RAILROAD-CAR BRAKING.

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A sketch of the practical development of brakes in railroad services; an analysis of the results of investigation of brake-shoe friction; a definition of the requirements and limitations of brake efficiency in practice; a consideration of the loss of braking efficiency through disturbance of the normal rail-pressures by brake retardation and a description of the means of compensating for it. The peculiar characteristics of the magnetic brake are also briefly described. Illustrated.

The braking problem is as old as railroading and, in the early conception of it, the problem appeared a simple one. Practical railroading had no sooner accomplished the motion of trains than the necessity for stopping them appeared, and it was early perceived that the "adhesion," as it was usually called, or the frictional resistance to sliding of the wheels upon the rails, through which the wheels continue to rotate in spite of considerable resistance, is the most available retarding force. The application of a brake to the wheel to excite the retarding influence of the rail friction has naturally suggested itself, and when, in addition, the simple mechanism was provided for manually applying the brake to the wheel with sufficient force, in an emergency, to cause the wheel to slide upon the rail, the braking problem appeared to have been satisfactorily solved. Under the operating conditions of railroading then existing, the application by hand of brakes to a portion of the wheels of a train was ordinarily sufficient for all practical purposes, and when, in an emergency, the brake resistance so acquired was sufficient to overcome the

rail friction and cause the wheels to slide upon the rails, it was quite generally conceded that the most that could be expected had been accomplished and a disaster not averted by such means was accepted as one of the unfortunate but unavoidable contingencies of railroading. This early method of controlling the speed of trains has since been modified only by the substitution of mechanical for physical power in the application of the brakes, which has been extended to all the wheels, and by so restricting the pressure of the brake shoes that the wheels shall *not* be caused to slide upon the rails; but the diversified and intricate character of the influences now recognized as constituent elements in the efficiency of the power-brake, have involved the subject in much complication.

The advantage of the use of power, under the control of the engineer, for promptly applying brakes upon all the wheels of a train, early became recognized, and the special advantages of compressed air for the transmission of the power throughout trains of considerable length, established the air-brake in a pre-eminent position. At first, the compressed air was stored in a reservoir upon the locomotive and, by means of a line of iron pipe, extending throughout the length of the train, with flexible hose and couplings between vehicles and an operating valve upon the locomotive, it was conducted to the brake-cylinder upon each car, by which the brakes were applied to the wheels, through the intervention of suitable rods and levers. The time required to convey the necessary volume of compressed air from the storage reservoir upon the locomotive to all the brake-cylinders of even a comparatively short train, and the total disability resulting from rupture at any point of the air-conduit, caused this form of air-brake to be supplanted by the automatic air-brake, in which an auxiliary storage reservoir, of sufficient capacity to operate a single brake-cylinder, was added upon each car. Through the operation of a triple-valve device, which connects the train-pipe or air-conduit, the auxiliary reservoir and the brake-cylinder upon each car, admission of compressed air into the train-pipe causes each auxiliary reservoir to become charged to operate the brakes, and discharge of air from the train-pipe, from any cause, causes communication with the auxiliary reservoir to be transferred from the train-pipe to the brake-cylinder, whereby a corresponding quantity of compressed air is discharged from the auxiliary reservoir into the brake-cylinder. By defining the reduction of pressure of air in the train-pipe, the pressure of the brake-shoe

upon the wheels may be graduated to any desired degree within the limit established by the ultimate equalization of air-pressure in the auxiliary reservoir and the brake-cylinder, and accidental rupture of the train-pipe instantly operates to stop the train and to prevent further progress without effective repair. In any case, restoration of the air-pressure in the train-pipe actuates the triple valve to re-establish access to the auxiliary reservoir, whereby it is recharged with air-pressure, and to transfer communication with the brake-cylinder from the auxiliary reservoir to the atmosphere, through which the brakes are released.

When the automatic air-brake became employed upon freight-trains, it was discovered that, in making a quick stop by venting the train-pipe at the locomotive, the interval of time required to cause an operative reduction of the air-pressure in the train-pipe at the rear end of the train was so considerable that effective application of the brakes upon the rear cars was delayed until sufficient retardation of the forward portion of the train had become effected to cause collision with the rear cars that damaged and often disabled the cars and did violence to the lading. To remove this obstacle, the automatic air-brake became superseded by the quick-action automatic air-brake, in which, when a quick stop is desired, each triple-valve opens a vent in the train-pipe, in addition to the vent upon the locomotive. By this means, an operative reduction of the air-pressure progresses throughout the train-pipe, of the longest trains, with nearly the velocity of sound, and damage from the serial character of the application of brakes by compressed air becomes practically eliminated. Incidentally, also, in the local venting of the train pipe air at each succeeding triple-valve, the utilization of this source of power, formerly wasted at the engineer's operating valve, was accomplished by conducting the vented air into the adjacent empty brake-cylinder, before it receives the ordinary supply provided by the auxiliary reservoir. Thereby, the ultimate air-pressure in the brake-cylinder is augmented about 20 per cent., and the character of the application of the brakes in disaster-threatening emergencies is further distinguished from that in ordinary service, where neither the violence nor the power of the emergency application is desirable or even tolerable.

While the character of the development of the compressed air brake has thus been chiefly dictated by conditions rendered conspicuous by increased length of trains, and would hardly have been suggested under the conditions to which electric railroad

operation appears best adapted, the high operative efficiency and other advantages thereby acquired are participated in by short as well as long trains, and are therefore of interest in dealing with the question of braking in electric service. Where even only two cars are operated together as a train, the automatic feature of the air-brake is still essential, in precisely the same way, if not in the same degree, as in longer trains, and, except where cars are always operated singly, the same is true of the increased efficiency secured through the quick-action feature of the automatic brake.

The first systematic or comprehensive investigation of the effect of brakes upon the wheels of moving vehicles was that of an elaborate series of experiments, conducted upon the London, Brighton and South Coast Railway, in 1878, with apparatus constructed and operated by George Westinghouse, under the supervision of Capt. Douglas Galton, representing the Institution of Mechanical Engineers, and which are now commonly known as the "Westinghouse-Galton" brake-tests. The ingenuity and perfection of the apparatus, and the scientific and unbiased character of the experiments, establish this investigation upon a high scientific plane and give its results an authoritative standing which has never been impaired or supplanted by subsequent investigations. Had the range of these experiments been extended but a little farther, the entire subject would have been elucidated. In fact, while later experiments have demonstrated the importance of additional information, that information has not yet been definitely supplied, and the Westinghouse-Galton investigation still stands as the only reliable source of definite knowledge concerning the important characteristics of brake-shoe friction.

More recent investigations have been conducted upon a machine devised for experimenting upon the friction of brake-shoes, but their purpose has been to determine the relative commercial values of brake-shoes of different materials, rather than to supplement the information furnished by the Westinghouse-Galton tests. While such investigations have doubtless been useful in indicating the relative advantage of different materials for brake-shoes, and have perhaps satisfactorily determined, within limits, the degree of efficiency which may properly be required of brake-shoes, they have added to previous knowledge of brake-shoe friction only in suggesting certain general characteristics, without furnishing adequate data to extend our knowledge of the action of brakes in stopping a train.

The Westinghouse-Galton experiments were conducted in a

moving four-wheeled car, provided with the necessary recording apparatus, the wheels being steel-tired and the brake-shoes of cast iron, and the results of the experiments may be summarized as follows: The coefficient of brake-shoe friction decreases with increased speeds and, at constant speed, it decreases with the increase of the time that the surfaces are in contact. The coefficient of friction does not appear to increase with increased pressure per square inch. When the wheels roll without sliding upon the rails, the rail-friction (or adhesion) is unaffected by the speed, but declines greatly with sliding of the wheels, when it varies inversely with the speed, corresponding with the brake-shoe friction, but is much inferior thereto. The rail-friction is very materially affected by the condition of the rail, being greatest when the rail is perfectly dry or very wet (as when washed by a hard rain) and least when the rail is quite moist; but, by the use of sand upon the rails, the effect of moisture is practically eliminated.

The actual variation of the brake-shoe friction with increased speeds is very regular, beginning with the friction of rest (static friction) and declining, very rapidly at first, but with continually decreasing rapidity, up to the highest speeds of the experiments. The undoubted fact that the friction cannot vanish for any finite speed, considered in conjunction with the peculiar character of the decline, suggested to the writer that the results of the experiments might be properly represented by a portion of an equilateral hyperbola, and upon applying the theory of probability to the mean results, the coefficient of friction is found to be represented by the equation

$$f = \frac{.326}{1 + .03532 V}$$

in which V represents the speed in miles per hour. The following table shows both the computed coefficients of friction from this formula and the means of the observations, up to a speed of 100 miles an hour.

Speed V.	Coefficient of friction, f .		Speed V.	Coefficient of friction, f .	
	Computed.	Observed.		Computed.	Observed.
0	.326	.330	45	.126	.127
5	.277	.273	50	.118	.116
10	.241	.242	55	.111	.111
15	.213	.223	60	.105	.074
20	.191	.192	65	.099	
25	.173	.166	70	.094	
30	.158	.164	80	.085	
35	.146	.142	90	.078	
40	.135	.140	100	.072	

It is worthy of note that the mean observed result for a speed of 60 miles per hour differs materially from the result of a law that applies with remarkable fidelity to the observed results at other speeds. This is perhaps attributable to two causes: first, while each of the other results (except for the lowest three speeds) is the mean of from 54 to 90 experiments, that for 60 miles represents but twelve experiments—much the smallest number at any speed—of which, the maximum observed result was .123 and the minimum was .058. The other probable cause is that, it having been discovered that the coefficient of friction is less at high speeds, the pressure was much increased at the very high speeds to compensate, and it is now known that the coefficient of friction is an inverse function of the pressure. The mean result at 60 miles an hour does not, therefore, appear to be fairly comparable with those at other speeds, and it has been rejected in determining the constants in the above formula.

In the tabulated results of these experiments, Capt. Galton gives the maximum and minimum, as well as the mean value of the coefficient of friction observed for each speed. It is observable that the maximum values very well follow a similar law to that above given for the means, while the minimum values are too irregular to suggest more than a general inclination to decline as the speed is increased. As the maximum values correspond more nearly with the results of more recent experiments, the writer has also formulated them. The most probable law is represented by the equation

$$f = \frac{.382}{1 + .02933 V}$$

and the calculated and observed values are given in the following table:

Speed V.	Coefficient of friction, <i>f</i> .		Speed V.	Coefficient of friction, <i>f</i> .	
	Calculated.	Observed.		Calculated.	Observed.
0	.382		45	.165	.179
5	.333		50	.155	.153
10	.295	.281	55	.146	.136
15	.265	.280	60	.138	.123
20	.241	.240	65	.131	
25	.220	.205	70	.125	
30	.203	.196	80	.114	
35	.188	.197	90	.106	
40	.176	.194	100	.097	

The decrease of the coefficient of brake-shoe friction from continued contact apparently follows the same character of law as does that from increase of speed; that is, the fall is rapid at first,

but the rate of fall quickly begins to decline. It was assumed by Capt. Galton that the decline of the friction is as the increase of the time of contact, but a careful analysis of the results shows that it is a function of the product of the speed and the time, or of the distance through which the shoe rubs upon the wheel. It is very unfortunate that the observed effect of continued rubbing does not extend over sufficient time to permit reliable formulation. The more recent experiments appear to show that, at the same speed, the friction becomes constant after a time, but it is probable that the coefficient of friction continues to yield to the influence of continued rubbing throughout stops from the highest speeds yet attained in practice. In formulating the incomplete data of Capt. Galton, therefore, the purpose is merely to present as comprehensive a view as possible of the complexity of the subject, and the results must not be accepted as trustworthy for practical use. Accepting the conclusion drawn from later experiments, that the coefficient of friction remains stationary after some definite period of continued rubbing at constant speeds, the question is still open as to whether the stationary point is the same for all speeds or is simply a function of the initial coefficient of friction, corresponding to the speed and brake-shoe pressure. While there is no definite information upon this point, beyond the limited observation of the Westinghouse-Galton experiments, that, so far as it goes, points to the latter view, and all our information suggests the probability that the ultimate level is different for each different initial coefficient of friction. The corresponding equilateral hyperbola is therefore represented by an equation of the form

$$f_1 = \frac{1 + h s}{1 + c s} f$$

where f is, as hitherto, the initial coefficient of friction, corresponding to the speed and pressure, s is the distance, in feet, through which the surface of the wheel has travelled in frictional contact with the brake-shoe, and h and c are constants. Applying the theory of the least squares to the observations of Capt. Galton (except those at a speed of 60 miles an hour, which are rejected for reasons already given), the values found for h and c cause the above expression to become

$$f_1 = \frac{1 + .000472 s}{1 + .002390 s} f$$

Thus the coefficient of friction, f , for a given speed, as indicated

in the tables already submitted, becomes reduced, after the brakes have been applied through a distance of 50 feet, to .914 *f*; after 100 feet, to .845 *f*; after 200 feet, to .740 *f*; after 300 feet, to .665 *f*; after 500 feet, to .563 *f*; after 1000 feet to .434 *f*; after 2000 feet, to .336 *f*; and the ultimate level, which is not yet reached after two miles, is $472/2390$ *f*, or about .2 *f*. The later experiments upon the efficiency of brake-shoes included some temperature tests, the results of which indicate that, under the conditions of those experiments, the stationary level is reached after from 2000 to 3000 feet, and there is good reason to believe that it is much higher than .2 *f*. Of the observations employed in determining the constants in the above expression, none was for a greater distance than 814 feet and the larger number were for short distances, which thus gave undue weight to the rapidly falling feature of the curve. In the absence of more extended observation, no practically useful mathematical application of this characteristic of brake-shoe friction can be realized. As it exerts a very important and effective influence upon the friction and the rate of retardation during every stop, the impossibility at present of determining the rate of retardation under varying conditions is obvious. Grade, for instance, not only affects the rate of retardation by directly modifying the effective retarding force, but also, in varying the distance in which a given reduction of speed occurs, it again indirectly affects the rate of retardation. It thus happens that experimental determinations of the rate of retardation are more misleading than useful under any other conditions than exactly those of the experiments. Experimental stops of the same train, made one right after another, from the same speed, at the same place, on the same track, differ very materially from obscure causes of this character, while the conditions apparently differ so little that they are looked upon as "practically" the same. The errors, accidents and other sins that occasionally result from the prevailing custom, in this practical, money-getting age, of founding conclusions upon premises that are "good enough for practical purposes," have no better illustration perhaps than in the matter of train stops, in which the enjoyment of confidence in one's conclusions generally varies inversely with accuracy of understanding of the subject.

In order to observe the effect of varied pressure in the Westinghouse-Galton experiments, a set of brake-shoes was made with projections cast upon their face, so that only about one-third the whole surface of the shoe came into contact with the wheels.

What actual pressure was then applied to the wheel is not recorded. From statements of Rennie, concerning the results of experiments upon the relation of pressure to static friction, it was expected that the coefficient of brake-shoe friction would be considerably increased by the increased pressure per square inch resulting from this modification of the brake shoes; but Capt. Galton says of these experiments simply that he is not prepared to say that any greater coefficient of friction was obtained, and that but a very small number of experiments were made with these special brake-shoes because the projections were entirely worn off in twelve applications. In view of the division of the rubbing surface into a considerable number of small surfaces with edges more or less sharp, it is doubtful if any reliable conclusions could have been drawn from pressure experiments in which the variation of the pressure per square inch is secured by such means. More recent experiments have shown very conclusively that, other conditions being the same, the coefficient of friction declines as the pressure increases and, while insufficient data has yet been presented to determine the character of the decline, there are substantial reasons for believing that it follows the same character of law as do the declines from increased speed and extended rubbing contact.

This view of the dependence of the coefficient of friction upon pressure, throws some light upon the considerable range of observed values recorded, at the various speeds, in the experiments to determine the influence of speed upon the coefficient of friction. In some cases, the brake-shoe pressure employed was more than three times the weight upon the rails, while at other times the pressure was but about one half that weight, so that the pressure varied from about 40 to about 250 pounds per square inch of brake-shoe surface. It is most significant, also, that in the cases cited in Capt. Galton's reports to illustrate special conditions, high pressures are accompanied by coefficients of friction that are about the minimum observed for the corresponding speeds, while the lowest pressures are accompanied by the maximum coefficients. As it is to be assumed, of course, that the coefficients of friction given in the tables for different speeds are those observed at the beginning of application only, it appears most probable that the quite uniformly large range of values for the different speeds is principally due to variation of pressure. Under such conditions, therefore, it can only be assumed that the table of mean coefficients of friction for different

speeds is representative also of the mean brake-shoe pressure during the experiments, which is probably about 90 pounds per square inch, though it cannot be definitely determined from the data given in Capt. Galton's reports, while the table of maximum coefficients appears to represent a pressure of from 40 to 50 pounds per square inch, or a total pressure of about 2,000 pounds upon the standard brake-shoe in this country.

The coefficient of rail-friction was found to vary for dry rails from .19 to .35, and averaged about .25. Upon wet or greasy rails, without sand, it fell to as low as .15 in one experiment, but averaged .18. With sand upon wet rails, it never fell below .20 and rose in some cases as high as to .40, so that, with the use of sand, the rail-friction of a wet rail is at least equal to that of a dry rail without stand. When the brake-shoe friction overcame the rail-friction, and caused the wheels to slide upon the rails, the coefficient of rail-friction immediately began to decline and then varied inversely as the speed, in much the same way as brake-shoe friction, but is much inferior thereto. This may be readily explained by the greatly reduced area of contact and consequent high pressure per square inch between wheels and rails. It may also, of course, be due in some measure to inferior frictional qualities of the steel wheel upon steel rails, as compared with cast-iron brake-shoes upon steel wheels.

The maximum retardation which may be utilized in stopping railroad vehicles by the customary means of brakes is therefore that which is realized by so applying brake-shoes to the wheels that the resulting brake-shoe friction shall be uniform, and just insufficient to overcome the constant, static rail-friction. If sand be suitably provided whenever the condition of the rail requires it, the coefficient of rail-friction always available (unless perhaps in the case of railroads running in streets) is at least .20 and may doubtless be safely regarded, in at least all cases where emergency calls for the highest efficiency, as .25 of the pressure of the wheel upon the rail. A brake system of ideal efficiency in the time of necessity is thus one in which the brake-shoes are so applied to the wheels that a retarding rail friction equal to one-fourth the weight of the train is instantly realized and continuously maintained throughout the stop. In such a case, the retardation (ignoring the resistances of rolling friction and the atmosphere) is one-fourth the acceleration of gravity, or 8.04 feet per second, which amounts to reducing the speed at the rate of almost $5\frac{1}{2}$ miles an hour per second. Stops would be made in a distance

represented by $.1338 V^2$, where V is the initial speed in miles per hour, or in 482 feet at 60 miles an hour, 214 feet at 40 miles an hour and only $53\frac{1}{2}$ feet at 20 miles an hour. The obstacles to the realization of a brake of such efficiency are apparent at once when the variable nature of the coefficient of brake-shoe friction is understood, and the difficulties which attend variation of the pressure of the brake-shoe upon the wheel to compensate for the fluctuation of the coefficient of friction through the simultaneous operation of such complex influences, appear insuperable. But the problem is not entirely hopeless and it is useful to consider what has been and what may yet be done to increase the efficiency of braking.

In the outset, while at first glance the coefficient of static rail-friction, which measures the maximum retardation, appears to be inflexibly established, such is not altogether the case. The inferiority of the coefficient of friction between the wheel and the rail to that of the brake shoe upon the wheel must be attributed chiefly to the very great difference in the areas of the surfaces in contact and the consequent difference in the pressure per unit area. The convenient doctrine of Morin, that the friction is independent of the area of the surfaces in contact, has been the cause of much misapprehension and of many errors of construction. Mr. P. H. Dudley has clearly demonstrated that broad-headed rails yield a materially greater tractive power to locomotives than narrow-headed ones, and it may be confidently assumed that any means of increasing the surface of contact between wheels and rails adds to the resistance which measures the maximum efficiency of the brakes. The theoretical *line* of contact between a wheel and a rail broadens out practically into a somewhat pear-shaped surface, which differs in form and extent with different materials and pressures. The head of the rail is locally depressed and the circular periphery of the wheel becomes flattened, resulting in a surface of contact, the extent of which depends upon the elasticity of the materials, the diameter of the wheel and the forms of the rail-head and wheel-tread. It is evident that a greater contact area occurs with a large than with a small diameter of wheel, and it is equally clear that greater elasticity of the material of either wheel or rail conduces to the same result. Steel is generally more elastic than chilled cast iron, and recent observation indicates a higher coefficient of rail friction with steel-tired wheels than with chilled iron. It is true that, while such a result should be expected because of a larger area of

contact, it may also be that the frictional qualities of the materials in contact constitute a factor of some importance. It has generally been understood that the dynamic friction of cast-iron brake-shoes upon chilled cast-iron wheels, exceeds that of the same shoes upon steel-tired wheels. While this conclusion does not appear to have been actually established, yet, even if it be correct, it does not follow that the static friction of steel-tired wheels upon *steel rails* may not be greater than that of chilled cast-iron wheels. But, whatever the fact may be in this respect, it is reasonable to expect that the most effective *surface of rail-contact* occurs with large, steel-tired, straight-tread wheels upon broad rails, and, so far as our information yet extends, observation confirms this view sufficiently to warrant the statement that it is a matter of considerable importance.

The utilization of the retarding force available as rail-friction, by means of brakes, involves the application of a brake-shoe pressure which shall (a) diminish as declining speed causes the coefficient of friction to increase, which shall (b) increase as increased distance of frictional contact causes the coefficient of friction to decline, and which shall (c), when diminishing or increasing for such purposes, further diminish or increase as reduction or increase of pressure itself causes the coefficient of friction correspondingly to increase or decline. The combined effect of declining speed and increasing distance is far from being uniform in stops from different initial speeds. The friction apparently declines from continued rubbing in about the same proportion, through a given distance—the first 100 feet of the application, for illustration—whether the initial speed is high or low; but the elevation of the coefficient of friction by declining speed during such first 100 feet of application, is much less proportionally when the initial speed is high than when it is low. The two opposing influences are thus not uniformly effective. In stops from low speeds, the coefficient of friction increases, slowly at first and rapidly at the close, but continuously from beginning to end. At high speeds, the elevating influence is proportionately less effective at first, so that, for a time, the friction remains about stationary, or even declines at first before becoming stationary; but it always subsequently rises with an increasing rapidity that becomes so great as to be almost abrupt at its termination. It is therefore a characteristic of all stops that the coefficient of friction is comparatively low during the early portion and much higher toward the close; and, while manipulation

of the pressure to compensate for the compound fluctuation of the coefficient of friction appears hopelessly complicated, a partial realization of the efficiency of such an ideal brake system may be accomplished by employing a comparatively high brake-shoe pressure during the early part of the stop and subsequently so reducing it that the high coefficient of friction near the end of the stop shall not cause the wheels to slide upon the rails. The provision of means by which this partial utilization of the advantage of compensating the pressure is practically realized constitutes the latest and highest progress thus far made in the practical development of the air-brake.

In accomplishing the purpose of applying an increased brake-shoe pressure during the early part of the stop, the quick-action automatic brake has been modified by the addition of an automatic pressure-reducing valve to each brake-cylinder, by the use of which a high air-pressure is utilized in the brake-cylinder in emergency applications of the brakes, and is gradually reduced to the level of that which, in earlier forms of the air-brake, is maintained continuously throughout the stop without sliding wheels at the close. That material progress in braking is marked by this step will be easily appreciated when it is understood that, with the use of this apparatus, called the "high-speed" brake, stops from the higher speeds are about 30 per cent. shorter than those attained by the quick-action brake. By the more prompt application of a greater brake-shoe pressure than immunity from injurious wheel sliding permitted in ordinary applications of the brakes, the "emergency stop" became clearly distinguished by the quick-action air-brake, and, in passenger train service, it was shortened to about 80 per cent. of the shortest stop of the older automatic air-brake. By the application of a greater brake-shoe pressure during the early period of stops, for utilizing a larger proportion of the retarding force realized at lower speeds, the high-speed brake shortens emergency stops from high speeds, to about 70 per cent. of those of the quick-action brake, or 56 per cent. of those of the old automatic brake. The increased brake-shoe pressure is secured by the use of a high air-pressure, which, through the operation of the automatic reducing valve, is available only in emergency applications of the brakes, the moderate pressures of the old automatic brake being still preserved in ordinary operation to prevent possible injury of wheels. But the higher storage-pressure of the auxiliary-reservoir air is equivalent to a correspondingly increased volume of air stored at the

pressure of ordinary service, and thus the high-speed brake also provides for repeated brake-applications without recharging the auxiliary reservoirs—an incidental advantage greatly increasing the security of trains under conditions of daily occurrence.

It is opportune to digress at this point to consider an application of the high-speed brake which concerns conditions to which electric railroad operation is peculiarly adapted. Hitherto, the ordinary stops of railroad trains have generally consisted of a preliminary reduction of speed at a long distance from the stopping point, to bring the train under full control, and then of a gradual reduction of the remaining speed, continuously or in stages, to suit the conditions or the operator's views, until the train comes to a standstill at the stopping point. The stops of express trains have been infrequent, and the speed of way trains has been moderate, so that, in both cases, the time occupied in the stop has not called for careful consideration. But with the rapid growth of suburban traffic in all large cities, particularly since the introduction of electric railroads, the changed operating conditions of suburban trains give great importance to the time consumed in the frequent stops. Both in steam and electric railroad traffic of this kind, large expenditures of thought and money have been made to secure high acceleration in starting trains, while almost nothing has been done to secure the equally important high retardation in stopping. Every start is accompanied by a stop, and if economy of time is important in one, it is equally important in the other. The neglect which efficiency of stopping has suffered is doubtless due to the fact that, while acceleration in starting has generally been limited to that which may be acquired from the rail-friction of a few wheels, the retardation resulting from brakes upon practically all the wheels is so effective in comparison that its inferiority to what it *might* be is overlooked. Moreover, the character of the train-stop of ordinary service has been so firmly established and avoidance of the use of the full power—the emergency application—of the brake has been insisted upon for such good reasons, that departure from the customary “service” application of the brakes does not readily suggest itself. But the conditions now under consideration are quite different. At low speeds, the violence of the emergency application is apt to result in discomfort to passengers, and the brake-shoe pressure is too near the wheel-sliding limit to be desirable. But at the high speeds attained between stations in efficient suburban service, the initial coefficient of brake-shoe

friction is so low that no perceptible shock or disagreeable effect accompanies an emergency application of even the high-speed brake; and the coefficient of friction near the end of the stop is so much lower, through the effect of continued rubbing through so much longer distance than in a stop from the lower speeds, that the danger of wheel sliding is not troublesome.

There is also another important time-saving feature of the emergency application for service stops in train service of this character. In making an ordinary service application, the personal equation of the operator is an important element. The proper point at which to apply the brake, the force of initial application and each subsequent increase or reduction of the braking force (to prevent over-running or stopping short of the station) are matters of personal judgment in which men differ materially; the consequence of which is that, to be safe, the brakes are usually applied too early and time is lost in drifting into the station at low speed. In an emergency application, the personal element is largely eliminated, as the full application is practically instantaneous and, where definite speed may always be depended upon, the point at which the brakes should be applied may be designated by a signal post. Thus, full speed might be obtained over much the larger part of the distance traversed during an ordinary service stop, followed by a quick stop of the high-speed brake, and fully half the time occupied by the service stop would be saved.

Thus far, consideration of means for approaching the theoretically possible retarding force has been confined to those for overcoming the obstacles due to the complicated variation of the coefficient of brake-shoe friction. This is a subject which, while of manifest interest and importance to those who operate railroads, chiefly concerns those who supply the motor mechanism for applying the pressure of the brake shoes upon the wheels. The distribution of pressure upon the rails, through the wheels, which defines the maximum rail friction itself, and the construction of running gear and brake-beam supports to permit the utilization of the rail-friction for retardation, constitute a branch of the subject which ought seriously to concern those who construct the running gear and apply the motive power, but which is almost invariably subordinated to every other consideration and is usually overlooked entirely. It is a strange fact that, although the brake occupies the uppermost position of importance among railroad safety appliances, the actual work of stopping trains when

disaster impends, depending entirely upon it—and almost as much depending upon it as upon the motive power apparatus for high efficiency in local and suburban train operation—every effort is apparently concentrated upon attaining the maximum acceleration in starting trains and the almost equally important matter of high retardation in stopping is so far neglected that efficient foundation brake-gear is not only usually not provided but often prohibited, and especially where electricity is the motive power. It is obvious that the force with which the brake-shoes should be applied to a pair of wheels depends primarily upon the pressure of those wheels upon the rails. In car construction, the weight is usually, but not always, equally distributed upon the various wheels. In electric car construction, it frequently occurs that the motors are so applied that greater weight is supported by one pair of wheels than by another. In any case where the wheel pressure materially differs, proper regard for brake-efficiency demands that the brake-shoe pressure shall correspondingly differ. But, entirely independent of the normal distribution of weight upon the various wheels of a vehicle, retardation itself, caused by the application of brakes to the wheels, involves a redistribution of the weight which, where the retardation corresponds to the maximum brake application, inevitably results in a very serious loss of braking efficiency, unless means be provided for varying the brake-shoe pressure to correspond with the changed wheel-pressures. Whatever the source of the retarding force, its operative effect in retarding the motion of the car is the same as that of an infinite number of small retarding forces, each engaged in retarding the motion of an elementary portion of the mass of the car, and therefore, in order that a single retarding force, or a combination of retarding forces, shall so operate, without either changing the direction of the car's motion, or producing rotation of the structure as a whole, or calling into operation other forces to prevent such deviation or rotation, the force or the resultant of the combination of forces must be so applied that it shall pass through the centre of inertia of the mass, in a direction opposite to that of the motion of the car. In utilizing the rail-friction for the retarding force, while it has the proper direction, it is applied at the lowest points of the mass of the car and must therefore either cause rotation of the entire structure or the interposition of other forces which combine with the retarding force to preserve the simple motion of translation. The car may be considered as a single mass or as being

composed of three separate masses, according as it is provided with one rigid or two swiveling trucks. Rotation of the car by the eccentric retarding force does not occur in either case; but, in the first, it is because the reacting pressure of the rails upon the forward pair of wheels exceeds that upon the rear pair of wheels in such measure that the contrary rotative moment, thereby introduced, just balances that of the eccentric retarding force, also applied by the rails to the wheels. In the case of the car with two trucks, the retarding force is applied at the lower extremity of the two trucks, being equally divided between them (if constructed alike); that portion of the retarding force necessary to retard the mass of the trucks is absorbed in so doing, and the remainder is applied by the trucks to the car body at substantially its lowest extremity. In consequence, the center of inertia of the car body being above the points of application of the retarding force, rotation through the eccentrically applied retarding force is prevented only by the resisting rotative moment of a greater supporting pressure from the forward than from the rear truck. Each truck is subject to the combined rotative moment of the eccentric retarding force at its lower extremity, and the eccentric reacting force from the car body at its upper extremity, and rotation is prevented only by the contrary rotative moment of a greater supporting pressure by the rails upon the forward than upon the rear pair of wheels. Thus the very act of applying the brakes to the wheels produces a new and very different system of wheel pressures upon the rails, and it is the wheel pressures under these conditions which determine the available retarding force. As the total pressure of all the wheels upon the rails cannot vary, it is obvious that the existence of a greater rail pressure for the forward than for the rear pair of wheels of the truck implies the virtual transfer of a portion of the normal pressure from one pair of wheels to the other. The brake-shoe pressure upon the rear pair of wheels must be insufficient to cause the wheels to slide upon the rails, and must therefore be cut down in proportion to the transfer of weight from the rear to the forward pair of wheels. But as the forward pair of wheels will become the rear pair when the car moves in the opposite direction, the brake-shoe pressure upon that pair of wheels must also be limited in the same way. Thus, the braking pressure upon each pair of wheels must be restricted to correspond with the minimum pressure of the wheels upon the rails, and when it is understood that this minimum rail-pressure, which

occurs in the maximum application of the brakes, is less than 85 per cent. of the normal, or, in other words, that the effective wheel-pressure, available for braking, of an ordinary eight-wheeled passenger car is but 85 per cent. of the weight of the entire car, the importance of the provision of means to compensate for such a serious loss of retarding efficiency becomes clearly manifest, and a solution of the problem, devised by the writer, is therefore submitted.

The first step in determining the rail-pressure is to find the reactions between the car-body and trucks, after which, investigation is confined to truck-conditions alone. The only means of the application of force to the car-body are the trucks and the draw-bars. In any case where, in a train of vehicles, the ratio of the retarding force of the brakes to the weight is greater for some vehicles than for others, the forces transmitted through the draw-bars require consideration; but in electric service, where cars are either operated singly or in trains that permit equally effective brake-retardation upon the several vehicles, no one produces a retarding effect upon another, so that draw-bar forces are eliminated and the problem is more simple than in the case of steam-railroad trains, where draw-bar forces accompany brake-application, because all vehicles are not uniformly braked. It is true that, in a train of electrically operated cars, defective apparatus upon one or more of the cars might result in the application of the brakes upon only a portion of the cars, whereby retarding forces would be transmitted through the draw-bars to the unbraked cars, but consideration can hardly be extended to such uncertain accidental conditions.

Fig. 1 represents the body of a car, from which the trucks have been removed and replaced by arrows representing the supporting and retarding forces which they apply to the car. The trucks being alike and subject to the same retarding rail-friction, each truck contributes the same horizontal force (H) to retard the mass of the car-body, and the forward truck applies a vertical supporting pressure P_1 , while the rear truck similarly applies the supporting pressure P_2 . The weight (W_1) of the car body is represented by an arrow acting downward through the center of inertia of the car, midway between the truck supports and at distance k above them. The distance between the truck supports is represented by l . These are all the real forces acting upon the car-body (in the absence of draw-bar forces), as the resistance of the atmosphere is so insignificant in comparison with the rail-friction that it may be neglected.

FIG. 1.

The rail-friction causes all parts of the car-structure to be retarded at a certain rate (p) and the horizontal forces (H) from the trucks are absorbed in overcoming the resistances of inertia of all the various elements of mass of the car-body, which latter thus constitute a system of imaginary forces that may be represented in the aggregate by an equivalent horizontal force, which is the product of the mass of the car-body (W/g , where g represents the acceleration of gravity) and the common retardation (p), acting at the center of gravity in a direction opposite to that of the retarding forces H . The operation of the real forces must thus be equal and opposite to that of the imaginary forces, and the following conditions are therefore evident:

$$P_1 + P_2 - W_1 = 0, \text{ or } P_1 = W_1 - P_2$$

$$2H = \frac{W_1 p}{g}, \text{ or } H = \frac{W_1 p}{2g}$$

$$W_1 \frac{l}{2} - P_2 l = \frac{W_1 p}{g} k, \text{ or } P_2 = \frac{W_1}{2} - \frac{W_1 p k}{g l}$$

Also, combining the first and last of these equations,

$$P_1 = \frac{W_1}{2} + \frac{W_1 p k}{g l}$$

By comparison of these values of P_1 and P_2 , it is evident that, of the weight $W_1/2$, normally supported by each truck, the weight $p k W_1/g l$ has, through the application of the brakes, been removed from the rear truck and transferred to the forward truck. The center of gravity of car-bodies, especially in electric-car construction, where apparatus of considerable weight is carried below the floor, is generally low, so that the fraction k/l is usually not great, and in some cases its consideration may be neglected in what follows. It is clear, however, that, inasmuch as the rail-pressure upon the rear truck is that which limits the brake-shoe pressure upon either, we are concerned only with the

conditions relating to the rear truck, and any conclusions reached regarding the application of brakes to that truck, are also applicable to the forward truck.

Fig. 2 represents the rear truck, to which is applied the vertical car-body pressure P_2 , midway between the wheels, and the horizontal reaction H from the car-body, opposing the retarding force applied by the truck to the car body, at a height h above the rails. The rails have been removed and in their place the rail-pressures R_1 and R_2 and the corresponding rail-frictions T_1 and T_2 are indicated by arrows, as applied, respectively, to the forward and rear pairs of wheels. It may be noted, in passing, that, although each wheel has a separate rail-pressure and rail-friction, the pair

FIG. 2.

of wheels mounted upon the same axle are rigidly secured together, so that, if one turns or slides, the other must accompany it, and the simpler method of treating the pair as a single element, operating under the sum of their rail-pressures and brake-shoe frictions, is subject to no objection of any kind. The weight (W_2) of the truck is represented by an arrow, acting vertically downward through the center of inertia, at a point midway between the wheels and at a height d above the top of the rails. These constitute all the real forces acting upon the truck from without. (Although other forces, due to the brake-shoes and apparatus, act coordinately, they are altogether internal forces and therefore have no influence upon the relation of the truck to external masses.)

Each elementary portion of the mass of the truck is retarded and, through the resistance of its inertia, absorbs a portion of the retarding force, such resistances constituting a system of imagin-

ary forces equivalent to the horizontal force $W_2 p/g$, acting at the center of gravity, in a direction opposite to that of the retarding forces T_1 and T_2 . From the equality of the real and imaginary forces, the following relations are evident:

$$R_1 + R_2 - W_2 - P_2 = 0, \text{ or } R_1 = W_2 + P_2 - R_2$$

$$T_1 + T_2 - H = \frac{W_2 p}{g}, \text{ or } T_1 + T_2 = H + \frac{W_2 p}{g}$$

$$W_2 \frac{b}{2} + P_2 \frac{b}{2} - R_2 b - H h = \frac{W_2 p}{g} d, \text{ or}$$

$$R_2 = \frac{W_2}{2} + \frac{P_2}{2} - H \frac{h}{b} - \frac{W_2 p}{g} \frac{d}{b}$$

Also, combining the first and last of these equations,

$$R_1 = \frac{W_2}{2} + \frac{P_2}{2} + H \frac{h}{b} + \frac{W_2 p}{g} \frac{d}{b}$$

Giving H and P_2 the values already found, the last three of these equations become, respectively,

$$T_1 + T_2 = \frac{W_1 + 2 W_2 p}{2 g}$$

$$R_2 = \frac{W_1 + 2 W_2}{4} - \frac{W_1 p}{2 g} \left(\frac{h}{b} + \frac{k}{l} \right) - \frac{W_2 p}{g} \frac{d}{b}$$

$$R_1 = \frac{W_1 + 2 W_2}{4} + \frac{W_1 p}{2 g} \left(\frac{h}{b} - \frac{k}{l} \right) + \frac{W_2 p}{g} \frac{d}{b}$$

Here again, comparison of these values of R_1 and R_2 shows that, beside reducing the rail-pressure upon each pair of wheels of the rear truck, to the extent of $p k W_1 / 2 g l$, the effect of applying the brakes has also been to transfer from the rear to the forward pair of wheels the weight

$$\frac{1}{2} \frac{p}{g} \frac{h}{b} W_1 + \frac{p}{g} \frac{d}{b} W_2;$$

and, as the front pair becomes the rear pair of wheels upon reversing the direction of motion of the car, the available wheel-pressure for braking purposes has been reduced from

$$\frac{W_1 + 2 W_2}{4}$$

for each pair of wheels, to

$$\frac{W_1 + 2 W_2}{4} - \left\{ \frac{W_1}{2} \frac{p}{g} \left(\frac{h}{b} + \frac{k}{l} \right) + W_2 \frac{p}{g} \frac{d}{b} \right\}$$

From the first of the last three equations,

$$p = 2 \frac{T_1 + T_2}{W_1 + 2 W_2} g$$

For simplicity and greater convenience, $W_1 + 2 W_2$, the weight of the entire car, may be designated by W and thus

$$p = 2 \frac{T_1 + T_2}{W} g$$

Applying this value of p in the expressions for R_1 and R_2 ,

$$R_2 = \frac{W}{4} - \left\{ W_1 \left(\frac{h}{b} + \frac{k}{l} \right) + 2 W_2 \frac{d}{b} \right\} \frac{T_1 + T_2}{W}$$

$$R_1 = \frac{W}{4} + \left\{ W_1 \left(\frac{h}{b} - \frac{k}{l} \right) + 2 W_2 \frac{d}{b} \right\} \frac{T_1 + T_2}{W}$$

It is to be observed that these expressions for the rail-pressures of the rear and forward part of wheels of the rear truck are true for any rail-frictions, T_1 and T_2 , that may be available. If f_1 denotes the coefficient of static rail-friction, the maximum values of T_1 and T_2 are $f_1 R_1$ and $f_1 R_2$, respectively. If then for R_1 and R_2 , their corresponding values T_1/f_1 and T_2/f_1 , respectively, be substituted, the above equations may be solved for T_1 and T_2 , and their values found for the condition of maximum efficiency. They are

$$T_1 = \frac{f_1 W}{4} \frac{W b + 2 f_1 (W_1 h + 2 W_2 d)}{\left(W + 2 f_1 W_1 \frac{k}{l} \right) b} \quad (1)$$

$$T_2 = \frac{f_1 W}{4} \frac{W b - 2 f_1 (W_1 h + 2 W_2 d)}{\left(W + 2 f_1 W_1 \frac{k}{l} \right) b} \quad (2)$$

In order to understand now what results from this transfer of weight in practice, where it is customary to apply to each wheel the greatest brake-shoe pressure that will slide none of them, let $T_1 = T_2 = f_1 R_2$ be substituted in the value of R_2 ; then, upon solving the equation,

$$T_1 = T_2 = \frac{f_1 W}{4} \cdot \frac{W b}{W b + 2 f_1 \left[W_1 \left(h + \frac{k}{l} b \right) + 2 W_2 d \right]}$$

The average conditions for the passenger cars of steam-railroad service are fairly represented by the values $b = 84"$, $h = 34"$, $k = 32"$, $l = 480"$, $d = 21"$, $W_1 = \frac{2}{3} W$ and $W_2 = 1/6 W$. Assuming that, in safe average practice, $f_1 = .25$, it is thus found that, instead of a maximum rail-friction of $f_1 W/4$, for each pair of wheels, as ordinarily supposed, the maximum is but $.834 f_1 W/4$.

This serious loss of available retarding force induced the writer some time since to investigate the conditions for the purpose of finding, if possible, a means of automatically proportioning the brake-shoe pressure upon the forward and rear wheels of the truck, so that they should more nearly correspond to the rail-frictions of equations 1 and 2. Using the above values of b, h, k, l, d, W_1 and W_2 , in those equations, $T_1 = 1.151 f_1 W/4$ and $T_2 = .806 f_1 W/4$, so that $T_1 + T_2 = .978 f_1 W/2$ might be realized, while only $.834 f_1 W/2$ is ordinarily attained in practice. Any means of making the possible rail-friction of the forward pair of wheels fully available would thus increase the braking efficiency more than 17 per cent.

The investigation disclosed the fact that a structural feature of truck brake-gear that had been found to complicate the application of brake-shoe pressure to the wheels, might be so employed that it would serve almost entirely to recover this lost efficiency in braking. It is, in brief, the angularity of the hanger-link, by which, if the brake-shoes be applied upon the inner face of the wheel—that nearest the center of the truck—and the hanger-link supporting the brake-shoe be inclined at a proper angle with the tangent to the wheel at the center of the bearing surface of the brake-shoe, the brake-shoe pressure is proportioned to the wheel pressure. This matter merits very careful consideration.

More as a matter of convenience than for any other apparent reason, it has generally been customary, in passenger-car construction, to suspend the brake-shoes from the end-timbers of the truck at the outer face of the wheels. It is true that the brake-

shoes are thus more accessible for renewals, but the arrangement is inconvenient in other respects, requiring the disconnection and often the removal of the brake-beams to remove the wheels. The application of the brake-shoes at the outer face of the wheels results in an upward thrust of the brake-hangers, proportional to the brake-shoe friction, upon the end-timber at the rear end of the truck, and a corresponding downward drag upon that at the forward end. It has already been shown that the retardation of the car by the rail-friction produces a rotative effect upon the truck, which is greatly augmented by this direct action of the brake-shoe friction through the hanger-links, and the result is that a considerable rotation or tilting of the truck frame actually occurs, compressing the forward equalizing-bar springs and relaxing those at the rear. The reaction or recoil of these springs is the cause of the frequently observed violent backward surge or shock, so disagreeable to passengers and sometimes throwing unguarded standing persons to the floor, at the instant of stopping. If, however, the brake-shoes were suspended at the inner face of the wheels, the upward thrust of the hanger-links would act upon the forward portion of the truck-frame, and the downward thrust upon the rear portion, so that the effect would be to counteract and neutralize, instead of aggravate, the disagreeable influence of the rail-friction.

Another and still more serious objection to this method of suspending the brake-shoes is the evil effect of the angular inclination of the hanger-links, which is not only desirable to insure clearance of the shoes from the wheels when the brakes are released, but is usually unavoidable for constructive reasons. This feature will be better understood upon further consideration of the effect of inclining the brake-beam hanger-links.

Fig. 3 represents the outlines of the usual form of passenger-car truck, in which are indicated by arrows the forces applied to each brake-beam. These forces are not all in the same lateral plane, the actuating brake-force P from the brake-cylinder being applied at the center of the brake-beam, while one-half of each of the other three forces is applied to the brake-shoe at each end of the beam; but it is simpler and without objection to treat them as represented. The forces Q_1 and Q_2 are the reacting pressures of the forward and rear pairs of wheels, respectively, upon the brake-shoes; the forces F_1 and F_2 are the corresponding frictional forces resulting from the pressures Q_1 and Q_2 ; and the forces V_1 and V_2 are the reactions upon the brake-shoes from the hanger-

links. The center of the brake-shoes should be about three and one-half inches below the center of the wheels, and the angle between the radial direction of the pressures Q_1 and Q_2 and the horizontal, is

FIG. 3.

designated by α . The mean frictional forces F_1 and F_2 consequently incline at the same angle with the vertical. The inclination of the hanger-links to the tangential direction of the frictional forces F_1 and F_2 is designated by ϕ . The forces being resolved into rectangular components, with reference to the direction of the hanger links of each pair of wheels, the following relations are evident:

$$\begin{aligned} Q_1 \cos \phi - F_1 \sin \phi - P \cos (\alpha + \phi) &= 0 \\ Q_2 \cos \phi + F_2 \sin \phi - P \cos (\alpha + \phi) &= 0 \end{aligned}$$

If the coefficient of brake-shoe friction be designated by f_2 , then, whatever be its value, $F_1 = f_2 Q_1$ and $F_2 = f_2 Q_2$. Replacing Q_1 and Q_2 by their values in terms of F_1 and F_2 , these equations may be combined and solved for ϕ and P , with the results that

$$\tan \phi = \frac{1}{f_2} \frac{F_1 - F_2}{F_1 + F_2} \quad (3)$$

$$P = \frac{(F_2 + F_1) \cos \phi - f_2 (F_1 - F_2) \sin \phi}{2 f_2 \cos (\alpha + \phi)} \quad (4)$$

In retarding the motion of the vehicle, the friction of the brake-shoes must resist rotation of the wheels to the same extent that the rail friction urges continued rotation; but, in addition, it must also destroy the rotative energy of the wheels. That is, the stored energy due to the rotary motion of the wheels, which is

entirely independent of the energy due to the motion of translation (and which would still consume considerable work if the simple motion of translation of the entire vehicle, wheels and all, could be overcome by means not affecting the rotation of the wheels), absorbs a part of the work done by the brake-shoes and only what then remains of that work operates to excite rail-resistance to wheel sliding. The peripheral velocity of the wheels being the same as the velocity of translation, if r_1 represent the radius of the wheel, a retardation p of the vehicle is accompanied by an angular retardation p/r_1 of the wheel's rotation. If, also, w represent the weight of each wheel and r_2 the radius of gyration, then, when the rail-friction causes the retardation of the motion of the car the corresponding retardation of the rotary motion of the wheels requires at the surface of each wheel a retarding force equal to $w r_2^2 p / g r_1^2$. By careful computation, the square of the radius of gyration of car wheels is found to be almost exactly six-tenths of the square of the wheel-radius. Therefore, the brake-shoe friction being required to supply both retarding forces, the relation between the wheel-forces of the two pair of wheels is obviously

$$F_1 = \frac{1.2 w}{g} p + T_1$$

$$F_2 = \frac{1.2 w}{g} p + T_2$$

If now the maximum rail-friction which may exist with the unequal distribution of weight upon the different pairs of wheels, as found in equations 1 and 2, is to be realized, it is necessary that the brake-shoe frictions F_1 and F_2 shall correspond with those rail-frictions. Giving T_1 and T_2 the values of equations 1 and 2, respectively, and giving p the value found for it, these brake-shoe frictions become

$$F_1 = \frac{f_1 W}{4} \frac{(W + 4.8 w) b + 2 f_1 (W_1 h + 2 W_2 d)}{\left(W + 2 f_1 W_1 \frac{k}{l} \right) b}$$

$$F_2 = \frac{f_1 W}{4} \frac{(W + 4.8 w) b - 2 f_1 (W_1 h + 2 W_2 d)}{\left(W + 2 f_1 W_1 \frac{k}{l} \right) b}$$

Applying these values in equation 3, it becomes

$$\tan \varphi = 2 \frac{f_1}{f_2} \frac{W_1 h + 2 W_2 d}{(W + 4.8 w) b} \quad (5)$$

The expression for the braking force applied to the brake-beam may be given a more convenient form for practical work by giving $F_1 - F_2$ the value $f_2 (F + F_2) \tan \varphi$, desired from equation 3, and, finally,

$$P = \frac{f_1}{f_2} \frac{W}{4} \frac{W + 4.8 w}{W + 2 f_1 W_1 \frac{k}{l}} \frac{(1 - f_2^2 \tan^2 \varphi) \cos \varphi}{\cos (a + \varphi)} \quad (6)$$

Thus, by hanging the brake-beams between the wheels, instead of outside, and inclining the hanger-links at a proper angle, the increased pressure and consequently the increased friction of the brake-shoes upon the forward pair of wheels and the diminished pressure and friction of the brake-shoes upon the rear wheels, due to the effect of the friction itself in causing the shoes to press more or less forcibly upon the wheels through the angularity of the hanger-links, are made to correspond with and compensate for the transferred weight from the rear to the forward wheels. In the same manner that running in the opposite direction causes a reversal of the conditions for the transfer of weight, so, too, the rotation of the wheels in the opposite direction causes a reversal of the effect of the inclined hanger-links, and the increased brake-shoe pressure is always applied to the wheels carrying the increased weight.

It will now be more clearly understood why inclination of the hanger-links of outside-hung brake-beams has an evil effect upon the efficiency of the brakes. As the link must necessarily be so inclined, if inclined at all, that the upward movement of the wheel surface upon the brake-shoe increases the brake-shoe pressure, the pressure is *increased* upon the *rear* pair and *reduced* upon the *forward* pair of wheels when the brakes are outside-hung. Thus, the high brake-shoe pressure is always applied to the wheel carrying the reduced weight and the reduced brake-shoe pressure to the wheels carrying the increased weight. The braking force P upon the brake-beam must therefore be so reduced that the rear pair of wheels shall not be caused to slide, and the combined friction of the brake-shoes upon the two pair of wheels is thereby reduced accordingly. The loss of more than 15 per cent. in

braking efficiency, which has been shown to result from the use of a uniform brake-shoe pressure instead of pressures proportioned to the rail-pressures, occurs when the hanger-links are not inclined; but it will now be understood that, with outside-hung brakes having the ordinary inclination of hanger-links, the loss is considerably greater, the retardation probably averaging at least 20 per cent. below that attainable by the expedient of inside-hung brakes with properly inclined hangers.

In practice, the application of this method of inclined hanger-links is not without some difficulty. The chief trouble is that no constant angle of the links can be maintained, as the wearing away of the brake-shoes, together with wearing and turning down of the tires of steel-tired wheels, causes constant and considerable variation. Thus, if the angle of inclination and the braking pressure be calculated for the conditions existing when the brake-shoes and wheels are new, the increased angle when the shoes become much worn and the tires have been well turned off, would probably cause the forward wheels to slide upon the rails. On the other hand, if the calculations be made for turned wheels and worn shoes, the rear wheels would probably slide when the wheels and brake-shoes are new. It is therefore necessary to compromise between the extremes, in reference to the angle of inclination of the hanger-links, by inclining the hanger-links at the angle determined from equation 5, when the brake-shoes and wheels are each *half-worn*, and the brake-beam force must then be so established that neither pair of wheels shall be caused to slide in the extreme positions of the hanger-links. The brake-beam force determined by equation 6 would be too great for the forward pair of wheels when the brake-shoes and wheels are fully worn, and also too great for the rear pair of wheels when the brake-shoes and wheels are new. The maximum safe brake-beam force for the forward pair of wheels may be determined from the rail-pressures and the values of the angles α and φ when the brake-shoes and wheels are fully worn, and that for the rear pair from the corresponding conditions when the brake-shoes and wheels are new, and the least of the two is the greatest brake-beam force that can be continuously employed. It is found that the angular variation from the position of the hanger-link when the brake-shoes and wheels are half worn is a little less when the brake-shoes and wheels are new than when they are fully worn, and the brake-beam force to be employed in practice must be determined by the conditions existing when the brake-shoes and

wheels are fully worn. It is obvious that the variation of the angularity of the hanger-links, through the allowance necessary for wear ($2\frac{3}{4}$ inches for steel-tired and $1\frac{3}{4}$ inches for chilled cast-iron wheels), is an inverse function of the length of the hanger-link itself, which should therefore be as long as practicable. With the use of that form of brake-shoe holder or head in which the hanger-link pin is located behind the center of the brake-shoe (illustrated in Figs. 2 and 3, above), the maximum length of hanger is secured. The form of brake-head in which the pin is considerably above the center, shortens the hanger materially, and its use should be avoided.

Whether the maximum rail-frictions are developed by corresponding brake-shoe frictions or not, it is to be observed that the relation between the frictions F_1 and F_2 expressed in equation 3 is one that always exists, being dependent simply upon the angle of inclination of the hanger-links and the coefficient of brake-shoe friction and independent of everything else. Therefore, even though the angle of inclination varies with wear, and the construction of the truck may be such that the full inclination of the hanger, necessary to the greatest efficiency, cannot be employed, a portion of the added efficiency of the system may still be realized by inclining the hanger-links as much as may be consistent with proper length and the limitations of the truck construction, and by employing a corresponding brake-beam force. As the inclination of the hanger-links is insufficient to compensate fully for the reduced rail pressure upon the rear pair of wheels when the brake-shoes and wheels are half worn, the brake-beam force must in this case be determined with reference to that pair of wheels when the brake-shoes and wheels are new.

The problem thus resolves itself into so laying out the truck brake-gear that the center of the half-worn brake-shoe shall be at the proper distance ($3\frac{1}{2}$ inches) below the center of the half-worn wheel, and that the hanger-link (as long as possible) shall stand at as great an angle to the wheel-tangent as possible (not exceeding the angle of full compensation, where it has the value found from equation 5 *when the brake-shoes and wheels are each half worn*, and then calculating the brake-beam force P for the angles φ_1 and α_1 that exist *when the brake shoes and wheels are fully worn*, if the angle of full compensation has been employed, or for the angles φ_2 and α_2 , that exist *when the brake-shoes and wheels are new*, if an angle less than that of full compensation has been employed. To facilitate the application of the principle of

inclined hanger-links, the various angles of inclination for half-worn brake-shoes and wheels, and the corresponding braking forces have been determined for different conditions, as to truck wheel-base and diameter of wheel, and are tabulated below.

It will be observed in equation 5 that the angle of inclination of the hanger-links varies inversely with the wheel-base of the truck, which is fortunate, since long wheel base is desirable also in all other respects (ease of riding, minimum wear and tear of wheel-flanges and truck-frames, etc.), except in cases where curves are so sharp that long wheel base interferes with curving. The full compensating inclination of long hangers may be employed, or any size of wheel, where trucks have a wheel-base of 8 feet; and, for wheels of 33 and 36 inches diameter, it may also generally be employed where the wheel-base is $7\frac{1}{2}$ feet. With passenger-car trucks of ordinary construction, partial compensation may be obtained, for wheels of larger diameter than 36 inches, where the wheel-base is $7\frac{1}{2}$ feet; and, for 33 and 36 inch wheels, it may also be obtained where the wheel-base is but 7 feet. Shorter wheel-base than 7 feet, while requiring a large angle of inclination of hanger-links, provide insufficient room for an inclination of any value, unless the hanger-links are so short that the variation of the angle impairs its utility, or the trucks are low, so that the height h is also reduced as the wheel-base b is reduced.

The method of applying the principle, in cases where the full compensating angle of inclination can be used, depends primarily upon the relations expressed in equation 3. As the braking force must be so established that the forward pair of wheels shall not slide when the wheels and brake-shoes are fully worn, the friction of the brake-shoes upon that pair of wheels must correspond with the rail-pressure at that time, and the friction of the brake-shoe upon the rear pair of wheels will necessarily be

$$F_2 = \frac{1 - f_2 \tan \varphi_1}{1 + f_2 \tan \varphi_1} F_1$$

as found from equation 3, in which φ_1 is the value of the angle of inclination the hanger-link to the wheel tangent when the brake-shoes and wheels are fully worn. It has been shown that

$$F = T_1 + \frac{2.4 w}{W} (T_1 + T_2)$$

and that

$$F_2 = T_2 + \frac{2.4 w}{W} (T_1 + T_2)$$

and, by combining these two equations,

$$T_1 + T_2 = \frac{W}{W + 4.8 w} (F_1 + F_2)$$

and

$$T_1 = \frac{(W + 2.4 w) F_1 - 2.4 w F_2}{W + 4.8 w}$$

The rail-pressure for the forward pair of wheels was found to be

$$R_1 = \frac{W}{4} + \left\{ \frac{W_1}{W} \left(\frac{h}{b} - \frac{k}{l} \right) + \frac{2 W_2 d}{W^2 b} \right\} (T_1 + T_2)$$

If the full rail-friction of this pair of wheels be utilized, $R_1 = T_1/f_1$ and the above values of T_1 and $T_1 + T_2$ being substituted in this equation for the rail-pressure, it may be solved, with the aid of the above value of F_2 in terms of F_1 and $\tan \varphi_1$, to yield values of F_1 and F_2 , which depend simply upon the available rail-pressure of the forward pair of wheels when the brake shoes and wheels are fully worn. These values of F_1 and F_2 being then applied in equation 4 and representing the value of the angle α when the wheels and brake-shoes are fully worn by α_1 the necessary braking force to be applied at each brake-beam becomes

$$\varphi_1 = \frac{f_1}{f_2} \frac{(W + 4.8 w) (1 - f_2^2 \tan^2 \varphi_1) \cos \varphi_1}{\left\{ W + 2 f_1 W_1 \frac{k}{l} - 2 f_1 \frac{W_1 h + 2 W_2 d}{b} + f_2 (W + 4.8 w) \tan \varphi_1 \right\} \cos (\alpha_1 + \varphi_1)} \frac{W}{4}$$

In practice, the application of this brake-beam force by the air-pressure in the brake-cylinder is through the medium of a series of levers and connecting rods, pinned together and forming the "brake-gear." A portion of the air-pressure upon the piston of the brake-cylinder is absorbed by the frictional rigidity of the brake-gear; and, as the leverage of the brake-gear is always calculated from the piston-pressure, it is necessary to make proper allowance for the frictional loss in the brake-gear. The only available results of investigation of such loss indicates that, in the ordinary form of passenger-car brake-gear, the brake-beam force is about 80 per cent. of that calculated from the piston-pressure.

As the total braking force of the car is the sum of the forces upon the four brake-beams, if the calculated braking force of the car be represented by B , it therefore becomes $B = 4 P / .80$.

Examination of the practice in passenger-car construction of ordinary American steam railroads indicates quite general uniformity in respect to most of the various features which determine the angle of inclination of the brake hanger-link and the braking force. In general, for practical purposes, the average of the conditions existing is sufficiently accurate and may be taken as follows: The coefficient of rail-friction may be regarded as $f_1 = .25$, particularly as the limiting conditions are those of emergency applications. The coefficient of brake-shoe friction most apt to slide wheels is that of low speeds, near the end of the stop, and experience indicates a safe value to be $f_2 = \frac{1}{3}$. Little error will, in any ordinary case, result from making $W_1 = \frac{2}{3} W$ and $W_2 = 1/6 W$. Generally, $h = 34$ inches, and $k/l = 1/15$. The height of the center of gravity of the truck varies with the diameter of the wheels; for 33-inch wheels, $d = 20$ inches; for 36-inch wheels, $d = 20.5$; for 38-inch wheels, $d = 21$; for 40-inch wheels, $d = 21.25$; and for 42-inch wheels, $d = 21.5$; the weights of wheels differ both with the diameter and the construction; a fair average appears to make $4.8 w = .071 W$ for 33-inch steel-tired wheels, $.080 W$ for 36-inch, $.089 W$ for 38-inch, $.094 W$ for 40-inch, and $.098 W$ for 42-inch. With these values (except those depending upon the diameter of the wheels), the maximum braking force, as ordinarily calculated, which may be used without injurious wheel-sliding, becomes

$$B = \frac{15 \left(1 + \frac{4.8w}{W} \right) \left(1 - \frac{1}{9} \tan^2 \varphi_1 \right) \cos \varphi_1}{16 \left\{ \frac{46}{45} \frac{68+d}{6b} + \frac{1}{3} \left(1 + \frac{4.8w}{W} \right) \tan \varphi_1 \right\} \cos (\alpha_1 + \varphi_1)} W$$

In cases where an angle of inclination of less than that required full compensation must be given the hanger-links when the brake-shoes and wheels are half worn, the same mode of procedure is followed to determine the braking force, except that the angles α_2 and φ_2 , existing when the brake-shoes and wheels are new, are employed instead of α_1 and φ_1 , and F_1 is found in terms of F_2 and φ_2 from equation 3, instead of F_2 in terms of F_1 and φ_1 . Thus T_2 and $T_1 + T_2$ are found in terms of F_1 and substituted in the expression for the rail-friction upon the rear pair of wheels, which yields values of F_1 and F_2 in terms of

simply the rail-pressure upon that pair of wheels and the angle φ_2 . These values of F_1 and F_2 , being substituted in equation 4, furnish the brake-beam force

$$P_2 = \frac{f_1}{f_2} \frac{(W + 4.8w)(1 - \tan^2 \varphi_2) \cos \varphi_2}{\left\{ W + 2f_1 W_1 \frac{k}{l} + 2f_1 \frac{W_1 h + 2W_2 d}{b} - f_2(W + 4.8w) \tan \varphi \right\} \cos(a_2 + \varphi_2)} \frac{W}{4}$$

From this brake-beam force, in terms of the rail-pressure of the rear pair of wheels and the angles a_2 and φ_2 , existing when the brake-shoes and wheels are new, the corresponding braking force, as customarily calculated from the pressure upon the brake-cylinder piston, is found. When the approximate average values of the various fixed symbols are applied, it becomes

$$B_2 = \frac{15 \left(1 + \frac{4.8w}{W} \right) \left(1 - \frac{1}{9} \tan^2 \varphi_2 \right) \cos \varphi_2}{16 \left\{ \frac{46}{45} + \frac{68+d}{6b} - \frac{1}{3} \left(1 + \frac{4.8w}{W} \right) \tan \varphi_2 \right\} \cos(a_2 + \varphi_2)} W$$

For convenience in applying this principle to the design of truck brake-gear, the following table has been prepared, showing the angle of compensation which the hanger-link makes with the vertical—that is, $a + \varphi$ —instead of the angle φ with the tangent to the wheel, when the brake-shoe and wheel are half worn. The angle is determined by placing the center of the half-worn shoe (assumed to be when the thickness has been reduced $\frac{1}{2}$ inch), $3\frac{1}{2}$ inches below the center of the half-worn wheel, the radius of which is assumed to have been reduced $\frac{7}{8}$ inch. For convenience, also, the tangent of the angle $a + \varphi$ is given, so that, a convenient definite distance having been laid off from the center of the lower hanger-link pin upon a vertical line through it, a horizontal distance from its upper end, equal to the product of this tangent and the vertical distance so laid off, will determine a point in the hanger-link, and consequently its direction. The upper end will of course be fixed by the position in which the upper pin can be supported by a bracket from the truck-transom, the length, however, being as great as possible. The braking force is also given, as determined from the angles a_1 and φ_1 when the brake-shoe has been reduced 1 inch in thickness and the radius of the wheel has been reduced $1\frac{3}{4}$ inches, or from the angles a_2 and φ_2 when the brake-shoes and wheels are new, according as the angle of inclination of the hanger-links when the brake-shoes and wheels are half worn is that of full compensation or a lesser one. The length of

each hanger-link employed in these computations is also given. As already stated, only partially compensating angles of inclination occur for a wheel-base of 84 inches and, with 38, 40 and 42 inch wheels, for a wheel-base of 90 inches. In the other cases, full compensation occurs.

Diam. of wheel.	$b = 84 \text{ in.}$				$b = 90 \text{ in.}$				$b = 96 \text{ in.}$			
	l	$\alpha + \phi$	$\tan \alpha + \phi$	B	l	$\alpha + \phi$	$\tan \alpha + \phi$	B	l	$\alpha + \phi$	$\tan \alpha + \phi$	B
33"	19½	28°-33½'	.5444	.956W	20	37°-28½'	.7668	1.025W	20	36°-6½'	.7296	1.028W
36"	18½	24-29½'	.4555	.929W	18½	36°-16½'	.7338	1.016W	20	34°-54½'	.6978	1.026W
38"					20	29°-10½'	.5582	.990W	20	34°-10'	.6787	1.024W
40"					19½	26°-37'	.5011	.972W	20	33°-33½'	.6635	1.022W
42"					19	23°-58½'	.4447	.954W	19½	33°-1½'	.6499	1.020W

NOTE:— b is the wheel base of the truck, in inches; B is the braking force, calculated from the brake-cylind. ter piston pressure, in terms of the weight of the car; and l is the length of the brake-beam hanger link, in inches. The evil effect of shortening the hanger link is illustrated in this table. For a wheel base of 96 inches, the braking force for trucks with 36-inch wheels is less than that for trucks with 33-inch wheels by only .002W, where the angle of inclination is that of full compensation and the hanger links are 20 inches long in each case; but, where the wheel base is 90 inches, the length of hanger link may, in many cases, be no longer than 18½ inches for trucks with 36-inch wheels, and the braking force in that case is .009W less than that of trucks with 33-inch wheels and 20-inch hanger links. Had 20-inch hanger links been employed, instead of 18½ inch, for 36-inch wheels, the braking force could probably be increased about .007W above that given (1.016W).

It would be a serious oversight to dismiss this subject of truck brake-gear construction without mentioning the pernicious break-beam release spring. The custom of hanging brake-beams from the end-timbers of passenger-car trucks has been attended—in the many cases where the inclination of the hanger-links is insufficient to cause the brake-beams to fall away from the wheels by gravity—by the necessary use of springs to insure clearance between brake-shoes and wheels when the brakes are not applied. The loss of brake-shoe pressure from the use of such springs might, of course, be readily compensated by increasing the braking force correspondingly, if such loss could be determined. But these springs vary to such an extent, even when made apparently alike and applied in the same way, that allowance for their influence is well-nigh impossible. Springs, so applied to trucks that they should keep the shoes uniformly away from the wheels, are found to operate so unevenly that, to prevent the brake-shoe at one end of a beam from dragging upon the wheel, that at the other end must be permitted to stand off so far from its wheel that excessive travel of the brake-cylinder piston is necessary to apply the brakes, whereby the air-pressure in the brake-cylinder

is reduced and the efficiency of the brakes correspondingly impaired. Even if these springs were so constructed and applied that they exert a uniform influence upon the brake-beams, the inequalities of brake-shoe material cause them to wear unevenly, so that a new shoe at one end of the brake beam is often accompanied by a considerably worn shoe at the other end, and adjustment of clearance that will avoid excessive piston-travel at the brake-cylinder is impossible. The impossibility of adequately measuring and providing braking force for the resistance of these springs, added to the loss of efficiency from excessive piston-travel, or the alternative trouble from dragging brakes, renders the brake-beam release spring one of the most serious evils of modern brake-practice. By the use of inside-hung brake-beams, where sufficient inclination of the hanger-links insures brake-shoe clearance through the action of gravity, both the expense and trouble due to the release-spring is avoided.

To enter into the detail of the air-brake apparatus employed to furnish the braking force, in a paper of this character, would unduly extend it and would also be a work of supererogation. The compressed-air supply generally implies a suitable compressor upon the car, or, if operated in trains, one or more upon each train. Storage of the compressed air in sufficient quantity has, however, been satisfactorily accomplished in some cases and possesses certain advantages. The air is usually stored at a comparatively high pressure (generally 150 pounds) in large reservoirs secured beneath the car, or in any other convenient place. It is delivered through a reducing valve into the "main reservoir" of brake-operation, at the desired pressure, where it is handled in the ordinary manner. In such a system, a single air compressor, of large capacity and high efficiency, compresses the air at a station, where it is stored and charged into the car storage-reservoirs from time to time. The advantages lie in avoiding the cost of installing and maintaining compressors upon all the cars, and in cheapness of operation. The disadvantages consist of the bulkiness of the storage-reservoirs and the time required to stop and charge them, and also the limited distance that may be traversed during the intervals. Where the air is compressed upon the car, the compressor must be accessibly constructed and placed upon the car, and supplied with clean, dry air. It may be operated by steam, by a separate electric motor, or by the car-motor, through suitable connection with the car-axle, as circumstances render it expedient. Its

operation should be so controlled by a governor that it shall cease whenever the maximum storage-pressure has been attained in the main reservoir, and shall be renewed when operation of the brakes has reduced the storage-pressure to the inferior limit.

Upon the motorman's operating valve, the satisfactory operation of the brake-system in large measure depends. It must not only present the means of accurately gauging the force of brake-application and of promptly releasing the brakes, but must also define, with precision, the pressure of the air with which the auxiliary reservoirs are charged, to insure the full efficiency of braking without exceeding it to the injury of wheels and detriment of efficiency; while, at the same time, it must provide a superior pressure—that may vary considerably under different conditions—in the main storage reservoir, to insure prompt release of the brakes and restoration of pressure in the auxiliary reservoirs, without any variation of the working pressure in the latter—an exacting combination of conditions, not easy of realization, but of capital importance.

Of the apparatus for the immediate application of the brakes to the wheels, sufficient has already been said, it having been indicated that, in the single case where the unit invariably consists of a single car, simply an air-cylinder, in communication with the motorman's valve, meets all the requirements, while the conditions of every other case justify nothing short of the efficiency of the quick-acting automatic apparatus, and, where characterized by high speeds and frequent stops, the superior efficiency of the high-speed brake is essential to high efficiency of service.

The application of other forms of power than compressed air to brake-service has been practically limited to the vacuum and electricity. The limited pressure and bulky apparatus have restricted the use of the vacuum to comparatively light vehicles and it is fast becoming a mere historical feature of the development of the art of braking. Electricity has been experimentally applied in various forms of apparatus, but has only recently become recognized as a means of promising utility in practical braking. The simplicity of employing the back torque of the car motors of electric railroads for retarding purposes has appeared very attractive to those unacquainted with the objection to dependence upon that means alone. In combination with other means of retardation, so that excessive

heating may be avoided, this means of braking cars has been used with some practical success. But the application of electricity to the purpose of braking that appears to overshadow all others is that of the magnetic brake, which embodies such novel applications of old devices, with results so phenomenal, that the use of electricity as the source of braking force in electric railroad service at once occupies an interesting position with a very promising future.

The magnetic brake forms part of a combined braking and car-heating system, in the latter of which, the diverters of the ordinary car-starting apparatus are so also used as resistances for grading the current employed for braking purposes that they serve to heat the car. This part of the system is itself of such interest that, while it is a separate purpose, a brief description is excusable.

It appears to be a well-established fact that the heating effect due to the use of the diverters in starting is not alone sufficient for the practical purpose of heating the car, under the usual varying conditions of service; but when, in addition, the heating effect accompanying braking is also diverted to that purpose, the heat supply is abundant. Two systems of diverters are therefore used, one beneath the car, radiating the heat into the external atmosphere, and the other suitably arranged within the car for heating purposes and called heaters. The two systems may be so combined, according to the temperature and weather that any desired portion of the generated heat, from none of it to all of it, may be employed for heating the car, and any remaining heat is dissipated. The heat supply is, of course, intermittent, but the heaters are so constructed that they readily absorb the intermittently generated heat, and gradually and continuously supply it to the car for uniform heating. In this manner, the cost of the trolley current ordinarily required for heating purposes—an item of no mean proportions—is entirely obviated.

The brake apparatus itself consists of a combination of a track-brake with the ordinary wheel-brake. Track-brakes have long been known and variously proposed for practical use, but no satisfactory application of these devices appears to have been made. The track-brake consists of a system of brake-shoes, arranged beneath the car, directly over the rails, which may be forced down upon the rails to produce the retarding friction. There are several serious obstacles to the

attainment of satisfactory efficiency from this method of retardation. In the first place, as the weight of the car must be lifted off from the wheels and transferred to the track-brake shoes, it is obvious that the shoes must be so formed that they will guide the car upon the rails as safely and surely as do the wheels. This appears to be impracticable and, where the track-brake has been employed, only so much of the weight of the car is transferred to the brake-shoes that there still rests upon the wheels sufficient weight to enable them to guide the car securely. Thus only a portion of the weight of the car is available for rail-friction.

Again, it is manifest that the friction of the track-brake-shoes is of the same dynamic character as that of the brake-shoes upon wheels, which varies inversely with the speed and is very much inferior to the static friction of the rotating wheels upon the rails, which is the measure of the available friction of brake-shoes upon the wheels. While it is undoubtedly true that the large area of contact between the rails and track-brake-shoes that may be employed results in a track brake friction which is much superior to that of the sliding wheels (provided the entire weight is transferred to the track-brake-shoes), it is doubtful if the conditions permit the realization of as high a coefficient of friction where brake-shoes rub upon the track as where they are applied with the same force to the wheels, and it is certain that a track-brake of any construction, working upon this principle, is much inferior to an efficient wheel-brake of the "high-speed" type. The combination of a track-brake of this kind with a wheel-brake is merely an added intricacy, as the efficiency of the track brake is entirely at the expense of the wheel brake, in robbing the latter of the rail-friction upon which its effectiveness depends. For these reasons, and for the additional one of constructive complication and cost, no practical application of the track-brake has ever been made, within the knowledge of the writer, except in certain cases of street railroads with exceptionally heavy grades, where an independent track-brake system has been installed for emergency use only, in the event that the wheel-brake system should become disabled.

But the magnetic brake introduces an entirely new element, through which the track-brake not only does not interfere with the efficiency of the wheel-brake but both adds to its normal retardation and provides the operating brake-force. The track-

brake-shoe is placed between the two pairs of wheels, and, instead of being forced upon the rails through an effort from the car, is drawn to the rails by an electromagnet suspended from the car, thereby not merely adding its friction to the unimpaired friction of the wheel-brake but also actually increasing the rail-pressure of the wheels to the extent that the supporting springs for the track-shoes and magnets are in tension through the descent of the track-shoes to the rails. The construction will be better understood by reference to the illustration in Fig. 4.

FIG. 4.

In Fig. 4, the magnetic brake apparatus is shown as applied to a four-wheeled, swiveling truck, the parts of which are in dotted lines, more readily to distinguish the brake apparatus. The electromagnet *a*, dividing the track-brake shoe *b* into two parts, is secured by pins to the two push-rods *c*, and suspended at a proper distance above the rails by the adjustable springs *h*. The push-rods are secured by pins to the lower ends of the brake lever *d*, which are connected at their upper ends by the adjustable rod *g* and are pivoted at an intermediate point to the brake-shoe holders *e*, carrying the wheel-brake shoes, and the hanger-links *f*, suspended from the truck-frame. The push-rods *c* are telescopic, as shown in the sectional view of the one at the left, so that a movement of the track-shoe toward the right, relative to the truck-frame, causes the wheel-brake-shoe at the right to be applied to the wheel and the connection *g* to be moved to the left, thereby applying the wheel brake-shoe at the left, the stop *i* preventing the lower end of the brake lever at the left from following the track-brake

shoe. A relative movement of the track-brake-shoe to the left is obviously accompanied by application of the wheel-brake-shoes through corresponding movement of the parts in the reverse order.

The brake-controlling device may be incorporated in the running controller or may be a separate device, placed by its side and operatively interlocked with it, so that neither can, through carelessness, be caused to interfere with the operation of the other. In the operation of the apparatus, the current is supplied by the motors, running in multiple as generators—the trolley current being entirely cut off, and is divided between the electromagnets and the diverter combination in such ratio as to cause the track-brake-shoes to be drawn upon the rails with a force proportionate to the braking requirements. The frictional resistance of the rails to the motion of the track-shoes causes the wheel-brakes to be applied with corresponding force. Thus, to the ordinary retardation of the wheel-brakes is added that of the track-brake and also the back torque of the motors, which latter, however, is practically limited to compensation for the rotative energy of the motor and car-wheels. The force of application depends primarily upon the current and upon the electromagnets operating the brake-shoes. The attractive force of the rails upon the magnets is under the control of the motorman up to a limit of about 150 pounds per square inch of brake-shoe surface in contact with the rails. The strength of the magnet is limited by the sectional area of the rail, acting as armature, and where the weight of the car makes a magnet of greater strength desirable, the track-shoe is divided into three parts, instead of two, and wound to form a three-pole magnet, or two combined two-pole electromagnets with one common pole.

The friction of the track-brake-shoe may also be adjusted to some extent through the angular inclination of the push-rods *c*, by which some of the weight of the car may be thrown upon the track-shoes, the levers *d* being correspondingly adjusted to reduce the wheel-brake-shoe pressure in proportion as the weight is transferred to the track-shoe. It will be observed also that the brake-shoes are hung at the inside of the wheels, so that the hanger-links may have an angular inclination to correspond with the transfer of weight from the rear to the forward pair of wheels through the influence of the retarding forces.

The incidental features of the magnetic brake are also of interest and great value. The current declines with the speed

during a stop, thereby offsetting the increased coefficient of friction at the lower speeds. In bad weather, when the condition of the rails is liable to be accompanied by wheel-sliding, the braking force operating the wheel-brake is correspondingly reduced, so that the force of application of the wheel-brake is automatically proportioned to the rail-friction which rotates the wheels. But, in addition to this valuable feature, if by chance the wheels should slide upon the rails, the interruption of wheel-rotation is accompanied by the extinguishment of the track-magnet current, through which the pressure of the brake-shoes upon the wheel is instantly relaxed and rotation of the wheels is resumed, without injury or serious loss of time.

In operation, the efficiency of the magnetic brake, under the various conditions of service, is phenomenal and far exceeds that of any other braking system for the retardation of trains, which, combined with the commercial advantage of dispensing with the use of trolley current or any other means for heating cars, seems to assure it a leading position in electric-railroad operation in the future.