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## WHEEL/TRUCK TOLERANCE EXPERIMENT AT FAST

MAY 1991

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03 - Rail Vehicles & Components

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The Wheel/Truck Tolerance Experiment at FAST (Facility for Accelerated Service Testing), Transportation Test Center, Pueblo, Colorado, investigated the effect of various mechanical factors on freight truck curving performance. This was accomplished through a series of tests that included (1) wheel wear tests, (2) measurement of wheel/rail forces, (3) rolling resistance testing, and (4) curving model analysis. This project was carried out to study a number of different parameters under known controlled test conditions. Past tests have been done that covered wheel wear, for example, but with no comparable measure of wheel/rail forces or rolling resistance for the same test vehicle(s). The goal was to measure several aspects of truck behavior and thus provide more consistent understanding of wheel/rail interaction as affected by various mechanical factors.

The wheel wear portion of the study demonstrated the role the wheel profile has on the initial wear rate at FAST. Under unlubricated running, the CN Heumann wheel profile reduced flange thickness wear by 27 percent in comparison to the standard AAR 1:20 profile during the "break-in" phase of operation. The GB cylindrical profile used by the Southern Pacific Railroad, in some applications, produced only a marginal reduction in flange wear when compared to the 1:20 profile. After the profiles had worn in, there was very little difference in wear rate, and the ultimate running surface geometries were nearly identical.

The effect of axle misalignment on wheel wear was significant. Working within the available tolerances of typical three-piece trucks, wheel wear was increased by more than an order of magnitude when held in extreme levels of radial misalignment on FAST.

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Similarly, the rolling resistance of trucks with various degrees of fixed misalignment increased considerably with the level of radial misalignment. In general the resistance was proportional to the wheel wear measurements.

Wheel/rail forces were measured for standard three-piece trucks and a retrofit steering truck on curves up to 7.5 degrees. Both lateral and longitudinal forces were radically reduced by the steering truck under loaded, semiloaded, and fully loaded conditions. In most cases the reductions were on the order of 60 percent to 80 percent depending on track curvature.

Forces between the wheel and rail under various states of radial axle misalignment were measured on curves up to 12 degrees. The misalignment of axles affected the lead axle L/V (lateral over vertical) ratio most significantly at curves up to 7.5 degrees. In the 10.5- and 12-degree curves there was little difference.

Trains of increasing weight were tested for tractive effort requirements on the FAST loop. These results were then compared to calculations using the modified Davis equation. The predictions were low for the longer trains. The wind conditions were not factored into the calculations but were documented at the time of the test to be in the 5 mph to 15 mph range. This may have contributed to the apparent discrepancy between measured and calculated resistance. During the train resistance test, rail-head temperature rise was measured and was found to be a rough indicator of energy dissipation depending on train weight and track curvature.

Curving model predictions of steady state forces in curves were in good agreement, with the exception of longitudinal forces and lateral forces measured under unloaded conditions on conventional three-piece trucks. The predicted longitudinal wheel/ rail forces for the conventional trucks were significantly greater than those measured. There appeared to be better agreement between predictions and measured data on the steering truck.

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#### 1.0 INTRODUCTION

The wear of freight wheels has been the topic of many investigations in recent years. Several of those studies have been conducted under the FAST Program at the Transportation Test Center (TTC) in Pueblo, Colorado. The FAST Program's Wheel/ Truck Tolerance Test Project results contained in this report are the latest in this series. They represent a culmination of information which addresses the tracking characteristics of freight equipment in North America.

Major factors which influence the rate of wheel wear have been well documented in past tests. In general, they fall into three major categories: mechanical, metallurgical, and external environmental factors.<sup>1</sup>

The mechanical effects influence the force-creepage environment at the wheel/rail interface. Axle load, wheel and rail running surface geometry, and vehicle design are three examples of mechanical effects. The metallurgical category is predominated by the material properties of wheels and rails. These are typically characterized in terms of surface hardness, chemistry, and microstructures. Lubrication and natural causes such as rain and moisture alter wheel/rail adhesion and form the last category, external or environmental factors.

The basic premise of the Wheel/Truck Tolerance Test was that dominating mechanical aspects can be altered in a controlled fashion and studied experimentally and analytically. The major variables were mechanically varied in testing as well as in a computer model which simulates steady-state curving performance. Predictions of parameters such as wheel/rail force and rolling resistance were then compared to measured results. Additional testing of wheel/rail forces was carried out under cars of different axle loads with standard three-piece equipment and one type of steering truck. Wear testing, wheel/rail force measurements, and rolling resistance tests were also carried out on a sample of freight cars with varying axle misalignment.

This was done to understand the performance implications of the rolling attitude of the axle and also to further validate the accuracy of the model.

For certain configurations, the response measured for specific variables was compared to predicted values. The tool used in calculating the predictions was a version of the Association of American Railroads (AAR) non-linear, steady-state curving model.<sup>2</sup> This model was developed and refined over a number of years at British Rail (BR) and in the U.S. It incorporates a general theory for prediction of the steady-state forces generated by railway vehicles. Major considerations within the model include the non-linear nature of the contact geometry between wheels and rails and prediction of non-linear creep forces. Load measuring wheels at BR were employed in the process of validating the model as well as displacement probes to measure angle of attack. Generally, good agreement was achieved, and the curving theory was accepted as adequate for the vehicles tested at BR.

Testing in North America with instrumentation to measure loads and angle of attack has also been accomplished.<sup>3</sup> Lateral wheel/rail forces for curves up to 7.5 degrees were measured and compared to predicted model values with good agreement. Likewise, angle of attack data was compared and found to lie within acceptable limits.

Further work has also been done at FAST in predicting wear using the curving model.<sup>4</sup> This was possible by calculating a wear index value for three-piece equipment at a range of axle loads. These values were then correlated with measured wheel wear at the same range of axle loads. Also, the relationship between wheel material property and wear was derived.<sup>5</sup> Given this statistical model, wear predictions for a revenue test train were made. These gave reasonable results.

#### 1.1 BACKGROUND

Having developed confidence in the overall reliability of the curving model, the tests covered in this report were designed to further the understanding of curving behavior. Specifically, the wheel wear tests were intended to further expand a wear index/wear rate relationship. Wear index is an expression of theoretical energy dissipation which takes into account all of the environmental and mechanical features of a wearing system. For rail vehicles, this includes: axle load, wheel and rail surface geometry, level of lubrication, and wheel set attitude on tangent and curved track.

In this study, the driving function was axle misalignment which greatly influences wheel set attitude. Figure 1 shows the typical wheel set attitude in a curve. The angle of attack is greatly influenced by the extent of misalignment and, therefore, it also influences the wheel/rail forces, wear, and rolling resistance. Fixed misalignment of axles to varying degrees was introduced to quantify the influence on wheel wear, wheel/rail forces, and rolling resistance. The study was expanded to include wheel/rail force measurements on test cars with varying wheel load and truck design. Also included were a limited number of cars with different wheel profiles. Axle alignment was not controlled in this portion of the test. This provided additional data for comparisons with model results.

Supplemental testing was carried out to measure locomotive power as it changes with train length, speed, and track curvature. Limited measurements were simultaneously made to detect rail head temperature change for the same test variables.

Overall, the basic test theme was to provide experimental and theoretical results (gathered under controlled conditions) in order to provide additional information regarding the steadystate behavior of freight cars. Although logistically complex, the operation of the full-scale test train and a variety of eight smaller mini-consists provided reliable and accurate information for the diverse range of configurations.

## STANDARD TRUCK ATTITUDE IN CURVE



## FIGURE 1. CONVENTIONAL TRUCK WHEEL SET ATTITUDE IN CURVE.

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This test data serves as a source of comparative information which can be generalized within the limitations of the test variables. Furthermore, implementation of the curving model over a wider range of conditions will increase overall confidence in this tool. Thus, situations not tested here or those which may be impractical to test due to time and money limitations could be reliably simulated.

#### 2.0 TEST CONFIGURATIONS

There were basically three different train configurations tested within this project.

- 1) Wheel wear was gathered over a 10,000-mile period while the standard FAST train was in operation.
  - 2) Several "mini" consists were used to gather instrumented wheel set and coupler data while the full train was not in operation. These are usually made up of a locomotive, lab car, buffer car, and test vehicle.
  - 3) The locomotive energy measurements were made with the FAST train during a dedicated one-week period. This train was configured from only 4 locomotives up to a full 80 cars. A more detailed description of each of these test scenarios is covered in the following sections.

#### 2.1 WHEEL WEAR TEST

As has been the case in past FAST sponsored wear tests, it was elected to run the wheels on unlubricated rails. This was done for two major reasons. One reason was to generate large absolute differential rates in wear for the range of conditions such as axle misalignment. The other reason was that a totally dry condition is the most consistent level of adhesion that is easily maintained for long periods of time. Although snow and rain do occur, the approach was to prohibit train operation during these times. Other FAST train operating guidelines were also continued for this test. These included turning the train at regular intervals as well as switching power from one end to the other. This had the net effect of evenly distributing wear on the four wheel positions per truck in between each measurement interval. Application of tread braking was kept to an absolute minimum so that tread wear could be attributed to interaction between the wheel and rail.

The test consist was made up of 12 fully laden hopper cars of similar design. All 12 were equipped with Barber S-2-C three-piece trucks. The distribution of misaligned axles and wheel profiles is shown in Table 1. Basically, it consists of four cars containing varying degrees of radial and shear misalignment, including one of parallel alignment, with the remainder CN Heumann and AAR 1:20 to serve as baseline profile. Class U cast wheels were installed under all test cars. This was done to insure uniform material property. This wheel class also provides more rapidly measurable differences in performance with less mileage as compared to other classes of wheel. The details of this test are covered in Section 3.0.

#### 2.2 <u>MINI-CONSIST TESTS</u>

As an adjunct to the wear tests, instrumented load measuring wheel set data was also taken. This was done for a variety of vehicle configurations as listed in Table 2. The objective was to observe the measured wheel/rail loads and compare these to the curving model's predictions. The intention was to discern the trends in vertical, lateral, and longitudinal forces which result from varying axle load, speed, truck type, curvature, and axle misalignment. Although some of these same test configurations have been investigated in past tests, this experiment presented the first opportunity to measure longitudinal wheel/ rail forces.

FAST CAR NO.	WHEEL PROFILE	MISALIGNMENT
124	HEUMANN	AS IS
121	HEUMANN	AS IS
120	HEUMANN	AS IS
122	HEUMANN	AS IS
16	AAR 1:20	AS IS
46	AAR 1:20	AS IS
102	CYLINDRICAL	AS IS
100	CYLINDRICAL	AS IS
103	HEUMANN	RADIAL
118	e HEUMANN	SHEAR
104	HEUMANN	SHEAR A-END
	· · · · · ·	RADIAL B-END
119	HEUMANN	SQUARE
P		

TABLE 1. CARS FOR WHEEL WEAR TEST

TABLE 2. MINI CONSIST VEHICLE CONFIGURATIONS

TRUCK TYPE	CAR LOAD	AXLE ALIGNMENT
CONVENTIONAL	LOADED	AS IS
CONVENTIONAL	HALF CAPACITY	AS IS
CONVENTIONAL	EMPTY	AS IS
DR-1 RADIAL	LOADED	AS IS
DR-1 RADIAL	HALF CAPACITY	AS IS
DR-1 RADIAL	EMPTY	AS IS
CONVENTIONAL	LOADED	SQUARE
CONVENTIONAL	LOADED	RADIAL 0.6 MRAD
CONVENTIONAL	LOADED	RADIAL 1.3 MRAD
CONVENTIONAL	LOADED	RADIAL 4.0 MRAD

Two separate cars and a variety of trucks were used in this exercise. Although it would have been preferable to use the same vehicle for all configurations, logistics and scheduling considerations precluded it.

For the axle misalignment tests, the addition of an instrumented coupler was made. The contribution of axle misalignment to train resistance on tangent and curved track was studied on these test runs. The original FAST test plan did not incorporate the coupler in the test. It was included at a later time and was sponsored through a cooperative agreement between the AAR and the Federal Railroad Administration (FRA). These results are in Sections 4.0 and 5.0.

#### 2.3 FULL TRAIN DYNAMIC TESTING

At the conclusion of the 10,000 mile wear test, a series of test runs were made to measure train resistance and rail head temperature change under a variety of conditions. The objective was to quantify the total resistance of a full-length train as well as consists of varying lengths. The intention was then to calculate average resistance on a per ton basis and compare this to results from the mini-test coupler series and modelling data.

The means of measuring this resistance was done using instrumented locomotives which sampled power around the entire FAST track. As this was done, wayside instrumentation was installed to measure the change in temperature for the high and low rail for a sampling of curves. The purpose was to develop an easy means of estimating the energy dissipated at the wheel/ rail interface. Since a manifestation of this energy is heat, the temperature change-train resistance curve may prove promising. This is because of the assumed interrelation of train resistance, fuel consumption, energy dissipation at the wheel/rail interface, and wear of wheels and rails. Section 7.0 contains the results of these tests.

#### 2.4 <u>CURVING MODEL ANALYSIS</u>

Certain selected test data was compared to predicted data from the curving model simulations. Theoretical description of the test conditions was as close as possible to conditions actually tested. The agreement between the two sets of data is evaluated in Section 8.0 of this report.

#### 3.0 WHEEL WEAR TESTS

This discussion focuses on the wheel wear test. The data is based on 10,000 miles of service on the 4.7 mile FAST loop. Measurement and analysis of the data used techniques developed over a number of years in support of the FAST research program.

#### 3.1 WEAR MEASURING DEVICES

To document and record the wear of wheels and rail, two basic forms of instrumentation were used. One provides a scalar or one-dimensional readings at three points on the wheel surface. This device, referred to as the wheel <u>snap gage</u>, is shown in Figure 2. The three dial gages that measure flange thickness, height, and rim thickness are spring loaded and are released into a locking position for recording of the dial reading. This is done at two positions on each wheel for each one thousand miles of service. These devices are used primarily to provide a quick indication of wear. The data requires essentially no processing and can be directly plotted against accumulated mileage.

Cross sectional area loss was also measured on every test wheel. The modified Canadian National (CN) profilometer is shown in Figure 3. Applied to the same location on the test wheels, a series of dial gage readings are taken across the wheel. Over the life of the test, cross-sectional profiles are calculated and graphically produced as shown in Figure 4. From this average, area loss per one thousand miles of service is calculated.



FIGURE 2. TTC WHEEL SNAP GAGE.



FIGURE 3. TTC MODIFIED CN WHEEL PROFILOMETER.

## TYPICAL GRAPHICAL OUTPUT - CN PROFILOMETER





A more detailed discussion of these measurement tools is included in the proceedings of the 1981 FAST Engineering Conference.<sup>6</sup>

#### 3.2 WEAR ANALYSIS

This analysis of measurements emphasizes area loss due to wear of the running surface of wheels. The area loss requires an intermediate step to calculate the area of the running surface at each subsequent measurement interval. The raw CN profilometer data consists of 60 data points which are taken transversely across the wheel. After being entered into the computer, the profiles are curve-fit to provide a continuous representation of the running surface. The program does this by creating a cubic spline fit to the individual data points. The area under the wear surface described by the curve is then integrated and separated into the flange area and tread area segments.

At each measurement interval, the area loss is calculated and tabulated along with the mileage. At the conclusion of the test, wear rates are generated for the major categories using a first order regression. Each wheel and rail site wear rate is treated as a separate data point in the subsequent averaging process.

#### 3.3 WHEEL WEAR FOR DIFFERENT PROFILES

To determine whether differences in wear that are attributable to wheel profile could be detected, three different profiles\* were tested: 1) AAR 1:20, 2) GB Cylindrical, and 3) CN Heumann.

Figures 5A through 5C are plots, respectively, of the rolling radius differences. These plots clearly illustrate the dissimilarity in profile characteristics and result from mathematically superimposing the wheels on a typical FAST worn pair of rails. The slope of the line relates to the rate at which the rolling

\*Note: This study was conducted prior to adoption of the AAR-1B profile as standard and thus this profile was not tested.



FIGURE 5A. ROLLING RADIUS DIFFERENCE CURVES FOR TEST PROFILES.



FIGURE 5B. ROLLING RADIUS DIFFERENCE CURVES FOR TEST PROFILES.



FIGURE 5C. ROLLING RADIUS DIFFERENCE CURVES FOR TEST PROFILES.

radius changes for the pair of wheels on an axle as the axle set is laterally shifted across the rails. In general, the greater this rate, the better is the curving (wear) behavior of the wheels.

The AAR 1:20 is a straight line, low tread taper wheel. There is a slight change in rolling radius across the potential wheel/rail contact range. At the extreme position where flange contact occurs, two-point contact exists. It is this condition which results in excessive "break-in" wear for the wheel and rail.

The rolling radius plot for the GB cylindrical wheel presents a similar situation except that while tread contact is maintained, there is no change in rolling radius with the lateral shift of the wheel. Two-point contact also occurs at the extreme right and left hand position.

The CN Heumann wheel/rail contact plot indicates a narrow, low taper region with two, 2-point contact regimes. The overall change in rolling radius is greater than the two other wheel profiles. The characteristics illustrated in the preceding figures may vary with different rail geometry.

The flange area (FAW) and tread area (TAW) wear rates are presented in Figure 6. Contrary to the original expectations of the wear rate after 10,000 miles of service, the area wear rates are all very similar. These rates are exclusive of the "breakin" period of the first 2,000 miles. The practice of omitting the first 2,000 miles has been used in prior tests to best establish the steady-state wear rate once the wheel has conformed to the rail.

The similarity in wear led to the suspicion that all profile shapes had conformed to the "average" rail profile in FAST sometime after the 2,000 mile point. An evaluation of the wheel shape for average worn 1:20, Heumann, and GB Cylindrical profiles confirmed this thinking. In Figure 7, an overlay of all three profile types is given for wheels taken out at the end of the wear test. As shown, differences between profile types cannot be discerned because the wheels have nearly identical shapes.



IN OOOL NI OS FIGURE 6. AREA LOSS RATES FOR TEST WHEELS.



FIGURE 7. OVERLAY OF WORN WHEEL SHAPES FOR AAR 1:20, CN HEUMANN, AND GB CYLINDRICAL PROFILES. Realizing that the break-in period may better reflect the wear rate inherent to the new profile, a second series of wear rates was calculated. These flange thicknesses (FTA) wear rates were based on wear from zero to 2,000 miles. The results are presented in Figure 8. Also presented are the FTA wear rates from 2,000 to 10,000 miles.

Not only are the initial wear rates different, but the higher conicity wheel (CN) with potentially better steering characteristics wore less during the beginning stages of the test. After 2,000 miles, the differences were not distinguishable given the accuracy of the measurements and the sampling frequency. These events seemed to reinforce the theory that any reasonable range of profile shapes run in the same environment may have different initial wear rates, but they will ultimately achieve the same final geometry. A result of this process is that the wear rates become very similar.

#### 3.4 AXLE MISALIGNMENT INFLUENCE ON WHEEL WEAR

The measurement of wheel wear was also carried out on trucks with various misalignments. Figure 9 illustrates the two types of misalignment termed <u>radial</u> and <u>shear</u>. Nearly all three-piece trucks possess some extent of these two components. It was generally assumed that by misaligning the axles, wheel wear would increase. Asymmetric wear had been documented in the past on FAST wheel wear tests. Analysis with the steady-state curving model indicated that misalignment exerted a strong influence on the asymmetry and extent of flange wear. A major objective of this test was to measure wheel wear in response to known fixed radial and shear misalignment and compare these rates to the wear index calculations generated by the curving model.

The misalignments consisted of two separate regimes: radial and shear. The extent of the side-to-side difference in wheel base affects the magnitude of the radial misalignments and degrades the overall performance of the truck. A range of side-to-side differences were tested, resulting in the



FIGURE 8. INITIAL WEAR RATES OF TEST PROFILES.

# RADIAL MISALIGNMENT

## SHEAR MISALIGNMENT



FIGURE 9. COMPONENTS OF FREIGHT TRUCK AXLE MISALIGNMENT.

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misalignments given in Table 3. These values are for individual axles ( $\theta_1$  or  $\theta_2$  in Figure 9) and do not represent the included angle between both axles.

TABLE 3. RANGE OF MISALIGNMENTS INCLUDED IN WEAR TEST

NO. OF WHEELS			MISALIGNMENT		
	4		4.2 MRAD RADIAL		
	12		3.6 MRAD RADIAL		
	4	,	2.8 MRAD RADIAL		
	4	14 1	1.6 MRAD RADIAL		
•	4		1.0 MRAD RADIAL		
	12		2.5 MRAD SHEAR (APPROX.)		
	8		0.0 MRAD SQUARE (APPROX.)		

Figure 10 illustrates the type of shim used to retain the axle in a known location. Axle spacing was measured using a large vernier caliper which contacted the axle ends at the lathe center. Side-to-side differences were then used to calculate the axle misalignment.

A second misalignment type was also tested which involved shimming a three-piece truck in shear. With this misalignment, the axles and side frame remain parallel but are skewed. Assuming the truck sides remain square relative to the bolster, approximate values of shear misalignment can be derived. The values of  $\theta_1$  and  $\theta_2$  are equivalent and are based upon the longitudinal displacement of wheels on one side of the truck with respect to the wheels on the other side. Steel plates welded to the side frame were also used here.



FIGURE 10. SHIMS WELDED INTO PLACE ON SIDE FRAME.

The last use of the plates to shim the axles included a square truck setup which was intended to prevent the presence of radial or shear-type misalignment. The side-to-side differences in this case were less than 0.020 inches or 0.1 milliradian.

As previously mentioned, train operation was varied so as to distribute the wear potential on all four wheels on a track.

Figure 11 shows the trend in wear with radial axle misalignment. The rationale for the sign convention is based on the orientation of the misalignment as depicted in Figure 12. The side with the increased wheel base is negative. The opposite side is positive. The basic shape of the trends in area loss shows a definite increase in wear with misalignment. This is particularly true for misalignments greater than 3.0 milliradians or less than -3.0 milliradians. However, there is a sizeable difference in the magnitude of wear between positively and negatively misaligned wheels. This is due to the fact that the wheels on the positive side of the truck take on a larger angle of attach when in flange contact in curves.

Shear misalignment was also tested. The intention was to see if uneven wear was produced by this condition which utilized the maximum clearance available for interchange-approved three-piece equipment.

A schematic of a shear misaligned truck in a right- hand and left-hand curve is given in Figure 13. The bias introduced by the shims for this condition resulted in a potential misalignment equal to the maximum radial value of 3.0 milliradians.

The shear misalignment wear rates, relative to the average wear generated by unconstrained axles with Heumann profiles in the baseline control group, are shown in Table 4. The comparison of flange area wear shows no significant difference due to the shear misalignment. The tread area loss, however, is higher. Therefore, the <u>total wear</u> has been influenced to the extent that the tread loss for the shear misaligned vehicles was higher.



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FIGURE 12. SIGN CONVENTION FOR AXLE MISALIGNMENT.







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PROFILE TYPE	FLANGE AREA	TREAD AREA	TOTAL AREA	AVERAGE SIDE-TO-SIDE DIFFERENCE		
AAR 1:20	0.028	0.023	0.053	0.015		
CYLINDRICAL	0.030	0.016	0.046	0.013		
HEUMANN	0.028	0.023	0.055	0.013		
HEUMANN W/SHEAR	0.029	0.031	0.062	0.015		
HEUMANN W/RADIAL	0.129	0.025	0.155	0.137		

(AREA LOSS SQ. IN./1000 MI.)

The result of shear misalignment was expected to be most apparent in terms of diagonal wear across the truck. This is due to the relatively high angle of attack that may result for two of the four wheels. By calculating typical side-to-side differences in total wheel wear on a per axle basis, comparisons to the data for different profiles and the radial misalignment data can be made. The slight increase in Heumann with shear compared to those Heumann with no truck alteration is insignificant. This implies that the shear misalignment is overcome by the clearances between bolster and sideframe, and the axles achieve a more square orientation.

#### 4.0 WHEEL/RAIL FORCES

To ascertain the curving properties of a range of vehicle configurations, the forces between the wheel and rail should be known. Load measuring wheel sets were employed to determine the absolute trends for changes in truck design, axle load, and lading. There was a total of seven channels of data collected for each wheel set. Lateral, longitudinal, and vertical forces, were recorded. Instrumented wheel sets were used in both lead and trail axle positions.

### 4.1 INSTRUMENTED WHEEL SETS

Two instrumented wheel sets on the same test truck were used for a range of vehicle configurations. The wheels provide force information via strain gages installed on the wheel plate. There are three strain gages for resolving vertical force and two for lateral force. There is also a wheel position gage. The signal from this gage provides an index of rotation used in the processing of the various strain gage outputs. The signals are sinusoidal in nature, but with the processing done in real time by a small microprocessor, rectified continuous vertical and lateral response is generated.

This basic approach has been used at the TTC in previous tests.<sup>7</sup> The recent development of microcomputers has now allowed for real time processing. In the past, raw analog data was transmitted on tape to IITRI for processing and statistical analysis.

An additional set of strain gages was added to the original design of these FAST wheel sets. They were located on the axle, 180 degrees apart, in order to measure axle torsion which was translated into longitudinal force at the wheel/rail interface. This element had been absent in previous tests where the objective was to derive a comprehensive understanding of curving mechanics and wheel wear.

### 4.2 ANALYSIS OF WHEEL FORCES

A major premise of the wear index philosophy for this test program is the assumption of steady-state behavior in curves. The nearly continuous measurement of wheel force from the instrumented wheel sets is intended to create a large sample which is then averaged to characterize the wheel/rail force environment. For example, the average lateral force through a curve at a constant known speed is a statistic of primary importance. Similarly, the mean L/V ratio can be observed for a given car at a range of speeds. These values can then be compared to the steady-state predictions from the curving model.

## 4.3 TRUCK DESIGN AFFECT ON WHEEL/RAIL FORCES

In addition to the conventional three-piece design, a retrofit steering truck was used. The Dresser DR-1 configuration was chosen because of its documented curving characteristics and reduced wheel wear. The results of curving tests have been covered extensively in prior FAST reports.<sup>7</sup> A simple line drawing of the DR-1 truck is given in Figure 14. The test data discussed in this report are unique in that the longitudinal wheel/rail forces are addressed for the first time in a FAST test.

Loaded hopper car data for DR-1 and conventional trucks in a 5 degree curve are shown in Figure 15. Lateral and longitudinal wheel/rail forces at balance speed are given in this example. It illustrates the difference in lateral force for the two designs. All four-wheel forces are given in this illustration. The dominant role that the lead axle plays in curve negotiation is clearly demonstrated for the conventional truck as well as the measured force reduction for a steering truck. In this case, the outside lead axle lateral force is reduced by 80 percent for the DR-1 equipped car. On the standard three-piece truck, the lead axle lateral force is approximately 65 percent higher than the trailing axle. This general relationship is true for conventional equipment in curves ranging from 2 to 10 degrees.

ARRANGEMENT OF RETROFIT RADIAL TRUCK



FIGURE

QF

DR-1

TRUCK

DIAGRAM 31





**Steering Arm Connector Post** 

**Steering Arm Center Connection** 

-Steering Arm Attached To Adapter



LATERAL DR-1 ANI AND AND LONGITUDINAL FORCE CONVENTIONAL TRUCK. DISTRIBUTION

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The longitudinal force between the wheel and rail for the two vehicles is also presented in Figure 15. A reduction similar to the lateral force example is illustrated here. These force reductions are somewhat proportional to the reductions in wheel wear and rolling resistance documented in previous tests. Figures 16 and 17 show the lead outside wheel lateral forces and the lead axle longitudinal forces for the full range of curves tested for different conventional and radial trucks. There is a progressive difference in lateral wheel/force up to 6,000 pounds for the 7.5 degree curve. Likewise, the longitudinal differential between the two test trucks is nearly 4,000 pounds for curvature of 3 degrees and higher.

# 4.4 AXLE LOAD AFFECT

Wheel set forces under three different axle loads were also measured over several curves. Figure 18 presents the lateral force change for the lead axle at balance speed for a 1.5-degree curve, and a 5-degree curve on the DR-1 and conventional truck. The conventional truck was tested under empty conditions (63,000 pounds), half loaded (163,000 pounds), and fully loaded (263,000 pounds). For conventional equipment on a 5-degree curve, the linear trend indicates an average increase in lateral force of 211 pounds per 1,000-pound increase in vertical wheel load. For the steering truck on a 5-degree curve, the linear trend indicates a 15-pound increase in lateral force for every 1,000-pound increase in wheel load. The 1.5-degree data show a similar relationship for both trucks. This is most likely due to the fact that the conventional truck with a Heumann profile is not in flange contact in a 1.5-degree curve.



LATERAL FORCE (K-LBS)

FIGURE 16. LATERAL FORCE IN CURVES - FULLY LOADED DR-1 AND CONVENTIONAL TRUCK.



LONGITUDINAL FORCE (K-LBS)

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FIGURE 17. LONGITUDINAL FORCE IN CURVES - FULLY LOADED DR-1 AND CONVENTIONAL TRUCK.



FIGURE 18. CHANGE IN LATERAL FORCE WITH VARIED WHEEL LOAD.

The data shown in Figure 19 is the longitudinal force distribution for the same range of axle loads for the test trucks. The absolute magnitudes are less with a corresponding reduction in the rate at which they change with wheel load. For the radial design, the longitudinal force measurement shows no significant increase. The conventional truck produced a 125 pound increase for every 1,000 pound increment in wheel load on the 5-degree curve. The 1.5-degree curve generated little increase over the range of axle loads.

### 4.5 AXLE ALIGNMENT INFLUENCE

The major variable was the level of axle misalignment. This was applied on four separate test cars using a steel plate to shim the axles into location between the side frame yoke and axle bearing. This had the effect of forcing the axles into a variety of alignments for the instrumented wheel set tests.

Recognizing the potential effect that axle alignment has on wheel/rail loads, a number of radial misalignments were tested under a loaded hopper car. Figure 20 contains the lateral force response for the lead outside wheel on tangent track as well as curves ranging up to 12 degrees. The five levels of misalignment were 0.0, 0.6, 1.3, and 4.0 milliradians, as well as one unshimmed "as is" truck. The lateral forces are similar for all trucks except the 4.0 milliradians misalignment. Here the forces are significantly higher, up to 10 degrees of curvature.

The significance of misalignment on lateral force is evident in the L/V trend given in Figure 21. As was the case for the previous wheel/rail load data, this is the estimate for balance speed conditions. Traditional L/V limits for incipient flange climb are arrived at using a formula developed by Nadal.<sup>8</sup> It considers the coefficient of friction and the maximum contact angle between the wheel and rail. The expression is given below.

L/V(lim) = (tan(theta)-mu)/(l+mu\*tan(theta))



LONGITUDINAL FORCE (K-LBS)



FIGURE 20. LATERAL FORCE WITH CURVATURE FOR MISALIGNED TRUCKS.





CURVATURE (DEGREES)

FIGURE 21. L/V WITH CURVATURE FOR MISALIGNED TRUCKS.







For the conditions of this test, the limiting L/V was calculated as 0.94 assuming a maximum contact angle of 70 degrees and a coefficient of 0.5. Although there is a profound influence on the steady-state L/V when the axles are misaligned, the critical value is still significantly higher than any average L/V observed. This is not to say that the potential for derailment is not significantly increased by misalignment, since there is a smaller margin of safety for misaligned trucks. Dynamic input such as track irregularity or an externally applied force that might occur in buff or draft could easily result in a critically high L/V.

An aspect of the test data that seemed a bit unusual was the magnitude of the differences in lateral force for the lead axle. This net lateral force is shown in Figure 22. The trucks with low misalignment values effectively show no major lateral force. The other two misaligned cases, however, are considerably greater.

The primary factor related to a high net lateral force on a single axle is a significant shear stiffness. Prior tests have indicated that for a typical three-piece truck, the shear stiffness under dynamic conditions is essentially zero. This is because the bearing adapter-side frame, and bolster-side frame stiffness is usually low under running conditions. A plausible explanation for the difference seen in this data is that much of the clearance has been taken up by the metal shims which hold the axles in their predetermined position. This could have the effect of greatly increasing the shear stiffness once the bearing is in contact with the shim.

### 5.0 MULTIPLE CAR RESISTANCE AND RAIL TEMPERATURE

At the time that the Wear Index test plan was being prepared, an understanding of wheel wear, resistance, and energy was just starting to be quantified through new testing techniques. Holding constant other factors such as aerodynamics and bearing resistance, a measurement of the relationship between curvature and train length with resistance was attempted.



FIGURE 22. NET LATERAL FORCE WITH CURVATURE FOR MISALIGNED TRUCKS.

This was done by examining the total train resistance at the locomotive for a range of operating situations. The technology that allowed for the accurate measurement resistance was developed jointly between the Burlington Northern Railroad and the FRA. This involved the calibration of a single locomotive on the TTC Roll Dynamic Unit (RDU) followed by a series of controlled short consist tests to evaluate the precision of the technique.<sup>9</sup>

The basic system involves the onboard acquisition of voltage and amperage at the traction motor. The voltage is measured by voltage dividers across the armature leads. Amperage is collected via calibrated shunts in the armature leads. Dividing the armature power by speed provides a measurement of resistance (or tractive effort) for the entire train including the locomotive.

Having calibrated the unit on the RDU, the efficiency of the traction motors was used in calculating the actual tractive effort and power required to convey a given consist at known speeds over known track. Compensation is made for the track grade and train handling, and the final product is an accurate estimate of train resistance.

In the process of AAR Energy Program revenue testing and other full train tests, uncalibrated locomotives were also used with the same approach. This was done under the assumption that the traction motor efficiency worst case would only vary from 2 percent to 4 percent with a corresponding influence on the accuracy of the data. Also, the utility of most tests was to make comparisons of condition "A" versus condition "B" and that required only relative accuracy. The additional cost of calibrating additional locomotives was also prohibitive.

For a small series of test runs, the FAST locomotives were instrumented to measure power over a range of controlled conditions. The onboard computer system shown in Figure 23 was used to provide rapid data reduction and monitoring of the instrumentation package.

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FIGURE 23. ONBOARD HEWLETT-PACKARD 9826 DATA AQUISITION SYSTEM.

As part of the energy testing with regard to lubrication effectiveness and its affect on rail wear, temperature relationships between curvature, state of lubrication, and wear were being developed (temperature data was collected) during some of the full train resistance testing.

The objective was to measure the rise in field side rail head temperature during a train pass from its ambient state. A thermocouple was attached to the rail at a number of locations for this test series. The relationship between this measurement and the average resistance through the test zone was then identified.

## 5.2 ANALYSIS OF RESISTANCE DATA - LOCOMOTIVE

Conversion to engineering units and real-time data processing is done with a Hewlett Packard 9826 desktop computer. An example of real-time statistics is given in Table 5.

Correction for train acceleration is done on a one second basis in order to remove extraneous forces due to train handling effects. Off-board analysis included the removal of track grade influence. This was done by creating a grade map on disk of the FAST track and by identifying known locations within the resistance data. The force contribution due grade was subtracted from the data. The tabulation of test zone and test trains are then used to calculate trends of overall resistance. This technique has been utilized extensively in several full train tests.<sup>9</sup>

### 5.3 TRAIN RESISTANCE

Train resistance measurements were taken with three consists as shown in Table 6. The instrumented locomotives were not specifically calibrated for this test and, therefore, an average efficiency was used in the calculations.

	TIME	00:22:35.2	HOURS
	DISTANCE	3.67	MILES
	POWER/MILE	12.41	KILOWATT-HOUR/MILE
	ACCUMULATED POWER	24.67	KILOWATT-HOUR
·	SPEED	36.00	MILES PER HOUR
	RAW TRACTIVE EFFORT	32.00	K-LBS
	TRACTIVE EFFORT W/ACCELERATION REMOVED	24.56	K-LBS
•	TRACTIVE EFFORT W/ACCELERATION & GRADE REMOVED	16.24	K-LBS
	· · ·		

(WHERE APPROPRIATE - MEAN, MINIMUM, AND MAXIMUM ARE AVAILABLE)

TABLE 6. TRAIN CONSIST FOR RESISTANCE MEASUREMENTS

	CONSIST NO.	NO. OF CARS	TRAIN WEIGHT (TONS)
. <u>.</u>	l	4 LOCOS	500
	2	4 LOCOS + 20 CARS	3661
	3	4 LOCOS + 41 CARS	6184
	4	4 LOCOS + 60 CARS	8529
		·	· · · · · · · · · · · · · · · · · · ·

For most test consists, the length of the train exceeded the lengths of the standard test zones on the FAST track. These zones are typically dictated by the length of the curve or tangent section. For this reason, the entire train is never in the body of a zone sufficiently long enough to capture average resistance measurement. The most valid approach was to calculate the tractive effort of the locomotives over the entire FAST track. By using the data for the entire test oval, corrections for grade were not necessary since these cancel out for a closed loop operation.

Figure 24 demonstrates the trend of tractive effort with increasing trailing tonnage for test trains run at 35 mph. The data consists of measured and theoretical predictions of tractive effort. In relative terms, there is a distinct tractive effort increase with added tonnage. It equates to a 6-pound increase for every additional ton. In comparing the observed train resistance to that predicted from the widely used Davis equation, there appears to be greater resistance for the FAST test consists. The formulation from Davis is given in the following equation:

 $R = A + BV + CV^2$ 

Where

- A = Speed Independent Resistance due to Bearing and Wheel/Rail Interaction Drag
- B = Linearly Speed Dependent Resistance
- C = Aerodynamic Resistance

R = Total Train Resistance

Using the most recent derivative of the Davis equation, the trend in Figure 24 has been developed for the FAST scenario. An average additional 4.8 pounds of resistance was calculated for every additional train ton.

Because the wind conditions were severe for the test, the actual validity of this comparison is questionable. Although the net effect of an unidirectional wind on a closed loop train





FIGURE 24. CHANGE IN TRACTIVE EFFORT WITH INCREASING TRAIN TONNAGE.

operation could be argued to be negligible, recent energy tests at the TTC have shown cross wind influence to be much higher than once thought. In order to have absolute confidence in the resistance data, totally calm conditions would have been necessary.

# 5.4 RAIL TEMPERATURE AND TRAIN ENERGY

At the conclusion of the ten thousand mile wheel wear test, a series of train runs was completed to measure locomotive energy on the full FAST track with various consists. Test trains were made up of 60, 41, and 21 freight cars, in addition to one consist with locomotives only. As this data was recorded, wayside data of the change in rail temperature was also recorded.

This testing was run just prior to the dedication of the High Tonnage Loop (HTL), and only a limited period of time was available. After the dedication, the priority was to run the FAST train exclusively for the HTL experiments. High wind conditions and rain reduced the total number of useful runs that could be made in the available time frame.

#### 5.5 RAIL TEMPERATURE DATA

The rail temperature data collected (on cassette tape) with a data logger was downloaded to the TTC VAX 780 computer. Time and temperature plots and tables were then generated. By comparing the simultaneous events of train resistance with the temperature data, the relative influence of train weight, speed, and resistance was investigated.

## 5.6 RAIL TEMPERATURE INCREASE

Wear index values produced by the curving model are discussed in Section 7.0. They are estimates of the mechanical work done at the wheel/rail interface. A by-product of this work is heat which can be measured empirically as the change in the temperature of the rail as a train passes a specific site.<sup>10</sup>

Changes in the rolling resistance of the train is also an indication of the work done at the wheel/rail interface-assuming constant conditions such as aerodynamics, etc. The following section brings these two measurements together to present the relationship for some limited testing with trains of varying length.

During the train resistance tests, wayside measurement of rail temperature data was recorded (Figure 25). This figure portrays the heating and cooling cycle for the rail head for several train passes. Lubrication studies at FAST have shown the rail head temperature change to be a valid indicator of lubrication effectiveness. The attempt here was to investigate the use of this technique in determining a simple work index for wheel/rail interaction. This is analogous to the theoretical wear index described in the basic ground portion of this report.

Trains of three different lengths, passing through three different curves, were studied. The test runs correspond to those described in the train resistance section of this report. Problems with data acquisition and weather reduced the number of potential train passes that were used. The data analyzed here is still very useful in exploring the method of rail temperature rise as an indication of work done at the wheel/rail interface.

Since both increased train tonnage and curvature will result in a greater change in rail temperature, a simple index is proposed to incorporate both influences. This involves taking the product of curvature and train weight. The results, as plotted against temperature change, is shown in Figure 26. A first order curve fit results in a predicted change in rail temperature of 0.2 degrees Fahrenheit for every additional 1,000 tons of trailing weight.

The scatter of data about the regression line is fairly significant and can be attributed to a number of factors. The first consideration is the ambient condition at the time the data was collected. Exposure to direct sunlight or cooling wind may alter the data. Also, the profile of the rail should be considered since it affects the efficiency with which the wheels track at the very point that data is being taken.



FIGURE 25. RAILHEAD TEMPERATURE TIME HISTORY.

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FIGURE 26. TRAIN AND CURVE WEIGHTED TEMPERATURE INDEX.

The objective of this effort was to explore the use of rail temperature data outside the realm of lubrication investigation. The results are promising since a significant trend was observed for both changes in consist weight and curvature. It appears that the results may allow for generalization of this approach to other areas where the mechanical changes to vehicle tracking are of interest.

### 6.0 SINGLE CAR RESISTANCE

Single car resistance tests were also carried out on a broad range of vehicle configurations. The rolling resistance was measured using a load cell coupler arrangement shown in Figure 27. The technique of continuously measuring the coupler force using a simple yoke arrangement and load cell had many advantages over previous approaches. This included the problems associated with the low sensitivity of a strain gaged freight car coupler and the problems due to environmental conditions such as wet strain gages, etc.

The test consist was made up of a locomotive, the AAR-100 test car, a buffer car, and the hopper car configured for test. The buffer car was always loaded to provide very close ( $\pm$  0.5 inches) agreement in coupler height. The "zeroing" of the coupler was done by setting the test consist on level track with the draft gear between the test vehicle and buffer in a neutral or drift state. This procedure later proved to be insufficient, and present test techniques call for the removal of the load cell at the beginning of each major test series to assure a true zero calibration.

The test plan called for running each test configuration forward and backward through the test zone. This was done primarily to assess the grade effect in the test zone. By calculating one-half the difference in the net resistance in both directions at the same speed, the grade effect was removed. The mechanical offset as a result of a false zero was also eliminated.



# FIGURE 27. SINGLE CAR LOAD CELL COUPLER.

The load cell was scaled to  $\pm$  37,500 pounds with a resolvable load of 36 pounds. The sample rate was 64 samples per second. Test speeds through the test zones were held to within  $\pm$  .25 miles per hour on average.

# 6.1 <u>ANALYSIS OF RESISTANCE DATA - SINGLE CAR COUPLER</u>

Digitizing the load cell data was carried out on the AAR-100 test car. Statistical tables (that included mean force and speed) were generated after each test run. Analysis using a personal computer and Lotus 1-2-3 software involved the grade and zero offset corrections. The specifics of these compensations were covered in the prior instrumentation section. The plotting of resistance with car weight and/or truck type at a variety of speeds was then possible.

### 6.2 ROLLING RESISTANCE RESULTS

The measurement of railcar rolling resistance has been conducted in a variety of ways for several years. The instrumented coupler designed for the tests at FAST allowed for very accurate resistance measurement within relatively short test zones (500 feet) with results available within one or two minutes of the conclusion of the test run. The value of this data is to understand the implication of axle alignment as it changes fuel consumption and wheel and rail wear. The incremental resistance with misaligned axles is assumed to be a result of the energy dissipated at the wheel/rail interface. The source of the energy is the diesel fuel used by the locomotive, and the product of the work done is assumed to be wheel and rail wear.

#### 6.3 CHANGE IN RESISTANCE WITH MISALIGNMENT

The plot in Figure 28 shows the change in resistance with curvature for all three misalignments. Each discrete point is the average coupler force for a single test configuration and



FIGURE 28. RESISTANCE WITH CURVATURE AND AXLE MISALIGNMENT.

curve. The test runs were conducted at speeds of 5, 10, and 15 mph corrected for offset due to track grade and averaged. Low speed runs were used to reduce to the greatest extent the dynamic losses which are speed dependent. The plotted lines are the result of second order equations fit for the three respective configurations. The sign convention used in the analysis is positive for right-hand track curvature and negative for left-hand track curvature.

A main feature of the plot is the offset for the estimated minimum resistance for the three levels of misalignment. The 4.0 milliradian misalignment has a minimum at minus 5 degrees. The 1.3 milliradian minima occurred at minus 1.0 degrees, and the 0.6 milliradian case was the least on tangent track. At these respective points, the trucks achieve an attitude most favorable to the curvature in which it is running. For curves greater or less than the minimum point, the trucks apparently are either oversteering or understeering.

The implication of the measured rolling resistance is the potential fuel saving that can be attained with better steering vehicles. This is true for radial type trucks which roll more efficiently in curves and presumably in tangent track. However, considering the extent of tangent track for typical U.S. railroads, a simple "square" truck would generate considerable savings.

7.0 CURVING MODEL APPLICATION

Use of the curving model for the range of test variables required simulating as closely as possible the conditions under which test data was collected. The model was developed to allow for the variation of all parameters that influence the steadystate performance of rail vehicles.
## 7.1 INPUT AND OUTPUT DATA

Values produced by the model address, in general:

- O WHEEL AND TRUCK DISPLACEMENT
- O WHEEL/RAIL GEOMETRY FOR THE DISPLACEMENT PREDICTION
- O WHEEL/RAIL FORCES
- O WEAR INDEX CALCULATION

A breakdown of these categories is given in Table 7. Input to the model falls into two basic categories. The first is a table containing wheel/rail contact geometry. This contains six predicted contact geometry values for both left and right wheels for several positions of the wheel set, as it may be shifted laterally across the rails. These calculated values are based on measured wheels and rails. This input table requires an accurate record of the relative orientation of the wheels on an axle (or a pair of rails) in addition to the running surface shape. Presently, a pair of profilometers based on a British Rail design are used for this. These are shown in Figures 29 and 30. Table 8 is an example of a contact geometry table for a typical test wheel on FAST rails.

The second category of input is made up of the parameters that describe the truck and car body. These are shown in Table 9 for a standard three-piece truck under a loaded hopper car. In general, they cover the masses, stiffnesses, and dimensions for a vehicle. Masses and dimensions are easily derived from design information. Stiffnesses are based on the force/ displacement characteristics measured in the laboratory. Shown in Figure 31 is a primary longitudinal stiffness characteristic as measured by the longitudinal force-longitudinal displacement curve. It is for a steering truck which has rubber shear pads to allow yaw in curves.

# TABLE 7. CURVING MODEL OUTPUT DATA

# TRUCK AND AXLE GEOMETRY

-- LATERAL SHIFT OF LEAD AND TRAIL AXLE

- -- ANGLE OF ATTACK OF LEAD AND TRAIL AXLE
- -- LATERAL DISPLACEMENT OF LEFT AND RIGHT SIDE FRAME
- LONGITUDINAL DISPLACEMENT OF LEFT AND RIGHT SIDE FRAME
- -- YAW ROTATION OF LEFT AND RIGHT SIDE FRAME
- -- YAW AND LATERAL ROTATION OF BOLSTER

# WHEEL/RAIL CONTACT GEOMETRY DATA

-- ROLLING RADIUS DIFFERENCE RIGHT-TO-LEFT WHEEL

-- ANGLE OF CONTACT PATCH BETWEEN WHEEL AND RAIL

# WHEEL/RAIL FORCES

-- LATERAL AND LONGITUDINAL FORCES PER CONTACT PATCH -- NORMAL FORCES AT EACH CONTACT PATCH -- DRAG OF WHEEL SET

### WEAR INDEX CALCULATION

-- LATERAL WEAR INDEX FOR EACH CONTACT PATCH -- LONGITUDINAL WEAR INDEX FOR EACH CONTACT PATCH

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FIGURE 29. BR DESIGN WHEEL PROFILOMETER.



FIGURE 30. BR DESIGN RAIL PROFILOMETER.

## TABLE 8. CONTACT GEOMETRY TABLE

SHIFT	THETA	DRL	DRR	DELTL	DELTR	AREAL	AREAR	A/BL	A/BR	RHOL	RHOR
-11.704	11.565	23.220	-1.305	52.27	3.44	18.2	87.8	19.61	1.03	0.029	0.89
-10.404	8.606	19.702	-1.212	59,90	3.45	21.5	92.4	19.04	0.93	0.038	0.96
- 9.103	3.935	12.498	-0.951	55.78	3.42	29.4	90.0	12.12	0.98	0.073	0.9
- 7.803	1.857	8.859	-0.783	48.01	3.38	29.7	89.9	8.93	0.98	0.090	0.9
- 6.502	0.599	2.341	-0.662	8,43	3.36	79.1	92.4	1.25	0.93	0.750	0.9
- 5.202	0.380	-0.490	-0.585	3.21	3.35	79.1	95.9	1.26	0.86	0.757	1.0
- 3.901	0.278	-0.577	-0.495	3.21	· 3.34	76.9	99.5	1.33	0.80	0.721	1.0
- 2.601	0.176	-0.650	-0.419	3.22	3.34	76.1	105.2	1.36	0.71	0.708	1.1
- 1.300	0.074	-0.723	-0.328	3.23	3,36	75.4	112.4	1.38	0.62	0.696	1.2
0.000	- 0.251	-0.797	2.504	3.24	5.32	75.5	37.8	1.38	4.32	0.698	0.1
1.300	- 0.506	-0.885	2.507	3.26	5.31	74.9	48.2	1.40	2.98	0.689	0.2
2.601	- 0.716	-0.959	2.510	3.27	5.29	75.3	74.5	1.38	1.43	0.696	0.6
5.202	- 2.645	-1.290	8.856	3.06	49.90	66.3	27.1	1.73	11.74	0.554	0.0
6.502	- 4.902	-1.415	12.348	3.13	56.80	64.8	27.0	1.80	14.29	0.532	0.0
7.803	- 8.778	-1.588	17.322	3.32	62.70	63.2	22.3	1.88	19.30	0.507	0.0
	-12.533	-1.773	22,880	3,53	53.41	62.7	18.0	1.91	19.99	0.499	0.0

#### WHEEL SET CR12203A MEASURED 01/11/85 ON RAILS - AVG. FIVE MEASURED 08/14/87

DRL,R ROLLING RADIUS LEFT AND RIGHT WHEEL

DELTL,R - CONTACT ANGLE LEFT AND RIGHT WHEEL

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- -- WHEEL BASE OF TRUCK
- -- HEIGHT OF C.G. ABOVE RAIL
- -- SEMI-GAGE OF CONTACT PATCH
- -- WHEEL RADIUS
- -- TOTAL VEHICLE MASS
- -- SPACING OF JOURNALS
- -- LATERAL STIFFNESS OF PRIMARY SUSPENSION
- -- YAW STIFFNESS OF WHEEL SET W/RESPECT TO TRUCK
- -- SHEAR STIFFNESS OF TRUCK
- -- AXLE BENDING STIFFNESS
- -- LATERAL STIFFNESS BETWEEN SIDE FRAME AND BOLSTER
- -- LONGITUDINAL STIFFNESS BETWEEN SIDE FRAME AND BOLSTER
- -- YAW STIFFNESS BETWEEN SIDE FRAME AND BOLSTER
- -- TRACK CURVATURE
- -- CANT DEFICIENCY OF TRACK
- -- COEFFICIENT OF FRICTION FOR CONTACT PATCHES
- -- ROLL STIFFNESS OF TRUCK AND CAR BODY
- -- TRACK GAGE
- -- YAW TORQUE OF TRUCK
- -- RADIAL AND LATERAL AXLE MISALIGNMENT

FIGURE 31. PRIMARY LONGITUDINAL STIFFNESS PLOT.







DISPLACEMENT (IN)

This type of characteristic is measured in a laboratory where the vehicle is loaded to the appropriate weight and the axles are floated on air bearings. The air bearings eliminate nearly all friction that normally exists for the axle when it is sitting on normal track.

Predictions from the curving model centered on the calculation of forces, drag, and wear for a range of configurations. Table 10 contains these results for a standard coal car, fully loaded in a 5-degree curve at balance speed. These forces represent the steady-state for the conditions stated in the input files. Drag, or rolling resistance, is for one truck and a half car body. The wear index is representative of the energy dissipated at the wheel/rail interface.

The predicted results from the curving model are presented within the following sections. For several test conditions, direct comparisons are made over a range of track curvature, car weight, etc. In all cases, the predictions are for a coefficient of friction of 0.5.

## 7.2 MODEL PREDICTIONS OF WEAR INDEX

Curving model calculations of wear index have been carried out for previous FAST experiments. From these calculations, an approximate relationship between wear index and wheel wear was established. In this original effort, the driving variable was varying axle load. In the case of the Wear Index Wheel Wear Experiment, the most powerful variable was axle misalignment.

The relationship in terms of wear index with wear based on cycles instead of miles is shown in Figure 32. Given are both the newly derived relationship and the association between the same variables from the earlier FAST test. The difference in the slopes was unexpected, based on the major considerations. It is due to the unexplained increase in wheel wear rates for the Truck Tolerance Experiment. Possible explanations for this increase include changes in the basic rail transverse profile.

# TABLE 10. CURVING MODEL RESULTS

(TYPICAL LOADED HOPPER CAR IN A 5° CURVE)

FLANGE LATERAL FORCE LEAD AXLE - OUTSIDE WHEEL	14.4	K-LBS
TREAD LATERAL FORCE LEAD AXLE - OUTSIDE WHEEL	-3.3	K-LBS
TREAD LATERAL FORCE LEAD AXLE - INSIDE WHEEL	11.5	K-LBS
TREAD LATERAL FORCE TRAIL AXLE - OUTSIDE WHEEL	0.7	K-LBS
TREAD LATERAL FORCE TRAIL AXLE - INSIDE WHEEL	0.2	K-LBS
LEAD AXLE DRAG	0.3	K-LBS
TRAIL AXLE DRAG	0.04	K-LBS
WEAR INDEX FOR LEADING OUTSIDE WHEEL	142.3	LBS
WEAR INDEX FOR LEADING INSIDE WHEEL	75.3	LBS
WEAR INDEX FOR TRAILING OUTSIDE WHEEL	22.7	LBS
WEAR INDEX FOR TRAILING INSIDE WHEEL	18.7	LBS



WEAR INDEX

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It was during the truck tolerance testing that a rail grinding experiment was being carried out. Also, the cleanliness of the rail may have been more vigorously enforced during the latter testing. Although the Variable Axle Load Test, which was used in the initial wear-index/wear-rate correlation, was run under dry conditions; however, there was no direct means of quantifying the level of adhesion.

# 7.3 CURVING MODEL PREDICTIONS - WHEEL/RAIL FORCES

Using measured transverse profiles for the instrumented wheels and the average FAST rail, predictions of forces have been carried out. The predicted lateral and longitudinal wheel/rail forces for the different truck and axle load configurations tested are presented. This includes retrofit radial trucks, different axle loads, and a range of axle misalignments.

Shown in Figure 33 are the observed and predicted lateral forces for fully loaded conventional and retrofit radial trucks over a range of track curvature. The level of agreement is quite high between the predicted forces and the observed forces for curvatures of 5 degrees and less. The plot of longitudinal forces for the same conditions is presented in Figure 34. The DR-1 prediction is good. The conventional truck predictions are considerably higher than the observed levels of longitudinal forces.

The same comparisons for lateral and longitudinal force are given in Figures 35 and 36 for half capacity (50 tons), and in Figures 37 and 38 for unloaded conditions. The same general trends exist in the extent to which the modelled results align themselves with the test data.

For the DR-1 results, the only major discrepancy occurs in sharp curves because the model results indicate flange contact for the high rail outside wheel starting at approximately 6.5 degrees. The test results, which exist for curves up to 7.5 degrees, suggest that no flange contact has occurred.



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FIGURE 34.

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The predicted and measured data for the conventional truck are not in close agreement. The predicted lateral forces for all three load conditions are greater than the measured levels. The difference is most pronounced for the empty case. It appears that flange contact is not occurring for the measured cases as early in terms of curvature as is apparently the situation for the model predictions. This produces a lateral offset between the two sets of data, most noticeable for the lighter cases but still present for the fully loaded case.

A probable explanation of this is that the wheel/rail adhesion level was considerably less than the assumed value of 0.5. The basis for using this level is based on previous agreement between model and experimental data for dry track using the 0.5 value. The test results indicate that the coefficient of friction was probably in the range of 0.25 to 0.3. More recent testing of adhesion using a tribometer which directly measures the coefficient of friction indicates that seemingly dry rail can have values as low as 0.25. This may also be a contributing factor for the differences in model and test data for the DR-1 truck.

In terms of the increase in wheel/rail force with curvature and misalignment, the differences are more extreme at lower curvature for fully loaded cars. Figures 39, 40, and 41 contain the measured and theoretical results which plot lateral force with misalignment for three levels of curvature. The agreement between the predicted and actual test data increase for the 7.5 and 12.0 degree cases.

The disagreement for the lower curvature is most likely a result in the offset seen in previous plots. The point at which flange contact occurs is at a higher level of curvature for the test data in comparison to the model data.

# 7.4 PREDICTED STEADY-STATE ROLLING RESISTANCE

Analytical predictions of rolling resistance have been carried out for the range of misalignments tested with the instrumented

FIGURE 39. LATERAL FORCE WITH MISALIGNMENT ON 3° CURVE.





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20 THEORETICAL MEASURED 0 15 LATERAL FORCE (K-LBS) 10 5 0 ו -1

FIGURE 41. LATERAL FORCE WITH MISALIGNMENT ON 12° CURVE.



coupler described in Section 3.4. Figure 42 contains both modelled and experimental data from 0.0 milliradians up to 4.0 milliradians. In general, the agreement is fairly good.

### 8.0 FINDINGS

In a broad sense, the Wear Index/Truck Tolerance Test Program was very successful. It brought into a single context a variety of measured parameters which mechanically influence freight truck rolling performance. It also allowed for the use of a steady-state curving model in predicting the measured values of wheel/rail force, wheel wear, and rolling resistance. Of major significance are the following observations:

- o There is a large difference in wheel/rail forces between standard and radial equipment.
- o Axle misalignment has a significant influence on wheel rail forces and a correspondingly profound effect on the rate of wheel wear.
- o Initial differences in wheel wear are attributed to the original wheel profile; however, the differences diminish quickly due to the extremely high level of adhesion and high average curvature on FAST.
- o Train resistance has been documented for trains of varying length. The relationship of the measured resistance to predictions have been made using the CN-Modified Davis equation.



ROLLING RESISTANCE (Ibs/ton)

FIGURE 42. MEASURED AND PREDICTED ROLLING RESISTANCE WITH MISALIGNMENT.

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