WOOD TIE FASTENER PERFORMANCE REQUIREMENTS THE USER'S POINT OF VIEW

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INTRODUCTION

The wood tie/cut spike fastener system has been in use on North American railroads since the 19th century (2). However, as the nature of the railroad-operating environment has changed over the last decade, there has been an increasing emphasis on long heavy trains with high axle loading. This has placed a severe burden on the ability of the traditional wood tie-cut spike system to perform its critical functions, particularly on the most severely loaded track locations, such as high curvature and heavy grade territory. In some cases, this traditional wood tie/cut spike system has proven to be inadequate. Furthermore, this trend towards increased axle loads is continuing, as the industry looks to heavier 286,000 lb. and 315,000 lb. cars in the near future.

As this increase in railroad loading continues to place an increasingly severe burden on the traditional cut spike fastener system, the very nature of the cut spike system has been examined from the point of view as to whether this system, or its individual components are appropriate for these heavy-duty load environments. This is particularly true since the failure mechanisms, associated with these severe track locations, appear to be more closely associated with the fastening system (i.e., the cut spikes), than with the wood tie itself. In fact, the basis properties of the wood tie, such as its natural resiliency and electrical isolation, appear to be quite desirable in the heavy duty-loading environment. This, in turn, suggests that the basic wood tie could function in even the most severe environment, provided that an appropriate fastener, or even more effectively, an appropriate tie/fastener system, is utilized. The "elastic track fastener" appears to represent such an improved fastening system.

This paper will attempt to define the performance requirements for such an elastic fastener system intended for use in a severe railroad environment.

CATEGORIES OF PERFORMANCE

The performance requirements for a freight railway elastic fastener system can be divided into two broad categories:

* Track strength requirements
* Track performance requirements

The track strength requirements relate directly to the ability of the fastener/tie system to adequately and effectively perform its functions under the defined traffic and environmental loading conditions. This is the "strength" or load carrying capacity of the system, and includes the full range of track loadings: longitudinal; lateral (both gage widening and lateral track); and vertical.

The track performance requirements, as used here, refer to those factors (often non-quantifiable) that relate to the ability of the system to accommodate itself to railroad practices and operations. These include system life requirements, maintenance considerations, and operating considerations.

In the case of the first category, the track strength requirements, it is necessary to first define the load environment under which the fastening system must perform. Based on this load environment, the actual "strength" requirements can then be defined.

TRACK LOADING ENVIRONMENT

The performance requirements for heavy axle load freight operations must be based, at a minimum, on the current maximum allowable vehicle weight (loaded) of 263,000 lbs for free inter-change service. This translates to a static wheel load of 33,000 lbs, which can be used as a starting point for the loading environment used in the determination of the fastener performance requirements. However, noting the current trends towards heavier axle load operations (3), these loadings should be used with the understanding that future freight car loadings may increase to 286,000 lbs and possibly 315,000 lbs.

The track-loading environment can be divided into three parts, corresponding to the three major loading directions: vertical load; lateral load; and longitudinal loads. These loadings are illustrated in Figure 1.

VERTICAL WHEEL AND FASTENER LOADS

The vertical load is the load applied to the top of the railhead of both rails, normal to the plane formed by the two railheads. The vertical load experienced by the track structure is in fact a combination of the static wheel loading (due to the stationary weight of the car), together with dynamic effects associated with vehicle speed, impacts, and curvature/elevation.

The actual vertical loading experienced by the track (and therefore by the fastener/tie system) is a combination of the loadings of each of the various car and locomotive types, and is therefore a distribution or spectrum of loadings (Figure 2, Reference 4). However, for the purpose of developing performance requirements, it is appropriate to simply define the maximum expected wheel loads. This can be done by using the maximum measured dynamic loading, or by using the nominal static loading and developing equations or factors to take the various dynamic effects into account.

As noted above, the maximum static wheel load currently permitted in free interchange is the 33,000 lb. wheel load. This wheel load is generally restricted to 36-inch diameter wheels.

In order to account for the dynamic effects, associated with speed, the AREA speed effect formula (5) is defined, where:

$$\text{Dynamic load} = \text{Static load} * (1 + 33V/100D)$$

where V= speed in mph
D= wheel diameter in inches

Thus for a 100 Ton car (33,000 lbs wheel load) operating at 55 mph, this produces a dynamic wheel load of 49,500 lbs.

In addition to this speed effect there can also be a dynamic impact effect, usually associated with anomalies or defects on either the wheel or the rail surface. These impact effects are specifically associated with wheel imperfections (such as wheel flats, out-of-round wheels, etc.), and rail defects (such as corrugations, engine burns, battered joints, battered welds, etc.). Severe defects have been found to result in dynamic impact factors or multipliers (6) of between 2 and 3 (or even higher in the case of extreme defects). This type of impact effect will result in a dynamic impact load of 66,000 to 99,000 lbs per wheel. However, the duration of these impact forces are extremely short, often as low as 5 to 15 milliseconds.

An alternate approach to the definition of the vertical wheel loading is the selection of a value, based from data measurements taken in the field. Such a set of load measurements is presented in Figure 2 (4), which shows the distribution of loading for various types of track and traffic conditions. Specifically noting the freight traffic on tangent wood tie track curve, the data shows that the dynamic wheel load value of 48,000 lbs has a cumulative probability of 99.8% (i.e. 99.8 % of all measured wheel loads were below this value), and the dynamic wheel load of 58,000 lbs has a cumulative probability of 99.99%.

In order to determine the corresponding vertical fastener load, the track can be assumed to be represented by a continuously supported structure, where the support is the ballast and sub-grade (7). Then, using the beam-on-elastic foundation theory, the load distribution between the rail and the supporting ties can be determined together with the proportion of the wheel load carried by the tie directly under the wheel. The results of such an analysis are presented in Figure 3, which shows this percent- age as a function of track support condition or track modulus and rail section. As can be seen from this figure, the individual tie (and thus fastener) receives 15% to 45% of vertical wheel load. This corresponds to a fastener vertical load of 22,500 lbs (dynamic) for the above noted dynamic wheel loads.

LATERAL WHEEL AND FASTENER LOAD

The lateral wheel load is the load applied in the lateral plane, as illustrated in Figure 1. This load can be applied either at the gage face of the railhead or at the top of the railhead, depending on the actual loading mechanism. The lateral loading is related to the manner in which the vehicle (and particularly the vehicle truck) negotiates a curve, and as such, there is a strong correlation between increasing lateral load and increasing curvature.

In order to define the lateral loading effectively, field measurement data is frequently used. It should be noted that test data shows that the lateral loading of a locomotive is the largest lateral loading generated, and as such it should form the basis of the loadings used in any performance requirement. Such locomotive field test data for differing equipment is presented in Figure 4 (8,9). As can be seen in this figure, the maximum dynamic lateral loads (total) can reach values as high as 35,000 lbs for commonly used three axle truck road locomotives, such as the SDP40F.

In order to calculate the portion of the total load applied to an individual fastener, it is necessary to determine both the load transferred to the tie and the load transferred to the fastener.

The actual transfer mechanism consists of several parts. The first part is the distribution of the load, through the railhead to the tie directly under the load (in a manner analogous to that for the vertical load distribution). This can be done through a beam analysis using the lateral section strength of the rail to determine the load distribution. The single tie distribution can vary from between 25% and 50% of the total load, depending on rail size, fastener type (and torsional resistance), tie stiffness, etc.

The second part of the load transfer deals with the transfer of load through the fastener itself. A portion of the load is taken up by the frictional resistance of the base of the rail and the fastener railseat. The remainder is resisted by the fastener itself. In this transfer, the net lateral fastener load is equal to the net lateral load at the tie, (note above) minus the vertical load multiplied by the "effective" coefficient of friction between rail and fastener rail seat (10). In this case the fastener receives between 20% and 45% of the lateral tie load combining these two effects results in a net lateral fastener load (maximum) of 15,000 lbs.

L/V RATIO

The L/V ratio is the ratio of the vertical wheel/rail force and the Lateral wheel rail force. This ratio is an indication of the overturning force on the rail/fastener system, as well as the tendency for the wheel to climb the rail. The former case is defined by the overturning limit of L/V > .6, while the latter case is defined by the "Nadal" limit for wheel climb, L/V > .8 (11). Thus the range of L/V of interest is:

$$.6 < L/V < .8$$

LONGITUDINAL TRACK LOADINGS

Longitudinal forces in the track structure are induced through two distinct mechanisms: mechanical (train action) and environmental (thermal effects).

Mechanically induced longitudinal forces are directly related to train handling and operations. These include train acceleration, specifically at the traction wheels of the engines, and train deceleration or braking, either in the locomotive wheels only (dynamic braking) or at all the wheels in the consist (train air braking). Noting Reference 12, maximum mechanical forces of up to 60,000 lbs per rail have been recorded. However, more typically these forces are in the range of 20,000 lbs per rail.

Thermally induced longitudinal rail forces are caused by the changes in ambient (and thus rail) temperatures and the difference between the instantaneous temperature and the force free, or laying temperature of the rail. These forces, which can be either compressive or tensile in nature, can reach significant levels, as is illustrated in Figure 5 (13). These forces are the primary cause of movement at the ends of rail strings or at gaps in the rail, such as due to joints (desirable) or pull-aparts (undesirable). In curved track these longitudinal forces also contribute to the moving in or out of the curves. As can be seen in Figure 5, a 100 degree temperature change in 132 RE rail can generate a longitudinal rail force of 250,000 lbs.

TRACK STRENGTH REQUIREMENTS

As noted previously, track strength refers to the ability of the fastener/tie system to effectively perform its functions under the defined traffic and environmental loading conditions. This is the "strength" or load carrying capacity of the system, i.e., its abi11ty to carry the vehicle and environmental loadings without "excessive" deflection or deformation.

There are four basic areas of track strength performance, corresponding to the three principal loading directions presented in Figure 1 (13). These are:

a. Longitudinal strength
The resistance of the fastener/tie system to longitudinal loading, both mechanical and environmental (thermal)

b. Lateral strength: Gage Widening Resistance
The resistance of the fastener/tie system to gage widening.

c. Lateral strength: Lateral Shift Resistance
The resistance of the fastener/tie system to the lateral deformation of the track structure (alignment or buckling).

d. Vertical Strength

The resistance of the fastener/tie system to vertical loadings.

For each of these four track strength areas a performance requirement must be specified, usually in terms of a maximum allowable deflection for a given level of loading.

LONGITUDINAL FASTENER STRENGTH

Longitudinal fastener strength is the ability of the fastener system to provide longitudinal restraint to the rail, and prevent rail movement or creepage under all loading conditions. As noted previously, longitudinal loading can be due to train action, such as train or engine braking and/or acceleration, and to environmental action, specifically the variation in temperature, both rail and ambient.

In this mode, the function of the fastener is to provide longitudinal restraint to prevent any movement of the rail with respect to the tie. Under mechanical loadings, this is simply the case of each fastener, picking up a portion of the load up to its maximum capacity, until the entire longitudinal load is restrained. In the case of thermal loadings, the distribution of longitudinal force consists of three distinct zones (Figure 6), the two end zones or "breathing" zones in which longitudinal movement of the rail takes place, and the center "constrained" zone in which no longitudinal movement occurs (12). Therefore, the fastener longitudinal restraint is most critical in these end or "breathing" zones. These zones are at the ends of the Continuously Welded Rail (CWR) strings or at each side of a rail gap, such as occurs during a rail pull-apart or break. Good longitudinal restraint strength is required to minimize the size of these gaps.

Noting this, the minimum longitudinal restraint value is the restraint necessary to prevent an excessive rail end opening or "gap" at any discontinuity in the CWR. In addition, the fastener restraint is intended to prevent any excessive rail movement (longitudinal) between the rail and the cross-tie. This is particularly important in a critical "failure" situation, such as in the event of a pull-apart or rail break, where the break or gap must be controlled to avoid an excessive gap in the rail. Table 1 presents a set of calculated rail end openings or "gaps" as a function of the fastener longitudinal restraint (13). Based on a 132 lb. rail and 19.5 inch tie spacing for a 75 degree temperature change (such as would be encountered in a large portion of continental US), the required minimum restraint per rail seat is 1814 lbs/rail seat for a gap of I" or 2720 lbs/rail seat for a gap of 3/4". This defines a range for the minimum longitudinal restraint.

In order to properly quantify the range of restraint, it is appropriate to also specify a maximum longitudinal restraint value. In order to do this, it is necessary to note that the longitudinal restraint of the rail/tie fastener system should not be significantly greater than the resistance of the tie in the ballast. This latter value is the "plowing" resistance of the cross-tie in the ballast, as presented in Table 2 (14).

By designing the fastener/rail restraint to be greater than the resistance of the tie in the ballast, the tie/ballast inter- face becomes the weak link in the track structure, not the fastener/tie interface (as has often been the case in the past, when the rail "runs", due to the inability of the fastener to hold the rail). Based on this tie/ballast resistance data (Table 2), a resistance of 4900 lbs per tie or 2450 lbs per rail seat is an effective maximum limit for the longitudinal restraint of the fastener. This corresponds to a value of 1500 lbs/ft per rail seat. It should be noted that AREA (5) specifies 2400 lbs fastener restraint for concrete tie fasteners on 24 inch spacing. This corresponds to 1950 lbs on 19.5 inch spacing, or 1200 lbs/ft per rail seat.

Finally, it must be noted that these are static values, without any dynamic excitation of the track structure. The dynamic longitudinal restraint should be no less than 75% of the static longitudinal restraint values.

LATERAL GAGE STRENGTH

Lateral gage strength refers to the ability of the fastener /tie system to limit the amount of gage widening, both static and dynamic. This is an import parameter, since a key fastener function is to maintain track gage under loading (i.e., to pre- vent dynamic gage widening).

Gage widening is associated with three distinct mechanisms (15) as follows:

* Rail wear
  Abrasive wear on the railhead, particularly the gage face of the high rail. While primarily outside the scope of the fastener system, it is affected by fastener stiffness.

* Rail translation
  Rigid body movement of the rail, without any rotation (i.e., lateral movement of the base of the rail).

* Rail rotation
  Rotation of the rail about its longitudinal axis (i.e. overturning).

As noted previously, fastener strength can be defined in terms of deflection under loading with defined lateral and vertical wheel/rail, or equivalent tie/fastener loadings. For the performance requirements presented here in, a lateral fastener loading of 15,000 lbs will be defined under a simultaneous fastener vertical load of 21,500 lbs. This corresponds to an L/V ratio of .70. Note that the potential for rail overturning exists at this L/V, however, it is below the wheel climb limit.

RAIL WEAR

In general, rail wear is not significantly affected by the fastener system, except that a change in the fastener rotational resistance (torsional strength or overturning strength) will affect the dynamic wheel/rail lateral interactions, and thus the gage face wear. Limited test data (16) suggests that fasteners that are very stiff, torsionally, will result in a greater amount of gage face wear, than those that are somewhat softer (see Figure 7). However, on the other extreme, fasteners that are too soft torsionally, and which have large variations in dynamic gage with vehicle dynamic loading, will also result in excessive and non-uniform gage face wear (13).

In order to minimize this effect, the following range of torsional stiffness about the longitudinal axis is recommended:

Range of torsional stiffness = 1500 to 4000 in-Kips/radian.

The stiffness is defined as the Overturning Moment (the Force in thousands of lbs or Kips times the moment arm in inches) divided by the rotation of the rail about the longitudinal axis (in radians). Table 3 presents a sample of rotational stiffness values for wood tie fastener systems (17).

RAIL TRANSLATION

Rail translation is the lateral movement of the rail section, without rotation (i.e. the lateral movement of the rail base). For fastener/tie systems in "good" condition", this is not a significant value. This is illustrated in Figure 8 (18), which shows that rail translation is significantly smaller than the corresponding rotation in good track. However, as the track begins to deteriorate, this mechanism becomes quite significant. Noting this, the following performance requirements can be defined; under the above loading, rail base deflection should be for "new" condition, less than 0.060 inches per rail seat; for "old" condition, less than .10 inches per rail seat.

RAIL ROTATION

Rail rotation is the relative movement of the railhead with respect to the rail base. This mechanism generally accounts for the largest amount of railhead movement, particularly in track in relatively good condition (Figure 8).

Rail rotation is a behavior that occurs only under combined lateral and vertical loading, when the vector sum of the two loadings falls outside the base of the rail (15). When this occurs, the rail is potentially unstable and the fastener system begins to resist the overturning movement of the railhead. This condition occurs when the ratio of the lateral and vertical forces, the L/V ratio, exceeds 0.6, the exact value being dependent on the geometry of the rail section (15). Rail rotation is given in lateral movement of the railhead (in inches) with respect to the rail base. [Note, for total dynamic gage widening (under load), rail rotation (inches) must be added to the rail translation.] Once again, referring to the above-defined loadings for new fastener/tie systems, railhead deflection with respect to the rail base should be less than .25 inches per rail seat. For "old" systems, it should be less than .4 inches per rail seat.

DYNAMIC GAGE WIDENING

In order to effectively define the performance of the entire track structure, it is necessary to define the total gage widening under load (i.e. the dynamic gage widening). This is the total gage widening of both rails under loadings. Note, that for total dynamic gage widening (under load), rail rotation (inches) must be added to the rail translation (in inches) for both rails. As defined here in, the dynamic gage widening values do not include rail wear. They are based on a new railhead only.

Once again noting the above levels of loading, the total dynamic gage widening, new, should be less than 0.4 inches. Likewise, the old value (i.e. after the defined "life") should be less than 0.75 inches.

LATERAL SHIFT RESISTANCE

Lateral shift resistance is second lateral strength mechanism, which is the lateral movement of the entire track structure (i.e. the lateral shift of the track). This lateral shift can be either slow, such as occurs with a loss of alignment, particularly on curves where the lateral loadings are most severe, or it can be abrupt, such as due to a sun kink or track buckle. Lateral resistance of the track structure is a key aspect of this deformation behavior in both cases.

Lateral track resistance is primarily related to the tie/ballast resistance mechanisms, rather than the rail/tie mechanisms (19,20). Improving the resistance of the tie in the ballast provides a significant increase in resistance to lateral shift. However, the fastener system provides a secondary effect in that it affects the frame strength of the track, and consequently its lateral resistance (21). This effect is due to the in-plane torsional resistance of the fastener (i.e. the torsion- al resistance about the vertical axis), which directly affects the frame strength of the track structure. While the effect is of a secondary nature, a fastener in-plane torsional resistance of 3000-5000 in-Kips/radian can provide an improvement to the lateral strength of the track frame.

VERTICAL STRENGTH

Vertical strength refers to the ability of the tie/fastener system to respond to loadings in the vertical plane. As noted earlier, Figure 2 presents a spectrum or distribution of vertical wheel rail dynamic loadings that is representative of the dynamic loading environment of the track structure. These loads are then transferred to the individual tie/fasteners.

In order to properly function in the vertical plane, the fastener system must possess the following "strength" capabilities: Uplift strength; Static Vertical Strength; and Dynamic Vertical Strength. These are defined as follows:

- UPLIFT STRENGTH

Uplift strength is the resistance of the fastener system to vertical uplift (upward) forces that can cause pullout or yielding of key fastener components. These can be due to either the vertical component of the rail rotation, or to the vertical uplift forces due to the track "uplift" wave (from beam on elastic analysis), and the corresponding ability of the fastener to support the track superstructure (rails, ties, fasteners). This latter uplift resistance is associated with elastic fastener systems, which do not permit a "floating" of the rail with respect to the tie.

The effect of track uplift, 'due to the track acting as a beam on elastic foundation, provides a force of approximately 600 to 850 lbs/foot of uplift, depending on the track modulus. [Note that this corresponds to 20% of the effective downward force.] However, for wood tie track, this is more than the weight of the rails, ties, and fasteners, whose weight is "of the order of 350 to 400 lbs/foot. Therefore there is a net upward force of over 100 lbs/foot. In addition, the resistance to rail overturning (i.e., the uplift component of rail rotation), must also be considered.

- STATIC STRENGTH

The static strength of the tie/fastener system is based on the crushing strength of the wood under the applied vertical loading. This loading, when converted to a per railseat vertical force, is used to determine the minimum bearing area of the fastener system on the tie necessary to avoid crushing of the tie fibers.

Noting that the crushing strength of hardwood (specifically those used in Railroad applications) ranges from 300 to 500 lbs/square inch (13) the vertical fastener load of 21,500 lbs requires a minimum fastener bearing area of 75 square inches. For softwood ties, this value increases to 100 square inches of fastener bearing area.

- DYNAMIC STRENGTH

As noted earlier, dynamic fastener loads (vertical) can be as high as 21,500 lbs per rail seat. This does not take any dynamic impact sources into account, such as defects on the rail surface (corrugations, engine burns, battered welds), rail joints, or imperfections in the wheel tread (i.e., flats or out- of-round wheels). These defects can magnify the dynamic loads by factors of 2 to 3, thus providing an instantaneous vertical load per rail seat of the order of 66,000 lbs, but of very short duration. These forces must be transmitted through the tie/fastener system without failure of any tie or fastener component. For stiff track, such as concrete tie track, soft rubber pads are required to reduce the effect of these impact forces.

In the case of wood ties, the tie itself acts as a resilient pad to reduce the effect of these impact forces. This is a particular feature of the wood material itself, in that it has a very high natural resiliency. In fact, the wood cross-tie acts as if it were a 6 or 7-inch thick pad with good dynamic attenuation capability. Thus these dynamic impact effects are immediately reduced.

However, the dynamic strength of the fastener system must be such, that it can endure short duration dynamic impact loads of up to 66,000 lbs vertically, for a duration of 10 to 15 milliseconds, without component failure.

PERFORMANCE REQUIREMENTS

The previous section has presented a set of track strength performance requirements for the fastener and/or fastener/tie system. These strength characteristics are the directly quantified values, which can be addressed through conventional design engineering. However, in addition to these strength requirements there is a separate set of requirements; some not readily quantifiable, which address the operational and maintainability characteristics of the tie/fastener system. This is a set of "practical" characteristics that relates to the ability of the fastener system to accommodate itself to railroad practices, and to be used by "typical" railroad personnel.

LIFE

Fastener life has been defined to be the "period of time or cumulative tonnage until the fastener, or its individual components, must be replaced". In actuality, fastener life is an economic criterion as well as a design criterion, since there is a tradeoff between increased cost and increased life. However, from an operational point of view, there is in fact a minimum "practical" life that should be achieved by any rail fastener, in order to avoid excessive efforts on the part of the local maintenance of way forces. It is this aspect of the operational considerations that helps provide a justification for a "premium" type fastener, whose first cost is higher than a conventional system, but whose overall (life-cycle) cost can be significantly lower.

As was previously noted, "life" can be defined in terms of either cumulative tonnage (in Million Gross Tons) or in years. In the case of high density, severe environment freight operation, it is anticipated, that the cumulative tonnage life will be reached well before the time based (year) life limit. Therefore, the fastener systems must be capable of maintaining a defined level of performance after a minimum of 1,000 MGT or 30 Million cycles of loading (at 33,000 lbs wheel load per cycle), based on the tangent or low curvature mainline service. [Note that on curves rail life is shorter, however the fastener life should be equal to several curve rail life cycles.]

At the conclusion of 30 Million cycles or 1000 MGT, the "old" strength values must be at a level equal to 90% of the "new" strength levels defined for all strength categories, unless indicated otherwise.

ELECTRICAL ISOLATION

The fastener/tie system must provide adequate electrical isolation of the rail to prevent interference with signal systems, and deterioration of the fastening system through electrical leakage. This is particularly important for high-density trackage, since it is most likely to be signaled territory. The electrical insulation properties must be such as to maintain a minimum resistance value between the two rails in a wet and dirty ballast environment.

It should be noted here that wood is a natural insulator material. As such, the natural electrical resistance of the wood cross-tie is such, that it should provide sufficient electrical insulation without the need for additional external fastener components. This should hold true except for those cases where the fastener/tie system is such, that it provides a conductive medium through the ties and thus short-circuits the insulative properties of the wood ties.

MAINTENANCE CONSIDERATIONS

Maintenance considerations or maintainability relates to ease of use and acceptability of the fastener systems by railroad forces. These are unquantifiable factors that enter into the design and application of the fastener system.

EASE OF USE

An important consideration in the acceptability of a fastener system is the ease in which that system can be handled by field forces. This relates only to that portion of the fastener system that must be handled by field or gang forces, specifically the rail holdown portion of the fastener.

The connection between the fastener and the tie does not have to be "easy to use" if it is separate from the rail holdown system. In fact, a secure fastener holdown assembly is an advantage (it is an important part of the strength requirement as well).

Therefore, the rail holdown portion of the fastener system should be readily installable or removable by local forces in the field. It is n2t necessary for the field forces to remove or install the fastener to the tie. However, it is important that the local forces be able to readily remove and install the rail hold-down portion of the system, so as to allow for various field maintenance activities.

This operation should be capable of being performed in the field by local forces, using convention and commonly available hand tools. If special hand tools are required, they should be kept to a minimum level.

Finally, the amount of physical effort required to remove the fasteners with these hand tools, should be kept to a minimum. However, the fastener system should be resistant to tampering in the field by unauthorized personnel using rocks or other crude tools.

In order to avoid confusion and improper or incorrect installation in the field, the portion of the fastening system that must be removed and reinstalled in the field should be kept as simple as possible. In addition, the fastening system components should be readily identifiable, as to their location and direction of installation, so as to avoid improper installation of components and assembly of system (in the field).

POTENTIAL FOR MECHANIZATION

For large production applications, the industry trend is towards mechanization of all large production gangs. Therefore, it is desirable for the fastener system to allow for mechanization of the installation and removal process. As a result, the fastener system should directly lend itself towards mechanized application. This should specifically permit the development and use of production fastener inserters/removers within the overall context of a railroad production gang.

FIELD ADJUSTMENT REQUIREMENTS

In the railroad environment there is very limited capability for "follow up" adjustment of systems. Such a follow up entails high manpower costs associated with specialized maintenance gangs or local force activities. In order to eliminate or minimize these follow up costs, the fastener/tie system should not require any periodic adjustments after the initial installation and adjustment, such as bolt tightening, re-torquing, clip re-tensioning, etc. The system should be installed and should maintain its performance at the above defined performance levels without any additional attention on the part of the maintenance forces.

EASE OF INSPECTION

In order to facilitate field inspection of the track structure in general, and of the fastener system in particular, the fastener system should allow for ease of visual inspection of all key parts by local inspectors. This should include the ability to observe all components for breakage and/or movement from "proper" position.

ENVIRONMENTAL EFFECT

The fastening system must retain its ability to perform (as defined previously under track strength) for the entire range of environmental conditions encountered in the railway, including ice, snow, head, and cold.

DERAILMENT AND DAMAGE

While it is impossible to expect that any track system should be able to survive a derailment in which the full brunt of the derailment force is concentrated on the fastener, it is possible to design a system that minimizes the potential for derailment or other damage (such as damage due to other mechanized maintenance operations). Such design features can include a "low" profile (i.e. a profile significantly below the bottom of the railhead), and an orientation such that the fastener, even when "hit" by a wheel or piece of maintenance equipment, will not completely fail. The ability to maintain a minimum level of performance under such adverse circumstances is extremely desirable.

DERAILMENT PERFORMANCE

Noting the above, the fastener system should have a high "survivability" rate in the event of a derailment, and should maintain a minimum "emergency" level of performance even in a derailment environment.

In conjunction with this, the fastener system should allow for the replacement only of those parts of the system damaged during the derailment, while retaining "undamaged" components as much as possible.

VULNERABILITY TO FIELD DAMAGE

The fastener system should not be vulnerable to damage in the field, due to the operation of conventional maintenance equipment. Particularly "low" profile maintenance equipment, such as brooms and ballast spreaders, should not cause damage to the fastener system, nor should they result in displacement or removal of the rail hold down fasteners themselves.

REFERENCES

1. Railway Tie Association, Wood Tie-Fastener Performance Specification; An Engineered Wood Crosstie System Project, Mars, PA 1989.

2. Joyce, James M., Railroad Spikes, A Collector's Guide, Sutter House, Lititz, PA 1985.

3. Newman, R. R., Zarembski, A. M., and Resor, R. R., "The Effect of Increased Axle Loads on Maintenance of Way and Train Operations at Burlington Northern", International Heavy Haul Association/Transportation Research Board Workshop, Vancouver, B.C., June 1991.

4. Zarembski, A. M. and Abbott, R. A., " Fatigue Analysis of Rail Subject to Traffic and Temperature Loading", Association of American Railroads Report R-315, July 1978.

5. American Railway Engineering Association, Manual for Railway Engineering, 1991.

6. Ahlbeck, D. R., "An Investigation of Impact Loads Due to Wheel Flats and Rail Joints", American Society of Mechanical Engineers, 80-WA/RT-1, November 1980.

7. Hay, W. W., Railroad Engineering, Second Edition, John Wiley and Sons, 1982.

8. Koci, L. and Marta, H., "Wheel and Rail Loadings from Diesel Locomotives", AREA Annual Meeting, 1971.

9. Koci, L. and Swenson, C., " Locomotive Wheel and Rail Loadings -A Systems Approach", Heavy Haul Railways Conference, Perth Western, Australia, September 1978.

10. Manos, W. et al, "Development of an Improved Vehicular Loading Characterization Associated with the Gage Strength of Track", AAR Report R-493, August 1981.

11. Weinstock, H., "Wheel Climb Derailment Criteria for Evaluation of Rail Vehicle Safety", American Society of Mechanical Engineers, Paper 84-WA/RT-1, 1984.

12. Kerr, A. D., "Thermal Buckling of Straight Tracks; Fundamentals, Analyses and Preventive Measures", Bulletin of the American Railway Engineering Association, Bulletin 669, September 1978.

13. Zarembski, A. M., "Performance Characteristics for Wood Tie Fasteners", Bulletin of the American Railway Engineering Association, Bulletin 697, October 1984.

14. McConnell, D. P., Letter to J.D. Born, Chief Engineer Maine Central Railroad, July 1980, describing Federal Railroad Administration/Transportation Test Center Tests on U.S. Railroads in 1974.

15. Zarembski, A. M., " Rail Rollover, The State of the Art", Bulletin of the American Railway Engineering Association, Bulletin 664, September October 1977.

16. Mirtchev, M. and Nedjalkov, G., "The Influence of Railway Track Parameters on the Intensity of Side Wear on Rails on Small Radius Curves", Rail International, April 1981.

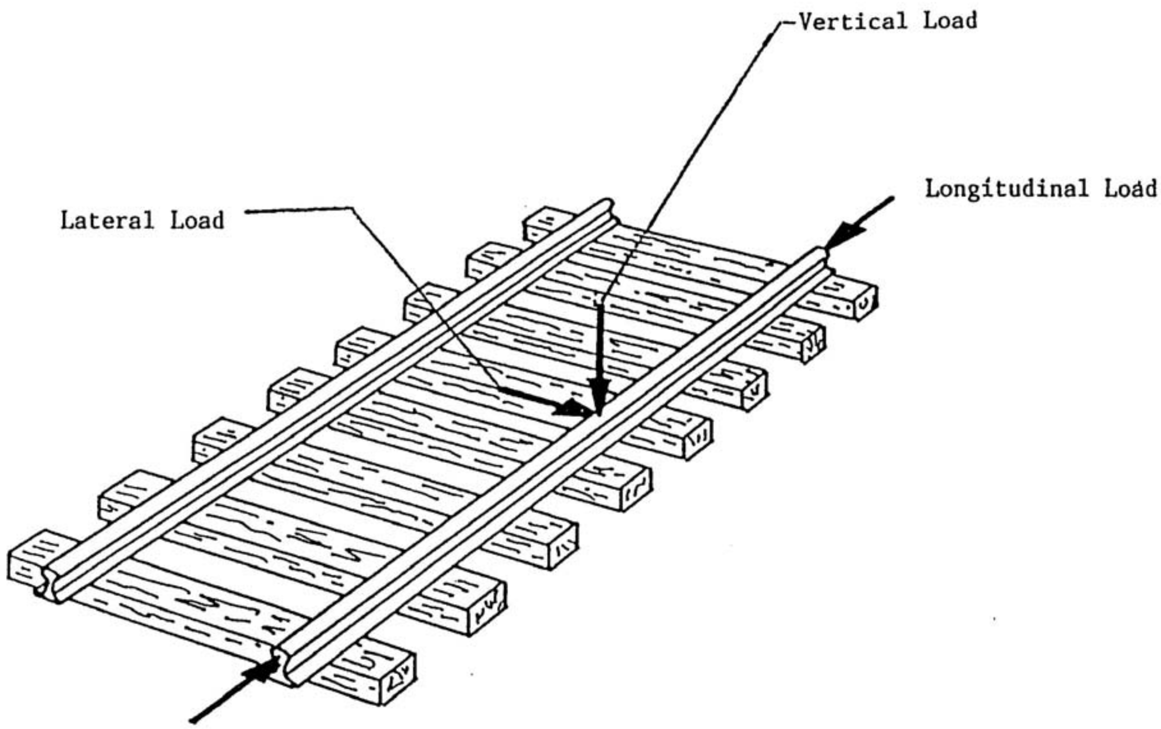
17. Choros, J. and Gitlin, I., "Track Component Property Tests, Volume II, Ties, Joint-Bars and Fasteners", Federal Railroad Administration Report, FRA/ORD-80/25, June 1980.

18. Zarembski, A. M. and Choros, J., "Laboratory Investigation of Track Gage Widening", AAR Report R-395, September 1979.

19. Choras, J., "Laboratory Tests on Three Alternative Track Structures", AAR Report R-614, September 1985.

20. Frederick, C. O., "The Effect of Lateral Loads on Track Movement", Railroad Track Mechanics and Technology, Pergamon Press, 1978.

21. Kerr, A. D., "Effect of Torsional Fastener Resistance on the Lateral Response of a Rail-Tie Structure", Federal Railroad Administration Report FRA/ORD 78/35, September 1978.



LOAD INPUTS TO RAIL

Figure 1

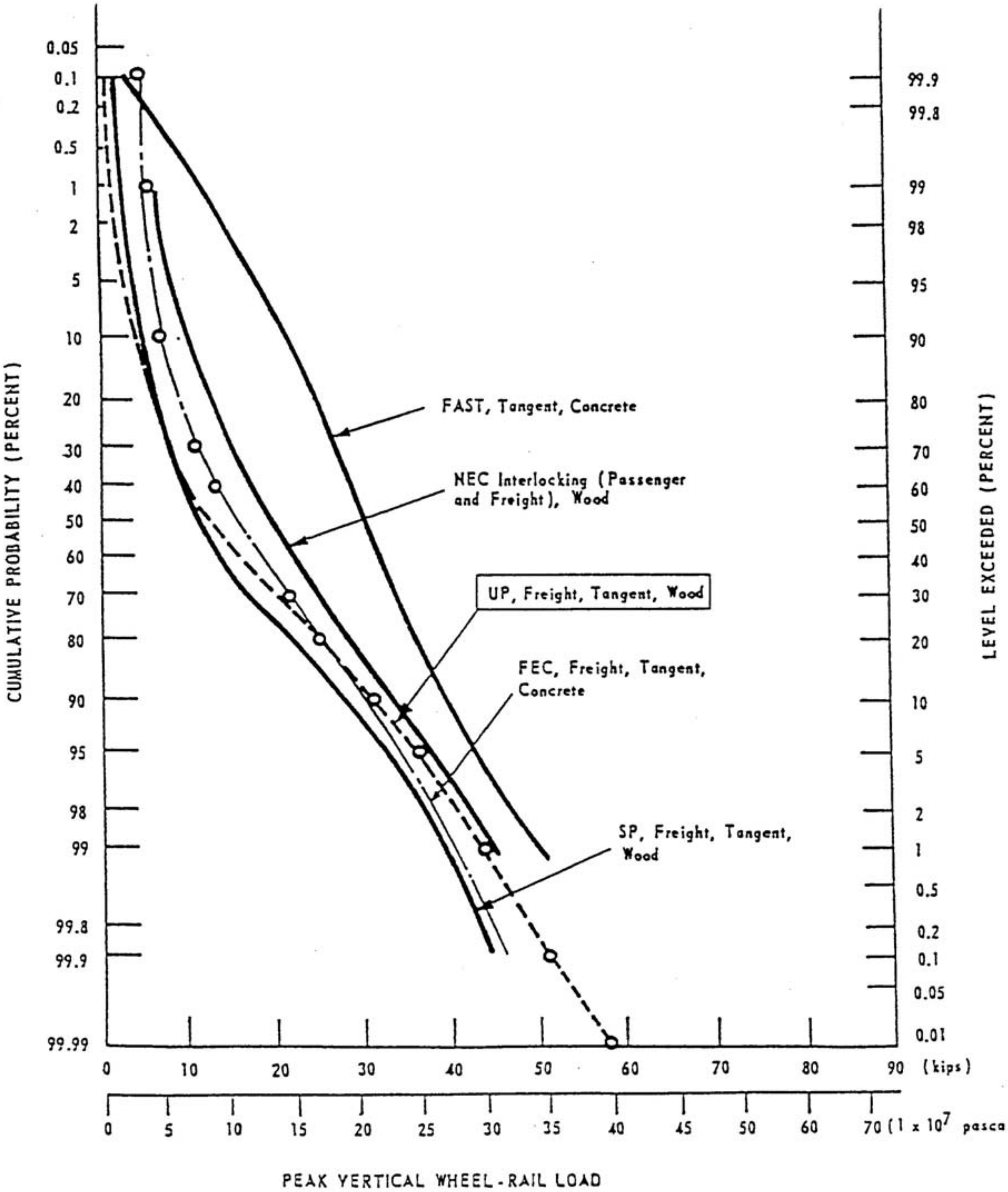
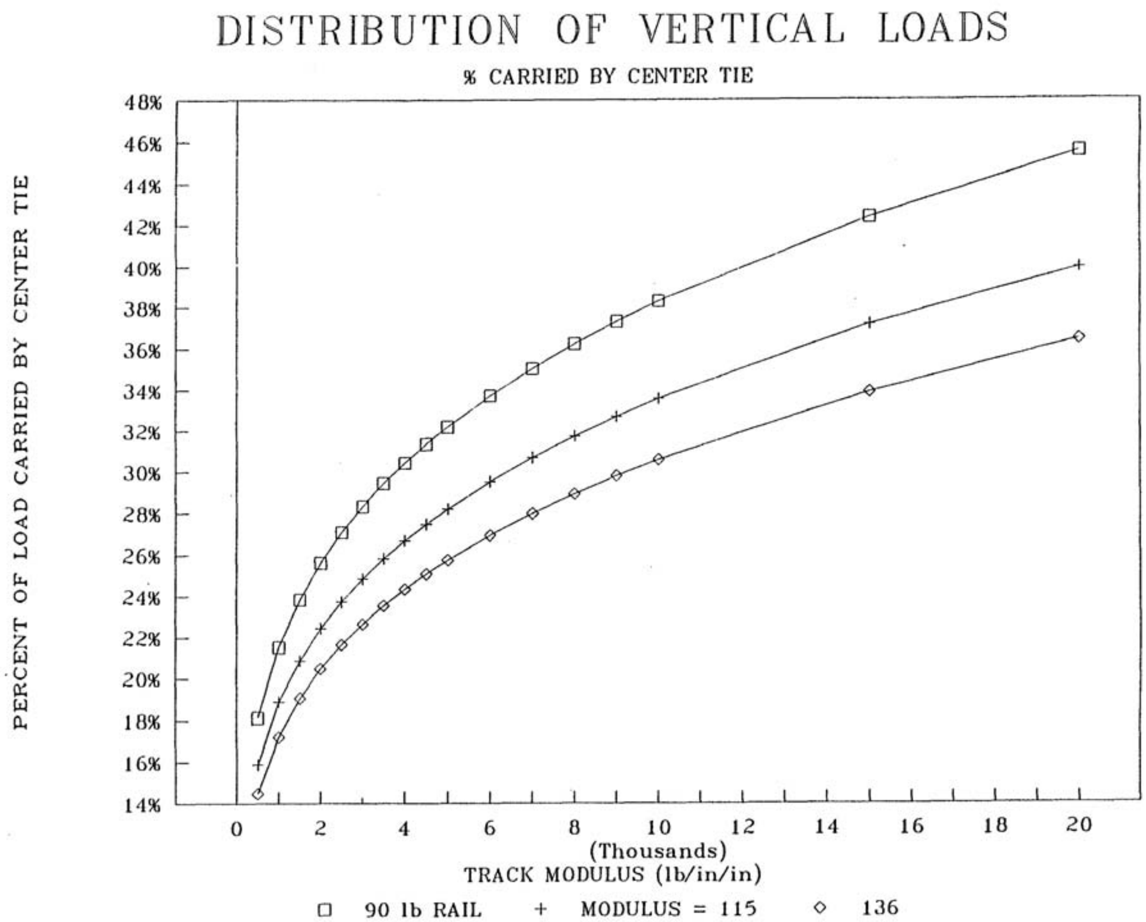


Figure 2

Level exceeded is the percentage of the loads (wheels) that exceed the peak vertical wheel-rail load (x axis).

Cumulative probability is the percentage of loads (wheels) that are less than the peak vertical load. (x axis).



DISTRIBUTION OF VERTICAL LOADS
% CARRIED BY CENTER TIE
PERCENT OF LOAD CARRIED BY CENTER TIE
48%
46%
44%
42%
40%
38%
36%
34%
32%
30%
28%
26%
24%
22%
20%
18%
16%
14%
0
2
4
6
8
10
12
14
16
18
20
(Thousands)
TRACK MODULUS (lb/in/in)
□ 90 lb RAIL
+ MODULUS = 115
◇ 136

Figure 3

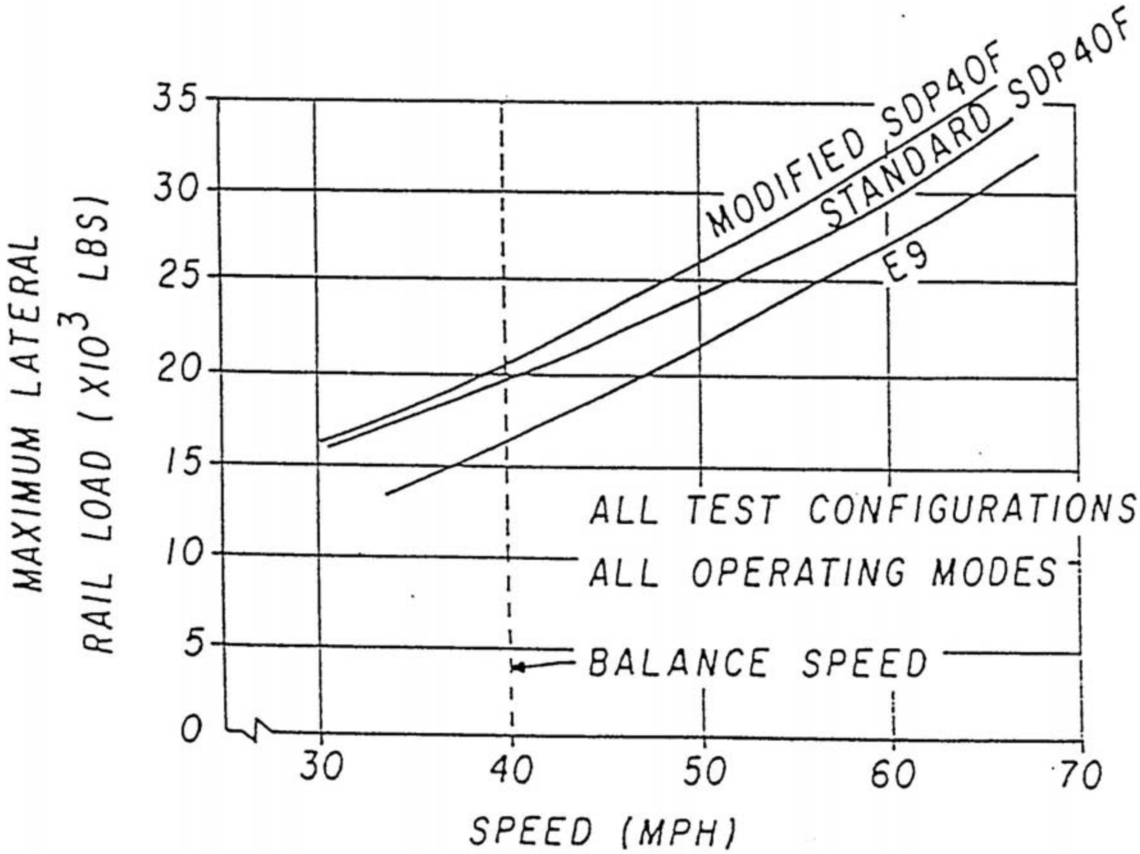
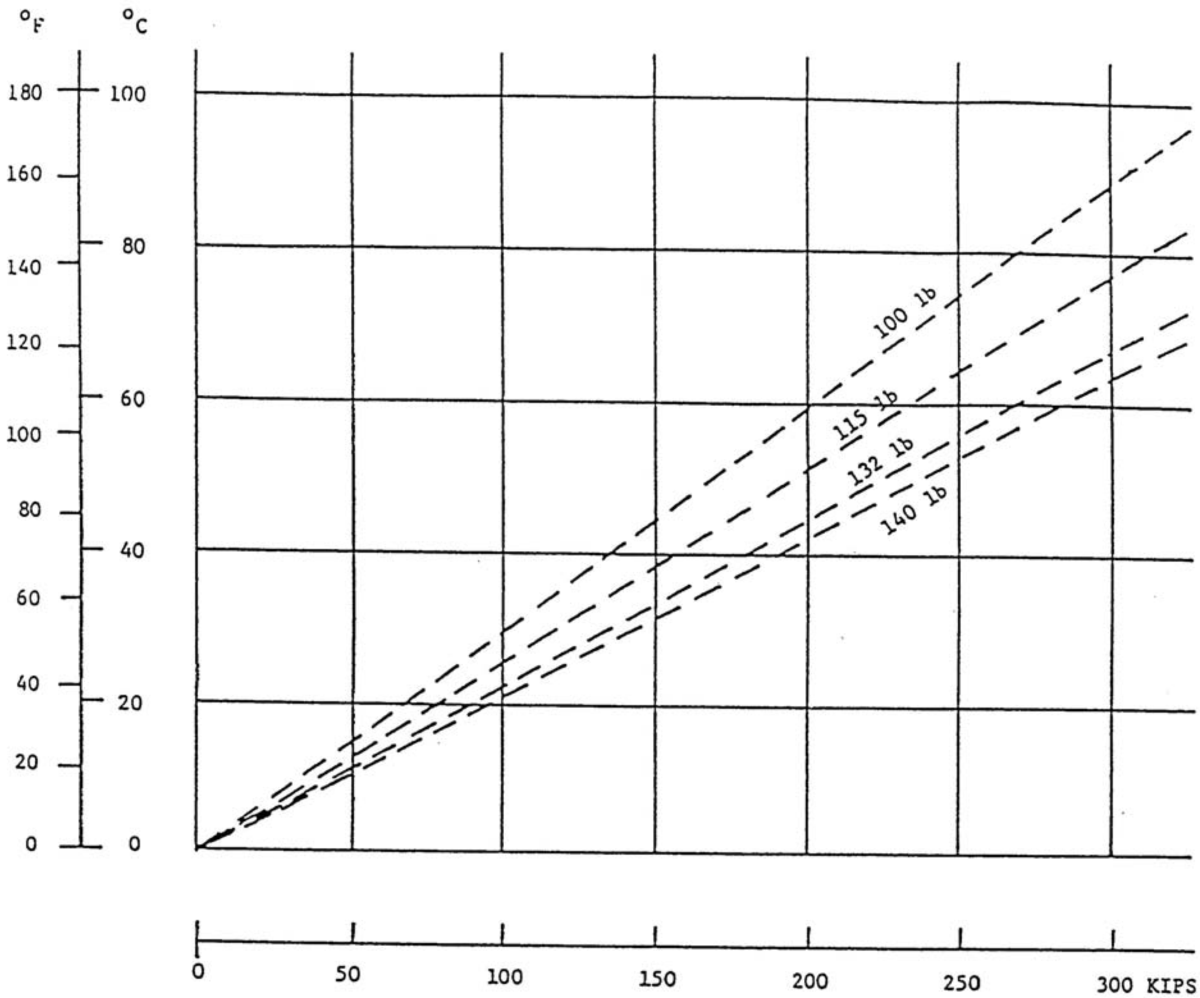


FIG. 4 MAXIMUM LATERAL RAIL LOADS- CURVE SITE



AXIAL FORCE IN EACH RAIL

FIGURE 5

RAIL TEMPERATURE INCREASE VS. AXIAL FORCE IN RAIL

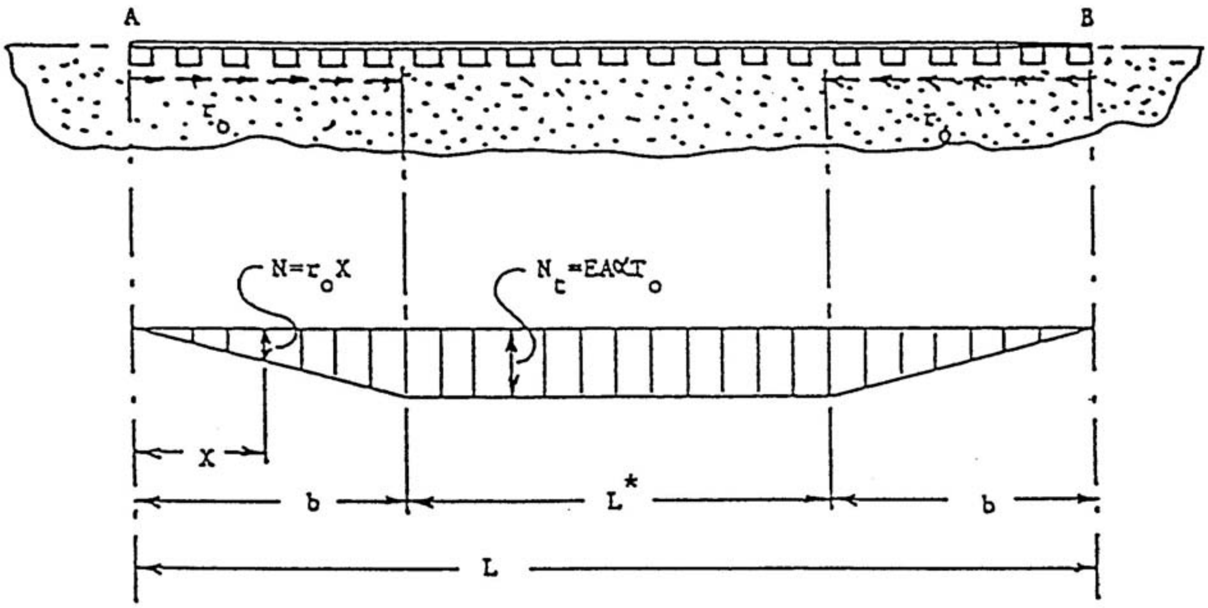


FIGURE 6

AXIAL FORCE DISTRIBUTION IN A TRACK OF LENGTH L

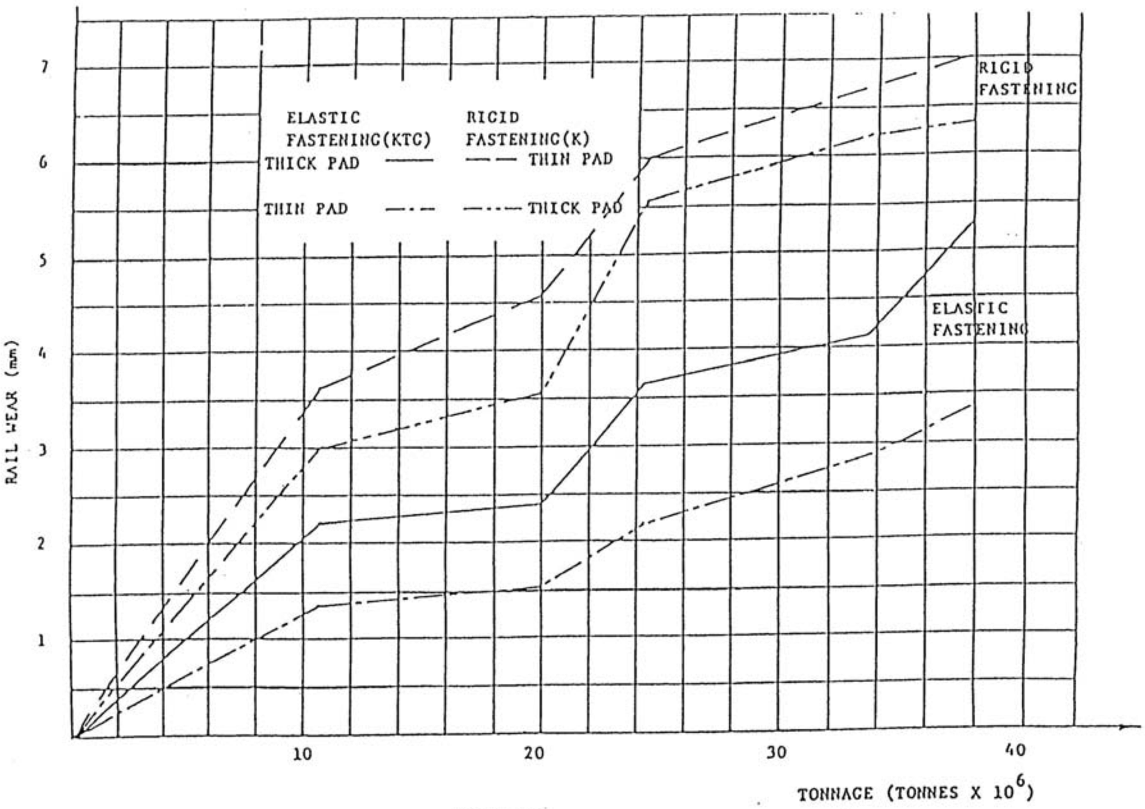


FIGURE 7

RAIL WEAR FOR RIGID VS ELASTIC FASTENINGS

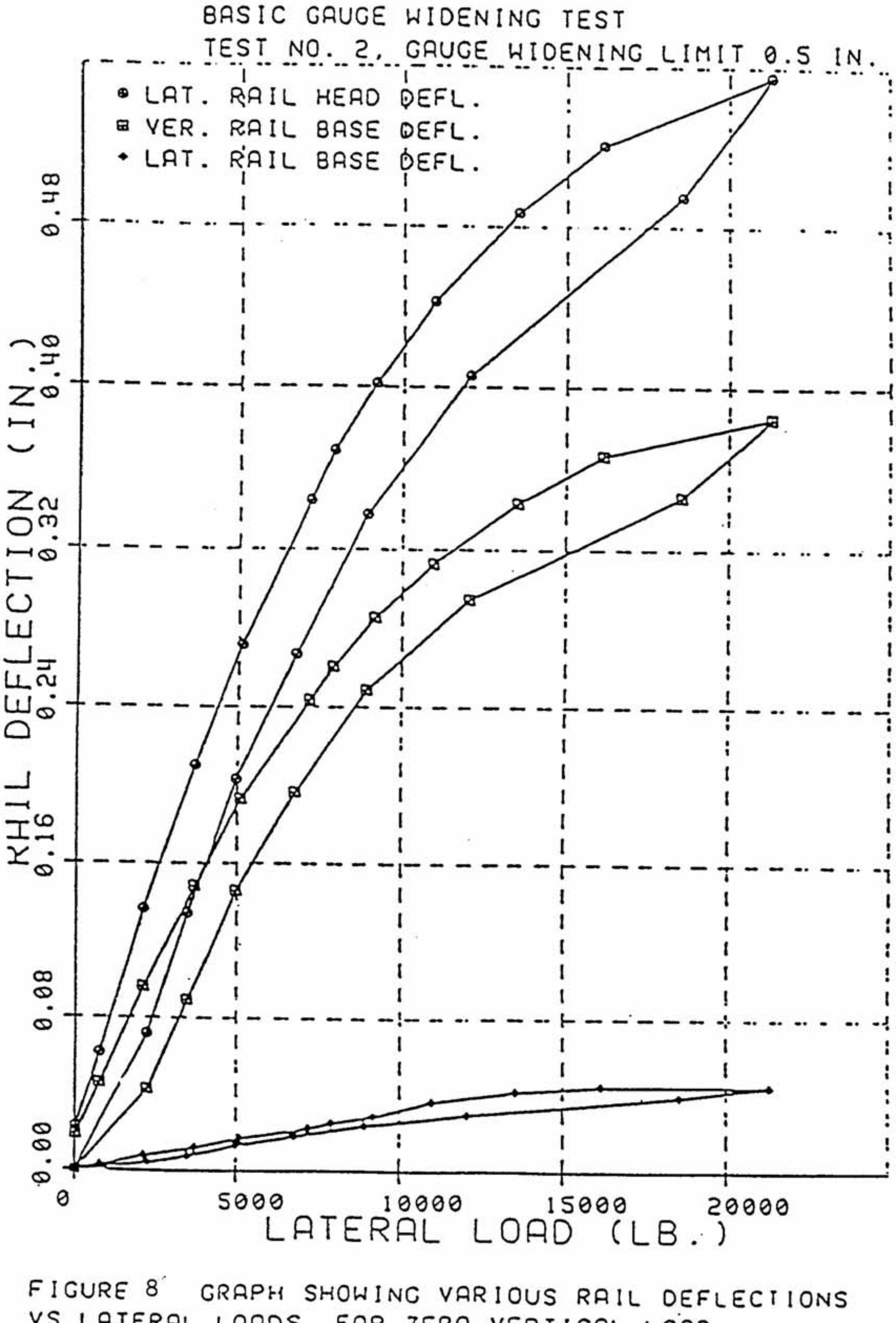


FIGURE 8 GRAPH SHOWING VARIOUS RAIL DEFLECTIONS VS LATERAL LOADS, FOR ZERO VERTICAL LOAD.

TABLE 1

FASTENER LONGITUDINAL RESTRAINT FOR PULL A PART

RAIL BREAK "GAP" (INCHES)	RESTRAINED LENGTH OF TRACK ON EACH SIDE OF GAP (FT)	TIE SPACING (INCHES)	LONGITUDINAL RESTRAINT PER FASTENER (LBS)
0.5	86	19.5	3627
0.75	129	19.5	2720
1.0	172	19.5	1814
1.5	258	19.5	1360
2.0	344	19.5	907

RAIL WEIGHT = 132 RE

MAXIMUM TEMPERATURE CHANGE = 75° F

TABLE 2

LONGITUDINAL RESISTANCE OF WOOD TIES IN BALLAST

LONGITUDINAL RESISTANCE* (LBS PER TIE)

	UNCONSOLIDATED **			CONSOLIDATED **	
MGT	TANGENT	CURVE		TANGENT	CURVE
0	2588	2763		3049	2712
.5 to 1	3030	3476		3279	3279
1 to 2	3194	3814		3530	3296
Maximum Individual Case			4900		
Average (all cases			3180		

* Defined to be longitudinal force necessary to displace tie .08 inches

** Average of test data from Boston & Maine, Southern, St. Louis-South Western, and Missouri Pacific Railroads

TABLE 3

FASTENER ROTATIONAL RESISTANCE TEST RESULTS FOR

115 RE RAIL SECTION ON WOOD TIES

ROTATIONAL STIFFNESS

10**6(LB.-IN/RADIAN)

TYPE OF FASTENER	VERTICAL AXIS	LATERAL AXIS	LONGITUDINAL AXIS
2 CUT SPIKES	0.92*	1.68	1.49
4 CUT SPIKES	3.32*	1.64	3.09
PANDROL	3.52*	0.73	1.66
SCREW SPIKE	1.27	1.89	2.60
COMPRESSION CLIP	2.79	0.94	1.99