Truck Performance -- Friction Snubber Force Measurement System

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ABSTRACT

With all its truck and carbody instrumentation, Phase I of the Truck Design Optimization Project found that adequate transducer systems for the measurement of forces in the friction snubber were non-existent. This paper documents the design, testing and potential utilization of such a system, fabricated and tested by Wyle Laboratories.

INTRODUCTION

Within the last decade, an increasing amount of research has been directed toward improving the performance of the three-piece freight car truck. Truck manufacturers have designed and built several new truck configurations as well as added special purpose components aimed at improving such performance parameters as ride quality, lateral stability, and curve negotiation. There has been a great deal of testing by both industry and government for the evaluation of the comparative advantages afforded by modified or new trucks under a range of operating conditions. Of course, testing was conducted on the standard three-piece friction snubbed truck to establish quantitative performance characteristics as a base for the evaluation of new or modified designs.

While the conclusions drawn from many of the test results were very often contradictory, the methods of truck performance evaluation have steadily grown more rational. Much of the technological advance can be attributed to the parallel use of mathematical simulation and full-scale testing. The mathematical models used in simulations are necessarily idealized as they are linearized to reduce computing time, and because many of the non-linear parameters in truck dynamics are not quantified.

In 1974, the Federal Railroad Administration awarded a contract to the Southern Pacific Transportation Company for the performance of Phase I of the Truck Design Optimization Project (TDOP) which had the objective of quantitatively characterizing the performance of the general purpose freightcar truck. The Contractor was primarily concerned with the evaluation of the two most commonly used three-piece trucks: the American Steel Foundries (ASF) "Ride Control" truck, and the Barber S-2 truck; the first incorporates constant snubbing friction, and the second, load-dependent snubbing friction. TDOP Phase I instrumented new 70-ton (63,502 kg) and 100-ton (90,718 kg) ASF and Barber trucks to measure accelerations, normal contact forces at the roller bearing adapters and relative linear and angular displacements between side frames and bolster. Relative rotation between truck and carbody bolsters was also measured.

It was soon recognized that instrumentation was not available to measure the forces transmitted through the spring loaded friction shoes or wedges between the side frames and bolster. In theory, it might have been possible to derive the friction forces by comparing measured accelerations and displacements with the calculated dynamics of a system without energy dissipation. It was also apparent that this approach would not only be cumbersome and expensive in computer time but would not lead to an accurate determination of the friction forces. In November 1974, a preliminary design concept of a transducer system to measure the forces between side frames and bolster was submitted by Wyle Laboratories for review.

ENGINEERING CONSIDERATIONS

Although the bolster-side frame connection is structurally and mechanically simple, it performs a multiplicity of functions:

- Vertical support of the carbody weight through the spring nest
- Centering of the bolster between the side frames through lateral spring forces
- Partial isolation of the carbody from shock and vibration through the springs in both vertical and lateral directions
- Dissipation of energy in both vertical and lateral directions, through the friction wedges and wear plates
- Equalization of wheel loads on uneven track, by permitting relative pitch and roll displacement
- Transmission of yaw torques between wheelsets and centerplate, required for curve negotiation, mainly through the friction shoes and wear plates
- Transmission of longitudinal braking forces, also through the friction shoes
- Limitation of excessive relative displacements through the bolster gibs

Examining these functions it is important to note that the friction shoes are involved in four of the eight interactions between side frame and bolster. The non-rigid connection between side frame and bolster permits relative motion in six degrees of freedom and consequently transmits six generalized forces--three forces and three moments--between the friction shoes and wear plates.

The tapered surface of the friction shoes presses against the mating surface of the bolster, and the wedge action results in a normal force between the vertical shoe surface and the wear plate. This is generally referred to as the column pressure. The two shoes at each bolster end load each other. Relative vertical or lateral displacement of the bolster gives rise to corresponding friction forces. Braking forces are transmitted by increasing friction forces on the rear shoe unless the column load is exceeded which results in gib contact. Relative bolster roll, in which plane contact between friction shoe and wear plate is maintained, applies a roll friction moment on the side frame column.

The remaining two rotations, relative pitch and yaw, give rise to more complex interactions as both the slanting and vertical surfaces can no longer remain in plane contact with the bolster and wear plate, respectively. The resulting edge-to-surface contact is an important cause of wear, both in the bolster pocket and at the upper and lower edges of the vertical shoe surface. The high restoring moment in yaw, also in pitch, occurs when the side frame is yawed with respect to the bolster and plane contact at either the sloping or vertical surfaces of the friction shoes changes to contact at diametrically opposite wear plates or contact points in the bolster pockets. Rotation of this diagonal into the center plane of the side frame thus requires that the distance between opposing friction shoes be shortened. The friction shoes thus move closer together, and in so doing slide inward along the slanted mating surfaces with the bolster. This causes additional compression on the snubber spring, and since the vertical load has not changed, there is a slight rise in the bolster with respect to the side frame in the case of load-dependent snubbing. The potential energy of elastic deformation is merely redistributed between the suspension and snubber springs. Thus, yaw rotation increases the potential energy of the system by raising the weight carried by the bolster. For load independent snubbing, where the snubber spring is based on the bolster, the entire potential energy is stored in the snubber spring. In either case, the reactions due to skew are applied at diagonally opposite edges of the side frame column, thus providing a yaw restoring couple.

All of the load paths discussed above were considered in the design of the Friction Snubber Force Measurement System (FSFMS).

MANUFACTURING TECHNOLOGY AND CONCEPTUAL DESIGN

The technology applied to the FSFMS was available in the field of towing tank testing where "force blocks" are utilized to measure the forces and moments applied between a moving carriage and a towed ship model. A force block is a hollow, roughly cubical block of alloy steel mounted at opposite sides to the objects between which forces are to be measured. The other four sides are machined so as to leave short cantilever beams instrumented with strain gages to measure bending stresses resulting from shears applied at the mounting surfaces. Additional strain gages are provided for nulling stresses due to normal forces.

In measuring several degrees of freedom, the load path must pass through each transducer in turn. In other words, the transducers must be in series or cascaded. Cross coupling of signals is minimized by making the blocks very stiff in both shear and compression along all axes not used for measurement. In the case of the FSFMS, an additional requirement was symmetry of the load path in order to preclude unsymmetrical deflections that would alter the contact geometry between the friction shoe and the wear plate. In addition, all force blocks had to fit within the envelope of the side frame.

The original design concept of the FSFMS is shown in Figure 1. The wear plate is welded to an adapter which in turn is bolted to a single vertical force block. The opposite face of the force block is bolted to an adapter to which a pair of lateral force blocks are mounted. Two normal force blocks are attached above and below, and are in turn bolted to an adapter rigidly mounted on the side frame. The adapter between the vertical and lateral force block divides the load path into two symmetrical sections which ensure that any tendency of the wear plate to tilt under unsymmetrical loading is minimized.

The five force blocks are capable of measuring two of the three moments applied by the friction shoes: The pitch moment is found from the differential loading of the upper and lower normal transducers, and the roll friction moment from the differential loading of the upper and lower lateral transducers. Space limitations precluded a transducer configuration capable of measuring a yaw moment, however, the increased column load due to yaw can be measured. An exploded view of the transducer as built is shown in Figure 2. Space limitations in the side frame also required that the normal and lateral force transducers be combined into a single unit. Each individual transducer is compensated against cross-coupling, however, it was not possible to eliminate the cross-coupling between the vertical and normal transducers. This is due to the fact that the plane of the mounting adapter to which the lateral and normal force transducers are attached is offset from the plane of the wear plate adapter carried by the vertical transducer. A vertical friction force thus produces a moment which is resisted by equal and opposite normal forces. The forces making up this couple must be distinguished from unequal normal forces due to a vertical offset of the center of friction shoe pressure from the center of the wear plate which occurs with bolster displacement. The correction factors were established for each transducer assembly by calibration and must be used in the reduction of data collected in road tests.

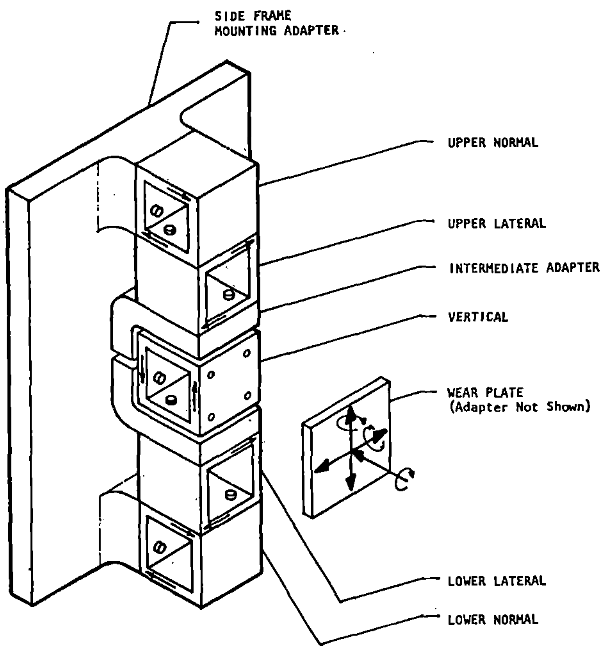

Figure 1 - Basic Concept of Force Transducer

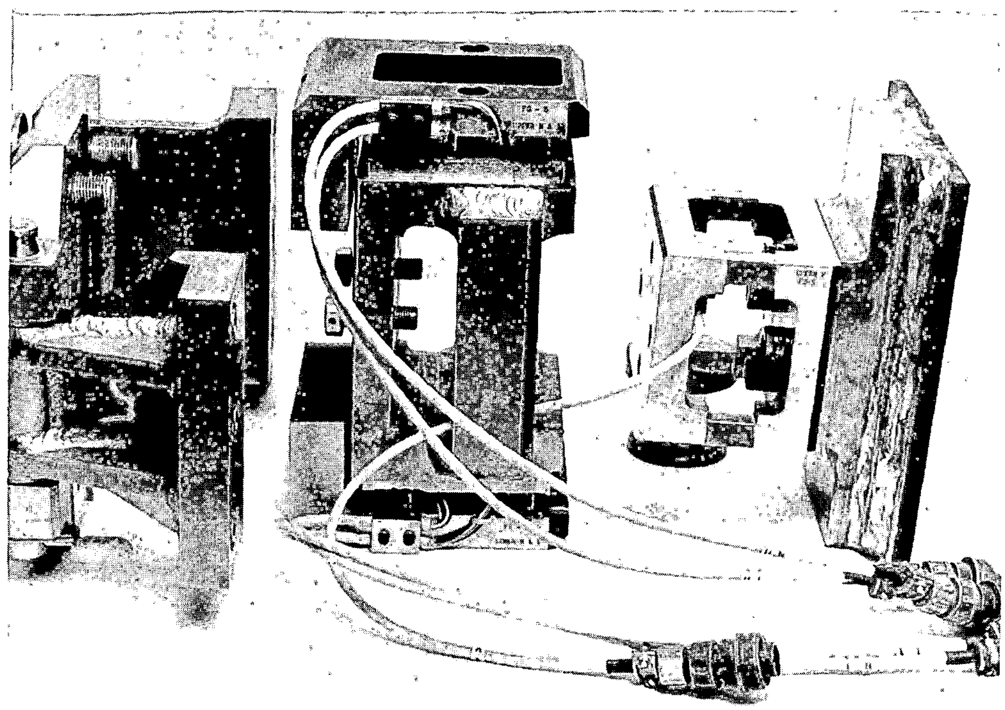

Figure 2 - Exploded View of Transducer Assembly

MODIFICATION OF SIDE FRAMES

To be capable of running across the country under cars in revenue service, the modified, instrumented trucks had to be able to withstand normal shock loads, resulting in stresses below the fatigue limit as specified in AAR M-203-65. This required the side frame to be tested under a lateral load of 35,000 lb (15,876 kg) without exceeding the deflections listed in the standard. This led to a modification of the orignal design.

An opening in the center of each column of the side frames was required to accommodate the wear plate adapter which transmitted friction shoe forces to the transducer assembly mounted behind it. Originally, the entire column was to have been removed and replaced by two heavy welded steel bars. The bars would serve the double purpose of providing both reinforcement of the open center and a mounting surface. Removal of the entire column might have caused more deformation in the side frame than could have been corrected. Therefore, only the column web behind the location of the wear plate was removed after the reinforcing bars had been welded. The modified side frames were then stress relieved with the center opening of each side frame stabilized by diagonal braces to preclude distortion. A modified side frame of the Barber S-2 truck is shown in Figure 3.

The critical dimension maintained in the modification of each side frame was the distance between wear plates: 17-3/4 in. (0.45m) in the ASF truck and 17 in. (0.43m) in the Barber S-2 truck. This spacing determines the column load with the given bolster and friction shoe geometry, and the spring characteristics. With the dimensions of the transducer stack between the wear plate and the mounting adapter flanges given, the offset between the face of the wear plate and the back surface of the reinforcing bars determined the spacing between the wear plates. Care was taken in the fabrication to prevent distortion.

A preliminary stress analysis indicated that the modified column when treated as a rigid frame with infinitely stiff girders had ample strength to resist a concentrated transverse force of 17,500 lb (7,938 kg) applied at the center of one of the reinforcing bars. This represented one-half of the specified 35,000 lb (15,876 kg). The ledges surrounding the wear plate where concentrated lateral force would be applied by one or the other bolster gibs were the weakest point in the modified column. Removal of the column web deprived the lip of a backup and caused the gib forces to be resisted by a portion of the lip in cantilever bending. A bar with tapered edges was welded to the inside of the cut to provide reinforcement, and the rear edges of the wear plate adapter were tapered to provide clearance in the reduced opening.

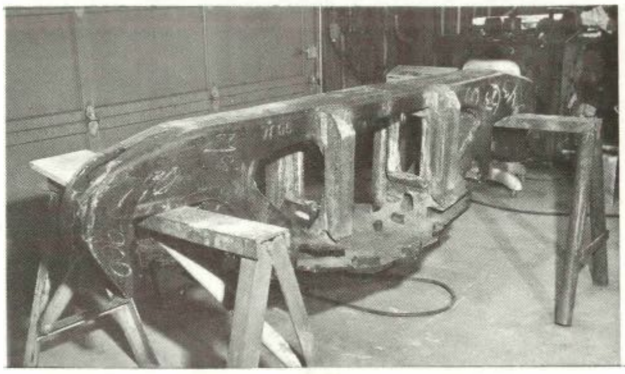

Figure 3 - Modified
Barber S-2 Side Frame

TRANSDUCER CONSTRUCTION AND ASSEMBLY

The transducer components were machined from 17-4 pH precipitation hardening steel with a yield strength of 140,000 psi (965,266 x 10^3 N/m^2). Simultaneous application of a normal load of 6000 lb (2722 kg) and vertical and horizontal friction forces of 3000 lb (1361 kg) each would produce stresses of only about 20,000 psi (137,895 x 10^3 N/m^2). Therefore, an ample margin of safety is provided for unforeseen overloads.

The 35,000 lb (15,876 kg) lateral load does not pass through the force blocks and therefore posed no problem for the transducer design. A portion of this load, however, must be resisted by the U-shaped mounting adapter which is flange mounted on the column reinforcing bars and forms a structural tie across the column opening. The stiffness in the lateral load path through the mounting adapter was therefore lowered. As shown in Figure 4, the thickness of one web between the transducer mounting plate and its flange was reduced so that it would act as a flexure. The opposite bracket is connected to the transducer mounting plate by a stainless steel pin assembled in self-lubricating bushings. This bracket transmits essentially all of the lateral friction forces from the wear plate to one column reinforcing bar. Under a lateral impact force high enough to decrease the distance between the reinforcing bars, the flexure will minimize the portion of the load transmitted through the mounting adapter.

The two-piece mounting adapter greatly eased the assembling of the transducers in the confined space of the side frames. Individual components were introduced one at a time and tightened with a torque wrench in a thread-locking compound. Some interferences between transducers and fillets in the side frame castings were found behind the column, above and below the cutout. Some of these interferences were due to variations between castings, and it was necessary to bevel the edges of the lateral and normal force transducers as well as the rear edges of the cutout. Modifications were also required in the same area of the lower two bolts of the mounting adapter in the case of the Barber truck to provide space for assembly. Figure 5 illustrates the completely assembled transducer in the ASF truck.

CALIBRATION TESTING

The tests conducted on the FSFMS were intended to demonstrate performance of the transducers in the truck under some simulated operating conditions without reproducing all aspects of the rail environment which would have required more complex and costly test equipment. Only vertical and lateral movements of the bolster were generated during testing, the later displacement considered essential to prevent the formation of vertical grooves in the friction shoes. To minimize the hydraulic power required to move the bolster, only two springs were installed in each side frame. More springs should have been used to prevent rocking of the side frames about their roll axes; however, this motion demonstrated the capability of the transducer assembly to identify friction torques due to roll.

Figure 6 depicts the Barber S2 test set-up which was, of course, identical for the ASF Ride Control Truck. An existing test frame was modified by adding four pedestals to support the pedestals of the side frames and to restrain them laterally. A beam simulating the carbody bolster was nested by a center plate in the truck centerbowl. The beam was raised and lowered by a pair of double-ended, double-acting hydraulic actuators controlled by electrohydraulic servo valves. Linear Differential Voltage Transformers (LVDT's) mounted on the actuators provided position feedback. A third horizontal hydraulic actuator mounted on a bracket atop the test frame provided lateral motion of the simulated carbody bolster. A central frame guided the bolster beam in a vertical plane through grease lubricated rubbing plates. Vertical and lateral relative displacements between the bolster and each side frame were measured by LVDT's.

Outputs of all 20 force transducers, the four bolster-side frame LVDT's, and the three actuator LVDT's were recorded on four oscillographs. All calibration factors for the force transducers were established with the friction shoes out of contact with the wear plates. In

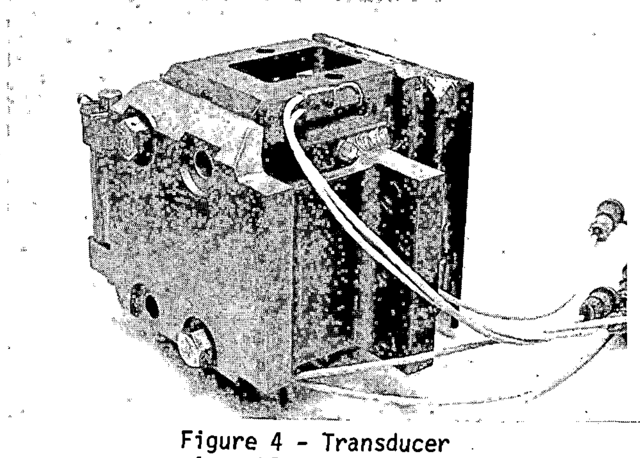

Figure 4 - Transducer Assembly Rear View

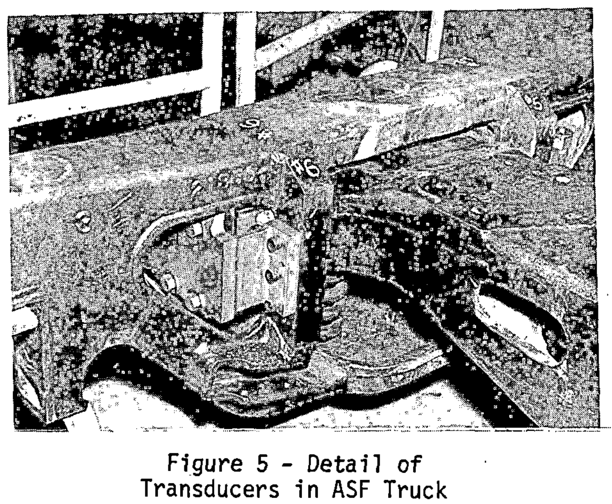

Figure 5 - Detail of Transducers in ASF Truck

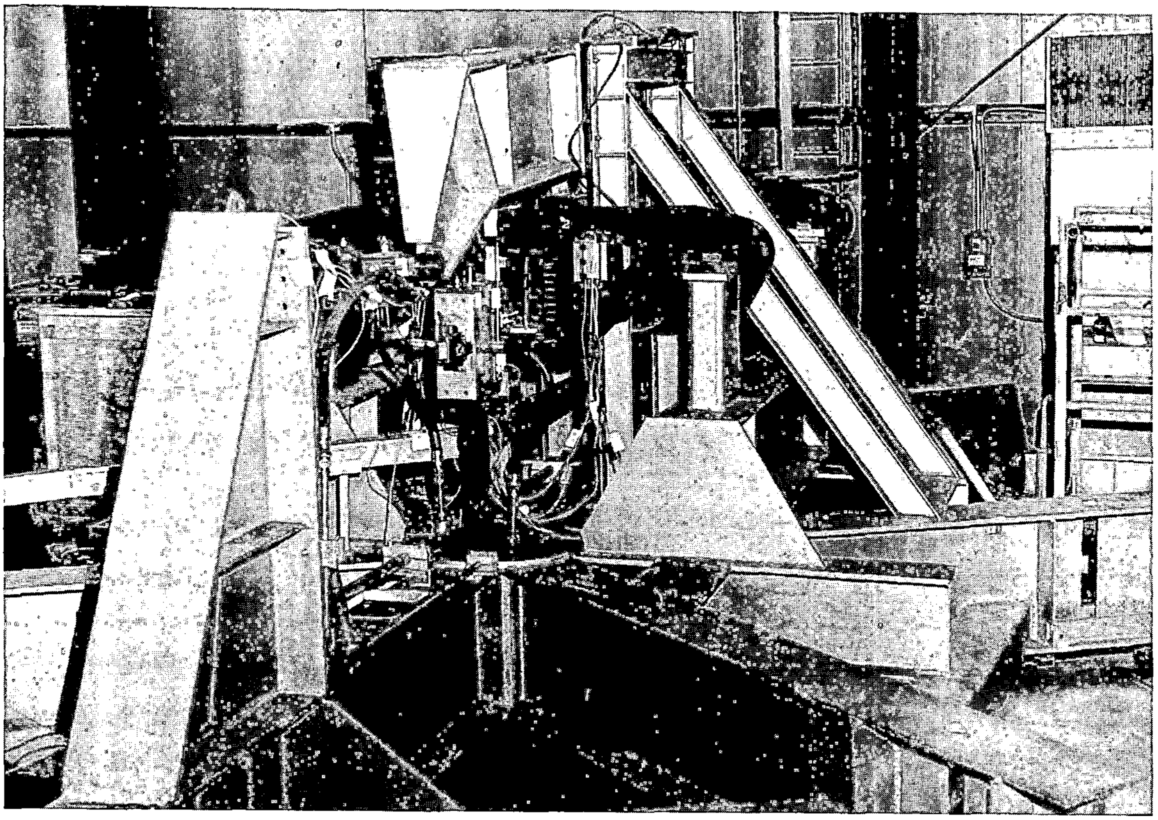

Figure 6 - Barber S-2
Truck on Test Stand

the ASF truck, the pins that lock the shoes against the springs were left in place until after calibration was completed. In the case of the Barber truck, the bolster was lifted by crane to unload the friction shoe springs to a point where the shoes could be moved manually away from the wear plates.

For the first test series, the bolster was lowered until the springs were compressed to about half their travel. The bolster was then oscillated about this position through an amplitude of ± 3/4-inch (± 0.19m) at a frequency of 0.1 Hz. Simultaneously, the bolster was displaced laterally through an amplitude of +1/4-inch (± .006m) at a frequency of 1.0 Hz.

Next, a sine sweep was performed, with the frequency gradually increasing and the amplitude decreasing. During this test considerable wear was taking place at the friction shoe-wear plate interface indicated by black powdery debris. The normal forces being measured were increasing beyond estimated levels, and gouging of the wear plates was noted. The sine sweep was terminated at 8.0 Hz as it was recognized that considerable time was required for the friction shoes to wear to service levels.

The trucks were then disassembled and the vertical surface of the friction shoes lightly ground to remove larger asperities so that the forces measured during the tests would be more representative of those occurring in service after wear-in. The first test described was repeated and the measured normal forces were in the expected range for both trucks. The sine sweep test was eliminated to prevent the localized wear at small amplitudes.

Static friction in the snubber system was checked by very slowly moving the bolster downward one inch from the centered position with the hydraulic actuators under manual control. Also, before disassembling the trucks for friction shoe grinding, the truck was forced out of tram and the increase in normal forces on the column measured.

TEST RESULTS

To reiterate, the main objective of these tests was to establish proper operation of the transducer system installed in the trucks. The objective was achieved. It was not intended to subject the truck to a full range of inputs such as would be observed on the track. The force distribution at the column is likely to change as the friction shoes wear and this phenomena should be monitored during future road tests. The following discussions are not intended to imply endorsement or critique of either truck design.

The highest recorded friction forces in the Barber S2 were approximately 1200 lb (544 kg) in both the vertical and lateral directions. The highest measured force at the lower normal transducer was about 7200 lb (3266 kg). Since the bolster was descending at this time, this must be corrected by subtracting the force due to the vertical friction moment, leaving a true lower normal force of about 6900 lb (3130 kg). The upper normal force is practically zero at this point or 275 lb (125 kg) with the correction factor. The total normal force is therefore about 7200 lb (3266 kg) and the friction coefficient is 0.167. The lateral coefficient appears to be about twice as high, but this may be due to a slight cocking of the friction shoes during the lateral bolster motion. The lateral friction forces were not as repeatable as the vertical and normal forces due more than likely to the rocking of the side frame which was supported only on two springs. Audible chatter was noticeable on the downstroke apparently due to vertical stickslip.

An attempt was made to measure the effect of forcing the truck out of tram by means of set screws at two adjacent pedestals. In this case, the normal column load is redistributed, increasing at the top and decreasing at the bottom, and the friction shoe moves downward, as expected, since the bolster is restrained by the actuators from moving upward. Because of the unsymmetrical distortion of the truck, there is some lateral sliding between bolster and side frames creating a friction force of about 200 lb (91 kg) each.

In testing the ASF Ride Control Truck, the highest vertical friction forces were 4500 lb (2041 kg) down, and 2500 lb (1134 kg) up. On the downstroke, the measured upper and lower normal forces were 7500 lb (3402 kg) and 5000 lb (2268 kg), respectively. Therefore, the corrected total normal force was 9910 lb (4495 kg) with a friction coefficient of 0.45. The lateral friction forces were about 4000 lb (1814 kg) and 2500 lb (1134 kg) on the downstroke, so the apparent lateral friction coefficient was 0.69. Vertical friction and normal forces were generally lower during the upstroke, but lateral friction forces were about the same in both directions. Therefore, the effective friction coefficient varied somewhat indicating some change in geometry which again may be due to the rocking of the side frames. There also appeared to be some rocking of the friction shoes indicated by a sharp rise in the lower normal and vertical friction forces as the direction of vertical motion reversed at the beginning of the downstroke. There was also heavy chatter implying additional energy dissipation and the distribution of the normal load was highly unsymmetrical with respect to the center of the wear plate.

It must be emphasized that the data discussed in the foregoing paragraphs are not necessarily typical of a friction snubber assembly worn in under actual operating conditions. These data are presented solely to illustrate the kind of information obtainable from the Friction Snubber Force Measurement System.

POTENTIAL UTILIZATION

Since the calibration testing of the FSFMS was not completed until March of 1977, utilization of the system in Phase I of the TDOP, for which it was designed, was not possible. To reiterate, the transducer system has been installed on two trucks commonly used in freight service in the United States. The most obvious difference between the two with respect to the snubbing force is the dependence on or independence of the load on the truck. A second difference relates to the change in snubbing friction as the truck parallelograms. The warp stiffness, and thus the friction force, is necessarily affected by the bearing width of the friction wedge which differs substantially in the two trucks. A third factor affecting snubbing friction is the frequency content and the vibrations applied to the side frame-bolster connection relating to the phenomenon of "breakout" friction.

All of the above suggest strong nonlinearities due to snubbing friction in the truck suspension system, the modeling of which is difficult and the effects of which on truck performance have not yet been quantified. Complete characterization of the general purpose freight car truck must involve the evaluation of these forces on both tangent and curved track, in both new and worn conditions. As part of Phase II TDOP, recently awarded to Wyle Laboratories, both the Barber S2 and the ASF Ride Control trucks will be re-tested under various load conditions and on several track types with the transducer equipped side frames. In addition, quasi-static friction forces will be measured at frequencies for which inertial effects are negligible by using the calibration test setup and supporting the pedestals on load cells to measure the vertical and lateral reactions transmitted from the actuators, through the snubbing components, to the test frame. The actual forces on the columns can then be derived from the known applied vertical force and the wedge angle.

Acquisition of these data will allow more detailed specification of the test conditions to be met in testing for conformance to recommended performance guidelines (developed under TDOP), will aid in the validation of mathematical simulation of truck performance, will complete the characterization of the general purpose freight car trucks, and will provide a technical baseline for the evaluation of special purpose trucks to be accomplished in Phase II TDOP.

ACKNOWLEDGEMENTS

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ACKNOWLEDGMENTS

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ERRATA

"Rail Dynamics Laboratory Requirements and Hardware Configurations"

Page 90 first sentence under Fig. 6, Vibration Test Unit should read as follows:

"The vertical excitation modules (each under independent servo control) are designed around a 60,000 lb (27,216 kg) hydraulic actuator, equipped with a 200 gpm (.0126 m^3/s) high performance servo-valve."

Page 90 first sentence of second major paragraph from bottom starting "The hydraulic flow demands ..." should be changed to read as follows:

"The hydraulic flow demands of the various excitation modules and hydrostatic bearing elements at peak excitation levels can be as high as 1000 gpm (.0631 m^3/s) @ 3,000 psi (20,684,271 N/m^2). This has been provided for via three 360 gpm (.0227 m^3/s) variable volume pumping systems each capable of delivering the rated flow at 3,000 psi (20,684,271 N/m^2)."