DISCUSSION

F. T. McInerney

The paper by Messrs. H. A. Marta and K. D. Mels on "Wheel-Rail Adhesion" is quite interesting and potentially useful, especially in correlating in one paper the results of other studies. The use of a single locomotive for these various test conditions helps to eliminate the locomotive as a variable, and the paper shows effectively the wide variations in adhesion coefficient that can exist because of rail surface conditions and track structural conditions.

There are two items we feel should have been included in this paper, especially in view of our own experiences with adhesion. The first is the relative value of weight transfer per axle in each direction for the modified SD-45, and for the standard SD-45; the second is an indication of which axle(s) tended to slip first. The passage of wheels over a rail tends to improve the rail surface condition, and if the lightest axle is behind the lead axle, the lead axle may be most prone to slip even though it is not the lightest axle. The passage of the wheels preceding the lightest pair may improve the rail surface so that the lightest pair can sustain higher tractive effort than a heavier lead pair.

Lead unit power reduction is used to take advantage of this effect on many locomotives. It can be either a fixed per cent reduction or continuously variable, at the customer's option. Power matching, which reduces the tractive effort at lower speeds, accomplishes a similar effect, although it is also intended to match the adhesion characteristics of higher horsepower locomotives to lower horsepower ones.

The effect of weight transfer decreases at increasing speed, while the effect of rail profile increases. Once the lightest axle is also the lightest, it becomes very prone to slip.

Ultimate wheel slip control may be quite different than we now know. It seems unreasonable to penalize a locomotive for weight transfer by reduction power when an axle slips. Locomotive weight does not change suddenly; rather, some axle is heavier by the amount that the slipping one is lighter, and the heavier axle might well absorb the power it is necessary to subtract from the slipping one. Preliminary work indicates that this approach is feasible and correct. With a wheel slip system that could control the power application to individual motors, a locomotive would be insensitive to weight transfer, changing track conditions as lead wheels improve the rail surfaces, and other aggravating conditions. It could operate with continuous motor imbalance as long as the current of the most heavily loaded motor was within the continuous current limit; and power would be modulated only as a last resort in synchronous slips.

D. R. Meier

The authors are to be complimented for their contribution to the literature on the subject of rail adhesion. These are the most comprehensive tests to be reported on diesel-electric locomotives under U.S. conditions of axle loadings and roadbed construction, and as such, are a valuable reference.

Although this subject has been investigated exhaustively, as the bibliography indicates, recent papers do not seem to indicate improved correlation of data. Chart 1 compares the data given by Borgeaud—in his 1967 paper, with that given in my paper ASME 62-WA-204, and with data given by Marta and Mels. Each of these sources is represented by upper and lower limits, identified by the color patterns. The upper band of Marta-Mels is their "dry-jointed" curve. The lower band is the welded-jointed curve.

Anyone will agree that the maximum upper and minimum lower limits shown here—amounting to more than a four-to-one variation—make it pretty difficult to decide just what constitutes practical operating limits. However, we must realize that these curves illustrate the composite influence of many variables such as different locomotive types, mechanical configurations, control systems, etc. One could conclude that there are probably not with higher horsepower per axle, especially now that it is common to begin reducing locomotive horsepower at lower speeds to follow an adhesion curve. Figs. 5 and 6(b) indicate that welded rail contributes more to high speed operation than to drag speed operation. We would agree that the wheel slip control system is a major factor in determining at what adhesion level a locomotive can operate. Careful design is necessary to prevent the wheel slip system from contributing to loss of adhesion by modulating power too abruptly. Good low speed sensitivity, with high speed oversensitivity is necessary.

The method of detecting slips can be significant. On systems that sense wheel speed, slightly different wheel diameters will give an increasing signal with increasing speed. This would seem to indicate that rate of change of wheel speed is a better and faster indication of slip than difference in speed. Decreased response time makes it possible for a slip to be arrested with only sand or slight power reduction. Higher speed slips require increasingly greater power reductions and the reduction must last until the slip ceases. In addition, a rate of change system offers an opportunity for detection of synchronous slip, apparently not encountered by EMD in these tests.

Systems of wheel slip control other than power modulation may eventually be desirable. The characteristics of shunt-wound motors used on some European locomotives provide inherent control of slip. The torque output of the motors at constant voltage decreases with increasing slip more than adhesion decreases for increasing slip speed. Thus the motor is inherently self correcting for wheel slip.
very many instances in which the maximum adhesion shown could be produced without sand, and probably the worst adhesion shown here would not be very often encountered. This lower curve, by the way, seems to be below that indicated by other investigators, as shown in the authors' Fig. 5. Perhaps some of these differences can be attributed to measurements made more carefully and with better equipment.

The concern expressed by the authors as to whether the horsepower-per-axle values are getting too high for adhesion conditions is expressed frequently. It is helpful to superpose curves of constant horsepower values, in terms of adhesion required at various speeds, on the adhesion versus speed curves so that one can readily appraise adhesion required against that available. Chart 2 shows curves of adhesion required at various speeds, to absorb horsepower per axle ranging from 300 through 1250 hp. The calculations are based on axle weights of 60,000 lb, which are typical of modern high horsepower four-axle locomotives. One interesting point is that the curves of Marta and Me$ are of less slope than has usually been indicated heretofore. The effect of this is that the 1250 hp curve doesn't require any better rail condition at 24 mph, than the 1000 hp curve requires at 19 mph. Of course, the adhesion required for 1250 hp at 19 mph is 25 percent more than that required for 1000 hp/axle at 19 mph. In any event, the flat slope helps the high horsepower/axle locomotive in the high speed range.

As a matter of interest, horsepower/axle values of 1000 to 1500 hp have been used since 1935, beginning with the GG-1 electric locomotives on the Pennsylvania Railroad. These units draw as much as 9000 hp from the catenary during acceleration. Even earlier than this, many steam locomotives were able to produce 1500 hp/axle. Many European locomotives have been built at these power levels. In fact, European developments indicate that 70 hp/long ton of weight on drivers can be supported by the adhesion between wheel and rail, on electric locomotives.

The situation regarding adhesion at higher power per axle is not as bad as one might first think. Certain qualifying considerations tend to relieve the problem of determining exactly how much adhesion rail conditions will support, for given levels of horsepower. One of these is that the high horsepower per axle units are most effective when dispatched on a horsepower/ton basis, rather than by tonnage rating. Expedited trains are usually dispatched on this basis. At levels of say three horsepower per ton, the adhesion required on a 3000 hp, four-axle unit is only about 15 percent, even on a 2 percent grade. At two hp/ton, about 18 percent adhesion would be required on a 11/2 percent grade.

Locomotives applied on a tonnage rating basis are generally six-axle, so that hp/axle ratings are lower, and this is the best application of units of this type. In these applications, the maximum adhesion available is not appreciably different at 1000 hp/axle than it is at 500 hp/axle, as we have seen. In other words, the problem in these applications is not caused by excessive power per axle, but just simply not enough maximum adhesion available. It is not a horsepower problem.

Another factor is that traction motor ratings correspond to about 20 percent adhesion rating, on a continuous basis. Measures taken to achieve this level of adhesion, such as sanding or use of wheel slip protective equipment, are normally adequate to fully utilize the motor rating.

Also, as the authors mention, wheel slip protection equipment is much superior to what it was ten years ago when locomotives had lower horsepower per axle. Automatic detection and correction of wheel slip is now rather effective in more successful application at high adhesion values.

In starting heavy trains, it is often necessary to develop values of adhesion as high as 30 percent. The usual method of doing this is to notch up from engine idle speed to a notch value sufficient to give the required adhesion, as illustrated on curve 3. Assume that notch 4 is sufficient to develop 30 percent adhesion, and the train starts to move. After a speed of 2½ mph is obtained, another notch can be taken, and so on, to notch 8, at which time a speed of 14 mph would be attained. This technique is normally used, regardless of hp ratings, and is also used on high horsepower electric locomotives. Its use will surely be continued to accommodate further increases in horsepower. It is also used to slack off power to avoid slips when adhesion conditions are poor.

The authors do not really answer the question as to whether there is a limit to the maximum hp/axle which might be used. However, it seems clear that the advantage in going to higher hp/axle on expedited train movements, together with improved methods for controlling wheel slip on high adhesion applications, will assure continued progress in horsepower per axle growth.

It is hoped that the authors will continue their investigation and reporting on this subject.
A way to gain insight into the problem of how to increase train traction is to consider the power balance for a whole train:

\[
\text{input power} = \text{dissipated power}
\]

The input power is limited at low speeds by wheel-rail adhesion, and at high speeds by the capacity of the propulsive device, be it a diesel engine or an electric motor, as the authors point out. When the train is not being accelerated, the dissipated power arises from two contributions:

1. The power to overcome aerodynamic drag, and
2. The power needed to overcome the rolling friction of the free wheels supporting the cars that make up the train.

At high speeds, the aerodynamic drag force dominates the rolling friction force, whereas at low speeds, such as the authors are concerned with, the rolling friction is at least as important as the drag.

As discussed by the authors, train traction may be increased by increasing the input power. At low speeds, the input power can be increased only by improving wheel-rail adhesion characteristics.

An alternative to this approach is to decrease the dissipated power. At high speeds, this is accomplished by streamlining, and a considerable amount of attention has been devoted to this method. At low speeds, the dissipated power would be most effectively decreased by decreasing the rolling friction of the free wheels. This method has not been adequately studied, and the discussers would like to make a few observations:

1. The resistance to rolling of a free wheel consists of the “true” rolling friction, exerted at the interface between wheel and rail, and the bearing friction in the bearing supporting the axle. The magnitude of these two friction forces is approximately the same [35]. For cast-iron wheels with machined treads, the magnitude of the coefficients of friction for bearing and rolling friction (of the “true” variety) is approximately 0.008 [35].
2. The bearing friction varies greatly with the design of the bearings used [33].
3. The true rolling friction of a free wheel depends on the surface finish of the wheel, and may be reduced by half by using wheels with a good finish [35].
4. A second factor influencing rolling friction in free wheels is the hysteresis loss in the material of the wheel and track [36]. The lower the hysteresis loss, the lower the rolling friction.
5. Yet another factor affecting the rolling friction of free wheels is the continuous subsurface plastic deformation that takes place in both the wheel and the track due to the rolling action of the wheel [37]. A reduction in the wheel load would decrease the amount of plastic deformation and, consequently, the rolling friction.

The compromises and trade-offs involved may best be appreciated by listing some of the desirable qualities of the propulsive and suspension systems of a train:

1. High tractive capacity; this is achieved by increasing the adhesion coefficient of the driving wheels of the locomotive.
2. Low power dissipation; this is achieved by decreasing the rolling friction of the free wheels.
3. Dynamic stability; this requires a high friction coefficient for transverse slip of all the wheels.

The problem, of course, is considerably more complex than the above outline indicates. This, I am sure, will prove to be more an incentive than a deterrent to further investigations. Meanwhile, the authors are to be complimented for a comprehensive study of one facet of the problem.

References


F. Nouvion

I have read with great interest the paper by Mr. H. A. Marta and Mr. K. D. Mels giving new information on adhesion.

This obviously applies to a locomotive of a given type, the GMD SD 45 (CC type diesel engine) with an axle load of 30 tons, each axle being individually controlled.

I understand from your information that the wheel diameter is 800 mm. I think it worth recalling these details, since adhesion is peculiar to each machine, and results in this matter can never be generalized.

I have noted the very favorable effect of sanding in all cases. This is well known, but sanding is all the more effective when it is used in suitable quantities and at the right point in the wheel-rail angle, as near as possible to the point of contact.

In France we adopt a flow of 1 liter per minute per stretch of rails, or an average of 0.5 liter per sandbox for traction engines with a sandbox ahead of each bogie. It would also be interesting to know the composition of the sand and its grain size, as well as the treatments it undergoes before use.

I have also noted what I believe to be the new information about the favorable effect of tracks made of long welded rails; this is a very important fact for the suppression of joints.

I have also noted the unfavorable effect of small radius curves. The loss of adhesion referred to in the paper by Mr. H. A. Marta and Mr. K. D. Mels is, however, much greater than that which we have recorded in a certain number of adhesion tests of electric locomotives. In our experience, the loss of adhesion was 10 percent on curves of 400 meters radius and can be regarded as approximately inversely proportionate to the radius of the curve, in the usual field.

The loss of adhesion observed on your curves of 425 meters radius (4 dog) with the SD 45 locomotive amounting to 50 percent, strike us as really quite considerable, but it is true that this phenomenon is very probably affected by the cant of the track, which is not specified, and the wheelbase of the bogies. On curves of equal radius, the loss of adhesion is higher with a big wheelbase than with a small one.

Now, our results referred to relate to a locomotive with a bogie wheelbase of 2.9 meters, whereas the SD 45 locomotive has a higher wheelbase.

We have found in our tests, that whatever the state of the rail, the fall in the coefficient of adhesion with increasing speed was less with single motor bogies with the driving axles geared between them than with individually controlled bogies. In particular, the effect of joints is certainly more adverse than in the latter case. Passage over a joint sometimes affects one axle which is held up by one or more of the others.

With regard to the variations of the coefficient of adhesion according to the state of the rail, we have found with a 50 Hz monophase current electric locomotive with individually controlled axles, the following results at 15 mph:

- Good dry rail, without sanding = 0.35 (100 percent)
- Good dry rail, with sanding = 0.37 (106 percent)
- Poor rail, without sanding = 0.16 (46 percent)
- Poor rail, with sanding = 0.225 (65 percent)

It should be noted that these results were obtained with sanding.
boxes of a then current pattern used at the time on all locomotives. With the improved sandboxes since perfected, a greater improvement in the coefficient with sanding would certainly be recorded.

The values of adhesion indicated by the authors related to the efforts after which sliding of the wheels on the rail is established, but we know that the characteristics of the driving part have a great importance in the use of this adhesion.

It is now well known, and we have often had occasion to point out, that it is essential that the axle which begins to skid should have a characteristic \( F_k \) as vertical as possible, so that the skid does not degenerate into a runaway, but is stabilized at a moderate value. This is why we have applied to our most recent diesel locomotives designs known as "hypermeshive" where the motors on a working point are fed at almost constant voltage (CC 72000).

One process for making the characteristics vertical and resistant to any skidding is to use separate excitation motors. We have abandoned this process, because the deflection of the motors is considerably slowed down on a flash and the subsequent destruction is greater.

In cases where the effort/speed characteristic is not spontaneously favorable, the solution which we use is to rapidly reduce the motor effort of the axle which is slipping, is to momentarily shunt the armature of the corresponding motor by appropriate resistance. But another solution which we have tried out is to stiffen the characteristics by a permanent shunting of motors at low speeds (starting). The effects of weight transfer can, if it is so desired, also be offset by applying this shunting only to the axles with the least load.

It should be noted that body load transfer is greatly reduced if not eliminated, in the case of a pusher locomotive by buffer friction with the European type of coupling. We do not know what happens with the U.S.A. type of coupling.

One interesting piece of information which I think I can give here is the adverse effect of certain types of shoes in composite material. In practice, this type of shoe, installed on some of our suburban railcar units is far from giving entire satisfaction. In these shoes, the surface between wheel and rail.

J. P. Van Overveen

Abstract

The authors have made a valuable contribution toward improved understanding of the mechanism of rail-wheel adhesion.

During the operation of the Test Track of BART, a number of adhesion observations were made with transit type cars, both in acceleration and deceleration. Some of the results would confirm, others would indicate contrary to the findings of the paper under discussion.

The characteristics of traction forces between wheel and rail of a locomotive will differ from that of a transit car, which has all axles powered, and does not transmit tractive effort from the coupling of a train made up of such cars.

In general, the adhesion factor as such should not change due to mechanical changes in vehicle or track. The tractive effort will change depending on how well we can use the available adhesion. The reference to adhesion percentages (over 100 percent) might lead to confusing tractive effort with adhesion.

The following discussion is offered as a suggestion for further practical tests within the capability of present technology.

Definition

Adhesion or the adhesion factor is the ratio of the maximum available tangential force at the tread of the wheel, over the wheel load perpendicular to the supporting surface, just before the wheel slips or slides. Slipping occurs in the event the wheel turns faster than the car motion and sliding occurs when the wheels do not rotate but the car keeps moving (Fig. 7).

Discussion

A tangential force can only be transmitted between railhead and wheel tread, if the circumferential speed of the wheel differs slightly from the linear speed of the wheel center. Without such difference, the wheel would be rolling in the true sense of the word, and this cannot produce a tangential force on the wheel tread. Investigators such as Nouvion, Kraft, and Steiner [9] refer to the relationship between tractive effort and wheel creep or slip, indicating an optimum difference of linear car speed and wheel circumferential speed below which the available adhesion is not fully utilized, and above which undesirable slip or slide occurs. No values have been arrived at. The effect of wheel diameter and the form of the contact between wheel tread and rail head of different profiles has been reported, but without emphasis.

Several tests made with newly designed electric locomotives indicate that early detection and rapid response to wheel slip improves the adhesion performance of these locomotives. The sensitivity and short reaction time of the controls, in case of slip, is pointed to as a factor in improving the tractive effort performance. There is every reason to believe that the same benefit can be achieved during braking.

A smooth torque input to the driving axle is important, in positive as well as negative tractive effort. The quality of the gear train, use of helical gearing and other torsional vibration reducing drives and connections contribute to dependable adhesion performance. The use of thyristors and other static controls for electric powered rail equipment have proven to be an asset to good adhesion performance.

Sliding, which usually occurs on wet or otherwise contaminated rail surfaces, is actually a lubrication phenomenon. Some agent acts as a lubricant. If a rail is clean, the adhesion is high, no matter if the rail is dry or wet. A moist rail is slippery because the moisture acts as a binder for small dust and dirt particles, and a lubricant is formed. As soon as a good rain has washed this film off the rail, adhesion equals that of a dry, clean rail. One problem is how to maintain a noncontaminated contact surface between rail and wheel.

A transit vehicle with a wheel load of 12,000 lbs. and a 30 in. diameter wheel will exert wheel-rail contact pressure close to 100,000 psi. These unit pressures are of a complex nature and have been studied by many. Under such a unit pressure, it is...
very unlikely that a lubricant can exist at that point, under static conditions. Under true rolling, the same would be true given a relatively slow speed which allows time for the contaminant (or lubricant) to be squeezed out. This relatively low speed is assumed at vehicle speed below 100 mph.

Tests made on the BARTD Test Track with sensitive slip-slide control systems and very rapid responses of a hydraulically operated friction brake system showed an adhesion factor of over 10 percent on a rusty and moist section of track. This one test result was not repeated and therefore must be verified by further tests when the opportunity presents itself. In any event, the "traditional" type of braking system would be expected to show an adhesion factor of not more than 4 percent under these circumstances.

The data just mentioned were produced by:
1. Frequency generated speed signals from a braking and a "free wheeling" truck,
2. A signal of slide control activation,
3. Brake cylinder pressure indication,
4. Car deceleration by accelerometer,
5. Brake command signal, and
6. Torque measurement through load cells mounted on brake rigging.

Other adhesion data were obtained at the Concord Test Track showing essentially the same values for tangent and curved track with a truck that had a very tight wheel gauge (less than 5/16 in. smaller than rail gauge). AAR standard gauged truck (wheel gauge 5/16 in. less than rail gauge) showed a lower adhesion on tangent track than in curves but the "in-curve" values were essentially the same as those of the "tight gauge" truck (somewhere in the 14 percent to 15 percent adhesion value on wetted track). This gave emphasis to the importance of proper track alignment and the effect of wheel-rail contact profile on the adhesion performance.

All these data were continuously recorded by an oscillograph. Moment of slip and all other pertinent events were defined clearly, Fig. 8.

The tests were not set up to determine the optimum differential velocity between rail and wheel for maximum adhesion, but analysis of the curve seemed to indicate that the order of magnitude was 0.5 mph at various speeds between 0 and 30 mph of car speed.

Future experiments may further bolster the theory that, given a steady vertical force, the differential speed for maximum adhesion is a constant.

Fig. 9 shows that the increment of wheel circumference travel for a given differential speed gets very small as the train speed increases.

This may explain one reason for observed reduced adhesion with increasing speed, as the time available to correct for slip or slide reduces as the train speed increases. Kraft [40] refers to this as well as the effect of a third agent contaminant between rail and wheel.
In future tests, efforts should be made to record the vertical force on the wheels. This can be done by load cells placed at the journal bearings under the pedestal springs or a similar location. The effect of axle alignment, the correct tramming of the tracks, the condition of the wheel tread, the flange clearances, all influence the experimental results.

The investigations of several researchers [40, 41, 42] into the phenomena of solid adhesion of metals indicate that when two metallic surfaces are forced together and then moved in relation to one another, they tend to stick together. The holding force resulting from this “sticking together” is said to be related to the alloys involved, with very little, if any, effect of the hardness or smoothness of the contact surfaces involved.

This should be basically applicable to the subject of wheel-rail adhesion problem. With instrumentation as mentioned herein, the adhesion theories could be proven and further developed for practical application.

In order to know how the rail-wheel adhesion can be used to the highest extent, it is necessary to know what the adhesion value is.

It is recommended that instrumentation be used, which can detect very small changes in rotational velocities, comparing these velocities to a “master” signal and recording the changes that occur together with changes in wheel load. The ‘master’ signal, insofar as speed is concerned, can come from a “free wheeling” truck or axle.

Data to be entered on a multichannel tape recorders for later analysis or on an oscillograph should include:

1. Freewheeling truck speed
2. Working axle speed
3. Speed differential readout
4. Friction brake cylinder pressure
5. Dynamic braking motor current
6. Propulsion motor current
7. Slip control signal commands
8. Slide control signal commands
9. Car accelerometer reading
10. Time reference
11. Car location reference on track
12. Wheel loads (both wheels of observed axle)

Other data to be noted are:

1. Wheel diameters
2. Wheel alloys
3. Wheel profile
4. Rail profile
5. Rail condition
6. Rail surface contamination
7. Track jointed or welded
8. Truck design
9. Disk brake or tread brake

Additional References


C. E. Tack

The authors are to be commended for the excellent presentation of their work which is in a very vital area. Wheel-rail adhesion is of equal interest to the brake designer. A comprehensive series of brake tests conducted in 1949 and 1950 disclosed the range of adhesion levels to be similar to those reported by Marta and Mels.

Later brake tests conducted on the Santa Fe with high speed passenger cars in a train further confirmed the wide range of adhesion available. Weight transfer is particularly troublesome when axles are individually driven. Care should be taken to assure that the best truck geometry is incorporated, as pointed out in this paper.

I appreciate the opportunity to offer these comments on Messrs. Marta and Mels’ excellent paper.

J. E. Weigel

The authors are to be congratulated for taking a major step toward dispelling the mystery and folklore surrounding the study of wheel-rail adhesion. Their well-planned, thoroughly instrumented, and carefully executed program has performed a needed service in compressing the dismaying spread of adhesion coefficients data shown in their Fig. 3, and in documenting the effects of significant variables on the adhesion available in modern railroad operation.

With regard to Fig. 3, the high adhesion attained by the rod drive locomotive (curve 5) is derived at least partially from the coupling effect of the drive arrangement, in that all drive wheels must lose adhesion before slip begins.

The following comments pertain to the indicated sections of the paper:

In Instrumentation. Reproductions of representative instrument records, admittedly troublesome to obtain, are always helpful in authenticating the conclusions reached. Perhaps such traces can be added when the paper appears in final form.

Test Results. The most important result, that adhesion is relatively independent of speed, was most gratifying to this discussor, in that it substantiates a similar, rather lonely conclusion based on his extensive adhesion test program conducted for the San Francisco Bay Area Rapid Transit District (BART) at the Diablo Test Track. Even though the BART studies involved braking of light weight transit car prototypes running on welded rail at speeds up to 80 mph, the adhesion coefficient versus speed relationships found were remarkably similar to those shown in Figs. 5 and 6 (10). The BART welded rail results with three identical, single cars equipped with three different trucks and braking systems are summarized in the accompanying BART figure (Fig. 10). Dry rail data were not obtained because the friction brakes demonstrated were incapable of producing repeated slips on uncontaminated rail. The atypical shape of the oiled rail BART curve may be explained by the fact that the fixed rate contaminant spray system allowed increasing amounts of oil to be retained on the rail head as the car speed decreased. On a curve of about 3 deg (2,000 foot radius), the BART results differed from those reported by Messrs. Marta and Mels, in that, depending on the track construction and braking system studied, the adhesion coefficients ranged from about 25 percent less to 25 percent more than those obtained on tangent track. The authors fail to mention an obvious contributor to lowered adhesion on curved track, that of the different travel distances imposed on the inner and outer wheels of a fixed axle.

Available Adhesion for Different Typical Conditions. The discussor heartily concurs with the authors’ recognition of the importance of a fast acting wheel slip control system. Control sensitivity is equally vital, as demonstrated at the Diablo Test Track, where
forces transmitted across the area of contact). It is obviously possible to conduct laboratory tests in which, say, all but two and longitudinal forces and the normal couple (specifying the specify six quantities which are the lateral and longitudinal creepages and spin (specifying the relative motion) and the lateral

Conclusions. The authors have generated a body of data that will be invaluable in predicting the adhesion performance of locomotives in main line operation. Not only have they indicated the expected limits of adhesion under a variety of realistic rail and track conditions, but, more important, they have shown that the magnitude of the coefficient is relatively independent of speed. No longer, let us hope, will engineers be cautioned against applying the brakes at speeds above that at which adhesion drops to zero!

A. H. Wickens

The results of the adhesion measurements carried out by the authors will be very useful particularly to those concerned with the design of locomotives which are generally similar to those used in the experiments. However, from the point of view of the research worker and those of us who are concerned with the investigation of possible vehicles which may be very different from the conventional locomotive of today, all adhesion measurements recorded in the literature leave much to be desired.

Much work has been done and is being done on understanding the physico-chemical state of the interface between wheel and rail and on cleaning the rail by chemical, mechanical or thermal means. These investigations are clearly very important, but in addition, it is necessary to recognize the relationship between the dynamics of the vehicle and the apparent adhesion. For adhesion is simply one aspect of the problem of rolling contact, and the forces induced during rolling contact are dependent upon the vehicle dynamics.

In order to specify the conditions in the contact area between wheel and rail at any given instant of time, it is necessary to specify six quantities which are the lateral and longitudinal creepages and spin (specifying the relative motion) and the lateral and longitudinal forces and the normal couple (specifying the forces transmitted across the area of contact). It is obviously possible to conduct laboratory tests in which, say, all but two quantities are made zero, and then to produce a characteristic representing, for example, longitudinal force versus longitudinal creepage. In this case, the maximum longitudinal force would be a measure of the adhesion. However, in track tests all six quantities are nonzero and are required to specify the conditions in the contact area. In all existing measurements only a few of these quantities are recorded and it may well be that one reason for the scatter of adhesion measurements is that the values of these unrecorded quantities vary considerably from case to case depending on the motion of the vehicle at the time of measurement.

A good example of this is provided in the paper being discussed. It is noted that there was a 50 percent loss of adhesion while negotiating the 4 deg curve in the welded rail test site (last paragraph of "Test Results" in paper). As the authors point out, in a curve there is not only a demand for longitudinal force for traction, but a demand for lateral force due to curving, and consequently the total available friction (which may be considered a vector quantity if Coulomb's laws are accepted) has to be shared. The physical nature of adhesion remains the same on curved track as on straight track but the demand has altered and this demand is a function of suspension design and hence vehicle dynamics.

Thus, in addition to the very useful measurements of the type made by the authors, there is, I believe, a very real need for fundamental work on this subject.

Authors' Closure

We appreciate the considerable interest given to this paper and would like to thank all those who have contributed comments to and discussions of the test results and other information reported in our presentation on wheel-rail adhesion.

The following will cover our comments regarding questions raised by the discussers.

Comments to F. T. McLernery. The standard SD-45 locomotive is arranged so that the lightest axle is in the No. 4 position and therefore it is not exposed to virgin rail. The relationship of axle position to anticipated axle weight, from lightest to the heaviest is as follows: 4, 1, 2, 5, 6 and 3. The nominal weight of the No. 4 axle is reduced by an amount equal to 14 percent of the locomotive tractive effort, the No. 1 by 8 percent and the No. 2 by 6 percent (Fig. 11).

This means that at 50,000 lb TE the No. 4 axle load of a 402,000 lb locomotive (SD-45) is 60,000 lb instead of its nominal 67,000 lb. At 100,000 lb TE the same axle weighs only 53,000 lb.

On the other hand the No. 3 axle, which becomes the heaviest, weighs 74,000 lb and 81,000 lb at 50,000 and 100,000 lb TE, respectively. This gives rise to higher wheel, axle and rail stresses.

The No. 4 and No. 1 axles are the most prone to slip. The fact that the No. 1 axle is always exposed to virgin rail (on a lead locomotive) increases its susceptibility to slip, especially on straight track and at higher speeds. The effect of the virgin rail
makes the No. 1 axle comparable to the No. 4 axle even though it has less tractive weight shift. It should be mentioned that on a trailing locomotive the rails are conditioned by the leading units, and the No. 1 axle becomes more stable, with the result that the No. 4 axle slips most frequently.

During the road tests the locomotives normal wheel slip detection and correction device was disconnected so that automatic corrections were not made. Under these conditions synchronous wheel slips were occasionally noted on two or more axles. The slip conditions are different enough during synchronous slips for a rate of change of wheel speed (current) detection device to sense a wheel slip.

Comments to Mr. D. R. Meier. The conditions surrounding the measurement of the published lower limit of available adhesion might make the presented data look more reasonable. Primarily, the data represents the highest adhesion level that the locomotive was able to operate on the given rail surface condition without the use of the wheel slip control, and just below which wheel slip would take place.

With regards to whether the horsepower axle levels are getting too high for prevalent adhesion conditions, the authors did not intend to indicate any undue concern. There is no question that the available wheel-rail adhesion levels govern the delivered tractive effort on U. S. railroads within the lower operating speed range. This is one reason why the power output of diesel electric locomotives is regulated below 20 to 25 mph. The main purpose of our paper was to review a number of factors that would take place.

Mr. Meier commented that “the authors do not really answer the question of whether there is a limit to maximum horsepower/axle which might be used.” Certainly some of the information in the paper may have partially answered this question. However, it is our opinion that it would be unwise to set such a limit because such data is dependent on the many physical parameters involved, the state of technology, and the economics of railroad operation.

Comments to Mr. P. R. Nayak. Mr. Nayak neglected to list the most demanding power dissipation required of today’s locomotives, that used to pull a train up a grade and to operate through curves.

Comments to Mr. J. E. Weigel. It was the desire of the authors to

Comments to Mr. P. R. Nayak. Mr. Nayak neglected to list the most demanding power dissipation required of today’s locomotives, that used to pull a train up a grade and to operate through curves.

The composition of sand used and it grain size was as follows:

White silica sand—50-80 mesh in size (0.005-0.007 in) cleaned and dried.

The quantity of sand used in our tests was 1 lb/min or approximately 22 cubic inches per min per trap.

The authors agree with Mr. Nouvion that, in general, the loss of adhesion on curves is inversely proportional to the radius of curvature, assuming that track gage and other parameters remain the same on different size curves. The SD truck has a longer wheel base (414 meters, 163 in) than the one mentioned in Mr. Nouvion’s comments, and it is a three axle truck. It is reasonable to state that the 50 percent loss of adhesion experienced with the SD-45 locomotive tests on a specific curve is higher than normally expected for a 4 deg curve with nominal 4 in. wheel to rail clearance. The extent of adhesion loss is very likely due to excessive gage widening which existed on the curve at the time of the test. Based on a standard track gage (561/2 in.), a 4 deg curve size, and friction—creep data obtained in laboratory tests at EMD, a 15 to 20 percent adhesion loss would be expected.

With regard to Mr. Nouvion’s final comments on the effect of composition shoes, EMD is not aware of any adverse effect on wheel-rail adhesion, and our experience to this date has been that the high friction composition shoes give considerably longer life and result in lower maintenance cost due to the longer life and fewer number of shoes used, i.e., one composition shoe per wheel is used compared to two cast iron shoes per wheel on SD locomotives. U. S. railroad experience indicates that composition shoes have reduced wheel slide flats significantly.

Comments to Mr. J. P. Van Overveen. Reference 2 of this paper, Vermolen and Johnson,12 and Kalker14 cover the effect of wheel diameter (actually, contact stress level is the very important parameter—as determined by wheel load and wheel geometry) and of contact area and shape. These references include experimental or theoretical treatment of one or more of the indicated parameters, and should be of considerable interest to Mr. Van Overveen.


Fig. 11 Weight shift of a standard SD-45 locomotive, where $T$ is the tractive effort
include some typical instrument records in this paper; however, although the data obtained from the x-y plotter records was accurate, the records are not considered presentable.

Mr. Weigel states that “the authors fail to mention an obvious contributor to lowered adhesion on curved track, that of the different travel distances imposed on the inner and outer wheels of a fixed axle.” This was not stated because, although it seems obvious, it is not really true. If the wheel pairs were not fixed on an axle, but were free to rotate independently while driving, similar losses in adhesion would prevail. The reason for this is that it is not the different longitudinal creepages imposed by a fixed axle that cause the loss, but rather the imposed lateral and spin creep and their associated forces which use a percentage of the available adhesive force. The contact area has a certain ability to do work (transfer energy) governed by a host of variables. If lateral forces due to curving exist in addition to the longitudinal driving forces, the vector sum of these two components makes up the total available adhesion force between wheel tread and rail.