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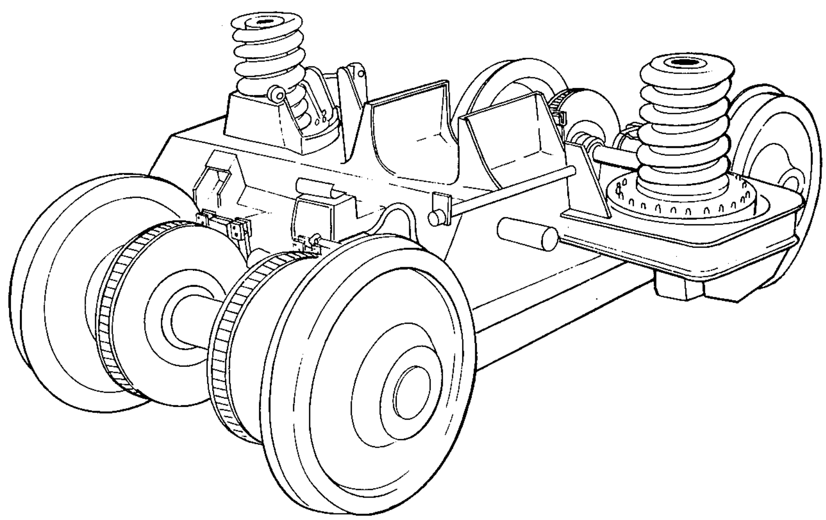
TECHNICAL REPORT

PASSENGER VEHICLE BRAKING STUDY



MAY 1978

DEPARTMENT OF TRANSPORTATION ★ UNITED STATES OF AMERICA ★

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16. Abstract

This report is a summary of currently available brake components and braking systems that might be applicable to 150-mph passenger service. The summary includes an analysis of the braking problem, a description of braking systems now in use and an evaluation of several advanced braking systems. The report reviews whether or not eddy-current brakes should be developed for use on Amcoaches and/or Metroliners in high-speed service in the Northeast Corridor. This report also considers what systems or components should be developed in the event that eddy-current brakes prove unusable.

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EXECUTIVE SUMMARY

This report is a summary of currently available brake components and braking systems that might be applicable to 150-mph passenger service. The summary includes an analysis of the braking problem, a description of braking systems now in use and an evaluation of several advanced braking systems. The purpose of this report is to review whether or not eddy-current braking systems should be developed for use on Amcoaches and/or Metroliners in high speed service on the upgraded Northeast Corridor (NEC). The report also considers what systems or components should be developed or adapted for this service in the event that eddy-current brakes prove to be unusable. The eddy-current braking systems developed by the European railroads and other advanced braking systems were studied so as to make recommendations concerning their use on present and future passenger consists. The study indicated that the braking systems now used on Amcoaches and Metroliners would be suitable for higher speed service if certain modifications or additions (all within reach of current technology) are made. Both the Amcoaches and the Metroliners were found to be capable of meeting the projected 120-mph service requirement of the first stage of the NEC upgrade program, assuming that the minimum block distances or required stopping distance is not less than 10,000 feet. The Amcoaches may incur even greater wheel problems as the operating speed is increased but indications are that the discs are capable of dissipating the added energy for 120-mph operation. Both the Amcoaches and the Metroliners are predicted to be near their limit if operated in 140-mph service. Both are capable of stopping from this speed but wheel and component problems will make maintenance costs very high.

Since both Amcoaches and Metroliners are basically capable of meeting the near term objectives set for the NEC with existing braking systems, eddy-current or other advanced braking systems

do not appear to be justified. For those Amcoaches operating on the NEC, the addition of tread brakes seems like an improvement which could be justified by reducing the cost of wheel maintenance. Wheel-slide-control appears to be an important factor in high-speed braking, and systems which react to variations in adhesion through the inherent physical characteristics of their operation seem best capable of providing the required braking at speeds in the 130-to 150-mph category. Systems like hydrokinetic brakes or air retarders were judged to be most promising for future use.

1.0 INTRODUCTION

1.1 GENERAL

This report has been prepared to support the development of test plans for evaluating eddy-current brakes. The study reported herein was made to evaluate the requirements for high-speed braking and to compare the performance of the eddy-current brake to other types of braking. In the process of developing detailed plans for procurement and subsequent testing of eddy-current brake devices, it became apparent that the eddy-current systems were very heavy, required huge amounts of electrical power and caused excessive heating in the rail. Eddy-current systems have been developed by both the French and German railroads for high-speed braking applications, and have also been extensively evaluated by Japan and England. The French equipped their Z-7001 train with prototype eddy-current hardware but later removed the system and have reportedly abandoned eddy-current brake development. The same negative reports have filtered out of the other nations. The Germans have dropped all active work on eddy-current brakes and the English decided that they could operate satisfactorily without eddy-current brake systems. The detailed reasons for these decisions are covered in the sections on eddy-current brake systems, but the basic complaint was that the eddy-current devices caused excessive heating in the rail. Reportedly, the rail facilities groups ultimately vetoed the use of eddy-current systems. The reports leave some indication that the equipment developers may have been relieved at this outcome and did not protest strongly. Each group who considered the use of eddy-current brakes apparently decided that they could operate satisfactorily without this system and even without any independent form of braking.

The attractive aspect of eddy-current brake systems is that the braking is directly between the rail and the truck and therefore, braking does not rely on wheel-to-rail adhesion. This means that the braking is not adhesion limited and the wheels are not loaded with all of the retarding force. With the systems now in use, however, the wheel slip increases as the braking force approaches the adhesion limit. If brake forces are not reduced, the wheel will stop rotating and slide, damaging the wheel and the rail surface. The linear version of eddy-current brakes is adhesion independent. Hence, these systems have the potential of providing higher braking rates and diverting some of the braking from the wheels.

This report reviews braking requirements, and both the conventional and the new braking systems which have been adapted by various other nations for use on high-speed consists. The intent is to put the eddy-current braking system in perspective relative to our requirements and to provide some insight as to whether we should proceed with eddy-current brake testing even though other nations have reportedly discarded application of eddy-current systems.

1.2 THE BRAKING PROBLEM

The Railroad Revitalization and Regulatory Reform Act of 1976 established a mandate that rail passenger service operating on a schedule of three hours and forty minutes between Boston and New York and two hours and forty minutes between New York and Washington, DC be established by 1981 (Appendix A). Two years after the date of enactment, a report must be submitted on the feasibility and other factors related to reducing the New York to Boston trip time to three hours and the New York to Washington, DC trip time to two hours and thirty minutes. The train performance calculations performed by Carnegie Mellon Institute and Transportation Systems Center indicated that in order to meet the initial objectives set for 1981, the maximum train speeds will have to be 140 mph. To maintain schedules, the trains will need to operate at a maximum speed of 130 mph to meet the two-hour and 40-minute requirement and 150 mph to meet the two-hour and 30-minute requirement.

After the maximum operating speed is defined, the braking problem breaks down into the following two fundamental elements:

- Adhesion.
- Kinetic energy dissipation.

The braking capabilities of high-speed trains affect both train safety and train performance. To avoid accidents, it would be ideal to stop high-speed trains in very short distances. However, if the train is slowed too quickly, the passengers can be thrown down and possibly injured. With conventional braking systems, this problem is seldom experienced because the limitations of wheel-to-rail adhesion will not allow braking rates which lead to passenger injury from stopping too quickly. The primary safety problem has traditionally been stopping soon enough to avoid collision. The braking rates established by recent Amtrak-equipment specifications stipulate that the maximum deceleration at any time should not exceed 2.75 mphps for Amfleet cars and 3.0 mphps for the new bi-level cars. These rates would correspond to 0.13 and 0.14 g, respectively. At this deceleration, a 150-pound person is pushed forward with a force of about 20 pounds. If this force were applied suddenly, it would cause a person to fall. Since the train does not respond quickly, the passengers should be capable of bracing against stops of even greater severity.

The second factor related to establishment of brake rates is train performance. This factor might prove to be critical in meeting future objectives (requiring 140 mph), but the studies on trip time for the current track plans do not indicate that the trip time can be improved significantly by variations in brake rate. The train performance calculations are normally based on brake rates of 1.46 mphps. These performance calculations indicated that time is only improved by one percent for a twenty-percent change in braking rate. For current programs, the improvement in brake rates obtainable by use of eddy-current or other improved brakes is not predicted to improve train performance.

Since train performance does not dominate the required brake rate, the next priority is stopping distance based on signaling requirements. The Northeast Corridor presently has block distances from 6,000 to 12,000 feet. The short blocks are not in high-speed-track locations; the shortest effective block for high speed service is 10,000 feet. Reports from the signaling group at the NEC Office indicate that the present plans are to maintain 10,000-to 12,000-foot blocks in the upgraded NEC. This would indicate that the high-speed trains should be capable of stopping inside of 10,000 feet. With this data, the braking problem can be placed in perspective by reviewing some basic physical principles.

If braking is accomplished entirely by the wheels, normal-service braking is limited by the coefficient of adhesion between the wheel and rail surfaces. The coefficient of adhesion varies considerably depending on the conditions of the contact surfaces. In practice, train deceleration is limited to about 1.86 mphps for speeds above 80 mph and 2.0 mphps for speeds below 40 mph.

The relationship between stopping rate and the coefficient of adhesion can be developed from Newton's law of momentum:

$$\text{Force } (F) = \text{mass} \times \text{acceleration} = ma$$

where

$$m = \frac{w}{g}$$

w = the weight of the mass

g = the acceleration due to gravity.

then

$$F = \frac{w}{g} \times a$$

If all force is developed by the wheels and

μ = coefficient of rolling friction or adhesion

$$F = w\mu = \frac{w}{g} \times a$$

therefore

$$a = \mu g$$

Values of μ for various rail conditions and train speeds are shown in Figure 1-1.

For a complete consist

maximum velocity (v_{max}) = acceleration x time

distance traveled (S) = v_{av} x t

assuming constant deceleration

$$v_{av} = v_{max} \div 2$$

and

$$S = \frac{v^2_{max}}{2a}$$

if a is expressed in mphps and v is expressed in mph

$$S = \frac{(1.46)^2 \ v^2}{1.46 \times 2 \times a} = 0.73 \ \frac{v^2}{a}$$

if a is expressed in g's and v is in mph

$$S = \frac{(1.46)^2 \ v^2}{2 \times 32.2 \times g} = \frac{0.033 \ v^2}{g}$$

when stopping rate is limited by adhesion

$$a = \mu g$$

and

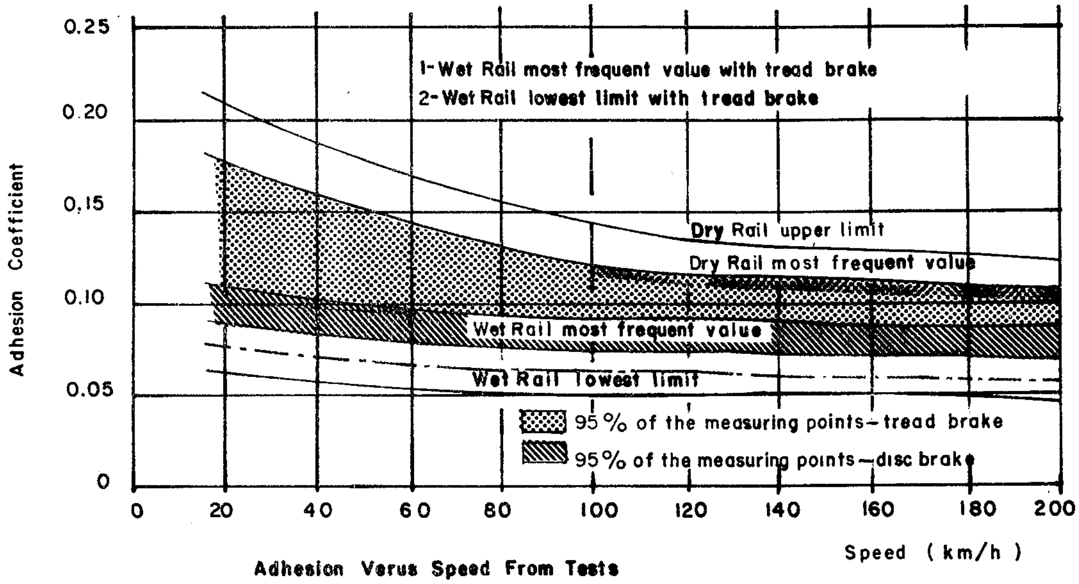
$$S = \frac{(1.46)^2 \ v^2}{2 \ \mu \ x \ g} \qquad = \frac{0.033 \ v^2}{\mu}$$

Using the wet-rail adhesion data shown in Figure 1-1,the limit of effective stopping distance versus speed can be illustrated as shown in Figure 1-2.

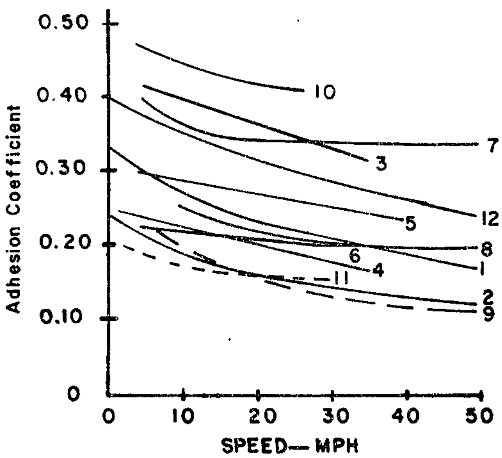
Figure 1-2 shows that the train will require about 8,000 feet to stop from 130 mph if all braking is developed by the wheel-to-rail interface under severe wet-rail conditions. When traveling at 150 mph, the train would require 10,500 feet to stop. These stopping estimates assume that the system is capable of dissipating the kinetic energy without damage to the braking apparatus.

The second curve on this graph shows the stopping distance versus train speed that would be obtained if the maximum brake rate of 0.14 g were maintained by an eddy-current or other non-adhesion-limited braking system. In this case, the train can stop from 130 mph in 4,000 feet and from 150 mph in 5,300 feet. This indicates that supplementary braking systems which do not depend on wheel-to-rail adhesion or some form of adhesion enhancement are not required if the 130-mph trains do not need to stop in less than 8,000 feet. For trains running at maximum speeds of 150 mph, the stopping distance is about 10,500 feet without a supplementary system.

This data is also illustrated in Figure 1-3 which shows stopping distance required (in feet) versus required deceleration, assuming that the wheels do not slide. This illustration shows that the required stopping rate for a train traveling at 150 mph

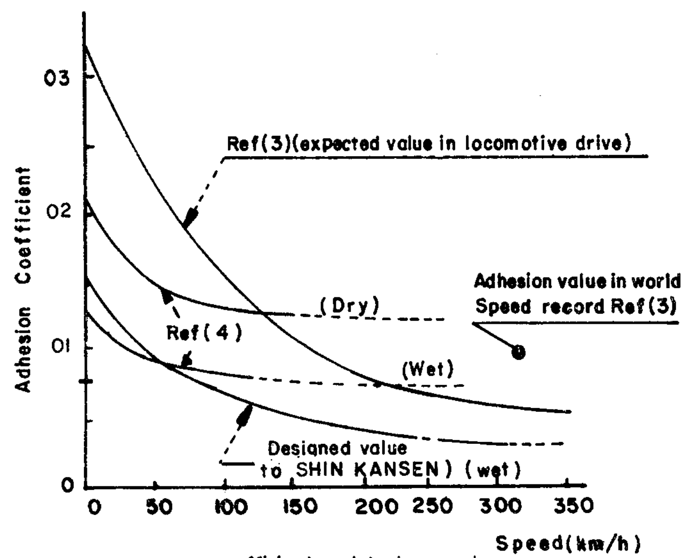


Adhesion Verus Speed From Tests
Performed by The SNCF (French National Railways)



Published data of adhesion as a function of speed

1 Muller – dry rail
2 Muller – wet rail
3 Andrews – dry rail
4 Andrews – wet rail
5 Bager and Ottoson-1-C-1 locomotive with rod drive
6 Bager and Ottoson-Bo-Bo locomotive with individual axle drive
7 Curtius and Kniffler – dry rail
8 Curtius and Kniffler – wet rail, upper limit
9 Curtius and Kniffler – wet rail, lower limit
10 Nouvion – maximum
11 Nouvion – minimum
12 Koffman



Adhesion coefficient and train speed

Figure 1-1. Typical Adhesion Data

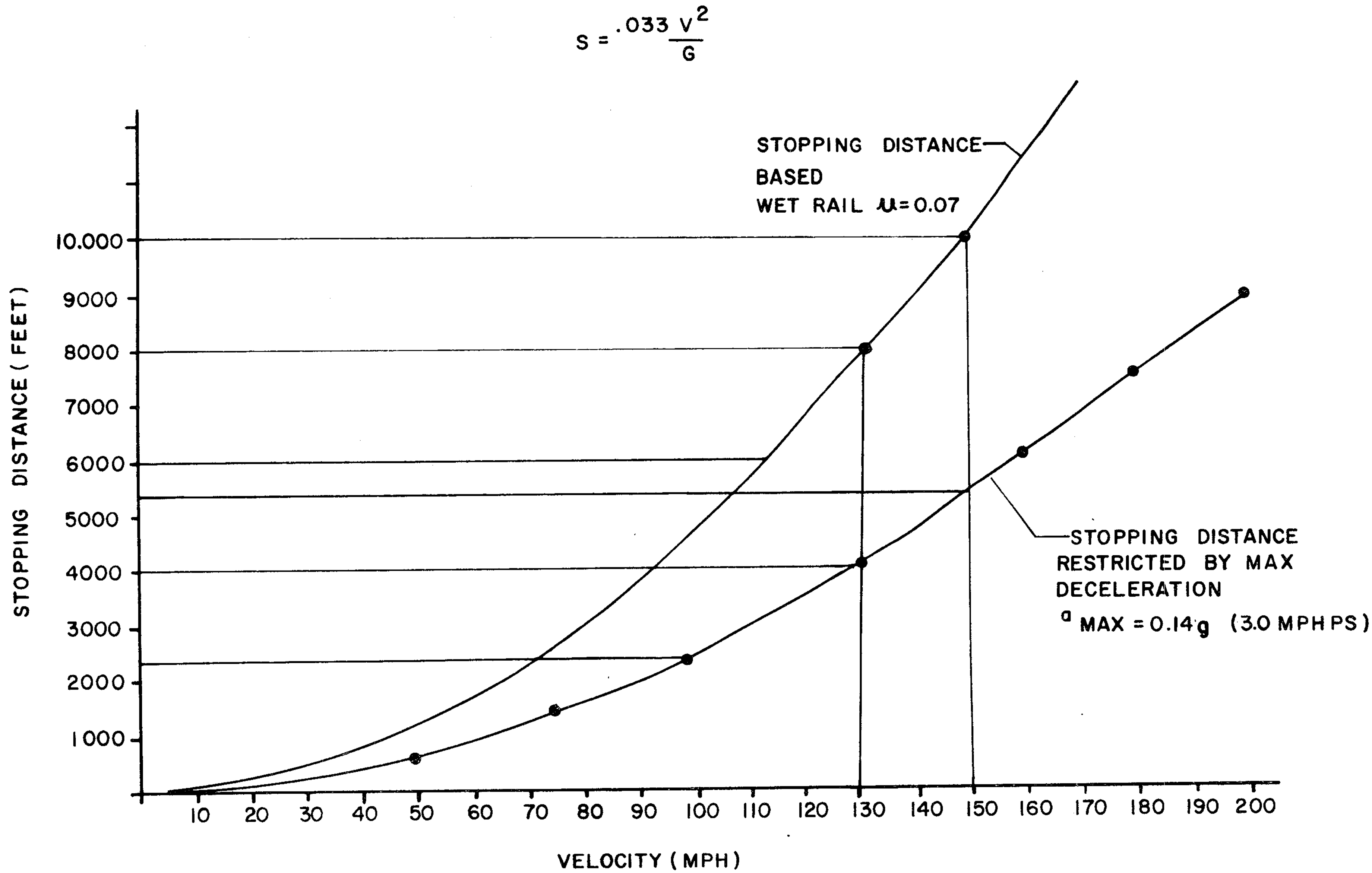


Figure 1-2. Stopping Distance Versus Velocity

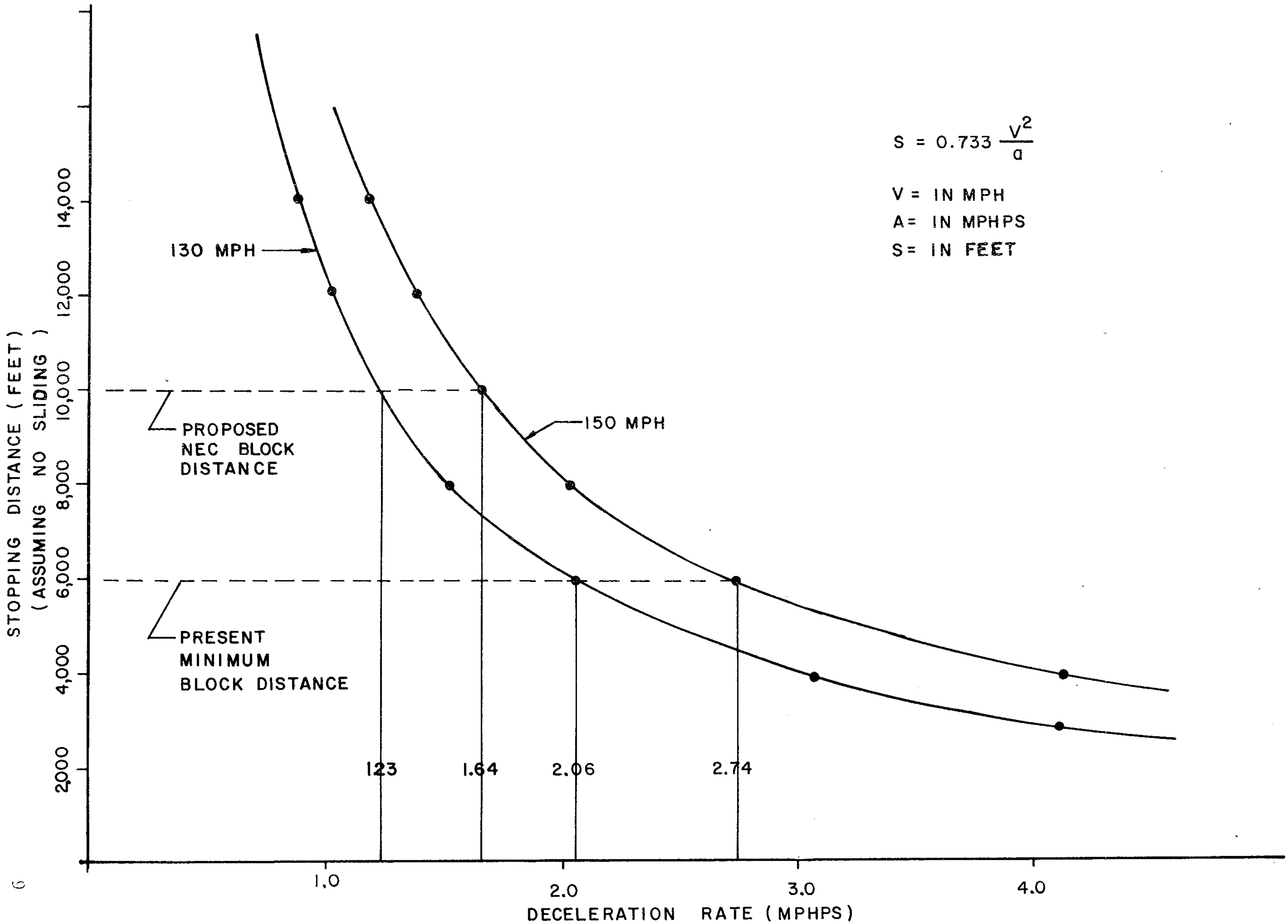


Figure 1-3. Stopping Versus Deceleration Rate

(assuming that the train is required to stop within the 10,000-foot block distance) is 1.64 mphps and the rate for stopping within a 6,000-foot block distance is 2.74 mphps. If the 10,000-foot requirement is maintained, a 130-mph train must stop with an average deceleration of 1.23 mphps. These rates can be obtained with conventional braking systems.

If the only reason for using eddy-current brakes is to achieve the desired stopping distances, the supplementary systems do not appear to be justified unless a stopping distance of less than 10,000 feet is required.

The additional kinetic energy (the energy of a body that results from its motion which is equal to one-half its mass times the square of its velocity) that must be dissipated during braking as a train's speed increases is shown in Figure 1-4.

The deceleration rates that are achievable without wheel-slip depend on the adhesion coefficient (μ). As μ decreases (as it does when the rail becomes icy, oily or wet),the train's deceleration rate is also reduced. If deceleration rate is not reduced enough, the wheels will slip and the slip may lead to sliding. Figure 1-3 shows the variation of stopping distance with deceleration rate. This is particularly important, since it sets a minimum standard of performance for braking systems, i.e., that the brakes be able to stop a train within the established block distance without sliding, thus preventing damage to the wheels.

These are some of the considerations that must be kept in mind when evaluating a braking system. The following sections expand on these topics in order to describe the state-of-the-art, i.e., the capabilities and limitations of present-day braking systems. Advanced braking systems and the problems involved with their implementation are also discussed.

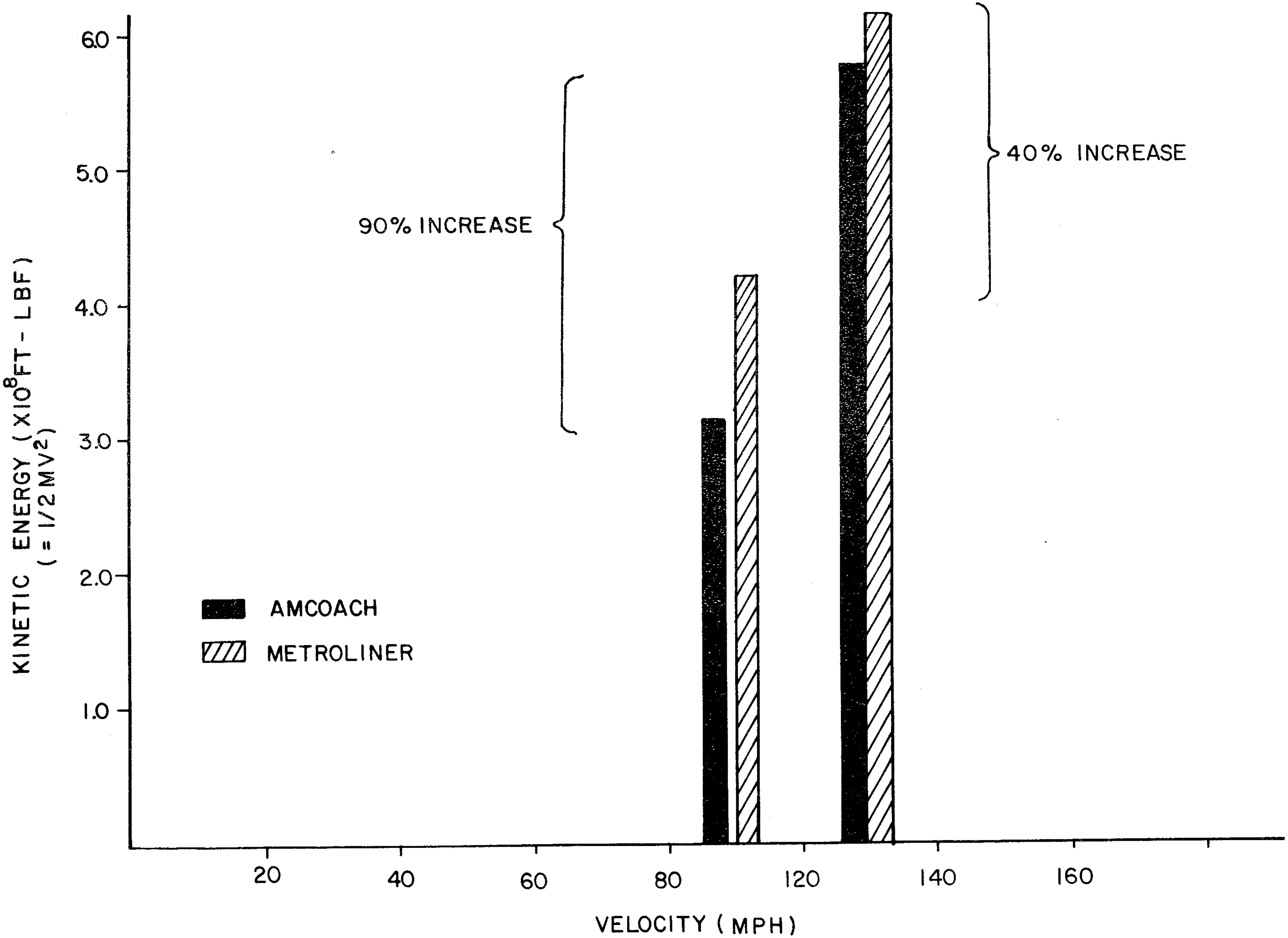


Figure 1-4. Kinetic Energy Versus Velocity

2.0 BRAKING PARAMETERS

2.1 ADHESION

There are two descriptions of adhesion. One is termed true adhesion, which is defined in the same way as the coefficient of friction: $\mu \equiv F/N$, where F is the tangential force developed by the wheel at its point of contact with the rail, and N is the normal force at the point of contact. Because of wheel-rail surface conditions, weight transfer during periods of acceleration or deceleration, wheel unbalance and eccentricity, track condition, spring stiffness, rail and roadbed resilency, and a number of other factors, the value of the adhesion found in actual practice (called apparent or practical adhesion) differs from true adhesion. Apparent or practical adhesion (referred to as the coefficient of adhesion) is defined as the ratio of the total braking force of the train to the weight of the train. This type of adhesion can be thought of as how slippery the wheel-rail interface is to an entire train, while true adhesion can be envisioned as the friction obtained in a laboratory test from a single wheel in perfect condition braking on a thoroughly-clean, unmarked rail. Both types of adhesion are illustrated in Figure 2-1. Unless otherwise noted, the term adhesion, as used in this report, refers to apparent adhesion.

Depending on the level of adhesion available, a rolling body will respond in one of three modes. Relatively high levels of adhesion are associated with the phenomenon known as creep, which is rolling with a very small degree of slipping.

As the adhesion coefficient decreases, this microslip phenomenon transforms into macroslip behavior commonly known as wheel-slip. This means that the tangential velocity of the wheel is significantly higher than its traveling velocity, i.e., there is relative motion between the wheel and the rail at the contact

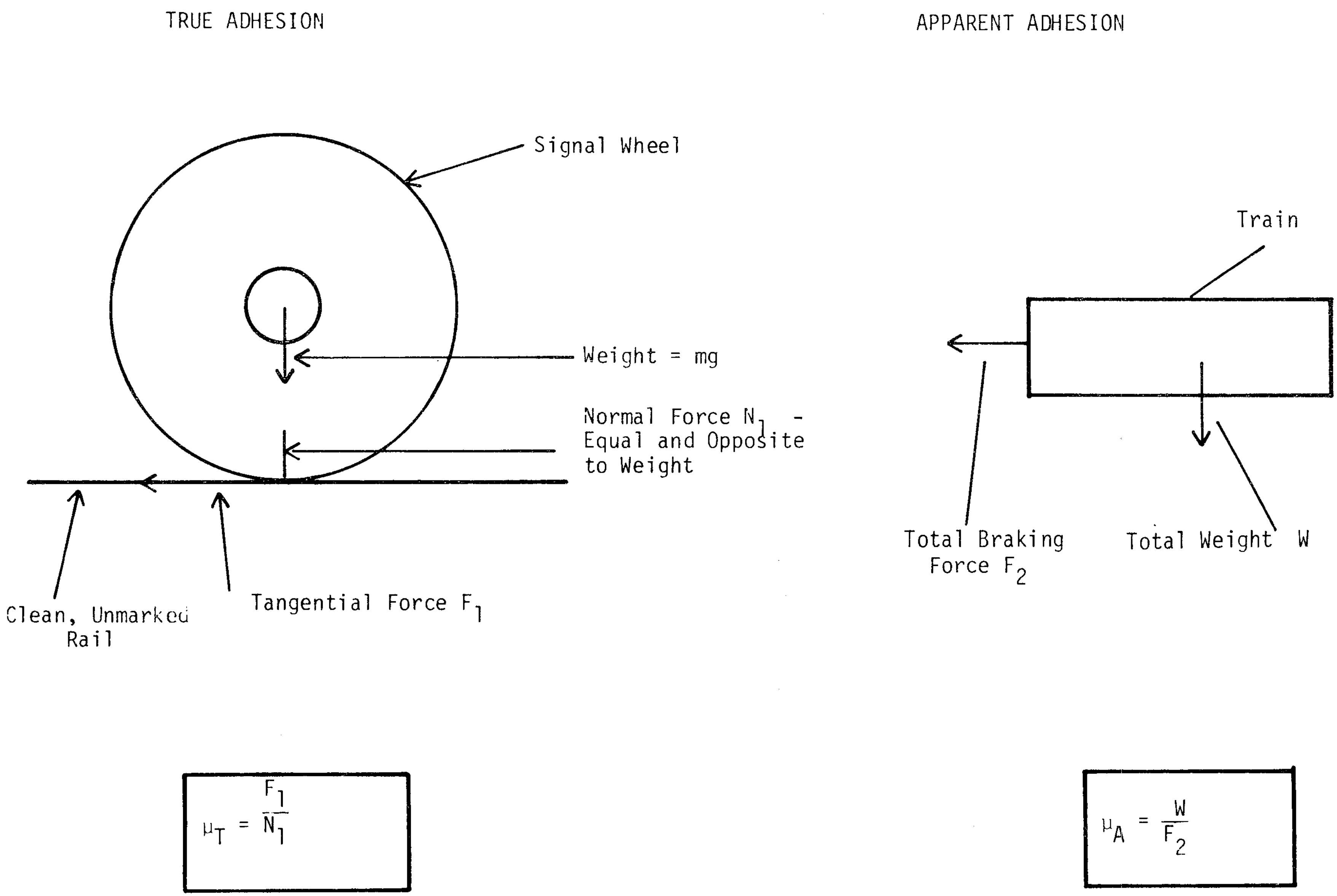


Figure 2-1. True Adhesion and Apparent Adhesion

surface. At low levels of adhesion, sliding takes place. In this case, the wheel moves forward without revolving as shown in Figure 2-2.

From the foregoing, it is evident that the adhesion coefficient is a primary factor in determining the maximum braking rate that can be achieved without causing slipping, which may quickly change to sliding. Although slip-slide detectors are installed to prevent sliding (Section 3.5), the time between the beginning of wheel slip and brake release may be sufficient to damage the wheels. In addition, the fact that the operation of these detectors releases the brakes tends to lengthen braking distances.

In general, there are three possible solutions to the problem of insufficient adhesion: to use an adhesion-independent braking system, to increase the available adhesion, and to use the available adhesion more efficiently.

2.1.1 ADHESION INDEPENDENT SYSTEMS

There are a number of adhesion-independent systems that have reached various stages of development. However, none of these seem to be readily applicable to the braking problem on the Northeast Corridor.

The eddy-current rail brake (Section 4-2) requires an unacceptably small gap (in the order of seven millimeters) between the rail and the brake for efficient operation. Its high weight (1322 pounds/brake) and the possibility of rail buckling associated with its operation are two other problems that would have to be investigated before the brake could be used.

The electromagnetic brake, as its name implies, consists of a row of electromagnets suspended between the wheels of a truck

ROLLING

No Relative Motion Between Wheel and Rail

SLIPPING

Tangential Velocity of Wheel Differs From Its Traveling Velocity

SLIDING

Wheel is Locked - Moves Forward Without Revolving

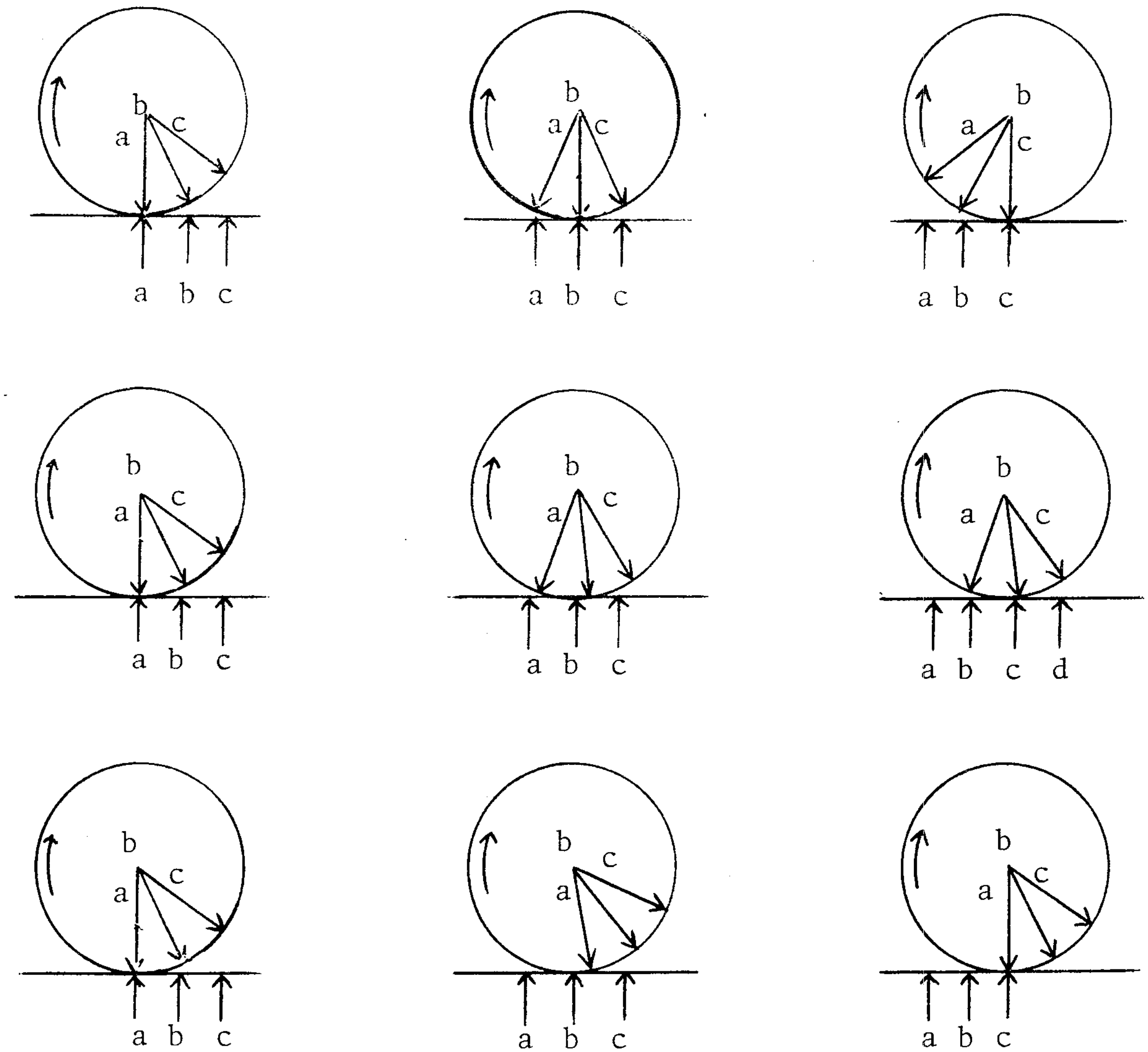


Figure 2-2. Response of a Wheel During Braking as a Function of Available Adhesion

(Figure 2-3). During braking, the electromagnets are lowered close to the rail and energized. The resulting magnetic field causes the brake to clamp down on the rail, thus producing a frictional retarding force.

Very high wear-rates are experienced by this brake at high speed because its operation involves contact with the rail. For this reason, its use is limited to special situations, i.e., emergency stops where a greater than usual amount of braking effort is required. This brake, which is not used in the United States, can also serve as a parking brake.

The linear induction brake can be used only on trains driven by a linear induction motor. Operating on the same principle as dynamic brakes (Section 3.4), this brake transforms the motor into a generator. The kinetic energy of the train is thus changed to electrical energy, which in turn, is changed into heat energy and dissipated.

The aerodynamic brake produces a retarding force by increasing the drag on the vehicle. One configuration, proposed by Calspan Corporation of Buffalo, NY, is to use cascade-type spoilers which are extended from the top of the car during braking (Figure 2-4). Deceleration rates produced by this type of braking fall exponentially with velocity; maximum anticipated performance of flat-plate spoiler brakes (which are less efficient than cascade brakes) exceeds the 4.0 mphps rate at speeds over 180 mph. At 100 mph the predicted brake rate is 2.6 mphps which is still 30 percent greater than the required 2.0 mphps. Another design was proposed for the 150-mph United Turbotrain (used in NEC service) but was never built.

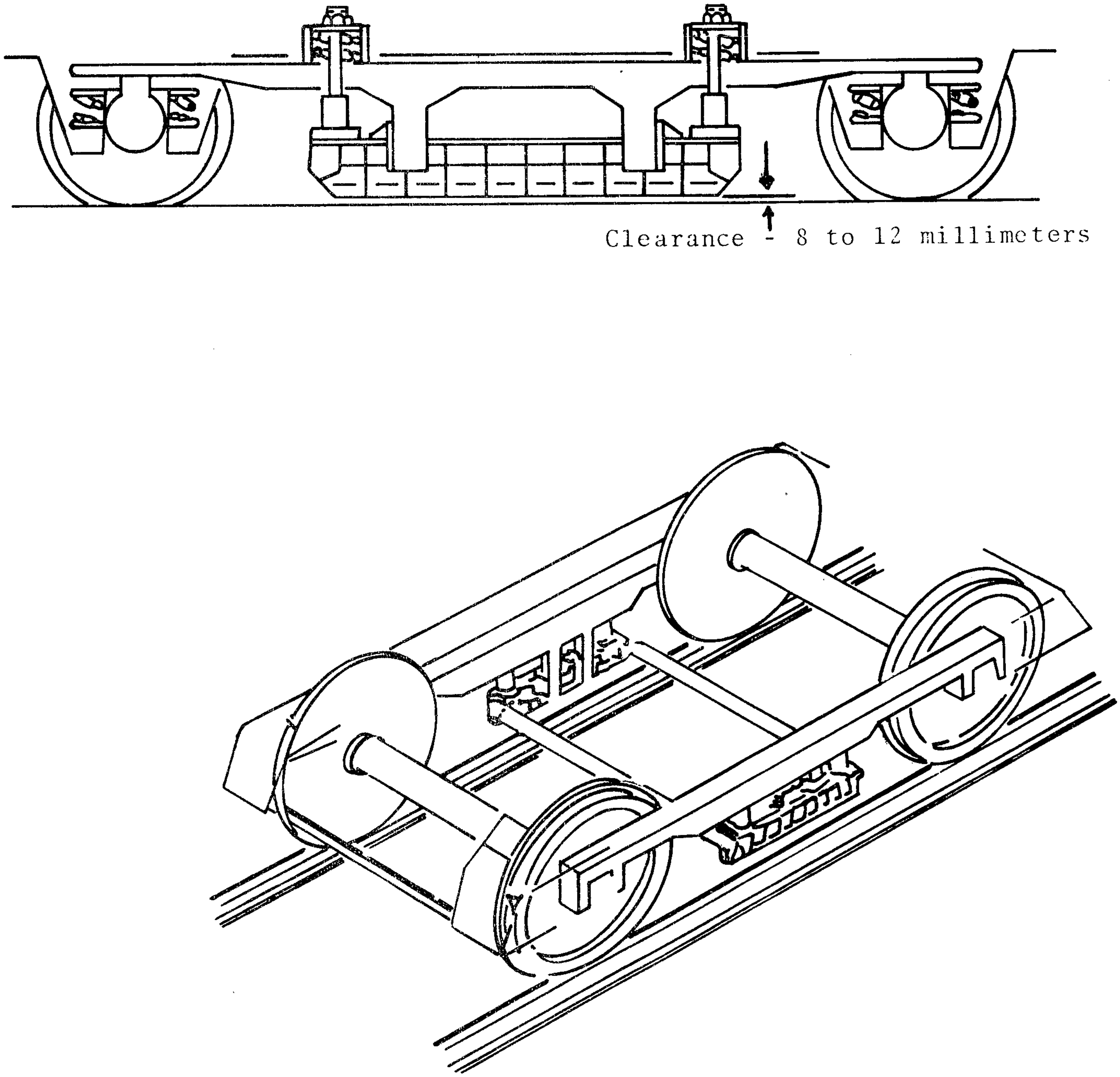


Figure 2-3. Electromagnetic Rail Brakes

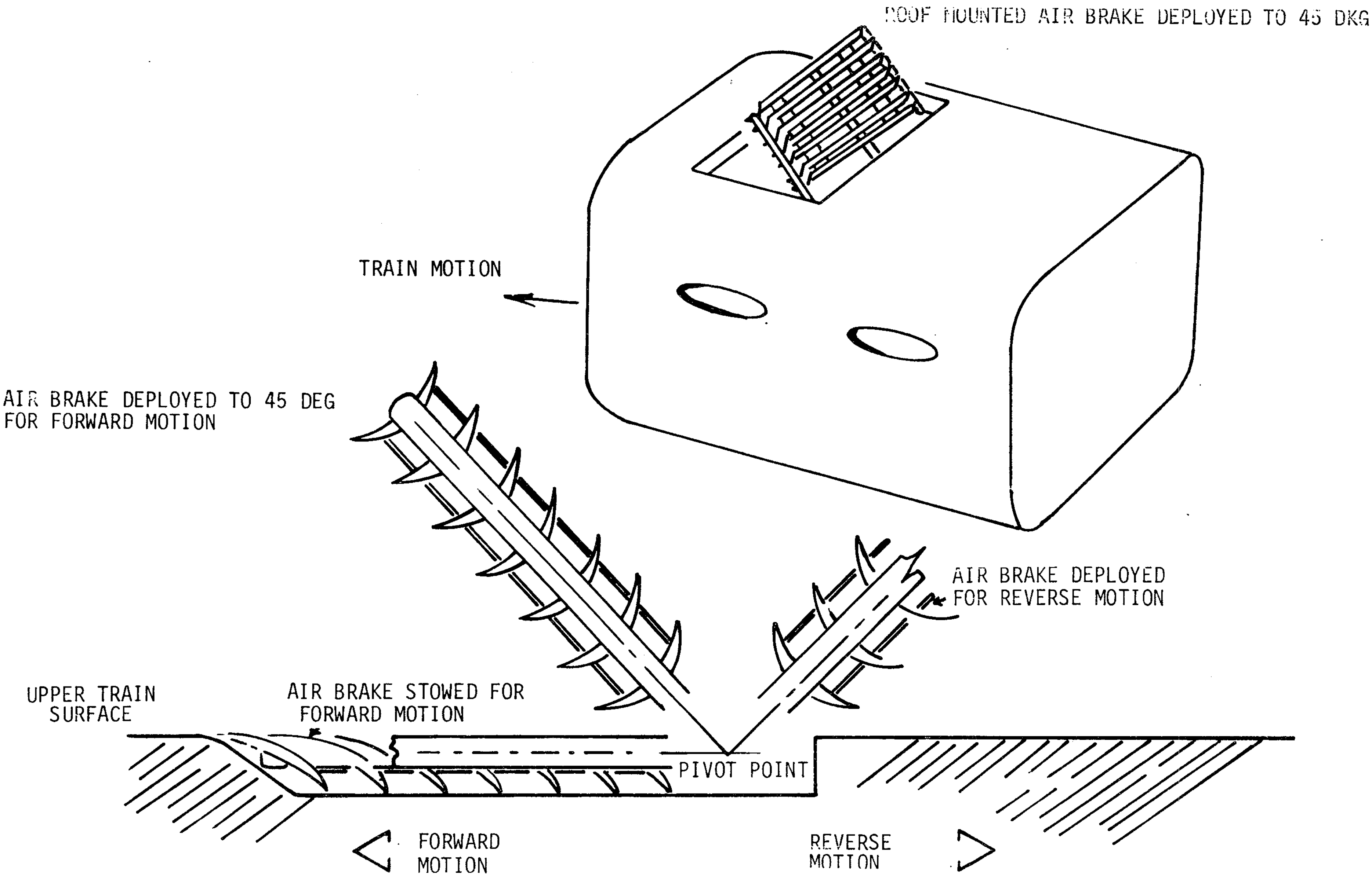


Figure 2-4. Bi-Directional Aerodynamic Braking Device - Two Dimensional Cascade With Stylized Airfoils

2.1.2 ADHESION ENHANCEMENT SCHEMES

There are at least four possible methods of increasing adhesion: the use of brake shoes containing iron particles, sand application, chemical rail cleaning, and the use of a plasma torch.

When brake shoes containing iron particles are used, adhesion increases by about 20 percent. This is probably due, at least in part, to the tendency for adhesion-reducing substances to be scrubbed off the wheel tread. In addition, cast iron particles leave the shoe during braking, and fall between the wheel and rail, which also has the effect of increasing the adhesion coefficient.

Sand has been used to increase adhesion for many years with some success. However, this method does have disadvantages, such as increased wear on wheels and rail, occasional failure of track circuits due to excess sand on the railhead, and the high cost of the system hardware. In addition, it becomes difficult to distribute the sand effectively as speeds increase.

To help solve the distribution problem, a fluid sand (a sand suspension) has been developed. Tests indicate that this material will be affected less by wind and will be more efficient in operation than dry sand.

Chemical rail cleaning involves applying a fluid to the rail to remove or displace oil-based contaminants from the railhead. The fluid used should not be poisonous, highly flammable, corrosive, cause environmental damage, or act as a lubricant. Many different compounds have been investigated in trying to meet these criteria and at the same time trying to design a suitable dispensing system. As far as is known, the results of this research have not been applied to the development of an operational rail cleaning system.

A plasma torch is a device that volatilizes contaminants from the rail by generating a high-energy spark between an electrode, which is placed just forward of the leading wheels, and the rail. While this system effectively cleans organic pollution from the railhead, it has several major disadvantages such as high power consumption (150 kilowatts start-up), an unacceptably small gap between the rail and the electrode, safety problems and high maintenance costs.

2.1.3 MORE EFFICIENT USE OF AVAILABLE ADHESION

The third approach to solving the adhesion problem is to make more efficient use of the adhesion that is available. This can be done by using a variable force braking system (Figure 2-5) or improving wheel-slide protection systems.

Variable-force braking systems include the eddy-current wheel brake, the retarder systems, and dynamic brakes. They make more efficient use of adhesion than the constant force systems (such as tread and disc brakes) because they are designed to brake up to but not beyond the adhesion limit. The constant-force systems may brake either under or over this limit, which opens up the possibility of wheel damage.

Improving wheel-slide protection systems is another way of using the available adhesion more efficiently. The problem with systems presently in use is that they can only release or apply the brakes. They have no control over the magnitude of the braking. This means that the train tends to stop in a series of jerks due to the brakes being constantly switched on and off, resulting in possible wheel damage, longer braking distances and a reduction in ride quality. All of these could be prevented if the braking force could be varied by the slip-slide detector. In this case, the train would be brought to a smoother stop,

<u>ADHESION INDEPENDENT</u>

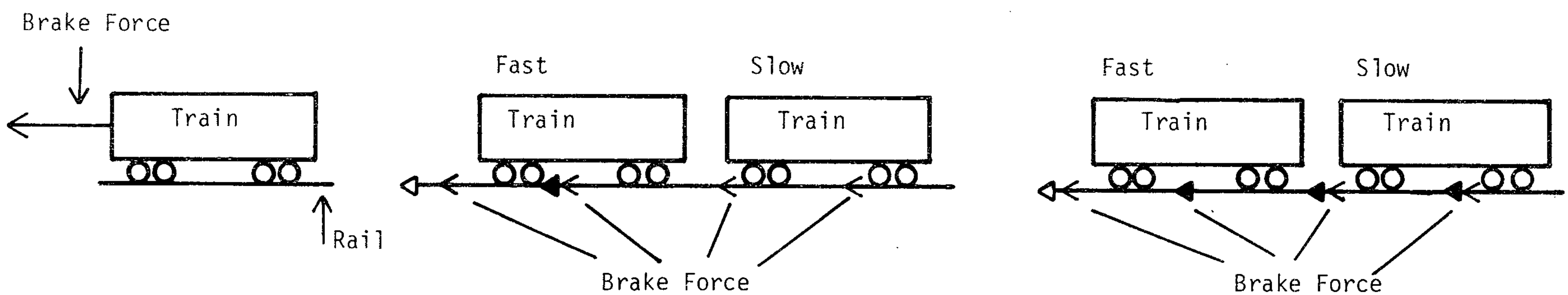
Braking Does Not Occur Through Wheel-Rail Interface.

<u>VARIABLE FORCE</u>

Brake Force Decreases Automatically With Speed - Wheels Cannot Lock.

<u>CONSTANT FORCE</u>

Amount of Brake Force is Manually Controlled - Wheels May Lock.



Examples: Eddy-Current Electromagnetic Track Brake Aerodynamic Brake.

Examples: Dynamic Brakes, Air Retarder, Hydraulic Retarder, Eddy-Current Wheel Brake.

Examples: Tread Brakes, Disc Brakes.

Figure 2-5. Categories of Braking Systems and less stress would be placed on the wheels (Figure 2-6). Another possibile way to upgrade wheel-slide protection systems would be to improve their reliability.

2.1.4 DISCUSSION

It is difficult to treat the adhesion problem on a quantitative basis. Because of the factors that make true adhesion different from apparent adhesion, it is virtually impossible to determine accurately the value of adhesion on a given length of track without measuring it under varying conditions at the location using the same vehicle as is used in service operations.

However, an assessment of the magnitude of the adhesion problem is necessary in order to determine whether or not new or improved brake hardware such as an adhesion-independent system, an adhesion enhancement scheme, or an improved wheel-slide protection system is required for safe and economical operation on the Northeast Corridor. The approach used in this study was to collect data from a number of sources and considering the conditions on the upgraded NEC, to arrive at average values for the adhesion coefficient for normal dry rail (slightly contaminated but in good mechanical condition) and for wet rail.

Based on the curves shown in Figures 1-1, 1-2 and 2-7, and on estimates provided by Knorr Brake Company and Amtrak, 0.12 was chosen as the adhesion coefficient for dry rail at high speeds (above 100 mph). The fact that adhesion decreases with increasing speed must be taken into account. Accordingly, this value was derived from coefficients at the upper end of the velocity scale. This figure can be checked by using Kraft's formula [3] for the change in adhesion as a function of velocity:

$$\mu = \mu_o \left[0.4 + \frac{0.6}{(1 + \frac{V}{300})} \right]$$

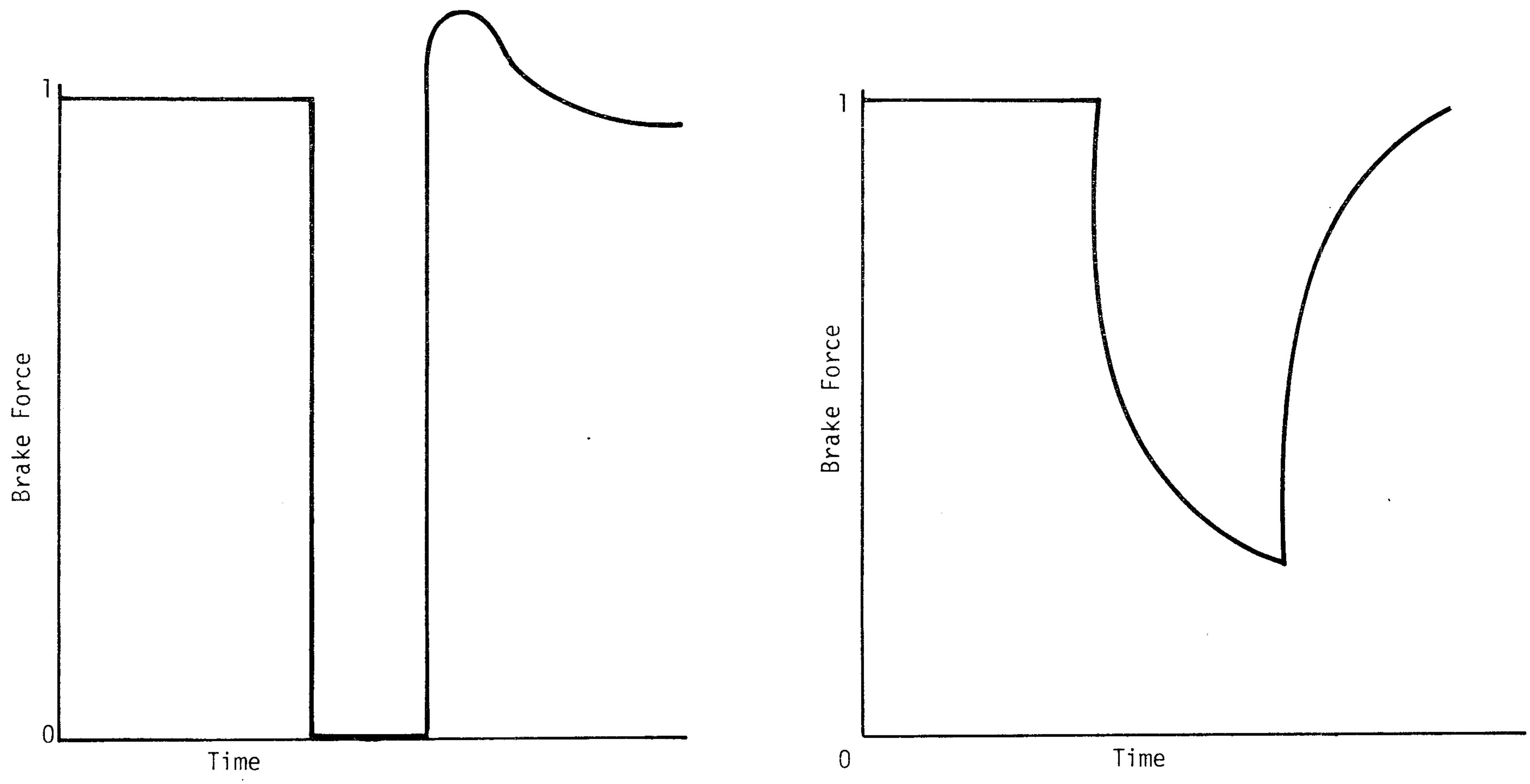


Figure 2-6. Response of Two Different Wheel-Slide Protection Systems

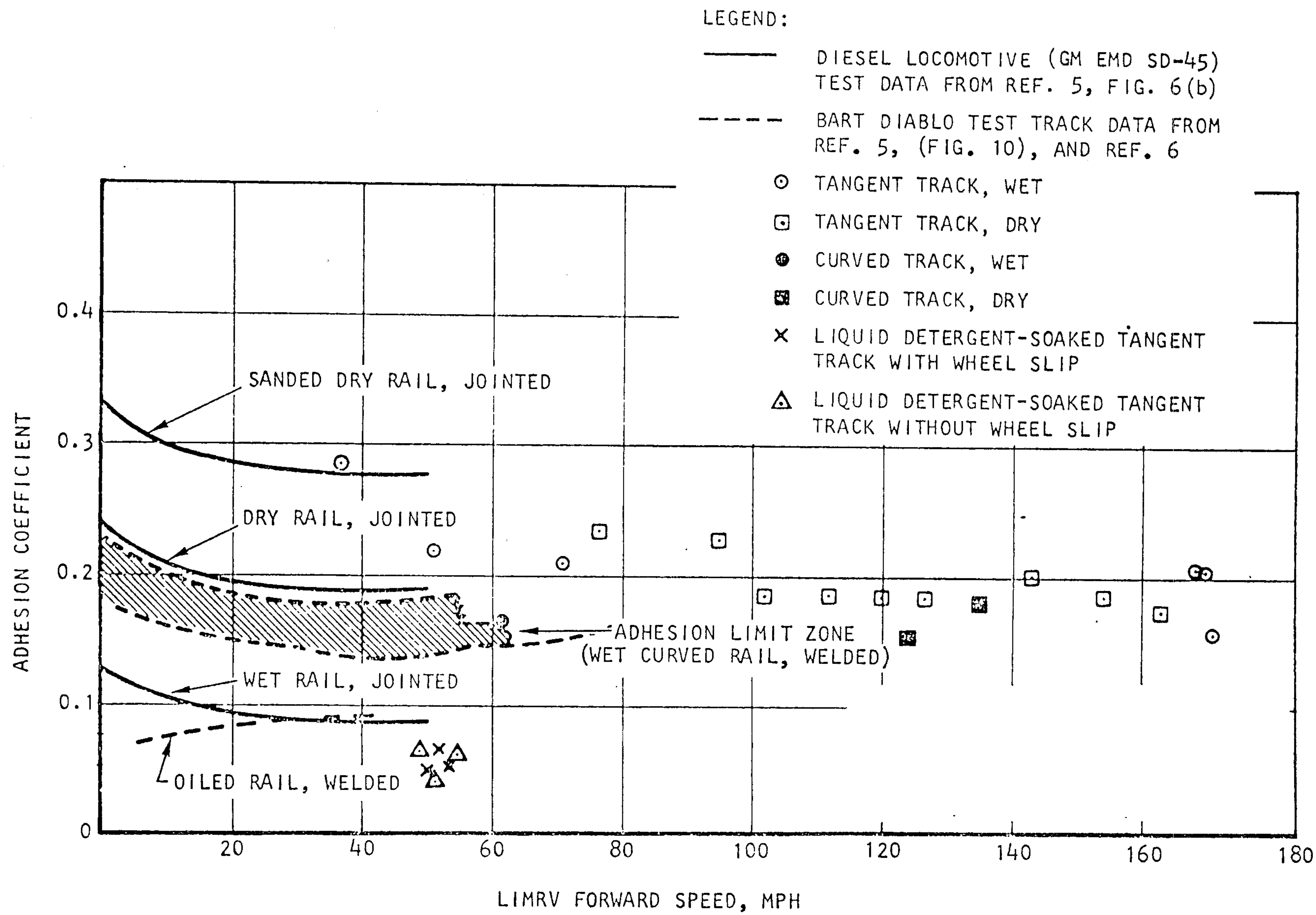


Figure 2-7. LIMRV Wheel/Rail Adhesion Data Measured at Axle 3

where v is the velocity in kph and μ_o is the coefficient of adhesion on normal dry track. If 0.218 is taken as the average μ_o and v is equal to 130 mph, μ is found to be approximately 0.16.

$$\mu = \mu_o \left[0.4 + \frac{0.6}{(1 + \frac{v}{25})}\right]$$

Assuming the same values for μ_o and v as before, μ is found to be 0.10.

Because of lack of information about adhesion on slightly dampened or heavily contaminated rail, it is difficult to analyze the worst-case situation. For example, values of 0.0 and 0.05 have been observed for adhesion on rails covered with leaves ground into a film. The approach used here will be to treat unusually low adhesion coefficients (below 0.05) as occuring infrequently enough to fall into the area of problems treatable by wheel-slide protection systems.

The use of the Kraft equation suggests that the values obtained by inspection of the graphs tend to be conservative. Using these values, a good approximation of the maximum deceleration rate possible without initiating wheel slip can be found by using the no-slip condition $\mu > a/g$ where a = deceleration of the train and g = acceleration due to gravity. The result for dry rail is a = 2.63 mphps and for wet rail a = 1.86 mphps.

An approximation for the shortest stopping distance possible with the wet rail deceleration rate can be found using $s = \frac{v^2}{2a}$ where s = stopping distance, v = initial velocity, and a = deceleration rate. For v = 130 mph, and a = 1.86 mphps, s = 6,663 feet.

The significance of the foregoing is that:

- Using present equipment and procedures, a certain amount of wheel-slip is likely to occur during wet rail conditions.
- From an adhesion standpoint, stopping within the proposed block distances on the upgraded NEC is well within the capabilities of currently used braking systems.

2.2 KINETIC ENERGY DISSIPATION

The other part of the braking problem is dissipating kinetic energy without causing excessive wear to the braking system or damaging the track. Since kinetic energy increases as the square of the velocity, trains traveling at 130 mph will have to absorb much more energy than presently-operating trains. As shown in Figure 1-5, Metroliners,which now operate at a top speed of 110 mph,face a 40 percent increase in the amount of energy absorbed, and Amcoaches,now running at a maximum of 95 mph,face a 90 percent increase.

Almost all of the braking systems presently in use transform kinetic energy into heat. This can seriously affect the system if the vehicle's velocity (which determines the amount of thermal energy produced) increases beyond a certain point. For example, the friction brake systems are limited by the amount of heat and the rate of its transfer into the wheel or disc. Wheels are especially susceptible to cracking caused by thermal stresses.

Although the problems are different, some non-friction brake systems are also constrained by thermal considerations. The resistor grids used as a heat sink for the dynamic brakes on Metroliners have burned out frequently in the past, thus impairing the operation of the braking system. However, they are currently being moved from underneath to the top of the car in order to improve cooling and therefore improve reliability.

Eddy-current rail-brakes provide another example of a thermally-limited system. Since the track is used as a heat sink, train headways that are too short may cause buckled rails (Section 4.1). Eddy-current wheel-brakes use both the wheel and the rail as heat sinks. Because the resulting distribution of heat is more even, this system is not expected to have any significant thermal problems. However, this system is not adhesion-limited.

Both retarder brake systems (hydraulic and air) are designed to avoid such problems. The action of the air retarder heats air, which is discharged after braking. The hydraulic retarder dissipates kinetic energy into a fluid which is cooled in an external radiator and recycled.

3.0 EXISTING SYSTEMS

3.1 GENERAL

The three systems presently used in the United States (alone or in some combination) are tread, disc, and dynamic brakes. Some advantages and disadvantages of these systems are listed in Table 3-1.

3.2 TREAD BRAKES

The limitations of tread brakes are associated with the thermal stress defects that may be produced in the wheel as a result of the heat generated by friction between the tread and the brake shoe during braking.

There are at least three types of tread defect caused by thermal stresses:

- Thermal cracks.
- Sudden-type thermal cracks.
- Fatigue-type thermal cracks.

Thermal checks, commonly known as crazing, are believed to be produced by tensile stresses that result from volumetric changes associated with microstructural transformations. In other words, the compounds created by high temperatures (such as martensite or pearlite) have a smaller specific volume than the surrounding carbon steel. This results in tensile stresses. Indications are that thermal checks do not propagate and are therefore, not believed to be dangerous except that they may serve as a starting point for other types of cracking. Worn away by the action at the brake shoe against the tread, they are continually replaced by new thermal checks.

Sudden-type thermal cracks are considered the most hazardous rim defects because of their large size when first formed and their

TABLE 3-1

EXISTING BRAKING SYSTEMS

	ADVANTAGES	DISADVANTAGES
TREAD	CLEANS WHEEL TREAD MAINTAINS TREAD TAPER CAST IRON SHOES CAN INCREASE ADHESION	CAN CAUSE THERMAL CRACKS IN WHEELS POSSIBILITY OF UNDESIRABLE WHEEL WEAR PATTERNS
DISC	DOES NOT HEAT WHEEL BETTER HEAT TRANSFER CAPABLE OF ABSORBING MORE ENERGY THAN WHEELS	DOES NOT CLEAN WHEELS NEED SPACE TO MOUNT DISC AND CYLINDER DOES NOTHING TO REDUCE SPALLING--MAY COME LOOSE FROM MOUNTING
DYNAMIC	WILL NOT LOCK WHEELS SIZE OF HEAT SINK IS ADJUSTABLE REQUIRES NO ACTUATING POWER	APPLICABLE ONLY TO TRACTION UNITS INEFFECTIVE AT LOW SPEEDS (CANNOT BE USED AS AN EMERGENCY BRAKE)
BLENDED	PROVIDES REDUNDANCY TAKES LOAD OFF SINGLE SYSTEM	INCREASES COMPLEXITY AND THEREFORE POSSIBILITY OF BREAKDOWN COMPLICATES MAINTENANCE SCHEDULE

tendency to propagate rapidly. It is believed that they are a result of tensile stresses induced by the thermal expansion of the rim during braking (at which time a large amount of energy is dissipated).

Fatigue-type thermal cracks propagate as deeply as sudden-type cracks. However, they form much more slowly. They are supposedly caused by cyclical thermal stresses developed by alternate heating and cooling. These three types of tread defects are illustrated in Figure 3-1.

Both sudden-type and fatigue-type cracks can become large enough to render the wheel unsafe and under certain conditions can cause explosive wheel failure. The frequency of occurrence and the severity of these cracks depend on a number of factors. Two of these factors are the rate of temperature change and the temperature gradient. This leads to an obvious limitation on tread brakes: they cannot be required to assume so large a part of the braking effort that wheel damage occurs.

Composition brake shoes can assume more braking effort than cast iron shoes because the heat they produce penetrates the rim more evenly and to a greater degree. This decreases the temperature and reduces the likelihood of hazardous-thermal-crack formation. However, the potential for damaged wheels still exists if the braking effort assumed by the tread brake becomes large enough.

There have been reports of tread brakes reducing the incidence of shelling.* However, no data has been found to substantiate this claim. As mentioned in Section 2.1.2, tread brakes incorporating brake shoes containing iron particles have the advantage

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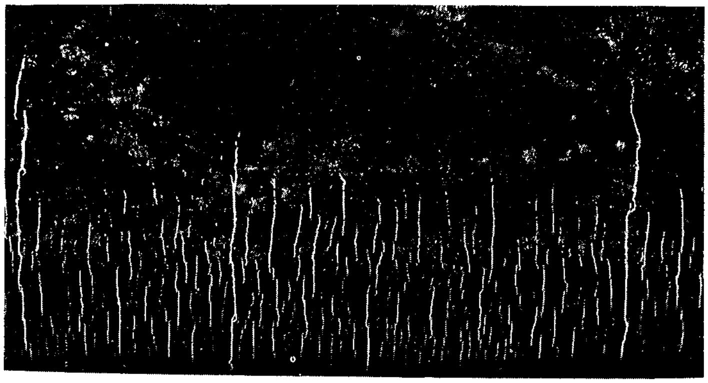
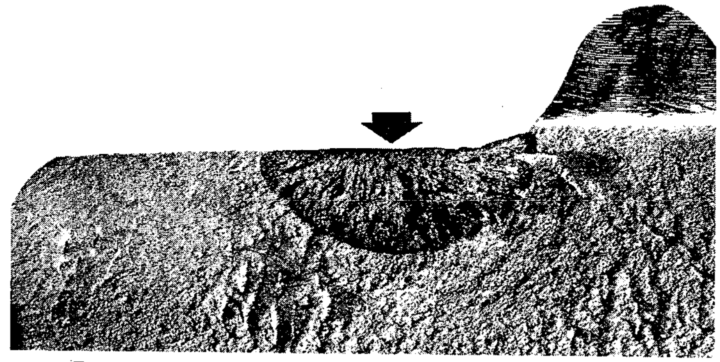
* Shelling is the expulsion of tread material resulting from cracks caused by rolling (not braking) stresses.



Front-rim face

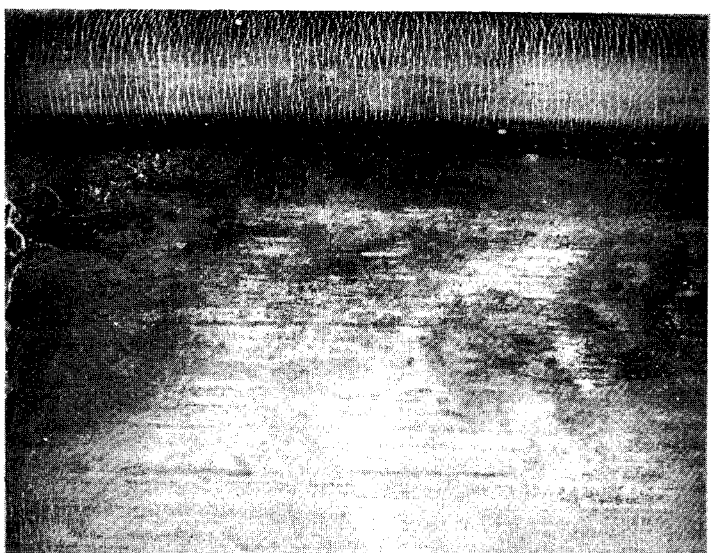
Appearance of fatigue-type thermal cracks in tread surface of wheel returned from service. Cracks revealed by fluorescent magnetic-particle inspection. About 2/3 actual size.

(Ref.9)



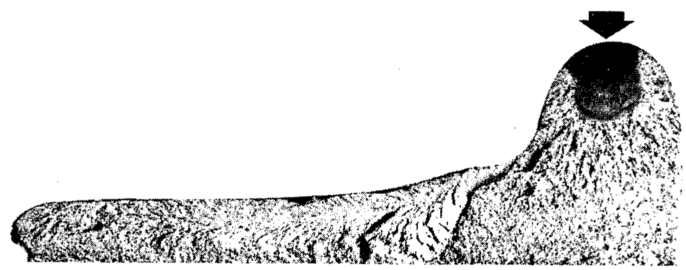
Fracture surface of sudden-type thermal crack (indicated by arrows) formed in the Class CR wheel (0.73 pct C). This wheel was subjected to one high-energy dissipation stop from 115 mph under simulated emergency conditions. About 2/3 actual size.

(Ref.10)



Thermal checks in tread and flange of wheel.

(Ref. 11)



(a) Fatigue-type thermal crack that nucleated near tip of flange. About 2/3 actual size.

(b) Fatigue-type thermal crack that nucleated near front edge of tread.

Fracture surface of fatigue-type thermal cracks in rims of wheels returned from service. Fracture surfaces were exposed by breaking sections of rim containing cracks.

(Ref. 9)

Figure 3-1. Fatigue-Type Thermal Tread Defects of enhancing the available adhesion by 20 percent. Also, they help to prevent spalled wheels (Section 3.2). Examples of spalling and shelling are shown in Figure 3-2.

3.3 DISC BRAKES

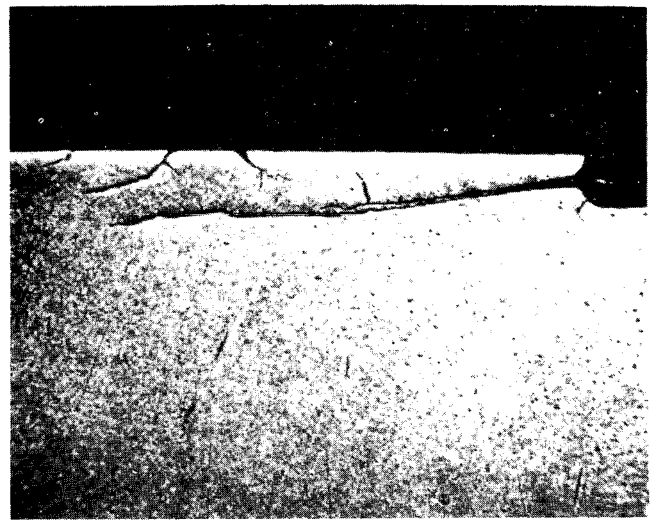
Disc brakes were introduced in an attempt to alleviate the wheel problems associated with thermal stresses that were increasing along with increased speeds. Cracking is not as serious a problem in discs as it is in wheels because the thermal diffusivity of cast iron is higher than that of carbon steel. This means that discs are able to absorb heat more efficiently than wheels.

Heat absorption in discs starts to become a problem in the 130-to 135-mph range assuming two discs per axle on cars of 120,000 pounds or less. The probability of thermal cracking occurring at these and higher speeds does not depend significantly on the braking rate, i.e., the rate at which heat is absorbed by the disc.

One of the problems associated with the use of disc brakes is the increased tendency for spalling of the wheels to occur. Spalling is a tread defect caused by wheels slipping, sliding or chain-sliding.* The resultant rapid heating and cooling of spots on the wheel tread produces patches of metallurgically changed rim. Small transverse cracks develop in these areas and lead to portions of the tread surface falling away. Evidence gained through testing indicates that this is much less of a problem in tread-braked wheels, possibly because of the polishing action of the brake shoe against the wheel rim.

Another problem related to the use of disc brakes is the difficulty in developing secure mounts for the discs as they grow in size and weight. Truck dynamics may also be adversely affected

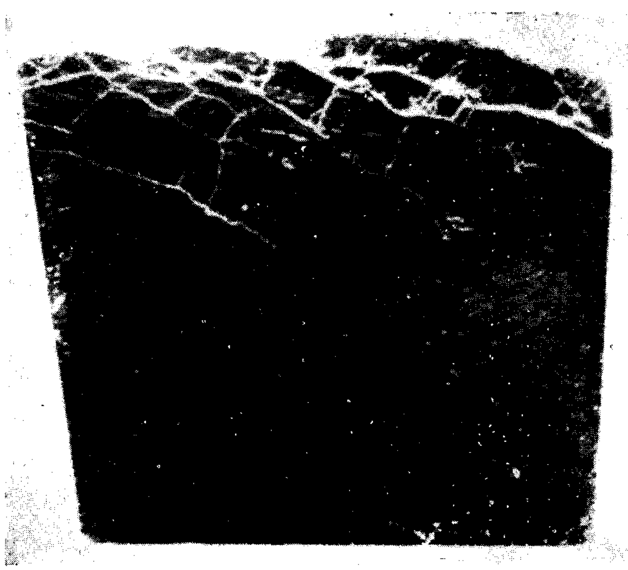
*Caused by the on-again, off-again action of a wheel slide detector in operation.



Cross section of wheel tread with spalling.
Magnification 15X.

Shelled tread,
medium stage.

Shelled tread,
full development.



Cross section of wheel tread with deep shelling.

Spalling condition.

Figure 3-2. Shelling and Spalling of Wheel Treads by these increases.

3.4 DYNAMIC BRAKING

There are two types of dynamic brakes: rheostatic and regenerative. Both systems are similar in that they convert kinetic energy to electricity, but there the similarity ends. Rheostatic brakes convert the electricity into heat by means of resistors mounted on the vehicle. Regenerative brakes, on the other hand, channel the electricity into the power supply.

Electromechanical energy conversion involves the interchange of energy between an electrical system and a mechanical system. The process is essentially reversible. When the energy conversion takes place from electrical to mechanical form, the device is called a motor. When mechanical energy is converted to electrical energy, the device is called a generator.

When dynamic braking is used on a motorized unit, the motor is converted into a generator. The rotating member (rotor) is driven by the axle, causing an induced voltage to appear across the stationary member (stator). When a load is applied to the stator, a current flows and delivers electrical power to the load. The current flow through the stator interacts with the magnetic field to produce a reaction torque opposing the applied torque originating in the axle.

3.5 WHEEL SLIDE PROTECTION SYSTEMS*

Wheel slide protection is needed to obtain shortest possible stopping distance despite variations in adhesion. There are two generally accepted objectives in the use of wheel slide protection:

- To avoid wheel damage.
- To improve braking efficiency.

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*Current Amtrak policy is to provide these systems on disc-braked coaches only.

The wheel slide protection system can be separated into four areas:

- Signal generation.
- Detection of incipient slides.
- Detection of the condition for restoration of braking.
- Implementation of brake control.

Wheel slide protection is used to eliminate wheel lock during braking by means of brake release signals. Slide protection affects only the air system; dynamic braking is slide protected because brake effort falls to zero if the axle ceases to turn.

A wheel slip protection system that is in wide use on both American and European passenger coaches is the Western Air Brake Company's (WABCO) E-5 Decelostat equipment. The E-5 provides wheel slip control for a four-axle, two-truck, non-propelled car. Braking effort correction is on a per truck basis. The equipment is operational for both service and emergency braking.

A functional block diagram of the E-5 Decelostat controller is shown in Figure 3-3. The velocities of each axle are measured by unloaded, journal-mounted, magnetic pick-ups. By sensing the rotation of special axle-mounted gears, the pick-ups generate a signal having a frequency proportional to speed. This signal is then converted to an analog signal proportional to axle velocity. The analog, axle-velocity signals are differentiated to provide acceleration or deceleration signals which are proportional to rate-of-change of axle velocities.

3.5.1 RATE REDUCTION

Whenever a deceleration voltage signal exceeds a present value equivalent to 8 mphps,* the decelostat valve for the affected

*Current Amtrack specification.

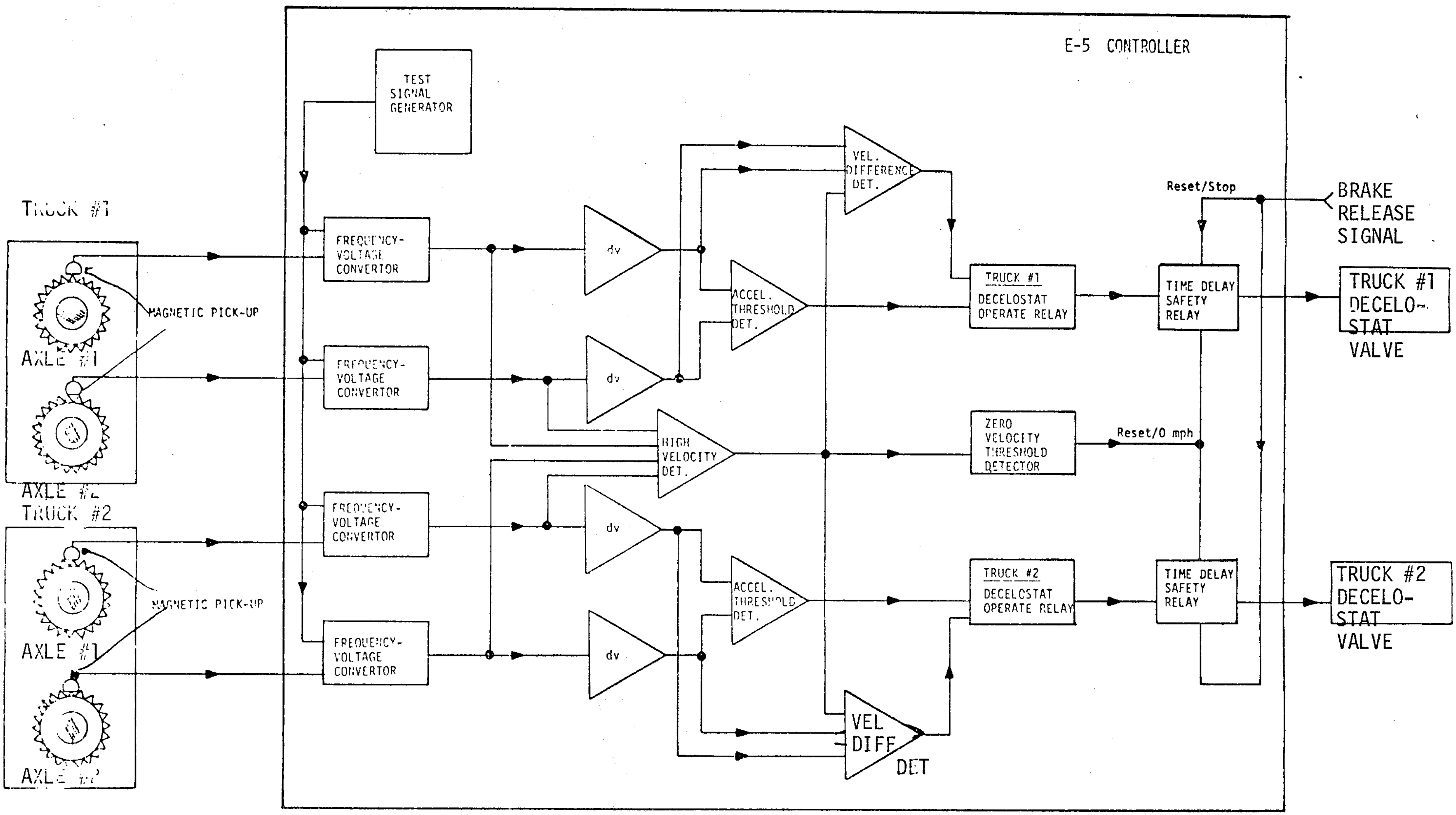

Figure 3-3. Functional Block Diagram - WABCO Decelostat Controller

truck is energized, causing a release of braking effort on that truck.

If the affected axle accelerates quickly because of an increase in adhesion, the acceleration signal voltage will cause a termination of the decelostat valve energization permitting a jerk-rate-limited reapplication of braking effort. If the affected axle does not accelerate quickly enough to produce an acceleration voltage of sufficient magnitude to terminate energization of the decelostat valve within a given time limit, energization will be terminated by a safety relay. Once the safety relay terminates the energization of the decelostat valve, the valve is inhibited until a stop is made or brake release is initiated.

3.5.2 VELOCITY DIFFERENTIAL DETECTION

The analog velocity signals are compared and if a speed difference exceeds a pre-established value, the decelostat valve on the truck experiencing a low axle-velocity is energized. When adhesion is regained and causes the difference in velocity voltages to be reduced, termination of decelostat-valve-energization permits a jerk-rate-limited reapplication of braking effort.

The pre-established values for velocity difference detection is a function of car velocity. The values are approximately 3 mph* near zero velocity and 14 mph near 120 mph.

3.6 BRAKE SYSTEMS--U.S. TRAINS

Inter-city passenger trains in the United States, except for Metroliners, seldom use in-train dynamic braking. All trains in use in the United States for inter-city passenger service depend on adhesion for braking. Tests have shown that adhesion decreases with increased speed. The use of electro-pneumatic brake systems

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*Current Amtrak specification.

with wheel-slip protection has reduced stopping distances by reducing lag time in the application and release of brakes. The following is a summary of the brake systems used in various passanger cars:

- Metroliner - used on Northeast Corridor only - operational speeds of 105 mph today - system is capable of 160 mph.

  - Type brakes - electro-pneumatic tread brakes with automatic dynamic brake blending.

  - Full-service rates - Figure 3-4.

  - Emergency rates - Figure 3-4.

  - Wheel slip/slide detection - detects slip/slides of approximately 5-mph difference in wheel speeds or deceleration of less than 8 mphps - correction is only on trucks experiencing slip/slide until same is corrected.

  - Problems - thermal problems caused by friction braking require wheels to be magnafluxed every 30 days.

- Amcoach - used on all Amtrak routes.

  - Type brakes - electro-pneumatic disc brakes.

  - Brake rates - full-service - 1.24 mphps at 120 mph to 2.00 mphps at 70 mph and a steady 2.00 mphps to stop. Emergency - no less than 2.50 mphps at speeds below 70 mph - above 70 mph within limits of wheel/rail adhesion.

  - Wheel slip detection - detects random and synchronous slides starting at greater than 5 mph and tapering to 14 mph at 120 mph - controlled so that maximum change in deceleration during initial braking does not exceed 1.5 mphps under normal conditions.

  - Problems - flat spots caused by decelostats which result in wheel slide at service speeds - causes wheels to be rejected at inspection.

- Amtrak Bi-Level Coach - used on Amtrak Western routes.

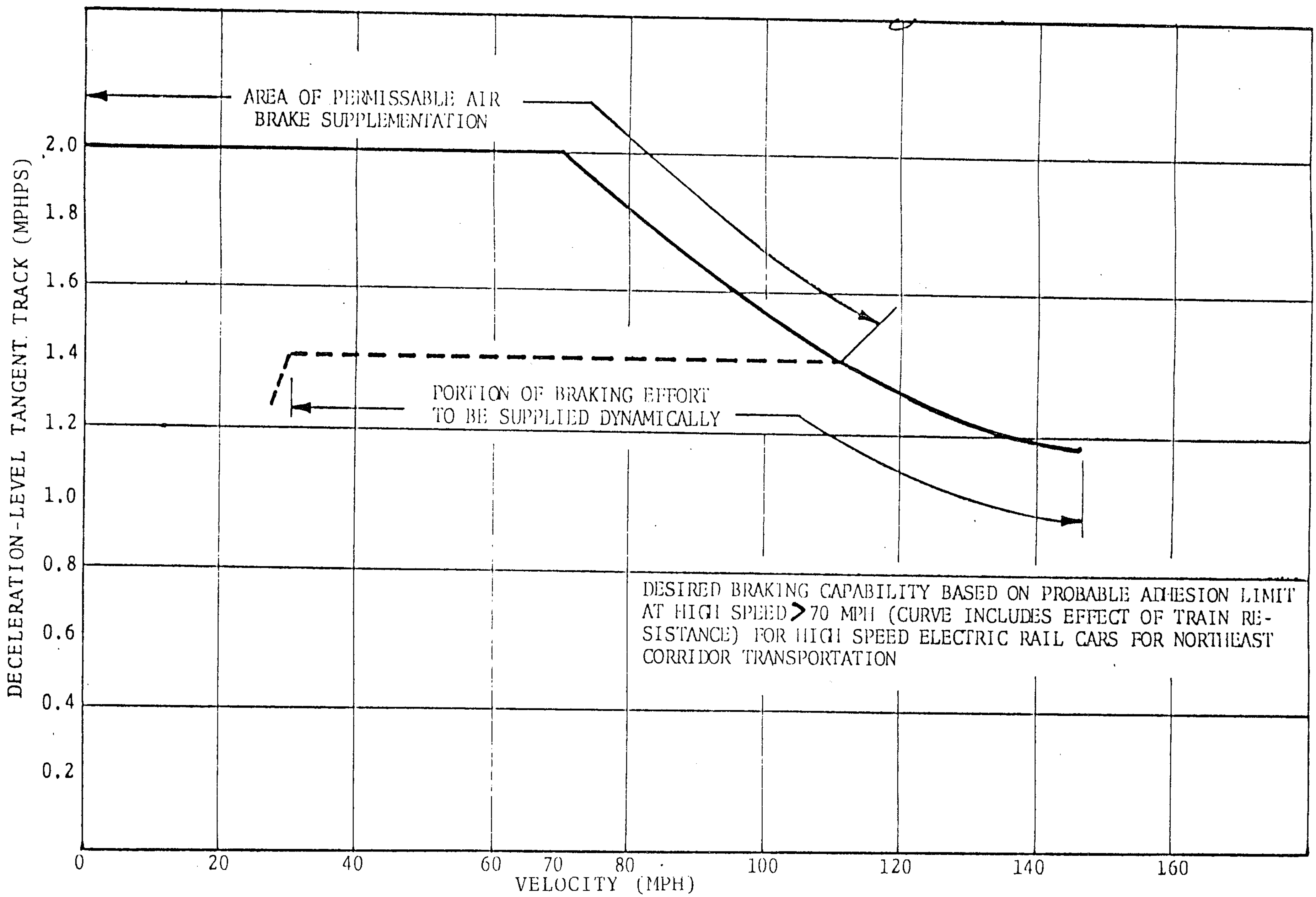

Figure 3-4. Metroliner Braking Rates

- Type brakes - disc-electro-pneumatic with or without blended dynamic braking of locomotive.

- Brake rates - full-service (air only) - Figure 3-5. Emergency - Figure 3-5.

- Wheel slip/slide detection - detects slides of at least 5 mph - difference in wheel speeds and deceleration rates equal to or greater than 8 mphps.

- Problems - flat spots caused by wheel slide at high speeds - causes wheels to be rejected at inspection.

- Amtrak Turboliners - maximum speed of 110 mph.

  - Type brakes - on-tread and disc-electro-pneumatic in combination with hydrodynamic braking.

  - Braking rates - at 90 mph full-service braking can stop a unit train (622,000 pounds) within 3,250 feet - acceptance tests indicated that rates varied from 1.14 mphps to 2.79 mphps depending on location, speed, brake application and mode.

  - Automatic slip/slide detection - detects random and synchronous slides greater than 5 mph and tapering to 14 mph at 120 mph - controlled so that maximum change in deceleration during initial braking does not exceed 1.5 mphps under normal conditions.

3.7 BRAKE SYSTEMS--FOREIGN TRAINS

The following is a summary of the brake systems used on various foreign trains:

- French - RTG Turbotrains designed for 125-mph service.

  - Type brakes - hydraulic retarder on powered trucks and magnetic track friction brake for emergency use with electro-pneumatic discs (magnetic brakes not used in U.S.).

  - Brake rates - detailed information was not available.

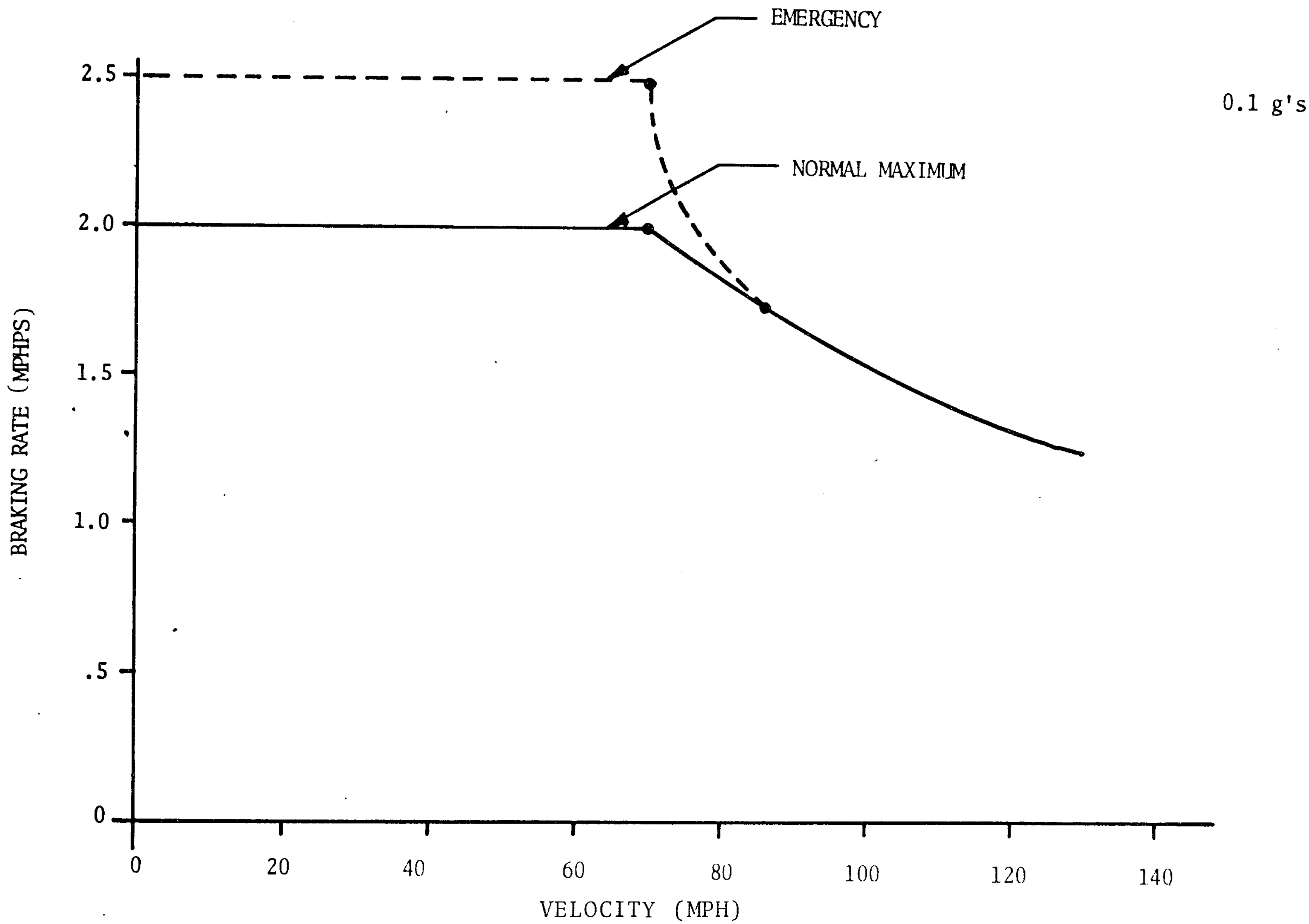

Figure 3-5. Amtrak Bi-Level Coach - Braking Rates

- French TGV - electric high-speed passenger trains.
  - Type brakes - electro-pneumatic disc with blended dynamic braking (four discs per axle).
  - Braking rate - detailed information was not available.
  - Automatic slide protection.

3.8 ADVANCED PASSENGER TRAIN (APT)

The APT was developed by the British Railway to improve the performance of existing intercity passenger trains by:

- Increasing maximum speed by 50 percent.
- Increasing speed in curves by 40 percent.
- Running on existing track.
- Using existing block signaling.
- Maintaining standards of passenger comfort.
- Being more energy efficient.
- Causing low environmental noise.
- Providing low cost per seat-mile.

The prototype APT-P is now in service and has undergone extensive testing. With a vehicle body that tilts up to 9 degrees, electric-powered traction, and an advanced anti-tilt pantograph design, the APT has met or exceeded its design goals.

The APT uses a hydrokinetic brake (Section 4.3) with auxilliary on-tread braking on its unpowered coaches. Each powered axle is hydrokinetically braked; the brake is fitted to the body-mounted gear box in the mechanical drive for the axle.

The braking system was designed to stop the train from 155 mph (250 km/h) in the existing signal block of 6,690 feet (2,040 m) including a 12.5-percent margin of safety. The tread brakes are automatically blended in at speeds below 50mph (80 km/h). The average deceleration rate is 3.05 mphps.

4.0 ADVANCED SYSTEMS

4.1 GENERAL

Higher speeds, wheel loads, and deceleration rates have stimulated research in new types of braking. This work has produced four new non-friction braking systems: eddy-current rail brakes, eddy-current wheel brakes, air retarders and hydraulic retarders. All are described and evaluated in the following sections.

4.2 EDDY-CURRENT RAIL BRAKES

The eddy-current rail brake operates by transforming the kinetic energy of the train into heat energy via a varying magnetic field. Basically, it consists of a row of electromagnets suspended between the wheels of a truck (Figure 4-1). Braking occurs when a current flows through the coils that make up the electromagnets (Figure 4-2). The resulting magnetic field passes through the top of the railhead. Since the train is moving, the magnetic field through any given section of railhead is always changing. The changing magnetic field produces an electromotive force which by Faraday's Law causes eddy currents to flow. The current flow is transformed into heat in the railhead through the Joule effect.

An interesting feature of this system is that it does not use an actual force to slow the train. What is commonly referred to as eddy-current braking force is actually the instantaneous rate of change of kinetic energy with distance. This leads to one of the most desirable characteristics of the eddy-current rail brake. Since it does not act through the wheel-rail interface, it is completely adhesion independent. This technology has been extensively tested in Europe and Japan. Prototype systems built in France and Germany are described in the following sections.

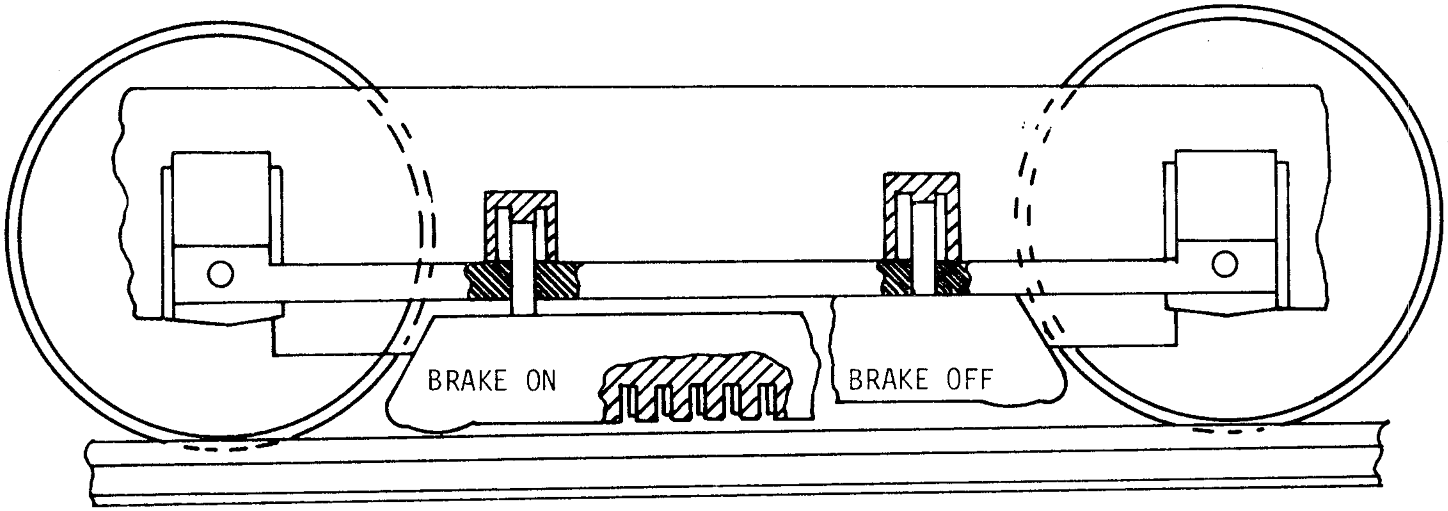

Figure 4-1. Baermann Eddy-Current-Track-Brake

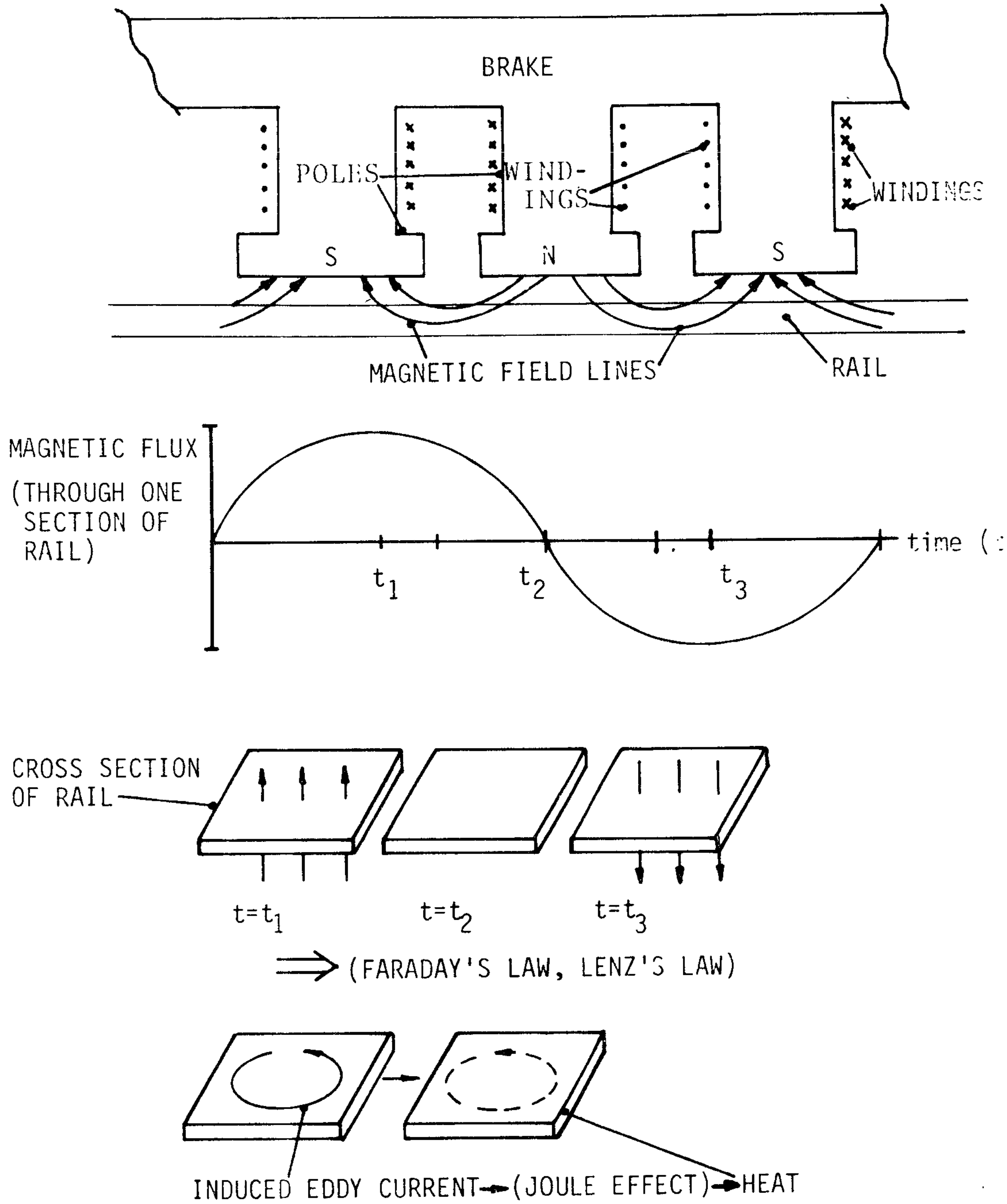

Figure 4-2. Rail Brake Physics

4.2.1 PROTOTYPE SYSTEM

- MTE - linear brake developed for the French National Railroad and installed on the Z7001 (a test car for new design of powered trucks) which has been tested at speeds up to 186 mph (300 km/h).

  - Weight - 1,322 pounds per brake including mounting and control hardware.

  - Length - 6.3 feet (1.9 m/h) per brake.

  - Deceleration rates - dependent upon airgap (distance between linear brake and railhead) and excitation current.

    - During test of the Z7001, an average deceleration rate of 2.0 mphps was obtained from 186 mph by sole use of the linear brakes with a 7-mm gap and an excitation of 650 amperes per brake.

  - Consistently achieved stopping distances using the MTE linear brakes on the Z7001 were:

    186 mph within 11,480 feet
    137 mph within 6,560 feet
    125 mph within 5,240 feet
    100 mph within 3,600 feet

  - Power requirements - during the Z7001 test, power for the linear rail brakes was supplied by the drive motors of the motorized truck. During braking, current that was supplied to the motors was regulated to maintain a constant voltage to the brakes down to a speed of 60 mph, after which the voltage decreased as a function of speed.

Battery fail-safe-backup for the current control system was provided. Power consumption per brake was equal to 50 kilowatts (77 volts at 700 amperes).

- BSI - West German system developed by Mr. Baerman of the West German Physics Laboratory has been tested on a dynamometer.

  - Weight - unknown.
  - Length - 4.26 feet (1.3 m/h).
  - Deceleration rates - depends on air gap and excitation. Current claims of 1,800 pounds (8 KN) of braking power per brake with a gap of 7 millimeters at 20 kilowatts.
  - Power requirements - using a patented coil, the Baerman system claims 158 percent of the braking power of the German MFE System for the same kilowatts of excitatation. Specifications claim 20 kilowatts for an equivalent of 2,000 pounds of braking power.

- Japanese National Railway. Test information was not available. Tests were conducted on a smaller version of the German system at lower power rates. Brake rates of 860 pounds of braking force from 20-100 mph were produced.

4.3 EDDY-CURRENT WHEEL BRAKE

The eddy-current wheel brake works on the same principle as the linear rail brake, but generates eddy currents in the wheel instead of the rail. In this configuration, a coil is suspended around the wheel (Figure 4-3). When a current passes through the coil, a magnetic field is generated creating a magnetic flux perpendicular to the plane of the truck. When the train is in motion, the turning of the wheel causes the flux through

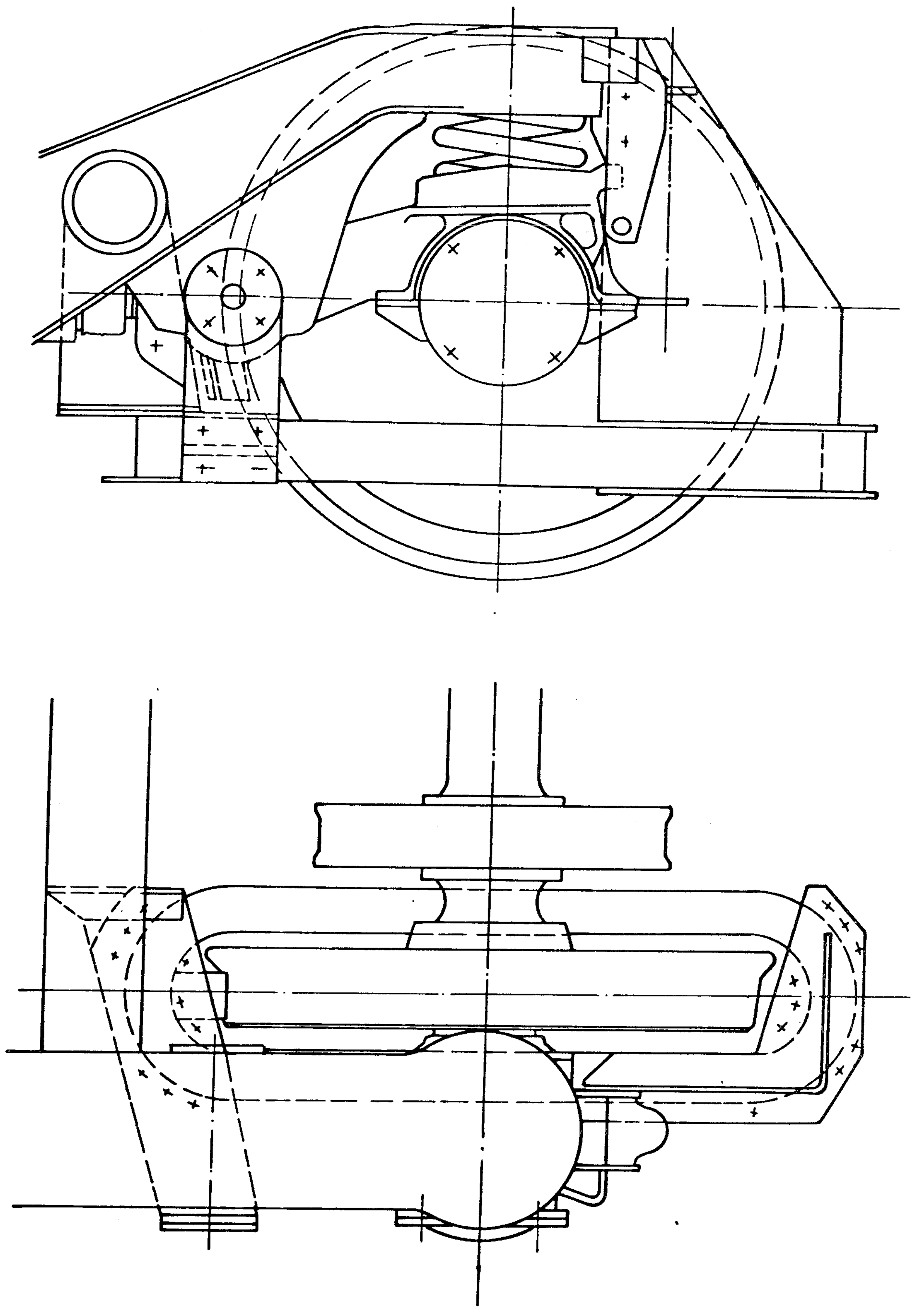

Figure 4-3. Physical Configuration of SENF Wheel Brake

any given radial cross section of the wheel to change. By Faraday's Law, this change in flux induces an electromotive force in the wheel that gives rise to eddy currents (Figure 4-4).

The flow of these currents causes electrical energy to be transformed into thermal energy through the Joule effect. The kinetic energy of the wheel decreases in an attempt to maintain the eddy currents. To be more specific, the kinetic energy of the wheels is transformed to electrical energy which is then transformed to heat.

An important difference between the wheel and rail brake is that the former is not adhesion independent. It is claimed that in cases of insufficient adhesion, an increased amount of flux reaches the rail. This decreases the dependence of the wheel brakes on adhesion, since it then operates in the same manner as the rail brake. However, this claim has not been substantiated by tests.

4.3.1 PROTOTYPE SYSTEMS

Two systems have been developed: one by the French National Railway (SENF) and the other by BSI, a German manufacturer.

- SENF - this system has only been laboratory tested using a 1/5-scale model.
  - Type - a coil surrounds the lower part of the wheel and contains no cores or yokes. A current is passed through the coil and generates a magnetic field.
  - Weight - 110 pounds per coil.
  - Braking rate - 5,620 pounds of braking force available for 80,000 AT (100 coil turns at 800 amperes) and is constant between 186 mph

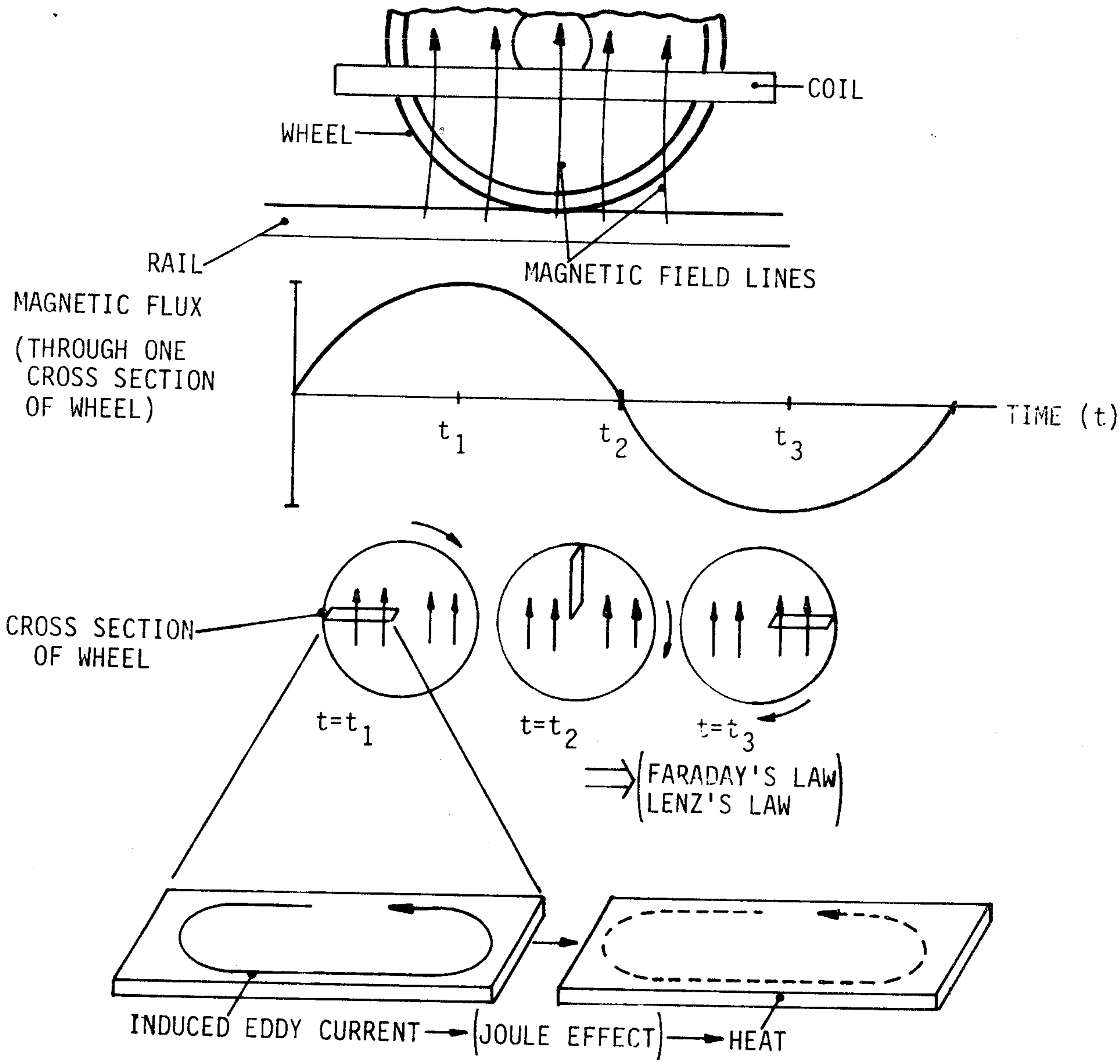

Figure 4-4. Wheel Brake Physics

and 20 mph. Average deceleration (computed) 2 mphps from 160 mph to 20 mph with a wheel load of 16.9 T/axle.

- Power - 800 amperes per coil (100 turns/800 amperes)
- Advantages
  - No minimum clearance between rails and coils required.
  - Lightweight.
  - Release of heat into rail decreased, and possibly controllable through use of shielding.
  - Impossible to lock wheels.

- Problem areas
  - Power requirements - 32,000 amperes per truck (4 wheels).
  - In the case of insufficient adhesion, is other adequate braking power available as the system transforms into a linear brake operation?
  - Coil heating/wheel heating. Scale model testing indicates that the wheel behaves as an excellent radiator when it is heated in this way and that the temperatures which it reaches(even outside of any braking on the part of the wheel) do not appear troublesome for operation. Also, the design of the coil tested proved inadequate. However, the design can be improved by using oxidized aluminum. The question of whether or not adequate braking power can be supplied in the case of insufficient adhesion does not appear to have been fully investigated.

- BSI - linear wheel brake - other than a proposed adaptation drawing, detailed specifications on the BSI wheel brake system were not available. It appears that the BSI system uses the same principle as the rail head brake, but uses the top of the wheel as a reaction member with a curved electromagnetic brake located on the inside of the wheel.

- Brake rates - for an axle load of 38,200 pounds, a retarding force of 2,250 pounds from 125 to 55 mph was specified for two eddy-current wheel brakes.

- Weight - not available.

- Power - not available.

- Advantages/disadvantages - requires more detailed specifications and test data.

4.4 AIR RETARDER

Basically, the air retarder consist of a pair of gears, driven by the axle, that pump air into a reservoir (Figures 4-5 and 4-6). During unloaded (non-braking operation, the reservoir is open to the atmosphere and free flow conditions exist throughout the system. The gears turn freely in this mode and no torque develops on the axle.

Braking occurs when the reservoir is sealed except for its inlet nozzle. This requires that the gears do work in order to force the incoming air into the reservoir. The resultant resisting torque that develops on the axle is used to brake the train.

In other words, the train slows down because its kinetic energy is used to compress air.

4.5 HYDRAULIC RETARDER

The hydraulic retarder (also known as the hydrodynamic or hydrokinetic brake) consists of a set of vanes (the rotor) which rotate inside a geometrically similar housing (the stator-Figure 4-7). During unloaded operation, the vanes which are driven by the axle revolve freely inside an empty stator.

Braking occurs when pressurized air forces a fluid into the

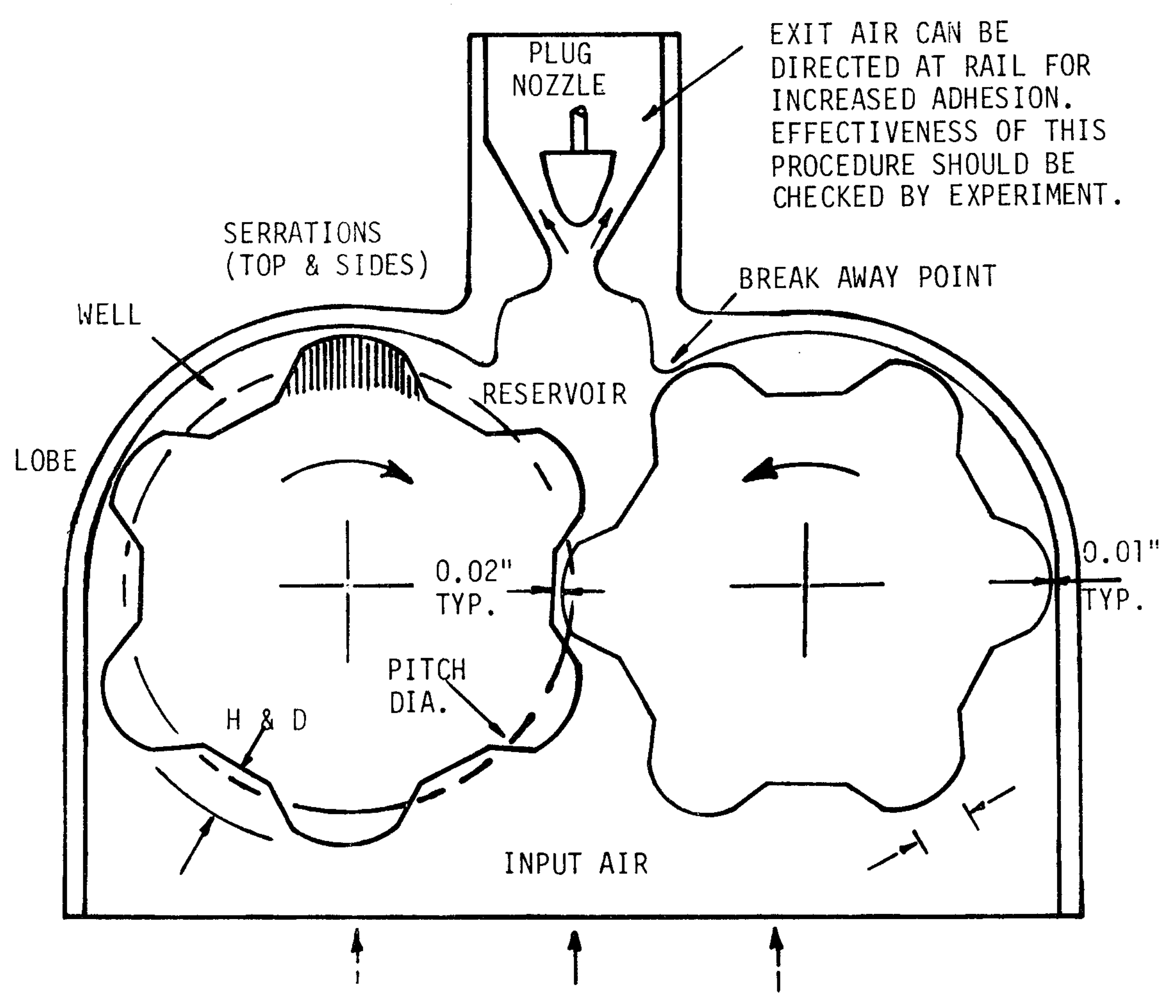

Figure 4-5. Air Retarder Configuration

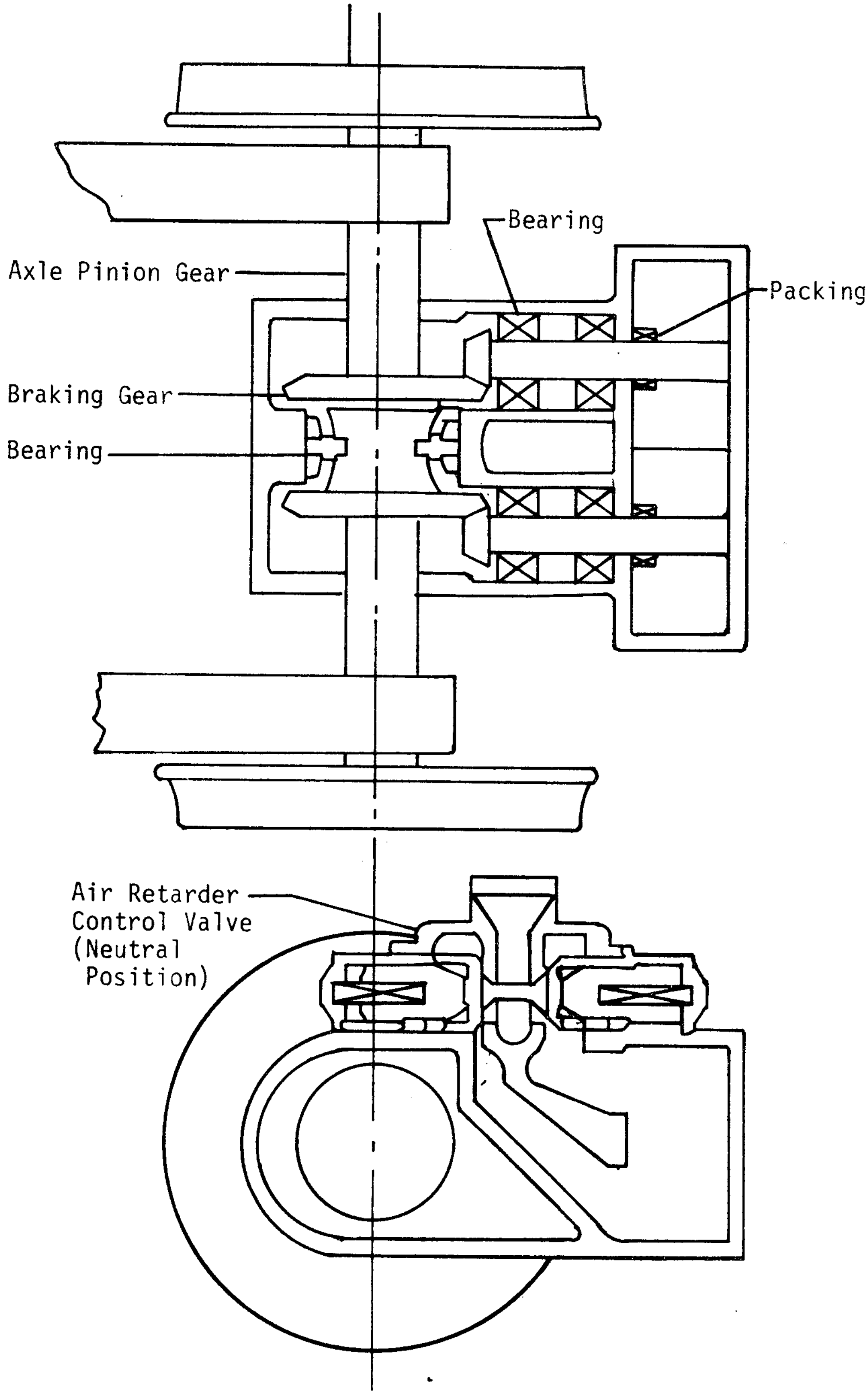

Figure 4-6. Installation of Air Retarder

spaces between the rotor and stator. The temperature of the fluid rises because of the resistance it offers to the motion of the rotor. The train's kinetic energy is thus transformed to thermal energy causing the train to lose speed.

Cooling is achieved by the pumping action of the brake which circulates fluid between the brake and the reservoir (which acts as a heat sink).

4.6 DISCUSSION

The evaluation of the eddy-current rail brake indicates that several serious problems exist. For efficient operation, the gap between the brake and the rail must be 7 millimeters (0.275 in). Increasing the gap causes the braking power to fall off very quickly, as illustrated by Figure 4-8. A brake that met the AAR standard of 2.5* inches minimum clearance would be useless because of the braking power that would be lost.

The possibility of rail buckling is another difficulty. The rail temperature rise and the corresponding buckling hazard depends on the headway between trains. Calculations show (see Appendix B) that 15-minute and 30-minute headways cause unacceptable risks, and 60-minute headways are, at best, marginal.

At first, the eddy-current wheel brake appears much more promising, as it was designed to eliminate both of the above problems. However, development of the brake was discontinued after a series of bench tests, presumably because 8000 amperes per wheel would be required for a full-scale prototype.

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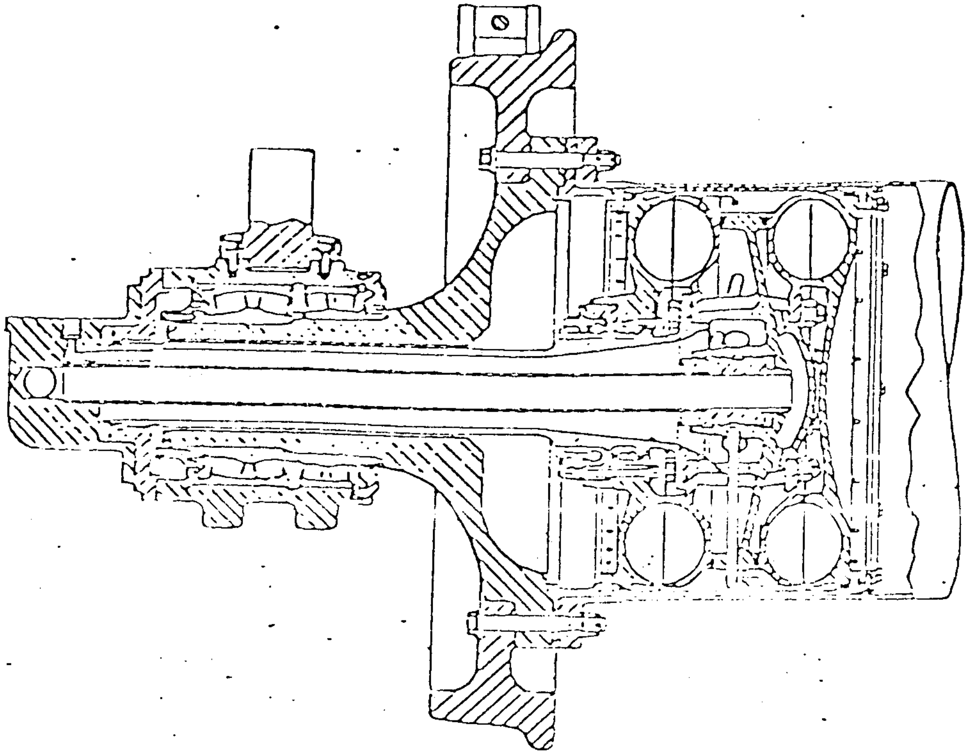
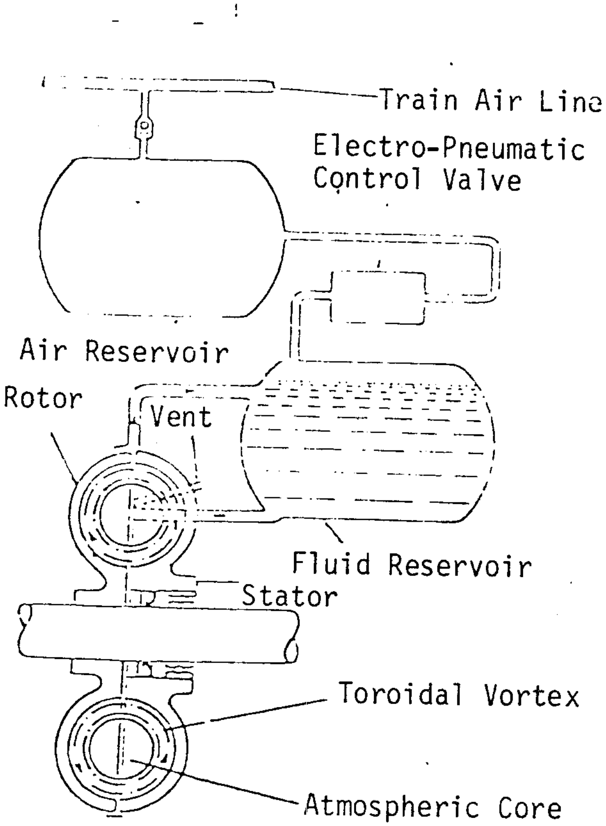
* This standard applies to all classes of track. See AAR Standard No. C-84A-1965.



ABOVE: A large-diameter thin-walled axle bolted to the wheels contains rotor and motor turbines which provide most of the braking torque for the APT: the stator runs in bearings inside the axle, the tube carrying the incoming fluid under pressure acting as a torque reaction member



RIGHT: The hydrokinetic brake is applied by admitting air from the train pipe into the fluid reservoir through an e.p. valve.

Figure 4-7. Hydrokinetic Brake

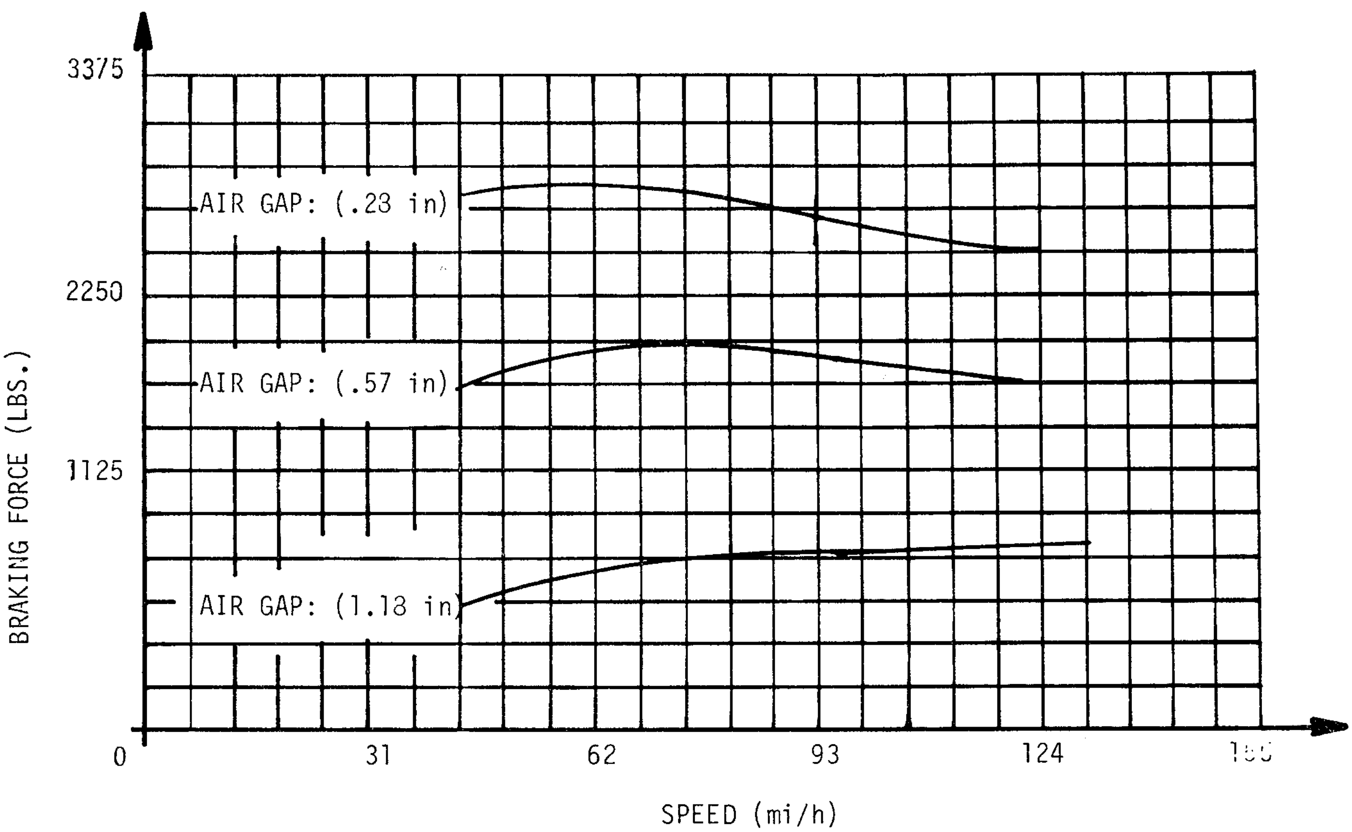


Figure 4-8. Effects of Varying Air Gap-Linear Rail Brake (SNCF)

A working model of the air retarder was built and bench tested at Calspan Inc. in Buffalo, NY in 1973. During these tests, this device exceeded its low-speed braking rate target (2.0 mphps) by about 75 percent which indicated that braking rate targets would be exceeded at all speeds. It also produced a great deal of noise because of its compressor-like action -- 130 dBA when applied at a speed of 50 mph. This was expected, and plans were made to attenuate the noise for the next phase of testing. Because of a decision to discontinue the program, however, no further testing was performed.

The hydraulic retarder was developed by British Rail for the APT because of its ability to dissipate large amounts of energy and its low mass. Full-scale tests have shown the brake to be free from wear, and have demonstrated its ability to slow the APT from speeds above 125 mph at a rate of 1.4 mphps.

Because it was designed to operate within the axle, the British design is unusable on present domestic trucks. The brake has been used outside of the axle by inserting it in the drive train of motorized units.

5.0 CONCLUSIONS AND RECOMMENDATIONS

On the basis of available data, the following conclusions may be drawn:

- The braking system now in use on Amfleet cars is sufficient to meet the demands of 130-mph service from a thermal standpoint regardless of the block distances involved. An analysis of the adhesion problem indicates that under certain conditions, slipping will occur. There are three possible ways of dealing with this problem:
  - Adding tread brakes, which would increase adhesion and reduce spalling.
  - Improving the effectiveness of wheel-slide protection systems. This would also reduce spalling.
  - Start quantifying the amount and the cost of wheel wear that results from sliding and spalling. Also investigate the possibility of relating the increase in ride quality resulting from the use of an improved braking system to a possible increase in revenue due to greater passenger satisfaction with the quality of the ride. This information could then be used to determine whether or not the expense involved in adding tread brakes and/or improving wheel-slide protection systems is justified.

- The braking system now in use on Metroliners will be suitable for 130-mph service from both the thermal and the adhesion standpoints, if the reliability and effectiveness of the dynamic brakes are improved to the point where tread brakes are able to assume only the amount of braking effort they were designed for. Otherwise, wheel damage resulting from thermal stresses and sliding will, in all probability, reach unacceptable levels.

- Because of problems related to clearances between the brake and the track, excessive power requirements, rail-buckling hazards, increased unsprung mass, eddy-current rail brakes are impractical for use on U.S. railroads at the present time. Eddy-current wheel brakes were found to be impractical because of excessive power requirements.

- Future rail vehicles operating above 130 mph should have air or hydraulic retarders incorporated into their braking system because of the wear-free anti-wheel-locking operation of these brakes and their ability to dissipate large amounts of kinetic energy.

- After years of experimentation with adhesion enhancement schemes, the Europeans are now concentrating on improving wheel-slide protection systems as a means of dealing with low adhesion. Except for the use of brake shoes containing iron particles, the costs and technical difficulties of enhancing adhesion proved to be too great to be practical.

- The purpose of this report is not to determine whether or not a given braking system can meet its braking-rate targets on a vehicle operating at 130 mph; rather, the question is whether it can do so safely and economically. The purpose of this paper has been to attempt to determine how this goal can be accomplished with the smallest investment in research and development, retrofitting, installation and maintenance. This approach has been illustrated by the evaluation of eddy-current brakes, which would be a very attractive system if the technical problems involved were solved. Although these problems are not insurmountable, the cost of finding and implementing solutions would make eddy-current brakes more expensive than any of the other systems.

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16. _Ibid_, p. 108.

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APPENDIX A

Railroad Revitalization and Reform Act of 1976

Title VII. Section 703: "Required Goals"

REQUIRED GOALS

SEC. 703. The Northeast Corridor improvement project shall be implemented by the Secretary in order to achieve the following goals: 45 USC 853.

(1) INTERCITY RAIL PASSENGER SERVICES.—(A)(i) Within 5 years after the date of enactment of this Act, the establishment of regularly scheduled and dependable intercity rail passenger service between Boston, Massachusetts, and New York, New York, operating on a 3-hour-and-40-minute schedule, including appropriate intermediate stops; and regularly scheduled and dependable intercity rail passenger service between New York, New York, and Washington, District of Columbia, operating on a 2-hour-and-40-minute schedule, including appropriate intermediate stops.

(ii) Improvements in facilities in accordance with route criteria approved by the Congress, on routes to Harrisburg, Pennsylvania, and Albany, New York, from the Northeast Corridor main line, and from Springfield, Massachusetts, to Boston, Massachusetts, and New Haven, Connecticut, in order to facilitate compatibility with improved high-speed rail service operated on the Northeast Corridor main line.

(B) The improvement of nonoperational portions of stations (as determined by the Secretary in consultation with the National Railroad Passenger Corporation) used in intercity rail passenger service and of related facilities and fencing. Fifty percent of the cost of such improvements shall be borne by States (or local or regional transportation authorities), except that the Secretary may, in his sole discretion, fund entirely any safety-related improvement.

(C) The improvements required by this section shall be accomplished in a manner which is compatible with the accomplishment in the future of additional improvements in service levels, and which will produce the maximum labor benefit in terms of hiring persons who are presently unemployed.

(D) The submission by the Secretary and the National Railroad Passenger Corporation to the Congress of annual reports on progress achieved and work in progress and planned (including the need for further improvements) with respect to the completion of this program, including an up-to-date accounting of intercity passenger ridership, revenues from such ridership, expenses, and on-time dependability of intercity passenger trains in the Northeast Corridor.

(E) Within 2 years after the date of enactment of this Act, the submission by the Secretary to the Congress of a report on the financial and operating results of the intercity rail passenger service established under this section, on the rail freight service improved and maintained pursuant to this section, and on the Report to Congress.

practicability, considering engineering and financial feasibility and market demand, of the establishment of regularly scheduled and dependable intercity rail passenger service between Boston, Massachusetts, and New York, New York, operating on a 3-hour schedule, including appropriate intermediate stops, and regularly scheduled and dependable intercity rail passenger service between New York, New York, and Washington, District of Columbia, operating on a 2½-hour schedule, including appropriate intermediate stops. Such report shall include a full and complete accounting of the need for improvements in intercity passenger transportation within the Northeast Corridor and a full accounting of the public costs and benefits of improving various modes of transportation to meet those needs. If such report shows (i) that further improvements are needed in intercity passenger transportation in the Northeast Corridor, and (ii) that improvements (in addition to those required by subparagraph (A)(i) of this paragraph) in the rail system in such area would return the most public benefits for the public costs involved, the Secretary shall make appropriate recommendations to the Congress. Within 6 years after the date of enactment of this Act, the Secretary shall submit an updated comprehensive report on the matters referred to in this subparagraph. Thereafter, if it is practicable, the Secretary shall facilitate the establishment of intercity rail passenger service in the Corridor which achieves the service goals specified in this subparagraph.

(2) Rail Commuter Services, Rail Rapid Transit, and Local Transportation.—To the extent compatible with the goals contained in paragraph (1) of this section, the facilitation of improvements in and usage of rail commuter services, rail rapid transit, and local public transportation.

(3) Rail Freight Service.—The maintenance and improvement of rail freight service to all users of rail freight service located on or adjacent to the Northeast Corridor and the maintenance and improvement of all through-freight services which remain in the Northeast Corridor, to the extent compatible with the goals contained in paragraphs (1) and (2) of this section.

(4) Passenger Radio Telephone Service.—To the extent compatible with the goals contained in paragraph (1) of this section, the continuation of and improvement in passenger radio telephone service aboard trains operated in high-speed rail service between Washington, District of Columbia, and Boston, Massachusetts. The President and relevant Federal agencies, including the Federal Communications Commission, shall take such actions as are necessary to achieve this goal, subject to the provisions of the Communications Act of 1934 (47 U.S.C. 151 et seq.), including necessary licensing, construction, operation, and maintenance standards for the radio service, as determined by the Federal Communications Commission to be in the public interest, convenience, and necessity.

APPENDIX B

CALCULATION OF RAIL TEMPERATURE RISE DUE TO THE USE OF THE EDDY-CURRENT RAIL BRAKE

PROBLEM

Determine rail temperature rise for a consist of six Amcoaches and two locomotives where all six Amcoaches are equipped with S.C.N.F. eddy-current rail brakes. Temperature rise is to be found as a function of headways and of the number of times a train equipped with eddy-current rail brakes passes over the section of track in question.

ASSUMPTIONS

Braking force = 1200 daN/brake (from tests performed with the French Z7001, a self-propelled rail research vehicle).
Specific heat = 0.458 $^{kj}/_{kg°C}$ (Materials Engineering, Selector Issue 1966-7.

The I.R.T. heat transfer coefficients and time constants can be used in calculations on U.S. rail. U.S. rail is 132 lb/yd $\Rightarrow m_1 = 22.4\ ^{kg}/_{m}$.

METHOD OF SOLUTION

Use the Giovanchini-Pascal mathematical model for heat transfer, and their experimentally determined heat-transfer coefficients and time constants (from Scale Modeling Analysis of the S.E.N.F. Braking System, Research Report I.R.T. No. 10, by J. Giovanchini and J. Pascal).

SOLUTION

$\theta_{1,\infty}$ = equilibrium temperature of head of rail.

$\theta_{2,\infty}$ = equilibrium temperature of web and sole.

$d\theta_1$ = instantaneous heating of mass m_1.

λ_1, λ_2 = time constants.

K, K_1, K_2 = heat transfer coefficients

F = braking force.

m_1 = mass of railhead per meter of length.

C = specific heat of rail.

T = headway

$$\theta_{1,\infty} = d\theta_1 \frac{1 - g_2}{1 - (f_1 + g_2) + f_1 g_2 - f_2 g_1} ;$$

$$\theta_{2,\infty} = d\theta_1 \frac{f_2}{1 - (f_1 + g_2) + f_1 g_2 - f_2 g_1} ,$$

where: $$d\theta_1 = \frac{F}{m_1 C}$$

$$f_1 = e^{-\lambda_1 T} - \frac{\alpha}{\beta - \alpha}\left[e^{-\lambda_2 T} - e^{-\lambda_1 T}\right]$$

$$g_1 = \frac{1}{\beta - \alpha}\left[e^{-\lambda_2 T} - e^{-\lambda_1 T}\right]$$

$$f_2 = \frac{\beta}{\alpha - \beta}\left[e^{-\lambda_2 T} - e^{-\lambda_1 T}\right]$$

$$g_2 = e^{-\lambda_1 T} + \frac{\beta}{\beta - \alpha}\left[e^{-\lambda_2 T} - e^{-\lambda_1 T}\right] \quad \text{and}$$

$$\alpha = \frac{1}{K}\left[K + K_1 - \lambda_1 m_1 c\right]$$

$$\beta = \frac{1}{K}\left[K + K_1 - \lambda_2 m_1 c\right]$$

Now, $K = 7.2\ {}^{W}/{}^{o}C$

$$K_1 = 1.4^{W}/{}^{o}C$$

$$K_2 = 3.6^{W}/{}^{o}C$$

$$M_1 = 22.4\ \frac{kg}{m}$$

$$C = .458^{kj}/kg^{o}C$$

$$\lambda_1 = \frac{1}{92\ min}$$

$$\lambda_2 = \frac{1}{12.5\ min}$$

$$T = 15\ min$$

$$F = 12\ \text{brakes} \times 1200\ \text{daN/brake} = 1.44 \times 10^5 N$$

$$e^{-\lambda_1 T} = e^{-(.0109).(15)} = .85$$

$$e^{-\lambda_2 T} = e^{-(.08).(15)} = .30$$

$$e^{-\lambda_2 T} - e^{-\lambda_1 T} = -.55$$

$$\alpha = \frac{1}{7.2}\left[7.2 + 1.4 - .0109 \cdot 22.4 \cdot .458 \cdot \frac{1000}{60}\right] = .94$$

$$\beta = \frac{1}{7.2}\left[7.2 + 1.4 - .08 \cdot 22.4 \cdot .458 \cdot \frac{1000}{60}\right] = -.70$$

$$f_1 = .85 - \frac{.94 \cdot -.55}{-.70 - .94} = .54$$

$$g_1 = .85 - \frac{1 \cdot -.55}{-.70 - .94} = .34$$

$$f_2 = \frac{(.94)(-.70)(-.55)}{.94 - (-.70)} = .22$$

$$g_2 = .85 + \frac{-.7(-.55)}{-.7 - .94} = .62$$

$$d\theta_1 = \frac{1.44 \times 10^5 N}{22.4 \text{ kg/m} \left(.458 \frac{\text{kj}}{\text{kg}^\circ\text{C}}\right) \times 1000} = 14.04$$

therefore, $\theta_{1,\infty} = 14.04 \left[\frac{1 - .62}{1 - (.54 + 62) + (.54)(.62) - (.22)(.34)} \right]$

$$= 14.04 \left[\frac{.38}{.1} \right] = 53.3°C$$

$$\theta_{2,\infty} = 14.04 \left[\frac{.22}{.1} \right] = 30.9°C$$

Results:

Headway	$\theta_{1,\infty}$		$\theta_{2,\infty}$	
	°C	°F	°C	°F
15 min	53.3	95.7	30.9	55.5
30 min	30.0	54.0	13.7	24.7
60 min	20.6	37.0	5.8	10.5

$\Delta 5°C = \Delta 9°F$

These are the peak temperatures reached after a sufficient number of tains have passed. The following set of calculations will show how long it will take to reach peak temperature for one-half hour headways.

Determining rail temperature rise after passage of nth train:

$$\theta_{1,n} = py_1^n + qy_2^n + \theta_{1,\infty}$$

$$\theta_{2,n} = \frac{1}{g_1} \left\{ p(y_1 - f_1)y_1^n + q(y_2 - f_1)y_2^n \right\} + \theta_{2,\infty}$$

where y_1 and y_2 are solutions to:

$$y^2 - y(f_1 + g_2) + f_1 g_2 - g_1 f_2 = 0$$

and $$p = \frac{1}{y_2 - y_1} \quad g_1\theta_{2,\infty} - \Big((f_1 - y_2)(d\theta_1 - \theta_{1,\infty})\Big)$$

$$q = \frac{1}{y_2 - y_1} \quad \Big((f_1 - y_1)(d\theta_1 - \theta_{1,\infty}) - g_1\theta_{2,\infty}\Big)$$

Now, for 1/2-hour headways,

$$f_1 = .36 \qquad f_2 = .25$$

$$g_1 = .38 \qquad g_2 = .45$$

$$\theta_{1,\infty} = 30.0 \qquad \theta_{2,\infty} = 13.7$$

$$d\theta_1 = 14.04$$

Using the quadratic equation to solve

$$ax^2 + bx + c = 0$$

where $a = 1$

$$b = (f_1 + g_2)$$

$$c = f_1g_2 - g_1f_2$$

we have

$$y = \frac{-b \pm \sqrt{b^2 - 4ac}}{2a} = \frac{+ .81 \pm \sqrt{(.81)^2 - (.4)(1)(.067)}}{2(1)}$$

$$= +.40 \pm .31 \Rightarrow y_1 = .71 \text{ and } y_2 = .09$$

$$\therefore \; p = \frac{1}{.09 - .71} \left[(.38)(13.7) - (.35 - .09)(14.04 - 30.0)\right] = -15.09$$

$$q = \frac{1}{.09 - .71} \left[(.36 - .71)\ (14.04 - 30.0) - (.38)(13.7)\right] = -.61$$

Now, $\theta_{1,1} = -15.09(.71)^1 + (-.61)(.09)^1 + 30.0 = 19.2°C$

$$\theta_{2,1} = \frac{1}{.38} \left[-15.09(.71 - .36)(.71)^1 + (-.61)(.09 - .36)(.09)^1\right]$$

$$13.7 = 3.8\ °C$$

$\theta_{1,2} = 22.4°C$	$\theta_{2,2} = 6.7°C$
$\theta_{1,4} = 26.2°C$	$\theta_{2,4} = 10.1°C$
$\theta_{1,7} = 28.6°C$	$\theta_{2,7} = 12.4°C$
$\theta_{1,10} = 30.0°C$	$\theta_{2,10} = 13.5°C$

For 1-hour headways,

$$\theta_{1,2} = 17.3°C \qquad \theta_{2,2} = 2.8°C$$

$$\theta_{1,2} = 18.9°C \qquad \theta_{2,3} = 4.2°C$$

$$\theta_{1,5} = 20.1°C \qquad \theta_{2,5} = 5.4°C$$

These results are illustrated in Figures B-1 and B-2, which show rail temperature rise as a function of time.

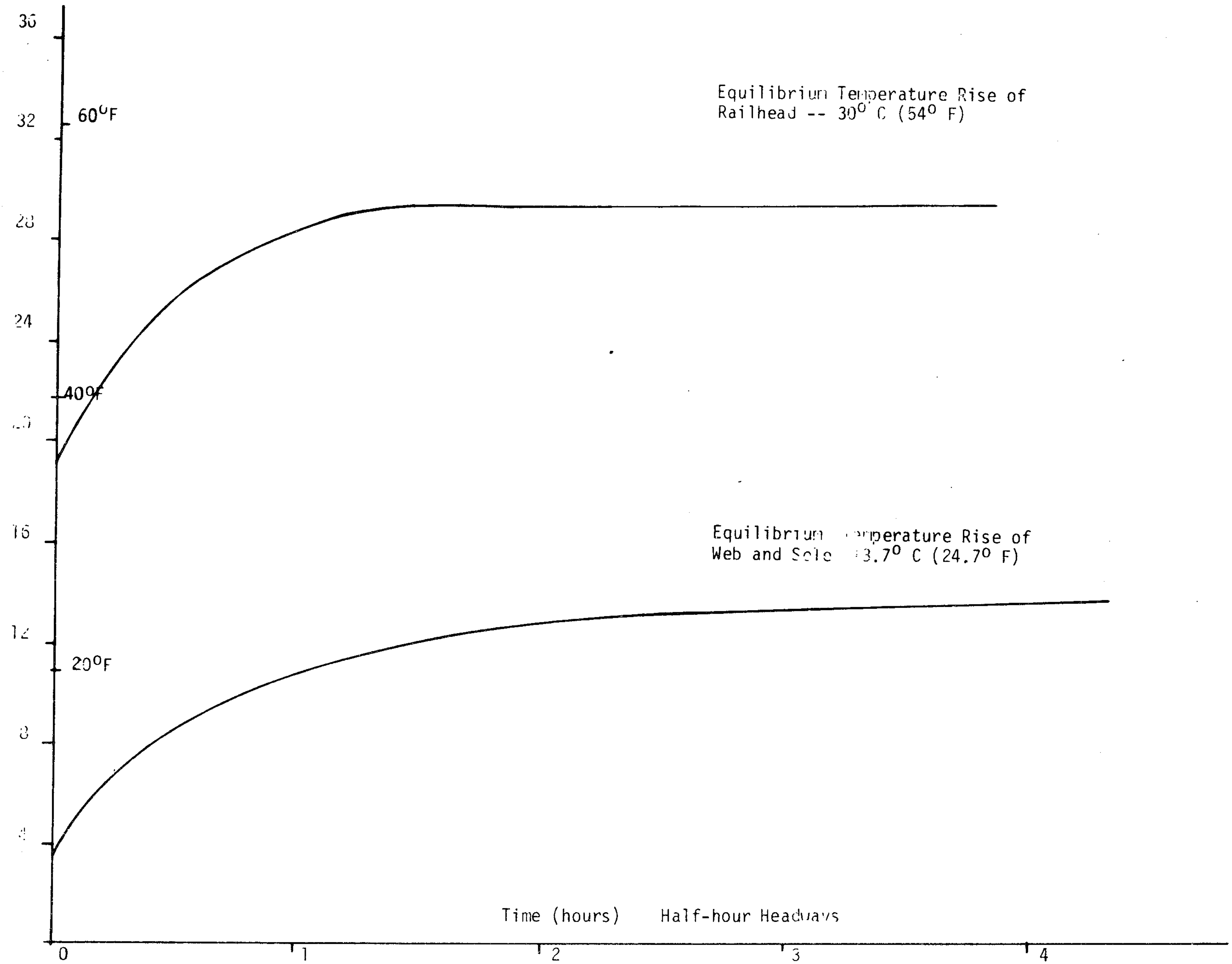

Figure B-1. Rail Temperature Rise

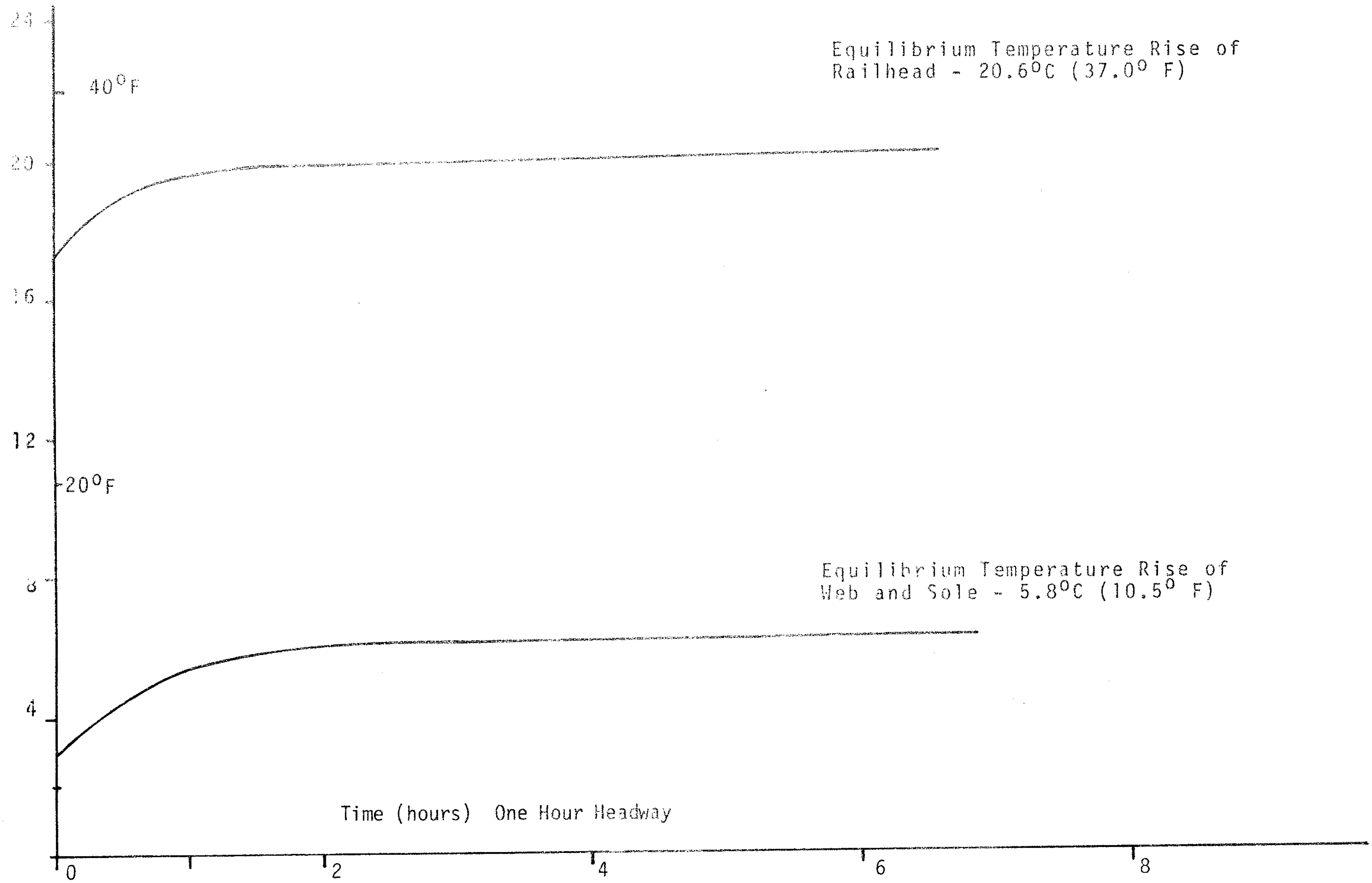

Figure B-2. Rail Temperature Rise

APPENDIX C
UPGRADED NORTHEAST CORRIDOR
TOP SPEEDS AND BLOCK DISTANCES

Transportation and Distribution, Inc. (TAD) has produced a speed profile for an upgraded Northeast Corridor (NEC) based on a two-hour and thirty-five minute schedule between Washington and New York. The maximum running speed required to meet this schedule is 120 mph. For the purpose of this study, 130 mph has been assumed as an upper limit to allow for possible over-speeding.

Precise block distance for the upgraded NEC have not been established as yet. However, the NEC Office of FRA has indicated that they will be on the order of 10,000 to 12,000 feet. Because they require much longer distances to come to a stop, freight trains are the determining factor behind such long blocks.

The information shown in Figure 1-4 can be used to find the no-slip deceleration rates associated with various stopping distances, which in turn, can help to describe the behavior of a rail system using shorter blocks.