Wheel and Rail Loadings from Diesel Locomotives

L. F. Koci

Chief Locomotive Development Engineer

H. A. Marta

Truck and Underframe Design Engineer

The area of wheel-to-rail loadings as they affect maintenance, wear, and potential train derailments has received increased visibility in recent years. In view of this and the considerable interest on the part of the railroads and the government, this presentation was prepared reviewing Electro–Motive Division's background in this area particularly regarding locomotives and activities in this area at the present.

The information in this paper has been presented to a number of railroads in the United States and Canada, and to the American Railway Engineering Association at the 1971 annual convention held in Chicago, Illinois. It has also been presented to the Railway Fuel and Operating Officers Association at their 1971 annual convention.

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Electro-Motive Division
La Grange, Illinois

Introduction

We, at Electro-Motive, understandably deal primarily with the railroad Mechanical Departments. In recent years as axle loads, the number of axles commonly used on a locomotive, locomotive weights, and operating conditions moved in the direction of representing more demanding applications, we found it necessary to be certain of just what the wheel-rail loadings were not only from the standpoint of locomotive equipment but also from the standpoint of the effect on the rail, since the "tracking" capability of the locomotive is one of our concerns. Over the last dozen years or so we have become involved in a number of tests run by our customers or cooperative tests between particular railroads and Electro-Motive. Consequently, the data presented in parts of this review, will certainly be recognized by people on the various railroads on which the tests have taken place.

The following report represents a brief review of some of the data accumulated in these various tests. It is certainly not all inclusive nor is all of it new. It is clear, however, that when studying the tracking of railroad vehicles and such aspects as derailments, it is impossible to separate the effect of the Mechanical, Engineering, or Operating functions of a railroad operation. The following information does try to "bring together" some of the information developed in each of these various areas.

Ludvik F. Koci
Chief Locomotive
Development Engineer

Mr. Koci joined General Motors as a Co-op student with General Motors Institute in 1954. He earned his Bachelor of Science there. He was conferred a Master of Science Degree in Applied Mechanics in 1960 at Michigan State. Much of his activity has been in the field of trucks and underframes with EMD. He is a member of LMOA.

Henry A. Marta
Truck and Underframe
Design Engineer

Mr. Marta earned a degree in Mechanical Engineering and a Master's in Engineering Mechanics from Wayne State University. He has since studied at Illinois Institute of Technology, and is a member of ASME and LMOA. Much of his activity since joining EMD in 1963 has been in the area of wheel-rail dynamics.

This review is divided into the following six areas:

1. Sample Derailment Data
2. Basic Curve Negotiation Mechanics
3. Experimentally Determined Wheel-to-Rail Forces
4. Rail Profile Data
5. The Effect of Dynamic Brake Levels
6. Mechanical Considerations

I. Sample Derailment Data

In preparing for one of the cooperative tests run with one of our major railroad customers, a brief summary was made of ten derailments which the railroad had experienced which either reflected no clear cause, or for which the cause seemed questionable. This would include such things as a rail break in which there did not appear to be sufficient old break to explain the derailment entirely. In examining this summary we were looking for guidance to the types of things which should be investigated to provide most meaningful results. Figures 1 and 2 summarize this information. It should be made clear that there was additional, detailed data available in many cases including crew comments; however, what is summarized in these figures is the rail weight, curve condition, locomotive consist, and the location of derailed cars. For example, in the first situation listed the locomotive consist consisted of an SD-45, (one of EMD's 6-axle locomotives), a second SD-45, a GP-35, (which is an EMD 4-axle unit), and a U-25B, (which is a competitive 4-axle unit). The x's in the illustration indicate what units or cars were derailed in the incident. The data represented as well as some of the details not indicated in the illustrations may be summarized as follows:

1. Either by crew comment, spectator comment, or locomotive speed tapes, change in train speed was indicated in almost all cases. In many cases locomotive braking was indicated. In one particular case the crew comment indicated that the train was being switched to a siding and the locomotive dynamic brakes were applied immediately after the locomotive consist crossed over the switch. As discussed in more detail later, this situation, in which the cars immediately behind the locomotive, (which do not have the vertical load always present which a locomotive does), are subjected to the maximum buff loads available in locomotive braking while crossing over a switch or crossover or on curved track, can carry with it the risk of derailment. The application of heavy buff coupler loads can result in substantial lateral wheel-to-rail reaction. The lighter the vertical wheel load present, the more "unstable" the condition may be. As noted, this is discussed further later. In most of the situations summarized the speeds were in the range of 35 to 50 mph.

2. The curve size involved was generally about 2 to 4 degrees. As will also be discussed later, the basic curve negotiation forces present on curves of this size are far lower than they are on tighter curves, such as those of 8 to 10 degrees.

3. Transient drawbar conditions were indicated throughout the train. An example of this is the fourth situation noted in which three different groups of cars ended up derailed in various locations in the train.

4. In some cases empty "TTX" type cars were placed immediately next to the locomotive consist in operation involving locomotive braking. Placing any empty cars adjacent to the locomotive in heavy locomotive braking conditions involves the risk of developing unsatisfactory wheel-to-rail conditions. However, the long-overhang type cars such as the TTX cars are particularly vulnerable to this and this type of operation if combined with a chance occurrence of a low joint, or bad track alignment, or train handling giving substantial dynamic buff "run-in", increases the risk of a problem.

5. In a number of cases the crew comment indicated buckled rail occurred, sometimes ahead of the locomotive. The comments pointed to the possibility of track thermal strains being a factor. While considerable investigation of the effect of thermal strains on track stability has taken place, little seems to be known about how thermal strains below the threshold of instability themselves, affect the ability of the track to withstand the not insignificant service loads applied to it during operation on curves. In other words, if occasionally the track thermal strains become great enough to be of concern from their own account, how is the ability of the track to withstand lateral loads(in the area of 10 – 15,000 lbs.,)reduced when those thermal strains are ½ as high or ¾ as high?

6. Of the ten derailments noted:

   a. Five occurred within the train only, involving no locomotives
   b. Two involved EMD 6-axle locomotives
   c. One involved a competitive 6-axle locomotive
   d. Three involved 4-axle locomotives

Although this summary involves one particular railroad, a number of similar summaries have been made on other railroads and in many ways they are not unlike. Particularly from the standpoint of involving curve sizes in which lateral loads are not as high as elsewhere; involving braking conditions; and involving a variety of vehicles including cars and both 4- and 6-axle locomotives, similarities are common.

II. Basic Curve Negotiation Mechanics

What is intended here is a brief review of what may be termed the "basic" lateral loading which occurs due to normal curve negotiation of a railway vehicle truck around a curve. The forces resulting from this normal negotiation are among the predominant forces involved, at least in the tighter curves. To fully understand how some of the lateral loads developed by various operating, track, and mechanical conditions discussed later combine, it is necessary to appreciate what causes the basic curve-negotiation forces. The following is a brief review of this and for further explanation, ASME paper No. 65-WA/RR-4 may be helpful.

These curve-negotiation forces are not related in any manner to centrifugal force. They are the forces which would be present if

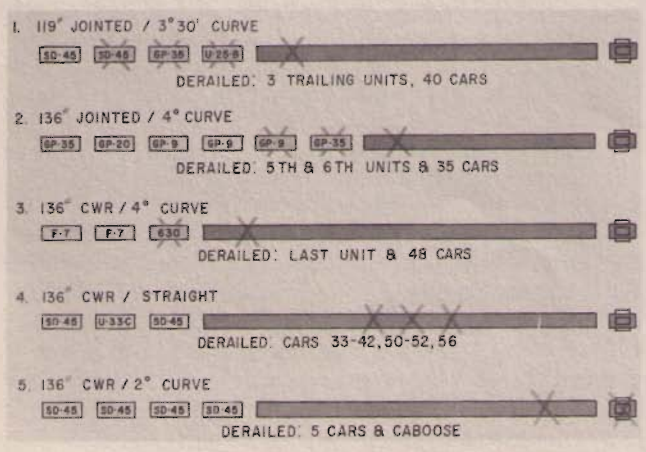

Fig. 1 – Sample Derailments (1 thru 5)

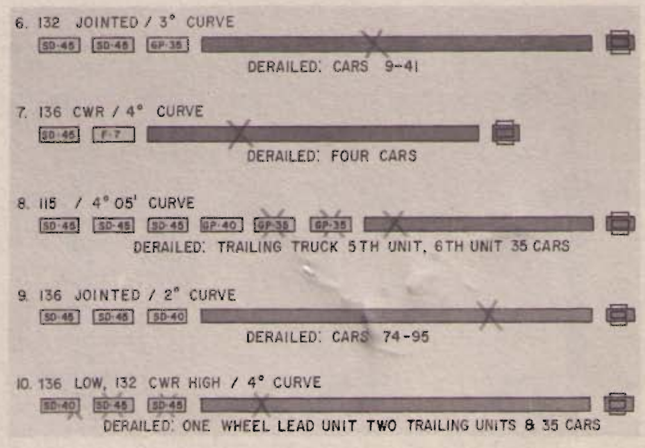

Fig. 2 – Sample Derailments (6 thru 10)

the operating speed involved were exactly balanced by the superelevation of the track. Perhaps the best way to make this clear is to state that these are the forces which would occur if the locomotive or railroad car were standing still and the track were on a giant turntable moving around underneath it.

It should also be noted that the basic curve negotiation forces are those that occur with no driving or braking taking place. They result strictly from the frictional forces between wheel and rail and are due to the fact that all wheels of the truck do not line up perfectly tangent to the rail, nor do they have a differential which would allow the inner and outer wheels to roll the different distances involved on the inner and outer rail of the curve.

Figure 3 illustrates schematically a 3-axle truck position while traveling around a curve. This position is typical in that the truck assumes an "angle of attack" between the leading axle and the rail, tending to skew within the clearances allowed between wheel flanges and rail. The curve size and the clearances involved were obviously exaggerated in this illustration and actually on curve sizes down to approximately 300 ft. radius the #3 or trailing axle of an EMD 6-wheel truck tends to be disposed toward or against the inner rail of the curve with the middle axle further away from the inner rail.

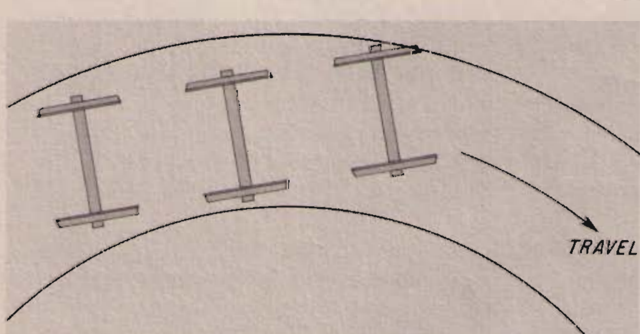

Fig. 3 – 3-Axle Truck on Curve

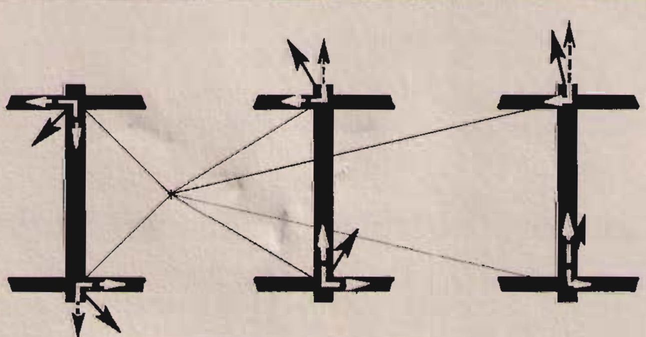

Fig. 4 – Wheel Tread Slip Components Due to Curve Negotiation

Figure 4 helps illustrate why this position results when a railroad truck negotiates a curve. As previously noted there are two separate circumstances which cause the wheel-to-rail frictional forces resulting in this curve negotiation action. First of all, the outer rail of the curve represents a longer distance to travel than the inner rail of the curve. Since the wheels are rigidly mounted on a common axle some longitudinal slip between the wheels and the rail must and does result. The arrows in the direction of the wheels on Figure 4 represent slip taking place in this direction; as can be shown, without driving or braking taking place, this results in equal and opposite slips on the inner and outer wheels of a given axle. (Wheel taper cannot provide the differential required, both because it would be correct for only a given size curve and because the three axles do not position themselves such that taper can serve this function.)

Secondly, since all three axles are held parallel to one another within the truck, it is clear that all three of them cannot be "aligned" with the track such that the wheels would be tangent to the direction of the rail at the point of contact rather than at an angle making it necessary to slip sideways. Since all three axles cannot be aligned with the rail, at least two of them must be at an angle either "running into or away" from the rail and, therefore, "dragging sideways" resulting in lateral slip between wheel and rail. In truth the equilibrium position which finally occurs results in none of the three axles being aligned with the rail and the arrows shown on Figure 4 at right angles to the wheels reflect the lateral slip which takes place at the point of contact between wheel and rail.

These two slip components at right angles to one another combine to give a resultant direction of slip between wheel and rail as shown in the diagonal arrows on Figure 4. With this relative motion or slip present, of course, frictional force from rail onto wheel results in the direction of the arrows shown. (These slip magnitudes are small and generally referred to as "creep" or microslip; however, for the purposes of this discussion it can be considered as slip or relative movement between rail and wheel.) As the truck passes through the curve these forces represented by the diagonal arrows cause the truck to turn or skew within the rails until flange forces at the leading outer wheel and sometimes at wheels of the second and third axle, balance these frictional forces and an equilibrium position results. The next illustration, Figure 5, shows these frictional tread forces balanced by the one or more flange forces which may be present. With the lateral clearances between the axles and the truck available on an SD-type 3-axle truck, the equilibrium position attained for curve sizes down to approximately 2100 ft. radius consists of the leading outer wheel obtaining a flange reaction from the outer rail, the trailing axle disposed toward the inner rail but without the flange contacting, and the second axle against the outer rail.
On curves between 2100 ft. radius and down to extremely tight curves (approximately 280 ft. radius) with sufficient lateral clearance designed into the truck between axles and truck frame, the trailing axle is actually against the inner rail. This is a desirable operating mode since it tends to reduce or limit the

amount of angle of attack and related forces which may be present otherwise. It should be noted that consequently, additional gauge widening between the rails merely allows the truck to skew or cock to a greater extent and increases the angle present between wheels and rail. It also increases the resultant lateral forces. Therefore, if there is sufficient lateral within a truck for curve negotiation without demanding that the axles "bind" in the rail (that is, with the center axle against the inner rail), unnecessary gauge widening is undesirable. Similarly, placing the wheel flanges closer together to provide more wheel-to-rail clearance can have an undesirable effect, although this is a much smaller factor merely because of the dimensions involved generally in gauge widening.

There are a number of specific factors associated with curve negotiation which may be considered important from the standpoint of tracking or wheel-rail operating conditions. The flange reaction itself is important from the standpoint of wheel and rail wear since it represents the force pressing the flange surface and the inside rail surface together while they have relative sliding taking place; and it is also of interest from the standpoint of contact stresses and local forces on the wheel flange and the rail. The angle between the leading outer wheel and the rail is commonly referred to as the angle of attack. This is important not only because the greater the angle the greater the resultant lateral creep and frictional forces, but also because the greater the angle the more "scuffing" there is between wheel flange and rail, and therefore, the increased amount of wear. Generally, it is accepted that wheel flange-rail wear is a function of the flange force and the angle of attack, although the manner in which they combine is not clearly defined.

The "net lateral load" between wheel and rail at any given wheel is the difference between the flange force and the lateral component of the tread frictional force. This net lateral load is what is important from the standpoint of tracking; that is, the likelihood of either wheel climbing the rail or the rail rolling over depends on the level of this net lateral load. In following discussion where lateral load or the ratio of lateral load to vertical load is involved it is this net lateral load referred to. This load is also of interest from the standpoint of wheel plate stress or axle stress resulting or lateral bending stresses produced in the rail.

The net lateral load at the entire axle instead of at a particular wheel is the result, or difference between, the flange force at one wheel of the axle and the two lateral components of the tread frictional force at both wheels of the axle. This then represents the total net lateral load on both rails at a given axle location and is primarily of interest from the standpoint of shifting the entire track structure within the ballast.

Figure 6 helps to clarify these forces as they act on the leading wheel axle set of a truck in passing through a curve.

The important relations and limits of these various forces considered from the standpoint of the effect on the rail and influence on possible derailments are generally as follows:

1. Wheel Climbing the Rail: Lateral-to-vertical wheel load ratio, L/V equal to, or greater than 0.9 at the leading outer wheel

2. Rail Rollover: L/V for one side of an entire truck $\left(= \dfrac{L_1 + L_2 + L_3}{V_1 + V_2 + V_3} \right)$ equal to or greater than 0.5

3. Track Shifting: Net lateral axle load equal to or greater than 40% of the vertical axle load on poor track conditions

The L/V limit of .9 is a value which is considered a minimum or lower limit for the possibility of wheel climbing to be considered likely. Most analytical or experimental work which has been done would tend to indicate this is probably in most cases required to be over 1.0 and with the flange configuration used in the United

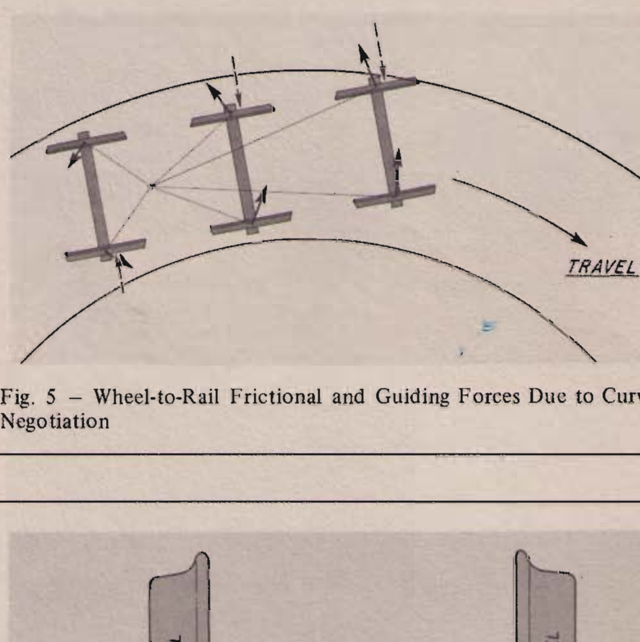

Fig. 5 – Wheel-to-Rail Frictional and Guiding Forces Due to Curve Negotiation

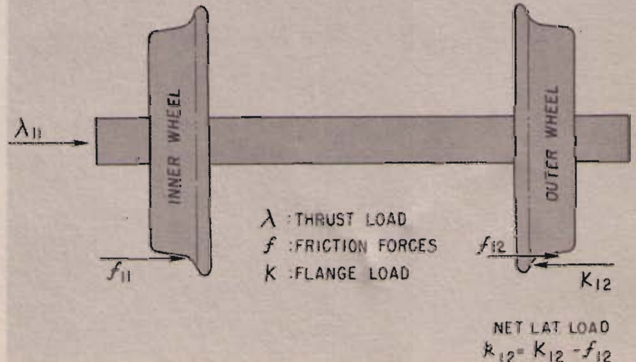

Fig. 6 – Lateral Forces on Leading Axle-Wheel Set

States as high as 1.5 before actual wheel climbing will occur. Also, in many specific operations it is known that ratios of approximately .9 do occur not uncommonly without derailment. However, because of the number of variables which can be and are involved in any derailment situation, the .9 figure seems a reasonable one to consider for the lower limit and for an indication of those load combinations at which this should start being of concern.

The ratio of 0.5 for total lateral load on one side of a truck to total vertical load on that side of the truck for rail rollover is merely the ratio of loads required to cause the resultant load on rail to fall outside of the edge of the rail base. On many common United States sections this occurs at a ratio of approximately .5 to .55. Obviously, this ratio can be exceeded at a single wheel without rollover occurring because the adjacent wheels provide vertical load which "holds" the rail down. The choice here of using the total for one side of a truck, whether it be a locomotive or a freight car truck, assumes no tiedowns or effective spikes, and a rail joint between the rail under that truck and the rail under adjacent trucks which provides no torsional restraint whatsoever.

The limit of 40% indicated for the entire axle relative to shifting of the track structure, is taken from work performed in France some years ago indicating that this was the lowest value on poor track which could cause this to occur.

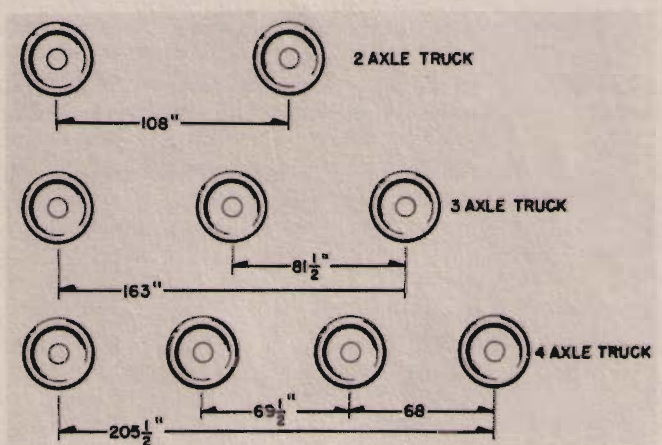

Fig. 7 – Wheel Bases of Tested Trucks

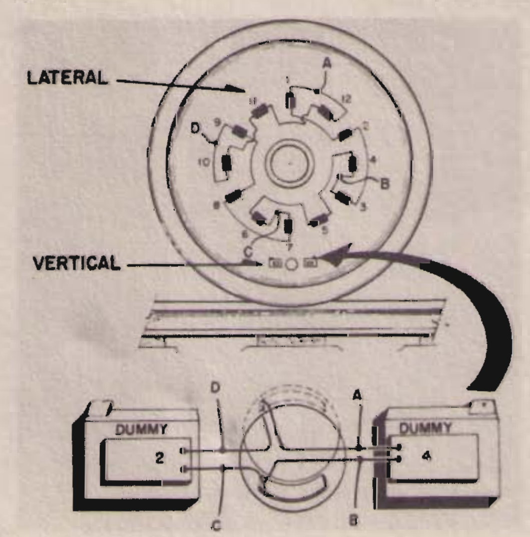

Fig. 8 – Strain Gage Locations and Connections for Lateral and Vertical Load Transducers

III. Experimentally Determined Wheel-To-Rail Forces

A number of tests have been run in recent years determining the lateral wheel-to-rail reactions which occur in operation on U.S. railroads. This data is discussed below as related to four general areas of interest: due to basic curve-negotiation mechanics, as influenced by speed, as influenced by sanding, and as influenced by coupler reaction.

A few years ago, EMD manufactured their first road locomotives using 4-axle trucks. While a considerable amount of analytical and laboratory work had been done previous to this time related to lateral wheel-to-rail reaction during curve negotiation, it was felt necessary with the first use of a 4-axle truck, (with all four axles in a common frame), to determine experimentally specifically what wheel-to-rail loads then existed in actual operating conditions and how the wheel-to-rail loads which would occur with the 4-axle truck would compare to them. Loads had been measured under certain specific conditions, as with the use of instrumented rail sections and roller bearing tie plates. This, however, had the disadvantage of measuring only one condition at one specific location and not reflecting the types of dynamic loading typical with normally found track irregularities.

Shortly prior to that time, tests had been run in Europe in which the wheel plates of a wheel-axle set were instrumented to provide a measure of lateral load on a continuous basis. Figure 7 shows the three EMD road locomotive trucks including the 2 and 3-axle in service and the 4-axle which was being tested at that time. Since all used a common wheel-axle-traction motor assembly, the use of an instrumented wheel-axle set would provide the opportunity to place it in three different trucks of widely varying wheel bases to obtain a good measure of what was desired. Therefore, it was decided to develop an instrumented wheel-axle set providing a continuous record of net lateral load between wheel and rail. Figure 8 illustrates schematically the strain gage instrumentation involved and Figure 9 shows photographs of the instrumented wheel. While this instrumentation gave a continuous record of lateral wheel-to-rail reaction it was also applied such to give an indication of vertical load once per revolution. Figure 10 shows the instrumented 4-axle truck with the slip ring assembly shown mounted on the far axle.

The experimental results are summarized in Figure 11. These curves show the net lateral load in thousands of pounds along the bottom scale versus the radius of curvature along the vertical scale. Degrees of curvature are shown for reference. The three curves indicate the loads measured for the 2-axle, 3-axle, and 4-axle trucks noted previously. It should be noted this data is based on coasting conditions; that is, there is no significant tractive effort or braking taking place, and little or no centrifugal force not balanced by track elevation. In other words this data reflects that loading occurring strictly from the curve-negotiation mechanics and the wheel tread frictional forces discussed previously. Also, these curves reflect the net lateral load between the leading outer wheel in the curve and the outer rail; the flange force itself would be higher.

There are a number of factors of interest apparent in examining these curves. For one, the loads are substantial, reflecting for example, approximately 10,000 lbs., 14,000 lbs., and 16,000 lbs. for a 10-degree curve with the 2-axle, 3-axle, and 4-axle trucks respectively. As would be expected the loads increase substantially with curvature; however, the loads at the 3 and 4 degree curve size area where most of the derailments discussed previously take place, are far lower than they are on other tighter but commonly found curves.

As the curves show, the same basic phenomena and same relationship of wheel-rail loads occurs with either the 2, 3, or 4-axle truck, although the "degree" of loading increases as the number of axles or the amount of tractive weight supported increases. As mentioned, these are the reactions from the wheel tread-to-rail friction, and therefore, the size of these forces depends on the adhesion conditions involved. These curves represent the average of many data points over a great deal of operation picked to represent average dry rail conditions. Under conditions which involve wet weather the adhesion and consequently the forces are lower, while under conditions which increase adhesion such as sanding, the frictional loads and correspondingly the lateral reactions are higher. These forces are always present during curve negotiation due merely to negotiating the curve, and other factors which affect rail reactions add over-and-above this. These would include such things as centerplate reaction resulting from a high coupler force acting at an angle, or centrifugal force.

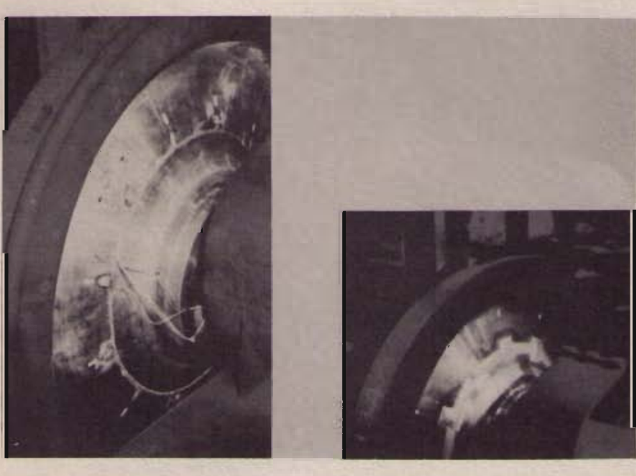

Fig. 9 – Photograph of Wheel Instrumentation for Lateral and Vertical Load Transducers

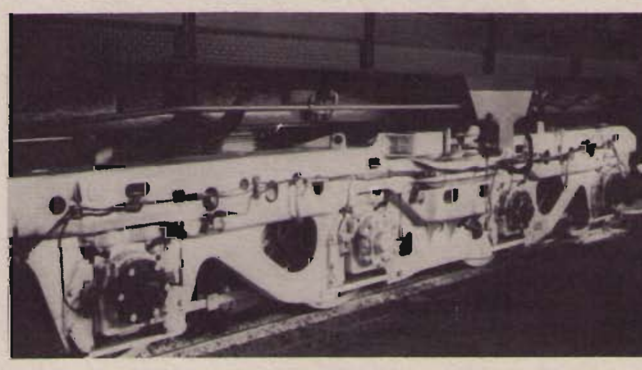

Fig. 10 – Instrumented 4-Axle Truck

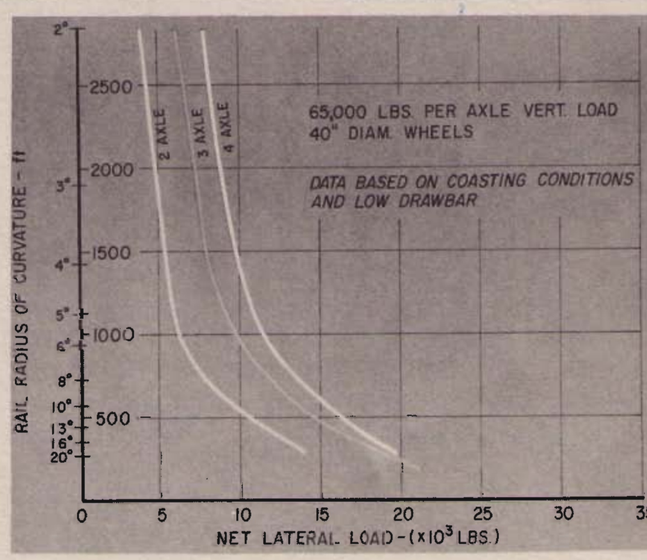

Fig. 11 – Experimental Results for the Net Lateral Load Between the Leading Outer Wheel and Rail

TRACK CURVE	2-AXLE TRUCK	3-AXLE TRUCK	4-AXLE TRUCK
19 (305'R)	14,000 LBS.	18,500 LBS.	19,500 LBS.
13° (440'R)	11,500	16,000	17,500
10° (573'R)	10,000	14,000	16,000
6° (955'R)	6,500	10,000	12,000
4° (1430'R)	5,500	8,000	10,000
2 (2860'R)	4,000	6,000	8,000

Fig. 12 – Table of Wheel-Rail Loads for EMD 2, 3, and 4-Axle Trucks Due to Curve Negotiation

A. ON CURVES:

GENERALLY 6,000 - 8,000 LBS. ADDITIONAL

OCCASIONALLY 12,000 - 15,000 LBS. ADDITIONAL

B. ON TANGENT TRACK (AT ABOUT 65 MPH):

GENERALLY 10,000 - 12,000 LBS.

OCCASIONALLY 15,000 - 20,000 LBS.

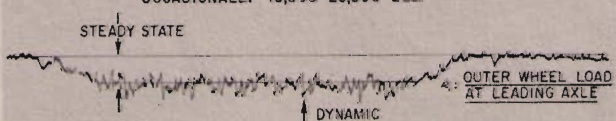

Fig. 13 – Dynamic Transient Loads (to 0.1 Sec. Duration) on Typical U.S. Railroad Track

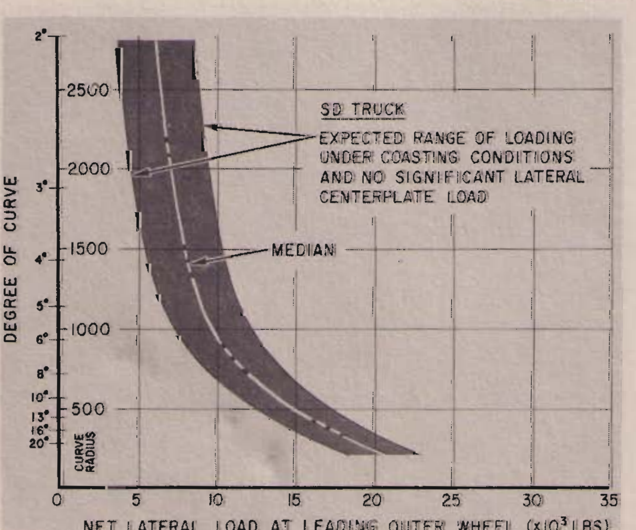

Fig. 14 – SD Truck Expected Range of Loading Under Coasting Conditions and No Significant Lateral Centerplate Load

Figure 12 is a table showing some of the values taken from the curves in Figure 11. It shows for example, that a 2-axle truck has a net lateral load reaction on a 10 degree curve the same as a 3-axle truck has on a 6 degree curve or a 4-axle truck has on a 4 degree curve. In making this comparison and evaluating different applications for different types of service it should be remembered that to pull a certain trainload a similar number of axles must be involved. In other words a given operation may require either six 2-axle trucks under a locomotive consist, or four 3-axle trucks.

The data summarized above represents the steady-state load; that is, it does not reflect the dynamic forces which occur at rail irregularities or at rail joints. Since the wheel and rail are being forced together with substantial loads, whenever a rail joint occurs it represents a change in path and dynamic loading over and above the base loads discussed occurs. The dynamic loads measured in typical operation are summarized in Figure 13. The sample lateral-load trace shown across the bottom of the illustration is typical, with the lateral reaction increasing as the vehicle travels through the spiralling into the curve, developing a "more-or-less" steady state level throughout the curve, and then decreasing in passing from the curve to tangent track. As is apparent, the dynamic loads occur fairly continuously over and above the steady state load. On curves these dynamic impacts are commonly 6,000 to 8,000 lbs. additional load occasionally reaching 12–15,000 lbs. On tangent track as indicated they are somewhat higher at the test speeds of 65 mph. These dynamic loads are very short in duration, mostly below 0.1 second long. Therefore, while they are of interest from the standpoint of rail "batter" and shock loads locally affecting the wheel and the rail, they are not of primary interest from the standpoint of tracking or derailments

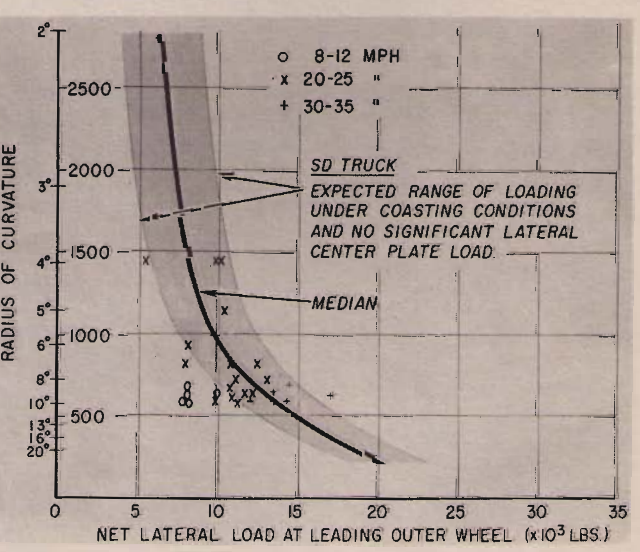

Fig. 15 – Influence of Speed on the Net Lateral Load

since they are not long-lived enough for any significant displacement or change in location to take place.

It is also of interest to note that the dynamic loads measured were quite similar with either the 2-axle, 3-axle, or 4-axle trucks, thus, indicating that they are primarily the result of the unsprung mass dynamics and represent too small a displacement to depend on the truck frame or the length of truck involved.

The next six illustrations show the effect of some other external loadings on these wheel-to-rail forces. Figure 14 shows the same curve previously shown for the 6-wheel truck indicating net lateral load versus degree of curvature. In addition to showing the average values, it also shows in the shaded area the range of loadings found under the considerable variables which exist in day-in and day-out operating practice. The following illustrations show the effect of speed, sanding, and different levels of coupler reaction upon these load levels.

Figure 15 illustrates the effect of speed in a given group of curves. The same median curve and range previously discussed is shown. In addition to that, actual data points are shown obtained while operating at three different speed ranges over a given series of curves. The x's indicate values obtained while operating at approximately the speed to balance track super-elevation; therefore, there is little or no effect of centrifugal force in these data points. The small circles represent data obtained while operating at about 10 to 15 mph under the balanced condition. As may be noted on this group of curves, the net lateral loading is reduced slightly but not to a major extent. Similarly, the crosses reflect data obtained while operating at 10 to 15 mph over the balanced condition and again, while these show a slight increase, it is not a major change in the load levels present. Speed, then, does not make a major change in the load levels which occur. Traveling at reduced speeds over restricted areas or temporary track does, of course, reduce the risk involved if a derailment occurs but does not substantially reduce the load to which the track is subjected.

As noted previously the lateral reactions result from the frictional forces between wheel tread and rail, and therefore, are a function of the amount of friction present. Figure 16 shows some representative data points and a shaded range of data obtained on various size curves while continuous sanding was taking place on the locomotive. As might be expected the increased friction level resulting from sanding increases the lateral reaction substantially. For example, it increases from a value of approximately 14,000 lbs. to one of about 18,000 lbs. for a 10-degree curve. In general, the actual friction coefficient measured under conditions of continuous sanding was found to be approximately .40 although it should be remembered that not all of this available frictional force goes into either driving or lateral reactions but a combination of both.

The next three illustrations illustrate the added effect of varying coupler buff reactions applied to the locomotive through the angle involved when curves are negotiated. This particular data is taken from tests run in a "pusher" operation; however, the reactions resulting would be present if this coupler reaction occurred due to braking or train "run-in" also. Figure 17 shows some data points and the range of data measured under conditions of moderate buff load (approximately 125,000 lbs.) while operating over the curve sizes indicated. This data also included continuous sanding and, therefore, shows the increased effect of the coupler reaction. Figure 18 is similar except the buff load coming through

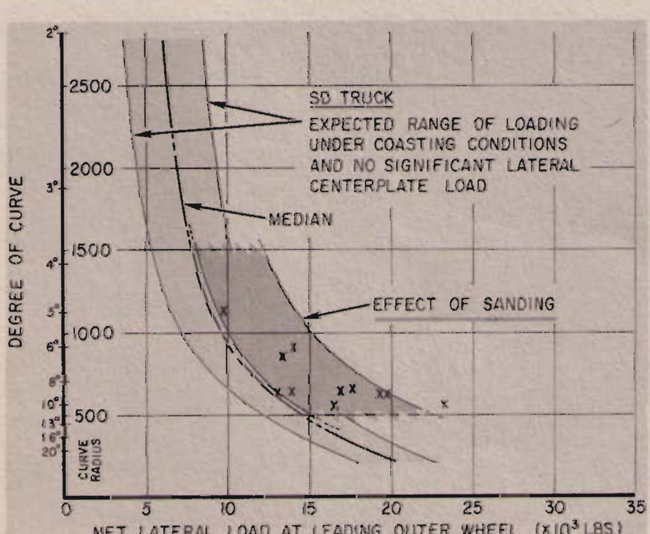

Fig. 16 – Effect of Sanding on the Net Lateral Load

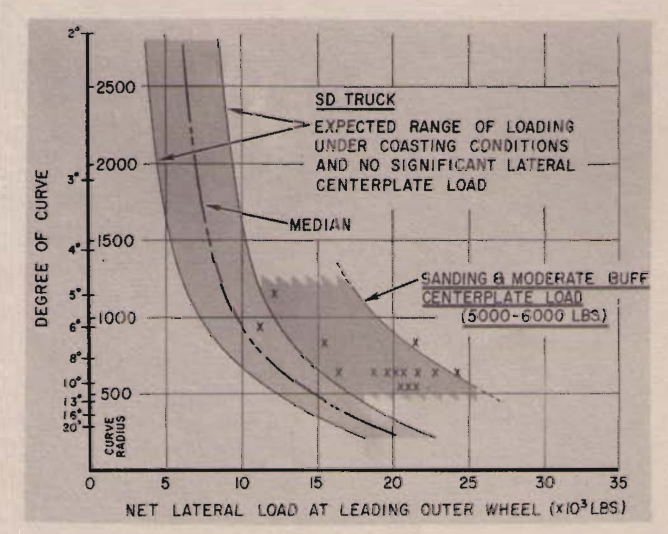

Fig. 17 – Effect of Sanding and Moderate Buff Centerplate Load

the coupler is still higher (in the range of 250,000 to 300,000 lbs.) and correspondingly the net lateral wheel-to-rail loads are seen to increase. Figure 19 shows all of these variables superimposed on the same curve. The cumulative effect of these external loads acting over and above the basic curving forces is apparent. It may be noted that lateral loads in the range of 25–27,000 lbs. for a wheel with average static loading of 32,000 lbs. occur here regularly. Under these particular conditions wheel and rail wear are quite high.

In many of the tests which have been run freight car wheel-axle sets on the cars immediately adjacent to the locomotive have also been instrumented. Primarily because of less vertical load available to "stabilize" the wheel and rail, it is generally found that the most vulnerable area for tracking difficulties or derailment is on the cars immediately adjacent to the locomotive. This is particularly so with some of the newer long-overhang cars.

Because the side frames of most freight car trucks are independent (that is, they are not held rigidly relative to one another) the basic frictional forces which occur in curve negotiation also tend to skew the freight car truck side frames producing a greater angle of attack than would be present for a truck of similar wheel base otherwise. Consequently, the wheel-axle position that develops is as shown in Figure 20. The axles shift relative to one another and after the trailing axle comes against the outer rail, the remaining frictional moment shifts the side frames and allows the axles

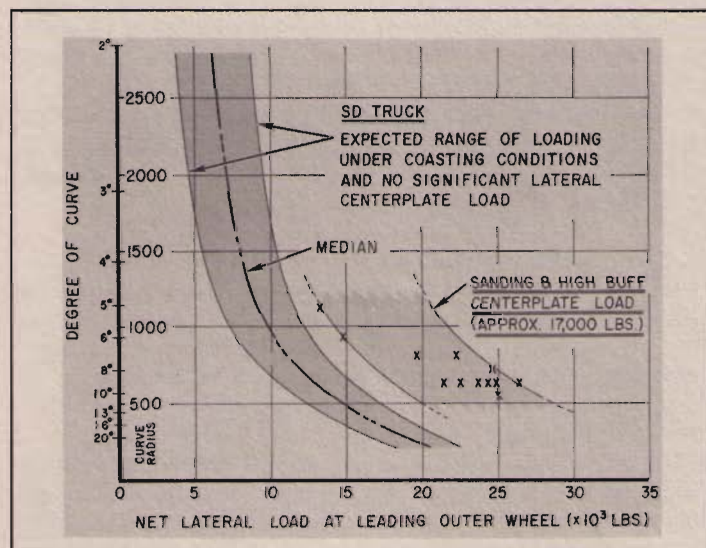

Fig. 18 – Effect of Sanding and High Buff Centerplate Load

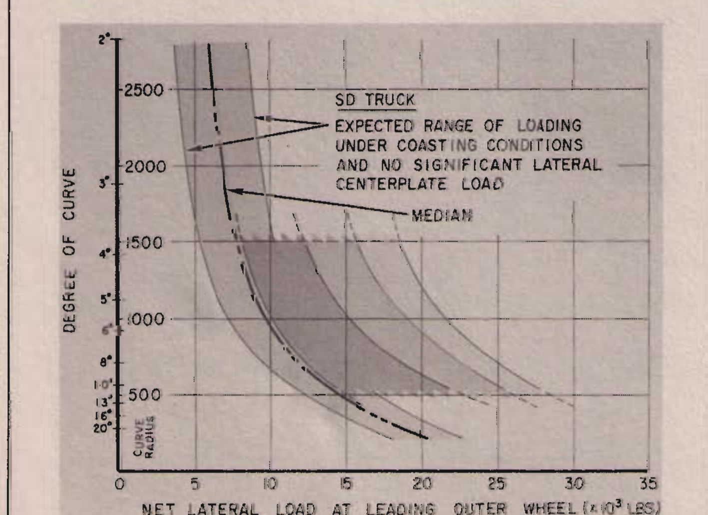

Fig. 19 – Influence of Sanding and Moderate or High Buff Loads

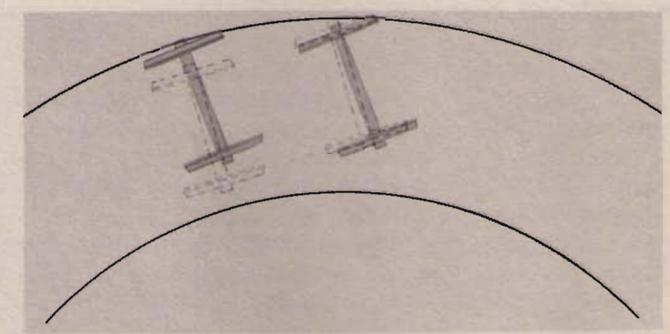

Fig. 20 – Conventional Freight Car Truck Position on Track Curve

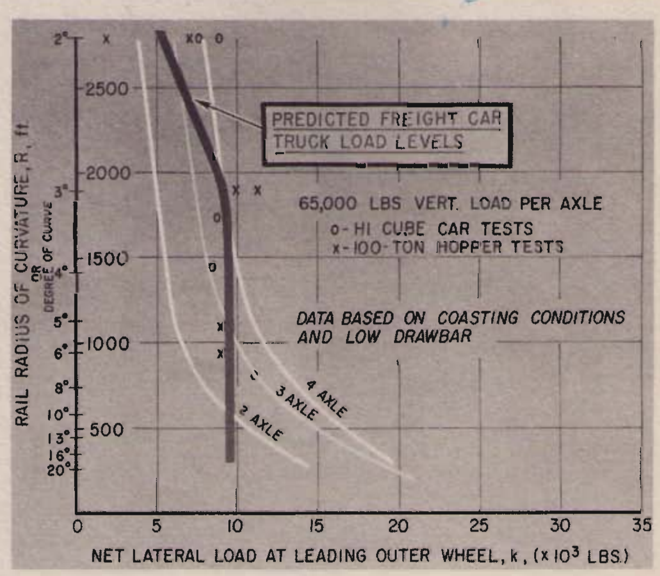

Fig. 21 – Predicted and Experimental Freight Car Truck Wheel-Rail Load Levels Superimposed on Load Curves of EMD Trucks

to skew further attaining a higher angle of attack and correspondingly somewhat increased lateral forces and reactions. Figure 21 shows what the predicted net lateral load on rail would be at the leading outer wheel of a freight car truck with 65,000 lb. vertical axle load. This illustration shows this curve overlayed on the 2-axle, 3-axle, and 4-axle locomotive truck data discussed previously. In addition, the circles and x's represent actual data points measured in railroad tests.

IV. RAIL PROFILE DATA

Since it is the ratio of lateral to vertical load present between a wheel and the rail which is of interest the vertical load present at any given time is important. This vertical load changes continuously as the wheel follows rail irregularities. To determine what vertical load was present during operation, it was considered mandatory to determine what rail profile the wheel followed in operation. That is, it was important to know what the "loaded" profile of the rail was when the wheel load was upon it, not what position the rail was in when there was no load present. The loaded profile can be and is substantially different from the unloaded shape of the rail. The instrumentation developed to obtain this, sample profiles obtained on U.S. railroads, and a summary of the data is discussed below.

Figure 22 illustrates the change in wheel load which results when the particular wheels indicated are subjected to a track irregularity of the size indicated. This track irregularity is defined as vertical rail profile which is intended to represent the change in profile from the middle of a particular rail length (that is, approximately halfway between the rail joints) to the lowest point encountered at the joint. This will be more clear after reviewing the subsequent illustrations. As can be seen the changes in wheel load which occur are substantial. A not unusual rail profile of 1½" can cause a change in wheel load of approximately 10,000 lbs. out of total nominal static wheel load of about 32,000 lbs. The effect on adhesion is obvious as well as the effect on wheel-rail stresses; also apparent should be the effect on tracking performance and possibility of derailments. Since this can be such a major factor it must be taken into account in considering the relationship of lateral-to-vertical loads present and what type of tracking condition is involved.

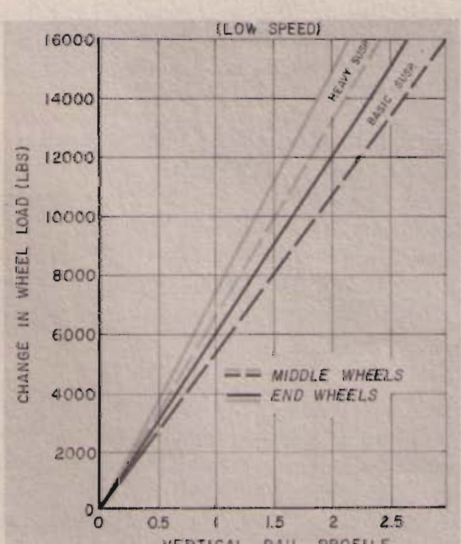

Fig. 22 – Change in Wheel Load Due to Vertical Peak to Valley Rail Profile for the SD Truck

To determine the loaded rail profile new instrumentation was developed which did not depend on relative displacements between different points of the rail but rather gave an indication of the absolute path "in space" the wheel had to follow in passing over the rail. General Motors Research had developed such instrumentation for use on highways in 1962 and it was first used in conjunction with locomotives and railroad track in 1965. Figure 23 shows the instrumentation schematically and it is more fully described and discussed in ASME paper 66RR/1. The following four illustrations show some typical traces of rail profile which have been accumulated in recent years.

The first sample trace, Figure 24, shows a typical mainline section on one railroad. What is reproduced here is the exact profile that the wheel follows in passing over the rail, or correspondingly the exact shape of the rail when the wheel load is applied to it except for the scale changes. The horizontal scale is condensed as shown by the 39 ft. distances shown in Figure 24 representing adjacent

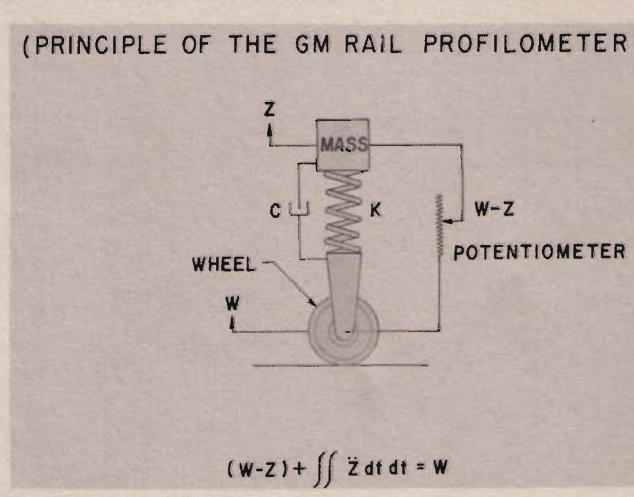

Fig. 23 – Mechanical Vibrometer

rail joints, and the vertical scale is expanded relatively, as indicated by the 2.1 inch peak-to-valley displacement shown. The uniform rail joint pattern is apparent and it is of interest that compared to the commonly considered unloaded rail profile the irregularities under the condition shown are substantial.

Figure 25 is a similar trace taken on a section of yard track. It is labelled "through" track to distinguish it from such facility tracks as sanding tracks or fueling tracks; that is, this represents track on which trains are made up and started, where demand for adhesion is often greatest. As can be seen and as may be expected, the uniformity of the rail joint pattern is not present because of the variations in rail pieces involved in the yard. Also of interest are the

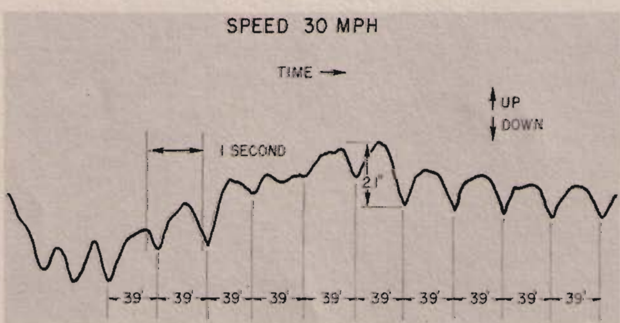

Fig. 24 – Dynamic Measurement of Rail Vertical Profile – Jointed Rail Mainline Track

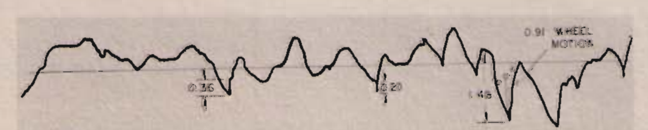

Fig. 25 – Typical Through Yard Track

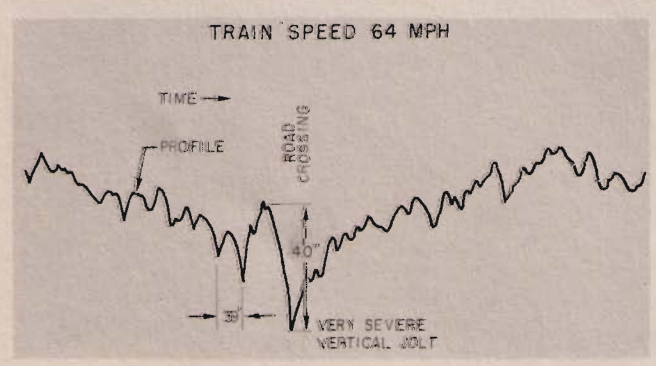

Fig. 26 – Highway Grade Crossing

areas of immediate mismatch where adjacent rail sections were placed without shimming. For comparison purposes a 6-wheel SD truck is shown to the same scales indicating that in passing over the rail lengths with 1.48 peak-to-valley displacements, the middle axle of the truck "feels" .91 inches of that displacement.

Figure 26 shows an additional sample taken on mainline track at 64 mph. Because of the higher speed the horizontal scale is "squeezed down" somewhat further as indicated by the 39 ft. dimension shown. Again, the rail joint pattern is apparent but in addition this sample includes passage over a highway grade crossing and reflects a 4 inch displacement within about a 20 ft. length.

The last sample trace indicated, Figure 27, shows passage from welded rail onto jointed rail. As may be expected the welded rail irregularities are far smaller than those found on the jointed track; however, it is interesting that to some extent the 39 ft. pattern is still present. This appears to be the result of the "memory" of the ballast and substructure due to the "pounding out" which took place when the joints were present.

As a summary of some of the rail profile data which has been accumulated, Figure 28 shows the peak-to-peak vertical displacement plotted against the number of times a displacement of that level occurs per thousand miles of operation. Data accumulated on four different railroads is shown. For example, on Railroad D a 1½" peak-to-valley irregularity would occur 3500 times per thousand miles or about 3½ times every mile. The rather substantial difference between data obtained on different railroads (or sometimes between different areas of a railroad) is apparent.

V. The Effect of Dynamic Brake Levels

There are indications that locomotive braking practice and, in particular, dynamic braking, often may be involved in some of the tracking problems which have been investigated. In some cases it appears personnel may not realize how "potent" a tool the dynamic brake level available to them today is. There are two aspects to this: the steady state dynamic brake level available, and the dynamic reactions (run-ins) which can often be involved in locomotive braking practice.

Figure 29 summarizes the braking effort available in pounds in dynamic braking a 4-unit consist of 6-axle units. The lower curve shows the braking available from four SD-7 units which are 6-axle units of approximately 1954 vintage. Use of the same number of axles, 24, in dynamic brake provides an increase as shown by the lighter intermediate shaded area on four units produced subsequently; the top curve shows the amount of braking effort available from the same 24 axles in dynamic brake on current locomotive units. This illustration reflects the significant increase in braking level available, an increase not apparent when the operator considers only the number of axles involved. In addition, the current models have available what is referred to as "extended range" dynamic brake which makes available the maximum level of braking down through the lower speeds which are normally involved in travelling over such areas as turnouts, crossovers, tight curvature, or temporary or poor track conditions.

In addition to these two factors, (the approximate 40% increase in braking effort available with the same number of axles, and the availability of the extended range braking), changes in the electrical control of the dynamic brake also make it possible to obtain increased braking levels from what was available in the past when multiple unit consists of more than four units are used. Initially, what is referred to as a "field loop" type control was used with which it was unable to provide increased braking effort if more than four units were used in multiple; the "potential trainline" type of control available today makes it possible that multiple units beyond four in number will add directly. On a 6-unit consist, this alone can represent a 14% increase in the amount of braking available for the same number of axles. For all these reasons, it is possible as mentioned earlier, that this does represent a greater "tool" than is realized by some operating personnel.

In regard to the dynamic aspects the following points are of importance:

1. Relatively high run-ins can be commonly measured, in the range of 250,000 to 300,000 lbs., in locomotive braking practice today, and there is little or no reaction felt in the lead cab where the crew is located. The reason for this is that a 6-unit consist can represent almost 2½ million pounds of mass, and the 300,000 lb. buff or run-in occurring between the trailing locomotive and the first cars in the train do not give much reaction to the lead cab when acting through that much mass and the eleven intermediate locomotive draft gears. Thus, it is impossible to determine whether braking practice is satisfactory by the "feel" in the lead cab.

2. It has not been possible to develop a universal dynamic handling procedure which would always be best. This is because it depends on the train makeup involved, and the

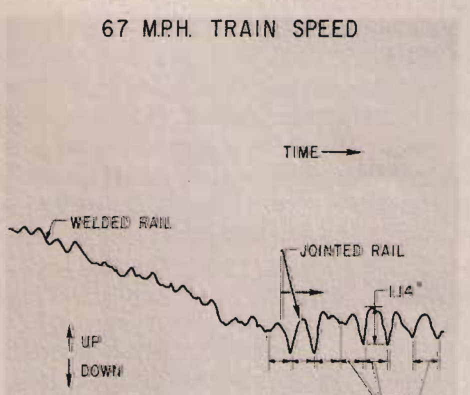

Fig. 27 – Welded to Jointed Rail

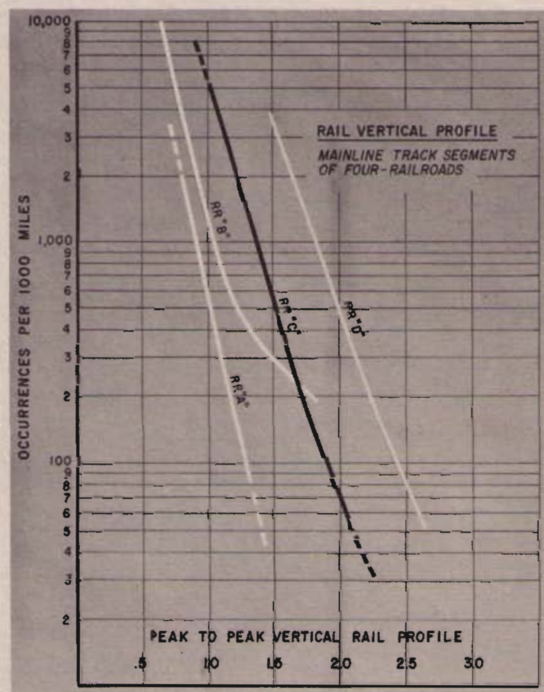

Fig. 28 – Rail Vertical Profile – Occurrence Vs. Profile – Mainline Track Segments of Four Railroads

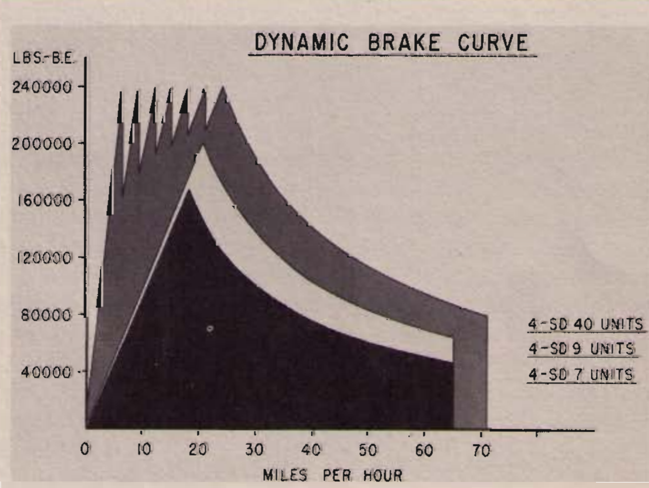

Fig. 29 – Dynamic Brake Curves

terrain being travelled over. There are some procedures, however, which are very undesirable:

a. It has been found that some personnel fail to provide the recommended 10-second delay in switching the selector handle from power to brake. If this switch is made rapidly the residual magnetism in the generator, can serve to give a high peak dynamic braking level immediately, without any throttle motion made by the operating personnel. This has been confirmed in tests with substantial run-ins measured.

b. Rapid buildup of the brake or "wiping out" of the handle can give high run-ins.

c. Applying or maintaining high braking levels at turnouts, curves or crossovers where there is reason to consider conditions may be less-than-desirable can result in unsatisfactory conditions.

3. In almost all tests, the worst conditions have consistently been found to be on cars near the locomotive consist. This is particularly so with empty cars and it is further exaggerated when the long-overhang TTX-type cars are placed adjacent to the locomotive. This is because cars in this position are subjected to the maximum drawbar values, often without having the substantial vertical load of the locomotive present helping to stabilize the wheel on the rail or the rail on the tie.

VI. Mechanical Considerations

There are a number of mechanical areas involved in the locomotive which can and do affect the wheel-rail loading. Among these (although there are many others) are: the alignment control draft gear, matching wheel sizes, and maintenance of truck bolster stops where necessary.

A number of years ago, when multiple-unit consist operation first became used, it was found that under conditions of significant locomotive buff, either braking or pushing, a phenomenon occurred referred to as "jackknifing". This is simply the condition in which the locomotives skew relative to one another within the clearances available between the locomotive carbody and the track. This includes wheel-rail clearances, axle clearances, and the lateral clearances between the truck bolster and the truck frame. As a result of this the couplers between adjacent units are at an angle to the unit and, therefore, the buff load develops a lateral force component reacted at the truck centerplate and subsequently at the rail. At that time the best manner determined to control this jackknifing effect, was to substantially reduce the lateral clearance between truck frame and truck bolster from the approximate 2¼ inch per side dimension present on the 4-axle trucks involved, to ½ inch per side by applying what is referred to as "bolster stops". By reducing the amount of shifting and skewing possible, these stops reduced the lateral component which could be developed and successfully limited the jackknifing forces involved, although they can increase the roughness of the ride laterally in higher speed operation.

In more recent years a coupler arrangement known as the "alignment control"-draft gear has been used on road locomotives. This coupler arrangement provides a resistance to coupler angling in heavy buff conditions, which more successfully combats the jackknifing effect which can occur in heavy buff whether on tangent or curved track. Under common operating conditions today, the use of the alignment control draft gear maintained in satisfactory operating condition, is mandatory to prevent significant lateral rail reactions. Figure 30 illustrates a locomotive consist and the first two cars of a train actually involved in a derailment which was investigated. The consist is three 6-axle locomotives and three 4-axle locomotives with TTX cars immediately behind the locomotive. The numbers indicated in the illustration are the lateral reactions resulting at the truck centerplate due to the skewing effect which takes place under the buff loading involved in this particular derailment. This centerplate loading combines with the curve negotiation loads to effect the wheel-to-rail reaction. In this particular consist and on this size curve the unit with the highest centerplate reaction is the trailing locomotive unit and of primary interest is the fact that this maximum loading increases from 17,400 lbs. at the point of maximum load to a value of 40,600 lbs. if alignment control were either not present or not maintained to engage properly. Also of interest is the fact as illustrated in the bottom portion of the illustration, that one inch of gage widening allows this maximum load to additionally increase approximately 20 percent.

As noted previously, prior to the use of alignment control draft gears, it was necessary to apply bolster stops to 4-wheel locomotive trucks used in multiple-unit locomotive braking conditions. Figure 31 illustrates another actual derailment condition which was investigated. In this case the three leading units were equipped with alignment control draft gear and the three trailing units, of older vintage, were not. The top portion of the illustration shows what centerplate reactions would be expected if all three of the trailing 4-axle units were equipped with bolster stops limiting the clearance to ½ inch per side. As may be noted the maximum reaction involved is approximately 30,000 lbs. After the derailment occurred actual measurements of the bolster lateral clearances present were made and it was found that the fifth unit in the consist did not have bolster stops applied while the fourth and sixth units did have bolster stops but these had become worn over the intervening years since application and resulted in lateral clearances per side of up to about 1-3/8 inch instead of 1/2 inch. The result of these actual lateral clearances involved was sufficient to increase centerplate reactions to 67,000 lbs., certainly sufficient to be of concern as a possible cause for the derailment. Two additional points may be noted about this situation: The three trailing units were being shipped "dead" in the train and there was some misunderstanding that they therefore did not require the same consideration as if powered. However, as long as they are subjected to the locomotive braking buff loads developed by the leading powered units, whether the units in question are dead or powered does not affect the center bearing reactions. Secondly, because some railroads have understandably chosen to keep 4-wheel trucks interchangeable under locomotives of different vintages, it has been found that in some cases trucks without bolster stops from units with alignment control have become applied to older units without alignment control without the bolster stops being added.

Wheel size matching within a truck and within the locomotive can be very important as illustrated in Figure 32. It is not uncommon to find 6-axle locomotives operating today in which wheel sizes within one of the trucks are mismatched well beyond recommended limits. This can and does make substantial change in the wheel loads present affecting the locomotive tracking conditions. Figure 32, for example, indicates that wheels mismatched by 1" in radius or 2" diameter on the middle axle can cause a change in axle load of approximately 12,000 lbs. In addition, this mismatch affects motor tractive effort distribution and axle load distribution both of which influence attainable tractive effort the locomotive can develop. It seems largely since the advent of wheel truing machines that difficulties in wheel size measurement and matching have become common and that this area deserves greater attention.

Conclusion

It is difficult to summarize all the factors discussed above. Considering what areas seem to deserve the most attention would indicate the following:

1. Locomotive Braking Practice
   a. Delay in power-to-brake transfer
   b. Gradual buildup of braking level
   c. Control of braking level over conditions such as crossovers, turnouts, and curves

2. Track
   a. Gage widening not excessive
   b. Level of rail irregularities
   c. Possible thermal strain investigations

3. Mechanical
   a. A 6-wheel locomotive truck supporting 195,000 lbs. adhesive weight has lateral loads 40 to 45% higher than a 4-wheel truck supporting 130,000 lbs.
   b. Freight truck forces measured due to curving are of similar magnitude to a 6-wheel locomotive truck
   c. Proper alignment control in draft gears
   d. Proper bolster stops on units without alignment control
   e. Long-overhang cars, especially empty cars, should not be near the locomotive consist in braking operation
   f. Wheel size matching

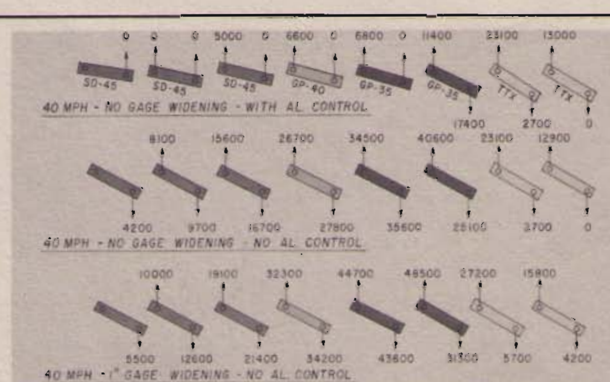

Fig. 30 – Influence of Alignment Control and Gage Widening on Consist Involved in Derailment on a 4° Curve

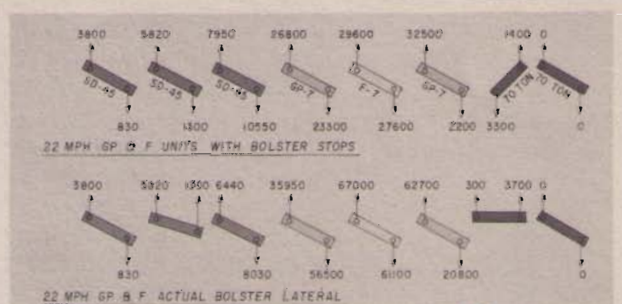

Fig. 31 – Influence of Truck Bolster Stops on GP and F Units in Consist Involved in Derailment on 3° Curve

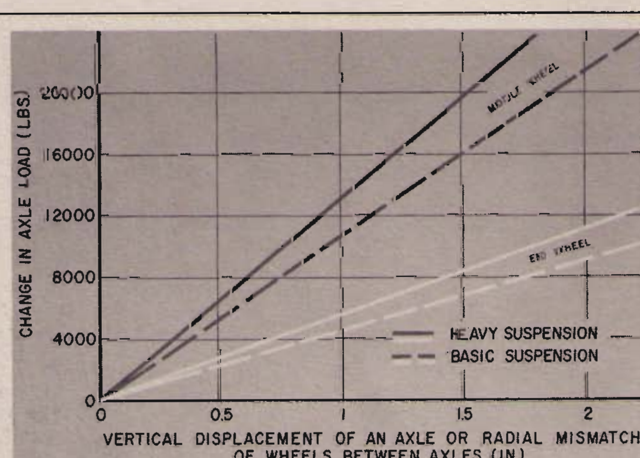

Fig. 32 – Change in Axle Load Due to Radial Mismatch of Wheels Between Axles or Vertical Displacement of Two Wheels on an Axle for the SD Truck

MARK OF EXCELLENCE