DYNAMIC STRUCTURAL CHARACTERIZATION OF STUB-SILL TANK CARS UTILIZING ADAMS & ANSYS SIMULATION MODELS

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ABSTRACT

A project on the dynamic structural characterization of a railway stub-sill tank car, using ADAMS and ANSYS simulation models, was carried out by the Centre for Surface Transportation (CSTT). ADAMS modelling was utilized to develop the dynamic system model. A simple interaction model was created to simulate the liquid-vehicle interaction. The calculated impact forces show a good correlation with the physical test data generated and measured by CSTT. Dynamic forces and accelerations obtained from the ADAMS model were input to a finite element (FE) model created in ANSYS to calculate the quasi-static stresses on the tank car body. The results from the FE analysis were compared with the physical data measured during tank car impact experiments. A reasonable correlation was achieved. The ADAMS and ANSYS models, their interfacing, and their validations are presented in this paper.

1. INTRODUCTION

Railway stub-sill tank cars are often used to transport hazardous materials. Tank damage may result in the leakage of contents which can have serious impacts on safety, health, and the environment. Fatigue cracking in the region of the stub-sill head brace has been identified as a significant factor that causes the failures of stub-sill tank cars. It is generally believed that high impact forces in the railway switching yard are the major sources that initiate the cracks or even directly cause damage to the tank cars.

In order to have a good understanding of the characteristics of the dynamic forces and stresses of the stub-sill tank cars under impact situations, a series of static and dynamic tests was recently carried out on a tank car by the Centre for Surface Transportation Technology (CSTT), National Research Council Canada (NRC) [1], and sponsored by Transport Canada. To further understand the behaviour of the stub-sill tank cars under impact situations, a project on the stub-sill tank car dynamic structure characterization using ADAMS and ANSYS simulation models was also carried out by CSTT. The ADAMS model was first developed and validated using the previous physical test data. The dynamic forces and accelerations obtained from the ADAMS model were input to a finite element (FE) model created in ANSYS using an ADAMS/FEA module. The quasi-static stresses on the tank car body were calculated using the ANSYS model and compared with the experimental data. The validated models were utilized to investigate the characteristics of impact forces on couplers and dynamic stresses on the car body. The ADAMS and ANSYS models, their interfaces, and their validations are presented in this paper.

2. SYSTEM DYNAMIC MODEL DEVELOPED IN ADAMS

To consider the flexibility of the tank car, the car body (tank and stub-sill assembly) is considered as two separated rigid bodies connected by a spring-damper system. Only longitudinal relative motion between the two rigid bodies is allowed. The stiffness of the spring-damper system is calculated from the finite element model of a tank car used to calculate the stresses. In the calculation, one end of the car is fixed at the location of the rear draft gear stops, and a static force is applied to the other end of the car. In this way, the bending flexibility of the car body is taken into account in the model. The car system model is shown in Figure 1.

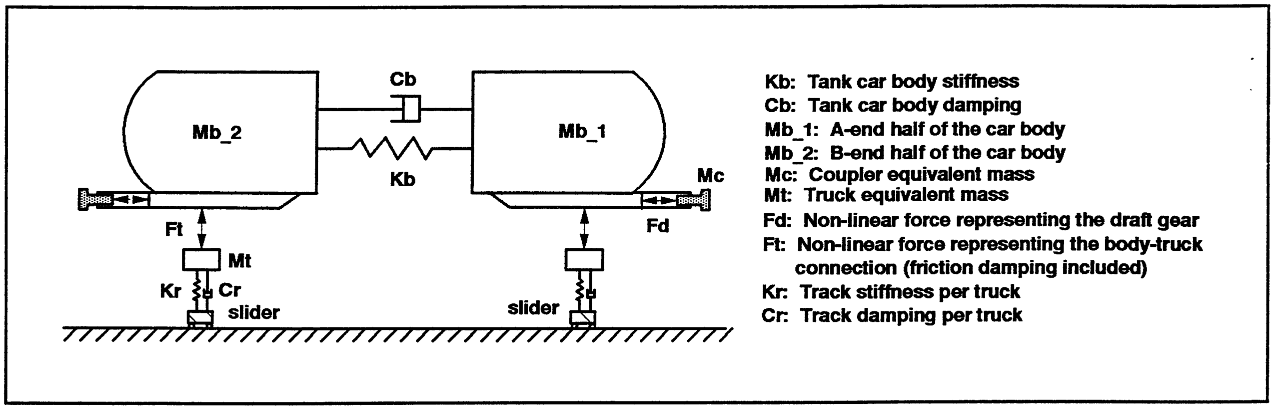

Figure 1. A model of a railway stub-sill tank car

The truck bolster is modelled as a rigid body. The rotational inertia of each wheelset is transformed to an equivalent mass and lumped with the truck body. The connection between the car body and truck bolsters at the centre plate is actually very rigid. A hinged joint is used for this connection. The rest of truck components are modelled as a lumped equivalent mass. A stiff spring-damper system is used to model the longitudinal stiffness of the truck. It links the bolster and the lumped truck mass in the longitudinal direction. In the vertical direction, a piece-wise linear spring is used to represent the truck suspension. If the upward displacement between the bolster and the sideframe exceeds the spring deflection under a static load at the truck location, then the car lifts off the truck springs. In such a condition, the reaction force on the truck is taken as zero. If the downward displacement under dynamic conditions exceeds the allowable travel of the truck springs, as specified by the designers and manufacturers, then the springs are considered to have bottomed out. Under this condition, a stiff bottom stiffness is used to compute the truck reaction force. The combined bottom stiffness consists of the sum of the bolster stiffness, the bottomed spring stiffness, and the sideframe stiffness.

The friction wedges used in the truck are energy dissipation elements used for reducing the amplitudes of vertical car body motion, as manifested at the truck bolster. In the model, the wedges are represented by dry friction (Coulomb) damping elements. At every time step, the vertical velocity of the carbody at the center plate relative to the truck sideframe is computed, then, a positive or negative sign is assigned to the calculated damping force such that the damping force always opposes the motion. The step function built into ADAMS was used to keep a smooth transition from one direction to the other one, which is very useful to maintain the stability of solutions.

In some impact cases, braking may be applied to the stationary vehicles being struck. A constant force is used to represent the braking force. The direction of the force is opposite to the direction of vehicle travel.

The track equivalent mass has little effect on the coupler impact forces and it is ignored in the model. The track is assumed to be an elastic, mass-less support and it is represented by a linear spring-damper system. The track stiffness is calculated based on the track model of a beam on an elastic foundation.

An equivalent mass is used to represent the coupler, yoke and a part of the draft gear mass. Their connection to the car body is modelled as a non-linear coupler force in the longitudinal direction, and is attached to the car body by a translational joint.

The longitudinal coupler force is critical and significantly affects the behaviour of the model. The coupler force depends on the draft gear performance characteristics and on the status of the draft gears on the two adjacent cars. The draft gears are energy dissipation devices and they have some special features. The force developed in the draft gear is not only a function of the distance traveled of the coupler relative to the car body, but also of the travel direction and speed. The influences of these factors on the force are also non-linear, and it is difficult to use a simple function to describe them accurately. The performance characteristics of MARK 50 and NY-11-F draft gears, two commercially-available models, are used in this investigation. The performance characteristics are usually obtained from drop hammer tests.

Several methods have been previously used to model the friction draft gear characteristics. The most popular one was to represent the characteristics by several segments of straight lines and compute the coupler force based on the coupler travel distance and status, as used in Ref. [2]. In our study, the travel and recoil curves of the draft gear characteristics are represented by two splines. When the direction of the relative motion changes, the curve for the computation is switched. The swich from the travel curve to the recoil curve is smoothed by the step function built into ADAMS, which gives good stability to the calculation. The available draft gear performance curve is usually up to 500 kips. Beyond this point, a stiffness of 1,200 kips/in is used to compute the increased part of the coupling force.

The vertical coupling force depends on the longitudinal coupler force, the relative displacement of the adjacent couplers, and the friction coefficient at the coupling interface. The vertical coupling force is calculated based on the following condition: the relative vertical displacement at adjacent couplers is calculated, which is dependent on the vertical and rotational or pitch orientation of the adjacent cars. When the relative displacement does not exceed the vertical coupler slack, the force is considered to be zero. If the slack is exceeded, the vertical coupling force is calculated using the vertical restraining spring stiffness between the coupler and the car body. The friction force is also calculated using the friction coefficient and the longitudinal coupling force. If the friction force is less than the vertical spring force, the friction force is taken as the force acting on the couplers. Otherwise, the spring force is used for the active force on the couplers.

In tank-car impacts involving a partially filled tank, part of the liquid cargo may undergo turbulent flow, and part of it may be separated from its main body. It is very difficult to model the liquid reaction accurately. In this study, it is assumed that the liquid inside the tank always takes a quasi-static shape, as shown in Figure 2. The angle (α) between the liquid surface and the horizontal plane depends on the vertical and longitudinal accelerations. With such an assumption, the pressure at the impacted end achieves a maximum value and gradually reduces as the distance from the end increases. This may not reflect the exact pressure distribution, but its basic tendency is consistent with the experimental data as reported in Ref. [3].

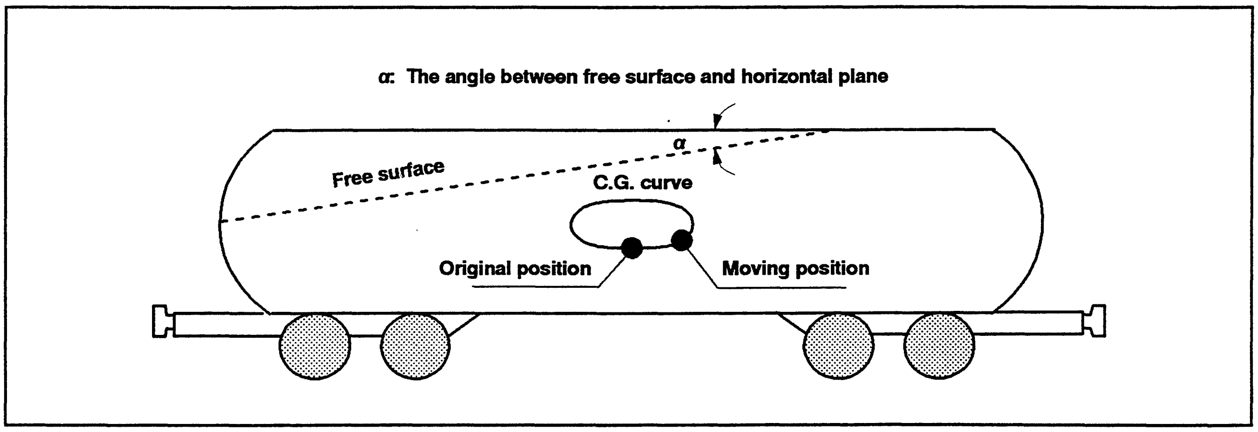

Figure 2. Liquid Model

Based on the assumption for the liquid reaction, the centre of gravity (CG) can be calculated for any shape of tank at any fill level using a geometry calculation program. The tank shape can be assumed to be the same as that in a static situation because it has little effect on the CG. For various longitudinal accelerations, the CG will move in the space and a CG curve or trace can be found. In the model, the CG curve is represented by a spline curve that can be easily created in ADAMS. The liquid inside the tank is modelled as a lumped mass and its CG is constrained to the CG curve using two PTCV constraints built into ADAMS. To consider the energy consumption and the compression effect of the liquid in the impact, a damper is attached to the liquid mass in the longitudinal direction.

An ADAMS model of an impact test setup is shown in Figure 3. In this setup, the hammer car is a stub-sill tank car. It moves at a certain speed and impacts the stationary "anvil" cars, which consist of an empty box car and three tank cars filled with concrete. The relationship between the two adjacent couplers on the anvil cars is represented by an impact element built into ADAMS. For purposes of this study, in order to simplify the computations, the pulling force on the couplers is not considered. The couplers can therefore separate freely, and in such a case, the longitudinal and vertical coupling forces are equal to zero. The coupler pulling force can be considered without any major difficulties.

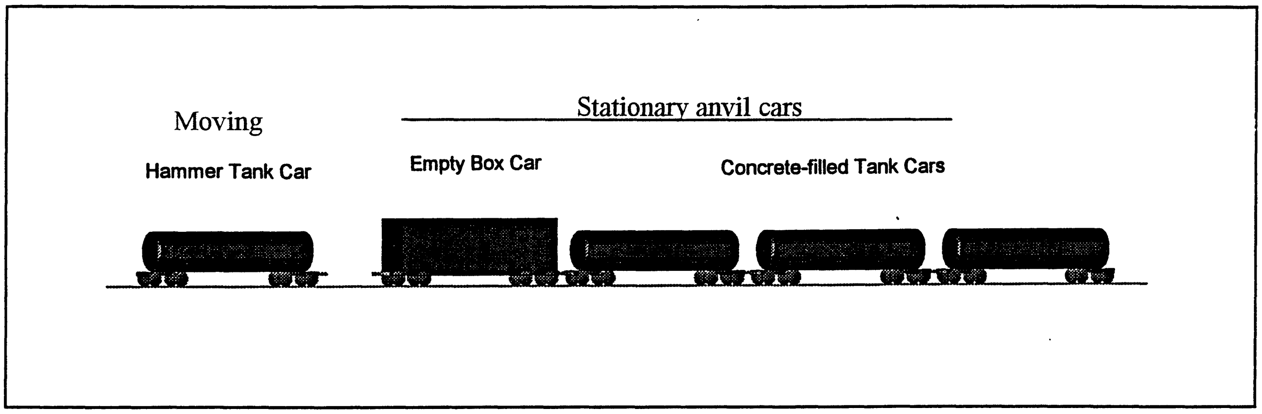

Figure 3 An ADAMS model of an impact test setup

3. ADAMS MODEL VALIDATIONS

A series of impact tests was carried out at the Centre for Surface Transportation Technology (CSTT), National Research Council of Canada, in August of 1995 [1]. The impact forces were measured using an instrumented coupler installed on the struck end of the first standing box car. The measured impact force data were then used to validate the system model in ADAMS.

Figure 4 shows several samples of comparisons between the theoretical time history predictions and the actual test data. As can be seen, both the calculated and the measured impact forces have two major peaks. We specify the first major peak as P1 and the second as P2. P1 results mainly from interaction between the hammer car and the first anvil car. When the first anvil car rebounds from the second anvil car, the second peak, P2, is generated. It can be seen that both the basic shape and the time period of each peak are in very good agreement with the experimental data. This suggests that the impact-tested rail car system is well represented by the ADAMS mathematical model.

The P1 values vs. the impact speeds from the model and the tests are shown in Figure 5. It is clear that the basic tendency of the calculated peak forces are the same as the experimental data. The agreement between the theoretical results and experimental data is excellent when the impact speed is greater than about 7 mph. At lower speeds, the experimental data values are usually greater than the theoretical results. A possible cause is the forces in this range being greatly affected by the friction characteristics of the draft gear. The fluctuation in friction forces usually causes some sharp spikes as the draft gear compresses, as can be seen in all the experimental data. This may be caused by vibration of the couplers and may be influenced by the draft gear travel speed.

The friction coefficient for a static situation is usually higher than for dynamic conditions. When the draft gear travels from a given speed to a stop, the friction force will usually rise suddenly to a sharp peak [4]. This is not considered in the present calculation and may be the main source that caused the differences. Fortunately, these

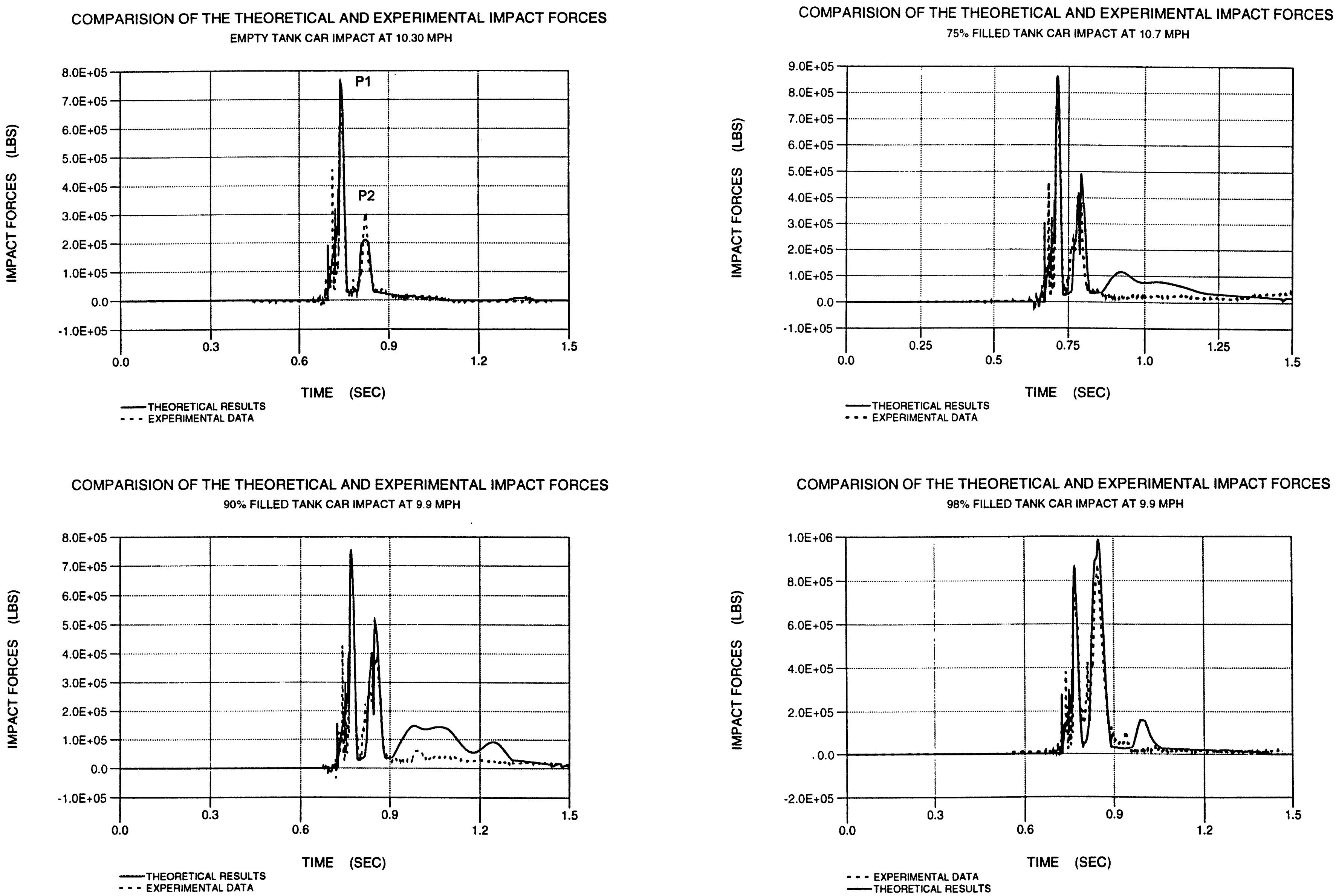

Figure 4. Comparisons of theoretical data and test results in time domain

sharp peaks usually have a high frequency and they may be attenuated as they travel through the car body structure, and may have little effect on the strength of the structure. Also, the forces in this range are usually lower than 350 kips and they tend not to threaten the integrity of the tank car structure. If these sharp peaks are filled out, the agreement between the experimental data the theoretical results would be better.

Both theoretical and experimental results indicate that the change in impact force is relatively small when the speed is less than about 7 mph. In this speed range, the draft gears are usually not bottomed and the impact forces mainly depend on the performance characteristics of the draft gears. Once the draft gear has bottomed out, the impact force increases rapidly. A small increase in the higher speed range (say, above 7 mph) could result in a significant increase in impact force and cause damage to a vehicle, especially if the hammer car and anvil car(s) are loaded.

Both theoretical and experiment results show that increasing the fill level of the tank generally increases the impact force. However, its influence on the first peak value is not really significant when the fill level is less than 95%. The liquid in such a case acts like bulk material with a soft cushioning effect. The surge force is always delayed from coupler impact and contributes little to the increase in impact force. Further increasing the fill level reduces the delay of liquid reaction and makes the surge force phase close to that of coupler impact so that the impact force is increased.

Comparisons of the calculated results and experimental data for the second peak (P2) are shown in Figure 6. Their agreement is not as good as that of P1 but their basic tendency is similar. Both theoretical and experimental data indicate that increasing the fill level significantly increases the second peak value. This is because increasing the fill level increases the surge force and reduces the phase difference between the surge force and the rebound action of the first standing car. If the first stationary car is empty, the second peak value may be larger than the first peak value at a high fill level, say larger than 95%,. In this case, the second peak usually has a lower frequency and may be more critical than the first one.

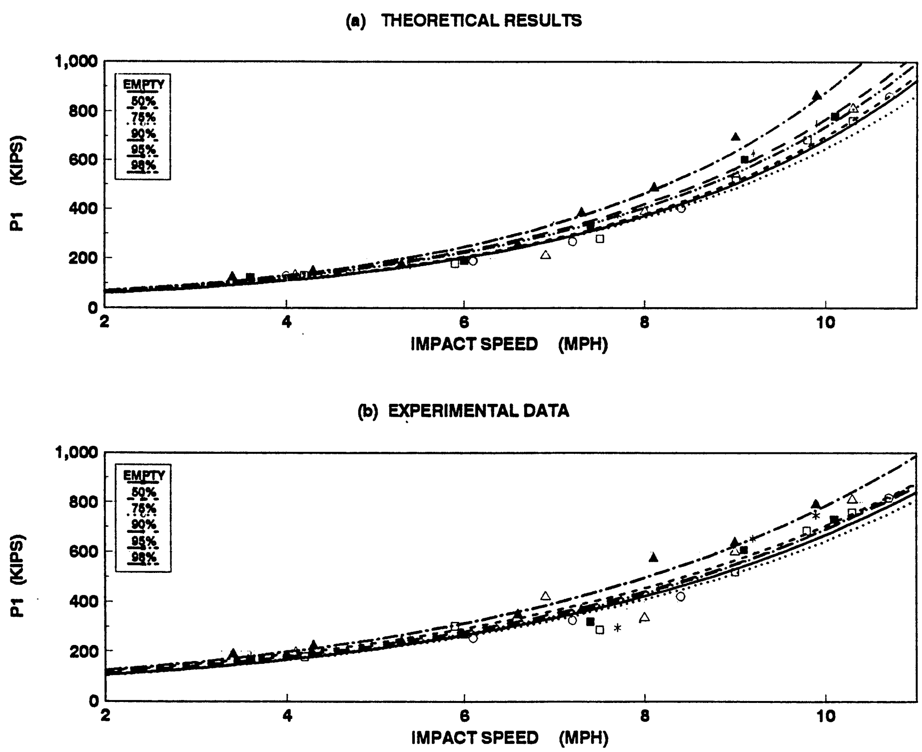

Figure 5. P1 comparisons

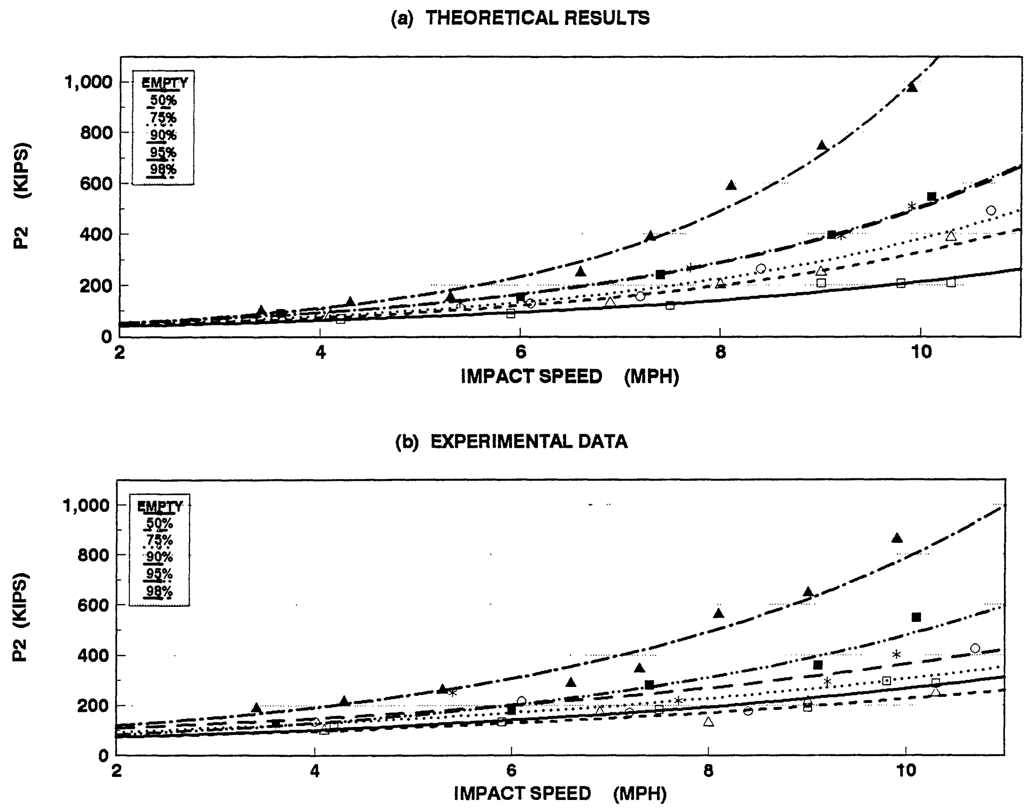

Figure 6. P2 comparisons

4. ANSYS FE MODELLING OF STUB-SILL TANK CAR BODY

For the purposes of this investigation, the stresses on the tank car body are calculated using an FE model developed in ANSYS. The ANSYS tank car body model is shown in Figure 7. The geometry is built from drawings provided by the tank car designer. The FE model is created using 4-node elastic shell elements and spring elements. The stub sill extension, bolster saddle plate and pipe are also accurately modelled. For simplification, the model is built with the tank pad integrated into the shell, avoiding the use of contact elements between the two surfaces. Neither the manhole nor the pressure relief valve is considered in the model.

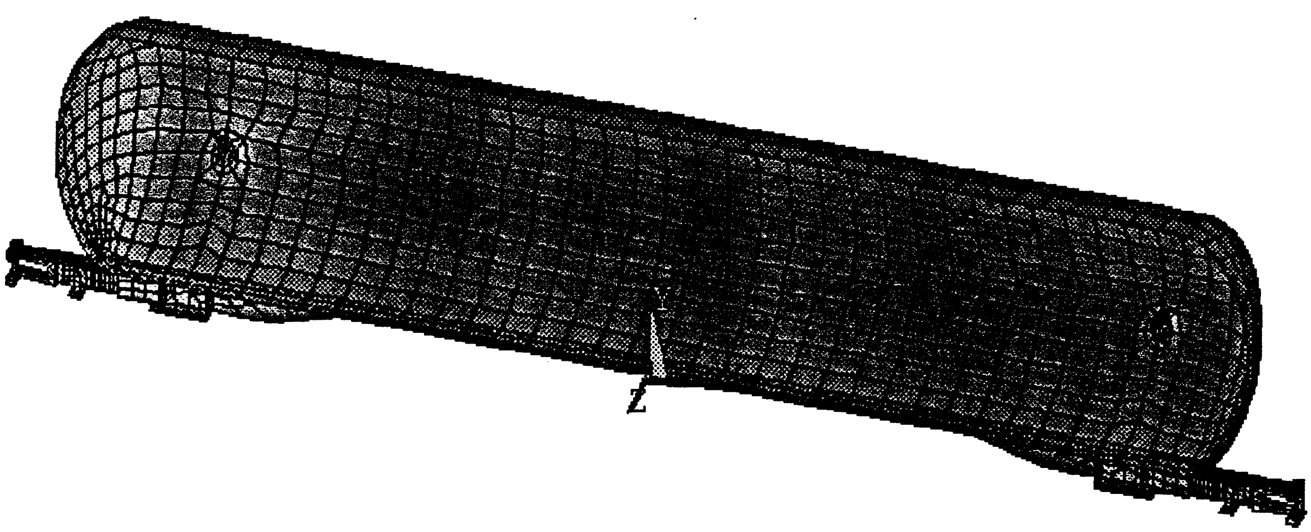

Figure 7. ANSYS tank car body model

Creating the mesh for the FE model is one of the most important steps in the analytical process. The accuracy of the results depends on how well the mesh is created. Therefore, adequate meshing techniques (using mapped meshes for better element shape and meshes with higher densities for high stress gradient areas) are applied particularly at the areas of anticipated high stress.

5. INTERFACING ADAMS AND ANSYS MODELS

A quasi-static analysis technique is employed to calculate the dynamic stresses on the stub-sill tank car body. The worst load condition at each impact speed is considered in the calculation. An ADAMS/FEA module is used to transform the basic forces and accelerations generated in the ADAMS model into the ANSYS model.

A single force is assumed for the connection between the coupler and car body in the ADAMS model and it acts at the geometrical centre of the stub sill. Because there is no FE element or node point at such a point in the ANSYS model, and the force is supposed to act on an area, the force is transformed to a pressure and is manually assigned to the corresponding area.

The liquid pressure in the tank is calculated based on the accelerations of the liquid equivalent mass. The free surface in the liquid model is determined using the ratio of the longitudinal and vertical accelerations of the liquid equivalent mass. The liquid boundaries are found after the free surface is determined. The pressure on each element area is calculated using the accelerations and the distance from the element area to the free surface, which is measured on the line perpendicular to the free surface. The pressure is manually applied to the tank shell.

Symmetrical boundary conditions are applied on the vertical centre plane, since half geometry is used in the calculation. To minimize any possible errors in the critical area, only a vertical constraint is applied to the centre plate at the struck end of the tank car. At the other end, vertical, lateral and longitudinal displacement constraints are applied on the car body bolster to eliminate singularity of the FE model. In the case of empty car, in which liquid force does not act on the car body, the reaction forces at the constraints are very small, as required. The reaction forces generally increase with the fill level. This is mostly caused by the errors in the calculated pressure and its distribution pattern. Its equivalent force might not be exactly consistent with the ADAMS modelling. However, the errors are usually small and the calculated stresses are qualitatively acceptable for understanding the basic characteristics of the stub-sill tank structure, as presented below.

6. VALIDATIONS OF THE ANSYS MODEL

6.1. Static stresses

Both static and impact tests were carried out by CSTT at its Ottawa vehicle laboratory. The tests were performed using a tank car of stub-sill design. This car had been previously used in service, but did not show any visible cracks or damage. This car is considered representative of those widely used by railroads across North America. The instrumentation and the locations of the strain gauges for the static and impact tests are basically the same.

The test data is utilized to validate the ANSYS model. Static validation is performed by comparing the strain gauge readings during compression tests at 800,000 lb. with the corresponding FE values. For comparison, a number of high stress points at various locations on the structure were chosen. The strain values of the rosette gauges are resolved into principal stresses. The gauges are generally located around the connection of the stub-sill with the head brace and pad/hemispherical head. The results of the FE analysis and static tests are generally within 10%, as shown in Figure 7. This validation therefore ensures proper development of the FE model and appropriate application of boundary conditions, which are then used for the dynamic analysis.

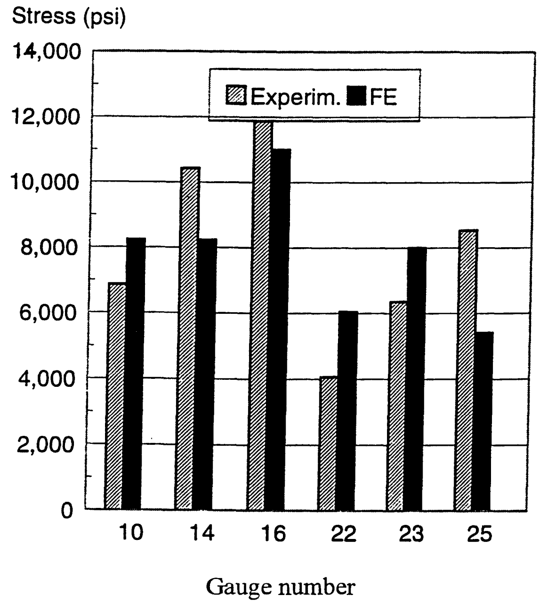

Fig. 8. Comparison of FE results and experimental data from static compression tests

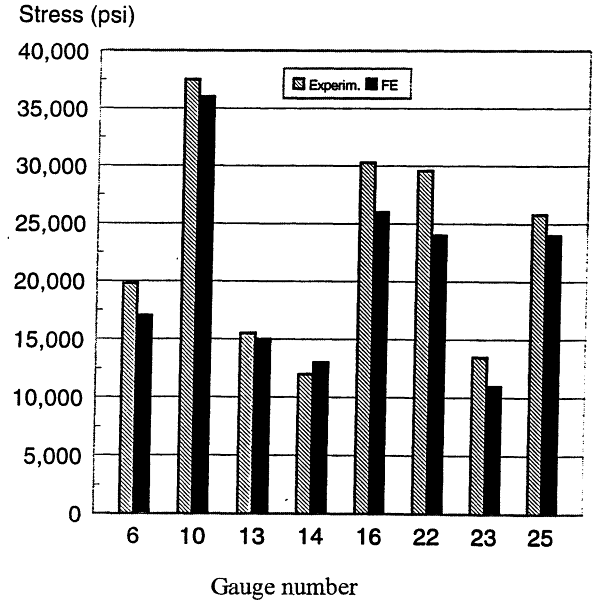

Fig. 9. Comparison of FE results and experimental data for 0% fill level at an impact speed of 9.8 mph.

6.2. Dynamic stresses

Dynamic analyses of the tank car FE model, using ADAMS results, are performed for three distinct cases: 0%, 50%, and 98% tank car fill levels. There is usually a delay in the dynamic strain from the impact force acting on the coupler. It is assumed that the maximum peak value corresponds to the worst load case calculated from ADAMS model.

For the 0% fill level (empty) impact case, the correlation between the test data and theoretical results is very reasonable when the impact force is high or the impact speed is relatively large, say larger than 7 mph, as shown in Figure 9. The correlation is not so good at low impact speeds or at low impact forces. The reason is probably that the oscillations in the coupler force at low impact speeds also cause a noisy response in the stresses. These peak values do not correspond to the calculated quasi-static stresses. Fortunately, the stresses in the low-speed range are not critical in our investigation, being well below the allowable level.

For the 50% fill level studies, the ANSYS - ADAMS model also provides good results. The principal stress values calculated from the physical test results are within 20% of the FE stress values, and closely reproduce the structural behaviour, as shown in Figure 10. The structural response of the 50% filled car and the 0% filled car do not differ greatly. The major difference is at the non-impacted end. At this end, the structure responds to actual impact by producing a peak value with about 100 ms delay in time referred to the impact force on the struck coupler. This peak is related to wave transmission in the tank car structure. It is impossible to properly predict the stresses at such a location using the quasi-static technique. As in the case of empty tank car, the stresses determined by the FE model are not in agreement with the experimental data at a low impact speeds.

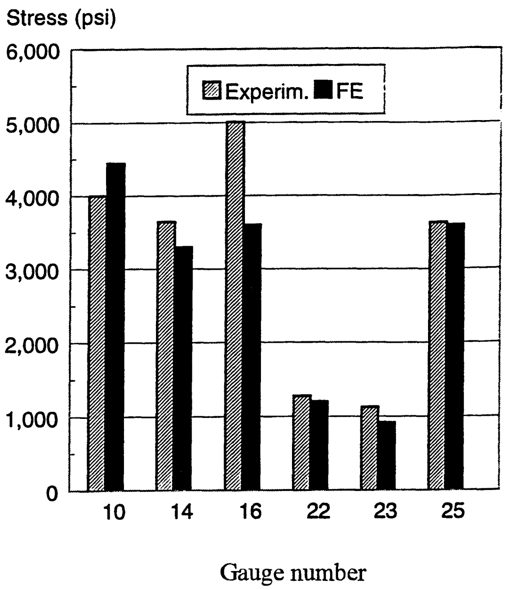

Fig. 10. Comparison of FE results and experimental data for 50% fill level at impact speed of 8.0 mph

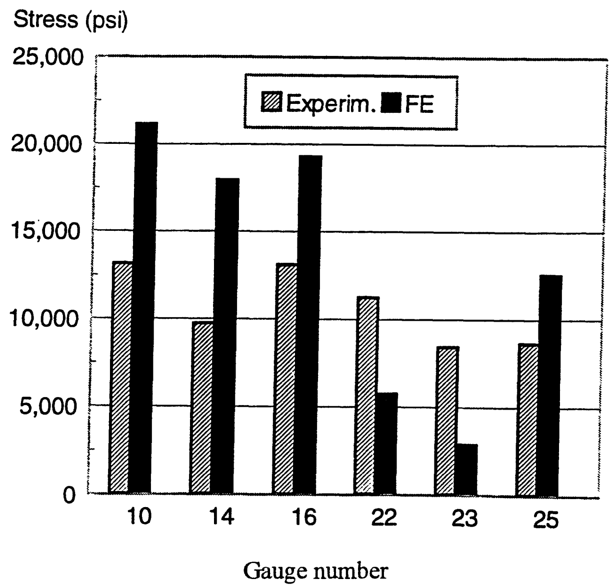

Fig. 11. Comparison of FE results and experimental data for 98% fill level at impact speed of 9.9 mph.

Figure 11 presents the principal stresses from the FE analysis and tests for a 98% fill level impact at 9.9 mph. From this plot, it can be seen that the experimental stresses only represent about 60% of the calculated stresses. The main reason is probably that the liquid model may not truly represent the nature of the liquid reaction at such a high fill level. A further improvement on the liquid model is required. Nevertheless, the basic pattern of the stress distribution and the critical area of the stresses is consistent with the experimental data.

7. CONCLUDING REMARKS

An ADAMS model of a railway stub-sill tank car was developed. The impact forces acting on the stuck couplers showed good correlation with the experimental data, especially at a fill levels lower than 95%. The ADAMS model was adequate to study the characteristics of impact forces acting on the couplers and car bodies.

The dynamic stresses on the tank car body were calculated using a quasi-static calculation method. The calculated results matched with experimental data fairly well when the tank fill level was relatively low. Even though the stress values were overestimated when the fill level was high, (say 98%), the basic stress distribution pattern and critical area could also be predicted using the ADAMS-ANSYS models.

The proposed liquid model for the liquid-vehicle interaction simulation was simple and efficient. It could be used to understand the basic performance of the tank vehicle. The impact forces and stresses might be overestimated when the tank fill level is high. Modifications of the liquid modelling are needed for future studies.

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