IDEA

Innovations Deserving
Exploratory Analysis Programs

High-Speed Rail IDEA Program

Vibration Measurement of Rail Stress

Final Report for High-Speed Rail IDEA Project 48

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TRANSPORTATION RESEARCH BOARD
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Vibration Measurement of Rail Stress

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ABSTRACT

Measurement of thermally-induced stress in railroad rails is among the most significant issues in rail safety and track maintenance management, with implications for operation of high speed passenger trains on freight railroad trackage. Current techniques for that measurement are imprecise, time consuming, and expensive. This project investigated a new technique, using rail vibrations and laser vibrometry, for the measurement of contained rail force. The concept is based on the known sensitivity of bending vibrations to contained longitudinal stress. The rail is excited at a specified frequency using an electromagnetic shaker. A laser vibrometer that is scanned along the rail is used for dynamic displacement measurements to determine the frequency and wavelength of the resulting rail vibrations. The resulting wavelength, along with the rail rigidity (which can be determined by the rail dimensions and modulus) can be used to determine contained longitudinal load.

This project, together with a previous concept exploration IDEA project (HSR-30) developed the measurement technique, hardware and software to infer longitudinal rail stress loads in rail. The laboratory test results were encouraging, but did not meet the accuracy requirements established for such a system. Field tests were conducted by installing the prototype system on rail at TTCI at a location that was also equipped with a strain gage system for measuring rail stress. Comparison of the strain gage data with the prototype data revealed large differences. Any further development of this concept will require an investigation of the causes of these differences and whether they can be corrected with system design changes and/or changes in the field test procedures. This report summarizes the results of the project and concludes with recommendations for further development.

Keywords: rail stress, rail force, track buckling, rail breaks, vibration, rail neutral temperature, nondestructive evaluation

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EXECUTIVE SUMMARY

Contained stress in rails, due principally to thermally-induced expansion and contraction, results in large longitudinal loads that can lead to broken rails and track buckling, or "sun kinks". Broken rails and track bucklings are leading causes of train delays and serious mainline derailments. The consequences for safety, reliability and rail efficiency are significant. The ramifications of both safety and service reliability are magnified with high-speed passenger train operations.

Rail stress can be measured by rail uplift or application of strain sensors. Both have significant limitations. Measurement of stress by rail uplift is cumbersome and not very precise [Kish *et al* 1993]. It requires removal of the fastening devices from a large number of ties and requires significant track time and effort. Use of strain gages limits measurement of strains to specific locations where the gages have been installed and where cutting/destressing have been implemented to provide a reference state to enable subsequent absolute force determination. Field-mounted strain gages can degrade or detach.

The concept explored in this project is based on the known sensitivity of bending vibrations to contained longitudinal stress. The rail is excited at a specified frequency using an electromagnetic shaker. A laser vibrometer that is scanned along the rail is used for dynamic displacement measurements to determine the wavelength of the resulting rail vibrations. The technique requires the precise measurement of rail vibration profiles as a rail is vibrated at prescribed frequencies after two or three fasteners are (temporarily) removed. The resulting wavelength, along with the rail rigidity (which can be determined by the rail dimensions and modulus or by additional vibration measurements) can be used to determine contained longitudinal load.

A previous HSR-IDEA concept exploration project, HSR-30, investigated the basic concept and found it had potential but identified some issues that needed to be addressed. Much of that work was reported in Damljanovic and Weaver 2004, 2005) These included the need for a more robust, rapid, and reliable laser scan system; a need to accommodate variations in rail shape; and a need to mitigate the effects of intermittent structural changes in the track structure observed during measurements. This follow-on effort included development of an improved laser scan system. Laboratory tests on rails loaded in compression to 100,000 pounds, and circular rods loaded in tension to 25,000 pounds, showed the expected dependence on stress. The laboratory test results were encouraging, but did not consistently meet the accuracy requirements of 25 kips established for such a system – especially for worn rail. Field tests (see figure 1) were conducted by installing the prototype system on CN rail in Rantoul, Illinois and at TTCI at locations that were also equipped with a strain gage system for measuring stress. Comparison of the strain gage data with the prototype data revealed large differences. Possible explanations for these differences include the presence of loud bursts of vibration noise observed in the rail during testing, possibly due to changes in rail temperature during the measurements, and small slow variations, or "drift," in rail loading observed in both the laboratory and field tests. The unexpected noise at TTCI resulted in data from only one of three excitation frequencies usable. As a result, the technique for determining intrinsic bending rigidity could not be used. The lack of this information contributed significantly to the large differences between the strain gage data and the prototype data.

Any further development of this concept will require an investigation of the causes of these inaccuracies and whether they can be corrected with system design changes and/or changes in the field test procedures.

Figure 1] Photograph of the Frame used in field tests to support the laser scan platform (hanging below I-beam).

BACKGROUND AND OBJECTIVES

The widespread adoption of continuous welded rail on railroad mainlines has improved ride quality and reduced both maintenance costs, and dynamic loads that can damage track structure. However, a problem that has long been recognized is the contained stress in rails that is due to thermally induced expansion and contraction, and the effects of locomotive traction and train braking. In continuous welded rail, bolted connections are unavailable to relieve that stress. This results in large longitudinal loads that can lead to a number of problems.

Tensile stresses due to decreased rail temperature can lead to broken rails, while compressive stress due to increased rail temperature can cause track buckling, or "sun kinks". The resultant effect on safety and service reliability is of significant consequence to the railroad industry. Analysis of recent FRA accident/incident statistics show that various types of broken rails and track buckling are leading causes of serious mainline derailments (as measured by average number of cars derailed per accident.) These causes alone accounted for over 20% of all the derailed cars over the five-year period 1994-1998. Although the majority of broken rails and track buckles are detected before they cause a derailment, they still have a serious impact on service reliability. The ramifications of both safety and service reliability problems are magnified where high-speed passenger trains operate on freight railroad lines. Furthermore, fluctuations in rail stress can contribute to fatigue and thus premature failure.

For all of these reasons, a means of measuring contained stress in rails is important for freight railroads in general, and lines with high-speed rail operations in particular. This is supported by Transportation Research Board (TRB) Committees A2M01 (Track Structure and Design), A2MO5 (Guided Intercity Passenger Transport), and A2MO6 (Railway Maintenance) who have all identified the problem as an important research need. For discussion, see the web sites at http://www.nas.edu/trb/about/a2m01ps.html, /a2m05ps.html, and /a2m06ps.html. Interest is not confined to North America; at the 1997 World Congress of Railway Research this problem was identified as high priority. Members of the railroad engineering and research community have told us that a portable practical means of measuring this type of stress would be ideal because it would enable track engineering personnel to spot-check locations where trouble is suspected. Currently there are no practical methods for this sort of field measurement of stress levels in rails. The objective of this proposal is to develop technology that will enable such measurements.

Rails are laid at a temperature (the initial "neutral temperature") calculated to minimize the total risks associated with rail breaks and buckling. If the neutral temperature is known, the stress is inferable from the current temperature and the thermal expansion coefficient of the rail. Conditions of high stress would then be predictable. Unfortunately neutral temperature can change after track is laid, due to longitudinal creep associated with locomotive traction and braking. The essential problem is therefore often described as determination of a rail's neutral temperature, related to stress and temperature by simple math. Stress is, however, the more natural quantity, and endeavors to assess neutral temperature are usually recast as simultaneous measurements of temperature (simple) and stress (difficult.)

Rail tension P, equal to longitudinal rail stress times cross sectional area, is related to temperature and longitudinal strain by equations of thermo-elasticity;

$$P = EA\,(\,\varepsilon - \varepsilon_o - \alpha\,(T-T_o)\,)$$

where A is cross-sectional area, E is Young's modulus, α is coefficient of thermal expansion, ε is the strain, and T and T_o are respectively the temperature of the rail, and the reference temperature at which the reference strain ε_o is the strain when rail force vanishes. A measurement of strain, together with a measurement of the temperature of the rail, and knowledge of the reference state, allows one to calculate the rail force. This is the principle behind some methods discussed below. Clearly, if the rail expands as temperature rises, with changes in strain equal to α times the increase in temperature, the rail stress does not change. Neutral temperature is defined as that value of T for which P vanishes when there is no strain, i.e., when lack of joints has prevented the rail from expanding or contracting as temperature changes.

$$T_{neutral} = T_o - \varepsilon_0/\alpha$$

It may be concluded that

$$P = EA\,\alpha\,(T-T_{Neu\square\square\square\square})$$

For 136-pound rail this is 2.5 kips per degree Fahrenheit; the coefficient depends only slightly on rail size.

Rail stress can be measured by rail uplift or by application of strain sensors. Both have significant limitations. The rail uplift technique is based on simple beam-column bending theory, namely that the vertical force required to lift a rail varies with the axial force contained within the rail (Kish and Samavedam, 1987). The Transportation Technology Center (TTC) first demonstrated the concept in the early 1990's using a track loading vehicle (Kish *et al*., 1993). The technology is currently marketed under the VERSE® (Vertical Rail Stiffness Equipment) name and is owned by Vortok International & AEA Technology based in the United Kingdom. The VERSE® setup consists of a loading frame that is positioned over the rail to be measured. The frame lifts the rail by applying a vertical force, Q, while transducers measure the vertical displacement of the rail. The structure behaves like a beam-column; its deflection is measurably influenced by the longitudinal load in the rail. Compressive longitudinal load will diminish the effective flexural rigidity and increase the deflection at a given Q. Tensile load will increase the rigidity (as it does in a guitar string) and decrease the deflection at specified force Q. Rail uplift provides accurate results of neutral temperature; however, the overall process is cumbersome. The lifted section of rail must be free of end effects; which requires that the rail must be unclipped for at least 35 yd (32 m). Riser blocks must also be added at the ends of the unclipped rail section to overcome the effects of friction. That the method cannot be applied when rail is in compression is a further problem.

The use of strain gauges to measure strain in structural members is common. A strain gauge measures the strain, or change in length per unit length of the member, either mechanically or by means of electrical resistance. For strain gauges to measure neutral temperature in CWR, the gauges must be calibrated to a strain-free state (Zarembski, 1980). The mechanical strain gauge uses two indentations spaced a fixed distance apart that are punched along the rail's neutral axis when it is unloaded (Zarembski, 1980). This procedure is done on both the rail and on a standard, usually a short segment of similar rail that is left in the field near the rail being measured. The standard is measured for thermally induced strain and the net strain in the standard (strain in the standard minus the strain in the rail) corresponds to the longitudinal stress experienced by the rail constrained in the track. The electrical resistance strain gauge simplifies rail longitudinal strain measurement (Zarembski, 1980). It can either be bonded or spot-welded to the neutral axis of the rail. More commonly, spot-welded gauges are used on rail because they are better suited for severe environments. The electrical resistance strain gauge installation typically utilizes a half Wheatstone-bridge circuit, with two strain gauges on each side of the rail, and two fixed resistors. The voltage across the galvanometer is proportional to the change in gauge resistance, which is proportional to the change in strain in the rail. An advantage of the strain gauge technique is that temperature and rail lateral bending are compensated for (Kish *et al.*, 1987). Its use, however, requires cutting and destressing of the rail or the use or another technology to correctly zero the gauges, otherwise they only measure the change in strain since installation. Several companies specialize in strain gauge installations on CWR including DataTraks, Inc. and Salient Systems, Inc. This is probably the most commonly used technique to monitor neutral temperature in North America today.

More quick, reliable and cost-effective methods for measuring stress in rail would improve both the safety and efficiency of railway operations. This is a long-standing problem, and with increasing axle loads and train speeds this problem is becoming even more critical. New methods for measuring stresses in metal members could also be valuable for evaluating stresses in other items of interest to the railroad industry, for example wheels and bridges and indeed throughout the national transportation infrastructure.

Another concept that has been considered is to take advantage of the known sensitivity of bending vibrations to contained longitudinal stress. Related investigations using rail vibrations with a view towards determining stress in rails have been by Beliveau and others, with no success.

Here we have approached the problem from a significantly different perspective. We have developed the principles needed for stress determination by measurements of rail vibration wave functions, as measured using laser vibrometry scans and lock-in amplifier techniques, the instrumentation for which has advanced substantially in recent years. Precise scans of vibration patterns have been shown to reveal the state of stress of the rail. The work reported here involved laboratory tests and development of methods, field tests, and software algorithm development.

IDEA PRODUCT

At present we have a prototype system that includes a laser vibrometer, a 2-meter scan platform upon which it rides, an 20 lb electromagnetic shaker, small accelerometers and related electronics, all controlled by an integrated PC. A version ready for wide-spread field proofing could be mounted on a high-rail vehicle or small truck, positioned either manually or automatically, and be capable of being operated by rail personnel. Total cost of each system would be of the order of $50,000. Tests require removing a small number (2 or 3) fasteners, or more if greater accuracy is desired. We speculate that the technique would demand 30 minutes of track time, including time to remove and re-attach fasteners.

Our conversations with rail engineers show potential for significant impact on rail operations. Stress management would become more efficient (in track time, personnel costs and track damage) with consequences for operation efficiency and safety.

CONCEPT AND INNOVATION

The basic physics of the proposed approach is similar to that exploited in the rail uplift method discussed above. There a vertical force is applied to the mid section of a (~30m) span of rail (released from fasteners) sufficient to raise the rail a specified amount. Contained tensile force increases the effective rigidity of the rail and increases the force needed to cause the specified displacement.

The decreased rigidity associated with contained compressive load (or increased rigidity associated with contained tension) also manifests itself as decreased (increased) dynamic rigidity and corresponding vibration frequency. Beliveau and co-workers [Beliveau *et al* 1995-1997] proposed using the effect to assess rail stress, but found that variability in rail supports was too great; the effect of interest was masked by the support variability.

The decreased rigidity associated with contained compressive load (or increased rigidity associated with contained tension) also manifests as a change in wavelength of vibrations at specified imposed frequency. Thus detailed measurement of rail vibration profiles at fixed frequency, as a function of position along the rail between supports, i.e., how the vibration profile varies in space, allows inference of the contained load. Scanned laser vibrometry can do this. Most importantly, this dependence of wavelength on load is independent of end conditions. It can be used to measure the stress. The need is for accurate measurements of wavelength, and accurate understanding of the reference wavelength (the value it would have in the absence of stress), and a comparison between them. The chief challenges are related to the need for high precision measurements, often in a noisy environment, of rails of various degrees of wear.

Figures [2] and [3] show the measurement system. Representative data is shown in figure [4].

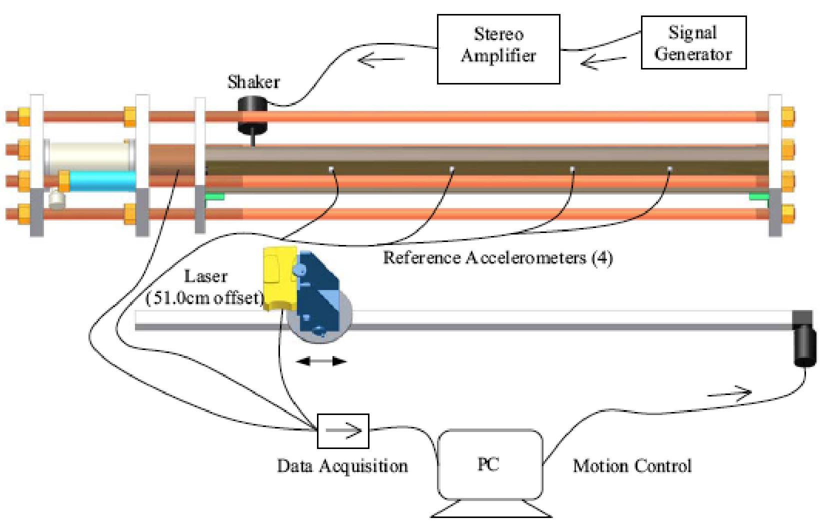

Figure 2] A schematic of the test system, showing a rail in its compression loading rig, the laser vibrometer on its linear scanning rail, the electromagnetic shaker, the computer's control of laser position, and sundry reference accelerometers.

INVESTIGATION

A previous HSR-IDEA concept exploration project (HSR-30) showed that rail vibration wavelengths indeed depend on stress as theoretically expected, and that the proposed laser vibrometery method was indeed capable of assessing that stress. The work on HSR-30 is summarized in previous reports and in the journal papers by Damljanovic *et al*. That project developed a finite element code to predict zero stress wavelengths and vibration profiles of the bending waves and torsional waves and evanescent waves. It numerically simulated measurement procedures in order to elucidate practical issues that might arise, and it demonstrated the ability of the technique to infer load in a test-bay setting. It showed that the proposed technique can be used to accurately measure contained load in rail, at least in the laboratory. It discovered and illustrated a number of aspects that would have to be understood and accounted for before the concept could be used in the field. The project showed that full development of the technique into a practical method for measuring rail stress in the field would require either knowing the intrinsic bending rigidity of the rail in the absence of load, or development of a method for inferring it from the vibrations. It was also clear that shifts in the rail structure during a test, particularly in its supports, contaminate the data. Practical application would therefore require either that the rail supports be fixed securely during the test or that a method be developed to control for this uncertainty. These observations formed part of the focus for HSR-48. The following subsections discuss the work done. Further details may be found in Koob (2005).

Figure 3] Photograph of measurement system.

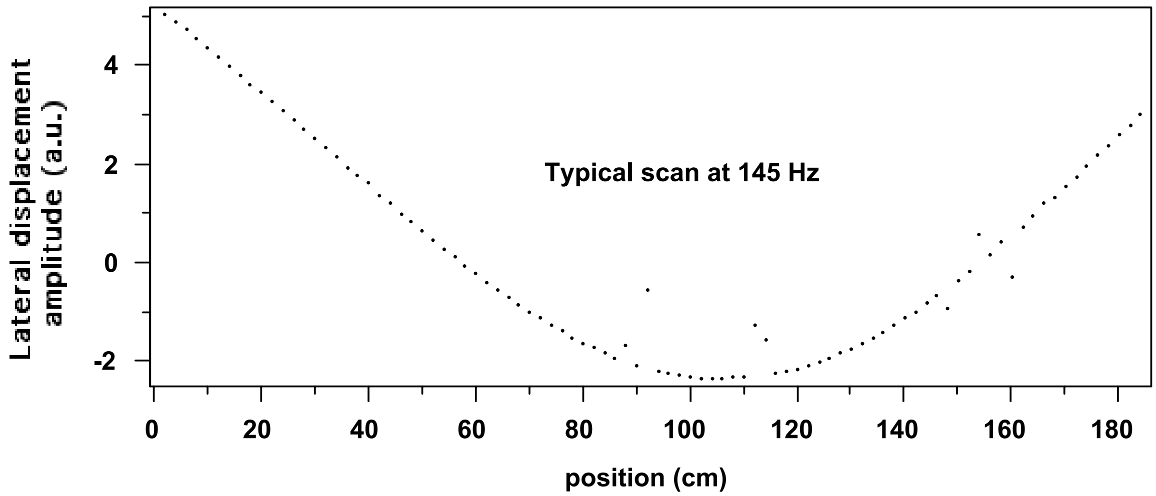

Figure 4] A typical vibration profile obtained by comparing the signal amplitude from the vibrometer with that from a reference accelerometer. The profile is nominally a simple bending wave. The nominal half-wavelength is the distance between successive zeros (at 57 and 154), or 97 cm. The actual bending wavelength is determined by a complex fitting process that includes the effects of torsional waves and evanescent bending waves. One can see the expected end effects, where the simple sinusoid is modified. One also sees the occasional "glitch" or bad data point, which the fitting software discards.

A. DEVELOPMENT OF ROBUST AUTOMATED SYSTEM FOR LASER SCAN OPERATION

A robust belt-drive track (see figure 3) for the laser, capable of automatic and precise positioning of the laser spot under computer control was constructed. The apparatus was designed to be independent of the vagaries of operator fatigue and loss of concentration (HSR-30 had used manual scanning where such concerns arose); it is capable of rapid and precise positioning with sub-millimeter accuracy. It has three independent drives, each under computer control: vertical (to permit measurements at different heights) and axial (along the laser beam direction that is, to permit re-focusing), and also length-wise (parallel to the rail, for the main scan.) The length-wise scan has a magnetic encoder to accurately report laser position in the length-wise direction.

The laser positioning system consisted of a Compumotor Zeta 83-93 motor driving a stage on an 8 ft (2.44 m) long Parker HLE-60 scan rail. Two smaller Compumotor HV-233 motors provided vertical and offset movement of the laser on a Parker 802-5470B platform. A Ramsey SG550 audio generator generated an excitation waveform that was amplified by a Crown DC-300A-2 stereo amplifier. A Labworks 12.5 lbf (56 N) electromagnetic shaker was used to vibrate the rail at a constant frequency. A 6.25-amp bus fuse protected the shaker from excessive current. A Shurite model 8505 ammeter was used to monitor the current to the shaker and to aid the operator in adjusting the shaker power. A household box fan provided cooling for the amplifier. Initially a magnet was used to fix the shaker rod to the rail. Later it was realized that more energy could be driven into the rail by clamping the shaker to the rail. A clamp was built by modifying a deep 3 in (7.62 cm) C-clamp. This allowed greater force to be transmitted without chatter, and thus permitted stronger and cleaner vibration signals. A Polytec PDV100 portable digital vibrometer was used to scan the response at discrete points along the rail based on the heterodyne interferometer principle and the Doppler Effect. Focusing the laser prior to each scan and maintaining a constant distance from the rail enhanced the laser signal quality. The offset distance from the rail also had to be changed when vertical movements were made; thus, three-axis motion was required for taking quality data. Three Compumotor Zeta-6104 microstepping drives were daisy-chained and operated via RS-232C serial port communications. The system was addressed so drive 0 controlled motions along the longitudinal axis (z), drive 1 controlled vertical motions (y), and drive 2 controlled offset motions (x) of the laser. Four PCB Piezotronics 353B33 ICP accelerometers, with model 080A27 magnetic bases, monitored the vibration of the rail and were attached to the rail or the jig at points *outside* the scan range. The accelerometers were powered by a model 482A04 power supply and 10 ft (3.05 m) long model 003A10 microdot cables were used to connect the two. A thermocouple was used to measure and monitor the rail temperature during the test and to take an ambient air temperature reading. A National Instruments BNC-2110 shielded connector block with signal-labeled BNC connectors bundled the signals and a single 12-bit National Instruments DAQCard-6062E in an IBM ThinkPad PC was used for the acquisition and digitization of the waveforms detected by the laser vibrometer and the accelerometers. A National Instruments LabVIEW 6.1 virtual instrument (VI) was written to command data acquisition and direct the laser motion via the ThinkPad's serial port.

B. CONSTRUCTION OF A RAIL LOADING RIG FOR TEST-BAY MEASUREMENTS

A frame for manual imposition of compressive rail stress was developed (seen in figures 2 and 3; illustrated in detail in figure 5). It also permitted tests on circular rods in tension. This loading jig was built to compress short segments of rail up to 100 kips. The jig was designed to accommodate rails of different sizes and wear patterns. The jig permitted repeatable laboratory tests on compressed rail at various load levels and complementary tests on circular rods in tension.
Four steel plates formed the bulkheads, A, B, C, and D, of the loading jig. Bulkheads A, B, and D were cut from 2 in (5.08 cm) steel plate. Bulkhead C was cut from 1.5 in (3.81 cm) steel plate. All bulkheads were 16 in (40.64 cm) square and had clearance fits for 1.75 in (4.45 cm) diameter steel rods at each corner. Nuts on the ends of the rods contained the load during load application; nuts on bulkhead B locked the load into the loading frame for testing. An Enerpac RC-1006 hydraulic jack between A and B applied the longitudinal force in the rail via a hand pump. A Lebow 3156 load cell between B and C was used to measure the longitudinal force in the rail. Sacrificial aluminum shims 0.0625 in (1.58 mm) thick were used between the faces of the load cell and the bulkheads B and C. L3x3x (inch units) steel angles 8 in (20.32 cm) long were used to support the rail on bulkheads C and D. The angles were slotted so their heights could be adjusted to accommodate different size rail. Lead sheets were used between the surface of the rail cross-section and the vertical wall of the support to prevent slipping and to ensure homogeneous end support conditions. A hollow cylindrical load cell was used between A and B to measure the tensile load in one of the circular rods for the rod study. The load cell outputs were wired to a strain gauge conditioner that connected to one of the inputs on the BNC-2110 shielded connector block.

C. CONSTUCTION OF AN I-BEAM FRAME FOR FIELD-TESTS

A frame was constructed, chiefly for use in field-tests (see figure 6) to which the laser scan track could be fixed**.** Steel frames with a 28.0 in (71.1 cm) wide footprint supported a steel W6x20 "I"-beam 11 ft (3.35 m) long (Figure 9.5). The scan rail was bolted to the beam and field tests were conducted as in the laboratory, but with the laser in the inverted position. Bolt holes were patterned into the beam 2 in (5.08 cm) on-center that allowed the length between the frames to be adjusted to accommodate different tie spacings. A Honda EU2000i generator powered all electrical equipment in the field.

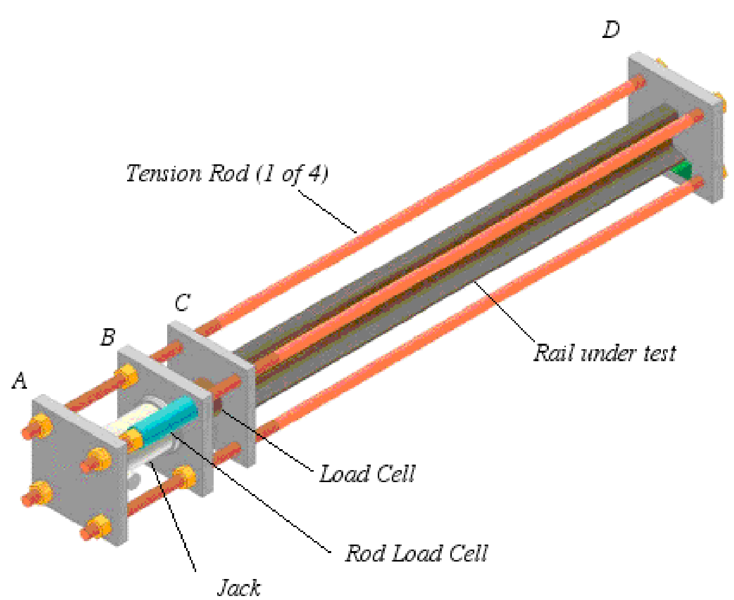

Figure 5]
The loading rig.

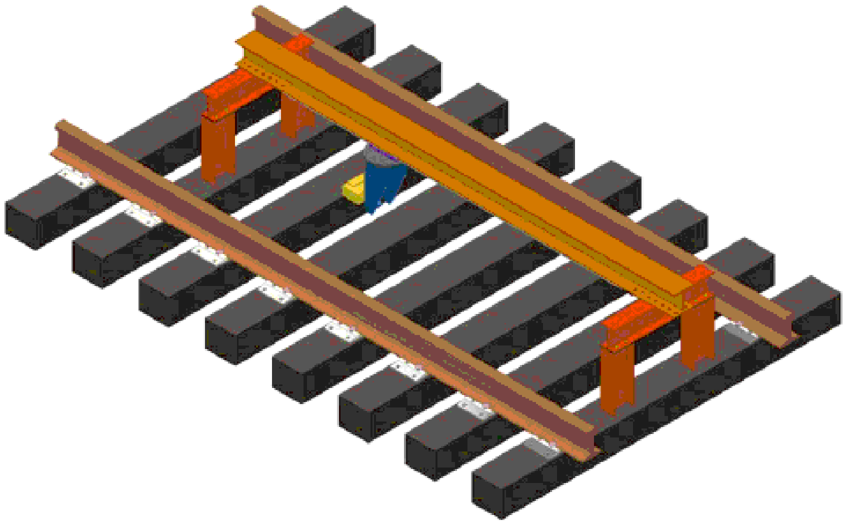

Figure 6]
The I-beam
frame to hold
the scanning
system for field
tests.

D. ADVISORY PANEL.

The advisory panel consisting of Hayden Newell (NS) Dave Davis (TTC, AAR), and Andy Kish (Volpe) met with Richard Weaver, Chris Barkan and Michael Koob by teleconference March 2004. The project staff summarized the conclusions of HSR-30 and outlined the plans for HSR-48.

Weaver noted that the requisite precisions, for measured and reference wavenumbers, may be relaxed by the simple expedient of scanning longer ranges; precision demands drop rapidly with longer scan regions. The panel was of the opinion that keeping the scan region small, enough so that one needs to remove no more than 3 fasteners, is a good idea. This continues to be the target.

The panel asked how the apparatus would be mounted in practice. Currently it is positioned by hand at the rail side, about 18 inches from the rail. We envision a practical device that would be cantilevered from a high rail vehicle or from a truck parked on the side. Mechanisms would be used to automatically align the scan direction parallel to the axis of the rail, thus assuring a scan's constant height in field tests. The Panel pointed out that worn rails have a) a shape very different from AREMA standard and b) lots of head-checks on gage corners, and 45-degree hairline cracks. The Panel questioned whether a reference wavenumber obtained in Finite Element predictions based on measured rail profiles, and an assumed Young's modulus, will be sufficiently accurate. Might this be a serious concern that would require tests with longer spans? How accurate is profilometry?

Sensitivity calculations establish that the technique (when conducted with ~ 1 meter scan lengths) will be able to tolerate a 1% error in the effective Young's modulus and a 1/2 % error in radius of gyration (corresponding to about half a mm in head width as measured by profilometry). Experts generally feel that profilometry is conducted well within the stated precision, and Young's modulus is known to the required precision. It was noted, however, that profilometry would need to be conducted at each test site, as profiles can vary from site to site.

The presence of hairline cracks is another matter. Deep or large enough cracks could diminish the effective natural bending rigidity and cause the FEM prediction of reference wavenumber to be in error. One Panel member thought it unlikely to be an issue.

It was further noted that the proposed "two-frequency" technique which promises to deliver stress and intrinsic rigidity without the need for a highly accurate FEM code, would obviate concerns regarding rail shape, wear and Young's modulus. There would be little or no need for profilometry.

The Panel urged that further tests be conducted on worn, and cracked, rail. Need for work on rails of very different standards was perceived as less critical.

The erratic shifts seen in the field tests were thought interesting. The panel whether such shifts are an indication of the state of the structure and if being able to see such shifts could be a fringe benefit.

The panel asked how long scans take. With the present apparatus, they take about half an hour for 100 scan points, at 2 seconds of data at each point. Thus there is about 200 seconds of actual data, and 27 minutes in which the operator is repositioning the laser. In principle, with automatic repositioning, this could be reduced to a total time of about 4 minutes. If electronic and other random noise was sufficiently low (or vibration amplitude was sufficiently high in comparison), the need for 100 points at 2 seconds of data each, could be reduced, further reducing the time needed. The minimum time needed for data taking is likely to be at least a minute per scan. Several scans may be needed. The use of multiple lasers could accelerate the process.

It was appreciated that installing strain gages, cutting and rewelding is expensive, but also appreciated that it will be necessary before the technique can be proofed. The Panel agreed that eventually the project personnel and apparatus should go to TTC and/or a Revenue Service mega test site to do extensive cutting strain gaging and rewelding for such proof tests.

There was a general sense of a preference for the two-frequency technique, as opposed to using a (probably huge) catalog of rail profiles, or a real time FEM run to predict reference wavenumber. The 2-frequency approach, is less dependent on assumptions about rail properties, and does not need profilometry, which could make the test procedure less complicated

It was also suggested that removing fasteners could change the load, thereby regularizing force distributions, and eliminating extreme values as the rail finds a local equilibrium. On the other hand, removing fasteners is unlikely to change load by large amounts, and would change it only by amounts that could happen anyway as the rail shifts. Of course uplift suffers from the same caveat.

E. LABORATORY WORK

Extensive work designing and developing and proofing hardware and software was carried out. Chief amongst the anticipated challenges was the presence of small slow variations, or "drift," in the response of our laboratory rail to a nominally steady dynamic loading. Uncontrolled variations (up to several percent) of its magnitude are unacceptable. The drifts were most severe in the field, (see figure 7) but also present even in well-controlled laboratory circumstances. Drifts were erratic and unpredictable. They were severe in a second floor laboratory where it was found that the position of persons in the lab changed transfer functions (the data points shown in fig 4) by 1% or more. This was a surprise, and eventually ascribed to changes in the structure consisting of rail, loading rig and floor as people moved. Drifts were seen also in the basement test bay, which had a thick concrete floor. Changes were less there, but remained problematic. Position of nearby persons did not affect the data there.

Drifts were ascribed to changes in structural supports. They were controlled in the test bay to a large extent by eliminating air-gaps between rail and bulkheads C and D by inserting lead foil. Gaps between bulkheads and the floor were filled by HydroCal (the white patches seen in fig 3.) Drifts were seen in the field also, where they were generally stronger, and ascribable to structural changes near the section in test, and maybe also to distant sections. Rather than eliminate them by rigidly fixing the rail supports, (probably impossible in the field if difficult in the lab) we hypothesized that by monitoring the rail at several (up to four) distinct positions with fixed accelerometers, we could monitor those changes. The algorithm with which wavelength was then determined from the vibrometer and accelerometer data was then modified so as to use the several accelerometers. It was found that the use of only one reference accelerometer was almost never satisfactory, but that the use of two or three generally sufficed to yield clean fits of the data to the hypothesized sum of guided waves. Thus changes in the relative amplitude and phase of the accelerometer signals permitted assessment of changes in the structure sufficient to correct for the drifts. We count this as a great success; the scheme will be vital to any continued development, as structural drifts are probably unavoidable.

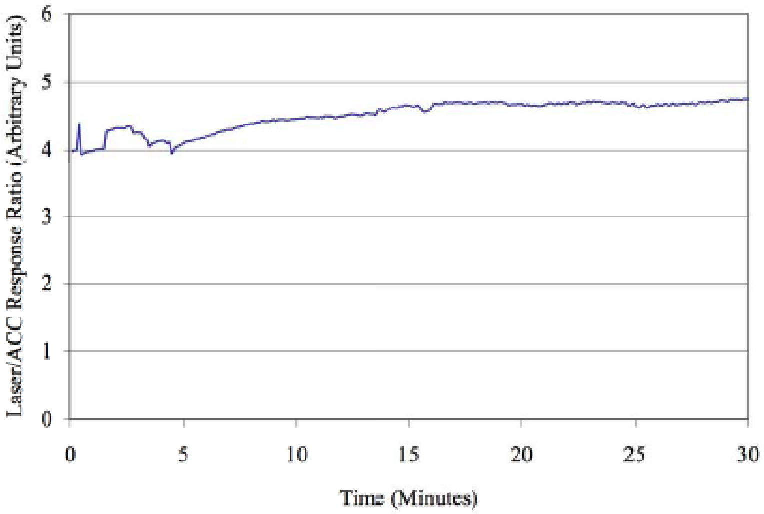

Figure 7] An illustration of "drift." These data were taken from field measurements with the laser supported by the I-beam frame on the ties, as in figure 1. The ratio between the amplitude of one accelerometer and the fixed vibrometer varies, sometimes erratically over a period of minutes.

The other chief anticipated challenge was the need to determine intrinsic bending rigidity as well as load. We hypothesized that the measurement of wavelengths at two or more different frequencies would allow this. Methods for using multiple frequencies were developed, and first proofed on circular rods in tension (figure 7) and then on rails in compression. (Fig 8)

The laboratory scan apparatus was further refined when we discovered and corrected a software error that led to a small inconsistency between the laser's position as reported by the encoder and its actual position. The vertical position motion controller was re-configured to allow it to operate when the scan was done from the I-beam frame. Data then appeared to be much cleaner and the fits were better. The reason for this improvement is not understood, however.

F. FITS OF THE LAB DATA

We successfully recovered stress in laboratory rods (figure 9) and rails (figures 10 and 11) in blind tests, where the person analyzing the vibration data was not told the actual stress until after he had reported his best estimate. Nor did the operator assume any intrinsic rod rigidity. For each test, the operator would set and measure (and monitor) compressive load. Two or three frequencies were chosen, often different for different loads. Frequencies were chosen to lie in the range from 70 to 200 Hz, but the precise value was chosen so as to have strong signals. For each frequency, the PC's LabView Virtual Instrument controlled the laser position, over scans at three different heights, each scan consisting typically of 90 two-centimeter steps over a distance of 180 cm. Each run produced a data file with identifying information, frequency chosen, heights chosen and, for each of up to 270 positions, the (locked-in) reading from each of the four accelerometers and the laser vibrometer. This data file was given to the operator of the fitting software, who analyzed it and extracted a best estimate for load.

Similar blind tests on unworn 136 pound rail (figure 4) recovered stress with the desired target precision of 10 degrees F, a figure chosen in the early stages of HSR-30**.** It was based upon what we were told the railroad engineers could live with, and what we deemed a technically achievable target. In worn 136 pound rail, (figure 5) there were occasional large differences, indicating need for further understanding and refinement of the algorithm or data taking. In the circular rods, stress was recovered with a precision corresponding to 4 degrees F (rods pose less of a challenge to the technique than rails, as they are more slender, and as their torsional waves may be neglected in the analysis).

These tests showed that the use of two or more frequencies did indeed accommodate one of the challenges initially anticipated. Bending wavelength depends, in a theoretically known way, on both intrinsic rigidity and contained load. That dependence is different at different frequencies. By measuring wavenumber (= 2π/wavelength) at two frequencies one in principle may deduce both intrinsic rigidity and contained load. This hypothesis was confirmed by these laboratory tests. A straight-line plot of frequency squared divided by wavenumber squared, *versus* wavenumber squared, indicates the intrinsic rigidity through its slope, and the contained load by its intercept. This is illustrated in figure [8] below, taken from test "E" on a rod.

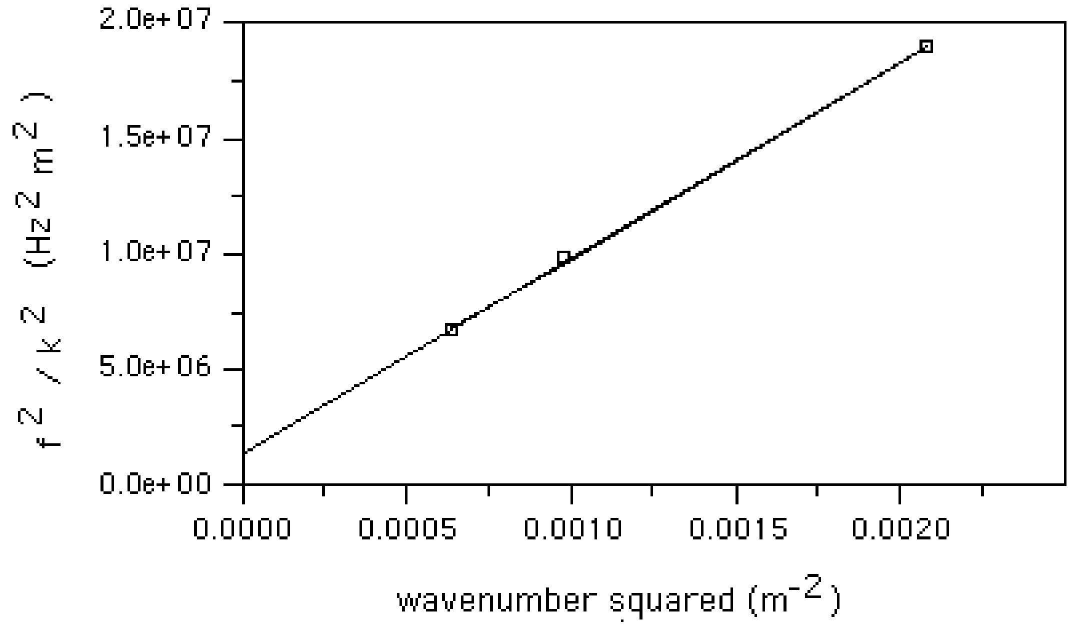

Figure 8] When recovered wavenumber k is plotted versus frequency in this format, the points will, according to theory, lie on a straight line with a slope that is a measure of intrinsic rigidity, and an intercept that is a measure of contained load. Here that hypothesis is well supported by a set of tests, at three different frequencies, on a rod in tension.

This work also established that the use of multiple reference accelerometers was a useful modification that provides compensation for drift. Rod and rail data from up to four fixed accelerometers were taken simultaneously with data from the vibrometer. When fits were attempted using only one of the accelerometer records, fits were almost uniformly bad. Residuals between the data (as in figure 4) and the fits were large, with systematic fluctuations of 2% or more. When two or more accelerometers were used in the fit, residuals were much lower, often as low as 0.2%, thus indicating a much better fit. That two accelerometers usually sufficed (use of three or four often did not lead to further improvements) is a strong indication that there were only two degrees of freedom to the secular changes in the vibrations. One is presumably due to unavoidable changes in force amplitude from the shaker, another due to an irregular one-degree of freedom change in the structure, for example, a change of stiffness at one point.

That blind recovery of contained load was more difficult on the worn rail (figure 10) deserves some comment. That the recovery would be less good, that the measured loads were going to be less reliable, was clear to the person doing the fits of the vibration data. That process saw greater residuals than with the unworn rail, especially at the higher frequencies. This observation, together with the better recovery on unworn rail, strongly suggests that the fitting procedure was using incorrect information on the "tilt" of the vibration modes. Bending waves are not purely composed of lateral displacements; they include some tilt. Similarly, torsional waves include some lateral displacements. The amount of mixing is greater at higher frequency, and varies with rail profile. The fitting procedures used mixings as predicted by the FEM code for an AREMA standard 136RE rail profile. Apparently this was accurate enough for the unworn rail, but not for the heavily worn rail used to generate figure 10. The fitting procedure adjusts its assumed torsional wave speed to get a best fit, so inaccurate FEM prediction for that parameter is not a culprit. Attempts to adjust the amount of tilt led to instabilities in the fitting. Further understanding of this effect is needed. Possibilities to correct for this include (a) FEM estimations of tilt for several degrees of wear, (b) theoretical estimations of tilt as a function of frequency and wear, or (c) making measurements only at lower frequencies where predictions of tilt are more reliable. Indeed, the fit qualities of both worn and unworn rail were best at the lower frequencies where this uncertainty is less.

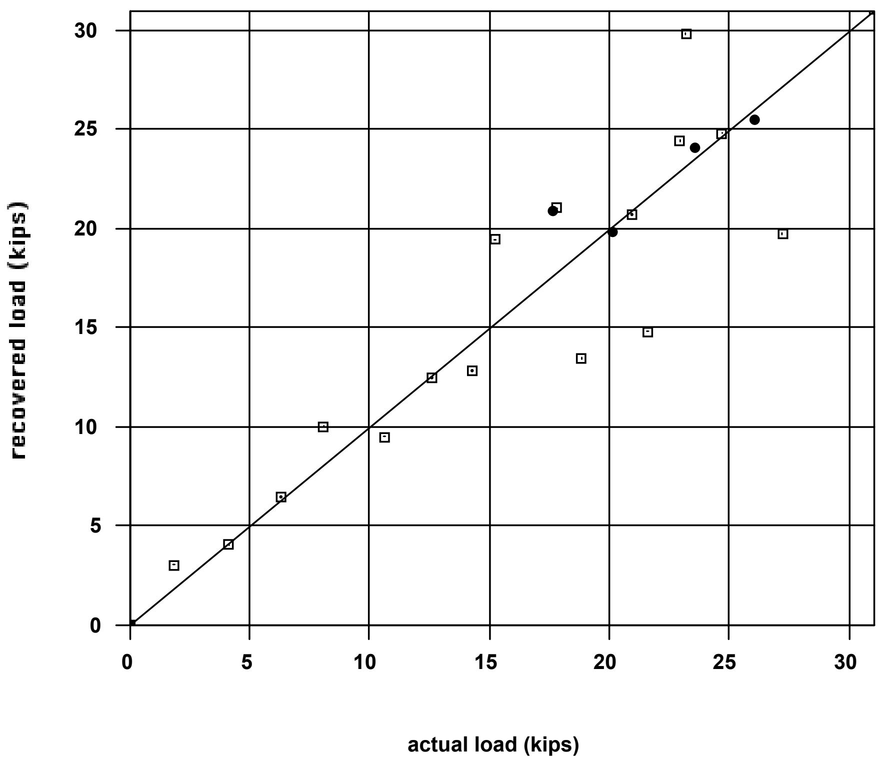

Figure 9] A plot of actual rod tension (in kips) *versus* the blindly recovered tension, for several distinct loads on a circular rod. Load is recovered with good accuracy using two or more reference accelerometers and scans at two of more frequencies. The four darker data points correspond to tests run with higher dynamic loads (and with a clamp to hold the shaker head to the rail in spite of the higher loads.) We conclude that higher dynamic loads improve the technique.

G. FIELD TESTS

Field tests began with attempts to study and control drifts in the field. They were found to be higher there than in the lab. See figure [7]. We attempted to minimize them by the use of HydroCal and by the use of shims between rail and tie plate, and by weighting the rails so that they made unambiguous acoustic contact with the ties. We did not succeed at eliminating them, but did learn how to make them smaller. Figure 7 shows a 15% drift. After various techniques were applied, this could be reduced to about 1%, comparable to that observed in the lab. As discussed above, we found that drifts could be corrected for, at least in the lab, by the use of multiple reference accelerometers. So we did not make further attempts to eliminate them in the field.

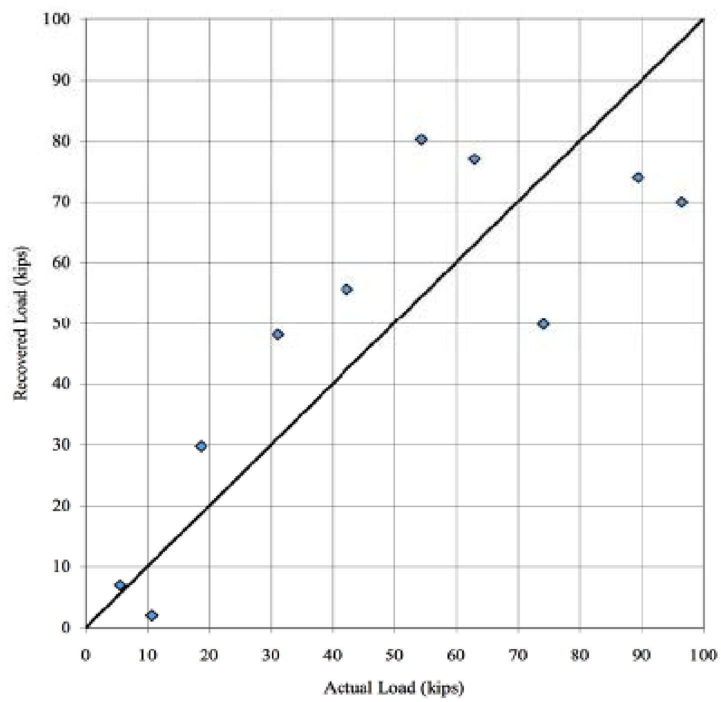

Figure 10] Blind tests on unworn 136 pound rail. The differences are almost all within the desired 10 degrees F (i.e. 25 kips). The technique used was the two (or more) frequency method with multiple reference accelerometers.

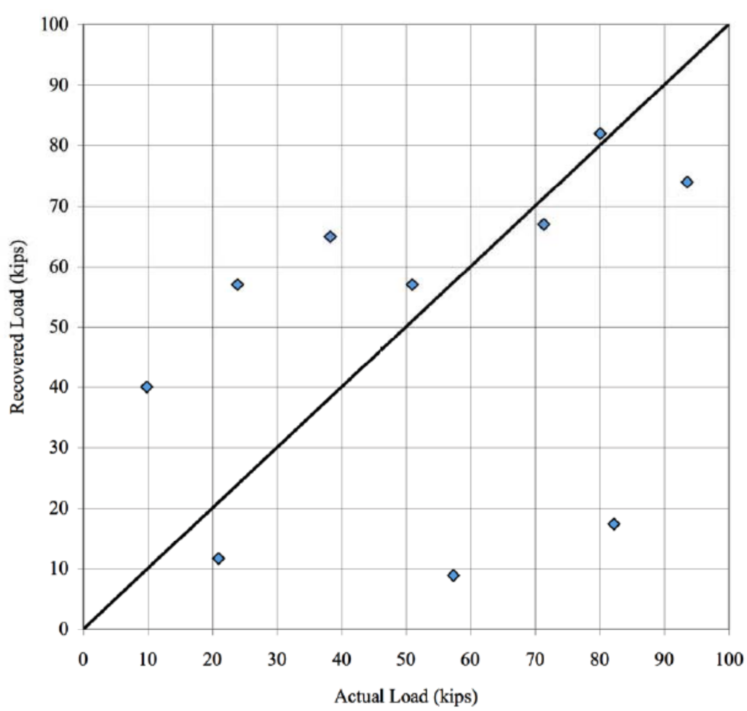

Figure 11] Blind tests on worn 136 pound rail. Two of the differences are well outside the desired 10 degrees F (i.e. 25 kips) target range.

We also found that vibrations were not confined to the rail, but managed to migrate into the ties. If the laser scan frame is supported on the ties (as in figure 1), there is a possibility that vibrations could enter the laser and contaminate the signals. Therefore it is recommended that the laser frame be supported on the ballast, or methods developed to better isolate the laser from the vibrations. (Vibration isolation is a well developed technology.)

Data were collected on a CN Railway mainline siding in Rantoul, Illinois. CN had installed strain gages on this section. The rail was supported on timber ties nominally spaced every 20 in (51 cm) with cut-spike fasteners. The ambient air temperature was 89°F and the rail temperature was 117°F. The rail neutral temperature was 92°F. This corresponds to a longitudinal force of approximately 60 kips compression at the time of the test. Fasteners and anchors were removed and data collected over a scan distance of 150 cm at three heights and at four frequencies, 50.00 Hz, 55.00 Hz, 136.00 Hz, and 155.00 Hz. As in the lab, four fixed accelerometers were used to monitor drifts, and the shaker was clamped (as opposed to magnetically attached) to the rail in order to enhance the maximum possible applied dynamic force. Data acquisition and control was conducted from the laptop in a small flatbed trailer next to the track. Use of the trailer enabled setup and configuration of the prototype next to the track while awaiting permission to be on the track. Actual track time available was short, as traffic that day was heavy.

Signals were very noisy, in large part due to sun glare in the laser vibrometer. The scan length was only 150 cm (as opposed to the lab where it was typically 190 cm, cf figure 4). We emphasize that accuracy is highly dependent on scan length. The scans at 50 and 55 Hz were too noisy to analyze. Fits were possible at the two higher frequencies. Being so close, though, extrapolation of the data at 136 and 155 Hz by a straight line fit to its intercept was unreliable. No reliable information could be recovered about rail load from such an extrapolation. The two data points at 136 and 155 Hz, together with an assumption of unworn rail's intrinsic rigidity, led to estimates of 20 kip tension and 78 kip compression respectively, comparable to the known 60 kip compression.

Data were also taken at the TTCI facility in Peublo for a few days in October, 2005 where blind-tests on lightly worn 141-pound rail were conducted. The vibration data from Peublo was striking. It showed intermittent loud clicks and bangs, features we had never before seen, in the lab or in the field. This was perhaps due to the rapid heating the rail experienced every morning. Had we had experience with this before, we would have known how to handle it (The data was such that it would have been easy to retake any datum that was corrupted by a bang.) It is a testimony to the robustness of the basic approach that the data was at all useable. Small bangs like this ought be, in principle, tolerable. They merely increase the strength of the vibration. Very strong bangs, however, will contaminate the signal by contributing significant energy at frequencies other than the driving frequency. The result of the contamination was that only one of the three chosen driving frequencies gave sensible fits. Thus the "two-frequency method" we developed could not be used. In order to infer stress, the operator had to *guess* the intrinsic bending rigidity of lightly-worn 141-pound rail. The stress inferred by this approach was later found to differ from the actual stress (as determined by referenced strain gages) by about 80 kips. See figure 12. That our measurement technique was working as properly as could be hoped under the circumstances can be seen by noting that the change in stress during the morning was correctly determined. It is therefore possible to attribute our error to the guess. Were the intrinsic rigidity assumed differently, all three data points could be made to lie within 20 kip of the known load.

In retrospect, the Pueblo testing was premature, as the problems identified in the lab testing and at Rantoul had not been resolved.

CONCLUSIONS

That the method works in the laboratory on unworn 136-pound rail is established. Data taken over a period of ~ 10 minutes reveal stress. It works less well on heavily worn rail or rails of other weight - due to our still incomplete understanding of the different shapes ("tilts") of the higher frequency guided waves in such rails.

The corresponding need is for improved algorithms for the extraction of wavelengths from the vibrometer and accelerometer data on worn rail and on other rail weights. The current algorithms work well on unworn 136-pound rail and on any rail at sufficiently low frequency. That stress in worn 136-pound rail in the lab was extracted with less precision than had been targeted (see figure 10) testifies to this. The difficulty is due to the presence of complicated torsional waves and "tilt" to the bending wave. In unworn 136-pound rail, we have a good understanding of these effects from our FEM code. At low frequencies, these effects are unimportant. Thus the need is either for restriction of our method to low frequencies (and perhaps to longer spans) or for improvement of the algorithm. In the PI's opinion, the latter approach is viable, and the former approach, though simpler, raises questions about how much unfastening is practical.

Little can be definitively concluded from the field tests. That the method worked as well as it did – correctly showing the changes in stress in the course of the day - in spite of the noisy environment and having to work on a rail weight with which we had no experience is a testimony, in the PI's opinion, as to the method's promise. Given further opportunity to learn how to accommodate such environments, we could no doubt do a better job.

PLANS FOR IMPLEMENTATION

Indications for the next stage of work are clear. There are two chief thrusts for such future work.

The algorithm by which the vibration data is analyzed needs to be further refined to reliably accommodate variations in torsional and bending wave shape (mostly their amount of "tilt") different from those of unworn 136-pound rail. Alternatively, the method needs to be restricted to lower frequencies, and corresponding longer span lengths. Extensive field experience is needed. We have no immediate plans to continue the development.

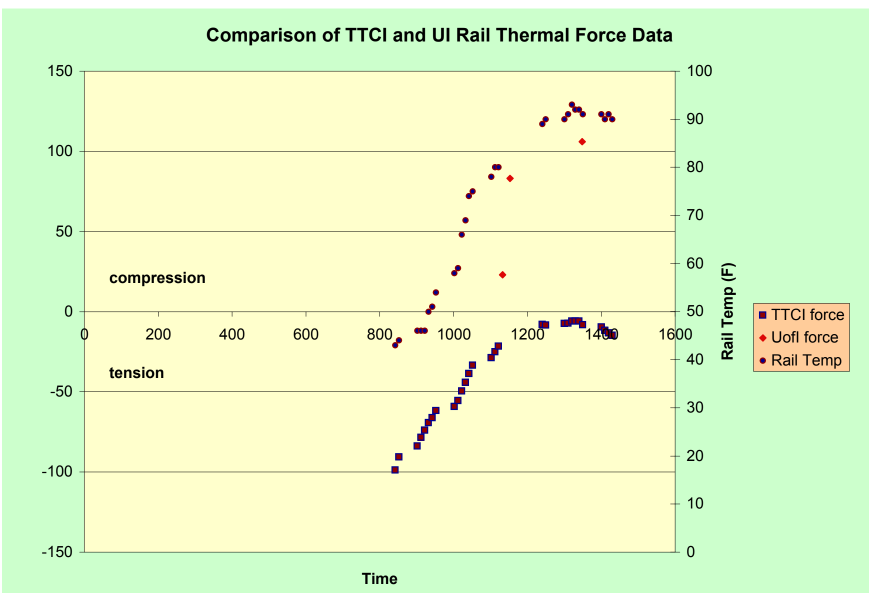

Figure 12] A plot of the rail force as determined by the vibration scan technique during one morning in October 2005. The temperature of the rail and the known load (from strain gages) are also plotted.

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