A Full Vehicle Modelling for Practical Handling Simulations

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ABSTRACT

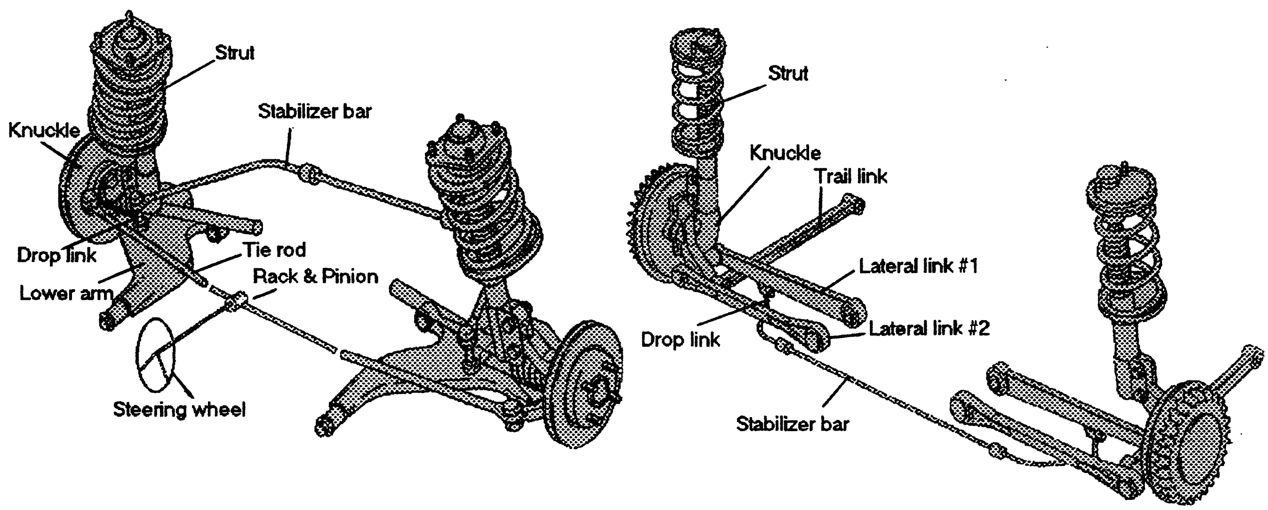
This paper presents an investigation of modelling effects on CPU time taken by ADAMS for the practical dynamic simulations of a vehicle. A vehicle contains numerous bodies, joints, bushes, springs and dampers. In general, these elements have highly non-linear characteristics. The dynamic simulation of a full vehicle model with these elements requires extensive CPU time. Sometimes, this model also yields significant numerical errors. Modelling methods replacing a steering wheel, tie rods and drop links are here developed. Then, the number of the equations of motion and the degrees of freedom reproduced by each method are studied. Modelling effects on CPU time depending on different cornering maneuvers are also examined.

1. INTRODUCTION

Since a vehicle consists of numerous elements such as bodies, joints, bushes, etc, its full modelling with all of those elements produces a large number of the equations of motion as well as the degrees of freedom. Especially, the bodies of small inertia properties connected by stiff spring elements such as bushes cause high frequency vibrations during computer simulations, which requires more CPU time and even produces integration errors. For practical handling simulations, the local high frequency motions of small bodies may not be important, because only the overall motion of a vehicle is generally concerned. In this paper, the effects of modelling methods on CPU time as well as the numbers of the equations of motion and the degrees of freedom are investigated. For this purpose, a steering wheel, tie rods and drop links are replaced by their simple artificial components which almost produce the same results to their original dynamic motions.

2. CONVENTIONAL MODELLING

A schematic representation of the front and rear suspension of a passenger car are shown as an example in Figure 1. The front suspension is composed of two MacPherson struts, and the rear suspension is furnished with two sets of tri-link types, four lateral links and two trail links. The conventional modelling of the car has 126 and 270 for the degrees of freedom and the number of the equations of motion, respectively.



(a) front suspension (b) rear suspension

Figure 1. A schematic representation of front and rear suspensions

3. PROPOSED MODELLING

The conventional modelling was modified in 3 different ways by eliminating a steering wheel and replacing tie rods and drop links by spherical-spherical joints. This joint is used to maintain constant length between two bodies by employing only one constraint [1]. Their effects on the degrees of freedom and the equations of motion depending on different modellings are compared in Table 1.

Table 1. Comparison of the number of equations of motion and the degrees of freedom

Contents \ Elements	Steering wheel --> removed	Tie rod --> sphr-sphr[1]	Drop link --> sphr-sphr
No. of PART[2] removed (①)	1	2	4
No. of Joint removed (②=No. of CONSTR[3])	revolute:1,coupler:1 (1×5+1×1=6)	universal:2,spherical:2 (2×4+2×3=14)	-
No. of Joint added (③=No. of CONSTR)	-	sphr-sphr:2 (2×1=2)	sphr-sphr:4 (4×1=4)
No.of CONSTR reduced (ⓐ=②-③)	6	12	-4
No.of COORD[4] reduced (ⓑ=①×6)	6	12	24
No.of DOF[5] reduced (ⓑ-ⓐ)	0	0	28
No.of EQN[6] reduced (ⓑ+ⓐ)	**12**	**24**	**20**

Note 1. sphr-sphr : Spherical-spherical joint
2. PART : Rigid body in ADAMS
3. CONSTR : CONSTRaint equation
4. COORD : generalized COORDinate (6 for a part)
5. DOF : Degree Of Freedom
6. EQN : EQuatioN of motion

3.1 Removal of a steering wheel

A schematic representation of a steering mechanism with kinematic joints is shown in Figure 2. A steering wheel is mainly used to change the direction of a vehicle during handling simulations. If its motion itself is not concerned, the steering wheel as well as a revolute joint and a coupler can be removed. Instead, translational steering input can be applied directly to rack/pinion. By doing this reduction, the 12 equations of motion can be totally reduced.

3.2 Replacement of tie rods

A tie rod, shown in Figure 2, conveys the motion between rack and knuckle. Since the tie rod has its small mass and maintains constant length, the tie rod with a universal joint and a spherical joint can be replaced with a spherical-spherical joint. Because there are 2 tie rods, the 24 equations of motion can be totally reduced.

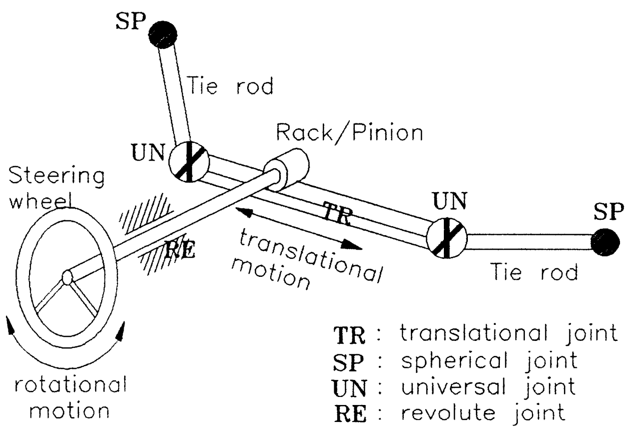


Figure 2. A systematic representation of a steering mechanism

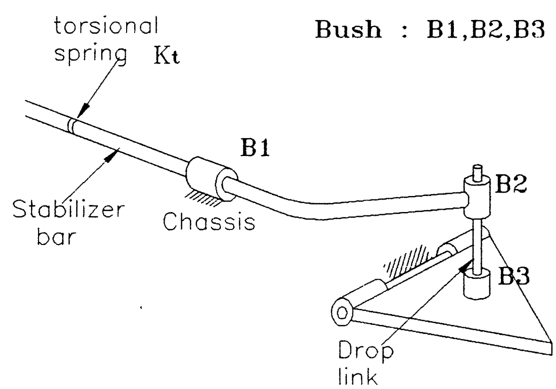


Figure 3. A systematic representation of a drop link

3.3 Replacement of drop links

In a conventional model, as shown in Figure 3, a drop link is located between an end of a stabilizer bar and a lower arm in the front suspension, and is connected by bushes, **B2** and **B3**. Because of its small mass, 0.13kg or 0.05kg, for the front or the rear suspension respectively, and stiff bushes at both ends of the drop link, it generally generates local high frequency vibrations which may cause to increase CPU time and even to produce numerical errors during dynamic simulations. As a means of avoiding these problems and reducing the numbers of the equations of motion, the drop link was replaced by a spherical-spherical joint. At the same time, the stiffnesses of bushes, **B2** and **B3**, were distributed to the torsional component of bush **B1** and torsional spring **Kt** to maintain the same vertical force compared to that of the conventional model with the same vertical movement of a tire.

For this purpose, at first, the same vertical motion was given at both right and left tires during quasi-static simulation. And the torsional component of bush **B1** was adjusted numerically to have the same vertical tire force compared to that of the conventional model. The vertical motions of the right and the left tires were then reversed, and torsional spring **Kt** was adjusted in the same manner. But the damping effects of bushes, **B2** and **B3,** were not considered here. In order to examine the effect of this modified model on suspension parameters, the simulation results of the suspension parameters between the modified and conventional models were compared in Figure 4. Since the 3 components of forces and moments of the drop link was replaced by an axial force along a spherical-spherical joint, there was a little difference in the longitudinal movement of a wheel center, but there was no any other significant difference between the simulation results of the two models.

4. DYNAMIC SIMULATIONS

By assembling the 3 modelling methods, 7 different sets of model types were made. Both the simulation results and CPU time were compared in order to find the effects of different modelling methods.

4.1 Definition of model types

Type_1 through type_7 were made by applying different sets of 3 modelling methods to the conventional model. Each of these model types had the different numbers of the equations of motion, and it was numbered in descending order of the equation numbers as shown in Table 2.

Table 2. Comparison of 7 model types and the conventional model

element / model type	Steering wheel --> removed	Tie rod --> sphr+sphr	Drop link --> sphr+sphr	No.of EQN	DOF
Conventional	-[1]	-	-	**270**	126
Type_1	o[2]	-	-	**258**	126
Type_2	-	-	o	**250**	98
Type_3	-	o	-	**246**	126
Type_4	o	-	o	**238**	98
Type_5	o	o	-	**234**	126
Type_6	-	o	o	**226**	98
Type_7	o	o	o	**214**	98

Note 1. - : not modified
2. o : modified

4.2 Comparison of CPU time

The handling simulations of the 7 types and the conventional model were carried out at different cornering maneuvers, i.e. straight-ahead, lane change and slalom. In the case of slalom simulation, the yaw rate, lateral acceleration and roll angle of the 7 types were compared with those of the conventional model (refer to Figure 5). The maximum differences between the results of the 7 types and the conventional model were within 1%. But CPU time was quite differently required depending on model types or severity conditions as shown in Figure 6.

At straight-ahead simulation, CPU time tends to be proportional to the numbers of the equations of motion. At lane change simulation, CPU time has also the same tendency to the straight-ahead. At severer condition such as slalom simulation, the tendency of CPU time is quite different from that of the straight-ahead. This tendency is characterized into 2 distinct groups. Group A includes conventional, type_1, type_3 and type_5. Group B is composed of type_2, type_4, type_6 and type_7. The average CPU time of Group B was 63% of that of Group A, and the main difference between the 2 groups was that Group A had drop links but Group B did not. It could be concluded that CPU time was significantly influenced by the existence of drop links but not so much by the numbers of the equations of motions. The effect of a steering wheel or tie rods on CPU time seems to be not so much as that of drop links.

5. CONCLUSION

The modelling methods used in this paper are as follows:

(a) A steering wheel was removed and instead translational steering input was applied directly to rack/pinion.

(b) A tie rod was replaced by a spherical-spherical joint.

(c) A drop link was replaced by a spherical-spherical joint and the stiffnesses of removed bushes were distributed in a way to produce the same vertical tire force.

The different model types composed of these 3 ways in different combinations were examined for their effects on the simulation results and CPU time. The results of practical handling simulations of these models showed that their differences were within 1% maximum. But the tendencies for CPU time were quite different depending on model types and driving conditions.

CPU time was proportional to the number of the equations of motions at less severe conditions. At severe conditions, however, CPU time was increased drastically when a model had parts of small mass connected by stiff bushes, like a drop link, regardless of the numbers of the equations of motion. It can be concluded that the proper treatment of the parts of small mass connected by stiff spring elements is very important in a full vehicle modelling to reduce CPU time without affecting the results of practical handling simulations.

REFERENCE

1. P.E. Nikravesh, Computer-Aided Analysis of Mechanical Systems, Prentice Hall, 1988.

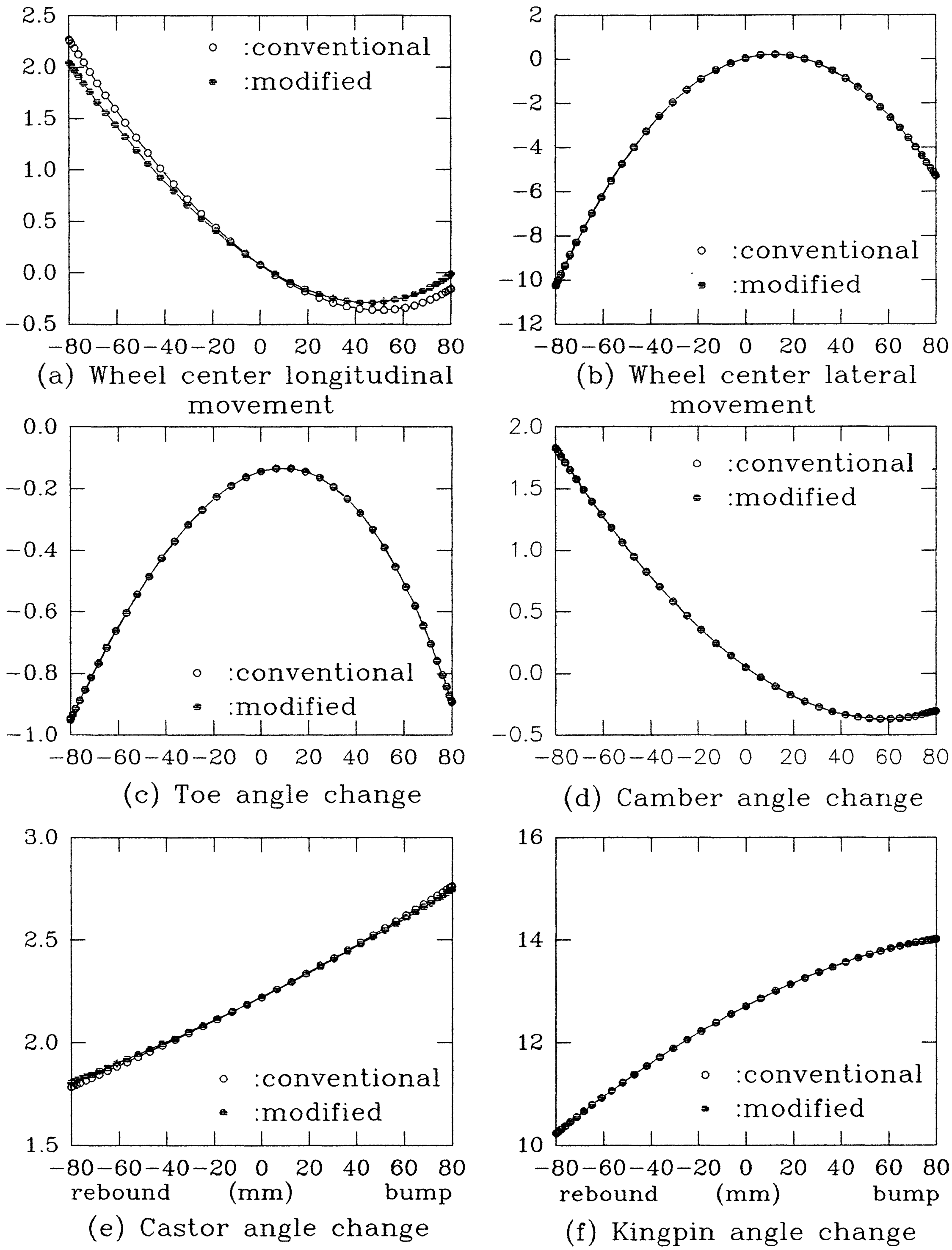


Figure 4. Comparisons of suspension parameters between modified and conventional models with respect to the vertical movement of a wheel center at the quasi-static simulation of a front suspension.

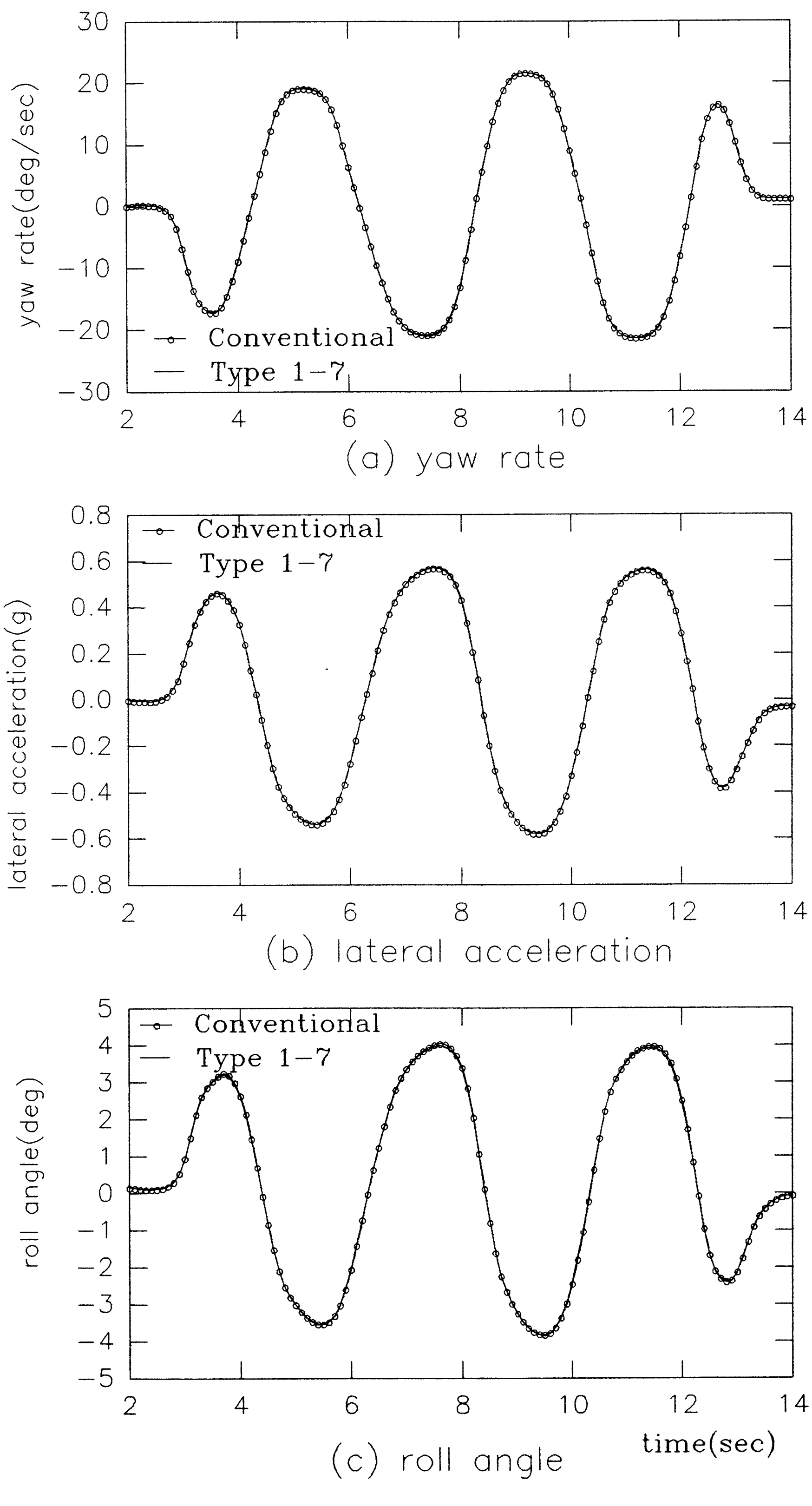


Fig 5. Comparison of slalom simulation results depending on model types

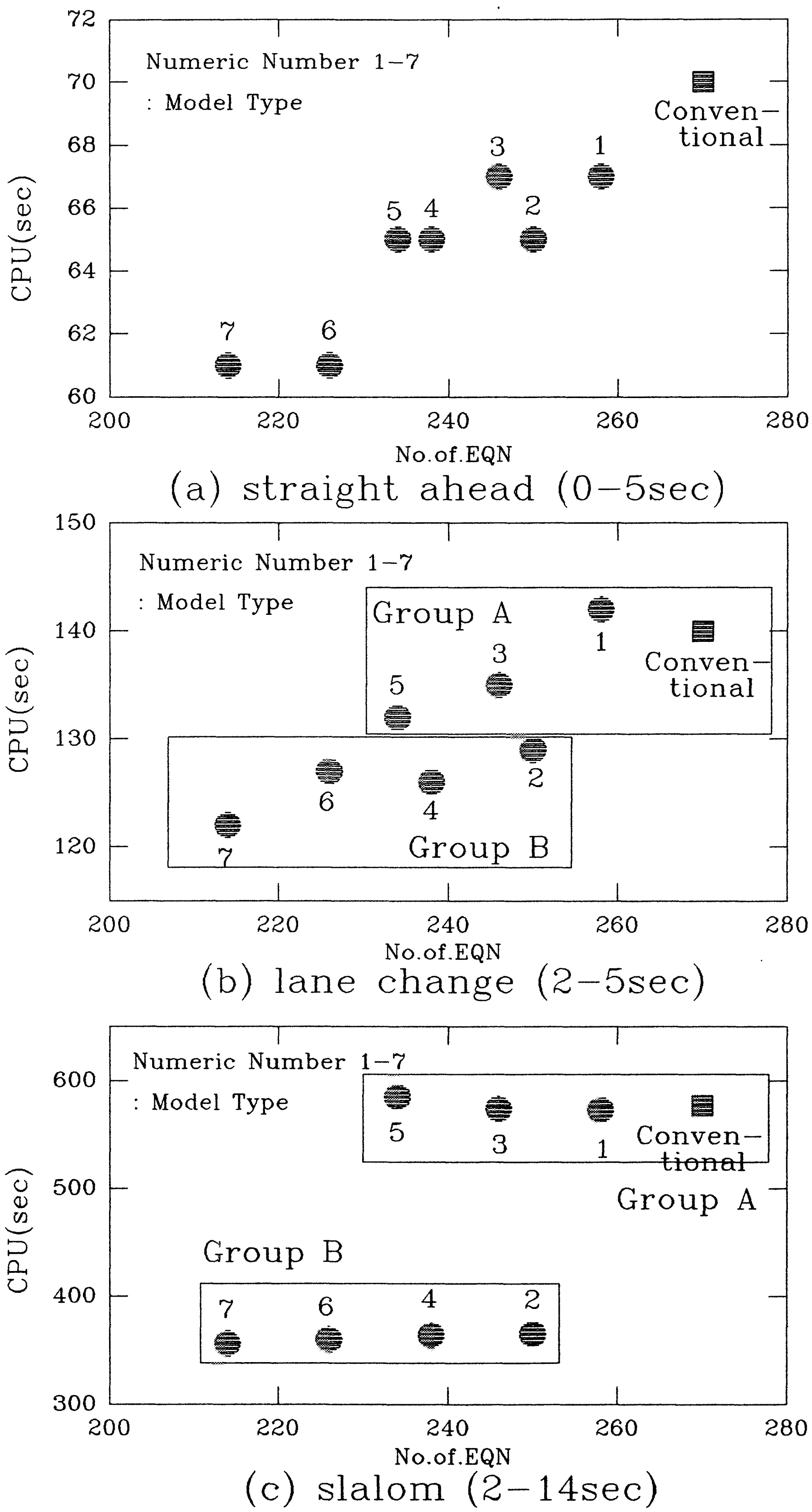


Figure 6. Comparison of CPU times depending on model types and driving conditions