CRASH ANALYSIS OF AUTO-BODY STRUCTURES CONSIDERING THE STRAIN-RATE HARDENING EFFECT

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ABSTRACT–The crashworthiness of vehicles with finite element methods depends on the geometry modeling and the material properties. The vehicle body structures are generally composed of various members such as frames, stamped panels and deep–drawn parts from sheet metals. In order to ensure the impact characteristics of auto–body structures, the dynamic behavior of sheet metals must be examined to provide the appropriate constitutive relation. In this paper, high strain–rate tensile tests have been carried out with a tension type split Hopkinson bar apparatus specially designed for sheet metals. Experimental results from both static and dynamic tests with the tension split Hopkinson bar apparatus are interpolated to construct the Johnson–Cook and a modified Johnson–Cook equation as the constitutive relation, that should be applied to simulation of the dynamic behavior of auto–body structures. Simulation of auto–body structures has been carried out with an elasto–plastic finite element method with explicit time integration. The stress integration scheme with the plastic predictor–elastic corrector method is adopted in order to accurately keep track of the stress–strain relation for the rate–dependent model accurately. The crashworthiness of the structure with quasi–static constitutive relation is compared to the one with the rate-dependent constitutive model. Numerical simulation has been carried out for frontal frames and a hood of an automobile. Deformed shapes and the impact energy absorption of the structure are investigated with the variation of the strain rate.

KEY WORDS : Split Hopkinson bar, Crashworthiness, Strain rate hardening

1. INTRODUCTION

The crashworthiness of auto-body structures is one of the important issues in the automotive industry. The automotive industry has made an effort to develop light-weight auto-body structures for the purpose of increasing the fuel efficiency and satisfying the emission gas regulation of vehicles. Since the weight reduction of an auto-body should not sacrifice the safety, the crash analysis has to be accurately conducted in order to efficiently reduce the auto-body weight. The vehicle body structures are generally constructed by sheet metals of deep-drawn parts. Since the strength of steel sheets depends on the rate of deformation, the dynamic behavior of sheet metals is a key to investigate the impact characteristics of the structure.

As the dynamic behavior of a material is different from the static one due to the inertia effect and the propagation of stress waves, an adequate experimental technique has to be developed to obtain the dynamic

response for the corresponding level of the strain rate. A high strain-rate testing apparatus was devised by Kolsky (1949), which is known as the split Hopkinson pressure bar (Follansbee, 1978). The stress-strain curves for the high strain-rate ranged from 1000 to 10000/sec can be acquired from the stress waves propagating through the incident and the transmitted bar in the apparatus. The split Hopkinson pressure bar apparatus can be modified for high strain-rate tensile tests, although there are some difficulties in design of grips and specimens as well as estimation of errors (Li *et al.*, 1993; Kang *et al.*,1998).

Various types of constitutive relations have been proposed to describe the dynamic behavior of materials. Johnson and Cook (1983) suggested a constitutive model and determined five material constants in the constitutive relation for materials subjected to large strains, high strain-rates and high temperatures. Zhao and Gary (1996) performed high strain-rate compression tests of sheet metals with a compression split Hopkinson bar apparatus. Kang *et al.* (1999) acquired the high strain-rate tensile properties of sheet metals with a tension split

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Hopkinson bar apparatus and proposed a modified Johnson–Cook model for sheet metals. Results showed that the model has to be described with a quadratic function for the strain-rate term in order to fit experimental results for sheet metal.

In this paper, numerical simulations are carried out with an elasto-plastic explicit finite element method. A shell element formulation has been derived for the crashworthiness of auto-body structures incorporating with a contact algorithm. The present algorithm adopts the plastic and predictor elastic corrector(PPEC) scheme (Nemat-Nasser and Chung, 1992) in stress integration in order to accurately keep track of the stress–strain relation for the rate-dependent model. The constitutive model obtained from dynamic experiments is imposed on the analysis for the dynamic response of auto-body structures instead of the static model that has been generally adopted in dynamic simulation. The crash analysis is performed for auto-body structures such as frontal frames of an automobile and a hood. The finite element model of an automobile is adopted from the web site of NHTSA (National Highway Traffic Safety Administration). Simulation results provide the deformed shape and the deformation energy in order to predict and evaluate the crashworthiness of a car.

2. FINITE ELEMENT FORMULATION

For an updated Lagrangian formulation, the configuration Ω^t at time t is taken as the reference state so that all static and kinematic variables are referred to this configuration. An incremental form of the principle of virtual work can be expressed as Equation (1) with an increment of Δt which is small enough that the first Piola–Kirchhoff is nearly the same as the Cauchy stress (Huh et al., 1990).

$$\int_{\Omega^t} \rho \ddot{u}_i^{t+\Delta t} \bar{u}_i \, d\Omega$$
$$= \int_{\Omega^t} \rho b_i^{t+\Delta t} \bar{u}_i \, d\Omega + \int_{\Gamma_t} \rho t_i^{t+\Delta t} \bar{u}_i \, d\Gamma - \int_{\Omega^t} \sigma_{ij}^{t+\Delta t} \bar{u}_{i,j} \, d\Omega, \forall \bar{u}_i \quad (1)$$

such that $\bar{u}_i = 0$ *on* Γ_d

where $\bar{u}_i$ is an arbitrary virtual displacement vector, σ_{ij} is the Cauchy stress tensor defined in the configuration, ρ is the mass density, b_i is the body force vector. The above equation is then approximated into finite dimension with the use of the shell element by the degenerated continuum approach.

The stress and the rate of deformation are expressed in the co-rotational coordinates at the centroid of an element. The local Cartesian coordinate system is defined element-wise by the local Cartesian unit

vectors $\hat{e}_1, \hat{e}_2, \hat{e}_3$. The superscript $\wedge$ denotes the local Cartesian coordinate system while e_1, e_2, e_3 are the global Cartesian unit base vectors. From the Mindlin–Reissner formulation that admits the transverse shear but assumes that the pseudo-normal remains straight, the velocity interpolation for the element in the co-rotational components can be written as follows:

$$\hat{v}_i(\xi, \eta, \varsigma) = N_I(\xi, \eta)\left[\hat{v}_{iI} + \frac{d}{2}(-\delta_{i2}\hat{\omega}_{1I} + \delta_{i1}\hat{\omega}_{2I})\right] \quad (2)$$

where d is the thickness, $N_I(\xi, \eta)$ are the isoparametric shape function and δ is the Kronecker delta. The rate-of-deformation tensor in the local Cartesian coordinate system is written as follows:

$$\hat{d}_{ij} = \frac{1}{2}(\frac{\partial \hat{v}_i}{\partial \hat{x}_j} + \frac{\partial \hat{v}_j}{\partial \hat{x}_i}) \quad (3)$$

The poor behavior of warped elements is corrected by superposing additional terms into the strain-displacement equations. The additional terms couple the curvature to the translation within an element (Belytschko et al.,1992; Belytschko and Leviathan, 1994). A reduced integration technique is employed to avoid locking phenomena and efficiently perform time integration with an explicit scheme. The physical stabilization scheme (Belytschko and Leviathan, 1994) is used to control the zero-energy mode that results from the reduced integration rule. To solve the governing equation, the internal forces and moments calculated in the local coordinate system are to be expressed in the global Cartesian coordinate system with the transformation matrix $\tilde{R}$.

$$f_I = \tilde{R}^t \hat{f}_I \quad (4)$$

where $\hat{f}_I$ is the global nodal force vector calculated from the principle of virtual power. The transformation matrix is expressed as follows:

$$\tilde{R} = \begin{bmatrix} R_{3\times3} & 0_{3\times3} \\ 0_{2\times3} & R_{2\times3} \end{bmatrix} \quad (5)$$

where

$$R_{ij} = \hat{e}_i \cdot e_j \quad (6)$$

Equation (1) can be approximated by spatial discretization with shell elements as a matrix equation as follows:

$$M_i \ddot{u}_i^{t+\Delta t} = P_i^{t+\Delta t} - F_i^{t+\Delta t} + H_i^{t+\Delta t} \tag{7}$$

where M_i is the lumped mass matrix, $P_i^{t+\Delta t}$ is the external force vector at time $t+\Delta t$, $F_i^{t+\Delta t}$ is the internal force vector at time $t+\Delta t$ and $H_i^{t+\Delta t}$ is the hourglass resistance vector.

The explicit finite element analysis solves a set of hyperbolic wave equations in the zone of influence of the wave front without coupling large number of equations. Explicit integration of the governing equation in the time domain is performed with the central difference scheme. The velocity and displacement vectors at the current time step are calculated as follows:

$$\dot{u}_i^{t+\frac{\Delta t}{2}} = \dot{u}_i^{t-\frac{\Delta t}{2}} + \Delta t \ddot{u}_i^{t} \tag{8a}$$

$$u_i^{t+\Delta t} = u_i^{t} + \Delta t \dot{u}_i^{t+\frac{\Delta t}{2}} \tag{8b}$$

The critical time step of the central difference integration is calculated from the Courant stability condition that is the period of the maximum natural frequency of the system.

The velocity gradient for each shell element is calculated with an explicit time integration scheme for dynamic elasto-plastic finite element analysis. A simple explicit algorithm that estimated the subsequent yield surface in each time increment without iteration was proposed by Nemat–Nasser and Chung (1992). Since the incremental deformation for a designated time step is assumed to be plastic at first and then corrected along an elastic integration path, this method is called the plastic predictor – elastic corrector (PPEC). The method is stable and accurate regardless of the time or strain increment and accurately calculates the stress in large plastic deformation state with much smaller time steps than that of the radial return method. For the reason, PPEC is adopted in the present shell formulation.

3. EXPERIMENT AND CONSTITUTIVE RELATIONS OF SHEET METALS

A tensile split Hopkinson bar apparatus is used to acquire high strain-rate behavior of steel sheets as shown in Figure 1. The signals from the strain gages are monitored and acquired by an oscilloscope. The reflected pulse measured by an oscilloscope is used to calculate the strain rate in a specimen with Equation (9). This strain rate is integrated with respect to time in order to obtain the strain in a specimen as represented by Equation (10). The transmitted pulse is used to calculate the stress in a specimen with Equation (11).
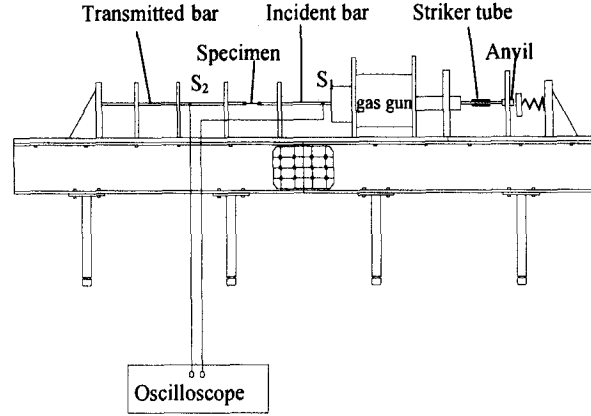

Figure 1. Schematic description of the tension split Hopkinson bar apparatus.

$$\dot{\varepsilon}(t) = 2\frac{C_e}{L_s}\varepsilon_R(t) \tag{9}$$

$$\varepsilon(t) = \int_0^t \dot{\varepsilon}(\tau)d\tau \tag{10}$$

$$\sigma(t) = E\frac{A_0}{A_s}\varepsilon_T(t) \tag{11}$$

where C_e is the speed of an elastic wave in a bar and L_s is the effective gage length of a specimen. The subscript R and T indicate the reflected and the transmitted pulse respectively. A_0 and A_s are the area of a bar and a specimen respectively. E is Young's modulus of a bar. Quasi-static tensile tests were carried out at the strain rate of 0.003/sec and 1/sec. The test result at the strain rate of 1/sec was chosen as the reference stress-strain curve to determine the constants in the Johnson Cook constitutive relation. The strain rate acquired in the present experiments was ranged from 0.003/sec to 10000/sec. Experimental results represented in Figure 2 are interpolated to construct constitutive relations of sheet metals. The conventional Johnson Cook model is written as Equation (12).

$$\bar{\sigma} = (A + B\bar{\varepsilon}^n)(1 + C\ln\bar{\dot{\varepsilon}})(1 - T^{*m}) \tag{12}$$

where T^* is the homologous temperature represented by Equation (13).

$$T^* = \frac{T - T_{room}}{T_{melt} - T_{room}} \tag{13}$$

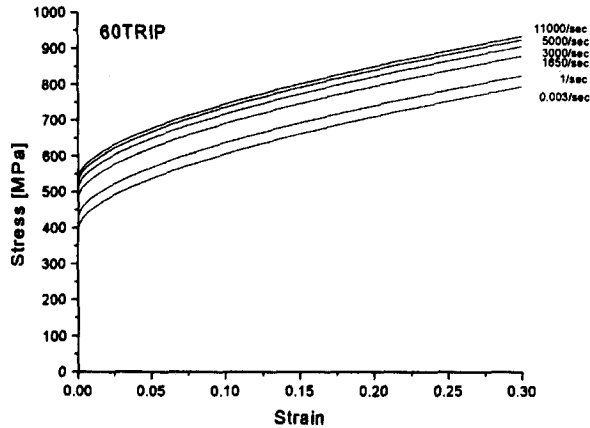where T is the temperature of the specimen, and T_{melt} is

Figure 2. Stress strain curves of 60TRIP from experiments.

the melting temperature of the specimen. The first term in the equation is the strain hardening term, the second is the strain-rate hardening term, and the third is the thermal softening term. The constant in the second term is determined by the initial yield stresses with respect to the strain rates as plotted in Figure 3 with a semi-log scale.

In the Johnson Cook relation, the stresses at the strain rate of 1/sec and a higher level are linearly interpolated as shown in the figure with a dashed line. Experimental results show that the linear interpolation is not adequate to sheet metals. For better description of the materials behavior, the experimental data is interpolated using a quadratic curve as shown in Equation (14).

$$\bar{\sigma} = (A + B\bar{\varepsilon}^n)[1 + C_1 \ln \dot{\bar{\varepsilon}} + C_2 (\ln \dot{\bar{\varepsilon}})^2](1 - T^{*m}) \qquad (14)$$
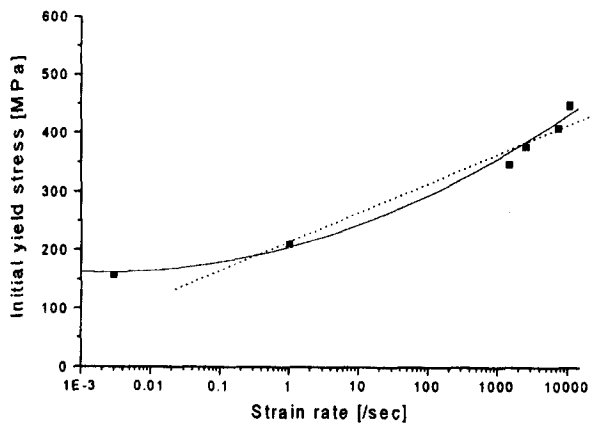

Figure 3. Initial yield stress of 60TRIP with respect to the strain rate.

Table 1. Constants in the original and modified Johnson–Cook constitutive relation for sheet metals.

	A (MPa)	B (MPa)	n	C	C_1	C_2	m
SPCEN	208	350	0.48	0.140	0.080	0.007	0.31
60TRIP	432	800	0.59	0.075	0.030	0.012	0.55
60C	463	800	0.63	0.036	0.037	0.004	0.63

The quadratic interpolation of the strain-rate hardening effect was proved to reduce the deviation from the experimental data as shown in Figure 3 with a solid line. The quadratic interpolation is especially suitable to the normal auto-body sheet metal such as SPCC and SPCEN (Kang et al., 1999). The constitutive relation presented is obtained from the adiabatic condition, which would well describe the dynamic behavior of sheet metals at the high strain rate. There will be, however, discrepancy between the curves and the metal behavior at the moderate or low strain rate where the condition is between the adiabatic and isothermal ones. The material constants are shown in Table 1 for a linear or a quadratic interpolation of the strain-rate hardening term.

4. CRASH ANALYSIS OF AUTO-BODY STRUCTURES

The crashworthiness of vehicles with finite element methods depends on the geometry modeling and the material properties. Although there is a large amount of information that is not clearly defined as the input for finite element codes, the finite element method is the most widely used numerical tool for the crash-worthiness of auto-body structures. The automobile is composed of various materials such as rubbers, composites and metals. Since the behavior of the materials is quite nonlinear, experimental techniques and the mathematical definition of the material behavior are quite complicated. The behavior of materials has to be verified to guarantee simulation results. One of the important parameters that have to be verified is the strain-rate effect of materials. In this paper, the rate-dependent constitutive relation of sheet metals is determined and used in numerical simula-tions. The crashworthiness of auto-body structures with the quasi-static constitutive relation is compared to the one with rate-dependent constitutive model such as the Johnson Cook model. The vehicle model that is used in simulation is Ford Taurus whose finite element

model is adopted from NHTSA web site. Auto-body structures such as frontal frames and a hood are separated from the full Taurus model and used in numerical simulation. The vehicle crashworthiness is greatly affected by side members and s-rails that are the energy absorbing structures with the axial and bending collapse mode. These structures are designed to effectively dissipate the vehicle kinetic energy in terms of plastic deformation in favor of the safety of passengers. Simulation results provide the deformed shape and the deformation energy in order to predict and evaluate the crashworthiness of a car. During the crash incident, the structure is subjected to the high impact load into large deformation and rotation with contact among various components. The deformation initially involves stress wave transition. Once the stress exceeds the yield strength of the material or its critical buckling load, localized deformation occurs during a few wave transits in the structure. This is followed by the inertial effect that dominates the subsequent transient response.

A finite element model of the frontal frame structure of Taurus is shown in Figure 4 with its boundary and loading conditions. Simulation is conducted by using the Johnson Cook model and the quasi-static model for SPCEN in order to compare the strain-rate effect of the two. SPCEN is one of the sheet steels that have been commonly used in auto body structures. The thickness of sheet metal is considered to be 1.2mm for all components. The initial impact velocity of a rigid wall is 48km/h, and the mass of the wall is 500kg. Deformed shapes from the finite element analysis are shown in Figure 5. The time–deformation energy and the time–displacement curves are shown in Figure 6. It is noted that the deformation is concentrated on frontal frames and s-rails while the passenger room remains undeformed. Although the amount of deformation that is calculated with the quasi-static model is larger than
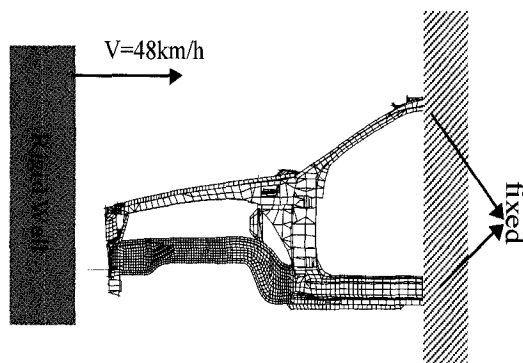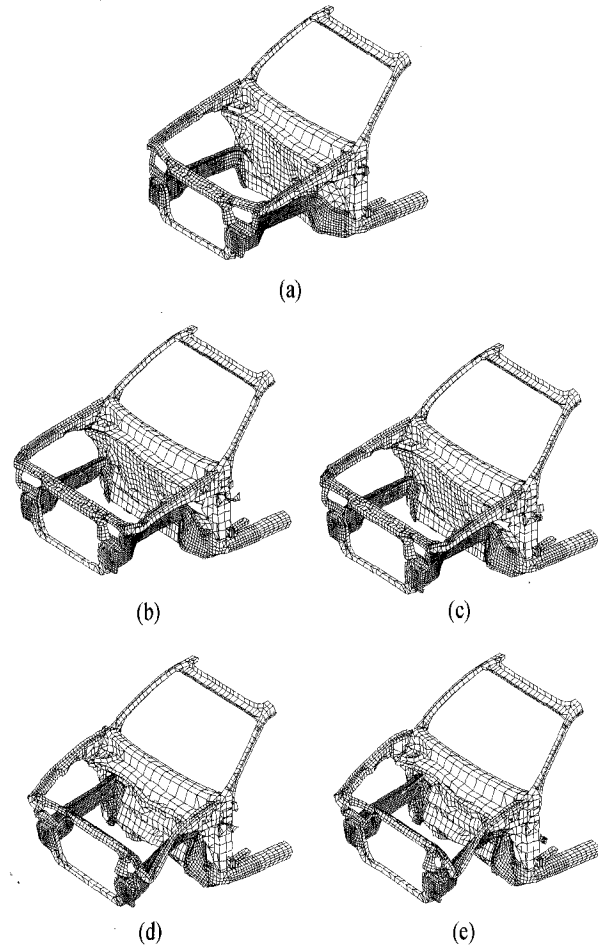

(a)

(b)   (c)

(d)   (e)

Figure 5. Initial and deformed shapes of frontal frames of an automobile: (a) initial shape; (b) displacement of 100 mm with the J C model; (c) displacement of 100 mm with the quasi-static model; (d) displacement of 300 mm with the J C model; (e) displacement of 300mm with the quasi-static model.

Figure 4. Finite element model of the frontal frame of an automobile.

that with the dynamic model, the deformation modes of the two are the same due to the controlled weak parts in the structures.

The hood of an automobile is composed of an inner and an outer panel, which contact each other as deformation progresses. The initial shape and the boundary condition of the hood are shown in Figure 7. In order to effectively absorb the impact energy, structural weak parts are formed in the inner panel as shown in Figure 7. The sheet metals used in crash analysis of the hood are SPCEN and 60TRIP. 60TRIP is one of the high strength steels made from the transformation-induced-plasticity process for the purpose of weight reduction. The initial impact velocity of a rigid wall is 48km/h, and the mass of the
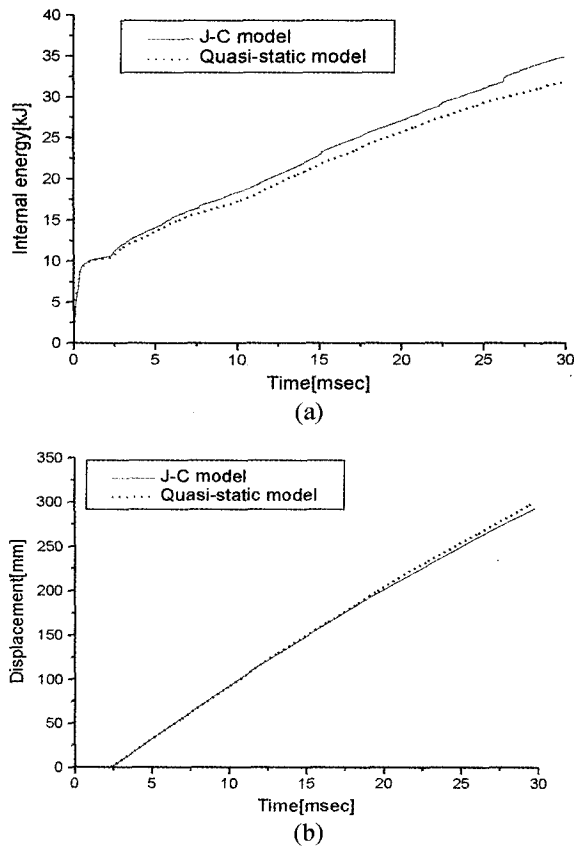
Figure 6. Deformation energy and displacement with respect to the time for the strain-rate effect: (a) Time–deformation energy curve; (b) Time–displacement curve.
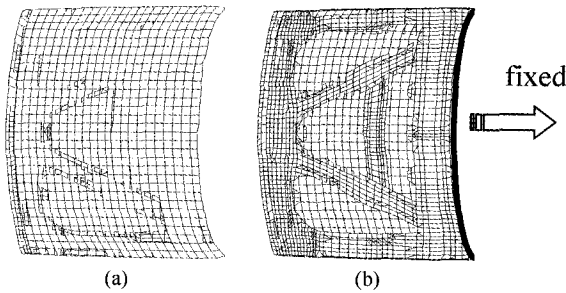

Figure 7. Finite element mesh of a hood and the boundary condition: (a) outer view; (b) inner view.

Deformed shapes are presented in Figure 8 with respect to the constitutive model. Deformed shapes wall is 500kg. Although the impact responses depend on the boundary conditions and the type of a barrier, the simplified fixed boundary condition is used to evaluate the rate dependency of a material.
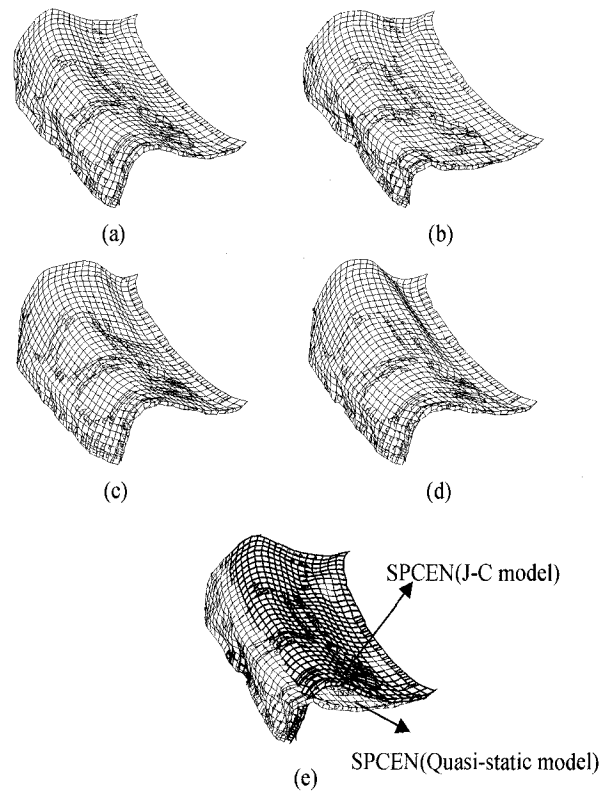

Figure 8. Deformed shapes of a hood: (a) displacement of 120mm with SPCEN(J–C Model); (b) displacement of 20mm with SPCEN(quasi-static model); (c) displacement of 350 mm with 60TRIP(J–C model); (d) displacement of 350mm with 60TRIP(quasi-static model); (e) comparison between (a) and (b).

become quite different depending on the material model as shown in Figure 8 (e). The displacement at
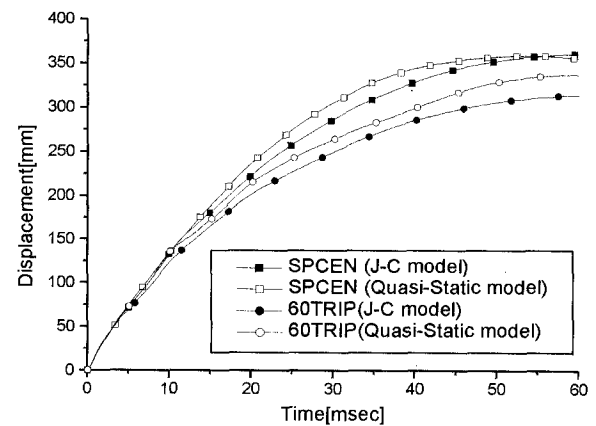

Figure 9. Displacement curves with respect to the time for the strain-rate effect.

the front end of the hood is shown in Figure 9.

When the strain-rate hardening effect is considered with the dynamic constitutive relation, the final displacement becomes relatively small compared to the one with the quasi-static model. It is noted from the result that the rate-dependent crash analysis must be carried out with its dynamic material properties identified for the purpose of the light-weight design of an auto-body with high strength steel.

5. CONCLUSION

The crash analysis of frontal frames and a hood of an automobile has been carried out with the rate-dependent constitutive model. The dynamic constitutive model of sheet metals is determined from the result of the static and dynamic material test with Instrons and a tension split Hopkinson bar apparatus respectively. The model takes the strain-rate hardening effect into account for numerical simulation of crash-worthiness of an auto-body. The simulation result with the dynamic model is compared to the one with the static constitutive model. Simulation results show that there is remarkable difference in the impact energy absorption between the static model and the dynamic one. The simulation also explores the difference in the deformed shape and the energy absorption ratio between the conventional steel sheet and the high strength steel sheet.

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