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COMPUTATIONS FOR TRUCK SLIDING
WITH TRUCK 3.1 CODE

ROMESH C. BATRA

AUGUST 1989

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The TRUCK 3.1 code developed by Lee, Hobbs and Atkinson has been thoroughly studied. Equations of motion used in the TRUCK code were found to be correct and accurately coded. The code gave good results for the problems studied.			
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I. INTRODUCTION

Lee, Hobbs and Atkinson [1] have developed a computer program TRUCK 3.1 to find the dynamic response of a vehicle subjected to different types of loads including such intensive loads as blast waves. The code is intended to yield gross motions of the vehicle body, and of tires, axles, shelters, and racks relative to the vehicle body. Large motions, including the overturning of the vehicle, are permitted. However, each individual element like an axle, shelter or rack is treated as a rigid body.

The purposes of this study are: (i) to check the validity of the governing equations of motion used in the TRUCK code that model the rigid body motions of the vehicle assembly and its components, (ii) to find out if the equations have been correctly coded and (iii) to investigate whether or not the TRUCK code models effectively the frictional force between the tires and the ground.

The third objective is achieved by analyzing a few sample problems and comparing their computed solution with either analytical solutions or those obtained by using the computer code ADINA.

II. VEHICLE MODEL

The vehicle and its primary components whose response can be studied by using the TRUCK code, are shown in Fig. 1. Each component is characterized by its mass, center of gravity position, inertias and products of inertias. Not explicitly shown in Fig. 1 are springs connecting the vehicle body to the axles. One might think that the mechanical properties of these springs will affect the motion of axles, racks etc. relative to the vehicle body but will not affect the overall motions of the assembly. But in the TRUCK code, the primary coordinate system is a body axis system attached to the vehicle body with its origin at the center of gravity of the entire system. Therefore, the mechanical properties of springs connecting the vehicle body to the axles influence the overall motions of the vehicle. However, the motion of the overall center of gravity of a frictionless system is not affected by the mechanical properties of these springs. The right handed rotations ϕ , θ , and ψ about the x , y , and z -axes, shown in Fig. 1, are referred to as pitch, yaw, and roll, respectively. Translational motions in the x , y , and z directions will be referred to as sideslip, heave, and fore-and-aft translation, respectively. If desired, the motions of the axles, shelters, and racks relative to the vehicle body can also be studied.

Each axle is assigned two degrees of freedom, namely its roll and heave relative to the vehicle body. A shelter is considered to have four degrees of freedom - its roll, heave, pitch and fore-and-aft translation relative to the vehicle body. A rack has six degrees of freedom - its roll, sideslip, heave, pitch, yaw and fore-and-aft translation relative to the vehicle body.

The TRUCK code has the option to consider the entire system as a rigid body with only three degrees of freedom involving vehicle roll, heave and sideslip only. However, this

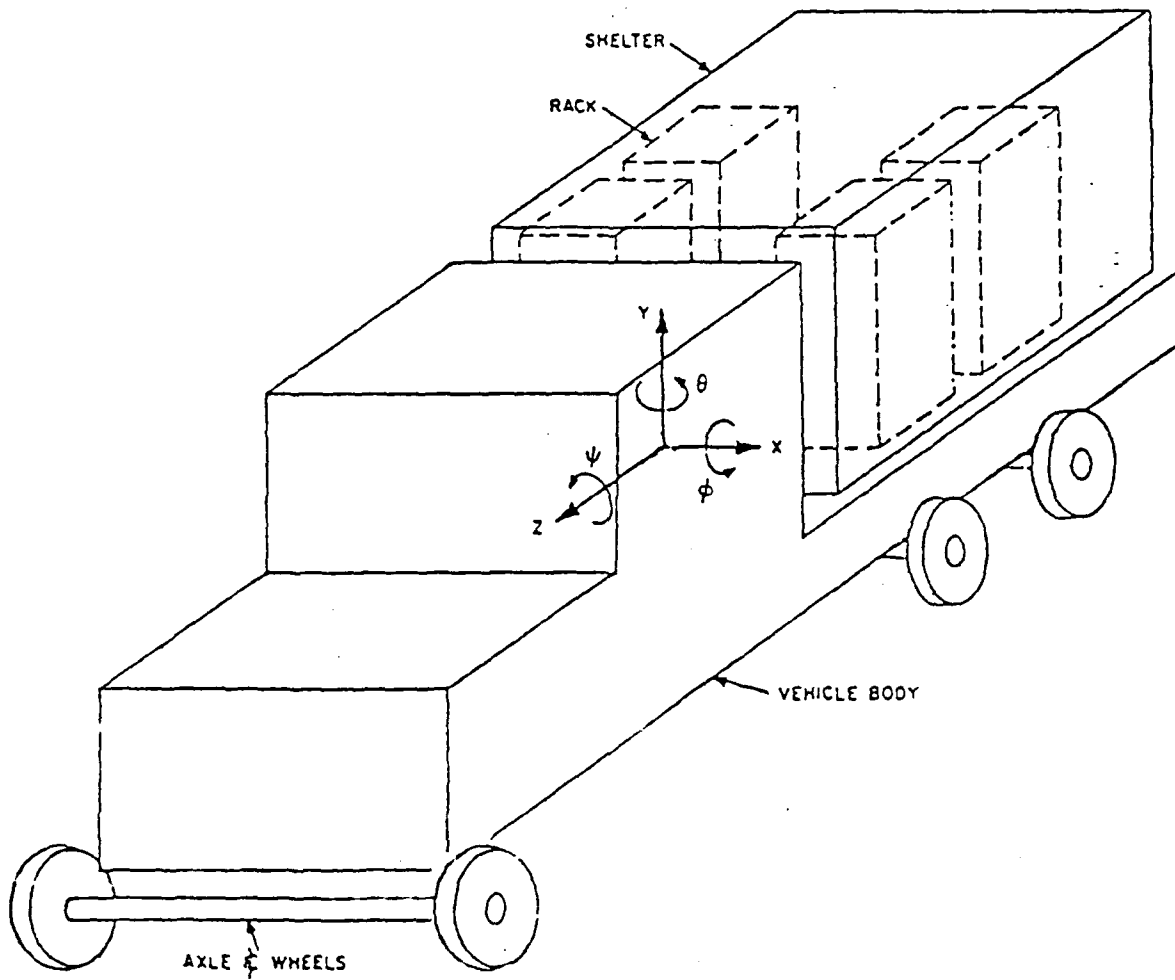


Figure 1. A Typical Vehicle Model for Study by TRUCK 3.1 Code.

problem still requires data on the tires necessary to define the tire/ground interaction forces.

III. FRICTIONAL MODELS IN TRUCK AND ADINA CODES

In the TRUCK code, the tire-ground interaction is presumed to be based upon Coulomb's law of friction. The normal force on the tire depends upon the normal deflection and normal velocity of the tire. For this purpose the tire is modeled as a non-linear spring and a non-linear dashpot connected in series. The magnitude of the tangential force depends upon whether the tire is sliding or not. When there is no sliding, the tangential force depends upon the tire tangential deflection and velocity and the spring and damping characteristics of the tire. The tire starts to slide when the tangential spring force equals the tire normal force times the coefficient of friction. The tangential spring force during sliding equals the tire normal force multiplied by the coefficient of friction and is in a direction opposite to the tangential deflection.

We note that in the TRUCK code the static and kinetic coefficients of friction are given the same value.

The tangential velocities and therefore tangential displacements of front tires are treated differently from those of other tires. This is because of the assumption that the front tires are free to roll in their planes. Therefore, for a front tire, the component of the tangential velocity which is normal to the intersection of the wheel plane and the ground plane is used rather than the total tangential velocity.

The possibility of a tire leaving the ground and coming in contact again is allowed. For a tire that is off the ground, tire-ground interaction forces are set equal to zero and its tangential deflection is also set equal to zero.

In the AE84 version of the ADINA [2] code one of the two contacting bodies must be deformable. For two-dimensional problems, the code sets the frictional force between the two bodies equal to that required to equilibrate the entire system provided that it does not exceed the limiting value of the coefficient of friction times the normal force between them. In the latter case, the frictional force is set equal to the limiting value and sliding is permitted between the two bodies.

In ADINA code too, there is no distinction made between the static and kinetic coefficients of friction.

Herein we will study those problems in which the vehicle undergoes planar motions.

IV. COMPUTATION AND DISCUSSION OF RESULTS

After having verified that equations governing the motion of the vehicle assembly and its components are modeled correctly in the TRUCK code we proceeded to find out if the code predicts well the response of an idealized system subjected to simple loading. The

vehicle considered henceforth is shown in Fig. 2 wherein are also listed values of various parameters. In order to minimize the axle roll relative to the vehicle body the axles were given arbitrarily high values of I_{zz} . When computing the numerical solution by using the TRUCK code, the subroutine that gives aerodynamic loading on the vehicle was bypassed.

As a first test problem, we applied a force \vec{F} (as a function of time t) to the rear of the vehicle so as to produce fore-and-aft translation only, and set the coefficient of friction μ between the tires and the ground equal to zero. The force is

$$\vec{F} = 4.45 t \hat{k} \text{ MN} = 10^6 t \hat{k} \text{ lb}$$

where \hat{k} is a unit vector in the z -direction. Thus the problem reduces to finding the motion of a rigid block of mass m (having units of kg [lb-s²/in]) sliding on a smooth plane and subjected to the external force \vec{F} . Its acceleration \vec{a} , velocity \vec{v} and displacement \vec{u} relative to the earth-fixed axes are given by

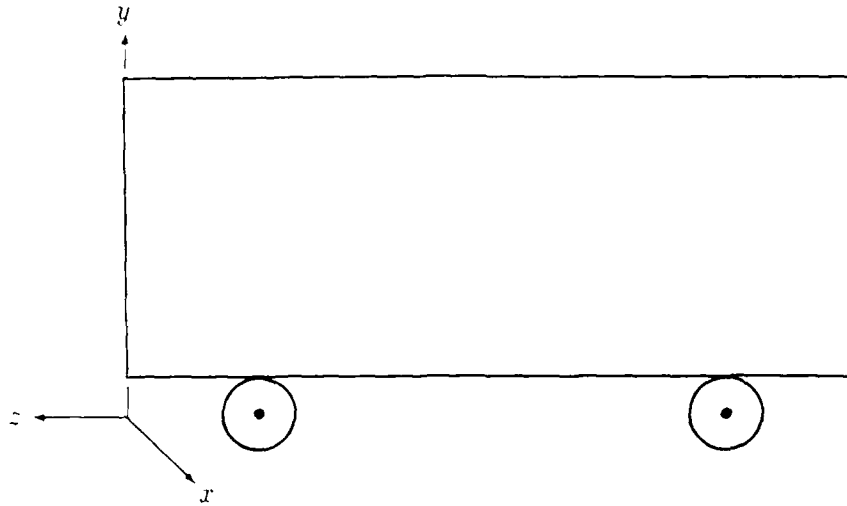
$$\begin{aligned}\vec{a} &= \frac{25.4}{m} t \hat{k} \frac{\text{km}}{\text{s}^2} = \frac{10^6}{m} t \hat{k} \frac{\text{in}}{\text{s}^2} \\ \vec{v} &= \frac{12.7}{m} t^2 \hat{k} \frac{\text{km}}{\text{s}} = \frac{10^6}{m} \frac{t^2}{2} \hat{k} \frac{\text{in}}{\text{s}} \\ \vec{u} &= \frac{4.23}{m} t^3 \hat{k} \text{ km} = \frac{10^6}{m} \frac{t^3}{6} \hat{k} \text{ in}\end{aligned}$$

The computed and analytical solutions for the fore-and-aft translation are shown in Fig. 3. The two solutions agreed with each other. Also the computed normal tire forces matched exactly with those found analytically. This exercise ensured that governing equations are integrated correctly at least for this simple loading.

Subsequently we changed the value of μ from 0 to 0.8. The maximum frictional force between the vehicle and the ground for the assumed weight of the vehicle assembly is 151.1 kN (33.970 lb). Thus, theoretically, the vehicle should not slide till the applied force exceeds 151.1 kN (33.970 lb). However, the frictional force between the tires and the ground need not equal the applied force in magnitude but it will act in the opposite direction. Fig. 4 shows the magnitude of the computed frictional force and the applied force. As dictated by Coulomb's friction law, the vehicle did not slide. The oscillations of the computed frictional force about the applied force are due to the modeling of the system as springs, dashpots, and connecting rigid masses.

In order to see how the assumed tangential tire stiffness will affect the computed solution, we repeated the above calculation with two other values of the tangential tire stiffness. The results depicted in Fig. 5 reveal that the computed frictional force depends upon the tangential tire stiffness.

We subsequently investigated the effect of the size of the time increment upon the computed solution. For time increment equal to one-tenth of its previous value, the computed frictional force for three values of k_t is shown in Fig. 6. The results shown in Figs. 5 and 6 indicate that the magnitude of the time-increment to be used in the TRUCK code



Component	Mass Mg (lb·sec ² /in)	Center of Gravity (x, y, z) m [in]	Products of Inertia Mg·m ² (lb·sec ² ·in)			Moments of Inertia Mg·m ² (lb·sec ² ·in)		
			I _{yz}	I _{xz}	I _{xy}	I _{xx}	I _{yy}	I _{zz}
Vehicle	15.8 (90)	(0, 1.02, -2.54) [(0.40, -100)]	-42 (-38×10 ⁴)	0	0	95,000 (84×10 ⁷)	56 (5×10 ⁵)	56 (5×10 ⁵)
Front Axle	1.75 (10)	(0, 0.025, -0.254) [(0, 1, -10)]	0	0	0	0.056 (500)	0.056 (500)	56 (5×10 ⁵)
Rear Axle	1.75 (10)	(0, 0.025, -4.83) [(0, 1, -190)]	0	0	0	0.056 (500)	0.056 (500)	56 (5×10 ⁵)

Notes:

- Damping constants for tires = 0.
- Tires are modeled as linear springs with spring constants of 17.5 GN/m (10⁸ lb/in) for both normal and tangential displacements.
- Time increment = 100μs.
- Damping constants for springs between axle and body = 0.
- Spring constants for springs between axle and body = 1.75 GN/m (10⁷ lb/in).
- Linear springs, used to model the tire stiffness normal to the ground, can slide on the ground.

Figure 2. Vehicle Model Studied.

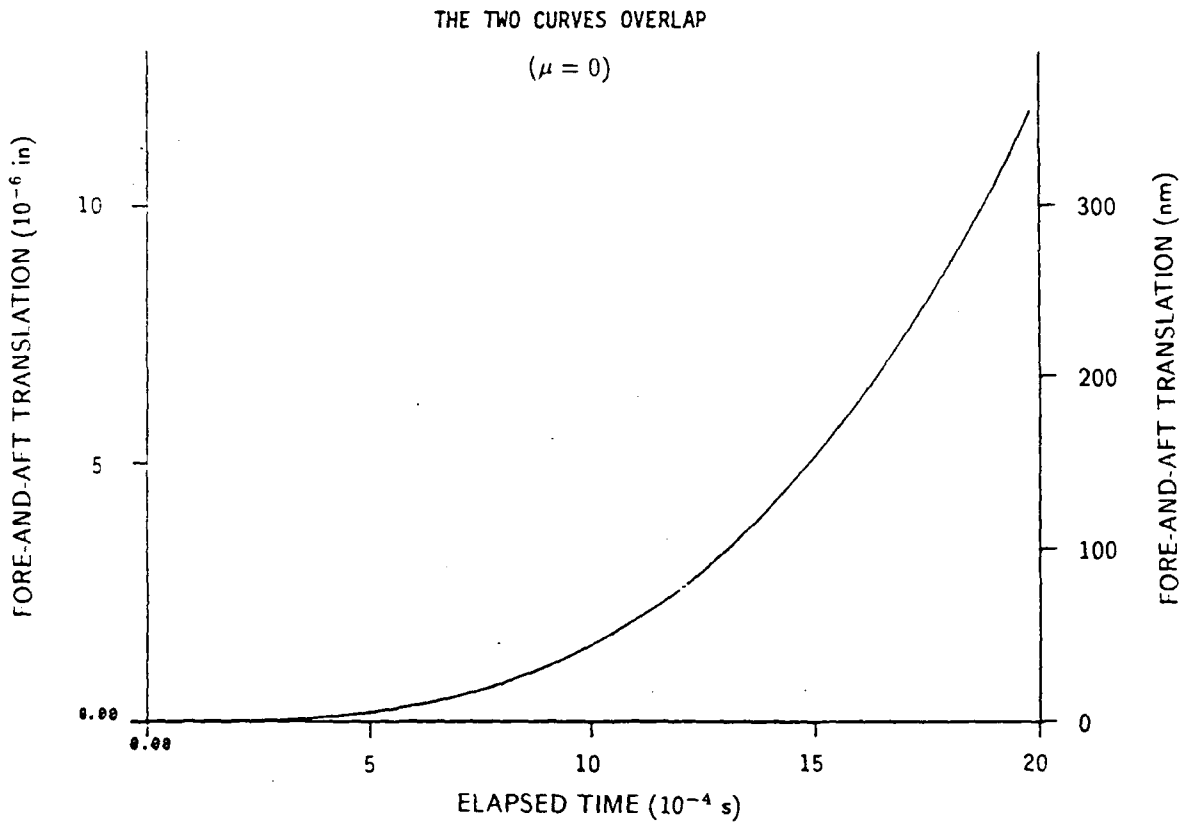


Figure 3. Comparison of the Analytical Solution with that Obtained by Using the TRUCK Code (No Friction Force between Tires and Ground).

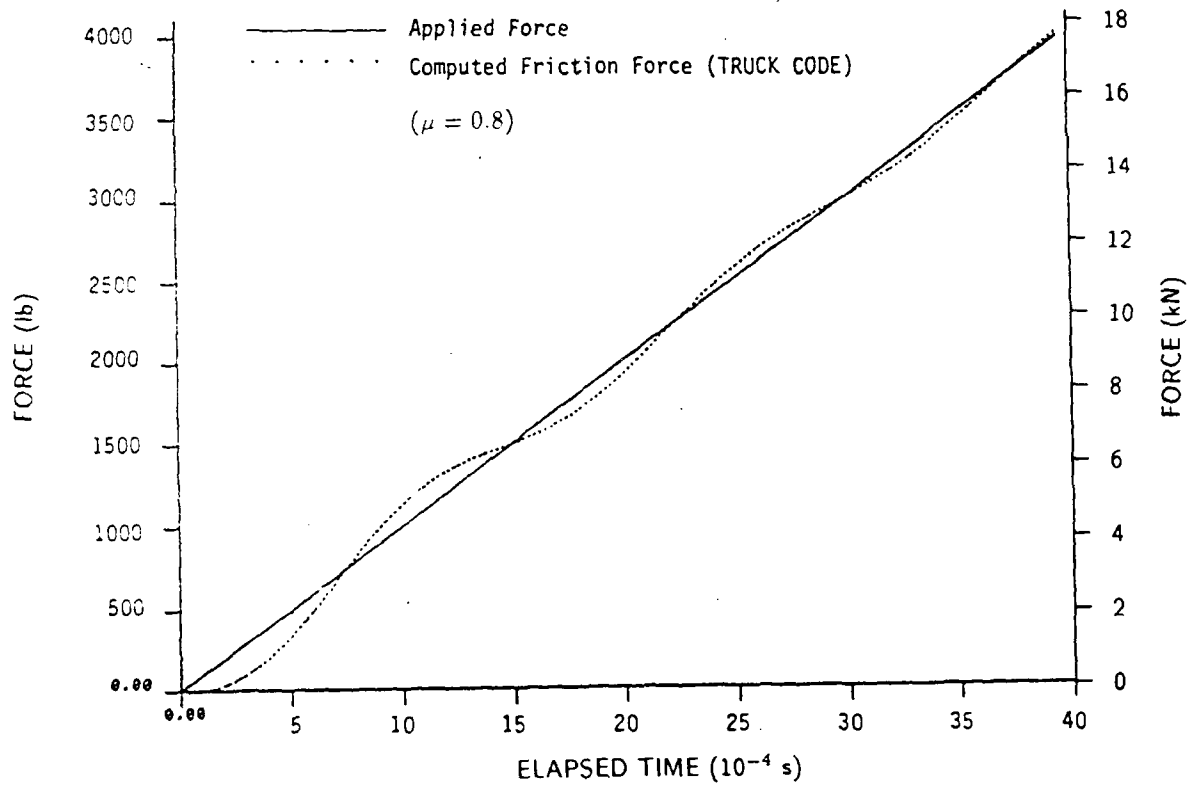


Figure 4. Comparison of the Applied Force and the Frictional Force Computed with the TRUCK Code.

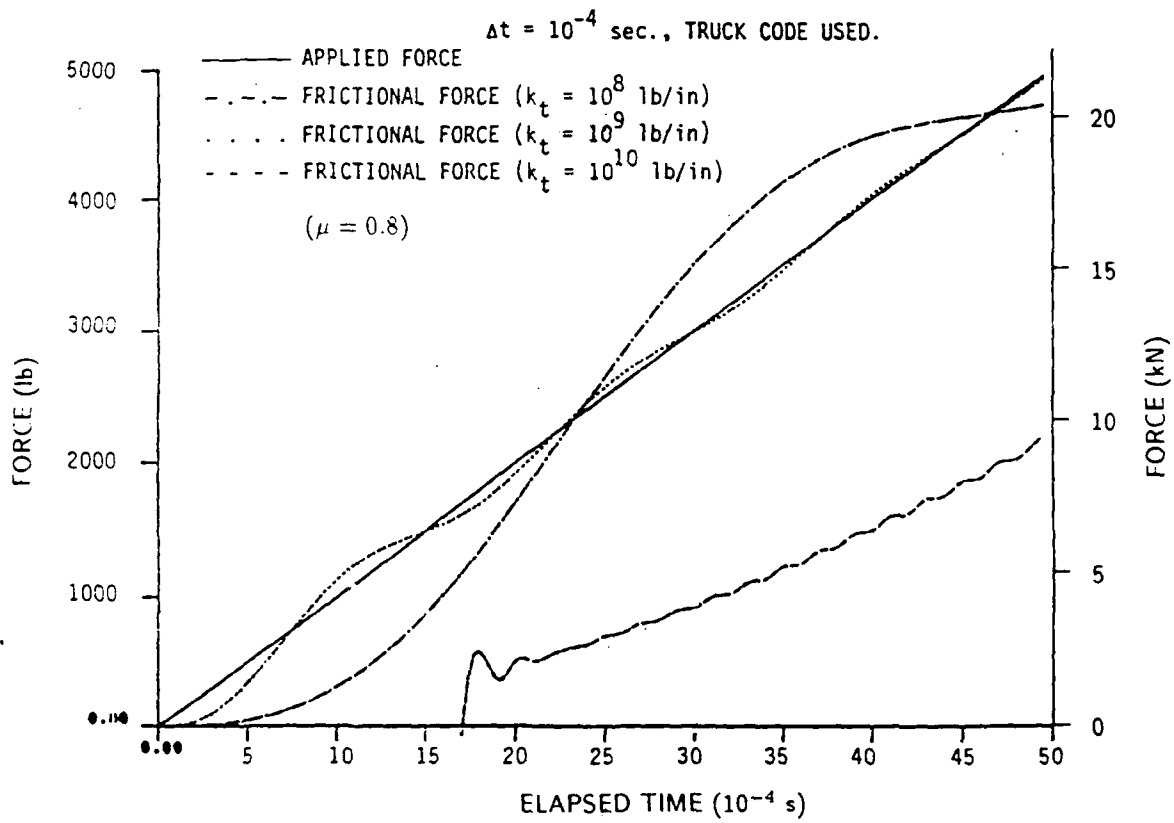


Figure 5. Effect of the Tangential Tire Stiffness upon the Frictional Force Computed with the TRUCK Code ($\Delta t = 100\mu s$).

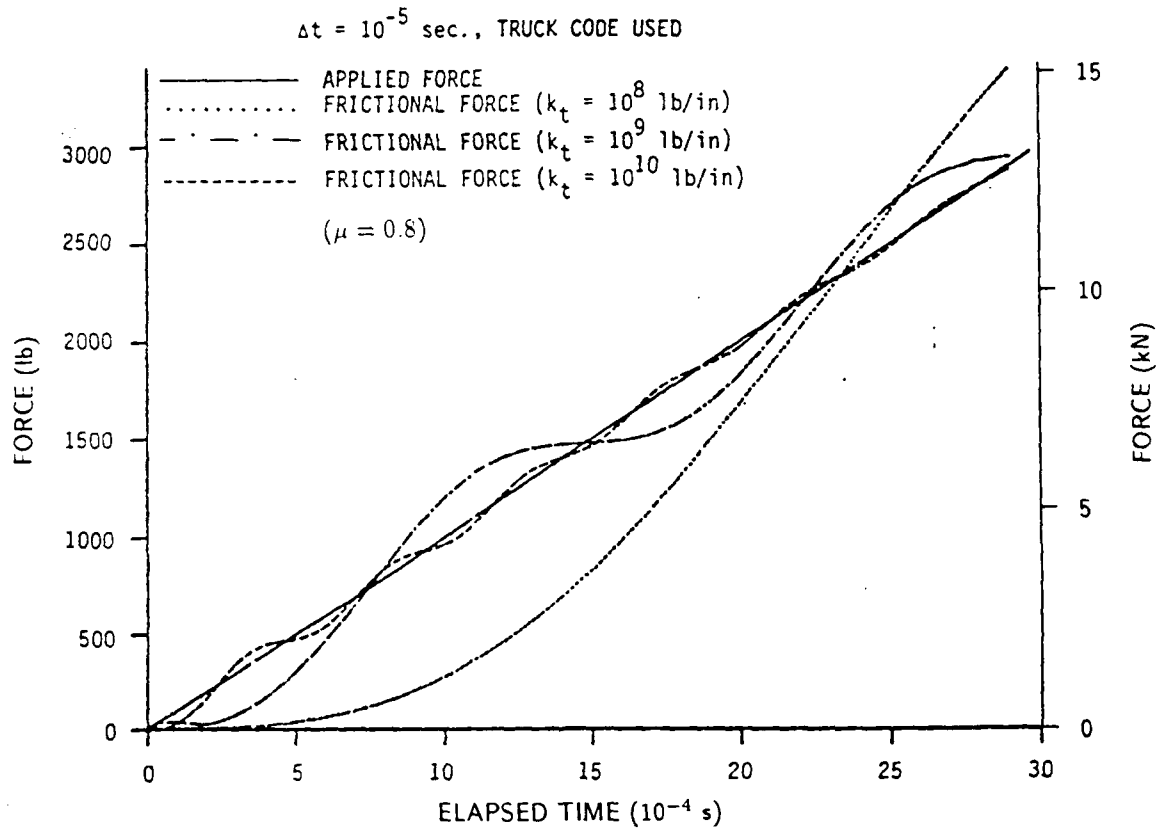


Figure 6. Effect of the Tangential Tire Stiffness upon the Frictional Force Computed with the TRUCK Code ($\Delta t = 10\mu s$).

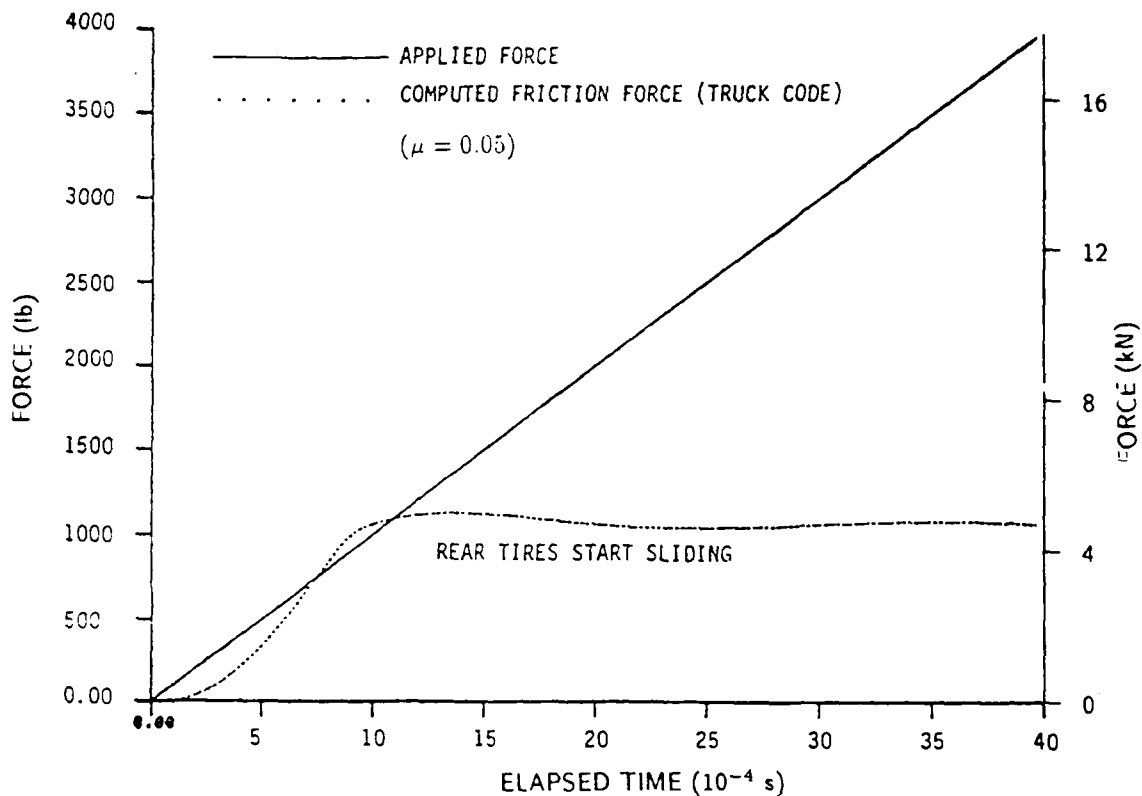


Figure 7. Comparison of Frictional Force Computed with the TRUCK Code and Applied Loads when Tires Slide.

depends strongly upon the tangential stiffness of the tire. The rather odd result for $k_t = 1.75 \text{ TN/m} = 10^{10} \text{ lb/in}$ could be due to improper choice of the time increment Δt . We have not investigated this any further. We note that our values for k_t are hypothetical and need not correspond to a real tire.

In Fig. 7 are plotted the values of the computed frictional force and the applied force when the coefficient of friction was lowered from 0.8 to 0.05. The limiting frictional force equals 9.448 kN (2,124 lb). Because of the symmetries in the assumed vehicle model, each wheel should provide 2.362 kN (531 lb) of frictional resisting force in the limiting case. However, because of the special treatment rendered to front tires, the TRUCK code gave the limiting frictional force to be 4.724 kN (1,062 lb) when the rear wheels started sliding. This is exactly half of the theoretical value and the difference between the two is due to the way frictional forces are computed for the front and the rear tires. Note that the front tires did not slide and apparently contributed nothing to the frictional force.

When solving the same problem with the ADINA code, the tires were assumed to have negligible radius. Thus, the entire bottom surface was in contact with the flat ground. The ADINA code gave values of frictional force which were in agreement with Coulomb's friction

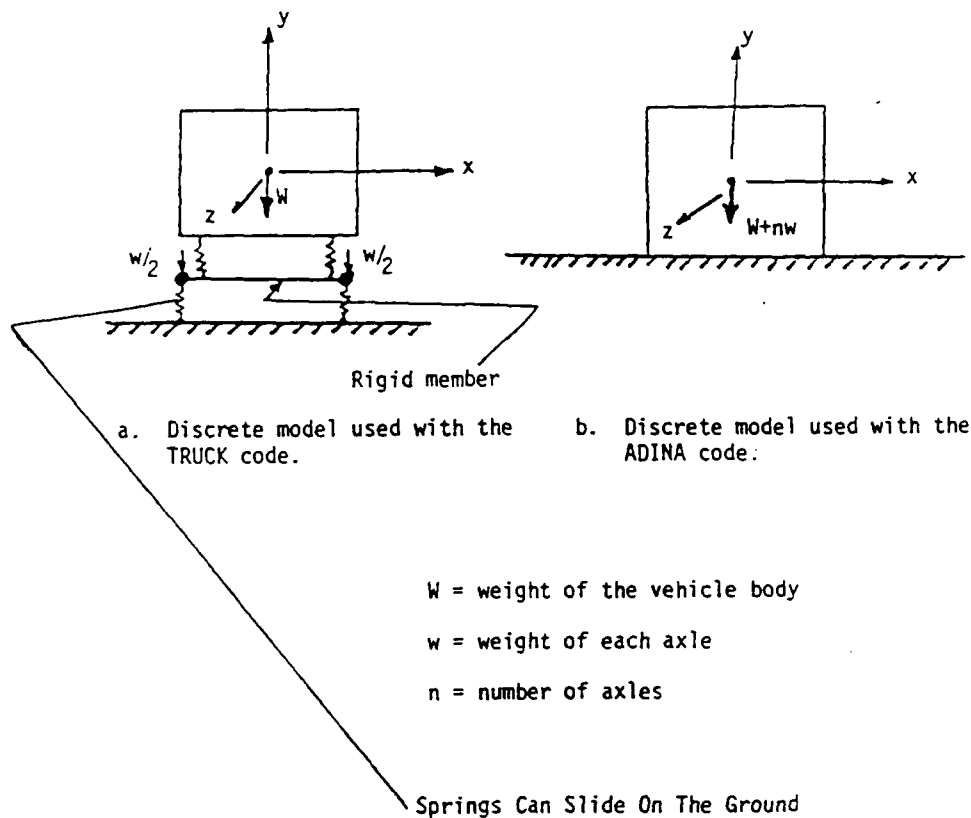


Figure 8. Discrete Models Used in the TRUCK and ADINA Codes.

law. That is, when the externally applied force was less than the limiting value of the frictional force between the vehicle and the ground, the code gave frictional force equal in magnitude and opposite in direction to the applied force. When the applied force exceeded the magnitude of the limiting frictional force, the computed frictional force equalled the limiting frictional force.

A possible explanation for the difference in the computed frictional force with the two codes is that the equivalent discrete models, shown in Fig. 8, used in the two codes are quite distinct. In the ADINA code the rigid block was modeled as a deformable body with very high value of Young's modulus. An attempt to model tires by equivalent truss elements in the ADINA code did not give satisfactory results. As already discussed modelling of tires as very strong springs in the TRUCK code necessitates the use of an extremely small time increment.

Note that in the discrete model used with the TRUCK code, there are springs connecting the axles to the body of the vehicle. In the ADINA code, the entire vehicle was modeled as a rigid body. The modelling of the vehicle by springs and dashpots with interconnecting masses contributes to the oscillations of the frictional force computed with the TRUCK code. An attempt was made to investigate the effect of the stiffness k_a of the springs between the

axles and the vehicle body upon the computed frictional force. The foregoing results were obtained by taking $k_a = 1.75 \text{ GN/m} = 10^7 \text{ lb/in}$ and $\Delta t = 100 \mu\text{s}$. When we increased the value of k_a to 1.75 TN/m (10^{10} lb/in), numerical instabilities were observed for $\Delta t = 100 \mu\text{s}$. These instabilities subsided for a while with $\Delta t = 10 \mu\text{s}$ and $\Delta t = 1 \mu\text{s}$ but appeared again at a later instant. It seems that one will need smaller time-step size to pursue this case further. We should add that during the time interval for which stable results were obtained with the TRUCK code, the computed frictional force essentially matched with the applied force and also with that computed with the ADINA code.

Even though there were no dampers included in either of the discrete models, the amplitude of the oscillations of the frictional force computed with the TRUCK code seemed to die out possibly due to the dissipation introduced by the numerical method used to integrate the equations of motion.

In the loading situation studied above with the TRUCK code, the rear tires slid whereas the front ones did not. One situation where all four tires are likely to slide simultaneously is when the force is applied to a side of the truck. We tried to simulate this by applying a force

$$\vec{F} = 4.45 t \hat{i} \text{ MN} = 10^6 t \hat{i} \text{ lb}$$

at the center of gravity of the truck assembly. Values of all other parameters are given in Fig. 2. The computed frictional forces for $\mu = 0.8$ and $\mu = 0.05$ are plotted in Fig. 9. As expected, all tires slid simultaneously in the latter case and the computed frictional force did eventually stabilize near the limiting value of 9.448 kN ($2,124 \text{ lb}$).

To see how well the TRUCK code predicts the rotational response of a vehicle, we changed the loading in the vehicle model of Fig. 2 to zero applied force but applied a yawing moment \vec{M} given by

$$\vec{M} = 1.13 t \hat{j} \text{ MN}\cdot\text{m} = 10^7 t \hat{j} \text{ lb}\cdot\text{in}$$

Results were computed for $k_t = 17.5 \text{ GN/m}$ (10^8 lb/in), $\mu = 0.8$; $k_t = 17.5 \text{ GN/m}$ (10^8 lb/in), $\mu = 0.1$; and $k_t = 175 \text{ GN/m}$ (10^9 lb/in), $\mu = 0.8$. For $\mu = 0.8$ and 0.1 , the limiting frictional moments equal $345.4 \text{ kN}\cdot\text{m}$ ($30.57 \times 10^5 \text{ lb}\cdot\text{in}$) and $43.16 \text{ kN}\cdot\text{m}$ ($3.82 \times 10^5 \text{ lb}\cdot\text{in}$), respectively. The computed values of the moment caused by frictional forces are plotted in Fig. 10. Results for $k_t = 17.5 \text{ GN/m}$ (10^8 lb/in) and $\mu = 0.8$ essentially agreed with those computed for $\mu = 0.1$. However, when k_t was changed from 17.5 GN/m (10^8 lb/in) to 175 GN/m (10^9 lb/in) with μ kept fixed at 0.8 , the computed values of the frictional moment varied widely. These are listed in Table 1.

These simple exercises indicate that with a suitable choice of the values of the tangential tire stiffness and the time step size, the TRUCK code gives good results. It will be desirable to build into the code an algorithm that optimizes the time increment and checks for the stability of the computed results.

We should add that neither the applied force nor the applied moment produced overturning of the vehicle in the simple problems studied.

For 3-dimensional problems, the AE84 version of the ADINA code models contact

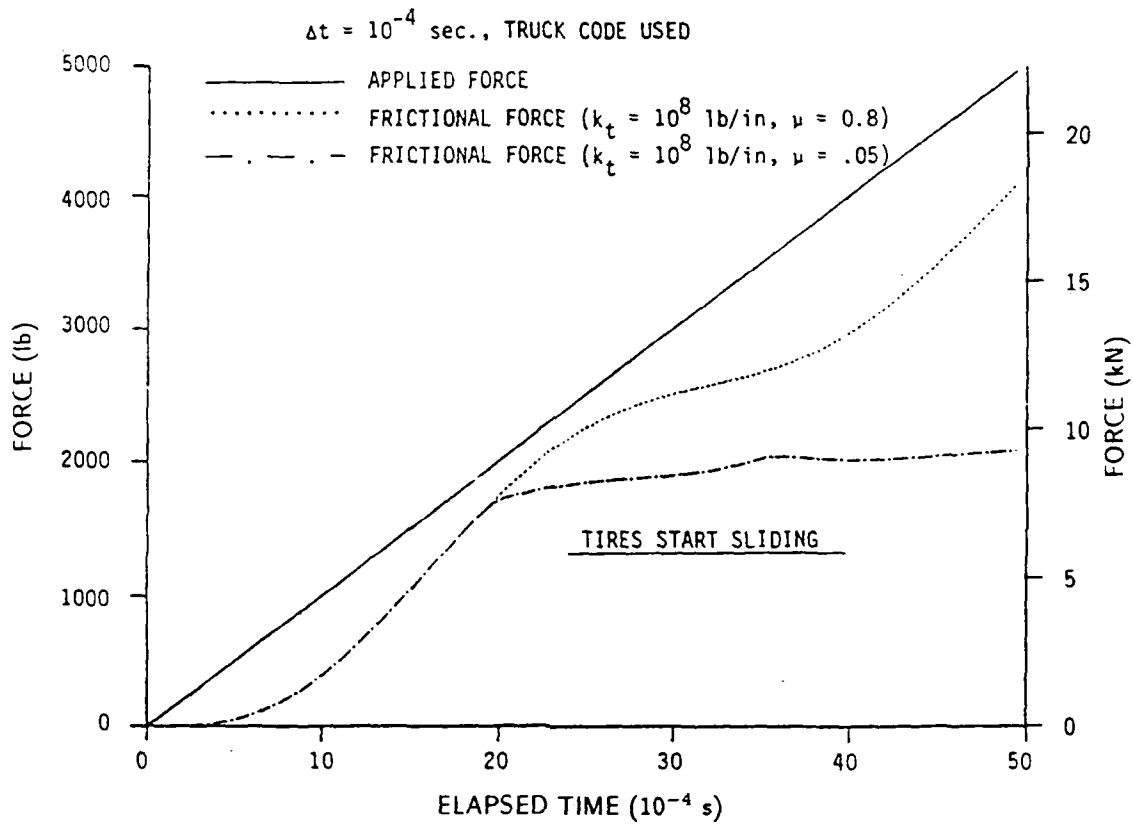


Figure 9. Comparison of Computed Frictional Force and Applied Force when the External Force Is Applied to the Side of a Truck.

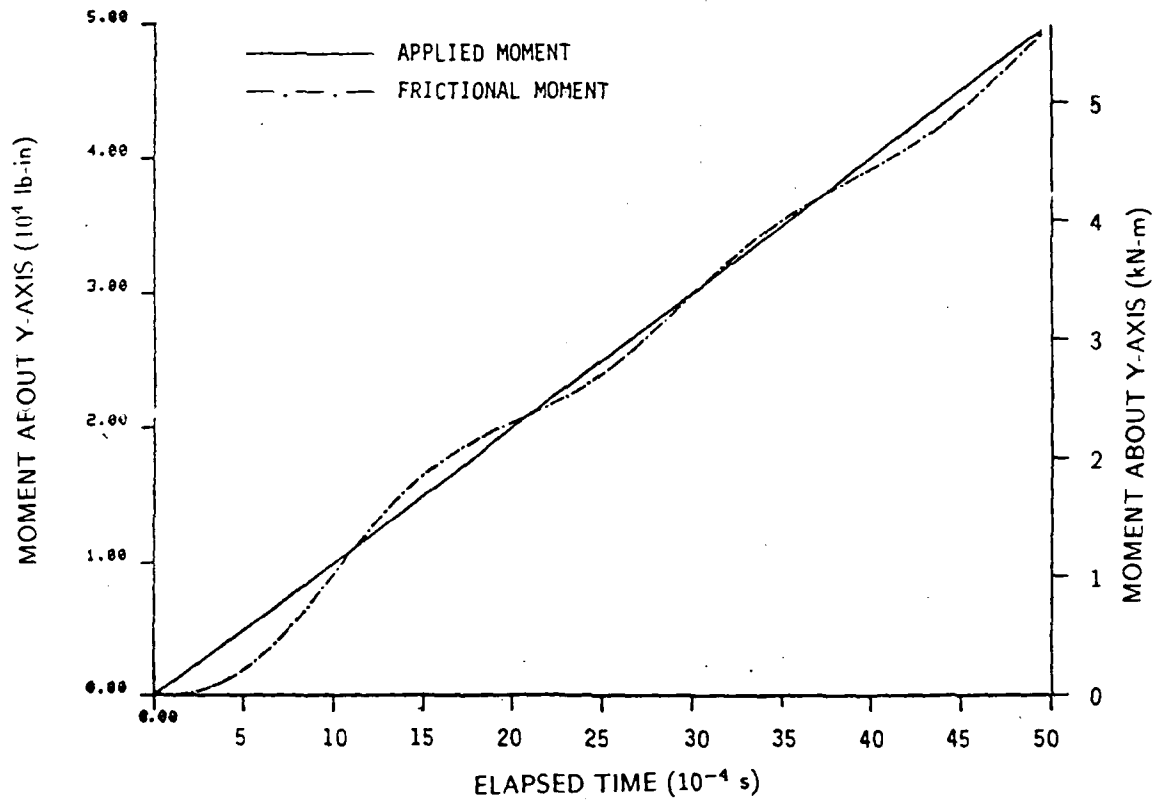


Figure 10. Comparison of the Applied Moment and Frictional Moment Computed with the TRUCK Code.

Table 1. Comparison of Computed Frictional Moment and Applied Moment. [$\mu = 0.8$, $k_t = 175 \text{ GN/m}$ (10^9 lb/in), $\Delta t = 100\mu\text{s}$]		
Time Elapsed ms	Applied Moment about y -axis $\times 113 \text{ N}\cdot\text{m}$ ($\times 10^3 \text{ lb}\cdot\text{in}$)	Computed Frictional Moment about y -axis $\times 113 \text{ N}\cdot\text{m}$ ($\times 10^3 \text{ lb}\cdot\text{in}$)
.0	0	0.
.1	1	0.
.2	2	-0.177
.3	3	-1.851
.4	4	-6.620
.5	5	+4.095
.6	6	-22.878
.7	7	+40.09
.8	8	-121.712
.9	9	+26.998
1.0	10	-701.29
1.1	11	+1691.1
1.2	12	-2671.
1.3	13	+3134.
1.4	14	-3135.
1.5	15	+3136.
1.6	16	-3136.
1.7	17	+3136.
1.8	18	-3136.
1.9	19	+3136.

The values kept on
alternating between
-3136 and +3136.

conditions as either frictionless or sticking i.e. with infinite friction. Once the additional capability to account for finite non-zero frictional forces at the contact surfaces has been included in the ADINA code, it should be possible to solve problems involving overturning of a vehicle with both the TRUCK and ADINA codes and compare the results. We note that all of the problems studied above were 2-dimensional and the ADINA code modeled frictional effects properly.

V. CONCLUSIONS

The equations of motion used in the TRUCK code were found to be correct and accurately coded in the computer program. The code gave acceptable values of frictional forces between the tires and the ground.

For the problems studied, the ADINA code gave frictional forces which agreed with those obtained by using Coulomb's law.

Problems involving overturning of a vehicle were not solved with either the TRUCK or the ADINA code.

The results indicate that the magnitude of the time-increment to be used in the TRUCK code depends strongly upon the tangential stiffness of the tire.

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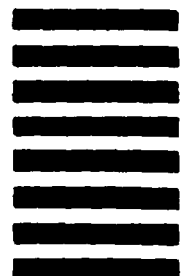
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