BRAKE PERFORMANCE
ON
MODERN STEAM RAILROAD PASSENGER TRAINS

BRAKE PERFORMANCE ON MODERN STEAM RAILROAD PASSENGER TRAINS

A DISCUSSION OF THE RESULTS OF THE PENNSYLVANIA RAILROAD BRAKE TESTS, 1913

By S. W. Dudley

SYNOPSIS OF PAPER

A train of 12 steel passenger cars and modern locomotive weighs nearly 1000 tons, is about 1000 ft. long and, at 60 m.p.h. speed has a kinetic energy of 224,000,000 ft.-lb.

With the ordinary high-speed brake apparatus such a train would be stopped by an emergency application of the brakes in a distance of from 1600 to 1800 ft. according to the truck rigging and brake shoe design and installation.

In making ordinary brake applications for slow-downs or station stops, skill and judgment must be carefully exercised in order to avoid shocks and make short and accurate stops.

The Pennsylvania Railroad brake tests of 1913 showed that such a train at 60 m.p.h. speed can be stopped by an emergency application in 1000 ft. or within the length of the train. They also showed that trains can be controlled by service applications without shocks at any speeds and with greater accuracy and promptness and still require less expert knowledge and skill on the part of the manipulator.

The improvement in emergency stopping power has resulted from applying the air brakes more quickly and to a higher pressure, holding this higher pressure without diminution toward the end of the stop, using a more efficient design and better installation of foundation brake rigging and providing a better method of applying the brake shoe to the wheel and more brake shoe metal to absorb the heat developed during the process of stopping.

The greater efficiency, economy and flexibility in service has been attained by making the air brake apparatus more positive and responsive in its operation, both in application and release, enabling full advantage to be taken of all the possibilities of these improvements through the quick, simultaneous and flexible action obtainable only with electric control, the maintenance of a high and uniform brake rigging efficiency, and the improved truck, journal and brake shoe action, less wear of brake shoes and better distribution of forces and reactions accompanying the use of the clasp brake having two shoes per wheel, instead of concentrating the heavy braking forces required by modern equipment on only one side of the wheel.

The tests constitute a scientific study of the brake as a whole; comparing in detail the characteristics of the ordinary high-speed air brake apparatus with the improved electro-pneumatic brake, the effect of low and high emergency braking powers, the clasp with the single shoe type of brake rigging,

the relative advantages of one and of two brake shoes per wheel and investigating the limitations of track and operating conditions commonly experienced.

The improved air brake apparatus, operating pneumatically, shortens the time of obtaining maximum emergency brake cylinder pressure on the train as a whole from 8 seconds, with the PM equipment, to 3.5 seconds and with electric control this is again shortened to 2.25 seconds. Moreover, 125%, 150% or 180% emergency braking power is available as may be thought desirable or found permissible according to circumstances when the installation is made. The PM equipment has an average of 100%.

Using 150% emergency braking power, the quicker and more powerful pneumatic emergency application shortened the stop at 60 m.p.h. from over 1600 ft. to about 1400 ft. and the simultaneous action of the electro-pneumatic brake still further shortened the stop to less than 1200 ft.

With the PM equipment the attempt to make an emergency application during the progress of or before releasing a partial or full service application will produce only the same stop as if merely a full service application had been made. Considering the ordinary full service stop from 60 m.p.h. (say 2000 or 2200 ft.) as 100%; with the improved apparatus, operating pneumatically, an emergency application following a partial service application will shorten about 14% and after a full service application about 10%; with electro-pneumatic operation the gain is 23% and 15% respectively. This is a safety factor of great importance.

Shocks during brake applications are due to slack action modified by speed. This was shown by pneumatic and electro-pneumatic stops from both high and low speeds. At high speeds, 60 to 80 m.p.h. the serial action of the pneumatic emergency application resulted in noticeable shocks which increased in severity at lower speeds and at 10 m.p.h. amounted, in effect, to a collision between the rear and forward end of the train, the train being stopped in 42 ft. The simultaneous application of just as great retarding forces by the electro-pneumatic brake entirely eliminated violent slack action at all speeds, the stop even at 10 m.p.h. (37 ft.) being without shock.

Station stops and emergency stops with the old and the new air brake apparatus mixed in various ways in the train showed harmony in operation and improved general results, particularly when releasing brakes.

Comparative tests showed that when the improved air brake mechanism is used on the cars, an arrangement giving a high emergency braking power on the locomotive, with blow-down feature, reduces shocks due to unequal braking power and shortens the stops.

With the electro-pneumatic brake a uniform increase in maximum per cent braking power results in a substantially uniform decrease in length of stop, the decrease amounting to about 2% for each 5% increase in braking power within the range of these tests.

The available rail adhesion varies through wide limits e.g., from 15% in the case of a frosty rail early in the morning to 30% for a clean dry rail at mid-day.

Wheel sliding depends more on the rail and weather conditions than on the per cent braking power. Some sliding was experienced with braking powers as low as 90% when rail conditions were unfavorable, but 180% braking power did not cause wheel sliding with a good rail.

The effect of excessive wheel sliding was to make the length of the stop about 12% greater than similar stops without wheel sliding.

An expression for the relation between coefficient of rail friction f_r, efficiency of brake rigging e, and per cent braking power P at the instant wheel sliding is about to occur derived from the data of the best tests made, is $P^{0.587} = \frac{8.33 f_r}{e}$

An efficient design of clasp brake rigging was shown to be necessary in order to avoid serious losses due to excessive piston travel, undesirable journal and truck reactions, brake shoe wear and variable shoe action.

The performance of the cast-iron brake shoe was shown to vary between wide limits, the ordinarily assigned causes for this variation (such as speed, pressure and time of action) becoming effective chiefly as they affect the temperature of the working metal of the brake shoe and wheel.

The amount of wheel and shoe metal in working contact during a stop is very small and is the most difficult factor to control. At the same time it is most potent in producing variations in brake performance.

The brake shoe bearing area and consequently the generation of heat in the working metal and the resultant coefficient of friction was shown to vary considerably during the progress of a single stop and to a greater degree as the fit of the shoe to the wheel changed due to warping or continued rubbing. Due to this effect alone the emergency stopping distance at 60 m.p.h. changed by as much as 20%.

This is evidence, however, that with reasonable attention to brake shoe maintenance the condition of the shoes on cars in ordinary road service is likely to be more favorable to making short emergency stops than during a series of tests in which the brake shoes are worked severely.

Flanged shoes provide more available area for bearing than unflanged shoes and, when worn in, shortened the train stops about 12%.

The use of two shoes instead of one per wheel will result in a higher coefficient of friction and less wear per unit of work done, the durability under clasp brake conditions being about 40% greater than under single shoe conditions.

The use of two shoes per wheel permits a design of rigging which will enable flanged shoes to be used without danger of pinching flanges and causing excessive flange wear or non-uniform brake forces which result when flanged shoes are used with rigid beam connections.

The wear of flanged shoes is about 20% less than unflanged shoes.

From tests at 60 m.p.h. with clasp brake rigging, the relation between coefficient of brake shoe friction f_s and per cent braking power P, was found to be $f_s = \frac{0.12}{P^{0.419}}$. No satisfactory separate determination of brake rigging efficiency e, and coefficient of brake shoe friction f_s, was made during the road tests. The combined effect $e \times f_s$, was obtainable with reasonable accuracy from the best of the several stops made under various conditions as given in the following table. The values for single shoe conditions are more uncertain than the clasp brake conditions due to less satisfactory data.

TABLE 10. VALUES OF $e \times f_s$

Kind of Brake Rigging		Clasp Brake		Single Shoe[1]	
Type of Brake Shoe		Plain	Flanged	Plain	Flanged
Speed m.p.h.	% Braking Power				
30	125	0.141	0.169	0.108	0.112
	150	0.129	0.154	0.099	0.103
	180	0.118	0.141	0.090	0.094
60	125	0.103	0.122	0.074	0.090
	150	0.094	0.112	0.068	0.082
	180	0.086	0.102	0.062	0.075
80	125	0.092	0.109	0.070	0.074
	150	0.084	0.100	0.064	0.068
	180	0.077	0.092	0.059	0.062

[1] Value of data uncertain due to non-uniform brake shoe conditions.

The length of emergency stop S_t, on a straight level track, neglecting air and internal friction on the one hand and the rotative energy of the wheels and axles on the other hand, can be calculated from the following formula:

$$S_t = 1.467\ Vt + \frac{V^2}{30\ Pef_s}$$

in which

V = initial speed in m.p.h.

P = per cent braking power actually realized

$e \times f_s$ as determined by the existing brake rigging and shoe conditions (see Table 10).

t = time at the beginning of the stop during which the brakes are to be considered as having no effect, to allow for the time element in the application of the brakes.

Kind of Air Brake Equipment

	PM	UC Electro-Pneumatic
For a 12-car train		
t ranges from	2.0	0.70
t ranges to	2.5	0.85

A train of 12 steel cars and locomotive with the electro-pneumatic brake, 150% braking power, clasp brake rigging, unflanged brake shoes can be stopped.

From 30 m.p.h. in 200 ft. equivalent to an average retarding force of 300 lb. per ton

From 60 m.p.h. in 1000 ft. equivalent to an average retarding force of 240 lb. per ton

From 80 m.p.h in 2000 ft. equivalent to an average retarding force of 214 lb. per ton

BRAKE PERFORMANCE ON MODERN STEAM RAILROAD PASSENGER TRAINS

A DISCUSSION OF THE RESULTS OF THE PENNSYLVANIA RAILROAD BRAKE TESTS, 1913

By S. W. Dudley[1], Pittsburgh, Pa.
Non-Member

INTRODUCTION[2]

Exactly one year ago today the Pennsylvania Railroad began the series of passenger train tests described herein. The tests continued until nearly the end of May, a total of 691 tests having been made. The compilation of the data and their preparation for publication in the official report occupied practically all of what was left of the year 1913.

A train of 12 steel passenger cars and modern locomotive weighs nearly 1000 tons (actually 932 tons), is about 1000 ft. long and, at 60 m.p.h. speed has a kinetic energy of 224,000,000 ft.-lb.

The amount and cost of the energy which must be expended in acquiring this speed have always received the most careful study. Every effort which experience and ingenuity can suggest has been put forth to produce a locomotive of greater capacity, or of higher efficiency, or a safer or more economical passenger car. Many difficult and important problems have been solved. But there are questions of greater difficulty and importance than these. The acquiring of speed is a matter of convenience, probably,—of economy, possibly. But the disposal of the kinetic energy thus developed involves not only the comfort of the passenger and the profit of the railroad but also the more vital considerations of the integrity of the equipment and service, and the safeguarding of human life itself.

Having the train now at full speed what does it amount to and what is to be done with it?

With the ordinary high-speed brake apparatus such a train would be stopped by an emergency application of the brakes in a distance of from 1600 to 1800 ft. (over $1\frac{1}{2}$ times the length of the train).

In making ordinary brake applications for slow-downs or station

---

[1]Assistant Chief Engineer, Westinghouse Air Brake Co.

[2]Read at the New York Meeting, February 10, 1914.

stops, skill and judgment must be carefully exercised in order to avoid shocks and make short and accurate stops.

The Pennsylvania Railroad brake tests of 1913 showed that such a train at 60 m.p.h. speed can be stopped by an emergency application in 1000 ft. or within the length of the train. They also showed that trains can be controlled by service applications without shocks at any speeds and with a high degree of accuracy and promptness and still require less expert knowledge and skill on the part of the manipulator.

The performance of brakes is usually discussed from the standpoint of the length of emergency stops. This is a vital consideration but not the *only* one. Countless station stops, slow-downs and applications while descending grades are made to one emergency stop. Consequently the action of the brake mechanism, its effect on the train, and the relation of the engineman to the brake, merit serious consideration if a high standard of efficiency is to be reached and maintained. Too often a well merited confidence in the effectiveness of the brake as a safety device obscures the fact that in every-day service, on every train on the road, an efficient and well maintained brake is returning good and regular dividends in freedom from delays and troubles that every engineman, round house foreman and master mechanic knows too well, but which may not appear as a direct item of expense on the balance sheet. Another distinction to be kept in mind is that when speaking of the brake we mean more than the air brake mechanism. Ordinarily the term *brake* is understood to be synonymous with the triple valve. This is a mistake. The air brake is but one element in the performance of brakes on trains.

The brake comprises not only the air brake valve device on the car but every element of the system between the hand of the engineman at the brake valve on the locomotive and the points of contact of the wheels on the rail. It is only as we succeed in obtaining a high efficiency in everything that lies between these two extremes that we can claim to have realized the possibilities of the brake.

For example, during the tests the length of emergency stops from 60 m.p.h. was shortened from 1700 to 1100 ft. by improvement in the air brake apparatus alone. But a still further reduction of 300 ft. was shown to be possible by taking advantage of every improvement in foundation brake rigging, brake shoes and locomotive brake equipment that was available during these tests. Similarly, although the use of the improved air brake with electro-pneumatic control of the service application and release, entirely eliminates shocks due to brake manipulation and insures a certainty and responsiveness of action impossible with less efficient apparatus, the clasp type of brake rigging brings such relief to journals, trucks and brake shoes and so reduces the false or ineffective travel of the brake cylinder piston, that without it satisfactory service and emergency performance on heavy cars cannot be expected.

The importance of the Pennsylvania Railroad Brake Tests of 1913 lies not so much in the record emergency stops made, but (to the railroad man especially) in the fact that never before have the inherent characteristics of the air brake, the brake rigging and the brake shoe been so clearly demonstrated, nor their direct and separate influence on the performance of the train in service as well as in emergency stops so definitely determined.

Many new problems were encountered during the investigations arising from the service operating conditions imposed by modern heavy rolling stock. In the formal record an attempt has been made to present these problems clearly, to develop the mechanical principles upon which their solution was found to depend, and to show the reasons for and significance of the results. Reference to this record should be made for the details of any particular problem that may be of interest at the moment.

The brake shoe has to dissipate energy (the $\frac{1}{2} MV^2$ of the train) in the form of heat through the medium of the friction and abrasion of the brake shoes. To effect improvement in this performance many avenues of approach are open, all of which can be classified under the following heads:

I Improvement in the *effectiveness* of the brake as a whole. *Effectiveness* depends upon the *amount* of retarding force developed by the brake shoes on the wheels and the *promptness* with which maximum retarding force is obtained.

II Improvement in the *efficiency* (which includes *economy*) of the brake as a whole. *Efficiency* (*including economy*) depends upon realizing the maximum possible amount of retarding force from a given expenditure of compressed air and brake shoe material.

III Improvements in the automatic and manual *control* of the brake operation both of which depend upon the characteristics of the *air brake mechanism.*

Before entering upon the discussion it seems to me proper that this Society and the railroad fraternity should know to whom we are all indebted for the successful outcome of the tests and liberty to present these results freely and fully for the benefit of all who may be interested.

Mr. J. T. Wallis, General Superintendent of Motive Power, P.R.R., keenly appreciating the braking situation and its needs, extended every opportunity for carrying the investigation through to a definite and satisfactory conclusion and stimulated development in all the lines investigated by a rare personal interest and enthusiasm.

Mr. C. D. Young, Engineer of Tests, P.R.R., was in active supervision of the work and the comprehensiveness and direct practical application of the results are largely due to his energy and insight.

To mention by name all who, by ingenuity, skill and hard work contributed to the final results of these tests, would be to name every individual interested in the obtaining of the data and compiling the report. It is seldom that a corps of over 40 men can be engaged on a work of this nature for over three months and maintain throughout as intense an interest and as effective and harmonious a working organization as was the case in this instance.

I. THE BRAKE PROBLEM

Consider a train of 12 steel passenger cars and a locomotive running at a speed of 60 m.p.h. The amount and the cost of the energy which must be expended in acquiring this speed have always received the most careful study. Every effort which experience and ingenuity can suggest has been put forth to produce a locomotive of greater capacity, or of higher efficiency, or a safer or more economical passenger car. Many difficult and important problems have been solved. But there are questions of greater difficulty and importance than these. The acquiring of speed is a matter of convenience, probably; of economy, possibly. But the disposal of the kinetic energy thus developed involves not only the comfort of the passenger and the profit of the railroad but also the more vital considerations of the integrity of the equipment and service, and the safeguarding of human life itself.

Having the train now at full speed what does it amount to and what is to be done with it?

Through daily familiarity, this situation has come to be accepted as one of these commonplaces too well understood and provided for to attract more than passing interest. The reverse, however, is true; as is recognized at once when the tremendous and, too often, distressing consequences of an accidental exhibition of this energy in all its destructive power are encountered. The train moving at 60 m.p.h. possesses a kinetic energy of 224,000,000 ft-lb. In amount this corresponds to a ball weighing one ton falling from a height of 21 miles, or a blast of dynamite powerful enough to raise the train itself 120 ft. into the air.

The brake problem therefore embraces all of the elements which have to do with the convenient, economical and harmless dissipation of the kinetic energy of a moving train by controlling or stopping its movement. The particular phase of the problem which will engage our present attention is that respecting the performance of the brakes in general use on typical modern steam railroad passenger trains, the elements affecting the stopping of such trains, the direction in which improvement in these elements is possible and the results accomplished by the improvement which has been made.

Present and Past Brake Performance

The typical modern train referred to above would be about 1040 ft. in length. If equipped with the brake apparatus, including foundation brake rigging and brake shoes, of the type in most common use throughout the country, an emergency application of the brakes at a speed of 60 m.p.h. would bring the train to a standstill in 1600 ft., or approximately one and one-half times its own length. By means of the improvements given attention in the recent Pennsylvania Railroad tests, it has been found possible to stop the train in less than 1000 ft. In other words, a modern 12 steel car steam railroad passenger train, as a result of the developments exemplified in these tests, can be stopped from a speed of 60 m.p.h. in less than its own length. How much of an accomplishment this is will appear in what follows, but it should not be overlooked that even with this improvement we have reached a point on the absolute scale of safety but little further advanced than was the case more than ten years ago, when the then existing brake apparatus, the same that stops the modern trains in 1600 ft., was able to stop the trains of that day in approximately 1050 ft.

A substantial factor of safety and a high efficiency in brake operation was assumed to exist as a matter of course (and rightly so) when cars were light, trains short, average speeds and frequency of trains low, and the general operating conditions of service less severe than are found all over the country today. But with the intensive development in motive power and rolling stock and the consequent continuous increase in weight and length of cars, length of trains and speed since the introduction of the quick acting triple valve a quarter of a century ago, the margin of reserve capacity in the brake apparatus, to meet the demands in excess of those for which it was originally intended, has dwindled until the ultimate capacity of the old brake scarcely serves to meet the common requirements of modern high-speed train service and leaves no adequate reserve available for present extremes, nor for the greater demands of the future.

Scope of the Tests

Realizing the significance of the knowledge and experience accumulated in recent years, the Pennsylvania Railroad, in conjunction with the Westinghouse Air Brake Company, instituted in the spring of 1913 the most scientific and comprehensive investigation of the different factors affecting the operation of brakes on steam railroad passenger trains that has been undertaken since the Galton-Westinghouse trials of 1878 and 1879. In addition to an examination of the characteristics of brake shoe friction throughout a wide range of laboratory and operating conditions, the test included also a study of the effect of various types of air brake mechanisms and foundation brake rigging and different degrees of emergency braking force.

The tests indicated the degree to which existing apparatus was suited to existing conditions, the direction in which improvement was necessary and could be made, and the amount of improvement actually accomplished. But brief mention of details will be made in this paper. All of the information is available in the official report of the tests compiled by the Test Department of the Pennsylvania Railroad.[1]

The limitations of the old brake apparatus are most marked in the following particulars: In the length of emergency stops; the uniformity of brake applications on different vehicles comprising the train; the safety and protective features demanded by service conditions of great severity and complexity; the flexibility and certainty in applying and releasing the brake during service application; and the increased difficulty of keeping the service and emergency functions separate, i.e., insuring quick action when required on the one hand, and preventing it, when not required on the other.

In considering the improvements desirable in the above particulars four factors require special attention:

A The characteristics of the mechanism available for controlling the pressure of the compressed air in the brake cylinders.

B The efficiency of the mechanical transmission of the force of compressed air developed in the brake cylinders, through the rods and levers of the brake rigging to the brake shoes.

C The efficiency of the brake shoe in transforming the pressure imposed upon it into retarding force at the rim of the wheel.

D The available adhesion between the car wheels and the rails.

The Galton-Westinghouse brake trials on the London, Brighton and South Coast Railway in England during 1878, constituted the first scientific investigations of the action of brake shoes in retarding the motion of railway vehicles. They have occupied a unique position in the railway art, as the classical and in fact, the only source of information regarding the characteristics of brake shoe friction under certain typical road service conditions.

[1]Copies obtainable on request from the Westinghouse Air Brake Company—EDITOR.

But the conditions under which these experiments were conducted represented an early state of the art when much lighter cars, simpler mechanisms and lower braking pressures were used than has been common practice in this country for many years. In consequence of this, although the results of the experiments remain conclusive and fundamental as to general principles involved, they are far removed, in degree, from modern railroad train operating conditions.

The Lake Shore emergency brake tests of 1909, which appear in the Master Car Builders' Association Proceedings for 1910, directed special attention to the important influence of the foundation brake rigging and brake shoe performance as affecting the stopping of modern heavy rolling stock. These tests showed clearly the necessity for realizing, as nearly instantaneously as possible, a retarding force as high as the limitations of track and equipment would permit, if emergency stops, especially at high speeds, were to be made in as short a distance as desirable. The requirements of present and anticipated practice in heavy car constructions were formally considered and placed on record by the unanimous adoption of the following resolutions at a meeting of railway officials and the Master Car Builders' Committee on Train Brake and Signal Equipment, Pennsylvania Station, Pittsburgh, July 1909.

Resolved, That it is the sense of this meeting that the air brakes provided for the heavier passenger cars now building shall be of such design, proportion and capacity as to enable trains of said heavier passenger cars to be stopped in practically the same distance after the brakes are applied as is now the case with the existing lighter cars and be it further

Resolved, That for the use of this committee and others interested in making calculations, we suggest that it be assumed that the theoretically desirable stop is one which requires the space of not over 1,200 feet after the brakes are applied, the speed of the train at the time of the application of the brakes being sixty miles per hour.

There are four factors which have a controlling influence on the length of stop: (1) the maximum brake cylinder force; (2) the time in which this is obtained; (3) the efficiency of the foundation brake rigging in multiplying and transmitting this force to the brake shoe; (4) the mean coefficient of brake shoe friction.

All but the last factor, viz., the mean coefficient of brake shoe friction, can be controlled or properly provided for in advance by correct design and installation. On the other hand the experience of recent years has repeatedly demonstrated that no one of these four factors can be neglected without a corresponding loss in effective retarding force. It is therefore of the greatest importance to distinguish and give due consideration to the controllable factors mentioned in order to compensate as far as possible for the unavoidable variations in brake shoe friction. That these variations have more than a merely nominal effect follows from the fact that the brake shoe, considered as an element of the mechanical system transforming the force of compressed air into retarding force at the rim of the wheels, is low in efficiency, averaging for stops from 60 m.p.h, in the neighborhood of 10 per cent. Consequently a slight variation in brake shoe performance can cause a considerable percentage of change in mean coefficient of brake shoe friction and a corresponding change in length of the stop, the latter being subject to a range of variation of as much as 20 per cent, or more *due to brake shoe condition alone.*

The object of the Pennsylvania Railroad Tests of 1913 was to make as thorough a study as might be found practicable of the variables mentioned above and their effects, with particular reference to:

A A determination of the maximum percentage of emergency braking power which can be adopted, considering:

- *a* The type of brake shoe to be used
- *b* The type of brake rigging to be adopted
- *c* The type of air brake mechanism and control to be adopted
- *d* The degree to which occasional wheel sliding is to be permitted under unfavorable circumstances
- *e* The variation in the condition of the rail surface for which it is considered necessary to provide

B A comparison of the relative performance of the clasp brake rigging (two shoes per wheel) and the standard brake rigging (one shoe per wheel) with regard to:

- *a* Maintenance of predetermined and desired piston travel
- *b* Efficiency of transmission of forces
- *c* Effect upon wheel journals, bearings and truck
- *d* Mean coefficient of brake shoe friction for the standard plain cast iron shoe

C A comparison of the performance of the improved air brake mechanism (type UC) with that of the commonly used "high speed" (type PM) brake equipment with regard to:

- a Efficiency and effectiveness, as shown by the length of service and emergency stops
- *b* Safety and protective features
- *c* Flexibility and certainty of response to any manipulation of the engineer's brake valve

d Uniformity of action of individual equipments associated in the same train and of any individual equipment at different times

e Smoothness of riding during stopping, slack action between cars, and the resulting shocks

f Capacity for future requirements

D The behavior of the brake shoes as the tests progressed and any variation in the results of similar tests which could not be accounted for by known changes independent of the brake shoe. One type of brake shoe was to be used thoughout the range of the tests. Relating to objects *A, B,* and *C,* advantage was taken of this opportunity to establish as definitely as possible the characteristics of this type of brake shoe under the influence of various combinations of speed, pressure, time, weather and the conditions of the brake shoe.

E The coefficient of friction between the wheel and the rail under varying weather conditions.

In addition to the investigations outlined in general above, it developed during the tests that additional data were desired regarding the performance of brake shoes under certain specific conditions. In consequence a series of experiments was carried out at the laboratory of the American Brake Shoe and Foundry Company, at Mahwah, N. J.

From the outset of the tests an endeavor was made to obtain data and develop methods by which the performance (as to stopping when placed in service) of any given air brake apparatus and its related equipment, could be predetermined on the basis of the observed action of the individual elements which go to make up the whole.

It will be seen that the desirable stopping distance of 1200 ft. may be obtained by improvement in some or all of the controlling factors, namely, the type of air brake mechanism, the foundation brake rigging, the nominal percentage of braking power, and the type of and condition of the brake shoe. Furthermore, a stop of 200 ft. or more shorter than this can be obtained when all the elements having an influence on the length of stop are disposed in the most favorable manner possible.

In proportion as the efficiency of any one or more of these factors can be increased, that of the others can be correspondingly reduced so that a lower maximum can be employed for the remaining factors when circumstances render this desirable. For example, the reduction in the time of action, secured by the use of the electric control of the brakes, increases their effectiveness and makes a shorter stop possible, thus permitting the use of braking power 30 per cent less than is required with a less effective brake for the same stop.

Features of Equipment and Apparatus Tested

Air Brake. The tests of the standard (type PM) air brake equipment were planned to determine the characteristic performance of this type of equipment throughout the range of service and emergency operating conditions typical of the ordinary service in which this equipment is in general use. Inasmuch as experience has shown that under the severe requirements of today the type PM equipment lacks many of the features which are necessary to obtain a desirable degree of stopping power in emergency applications and prompt and certain response at all times in ordinary service brake manipulation, one of the objects of the tests scheduled for this type of equipment was to bring out its limitations and serve as a standard of reference to measure the betterment made possible by the improved features of the new air brake apparatus, the more efficient design of foundation brake rigging and more satisfactory brake shoe performance.

The special features of the improved air brake equipment (type UC) which received more or less attention during the tests may be summarized as follows:

A The electro-pneumatic brake equipment is adapted to meet any requirement, from that exemplified in the PM brake equipment to the more exacting requirements of present conditions, with a degree of efficiency as high as the existing physical conditions will permit.

B Considering cylinder pressure alone the equipment may be installed so as to produce any desired pressure, either in service or in emergency.

C The gain by use of the electric control, in addition to the pneumatic, is the elimination of the time required for the pneumatic transmission of the action of the brake from car to car and, in addition the elimination of shocks and uncomfortable surging which results from the non-simultaneous application of the brakes on all cars.

It is apparent that the gain from the electro-pneumatic control is not so much in the shortening of the stop, particularly in emergency, as it is in the increased flexibility and certainty of control of the brake and the assurance that modern long heavy trains can be handled smoothly and accurately.

D The troubles and inconveniences due to brakes failing to release, as, for example, after light brake pipe reductions, as well as the undesired application of brakes due to unavoidable fluctuations of brake pipe pressure when running over the road, are eliminated.

E An adequate supply of air is available at all times.

F The emergency braking power is available at any time, even after a full service application of the brake, since it is impossible for the engineman to use up the reserve emergency pressure without making an emergency application.

G The equipment is adaptable to all weights of cars and to any desired percentage of braking power. Two brake equipments for heavy cars are not necessary nor are two service brake cylinders required, except for cars weighing more than the limit of the service capacity of one brake cylinder. Provision is made for using one brake cylinder up to the maximum percentage of emergency braking power which it can provide, and for using two cylinders when a higher emergency braking power is desired. When using one brake cylinder, the maximum service pressure is controlled by means of a safety valve. When two cylinders are used, equalizing pressure from 110 lb. brake pipe pressure is utilized for the service brake (instead of blowing the air away at a reducing valve) and another brake cylinder is used for the additional power required in emergency applications. The use of one or two cylinders is optional, depending upon the amount of braking power to be employed.

Brake Rigging. Duplicate tests were made with the clasp brake rigging, two shoes per wheel, for every test made with the standard brake rigging, one shoe per wheel, in order to bring out the advantages of the clasp brake in the following desirable features: (*A*) constant piston travel for all cylinder pressures; (*B*) smoothness of action during stopping; (*C*) greater certainty of obtaining and maintaining the predetermined braking force contemplated in the design of the air brake equipment and foundation brake rigging; (*D*) less displacement of journals, bearings and trucks, tending toward greater mechanical efficiency and less cost of maintenance; (*E*) a coefficient of friction equal to or greater than that with the single shoe brake with less wear of brake shoe metal and lower brake shoe temperatures.

The original plan contemplated two 12 car trains of standard P-70 cars (see Fig. 1). These cars have 4-wheel trucks with one 16-in.

brake cylinder per car. One train was equipped with the clasp type of brake rigging (two shoes per wheel) and the other with the type of standard brake rigging (one brake shoe per wheel) existing on these cars since they were built, but modified by increasing the strength of

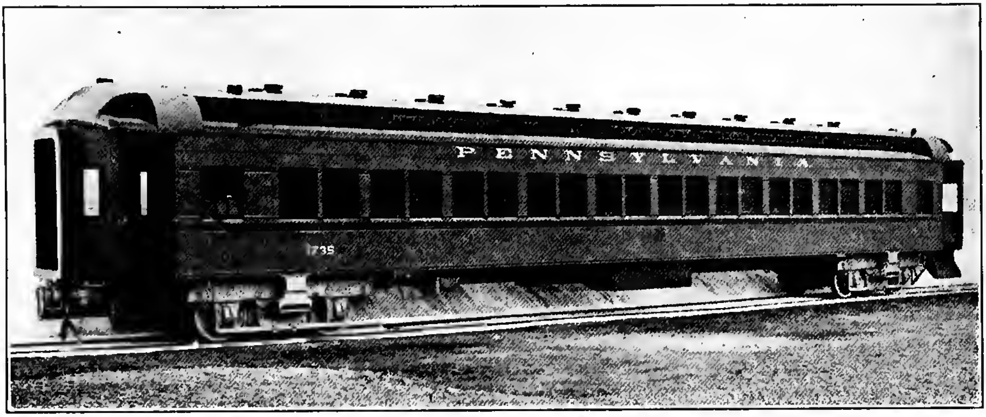


Fig. 1 Class P-70 Steel Car
Twelve cars of this type were used in the test train

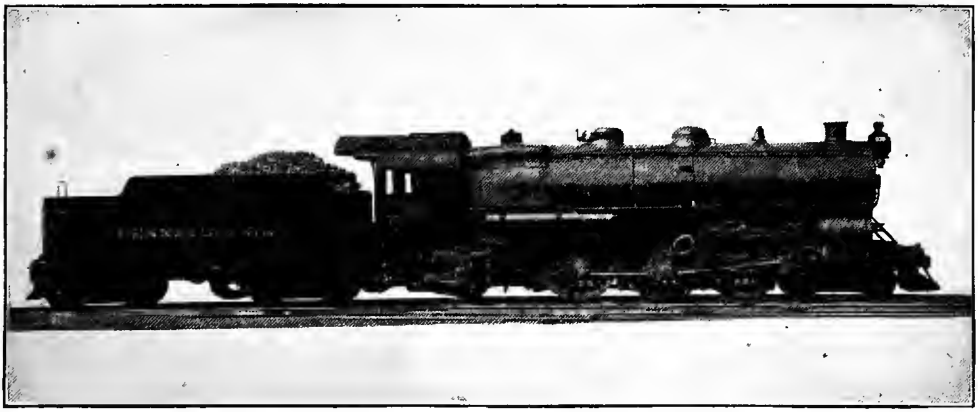


Fig. 2 Class K2sa Locomotive
A locomotive of this class was used in the test train

the members to be suitable for 180 per cent braking power which necessitated lowering the brake shoes 1⅛ in. below their former position and by anchoring the truck dead lever to the car body, instead of to the truck.

The standard brake rigging as originally applied to the cars of the test train is shown in Fig. 3.

With the design of clasp brake rigging shown in Fig. 4 the best

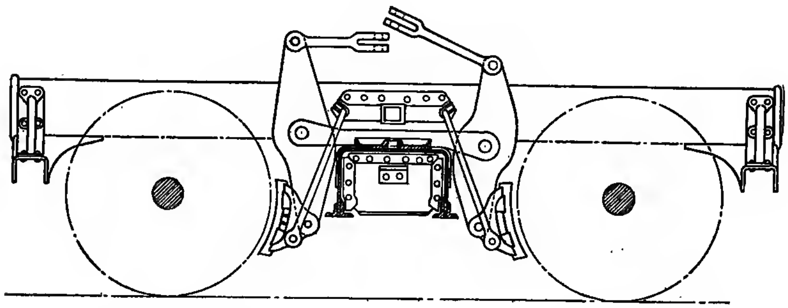


Fig. 3 Brake Rigging, the Standard as Modified for Test

This brake rigging was used on the twelve car train. It has one brake shoe per wheel

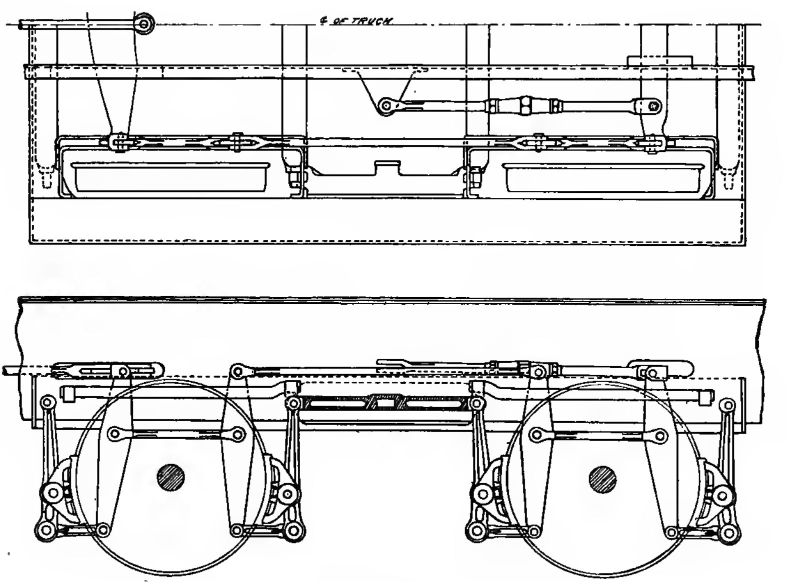


Fig. 4 Brake Rigging, the No. 3 Clasp Brake

The third form of clasp brake; this rigging was used only in single car break-away stops. It was with this clasp brake rigging that the best stops were made

stops were made fulfilling what it was anticipated should be obtained from the application of two brake shoes per wheel.

Diagrams of the lever arrangement of the various types of brake rigging tested are shown in Fig. 5.

Train Make-Up and Equipment

Before describing the tests and their results it will be in order to refer briefly to the composition of the trains used, the apparatus employed, and the methods of making observations.

In order to obtain the best data possible, instruments were devised for taking records of the friction of the rail, wheel sliding, retardation of the train, and slack action between cars as well as for a number of minor observations.

The test train was 1040 ft. long, consisting of a Pacific type locomotive and tender of the P. R. R. K2s class, weighing in working order about 200 tons, and 12 P-70 steel passenger cars averaging about 61 tons each.

The ET air brake equipment was used without any modification on the locomotive, except that in some tests an auxiliary device was used which increased the braking power obtained during the early portion of the stop.

All tests were made under road service conditions, except where otherwise noted, the air brake regulating devices on the locomotive and cars being adjusted as follows:

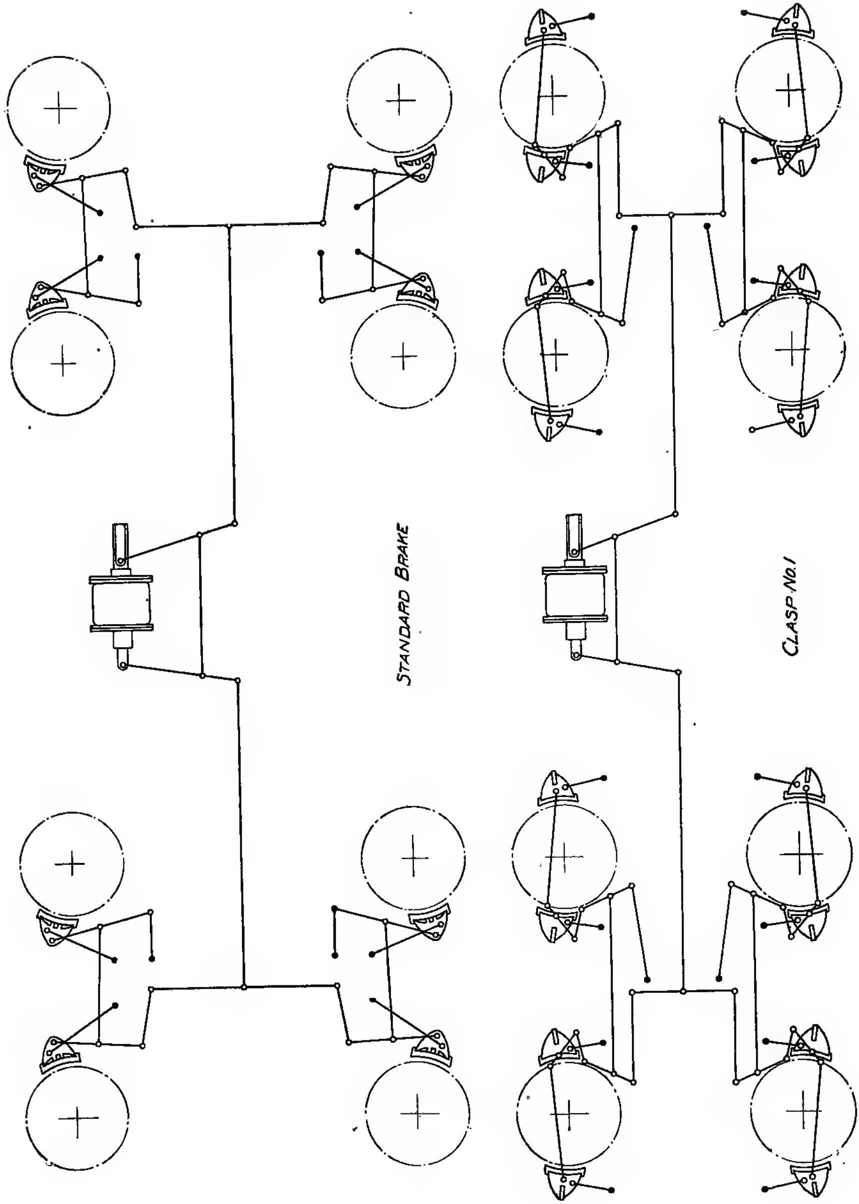
Pump governor, low-pressure head 130 lb. Maximum pressure head 140 lb.

Feed valve, 110 lb.

ET distributing valve safety valve, 68 lb.

The cars were equipped with the present standard air brake apparatus (PM) and with the improved type of air brake equipment (UC), these installations being so arranged that a complete change from the standard equipment (PM) to the new equipment (UC) having PM features only or the complete pneumatic features of the new equipment or to the new equipment with complete electrical control could be quickly made.

The standard plain cast-iron brake shoe was used in most of the tests. In several tests flanged, slotted and half area shoes were employed. Special care was taken to insure uniformity in quality and the condition of all shoes at the beginning and during the progress of the tests.


STANDARD BRAKE
CLASP No. 1

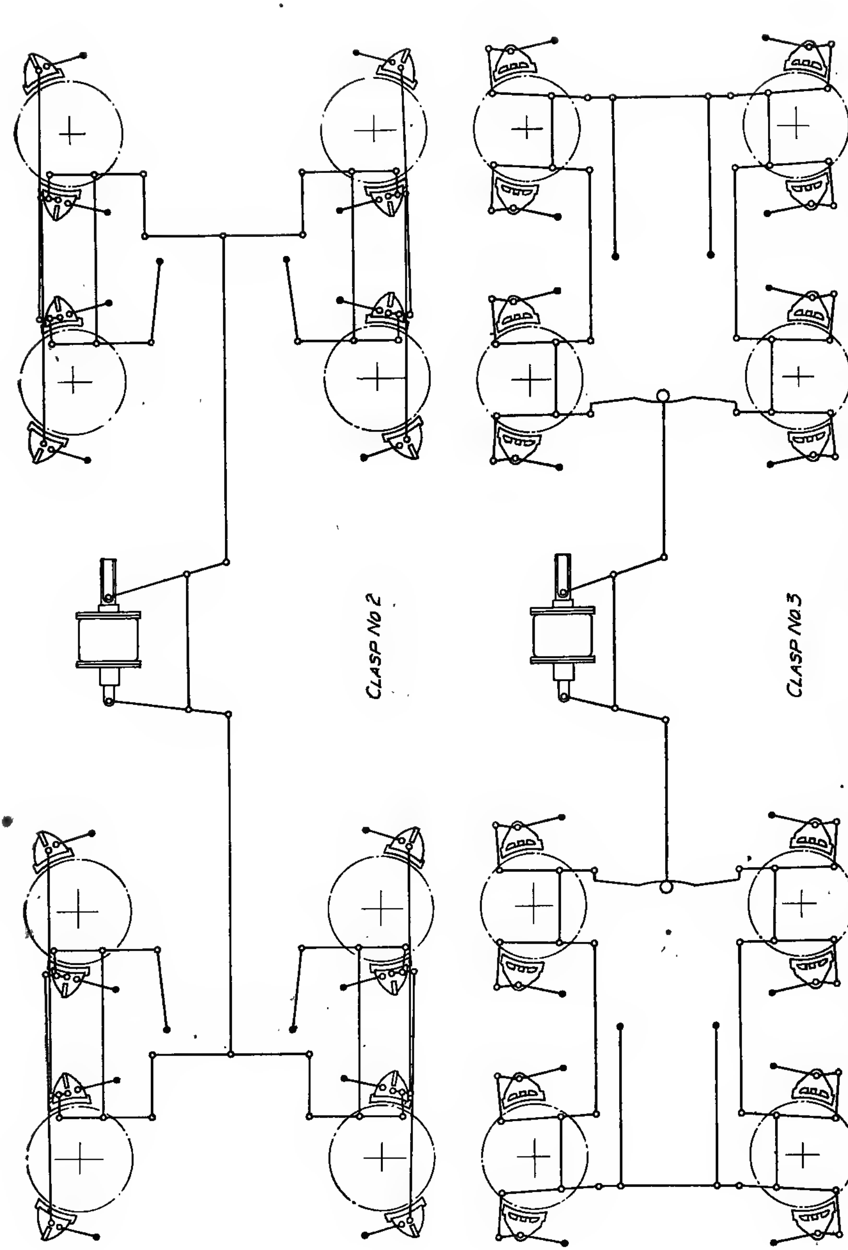


FIG. 5 OUTLINE DIAGRAM OF LEVER ARRANGEMENTS OF THE FOUR BRAKE RIGGINGS USED

The high-speed reducing valves of the PM equipment were adjusted to open at 62 lb. brake cylinder pressure.

The standing piston travel was adjusted before each run to $6\frac{1}{2}$ in. with a full service brake application.

FIG. 6 BRAKE CYLINDER PRESSURE INDICATOR

One of these instruments was used on each car. The drum is driven by a spring motor in the box. A typical record is shown in Fig. 47

TEST APPARATUS AND OBSERVATIONS TAKEN

Locomotive. The apparatus on the locomotive consisted of the usual gages which indicated main reservoir, brake pipe, and brake cylinder pressures, and in addition a brake cylinder indicator was used on the tender brake cylinder and served to measure the pressure in all of the brake cylinders of the locomotive and tender, viz., one engine truck, two driver brakes, one trailer truck and one tender brake

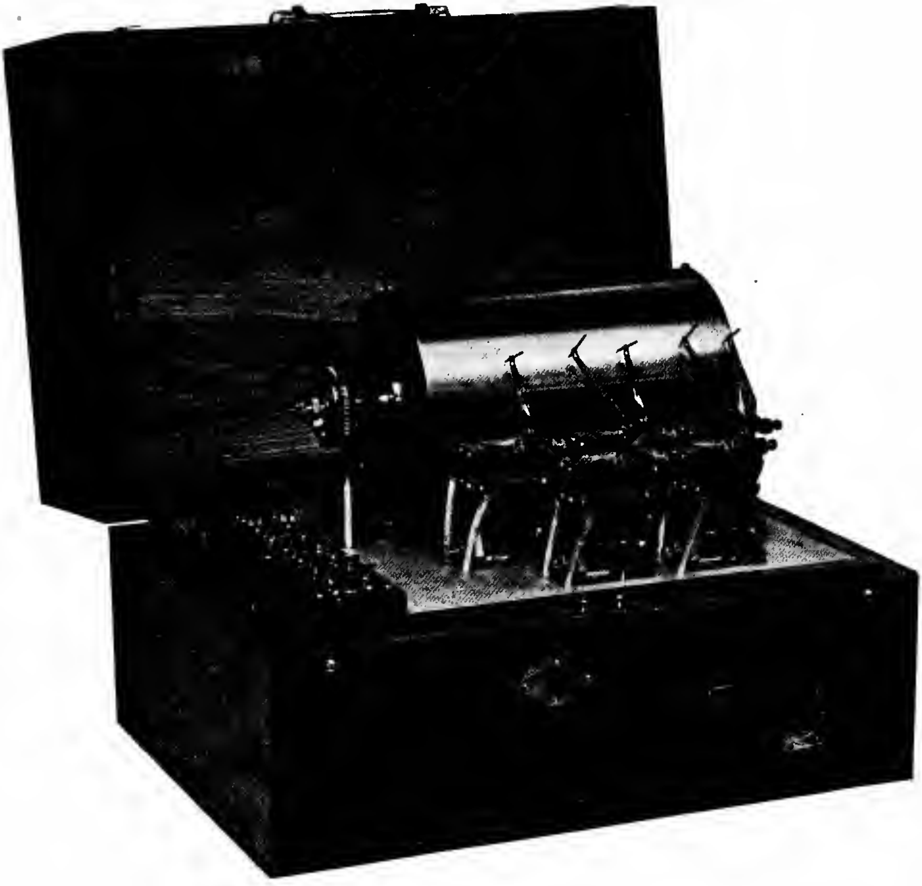


FIG. 7 WHEEL SLIDING INDICATOR

One of these instruments was on each car. There is a recording pencil for each axle and one distance pencil. A typical record is shown in Fig. 9

cylinder. A voltmeter, calibrated in m.p.h. was connected to a generator, belt-driven from the right front engine truck wheel, and served as a guide to the engineman in obtaining the desired speed.

A device for recording automatically the distance traveled by the train beyond the point of brake application was driven from the left engine truck wheel and was used in connection with the wheel sliding indicators on the cars.

Devices similar to those used in former brake tests were employed to operate the track circuit breakers and to automatically apply the brakes at the zero circuit breaker.

On the locomotive, observations were taken of the time of stop and the main reservoir and brake pipe pressure, the tender piston travel and the amount of coal and water on the tender.

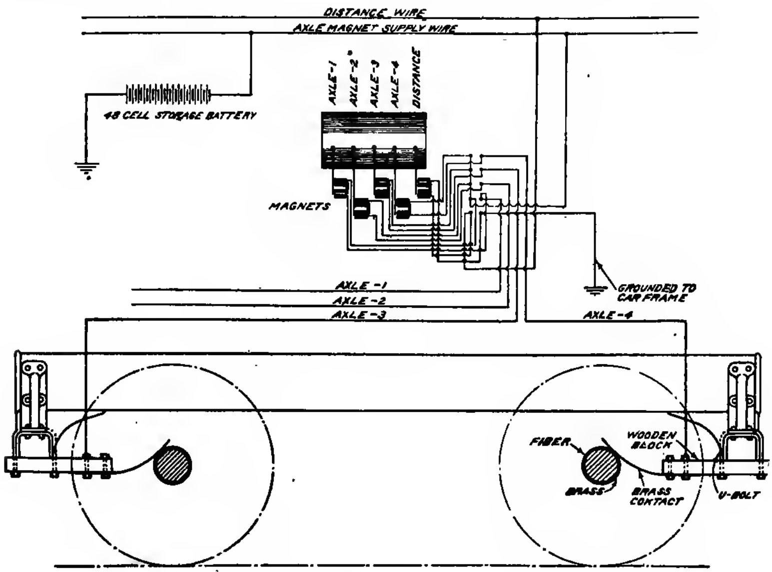


FIG. 8 WHEEL SLIDING INDICATOR
Diagram of electric circuits

Cars. Each car was furnished with a brake cylinder indicator, Fig. 6, and a wheel sliding indicator, Figs. 7 and 8, with the necessary wiring and connections. A typical wheel sliding indicator record is shown in Fig. 9.

A chronograph, Fig. 10, recording the distance of stop, time of stop, deceleration of train, the brake cylinder pressure and the brake pipe pressure, was located on car six. In connection with this chonograph a record was made of the action of the brake shoes with respect to sparking.

Indicators (Fig. 11) for measuring the slack action between the cars were used at different points in the train.

The construction and the data obtained from the automatic recording devices mentioned will be sufficiently clear from the illustrations shown to render further explanation unnecessary. Complete information regarding these devices is included in the complete report of the tests.

Specially designed apparatus was used to measure the pressure

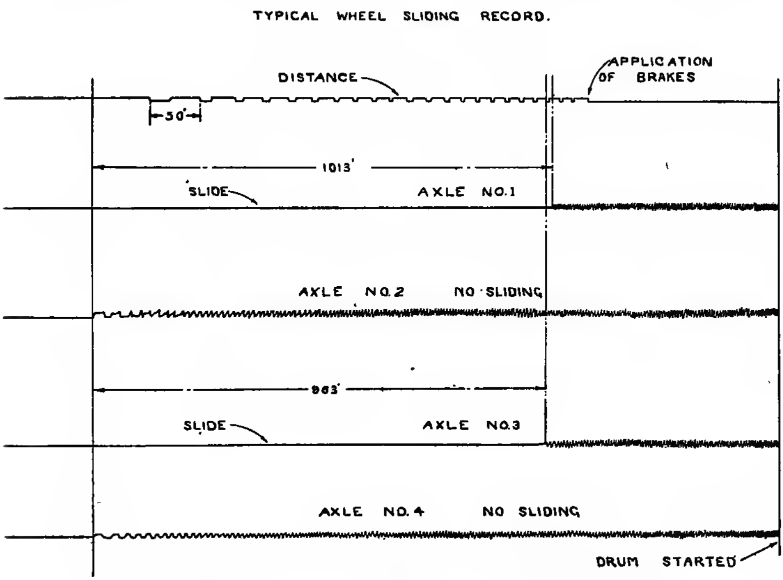


Fig. 9 Wheel Sliding Indicator, Typical Record

When the wheel slides the circuit is not interrupted and a straight line is drawn

delivered to the brake shoes during some of the tests the object of which was to determine the efficiency of the brake rigging.

Telephones were located in the first, third, sixth, ninth and twelfth cars and greatly facilitated the issuing of instructions.

Track. The tests were made on the south bound track of the Atlantic City Division of the W. J. & S. R.R. The portion of the track over which the braking was done was level, and part of a tangent about 25 miles long terminating at Absecon Station. A slight descending (0.3 per cent) grade approaching the measured test track was in favor of the train attaining speed. The point at which the brakes were applied was 2880 ft. north of mile post 9.

The track for a distance of 5000 ft. south of the zero point was wired for circuit breakers, which were placed at intervals of 25 ft. up

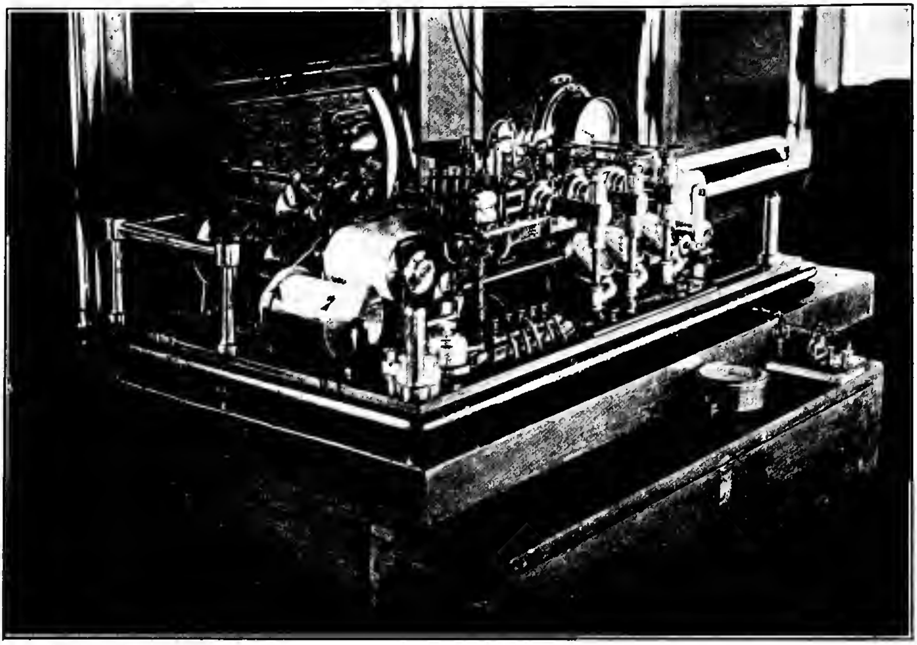


FIG. 10 KAPTEYN CHRONOGRAPH

This instrument was used to make a parallel record of brake cylinder pressure, time of stop, distance of stop and in addition a record of speed and deceleration

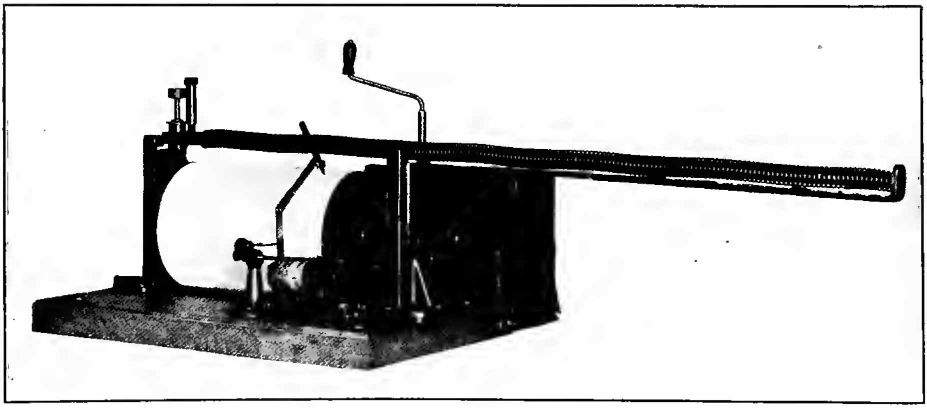


FIG. 11 SLACK ACTION RECORDER

This instrument was used to record the extension and recoil of the drawbars or the motion between the cars. A typical record is shown in Fig. 56

to 1200 ft. from the zero point, and at intervals of 50 ft. from there on to the 5000 ft. point. Preceding the zero point, eight circuit breakers were located, 66 ft. apart from which the initial speed of train (speed at the trip) was determined.

A cabin, located near the zero circuit breaker, contained the clock and chronograph from which in connection with the track circuit

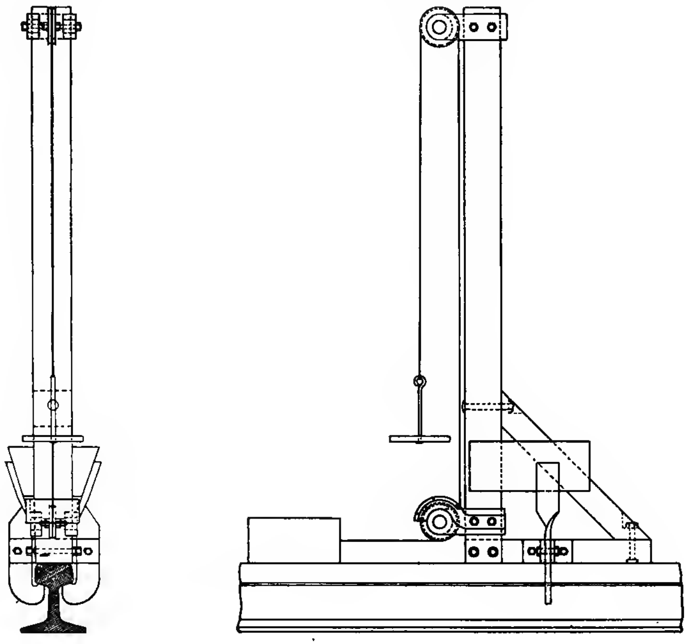


Fig. 12 Rail Friction Machine
This apparatus was used on one of the rails of the test track

breakers, the speed of the train before and during the stop was obtained.

After each test measurements were taken of the total length of the stop, and also the running-piston travel on each car.

Rail Friction Machine. Of the devices used on the track, the only one which requires special mention is shown in Fig. 12. This machine measured the force required to move or keep moving a block of tire steel resting upon the rail. The pressure of this block on the rail could be varied by means of weights of 20, 40, 60, 80 and 100 lb. Readings were taken with each of these weights and the coefficient

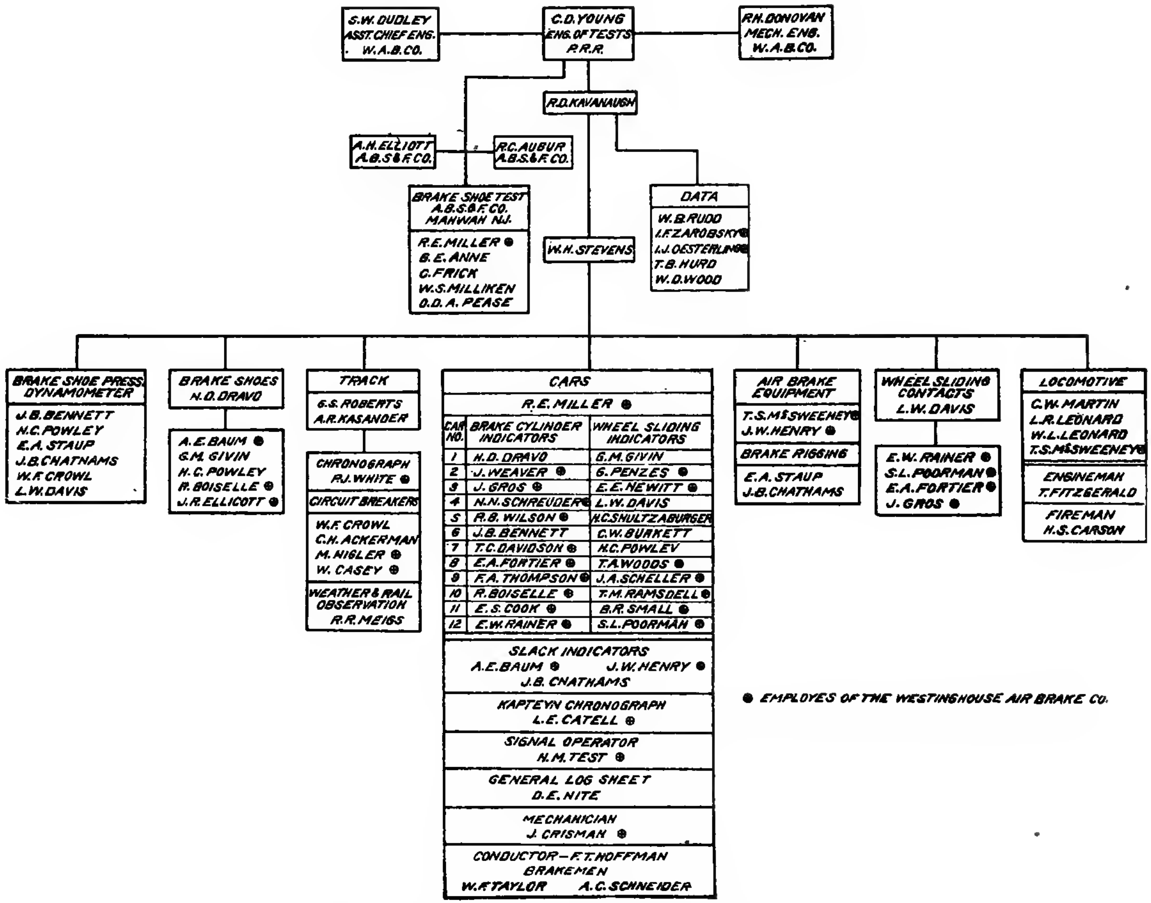


FIG. 13 ORGANIZATION CHART OF THE TEST FORCE

There were 44 observers who recorded 160 separate observations for each regular test run of rail friction recorded was derived from the average of the five readings.

When making a test run the engineman endeavored to reach a speed slightly above that desired, just before entering the measured track. The throttle was closed just before reaching the circuit breakers preceding the zero point, no change being made in the position of the reverse lever. The train then drifted over the circuit breakers preceding the zero point at which point the brake was automatically applied by the trip mechanism. At the instant the brake pipe exhaust started at the trip, the brake valve handle was moved to emergency position for all emergency tests and to lap position for all service application stops. When the engine and cars were to be stopped separately (breakaway tests), the same procedure as above

TABLE I. SUMMARY OF TESTS

Test No.		Date	Working Days	Brake Rigging Equipment	Kind of Brake Shoes
501	Starting..........	February 10.......	21	Clasp brake	Plain C. I.
718	Ending...........	March 5..........		No. 1	
1	Starting..........	March 13.........	14	Standard....	Plain C. I.
143	Ending...........	March 28.........		(Single shoe)	
1001	Starting..........	March 29.........	5	Standard....	Flanged C. I.
1040	Ending...........	April 6............		(Single Shoe)	
01	Starting..........	April 20..........	10	Clasp brake	Plain C. I.
0128	Ending...........	April 30...........		No. 2	
1501	Starting..........	May 11...........	6½	Clasp brake	Plain C. I.
1589	Ending...........	May 20...........		No. 3	
1101	Starting..........	May 16...........	3½	Clasp brake	Flanged C. I.
1160	Ending...........	May 19...........		No. 3	
401	Starting..........	May 22...........	1	Locomotive	
413	Ending...........	May 22...........		Alone	

Total number of working days.. 91
Total number of tests made.. 691
Average tests per working day.. 11
Maximum tests in one day.. 22

was followed, except that the coupling pin between the engine and tender was pulled out as soon as possible after steam was shut off. This permitted the engine to pull away from the train as soon as the brake application was made, providing the retardation of the cars was higher than that of the locomotive.

However, the engine did not always separate from the train when making stops with low braking powers on the cars. On this account it was decided to use steam on the locomotive in such tests as soon as the coupling pin was pulled out, so as to get the locomotive away from the cars and permit the cars to stop without any possible interference on the part of the locomotive, the stop of the locomotive in such cases being disregarded. For such stops the flexible wiper and the tripping mechanism were on the first car instead of the locomotive.

In all 691 tests were made, at Absecon, covering a period of time from February 10 to May 22, 1913. These were divided as shown in Table 1, which indicates that the average day's work consisted in making from 10 to 12 runs. A maximum of 22 tests were made in one day.

Organization. One hundred and sixty observations were taken on each regular test requiring an organization of 44 observers, distributed as shown on the organization chart (Fig. 13).

II. AIR BRAKE EQUIPMENT

The tests relating especially to the air brake apparatus were for the purpose of demonstrating the characteristic performance of the brake equipment in most common use throughout the country (the quick action automatic brake, type PM) with which the P-70 cars of the P.R.R. were equipped at the time of the tests; and for comparison with this to investigate what improvement, if any, would be afforded by the universal control (UC) electro-pneumatic equipment, both with respect to similar functions possessed by both the old and the new types of brake apparatus and the new features provided by the improved brake equipment.

It should be observed that the air brake apparatus has to do only with the controlling of the compressed air pressure in the brake cylinder. It has no control over what is beyond nor can it be affected by the performance of the other elements of the brake except as the position of the brake cylinder piston may be altered. In comparing different air brake mechanisms, therefore, it is preferable to consider their characteristic effects on the pressure realized on the brake cylinder piston, rather than with respect to the length of stop.

Present Equipment on P.R.R. P-70 Cars

The quick action automatic brake (PM equipment), is illustrated in Fig. 14. It comprises a 16-in. brake cylinder with automatic brake slack adjuster set for 8-in. running piston travel, a 16-in. by 42-in. auxiliary reservoir and a quick action triple valve (Type P-2, Fig. 15), which controls the flow of compressed air:

a From the brake pipe to the auxiliary reservoir for charging the system.

b From the auxiliary reservoir to the brake cylinder for applying the brakes.

c From the brake cylinder to the atmosphere when releasing.

d From the brake pipe to the brake cylinder, as well as from the auxiliary reservoir to the brake cylinder when a quick action application of the brakes is desired.

A high speed reducing valve designed to perform the functions of a safety valve during service brake applications, limits the brake

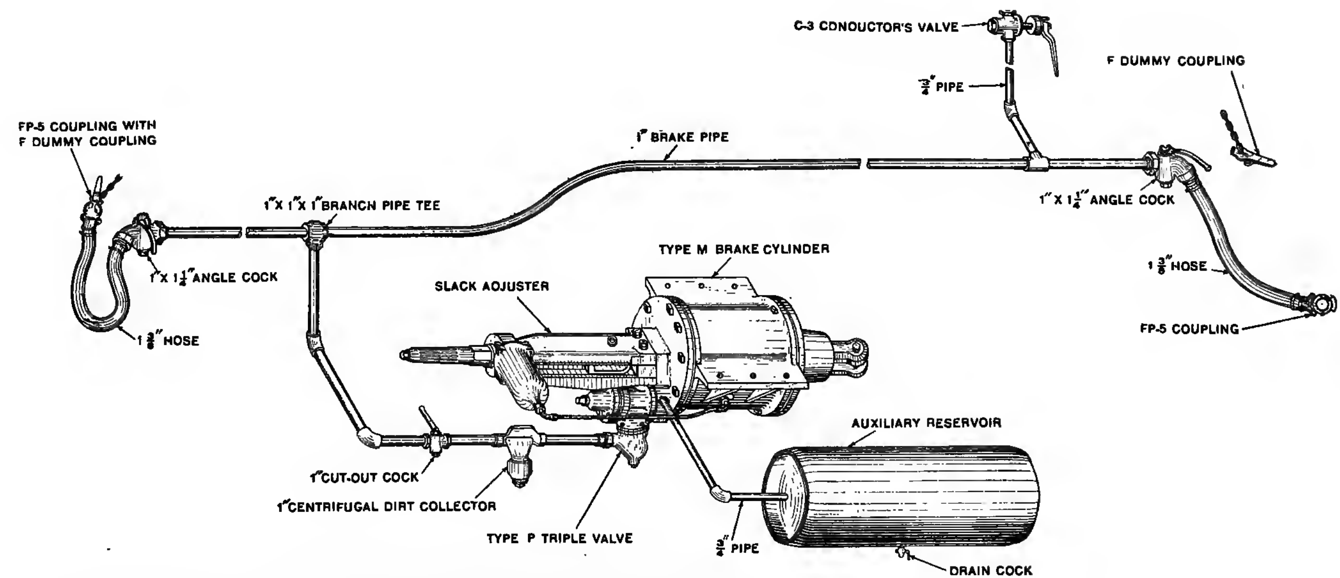


FIG. 14 QUICK ACTION AUTOMATIC BRAKE (STANDARD PM EQUIPMENT)

This shows the brake equipment in use on P-70 cars. It includes such well-known parts as the quick action triple valve (Fig. 15) and a high-speed reducing valve cylinder pressure to a maximum, predetermined as satisfactory for service operations (62 lb.). In emergency applications the high speed reducing valve retains the maximum cylinder pressure practically constant for a period of time and then by an accelerating blow down,

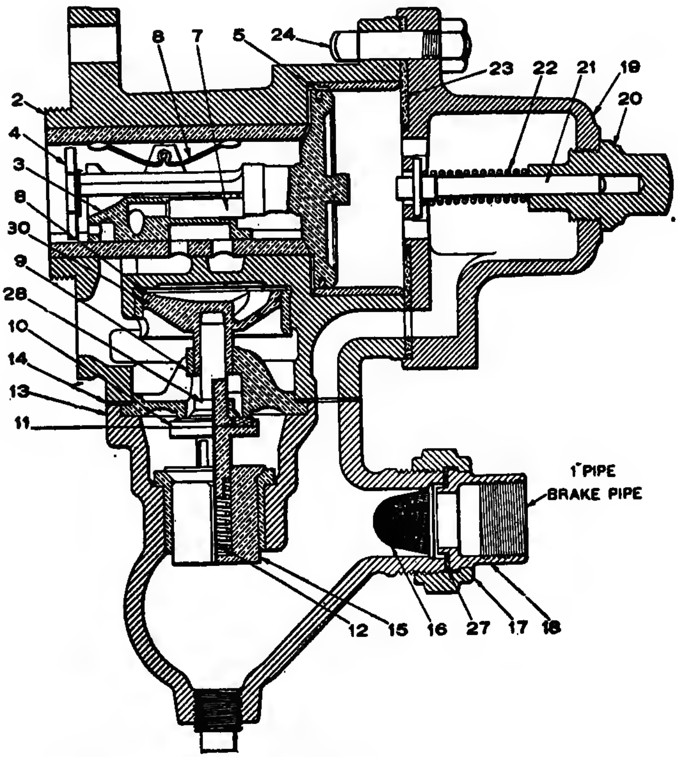


Fig. 15 Quick Action Triple Valve
Used with the standard PM brake equipment

reduces the brake cylinder pressure to 60 lb. This reduction is designed to compensate for the increased effectiveness of the brake shoes as the speed diminishes.

The compressed air required to charge the auxiliary reservoir is all supplied from the brake pipe through the feed groove around the triple valve piston when in release position only.

SERVICE BREAK APPLICATION

In response to a given reduction in brake pipe pressure the triple valve automatically reduces the pressure in the auxiliary reservoir an equal amount. The total volume of compressed air thus measured out from the auxiliary reservoir is delivered to the brake cylinder, where it produces a pressure on the brake cylinder piston proportional to the volume of the brake cylinder as determined by the piston travel.

The piston travel will vary with the brake cylinder pressure due to the deflection of members, lost motion in the rigging and so on. The relation between brake cylinder pressure and piston travel will be different with each type of brake rigging or the same rigging on different cars. Consequently, it is not practicable to establish the average for what is likely to be found in service. It is desirable, however, to establish the ideal relation between brake cylinder pressure and piston travel, which would exist if the rigging were well designed, the best material used and the installation made throughout in as satisfactory and efficient a manner as practicable.

As a guide to what this ideal relation should be, records were taken on some of the cars used during the test from which the performance of the brake rigging under test was determined. (See Figs. 51 and 52). As a result of a study of the subject in connection with these cards, the curve shown in Fig. 31 has been drawn to express the characteristic relation between brake cylinder pressure and piston travel for an ideal brake installation. The performance of existing types of brake rigging will approach more or less closely to that illustrated in Fig. 31 according to the degree in which sources of loss resulting in so-called "false" piston travel are eliminated.

Assuming the piston travel to vary with the brake cylinder pressure, as shown in Fig. 31, the characteristic relation between brake pipe reduction and resulting brake cylinder pressure, when using the PM equipment with the size of auxiliary reservoirs standard on P-70 cars, is shown by the dotted line Fig. 32. This curve is to be understood as characteristic and not necessarily exactly representative of the performance of any particular car brake installation in service. For comparison, a scale of per cent braking power is also shown, based on the same relation between cylinder pressure and percentage of braking power, as in the case of the P-70 car, namely 80 per cent braking power with 60 lb. brake cylinder pressure.

The amount of brake cylinder pressure and braking power corresponding to various brake pipe reductions with the type UC equipment are shown by the full line in Fig. 32. This does not coincide with the dotted curve for the PM equipment because the UC equipment has one size smaller auxiliary reservoirs, being designed to give a brake cylinder pressure of 50 lb. per sq. in. when a 20 lb. brake pipe reduction is made; and furthermore, the braking power basis is 90 per cent instead of 80 per cent as with the PM equipment.

The effect of the lower brake cylinder pressure per pound brake pipe reduction with the UC equipment in connection with a higher per cent braking power per lb. brake cylinder pressure than for the PM equipment is to provide a greater flexibility for service brake. For example: a 24-lb. brake pipe reduction is required to obtain a maximum (90 per cent) service braking power with the UC equipment, whereas, with the PM equipment the maximum service braking power (80 per cent) is obtained with the brake pipe reduction of approximately 19 lb.

Thus, with the smaller reservoirs, a longer time is available in which the engine man can exercise his judgment as to what rate of retardation is being obtained and how best to control the speed of the train.

This fact should be borne in mind whenever a comparison is made between the service applications of the PM and the UC brake equipments. The difference in reservoir volumes used necessarily results in a faster rate of building up of brake cylinder pressure with the PM equipment and large reservoirs than for the UC equipment with smaller size reservoirs for the same rate of brake pipe reduction. The effect of this was to provide a high minimum braking force and a low degree of flexibility.

RELEASING AND RECHARGING

When the brake pipe pressure is increased above that of the auxiliary reservoir the triple valve operates so as to:

(*a*) Connect the brake cylinder to the atmosphere through the release cavity in the triple valve slide valve, thus releasing the brakes.

(*b*) Connect the brake pipe to the auxiliary reservoir through the feed groove around the triple valve piston and so permit the auxiliary reservoir to be recharged.

After having moved to release position the triple valve piston remains there until a subsequent reduction of brake pipe pressure below the auxiliary reservoir pressure. Consequently, no graduation of the release is possible.

The certainty of releasing all brakes in the train depends on the possibility of establishing the differential pressure required to move the triple valve parts to release position. No difficulty is experienced in this direction so far as the cars at the head end of the train are concerned, but the increase in brake pipe pressure is necessarily slower at the rear than at the head end. Large auxiliary reservoir volumes, requiring a large amount of air for recharging (all of which must be drawn from the brake pipe), long trains, leaky brake pipe, poor condition of triple valve piston packing rings or slide valves, low main reservoir pressure, or light brake pipe reductions, all tend to bring about the slow rise of brake pipe pressure at the rear end of the train, which results in failure to release brakes, slow release, stuck brakes and dragging brake shoes.

Briefly stated, the compressed air entering the brake pipe is being called upon to perform two functions at the same time, viz.:

(*a*) Increase the pressure throughout the entire brake pipe throughout the train at a sufficiently rapid rate to insure the releasing of triple valves in whatever condition or position in the train they happen to be.

(*b*) Recharge the auxiliary reservoirs.

These two conditions are mutually antagonistic and conditions frequently arise where the release function is partially or almost completely nullified in the performance of the recharging function, which after a certain stage is reached may be the controlling factor in the release of the brake.

EMERGENCY BRAKE APPLICATION

In response to a rate of brake pipe reduction considerably more rapid than that established for service brake application the triple valve parts move to their emergency positions, in which the quick action parts of the triple valve are actuated so as to vent air from the brake pipe to the brake cylinder, thus

(*a*) Causing a local venting of brake pipe air on each vehicle and so transmitting serial quick action rapidly from car to car throughout the train.

(*b*) Supplementing the air flowing from the auxiliary reservoir to the brake cylinder, thus increasing the brake cylinder pressure, by the amount derived from the brake pipe.

The brake cylinder pressure in emergency is under the control of the high-speed reducing valve at all times. At first the reducing valve blows down the brake cylinder pressure at a very slow rate, but this rate gradually increases, being timed to become relatively rapid as the train nears its stopping point and the valve then closes at 60 lb. brake cylinder pressure.

It is impossible to obtain a quick action application with the PM equipment after a service application of any consequence has been made.

Features of the UC Equipment

The manner in which the functions of the universal control equipment are performed is described in full in the report of the tests. The valve mechanism which is the distinguishing feature of this equipment is of the "built-up" type which makes it possible to install and operate this equipment if desired in stages, by adding to the simplest arrangement of apparatus, including only those features required to give an operation equivalent to that of the PM brake, up to the complete form of the device.

Three arrangements of this apparatus were used in these tests:

Partial—equivalent to PM brake equipment.

Complete pneumatic equipment.

Complete electro-pneumatic equipment.

The partial form of the UC equipment was tried out chiefly to demonstrate the similarity of its action to that of the PM brake equipment. Aside from the fact that this was thoroughly demonstrated the tests with the partial equipment were of no special importance. Moreover, the trials with the complete pneumatic equipment, especially when mixed with PM equipments in the same train, did not disclose any reasons why the UC equipment in its complete pneumatic form could not be operated with the PM equipment during the transition period. Consequently, the arrangement of apparatus and the operation of the complete pneumatic and complete electro-pneumatic equipments were chiefly considered during the tests.

COMPLETE EQUIPMENT

The UC equipment, Figs. 16 and 17, in its complete form comprises a valve mechanism called the *universal valve* with its permanent pipe bracket and three reservoirs, the auxiliary, the service and emergency reservoirs.

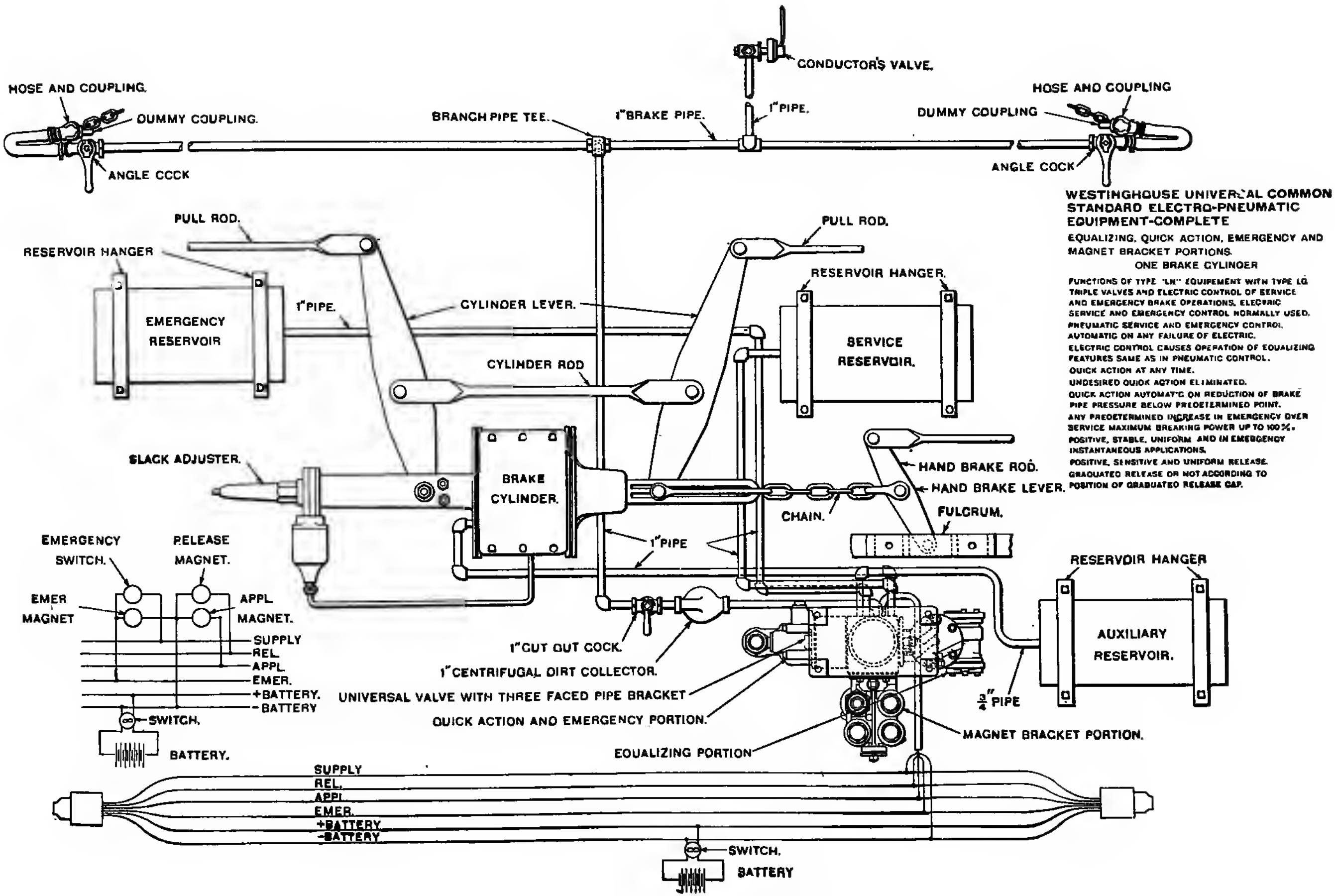


Fig. 16 The UC Equipment
A diagram of the complete electro-pneumatic brake

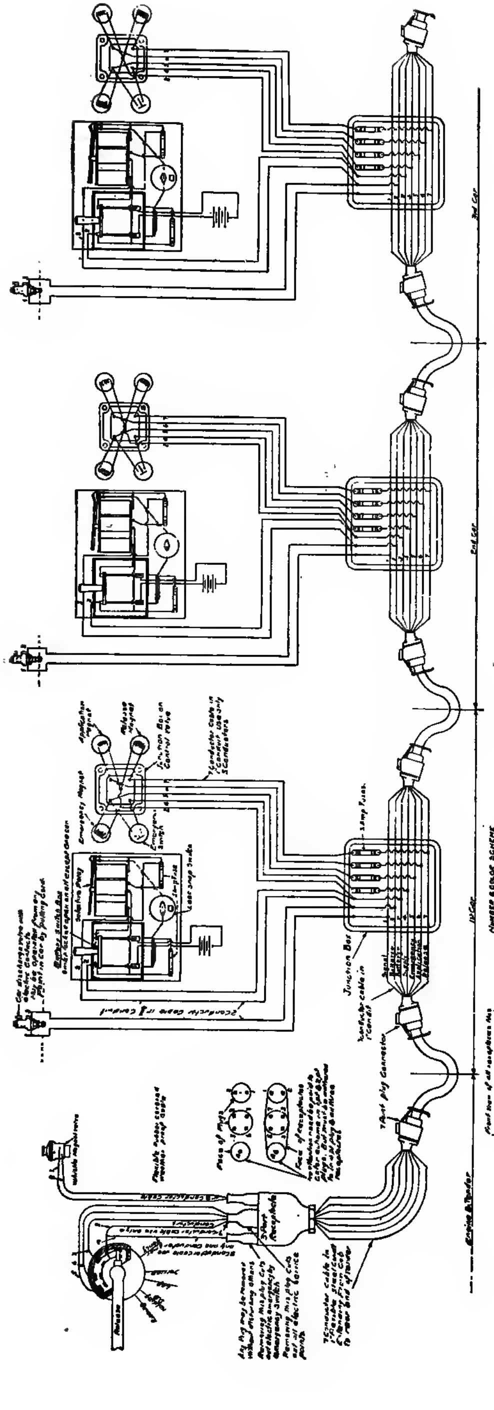


FIG. 17 UC EQUIPMENT
The electric circuits of the electro-pneumatic brake on locomotive and cars

The universal valve (Figs. 18 to 21), consists of an *equalizing portion,* which primarily controls the charging and recharging of the reservoirs of the equipment, the service application of the brakes and the releasing of the brakes.

A quick action portion with high pressure cap, which controls the

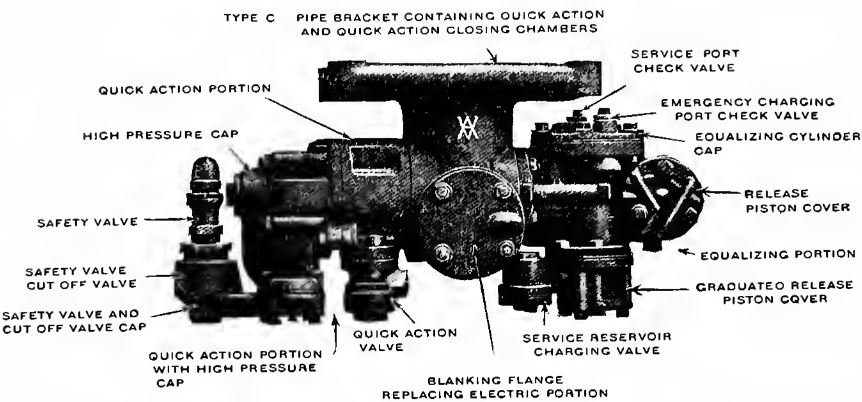


FIG. 18 UNIVERSAL VALVE, UC EQUIPMENT, FACE VIEW, WITHOUT ELECTRIC MAGNET PORTION

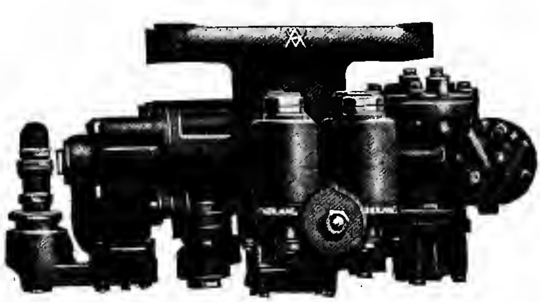


FIG. 19 UNIVERSAL VALVE WITH ELECTRIC MAGNET PORTION

transmission of serial quick action and obtaining of high emergency pressure in the brake cylinders when an emergency application of the brakes is made.

An electric portion, which comprises the magnets, switch, etc., controlling the electric service application, electric release and electric emergency applications of the brakes.

A pipe bracket, to which all pipe connections are permanently made and to which the various portions of the valve device are bolted. This bracket contains two small chambers, the quick action chamber and quick action closing chamber.

The *quick action closing chamber* provides means whereby the quick action outlet from the brake pipe to the atmosphere is open when an emergency application is made and is closed when a predetermined time thereafter has elapsed.

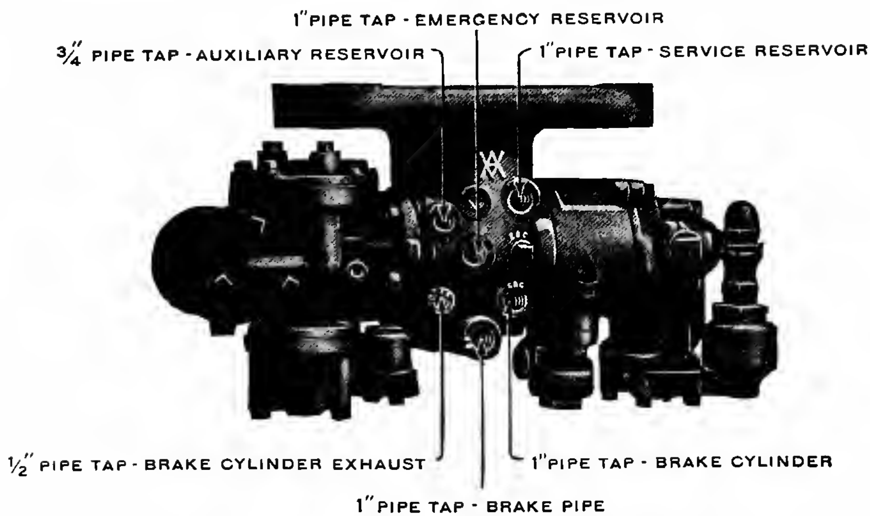


FIG. 20 UNIVERSAL VALVE, UC EQUIPMENT
The reverse side of valve showing pipe connections

The *quick action chamber* in connection with the quick action closing chamber controls the operation of the quick action parts of the valve in accordance with the rate of brake pipe reduction.

In addition to the above the equipment on each car comprises:

An auxiliary reservoir which is the same size for all sizes of brake cylinders, the pressure in which controls the movement of the equalizing piston and slide valve of the universal valve and supplies air to the brake cylinder.

A service reservoir which varies in size with the size of the brake cylinder. This, together with the auxiliary reservoir, supplies air for operating the brake cylinder in service and emergency brake applications.

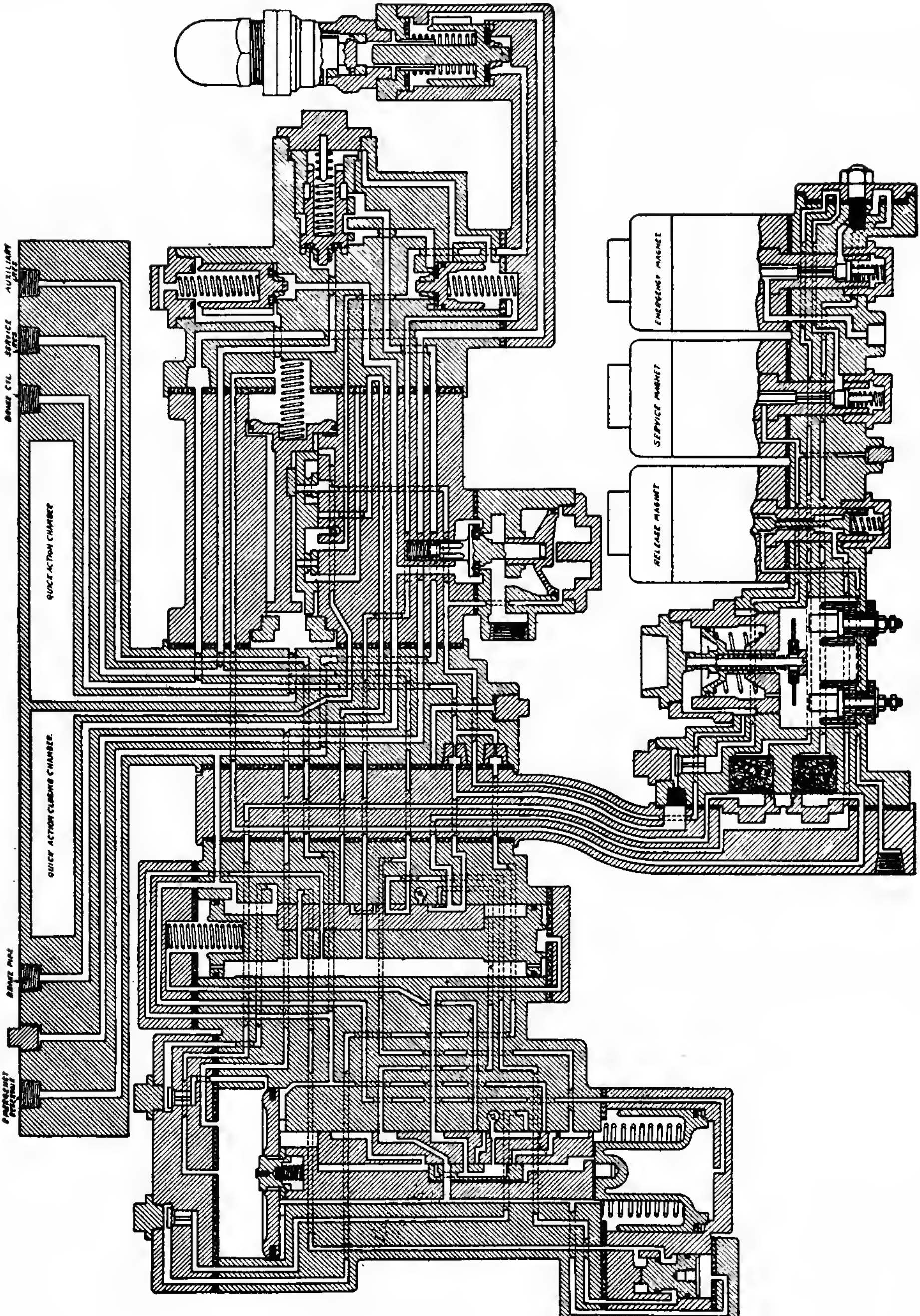


FIG. 21 UNIVERSAL VALVE, UC EQUIPMENT
Diagrammatic view of the valve showing sequence of parts and passageways

An *emergency reservoir* which varies in size according to the size of brake cylinder used and the amount of emergency brake cylinder pressure which the installation is designed to afford. This reservoir supplies air required to graduate the release of the brakes and to obtain a quick recharging of the service and auxiliary reservoirs after a service application of the brakes. It also provides the additional supply of air required to obtain the increased brake cylinder pressure desired for emergency applications.

In addition to the above, various electrical details are used on the locomotive and cars as illustrated in Figs. 17 and 22.

The valve mechanism is designed to require a drop in brake pipe pressure of approximately 4 lb. before it is possible to obtain an application of the brakes. The equalizing piston moves on a differential much lower than this, however, so as to close the feed groove and thus prevent back leakage from the auxiliary reservoir. Thus a service application of the brakes is positively insured when the required 4 lb. brake pipe reduction is reached. From this point the rise in brake cylinder pressure corresponds to the reduction in brake pipe pressure in the proper relation to produce a full service brake application (90 per cent braking power) for a brake pipe reduction of 24 lb.

The maximum brake cylinder pressure obtainable in a service application is limited by the setting of a quick blow-down and positive acting safety valve which is connected to the brake cylinder through the emergency portion of the universal valve at all times except when an emergency application of the brakes is made. When making an emergency application, the safety valve is automatically cut off from communication with the rest of the equipment. This safety valve is adjusted to limit the maximum obtainable service brake cylinder pressure to 60 lb. per sq. in.

EMERGENCY BRAKE APPLICATION AFTER SERVICE BRAKE APPLICATION

Whenever a predetermined emergency rate of brake pipe reduction is established the quick action and high pressure parts of the valve will operate as above described to start serial quick action and increase the brake cylinder pressure up to its full emergency value even though a partial or full service brake application had been completed or was in progress. That is to say, the obtaining of an emergency application of the brakes depends only on the functioning of the quick action parts and is entirely independent of the service operation of the valve.

EMERGENCY BRAKE APPLICATION AUTOMATIC ON DEPLETION OF BRAKE PIPE PRESSURE BELOW A PREDETERMINED POINT

Whenever, from any cause, the brake pipe pressure is reduced to a predetermined value (30 lb.), the protection valve included in the emergency portion of the universal valve will operate and cause the parts of the emergency portion to move to their quick action positions and so start a quick action application of the brakes—the operation of the equipment then being as already explained.

Within certain limits the percentage of emergency braking power and the brake pipe pressure to be used may be chosen as the conditions of operation and installation may dictate without requiring a change in any essential part of the apparatus and without affecting the fundamental and proper relations between the different reservoirs, cylinders and operating parts of the equipment. For example, one or two brake cylinders per car may be used as the weight of the car and the percentage of braking power desired may require, the only change necessary being the use of a special cap on the high pressure portion of the valve designed to handle two brake cylinders instead of one. The amount of emergency braking power can be fixed to suit special limits or requirements by proper choice of reservoir volumes and the arrangement of their connections. The equipment is designed to give normally an emergency braking power of 150 per cent when using 110 lb. brake pipe pressure; this insures a satisfactory stop on the one hand without the likelihood of injurious wheel sliding on the other, with an average condition of foundation brake rigging, track, etc. For the transition period, or where conditions of installation do not permit, or where the service requirements do not necessitate a braking power as high as this, a lower emergency braking power is available by the arrangement of reservoirs mentioned.

Complete Electro-Pneumatic Equipment

ELECTRIC SERVICE APPLICATION

When operating electrically, the service application of the brakes is actuated by a reduction in brake pipe pressure as when operating pneumatically. The equalizing portion of the universal valve causes the brakes to apply in response to a brake pipe reduction as when operating pneumatically. But this brake pipe reduction is made locally on each car (instead of all at one place, namely, the engineer's brake valve). This local reduction of brake pipe pressure is accomplished by means of the service magnet valves which open simultaneously on each car and vent brake pipe air to the atmosphere at the proper rate to produce a service brake application when the service magnets are energized by the engineer's brake valve handle being placed in service position. *The fact that the pneumatic and electric service position of the brake valve handle are the same insures the elimination of any delay in starting a pneumatic application of the brakes in case the electric control is, for any reason, inoperative.*

The movement of the brake valve handle back to lap position de-energizes the service magnets, which permits their valves to close and thus stop the brake pipe reduction. The valve parts then assume their lap positions as when operating pneumatically.

ELECTRIC RELEASE

Whether the graduated release cap is in direct or graduated release position, the release of the brakes can always be graduated electrically by alternately energizing and de-energizing the release magnets which control the flow of air from the brake cylinder exhaust ports to the atmosphere. These release magnets are energized and prevent the release of the brakes when the brake valve handle is in either release or holding position. At this time the brake pipe and reservoirs on the car are being recharged and the universal valve parts are in their release and charging positions. The outlet from the brake cylinder to the atmosphere is closed and the brakes cannot release so long as the release magnets are thus energized. The release magnets are de-energized and the exhaust of air from the brake cylinders permitted when the brake valve handle is in running position.

ELECTRIC EMERGENCY FROM ENGINEER'S BRAKE VALVE

In an electro-pneumatic emergency application, the emergency magnets on all cars are simultaneously and instantaneously energized. These magnets open their respective emergency magnet valves which in turn cause the quick action parts of each universal valve to operate and produce an emergency application of the brakes.

AUTOMATIC ELECTRIC EMERGENCY

In case a hose bursts or a conductor's valve is opened, the first universal valve to be affected by the resulting drop in brake pipe pressure will operate pneumatically.

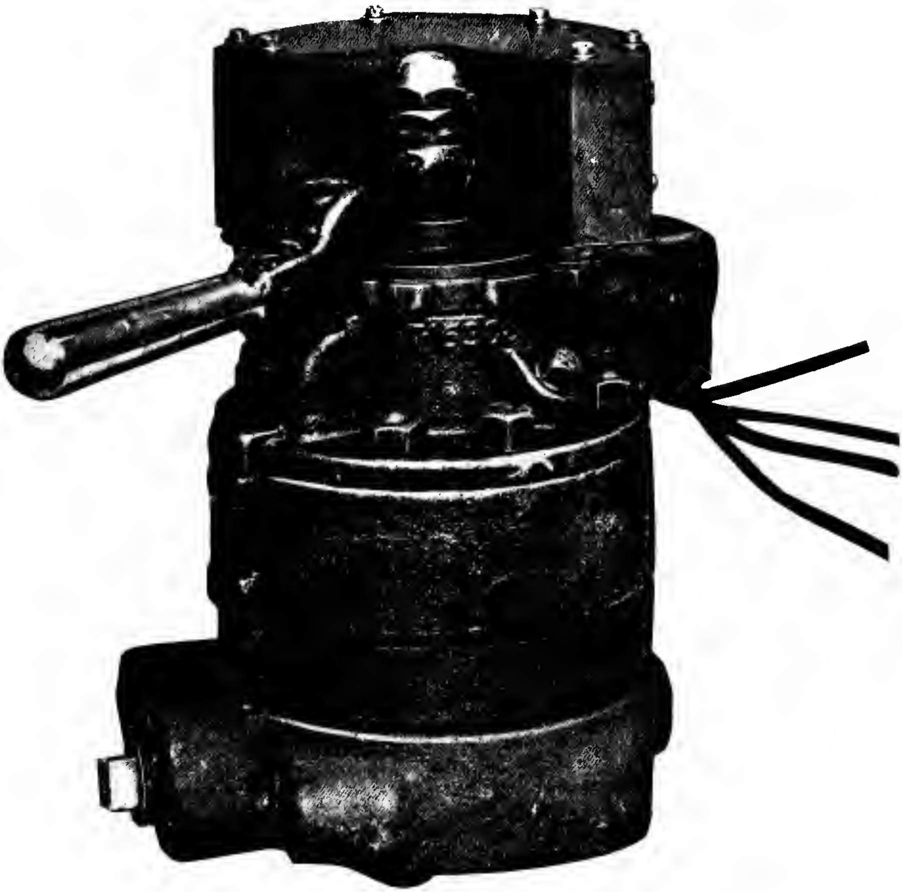


Fig. 22 Engineer's Brake Valve, UC Equipment
The valve is fitted with electric contacts

In so doing its emergency switch is closed which then energizes the emergency magnet circuit throughout the train, thus causing an electric emergency application on the rest of the cars as described.

The service and emergency magnets may be cut out if necessary by placing the magnet cut-out cap on the electric portion of the universal valve in the proper position.

Device for Obtaining Higher Brake Cylinder Pressure on Locomotive and Tender

The ET brake equipment as installed and in regular service on the locomotives previous to the tests was used at first without any change. As is well known, the standard braking power on the locomotive and tender (considering the ordinary working loads carried) is relatively a little less or about the same in stopping effectiveness as that of the PM equipment on the P-70 cars. This is shown by the fact that in breakaway tests under these conditions the locomotive ran but little if any farther than the cars.

When using the new and more effective car brake equipments at braking powers higher than those obtained with the PM equipment, the difference between the stopping force on the locomotive and cars was marked. The effect of this was to produce a noticeable running out of slack in ordinary train stops and in breakaway stops to cause the locomotive to run several hundred feet farther than the cars. This was especially true when the electro-pneumatic equipment was used.

This fact made it desirable to find out what could be done to make the locomotive and tender brake more nearly equal in effectiveness to that of the improved car brake equipments. Accordingly apparatus was devised while the tests were in progress and applied in an experimental form to the standard locomotive brake equipment. No modification was required in the existing apparatus except in the rearrangement of piping necessary to install an additional valve device.

This device was a "bypass valve," so arranged that the service operations of the ET equipment were not affected in any way, but when an emergency application of the brakes was made the bypass valve operated so as to short circuit compressed air directly from the main reservoirs to all the brake cylinders on the locomotive and tender. This resulted in a much quicker rate of rise of emergency brake cylinder pressure and a much higher maximum pressure being obtained than is the case with the standard ET brake. A possible effect of this high emergency braking power, if held until the speed becomes low, is to cause the drivers to slide. To protect against this the by-pass valve was arranged to hold the high initial brake cylinder pressure for a specified period (about ten seconds) and then, by a gradually accelerating blowdown, reduce this pressure so that as the speed of the train diminished the brake cylinder pressure would finally reach the normal emergency pressure standard with this equipment, namely, about 75 lb.

The results obtained with this device will be referred to later, but so far as its operation is concerned it may be stated here that the tests showed that such a device could easily be provided and made to give any desired rate of application, maximum initial pressure or rate of blowdown.

TESTS MADE AND RESULTS

Perhaps the most crucial and certainly one of the most conclusive tests to which the electro-pneumatic equipment was subjected was not on the list of scheduled comparative tests. The first test train completely equipped with the electro-pneumatic brake apparatus and clasp brake rigging in the P. R.R. shops at Altoona and with only a general trying out of connections and circuits in the yards, was sent over the road from Altoona to Atlantic City on a regular passenger train schedule. This was the first time that a steam railroad passenger train had been handled by means of electro-pneumatic brakes. The trip was made on schedule time without anything unusual occurring.

Full Service Brake Application

STANDARD PM BRAKE EQUIPMENT

The action of the PM equipment during a service application of the brakes, is illustrated by the curves (Fig. 33), plotted to show the brake cylinder pressure and corresponding percentage of braking power on the train as a whole, developed during the time required to make a full service brake pipe reduction.

The cars at the head end of the train *begin to apply* and *reach their maximum pressure* before those at the rear of the train, which is true of any form of pneumatically controlled brake. The time of commencing to apply on different cars varies through a range of about four seconds, which is an indication of the relatively slow serial response of the brake mechanism to a gradual fall in brake pipe pressure controlled by pneumatic means alone. From these cards and similar cards for the UC pneumatic equipment (Fig. 34), it is easily seen that there is a considerable time element involved in starting the service application of all the brakes in the train when operating pneumatically. It follows from this that the rate of "build-up" of brake cylinder pressure in a service application must be relatively slow in order to avoid shocks which result from the brakes applying with a slow serial action combined with too rapid "build-up" of pressure on the individual cars.

The *total time* required to *reach maximum full service brake cylinder pressure* is nearly 12 seconds, which shows clearly the effects of the long train (large brake pipe volume) in extending the time required to make a pneumatic full service brake application beyond a minimum which is fixed by the design of the *equalizing discharge feature of the brake valve* which requires that about 6 seconds at least be occupied in making a full service brake pipe reduction.

The *rate of rise of brake cylinder pressure* is more rapid than it would otherwise be, however, because of the use of the larger size auxiliary reservoirs with the PM equipments. For a given brake pipe reduction this results in a higher brake cylinder pressure than is obtained with a smaller size reservoir as used with the UC pneumatic equipment for the purpose of insuring flexibility of service operation of the brakes.

The maximum full service brake cylinder pressure is a trifle over 60 lbs., which is equivalent to a nominal braking power of 80 per cent. The variations in the setting and the individual action of the different high-speed reducing valves is the cause of the varying degrees of maximum braking power obtained on the different cars.

New UC Brake Equipment

The universal valve as originally applied was used throughout the tests without any modification whatever. Special attention to certain features of its operation, suggested the possibility of improvement without material modification.

These indications received the attention of the manufacturers and as a result slight modifications in the construction of the universal valve were made. However, rather than introduce factors which would affect the comparative value of the test results it was decided to complete the official program of the tests with the valves as first supplied, it being considered that the improvement accomplished by the modifications referred to could be amply illustrated by the results of the performance of the improved valve in laboratory tests. In fact, it is pertinent to point out here that laboratory test records, when properly analyzed, furnish all the information that can be desired with regard to the efficiency and effectiveness of the air brake apparatus so far as its immediate function (controlling of the air pressure in the reservoirs and brake cylinders) is concerned.

Therefore only Figs. 34, 35 and 36, which are diagrams showing the service action of the UC equipment obtained in laboratory tests subsequent to the road tests, will be described here. They correspond in all significant characteristics with the similar diagrams obtained from the valves used during the tests as shown in the complete report, but in addition these records show the performance of the universal valve improved in constructional details as an outcome of the tests.

Due to the service stability feature of the UC equipment the brake cylinder pressure does not start to rise quite as quickly as with the PM equipment, but the time of obtaining an effective brake cylinder pressure is the same with the UC and PM equipments. The time to the beginning of the rise of brake cylinder pressure is slightly longer for the UC equipment, Fig. 34, than for test 034 (Fig. 33) (PM equipment), but reference to the indicator cards showing other service applications for the PM equipment, e. g., Fig. 37, will show that on account of the ordinary variations to be expected in the action of the type *P* triple valve, due to the condition of the mechanism, it is likely to take just as long to start an application with the PM equipment as with the UC equipment having the service stability feature. Once having started, however, the UC equipment builds up pressure in the brake cylinder at a rate which is properly proportioned to the rate of brake pipe reduction. From this point on, it is similar to the action of the PM equipment installed on an equal basis.

Comparing Fig. 33 and Fig. 34, the effect of the different size reservoirs used when making a service application of the brakes, as already mentioned, is clearly seen in the different slopes of the application lines. This results in the PM equipment (Fig. 33) reaching its maximum pressure in 12 seconds, whereas, the UC equipment requires 16 to 17 seconds. These rates, it should be noted, are determined primarily by the rate of fall of brake pipe pressure which is relatively slow (pneumatic operation) on long trains because of the large brake pipe volume.

The maximum full service brake cylinder pressure averages 60 lb. which is equivalent to a nominal braking power of 90 per cent. This maximum pressure is limited to approximately 60 lb. by the safety valve used with the equipment.

When proper allowance is made for the difference in reservoir volumes used with the UC and PM equipments, it will be seen that the rate of development of brake cylinder pressure in service is substantially the same in each case, the rate of brake pipe reduction being the controlling factor.

This shows that there can be no undesirable difference in the action of the PM and UC equipments when operating together in the same trains in ordinary service. The results of tests made with trains of mixed UC and PM equipment, for the purpose of bringing out this particular point, confirm this statement.

ELECTRO-PNEUMATIC EQUIPMENT

The advantages of the electro-pneumatic control of the service brake are apparent from a comparison of Fig. 35 with Fig. 34. With the electro-pneumatic brake (Fig. 35), the application started almost simultaneously on all cars and built up to maximum brake cylinder pressure at a uniform rate. Furthermore, the rate of build-up of brake cylinder pressure is not dependent upon the length of train as it is in the case of any pneumatically controlled service application. The brake pipe reduction is accomplished locally on each car by the operation of the service magnet valves.

The maximum service brake cylinder pressure is obtained in about 8 seconds instead of 16 seconds, required by the same brake equipment operating pneumatically. As would be expected this more prompt and uniform action of the brakes produces a shorter stop.

The important advantages of the electro-pneumatic control of the service brake application are clearly brought out by a study of the curves of Fig. 35. They show the almost absolutely uniform and simultaneous action of all the cars in the train. In fact, when making an electro-pneumatic service application of the brakes, the application on different cars in the train is much more uniform than in the case of a pneumatic emergency application. This diagram also shows the great advantage of eliminating the time element in starting the application and the ability to quicken the rate of brake application on the entire train, and at the same time retain the necessary flexibility which enables the engine man readily to control the brake application as desired.

General Comparison of Service Application—PM and UC Equipment

Fig. 36 shows data from the same brake cylinder cards as plotted in Figs. 33, 34 and 35, but in this case plotted to show the relative time to start the application of the brakes and the time to obtain full brake cylinder pressure with the PM equipment and the UC pneumatic and electro-pneumatic equipment. The quicker and more uniform application of the electro-pneumatic equipment is still more clearly brought out by these curves.

Partial Service Followed by Emergency Application

PM BRAKE EQUIPMENT

Fig. 37 shows the results obtained with the PM equipment when a partial service brake application is made followed immediately by the movement of the brake valve handle from service to emergency position.

It is important to note in the first place the way in which the triple valves began to respond to the reduction in brake pipe pressure compared with the action shown in Fig. 25. The same type of triple valves were used, but the applications were on different days, being for Fig. 37 February 13 and for Fig. 25 April 23. There is as much difference between the time of starting to apply the P triple valves (Fig. 33 and Fig. 37), as between the P triple valves and the universal valve of the UC equipment (Fig. 33 and Fig. 34). All these examples show that the condition of the triple valve (which is a variable depending upon many things such as the weather, the lubrication of the valves, the amount of use they have had and so on) causes considerable variations in the results.

It is barely possible to distinguish signs of the emergency application from the shape of the curves of Fig. 37 when compared with a continuous full service application without any emergency with this equipment (Fig. 33). Slightly higher cylinder pressure was obtained on most of the cars sufficient to operate the high-speed reducing valves so as to cause the characteristic blow-down of emergency brake cylinder pressure, but the rate of obtaining brake cylinder pressure is the same as if no emergency application had been made.

It required about the same time to reach this 60-lb. brake cylinder pressure in this case as when no emergency application was made.

Moreover, it should be noted that the serial action of the valves remains the same. There was no serial quick action effect produced by the emergency application following partial service application. The length of the stop, as would be expected, is but little different from that which was obtained with a full service application of the brakes, without any emergency application.

UC PNEUMATIC EQUIPMENT

In the case of the UC pneumatic equipment an emergency application produces serial quick action and full emergency brake cylinder pressure, whether preceded by a service application or not, Fig. 38. Consequently, when the emergency application was made all the brakes applied simultaneously, the brake cylinder pressure rose at the usual emergency rate and the usual emergency maximum cylinder pressure was obtained.

From this it is plain that a very material increase in stopping power is possible by making an emergency application following a partial service application with the new complete pneumatic equipment. The result of this is to shorten the stop by about 300 ft. compared with that obtained with a full service application, no emergency application being made.

ELECTRO-PNEUMATIC EQUIPMENT

The action of the electro-pneumatic equipment (Fig. 39) is similar to that just described (Fig. 38), except that the time element due to the serial pneumatic application, both service and emergency, is eliminated, and the quicker rate of rise of brake cylinder pressure during the service application, which is due to the local venting of the air from the brake pipe on each car, produced by the electro-pneumatic service application feature. Both the service and the emergency applications occur on all cars simultaneously and the brake cylinder pressure rises as promptly on each car of a twelve car train as it would on a single car.

A direct result of the quicker rate of brake pipe reduction is that the partial service reduction determined upon is completed sooner and therefore the emergency application is made earlier in the stop than with the pneumatic equipment.

The result of these several advantages is to produce a much shorter stop (about 500 ft., Fig. 61) than with a full service electro-pneumatic application. This shows clearly the increased safety factor of the improved brake equipment over that now in service for conditions requiring the greatest possible stopping power after a service application of the brakes has been started.

As a matter of interest a composite brake cylinder indicator card and deceleration curve for both the PM and electro-pneumatic equipment have been plotted on Fig. 40 to show the effect of emergency application following a partial service application.

A comparison of the curves for the PM and electro-pneumatic equipment shows clearly the quicker rate of rise of service brake cylinder pressure and the quicker and much more effective brake cylinder pressure obtained with the new equipment when the emergency application is made. The results of this action are shown in the more prompt rise of and the higher value reached by the deceleration curve for the electro-pneumatic equipment and the correspondingly shorter stop obtained.

Figs. 38 and 39 are important as they demonstrate that with the UC equipment operating either electrically or pneumatically the engineer has available for use in any emergency that may arise a quick acting and fully effective emergency brake, no matter what manipulation he may have made previously. The additional safety factor insured by this means as compared with the absence of any such safety factor with the PM equipment is proportional to the difference in the stopping distance already discussed.

Emergency Application

PM EQUIPMENT

Fig. 41 shows characteristic brake cylinder indicator cards for PM equipment emergency applications. The rate of rise of brake cylinder pressure is slightly faster and the maximum pressure obtained slightly higher than would ordinarily be the case on account of the larger size of auxiliary reservoirs used.

The characteristic blow-down action of the high-speed reducing valve is clearly shown by the shape of the curves. The cylinder pressure is reduced from an average of about 78 lb. at the beginning to nearly 60 lb. at the end of the stop.

UC PNEUMATIC EQUIPMENT

Fig. 42 shows characteristic brake cylinder pressure cards obtained with this equipment, emergency application. The brake cylinder pressure rises almost instantly to its maximum value and is held without blow-down throughout the stop thus utilizing the air pressure available on each car to its fullest extent and effect. The results to be expected from this quick rise of brake cylinder pressure are offset to a certain extent, however, by the relatively slow rate of transmission of serial quick action which resulted in the stop being somewhat longer and not as smooth as would have been the case otherwise.

The brake cylinder indicator cards (Fig. 42), show that the time of transmission of serial quick action was slightly longer than with the PM equipment. This was due to operation of the valve mechanism and it was found possible to quicken the pneumatic serial quick action feature without any material change in the design of the parts. When so modified the time of transmission of quick action with the UC pneumatic equipment is practically the same as that with the PM equipment. It was not thought of sufficient importance to try out any of these valves so modified in the series of tests under discussion. But the universal valves being supplied for cars going into service at the present time are improved in this particular and the cylinder cards shown in Fig. 43 illustrate the results obtained with this improved valve in rack tests. Considering the improvement in the time of transmission of quick action indicated by these cards the stops would be improved correspondingly.

UC PNEUMATIC EQUIPMENT—IMPROVED

The relatively slow rate of propagation of serial quick action with the UC pneumatic equipment as used during the tests (Fig. 42), tended to produce undesirable slack action and rough emergency stops as already explained.

Improvement in this particular is especially desirable and has been accomplished as shown by indicator cards (Fig. 43). It will be seen that the pneumatic serial action with the improved valves Fig. 43, is slightly better than for the PM equipment as shown in Fig. 41 and that this has been accomplished without any sacrifice in the time to build up the maximum cylinder pressure which was characteristic of the valves used during the test (Fig. 42).

This material reduction in the time required for the transmission of quick action throughout the train will correspondingly minimize the effects of such slack action as may be unavoidable due to the effect of the locomotive and whatever slack may exist between the cars.

The quick rise of cylinder pressure and the high emergency pressure obtained with the UC pneumatic equipment is effective in producing a materially shorter stop, namely, from 200 to 250 ft. shorter than with the PM equipment. (Fig. 59).

ELECTRO-PNEUMATIC EQUIPMENT

With the electro-pneumatic equipment a simultaneous and almost instantaneous application of the brakes in the train is obtained. The valve mechanism on each of the cars causes the brake cylinder pressure to rise to its maximum as quickly as the physical limitations of the air brake and foundation brake gear installation as a whole will permit and the maximum cylinder pressure thus obtained is maintained without blow-down (Fig. 44). Thus, by means of the electro-pneumatic control, an ideal emergency application of the brakes is obtained.

It should be noted that the only difference between the UC pneumatic and electro-pneumatic emergency application is in the elimination of the time element in starting the application of the brakes on the various cars in the train. That this is an important gain, is shown by the fact that the emergency stops with the electro-pneumatic brake were from 200 to 275 ft. shorter than with the pneumatic equipment.

The electro-pneumatic emergency stops are from 350 to 550 ft. shorter than those obtained with an emergency application of the PM equipment. Furthermore, the very short stops made with the electro-pneumatic equipment were accompanied by the absolute elimination of shocks, due to the action of the brakes, except in so far as the effect of the locomotive brake differed from that of the brake on the cars. The effect of the locomotive was the only reason why the emergency stops with the electro-pneumatic brake were not absolutely without shock even at speeds as low as ten miles per hour.

GENERAL COMPARISON OF EMERGENCY APPLICATIONS. PM AND UC EQUIPMENT

Fig. 45 shows the data from the valves as used during the tests plotted to show the relative time to start an emergency application of the brakes and the time to obtain full emergency brake cylinder pressure with the PM equipment, and the UC pneumatic and electro-pneumatic equipments. The rapid and uniform rate of application of the electro-pneumatic equipment is very clearly shown by the curve when compared with the longer time required to start the application of the brakes and to reach the maximum brake cylinder pressure, especially toward the rear end of the train when using the PM equipment. It will be noted that the UC equipment operating pneumatically was somewhat slow in starting compared with the PM equipment and that although it reached its maximum pressure in about one-half the time required by the PM equipment on the train as a whole there was about three seconds required to transmit the quick action from the head to the rear end of the train. This sluggish action of the valve when operating pneumatically was the same throughout the series of tests covered by this report. The pneumatic emergency function of the valve has subsequently been improved, however, by a slight modification of the emergency portion of the device as already mentioned and the improved results thereby obtained are shown in Fig. 46.

The performance of the electro-pneumatic equipment as shown in Figs. 45 and 46, is substantially identical while in the case of the pneumatic equipment a marked improvement is shown by the material reduction in time element for the curves of Fig. 46 compared with those of Fig. 45.

MIXED EQUIPMENT STOPS

In order to tell how well the operation of the UC pneumatic equipment harmonizes with that of the PM equipment, when mixed in various ways in the same train, a number of tests were made, both service and emergency stops, and the action of the brakes and of the train as a whole during a stop carefully noted. Only a few of the typical tests made will be described here.

All service stops, no matter what the arrangement of the UC and PM equipments, were free from objectionable shock. They indicated that the gradual introduction into service of cars with UC equipment will result in an improvement in the handling of trains generally, without introducing any undesirable features of operation or manipulation.

The improvement in the release of the brakes was particularly noticeable, showing that not only was the release improved in proportion to the number of UC equipment cars in the train, but also that the more UC equipment cars, the greater was the certainty in releasing the PM equipment cars used in the same train. That is to say, the UC equipment actually assisted to insure a certain and prompt release of the PM brakes.

EMERGENCY TEST WITH MIXED EQUIPMENT TRAINS

The UC equipment was used with its standard nominal braking power of 150 per cent. Under these conditions shocks and somewhat rough stops were experienced with certain train make-ups and at some speeds, which, although more severe than would be desirable in ordinary train service, were not severe enough to be prohibitive, considering the necessity for and the infrequency of emergency applications and the material share which the relatively low braked locomotive contributed during these shocks.

TRAIN STOPS

EMERGENCY STOP AT 30 M.P.H.

High braking power on locomotive and mixed UC and PM equipments on cars as follows:

Car Nos..........	1	2	3	4	5	6	7	8	9	10	11	12
Equipment......	PM	PM	UC	UC	PM	PM	UC	UC	PM	PM	UC	UC

Fig. 47 shows separate indicator cards for the brake cylinder pressure on the locomotive and each car in the train, also the rate of reduction in brake pipe pressure on car six during the application.

The characteristic emergency action of the PM and the UC equipments is clearly shown. In the first half of the train the PM equipments appear to have a slight advantage in the time of starting the brake application, but by the time that any effective brake cylinder pressure is reached all the brakes are applying practically uniformly throughout the train. The more rapid application of the new equipment results in its maximum pressure being obtained before the maximum (but lower) pressure is obtained on the PM equipment cars. This stop was somewhat rough but the shocks were not sufficient to be called severe nor objectionable enough to indicate that such a combination would give rise to any trouble in general road operation.

GRADUATED RELEASE STATION STOP AT 45 MILES PER HOUR—UC PNEUMATIC EQUIPMENT

This test was made to illustrate the proper method of making a graduated release stop (Fig. 48). The initial application was made with one continuous full service brake pipe reduction, was then held a few seconds after which the release was graduated by moving the brake valve handle from lap to running position and then back to lap position, four distinct graduations being made. This resulted in the braking power being reduced substantially in accordance with the decrease in the speed of the train.

Except for the first three cars in the train, which felt the effects of a graduation of the release most promptly and consequently tend to release sooner than the cars toward the rear, the amount of brake cylinder pressure remaining when the train stopped was uniform throughout the train.

The indicator card shows that no graduation of the release was obtained on car seven. It was found on inspection that this was caused by dirt on the seat of one of the valves which condition prevented the emergency reservoir from performing its graduated release function.

This stop was very smooth from the beginning of the application until the train came to a standstill and the low cylinder pressure remaining at the end entirely eliminated the unpleasant surging usually experienced when the train comes to a stop with high pressure in the brake cylinders.

This shows that the brakes can be applied to give a high braking power when the speed is high, which is essential if time is to be saved in making stops and yet the stop made as smoothly as if only a very light retarding force had been applied.

STATION STOP

Mixed equipment on cars as follows:

Car Nos.	1	2	3	4	5	6	7	8	9	10	11	12
Equipment	PM	PM	PM	UC	UC	UC	PM	PM	PM	UC	UC	UC

The stop was made with a full service application followed by a release and then a second application to bring the train to a standstill, in other words the usual two-application method. The only feature requiring special mention is the behavior of the PM equipment on cars 7, 8 and 9 when the release was attempted after the first application (Fig. 49). The brake pipe pressure, as indicated by the record taken on car 6, was increased by a slight amount (about 3 lb.) when releasing from the first application, and preparing for the second application. This slight increase in brake pipe pressure released the three PM equipments and the three UC equipments in the first half of the train. The three UC equipments at the rear of the train were also released. But the PM equipment on cars 7, 8 and 9 failed to release.

This is a clear demonstration of the fact that the UC equipment was more sensative to release (as it was designed to be) than the PM equipment. In other words, the UC equipment on the three cars at the rear end released on a slight rise in brake pipe pressure which was not sufficient to release the PM equipments on the three cars preceding. These PM equipment cars, so far as the brake pipe pressure was concerned, were more favorably placed than the three UC equipment cars at the rear end which did release.

It is important to note that this stop was without any roughness or shock even though the braking power was considerably higher on some cars than on others. But as will also be evident from the indicator cards (Fig. 49), this difference in pressure was set up gradually and because of this the readjustment of slack took place gradually which means that no shocks or roughness could result.

III. BRAKE RIGGING

The brake rigging has only one operative function to perform but that one is important and not always easily accomplished, viz., to transmit the force of compressed air developed in the brake cylinder through the medium of rods and levers to the brake shoes in such a manner that the full amount of multiplication contemplated in the design of the lever system is realized in normal pressure between the brake shoe and the wheel.

The use of two brake shoes per wheel (clasp brake rigging) works partly to the advantage of the foundation brake rigging and partly to an improved brake shoe and journal condition particularly with regard to shoe, rail and journal reactions. This dual character of the foundation brake rigging should be clearly distinguished.

The only trustworthy indication we have of the relative performance of different brake riggings is in the length of the stop produced. But the stop is a resultant of both brake shoe and brake rigging performance, other conditions remaining the same. During these tests no satisfactory separation of the brake rigging and brake shoe performance during the process of stopping was effected. Consequently, when the stops produced by clasp brake rigging are being compared with those made by a rigging having but one shoe per wheel it is important to keep in mind that the performance in each case is a resultant of both brake rigging and brake shoe characteristics. The test results under consideration afford notable examples of the errors which would result if an attempt should be made to evaluate the effect of one of these factors without making due allowance for the influence of the other.

Existing Type of Brake Rigging on Cars

The brake which has been standard on P-70 cars since they were first built in 1907, is of the single shoe type without brake beams, the brake heads being suspended from the truck levers and hangers as shown in Fig. 23.

The brake heads are spaced by tie rods to give the shoes full bearing and to prevent the tendency of the shoes to run off the wheel.

The nominal braking power ratio of total nominal brake shoe pressure to empty weight of car of 80% in service application and 113% in emergency is obtained with a 16-in. cylinder and a truck lever of two to one and a total ratio of 7.8 to 1 for a car weighing 120,000 lb.

Previous to these tests it had been found that with the brake rigging anchored to the truck side frame there is a tendency for the trucks to be pulled toward the center of the car when the brake is applied. The pull of the body brake rod is resisted by the center plate bearing, and this resisting force, acting below the line of pull of the brake rod, results in a turning moment on the truck, tending to tilt them more or less inward. One observed result of this tilting has been to render the outside pairs of wheels more susceptible to wheel sliding. The standard brake as used in the tests had the truck

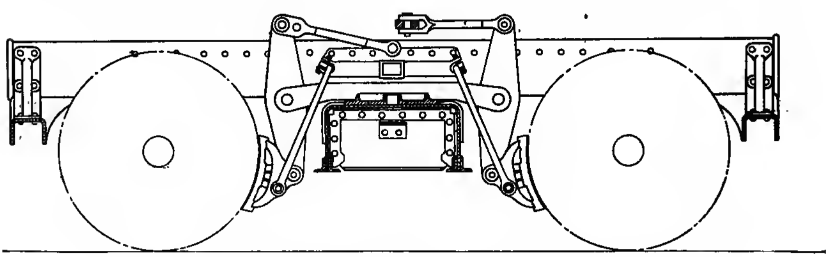


Fig. 23 Brake Rigging, Standard Single Shoe Brake
Standard rigging used on P-70 cars

rigging anchored to the car body and, this, as will be shown, effectively eliminated the cause of the tilting mentioned above.

Features Open For Improvement

To prevent the tendency for shoes to slide off the wheel it is necessary to use heavy tie rods and these at best do not offer a satisfactory solution of this difficulty.

To prevent the tendency to displace the journals due to the heavy unbalanced shoe pressures resulting from the use of a single shoe brake on a car of this weight, it is necessary to resort either to deep brasses or to hang the shoes so low that the resulting force passes within the bearing area of the brass. Both of these methods are objectionable. Lubricant must be supplied to deep brasses and the tendency of low hung shoes is to cause considerable false piston travel and consequently a low efficiency of brake rigging.

The amount of energy which must be absorbed per shoe by a single brake shoe when stopping modern heavy cars at high speed, taxes the material in the shoe beyond its capacity. Therefore, there is a tendency for the single shoe to break down and wear away more rapidly than was formerly the case with lighter cars, resulting in a correspondingly longer stop. These conditions are more fully discussed under the heading of brake shoes.

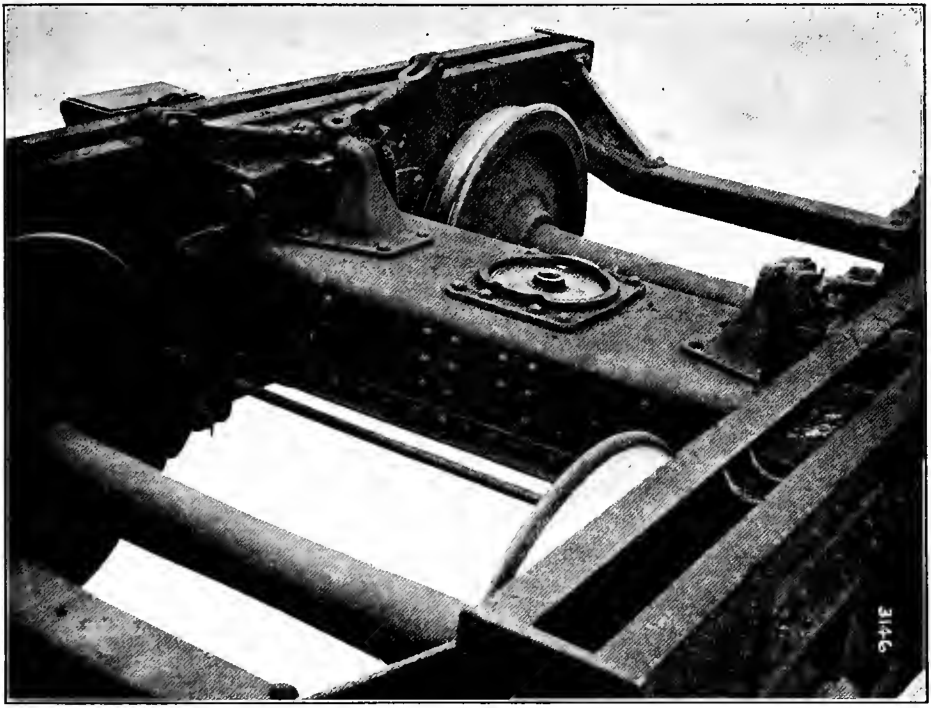


Fig. 24 Brake Rigging, Standard Single Shoe Brake
In the test the truck dead lever was anchored to the car body instead of to the truck as shown

Under the heavy loads imposed on the various members of the brake rigging system, the effect of deflection, pin wear and lost motion between parts requires especial attention when applying a rigging to a car weighing upwards of 120,000 lb., as these factors directly affect running piston travel.

The present practice in design, governing the unit fiber stress, although it provides ample strength for the brake rigging parts, causes deflection in some of the members resulting in an increase in piston travel. The present investigations point clearly to the advantage of using a fiber stress which reduces deflection and elongation to a minimum with as little increase in weight as possible.

Brake Rigging Requirements

These tests have developed certain principles, most of which were known before the test, but their relative importance was not established until a satisfactory design of clasp brake had been developed and stops obtained, which were anticipated from the application of

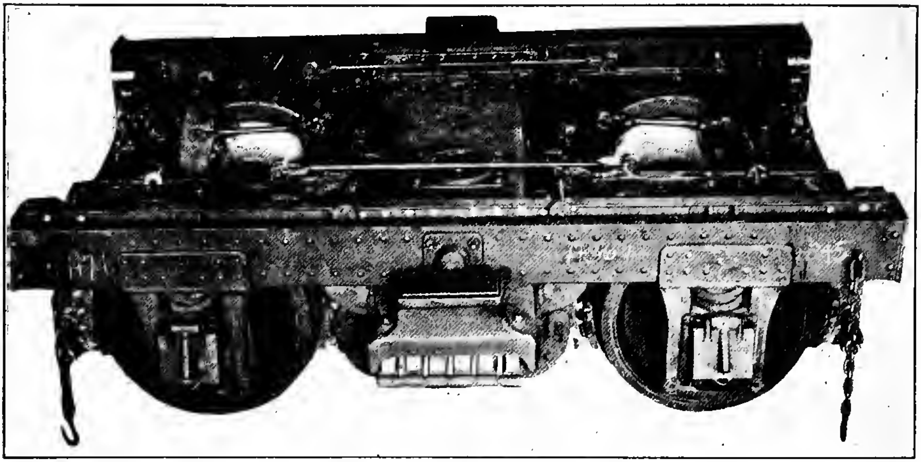


Fig. 25 Brake Rigging, No. 3 Clasp Brake

two brake shoes per wheel. The complete design of any brake rigging must be a compromise in which the relative values of each of the items herein enumerated have been given their proper consideration with respect to each other.

A Precaution against accidents that may result from parts of the rigging dropping on the track.

B Maximum efficiency of brake rigging at all times to insure the desired stop with a minimum nominal per cent of braking power, thereby reducing to a minimum the cost and weight of brake rigging and air brake equipment.

C Uniform distribution of brake force, in relation to weight braked, on all wheels, to insure the use of maximum retardation with minimum chances for wheel sliding.

D With a given nominal per cent braking power, the actual braking (which depends on the brake rigging efficiency) to remain constant throughout the life of shoes and wheels.

E Piston travel to be as near constant as practicable under all conditions of cylinder pressure, to insure maximum stopping efficiency for emergency braking and desired flexibility for service braking at low speed.

F Minimum brake shoe wear in doing a given amount of work in a minimum time.

G Minimum expense of maintenance and "running repairs" of the brake rigging between shopping of cars, for the purpose of expediting train movements.

H The parts to be designed so that they cannot be applied improperly, for the purpose of minimizing the possibility and probability of making wrong repairs.

I The initial and maintenance cost to be as low as consistent with, but secondary to, the points mentioned above.

Types of Rigging Tested

The various designs of rigging tested were the standard brake single-shoe type and three types of clasp brake known as Nos. 1, 2 and 3, respectively. The single-shoe brake shown in Figs. 3, 6 and 24 as used in these tests had the strength of various rigging members increased to allow for the use of 180 per cent emergency braking power, the truck dead lever anchored to the car body center sill and the brake shoes hung about 4 in. below the horizontal center line of the wheel. The effect of the position of the brake shoes was to cause a noticeable compression of the truck springs during brake applications, resulting in a corresponding horizontal movement of the brake shoes and a consequent increase in piston travel, tending to reduce the effectiveness of the brake.

In the improved (No. 3) type of clasp brake, Fig. 25, the members were so located that when the brake was applied, all the rods pulled perpendicularly to their respective levers and the pull rods rested on rollers, reducing the friction to a minimum. The shoes were hung as high as conditions would permit, being 2½ in. below the center line of the axle, the brake heads were pin connected to the hanger levers, enabling the shoe to adjust itself readily to the wheel.

Preliminary tests had demonstrated that even though the desired brake cylinder pressure could be obtained in an average minimum time of 2.2 seconds, the stops were longer than anticipated and in the development of the final design of clasp rigging consideration was given to all possible source of loss in the transmission of forces from the brake cylinder to the brake shoe, so as to provide as nearly as possible an instantaneous transmission of force with minimum loss.

Tests Made and Results

The standard (single shoe) brake rigging was tested under various conditions of speed, air brake equipment and braking power, using the complete train of 12 cars and locomotive, and also in 12 car breakaway stops. The clasp brake rigging was tested in single car breakaway tests only, there being but one of the test cars equipped with this type of rigging. Therefore, in comparing the different types of rigging it will be necessary to make the comparisons accordingly so far as the actual stops are concerned, although a method has been developed whereby the probable stop of a complete train can be computed with what is believed to be reasonable accuracy.

On account of the many different conditions of air brake equipment, per cent of braking power, and manipulation used with the different types of brake rigging, it is necessary to choose arbitrarily some representative combination of these factors and compare the different riggings all on the same basis. For this purpose the best available records are those of the so-called check runs, namely, emergency stops made at 60 m.p.h. with the complete train of 12 cars and locomotive using the electro-pneumatic air brake equipment and 150 per cent nominal braking power, and the breakaway stops under similar conditions. Furthermore, as the results obtained in these runs varied considerably, it will also be necessary to choose only those runs which represent the average of those stops that were not influenced by conditions which would tend to vary the stops on account of influences other than those resulting from the action of the different types of brake rigging. Table 2 has been made up on this basis.

In the above tabulation the average, the maximum and the minimum stops for both train and breakaway tests are shown with the different types of brake rigging. All tests which were affected by wheel sliding, by high braking power on locomotive or by other variable factors have been disregarded in making up this table. In order to make the comparison a fair one, the average per cent braking power for the various conditions is included in the table and an inspection of these data shows that the per cent braking power varied somewhat for the different types of rigging tested. This variation, so far as the difference between train and breakaway is concerned, necessarily follows from the effect of the lower braked locomotive.

Comparing the braking powers for the train using the No. 1 clasp, the standard, and the No. 2 clasp brake and for the 12 car breakaway stops with these types of rigging it will be seen that the variation is not sufficient to materially affect the average results obtained.

The No. 3 design of clasp brake was tried on a single car only. Consequently its performance has to be compared with that of the No. 2 clasp brake on a single car. The braking power per pound of

TABLE 2. EMERGENCY STOPS 60 M.P.H. WITH ELECTRO-PNEUMATIC BRAKE, 150 PER CENT NOMINAL BRAKING POWER

Kind of Brake Rigging	Kind of Stop	Stop Distance in Ft.			Average Braking Power, Train	Per Cent. Braking Power per Lb. Cylinder Pressure, Cars
		Average	Maximum	Minimum		
Standard	Train	1160	1298	1049	133	1.473
	12 car breakaway	1228	1291	1178	143	1.473
No. 1 clasp	Train	1204	1273	1157	135	1.499
	12 car breakaway	1145	1183	1112	148	1.499
No. 2 clasp	Train	1145	1213	1097	134	1.450
	12 car breakaway	1136	1178	1073	141	1.450
No. 2 clasp	Single car breakaway	1014	1027	1007	140	1.430
No. 3 clasp	Single car breakaway	924	1010	873	149	1.505

brake cylinder pressure for the No. 3 clasp brake was higher than that of the car having the No. 2 clasp brake as shown in the table and due allowance must be made for this, but after making due allowance, the stops with the No. 3 clasp brake are still materially shorter than corresponding stops with the No. 2 clasp brake.

To sum up, therefore, the relative performance of the several different types of rigging tested on the basis of stopping distance alone, would on the whole be arranged in the following order: Best, the No. 3 clasp brake, next, the two experimental designs of clasp brake, Nos. 1 and 2, and lastly the single shoe brake.

Brake Rigging Efficiency

The device shown in Fig. 26 was used to measure the efficiencies of the various types of brake rigging tested. The principle on which this device is based was the same for all riggings, modifications being necessary to make its application suitable to the various types of truck and brake rigging arrangements used.

A soft steel plate of known hardness and uniform structure was used to record the pressure with which a hardened steel ball was pressed against it, the depth of the impression being proportional to the force.

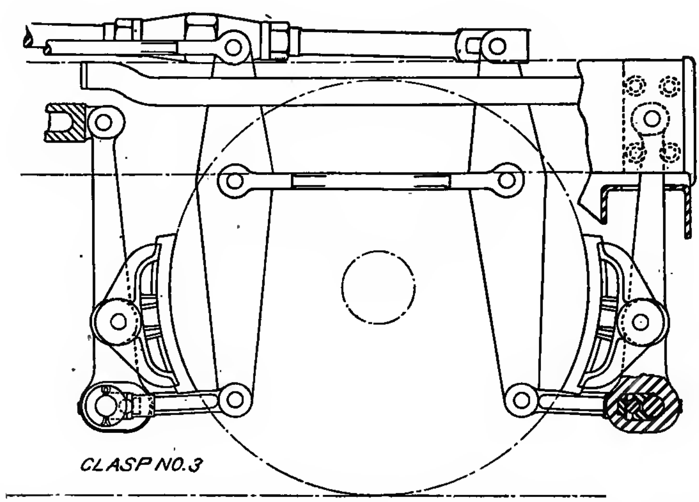


Fig. 26 Brake Shoe Pressure Measuring Device as Applied to the No. 3 Clasp Brake

The device holding the ball and plate was located in the brake rigging as near the shoe as possible so that the force transmitted by the brake rigging passed through the ball to the plate on its way to the brake shoe. The diameter of the impression was measured with a micrometer microscope and the corresponding pressure determined from a calibration curve of similar impressions made under direct known pressure on a testing machine.

The ratio of the pressure at the brake shoe as found by this method, to the brake shoe pressure which should result from the cylinder pressure and the total lever ratio known to exist, represents the mechanical efficiency of the rigging in per cent.

Tests of this character were made both when the cars were running and standing but on account of the disturbing influences encountered during the running tests, it was decided to consider only the standing tests, for which the data obtained were more consistent.

The data plotted in Fig. 50 should not be understood to signify anything with respect to the rigging efficiencies in running tests. In fact, from observation of the behavior of different types of rigging when making stops, it was evident that the efficiency of the transmission of the forces through the brake rigging were considerably different when making a stop than when a standing brake application was made, due to the different positions assumed by the brake shoes and levers, caused by pulling down truck springs and increase of piston travel. Furthermore, with the standard brake rigging considerable binding took place in the measuring device. This affected the accuracy of the readings and is probably one of the causes of the low efficiency shown in Fig. 50 for the standard brake rigging. Nevertheless, this curve is given here as a matter of information and to make the record of what was done complete.

With the other types of brake rigging, however, the results were remarkably consistent. In fact, by the aid of these records, it became possible for the first time to fix upon a logical basis for harmonizing the results obtained in road tests with those obtained in laboratory tests of brake shoes.

Piston Travel

One of the factors, affecting brake rigging efficiency, which was given particular attention during these tests was the variations in piston travel with different cylinder pressures. Records were taken of the length of piston travel on a time basis using for this purpose a piston travel indicator which was a slack action indicator (Fig. 11), adapted for this special purpose. The indicator drum was driven at a constant speed, as before, while the pencil mechanism was arranged to move in proportion to the travel of the brake cylinder piston. A typical indicator diagram from this device and curves derived from them are shown in Fig. 51. The piston travel time curves and the brake cylinder pressure time curves from the brake cylinder indicators were combined to form the brake cylinder pressure piston travel curves.

As a further study of the movement of the brake cylinder piston during the development of the brake cylinder pressure, diagrams were taken by means of a steam engine indicator screwed into the brake cylinder pressure head. The reducing motion of this indicator was connected to the piston cross head. Reproductions of typical indicator cards are shown in Fig. 52.

In Fig. 53 is shown the increase in running emergency piston travel over standing service for the different types of rigging tested and at various percentages of braking power. It will be noted that in these figures the increase in piston travel for the No. 1 and No. 3 clasp brake is less than for either the standard or the No. 2 clasp brake. The excessive piston travel on the No. 2 clasp brake was contributed to by low hung brake shoes and is an element tending toward a reduced over-all efficiency of the brake.

IV. PER CENT OF BRAKING POWER

Definition. The total brake shoe pressure to be provided by a steam road passenger brake installation is determined according to the empty weight of the car. For convenience the pressure afforded by a full service application of the brakes is chosen as the basis upon which different installations are classified and (also for convenience) the total brake shoe pressure is usually expressed in terms of its ratio to the empty weight of the car. The ratio of total brake shoe pressure to total car weight has always been termed "braking power" and is usually expressed as "per cent braking power." This misnomer might properly be replaced by the term "braking force" if the notion of the shoe pressure is uppermost or "braking ratio" if the idea of ratio alone is all that is to be implied. However, as "braking power" has been the universally accepted term for many years it will be used according to the common understanding of the term throughout this paper.

The term "per cent of braking power" requires further explanation in order that its significance in the different ways it has been in use may not be confused. Three distinct usages are:

A Nominal per cent braking power has been used in the common acceptance of the term, viz., the ratio of the total shoe pressure, calculated from the full service brake cylinder pressure, to the light weight of the car.

B Actual per cent braking power, based on brake cylinder pressure obtained, is referred to in several cases and is the ratio of the total brake shoe pressure calculated on the basis of the brake cylinder pressure actually obtained in any given test, to the weight of the car during that test.

C When referring to the ratio of the net resultant normal brake shoe pressure to the actual weight of the car in any given case the term braking power involves an allowance for the actual cylinder force developed and all losses in transmission included in the general term brake rigging efficiency. This ratio which is, properly, the *net* or *effective per cent retarding force,* is seldom referred to, except in connection with the study of the performance of brake shoes under known conditions of pressure and wheel loading, as in the case of the laboratory tests of brake shoes.

It has been common practice to design steam road passenger brake installations to produce a full service nominal braking power of 80 or 90%; the braking power obtained in emergency applications will then depend upon the characteristics of the installation. In the case of the P. R. R. P-70 cars with standard PM brake equipment the standard nominal braking power is 80%. In emergency application nominal 113% braking power is figured upon but in practice less than this is always obtained, due to the characteristics of the PM equipment whereby the maximum brake cylinder pressure is diminished by the action of the high-speed reducing valve, the effect of excessive piston travel and so on.

With the improved brake equipment the standard nominal braking power is 90%. This, although 10% higher than standard with PM equipment, is associated with a smaller size auxiliary reservoir which results in a slower rate of increase in braking power during the progress of a service application of the brakes. This affords greater flexibility in the manipulation of the brake in service and at the same time makes available a higher maximum service braking power. In emergency stops braking powers ranging from 90 to 180% were employed.

The objects of making emergency tests at different percentages of braking power were:

a To establish the relation between per cent of braking power and length of stop.

b To determine the limiting conditions which should be considered in fixing the maximum practicable emergency braking force consistent with reasonable protection against undue wheel sliding under the unfavorable conditions commonly experienced.

Length of Stop

The curves, Fig. 54, show the relation between the percentage of braking power and the length of stop, other factors being substantially constant. The curves are plotted for single-car breakaway stops from 60 m.p.h. with the No. 3 clasp brake, electro-pneumatic air brake equipment, plain brake shoes well worn in and cracked.

Taking from these curves the stop at 90% as the basis, the stop at 125% is 17.5% shorter, that at 150% braking power 25.5% shorter and that at 180%, 33% shorter. An interesting development from this is that for a given increase in braking power anywhere throughout the range, a constant decrease in length of stop will result, this constant decrease, however, not being equal to the corresponding increase in braking power. For example, an increase of 25% in braking power (from 90% up to 112.5% braking power) results in a decrease of about 12% in the length of stop (from 1177 ft. to 1033 ft.). Similarly, a 25% increase in braking power, from 144% up to 180% results in the same proportionate decrease in length of stop, viz.: from 896 ft. to 787 ft., which is a 12% decrease, as at the lower braking power.

An analysis of the curves on Fig. 54 shows that for single car breakaway stops from a speed of 60 m.p.h., using the electro-pneumatic brake the relation between the percentage of braking power and length of stop can be expressed by the following equation:

$$S = \frac{K}{P^x}$$

in which

S = the length of stop

K = a constant determined by the character of the air brake equipment, brake shoes and brake rigging.

P = percentage of braking power corresponding to the cylinder pressure obtained.

x = a fractional exponent, depending upon the effect of the percentage of braking power on the brake rigging efficiency and the coefficient of friction of the brake shoes.

This law is found to apply to both of the curves in Fig. 54 and that it holds for still lower braking powers than here shown was proven by both road and laboratory tests at low percentages of braking power (50 to 60%), the results of which satisfied the relation which had previously been found to exist between stops under similar conditions, but at percentages of braking power ranging between 90% and 180%. Manifestly, however, this law could not hold at very low percentages of braking power.

Referring to Fig. 54 the equation for the line showing the best 60 m.p.h. stop is

$$S = \frac{1107.5}{P^{0.581}}$$

and for the line showing the average 60 m.p.h. stops

$$S = \frac{1169.2}{P^{0.583}}$$

It should be noted here that these relations cannot be applied to other speeds, types of brake rigging or different brake apparatus, unless these characteristics are known to correspond to those which resulted in the curves in Fig. 54.

An approximate expression for the variation of percentage of braking power and length of stop with the electro-pneumatic equipment, is that, for an increase of 5% in the braking power, the stop is decreased 2%.

The above equations correspond to the theoretical relations which result from a consideration of the primary factors involved. Neglecting air and internal resistance on the one hand and the rotative energy of the wheels and axles on the other hand and considering the stop to be made on a straight level track, we have, for the portion of the stops that the brakes can be considered fully applied.

$$FS_2 = \tfrac{1}{2}\,\frac{W}{g}\,v^2$$

$$PWefS_2 = \frac{Wv^2}{2g}$$

$$S_2 = \frac{v^2}{2gPef}$$

or

$$S_2 = \frac{0.0334V^2}{Pef} = \frac{V^2}{30Pef}$$

in which

F = total retarding force in lb.

S_2 = that portion of the stop in ft. during which the brakes can be considered fully applied

W = weight of train in lb.

v = speed of train in ft. per second

V = speed of train in miles per hour

P = nominal percentage of braking power corresponding to average brake cylinder pressure during time brakes are applied

e = efficiency of brake rigging

f = mean coefficient of brake shoe friction

g = acceleration due to gravity 32.2 ft. per sec. per sec.

For the short time t, at the commencement of the stop during which the brakes may be considered as having no effect, the distance travelled will be $S_1 = vt = 1.467\ Vt$

The total length of stop S, is then

$$S = S_1 + S_2 = 1.467\ Vt + \frac{V^2}{30Pef}$$

For a given type of brake equipment and at a given speed, V and t are constants and the theoretical relation between the length of stop and percentage of braking power, other conditions being equal and as assumed above, is then

$$S = k + \frac{C}{Pef}$$

in which e, the efficiency of the brake rigging, remains practically constant throughout the range of values of P used in these tests (see Fig. 50). Let

$$\frac{C}{e} = C_1$$

f, the mean coefficient of friction is known to decrease as P increases, other conditions being the same. It may therefore be assumed that

$$f = \frac{c}{P^z} = cP^{-z}$$

where z is less than unity.

The expression for length of stop can now be written

$$S = k + \frac{C_1}{PcP^{-z}} = k + \frac{M}{P^{1-z}} = k + \frac{M}{P^x} = \frac{M + kP^x}{P^x}$$

In this expression k is a constant term resulting from the fact that the brake applies gradually instead of instantaneously to its maximum value.

But for practical purposes the formula

$$S = \frac{K}{P^x}$$

can, for the particular purpose in hand, be used as substantially the equivalent of the more accurate but less simple formula, provided that the proper value of K, which will make this possible, is determined.

This can be shown from the test data to be possible without involving an error of more than 2% at the extremes of the range of these experiments, the error being still less at intermediate points. Consequently, the simpler expression will serve as a satisfactory basis for a general approximation.

The expression is in the same form as that found empirically from the curves of Fig. 54 which is therefore justified from a theoretical as well as an empirical standpoint.

In the relation $f=\frac{c}{P^z}$ the values of c and z can be evaluated from the data furnished by the curves, Fig. 54. We have

$$S=\frac{1107.5}{P^{0.581}} \text{ and } S=\frac{K}{P^{1-z}}$$

Therefore

$$1\text{-}z=0.581$$
$$z=0.419$$
$$f=\frac{c}{P^z}=\frac{c}{P^{0.419}}$$

For $P=1.5$ (60 m.p.h. stops being considered now) we know that a fair average value for f is 0.10. Therefore

$$0.10=\frac{c}{1.5^{0.419}}$$

for which

$$c=0.12$$

Therefore an approximate relation between the mean coefficient of brake shoe friction and per cent braking power, resulting from the data of these tests (and applies therefore only to the range covered by these tests) for speeds of 60 m.p.h. plain cast iron brake shoes, clasp brake rigging, is

$$f=\frac{0.12}{P^{0.419}}$$

Wheel Sliding

Tests were considered as having excessive wheel sliding when the sum of all slides was 3000 ft. or over in 12 car train tests and 250 ft. in single car tests. These arbitrary figures were chosen when an analysis of the wheel sliding data developed that the sum of all slides, when less than 3000 ft. was not sufficient to have any material effect on the length of stop and was usually made up of relatively short slides on a number of pairs of wheels, but when more than 3000 ft. several pairs of wheels usually slid for the greater part of the stopping distance making the sum of all slides well above this figure. Slides under 15 ft. were not recorded.

Tables 3 and 4 show the number of tests made at each percentage braking power with each type of brake rigging and the number of tests in which the wheels slid 15 ft. or over.

It will be noted that with the No. 1 clasp brake wheel sliding occurred at low as well as at high percentages of emergency braking power. These tests, however, were run between February 10 and March 5 when there was a comparatively low prevailing air temperature which, with the high humidity characteristic of the locality brought about an adverse rail condition. The bad rail condition was especially marked in the first tests of the day which, with few exceptions, were electro-pneumatic emergency applications at 150% braking power.

Out of a total of 90 emergency tests at 150% braking power made with the No. 1 clasp brake, 62 tests developed no wheel sliding at all. Of the 28 in which there was wheel sliding, 13 were the first runs of the day, previous to which there was a period of two hours when the track had not been run over by other trains. This permitted an accumulation of frost or moisture on the rail during that interval. Few tests, made subsequent to the first run of the day, showed excessive wheel sliding, and it may be concluded that the rail condition referred to was chiefly responsible for the sliding that occurred.

The tests with the other types of brake were made later in the spring when the weather was more favorable to good rail condition, and it will be noted that in these tests there is a marked decrease in excessive wheel sliding.

An analysis of the percentage of runs with wheel sliding at various percentages of braking power shows that with plain shoes the amount of wheel sliding depends rather on the rail and weather conditions than on the percentage of braking power.

With the standard brake train flange shoes a high percentage of wheel sliding is shown, but on only one test was there excessive sliding. This may be partially attributed to the fact that it was run at 6 a.m. and subjected to rail conditions different from those prevailing during other tests made with this type of brake rigging.

A total of 282 emergency tests at 150% braking power were made

TABLE 3. CLASSIFICATION AND NUMBER OF TESTS WITH WHEEL SLIDING OVER 15 FEET

Type of Brake Rigging	Kind of Brake Shoes	90%				113%				125%				150%				180%			
		No. of Tests				No. of Tests				No. of Tests				No. of Tests				No. of Tests			
			With Sliding				With Sliding				With Sliding				With Sliding				With Sliding		
		Run	Over 15 Ft.	First Run of Day	Over 3000 Ft.	Run	Over 15 Ft.	First Run of Day	Over 3000 Ft.	Run	Over 15 Ft.	First Run of Day	Over 3000 Ft.	Run	Over 15 Ft.	First Run of Day	Over 3000 Ft.	Run	Over 15 Ft.	First Run of Day	Over 3000 Ft.
Standard (single shoe)	Plain C. I.	34	....	....	....	15	....	....	....	22	....	....	....	54	15	6	....	1	....	....	....
	Flange C. I.	....	....	....	....	4	....	....	....	8	2	....	....	12	8	2	1	14	10	....	....
No. 1 clasp	Plain C. I.	27	2	1	....	29	11	....	....	12	4	....	1	90	28	13	9†	20	5	....	1
No. 2 clasp	Plain C. I.	12	....	....	....	4	....	....	....	5	1	....	....	73	4	2	....	8	2	....	....
No. 3 clasp	Plain C. I.	27	1	....	....	3	....	....	....	1	....	....	....	38	2	1	....	7	....	....	....
	Flange C. I.	29	....	....	....	....	....	....	....	3	....	....	....	15	6	....	3*	9	1	....	....

†Seven of these were first runs of day.

*Single car, wheel sliding over 250 feet.

TABLE 4. PER CENT OF RUNS WITH WHEEL SLIDING OVER 15 FEET AT VARIOUS PERCENTAGES OF BRAKING POWER

Kind of Shoes	Clasp Brakes						Single Shoe Brake				All Brakes				
	90%	113%	125%	150%	180%	90%	113%	125%	150%	180%	90%	113%	125%	150%	180%
Plain C. I.	3	31	28	17	20	0	0	0	28	0	3	22	13	19	19
Flange C. I.	0	—	0	40	11	—	0	25	67	71	0	0	18	52	48

with the various types of brakes; of this number 22% had wheel sliding, 10% occurring during the tests of the No. 1 clasp brake.

At 180% braking power with plain shoes, wheel sliding occurred on but 7 out of 36 tests. With flange shoes sliding occurred in 11 out of 23 tests. In only one test of the 59, with 180% braking power, was there wheel sliding amounting to over 3000 ft.

From the above it follows, that the determining factor in wheel sliding is not high braking power alone but rather the uncontrollable conditions of rail and weather in connection with it, against which no permanent provision can be made without a sacrifice in the length of emergency stops during those favorable periods of the day or seasons of the year when conditions warrant the use of high braking power.

Whether the sliding of wheels will or will not cause flat spots of a size sufficient to produce rough riding of the car depends entirely on circumstances; for example, a condition of rail surface which will cause a considerable amount of wheel sliding, with relatively low percentages of braking power, is a condition which at the same time will permit long slides to occur without producing noticeable flat spots. On the other hand, when the rail is in good condition, or in the extreme case of a sanded rail, a very short slide may produce flat spots of a size requiring prompt attention. No flat spots were obtained of sufficient size to necessitate changing wheels during the tests, although, on account of the number of small spots accumulated upon the wheel tread, it was found advisable to change some wheels before the cars were put back into regular service.

Coefficient of Rail Friction

In using the machine shown in Fig. 12 to determine the relative condition of the rail surface during different tests, the same section of rail was used at all times. The kinetic coefficient of friction, or the ratio of the force required to keep the weights moving slowly and the pressure of these weights upon the rail, was found to give more consistent readings than observations of the "static" coefficient.

The kinetic values determined range between 12% and 35%, with the great majority of readings ranging between 22% and 30%. The records, when taken in connection with simultaneous readings of air temperature and relative humidity, show that the coefficient of rail friction decreases with an increase in the relative humidity for temperatures below the freezing point, whereas the coefficient of rail friction is not greatly affected by high humidities as long as the temperature is high, but begins to fall as the temperature approaches the freezing point.

As an interesting study of the effects of these various factors, consecutive observations were taken of the coefficient of rail friction, air temperature, relative humidity and barometric pressure for a period of twenty-four hours. These records are shown in Fig. 55 and indicate clearly the tendency of the coefficient of rail friction to increase with increasing temperature and decreasing relative humidity, and to decrease with increasing relative humidity and decreasing temperature. It is significant to note that the coefficient of rail friction obtained on the first reading in the early morning was practically the same as that found for a *well greased rail* later in the day.

While the absolute value of the data obtained may be open to question on account of the difference between the condition of the surfaces in contact in the case of the rolling wheel and the rail, and those which existed between the sliding weight and rail when taking the observations referred to, it is safe to assume that the values obtained at different times are comparative. It is apparent that the average rail condition changes with the seasons of the year (and, to a less degree, with the time of the day), and advantage can be taken of this fact by using a higher braking power in the summer than could be used in the winter without the likelihood of a material, if any, increase in wheel sliding.

The above conclusion logically follows from a knowledge of rail conditions in general and their effect irrespective of the readings of the apparatus used during the tests. As a matter of fact there was no great consistency between the readings obtained for coefficient of rail friction and the amount of wheel sliding experienced. While as a rule the greatest amount of wheel sliding occurred during the first runs of the morning, at which time the coefficient of rail friction was usually low, it was also a fact that occasionally considerable wheel sliding would be experienced when the coefficient of rail friction observed previous to such tests had been about at its average value.

A study of the action of the train where wheel sliding occurred and the coefficient of rail friction noted at the same time, led to the conclusion that other factors, such as shock, slack action, and foreign matter on the rail surface, have a controlling influence in causing wheel sliding. It was, therefore, concluded that the readings obtained for coefficient of rail friction at the particular point on the rail used for this purpose, could not be depended upon to indicate the probability of wheel sliding.

High Braking Power on Locomotive

To overcome the shock resulting from the maximum emergency braking power on the cars being higher and much more quickly obtained than that on the locomotive, an experimental device was applied to the locomotive which gave a higher maximum emergency braking power in a shorter time than is obtained with the standard ET equipment. The brake cylinder pressure was blown down to normal toward the end of the stop.

Fig. 56 shows comparative car and tender brake cylinder cards and slack action diagrams for 60 m.p.h. emergency stops, electro-pneumatic equipment, 150% braking power on cars, with and without the higher braking power on the locomotive. The sudden and considerable slack action on the records taken between cars 1 and 2 indicates clearly the shock received at the draft gears about two seconds after the brakes are applied when ordinary braking power is used on the locomotive. The slack action is much less severe on the diagram taken when the locomotive is braked higher than normally. A gradual change in the slack curves represents a comparatively slow relative movement between cars, which is not noticeable to passengers.

By referring to the time-pressure diagrams it is noted that at twelve seconds after the application, the pressure in the brake cylinder of the locomotive equipment is about the same with either the standard or modified locomotive brake apparatus. Therefore, such a method of controlling a high braking power feature would tend to prevent objectionable wheel sliding on the locomotive.

V. GENERAL DISCUSSION OF STOPS

By reason of the many combinations of conditions under which different tests were run, including different types of air brake equipments, methods of applying the brakes, nominal percent of braking power, types of brake rigging, brake shoe conditions, speeds and train make-up, a great variety of general comparisons might be made to illustrate the effect of these various conditions singly or in combination on the length of stop and the behavior of the trains during stopping. It will be possible to point out in this paper only the more significant and important comparisons which emphasize the salient features of the tests.

The shortest 60 m.p.h. emergency stop was made with a single car (locomotive not attached) with the No. 3 clasp brake electro-pneumatic equipment, 180 per cent braking power, and flanged brake shoes. The car was stopped under these conditions in 725 ft. The average retarding force for this test was 332 lb. per ton. This is equivalent to the resistance offered by a 16.6 per cent grade on which one end of a P-70 car (80 ft. long) would be 13.3 ft. higher than the other end.

This stop of 725 ft. from 60 m.p.h. made with a modern heavy passenger equipment car establishes a new record for a railway car stop.

Assuming a rail adhesion of 25 per cent, the shortest possible stop which could be obtained, by utilizing this adhesion to its maximum throughout the period of braking, would be 481 ft. This would require an ideal brake shoe and a controlling mechanism which would automatically adjust the retarding force of the brake, so that it would be at all times the maximum which could be used just short of producing wheel sliding.

The shortest 80 m.p.h. stop was made, with conditions the same as mentioned above, in 1422 ft. This is equivalent to an average retarding force of 310 lb. per ton.

From the data of stops made with locomotive alone and single car breakaway stops it is possible to calculate the approximate length of stop which would be obtained with a locomotive and train of twelve cars equipped with the electro-pneumatic brake.

Calculated from the results of single car breakaway tests, the best 60 m.p.h. train stop that could have been obtained with the means available during these tests is about 800 ft. and the best 80 m.p.h. stop about 1,570 ft.

The shortest 60 m.p.h. train stop with a locomotive and train of twelve cars was 1021 ft. This was made with high braking power on the locomotive and No. 1 clasp brake, electro-pneumatic equipment, 180 per cent braking power and plain shoes on the cars.

The shortest 80 m.p.h. train stop was made in 2197 ft. with high braking power on the locomotive and with No. 1 clasp brake, electro-pneumatic equipment, 150 per cent braking power and plain shoes.

Check Runs and Averages

In previous brake tests the average of two, or at the most three, stops under a particular set of conditions was thought sufficient to establish the average performance of the train, but a study of the situation revealed many variations in performance which could not be accounted for by any known differences in the equipment, adjustment, or manipulation. In order to determine the amount and cause of such variable performances as might result under supposedly constant conditions, a series of so called check runs was scheduled, one test to be made at the beginning and another at the end of each day's work, all conditions being kept the same throughout the entire series of tests as far as possible. These stops were all made from a speed of 60 m.p.h. with the complete train, standard braking power on the locomotive, electro-pneumatic equipment, 150 per cent braking power, and plain shoes on the cars.

As anticipated, it was found that even when the tests were run under supposedly identical conditions a variation in performance was obtained. It was observed that after the brake shoes were well worn in and no change was made in apparatus or manipulation for a considerable period of time the check runs of such a group of tests would show but little variation (for example five such tests averaged 1181 ft., with a maximum only eight ft. longer and a minimum only eleven ft. shorter than the average). However, when any change was made, such as in locomotives used, in per cent braking power of locomotive and tender, or in brakes shoes such as the replacement of a number of worn shoes by new ones, or the gradual wearing in of the brake shoes on the whole train, during the early part of a new series of tests, the length of stop obtained would vary, showing that the effect of new factors so introduced might be considerable.

A typical graphical record of the results of one complete series of check runs is shown in Fig. 57. This plot indicates clearly the variations which are unavoidable, except by the most careful provision for the constancy of all factors which have an effect on the stop. The brake shoe bearing is the most difficult factor to control and at the same time it is the most potent in producing variations in brake performance.

An example of the importance of the brake shoe condition and the manner in which it can be affected is afforded by one of the check runs with the standard (single shoe) brake rigging. This stop of 1049 ft. was the shortest 60 m.p.h. *train stop* actually made with 150 per cent braking power. Had the cars been stopped alone, i.e., without the locomotive, this stop would have been approximately 960 ft. This was not only the shortest stop made with this train, but was shorter than any check run with trains equipped with clasp brake rigging. An inspection of the record will show that this stop was made under peculiarly favorable conditions. While it was the first run of the day, the rail condition was good and the test followed nine tests at a low (90 per cent) braking power which insured the best possible shoe bearing. The favorable shoe bearing was further contributed to by the light service applications made during the movement of the train from and to the test ground and by the standing of the train over night. In this connection it is of interest to note also that the two other check run stops with this train (1091 ft. and 1076 ft., respectively) were each preceded by several runs at 90 per cent braking power. All of these tests were made while the brake shoe condition was, as a whole, satisfactory on the single shoe brake train.

On the other hand the *longest stops* of the check runs with any arrangement of brake rigging were also made with *this same train* (single shoe rigging), under identical conditions so far as could be provided, of air brake mechanism, brake rigging and all other controllable factors, but after the shoe conditions became unsatisfactory, due to many shoes running partially off the wheel. Three such stops were 1359 ft., 1361 ft., and 1389 ft. respectively from 60 m.p.h., or over 300 ft. longer than the short stops mentioned in the preceding paragraph. This shows that for a constant set of conditions (other than that of the brake shoes) the shortest and also the longest stops of the entire series of comparative tests were brought about by variations in brake shoe condition alone.

From the preceding it follows that the quality of the comparisons of air brake equipment and brake rigging may be considerably affected by the performance of the brake shoes. A knowledge of brake shoe performance under various conditions, therefore, becomes of prime importance in order to properly interpret the results of road tests. Furthermore, the knowledge gained during the road tests indicated the desirability of a more searching laboratory investigation along several original lines. Such investigations were made and are discussed in the chapter on Brake Shoes.

Tests with No. 3 Clasp Brake on Single Car

The No. 3 clasp brake was applied to but one car and in making tests this car was separated from the locomotive before reaching the point on the test track at which the brakes were automatically applied, so that the stop of the car alone might be observed. Such a method of making comparative tests of air brake mechanism, brake rigging, percentage of braking power and brake shoe performance affords an opportunity for controlling the conditions and insuring a freedom from influences other than those under investigation or beyond control which is attainable to a very much less degree in making breakaway stops with a number of cars and to a still less degree when making stops with a complete train of locomotive and cars. On the other hand, however, the fact that the individual cars necessarily differ in performance must not be overlooked, consequently the performance of a single car cannot be accepted unreservedly unless it is known that the car tested is fairly representative in every way of all cars in its class.

All of the stops made with the No. 3 clasp brake are shown in Fig. 58, the distance of the stop being plotted against the actual per cent braking power realized, calculated from the brake cylinder pressure observed for each test. Here again is shown a variation in length of stop of *nearly 300 ft.*, which can be attributed to variable brake shoe action alone, since other conditions were maintained substantially consistent.

An important consideration in this connection is that this variation occurred when even the longest of the stops were themselves relatively short. This is a condition that renders any further shortening of the stopping distance by any of the means *within control of the designer*, an exceedingly difficult task.

General Comparisons of Stop

Figs. 59-63 are chosen from the large number of similar comparisons contained in the complete report to illustrate how the data obtained permits of making a great variety of comparisons, according to the kind of comparative data desired. The diagrams are self explanatory, showing average results expressed in terms of the length of the stop under different conditions of train make-up, air brake equipment on cars and locomotive, kind of brake application, per cent of braking power, type of brake rigging, type of brake shoes, and speed.

Certain variables were encountered during different tests which were found to have more or less effect on the comparative value of the averages. As there was no satisfactory means for compensating for these variables, however, such averages as are so affected have been distinguished by Nos. 1, 2, 3, etc., which have the following significance:

(1) Car stop affected by locomotive.

(2) Braking power decidedly lower than tests of other brake rigging or of the same brake rigging at other speeds, with the same air brake equipment.

(3) Braking power decidedly higher than on tests of other brake rigging or of the same brake rigging at other speeds, with the same air brake equipment.

(4) Speed 4 m.p.h. or more, less than nominal.

(5) Bad shoe condition.

Comparisons Between Single Car and 12-Car Trains

In making comparisons of the effects of the various percentages of braking power on the length of stop of the complete train of locomotive and 12 cars it should be remembered that the nominal per cent braking power (unless qualified) represents the braking power of the cars. In train tests due to lower braking power of the locomotive and tender the per cent braking power of the train, as a whole, is somewhat reduced.

For example with a K locomotive and tender half loaded the following table shows the braking power of the cars and of the train as a whole:

Twelve Cars	Twelve Cars and K Locomotive (Tender half loaded)
90%	90%
113%	100%
125%	117%
150%	137%
180%	160%

The load on the tender was observed for each test, and due allowance was made for this in calculating the per cent braking power based on the actual brake cylinder pressure obtained for the entire train.

If of sufficient importance the effect of these variations must be studied by a reference to individual tests making up the average, complete information regarding which will be found in the original log sheets and tabulation of averages in the complete report. For general comparative purposes, however, the averages as shown in the accompanying illustrations are the best that the data afford.

Figs. 64 and 65 illustrate the characteristic change in length of stop for different initial speeds of the train, other conditions being substantially standard. Fig. 64 shows the average results obtained with locomotive and 12 car train using the ordinary standard (single shoe) brake rigging, electro-pneumatic air brake equipment and ordinary unflanged brake shoes with various percentages of braking power on the cars.

The curves shown for train stops from various speeds with the No. 3 clasp brake (Fig. 65) have been derived from the single car stops by assuming that the same ratio exists between the single car and train stops with the No. 3 clasp brake as was found to exist between the single car and train stops of the No. 3 clasp brake. This is believed to be the best possible calculation that can be made of the probable train stops with the No. 3 clasp brake (for which only single car breakaway tests were available) which would compare with those shown in Fig. 64.

Fig. 66 for the complete train and Fig. 67 for the locomotive illustrate the characteristic change in length of stop as the percentage of braking power is increased, other conditions remaining the same. A similar comparison is illustrated in Fig. 54 already referred to and in connection with which the general relation $S = \frac{K}{P^x}$ was derived. Comparing Fig. 54 and Figs. 66 and 67, it will be evident at once that the values of K and x will be different for each curve corresponding to the several combinations of conditions illustrated in the three figures mentioned.

Suppose that the locus of $S = \frac{K}{P^x}$ were plotted on logarithmic cross section paper. The locus is evidently a straight line. What has just been said then means that for different conditions of brake efficiency, brake rigging, speed and so on, the logarithmically plotted locus of the stopping distance corresponding to different percentages of braking power (within the range of the series of experiments under consideration) will always be a straight line having a characteristic slope corresponding to the characteristics of the brake rigging, brake shoe and (to a lesser extent) the type of air brake equipment used, and an intercept on the S (horizontal, logarithmic) axis corresponding to the speed and (to a lesser extent) to the brake rigging, brake shoe and air brake equipment characteristics.

From the data of all the single car, 60 m.p.h. breakaway stops the curves of Fig. 68 have been plotted to show not only the observed relation between the length of stop and per cent braking power but also how this relation changed for different types of brake shoes and brake rigging tested and again according to whether the several best or the average of all fairly comparative stops were considered.

Estimated Train Stops, No. 3 Clasp Brake

The various types of car brake rigging used during the tests were applied to the 12 cars of the test trains, with the exception of the No. 3 clasp brake rigging. The No. 3 clasp brake was applied to but one car and all tests made in connection with this rigging were single car breakaway stops. In order to make the data of these tests comparable with the tests of other types of brake rigging it would be necessary to compute the probable 12 car train stops of the No. 3 clasp brake from the data of the actual single car breakaway stops. With this object in view a series of separate locomotive tests were made in order to determine accurately the performance of the locomotive.

When the electro-pneumatic equipment is used the following approximate method may be employed for calculating the probable stop of a train of any number of cars and locomotive when the length of stop and weight of a single car and of the locomotive are separately known. This method does not take into account the difference between the time elements of the brake action on the locomotive and on the cars, which, however, is very small with this equipment, its effect amounting to about three ft. in the computation of a 1000-ft. stop. Let

W_t = weight of train, locomotive and cars in lb.
W_l = weight of locomotive in lb.
W_c = weight of a single car in lb.
V = initial speed in m.p.h.
S_t = length of stop of train in ft.
S_c = length of stop of a single car in ft.
S_l = length of stop of locomotive in ft.
F_t = average retarding force for entire stop of train
F_l = average retarding force for entire stop of locomotive
F_c = average retarding force for entire stop of a single car

The work done during the stop by the retarding force F_c acting through the distance S_c must be equal to the initial energy of the car, which is $\frac{1}{2}\,\frac{W_c v^2}{g}$, where v = initial speed in ft. per sec. and W_c = weight of car.

Therefore neglecting train resistance on the one hand, and rotative energy on the other

$$F_c\,S_c = \frac{W_c\,v^2}{2g}$$

If the velocity be expressed in miles per hour instead of feet per second, then

$$F_c\,S_c = \frac{W_c\,V^2}{30}, \text{ or } F_c = \frac{W_c\,V^2}{30\,S_c}$$

If more than one car is used let N = number of cars, then the total average retarding force on the cars will be

$$NF_c = \frac{NW\,V^2}{30\,S_c}$$

also for the locomotive $F_l \times S_l = \frac{W_l V^2}{30}$

and

$$F_l = \frac{W_l V^2}{30\,S_l}$$

But the total retarding force F_t acting on the trains of N cars and locomotive $= NF_c + F_l$

$$\therefore\ F_t = NF_c + F_l$$

also

$$(NF_c + F_l)S_t = \frac{W_t V^2}{30}$$

Substituting the values of NF_c and F_l just obtained

$$\left(\frac{NW_c V^2}{30\ S_c} + \frac{W_l V^2}{30\ S_l}\right) S_t = \frac{W_t V^2}{30}$$

Dividing this expression by $\frac{V^2}{30}$, we have

$$\left(\frac{NW_c}{S} + \frac{W_l}{S_l}\right) S_t = W_t$$

Solving for S_t, we have

$$S_t = \frac{W_t S_l S_c}{S_c W_l + N S_l W_c}$$

It will be noted that by means of the above formula a 12 car and locomotive train stop can be computed for any initial speed, provided the necessary single car and separate locomotive test data are known for the same speed.

For example, from the data of the best stops with the No. 3 clasp brake, flanged shoes, single car breakaway tests, we have:

W_c = weight of car = 125,200 lb.
$12W_c$ = 1,502,400 lb.
W_l = weight of locomotive = 414,800 lb.
$W_t = W_l + 12\ W_c$ = 1,917,200 lb.
S_l = 1552 ft., stopping distance of locomotive alone from a speed of 60 m.p.h.
S_c = 991 ft., stopping distance of single car alone at 90 per cent braking power from a speed of 60 m.p.h.

Substituting these values in the following expression

$$S_t = \frac{W_t S_c S_l}{S_c W_l + S_l\ 12W_c}$$

we have

$$= \frac{1{,}917{,}200\times 991\times 1552}{991\times 414{,}800+1552\times 1{,}502{,}400}$$

$$S_t=1075 \text{ ft.}$$

Where the special high-pressure emergency bypass valve was used with the ET equipment on the locomotive, a shorter stop was obtained and in order to work out the train stop according to the above formula for this condition it would be necessary to substitute for the value of S_1 = 1552 ft., the value of S_1 = 1228 ft., which was the average stopping distance of the locomotive from a speed of 60 m.p.h. when using the special high-pressure emergency bypass valve.

The data of the best single car breakaway stops made with the No. 3 clasp brake from a speed of 60 m.p.h. at various percentages of braking power, with both plain cracked and flanged cracked shoes, are shown in Fig. 68. Using these data as a basis and combining with them the data of the separate locomotive tests, complete train stops were computed as described and have been plotted in Fig. 69. It will be noted that for each type of shoe two curves are shown. In each case the dotted and solid lines represent the train stops which would be obtained if the locomotive were operated respectively with or without the special high-pressure emergency bypass valve. The difference between the solid line and the dotted line for the corresponding brake shoe condition shows directly the effect of the difference of using the locomotive with or without high pressure on a 12 car train.

Fig. 69 is also of interest in that it shows the gain from the use of flanged instead of unflanged shoes. The best single car breakaway stop with flanged shoes from 60 m.p.h. and 180 per cent braking power was 725 ft. This is equivalent to a 12 car train stop, using the standard ET locomotive equipment, of 795 ft. (Fig. 69).

Diagrams of Typical Stops

A considerable number of the data taken during tests were obtained by autographic recording devices, and can be combined on a single diagram to afford a convenient means of analyzing the various factors, and the effects which they produce with respect to the general action of the train during the stop. Numerous diagrams of this description for typical test runs are shown in the complete report from which examples, Figs. 70-78, have been chosen to illustrate the characteristic data thus obtained. Curves are given for the speed of the train, the building up of brake cylinder pressure and the braking power corresponding thereto, the deceleration in miles per hour per second, and the relative slack movement between cars at various points in the train as observed, all of which are plotted on a distance base with the corresponding time scale for comparison.

The deceleration curves are shown only on a sufficient number of representative diagrams to establish clearly the character of these curves. The slack action records are given wherever available. In some cases, no slack action device was used, and in other cases the record is imperfect.

These curves show graphically the effect of different rates of retardation at different points in the train. If all vehicles were being retarded alike, there would be no change in the slack between cars, and all of the slack indicator diagrams would be horizontal lines. As a matter of fact, however, as soon as the steam is shut off, the retardation on the locomotive when drifting is greater than that of the cars. Consequently, the locomotive holds back against the train, bunching the slack most at the head end of the train and less toward the rear. This is well illustrated in Fig. 75, and the action is similar in all cases. When the brakes are applied, in all cases except when the electro-pneumatic emergency application is used, the effect of the brake application is first felt on the locomotive, causing a further running in of slack, as shown by the more or less abrupt rise in the slack indicator diagram taken between cars Nos. 1 and 2.

It is to be noted that the slack indicator diagrams show merely the relative movement between cars, brought about by the different rates of retardation on the different cars in the train, which may or may not produce more or less severe shocks, depending upon whether the action takes place slowly or suddenly. The speed of this movement is shown by the abruptness of the rise or fall in the slack action curves. Consequently, a considerable amount of slack action may be shown by the diagram, indicating a considerable difference in rate of retardation at different points in the train, but if these differences are brought about gradually, there may be little or no shock noticed. On the other hand, a much smaller amount of movement between cars, if taking place suddenly, as shown by the more nearly vertical line on the slack action curve, might be accompanied by a noticeable shock. This relation between rate of slack action and shock experienced was marked throughout the tests and is particularly noticeable in the case of the service application stops. These show considerable changes in slack, but these changes occurred relatively slowly. Very little shock was felt during any service stops.

These slack curves, as a whole, are consistent in illustrating the action of the slack progressively throughout the train. Some apparent inconsistencies may be noticed, which are due to bunching of the slack previous to the brake application.

In the case of the electro-pneumatic equipment, the more prompt and effective application of the brakes on the cars results in the locomotive tending to run out as soon as the brake application becomes effective. The suddenness and degree of shock resulting from this run out depends largely upon the percentage of braking power being used on the cars and locomotive. After this initial adjustment of slack, however, the remainder of the stop is made with but relatively little movement between cars. When using high braking power on the locomotive (Fig. 75), the increased locomotive retardation tends to reduce materially the amount of slack action, compared with that produced with the ordinary braking power on the locomotive (Fig. 74), and for this reason it is desirable, when the cars are equipped with the electro-pneumatic brake.

Fig. 72 shows the result of the relatively slow serial action of the UC pneumatic equipment, for which the action of the slack was more nearly like that experienced with the PM equipment (Fig. 71).

Braking Power, Serial Action and Shocks—Low Speed Stops

There is but little on record concerning the likelihood of shocks due to a high emergency braking power quickly applied at low speeds, although the general impression is that emergency applications, even at sixty miles per hour or over, are likely to be rough or even dangerous to passengers. This is a wrong impression. It is well known to all who have observed the action of brakes at high speeds (and it was the invariable experience during these tests) that the higher the speed the less noticeable is the application of the brakes.

The use of the UC pneumatic and electro-pneumatic equipments at the same speeds, percentages of braking power and otherwise similar circumstances, afforded an opportunity to demonstrate the most important fact in this connection, namely, that the amount of braking power, by itself, has but little to do with the shock experienced. The rate of transmission of serial brake application in relation to the rate of build up of brake cylinder pressure on each car or, in other words the action of the slack between the different vehicles in the train, is the controlling factor. With the electro-pneumatic brake, in which simultaneous quick action of the brakes on all vehicles is obtained, there was no shock at any speed or percentages of braking power except the slight shock on the first few cars due to the running out of slack which was to be expected on account of the relatively low braking power on the locomotive. On the other hand with the pneumatic equipment, having an appreciable time interval between the application of the successive cars in the train, shocks were experienced.

The most marked evidence of the effect of the time element in the serial action of the brakes and resultant shocks, occurred during the emergency stops made from very low speeds. In four tests made with the UC pneumatic equipment at ten and at twenty miles per hour, the resulting shocks, especially in the last third of the train, were extremely severe, being in effect a collision between the forward end of the train, which was almost stopped, and the rear end upon which the brake application was having but little, if any, effect at the time the rear end run-in occurred. The fact that this test was made with the UC pneumatic equipment, which was relatively slow in transmitting serial quick action, undoubtedly caused more severe shocks than will be experienced with the considerably smaller time element of the universal valve as subsequently modified.

That shocks disappear entirely when the time element in the application of the successive brakes in the train is eliminated was shown by repeating the 10 m.p.h. stop using the electro-pneumatic equipment. Notwithstanding that the stop was made in a shorter distance than before (37 ft., instead of 42 ft., and 45 ft. respectively), the difference in the action of the train was marked. There was no shock or violent slack action although a very high rate of retardation was produced.

Characteristic Speed, Resistance, Deceleration and Power Curves

During any stop the retardation at different instants is dependent upon the resultant normal pressure of the brake shoes on the wheels, the instantaneous value of the coefficient of brake shoe friction, and the effect of the air and internal resistances. The difference between the effects of the rotative energy of the wheels and the opposing air and internal resistances is relatively so small, compared with that due to the action of the brakes, that it does not require consideration here. The brake resistances increase as the brake cylinder pressure increases, during the time the brakes are being applied. After the maximum brake cylinder pressure is reached with the UC equipment, no further change in brake cylinder pressure takes place. Consequently, the resultant normal pressure is substantially constant from that point to the end of the stop. If the coefficient of brake shoe friction was constant throughout the stop, the resultant resistance and retardation would then be constant, and the speed-distance curve would be a true parabola, while the speed-time curve would be a straight line. It will be of interest to check these conclusions with the results obtained in several typical tests.

In order to study the relations of the factors above mentioned to the best advantage, typical single car breakaway stops, and stops with the locomotive alone have been chosen. This eliminates the disturbing influences of slack action (which produce apparent changes in retardation which are not characteristic of the train as a whole) variations in the retardation on the different vehicles comprising the train (which would require the averaging of all data for all cars in order to arrive at a satisfactory relation between cause and effect) the non-uniformity of brake rigging and brake shoe conditions (which can not be as satisfactorily controlled on a train of a number of cars as on a single car) and the variable effect of the locomotive. With this in view, Figs. 76, 77 and 78 have been plotted to show the speed-time, and speed-distance curves plotted from the track chronograph records of the various stops chosen. From the speed-time curve, the time-deceleration and resistance curves have been plotted, the ordinate at any point on the deceleration curve being proportional to the slope of the tangent to the speed-time curve at the corresponding point. The fact that the same curve, referred to different scales, shows both the deceleration of the train and the instantaneous resistances to the motion of the train at any point during the stop, follows from the fact that the relation between resisting forces and resultant deceleration is constant, so long as the mass of the train is unchanged.

It will be noted that the curves showing deceleration and resistance are not horizontal lines beyond the point where the brake becomes fully applied. The resistance is not constant but changes more or less as the speed of the train is reduced, especially toward the end of the stop, where a considerable and continual increase in retardation is experienced. As pointed out above, this occurs during the time the brake cylinder pressure is constant, and consequently the resultant normal brake shoe pressure is substantially constant. The conclusion, therefore, follows that the other factor, viz., the coefficient of brake shoe friction is changing and that the character of the changes which it undergoes is accurately portrayed by the deceleration curves derived as explained. This characteristic change in the coefficient of brake shoe friction has been a matter of common observation in every instance where train tests or laboratory tests of brake shoes have been studied from this point of view. A further analysis of the influences which bring about the characteristic variations in brake shoe friction will be found in the chapter on Brake Shoes, but it may be stated here that with constant brake shoe pressure, the observed changes in retardation are brought about by changes in the character of the bearing surface between the brake shoe and the wheel. This in turn depends upon the temperature of the metal doing the work, which is dependent upon the rate at which energy is being absorbed; and this varies with the decreasing speed of the train.

Having the deceleration and resistance curve thus plotted on a time basis, the value of the resistance at different distances from the point of brake application can be determined by the aid of the speed-time and speed-distance curves. In this way, the distance-deceleration and resistance curves were plotted. The work done during any portion of the stop is proportional to the area under the distance-resistance curves.

In order to obtain the average resistance to the motion of the train, it is necessary to integrate this curve. The shape of the curve at the beginning is determined by the more or less rapid rate of rise of brake cylinder pressure, which depends upon the kind of brake equipment being used. In order to arrive at a relation between the force developed in the brake cylinder or at the brake shoe, and the resulting retarding force, it is advantageous to consider this relation from the standpoint of constant brake cylinder, or shoe pressure, throughout the stop. Consequently, it is desirable to replace the effect of the variable pressures acting during the time the brake is being applied by their equivalent effects, had the maximum pressure developed, and held throughout the stop, been realized instantaneously. This can be done by determining the point of equivalent instantaneous application of retarding force, which is the point at which the maximum retarding force initially developed could have been applied instantaneously to produce the same effect on the speed of the train, as was realized from the gradual building up of retarding force that actually occurred. The amount of work required to reduce the initial speed of the train to the value which existed at the time the retarding force reached its initial maximum value, is proportional to the area under the distance-resistance curve up to this point. The same amount of work would be represented if the gradually rising resistance curve were replaced by a vertical line, representing the development of the same maximum retarding force instantly, but at a point such that the work done in the two cases is the same.

Having determined the point of equivalent instantaneous application of the retarding force, the average resistance, *during the time that the brake may be considered fully applied,* can be found by integrating the area under the entire distance-resistance curve and dividing this area by the total length of stop minus the distance from the start to the point of equivalent instantaneous application. The average resistances in pounds per ton, shown in Figs. 76, 77 and 78, were determined in this manner.

TABLE 5. EMERGENCY 12 CAR TRAIN STOPS FROM 60 M.P.H.

Brake Rigging	Brake Shoes	Air Brake Equipment	Nominal Braking Power	Stop	
				Best	Avg.
Standard	Plain	PM	113	1659	1677
Standard	Flanged	PM	113	1453	1453
No. 1 clasp	Plain	UC pneumatic	125	1405	1443
No. 2 clasp	Plain	UC pneumatic	125	No tests	
No. 1 clasp	Plain	UC electro-pneumatic	125	1338	1339
No. 1 clasp	Plain	UC electro-pneumatic	150	1157	1204
No. 2 clasp	Plain	UC electro-pneumatic	125	1317	1332
No. 2 clasp	Plain	UC electro-pneumatic	150	1097	1145
No. 3 clasp	Plain	UC electro-pneumatic (estimated)	125	1055	
No. 3 clasp	Plain	UC electro-pneumatic (estimated)	150	965	
No. 3 clasp	Flanged	UC electro-pneumatic (estimated)	125	935	
No. 3 clasp	Flanged	UC electro-pneumatic (estimated)	150	860	

The amount of work done during any interval is equal to the corresponding change in kinetic energy. If the change in kinetic energy for consecutive intervals throughout the stop is calculated from data afforded by the speed-time curves, and these values plotted at times corresponding to the average speeds during these intervals, the locus of the points so determined will be a curve representing the rate at which work is being done throughout the stop or, in other words, the power being developed at each unit interval of the stop. Curves so determined are indicated on Figs. 76 and 77, as power-time curves. This curve is referred to four different scales, designating thousands of foot pounds per second, horsepower, total B.t.u. per second, and B.t.u. per brake shoe per second. The constant relation between these four scales is obvious.

These power-time curves are of particular interest in the study of the rate at which energy must be absorbed by the brake shoes during the progress of the stop, as will be discussed later, under Brake Shoes.

Actual and Estimated Stops

A SUMMARY OF GENERAL RESULTS WITH A TWELVE CAR TRAIN

Table 5, giving emergency 12 car train stops from 60 miles per hour summarizes the stops obtained during the road tests and shows briefly the gain in stopping distance to be obtained by the use of flanged shoes, clasp brakes and the UC pneumatic or electro-pneumatic equipment.

VI. BRAKE SHOES

Road Tests

The condition of the material in the brake shoes, the manner in which they are adjusted to fit the wheel, and the bearing which the shoes have upon the wheel will materially influence the length of stop, having the greatest effect at the higher speeds. These agencies, although previously observed and recognized in a theoretical way, had never been so forcibly impressed upon observers as during the present series of tests and the information gained from the performance of the brake shoes in the tests has developed some noteworthy facts in regard to them.

Prior to starting the tests it was arranged to have the brake shoes made from special heats and to be as near as possible to the normal composition of our standard cast iron shoes in order that the variation in brake shoes might not affect comparisons of air brake equipment and the brake rigging.

Before applying these shoes to the test train they were broken in on other trains to wear off the face surface or slight chill incident to foundry practice. When so broken in, with an average of ⅛ in. of thickness removed from the face, Brinell hardness readings of the surface were taken and the shoes grouped according to hardness numbers. The results of this preliminary work, showing the grouping as to the number of shoes obtained from two heats in range of hardness and the chemical analysis are shown in Table 6.

At the time of breaking-in the shoes for the test train a series of tests was made to determine the variation in hardness with respect to wear. The results of this investigation are shown in Fig. 79 which indicates that when the shoe is new a wide variation in hardness may be expected between shoes made from the same heat, but that as the shoe wears down the hardness becomes fairly uniform and remains so, decreasing slightly as the shoe wears out. Although at that time no data were available, it was thought that Brinell hardness was an indication, not only of the ultimate strength of the face metal of the shoe but also was related in some way to the coefficient of friction.

Later tests confirm this belief but at this time the degree of this relation is not known. The indications are, however, that the harder the cast-iron material the lower will be the coefficient of friction and that the highest possible coefficient of friction from cast iron (at least under clasp brake conditions) is obtained with a material approaching the condition of the shoe when about three-fourths worn out and having a Brinell hardness number of about 190.

The greatest uniformity of action and the highest friction seem

TABLE 6. RANGE OF HARDNESS AND NUMBER OF SHOES UNDER EACH GROUP

First Heat

Range of Brinell Hardness Numbers	160 to 170	170 to 180	180 to 185*	185 to 190*	190 to 195*	195 to 200*	200 to 205*	205 to 210*	210 to 215	215 to 220	220 to 230	230 to 240	240 to 300	Total
No. of shoes	6	29	39	27	54	73	99	102	95	31	21	8	8	59
Second Heat														
No. of shoes	1	136	474		38		....	....	....	....	....	....	....	649

*Shoes placed on test train

Analysis of Shoe

	Per Cent		Per Cent
Graphitic carbon	2.71	Phosphorous	0.53
Combined carbon	0.69	Silicon	0.92
Manganese	0.46	Sulphur	0.13

to be obtained when the brake shoe bearing on the wheel is best. This condition would naturally be expected to follow a series of relatively light applications in the course of which the effects of temperature in warping the shoe are kept a minimum.

It may be seen here that this most desirable condition is precisely what results from the continued use of the brakes during service stops on the road and consequently the brake shoes as worn in ordinary train service are in the most favorable condition for making short emergency stops. In the tests this condition appeared at times to have been reached after several light braking power runs. Stops following such a series of runs were then shorter than similar stops following several tests at higher braking powers, other conditions being the same. This result seemed to be most consistently obtained with the single shoe train.

The schedule of tests with the No. 3 clasp brake was arranged with this in mind but the effect was not so noticeable in this case, there being indications that a good shoe bearing with the clasp brake might follow a small number of tests made at high braking power as well as a much larger number of tests made at lower braking power. The influence of previous tests on the shoe bearing for any particular test under consideration is so obscured by other conditions, such as the general wearing in of the shoes that necessarily results at any braking power, that the observations of the effect of the shoe dressing runs are of questionable value.

The influence of shoe bearing on the length of stop is more fully referred to in the data obtained on the check runs which have already been discussed.

Cracked or Slotted Shoes

A consideration of the results of the warping of the shoe led to the conclusion that the warping could be largely eliminated by slotting or cracking the shoes so that they would be more free to conform to the contour of the wheel.

As a matter of fact, this cracking takes place with either plain or flanged shoes after a number of runs have been made, and while the effect of this cracking on the length of stop is not accurately obtainable from the data of the road tests it was made the subject of further investigation in the laboratory tests of brake shoes.

Plain Shoes—One-Half Area

As a further study of the influence of bearing area on the performance of brake shoes, tests were made with the same shoes as in the slotted shoe tests but with the ends broken off and their area reduced by 50 per cent. The first stop was made in 1031 ft. This was almost as short as the shortest stop made under similar conditions with the full area unslotted shoes (1007 ft.). Subsequent tests with these partial area shoes resulted in stops of 1210, 1190, 1193 and 1134 ft. All of these stops were with the brake shoes at a high temperature. These results tend to confirm the conclusion that the bearing area rather than the total face area of the shoe is the important factor in brake shoe performance, and that the bearing area on the first test with half area shoes was substantially as effective as that of the full area solid shoes which were undoubtedly affected by warping to a considerably greater extent. Furthermore, the much longer distances run in the four stops following the first with the one-half area shoes demonstrated the effect of shoe temperature which was offsetting the probable tendency of the better bearing area condition originally secured as a result of the reduced warping effect.

Flanged Shoes

The advantage of an increased bearing area was demonstrated beyond question by the fact that the use of flanged brake shoes after being worn to a satisfactory bearing resulted invariably in a shorter stop than under similar conditions with unflanged shoes. The shortest stops made in the entire series of tests were with flanged brake shoes and their use shortened the stop approximately 12 per cent as compared with the best similar tests in which unflanged shoes were used under similar conditions. This comparison is illustrated graphically in Fig. 69.

LABORATORY TESTS OF BRAKE SHOES

Up to the time of these tests there was no definite laboratory test information which would apply to the particular braking conditions under investigation, especially with reference to the actual brake shoe performance as distinguished from the brake rigging performance.

To supplement the road tests, a series of laboratory tests was carried out on the brake shoe testing machine of the American Brake Shoe and Foundry Company, at Mahwah, N. J.

The shoes used in the laboratory tests were selected from the group of shoes provided for the road tests so that as far as uniformity of shoe metal is concerned the laboratory test results would be comparable with the results of the road tests. The schedule of the tests was devised to develop information concerning the following:

A The effect of bearing area upon the mean coefficient of friction.

B The effect of temperature upon the mean coefficient of friction.

C Variation in the mean coefficient of friction through a range of speeds at constant braking power.

D Variation in the mean coefficient of friction through a range of braking powers at constant speed.

E The effect of clasp and standard brake conditions on the mean coefficient of friction at constant speed with various braking powers and at constant braking power with various speeds.

F The rate of wear of shoes as affected by clasp and standard brake conditions on the basis of wear per unit of work done.

G The rate of wear under clasp brake conditions as affected by various types of standard and reduced area brake shoes on a basis of wear per unit of work done.

H The effect of a difference in the relative hardness of the brake shoe metal.

I The effect of the rate of cooling of brake shoes after test.

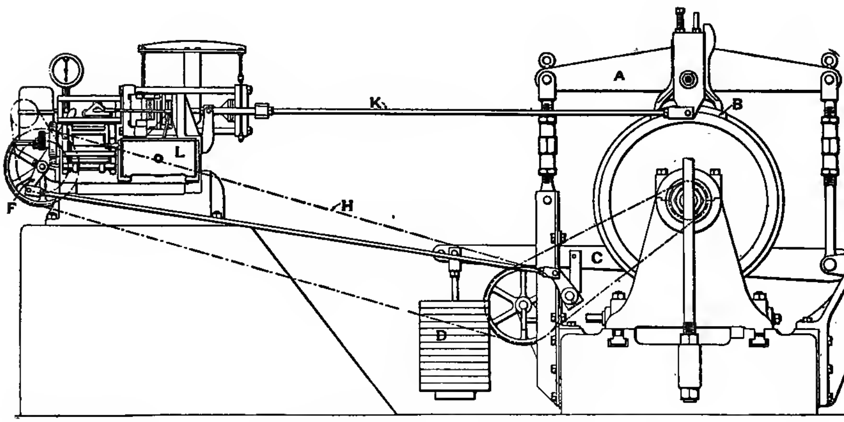


FIG. 27 BRAKE SHOE TESTING MACHINE

At *A* is shown the main lever holding the brake shoe *B*. An adjustable weight *D* is carried on the lever *C*. The mechanism for applying the load to the shoe is shown at *F*. The rod *K* transmits the pull of the brake shoe to the weighing mechanism at *L*.

J The effect of the difference in conditions of brake shoe machine and road tests.

All of the brake shoes were of the cast-iron steel back type, selected from the shoes provided for the road tests. Two or more shoes of each type were tested. There was some variation between different shoes of the same type when tested under the same conditions, but this variation was not greater than that noted when making successive tests on the same shoe and under the same conditions.

The brake shoe testing machine used is illustrated in Fig. 27 and typical diagrams obtained from this machine in Fig. 80.

The time of application of the brake shoe to the wheel was made as nearly as possible equivalent to the rate of development of the braking power on the test train during an electro-pneumatic emergency application. The time from the instant the shoe first began to touch the wheel to the instant that the full pressure came upon the shoe was approximately 2.25 seconds.

The method of starting and conducting the test was as follows: The surface of the shoe was chalked to assist in the observation of the bearing area after the test. The wheel was accelerated to a speed slightly above that desired for the test and allowed to drift until its velocity had decreased to that desired for the test when the shoe was applied to the wheel.

After every test (except those at 30 m.p.h.) the wheel was cleaned by making two stops with cleaning shoes from a speed of 40 m.p.h. The first cleaning shoe was sand filled and was used for the purpose of removing any rough spots of brake shoe metal which might have adhered to the test wheel. The second cleaning shoe was of a special composition which polished the wheel and removed any sand or grit remaining. After the 30 m.p.h. tests, only the composition polishing shoe was used. This program for the cleaning of the shoe was determined upon after many experiments had been made to find a method for cleaning the wheel which would insure the most consistent and uniform wheel surface conditions.

An important difference between the conditions of machine and road tests was that of the wheel treads. During the road tests the tread of the wheel is continually rolling on the rail but during a machine test the wheel is subjected to the action of the brake shoe only. It was impractical to produce any condition which would be equivalent to rolling the wheel tread during machine tests. The effect of this difference in wheel surface conditions is one of the factors which go to make up the difference between machine and road tests described in the discussion of the relation of road tests to machine tests.

The brake shoes to be tested were at all times arranged to come on test in a regular order so that all shoes were given time to cool and were at atmospheric temperatures at the start of every test.

Preliminary stops were made for the purpose of wearing the shoes in to the best possible bearing on the wheel.

MEAN COEFFICIENT OF FRICTION

The application of a brake shoe to a revolving car wheel results in the generation of friction between the surfaces of the shoe and the wheel tending to retard the motion of the wheel. Referring to Fig. 80 which shows a number of characteristic brake shoe test machine dynamometer cards, it will be noted that these cards give a continuous record of the force of resistance or friction produced between the brake shoe and the wheel on a distance base throughout the stop.

Now the friction developed between the surface of the wheel and the shoe will always be proportional to two factors:

A The normal pressure which forces the shoe against the wheel.

B The condition of the wheel and shoe surfaces in contact.

The cards (Fig. 80) show that the friction between the brake shoe and the wheel was changing, although the normal pressure on the brake shoe remained constant after the shoe had been fully applied to the wheel (2.25 seconds from the start). The fact that the friction developed was variable, while the normal pressure on the brake shoe was constant, means that the ratio between the resisting force and the normal pressure on the shoe (i.e., the coefficient of friction) was varying during the stop; which in turn indicates that the *nature or condition of the two surfaces in contact was changing.* This most important characteristic of the action of brake shoes was first observed and recorded in connection with the Galton-Westinghouse tests of 1878 but it is believed that until the present investigations no detailed study or satisfactory explanation of this phenomenon with respect to road test results has been made.

The mean value of the resisting force can be obtained by dividing the area under the force-distance curve by the length of the card from the point of equivalent instantaneous application to the point of stop and when this mean resisting force is divided by the normal pressure on the shoe the result is the mean coefficient of friction. This factor expresses the average frictional quality of the brake shoe surface in contact with the wheel tread surface during the entire stop and is convenient for use in comparing tests made on different types of shoes at various normal shoe pressures.

TEMPERATURES

While testing the plain solid and slotted shoes, temperature measurements were made of the brake shoes throughout the tests and the temperature of both the wheel and the shoe, before the test, was noted. The initial temperature of the shoes was usually taken as that of the room, the shoes having plenty of time to cool back to room temperature between tests. In a few cases the shoes did not have time to cool and then the initial temperature was noted by means of an electric pyrometer or a mercury thermometer. The temperatures during the tests were observed by means of an electric pyrometer, the thermo-couple of which was inserted in a hole drilled from the back of the shoe diagonally through to the face. The elliptical shaped opening marked "For Pyrometer" (see Figs. 28, 81 and 82) marks the position of the thermo-couple in the shoe. The thermo-couple was adjusted so that it was close to or touching the wheel during the

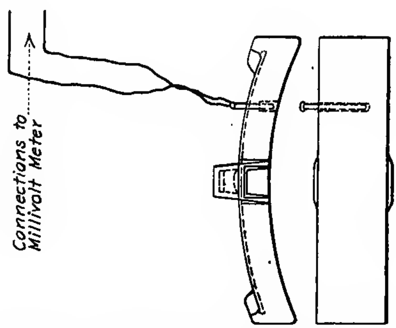


FIG. 28 BRAKE SHOE TEMPERATURE

Method of applying thermo-couple to brake shoe

stop, the object being to obtain the actual temperature of the working metal at the surface of the shoe. The diagram of the arrangement of the thermo-couple in the drilled opening through the shoe is shown in Fig. 80.

During the tests the time of each rise of 40 deg. Fahr. as indicated by the thermo-couple, was noted on the dynamometer record as shown by the upper line marked "Temperature rise" in Fig. 80. The temperature 30 seconds after the maximum temperature occurred was noted as well as the initial and maximum temperatures of the brake shoes. The temperature of the wheel was also measured previous to each test by means of a mercury thermometer placed in contact with the wheel and insulated from the atmosphere.

SPARKING

Observations of the sparking at the brake shoe were made during the first series of tests on plain solid and plain slotted shoes. A magnet similar to the one used for indicating rise in temperature was mounted on the bridge of the dynamometer and its circuit was controlled by an observer stationed near the test wheel. The line marked "Sparks" in Fig. 80 illustrates characteristic sparking records.

Five observations are indicated having the following significance: Application of brake; sparks start; maximum sparking; decrease of sparking; sparks stop.

It was noted that the action of sparking during machine tests was not similar to the same action observed during the road tests. The sparking of the brake shoe began simultaneously with the application of the shoe in the machine tests whereas during road tests an appreciable time was noted between the instant the shoe was applied to the wheel and the time the sparks appeared. This difference was due to the strong air currents experienced in the road tests when the speed of the train is relatively high, and could not be overcome in the machine tests although it was attempted by blowing a jet of air on the wheel.

BEARING AREA

The brake shoe bearing area was observed by chalking the shoe before the test and after the test making a sketch of the surface of the shoe from which the chalk had been removed. The bearing area thus shown was measured by means of a planimeter and the total shoe area, which had been in bearing at some time during the stop, computed.

Reproductions of characteristic bearing area sketches are shown in Figs. 81 and 82.

HARDNESS

Many Brinell hardness readings were taken previous to different tests. The observations differed very little for the various shoes throughout the entire series of tests and indicated a uniform material.

WEAR

The shoes were weighed before and after each series of tests and the loss in weight noted.

Results of Tests—Effect of Bearing Area on Results

The measurement of brake shoe bearing area did not prove to be as valuable in connection with the study of its effect on mean coefficient of friction as was desired, as the diagrams taken showed only the total shoe surface which had been in bearing contact at some time during the stop. The mean coefficient of friction is a function of the mean average of all the bearing areas which may have existed at each instant of the stop and, therefore, this average should be considered if a true relation between bearing area and mean coefficient of friction is to be established. The diagrams shown in Figs. 81 and 82, indicate the total bearing area which came into contact during the stop but they do not give any information as to its value at any instant of the stop or its average value throughout the stop.

The unsatisfactory nature of the determination of bearing area was appreciated and as a means of obtaining additional information a special series of "Intermittent" tests was made in the following manner:

The wheel was brought up to the speed desired and the shoes applied in the usual way. As soon as full pressure was obtained the shoe was released and the diminished value of the speed at the instant of release was noted. The bearing area of the shoe was then measured. This operation was repeated five times, each application being made at an initial speed equal to the final speed of the preceding application, starting with an initial speed of 60 m.p.h. and allowing the wheel to stop during the application of the pressure for the fifth time, after which the shoe was released, removed from the wheel and its final bearing area measured.

Typical results of these tests are shown in Fig. 83. The shoe bearing area is expressed in per cent of the total face area of the shoe, and is plotted against the speed of the wheel, noted at the instant the shoe was released.

From Fig. 83 it is apparent that the bearing area varies widely throughout the stop, the bearing area being small at the beginning but increasing to a maximum at or near the end of the stop. The results, however, are only indicative, because there is necessarily some difference between the conditions of the intermittent and regular stops. Some time was required to remove the shoe, measure its bearing area and replace it, and consequently the temperature of the shoe as a whole was much lower than it would be for the same stop made continuously. The heating of the shoe during the intermittent stop was not only less but was more evenly distributed than in a continuous stop. This results in a decreased tendency to warp. The instantaneous values of coefficient of friction were obtained directly from the dynamometer record. It is apparent that these coefficients of friction, considered either singly or averaged, are very much greater than the mean coefficient of friction obtained for similar stops made continuously.

This indicates that, although the intermittent tests were made as nearly as possible equivalent to continuous tests, the unavoidable difference in the conditions had a decided influence and this influence was in the direction of producing better shoe bearing conditions and consequently a higher coefficient of friction.

However, the data are such that it is possible to conclude that *the magnitude of the bearing area does change throughout the stop* and is greatest near the end of the stop.

During these tests it was noticed that the bearing area shifted; that is to say, a spot which was found in the bearing area early in the stop, was found in the non-bearing area later in the stop.

A study of all the bearing area measurements both for continuous and intermittent stops leads to the conclusion that none of the values of the bearing areas determined are sufficiently accurate to establish a true relation between bearing area, pressure density and mean coefficient of friction.

To determine the effect of bearing area upon the mean coefficient of friction, the mean value of the brake shoe bearing area throughout the stop is necessary, rather than the total and relatively high value observed after the stop. Obviously, it was impossible to measure the instantaneous brake shoe bearing area continuously, and therefore the data can be considered only as relative.

A study of the data in a general way shows that the greater the pressure per square inch of bearing area, the lower will be the mean coefficient of friction.

The temperature observations taken cannot be used to establish a definite relation between the temperature of the working metal and the mean coefficient of friction, for the actual temperatures existing *at the bearing area* were not determined.

The readings evidently indicated only the temperature of that portion of the shoe surrounding the pyrometer element. If this portion of the shoe happened to be in working contact with the wheel, the temperatures would be correspondingly high, but when the bearing area shifted to some other place on the face of the shoe the pyrometer indicated merely the temperature increase due to the conduction from that part of the shoe at which the heat was being generated.

For this reason the maximum temperature readings were very erratic and it was found that the only method which would give uniform temperature readings was to allow sufficient time after an observed maximum reading to permit the heat to become uniformly distributed by conduction throughout the whole shoe. It was found that the shoe temperature thirty seconds after maximum was consistent and proportional to or a function of the initial test speed, because the total amount of energy dissipated by the shoe in any stop will be proportional to the square of the speed and while some of this energy will pass into the wheel in the form of heat and some will pass off with the sparks, a proportional amount will produce a rise in the temperature of the shoe as a whole.

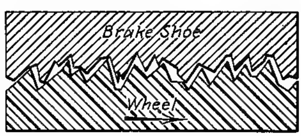


FIG. 29 FRICTION BETWEEN BRAKE SHOE AND WHEEL
Showing Line of Contact between Wheel and Shoe

However, it is not reasonable to look for evidences of the effect of the temperature of the whole brake shoe on the coefficient of friction, when it is appreciated that *the temperatures at the working surfaces greatly exceed the maximum temperatures ever reached by the shoe as a whole.* Consequently for a correct understanding of the relation between temperature and coefficient of friction it is necessary to examine minutely the phenomena which occur during the development of brake shoe friction and to study the action of the materials immediately concerned in this process.

Fig. 29, to an enlarged scale, illustrates how the slight inequalities of the two surfaces in contact can interlock and resist relative movement.

If the top surface is regarded as stationary any movement of the lower surface in the direction indicated by the arrow will be resisted by a force made up of two components:

a Tearing or abrasion of some of the surface projections which are interlocked. In other words, the motion of one surface with respect to the other can be accomplished by shearing off the interlocking projections as indicated by the dotted lines in the sketch:—

b By the lifting or unlocking of some of the surface projections which would require forcing the surfaces apart against the normal pressure until the minute projections on the surface successively cleared. This resistance component involves the performance of work

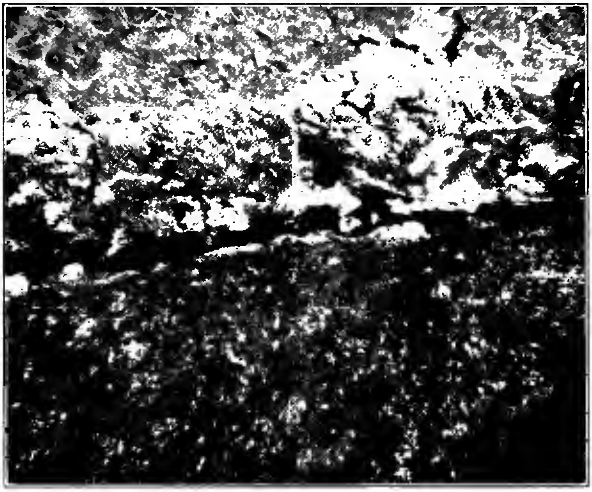


Fig. 30 Contact Surface of Wheel and Shoe

A brake shoe was applied to a steel wheel with a pressure of 15,000 lb. The line of contact was magnified to 100 diameters to show the interlocking of the surfaces.

in separating the surfaces against the normal pressure through the medium of the (for this purpose) inefficient surface angles of the minute projections.

In brake shoe friction the first component, namely, tearing or abrasive action between the minute interlocked portions of the bearing surfaces is the important factor. Consequently the resistance developed has frequently been spoken of as the "friction of abrasion." The conditions just mentioned are strikingly illustrated by Fig. 30 which is a microphotograph of the line of contact between a shoe and wheel well rubbed in and under an ordinary working load. The observed phenomena of brake shoe friction are at once explained when the influence of condition of bearing surface disclosed by this illustration is appreciated.

The retarding force developed bears some relation to the ultimate strength of the metal which undergoes abrasion, because the action of abrasion consists of the pulling apart, or crushing of the particles of the contact surfaces. Practically none of the abrasive action takes place on the wheel, due to the harder and tougher nature of the wheel surface and the fact that the surface of the wheel is not continuously in contact with the brake shoe while the brake shoe *is* in continuous contact with the wheel. Inasmuch as the tearing or abrasion of the metal particles can take place only in the thin layer of the shoe metal which is in contact with the wheel and as the actual contact area is, as has already been shown, but a portion of the total face area of the shoe, it is evident that the generation of the retarding forces and consequent absorption of the energy of the moving train is dependent upon but a *very small quantity of brake shoe metal.*

The result of this condition is that the metal in the state of abrasion undergoes a very rapid rise in temperature; and the indications are that the temperature of the working areas of the brake shoe is changing continuously, because when one set of particles is torn off energy is dissipated in the form of heat so that the next particles to be torn off are at a higher temperature than the first. This process appears to continue until the surface of the metal in abrasion reaches such a temperature that the force required to tear the particles away is greatly reduced. The abrasive action then seems to be extremely rapid until the bearing of the shoe shifts to some other and cooler spot on its surface, not previously in the same degree of contact with the wheel. The action may then be repeated and the bearing area may thus shift from one portion of the surface to another during the stop.

Unquestionably the constantly changing temperature of the contact surface has an important relation to the force of retardation developed by the tearing down of the metal particles. The resistance due to abrasion is dependent on the ultimate strength of the cast iron. It is well established that above a critical temperature (approaching 900 deg. fahr.) the ultimate strength of cast iron decreases rapidly and that at or above red heat temperatures (1400 deg. to 1800 deg. fahr.) its ultimate strength is greatly reduced.

The force of retardation due to abrasion and the corresponding mean coefficient of friction will therefore be a function of the constantly changing, but always high, temperature of the working metal.

Obviously the lower the mean temperature can be maintained, the higher will be the mean coefficient of friction as long as this average temperature is above the critical temperature at which the ultimate strength of cast iron is a maximum.

Unless the brake shoe is of such a nature or in such a condition that a large proportion of its face area is in working contact with the wheel, and remains so, the principal factor in producing high friction for any given braking condition appears to be the frequent shifting of the bearing area from the heated to the cooler spots over the face of the shoe. If a shoe has a relatively small bearing area, that cannot shift, the wheel is forced to wear the highly heated and ineffective shoe metal rapidly away, and successively exposes cooler metal, which is better able to offer resistance and absorb energy. This condition existed for the reduced area shoes of both plain and flanged types, the rate of wear being higher for these shoes than for those of full area.

If the friction characteristics developed by a given brake shoe under a given combination of conditions depend, as appears probable from the foregoing considerations, upon the frequent shifting of the contact areas of the shoe surface, the instantaneous as well as the average values of the frictional resistance of the brake shoes during a stop are functions of: (*A*) The fit of the shoe to the wheel; (*B*) the flexibility of the shoe; (*C*) the available bearing area.

A The fit of the shoe to the wheel has an important influence on the facility with which the contact area can shift. If an ordinary metal brake shoe fits well, according to the ordinary understanding of this expression, it necessarily cannot bear on the wheel equally at all portions of its surface. But when wear takes place, only a comparatively small amount of metal has to be worn off the bearing spots in order to bring cooler spots into contact, and consequently a good fit of the shoe will guarantee against the possibility of the bearing area being held concentrated at any particular spots, after they have become heated to a point where the metal breaks down rapidly and offers a comparatively poor resistance.

B The effect of warping is clearly shown by the comparative performance of a solid shoe of any type and a similar shoe slotted so as to make it more flexible. In every case the slotted shoe has a decided advantage. With a warped shoe the bearing area cannot shift readily and consequently the average temperature of the working metal is high and a correspondingly low mean coefficient of friction is obtained. During the tests of the solid shoes, a shoe would often be observed to be warped in such a manner that it touched the wheel only at each end while at the center of the shoe the two surfaces were 1/32 in. apart, although the full braking power was applied to the shoe to force it against the wheel. Under such conditions the temperature of the bearing area would rise until the red heat had penetrated the shoe fully 1/8 in. In such cases the bearing area would not shift to a cooler part of the shoe until sufficient metal had been worn off to offset the effect of the warping. The uneven heating effect caused by the concentration of the bearing area at the two ends of the shoe would in turn cause the shoe to warp so that the ends of the shoe would be drawn away from the wheel and the bearing area again held for a comparatively long time at one spot at the center of the shoe. Such tests invariably resulted in longer stops or lower mean coefficient of friction than was the case with the same type of shoe slotted.

The resulting higher average temperature of the working metal due to the action of warping also resulted in a greater rate of wear. All the data of shoe wear show that shoes of the same type and hardness had a high rate of wear per unit of energy absorbed when a low coefficient of friction was developed and likewise a lower rate of wear when a higher coefficient of friction was developed. In other words when the coefficient of friction is high, the average temperature of the working metal must be comparatively low and therefore the metal torn from the surface of the shoe works more effectively because it is at a lower temperature and consequently less metal is required to do a given amount of work.

C The shifting of the bearing area will tend to be more rapid if the size provides more available area for shoe bearing. The average working temperature will also be reduced because a shoe of large area, such as a flange shoe provides better facilities for radiating and conducting heat away from the working surfaces. This is borne out by the better performance of the flange shoes in comparison with the plain shoes, both solid and slotted.

Variations in Successive Tests Made With the Same Shoes and Under Same Braking Conditions

In all brake shoe machine tests considerable variation is found in the results obtained on successive tests and under supposedly similar conditions (Fig. 84). From what has gone before it is now possible to account for these variations on the basis of the changes in the bearing surface conditions as the mean force of retardation depends directly on the mean average temperature of the working metal and as the chief factor affecting the temperature of the working metal is the manner in which the bearing area shifts from point to point on the surface of the shoe, it follows that when this shifting is accomplished with the greatest promptness (and therefore the least amount of excessive heating at one spot) the average temperature of the working metal will be lower and the mean coefficient of friction higher. The rate at which the bearing area will shift depends, as has been stated, on the existing tendency for the shoe to warp. Warping is caused primarily by the uneven heating of the shoe but so many other factors, dependent on the quality and structure of the metal, are involved that the effect of warping will vary a great deal, even though the initial speed and braking power are the same throughout a series of tests.

Furthermore, the heating effect may be distributed over the face of the shoe in a different manner during one test than during another test, and so on. Consequently when successive tests are made on one brake shoe variations in the results are to be expected and it is therefore imperative that deductions be based on the results of a number of tests.

The variation in the results of successive machine tests will of course be greater than in road tests because in the case of a car or train an average is obtained from the action of all the brake shoes involved.

The Effect of Speed on Coefficient of Friction

The results of both road and machine tests on brake shoes have shown that the coefficient of friction under any condition of braking power is dependent upon the initial speed, and the coefficient of friction tends to decrease as the speed is increased. This condition can be expected from the fact that the total energy to be absorbed during a given stop is always proportional to the square of the initial speed. The effect of the higher speeds is to increase the rate at which the temperature of the working metal changes, and although the actual bearing areas will shift more rapidly, the greater tendency to a general heating of the shoe reduces its ability to conduct or radiate heat from the face of the shoe. For these reasons the average temperature of the working metal is consistently higher at higher speeds, and the coefficient of friction correspondingly lower.

If the foregoing correctly describes the effect of a continually changing speed on the instantaneous values of the coefficient of friction during the stop it follows that the mean coefficient of friction is going to vary with the average temperature of the working shoe metal. Consequently, since the rate at which energy is absorbed by the brake shoe is higher and the total amount of energy absorbed greater for high speeds and the converse of this true for stops from lower speeds, it follows that the mean coefficient of friction should vary inversely with the initial speed.

The relation between initial speed and mean coefficient of friction for various shoes and braking conditions indicated by the test results is shown on Fig. 85. These curves show the characteristically lower mean coefficient of friction for the high initial speeds. In considering the significance of these curves the conditions under which the data were obtained must be taken into account. That these conditions have an important bearing on these results is evident from the marked difference between the results obtained under single and clasp brake conditions.

It should not be supposed that because straight lines have been drawn that this relation holds outside the range of speeds tested because we would not expect to find a zero mean coefficient of friction at any finite speed however high.

The most that can be claimed for these straight lines is that they fit the results of the tests as well as any other lines that could be drawn, and that sufficient data were not available to warrant the plotting of curves instead of straight lines nor the extension of these curves beyond the speeds involved.

An extreme effect of speed is well illustrated by the dynamometer diagrams (Fig. 80), and on the braking force curves (Fig. 76). With constant brake shoe pressure, the force of retardation generated by the brake shoe begins to increase near the end of the stop and continues to increase at an accelerating rate, the retarding force reaching its maximum at the end of the stop. This inverse relation between the instantaneous coefficient of friction and the average temperature of the working metal as influenced by the rapidly decreasing speed near the end of the stop is characteristic (see Figs. 76 and 77).

The rate at which the working metal of the surface of the brake shoe must absorb energy is dependent on the speed. The average temperature of the working metal is affected by the speed in combination with the ability of the brake shoe as a whole to radiate or conduct the heat absorbed away from the surface of the shoe. The temperature of the brake shoe as a whole is, of course, always greater near the end of the stop and would have some tendency to increase the average temperature of the working metal, but on the other hand, the rapidly decreasing rate at which energy is delivered to the shoe for absorption, due to the decrease in speed, reduces the tendency of heating to such a degree that the effect of a higher general shoe temperature is offset and the average temperature of the working metal reduced.

Confirming this it is always observed that vigorous sparking decreases as the speed begins to decrease more rapidly near the end of the stop and finally is replaced by ground off metal not incandescent. This indicates that although the general shoe temperature is undoubtedly higher near the end than at any other part of the stop, the average temperature of the working metal is correspondingly lower and consequently the force of retardation and coefficient of friction increases. This also accounts for the relatively high coefficient of friction always developed during the low speed tests.

The conditions which bring about the temperature change mentioned are graphically illustrated in Figs. 76 and 77, in which the power curves referred to the scale of B.t.u. per shoe per second indicate a very high heat input into the working metal during the early part of the stop, this rate gradually decreasing to zero at the end of the stop. Comparing this curve with the retardation curve it is seen that the shoe metal seems to be incapable of exceeding a certain maximum resistance so long as the heat input is above that which can be expressed as 40 or 50 B.t.u. per brake shoe per second. But when the heat input becomes less than this the coefficient of friction rises and continues to rise more and more rapidly as the heat generated decreases until the end of the stop.

The maximum resistance limit encountered in the brake shoe action would seem to be accounted for by the more rapid abrasion which takes place at the higher temperatures. In other words, when the heating of the working metal exceeds a certain value the effect of the molten condition thus produced is chiefly to increase the rate at which the metal is worn away, but for this very reason, to maintain a substantially uniform condition of the surfaces successively presented to the action of abrasion.

Effect of Severe Braking Conditions

If the influence of the rate at which energy is being absorbed has been correctly described in the foregoing, it follows that the higher the rate the greater the heating of the working surface and the more rapid the abrasion and that this effect would be most marked under severe braking conditions, such as obtained when heavy cars equipped with one brake shoe per wheel are stopped by the application of high braking powers from high initial speeds. In such cases the rate of energy absorption and consequent temperature rise at the working surfaces becomes very high and the coefficient of friction is correspondingly low. The conditions are further aggravated by the inability of the shoe body metal to conduct heat away from the working surface with anything like a proportionate rate at which such conduction can take place when the total heat developed is much less. Abrasion in such cases occurs at a very rapid rate and when the time of such action is prolonged, as in high speed stops, instances have been observed when early in the stop the entire shoe surface becomes red for some distance from the surface and metal is discharged from the shoe at such a rate and in such a plastic condition that it is found deposited on the car body above the shoe in the form of recast metal.

That this action is a function of the amount of shoe metal available for the absorption to heat follows from the fact that under clasp brake conditions, and especially when flanged brake shoes were used, there was only moderate sparking, while on the other hand, under single shoe conditions at high braking powers and high speeds the sparking was vigorous and in many instances, especially when the shoe had only a partial bearing on the wheel, the molten metal was discharged as mentioned above. The latter action in many cases resulted in wearing through a shoe newly applied to the back and sometimes into the brake head in a single stop with high braking power from high speeds.

One Versus Two Shoes Per Wheel

The gain in brake shoe performance with two shoes per wheel (clasp brake conditions) as compared with one shoe per wheel (standard brake conditions) is most clearly shown in the comparison of the brake shoe machine test results. This difference is less easily determined from the results of the road tests because of the presence of a greater number of variable factors not present in machine tests.

Curves showing the average results of tests of various types of brake shoes under both single shoe and clasp brake conditions are shown in Fig. 86. A comparison of the values of mean coefficient of friction for standard and for clasp brake conditions shows a decided advantage for the clasp brake throughout the entire range of braking powers. The gain in favor of the clasp brake with slotted shoes amounts to about 40% at a braking power of 180% and 100% at a braking power of 40%, an average gain for the whole range of braking powers of about 70%.

When the action of brake shoes under standard and clasp brake conditions are considered in the light of the influence upon the working metal, the amount of energy to be absorbed per unit of working area, and the average temperature of the working metal, it is apparent that the clasp brake must have some advantage over the standard brake both in mean coefficient of friction and shoe wear.

The clasp brake provides two brake shoes to do the work which the standard demands of one shoe and this difference may be considered from three standpoints:

a The clasp brake shoe has but one-half the wheel load and consequently one-half as much energy to absorb.

b The clasp brake shoe is working at only one-half the shoe pressure at which the standard shoe must work under the same braking power.

c The available area for the same amount of energy to be absorbed is double. The important fact here is that it is only the *available* shoe surface and not the *actual* working area which is double. The tendency of a shoe to warp will be more or less influenced by the pressure forcing it against the wheel.

It follows, therefore, that on account of the lower pressure per shoe the clasp brake is at a disadvantage from the standpoint of warping when compared with the standard brake. However, with clasp brakes the intensity of the local heating of isolated bearing spots is not so great on account of less energy to be absorbed per shoe and consequently the cause of warping should be somewhat decreased. This tends to minimize the warping effect under clasp brake conditions. These are the chief factors which result in the performance of the brake shoe under clasp brake conditions, showing less improvement over that with standard brake conditions than might be expected, if only the amount of energy to be absorbed per shoe were considered.

In view of the above analysis of the factors affecting shoe performance under the two conditions, it follows that the use of two shoes instead of one will result in a higher coefficient of friction and less wear per unit of work done. This conclusion is supported by the results obtained during the road tests, but the difference, in so far as its effect on the length of stop is concerned, is less in the road tests than is shown by the machine tests, the former being a function

TABLE 7. COMPARISON OF SINGLE AND CLASP BRAKE SHOE WEAR PER 100,000,000 FT.-LB. WORK DONE

Brake Conditions	Kind of Shoes Plain		Ratio of Wear Solid to Slotted, Per Cent	Ratio of Wear Single Solid to Clasp Solid Per Cent	Ratio of Wear Single Slotted to Clasp Slotted, Per Cent
	Type A Solid	Type B Slotted			
Single	4.382	3.921	111.7	141.1	133.5
Clasp	3.105	2.937	105.9	......	......

TABLE 8. COMPARATIVE COST AND DURABILITY OF PLAIN AND FLANGED SHOES, UNDER CLASP BRAKE CONDITIONS

Kind of Shoe	Weight Lb.	Scrap Weight Lb.	Lb. Wt. Available for Wear	Cost Cents (Assumed)	Scrap Value Cents	Net Cost	Cost per Lb. Available for Wear, Cents	Ratio of Cost per Lb. Available for Wear, Per Cent
Type A plain	20	9	11	35	4.5	31.5	2.863	100
Type E flanged	36	15	21	70	7.5	62.5	2.976	104

Kind of Shoe	Lb. Wear per 100,000,000 Ft-Lb. Work Done or Wear Factor		Per Cent Slotted to Solid	Cost in Cents per 100,000,000 Ft-Lb. Work Done		Per Cent Slotted to Solid
	Solid	Slotted		Solid	Slotted	
Plain	3.105	2.937	94.6	8.89	8.41	95
Flanged	2.502	2.170	86.8	7.45	6.46	87
Per cent flanged to plain	81	74	......	84	77	......

Kind of Shoe	No. of 60 m.p.h. Stops at 150 Per Cent Braking Power Necessary to Wear Out Shoe		Per Cent Slotted to Solid
	Solid	Slotted	
Plain	378	397	105
Flanged	891	1024	115
Per cent flanged to plain	236	258	......

of the combined performance of the brake shoes and brake rigging, while the latter is a function of brake shoe performance only. Consequently, until the effect of the brake rigging and brake shoe can be observed separately, a comparison of the coefficient of friction with one and two shoes per wheel in road tests is always open to question.

Rate of Wear of Brake Shoes

A comparison of the shoe wear per 100,000,000 ft.-lb. of work done under single shoe and clasp brake conditions is shown in Table 7.

A comparison of plain solid and plain slotted brake shoes under single and clasp brake conditions (Table 8) shows:

A That the superior durability of the plain slotted shoe as compared with the plain solid amounts to 11.7% under single shoe brake conditions and 5.9% under clasp brake conditions.

B That with plain solid shoes the durability will be increased 41.1% under clasp brake conditions as compared with that under single shoe conditions.

C That with plain slotted shoes the durability will be increased 33.5% under clasp brake conditions as compared with that under single shoe conditions.

Table 8 gives the data of comparative wear and cost of plain and flanged shoes.

A comparison of flanged and plain shoes on the basis of durability with the costs assumed shows:

A That the net cost per pound of metal available for wear is 4.0% more for flanged shoes than it is for plain shoes.

B That the wear of the flanged solid shoes per unit of work done is 19% less than for plain solid shoes, and for flanged slotted 26% less than for plain slotted shoes, or 30% less than plain solid shoes.

C That the wear of plain slotted shoes per unit of work done is 5% less than the wear of plain solid shoes, and the wear of the flanged slotted is 13% less than the wear of flanged solid shoes.

D That for the same amount of work done flanged solid cost 16% less than plain solid shoes, and flanged slotted cost 23% less than plain slotted or 27% less than plain solid shoes.

E That approximately 136% more stops will be required to wear out the flanged solid than will be required to wear out the plain solid, and 158% more stops will be required to wear out the flanged slotted than the plain slotted shoe and 171% more stops to wear out the flanged slotted than the plain solid shoe.

Hardness

An illustration of the relation between Brinell hardness and the mean coefficient of friction is shown on Fig. 87. As this investigation was not carried out through a wide variation of hardness numbers, no definite data as to the most desirable hardness can be given. However, from observations made during these tests it is believed that under clasp brake conditions with cast iron brake shoes, the mean coefficient of friction is a maximum and shoe wear a minimum with a Brinell hardness of about 190.

RELATION OF MACHINE TO ROAD TESTS

It has always been observed that stops with a car or train were much longer than brake shoe machine test stops under similar conditions. Some of this difference in stopping distance could be accounted for by the relatively low efficiency of the car brake rigging. There are other sources of difference, however, which are inherent and of such a nature that their individual and combined effects could not be measured with the means available during these investigations. The determination of a factor which will express the difference between the performance of a brake shoe in machine tests and in road tests is of importance, because with such a factor established it will be possible to predict within the limits of ordinary variations of brake shoe performance the probable braking performance of a car or train under any given set of conditions.

The difference due to the performance of brake rigging can be fully allowed for if the running rigging efficiency can be determined, but in the first place the determination of the running efficiency would involve the making of road tests which it is desired to avoid, and in the second place all means employed up to this time to measure the running efficiency of the brake rigging have been rendered more or less unreliable by the erratic movement induced within the measuring apparatus. The measurements of standing efficiency, however, have given results which are consistent within themselves and with theoretical deductions and will therefore afford a convenient and satisfactory basis for establishing the proper allowance to be made for the characteristics of any given brake rigging. It is recognized that there may be a difference between the performance of a brake rigging when standing and when running, but for the reasons just mentioned, the standing efficiency is the most convenient to obtain and whatever difference may result from the motion of the train will be taken care of by the ratio or difference factor established from the comparison of road and machine tests, as will be explained.

Having the data of sufficient number of comparative road and machine tests, it is possible to determine the average ratio of the road to the machine test performance, after allowance has been made for the standing brake rigging efficiency. This ratio then furnishes a factor involving both the inherent difference between machine and road tests and the difference between standing and running efficiency which can be used in calculating the probable car or train stop after having determined the other controlling factors from machine tests and measurements of the car.

The results of tests of standing efficiency are plotted in Fig. 50. Applying the brake rigging efficiency thus determined and the stops

TABLE 9. RELATION OF ROAD AND LABORATORY RESULTS

Brake Shoe	Type of Brake	Ratio Between Road and Machine Stops Calculated on the Basis of 100 Per Cent Rigging Efficiency		
		Mean	60 Per Cent B. P.	180 Per Cent B. P.
Plain slotted or broken	No. 3 clasp	1.280	1.290	1.270
Flange slotted or broken	No. 3 clasp	1.250	1.210	1.290
Averages		1.265	1.250	1.280

determined in road tests at different percentages of braking power as shown by curve C, Fig. 88, curve B has been obtained as the probable relation between per cent breaking power and length of stop with 100% rigging efficiency. The difference between curve B and curve A which shows the relation between per cent braking power and stop as obtained on the machine tests is then proportional to the ratio or factor desired. Table 9 shows various values for this ratio which have been derived from the figures shown.

To illustrate the method of using the machine and road test ratio suppose that it is desirable to know the performance of a certain car in a breakaway stop from 60 m.p.h. at 150% braking power and that the standing rigging efficiency of this car has been measured. Assume that this measurement of rigging efficiency is expressed by the solid line curve on Fig. 50 and that slotted or cracked plain brake shoes will be used. Consulting the curve referred to, it will be noted that the efficiency of the rigging at 150% braking power is 83.2% which means that the actual braking power at the shoe will be

$$150 \times 0.823 = 123.45\%$$

Referring to Fig. 88, it will be seen that at 123.45% braking power the curve representing the results of machine tests of slotted plain shoes gives a value of 640 ft. as the stopping distance. Now applying the ratio or difference factor 1.265, the actual stop of the car will be $640 \times 1.265 = 810$ ft. at the actual braking power of 123.45% (nominal braking power 150%).

It must be borne in mind that in the example given above the brake application on the road is assumed to be an electro-pneumatic emergency which requires an average time from point of trip to maximum cylinder pressure of about 2.25 seconds. The time to point of equivalent instantaneous application of brake is 0.75 second. The ratio given for the machine test results is for this same time of application with a different time to point of equivalent instantaneous application. If it is desired to use this factor for any other brake application, then the difference in time must be multiplied by the initial speed in feet per second of the car and this difference added to or subtracted from the distance (810 ft. mentioned above) depending upon whether the time is greater or less.

CONCLUSIONS

AIR BRAKE

The characteristics of the mechanism available for controlling the air pressure in the brake cylinders determine in a large measure the length of the emergency stop, the reliability and flexibility of the brake operation in service applications and in general the safety, convenience, comfort and economy of train control.

The state of the art has been advanced to a marked degree in all of these directions by the developments of recent years as exemplified in the apparatus used in these tests. What has been accomplished can be broadly summarized as follows:

(*A*) Desired results are insured with greater certainty.

(*B*) Undesired results are guarded against more effectively.

(*C*) An adequate capacity for present and future requirements is provided.

In service applications with the improved (UC) equipment a greater flexibility of operation is provided. That is, the braking power per pound of brake pipe reduction is lower thus giving the engineer a greater time in which to use judgment when manipulating the brakes. At the same time, however, the maximum braking power obtainable in a full service application is higher.

A more sensitive and prompt release of the brakes is insured, tending to improve the releasing action of all brakes in the same train of mixed old and new equipments.

The action of the old and the new equipments mixed in the same train is harmonious and free from rough slack action or shocks both in service and emergency operation.

The UC equipment is adaptable to any weight of car and may be installed to furnish any desired nominal per cent of braking power.

With the new equipment operating electrically or pneumatically, there is always available a quick acting and fully effective emergency brake. This is not the case with the old equipment, in which the relation of the service and emergency functions is such that a quick action application could not be obtained after a service application of any consequence. The following average results indicate the degree to which this difference has an effect on the length of stop. Considering the ordinary full service stop from 60 miles per hour with both brakes (say 2000 or 2200 ft.) as 100%, the attempt to make an emergency application with the old equipment does not produce any shorter stop than if only a full service application were made. With the improved apparatus operating pneumatically, an emergency application following a partial service application will shorten the stop about 14% and after a full service application about 10%.

With the electro-pneumatic brake these figures are respectively 23% and 15%.

An electrically controlled brake application has been recognized as ideal ever since the report to this effect presented by the Master Car Builders' Committee in charge of the famous Burlington Freight Brake Trials 1886 and 1887, for the reason that thereby the time element in starting the application of the brakes on various cars in the train is eliminated, a correspondingly shorter stop made, and the possibility of shocks at any speeds removed. With the new brake apparatus the effectiveness of the pneumatic emergency application is so considerably increased that the saving in time due to electric control has proportionately less influence on the length of stop, but its effect in eliminating serial action and consequently the possibility of shocks due to brake application is of correspondingly greater importance.

The graduated release feature of the improved brake apparatus permits stops to be made shorter, smoother and with a greater economy in time and compressed air consumption.

The new apparatus can be applied to give only the equivalent of the old standard apparatus if desired but in such a form the complete new apparatus can then be built up by the addition of unit portions to the simplest form of the mechanism.

The electro-pneumatic brake acts as an automatic telltale in cases of malicious or accidental closing of an angle cock after the train is charged by permitting all the brakes to apply, it being thereafter impossible to release the brakes behind the closed cock until the cock is opened.

The PM equipment will start to apply on a brake pipe reduction of 2 lb. A 4-lb. brake pipe reduction is required to start an application with the UC equipment, thereby preventing undue sensitiveness to application on slight, unavoidable fluctuations in brake pipe pressure. As a bona fide service reduction of more than 4 lb. continues, the rate of attainment of braking power is the same as if no stability feature had existed.

The attainment of full service braking power on the entire train with the UC equipment operating pneumatically was 16 seconds, 33% longer than with the PM equipment because of the smaller size reservoirs used for greater flexibility.

Full service braking power was obtained in nine seconds with the electro-pneumatic brake but without sacrificing desirable flexibility because of the increased sensitiveness of control when operating the brakes electrically.

The time of transmission of serial quick action through the brake pipe is practically the same with UC and PM equipments.

The time to obtain full emergency braking power with the PM equipment on the entire train was 8 seconds; with the UC equipment operating pneumatically 3.5 seconds or 56% shorter; with the electro-pneumatic equipment 2.25 seconds or 72% shorter.

The gain in emergency stopping power of the electric pneumatic equipment over the PM equipment results from: (*a*) the shorter time occupied in applying the brakes; (*b*) a higher brake cylinder pressure obtained; (*c*) the holding of the pressure as obtained, without blow-down, as with the high-speed reducing valve of the PM equipment.

Designating the time of equivalent instantaneous application of retarding force by t, and the braking power, corresponding to the brake cylinder pressure obtained, by P, the values of t for emergency applications with the PM equipment 12 car train range from 2 to 2.5 seconds, for the UC pneumatic from 2 to 2.5 and for the electro-pneumatic from 0.7 to .85 seconds.

The observed average value for P, with the PM equipment (for a nominal 113% braking power on the cars) ranges from 95% to 100%. With the UC pneumatic equipment and electro-pneumatic equipment nominal emergency braking powers of 90, 125, 150 and 180% were used, which, due to locomotive effect, become for the complete train 90%, 117%, 137% and 160% respectively.

With the electro-pneumatic brake a uniform increase in per cent of braking power results in a substantially uniform decrease in length of train stop. An increase of 5% in braking power reduces the length of stop about 2% within the range of braking powers tested.

The available rail adhesion varies through wide limits, e.g., from 15% in the case of a frosty rail early in the morning to 30% for a clean, dry rail at mid-day.

The amount of wheel sliding depends more on the rail and weather conditions than on the per cent braking power. Some sliding was experienced with braking powers as low as 90% and 113% where rail conditions were unfavorable, but 180% braking power did not cause wheel sliding with good rail conditions.

The effect of excessive wheel sliding was to make the length of the stop about 12% greater than similar stops without wheel sliding.

A braking power low enough to eliminate the possibility of wheel sliding on a bad rail results in longer stops than could be considered satisfactory for general service. Since good rail conditions prevail a large part of the time, the preferable emergency braking power is that which, considering the installation conditions, will stop trains at all times in as short a distance as can be accomplished without trouble from wheel sliding in such cases as are to be anticipated when emergency stops have to be made under unfavorable rail conditions. Advantage might be taken of this fact to use a higher braking power in summer than could be used in the winter with the same degree of freedom from objectionable wheel sliding.

The relation between the opposing forces on the wheel when sliding is about to commence can be expressed as follows:—

$$F = H_m$$

$$Pwef_s = wf_r$$

$$P = \frac{f_r}{ef_s}$$

in which

F = tangential retarding force of shoe on wheel

H_m = maximum force of friction between wheel and rail (at instant of slipping)

P = nominal per cent braking power

w = weight on wheel, lb.

e = transmission efficiency of brake rigging

f_s = coefficient of brake shoe friction

f_r = coefficient of rail friction

This establishes the value of the per cent braking power which must exist at each instant of the stop if the vehicle is to be brought to rest in the *shortest possible* distance.

But it has already been shown that if other conditions are constant a change in braking power will produce a corresponding change in the mean coefficient of brake shoe friction, the relation derived for the conditions under consideration being

$$f_s = \frac{0.12}{P^{0.419}}$$

Substituting this value of f_s in the above equation for P, we have

$$P=\frac{f_r\ P^{0.419}}{0.12e}$$

$$P^{0.581}=\frac{f_r}{0.12e}=\frac{8.33\,f_r}{e}$$

For a 60 m.p.h. stop on a good rail f_r can safely be taken at 25 per cent and e for four-wheel trucks can be made at least 85 per cent.

Then

$$P^{0.581}=\frac{8.33\times0.25}{0.85}$$

for which

$$P=470 \text{ per cent}$$

This of course, is beyond the range of the tests which furnished the data from which the relation $f_s=\dfrac{0.12}{P^{0.419}}$ was derived, and consequently the result signifies nothing more than that, under the conditions assumed above, an extremely high nominal braking power would be necessary in order to cause the wheels to slide. As a matter of fact, tests at about 400 per cent nominal braking power are on record in which practically no sliding occurred.

On the other hand considering the variation in coefficient of brake shoe friction and rail friction that must be counted upon in service, it is advisable to limit the tangential retarding force, F, to a value less than Hm by an amount I which is proportional to the degree of protection against wheel sliding considered necessary under the ordinary condition of rail likely to be encountered, including also due allowance for the effect of surges in the train, non-uniformity of braking conditions on different vehicles, reasonable protection against excessive wheel sliding under bad rail conditions and all other causes tending to cause the wheels to slide.

The equation of condition is then

$$F = Hm-I$$

The value of I is entirely arbitrary, due to the wide range of variation of its elementary factors, according to locality, time of year, state of weather, character of train and equipment and so on.

For the purpose of illustrating the extreme opposite to that given above suppose that I be taken as $0.25\ Hm$. Then

$$F=0.75Hm$$

$$P=\frac{0.75f_r}{ef_s}$$

Again assuming the relation $f_s=\frac{0.12}{P^{0.419}}$, as above, we have

$$P^{0.581}=\frac{6.25f_r}{e}$$

Considering $e=85$ per cent as before and taking $f_r=12$ per cent which was the lowest observed rail friction, under as poor rail conditions as could well be imagined, we have

$$P^{0.581}=\frac{6.25\times 0.12}{0.85}$$

from which $P=81$ per cent.

As a matter of fact some wheel sliding was obtained during the tests with 85 per cent to 90 per cent braking power when very bad rail conditions prevailed. For practical purposes a conservative value for f_r would be 15 per cent. If now a nominal braking power of 150 per cent is used the corresponding margin I against wheel sliding can be calculated from the formula $I=kHm$ and $F=Hm$ $-kHm=(1-k)Hm=CHm$; *from* which $C=\frac{F}{Hm}=\frac{Pef_s}{f_r}=\frac{P^{0.581}e}{8.33\,f_r}$

Substituting the values $P=1.5$, $e=0.85$ and $f_r=0.15$, we have

$$C=\frac{(1.5)^{0.581}\times 0.85}{8.33\times 0.15}$$

from which

$$C=0.86$$

therefore

$$F=0.86\ Hm$$

and

$$k=1-C=0.14$$

That is to say, *under the assumed conditions*, when using a braking power of 150 per cent with as bad a rail condition as is represented by the extremely low value $f_r=15$ per cent, the mean retarding force developed by the brake shoe is still 14 per cent less than the adhesion of the wheel to the rail.

The amount of wheel flattening when sliding occurs depends upon the weight upon the wheels, the materials in the wheels and rails, and the condition of the rail surface. The rail surface may be such that relatively long slides will produce but small flat spots, or, conversely short slides may produce flat spots of a size requiring prompt attention.

When the UC equipment is used on the cars an arrangement giving a high emergency braking power on the locomotive, with a blow-down feature, has advantages as follows:

(*a*) Shocks between locomotive and cars practically eliminated
(*b*) Shorter stops
(*c*) No more wheel sliding than to be expected with the present installation of ET equipment.

Brake Rigging

An efficient design of brake rigging must be produced before the advantages of improved air brakes or brake shoes can be fully utilized.

The use of the clasp type of brake rigging eliminates unbalanced braking forces on the wheels and so avoids the undesirable and troublesome journal and truck reactions that come from the use of heavy braking pressures on but one side of the wheel. This has an important effect not only on freedom from journal troubles but also in enabling the wheel to follow freely vertical inequalities of the track.

The clasp brake also improves the brake shoe condition materially, both as to wear and variability of performance.

Although the clasp brake rigging will produce better stops than a single shoe brake rigging equally well designed (other conditions being equal), its advantage in this direction is of less importance than in the improved truck, journal and shoe conditions mentioned above.

The tests indicated that at least 85% transmission efficiency could be obtained with either single shoe or clasp brake rigging.

The following features were observed to be of importance if maximum overall brake rigging efficiency is to be secured:

(*a*) Protection against accidents that may result from parts of rigging dropping on the track.

(*b*) Maximum efficiency of brake rigging at all times to insure the desired stopping with a minimum per cent of braking power.

(*c*) Uniform distribution of brake force, in relation to weight braked, on all wheels.

(*d*) With a given nominal per cent braking power, the actual braking power to remain constant throughout the life of the brake shoes and wheels.

(*e*) Piston travel to be as near constant as practicable under all conditions of cylinder pressure.

(*f*) Minimum expense of maintenance and running repairs of brake rigging between the shopping of cars.

Brake Shoes

The brake shoe bearing was the most difficult factor to control and at the same time the most potent in producing variations in brake performance.

The tests established the possibility of a variation of 15% to 20% in length of stops from 60 m.p.h. with all factors except brake shoe condition remaining substantially constant. Continued stopping with moderate braking pressures produced a constantly improving brake shoe condition and shorter stops. This is evidence that with reasonable attention to brake shoe maintenance the condition of the shoes on cars in ordinary road service is likely to be more favorable to making short emergency stops than during a series of tests in which the brake shoes are worked severely.

The difference in the efficiency of the clasp and single shoe rigging may offset the gain which might be expected from difference in coefficient of friction and vice versa. Consequently as neither of these factors could be observed uninfluenced by the other, a satisfactory comparison of the mean coefficient of friction under different rigging conditions or of different types of rigging or air brake apparatus under variable shoe conditions in road tests, is impossible.

High braking powers from high initial speeds result in a great heating of the working surface of the shoe and a rapid abrasion. This effect is most marked under severe braking conditions such as obtained when heavy cars equipped with one brake shoe per wheel are stopped.

Shoes of the same type and hardness had a high rate of wear per unit of energy absorbed when a low coefficient of friction was developed and, conversely, a lower rate of wear when a higher coefficient of friction was developed.

Both the road and the laboratory tests confirmed previous tests and conclusions from analysis that the temperature of the working metal is the determining influence in coefficient of brake shoe friction.

The other factors that may be involved become effective chiefly as they affect the change of temperature of the working metal.

The general performance of the shoes as observed during the road tests formed the basis of the program established for laboratory tests, which resulted in the following deductions:

(*a*) The generation of the retarding forces and consequent absorption of the energy of the moving train is dependent upon but a very small quantity of brake shoe metal.

(*b*) The actual bearing area rather than the total face area of the shoe is the important factor in brake shoe performance.

(*c*) The magnitude of the bearing area changes throughout the stop and is greatest near the end of the stop.

(*d*) The bearing area shifts continuously from one portion of the surface to another during the stop.

(*e*) The principal factor in producing high friction for any given braking condition is the frequent shifting of the bearing area from the heated to the cooler spots over the face of the shoe.

(*f*) Slotted shoes or shoes that are cracked are more flexible than solid shoes and the bearing area shifts more readily than in the case of solid shoes.

(*g*) With shoes of the same type and approximately the same hardness, the wear per unit of work done is less with the slotted shoe than with the solid shoe. The stops with slotted shoes were always shorter and the mean coefficient of friction higher than with solid shoes.

(*h*) The shifting of the bearing area will tend to be more rapid if the size provides more available area for shoe bearing.

(*i*) The greater the pressure per square inch of bearing area, the lower will be the mean coefficient of friction.

(*j*) Flanged shoes provide more available area for bearing than unflanged shoes.

(*k*) The use of two shoes instead of one per wheel will result in a higher coefficient of friction and less wear per unit of work done.

(*l*) A comparison of the values of mean coefficient of friction for standard and for clasp brake conditions indicates a decided advantage for the clasp brake throughout the entire range of braking powers The gain in favor of the clasp brake with slotted shoes amounts to about 40%, at a braking power of 180%, and 100%, at a braking power of 40%, an average gain for the whole range of braking powers of about 70%.

(*m*) From a brake shoe standpoint the advantage of using two shoes instead of one shoe per wheel may be summed up as follows:—

First. The clasp brake shoe is associated with but one-half the wheel load and consequently has but one-half as much energy to absorb.

Second. The clasp brake shoe is working at only one-half the shoe pressure at which the standard shoe must work under the same braking power.

Third. The *available* working area for the same amount of energy to be absorbed is double.

A possible source of disadvantage when using two shoes per wheel is that a warped or poorly bearing shoe is subjected to less pressure tending to force it into a good contact with the wheel. For this reason, though the available shoe area is doubled when using clasp brakes, the actual *amount of working metal* throughout the stop may be less than with a single shoe, which is less capable of resisting the tendency of the heavier pressure to cause a better fit of shoe to wheel.

This, and an especially good shoe condition due to previous moderate pressure tests in each case, is an explanation why three of the 60 m.p.h. 150% B.P. electro-pneumatic stops with the single shoe train were shorter by 50 ft. than the best stops of either of the first or second clasp brake trains.

On the other hand the disadvantage and greater variability of the single shoe brake is evidenced in the fact that under the same conditions as cited above two stops with this train were longer than the longest stops without material wheel sliding made with either of the two clasp brake trains.

With plain solid shoes the durability will be increased 41.1% under clasp brake conditions as compared with that under single shoe conditions.

With plain slotted shoes the durability will be increased 33.5% under clasp brake conditions as compared with that under single shoe conditions.

The superior durability of the plain slotted shoe as compared with the plain solid amounts to 11.7% under single shoe brake conditions and 5.9% under clasp brake conditions.

The wear of the flanged solid shoes per unit of work done is 19% less than for plain solid shoes, and for flanged slotted 26% less than for plain slotted shoes, or 30% less than plain solid shoes.

The wear of plain slotted shoes per unit of work done is 5.4%

less than the wear of plain solid shoes, and the wear of the flanged slotted is 13.2% less than the wear of flanged solid shoes.

For the same amount of work done flanged solid cost 16% less than plain solid shoes, and flanged slotted cost 23% less than plain slotted, or 27% less than plain solid shoes.

Approximately 135% more stops will be required to wear out the flanged solid than will be required to wear out the plain solid shoe; 158% more stops to wear out the flanged slotted than the plain slotted shoe, and 171% more stops to wear out the flanged slotted than the plain solid shoe.

For any given braking condition with cast-iron brake shoes the indications are that the best relation will exist between shoe wear and mean coefficient of friction when the Brinell hardness of the cast iron is about 190.

Machine and road tests show a difference in stopping distance for the same type of shoe under the same braking conditions.

The effect of the difference in wheel surface conditions is one of the leading factors which go to make up the difference between machine and road tests. The difference in braking performance can be established and the factor expressing this difference be applied to laboratory results to predict the performance of a car or train.

Length of Stop

The stops and observed performance of the air brake, brake rigging and brake shoe are in agreement with the relation generally assumed to exist between the speed and other variables mentioned and resultant length of stop. This relation for straight, level track and neglecting air and internal friction on the one hand and the rotative energy of the wheels and axles on the other hand, is:

$$S_t = 1.467Vt + \frac{V^2}{30Pef}$$

in which the terms have the following significance and range of values according to conditions

S_t = length of stop to be expected in ft.

V = initial speed of train in m.p.h.

t = time at the beginning of the stop during which the brakes are to be considered as having no effect, to allow for the time element in the application of the brakes

	Kind of Air Brake Equipment	
	PM	UC Electro-Pnuematic
For a 12-car train		
t ranges from	2.0	0.70
t ranges to	2.5	0.85

P = nominal per cent braking power corresponding to the average cylinder pressure existing for that portion of the stop after the brake is considered fully applied,

With a single car or several similar cars, stopping without the locomotive attached, the value of P can be obtained from an average of all brake cylinder indicator cards or taken from one typical brake cylinder card, provided all cylinder pressures and foundation brake installations are substantially alike.

Where car weights differ or where the locomotive is attached, the average per cent braking power (P) for the train (total weight W) may be calculated from the formula

$$P_t = \frac{\Sigma P_c W_c + P_n W_n + P_e W_e}{W_t}$$

in which

P_c = nominal per cent braking power corresponding to average cylinder pressure of cars as noted above

W_c = weight of car in lb.

W_n = actual total weight of tender in lb.

P_n = nominal per cent braking power corresponding to tender cylinder pressure and to W_n

W_e = weight of engine in lb.

P_e = nominal per cent braking power corresponding to engine cylinder pressure (aggregate for driver, truck and trailer wheels according to distribution of weights and nominal braking power)

$e \times f$ = product of efficiency of brake rigging and mean coefficient of brake shoe friction, in which

e varies with the type and installation of brake rigging and with the per cent braking power, though the latter effect can be but slight throughout the range of ordinary emergency braking powers

f varies with the kind and initial condition of the brake shoe bearing surface, the initial speed and the various influences affecting the conditions of the bearing surface during the stop.

As no satisfactory separate determination of e and f was found possible the combined factor $e \times f$ is given in the following table, the values being representative of the best performance that might reasonably be expected under conditions comparable with those of this test.

TABLE 10. VALUES OF $e \times f_B$

Kind of Brake Rigging		Clasp Brake		Single Shoe[1]	
Type of Brake Shoe		Plain	Flanged	Plain	Flanged
Speed m.p.h.	Braking Power				
30	125	0.141	0.169	0.108	0.112
	150	0.129	0.154	0.099	0.103
	180	0.118	0.141	0.090	0.094
60	125	0.103	0.122	0.074	0.090
	150	0.094	0.112	0.068	0.082
	180	0.086	0.102	0.062	0.075
80	125	0.092	0.109	0.070	0.071
	150	0.084	0.100	0.064	0.068
	180	0.077	0.092	0.059	0.062

[1]Value of data uncertain due to non-uniform brake shoe conditions.

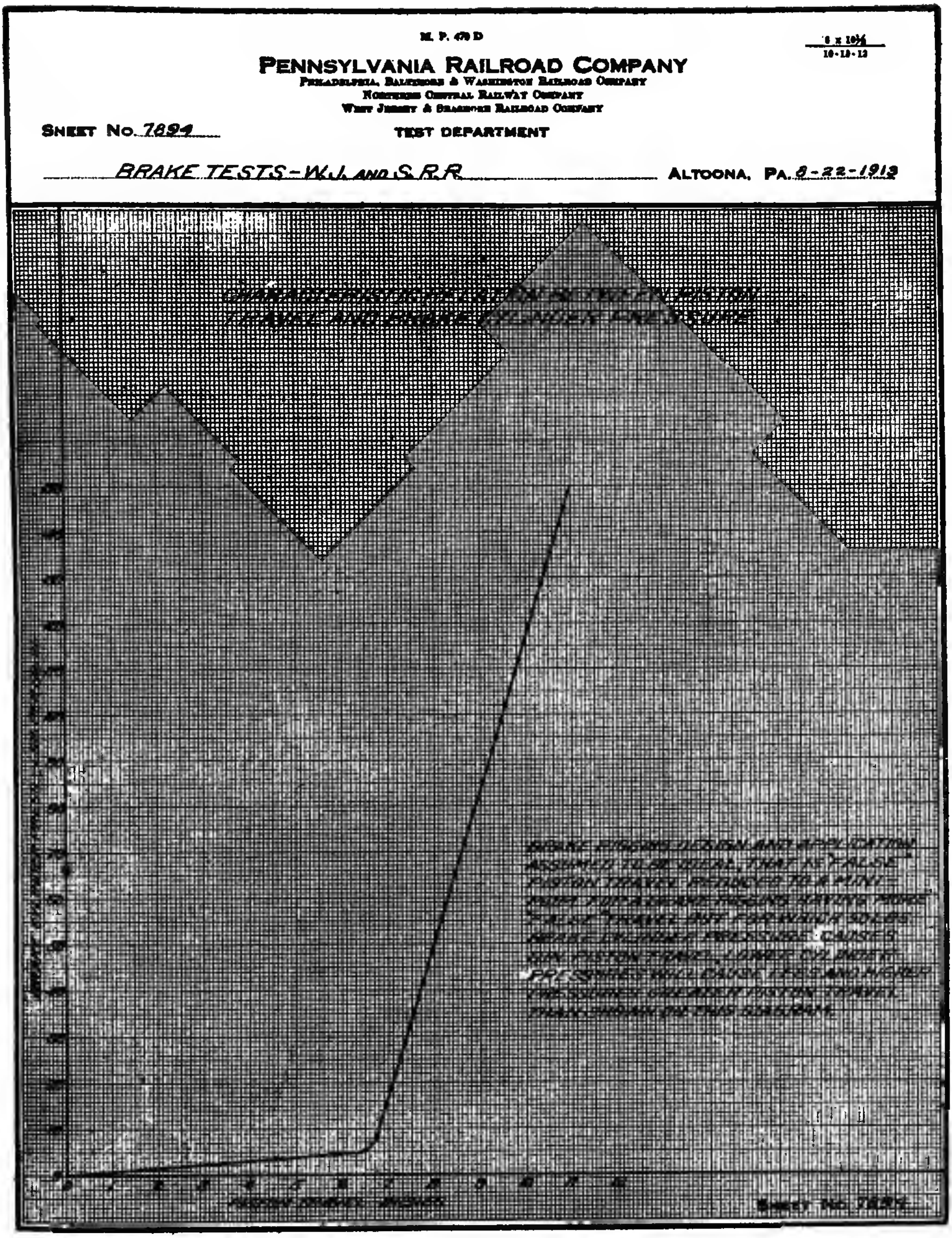


Fig. 31.

PISTON TRAVEL, BRAKE CYLINDER PRESSURE.

The relation between piston travel end breke cylinder pressure for an ideal breke installation.

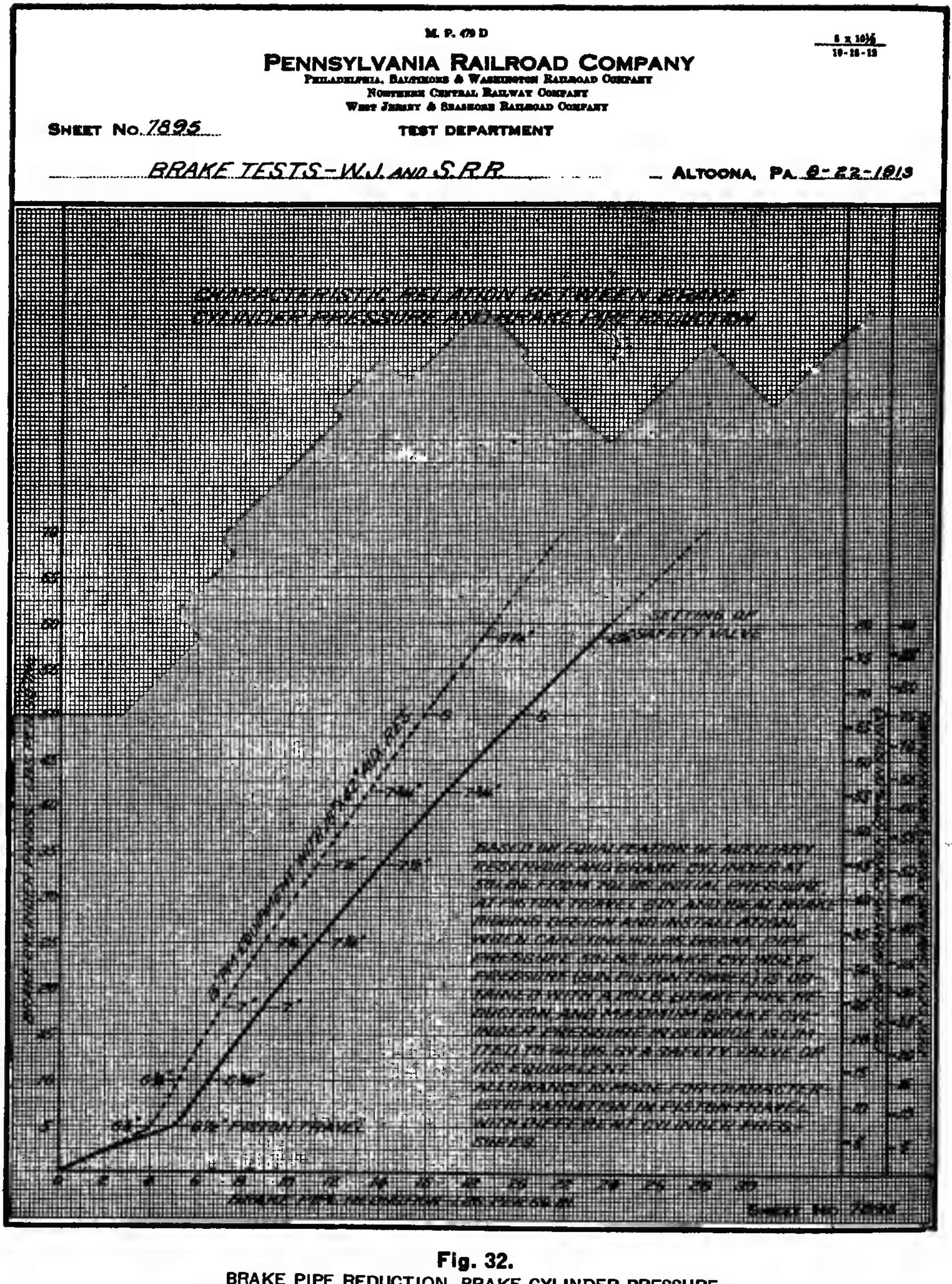


Fig. 32.

BRAKE PIPE REDUCTION, BRAKE CYLINDER PRESSURE.

Characteristic relation showing lack of flexibility with large size auxiliary reservoir.

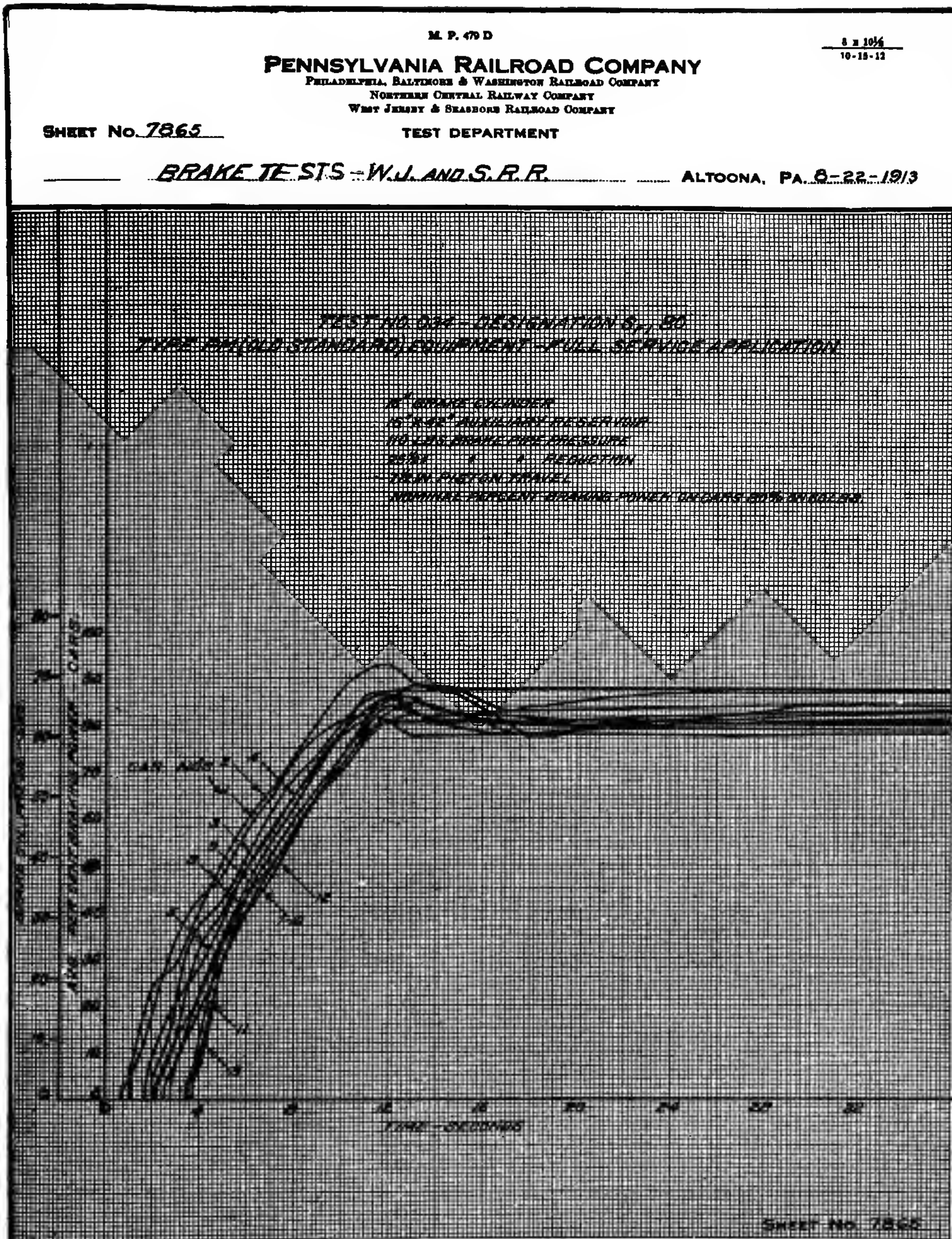


Fig. 33.

BRAKE CYLINDER PRESSURE, PM EQUIPMENT.

This shows the rate of building up of brake cylinder pressure in a full service application.

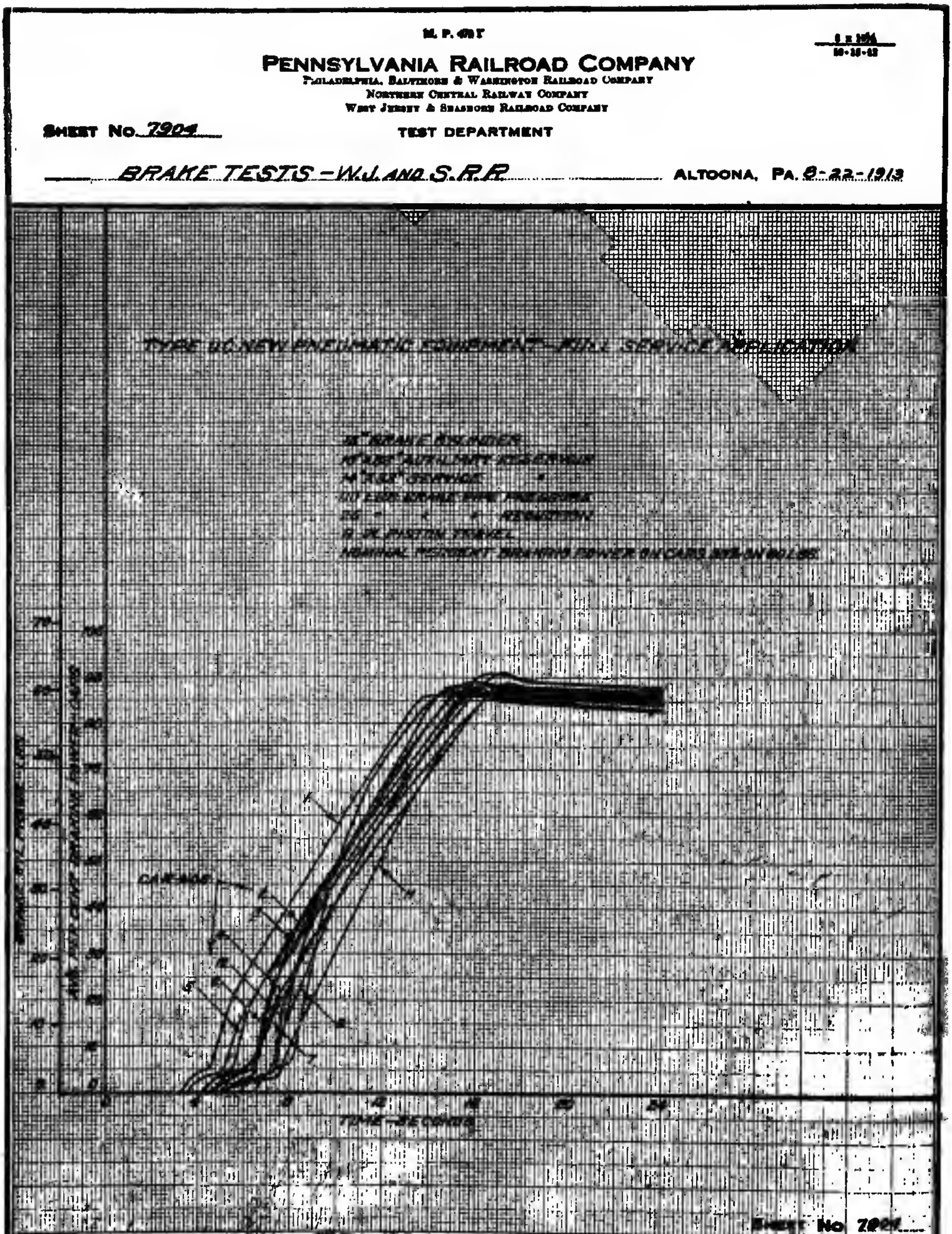


Fig. 34.

BRAKE CYLINDER PRESSURE, UC PNEUMATIC EQUIPMENT.

This diagram was taken in rack tests and shows the improvements which were made in the universal valve as a result of the earlier tests.

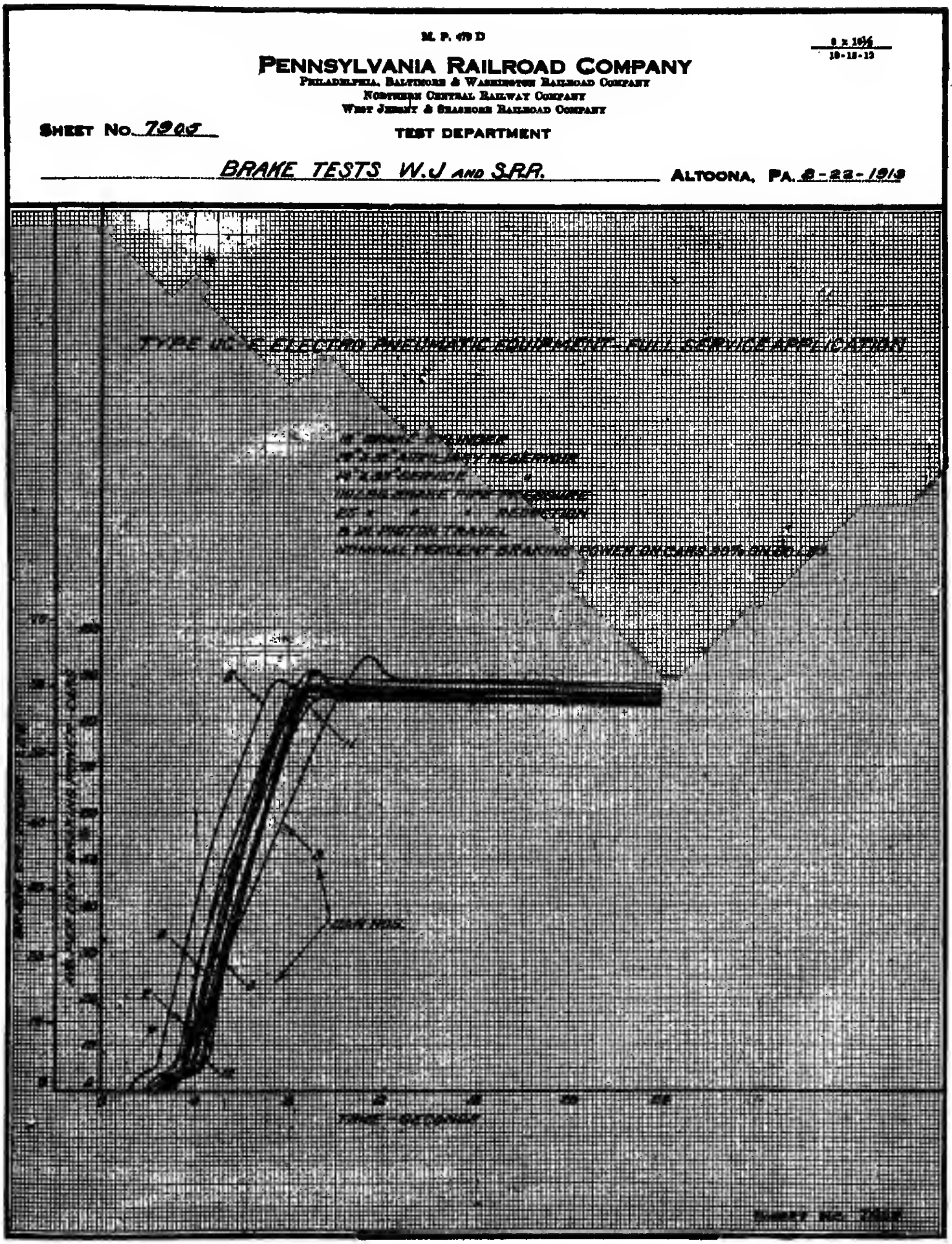


Fig. 35.

BRAKE CYLINDER PRESSURE, UC ELECTRO-PNEUMATIC EQUIPMENT.

The rack tests for a full service application with the electric control are shown in this diagram. The improved construction of the valve causes a more regular action when maximum brake cylinder pressure is reached.

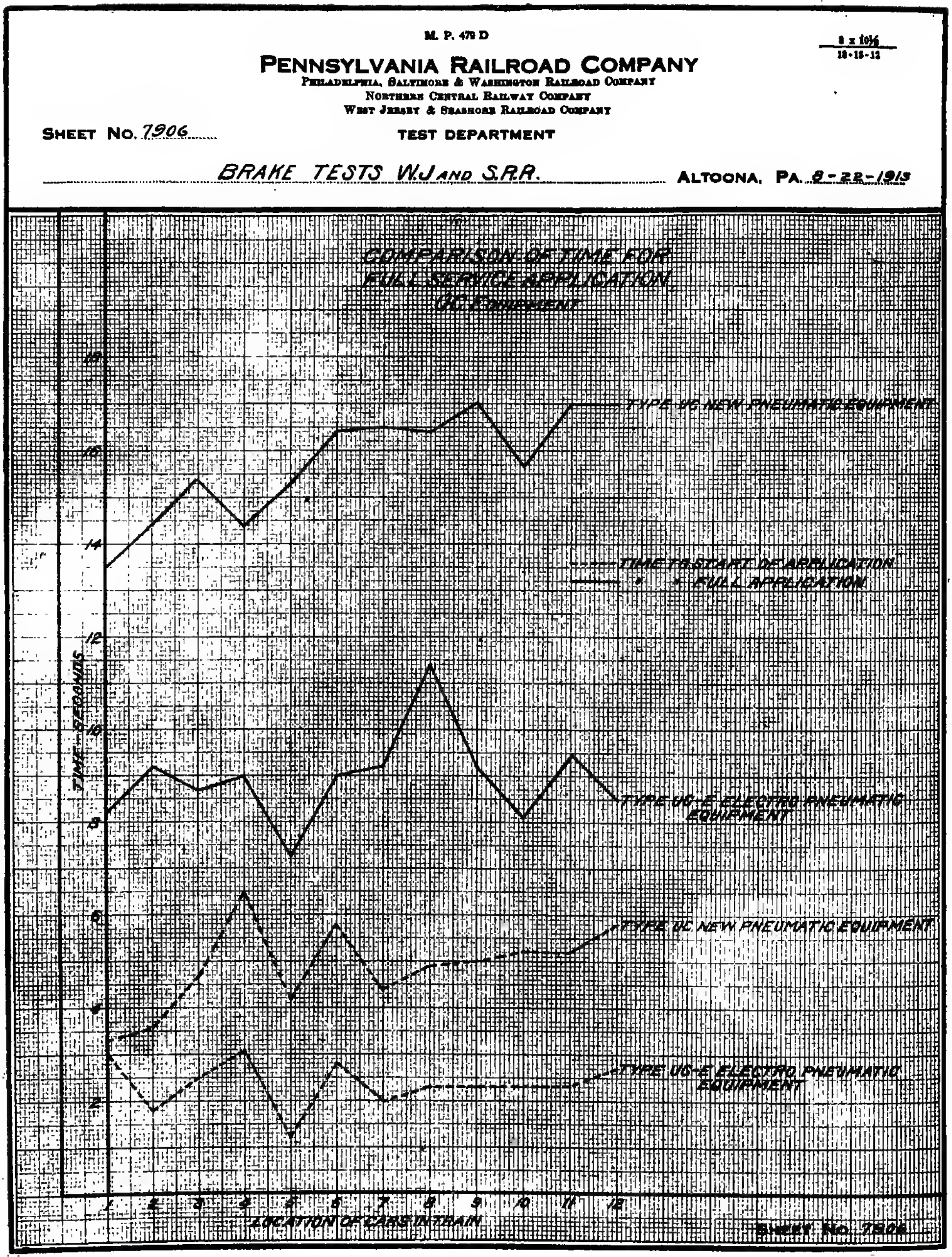


Fig. 36.

TIME TO APPLY BRAKES.

The time to start of application and the attainment of full pressure on all cars in a service application is shown in this figure made up from rack tests. An improvement in the universal valve is indicated as compared with the performance during the tests.

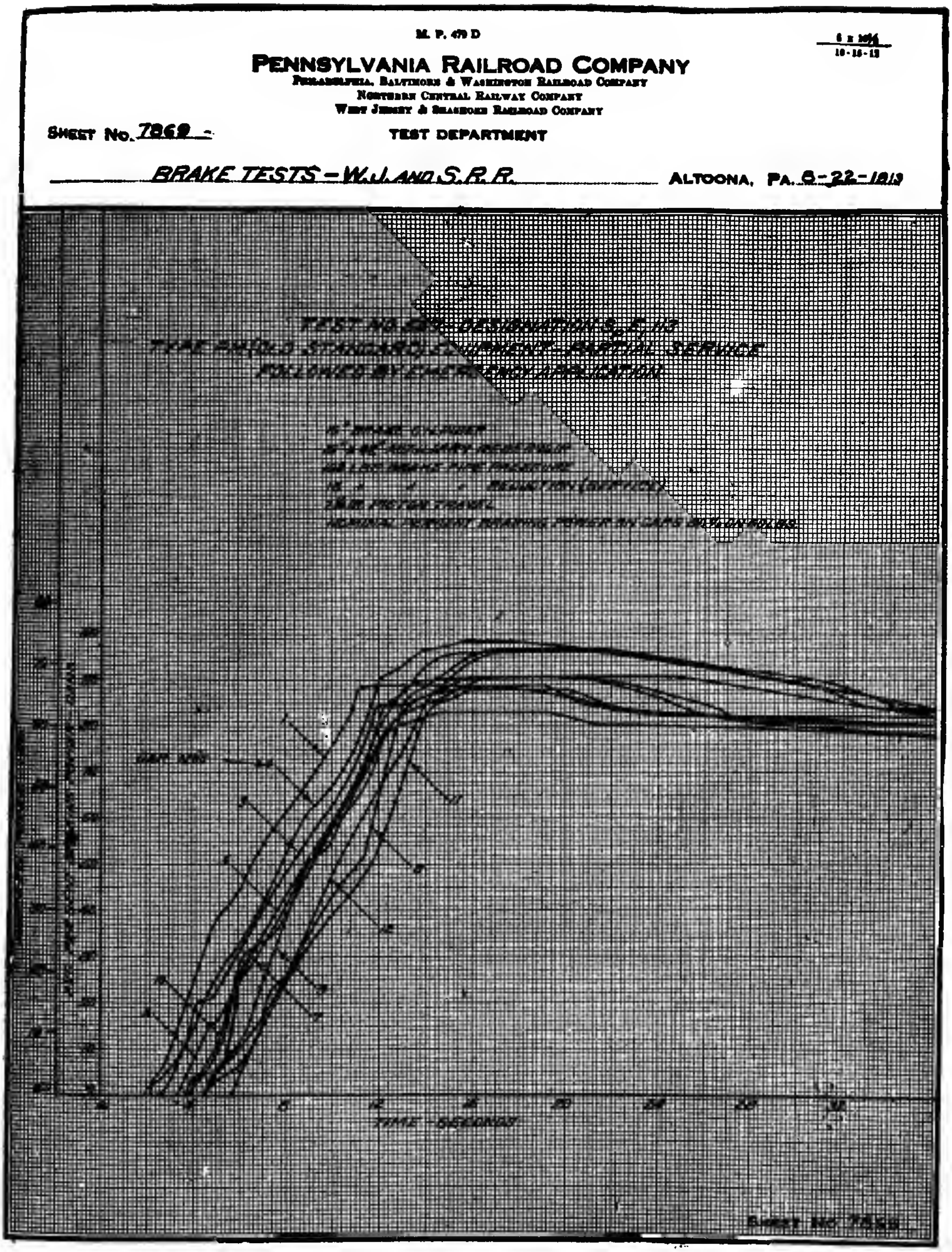


Fig. 37.

BRAKE CYLINDER PRESSURE, PM EQUIPMENT.

This diagram shows the rate of building up of brake cylinder pressure in a partial service followed by an emergency application. The emergency action is scarcely distinguishable.

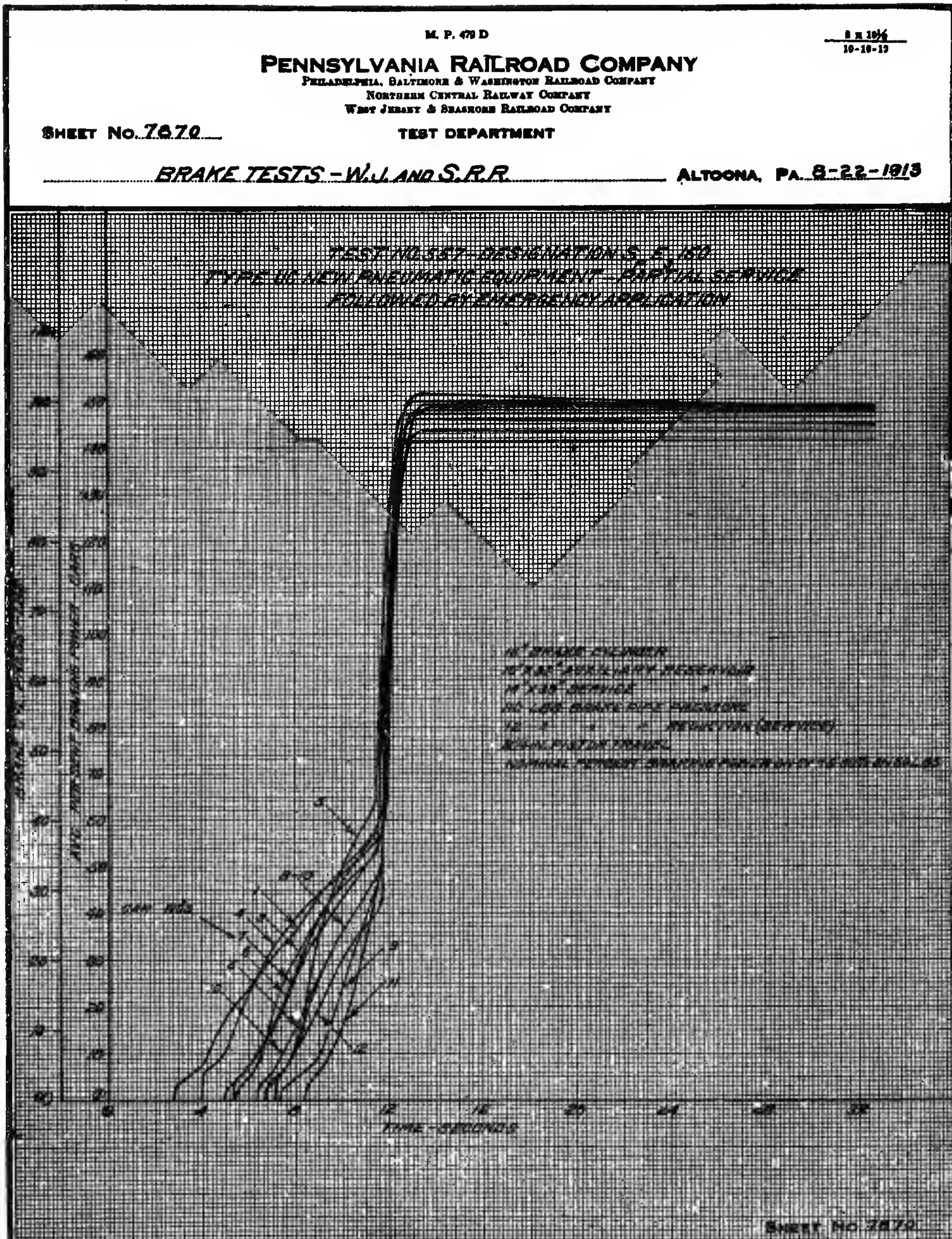


Fig. 38.

BRAKE CYLINDER PRESSURE, UC PNEUMATIC EQUIPMENT.

This diagram shows a partial service followed by an emergency application. The rapid rise in pressure clearly marks the emergency application. The non-uniformity of the service application has been greatly reduced as will be seen in Fig. 34.

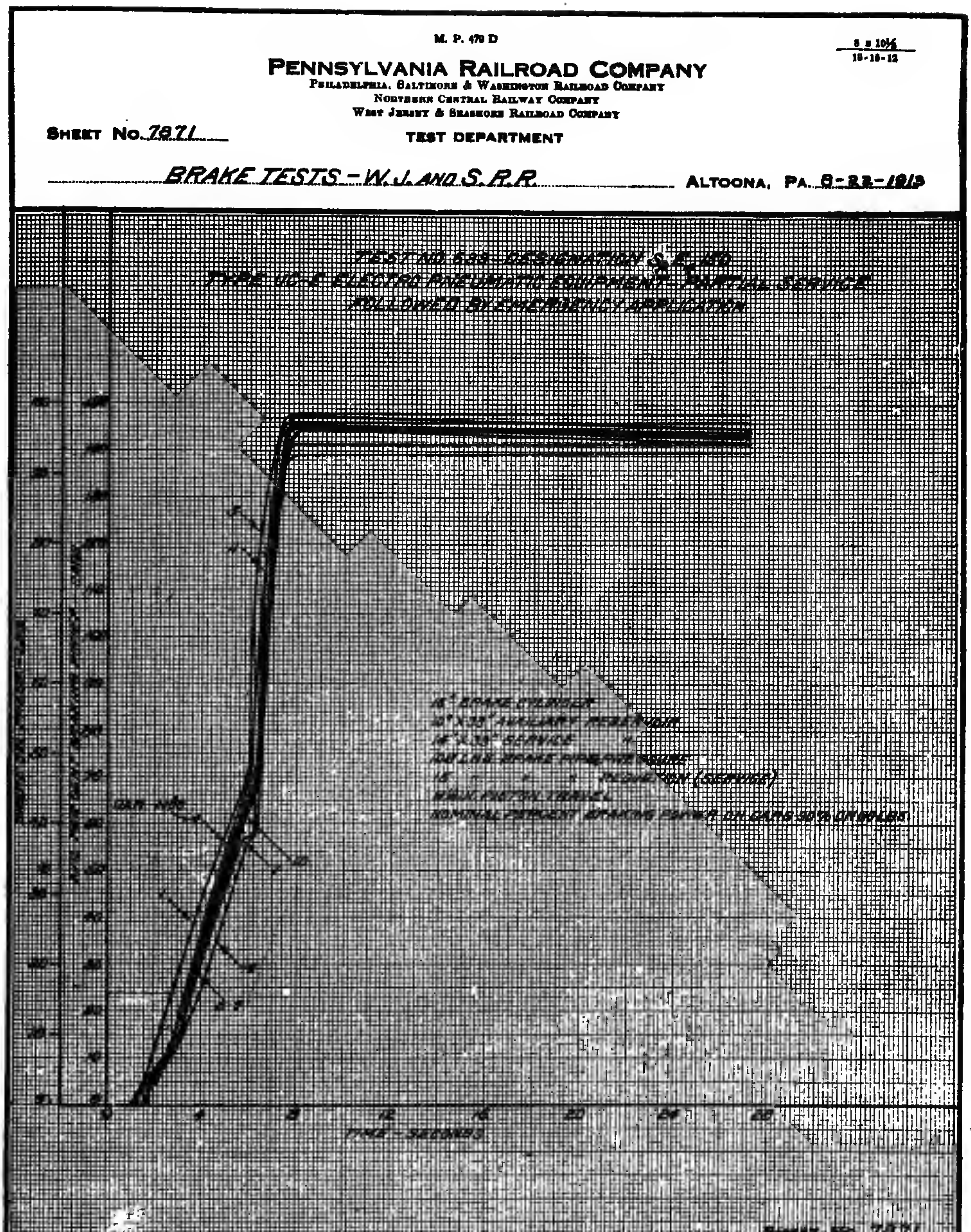


Fig. 39.

BRAKE CYLINDER PRESSURE, UC ELECTRO-PNEUMATIC EQUIPMENT.

The diagram shows a partial service followed by an emergency application. The electric control produces a nearly simultaneous application on all cars of the train.

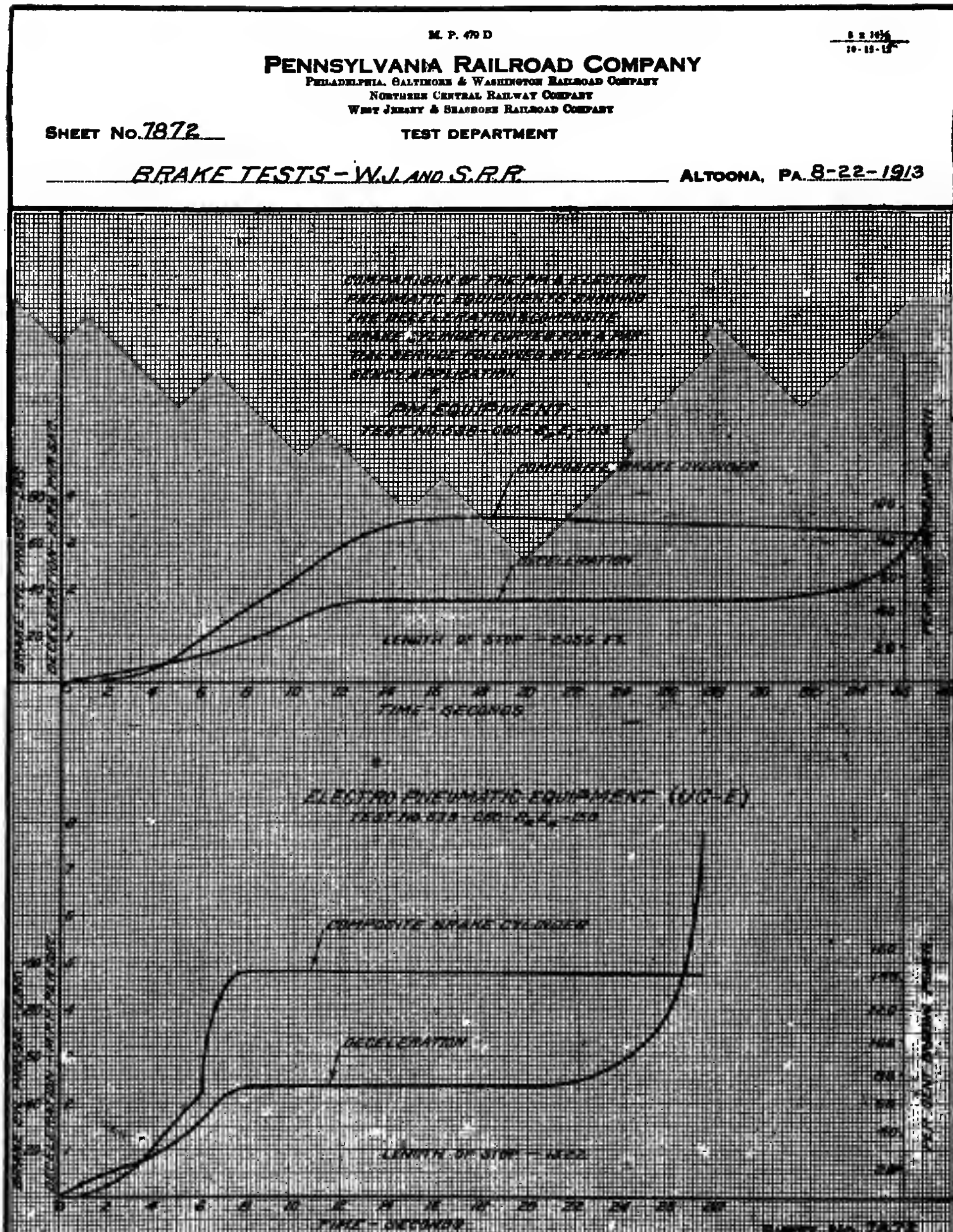


Fig. 40.

BRAKE CYLINDER PRESSURE, DECELERATION.

This shows the combined brake cylinder pressures for the entire train and the resulting improvement in deceleration due to improved pneumatic features and the electric control.

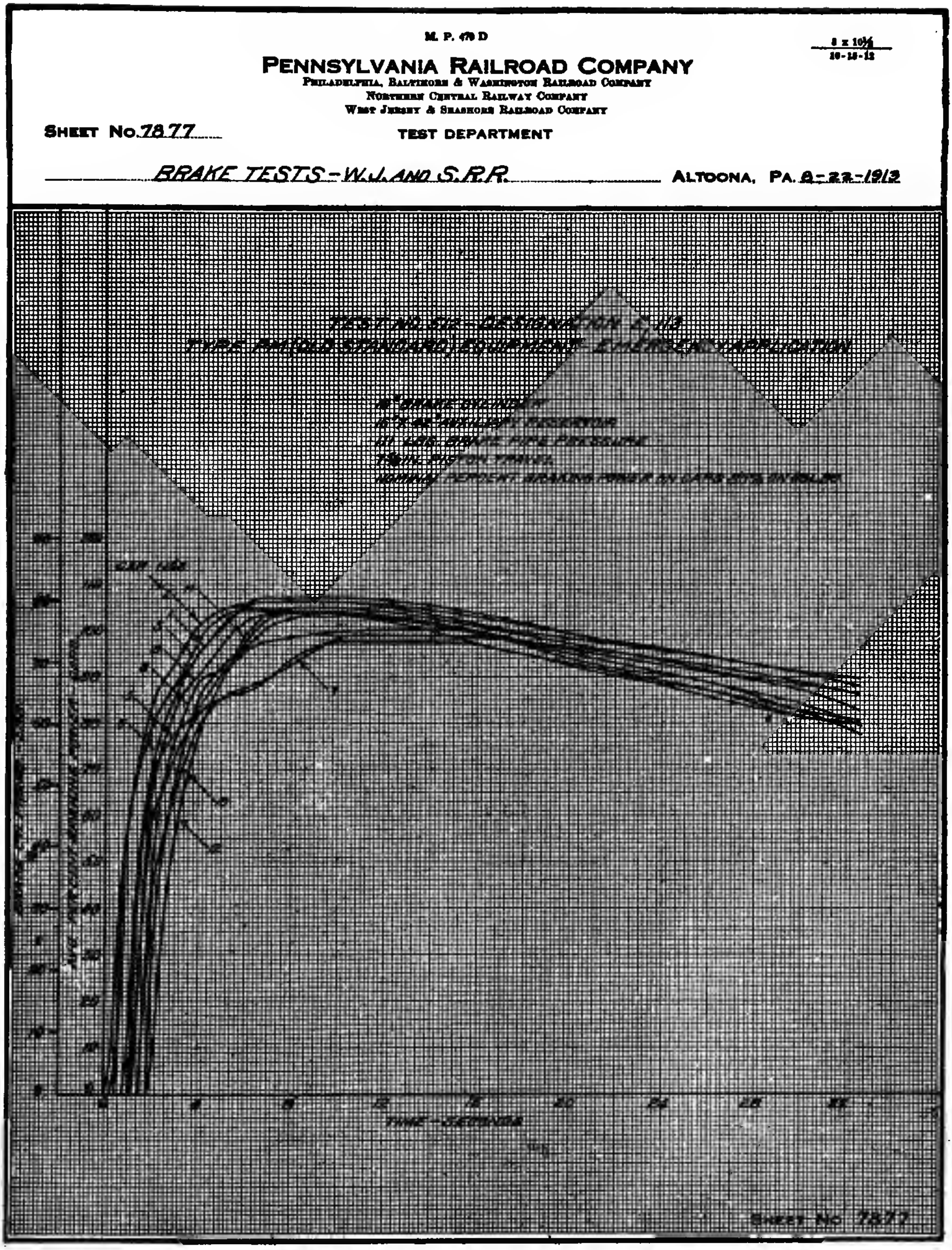


Fig. 41.

BRAKE CYLINDER PRESSURE, PM EQUIPMENT.

Maximum emergency cylinder pressure is obtained in about 8 seconds on the entire train. The blow down action of the high-speed reducing valve is well illustrated.

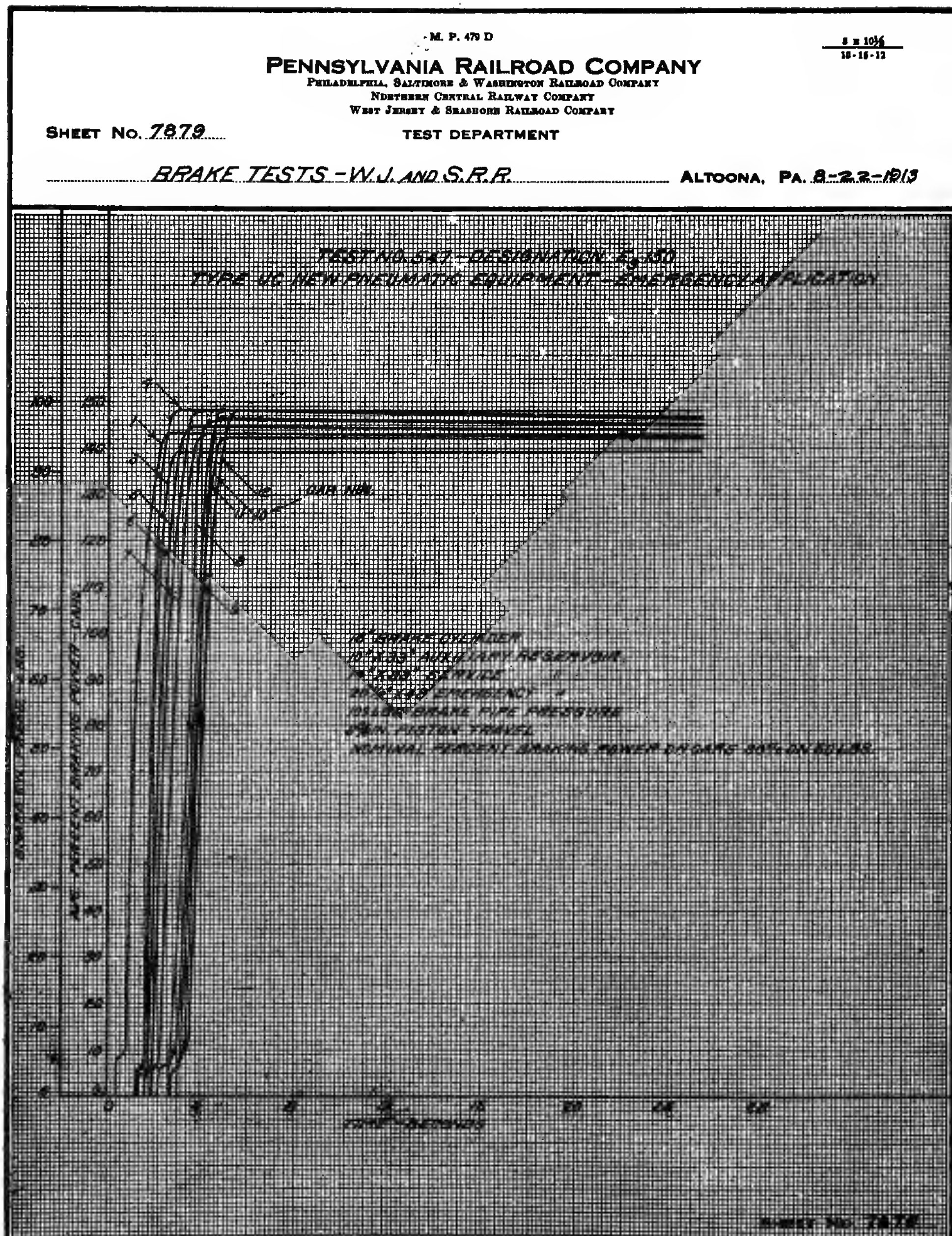


Fig. 42.

BRAKE CYLINDER PRESSURE, UC PNEUMATIC EQUIPMENT.

The emergency application without the electric control is here illustrated. The slow serial action shown by the spacing of the vertical lines has been improved by a slight alteration in the valves subsequent to the test here shown. Fig. 43 shows this improvement.

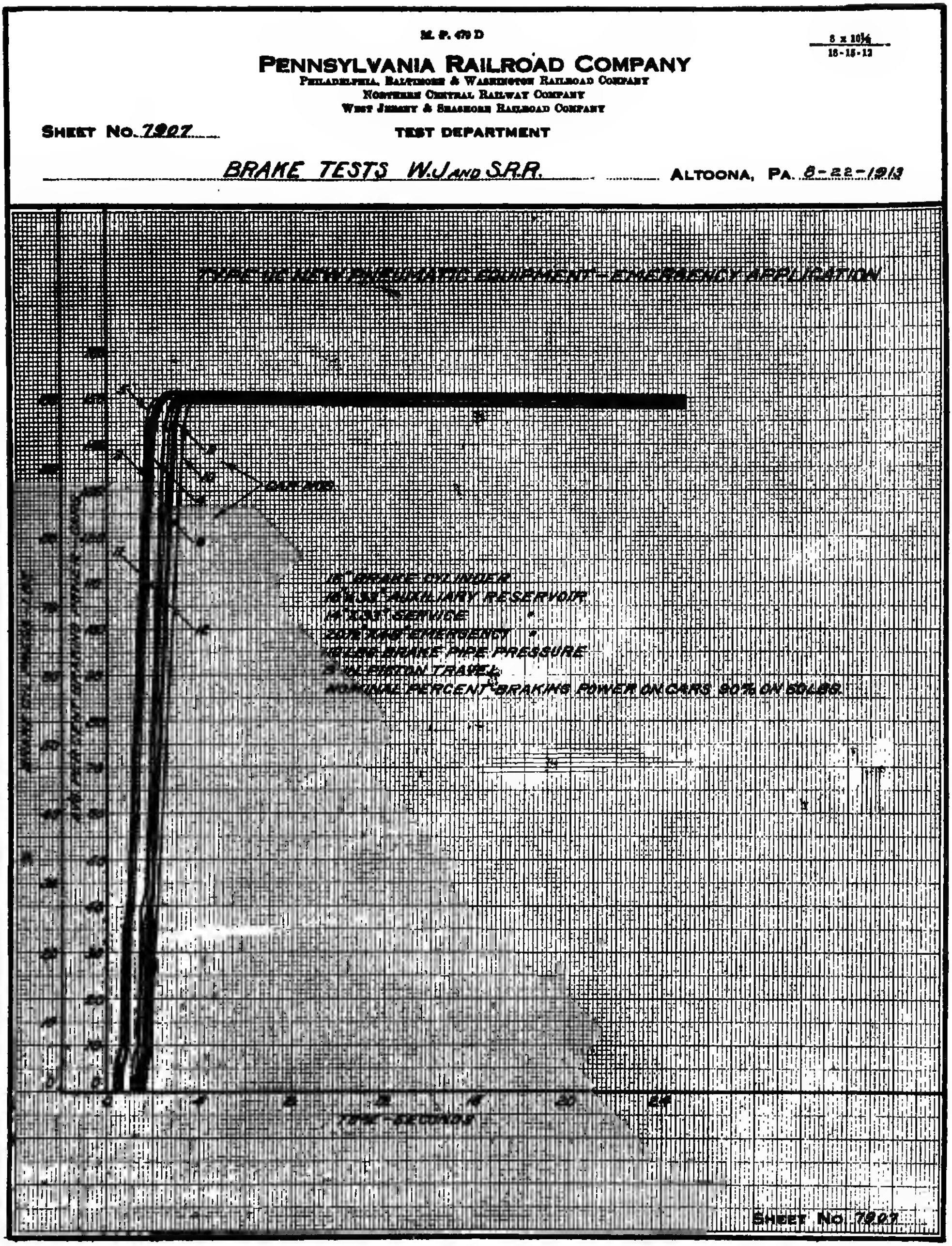


Fig. 43.

BRAKE CYLINDER PRESSURE, UC PNEUMATIC EQUIPMENT.

This diagram shows an emergency application taken in rack tests on the universal valve improved as a result of the road trials. A comparison with Fig. 42 shows the elimination of slow serial action which has been made.

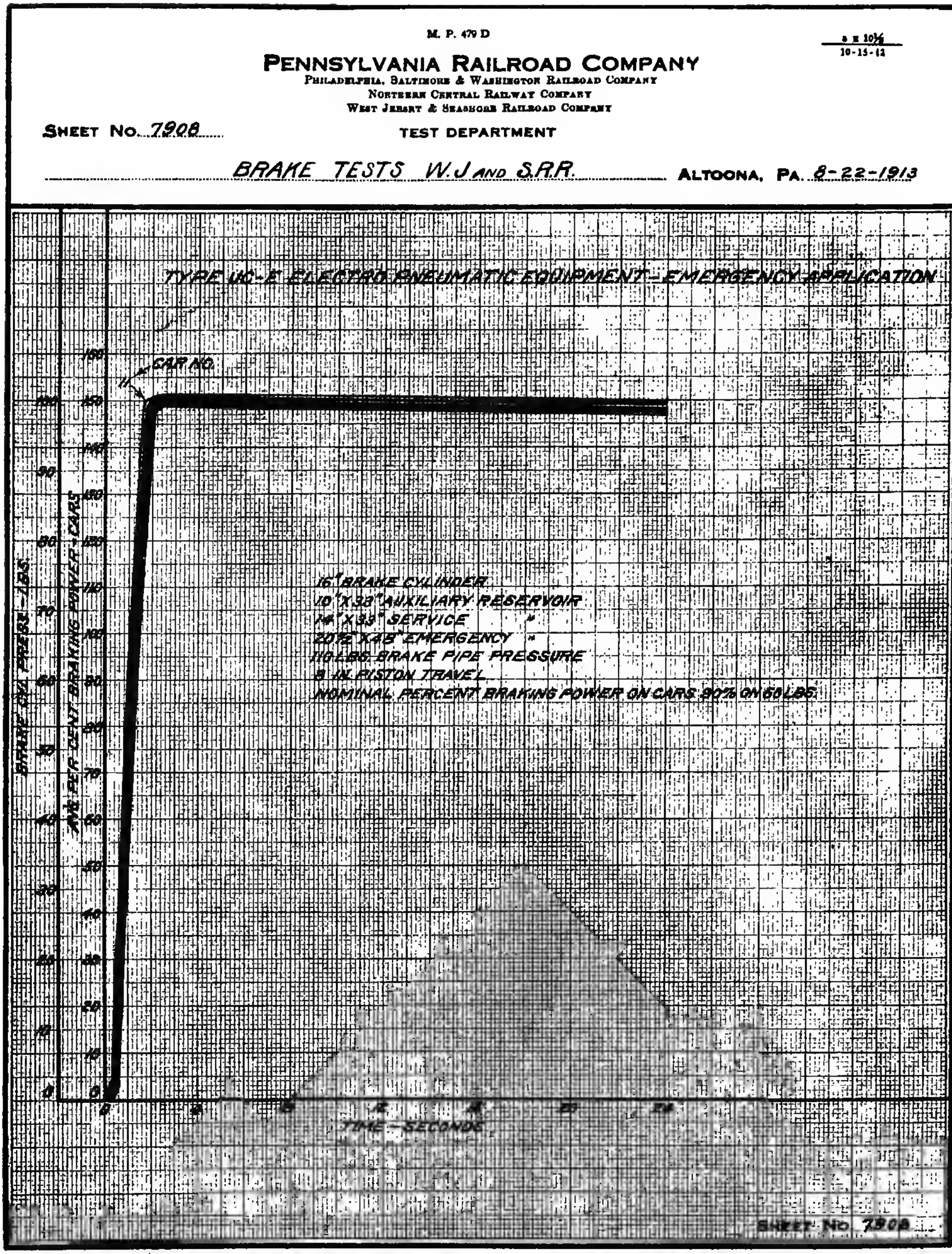


Fig. 44.

BRAKE CYLINDER PRESSURE, UC PNEUMATIC EQUIPMENT.

The rapid simultaneous rise of brake cylinder pressure on all cars, will result in the heaviest steel passenger train stopping, with an emergency application from a speed of 60 miles per hour, in less than its own length.

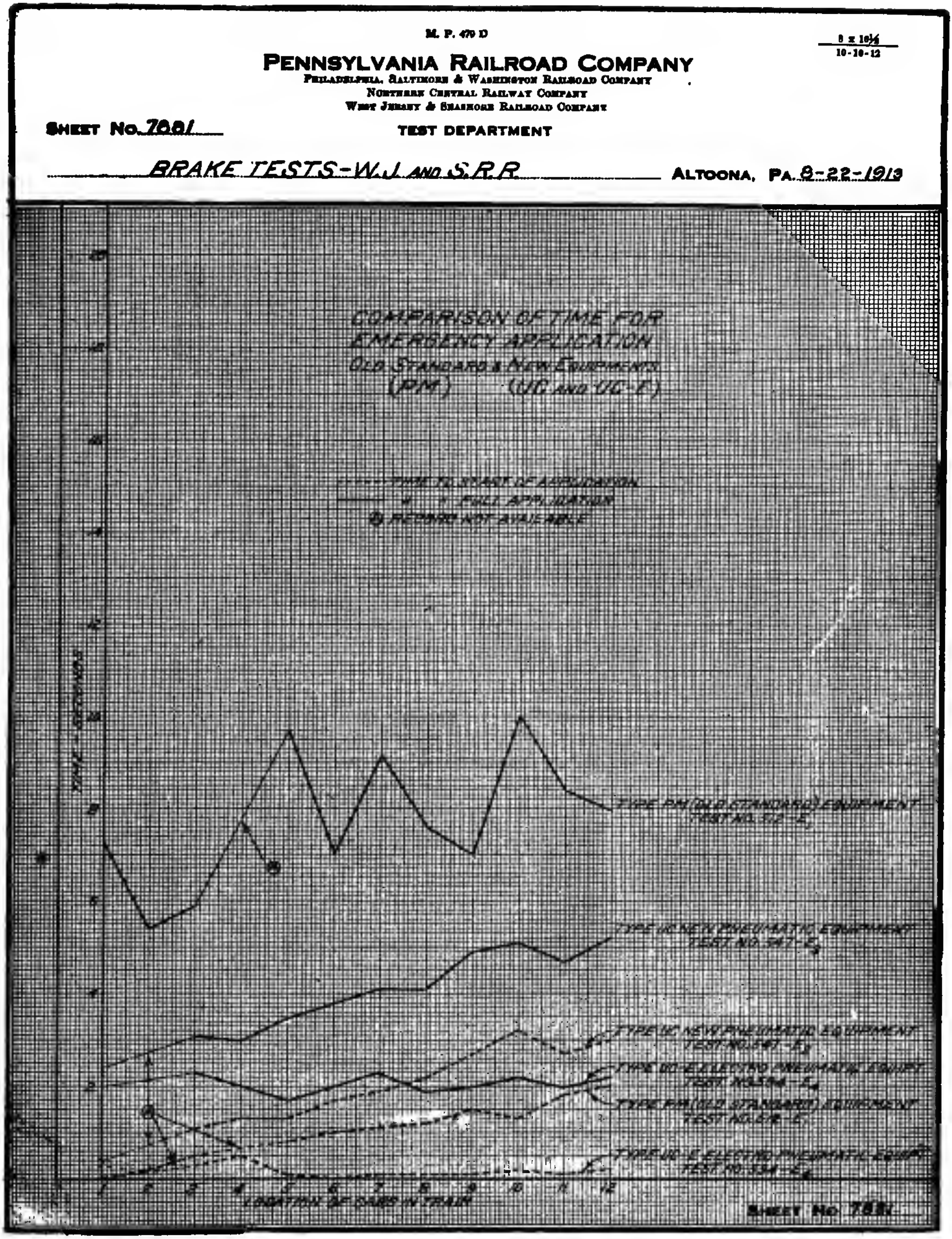


Fig. 45.

TIME TO APPLY BRAKES.

The lower dotted line and the lower full line for the electro-pneumatic equipment shows the elimination of serial action and a short time to attain maximum braking power on all cars.

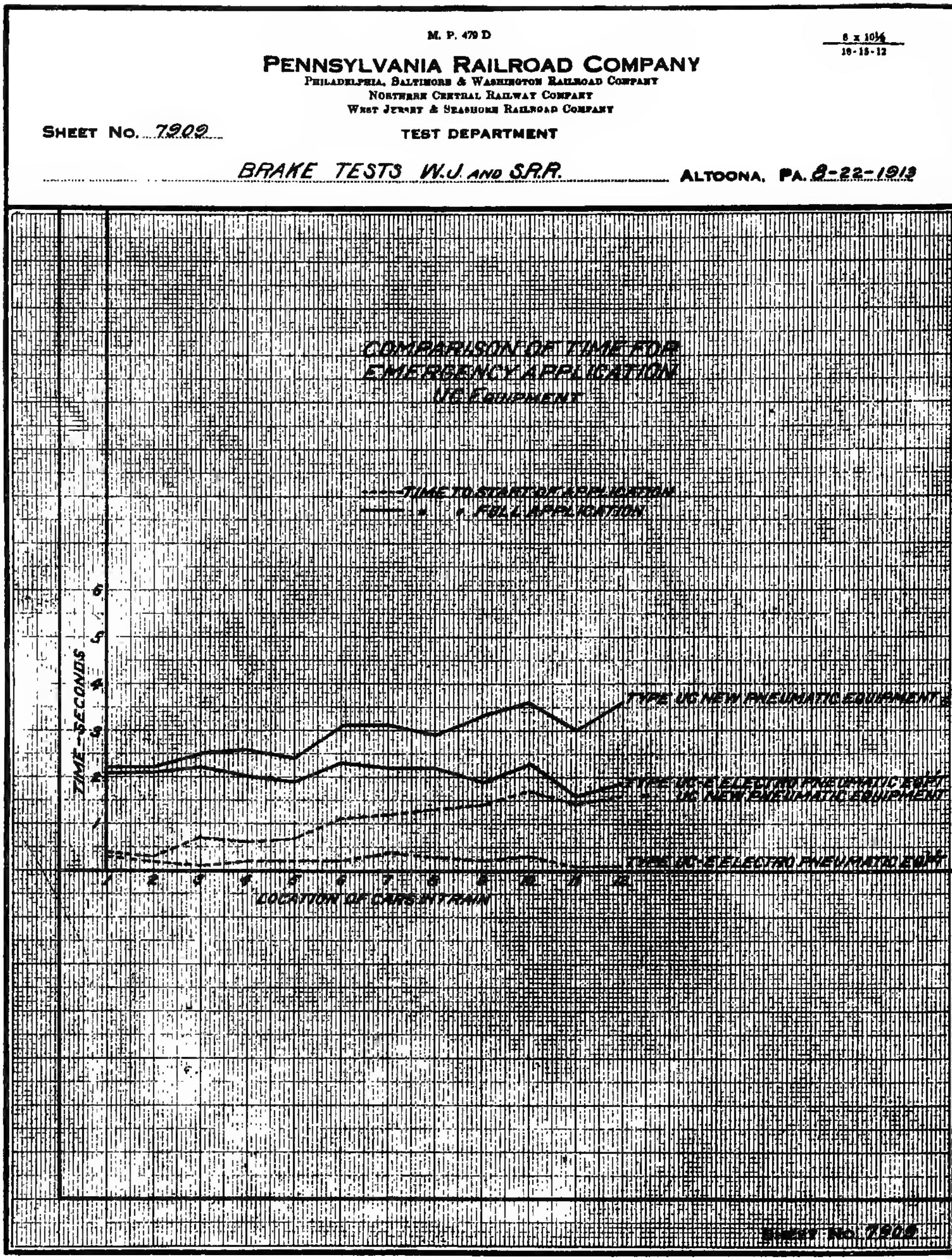


Fig. 46.

TIME TO APPLY BRAKES, UC PNEUMATIC AND ELECTRO-PNEUMATIC EQUIPMENT.

The time to start of application and the attainment of full emergency braking power on all cars is shown in this diagram which is representative of the equipment now being furnished.

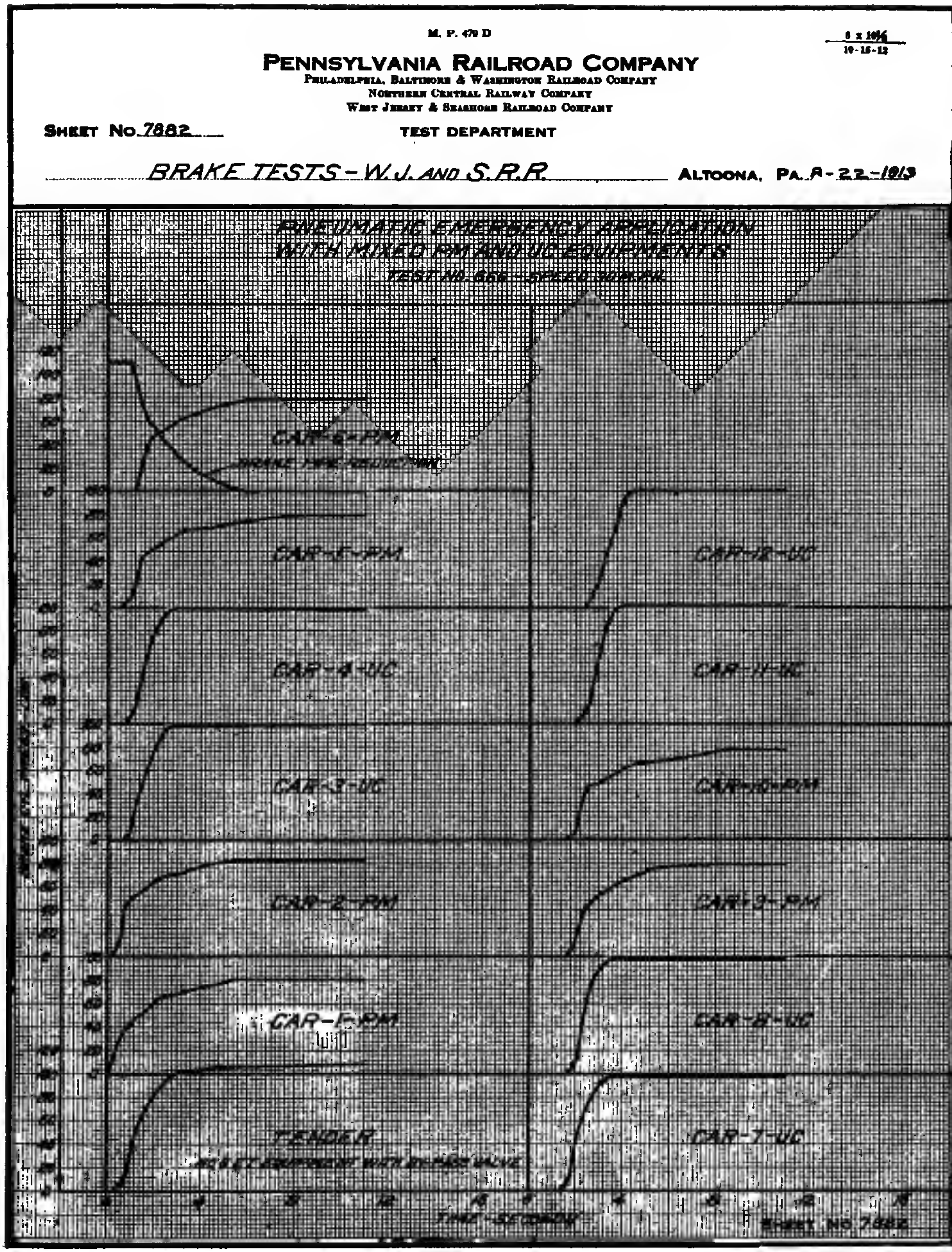


Fig. 47.

MIXED EQUIPMENT STOP, EMERGENCY APPLICATION.

The UC pneumatic equipment when mixed with the PM equipment in a train did not cause objectionable shocks. This stop was mede in 287 feet, from 30 m.p.h.

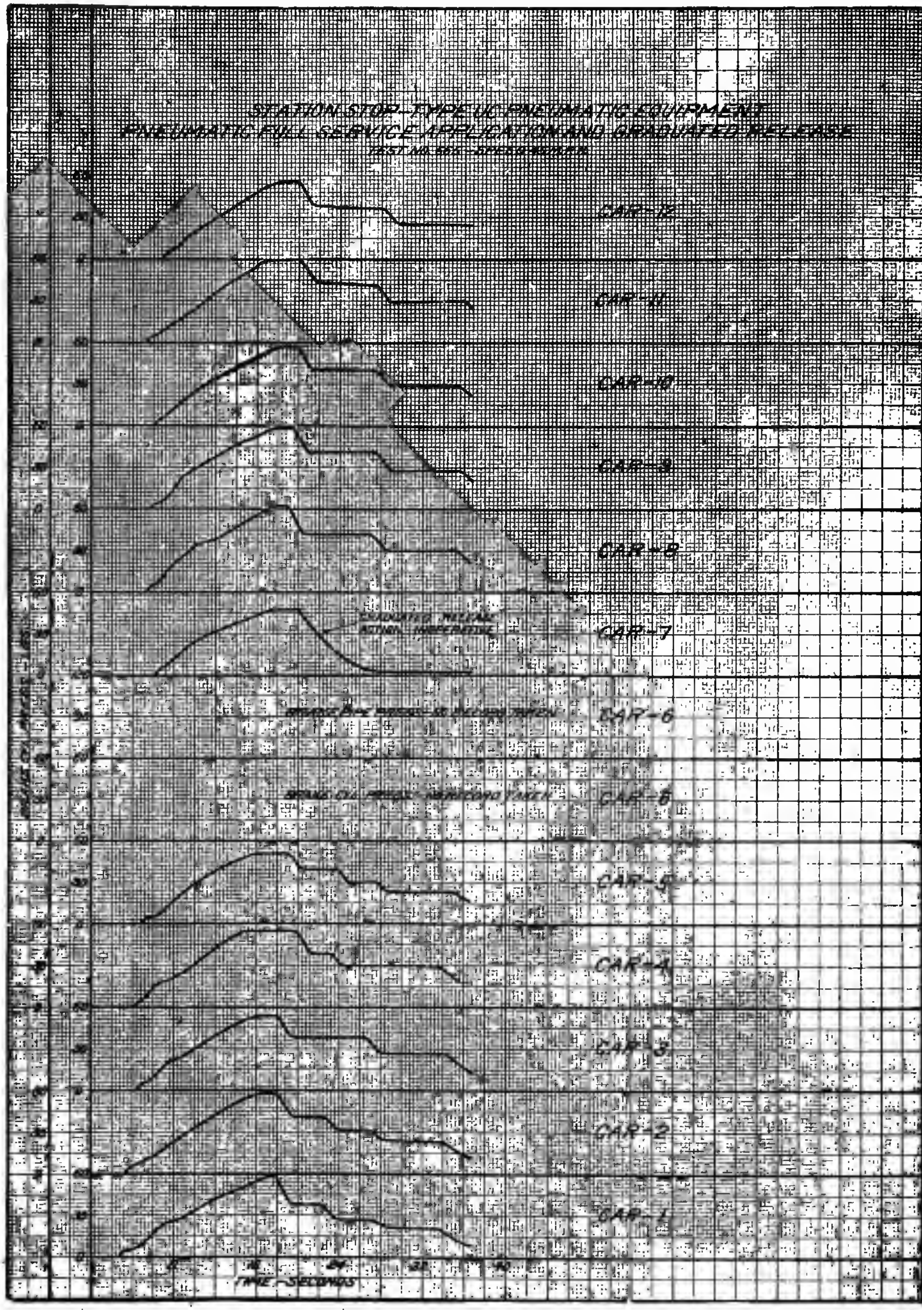


Fig. 48.

FULL SERVICE STOP, GRADUATED RELEASE. UC PNEUMATIC EQUIPMENT.

The graduated release is clearly shown.

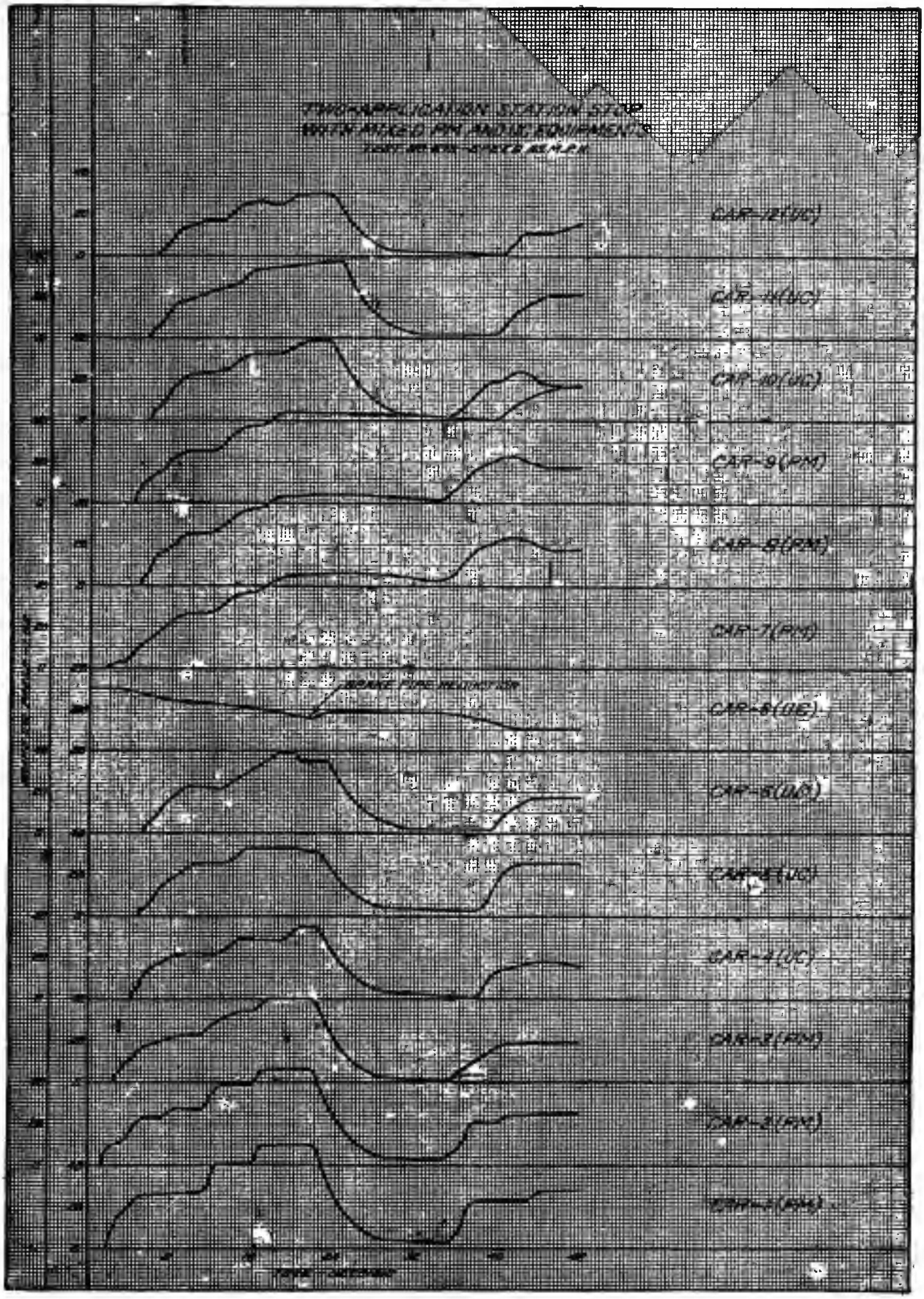


Fig. 49.

SERVICE STOP, TWO APPLICATIONS. MIXED PM AND UC PNEUMATIC EQUIPMENTS.

The failure of the PM equipment to release on cars seven, eight and nine is characteristic on long trains.

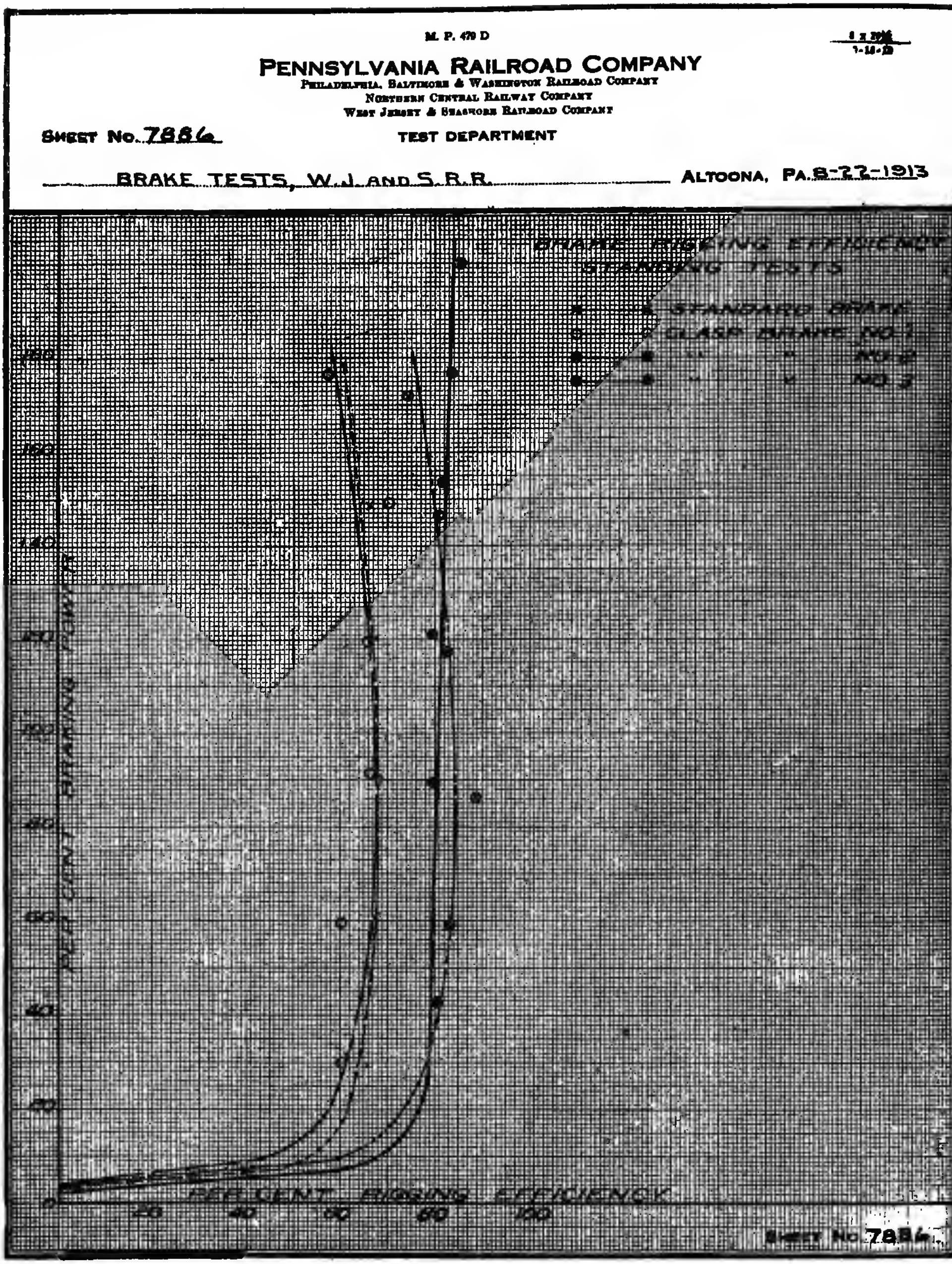


Fig. 50.

BRAKE RIGGING EFFICIENCY.

The efficiency taken in standing tests ranged from 60 to 85 per cent.

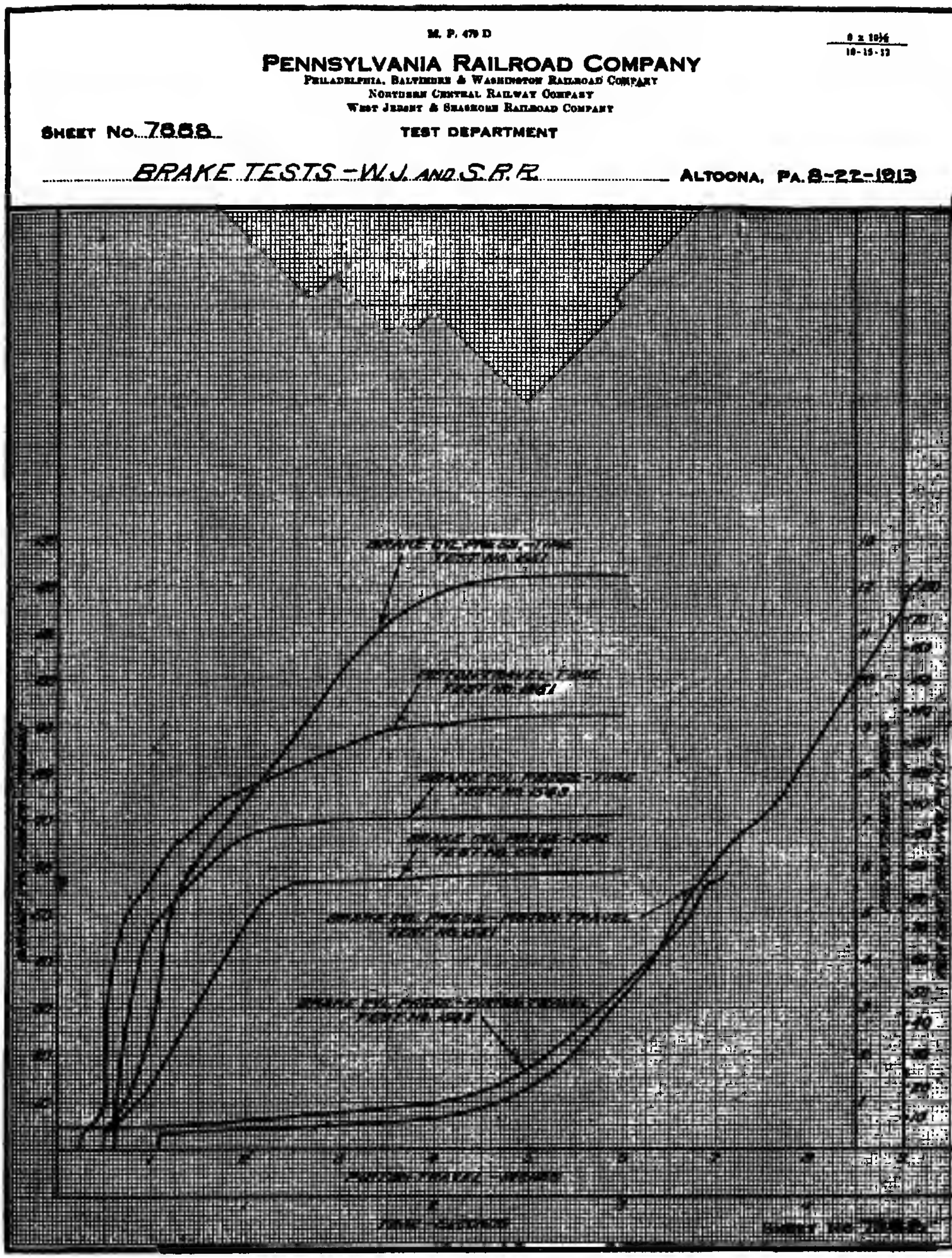


Fig. 51.
PISTON TRAVEL AND BRAKE CYLINDER PRESSURE.

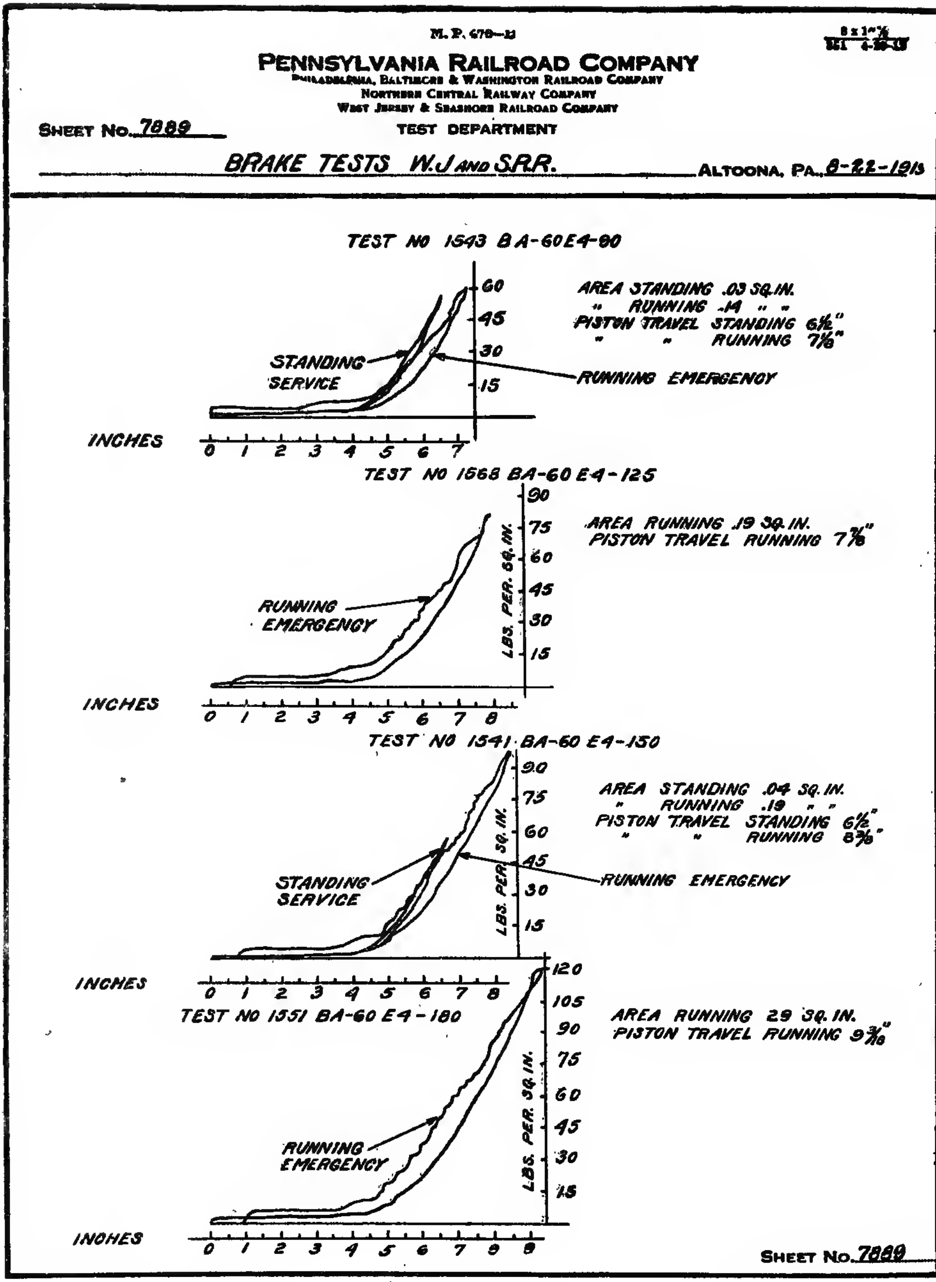


Fig. 52.

BRAKE CYLINDER CARDS.

These indicator cards were taken during standing service tests end in running emergency tests.

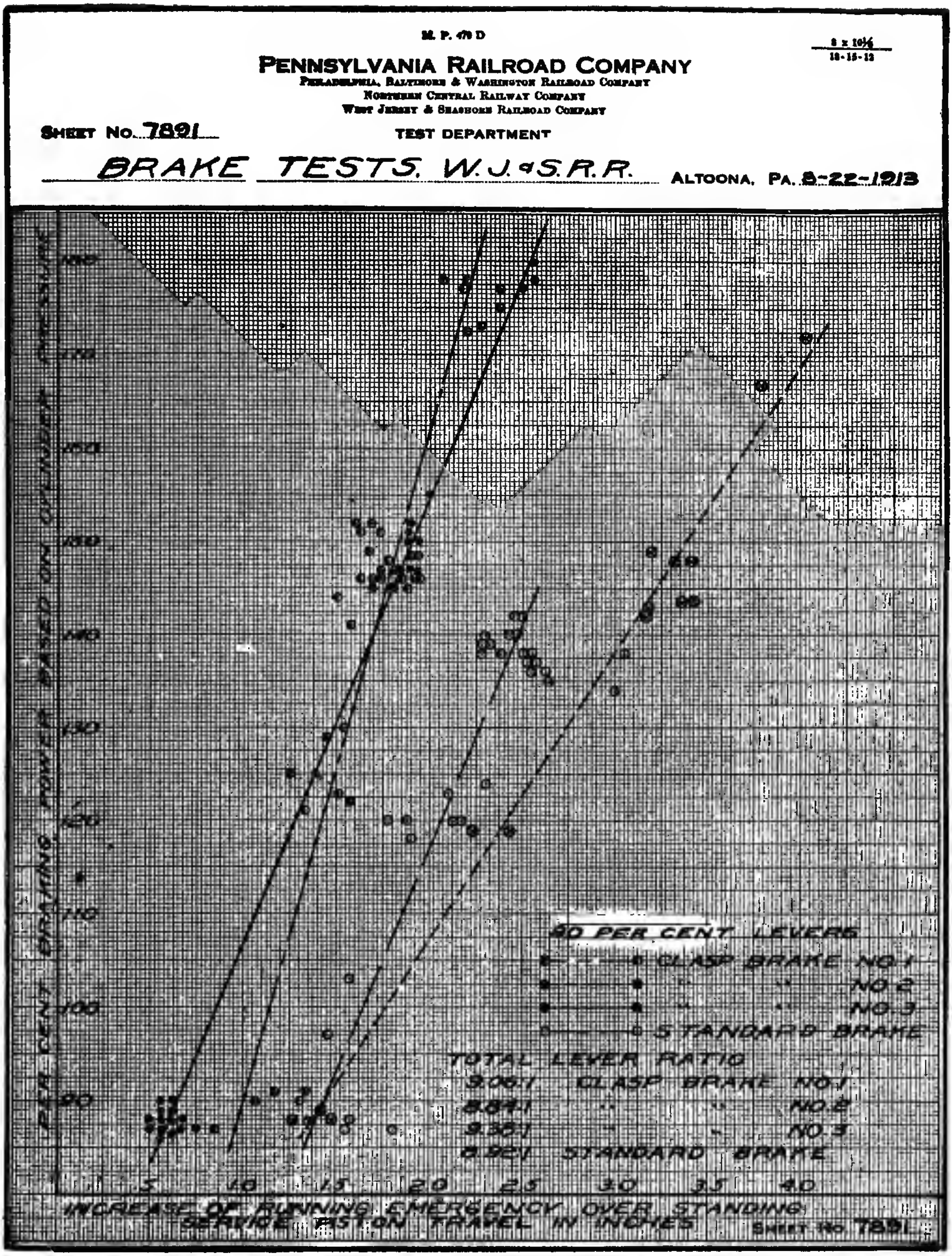


Fig. 53.

PISTON TRAVEL.

The increase in piston travel for various percentages of braking power with the 90 per cent. cylinder levers.

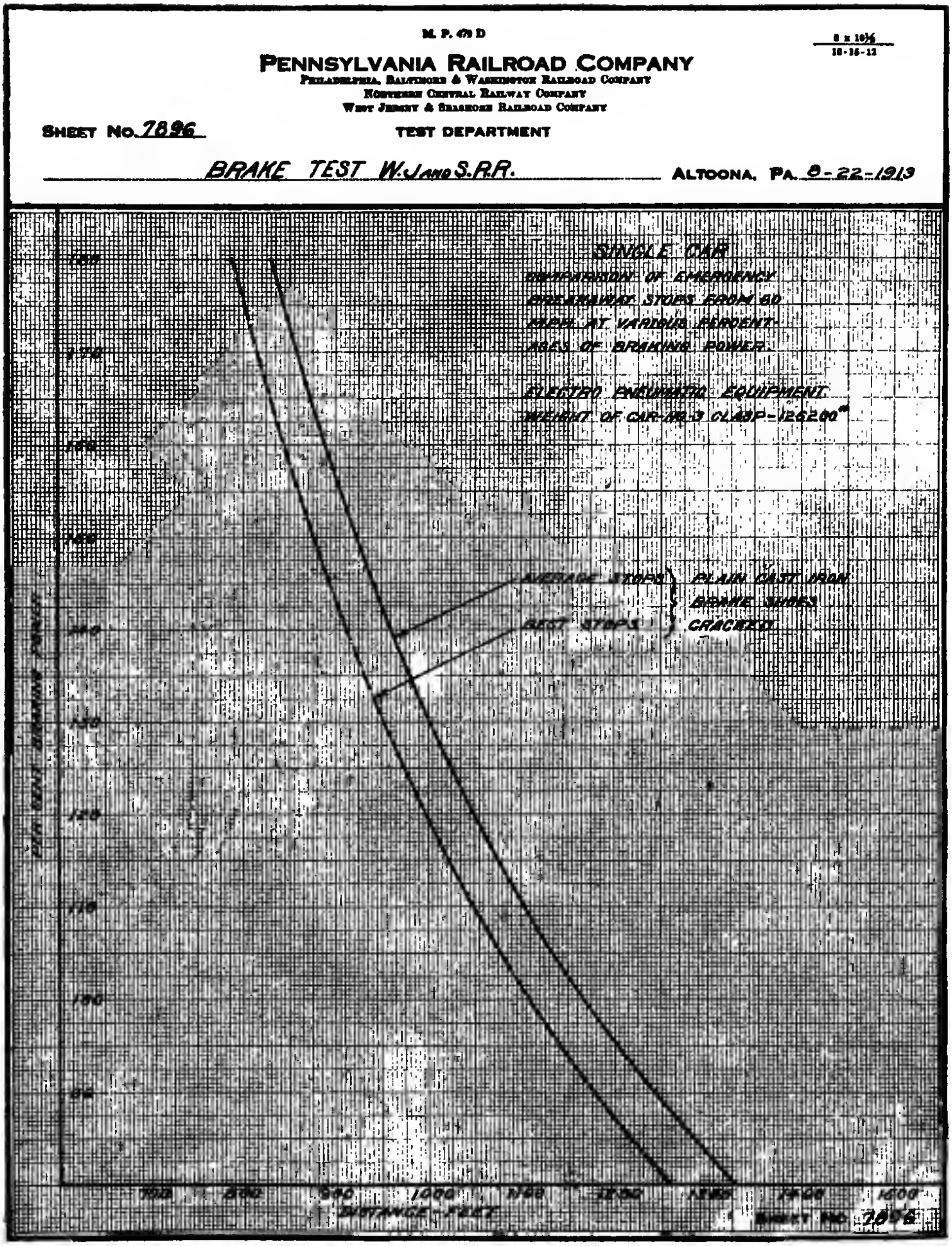


Fig. 54.

LENGTH OF STOP.

Single car, emergency breakaway stops made with the No. 3 clasp brake, from 60 m.p.h. with electro-pneumatic equipment.

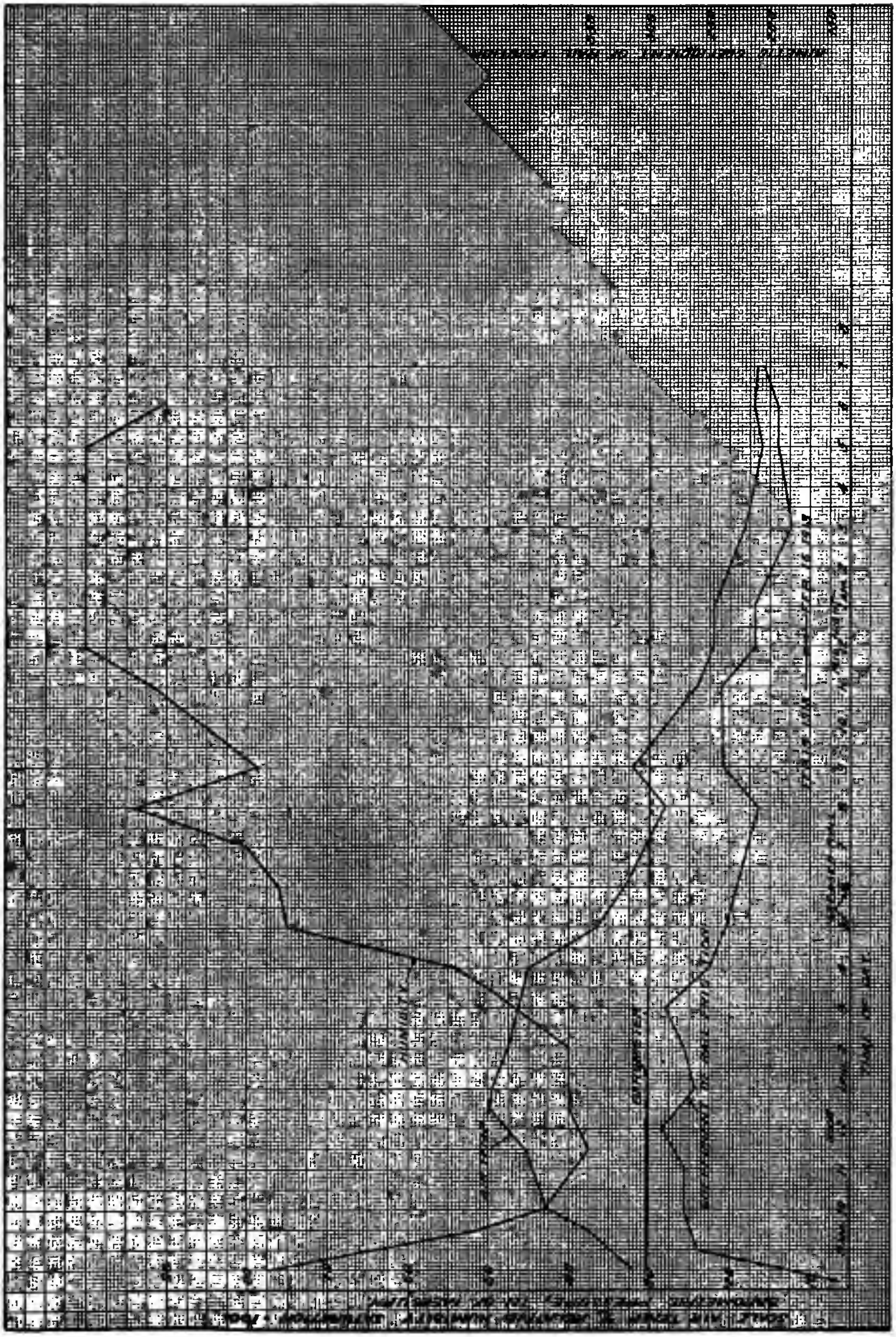


Fig. 55.

COEFFICIENT OF RAIL FRICTION.

A 24-hour record of weather conditions and rail friction.

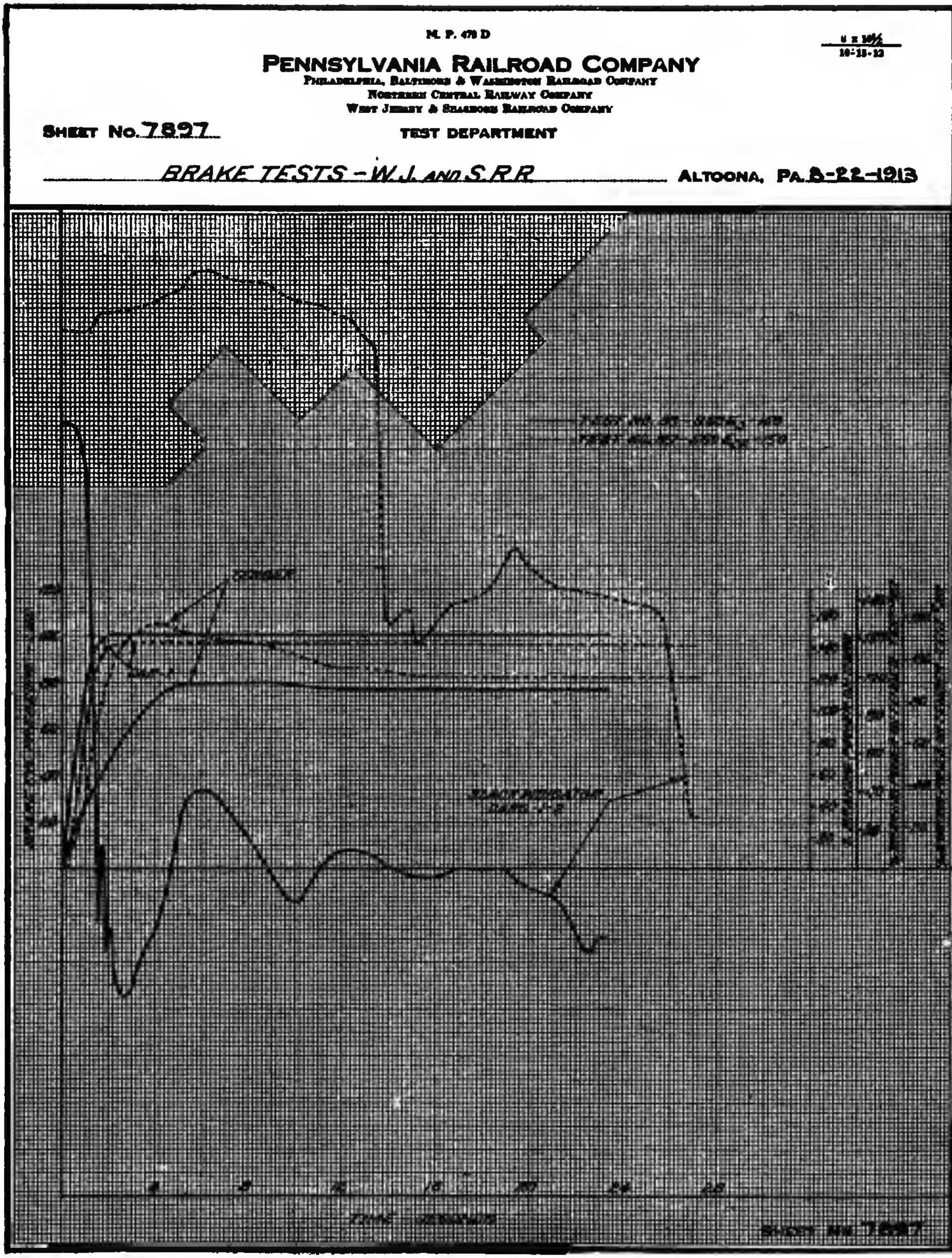


Fig. 56.

HIGH BRAKING POWER ON LOCOMOTIVES.

The slack action between the locomotive and cars is reduced when the braking power of the locomotive is increased.

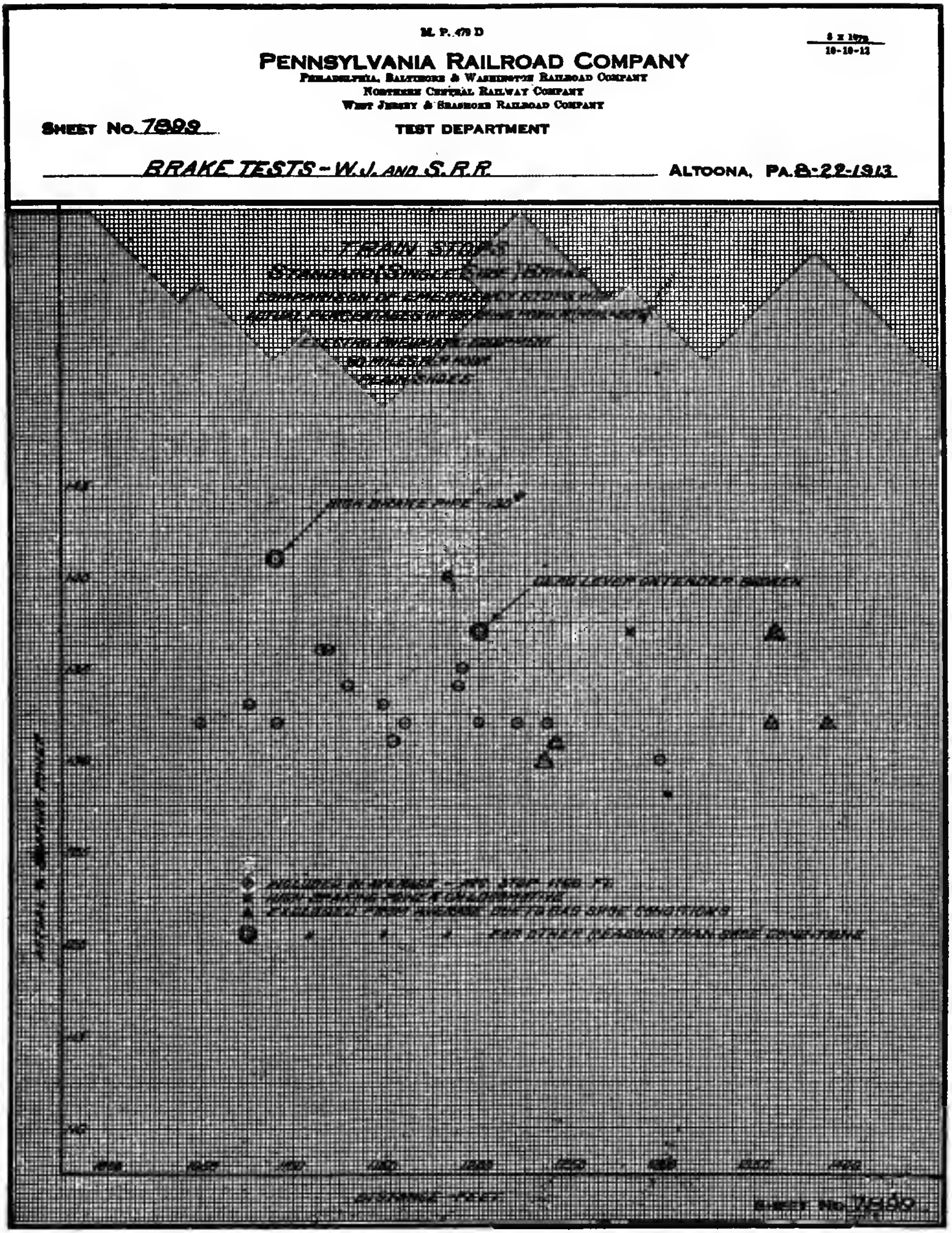


Fig. 57.

CHECK RUNS—TWELVE CARS.

STANDARD SINGLE SHOE BRAKE.

The emergency stopping distances from 60 m.p.h. cover a wide range due principally to variations in shoe condition.

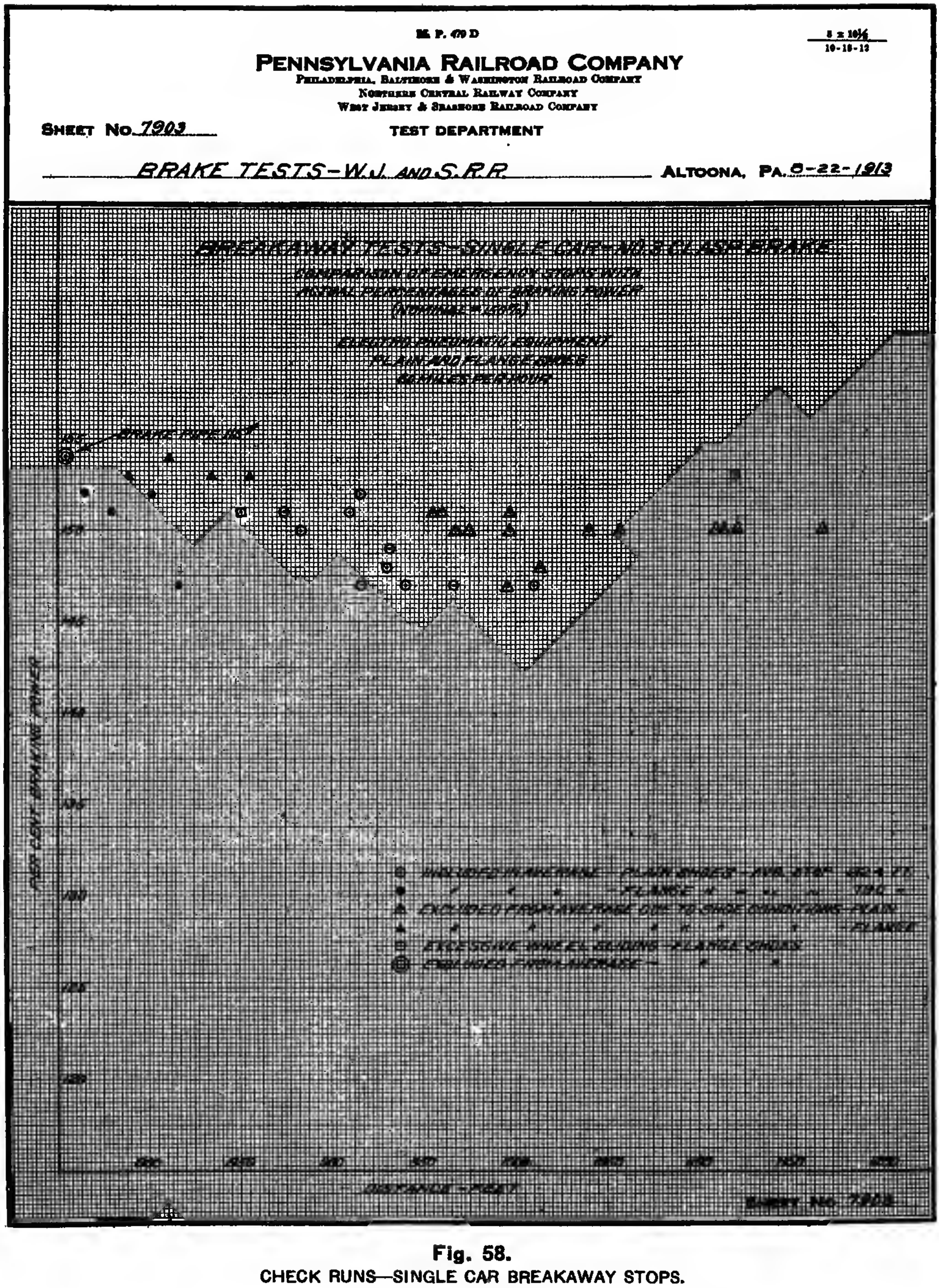


Fig. 58.

CHECK RUNS—SINGLE CAR BREAKAWAY STOPS.

No. 3 CLASP BRAKE.

The average emergency stop from 60 m.p.h. with plain brake shoes was 1014 feet, 140 per cent. actual braking power, electro-pneumatic equipment.

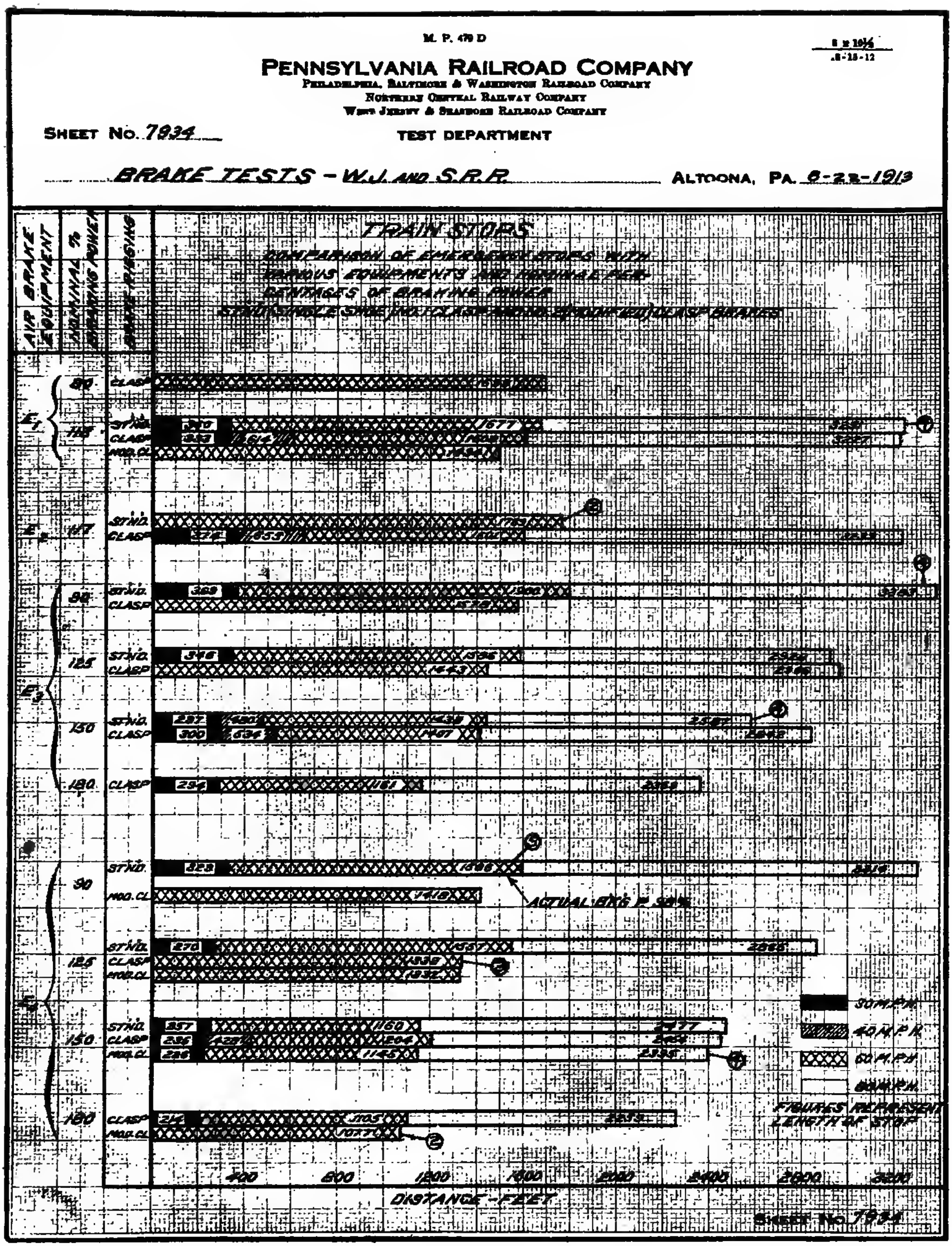


Fig. 59.

EMERGENCY STOPS—VARIOUS EQUIPMENTS.

From 60 m.p.h. the train, fitted with PM equipment and standard brake, stopped in 1677 feet. With electric control and 150 per cent. nominal braking power on the cars, the train was stopped in less than 1200 feet.

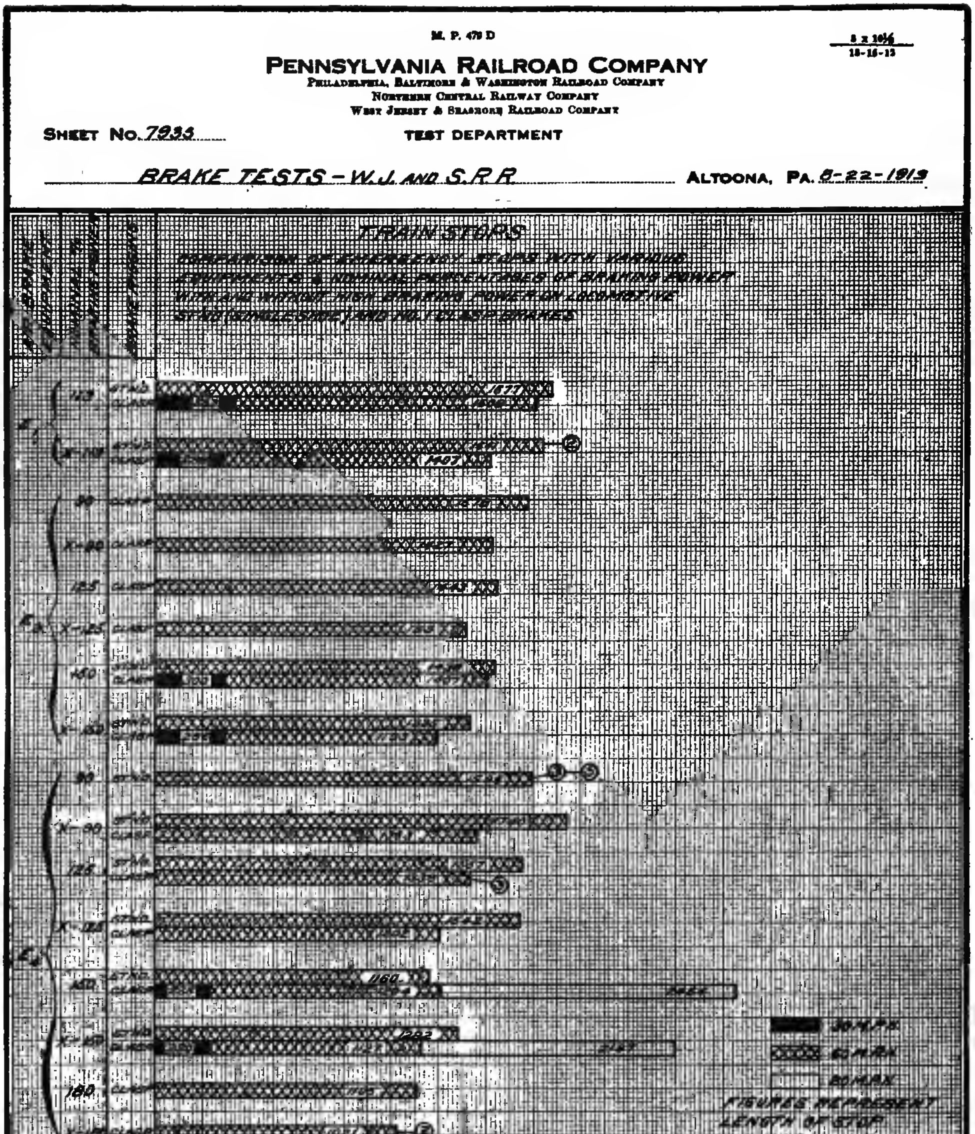


Fig. 60.

EMERGENCY STOPS.

HIGH AND ORDINARY BRAKING POWER ON LOCOMOTIVE.

With high braking power on the locomotive during the early portion of the application the stop shortened. Less slack action between locomotive and cars was also obtained.

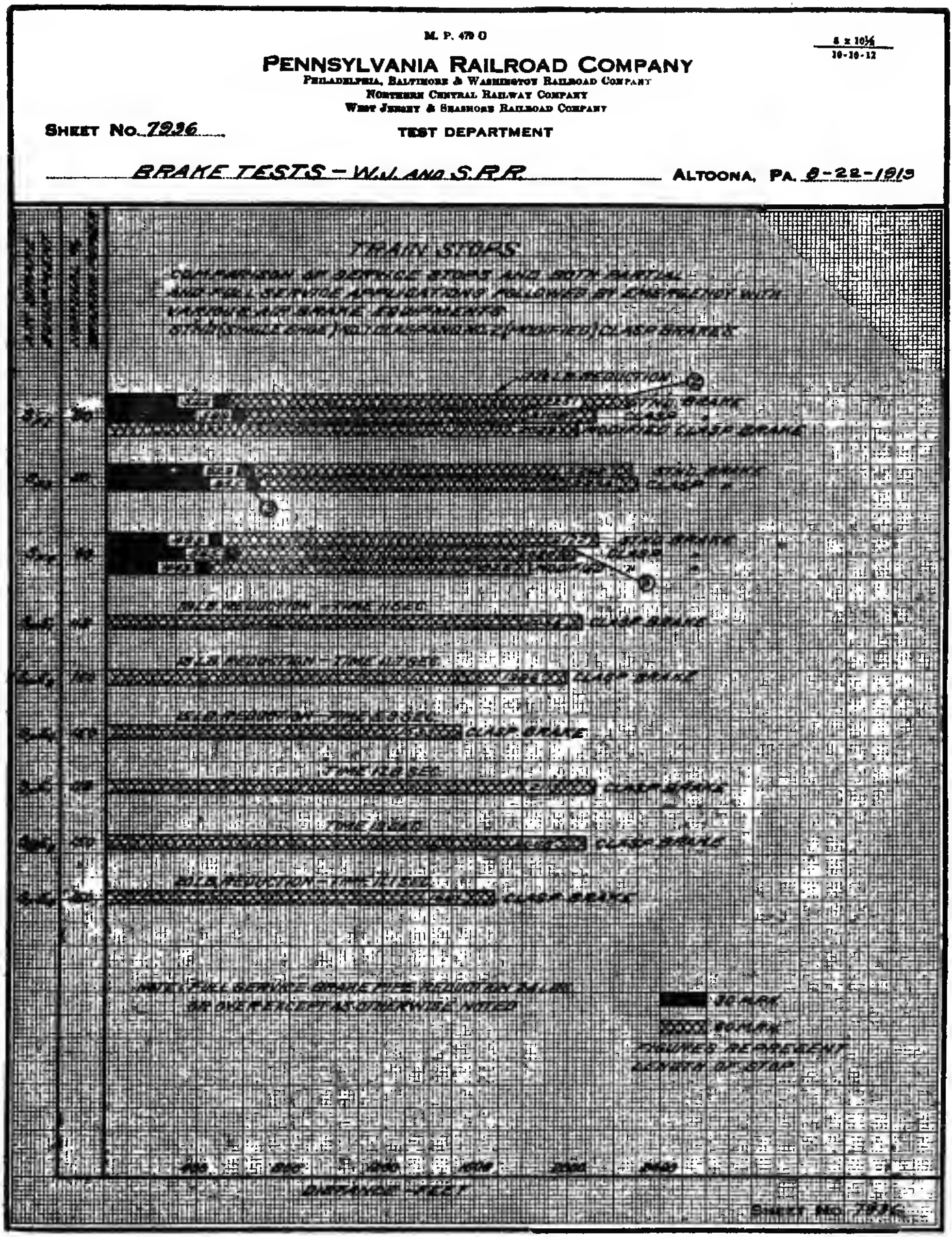


Fig. 61.

SERVICE STOPS—VARIOUS EQUIPMENTS.

A full service application at 60 m.p.h. with the PM equipment and standard brake, stops the train in 2250 feet. With electric control and clasp brake the stop is made in 1825 feet. With electric control, partial service, followed by an emergency application, stops the train in 1533 feet.

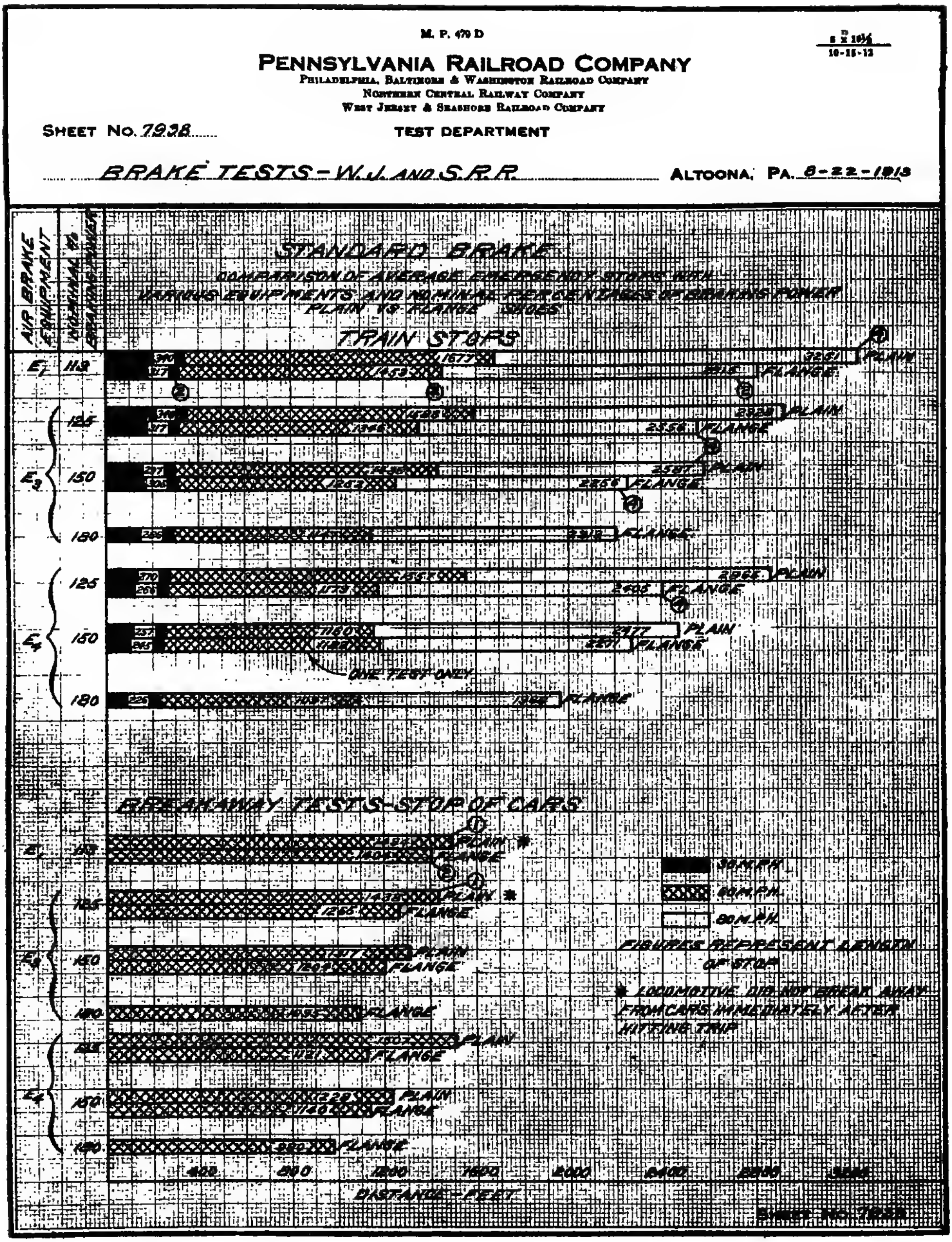


Fig. 62.

EMERGENCY STOPS—STANDARD BRAKE.

With the standard brake and PM equipment the stop at 60 m.p.h. is shortened 224 ft. by the use of flanged shoes.

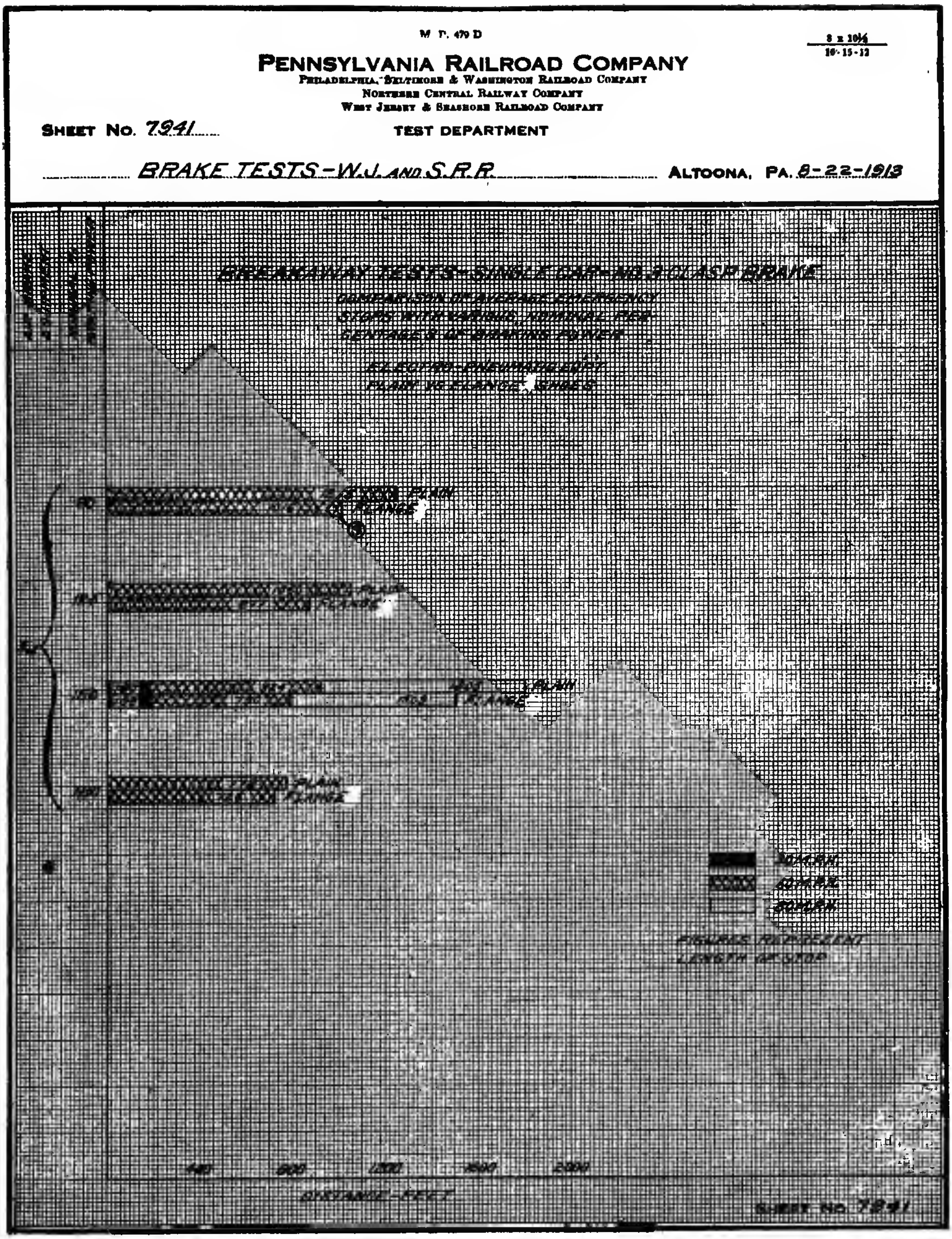


Fig. 63.

EMERGENCY STOPS—SINGLE CAR BREAKAWAYS.

The car was stopped in 790 feet from 60 m.p.h. at a nominal braking power of 150 per cent., electro-pneumatic equipment and flange shoes.

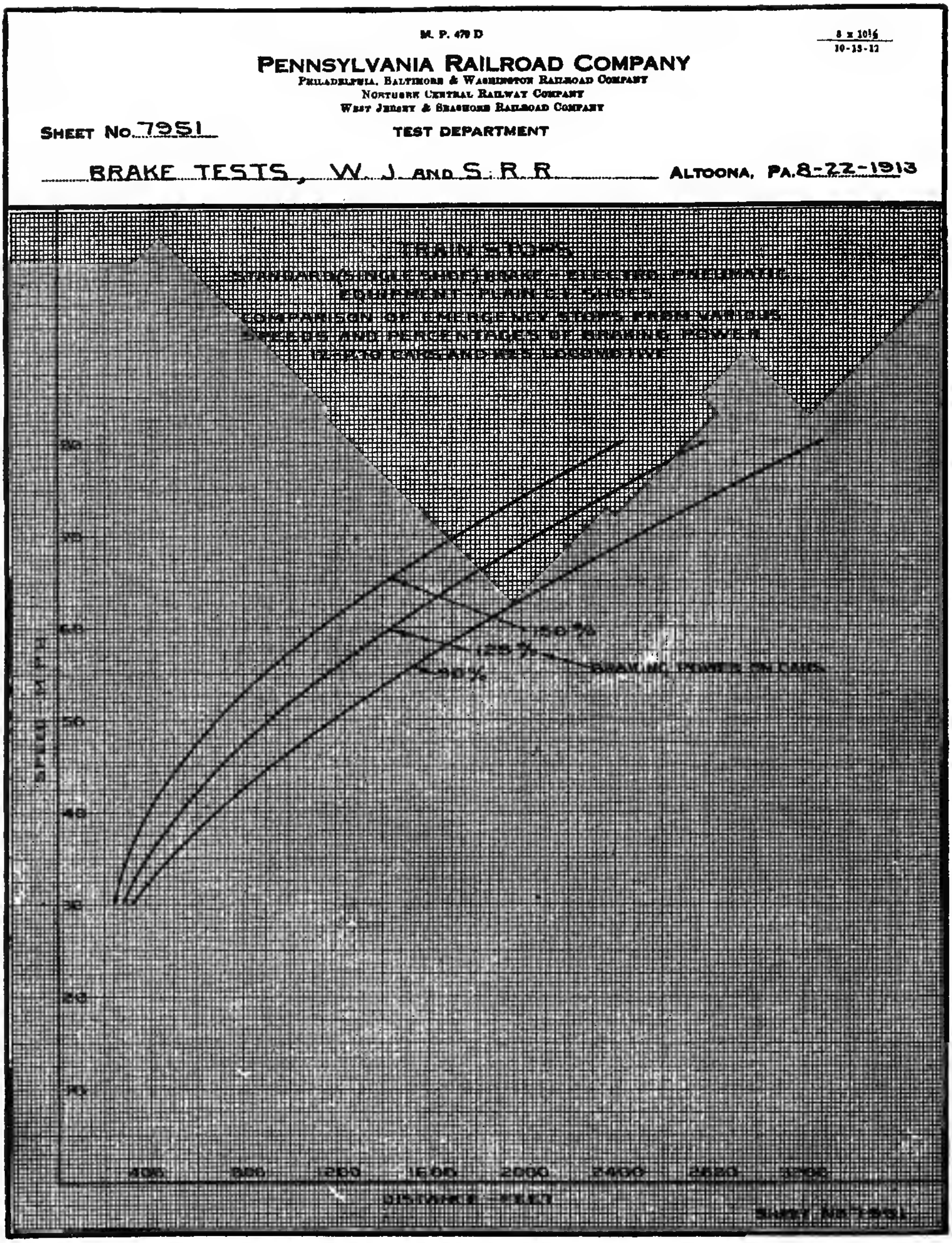


Fig. 64.

SPEED—STOP DISTANCE.

The diagram shows the average emergency stops from speeds of 30 to 80 m.p.h. which the standard brake rigging will give with the electro-pneumatic equipment.

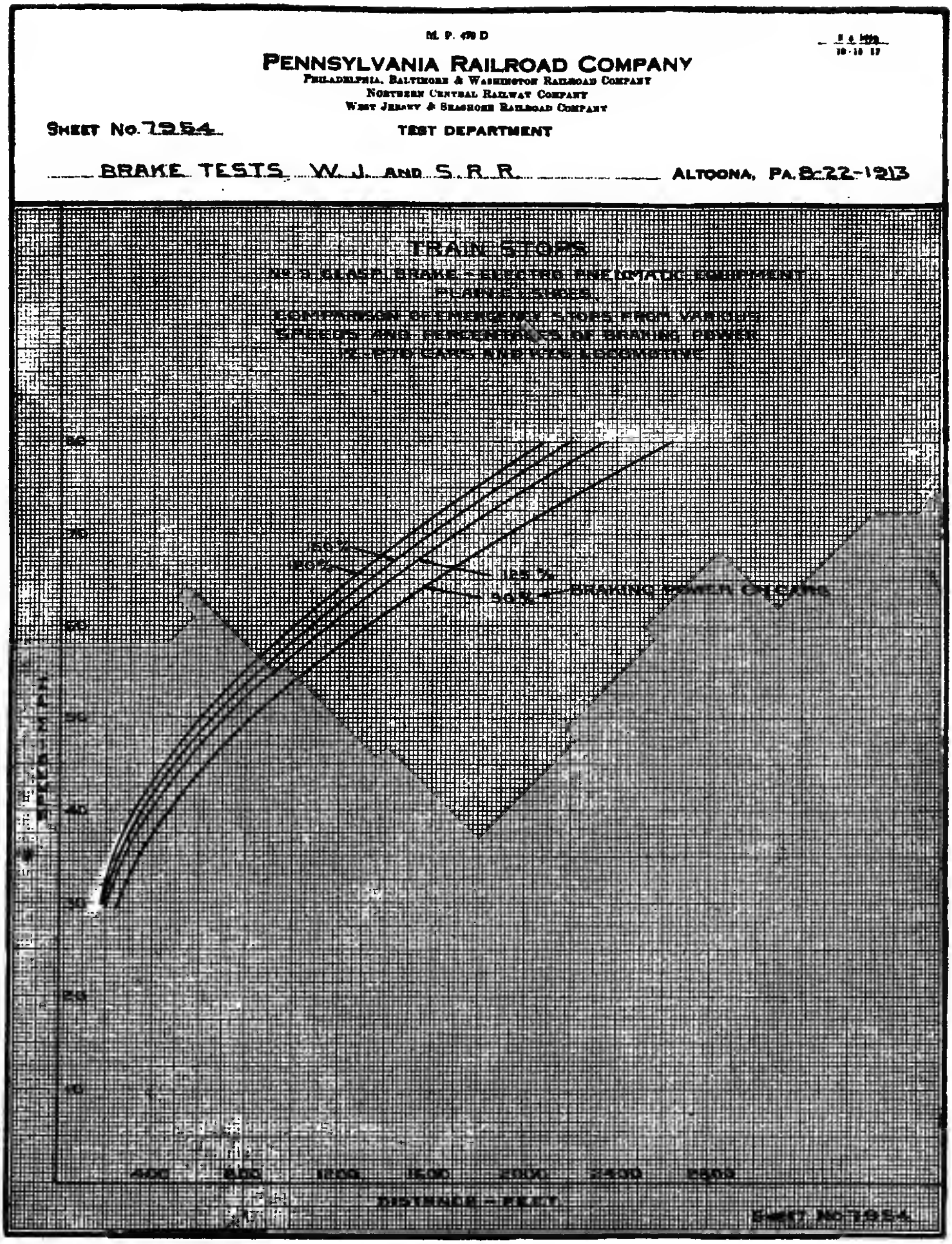


Fig. 65.

SPEED—STOP DISTANCE.

The calculated average stops which would be obtained with a train made up of 12 cars fitted with the No. 3 clasp brake and electro-pneumatic equipment.

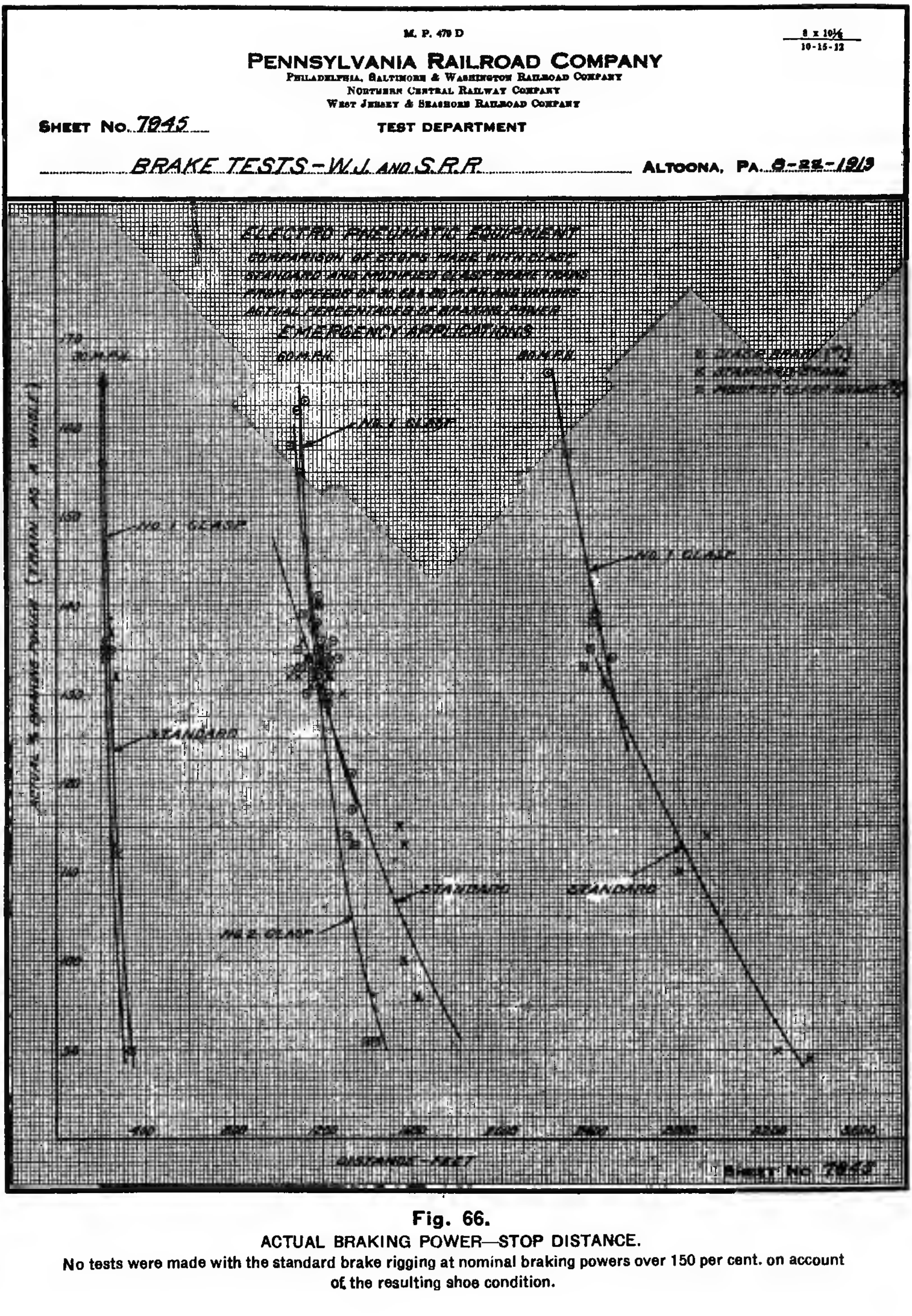


Fig. 66.

ACTUAL BRAKING POWER—STOP DISTANCE.

No tests were made with the standard brake rigging at nominal braking powers over 150 per cent. on account of the resulting shoe condition.

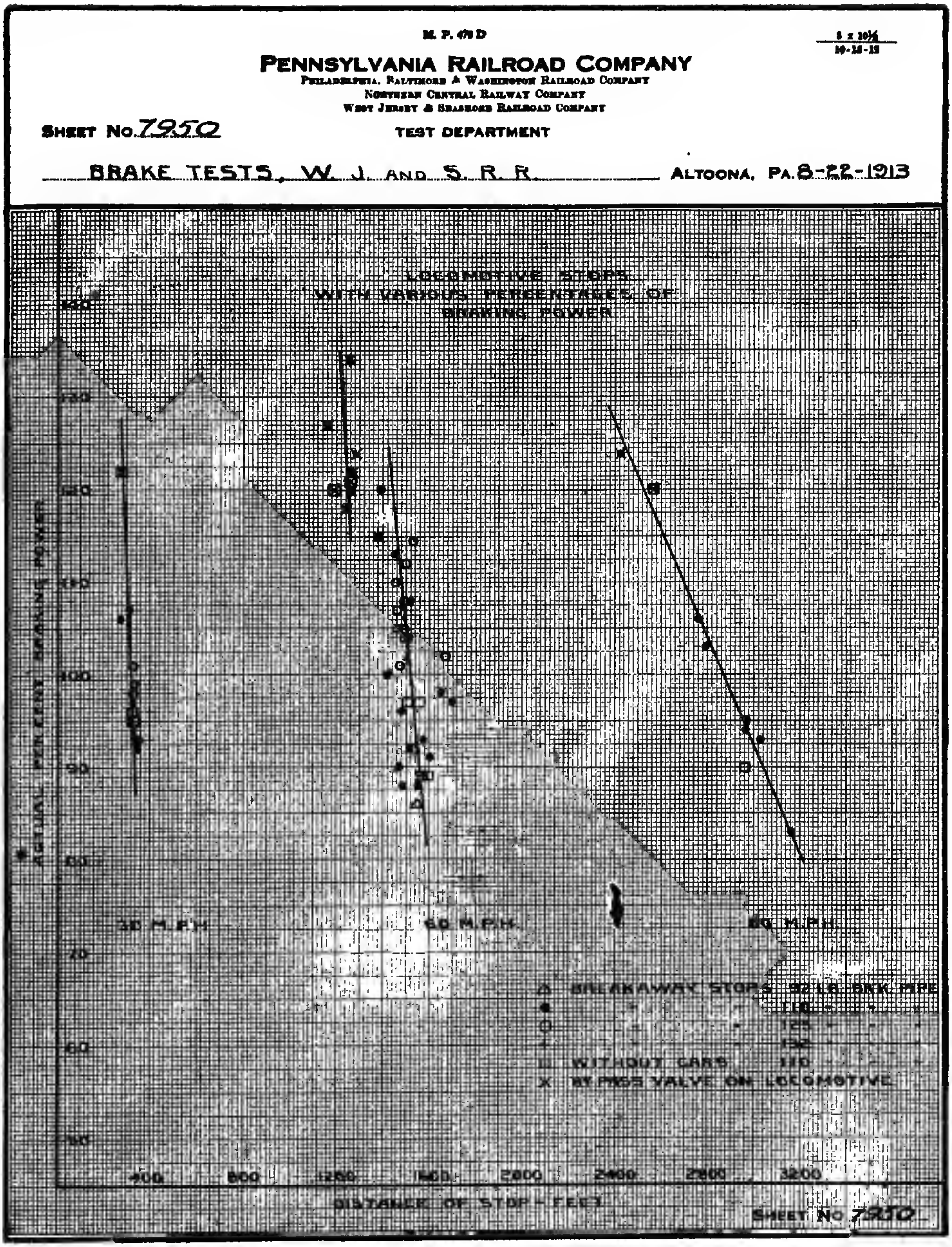


Fig. 67.

ACTUAL BRAKING POWER—STOP DISTANCE.

The advantage of a high braked locomotive is apparent.

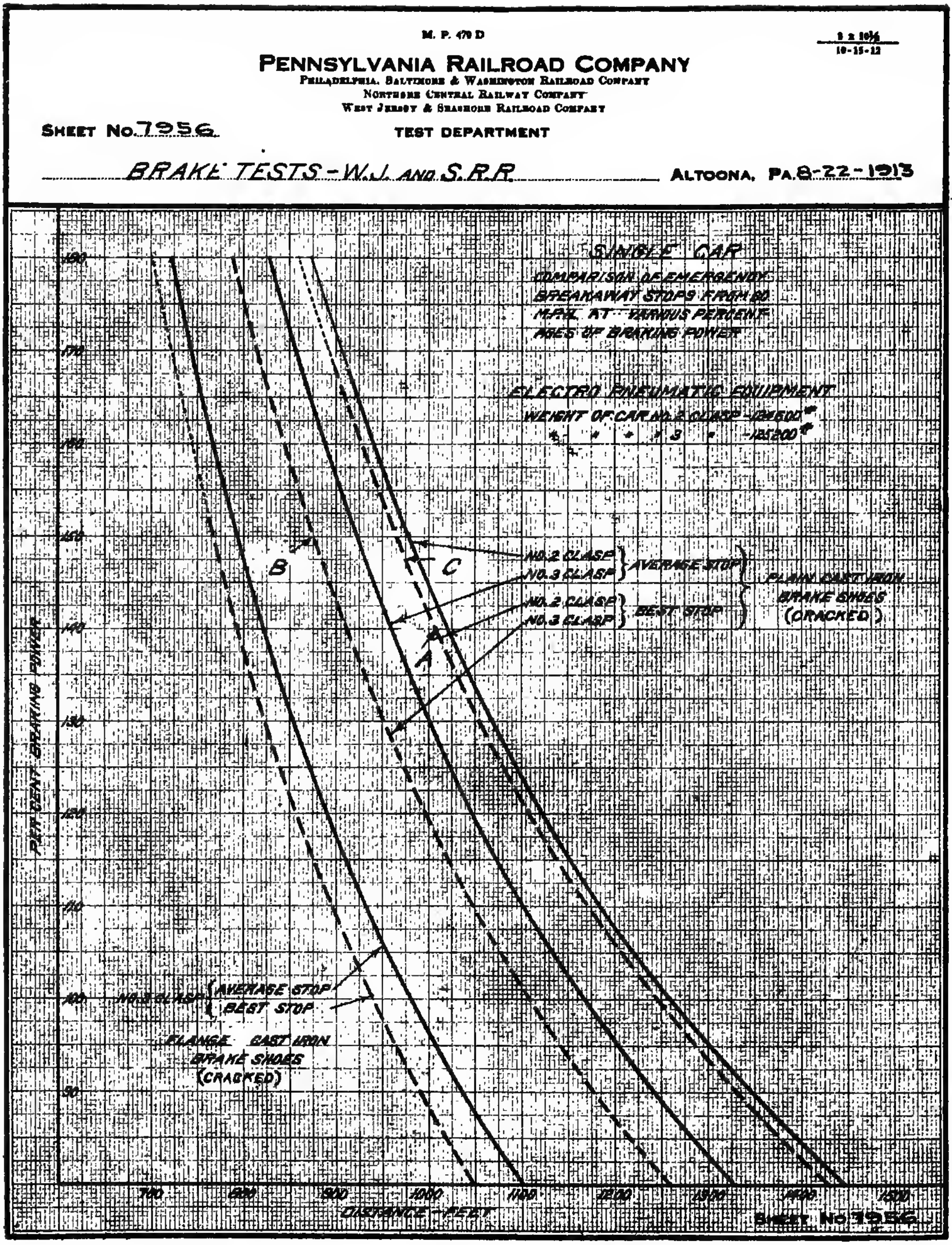


Fig. 68.

STOP DISTANCE—SINGLE CARS.

No. 2 AND No. 3 CLASP BRAKE.

The variation between the best and the average stops is due largely to shoe condition.

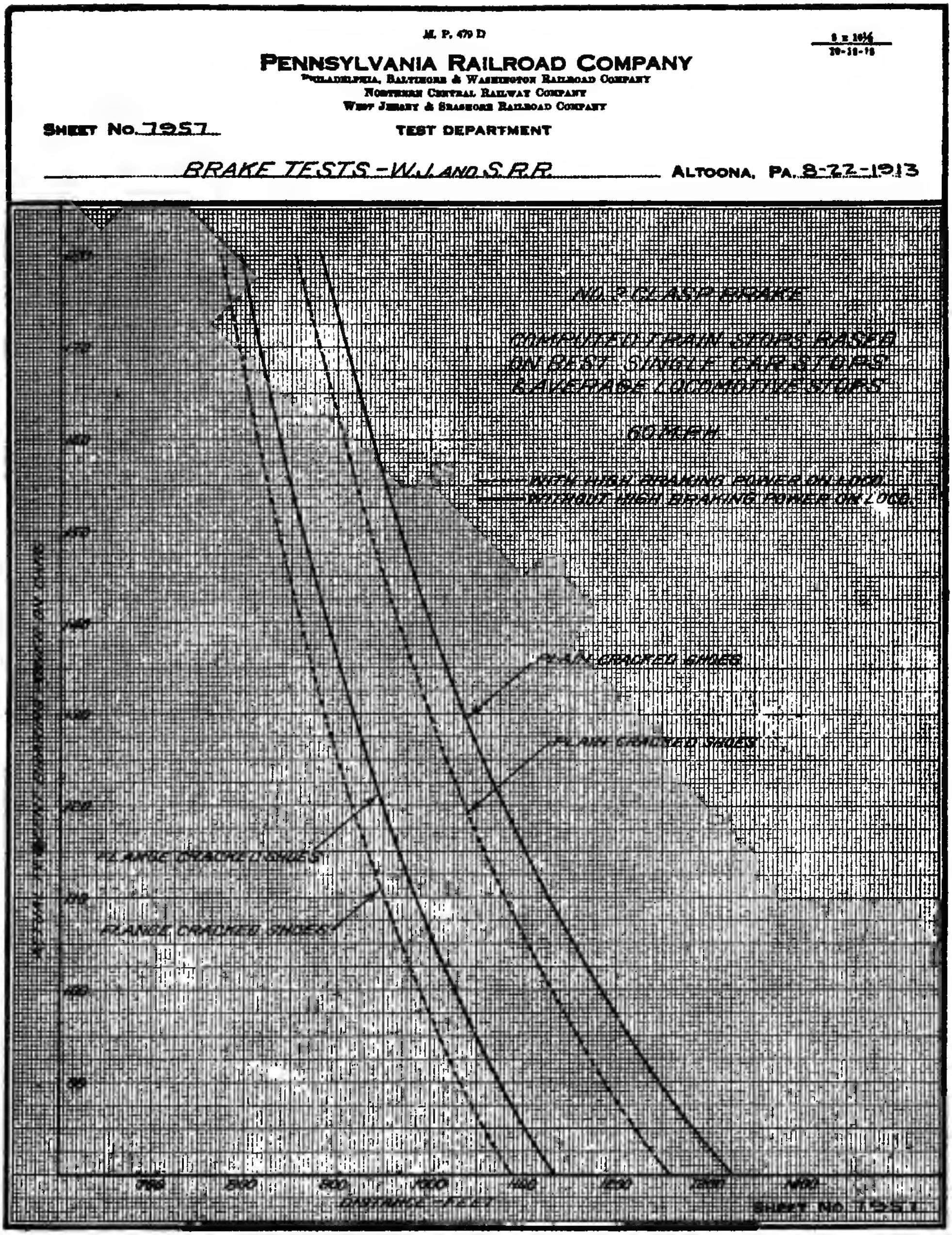


Fig. 69.

COMPUTED BEST TRAIN STOPS.

No. 3 CLASP BRAKE.

The gain in stopping distance with high braking power on the locomotive during the early portion of the stop (Figs. 56) is well illustrated.

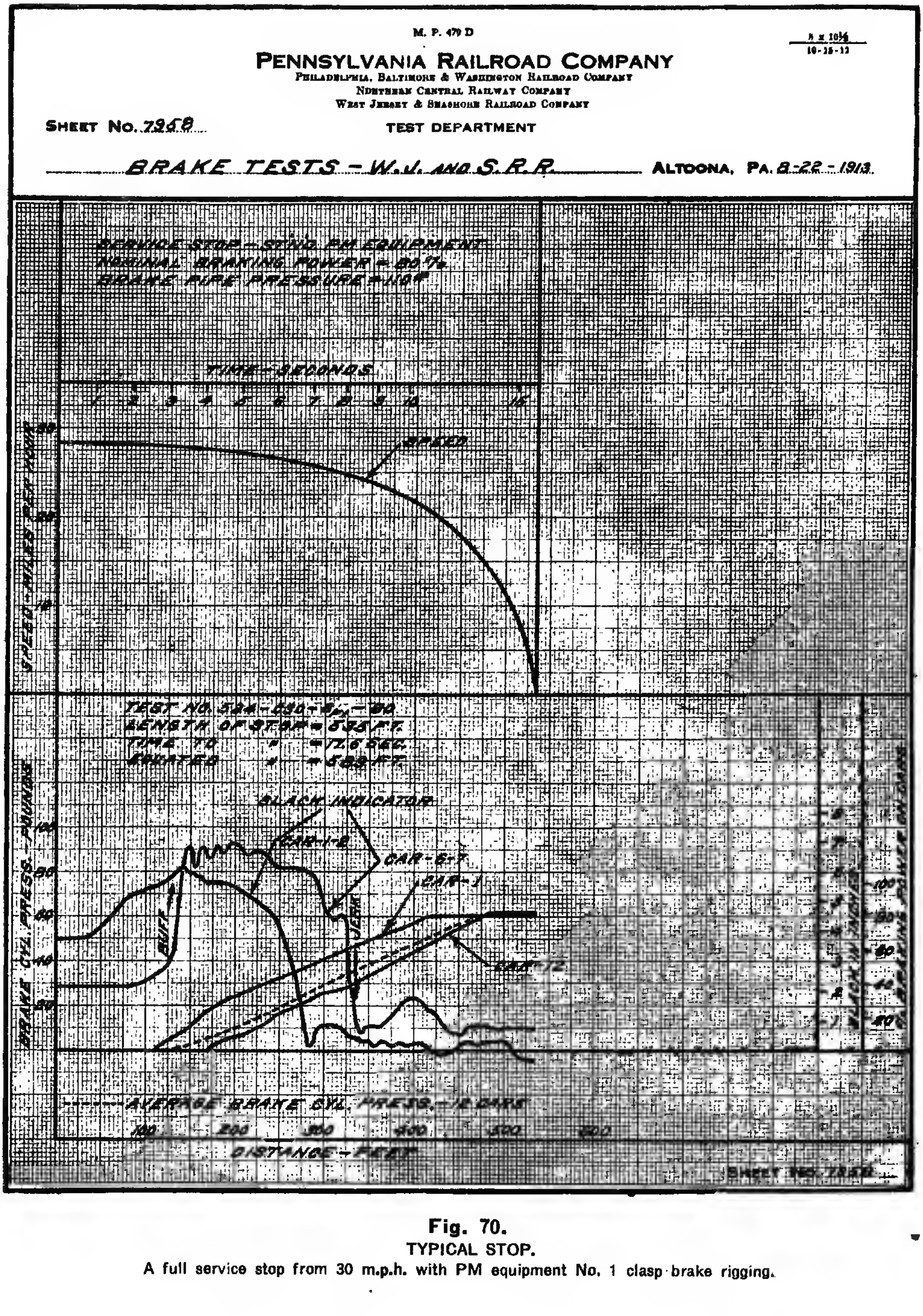


Fig. 70.

TYPICAL STOP.

A full service stop from 30 m.p.h. with PM equipment No. 1 clasp brake rigging.

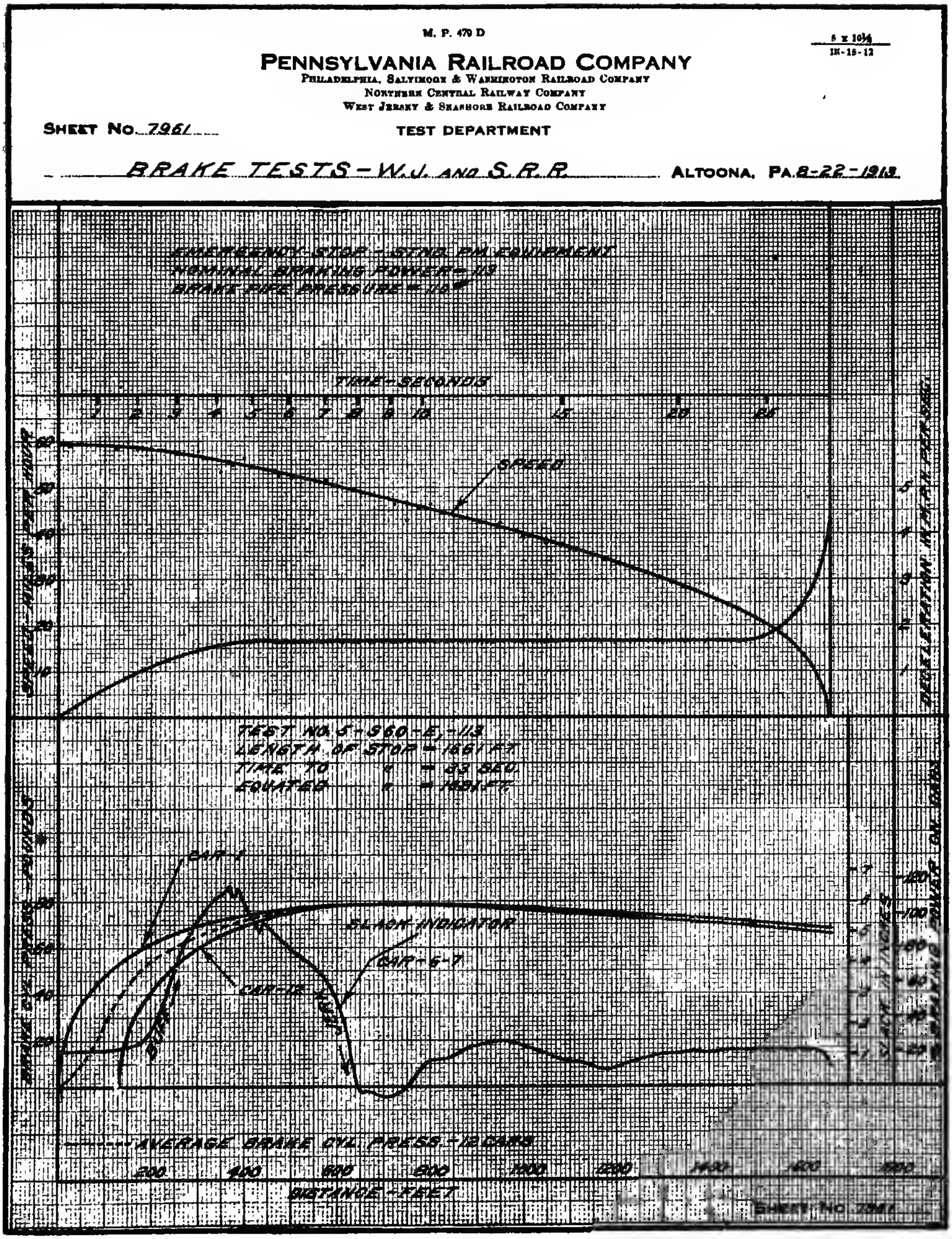


Fig. 71.

TYPICAL STOP.

An emergency stop from 60 m.p.h. with standard single shoe brake rigging.

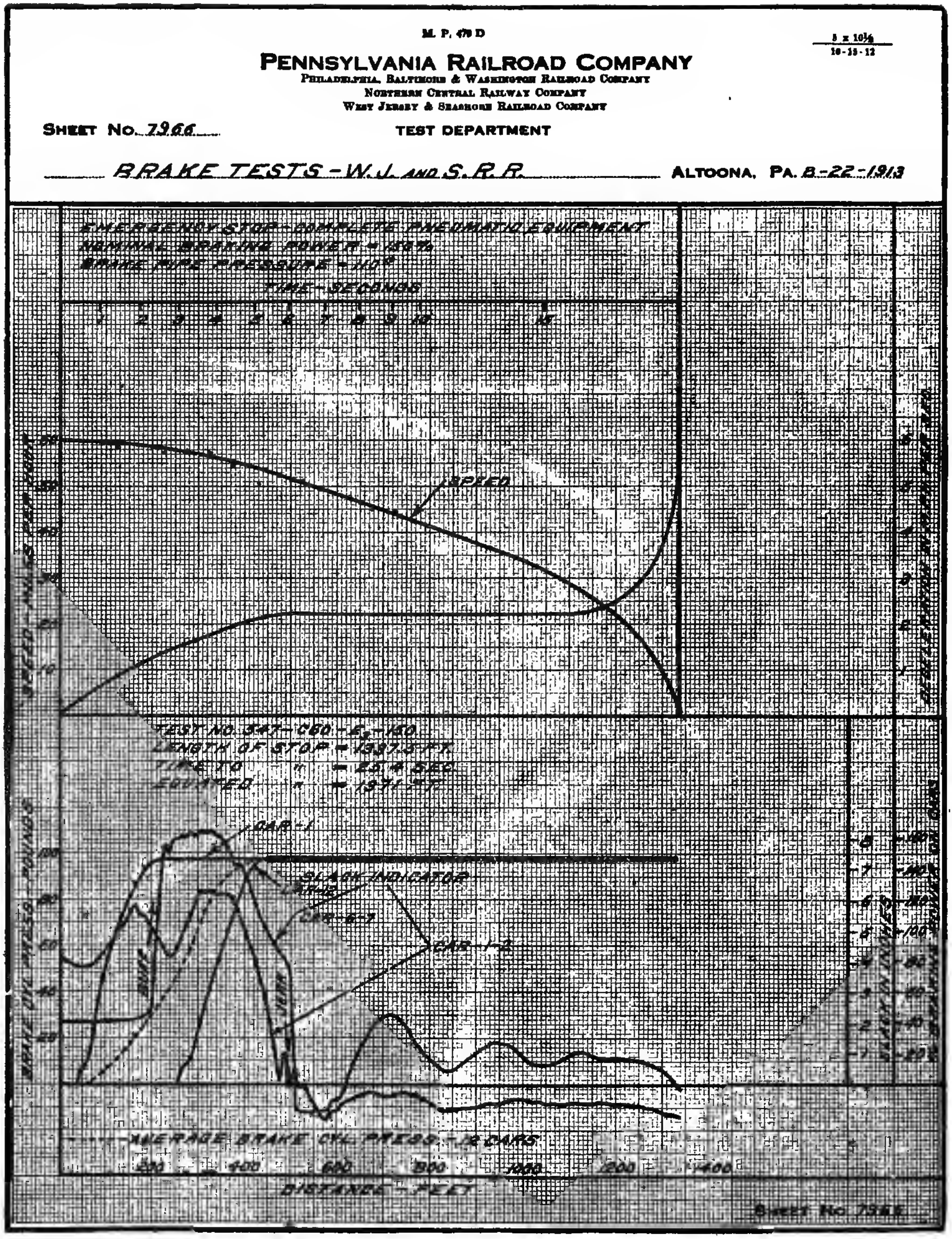


Fig. 72.

TYPICAL STOP.

An emergency stop from 60 m.p.h. with the UC pneumatic equipment No. 1 clasp brake rigging.

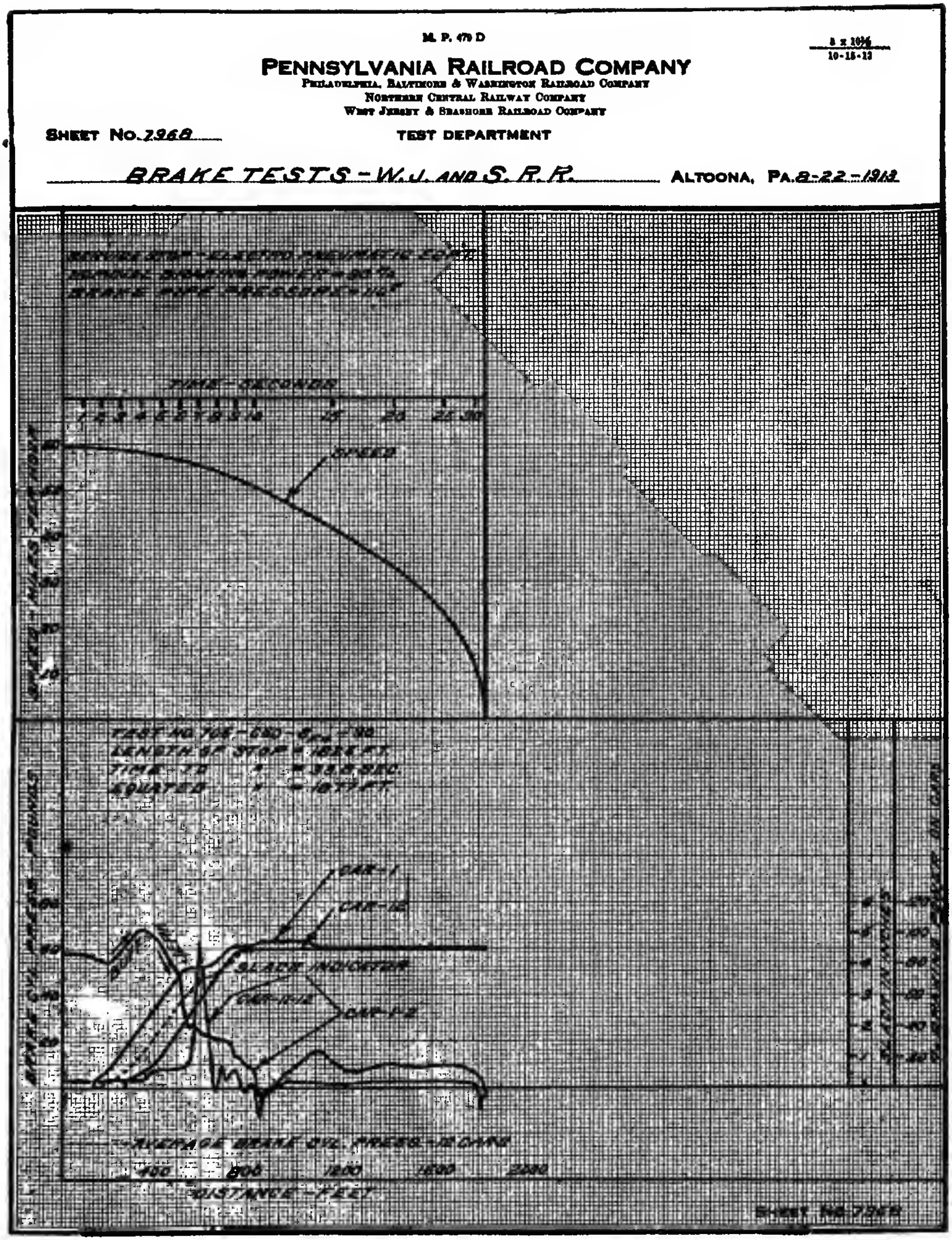


Fig. 73.

TYPICAL STOP.

This stop of 1877 feet from 60 m.p.h. was made by a full service application electro-pneumatic equipment No. 1 clasp brake rigging.

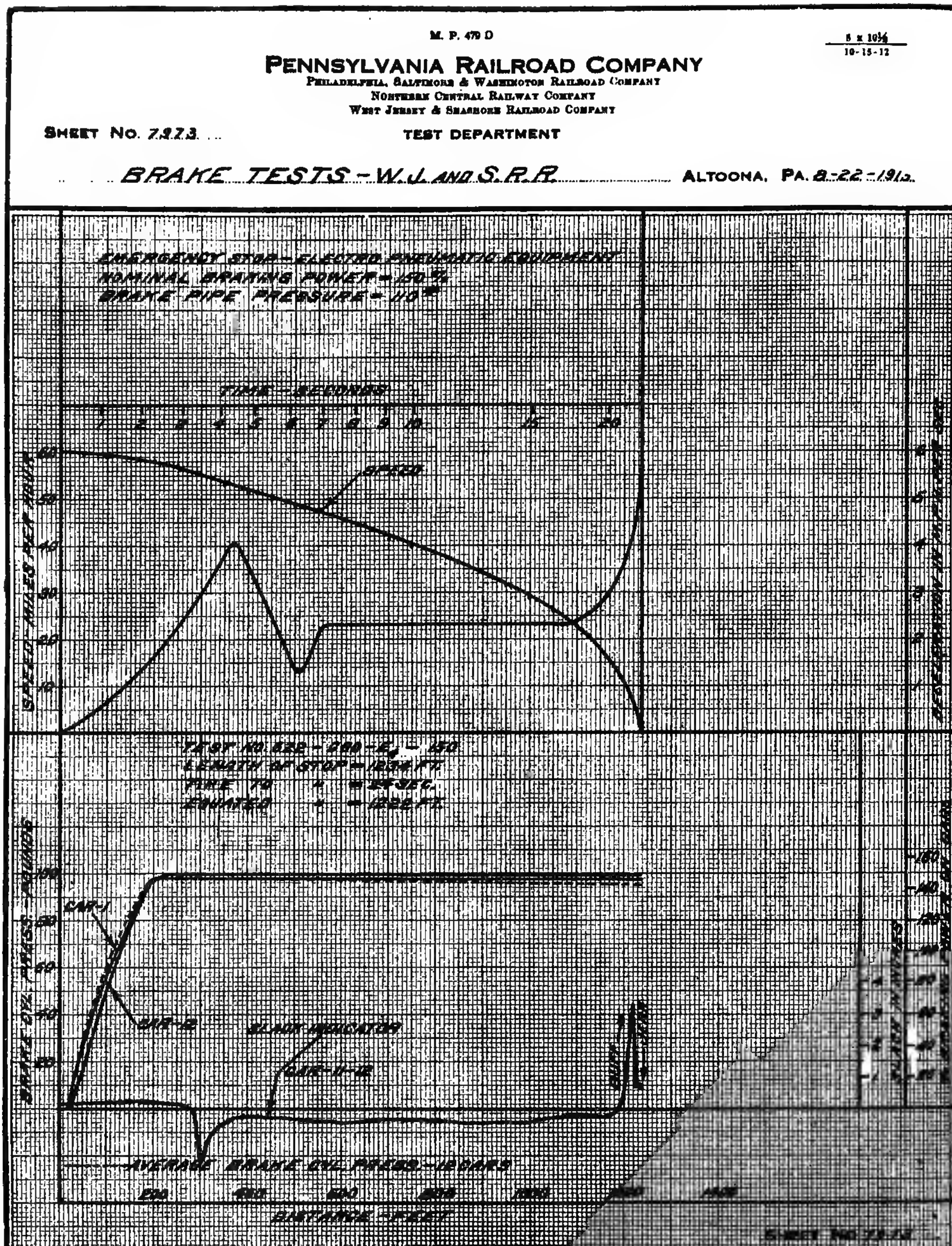


Fig. 74.

TYPICAL STOP.

A 60 m.p.h. emergency stop electro-pneumatic equipment No. 1 clasp brake rigging. The nominal braking power of the cars was 150 per cent.

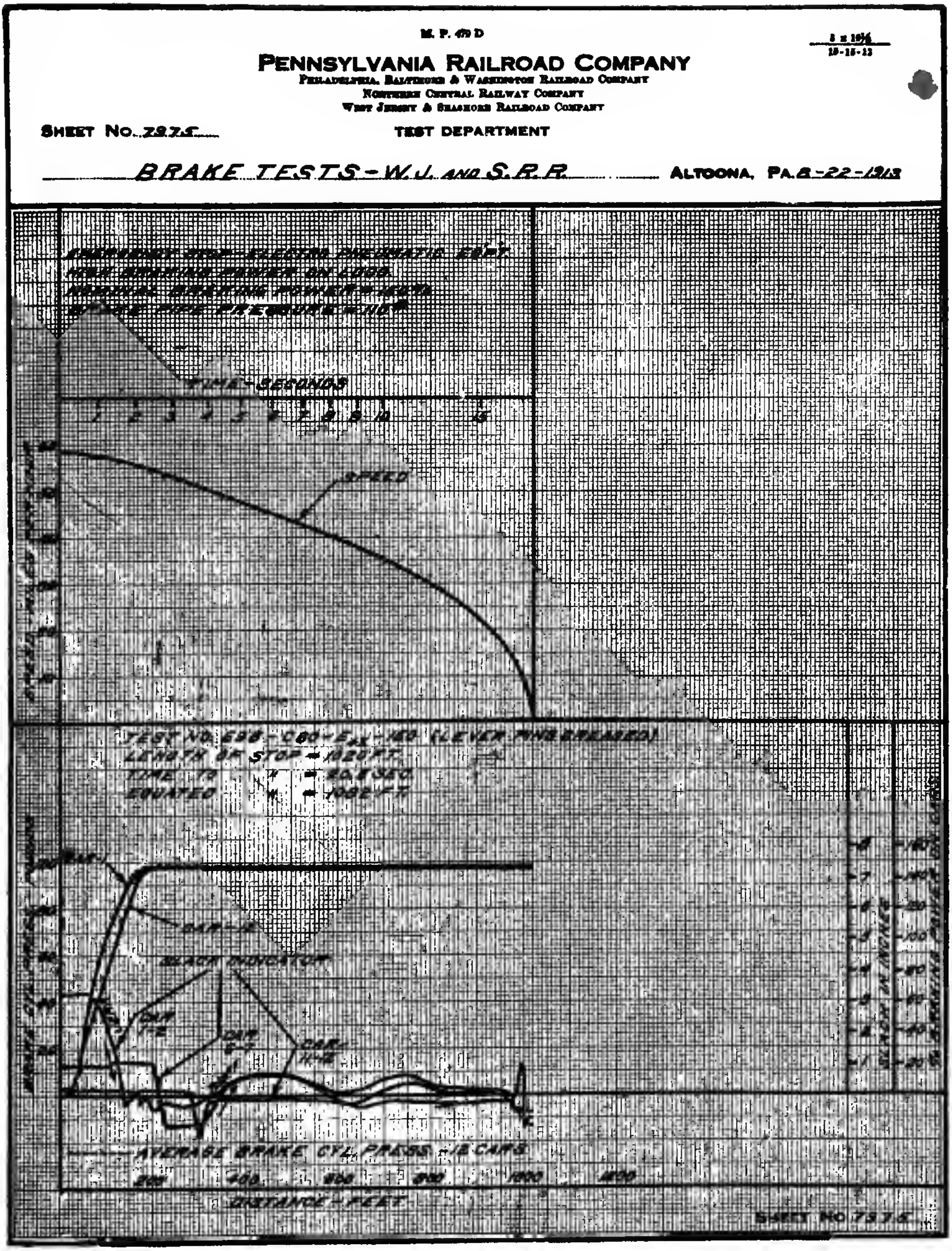


Fig. 75.

TYPICAL STOP.

High braking power on the locomotive has partially eliminated the slack action between the locomotive and the cars as shown by the nearly horizontal lines drawn by the slack indicators.

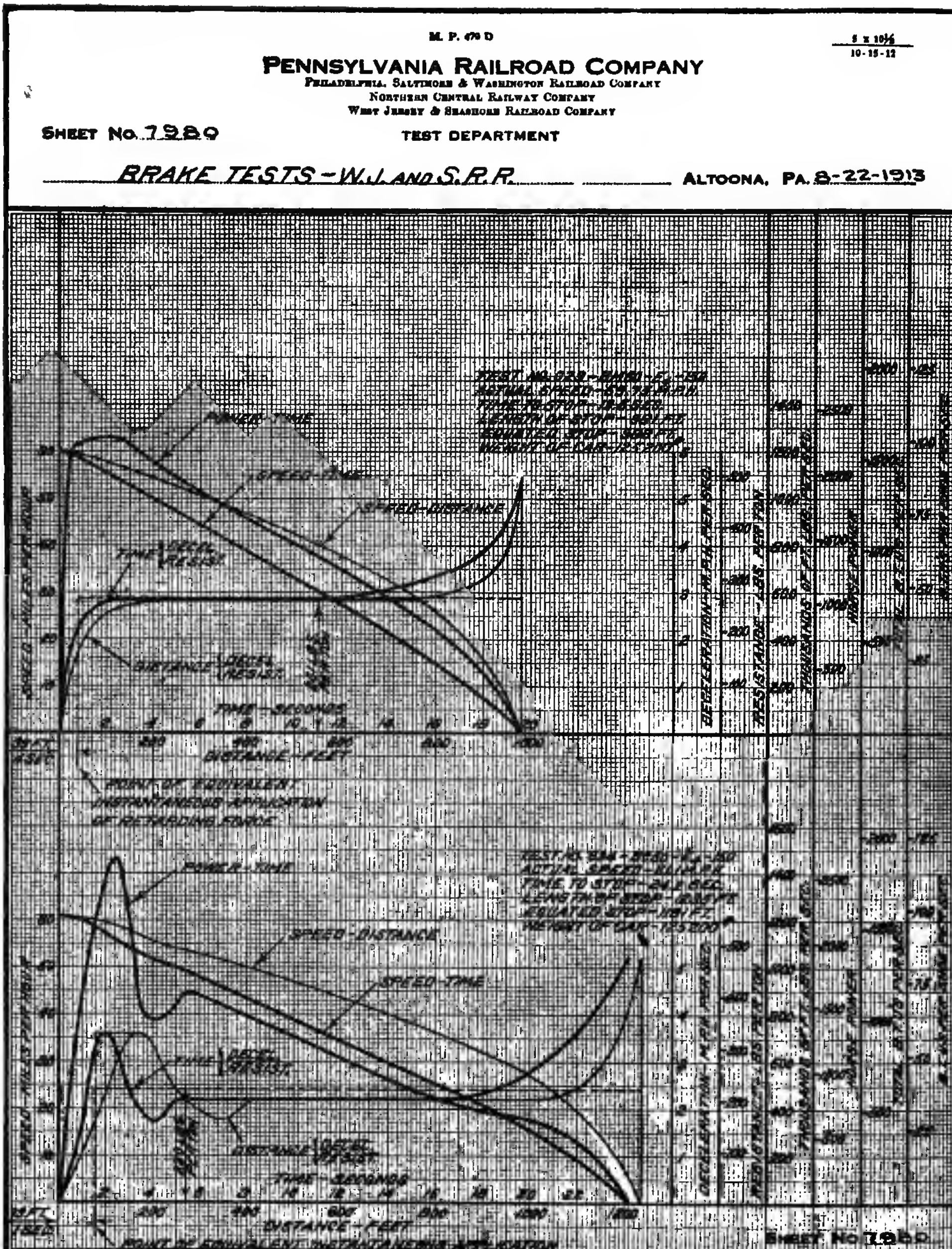


Fig. 76.

SINGLE CAR BREAKAWAY STOPS.

CHARACTERISTIC CURVE.

Electro-pneumatic emergency stops made with the No. 1 and No. 2 clasp brake rigging.

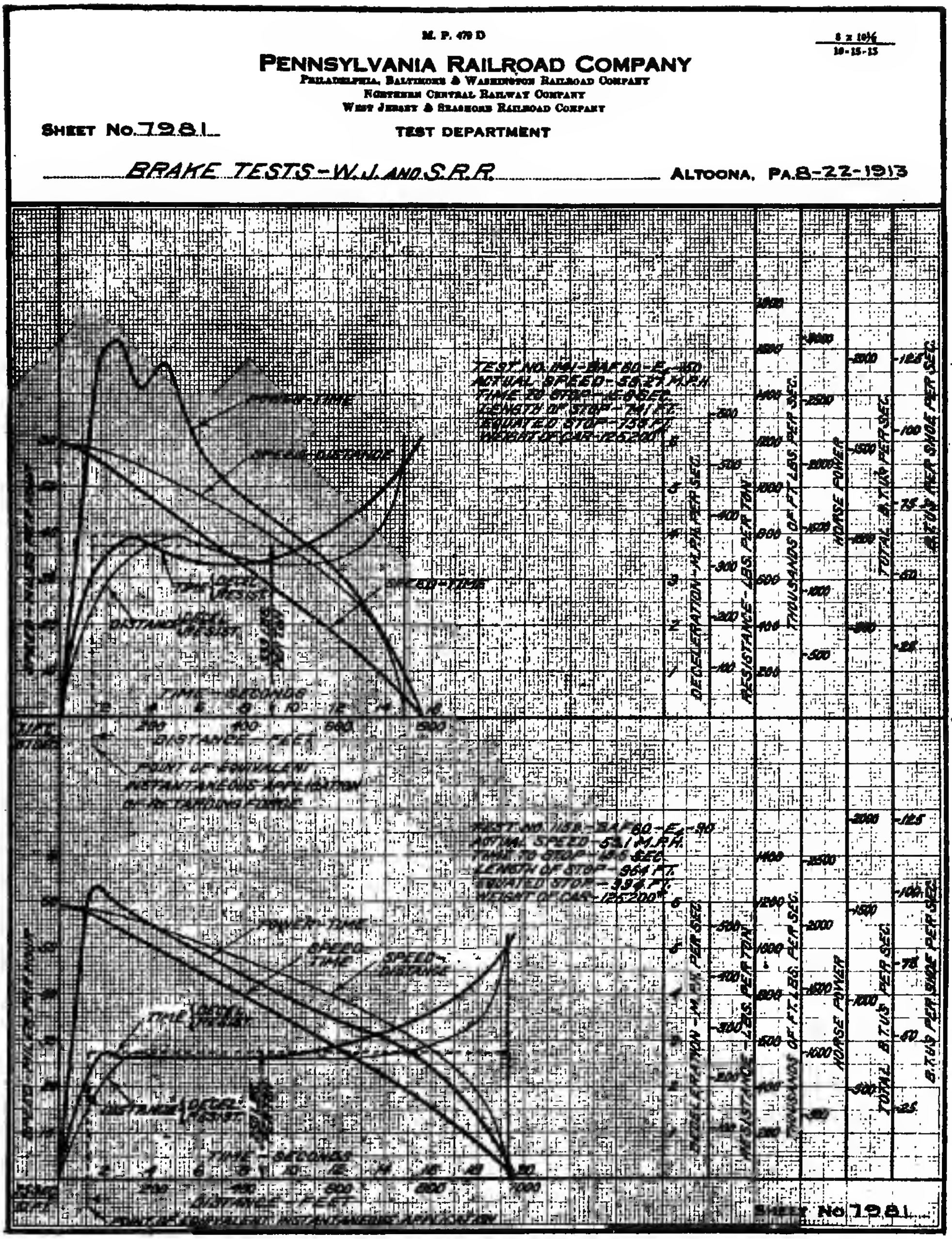


Fig. 77

SINGLE CAR BREAKAWAY STOPS.
CHARACTERISTIC CURVE.

Electro-pneumatic tests, similar to those shown in Fig. 76, except that in this case the car was fitted with flanged shoes and No. 3 clasp brake rigging.

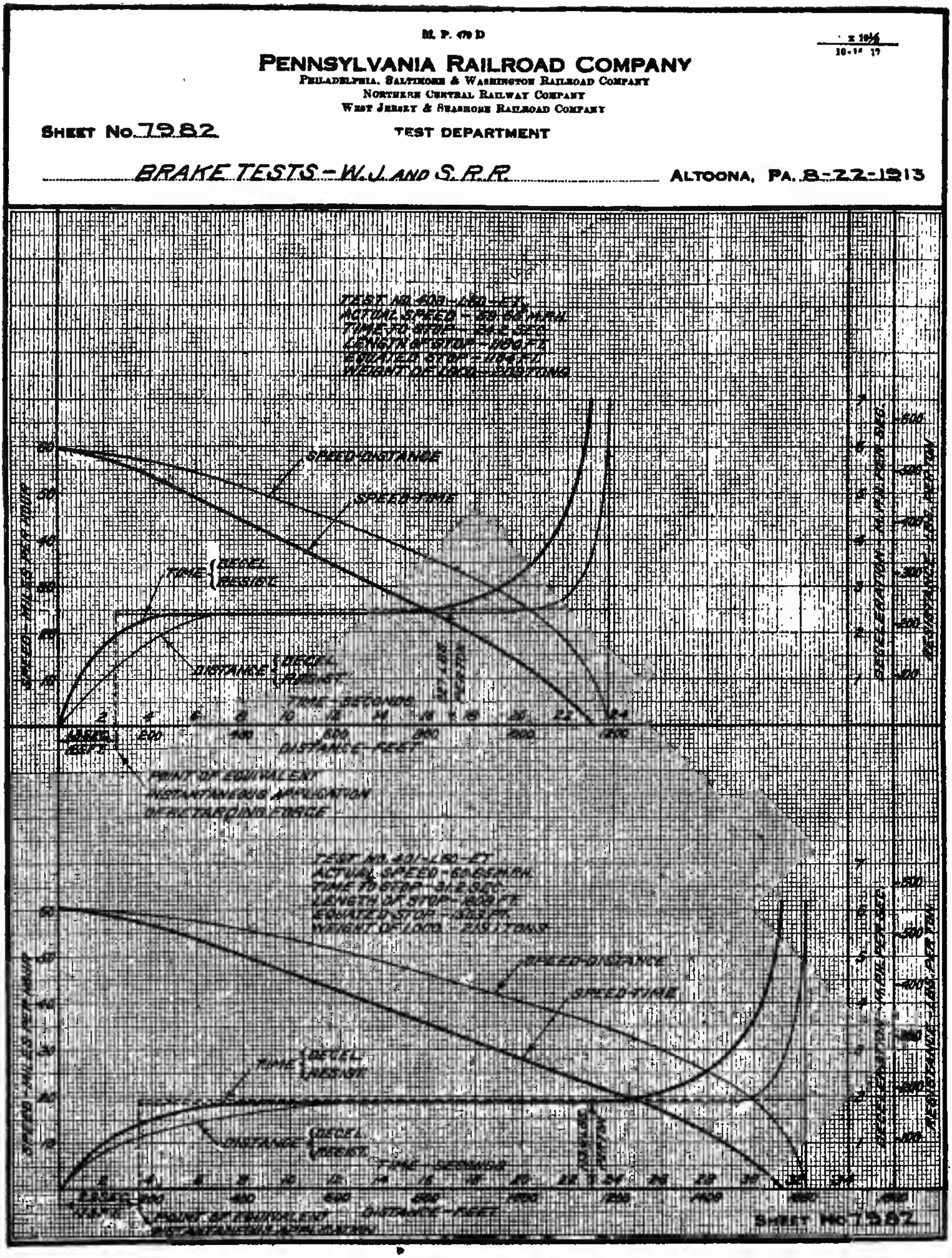


Fig. 78.

LOCOMOTIVE STOP.

CHARACTERISTIC CURVE.

When high braking power was used on the locomotive the stop from 60 m.p.h. was made in 1194 feet. The lower portion of the diagram shows a test with ordinary braking power on the locomotive. The stop was made in 1563 feet.

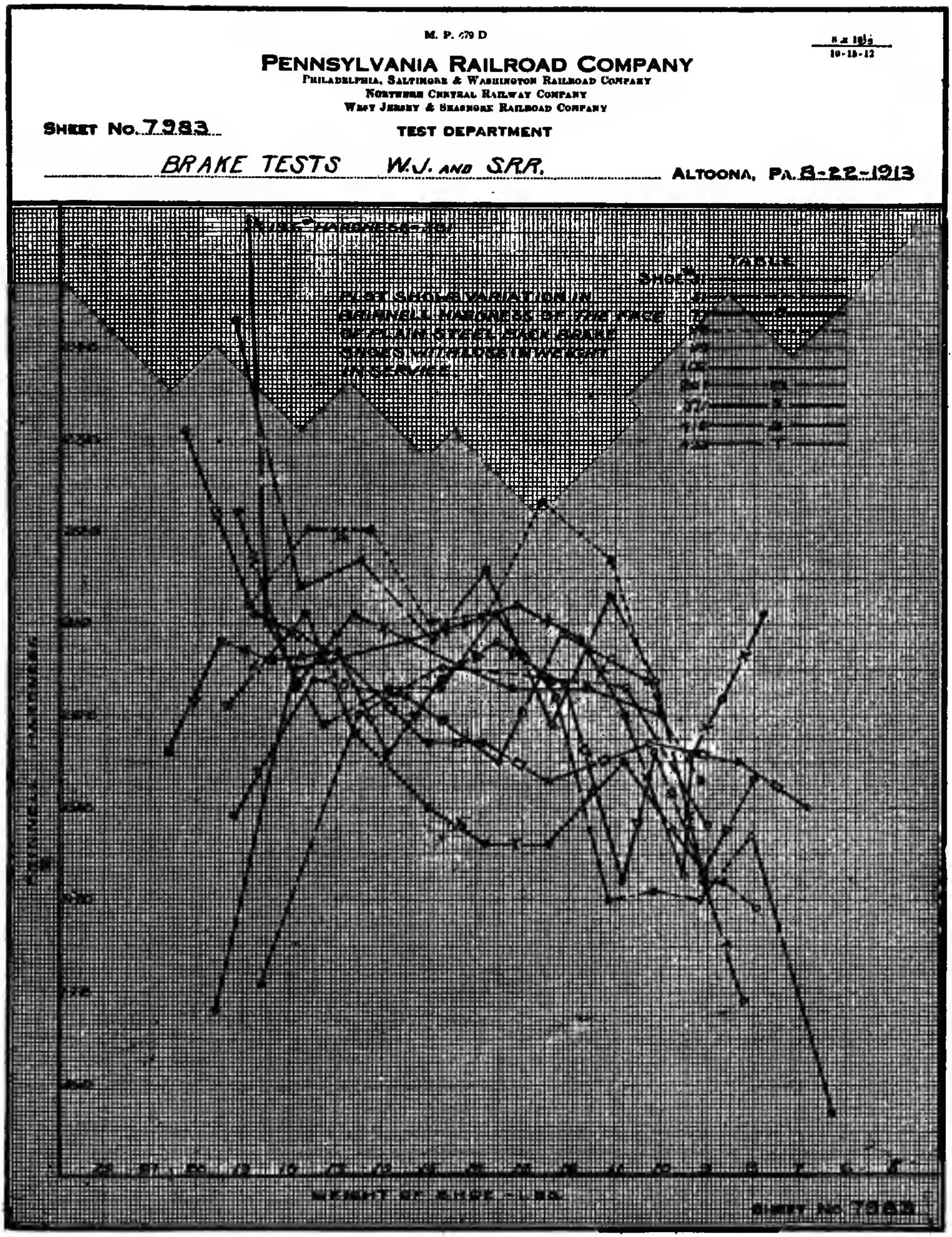


Fig. 79.

HARDNESS OF BRAKE SHOE (BRINELL METHOD).

The change in hardness of shoes as they are worn in service.

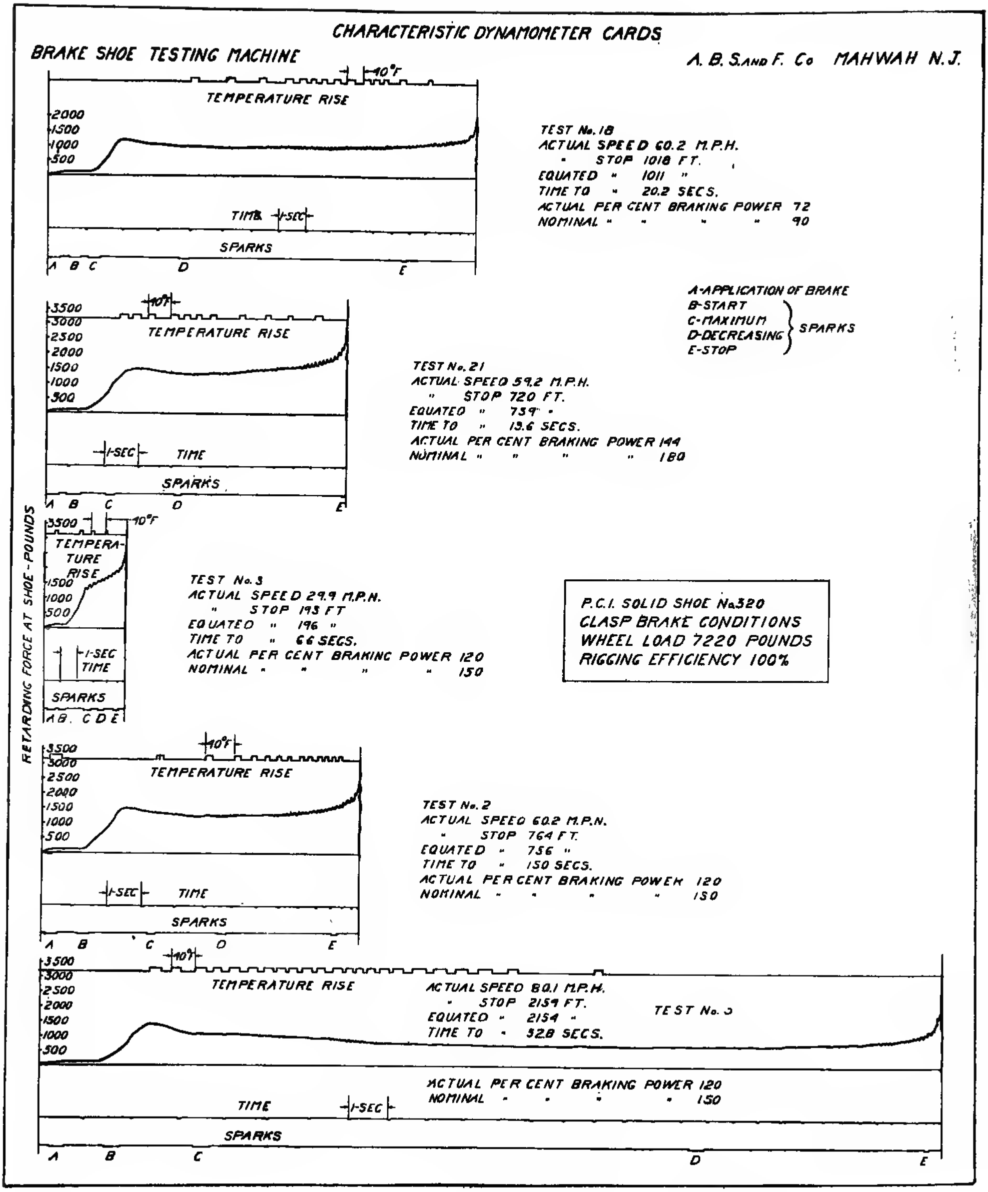


Fig. 80.

BRAKE SHOE TESTING MACHINE.

Typical records.

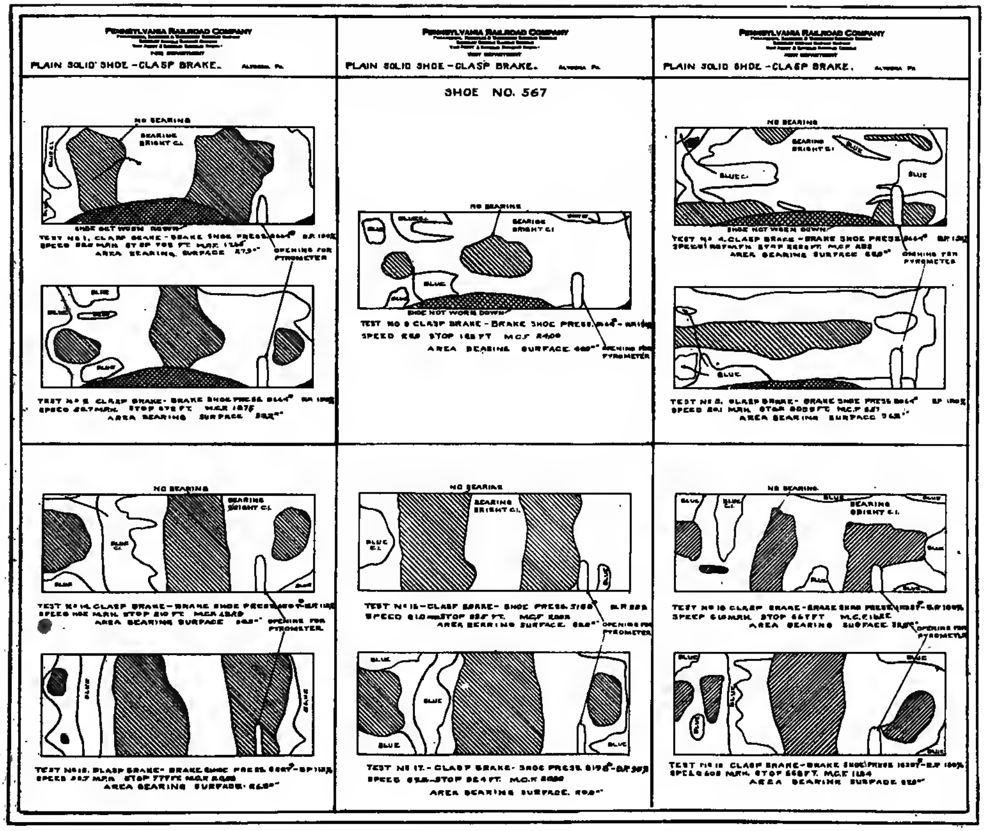


Fig. 81.

BRAKE SHOE BEARING AREA—PLAIN SOLID SHOE.

The shaded portion shows where the shoe does not come in contact with the wheel.

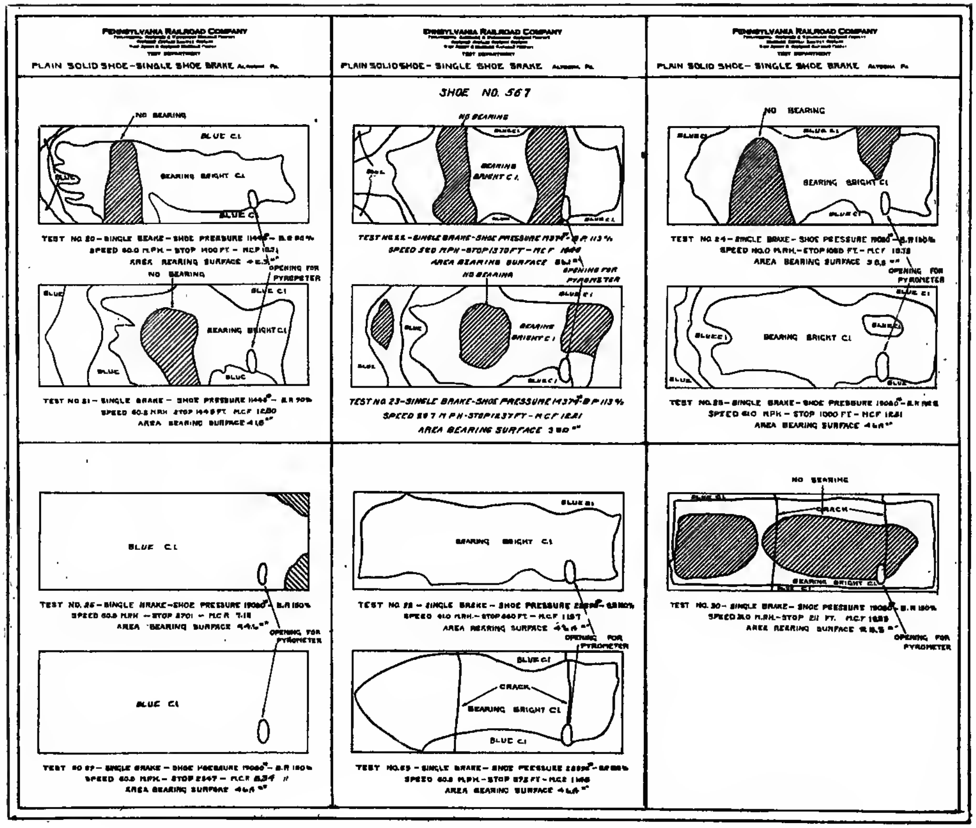


Fig. 82.

BRAKE SHOE BEARING AREAS—PLAIN SOLID SHOE.

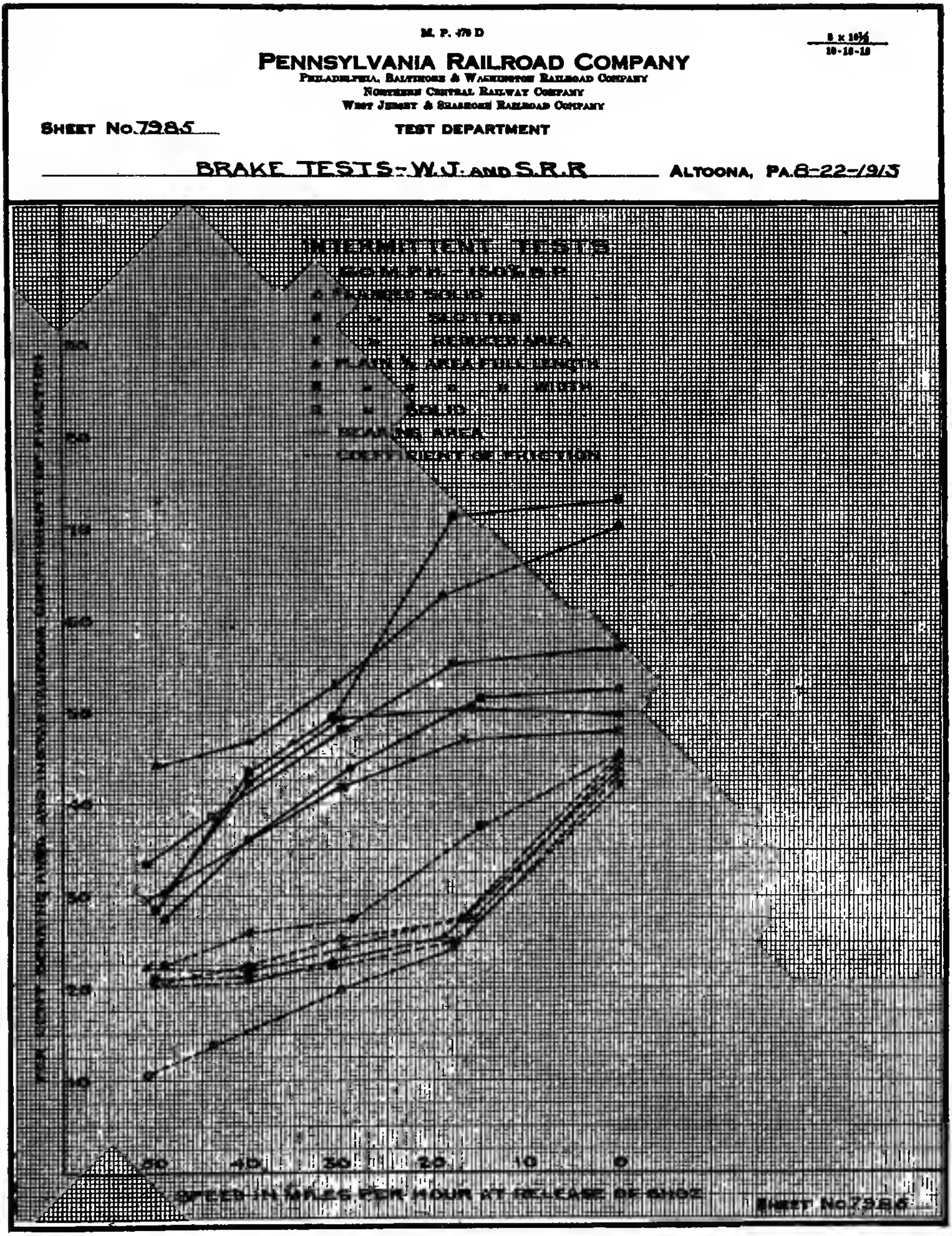


Fig. 83.
BEARING AREA AND COEFFICIENT OF FRICTION.

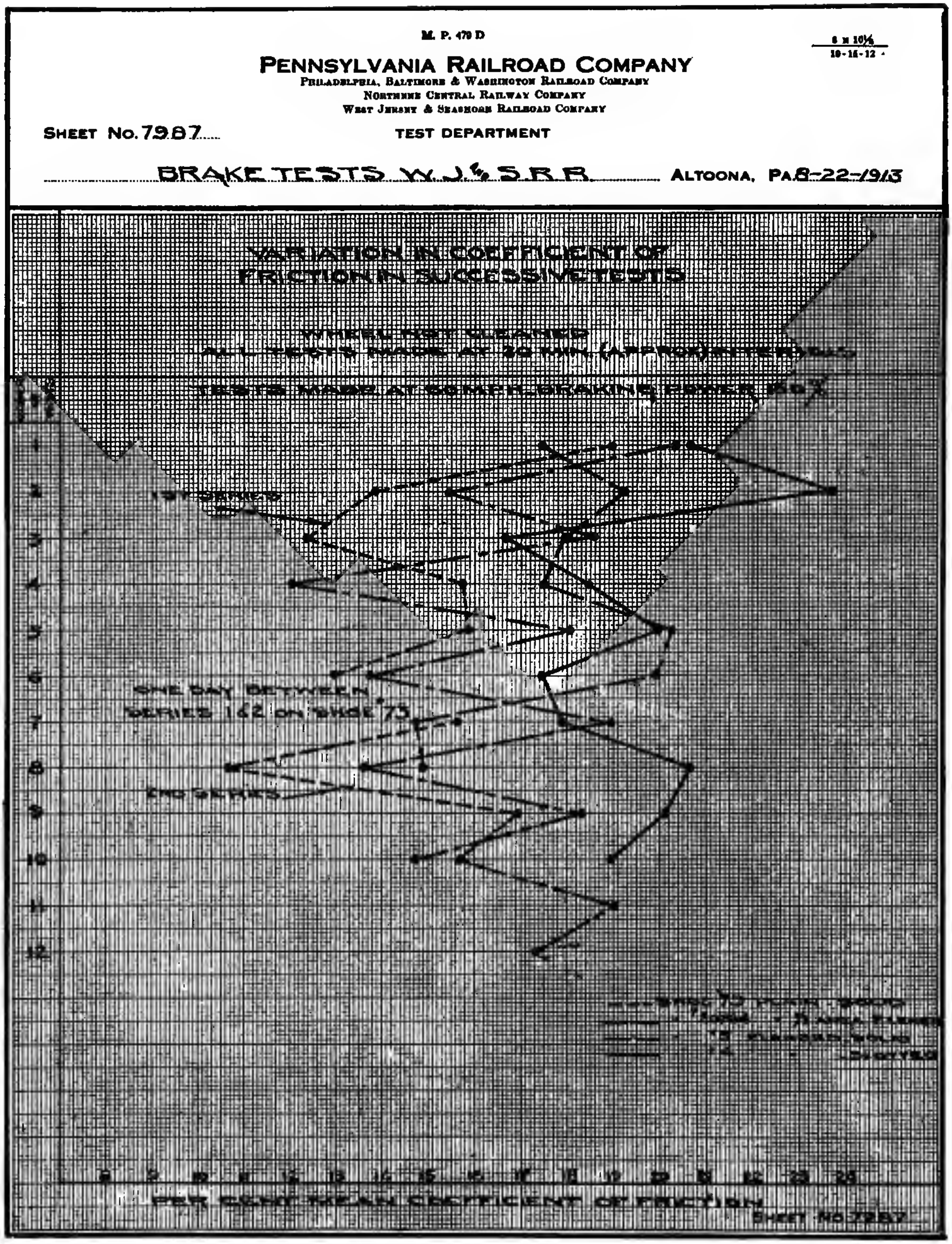


Fig. 84.

COEFFICIENT OF FRICTION OF BRAKE SHOE.

The coefficient varies through a wide range in successive tests.

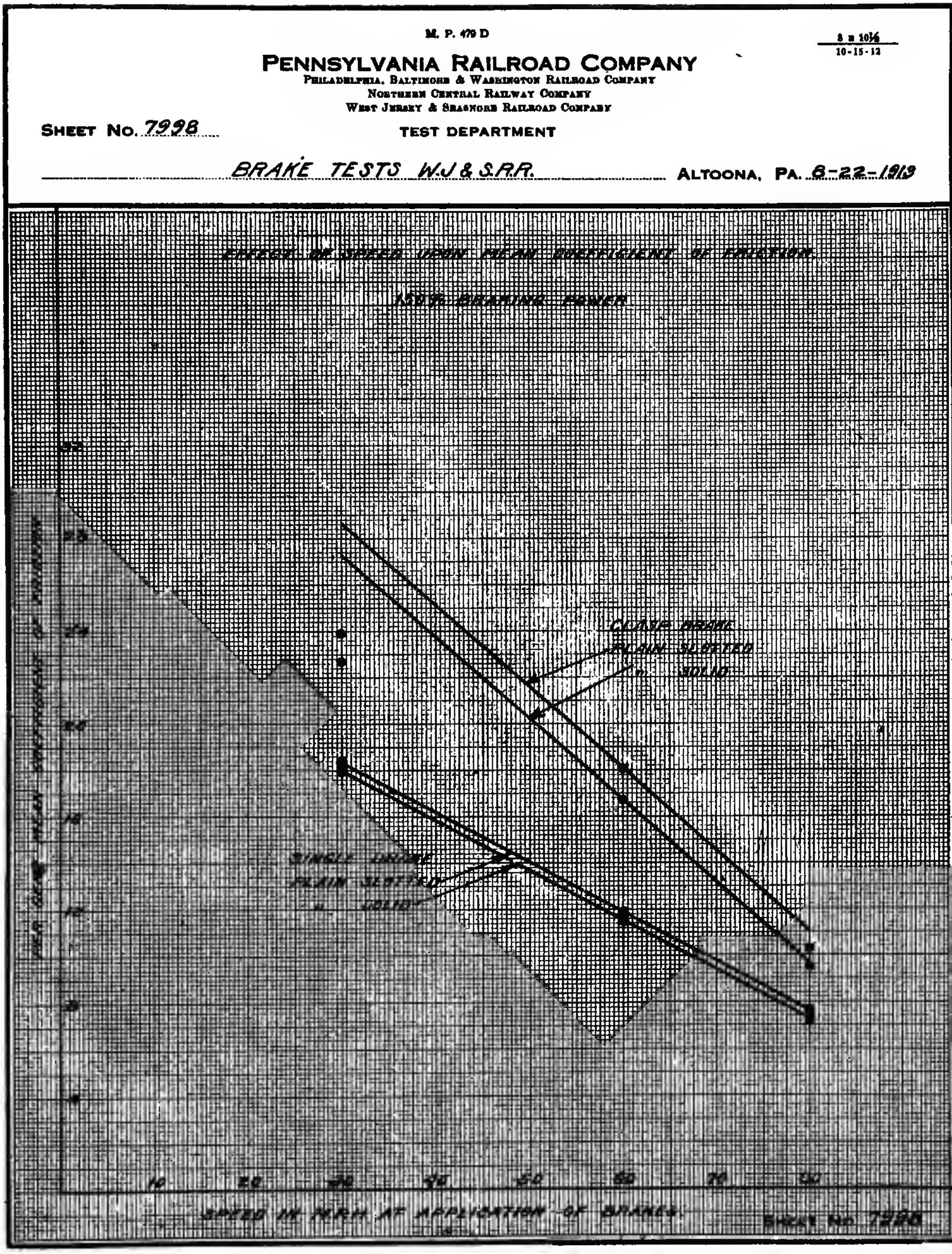


Fig. 85.

COEFFICIENT OF FRICTION AND SPEED.

The slotted shoe shows a higher coefficient of friction than the solid.

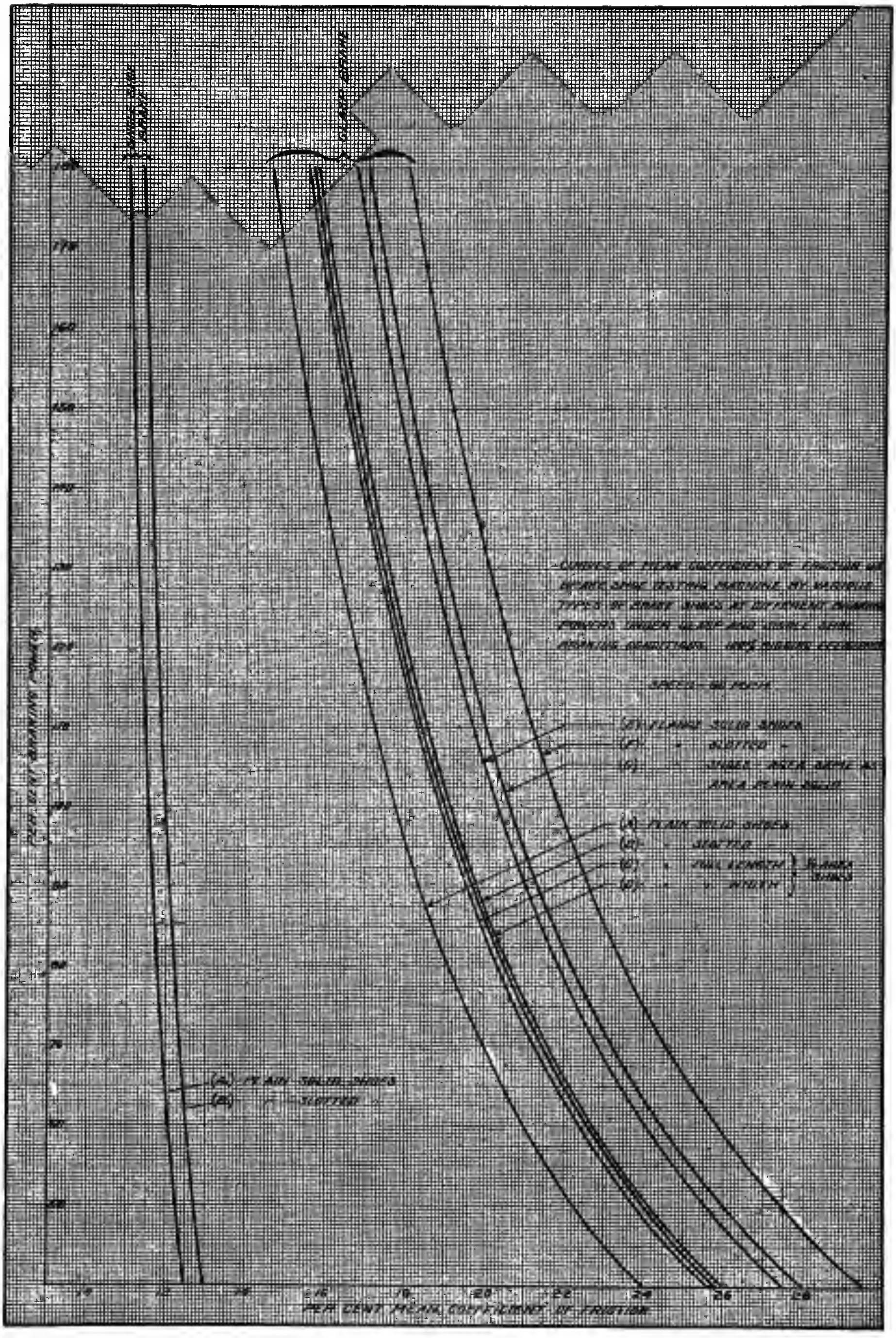


Fig. 86.
COEFFICIENT OF FRICTION—PER CENT. BRAKING POWER.

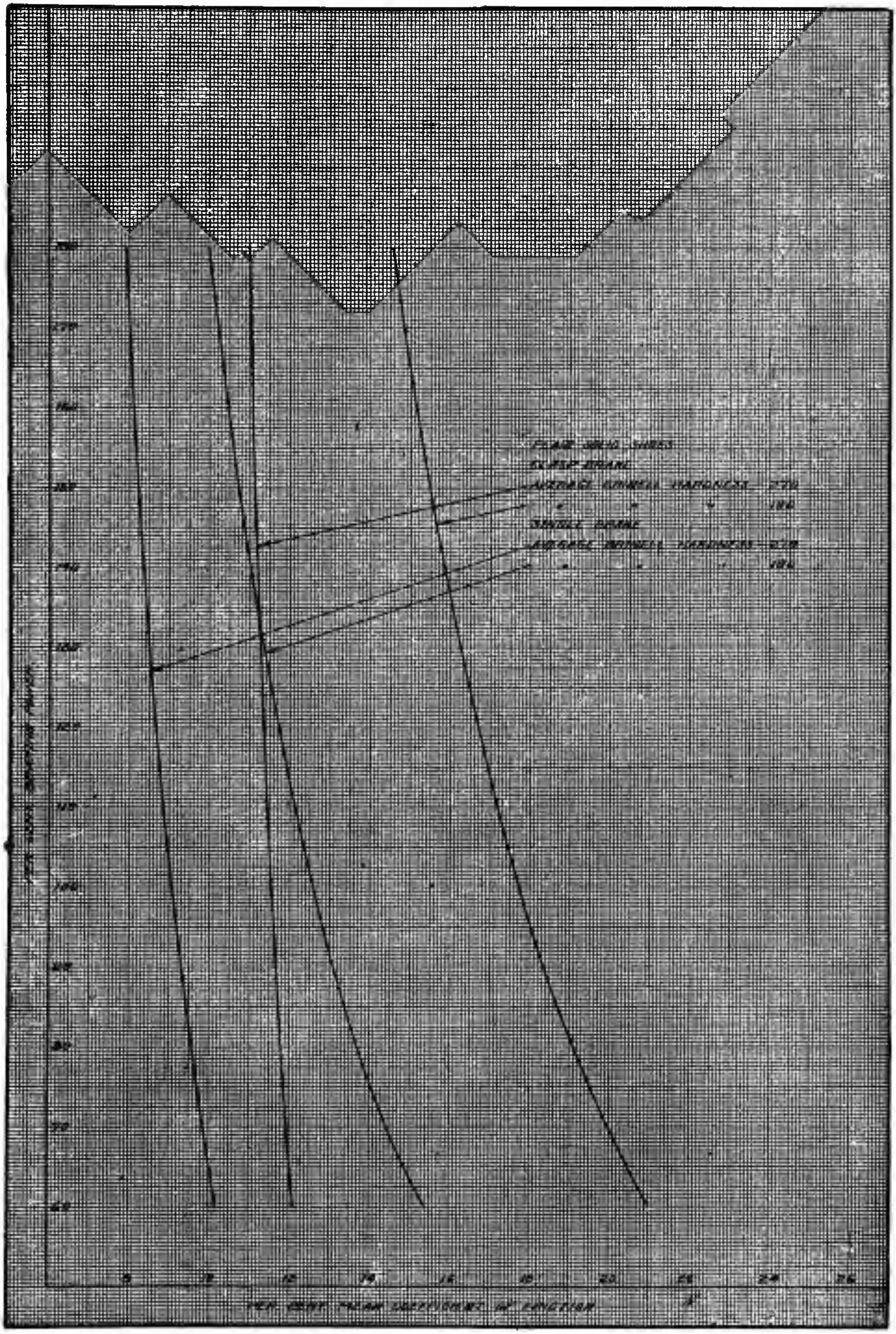


Fig. 87.

COEFFICIENT OF FRICTION—PER CENT. BRAKING POWER.

When the shoes are hard, the coefficient of friction is low throughout the range of braking powers.

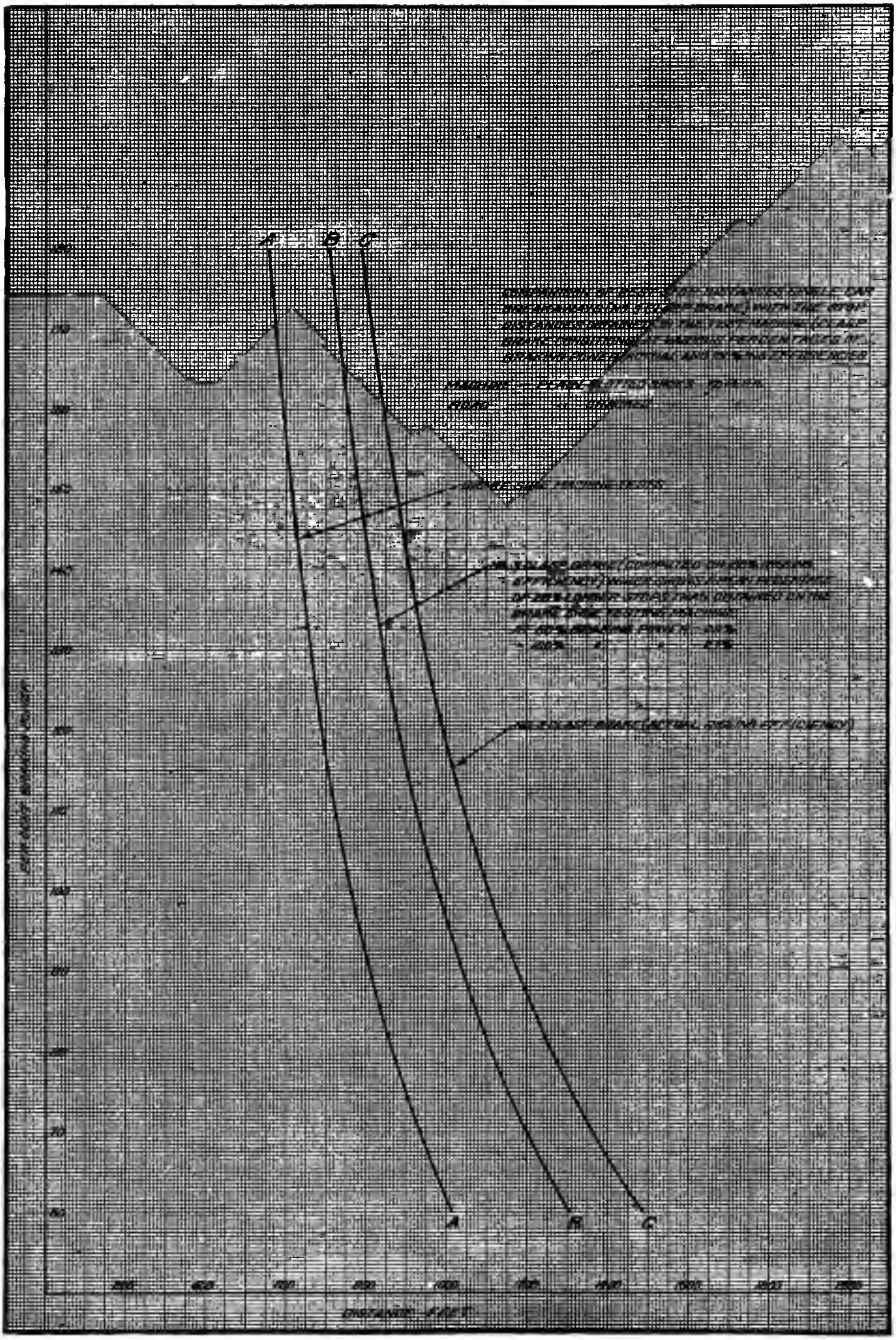


Fig. 88.

RELATION OF MACHINE TO ROAD TESTS.

Plain brake shoes.

DISCUSSION

J. P. Kelley.[1] To the author's statement that the sliding of wheels under brake applications is due largely to rail adhesion, I am inclined to add that the percentage of braking power employed is not of itself the cause of sliding. With the trucks and gears that are familiar to us, the braking forces are not evenly distributed over all the wheels and it is possible for the weight to be momentarily shifted from one or more of the wheels during the progress of a stop with brakes applied.

From a study of clasp brake design and foundation brake gear, as well as from experience, I should say that the clasp brake is the only suitable one to use on present day passenger car equipment. While it will not do away with the shock, it will go a long way towards keeping the wheels in their normal position with relation to the other parts of the truck, and to the rails, and so aid in making available whatever spring action is there to keep the wheels where they belong and to keep the rail adhesion uniform and constant. By this means the pressure on the wheels is balanced, also the number of hot boxes is reduced.

The author has traced the increase in weights of cars from the time when the wooden car was used, weighing about 30,000 lb., up to the present time when steel cars weigh 150,000 lb. or more. With the wooden cars, the braking power could be run up considerably higher than the total weight of the car, and the wheels could be allowed to slide some little distance without injury. There was a certain amount of resiliency in the wooden cars and the wooden trucks which tended to absorb and deaden the shocks thrust upon the wheels which are a fruitful source of wheel sliding by the sudden application of the brakes on long trains. With steel equipment this resiliency is entirely gone, and the wheels of today have to stand more direct shock under heavy brake applications than when wooden cars were used. It is not surprising that there are more cases of wheels sliding under the heavy steel equipments.

We have had considerable experience with clasp brakes on quite a number of our cars during this present winter, and with a few of them the previous winter, and find that as the shoe is higher above the rail in an emergency application than with the common type of gear, there is less chance for the accumulation of snow and ice on the shoe to drip off upon the rails when the shoe gets hot and so reduce the friction.

[1] Cons. Air-Brake Engr. of the N. Y. C. & H. R. R.R. Co.

To show that uneven braking power rather than the per cent of braking power considered in itself is productive of wheel sliding, a recent case may be cited of a train of nine cars, where the highest braking power would probably run 115 per cent, the lowest about 65 per cent, and the average through the train about 85 per cent. Through an accident the brake pipe was broken off on the fourth car from the rear. Every car ahead of the rear car came to a stop without injury to the wheels. They were the ones on which the braking power was very low. The rear car had normal braking power, and all but two pairs of its wheels were slid flat. This could be explained on no other principle than that when the brake application was initiated at the fourth car, and propagated in opposite directions, there was not only the braking power acting on the wheels, but also the effect of shock set up because of the difference in time and the difference in the power with which the brakes went on, which "jumped" the wheels along and caused them to skid to such an extent as to break or reduce very materially their rail adhesion.

Other cases have been noticed where the brakes were set through the accidental parting of the engine from the train, and although there was a uniform braking power throughout the whole train, the brakes on the front part went on with full force, almost instantly, while with those at the rear, there was a delay of one and one-half or two seconds before they began to act, producing a heavy thrust on the two forward cars, with the result that the wheels of those cars went flat.

There can be no question in the mind of any student of air brake science but that the pneumatic action of the brake on the long, heavy modern trains is too slow. If an electric apparatus can be had that will operate the brakes simultaneously from one end of the train to the other, combined with a clasp brake properly designed, and a suitable brake shoe, many annoying troubles will be eliminated.

With reference to emergency application, if a brake will operate instantaneously and simultaneously throughout the train, with almost any degree of flexibility, emergency stops, as such, can be very materially reduced. A brake that will operate instantaneously will bring a train to a stop with a service application, within a distance for which another brake equally powerful but much slower in action would require an emergency application.

H. H. Vaughan. The first thing that strikes me about this paper is the remarkably comprehensive way in which the series of tests has been carried out. The United States should be proud of having a railroad company in its domain with energy enough and interest enough on the subject of brakes to devote the time and money necessary to carry out a series of tests of this kind; and also proud of the fact that a railway company is equipped with a sufficient corps of trained men to take the observations required in the investigations reported in this paper. It is a magnificent testimonial to the scientific side of the operation of American railways.

A point that I wish to refer to, which is brought out in these tests, is the trouble with wheels sliding. About a year ago we went into the question of flanged shoes to reduce the brake shoe pressures, and found an extraordinary increase in wheels slid, especially in cold weather. We then tried unflanged shoes, without any other change, and the sliding reduced. I have no reason to give for this, but it is a point worth bringing up, and the sliding of wheels, while shown in rather a curious way in this paper, is really a very serious and expensive thing, especially in cold weather, in which I am pleased to say I have had a large and most unlimited experience. Wheels sliding with us is undoubtedly a prominent cause of shelled out tires. When it is known that we rarely run a car over 6000 miles in winter without having a wheel with shelled out tires, it can be imagined how serious it is.

Referring to Figs. 76, 77 and 80, the first two show the deceleration plotted from the speed curves of train tests, while Fig. 80 gives dynamometer cards taken at the testing plant. The extraordinary increase in the deceleration of the train tests is very curious. It will be noticed that it goes up almost in the ratio of 3 to 1 toward the end of the stop, while there is very little increase shown on the cards from the brake shoe testing plant. It is a point which has worried me and I ask if there is any explanation for it.

Regarding the paper in general, one is impressed by the marvelous ingenuity of the whole apparatus. It is a wonderful piece of mechanical invention, and one cannot help wondering where we are going to get to in this air brake matter. Probably there was never a more ingenious machine than the plain triple valve invented by Mr. Westinghouse, which made possible the use of automatic air brakes on passenger equipment. In 1887, when the Burlington tests were being conducted, Mr. Westinghouse supplemented his plain triple valve by the quick-action triple, which again was a peculiarly brilliant invention—the idea of knocking down one brake after another all through the train was remarkable and the fame of Mr. Westinghouse as an inventor will probably rest more on these two inventions, which are the foundation of our modern air brake system, than on all the other things he has done.

The quick-action triple valve behaved very well during the early years of its life, but about the time when it was twenty years of age, like many of us, it begun to branch out, and there was attached to the quick service application the retarded release and then a reservoir or two was annexed to the air brake system, giving the graduated release and higher pressure in emergency. There was the LN, and then the PC, and now we are to have the UC, the greatest development of all, which will work either pneumatically or electrically. If the electrical part breaks down, the pneumatic part comes on and works, and the whole thing is wonderful, all the way down from the plain triple, as a monument of inventive genius.

At the same time—I do not wish to raise a discordant voice—are we wise, are the railroad companies wise, is the Westinghouse Company wise, in this development in which the apparatus is continually being revised to interchange with all the other developments that have gone before it? Would it not be possible to design the electric-pneumatic brake without this complication? Mr. Dudley did not attempt to describe the figure showing the parts and passageways of the universal valve. It is a wonderful thing, but should it be put on every passenger car without very serious consideration?

I consider that the Westinghouse Company, supplementing this paper, should tell us what is necessary in the straight electro-pneumatic brake in which the train pipe carries the air and the electricity does the application, and I venture to say, without knowing exactly what would be the answer, that it would be an exceedingly simple and highly efficient apparatus. I cannot help thinking that an electro-pneumatic brake, even with the safety automatic feature on the air pipe, so that a fracture of an air pipe would cause the application of the brakes, would not be anything like the complicated apparatus we are getting today in order to make these things interchangeable and universal.

It would not be a serious matter to add such an equipment to the brake equipment already in use; nor to look forward to carrying the two sets of brakes on our trains during the transition period. I am prepared to place and keep the equipment on certain trains and to take it off of certain trains for a few years, rather than to continue this enormous increase in complications.

It would be an error, I acknowledge, to question the work that has been put on apparatus of this sort, but speaking candidly as a railroad man, I am afraid of it. It is a beautiful piece of apparatus, and all that, but if we can by simple electric control (and it does not seem that the difficulties are insuperable) get a simple, cheap brake, that our men can understand, and upon which they can locate the troubles, and know what they are doing, it would be very much better.

W. B. Turner[1] said in response to Mr. Vaughan's remarks on the complexity of air-brake apparatus that the electro-pneumatic brake has been used for traction service for the past seven years under several thousands of cars; and has been used on the subway system in New York for several years, which is the most severe service in the world, where it has given good results. The electro-pneumatic apparatus can be applied to practically any equipment that has been built up to the present time and will function in a satisfactory manner, but it will not do what the improved air brake equipment (type UC) described in the paper, will accomplish. To attain the results which have been detailed it is essential to have this equipment or its equivalent. While he would welcome any apparatus that was less complex, this would be impossible if it is desired to do the things which are necessary to control the trains of today in the best manner possible. If Mr. Vaughan had his double brake equipment and the supplemental devices for cutting-out the brakes, etc., he would find that the equipment would be vastly more complex than that shown in the paper—it would be so complex that he would be afraid of it himself.

Mr. Turner then presented the following written discussion:

The present day conditions in railroading that affect the air brake problem are:

a Higher Speed, because the energy to be dissipated increases as the square of the speed, which means, other things being equal, that the length of the stop will also increase as the square of the speed.

b Heavier Vehicles, because doubling the weight doubles the energy to be dissipated. The combined effect of doubling the speed and doubling the weight is to increase the energy to be dissipated eight times.

[1] Chief Engineer of the Westinghouse Air Brake Company.

c Longer and Heavier Trains. These increase the difficulty of control and greatly magnify reactions due to "slack" between cars.

d Greater Frequency of Trains. This increases the probability of accident and therefore the necessity for more efficient control.

e Parallel Tracks. This also increases the danger, as an accident on one track may cause one on any of the other tracks.

Any one of these changes in conditions makes imperative a compensating change in effectiveness in the means for controlling trains, but when they are all taken collectively, as they must be, the seriousness of the problem is even more apparent and, to one who knows the potential, alarming. But this is not all, for the very conditions that thus called for more effective brake appliances also decreased the efficiency or effectiveness of those existing by almost one-half. The predicament, therefore, is one requiring prompt attention and the creation of means not heretofore in existence.

The reasons why the above-mentioned conditions decrease the effectiveness of the brake control are:

Increased speed (work) causes a decrease in the mean coefficient of friction; and as the stopping distance is proportional to the coefficient of friction, the stop becomes longer.

Heavier vehicles also increase the work required of the brake shoe in a given time, and this further reduces the coefficient of friction. Also, as a result of the greater forces exerted, there are distortions and consequent losses in foundation brake gear efficiency. Again, much larger brake cylinders are necessary and this increases the time required to get full braking power, these three losses contributing, as is apparent, to a further material increase in stopping distance.

Longer trains lead to another decrease in effectiveness by adding to the time required to get the brakes on, while the combination of greater weight and length increases the volume of air required with consequent difficulty of recharging quickly. There is also the difficulty of controlling trains smoothly because of the bunching and stretching of such a number of heavier cars and the time lapse between the brakes applying on the front and rear end of the train.

Greater frequency of trains and parallel tracks make improvement necessary by requiring much more frequent brake applications, thus exhausting the air supply, and also by making it imperative to be able to obtain maximum emergency braking force at any time. Collectively, it will be seen that these changes mean much, nor must it be overlooked that one serious effect is the increased risk we must take with the human element.

To illustrate the foregoing by example, in 1890 with a train weight of 280 tons and a speed of 60 miles per hour, the energy to be dissipated was about 33,000 ft-tons and the stopping distance was 1000 ft.

In 1913, with a train-weight of 920 tons and a speed of 60 miles per hour, the energy to be dissipated is 111,000 ft-tons, or almost four times greater than that of the first mentioned train. With an old-style brake the collision energy as this train passed the point where the first mentioned train stopped would still be 48,000 ft-tons, or one and one-half times what the first mentioned train had before the brake was applied and there would still be 760 ft. to run.

This modern train, but now equipped with the new brake apparatus, running at a speed of 60 miles per hour, can be stopped in 860 ft., at which point, with the old brake equipment, it would still be running at 43 miles per hour and with a collision energy of 57,000 ft-tons or still about twice that contained in the train of 1890 at the beginning of the stop.

When the full force and meaning of the increased stopping distance is realized, in connection with the possibility of accidents arising from the greater frequency of trains and number of parallel tracks, it will be seen that not only are improvements in brake equipment really necessary, but also that the best that can be designed is none too good. It was a realization and an analysis of the effects of the changed conditions mentioned that led to the creation of the brake employed in the tests with which this paper deals.

The illustrations no doubt impress one with the complexity and comparative size of the apparatus, but when the net result measured by control requirements and stopping distance, is more effective than with the old brake, it will be admitted that these things must be accepted. Nothing less will suffice if reasonable capacity of track and rolling stock is to be had and general advance in transportation and safety are to keep equal pace.

F. J. Barry[1] said that after reading the paper over five times he not only felt he could not discuss it, but that what knowledge he had of electro-pneumatic brakes was very small in comparison. It had been his observation that when a paper was in such good shape

[1] Genl. Inspr. Air Brakes, N. Y. O. & W. R. R. Co.

as this one there was little to say about it, and that the more it was re-read the less one found to say. His own experience had extended over a period of about 20 years, and he was not familiar with the UC equipment. He was glad to state that his company had some new passenger equipment coming in the near future and that, through his recommendation, they would put on the clasp brake but not the UC.

F. W. Sargent. In conducting machine tests of brake shoes it is our practice to make a number of runs to get the shoes down to a good bearing and then make a number of records and average them. The average figures derived from repeated tests upon different shoes should be comparable with the results that might be obtained from the brake-shoe equipment of a train. The great variation in the train records as presented in the paper must be due, I think, to some neglect in the foundation brake rigging, rather than to a bad condition of all the brake shoes at one time.

The agreement between the results of the brake shoe testing machine and the road tests is extremely gratifying. When we first began testing brake shoes on the machine we encountered very large variations in the results, particularly with the heavy solid new shoes. But as the shoes wore down, and especially when they cracked and fitted the wheels better, the results became more uniform. When the shoes became thin and near the point where they ought to be removed, it developed that under the extreme pressures common to modern high-grade equipment, with the single brake, we were reaching the danger point and there was a possibility of the shoe melting. Such a condition would be realized on the road if the brakes were applied on a grade and then an emergency stop followed while the shoes were still warm. Under such a condition the shoes might be heated to a point where they would lose their grip on the wheels to a large extent, even if not melted away, and cause the brake to fail. It seems to me that the only thing which will prevent this is the adoption of the clasp brake. Besides saving the brake shoe from possible destruction, the test records indicate that with the two-shoe combination there is an actual gain in the retardation.

The machine test also demonstrates that the flanged brake shoe has a greater retarding effect with the same load applied, than the unflanged shoe, besides rendering better service in conducting away the heat. I think the objection made by Mr. Vaughan to the flanged shoe as compared with the unflanged shoe was due not to any inherent defect of the flanged shoe, but to the fact that the brake rigging was not adapted to that kind of shoe. We certainly get increased braking efficiency by the use of the flanged shoe.

R. R. Potter.[1] It may be of interest to speak of some tests made recently on the New York, Westchester & Boston Railway, to determine the efficiency of the clasp brake. The cars, which are operated electrically, weigh 60 tons, and are run as high as 60 miles an hour. In making the test, a motor truck without motors, which has clasp brakes, was put under the trailer end of the car, which normally has a simple truck. At 35 miles per hour the deceleration under an emergency application of the brake was found to be 4 miles per hour per second with the clasp brake; with the simple brake, it was 3.3 miles per hour per second. At 50 miles per hour, the deceleration was 3.65 miles per hour per second with the clasp brake, and with the standard brake, 3 miles per hour per second. The length of stop with the clasp brake, at 50 miles per hour, was 585 ft., and with the simple or standard brake 690 ft. These were the average lengths of stop for about 40 stops. Invariably the stops were shorter when the clasp brake was used. Stops were made at various speeds, principally at 35 and 50 miles per hour.

Another point of observation was the effect on the journal bearings. With the simple brake we experienced trouble with the journal leaving the brass, when brakes were applied to a pressure of 60 lb. in the brake cylinder. In an emergency application, we have 90 lb. pressure in the brake cylinder while the normal full service application gives a pressure of 50 lb. The braking power in the motor truck, service, is 100 per cent, and emergency, 180 per cent. In the trailer truck, service is 90 per cent and emergency 162 per cent.

S. G. Thomson.[2] I am particularly interested in the business end of this new brake, the part which involves its practical application. After listening to the discussion, the important thing to me seems to be the development of the foundation brake. Have we tried out the foundation brake far enough? Have we gone far enough with the clasp brake in increasing the friction applied on the wheels and are we not going too far for the present in this development of the electric control? I am inclined to agree with Mr. Vaughan in respect to the matter of complications.

[1] Supt. of Equipment, N. Y. W. & B. Ry.

[2] Supt. M. P. and Rolling Equip., Phila. & Reading R.R.

We have had considerable experience with the clasp brake on our road for a number of years, running it on 100 or more cars, with four and six-wheel trucks, high-speed service. All are giving excellent results, so that it seems to me more important at the present time to extend the use of the clasp brake than to hasten the electric development. Let the electric control be a matter of gradual unfolding.

How much benefit would this electric control as presented here tonight be to us? I figure that the principal gain would be in the little time saved by applying a considerably greater pressure in the brake cylinder; also that there would be a considerable advantage in making an emergency application after the service application has been made. Is that really of sufficiently great importance in actual train service to warrant all this complication and expense? An engineer may occasionally run by his station and may even bring his passengers up standing in applying the brakes; but does even such a matter warrant us in going to the great expense of equipping all of our cars with this extra braking equipment, and in trying to get men who could take care of it?

It may be that the electric controlled air-brake apparatus could be applied to advantage on a few long trains, but we all run many three-car and five-car trains where such apparatus is not required. I want, therefore, to raise the question for discussion as to how generally we need this further complication at the present time, and whether there is not more need for the development of the foundation brake including the clasp type.

T. L. Burton.[1] Both Mr. Sargent and Mr Vaughan have spoken of the wheel sliding effect of the flanged brake shoe. I have had considerable experience with this and, as the former stated, have found the trouble with the flanged shoe to be due largely to the brake gear, especially the spacing of the heads on the brake beams and the deflection of the beams under various loads.

There has been a question in the minds of some as to the relative stopping effect of the clasp and single-shoe types of brake, all conditions, excepting the performance of the gear and the shoe, being the same. For any set of conditions such as stated in Table 10, the length of the stop should be inversely proportional to the value of

[1] Westinghouse Air Brake Co., 165 Broadway, New York.

$e \times f_s$; consequently, so far as the brake shoe and rigging are concerned, Table 10 affords a precise means of comparing the length of stops obtained with the various types of brake gear and brake shoes used in the test with which the report deals.

I have prepared some supplementary tables, reducing Mr. Dudley's comparison to a percentage basis. Accepting the values of e and f_s as shown in Table 10 for clasp brakes and flanged shoes as 100 per cent, the relative length of stops for clasp and single shoe brakes should be as follows:

RELATIVE LENGTH OF STOP

Speed in M.P.H.	Per Cent Braking Power	Clasp Brakes Plain Shoes	Clasp Brakes Flanged Shoes	Single Shoe Brake Plain Shoes	Single Shoe Brake Flanged Shoes
30	125	$\frac{0.169}{0.141}=1.199$	$\frac{0.169}{0.169}=1.000$	$\frac{0.169}{0.108}=1.565$	$\frac{0.169}{0.112}=1.509$
30	150	$\frac{0.154}{0.129}=1.194$	$\frac{0.154}{0.154}=1.000$	$\frac{0.154}{0.099}=1.556$	$\frac{0.154}{0.103}=1.495$
30	180	$\frac{0.141}{0.118}=1.195$	$\frac{0.141}{0.141}=1.000$	$\frac{0.141}{0.090}=1.567$	$\frac{0.141}{0.094}=1.500$
60	125	$\frac{0.122}{0.103}=1.184$	$\frac{0.122}{0.122}=1.000$	$\frac{0.122}{0.074}=1.648$	$\frac{0.122}{0.090}=1.356$
60	150	$\frac{0.112}{0.094}=1.191$	$\frac{0.112}{0.112}=1.000$	$\frac{0.112}{0.068}=1.647$	$\frac{0.112}{0.082}=1.366$
60	180	$\frac{0.102}{0.086}=1.186$	$\frac{0.102}{0.102}=1.000$	$\frac{0.102}{0.062}=1.645$	$\frac{0.102}{0.075}=1.360$
80	125	$\frac{0.109}{0.092}=1.185$	$\frac{0.109}{0.109}=1.000$	$\frac{0.109}{0.070}=1.557$	$\frac{0.109}{0.074}=1.473$
80	150	$\frac{0.100}{0.084}=1.190$	$\frac{0.100}{0.100}=1.000$	$\frac{0.100}{0.064}=1.563$	$\frac{0.100}{0.068}=1.471$
80	180	$\frac{0.092}{0.077}=1.195$	$\frac{0.092}{0.092}=1.000$	$\frac{0.092}{0.059}=1.559$	$\frac{0.092}{0.062}=1.484$

INCREASE IN LENGTH OF STOP WITH THE SINGLE SHOE BRAKE AS COMPARED TO THE CLASP BRAKE

Speed in M. P. H.	Per Cent Braking Power	Plain Shoes Per Cent	Flanged Shoes Per Cent
30	125	$\frac{1.565}{1.199}-1=0.305=30.5$	$\frac{1.509}{1.000}-1=0.509=50.9$
30	150	$\frac{1.556}{1.194}-1=0.304=30.4$	$\frac{1.495}{1.000}-1=0.495=49.5$
30	180	$\frac{1.567}{1.195}-1=0.311=31.1$	$\frac{1.500}{1.000}-1=0.500=50.0$
		$\frac{1.648}{1.184}-1=0.392=39.2$	$\frac{1.356}{1.000}-1=0.356=35.6$
		$\frac{1.647}{1.191}-1=0.383=38.3$	$\frac{1.366}{1.000}-1=0.366=36.6$
		$\frac{1.645}{1.186}-1=0.387=38.7$	$\frac{1.360}{1.000}-1=0.360=36.0$
		$\frac{1.557}{1.185}-1=0.314=31.4$	$\frac{1.473}{1.000}-1=0.473=47.3$
		$\frac{1.563}{1.190}-1=0.314=31.4$	$\frac{1.471}{1.000}-1=0.471=47.1$
		$\frac{1.559}{1.195}-1=0.304=30.4$	$\frac{1.484}{1.000}-1=0.484=48.4$
Average Gain in favor of Clasp Brake, Per Cent		35.5	44.6

Tabulated results show the following *average* comparative lengths of stops:

Average stop, clasp brake, plain shoes 1.191
Average stop, clasp brake, flanged shoes 1.000
Average stop, single-shoe and plain shoes 1.590
Average stop, single-shoe and flanged shoes 1.446

Length of stop with clasp brake, plain shoes is 19.1 per cent greater than stop with clasp brake, flanged shoes.

Length of stop with single shoe brake, plain shoes is 59.0 per cent greater than stop with clasp brake, flanged shoes.

Length of stop with single shoe brake, flanged shoes is 44.6 per cent greater than stop with clasp brake, flanged shoes.

Also, length of stop with single shoe brake, plain shoes is 33.5 per cent greater than stop with clasp brake, plain shoes.

Whence the *general* average length of stop with the single shoe brake is $\frac{1}{2}(44.6+33.5)=39.05\%$ greater than the general average length of stop with the clasp brake.

The relative stopping distance of the two types of brake gear, however, tells but half the story. Mr. Dudley refers to the sliding of wheels as affected by the adhesion between the wheels and the rail. The adhesion is influenced, not only by the condition of the rail, but by the shifting of the wheel loads as affected by inertia forces, and shocks and lines of action in the brake gear.

Unfortunately, we have no control over the condition of the rail, or the effect of the inertia. The use of electro-pneumatic brakes will, however, minimize the shifting of wheel weights resulting from end shocks, and a careful analysis of many brake gears in use will disclose some remarkable facts on the equalization of wheel weights and shoe pressures as influenced by the design and application of the gear.

With the assistance of Mr. H. M. P. Murphy of our organization, we have made an analysis of the brake forces of some trucks in use on a road in this territory and their effects on the equalization of weights at wheels and shoe pressures. I will add in brief form some results of the analysis for the brake rigging of a six-wheel passenger car truck.

The tabulation relates to braking forces, wheel pressures on rail, percentage of braking power, etc., for service application, with all parts standard; and for emergency application, with shoes and tires worn and journals displaced in accordance with actual conditions observed. The normal braking power in service was 85 per cent and in emergency, 160 per cent. The total actual weight of car under the standard conditions assumed for service is 142,000 lb. Under the worn conditions assumed for emergency the total car weight is 138,000 lb.

NORMAL BRAKE SHOE PRESSURES: SERVICE

	Leading Truck	Rear Truck
Outside wheel	9155 lb	8825 lb.
Middle wheel	9775 lb	9175 lb.
Inside wheel	8130 lb	8215 lb.

Variation in normal brake shoe pressure = 1645 lb.; per cent variation = 20.2.

Normal Brake Shoe Pressures: Emergency

	Leading Truck	Rear Truck
Outside wheel	15,785 lb	16,090 lb.
Middle wheel	15,605 lb	15,555 lb.
Inside wheel	10,835 lb	9,840 lb.

Variation in normal brake shoe pressure = 6,250 lb.; per cent variation = 63.5.

Actual Wheel Pressures on Rail: Service

	Leading Truck	Rear Truck
Outside wheel	13,727 lb	11,622 lb.
Middle wheel	10,697 lb	11,826 lb.
Inside wheel	11,076 lb	12,052 lb.

Variation in wheel pressure on rail = 3030 lb.; per cent variation = 28.3.

Actual Wheel Pressures on Rail: Emergency

	Leading Truck	Rear Truck
Outside wheel	14,735 lb	11,956 lb.
Middle wheel	9,816 lb	11,234 lb.
Inside wheel	10,056 lb	11,435 lb.

Variation in wheel pressure on rail = 4,937 lb.; per cent variation = 50.3.

Actual Percentage of Braking Power: Service

Leading Truck

Outside wheel: $\frac{9155}{13727} \times 100 = 66.7$

Middle wheel: $\frac{9775}{10697} \times 100 = 91.4$

Inside wheel: $\frac{8130}{11076} \times 100 = 73.4$

Rear Truck

Outside wheel: $\frac{8825}{11622} \times 100 = 75.9$

Middle wheel: $\frac{9175}{11826} \times 100 = 77.6$

Inside wheel: $\frac{8215}{12052} \times 100 = 68.1$

Variation in percentage of braking power $= 91.4 - 66.7 = 24.7$. Per cent variation in percentage of braking power $= \frac{24.7}{66.7} \times 100 = 37.0$.

Force Efficiency of Brake Apparatus: Service

For leading truck,

$$\frac{2\times(9155+9775+8130)}{6\times9052.5}\times100=99.6\%$$

For rear truck,

$$\frac{2\times(8825+9175+8215)}{6\times9052.5}\times100=96.5\%$$

Force Efficiency of Car Brake, Exclusive of Cylinder Losses, Etc.: Service

$$\frac{54120+52430}{2\times54315}\times100=98.1\%$$

Force Efficiency of Car Brake, Including all Losses

$$\frac{106550}{0.85\times142000}\times100=88.3\%$$

Actual Percentage of Braking Power: Emergency

Leading Truck

Outside wheel: $\frac{15785}{14753}\times100=106.9+$

Middle wheel: $\frac{15605}{9816}\times100=159.1+$

Inside wheel: $\frac{10835}{10056}\times100=107.7$

Rear Truck

Outside wheel: $\frac{16090}{11956}\times100=134.6$

Middle wheel: $\frac{15555}{11234}\times100=138.5$

Inside wheel: $\frac{9840}{11435}\times100=86.1$

Variation in percentage of braking power $=159.1-86.1=73.0$. Per cent variation in percentage of braking power $=\frac{73.0}{86.1}\times100=84.8$.

Force Efficiency of Brake Apparatus

For leading truck,

$$2\times\frac{(15785+15605+10835)}{6\times17040}\times100=82.6\%$$

For rear truck,

$$2\times\frac{(16090+15555+9840)}{6\times17040}\times100=81.2\%$$

Force Efficiency of Car Brake, Exclusive of Cylinder Losses, Etc.

$$\frac{84450+82970}{2\times102240}\times100=81.9\%$$

Force Efficiency of Car Brake, Including all Losses

$$\frac{167420}{1.60\times142000}\times100=73.7\%$$

Mr. Dudley speaks of the variation in piston travel with the single shoe type of gear and the ill effect resulting therefrom, in the overall efficiency of the emergency brake. It also seriously affects the service brake application.

Other things being equal, the ability to release the brake satisfactorily is directly proportional to the extent of the reduction made in brake pipe pressure during an application.

With the average six-wheel truck gear, having the shoes hung well below the wheel centers, an 8 or 10 lb. application of the brake will scarcely give over 4 or 5 in. piston travel, and instead of developing the breaking power for which the apparatus is designed and proportioned, it develops more nearly 70 to 80 per cent braking power, and thereby effects a high rate of retardation, especially at low speed, which limits the ability of the operator to make a brake application sufficiently heavy to insure a satisfactory release.

The next question is why not let the piston travel out further? When it is so adjusted that an 8 to 10 lb. service application will develop only from 4 to 5 in. piston travel, it is adjusted so that an emergency application will cause the piston to travel full stroke or practically so, therefore the travel cannot be lengthened.

We also made a similar analysis of force actions of the brake gear referred to in the paper as Clasp Brake Design No. 3, and were able to find no shifting of weights at the wheels as a result of horizontal forces in the brake gear, and the normal brake shoe pressures varied but 70 lb. per wheel, an average of 35 lb. per shoe, with 36-in. wheels and new shoes and 33-in. wheels with shoes ⅜-in. thick.

P. J. LANGAN[1] said that Mr. Vaughan had touched upon the very points that he had been dwelling on for a long time and that Mr. Thomson also had covered some of these points. As the latter stated, we have trains of three, four and five or more cars, besides the longer trains of 12 and 14 cars, but in the installation of a brake equipment one cannot be governed by the number of cars per train because these cars do not remain constantly in the same train. Of course if there are 12 or 14 cars in a train, they must be provided with the best possible brake. A practical question in relation to the electro-pneumatic brake is How are we going to educate the repair-man and the inspector to handle it? Mr. Thomson struck the proper note when he advised us to get the foundation gears into proper shape before we started in on any advancement in the electro-pneumatic line. There is no question about the value of the clasp brake. We have not jumped into the use of the PC nor the UC types of brakes as we have been waiting until it is demonstrated that the later types are better than those that we now have. All honor is due the railroad companies who are advancing these tests and providing the information needed.

N. A. CAMPBELL.[2] After the Central Railroad of New Jersey high-speed brake tests in 1903, a relatively short time ago, it was thought that the last word had been said in high-speed braking. Stops were made from 60 miles per hour in less than 1000 ft. and until the Pennsylvania Railroad tests last year they had not been equalled. The locomotives and cars, however, were about half the weight of those now in use. The problem of braking the heavier modern cars so as to make the shortest possible stop in emergency applications has been a much more difficult one than that with which we were confronted ten years ago. Few fast passenger trains at that time averaged more than six cars in length or more than 500 tons including the locomotive. Trains of from 10 to 12 cars that weigh over 1000 tons, including the locomotive, have been a source of much trouble to some roads on account of the severe shocks and wheel sliding that resulted from emergency applications. The time required to obtain the maximum brake cylinder pressure, in emergency, as well as the time of propagating quick action through the train, is a very important factor in the length of a stop.

With the pneumatic brake, the brake cylinder pressure reaches the maximum on the forward portion of the train before the brakes

[1] Trav. Air Brake Instr., D., L. & W. R.R.

[2] Rep., N. Y. Air-Brake Co., 165 Broadway, New York.

have been applied on the rear. The result is that the slack has time to run in, causing a shock, more or less severe, according to conditions, and exerting a force against the forward portion of the train which has a tendency to cause slid flat wheels. Some railroad men have been protesting against the rapid rise of brake cylinder pressure in emergency applications. They said that they would prefer a slower rise of pressure in the cylinders and longer stops than the trouble due to shocks and slid flat wheels. They are justified in their complaint as far as the pneumatic brake is concerned, but I believe the electro-pneumatic brake will tend to overcome the trouble.

The use of the electric control in combination with an air brake equipment of more modern design, which has port areas sufficiently large to permit the maximum cylinder pressure to be obtained in less than half the time required by the older equipments, will very materially reduce the length of emergency stops.

However, as there are very many service stops made to one in emergency, it is just as important, if not more so, to take care of the service applications. The time required to make a service reduction of brake pipe pressure increases with the increased length of the train, and as the brakes are applied on the forward portion of the train before they are on the rear, causes shocks or surges. Electric control of the brakes causes the brakes on all cars to operate simultaneously, thus eliminating the cause of these shocks by eliminating the time element existing with pneumatic operation.

With the electro-pneumatic brake, even the indifferent or unskilled engineer can handle a long train as smoothly and accurately as a single car, while without the electric control the careful and skilled engineer cannot always make a smooth, accurate stop, no matter how hard he tries.

The author shows very clearly the better results obtained with the clasp brake. It is not, however, a recent type of foundation brake gear. It was used with the vacuum brake many years ago, and should never have been discarded. The fact that with it stops can be reduced 12 per cent should be sufficient to recommend it, but that the losses due to false piston travel, journal and truck reactions, brake shoe wear and variable shoe action are eliminated by it, are much more important.

The Author. Mr. Vaughan's graceful tribute to the organization responsible for these tests is particularly gratifying on account of his own long and intimate familiarity with the technical, engineering and economic phases of the brake problem.

Wheel sliding, to which he refers, is perhaps one of the most elusive limitations to successful and satisfactory brake operation. As Mr. Kelly explains, the factors involved in finally determining whether wheels slide or not in any given case are so numerous, complicated and oftentimes obscure that the assignment of a definite cause or causes or the adoption of practicable preventative measures, at the same time maintaining the desired brake effectiveness, is almost entirely a matter of judgment rather than of fixed rule.

The effect of flanged brake shoes as observed by Mr. Vaughan has been a more or less common experience on many roads in this country. On the other hand a large railroad system in the East has used flanged brake shoes of a highly efficient type as standard on all their passenger cars for years. The cars are braked from the ordinary high-speed brake standard up to as high as 180 per cent braking power in emergency, trains being made up as the cars may come. This road has not experienced any inconvenience from wheel sliding under these conditions. As pointed out by Mr. Sargent and Mr. Burton it has sometimes been found that the brake rigging and brake head construction has been responsible for trouble experienced with flanged brake shoes. The use of worn shoes on new wheels has also been a cause of trouble.

There appears to be but one circumstance which can unhesitatingly be blamed for sliding wheels, namely, *an excess of resistance to rotation over rail adhesion.* The causes contributing to diminished rail adhesion or increased resistance to rotation are many and variable.

The considerable increase in train deceleration during the last few hundred feet before the stopping point is an evidence of the characteristic action of frictional resistance at constant pressure, when the speed of relative motion of the rubbing surface is being continually reduced from a relatively high value to zero. This phenomenon was first observed and commented upon in the report of the classical Galton-Westinghouse Brake Trials on the London, Brighton & South Coast Railroad in 1878 and 1879, and was at that time attributed to the effect of speed.

All subsequent experiments have developed this characteristic action. The present experiments as explained in the paper show that it is after all, a temperature effect, speed being one of the factors determining the resultant temperature of the working surfaces. This can be visually observed in the disappearance of sparking (showing a high temperature of the rubbing surfaces) and the appearance of non-incandescent brake shoe metal ground off toward the end of the stop. Invariably the cessation of sparks and appearance of the non-incandescent metal dust are coincident with the beginning of the rise in the deceleration line referred to by Mr. Vaughan.

The higher rise of the train deceleration line than that shown for the test of individual brake shoes at the test plant may be due in part to whatever different temperature conditions existed in the two cases. It may also be a result of the different methods of obtaining the curves in the two cases. The laboratory records are autographically produced by the mechanism of the testing machine itself. To avoid excessive fluctuations due to inertia and impact effect the action of the automatic machanism must be more or less dampened, so that absolute accuracy of the final value indicated is scarcely to be expected nor is it considered significant. The train deceleration curves are plotted from track chronograph records and owing to the extremely slow speed and rapid increase in deceleration of the train during the last few feet of its motion the final deceleration values plotted can claim no more accuracy or significance than is the case of the testing machine record. The uncertainty due to these methods tends in opposite directions, however, so as to accentuate the contrast referred to by Mr. Vaughan.

The question of complication mentioned by Mr. Vaughan and Mr. Thomson depends largely on the functions demanded of the brake. If simple and moderate functional performance can serve the purpose, complication of apparatus can largely be dispensed with. But if a multiplicity of functions, a high degree of protection against undesired action, and a high efficiency and effectiveness are demanded a certain amount of elaboration of the apparatus is necessarily implied. Mr. Turner has pointed out the requirements of modern railroad service which are constantly demanding more and more of the brakes along these lines.

Mr. Thomson is quite correct in so far as the obtaining of the highest effectiveness from a given air brake design is concerned. This is but one of many channels along which improvement is necessary if we are to meet the demand of modern railroad service to the highest possible degree.

Common experience has shown that if a more elaborate mechanism is necessary for a given purpose the questions of manipulation and maintenance referred to by Messrs. Vaughan, Langan and Thomson will take care of themselves.

It might be suggested here that the impression of complication is heightened by the multiplicity of lines required to illustrate the sequence and connections of the working parts of the valve device diagrammatically and as if all in one plane. This by no means indicates an undue complexity of the devices itself however, partly because in the actual construction the connections can be made much more direct and simple than on a single flat surface and partly because ports and passageways once properly fixed in the metal do not create in fact the working complication suggested by the lines on the drawings.

The New York, Westchester & Boston tests mentioned by Mr. Potter are probably the best comparison of the action of one and two brake shoes per wheel which we have on record up to date. This is because the rigging effect was practically eliminated by reason of the use of the same rigging (with proper adjustment) for the single shoe tests as was used for the tests with two shoes per wheel. Thus, the tests fairly compare the action of one shoe with that of two shoes per wheel, without introducing the additional uncertainty of the action of two different types of foundation brake gear which is always present when testing differently designed riggings for clasp and for single shoe brake equipment.

With reference to the cause of the variation in train records mentioned by Mr. Sargent we would refer to page 76 where it is shown that the length of stop varied from 1049 ft. to 1389 ft. No assignable cause for this variation has been discovered aside from the known difference in brake shoe bearing and shoe temperature. It therefore seems fair to assume that the different shoe metal conditions resulting from the different manipulations of the test train were capable of producing the differences observed. Moreover, by the same manipulation the same variation in results could be reproduced at will.

The data submitted by Mr. Burton on braking force and wheel pressures disclose conditions of the greatest importance and significance. This phase of the situation deserves the careful consideration of all having to do with the design and maintenance of brakes and foundation brake rigging.

The per cent difference in stopping distance deduced by Mr. Burton requires a word of caution to prevent misunderstanding or misuse.

For electro-pneumatic brake operation the almost instantaneous application of the brakes makes it permissible to assume, as Mr. Burton has done, that the stops will vary inversely as the product of $e \times f_s$. But on account of the longer time occupied in reaching full effectiveness with pneumatic operation and especially if the performance of the PM brake equipment is being considered ($t = 2.5$ for PM equipment) more or less error is involved if the same assumption is made in such cases. The stops with the PM brake equipment, for instance, would all be longer than those on which the table was based and the influence of the time element (t in the formula given in the paper for length of stop) could no longer be disregarded as is the case when the stops are assumed to be inversely proportional to $e \times f_s$.

It is particularly gratifying that the discussion has added the force of unquestioned authority and experience to the main thesis of the paper, namely, that the performance of brakes on railroad trains is no longer to be accepted as a function of the air brake device alone. With respect to facility and economy as well as safety, the truck design, the design and application of the foundation brake gear and the characteristics of the brake shoes must be given consideration as equally potent factors.