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Improved Railroad Freight Car Truck Performance and Safety: Phase 1

DOT/FRA/ORD-22/39

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NOTICE

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14. ABSTRACT
The Federal Railroad Administration contracted Transportation Technology Center, Inc. (TTCI) to explore the development and status of the Improved (or Integrated) Freight Car Truck (IFCT) and trace the development of the modern three-piece truck into the M-976 truck into the IFCT over the last 30 to 40 years. In addition, the report collects thoughts from industry representatives regarding the status of the IFCT. Finally, using parametric simulation results, the report attempts to optimize the design suspension parameters of the IFCT.

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Warp resistance, truck turning resistance, interaxle yaw stiffness, Improved Freight Car Truck, IFCT, rolling stock

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METRIC/ENGLISH CONVERSION FACTORS

ENGLISH TO METRIC

LENGTH (APPROXIMATE)

1 inch (in)	=	2.5 centimeters (cm)
1 foot (ft)	=	30 centimeters (cm)
1 yard (yd)	=	0.9 meter (m)
1 mile (mi)	=	1.6 kilometers (km)

AREA (APPROXIMATE)

1 square inch (sq in, in^2)	=	6.5 square centimeters (cm^2)
1 square foot (sq ft, ft^2)	=	0.09 square meter (m^2)
1 square yard (sq yd, yd^2)	=	0.8 square meter (m^2)
1 square mile (sq mi, mi^2)	=	2.6 square kilometers (km^2)
1 acre = 0.4 hectare (he)	=	4,000 square meters (m^2)

MASS - WEIGHT (APPROXIMATE)

1 ounce (oz)	=	28 grams (gm)
1 pound (lb)	=	0.45 kilogram (kg)
1 short ton = 2,000 pounds (lb)	=	0.9 tonne (t)

VOLUME (APPROXIMATE)

1 teaspoon (tsp)	=	5 milliliters (ml)
1 tablespoon (tbsp)	=	15 milliliters (ml)
1 fluid ounce (fl oz)	=	30 milliliters (ml)
1 cup (c)	=	0.24 liter (l)
1 pint (pt)	=	0.47 liter (l)
1 quart (qt)	=	0.96 liter (l)
1 gallon (gal)	=	3.8 liters (l)
1 cubic foot (cu ft, ft^3)	=	0.03 cubic meter (m^3)
1 cubic yard (cu yd, yd^3)	=	0.76 cubic meter (m^3)

TEMPERATURE (EXACT)

[(x-32)(5/9)] °F = y °C

METRIC TO ENGLISH

LENGTH (APPROXIMATE)

1 millimeter (mm)	=	0.04 inch (in)
1 centimeter (cm)	=	0.4 inch (in)
1 meter (m)	=	3.3 feet (ft)
1 meter (m)	=	1.1 yards (yd)
1 kilometer (km)	=	0.6 mile (mi)

AREA (APPROXIMATE)

1 square centimeter (cm^2)	=	0.16 square inch (sq in, in^2)
1 square meter (m^2)	=	1.2 square yards (sq yd, yd^2)
1 square kilometer (km^2)	=	0.4 square mile (sq mi, mi^2)
10,000 square meters (m^2)	=	1 hectare (ha) = 2.5 acres

MASS - WEIGHT (APPROXIMATE)

1 gram (gm)	=	0.036 ounce (oz)
1 kilogram (kg)	=	2.2 pounds (lb)
1 tonne (t)	=	1,000 kilograms (kg)
	=	1.1 short tons

VOLUME (APPROXIMATE)

1 milliliter (ml)	=	0.03 fluid ounce (fl oz)
1 liter (l)	=	2.1 pints (pt)
1 liter (l)	=	1.06 quarts (qt)
1 liter (l)	=	0.26 gallon (gal)
1 cubic meter (m^3)	=	36 cubic feet (cu ft, ft^3)
1 cubic meter (m^3)	=	1.3 cubic yards (cu yd, yd^3)

TEMPERATURE (EXACT)

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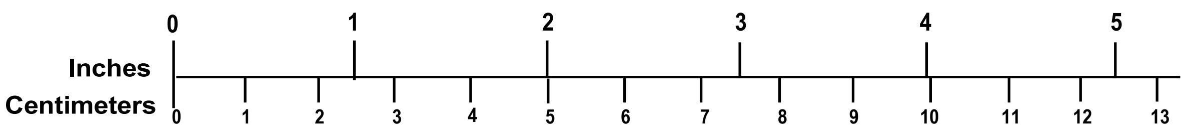
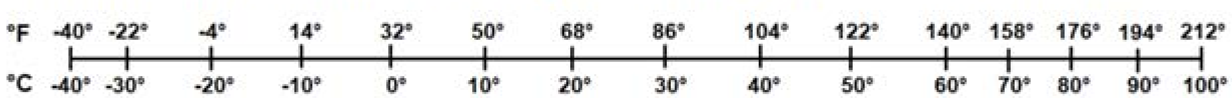
QUICK INCH - CENTIMETER LENGTH CONVERSION



QUICK FAHRENHEIT - CELSIUS TEMPERATURE CONVERSION



For more exact and or other conversion factors, see NIST Miscellaneous Publication 286, Units of Weights and Measures. Price $2.50 SD Catalog No. C13 10286

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Executive Summary

From September 2020 to September 2022, the Federal Railroad Administration (FRA) contracted Transportation Technology Center, Inc. (TTCI) to explore the development and status of the Improved (or Integrated) Freight Car Truck (IFCT) and trace the development of the modern three-piece truck into the M-976 truck into the IFCT over the last 30 to 40 years. In addition, the report collects thoughts from industry representatives regarding the status of the IFCT. Finally, using parametric simulation results, the report attempts to optimize the design suspension parameters of an IFCT.

The ubiquitous four-wheel freight car truck has developed dramatically in capacity and serviceability over the last 150-plus years. In the last 30–40 years, the three-piece freight car truck has improved in terms of performance and safety while not all achieved M-96-76 certification in ways barely perceived a generation ago. Not only can today's three-piece truck handle excepted (10 mph) track and 70+ mph operations, but the highest-performing trucks can do so with less rolling contact fatigue, risk of wheel failure, risk of derailment, and curving resistance than believed possible with 286,000-pound gross rail loads (GRL)—frequently denoted as "286 K"—and double-stack container cars (double-stacks) in their infancy.

The classic trade-off in rail car truck design is the balance between high-speed stability performance and curving performance, but optimum interaxle shear stiffness can benefit both performance areas. The formula for developing the IFCT involves finding the best combination of three fundamental characteristics:

- Truck turning resistance, expressed as a moment or torque. This moment must be minimized to enable truck rotation into and out of curves and turnouts while remaining large enough to maintain stability on tangent track at high speeds.
- Warp resistance. The truck frame, composed of two side frames and a bolster, must remain square (90-degree angles). It must have sufficient warp stiffness to return to square after the removal of a warp moment. A truck that warps, either in curving or in high-speed hunting oscillations, will generate very large gage-spreading forces. These forces can lead to wear and metallurgical damage to wheels and rails, and, in extreme cases, they can lead to wheel climb or rail rollover derailments.
- Interaxle yaw stiffness. The two wheelsets must be allowed to take the most favorable angle of attack possible. This positioning requires the proper amount of primary suspension shear compliance. Specifically, naturally occurring steering moments must be able to move the lead wheelset into as much of a radial position in the curve as possible. However, the primary suspension must be stiff enough to prohibit wheelset lateral oscillations at high speeds.

These characteristics are interrelated, and while each is important, this report focuses primarily on warp resistance. It reviews both the development of the so-called "standard warp test" and the results for many of the trucks that were tested using this method. Moving from laboratory to on-track testing, in recent years TTCI researchers developed the "traction ratio test" as the current best practice to validate IFCT performance levels. This test used instrumented wheelsets to verify performance.

In addition, this report also presents the results of a parametric simulation analysis. The

simulations covered a range of primary and secondary stiffness characteristics for a reference generic three-piece truck, a base IFCT, and a truck with optimized characteristics. The simulation results agree with the structure of on-track testing because the base IFCT is already optimized. While variations in the parameters have little to no improvement over the base IFCT, warp stiffness enhancement components, such as side frame cross bracing, can offer some marginal improvement. Industry experience agrees with simulation and testing, indicating meaningful improvement is difficult to achieve, and the cost, complexity, and weight penalties seem to nullify any gains.

It appears that development of passively steered three-piece trucks may have reached their best performance. In fact, it may be very difficult to achieve significantly better performance from a truck comprising two side frames and a bolster without the use of additional features such as cross links between side frames or steering links between axles.

1. Introduction

From September 2020 to September 2022, the Federal Railroad Administration (FRA) contracted Transportation Technology Center, Inc. (TTCI) to explore the development and status of the Improved (or Integrated) Freight Car Truck (IFCT), which is also known as the "improved freight car truck." The IFCT is essentially the state-of-the-art form of the traditional ubiquitous three-piece freight car truck (i.e., the M-976 truck) used in North American railroading (Figure 1). The term "integrated" suggests that the design should address the complete truck including the brake system and truck-to-carbody interface (Tournay, Duran, & Anankipaiboon, 2009). Note that Tournay et al. (2009) used the term "improved truck design" as well. The aim of using the IFCT is to reduce curving resistance, reduce wheel and rail metallurgical damage, and provide high-speed stability, while supporting a 286,000-pound gross rail load (GRL), frequently denoted as "286 K," operating over a range of track conditions and rail profiles.

1.1 Background

The classic trade-off in rail car truck design is the balance between high-speed stability and curving performance. Optimal interaxle shear stiffness can benefit both these performance areas. In three-piece railroad freight car trucks, warp restraint is a key component of interaxle shear stiffness. Warp restraint prevents the axles and side frames of a truck from moving from a rectangular orientation to a parallelogram orientation. Most trucks depend on friction wedges for warp restraint and to keep the side frames perpendicular to the bolster. The typical current freight car truck wedge provides adequate warp restraint in the empty car condition and quasi-static loaded car condition, but it sometimes falls short in loaded car dynamic conditions like loaded hunting.

Better freight truck warp restraint will improve high speed stability (hunting stability) and curving performance. Many previous attempts to improve warp restraint by adding stiffening components have been subject to fatigue and other types of failure. Documentation is needed for the performance requirements, load environment, failure modes, and other operational deficiencies of these trucks. One objective of this work was to evaluate warp restraint systems to provide that documentation.

Some freight car truck designs completely remove warp restraint from consideration and, instead, control interaxle motions with steering mechanisms. In this project, these trucks are not considered conventional three-piece trucks, and therefore, they were not evaluated. The scope of this project was limited to three-piece trucks, each having a bolster and two side frames, with primarily vertically oriented suspensions between them. The scope included trucks with additional stiffening components, such as a transom or cross braces between side frames.

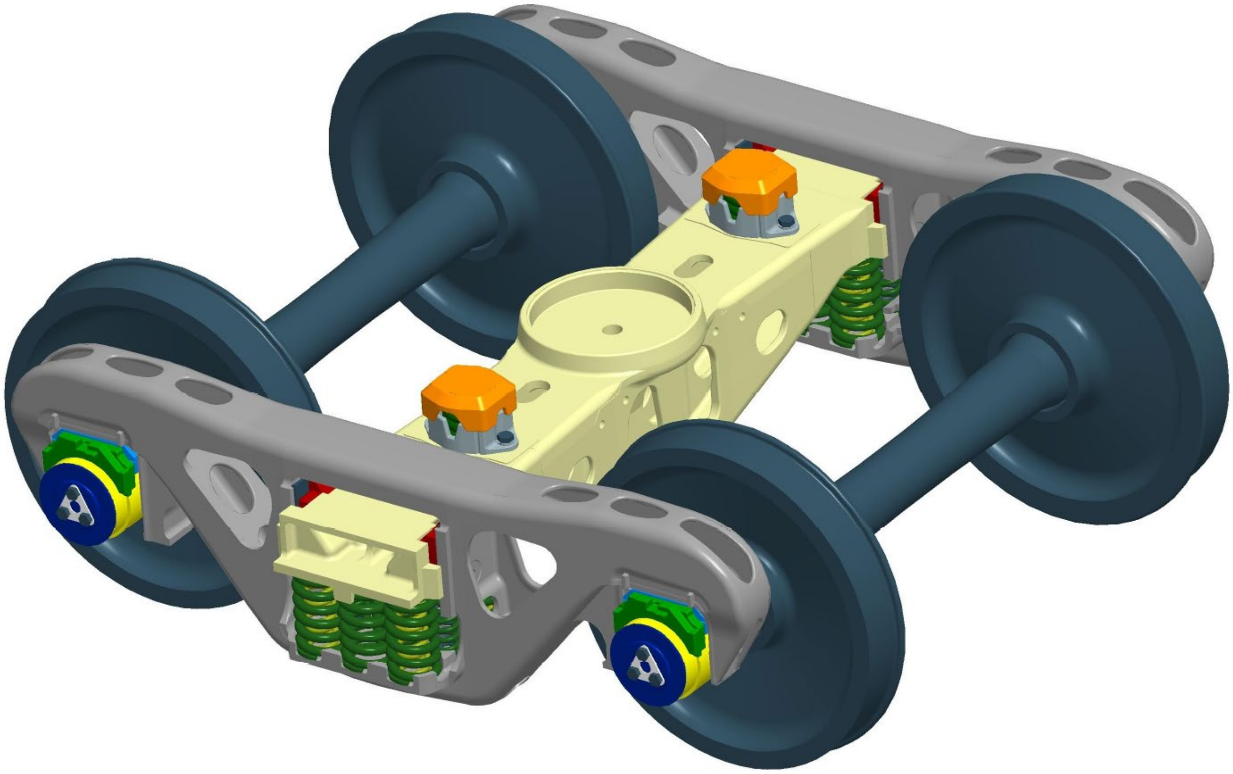


Figure 1. Typical three-piece truck (courtesy Amsted Rail)

1.2 Objectives

The primary objective of this work was to document the warp restraint performance requirements, load environment, failure modes, and other truck operational requirements necessary for a freight car three-piece truck to be successful as an IFCT.

1.3 Overall Approach

The research work encompasses three phases:

- Phase 1, documented in this report, includes a literature search and parametric vehicle dynamics modeling to define the realm of conditions over which warp restraint is effective.
- Phase 2 includes on-track testing to verify parameters and conditions predicted in Phase 1 and to quantify the load environment of secondary suspension yaw. This work will be presented as a written report and displayed at an appropriate forum.
- Phase 3 begins when prototype equipment, components, appliances, or designs capable of fabrication are available for laboratory or on-track testing as appropriate.

1.4 Scope

Phase 1 work was meant to establish 1) the current, state-of-the-art IFCT performance through a literature search and 2) a study of IFCT performance through parametrically modeling interaxle stiffness characteristics.

1.5 Organization of the Report

- Section 1 introduces the background of freight car truck research and development, particularly in the context of rail industry needs.
- Section 2 presents a literature review of the research and development of advanced three-piece trucks.
- Section 3 summarizes the results of the NUCARS®[1] simulations designed to explore three-piece truck performance as a function of suspension parameters.
- Section 4 listed future research topics arising from this work, and provides a statement of what has been learned about the status of the IFCT.

[1] NUCARS® is a registered trademark of TTCI.

2. Literature Review

Arch bar trucks were in widespread use from Civil War times until the early 20th century, when cast side frames were developed. Development continued with the American Steel Foundries' Andrews cast side frame truck, through the Bettendorf side frame with integrally cast journal boxes, with increasing focus on ride quality and maintainability (Model Railroad Hobbyist, 2022).

The literature on freight car trucks is extensive due to nearly two centuries of development. One of the earlier efforts to record the performance of various three-piece trucks was conducted by Canadian Pacific (Ghonem & Gonsavles, 1978). Predating Wheel Impact Load Detector -type force measurement sites and optical geometry detectors, this work was remarkable because it evaluated both curving and high-speed performance using instrumented rail cribs to measure wheel-rail forces and lasers to measure the angle of attack.

The literature showed that recent improvements in three-piece freight car trucks were necessary. Beginning in the late 20th century, the primary driver likely became the introduction of heavy axle loads (HAL) associated with hauling bulk commodities (Association of American Railroads, c. 1997). Given a HAL environment, research focused on the many areas where optimizing the wheel-rail interface and truck design could yield safety improvements and reduce operating costs. As a result, truck designs progressed from standard three-piece configurations to the current M-976-IFCT designation (Association of American Railroads, 2020).

2.1 Fundamentals and Origins of Freight Car Truck Improvements

As early as the 1990s, poor wheelset steering and truck warp, which together resulted in gage-widening forces, are two predecessor concepts central to the IFCT (Mace & DiBrito, 1994). These concepts were reviewed extensively in an FRA report concerned with gage-widening behavior observed on the High Tonnage Loop (HTL) at the Facility for Accelerated Service Testing (FAST) near Pueblo, CO (Mace, 1993).

As early as 1995, common drivers of truck warp could be excessive truck turning resistance and the reversal of normally beneficial steering moments (Hannafious & Mace, 1995). The same research attributed the loss of steering moments to adverse lubrication and wheel profiles. Research in 1996 provided higher axle loads as the impetus for improvements in three-piece freight car trucks (Read & Kalay, 1996). Phase III of the testing at FAST evaluated "premium trucks" to reduce the track maintenance cost impacts of 39-ton axle loads (315 k equipment). Testing included three premium trucks, which generally reduced the lead axle lateral forces and the angle of attack by about 50 percent, resulting in a 50–60 percent reduction in rail wear. Testing also showed reduced rail corrugations, reduced gage widening, and reduced fuel consumption.

2.1.1 Terminology

The following components and designations are associated with freight car trucks, such as the IFCT, and they will benefit from a detailed explanation.

Bearing Adapter Pads

Bearing adapter pads are also known as primary suspension pads, primary shear pads, or combinations of these terms. In conventional three-piece trucks, the cast steel bearing adapters fit the cylindrical shape of the journal bearing cup to the square shape of the side frame pedestal opening. The pedestal roof bears on the flat top surface of the bearing adapter in steel-on-steel frictional contact. The lugs on the corners of the bearing adapter engage the pedestal jaws, i.e., the vertical sides of the pedestal opening. The resulting primary suspension is very stiff, and longitudinal and lateral motion are constrained by friction within nominal total clearances of 3/32 and 5/16 inch, respectively. Bearing adapter pads are typically an elastomeric material that fills the interface between the top and lugs of the adapter and the pedestal opening. The resulting vertical stiffness is moderately high, but more importantly, shearing and compression of the elastomeric material governs longitudinal and lateral motion. This adds an important degree of control over wheelset yaw within the truck and truck steering characteristics.

Constant Contact Side Bearings

Constant contact side bearings (CCSB) have a stiffness element that keeps a side bearing pad in contact with the corresponding body wear pad on the body bolster of a carbody. The intent of the contact is to offer a controlled frictional sliding resistance as the truck bolster rotates under the car. This nominal truck turning resistance improves high speed stability by increasing the hunting onset threshold speed. The performance of these bearings is sensitive to their setup height: the distance from the top of the truck bolster on which they are applied to the bottom of the body bolster wear plate. If set too tight, the additional compression of the stiffness element will increase the sliding friction to the point of impeding truck rotation in curving. High truck rotational resistance can force the truck to warp instead of rotating (Hannafious & Mace, 1995).

M-976 Specification

As of January 1, 2003, the M-976 specification provided in the Association of American Railroads' (AAR) Manual of Standards and Recommended Practices (MSRP) volume D covers approval, performance requirements, and design validation of truck systems, devices, and suspension systems applied to railcars (Association of American Railroads, 2020). When required in the interchange rules, freight cars with two four-wheel trucks must be equipped with suspensions fully approved, conditionally approved, or preliminarily approved per this specification by the AAR Equipment Engineering Committee (EEC) for free/unrestricted interchange. The M-976 specification test load requirements imply that trucks approved under the M-976 specification are intended to operate with a maximum static wheel load of 36,000 pounds.

Wedges

Wedges, also called friction shoes or friction castings, are the key element in controlling the secondary suspension of a freight car truck. Fitted between the ends of the bolster and the corresponding side frames, wedges occupy a triangular-shaped volume (known as the wedge pocket) between the two. They provide a friction connection between the bolster and each side frame, damping vertical and lateral motions. Damping lateral motion is the most important factor in restricting truck warp.

2.2 Developments Driven by HAL

The early 2000s saw the articulation of the "Stress State of the Railroad" concept (Kalay & Samuels, 2002). This concept described the stresses that equipment placed on the track as a normal distribution (bell curve) and included the admonition that the industry needed to shift at least the upper end of the bell curve down to a lower stress state. About the same time, TTCI began evaluating improved ("premium") trucks in a series of tests titled "Improving the Economics of Bulk-Commodity Service." A 2002 AAR/TTCI *Technology Digest* summarized these tests (Rownd & Iler, 2002).

The three "bulk-commodity suspensions" are described below.

- Standard Car (Barber) Truck's S-2-HD with FrameBrace® components. FrameBrace® components provided substantial mechanical rigidity between the side frames, greatly reducing truck warp. Rownd & Walker (1999) also featured rubber bearing adapter pads.
- ASF's (now Amsted Rail) "Bulk Truck," an ASF Ridemaster truck with "high performance machined features, which produce consistent three-button side frames" as well as shear pads at bearing adapters (Rownd & Iler, 2000).
- The Barber S-2-E truck, equipped with split wedges, "50-286 dual rate" spring groups, and "945-SW" bearing adapter shear pads (Rownd & Iler, 2000).

All these truck tests showed a reduction in lateral curving forces around 50 percent over a base conventional Barber S-2-HD truck. The summary TD reviewed the Barber S2E and ASF Bulk trucks but substituted ABC-NACO's Bulk Truck and RESCO Engineering's Bulk Truck in place of the Barber FrameBrace® truck. These trucks showed improvements generally similar to the S2E and ASF Bulk trucks. A Jonsson (2002) paper published by Royal Institute of Technology in Stockholm indicates worldwide interest in premium trucks for HAL service.

2.3 The Importance of Warp Resistance

The premium truck performance testing made it clear that minimizing the warp angle between the truck bolster and the side frames was essential for improved truck performance. In 2003, the research team tested the warp stiffness characteristics of three premium trucks and published the results in a series of TDs titled "Warp Characteristics of Bulk Commodity Suspensions":

Barber S-2-HD FrameBrace® (Rownd & Pasta, 2003)

- Barber S-2-E (Rownd & Pasta, 2003)
- ASF Super Service Ride Master (SSRM) (Rownd & Pasta, 2003)

While bolster-side frame warp had to be minimized, interaxle yaw associated with wheelset motion in response to steering needed to be promoted. Researchers and the industry found that much of the steering benefit of premium trucks came from the bearing adapter pads able to shear longitudinally that afforded a degree of natural steering but reducing truck warp was necessary to allow the pads to survive (Rownd & Pasta, 2003). These tests described warp restraint in terms of three characteristics:

- Yaw stiffness: The torque developed between the bolster and side frames as a function of yaw rotation between those two bodies (in-kip/mrad).

- Yaw damping: The difference in torque developed between the bolster and side frames by warping the truck in one direction vs. the other (in-kip).
- Yaw moment: The torque developed between the bolster and side frames at a yaw (warp) angle of 10 mrad in the empty-car condition, 5 mrad when loaded.

2.3.1 Measuring Warp Resistance of Premium Trucks

The research team developed a specific apparatus and setup to test these yaw characteristics (Figure 2) (Rownd & Pasta, 2003). Table 1 summarizes the warp restraint characteristics measured by this method. Table 1 compiled the key performance metrics in the tests described above (Rownd & Pasta, 2003; Rownd & Pasta, 2003; Rownd & Pasta, 2003). In the empty-car condition, the premium trucks were an order of magnitude stiffer than the base conventional truck, and in the loaded condition, they were on the order of four times as stiff.

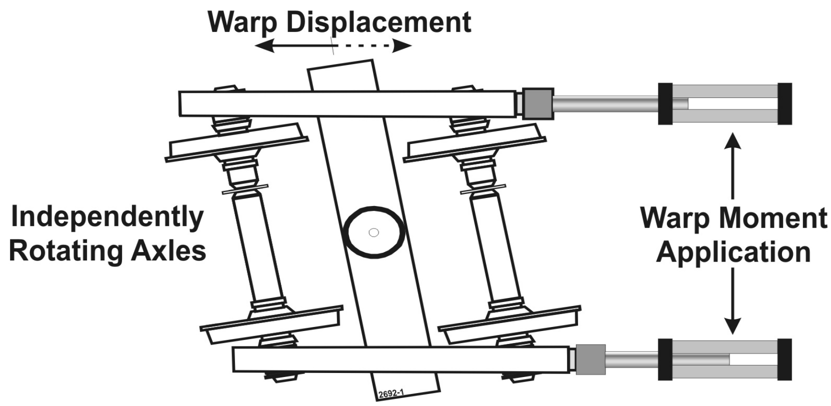


Figure 2. Warp restraint test setup

Table 1. Summary of measured warp restraint metrics

Condition	Metric	Base	FrameBrace®	S-2-E	SSRM
Empty	Stiffness, in.-kip/mrad	4.3	77.5	203	124
	Damping, in.-kip	140	209	673	346
	Moment, in.-kip @10mrad	182	984	2,700	1,590
Loaded	Stiffness, in.-kip/mrad	26.3	113	388	184
	Damping, in.-kip	438	469	928	975
	Moment, in.-kip @5mrad	580	1,030	2,871	1,893

2.3.2 High Level Radioactive Material Specification Development

Concurrent with the testing of new premium freight car trucks, the AAR was developing a specification for cars to carry high-level radioactive material (HLRM), sometimes called spent nuclear fuel (SNF). In the preliminary drafts, the specification required truck warp testing, though that requirement was not included in the final version of the specification (Wilson N. G., 2003). The testing used the same setup and methodology described in Section 2.3 and Figure 2. Figure 3 is a slide from a presentation to the AAR EEC showing photographs of the warp testing (Wilson N. G., 2003).

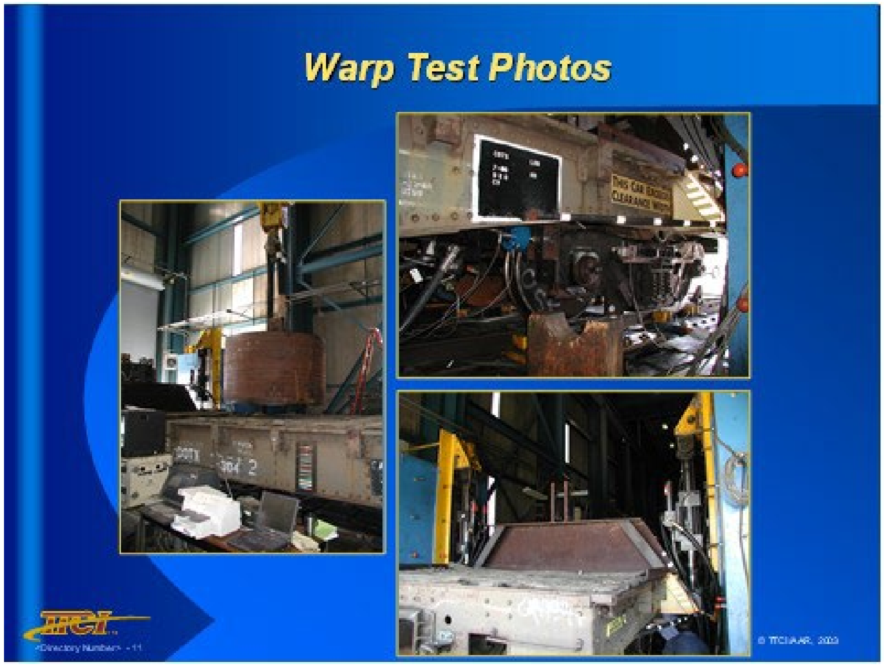


Figure 3. Compilation of photographs of warp restraint testing for HLRM service

The testing in 2003 applied a quasi-static warp input moment to reach a maximum warp angle and a cyclical warp moment. The test subjects were a Barber S-2-E, a Barber FrameBrace® S-2-HD, an ASF (Amsted) SSRM, and a conventional Barber S-2-HD as a baseline for comparison. At 2,000 in-kips of warp moment, all three improved trucks had warp angles less than 15 mrad, while the base S-2-HD exceeded 30 mrad (Wilson N. G., 2003).

Sudden removal of the warp moment revealed a self-restoring capacity (called warp resiliency). The S-2-E and FrameBrace® S-2-HD returned to less than 5 mrad. The SSRM returned to about 7 mrad. This was an important test because the drawback of adding warp stiffness by means of additional friction is creating a deadband about the neutral state, effectively adding hysteresis to the connection. In practical terms, it meant the truck might not return to neutral and steer straight down tangent track.

In 2006, TTCI followed up the 2003 testing by presenting to the EEC the effects of wedge rise and vertical input to the secondary suspension (Wilson N. G., 2006). Testing found "large amounts of wedge rise significantly reduce warp restraint" without numerically specifying the large amounts or the significant reduction. Worn truck conditions (in terms of wedge rise) tripled the equilibrium warp angle of restoration. Applying cyclical vertical loads, such as what a car would experience in pitch, bounce, and roll modes, further doubled the equilibrium warp angle.

Wilson (2007) wrote a summary assessment of the warp specifications that stated:

> *An initial version of the…truck warp specification was proposed in 2003... It required self-restoration to ±5 milliradians (mrad) of warp, without cyclic vertical load. Subsequently, a test was conducted with a cyclic vertical load, which showed that the proposed (2007) test method was feasible.*
>
> *Only one of the four truck designs, including some premium "M976" designs, could barely meet the self-restoration requirement of ±5 mrad when new. A test*

conducted on a simulated worn (high wedge rise) truck also could not meet the self-restoration specification. The revised (2007) specification relaxes the warp limit to ±7.5 mrad and includes specific definitions for the measurement of warp angle (Wilson N. G., 2007).

2.3.3 Testing Service-Worn Warp-Stiffened Trucks, "Standard Warp Characterization Test"

In 2004, AAR/TTCI published a *Technology Digest* on testing four trucks with approximately 500,000 miles of revenue service (Tunna & Walker, 2004). The testing itself was not as remarkable as the description of the test setup and parameters, described as a "standard warp characterization test:"

> *The standard warp characterization test was performed on the trucks. In this test, static and dynamic vertical forces are applied to simulate carbody movement, while equal and opposite longitudinal forces are applied to the side frames. The rotation of the bolster is measured with displacement transducers. The test produces a graph of warp moment plotted against warp displacement. From this graph, the truck's warp damping and stiffness can be calculated* (Tunna & Walker, 2004).

The test parameters were as follows, using the test setup and configuration as shown in Figure 4.

Constraints:

- Low friction center "lazy susan" bearing
- Independently rotating wheelsets
- CCSBs removed
- Resulting stiffness and damping provided only by spring nests, wedges, and pedestal bearing adapters

Loading:

- Longitudinal force applied in equal, opposite directions to each side frame at 0.2 Hz
    - Force control on each side frame, 15 kips empty, 25 kips loaded
- Static vertical load applied by empty car weight (67,000 pounds), plus two 50-kip dead weights
    - Loaded = 67,000 pounds / 2 + 100,000 pounds weights – 10,000 pounds truck weight = 123,500 pounds
    - Empty = 67,000 pounds / 2 – 10,000 pounds truck weight = 23,500 pounds
- Dynamic vertical load: applied ±0.2-inch displacement control
    - Loaded: at 0.5 Hz
    - Empty: at 2 Hz

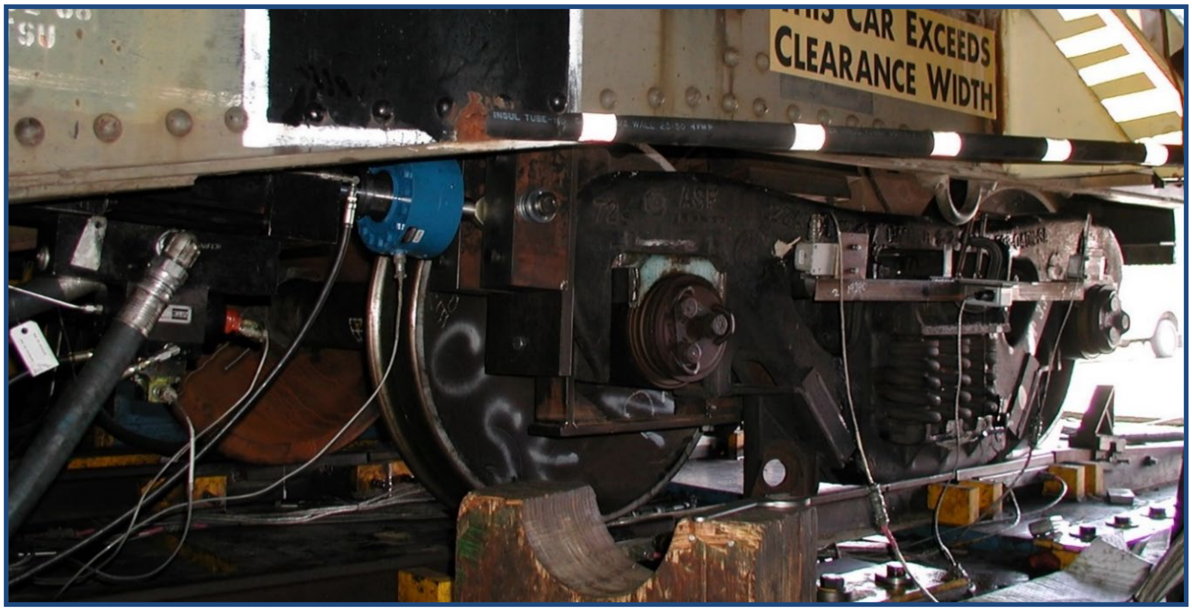
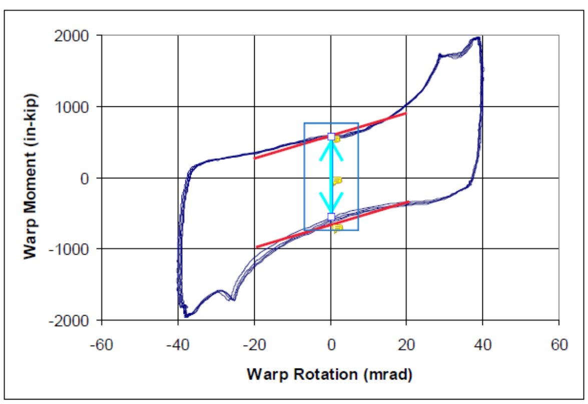


Figure 4. "Standard Warp Test" setup and configuration

Note: The blue load cell of a longitudinal actuator is seen between the side frame and carbody.

Tests were performed on two cars, in both empty and loaded conditions. Figure 5(a)(b) show the results from that testing. The base plot of the hysteresis loop measuring the warp moment as a function of warp angle was taken from the *Technology Digest*. This appears to be one of the first publications of a warp moment hysteresis loop. The two red lines and blue arrows, identifying warp stiffness and damping, respectively, were added in the current report.



Empty Car Warp Test Results

Truck	Warp Stiffness (in-kip/mrad)	Warp Damping (in-kip)	Moment @ 10-mrad Warp
UP35668 A	9.6	231	327
UP35668 B	11.4	111	225
UP35683 A	7.6	140	216
UP35683 B	18.8	197	385

Loaded Car Warp Test Results

Truck	Warp Stiffness (in-kip/mrad)	Warp Damping (in-kip)	Moment @ 5-mrad Warp
UP35668 A	16.3	624	706
UP35668 B	69.2	767	1113
UP35683 A	35.5	561	739
UP35683 B	52.0	573	833

(a) **(b)**

Figure 5. Test results for service-worn trucks: (a) hysteresis loop of warp moment and warp angle, with warp stiffness and damping identified; (b) warp resistance characteristics

The slope of the two red lines indicates the warp stiffness in the central portion of the hysteresis loop. The measurement units would be inch-kips per mrad of truck warp. Warp damping, which is frictional and thus not a function of velocity, is the difference in the warp moment between positive and negative warp rotation and has units of inch-kips. The warp moments at 5 mrad (loaded car) or 10 mrad (empty car) would be taken from the appropriate points on the hysteresis loop.

2.3.4 Another Warp Testing Methodology

Another work reported on measuring warp characteristics in a different manner than stated above. Instead of moving the side frames longitudinally, one axle was displaced laterally with respect to the other (Ren, Shen, & Hu, 2006). The calculated warp stiffness of a "worn Z8A truck, empty car condition" was 0.4 MNm/rad (3.54 kip-in/mrad), while that of a "new Z8A bogie under heavy load" exceeded 2 MNm/rad (17.7 kip-in/mrad). These values are comparable to warp stiffnesses reported by Rownd and Pasta for conventional three-piece trucks (Rownd & Pasta, 2003).

2.4 Performance Characteristics of Improved Freight Car Trucks

The key to IFCT performance is managing the interaxle shear stiffness within the truck. The management of interaxle shear stiffness has two components: 1) the warp or rotational resistance between the bolster and side frames and 2) the longitudinal and lateral compliance of the primary suspensions to allow the wheelsets to take up a radial position in a curve.

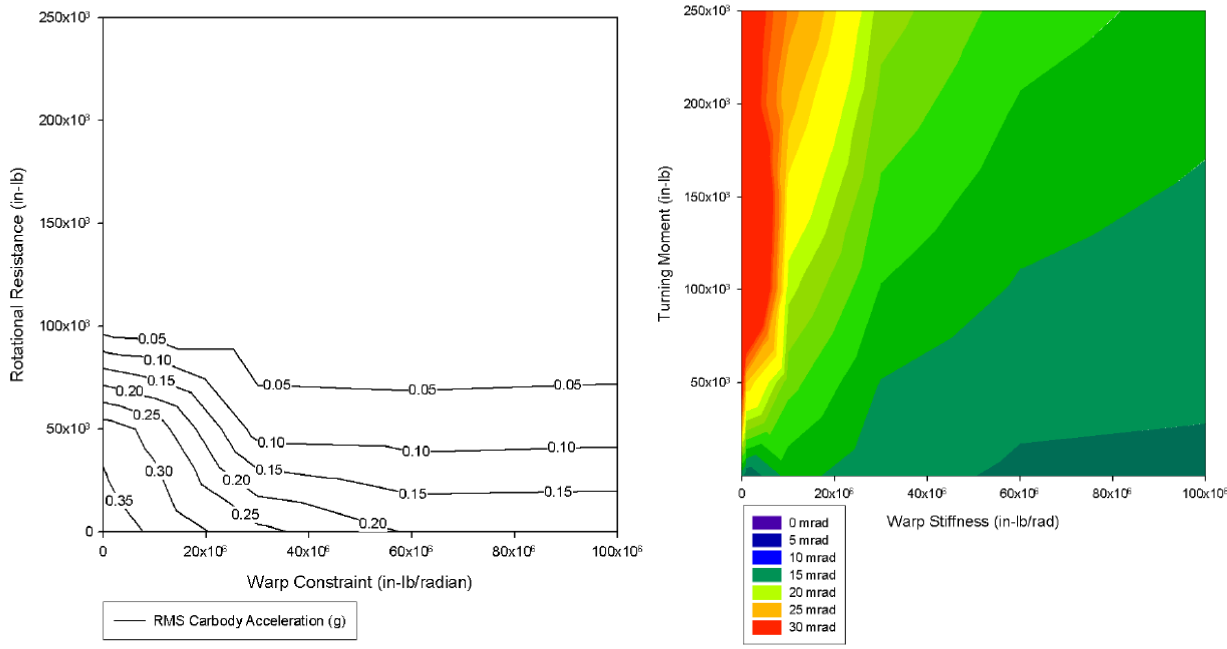
2.4.1 Performance as a Function of Warp Stiffness

Some of the first work to comprehensively characterize the performance envelope of IFCTs was given in an AAR report (Patsa, Tournay, & Urban, 2004). This work updated the warp performance of the premium truck by testing under a cyclic vertical load on the carbody, as shown in Table 2.

Table 2. Empty-car warp testing results—2 Hz vertical load applications

Metric	Standard Design	FrameBrace®	Split Wedge	Wide-Tapered Wedge
Warp Stiffness (in-kips/mrad)	3	117	45	38
Warp Damping (in-kips)	79	129	201	228
Warp Moment at 10 mrad (in-kips)	105	1,300	651	606

Going further, the work explored the development of an empty car performance envelope. Figure 6, taken from the AAR report, shows empty car performance in terms of the two diverse performance requirements: high speed stability and curving.



(a) **(b)**

Figure 6. Empty car performance envelopes: (a) contours of hunting behavior in terms of carbody lateral acceleration; (b) contours of angle of attack in a 5-degree curve

Stability is plotted as contours of Root Means Square (RMS) carbody lateral acceleration over a region defined by truck rotational resistance and warp stiffness. Higher accelerations indicate lower lateral stability. Curving performance is plotted as contours of the wheelset angle of attack over a region defined by truck rotational resistance and by warp stiffness. Greater angles of attack indicate poorer curving performance.

The culmination of the work's investigation into truck performance is shown in Figure 7, showing the work produced using NUCARS® vehicle dynamics simulations. This complex chart maps truck performance over the region of warp stiffness and turning moment under an empty carbody. The acceptable performance region is bounded 1) above by high single-wheel lateral over vertical (L/V) force ratio and, therefore, the risk of wheel climb, 2) below by decreased high-speed stability and greater hunting risk, and 3) on the upper left by a 10-mrad lead axle angle of attack.

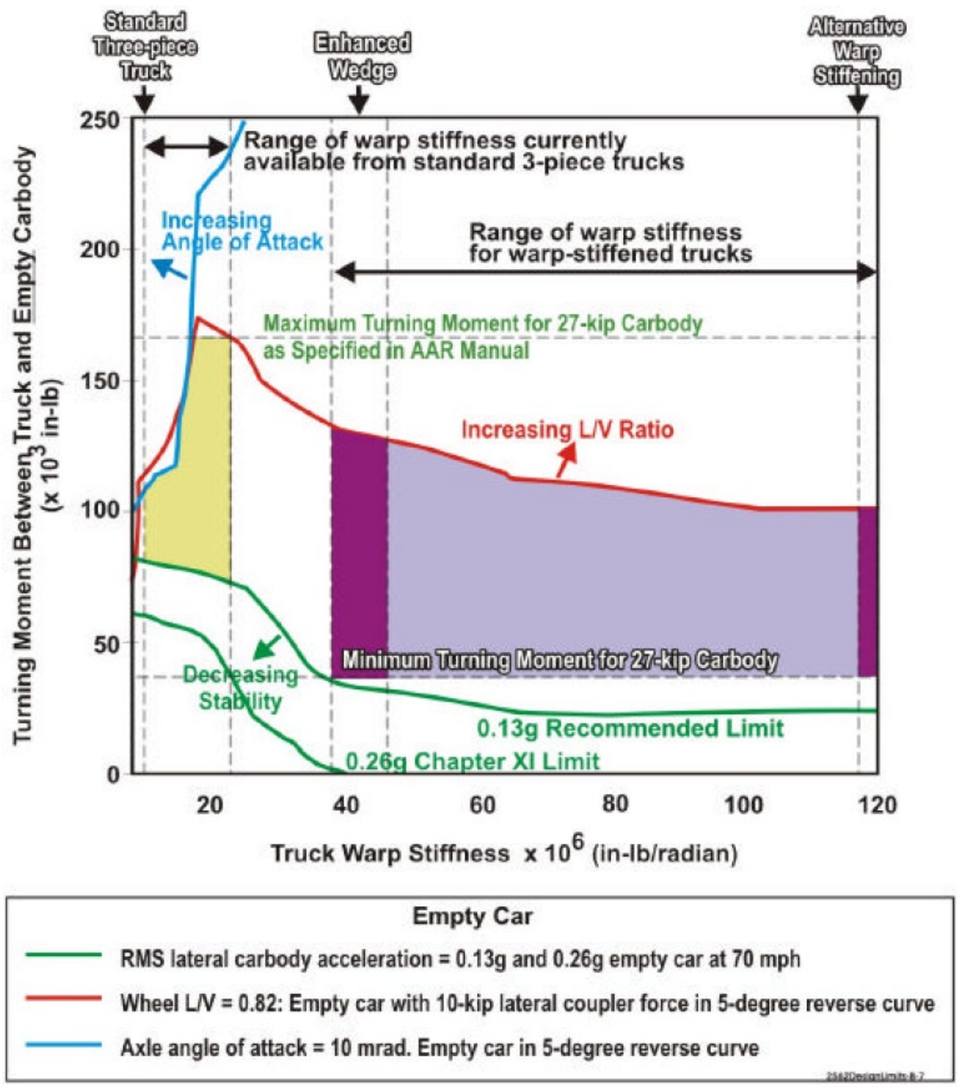


Figure 7. Truck performance map for empty-car condition

Work conducted by Tunna et al. (2006) expanded on the warp constraint performance envelopes. This work defined the performance envelope for hunting and curving in terms of warp stiffness, warp damping, and rotational resistance for three types of freight cars, each in loaded and empty conditions:

- 53-foot, 286,000-pound GRL hopper
- 65-foot, 263,000-pound GRL tank
- 89-foot, 206,850-pound GRL flat

The results were comprehensive due to the number and range of parameters that were varied. Among other findings, the results indicated that warp stiffness less than 10 kip-in./mrad can result in significant angles of attack. Appendix A1 in Tunna et al. (2006) explained how to modify a SYS file to model the "standard truck warp test."

The authors also developed a Windows-based application (Figure 8) that would calculate a performance output variable—angle of attack (AoA) (shown), L/V, and others—over a contour of truck characteristics, such as turning resistance, warp stiffness, warp damping, etc. Unfortunately, this work has been lost.

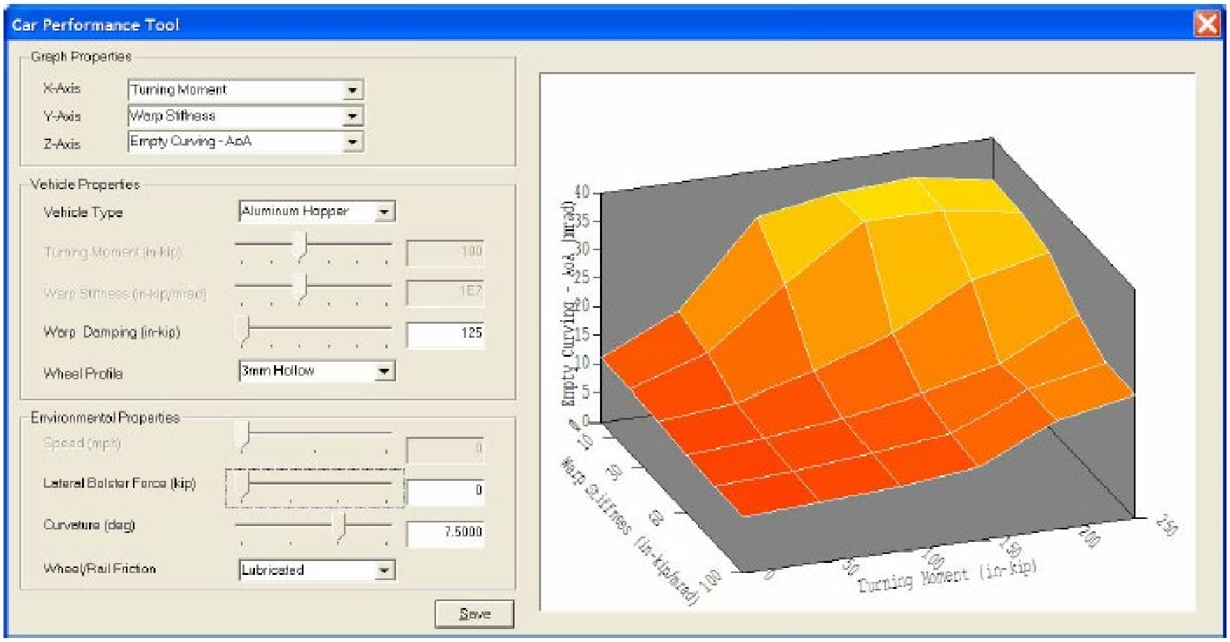


Figure 8. Example application window from the Car Performance Tool

2.4.2 Performance as a Function of Primary Suspension Characteristics

In Boronenko et al (2006), reported on the influence of various parameters on the curving performance and bogie stability. They found that "One of the most important parameters that influence curving is the total longitudinal clearance between the axle box and the side frame" (Boronenko, Orlova, & Rudakova, 2006). The experimental simulations showed a steep decline in angle of attack and wear number as [longitudinal] clearance in the axle box increased from 0 to 10 mm. L/V ratios decreased slightly over that range (Boronenko, Orlova, & Rudakova, 2006). North American researchers also found that allowing controlled axle motion in the pedestal was key to premium truck performance. However, they also found that warp stiffness in the secondary suspension was necessary for bearing adapter pads to survive:

> *Successful M-976 suspensions (to date) include elastomeric pedestal pads to reduce lateral curving forces and rolling resistance. Industry experience shows that suspensions with high warp resistance will allow these pads to survive and that the same pads applied to conventional suspensions can quickly fail. Long-term dynamic performance and warp restraint will be dependent on the wear life of the wedges and the integrity of the elastomeric pedestal pads* (Rownd & Walker, 2005).

Observations of curving performance measured by truck performance detectors (TPDs) confirmed the advantage of frame-braced trucks with primary suspension pads. Average high rail lateral forces were reduced to 3.6 kips (with a standard deviation of 0.33 kips) from 14.8 kips for conventional three-piece trucks (Pinney & Yoshino, 2005). These measurements came from the TPD installed on the HTL. The equipment population was Barber S-2 trucks, with and without frame bracing and primary adapter pads, in the FAST train. The researchers attributed the improved performance largely to the primary suspension pads.

2.5 Implementing and Applying Characteristics of IFCT to Solve Performance Problems

Without warp restraint exceeding that of conventional trucks, primary suspension pads tend not to survive. Three methods of increasing warp restraint have been tested: 1) FrameBrace®, 2) split wedges, and 3) wide wedges (Rownd, 2006). These tests show that "bulk commodity suspensions" (i.e., M-976 trucks) generate lateral forces that are about half that of conventional three-piece trucks in up to 6 to 7-degree curves. It is important to note that data for this comparison came from TPDs in revenue service.

As stated by improved truck performance research, improved truck performance depends on the ability of a wheelset to yaw in a controlled manner relative to the side frames, while keeping the side frames essentially square. The key to delivering good curving performance ensures that the trucks did not warp. This role of TPDs could monitor warp performance in revenue service and assess truck performance over a range of operating criteria.

In 2007, researchers proposed the concept of a warp index (WI), which is largely based on the lateral force of the trail axle as measured by TPDs. In an un-warped truck, the trail axle lateral force is very low. In a warped truck, it approaches the value of the lead axle (Madrill, Tournay, Wolgram, & Chapman, 2007). Mathematically, WI was proposed as the equation below:

$$WI = \frac{sqrt(warp_A^2 + warp_B^2)}{sqrt(2)}$$

Where $warp_A$ and $warp_B$ are warp indices as calculated in the Madrill publication.

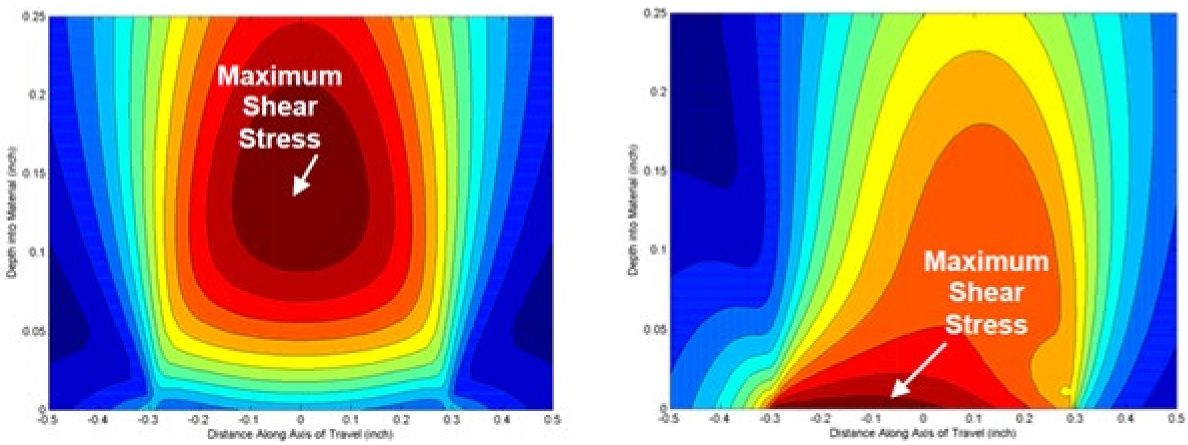
The authors further studied the WI in comparison to gage spreading forces, given curvature, wheel-rail friction, and axle load (Madrill, Tournay, Wolgram, & Chapman, 2007). The WI was further studied in the context of loaded car hunting (Tournay, Wu, & Wilson, 2008; Tournay, Wu, & Wilson, 2009). The WI shows that warped trucks produce high gage spreading forces, and trucks that can warp tend to produce loaded car hunting. However, this report did not indicate how to relate WI to actual truck characteristics.

In 2009, researchers made a subtle but significant adjustment to focus on the traction ratio (T/N), the ratio of tangential to normal forces acting at the wheel-rail contact patch, of specific wheels on the rail as opposed to the WI (Tournay, Anankitpaiboon, & Cummings, 2009). While the WI can be thought of as a macro truck performance indicator, T/N is related to mechanical wheel damage, and this appears to have been the seed-thought of looking into T/N as a truck performance criterion.

Continuing research in 2009 took the important step of relating truck suspension design to wheel damage via the T/N (Tournay, Duran, & Anankipaiboon, 2009). The T/N method gives a first-cut approximation of the interaxle shear stiffness required to limit forces on the low rail wheel and by extension on mechanical damage, rolling contact fatigue (RCF), and high impact wheels. The goal of the work was to develop new suspension designs as part of an integrated freight car truck design.

2.6 Reducing Metallurgical Damage: The Goal of IFCT

Throughout the research and development associated with “bulk commodity trucks,” M-976 trucks, and “integrated” or “improved” freight car trucks, large lateral wheel forces and longitudinal forces exceeding reasonable truck rotation moments have been considered adverse. More than adverse, it was known that over 50 percent of high impact wheels (HIW) were due to thermal mechanical shelling (Cummings, 2008). However, in 2009 investigators related measured surface tractions to wheel tread surface cracking. They proposed a hypothesis regarding how this cracking can progress to HIWs and proposed a way forward to an improved truck design (Tournay, Anankitpaiboon, & Cummings, 2009). They also found that T/N ratios moved the locus of maximum shear stress from below the surface of the wheel with no traction forces to the surface for T/Ns exceeding 0.3. Figure 9 reproduces Figure 4 and Figure 5 from Tournay, Anankitpaiboon & Cummings (2009).

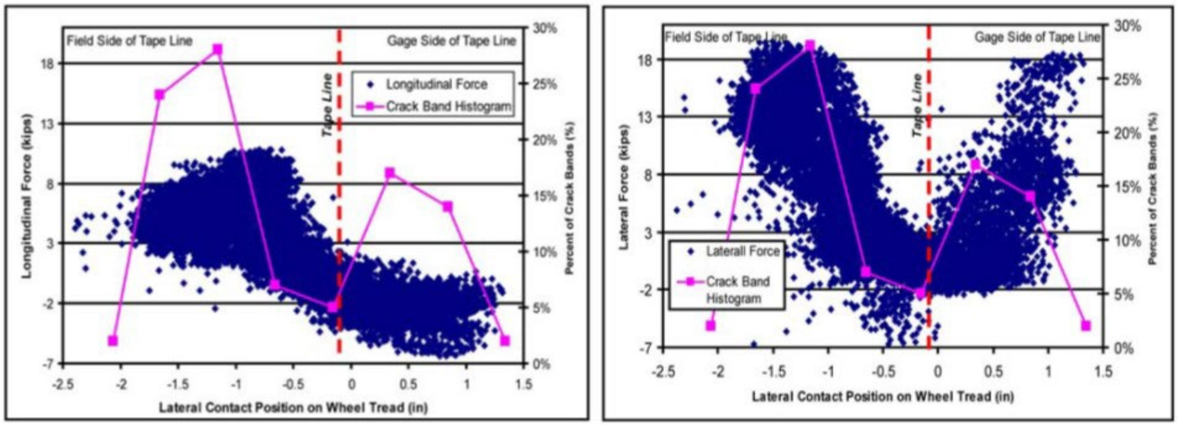


Position of the Maximum Shear Stress for Zero Traction

Position of the Maximum Shear Stress for a Traction >0.3

Figure 9. Locus of maximum shear stress as function of traction ratios

Further investigation showed that the magnitude of surface tractions (as measured on instrumented wheelsets [IWS]) correlated with the observations of surface cracks, as a function of location along the tread surfaces of lead axle low rail (LALR) wheels (Tournay & Anankitpaiboon, 2009). Figure 10 reproduces these scatter plots of surface traction values co-plotted with histograms of crack occurrences. Longitudinal forces (Figure 10a) varied linearly with distance from the tapeline, ranging from -7 kips at the flange root to 11 kips toward the rim face. Lateral forces (Figure 10b) drove the scatter plot shape that correlates with the incidence of fatigue cracks in the wheel tread.



Longitudinal Traction Forces on the Tread of the Lead Wheel Contacting the Low Rail of a Curve

(a)

Lateral Traction Forces on the Tread of the Lead Wheel Contacting the Low Rail in the Curve

(b)

Figure 10. Relationship between measured surface tractions and observed crack bands on the lead axle low rail wheel: (a) longitudinal, (b) lateral

The objective of this research was to further evaluate these tractions and relate them to suspension design attributes. The methodology for this evaluation was to measure the LALR lateral T/Ns for trucks being tested (Tournay & Anankitpaiboon, 2009).

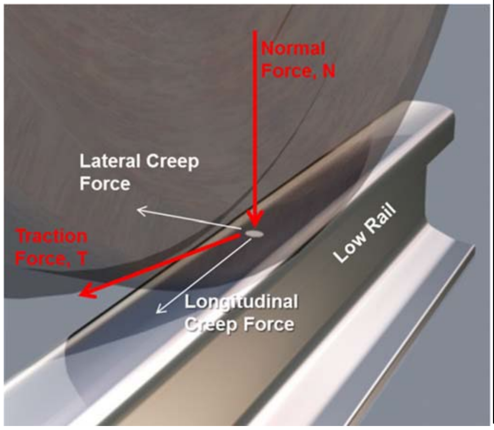
2.7 Current Development Status

To implement the T/N test, in 2016 through 17, the research team published a pair of TDs that described the test procedure and the data analysis and proposed adoption of T/N as the best method for identifying IFCT performance.

The IFCT research program aims to take the improvements found with the M-976 trucks one step further. Design improvements of the IFCT include optimizing the longitudinal inter-axle stiffness and improving warp restraint stiffness and clearances without compromising the life of the components (Cakdi & Tournay, 2016; Cakdi, Tournay, Walker, & Jones, 2017).

The work in both Cakdi TDs formalized the development of the T/N as shown in Figure 11. In physical testing, the T/N relies on IWS to provide the longitudinal, lateral, and normal forces at the contact patch. The contact angle must also be known. Vehicle dynamics software, such as NUCARS® and VAMPIRE®,[2] readily provide this calculation.

[2] VAMPIRE® is a registered trademark of DeltaRail Group Ltd.



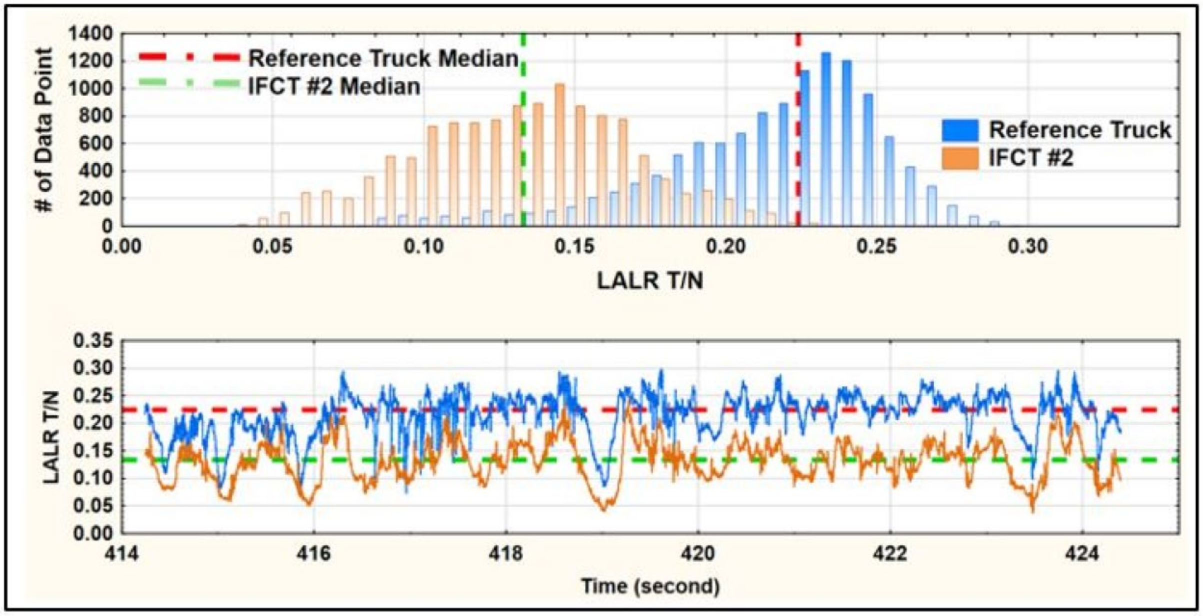
$$Vert = Vertical\ Load\ from\ IWS$$
$$Lat = Lateral\ Force\ from\ IWS$$
$$Long = Longitudinal\ Force\ from\ IWS$$
$$\alpha = Contact\ Angle\ (rad)$$
$$N = Normal\ Force\ Perpendicular\ to\ the\ Wheel\ Surface$$
$$T = Total\ Tractive\ Force$$
$$T = \sqrt{(Long)^2 + ((Lat) * \cos(\alpha) - (Vert) * \sin(\alpha))^2}$$
$$N = (Vert) * \cos(\alpha) + (Lat) * \sin(\alpha)$$
$$Traction\ Ratio = T/N$$

Figure 11. T/N Development

Compared to a base reference truck, the testing and analysis of IFCT performance show a compelling reduction in lead axle low rail T/Ns (Figure 12) (Cakdi, Tournay, Walker, & Jones, 2017). The reference truck was as prescribed in M-976, i.e., it used standard bearing adapters without pads, shimmed for longitudinal clearance not to exceed 1/16 inch. The intent of the reference truck is to provide a consistent baseline truck performance that both M-976 and IFCT trucks could use for comparison, regardless of day-to-day variations in wheel-rail friction conditions. Wheel-rail friction can be very difficult to control under test conditions, and including the baseline truck in the test consist serves two purposes:

1. It verifies that wheel-rail friction conditions are sufficiently high for a valid test.
2. It ensures that the differences in curving performance and T/N ratio results due to day-to-day variations in friction condition are removed from the analysis by normalizing the performance results against the performance of the baseline truck.

The three IFCTs were all developmental, none were available commercially. IFCT#1 employed a transom plate for increased warp stiffness and a proprietary-design primary pad. IFCT#2 used cross braces designed as part of the truck (though-bolster design) and proprietary-design primary pads. IFCT#3 was a commercially available M-976 truck with enhanced wedges that had cross braces applied additionally.



Sample Distributions and Time Series of LALR T/N: Reference and IFCT #2, 5-degree Curve, CW

Figure 12. Improvement in T/N of IFCT over reference truck

The current AAR MSRP M-976 specification added an "IFCT" designation in April 2020 for trucks that meet specific T/N criteria (Association of American Railroads, 2020). The limits that TTCI proposed for the M-976-IFCT designation appear in Figure 13, including the data points acquired in testing to support this development (Cakdi, Tournay, Walker, & Jones, 2017). AAR added these limits to the M-976 specification in 2020, appearing as Figure 4.2 in that standard (Association of American Railroads, 2020).

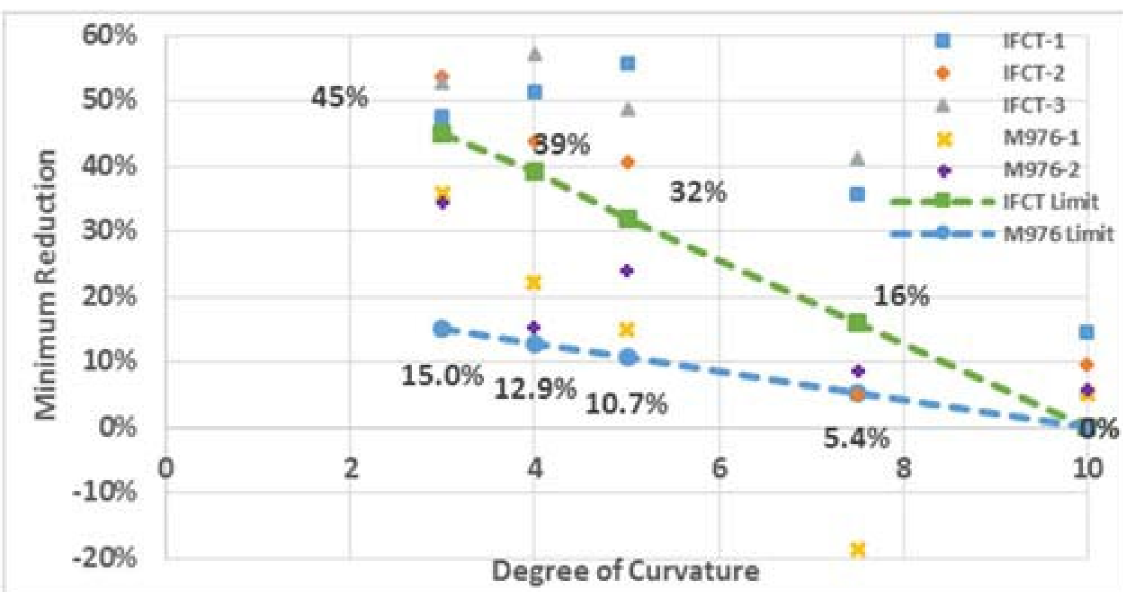


Figure 13. IFCT performance limits, as proposed prior to implementation

2.8 Comments from Industry Stakeholders

TTCI reached out to several industry stakeholders for an assessment of improved freight car truck technology and the issues surrounding it, including performance, testing, and AAR approvals. Unless noted otherwise, the information is quoted from emails. The prompting questions are enclosed in [square brackets], as are any minor edits for clarity and anonymity.

2.8.1 One Car Owner's Perspective

This section provides the perspective of one car owner in a first and second person narrative as received.

- With regard to our use of different truck technologies, we've worked quite a bit with different premium trucks and warp restraint systems over the years. We've followed the IFCT program for quite some time and have some thoughts and opinions as well.
- We like the direction that the IFCT project work drove to using IWS measurements to quantify curving performance. This is a more repeatable and objective measurement than the curve resistance test with coupler force. I think this was one of the most important outcomes of the IFCT project to date.
- I believe the number of FRA reportable derailments due to truck systems has decreased dramatically over the years. I would argue this has somewhat to do with improvements in performance of truck systems. We have seen performance and durability improvements with the evolution from constant damped trucks to variable damped, and eventually split wedge, then M-976 trucks. I see the IFCT as exploring this next evolution of performance benefits for the future. The challenge of IFCT, however, is there isn't as much improvement left to be gained without significantly altering the truck designs.
- If I were to offer some other thoughts which might help you think about the next direction of this program:
    - We have tested many different truck designs over the years. Many were tested at the [Transportation Test Center] TTC and if met performance criteria, were expanded to interchange service. A few of these truck designs we have track/field tested are as follows:
        - Swing Motion (various iterations of the design, 70T–125T capacities)
            - Performance of this product was very good on track, but not the best curving/high speed performance of products we have tested. This truck system is easy to maintain, and we've experienced component life of 1M miles +. This became our standard for high performance trucks because it offers the best balance of performance, durability, and ease of maintenance.
            - We worked with NACO to evolve many of the subcomponents – rocker seats, wedges (had lots of issues with the composition face breaking and falling off). Since this evolution, the durability and performance balance has been very good.
            - We have seen a disconnect in THD [truck hunting detector] performance (in service) compared to on-track hunting performance (on track at TTC).

- Frame Bracing (various iterations, 70T–100T capacities)
    - Performed well in tests at TTC using the LORD rubber adapter pads and the frame brace. Field issues were apparent from the beginning, mostly due to broken frame bracing from hitting things on the track. There was also a fatigue issue with the turnbuckles in the center of the frame brace fatiguing and breaking. This was a main reason for not using more than a few hundred cars – the frame bracing was as much of a maintenance liability as a performance benefit.
    - There were also issues with the LORD pads being too thick and causing the side frame height to be excessive. To account for this, the side frame manufacturers made a lower profile side frame specific for this application, but that always created a problem in supply of components and in maintenance.
- TI7 (various iterations, 70T)
    - Performed very well at TTC (some of the best performance we have seen in a truck design) but suffered from early component failures in the field. Mostly the leaf spring buckles would fail, but also had other nagging issues.
- North American Bogie / Wagon Union
    - Performed well in testing. We built 50ish standalone cars. These trucks required a special car body. Performed well but are so non-standard that it was hard to adopt these as a new truck. There were also wear related issues and component availability after the company went out of business. These cars have slowly been scrapped over time.
- AR-1 Steering Arms
    - Performed very well in testing at TTC. Had issues with the center bushing wearing out and losing performance. The field also had problems changing brake shoes on the car due to interference. There was a significant weight addition as well.

- In general, we have tested (track and service) various truck systems that exceed AAR standards for performance. The challenge has always been managing the truck performance with component durability and maintainability. We have also struggled with the repeatability of maintenance on any truck systems more complex than our normal 3-piece trucks. We still have issues with proper maintenance at all facilities on Swing Motion trucks, even though we've had a long history and large quantity of cars equipped with these. We are concerned with the complexity of the entrants into the IFCT test program because they appear to be similar to the design types listed above – we are skeptical whether or not these products could be a commercial success in our environment. It almost feels like advancing these further should include a laboratory durability test similar to transit truck system testing.
- While the FRA reportable derailments have decreased, we have heard the number of non-reportable slow speed derailments, usually in yards is still high. As I'm sure you're aware, this is usually flat, ladder track with less consistency than mainline track. I don't

think the IFCT project will do anything to influence these low-speed derailments that seem to plague RR operations.

- We have adopted S-2-HD trucks with split wedges and constant contact side bearings on a significant portion of our fleet. On M-976 equipment we use adapter pads in conjunction and are starting to apply 125T adapter pads to articulated intermodal trucks and it's proving to have a great improvement in wheel life. We have also started a test on 70T HD Barber style trucks with split wedges. Our strategy with truck systems is to continue to evolve the existing 3-piece systems to increase wear life of wedges, wear plates, and other components of the trucks. Small enhancements to the 3-piece system seem to be well received and easy to manage the change…the revolutionary truck changes are the ones this industry struggles to implement.
- A few years back, [a colleague] and I recommended to the M&VTS committee (now evolved into [the AAR Rolling Stock] Task Force) to consider developing a laboratory test of full truck systems. It's almost at a point where further evolution of a truck system (more refinement of rolling resistance, or warp restraint, or component durability) should be done in a laboratory setting as opposed to on track. We have felt for some time that the environmental factors are as strong of a test input on modern trucks as anything under test.

2.8.2 A Carrier Perspective

This section provides the perspective of a carrier supplier in a first and second person narrative as received.

- The focus of IFCT performance should be related to wheel (or other associated truck components) and associated rail wear rates.

2.8.3 A Truck Supplier Perspective (1)

This section provides the narrative from truck supplier from two different perspectives. [Regarding the development of IFCTs thus far:]

- No IFCT trucks have been approved to date (both traction ratio and loaded car hunting aspects under M-976).
- Traction ratio test is better than curving resistance test in previous version of M-976 as a more realistic measurement of curving performance.
- Trucks that may meet IFCT criteria may not meet all other M-976 criteria.

[Regarding the benefits of IFCTs, both documented and perceived:]

- Cost-benefit analysis conducted by AAR could not justify the extra cost of components for IFCT trucks.
- Lower traction ratio theoretically results in reduced RCF on wheel and rail.
- IFCT threshold for traction ratio does reduce rolling resistance to improve wheel and rail wear and save on energy consumption.

[Regarding the need for any further developments in IFCTs, or any type of freight car truck, what problems related to truck performance, effects on track and train operation, and ownership remain to be solved?]

- Lower traction ratio and loaded car hunting control have opposing requirements; additional components may be needed to satisfy both thus impacting cost.
- Effects of AAR-2A wheel on IFCT performance should be evaluated further since the years of testing under the SRI and testing of proponents' solutions was performed with AAR-1B profiles.
- Further study should be performed to understand how component wear of IFCT trucks affects performance over time.
- Loaded car hunting test car should be loaded to a "representative standard load" with lading impervious to degradation due to aging or environmental conditions.

2.8.4 A Truck Supplier Perspective (2)

- IFCT was to address:
  - high impact wheels (HIW) related to rolling contact fatigue initiated in curves
  - empty and loaded car hunting
  - rolling resistance test issues
  - asymmetric flange wear

[Regarding the development of IFCTs thus far:]

- To meet the IFCT levels, "premium" trucks are required. Based on presented test data, three-piece trucks with auxiliary warp restraint devices (Frame Brace, spring planks, etc.) with passive steering, and with active steering mechanisms with primary suspension pads, can meet current M-976-IFCT empty & loaded hunting and traction ratio levels. However, economic studies have concluded the price and weight penalty for steerable trucks deters application. Although typically lower in weight, the cost of components, installation, and maintenance of auxiliary warp restraint devices (AWRD) are also a deterrent in the North American market. The AAR HAL & Bulk Commodity Testing proved premium trucks perform very well but with weight and cost penalties.

[Regarding the benefits of IFCTs, both documented and perceived:]

- A considerable amount of effort was made measuring the traction ratio through various curvatures. The curving traction ratio values were then used to predict wheel RCF which leads to HIWs. After several years, SRI tests indicate that it is more likely RCF is generated in tangents vs. curves since a high number of cycles and a more constant wheel/rail position are required.

[Regarding the need for any further developments in IFCTs, or any type of freight car truck]

- Currently, improved high warp restraint 3-Pc trucks (without AWRD's) but with primary suspension pads have a difficult time meeting IFCT levels. 3-Pc truck T/N data has shown a small % difference, both positive and negative, between the reference truck and

M-976 test trucks for 10-degree curves. This introduces the possibility for random disqualifications of truck designs based on randomness within the band of test variability.

[What problems related to truck performance, effects on track and train operation, and ownership remain to be solved?]

a. Still to be refined is an improved rolling resistance test. The AAR reviewed the benefits of T/N over rolling resistance; however, with a new facility coming online shortly, past testing issues may be resolved.

b. Quantify acceptable truck warp stiffness for new and reconditioned trucks. Can warp stiffness tests eliminate expensive hunting and curving tests? How do truck warp test angles compare with on-track testing?

c. Asymmetric wheel wear. Recent SRI tests with M-976 trucks without brakes were inconclusive or suggest another mechanism other than the brake system still need to be investigated.

3. Parametric Modeling

This task involved building a parametric model of a conventional North American freight car, focusing on the trucks. Simulations were run in curving and high-speed stability (hunting) regimes using NUCARS® vehicle dynamics simulation software.

3.1 Parametric Simulations Objective

The objective of this task was to use NUCARS® to perform a parametric study of improvements, especially stiffness characteristics, required to achieve improved hunting and curving performance in IFCT.

3.2 Parametric Modeling Details

3.2.1 Vehicle Model

The vehicle was modeled as a loaded, generic 286 K GRL hopper car equipped with M-976 trucks in new, nominal condition. The hopper was 46 feet, 3 inches in overall length with 40-foot, 6-inch-long truck centers, 36-inch wheels, and a 70-inch truck wheelbase. Table 3 lists the detailed vehicle parameters.

Table 3. Vehicle design parameters, fixed

Carbody Mass	CCSB Preload	Secondary Suspension	Primary Suspension
265,440 lbm	4,500 lbf	7 D5 outer coils, 7 D5 inner coils, dual control coils	500,000 lb./in. vertical 40,000 lb./in. with clearances ±0.07 in. longitudinal ±0.031 in. lateral ±30 mrad yaw

Table 4 lists the detailed vehicle parameters that were varied as part of the parametric optimization task. As the research indicates, the two key characteristics that were modeled parametrically were the primary bearing adapter stiffness and the secondary yaw stiffness. Two separate connections, the adaptor pad tangential friction element stiffness (NUCARS® connection type 6.5 lateral and longitudinal) and the parallel spring damper element for the lateral primary pad stop (NUCARS® connection type 1.1), were varied for the primary bearing adaptor stiffness.

Table 4. Vehicle design parameters, variable

Parameter	Primary Adapter Pad Longitudinal Stiffness (lb./in.)	Primary Adapter Pad Lateral Stiffness (lb./in.)	Secondary Yaw Stiffness (in.-lb./rad)
Base Vehicle	40,000	65,000	7,500,000
Primary Stiffness 50%; Secondary Nominal	20,000	32,500	7,500,000
Primary Stiffness 200%; Secondary Nominal	80,000	130,000	7,500,000
Primary Stiffness 2000%; Secondary Nominal	800,000	1,300,000	7,500,000
Primary Nominal; Secondary Yaw Stiffness – 50%	40,000	65,000	3,750,000
Primary Nominal; Secondary Yaw Stiffness – 200%	40,000	65,000	15,000,000
Primary Nominal; Secondary Yaw Stiffness – 2000%	40,000	65,000	150,000,000
Primary Stiffness 50% Secondary Yaw Stiffness – 50%	20,000	32,500	3,750,000
Primary Stiffness 50% Secondary Yaw Stiffness – 200%	20,000	32,500	15,000,000
Primary Stiffness 200% Secondary Yaw Stiffness – 50%	80,000	130,000	3,750,000
Primary Stiffness 200% Secondary Yaw Stiffness – 200%	80,000	130,000	15,000,000

Table 5 shows the run environment matrix. Curvatures varied from tangent (0 degree) to 12 degrees in 2-degree increments. Superelevation was based on typical curve design conditions. Speed calculations were based on the equation below for 0, 2, and 4 inches of cant excess. When the cant excess of 4 inches would result in balance speed under 5 mph (or result in negative superelevation), the simulation speed was defaulted to 5 mph and the resulting cant excess was calculated and reported with an asterisk. Hunting and Dynamic Curve simulations were also run for the base vehicle and the primary stiffness 50 percent, secondary stiffness 200 percent cases over measured track geometry from the Railroad Test Track (RTT) tangents section and Wheel Rail Mechanism (WRM) track at the Transportation Technology Center (TTC), respectively. The wheel-rail geometry was AAR-2A narrow flange (NF) wheels on 132 RE rail with 10 in crown radius and 1:40 cant. A friction coefficient of 0.5 was used on the gage face and running surfaces.

$$Speed = \sqrt{\frac{Superelevation - Cant\ Excess}{0.00069 * Curvature}}$$

Table 5. Simulation matrix

Curve Degree	Cant Excess (inches)	Superelevation (inches)	Speed (mph)
0 (Tangent)	N/A	0	30
2	0	4	54
2	2	4	38
2	3.97*	4	5
4	0	5.5	45
4	2	5.5	36
4	4	5.5	23
6	0	5	35
6	2	5	27
6	4	5	16
8	0	5	30
8	2	5	23
8	4	5	13
10	0	2.5	19
10	2	2.5	9
10	2.32*	2.5	5
12	0	3	19
12	2	3	11
12	2.79*	3	5

*Note: When the cant excess of 4 inches would result in balance speed under 5 mph (or result in negative superelevation), the simulation speed was defaulted to 5 mph and the resulting cant excess was calculated and reported

3.3 Parametric Simulations Results

Simulation results are shown in terms of one or more of three key parameters:

1. Traction Ratio: defined and computed as the ratio of the vector sum of the longitudinal and lateral wheel forces at the contact patch, divided by the normal wheel force (vertical) in the contact patch.
2. Traction Ratio Improvement (percentage): This is the change in traction ratio value between the subject (test) truck and a reference truck, divided by the traction ratio of the reference truck.
3. L/V: This is the traditional ratio of lateral wheel on rail force divided by vertical wheel on rail force. The largest unsigned value is reported for any contact patch on either rail.

The results were summarized by changes in suspensions stiffness, as indicated above. The values shown in the following figures include 50 percent, 200 percent, and 2,000 percent of nominal value for the primary (bearing adapter) pad stiffness (“Prim”) and the secondary (warp) stiffness (“Sec”). These examples were all run at balance speed, as indicated by “Balanced” in the legends of Figure 14 through Figure 16. Figure 14 shows the T/Ns for an example cross-section of simulation conditions.

For these results, the "reference" truck is a conventional (non-IFCT, non-M-976) truck as specified in the AAR M-976-IFCT. Section 2.7 of this report includes an explanation of the reference truck in current truck testing. The "base" truck is an IFCT-qualified truck with nominal suspension parameters. The various results represent variations of those nominal IFCT parameters. Figure 14 shows the progressive increase in the T/Ns as primary stiffness increases relative to the base case and some reduction (improvement) with a 50 percent primary stiffness. Figure 15 shows the same results in terms of percent of improvement in T/N.

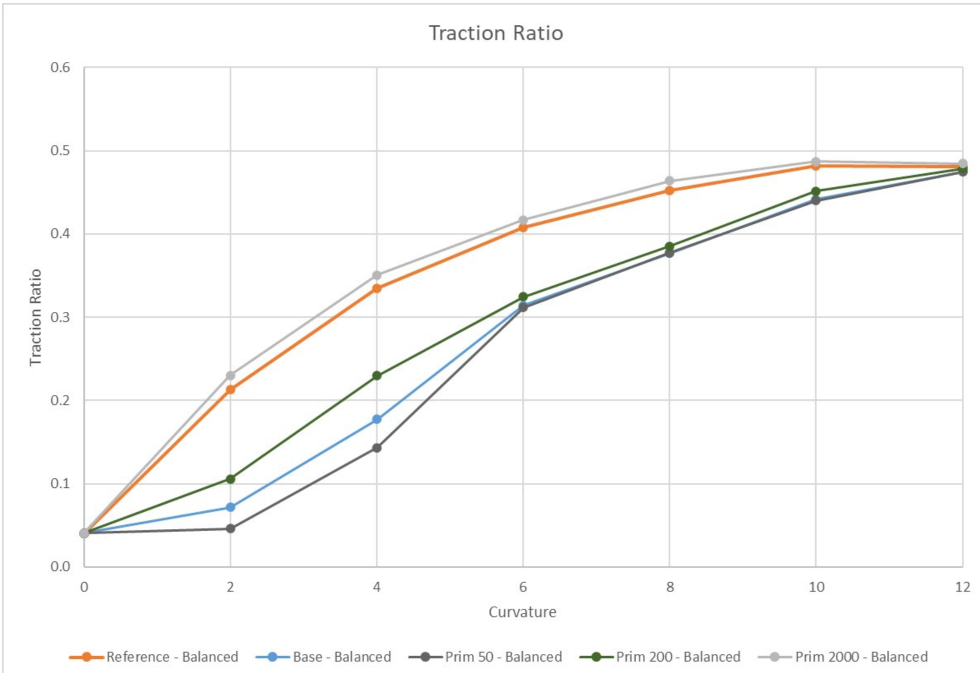


Figure 14. T/N response to primary suspension changes (lower is better)

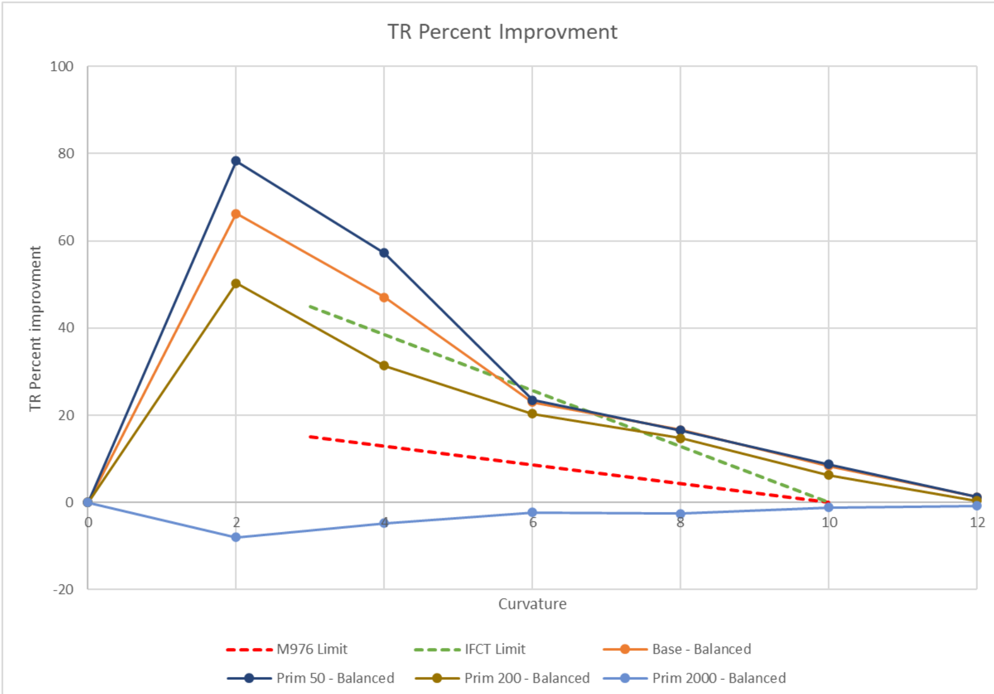


Figure 15. The percent improvement in T/N response to primary suspension changes (higher is better)

Similar summary simulations are shown for secondary stiffness variations. Figure 16 shows the changes in T/N for varying changes in warp stiffness. There is little to no change in T/N from the base case, except for 4 degrees of curvature.

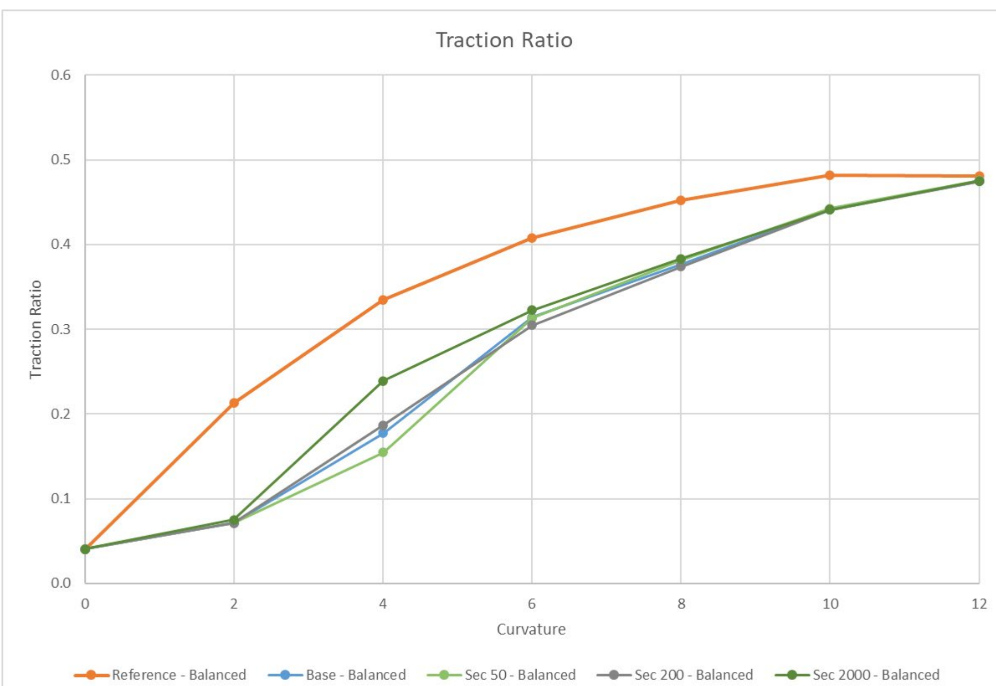


Figure 16. T/N response to secondary suspension changes

Based on these results and the literature review, Prim50_Sec200 was chosen as the "optimal design," i.e., primary suspension stiffness was reduced 50 percent from the base case, and secondary suspension was increased 200 percent from the base case. To verify the performance of this optimal design, the vehicle model with these parameters was simulated in the AAR's M-976 Dynamic Curving and Loaded Car Hunting test regimes. Figure 17 shows the performance of the loaded car in the Dynamic Curving regime. Dynamic Curving usually elicits the greatest response from a vehicle and is typically the most difficult test to pass. The results are given in terms of the two most important performance indices: minimum vertical wheel load and single wheel L/V. The designations of clockwise (CW) and counterclockwise (CCW) refer to the direction of travel through the loop track containing these two test regimes as the simulation input models the author's physical test track.

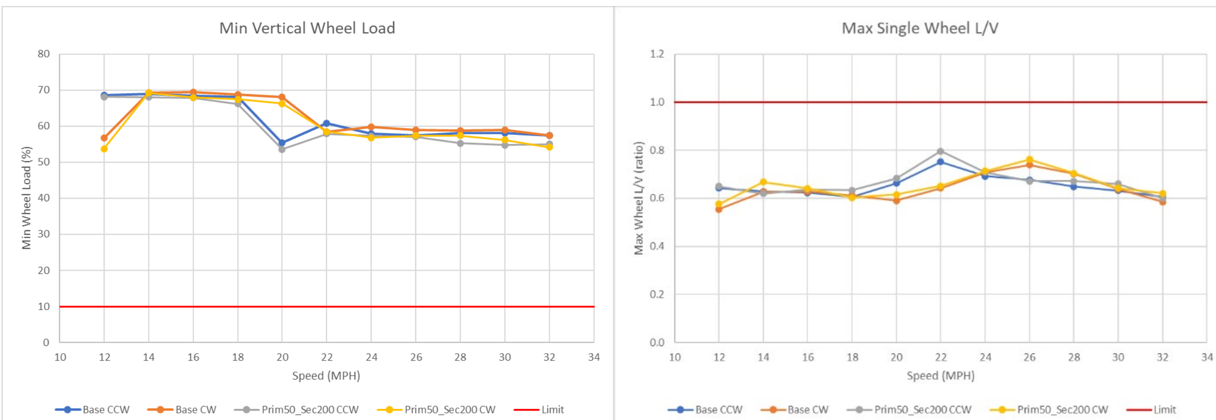


Figure 17. Base case and possible optimized suspension parameters simulated in Dynamic Curving regime

Figure 18 shows the performance within the Loaded Car Hunting regime in terms of standard deviation of lateral carbody acceleration and peak-to-peak lateral carbody acceleration.

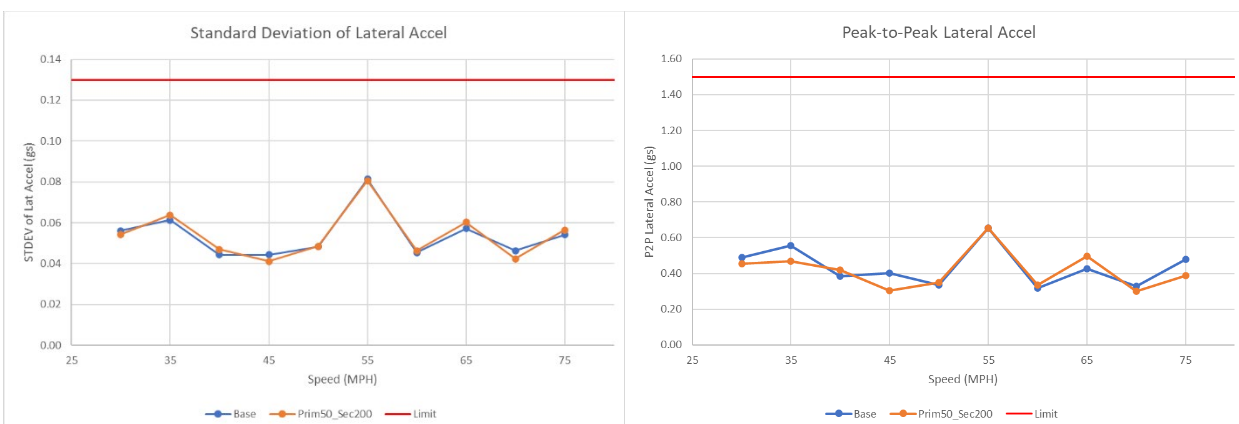


Figure 18**. Base case and possible optimized suspension parameters simulated in Loaded Car Hunting regime**

The key finding of the Loaded Car Hunting regime is that there is very little difference between the performance of the base case and the optimized parameters. This performance is consistent with the small amount of variation in T/N performance of the various simulated suspension parameters.

4. Conclusion

In this project, TTCI traced freight car truck improvements from generic three-piece trucks through M-976 trucks to the IFCT. Additionally, the project showed how the best performing freight car trucks depend on an optimal mix of truck turning resistance, warp resistance, and interaxle shear stiffness. The literature review focused on warp resistance and showed how researchers measured warp stiffness and damping. It summarized warp resistance in terms of warp stiffness and warp damping for a range of commercially developed three-piece trucks. Trucks that meet the AAR's M-976 performance specification operate very well in terms of curving and high-speed stability.

Further improvement of trucks to IFCT levels requires finely tuned suspension parameters to create optimized interaxle shear stiffness. Assessing IFCT performance levels requires performance methods that grew out of laboratory testing. State-of-the-art performance methodology is traction ratio testing. Remaining performance improvements may be diminishing to the extent that on-track testing cannot measure them within normal levels of measurement uncertainty, even with traction ratio testing.

Based on a review of truck development literature, parametric modeling of suspension parameters, dialog with industry leaders, and industry testing to date, it appears that IFCT performance, as described in the literature, is the zenith of three-piece truck design with passive steering. It may be difficult to achieve significantly better performance from a truck comprising two side frames and a bolster without additional features such as cross links between side frames or steering links between axles.

4.1 Future Work

Based on the literature review, receipt of stakeholder comments, and parametric modeling the following recommendations for future work related to IFCTs can be made:

1. Investigate physical wedge performance. Testing over the years indicates that trucks with auxiliary warp restraint perform much better in dynamic curving than trucks that depend only on wedges for warp restraint.
2. Investigate wedge connection performance in vehicle dynamics programs (e.g., NUCARS®) to attempt to model the binding or lock-up behavior mentioned in point 1.
3. Model IFCT performance with a range of service conditions. Different combinations of worn wheel-rail profiles and lubrication conditions should show how the difference in warp restraint affects curving performance and high-speed stability.
4. Investigate IFCT curving performance at 6 degrees of curvature and greater. The parametric vehicle dynamic simulations indicate an inflection in performance around six degrees. Unfortunately, wheel-rail mechanism at the TTC test track does not have a six-degree curve from which to obtain test data.
5. Investigate transient response of IFCT to abrupt alignment changes.

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Abbreviations and Acronyms

ACRONYM	EXPLANATION
ASF	American Steel Foundries (now Amsted Rail)
AOA	Angle of Attack
AAR	Association of American Railroads
AWRD	Auxiliary Warp Restraint Devices
CW	Clockwise
CCSB	Constant Contact Side Bearing
CCW	Counterclockwise
EEC	Equipment Engineering Committee
FAST	Facility for Accelerated Service Testing
FRA	Federal Railroad Administration
GRL	Gross Rail Load
HAL	Heavy Axle Load
HIW	High Impact Wheel
HLRM	High Level Radioactive Material
HTL	High Tonnage Loop
IFCT	Improved (or Integrated) Freight Car Truck
IWS	Instrumented Wheel Set
L/V	Lateral wheel-rail contact force
LALR	Lead Axle Low Rail (wheel)
lbf	Pounds Force
lbm	Pounds Mass
L/V	Lateral/Vertical
LVDT	Linear Variable Differential Transformer
MSRP	Manual of Standards and Recommended Practices
mrad	Milliradians
NF	Narrow Flange
RTT	Railroad Test Track
RCF	Rolling Contact Fatigue
RMS	Root Mean Square (average)

ACRONYM	EXPLANATION
SNF	Spent Nuclear Fuel
SRI	Strategic Research Initiative
SSRM	Super Service Ride Master
T/N	Traction Ratio
TTC	Transportation Technology Center
TTCI	Transportation Technology Center Inc.
THD	Truck Hunting Detector
TPD	Truck Performance Detector
WI	Warp Index
WRM	Wheel Rail Mechanism
WF	Wide Flange