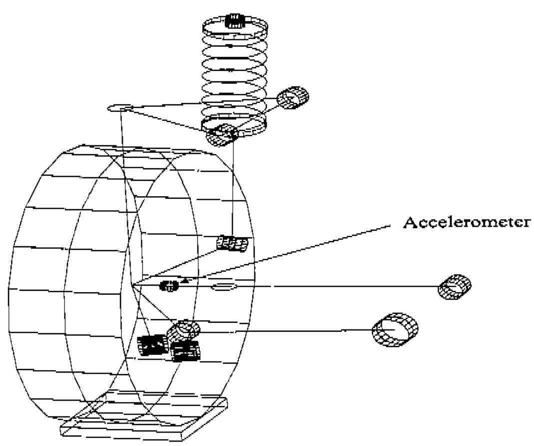


Fig. 2 Mounting location of an accelerometer

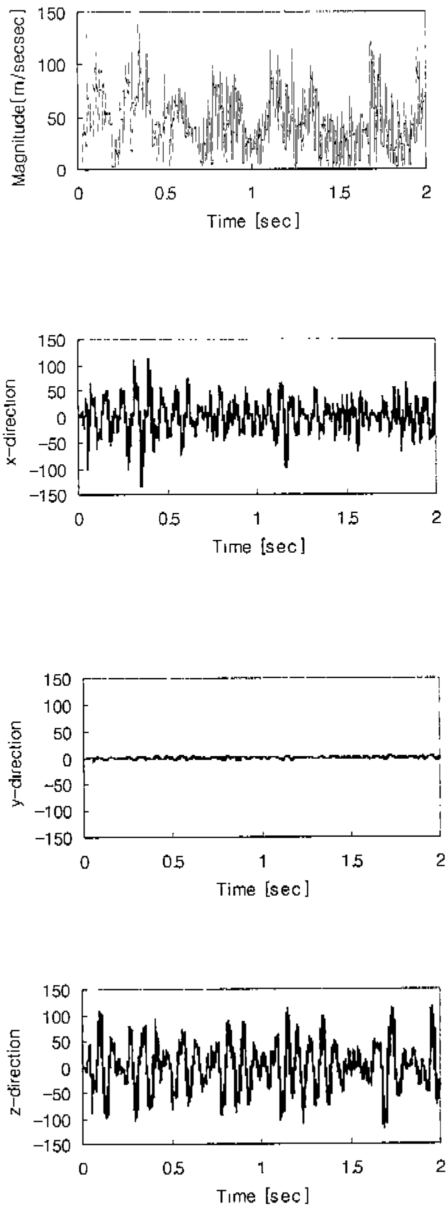


Fig. 3 Acceleration data measured in a vehicle test

The measured data of accelerations on Belgian road is inevitably contaminated with various noises. They include high frequency noises such as a sensor noise and a charge amplifier noise, and low frequency noise such as one from the rigid body motion of a vehicle. These unwanted signals must be separated from the acceleration data. Thus, the raw signal must be processed with a filter. Butterworth filter is used for signal processing and its frequency response is shown in Figure 4. The z-directional acceleration includes rigid body motion of a vehicle, which ranges from 0 to 4 Hz. Thus the lower limit is selected to be 4 Hz. The upper limit is set at 50 Hz because a signal above 50 Hz is not considered a signal from the suspension system.

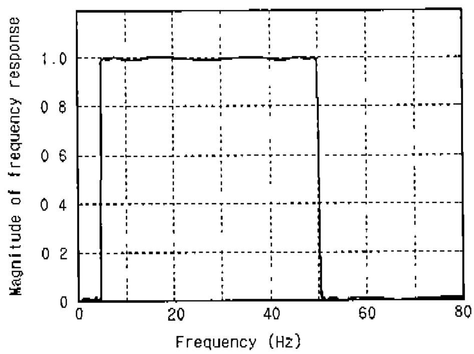


Fig. 4 Frequency response of Butterworth filter for signal processing of measured data

3. Computation of dynamic loadings to suspension parts using ADAMS

Since vehicle suspension consists of complicated kinematic and dynamic structures, a considerable amount of computations is required for the analysis. For the development of vehicle suspension many world class auto makers currently use ADAMS, a multi-body dynamics software. [9] ADAMS is also used in this study for the computation of dynamic loadings acting on each suspension part. Input data to the software are filtered accelerations experimentally measured from the actual vehicle test. Suspension can be modeled as a simple combination of spherical, universal and translational joints.

3.1 Multi-body dynamics

The accurate modeling of forces acting on various components is of importance for the analysis of vehicle dynamics. Forces acting on a system is divided into externally applied forces and internally constrained forces. Without friction, the latter does not do any work in a system comprised of kinematically compatible rigid body. Consequently, only the externally applied loads such as gravitational force, spring forces between rigid bodies, and damping forces are considered. The forces considered in vehicle suspension system are translational spring damper elements(coil spring, shock absorber), rotational spring damper elements(torsion bar, friction), stabilizer, tire and bushing. These forces are input to the equation of motion of a vehicle, and the motion of a vehicle is characterized in the result.

There are numerous methods to represent the equation of motion of a kinematically constrained rigid body. In this study, however, the equation of motion is derived by introducing the differential equations for n rigid bodies using Newton-Euler equation, and Lagrangian multipliers. Equations of motion are represented as

$$\sum_{i=1}^{n} \delta z_i^T (M_i Y_i - Q_i + \Phi_{Z_i}^T \lambda) = 0 \qquad (1)$$

where, M_i :6x6 inertia matrix of body i

Y_i :6x6 acceleration matrix of body i

Q_i : external force vector applied to rigid body i

Φ_{Z_i} : Jacobian matrix equivalent to kinematically constrained condition

λ : Lagrangian multipliers equivalent to kinematically constrained condition

Z_i : displacement vector of rigid body i

n : number of rigid bodies consisting of a system

3.2 Spherical(ball), universal and translational joints

Vehicle suspension is composed of various joints, which have their own conditions of constraint. It can be modeled as a simple combination of spherical, universal and translational joints. Spherical(ball) joints representing upper and lower arm have the following constraint equations and their schematic representation is shown in Figure 5. [10]

$$\Phi^{(s,3)} \equiv r_i + A_i S_i'^p - A_j S_j'^p - r_j = 0 \qquad (2)$$

,where r, S, and A represent position vectors in the global and body coordinates, and transformation matrix, respectively.

r_i : global coordinate vector to the origin of body fixed coordinate for each body i

r_j : global coordinate vector to the origin of body fixed coordinate for each body j

S'^p_i : local coordinate vector of body i

S'^p_j : local coordinate vector of body j

A_i : rotational transformation matrix of body i

A_j : rotational transformation matrix of body j

There are three relative degrees of freedom between two bodies which are connected by a ball joint.

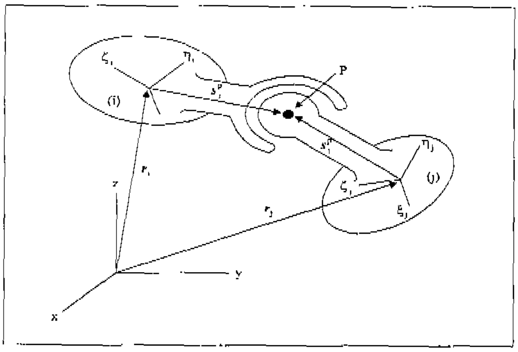


Fig. 5 Spherical joint

A part of steering system attached to suspension system is modeled as a universal joint. As shown in Figure 6, both pins are maintained in vertical manner and the constraint equation is expressed as

$$\Phi^{(s,3)} = 0$$
$$\Phi^{(n1,1)} \equiv S_i^T S_j = 0 \qquad (3)$$

There are two relative degrees of freedom between a pair of bodies which are connected by a universal joint.

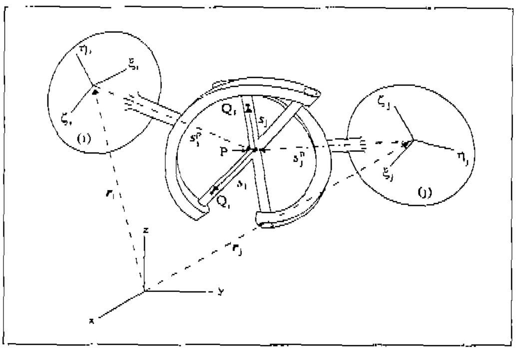


Fig. 6 Universal joint

A translational joint is modeled as the pinion and rack of steering system. As shown in Figure 7, two perpendicular vectors, h_i and h_j on bodies s_i and s_j, must remain perpendicular. Therefore, there are three constraint equations for a translational joint:

$$\Phi^{(p1,3)} \equiv S_i^T S_j = 0$$
$$\Phi^{(p2,2)} \equiv S_i^T d = 0 \qquad (4)$$
$$\Phi^{(n1,1)} \equiv h_i^T h_j = 0$$

There are one relative degree of freedom between two bodies which are connected by a translational joint.

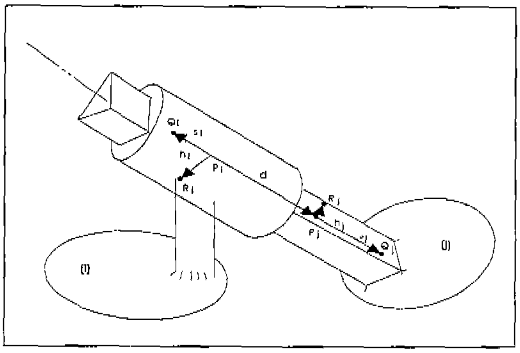


Fig. 7 Translational joint

3.3 Modeling of suspension using ADAMS

ADAMS can model suspension parts with consideration of their characteristics. Also, performance change of suspension can be predicted according to the changes of dynamic behaviors of a vehicle and the characteristics of parts. Suspension parts such as upper and lower arms, strut, upper and lower bushes, ball joints, spring system and trailing arm, constitute force elements and constraint elements. In this study, the suspension system of a mid-sized car newly introduced in the domestic market is investigated. The front suspension is the double wishbone type and the result modeled by ADAMS is shown in Figure 8. Since it also includes steering system the whole system is modeled. Upper and lower ball joints are modeled as spherical joints. However, the part of lower and upper arms mounted on the vehicle body is modeled as a non-linear bush element. Steering system is modeled as a combination of universal and translational joints. Rear suspension is multi-link type and considered as a variation of double wishbone type. The analysis for computing dynamic load is conducted in dynamic state during 2 seconds.

In ADAMS/Solver, a motion can be applied to a translational, to a revolute, or to a cylindrical joint. For the velocity and acceleration motions, ADAMS/Solver internally generates one and two differential equations, respectively. By numerically integrating the internal integrator, ADAMS/Solver obtains the displacement of the motion. It is possible to obtain a relationship between the constraint reaction forces and the constraint equations if (1) a proper vector of coordinates is defined and (2) the constraint forces are expressed with respect to the same coordinates systems as the vector of coordinates. The reaction force is described as

$$g^{(*)} = \Phi_q^T \lambda \tag{5}$$

where Φ_q: Jacobian matrix

λ: Lagrangian multiplier

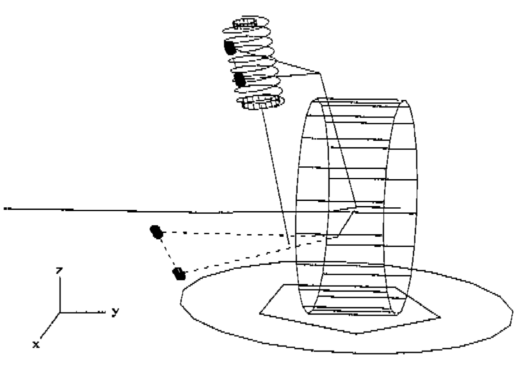


Fig. 8 Front suspension

4. Results and discussions

In this analysis, one of the suspension components, ball joints of lower and upper arms, is investigated to compute dynamic loadings acting on them. From the simulation with experimental results, Figure 9 shows the displacements of the wheel center in x, y and z directions and the magnitude. The maximum value of displacement is about 0.02 m, and the same magnitude of displacement is repeatedly input to the wheel center. Figure 10 and 11 show dynamic loadings in x, y and z directions and the magnitude acting on ball joints of upper arm and lower arm, respectively. Since the ball joint of upper arm does not carry the weight of the vehicle, the average value of loading in z direction is nearly zero. Meanwhile, the ball joint of lower arm always carries a quarter of the weight of the vehicle, a certain level of loading is always applied. From Figure 9 and 10, it is natural that the frequency range of dynamic loadings be higher than that of displacements. Also, from the peak values of Figure 10 and 11, the ball joint of lower arm is more severely loaded than that of upper arm. In the design stage, therefore, a designer should consider this uneven level of loadings. These dynamic loadings can be input to finite element analysis to compute stresses of suspension components. Computational results of stresses can also be used to predict their fatigue life with the rainflow counting method. Thus, suspension can be optimally designed to reduce its weight and cost. [11), 12)]

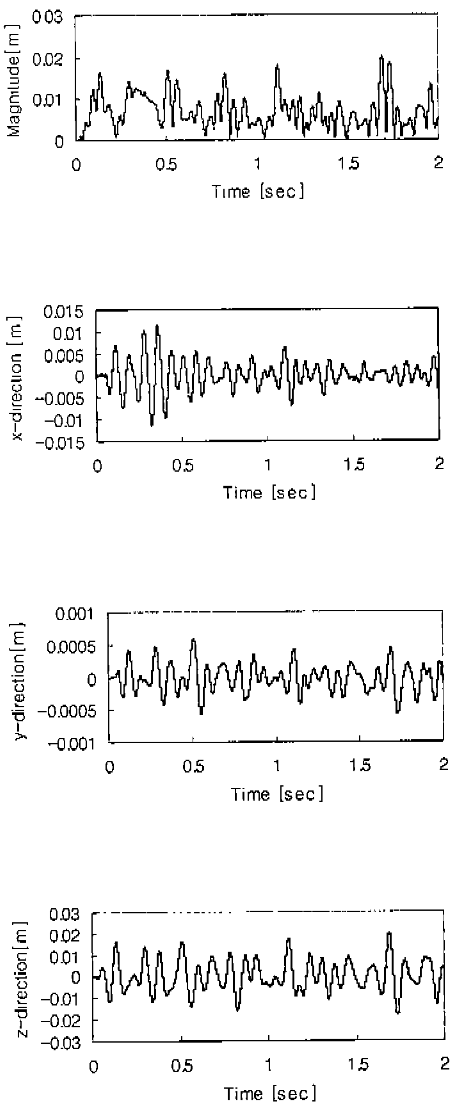
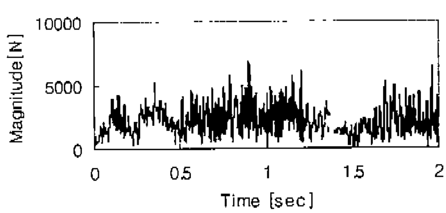


Figure 9 Displacements of wheel center



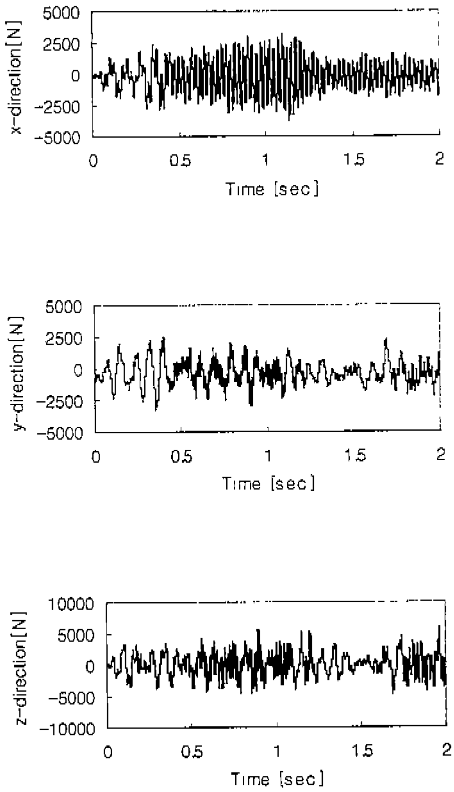
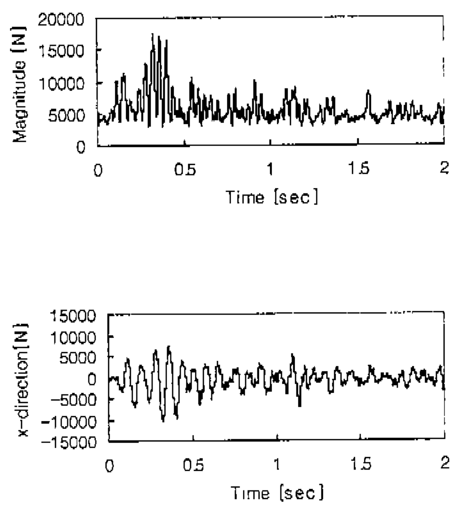


Figure 10 Dynamic loadings acting on upper arm ball joint



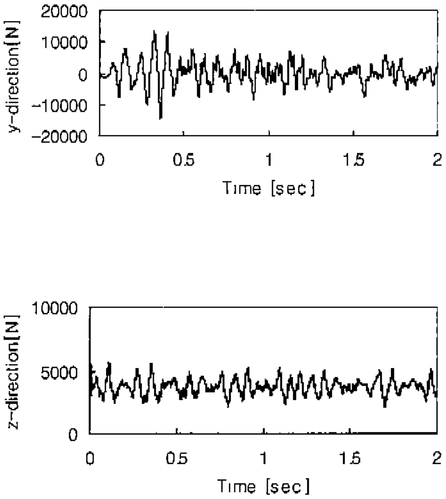


Figure 11 Dynamic loadings acting on lower arm ball joint

5. Conclusions

In this study, systematic methodology for computing the dynamic loadings acting on each suspension component has been proposed by using the commercial software and experimental data. Accelerations of wheel center in lateral, longitudinal, and vertical directions are measured under the most severe condition i.e., Belgian road. For instance, dynamic loadings acting on the ball joints of upper and lower arms are computed by simply modeling the suspension system as the combination of spherical, universal, and translational joints. These loadings can be used not only for the optimal design of suspension parts in terms of strength but also for predicting its fatigue life without performing a bench test.

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