THE DESIGN AND OPERATION OF LOCOMOTIVE LABORATORIES

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THIS IS TO CERTIFY THAT THE THESIS PREPARED UNDER MY SUPERVISION BY

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INTRODUCTION

From the earliest introduction of the stationary steam engine into commercial use, its testing became of great importance. Any ratings obtained on the engine in industrial service for coal or water consumption or other elements of its performance must, however, be only approximate and unsatisfactory when made on the engine running under actual operation conditions, because of the continual variations encountered in load, speed, boiler pressure, and firing. The testing of a locomotive on the road is manifestly much more difficult and much less satisfactory in its results; hence a means of testing locomotives under fixed and determinate conditions becomes of even greater value, and such means - the testing plants heretofore constructed - have added greatly to the bulk of knowledge concerning the performance of the locomotive engine.

While these conditions were early recognized in the road-testing of locomotives, the problem of a method of making stationary tests remained unsolved until 1881, when Alexander Borodin devised a method for stationary running, so that the action of the engine might be the more closely observed. There are on record, however, instances of the running of the wheels without propulsion along the track, such as belting the drivers to shop machinery, or greasing the track so that the drivers might be slipped in order to operate the old style feed pump. It remained for Mr. Borodin, Chief Engineer of the Russian Southwestern Railway, to conduct tests by this means.

He mounted small locomotives in turn at right angles to the wall of a machine shop with the driving wheels blocked clear of the rails, and belted to the countershaft of the shop. The load that could be thus imposed was about ninety horse-power, and the capacity of the apparatus was therefore very limited, thirty per cent. cut off and one hundred pounds of steam pressure being the maximum which could be used. With this simple and not very convenient arrangement he succeeded in conducting a very profitable and instructive series of tests on simple, compound and steam jacketed engines for comparative purposes. He thus anticipated the first practical plant by nearly ten years.

To Dr. W. F. M. Goss belongs the honor of developing the far more substantial and satisfactory arrangement characteristic of the first plant operated in this country, and it is only fair to state that this work was done with no knowledge of the previous experimenting.

For further information concerning the work of Borodin see "Locomotive Testing Plants", (Trans. A.S.M.E., Vol. XXV, p. 827)

The Purdue University Testing Plant.

Entirely independent of this early work of Borodin was developed the plant which was destined to do the pioneer work in testing locomotives under laboratory conditions, - the Locomotive Laboratory of Purdue University, at Lafayette, Indiana. In 1890 there was installed at that school a large Harris-Corliss engine completely fitted up for testing purposes, and at the same time the advantages of having a locomotive similarly arranged presented themselves.

Accordingly, in May 1891 an order was given to the Schenectady Locomotive Company for an eight-wheel engine with seventeen by twenty-four inch cylinders, for experimental purposes. This engine was delivered in September of the same year, and during the intervening time the details of the mounting were designed and put in place ready for the engine's reception. Professor Goss ("Locomotive Performance" p. 6) thus summarizes the Purdue plan of mounting, which in its essentials has been adopted in all testing plants built since that time:

"The plan of mounting, in its inception, involved (1) supporting wheels carried by axles mounted in fixed bearings, (2) brakes which would have sufficient capacity to absorb continuously the maximum power of the locomotive, and which should be mounted on the axles of the supporting wheels; (3) a Traction Dynamometer of such form as would serve to indicate the horizontal moving force and at the same time allow but slight horizontal motion of the engine on the supporting wheels."

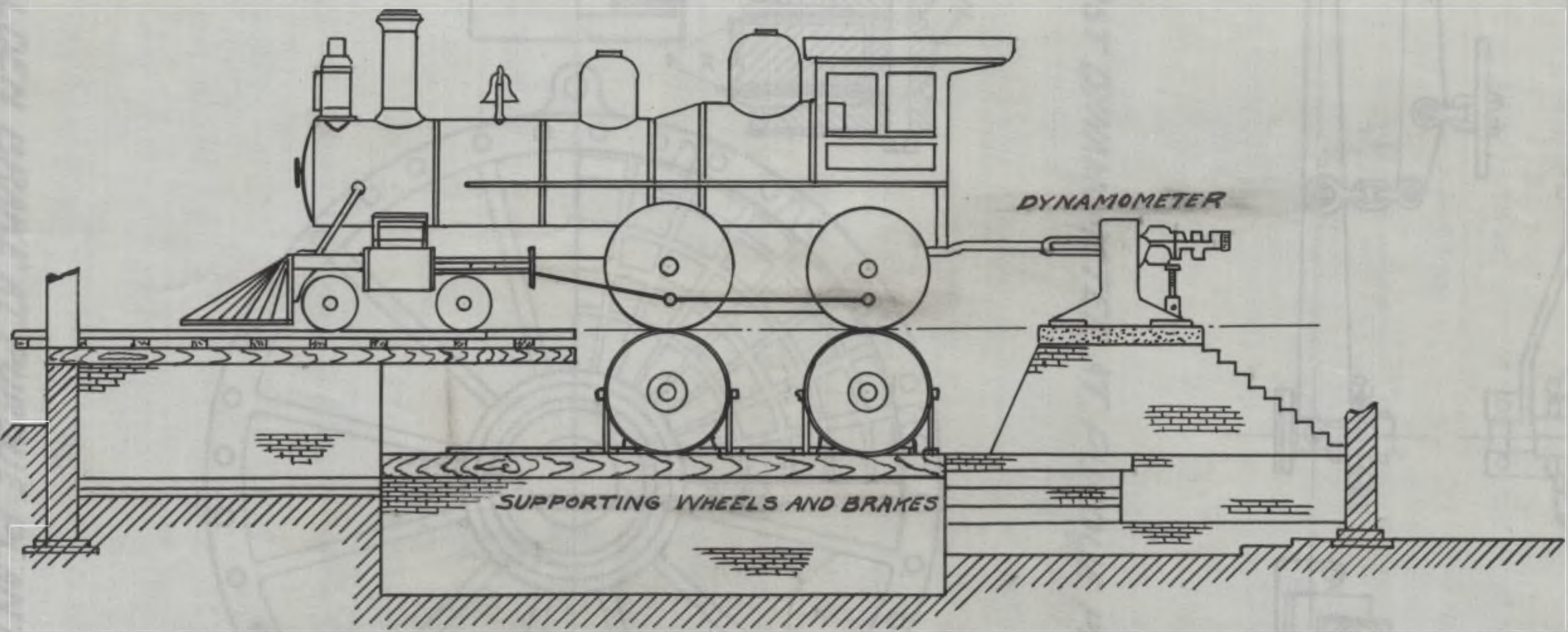

FIG. 1 LOCOMOTIVE TESTING PLANT AT
PURDUE UNIVERSITY - LAFAYETTE IND.
SECOND DESIGN

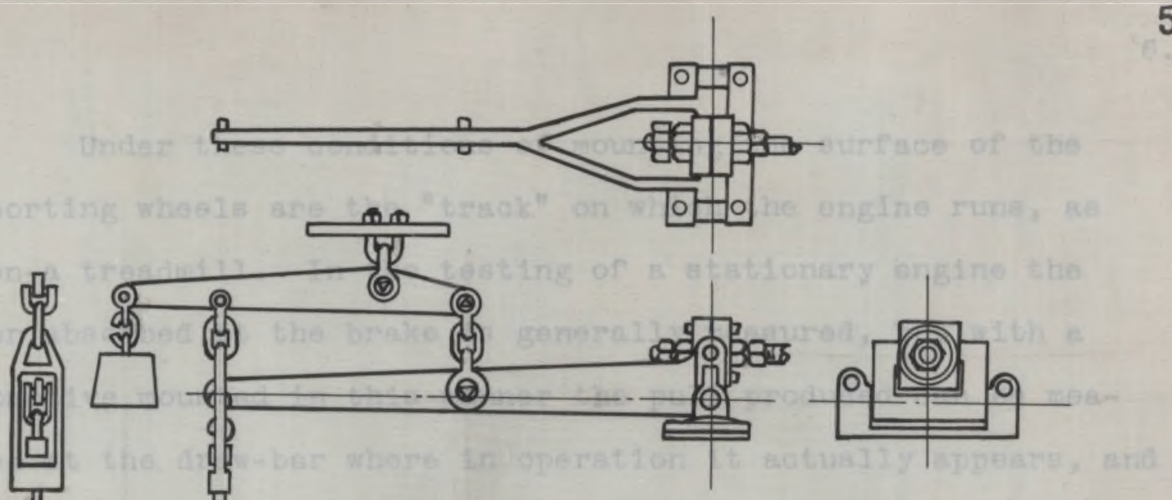

FIG. 3 FIRST DYNAMOMETER AT PURDUE PLANT

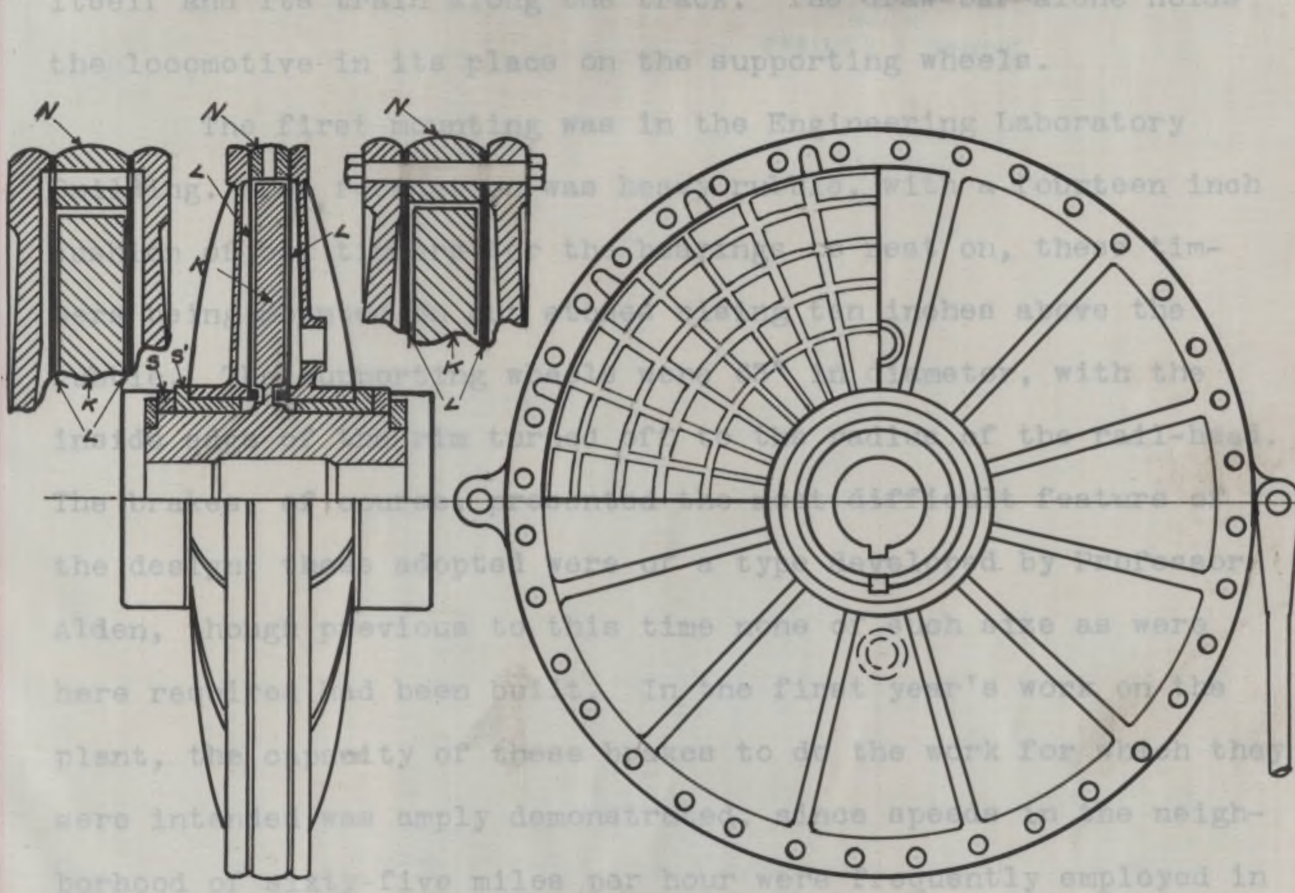

FIG. 2 ALDEN BRAKE AT PURDUE PLANT

Under these conditions of mounting the surface of the supporting wheels are the "track" on which the engine runs, as if on a treadmill. In the testing of a stationary engine the power absorbed at the brake is generally measured, but with a locomotive mounted in this manner the pull produced can be measured at the draw-bar where in operation it actually appears, and the varying retarding force which may be applied at the brakes merely represents varying degrees of difficulty in propelling itself and its train along the track. The draw-bar alone holds the locomotive in its place on the supporting wheels.

The first mounting was in the Engineering Laboratory Building. The foundation was heavy rubble, with a fourteen inch cushion of oak timbers for the bearings to rest on, these timbers being mounted on cut stones rising ten inches above the rubble. The supporting wheels were 63" in diameter, with the inside edge of the rim turned off to the radius of the rail-head. The brakes, of course, presented the most difficult feature of the design; those adopted were of a type developed by Professor Alden, though previous to this time none of such size as were here required had been built. In the first year's work on the plant, the capacity of these brakes to do the work for which they were intended was amply demonstrated, since speeds in the neighborhood of sixty-five miles per hour were frequently employed in the tests.

The essential features of the Alden brake are as follows: Mounted on and keyed to the supporting axle, and hence turning with it is the cast iron disc K (See Fig. 2). The power

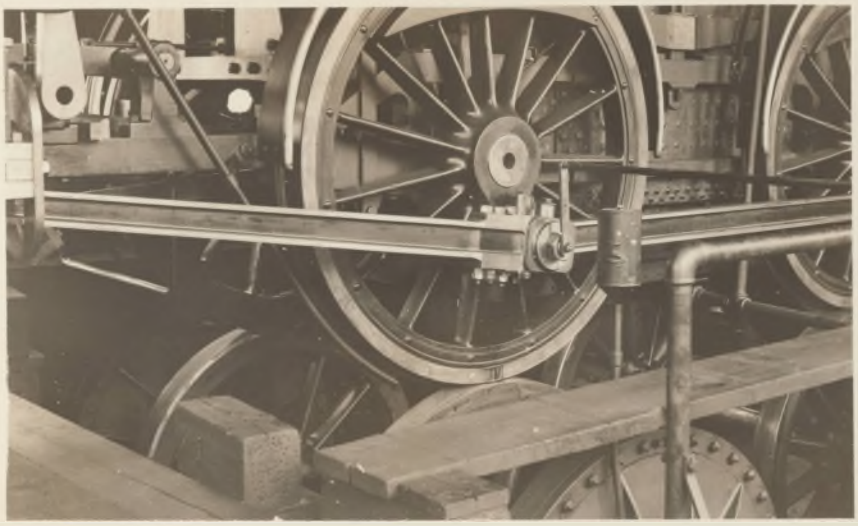

Fig. 4. Mounting at the Purdue Plant.

Fig. 5. Brake and Supporting Wheel,
Purdue Plant.

of the locomotive is transmitted to this disc, on either side of which are the thin copper plates L, forming a part of the enclosing case. The sides of the case bear outward on a collar attached to the hub of the revolving disc; the two sides are connected by the ring N, being held together by through-bolts. The rings S and S' are fitted to the hub of the revolving disc over a feather to prevent the water pressure from forcing the two sides of the case outward. The space between the copper plates and the cast iron disc is filled with oil when the brake is in use. The uniform distribution of the oil is secured by cutting radial and spiral grooves in the face of the disc; the surface is thereby cut up into sections, the largest of which are about four inches each way. Oil which is forced out at the top of the brake case by the pumping action of the grooves is filtered and delivered at the center by circulation pipes. As a result of preliminary experimenting with a smaller brake it was found that lubrication could be maintained under a water pressure of forty pounds per square inch, giving a coefficient of friction of about 3.5 percent. The dimensions chosen for the rubbing discs were twenty-eight inches outside diameter and ten inches inside diameter. (For method of calculation and equations see Part II, p. 53).

The Traction Dynamometer consisted of a series of levers so designed that either a push or a pull would have the same effect at the weight end. (See Fig. 3). The levers had an overall ratio of 1 in 100, so that a ten-pound weight balanced 1000 pounds of draw-bar pull or push.

Much work was done with this plant in the school years of 1891-2 and '93-'94, both in making of mechanical improvements and in collecting test data, until the disastrous fire of Jan. 23, 1894, made a total wreck of the plant and laboratory.

Every item of experience and information gained was made to serve in the design of the new plant; only four months after the fire the work of rebuilding the plant was complete. The engine was meanwhile reconstructed at the Indianapolis shops of the Big Four Railway. The new plant occupies a building of its own, with very much greater convenience than a corner of the general laboratory afforded.

In rebuilding, provision was made for the accomodation of any locomotive which had been built up to that time, by making the foundation twenty-five feet long, allowing a driving-wheel base of eighteen feet and six inches. It was planned to add one or two more supporting axles and brakes complete when they should be needed. The new foundations were of brick and finished stone work; the long timber cushion having proved of doubtful benefit, was omitted. Mounted on the stone-work were heavy cast iron bedplates carrying the pedestals for the axle boxes. By means of slotted flanges on the pedestals and threaded holes in the bed-plates, provision is made for placing the supporting wheels to correspond with the spacing of the drivers of any locomotive. The anchor bolts of the brakes are secured to timbers lying along the edge of the foundation. The boxes used are babbitted shaft-bearings; there is a wooden cushion between the bearing and its pedestal. In the fire, the brakes were uninjured, and the same set in use in the first plant are employed in the second.

The most important change in the general make-up of the plant was in the new dynamometer. The new pier is of massive brick construction, stayed with iron rods, and on this is mounted the dynamometer, which connects with the draw-bar at the rear of the locomotive. The pull transmitted to the draw-bar is received by the weighing head of an Emery testing machine, which is of 30,000 pounds capacity. The hydraulic support is capable of receiving this pull and also of withstanding the vibratory character of the stresses, and yet the instrument is sufficiently delicate in its registering to show the effect of a push on the pilot with the fingers. There is a ball joint connection between the apparatus and the engine draw-bar, and there is provision for raising and lowering the whole dynamometer-head on its frame.

Firing is done from the "tender-floor", the front of which is placed at a height corresponding to that of an actual tender deck: the back of this floor slopes off to the floor level of the coal-room, serving as a runway. The injectors of the locomotive draw their supply of water from calibrated tanks, and a steam pump is provided to keep up the circulation of cooling water in the brakes.

Views of this plant are shown on page 7, including a near view of the mounting and one of the right forward brake.

There have so far been four different engines on this plant: the University's first locomotive, "Schenectady No. 1", which was used until it ceased to be representative of good practice in 1897; "Schenectady No. 2", which at that time took its place; a New York Central Atlantic type passenger engine, on which a series of front-end tests were run in 1906; and the Strong Balanced Compound locomotive, which was thoroughly tested.

The Plants of the Chicago and Northwestern.

In 1893 there was a committee appointed by the American Railway Master Mechanics' Association for the purpose of investigating proper proportions for exhaust nozzles and steam passages. This committee arranged to make use of the plant at Purdue University in its tests, but the fire which destroyed the Engineering Laboratory there in January, 1894, forced the abandonment of this plan. The Chairman of this committee, Mr. Robert Quayle, (Master Mechanic of the C. & N. W. Ry.) erected temporary testing apparatus at South Kaukauna, Wisconsin, this same year, in order to carry forward the investigations.

Mr. Quayle utilized an ordinary four-wheeled passenger car truck for this purpose, lengthened out to equal the spacing of the drivers of the eight-wheel engine that was to be used on the plant. The flanges of the wheels were turned off, the whole truck secured to heavy timbers and sunk in a pit so that the tops of the supporting wheels were level with the rails. Braking was accomplished by means of two shoes on each of the four wheels, these being brought into action by a system of levers actuated by an ordinary air-brake cylinder, but in order that the braking force might be more easily maintained at a constant value, water in place of air was used in the cylinder. The cooling was done by jets of water which were so arranged as to play on the brakes,-

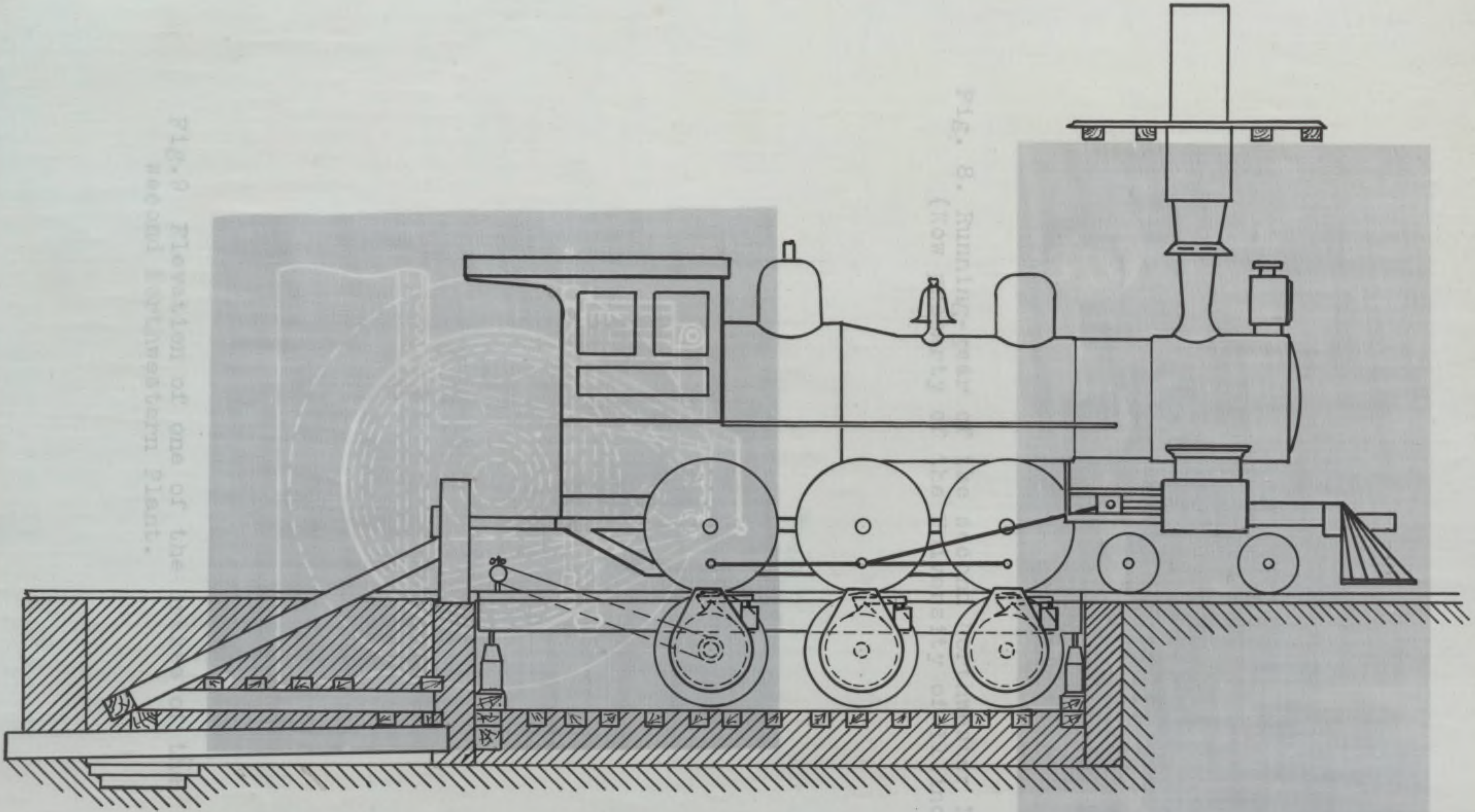

FIG. 7 LOCOMOTIVE TESTING PLANT OF THE
C. & N. W. R.R AT THE 40th. STREET SHOPS
CHICAGO ILL.

Fig. 8. Running-gear of the second Northwester Plant.
(Now property of the University of Illinois)

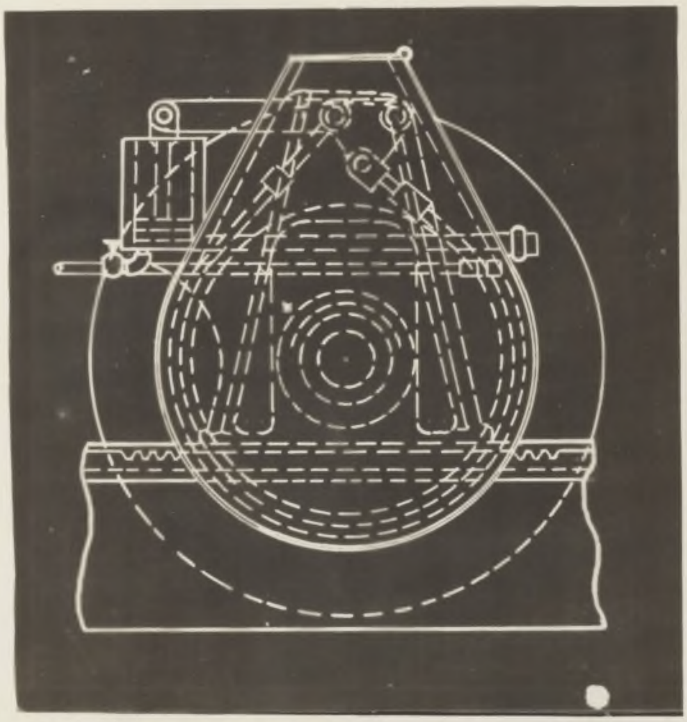

Fig.9 Elevation of one of the brakes of the second Northwestern Plant.

these being simply hose streams, throwing any amount of water that was found necessary to keep the brake-shoes cool. Since no measurements of draw-bar pull were desired it was only necessary to provide sufficient bracing to prevent any front or backward motion of the engine. It was found that when running the heavy bracing provided was insufficient, and the fastenings were in danger. To remedy this, a second locomotive was backed against the test engine: this second one, with the brakes set, proved an effective stop. The locomotive was operated at maximum power and at speeds as high as forty miles per hour satisfactorily. The cylinder power was measured and the values of draft and back pressure were determined for various arrangements of the front-end parts.

The next year Mr. Quayle became Superintendent of Motive Power of the C. & N. W., and a more permanent plant was set up in the roundhouse at West Fortieth Street, Chicago. Having a much broader scope and purpose than the previous plant, the new plant had far greater capacity, and the supporting wheels were arranged for adjustment to the wheels of any four- or six-driver locomotive. An elevation of the plant, and also a detail of one of the brakes are shown in accompanying drawings. (Figs. 7 and 9).

The plant occupied a stall of the roundhouse, and adjacent tracks were available for cars of coal and water. Permanent calibrated water tanks soon replaced the tank-car, and the tender of the engine was generally used on the coal track. There was a platform extending from the tender or coal car to the foot-board of the locomotive, over which the coal was handled in barrows, and also weighed. The pit in which the testing-gear was mounted was fifteen feet wide and twenty-eight feet long. The bottom was ballasted with gravel; in this were laid longitudinal timbers 16" x 16" x 25' long carrying the bed-plates, over 8" x 10" x 14' cross-ties. The bearings were attached to the bed plates by bolts working in tee-slots, and moved along by racks and pinions. The brakes were of the band type. The brake bands were one-fourth inch thick by five and one half wide; the brake-drums were 33" cast iron wheels with chilled tires, and the wheels and brakes were applied by air acting in six-inch cylinders, and the braking force was measured by a system of levers.

The supporting wheels were old fifty-six inch drivers with plain steel tires, the inside of the rim being turned off to the same radius as the head of a rail. There were but three sets of the wheels, though provision was made for a fourth set. All of the piping was so arranged as to facilitate any change which might be desirable in the spacing of the supporting wheels. The journals were 8" x 16"; a packing-box was provided to keep the cooling water from getting into the journals.

The engine ran onto the plant on its flanges on a temporary track consisting of I-beams. This temporary track was removed when the engine's drivers were above the supporting wheels, letting it down into place. The track was raised or lowered by air pressure working through a cylinder and a system of levers.

This plant served its purpose for a time, and then the space it occupied in the roundhouse being needed for other purposes, it was dismantled and stored. The running-gear is now the property of the University of Illinois, and a picture of it is here presented. (See Fig. 8).

The Plant of Columbia University.

In 1893 the Baldwin Locomotive Company sent to Chicago as a part of its exhibit at the Columbian Exposition the locomotive "Columbia", a high speed passenger engine of an entirely novel type. The wheel arrangement was "two-four-two" -- that is a two wheeled leading truck, four drivers, and a pair of trailing wheels. The engine weighs in working order 126,600 pounds, and is of the Vauclain Coumpound type. In 1899, under the condition that it should be mounted in some suitable manner, this engine was presented to Columbia University, of New York City. A testing plant after the general "Purdue plan" was accordingly built on the campus and the locomotive set up on it, it having been necessary to dismantle the engine in moving it into place, as the University is remote from railroad connections.

The pit for the supporting wheels in the case of this plant is made only sufficiently long enough to accommodate two axles, as it was deemed entirely unlikely that any other engine would be used on the plant. The supporting wheels are sixty inches in diameter. The locomotive is coupled to a buffing post to prevent its forward or backward motion. The power in this plant is absorbed by four Alden brakes, each with a capacity of four hundred horsepower. The force at the drawbar is measured by a system of weighing levers; the calibration is such that the amount of power absorbed in the journals of the engine is corrected for, and the scale readings give the total power of the cylinders. No traction dynamometer is included. The smoke duct is provided with a steam-driven Sturtevant Blower; other arrangements are usual -- calibrated weighing tanks, coal scales and platforms, etc. The locomotive is also arranged so that the boiler may be supplied with steam from a stationary boiler in the building.

The Columbia plant, while is serves a useful purpose in the laboratory, is unfortunately situated in that in its locality any avoidable smoke is intolerable, and for this reason the plant has been run very little.

"Locomotive Testing Plants" -- Proc. A.S.M.E., Vol. XXV, p. 855.

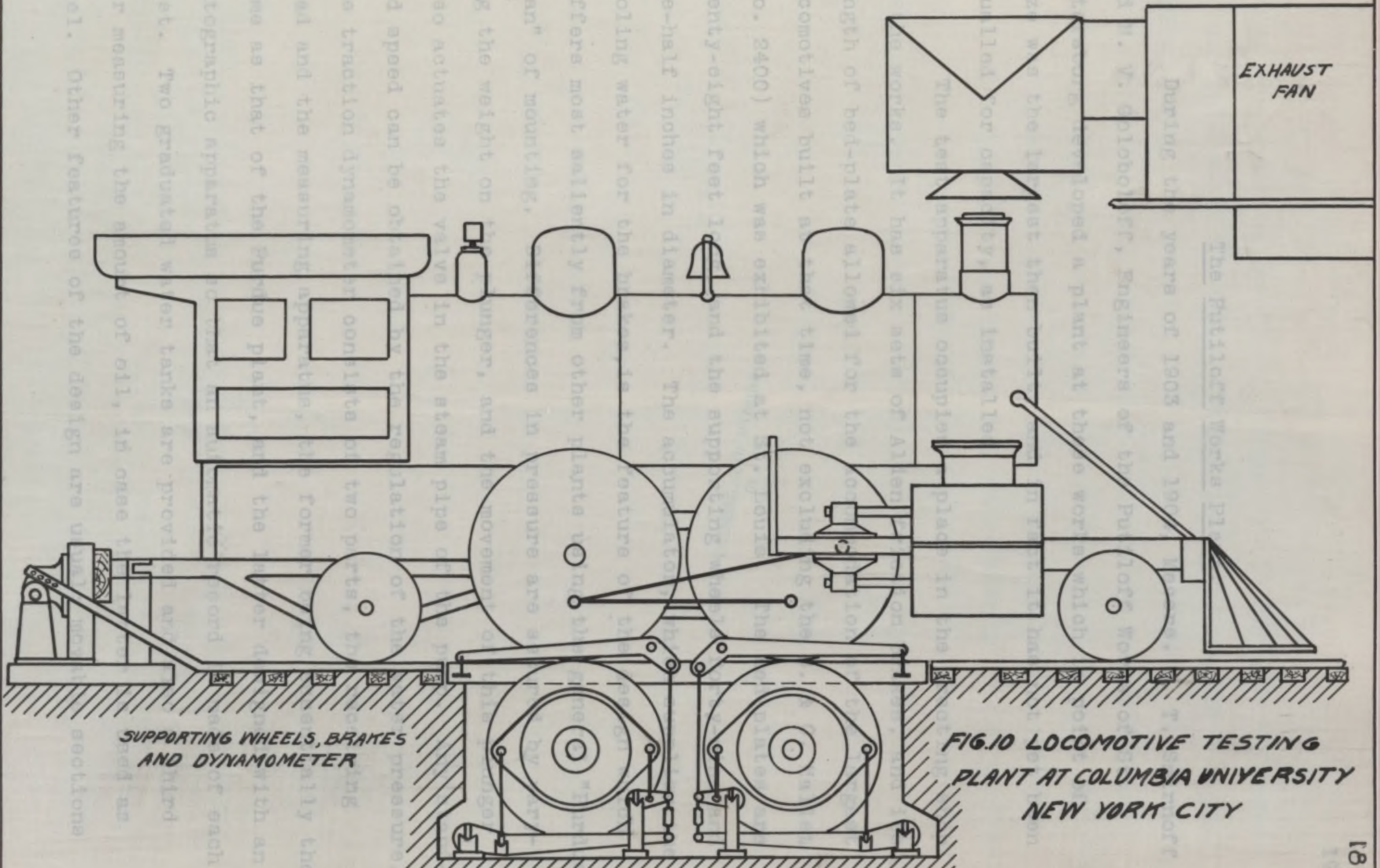

FIG.10 LOCOMOTIVE TESTING PLANT AT COLUMBIA UNIVERSITY NEW YORK CITY

The Putiloff Works Plant.

During the years of 1903 and 1904, Messrs. S. T. Smirnoff and M. V. Goloboloff, Engineers of the Putiloff Works of St. Petersburg developed a plant at these works which in point of size was the largest then built, and in fact it has not yet been equalled for capacity, as installed.

The test apparatus occupies a place in the erecting shop of the works. It has six sets of Alden friction brakes, and its length of bed-plate allowed for the accommodation of the largest locomotives built at that time, not excluding the B. & O. Mallet (No. 2400) which was exhibited at St. Louis. The bed-plates are twenty-eight feet long, and the supporting wheels forty-nine and one-half inches in diameter. The accumulator, which supplies the cooling water for the brakes, is the feature of the design which differs most saliently from other plants using the general "Purdue plan" of mounting. Differences in pressure are secured by varying the weight on the plunger, and the movement of this plunger also actuates the valve in the steam pipe of the pump. Any standard speed can be obtained by the regulation of the water pressure. The traction dynamometer consists of two parts, the receiving head and the measuring apparatus, the former being essentially the same as that of the Purdue plant, and the latter designed with an autographic apparatus so that an automatic record is made of each test. Two graduated water tanks are provided and also a third for measuring the amount of oil, in case the latter is used as fuel. Other features of the design are usual; movable sections of track between the supporting wheels lifted by jacks and taper-blocks; a telescopic steel stack, and a coal-chute from a large hopper outside the building to the weighing bucket.

The first published results obtained on this plant were printed in 1910, concerning tests of similar locomotives with and without brick arches in the fire-box. Diligent search fails to reveal any other publication of work done here, or of the realization of the plans of the Russian State Railways or of the South-western Railway of Russia to build plants of a similar nature.

"Locomotive Testing Plants" (Proc. A.S.M.E., Vol. XXV - p.866)

"Bulletin of the Int. Ry. Congress" (Nov. 1910 - p. 292)

The Pennsylvania Plant.

In 1904 the Pennsylvania Railroad exhibited at the Louisiana Purchase Exposition at St. Louis a testing plant which in its turn surpassed even the Putiloff plant for flexibility and capacity. In the few months in which this plant was in operation at St. Louis an immense amount of work both in volume and variety was accomplished, and this work has been steadily continued in the new location -- Altoona, where it was moved after the close of the exposition. Eight representative locomotives were tested: both passenger and freight types, as follows:

TYPE	CHARACTER	ROAD	BUILDER
2-8-0	Simple Consolidation	Pa.R.R	Juniata Shops Pa.R.R.
2-8-0	Simple Consolidation	L.S.&M.S.	Brooks Works.
2-8-0	Cross-Compound Consolidation	M.C.R.R.	American Locomotive Co(Schenectady)
2-10-2	Tandem Compound Santa Fé	A.T.&S.F.	Baldwin Works.
4-4-2	De Glehn Balanced Compound	Pa.R.R.	Societe Alsacienne de Construction Mecaniques, Belfort France
4-4-2	Balanced Compound	A.T.&S.F.	Baldwin Works.
4-4-2	Cole Balanced Compound	N.Y.C.	American Locomotive Co.
4-4-2	Balanced Compound, Superheater	Prussian State R.R.	Han. Mach. Act. Gesellschaft.

The Pennsylvania Plant was designed by Mr. A. S. Vogt, Mechanical Engineer of the road, under the direction of Mr. A. S. Gibbs, General Superintendent of Motive Power. The following description applies specifically to the installation at St. Louis, but no important changes have been made in the arrangements at Altoona.

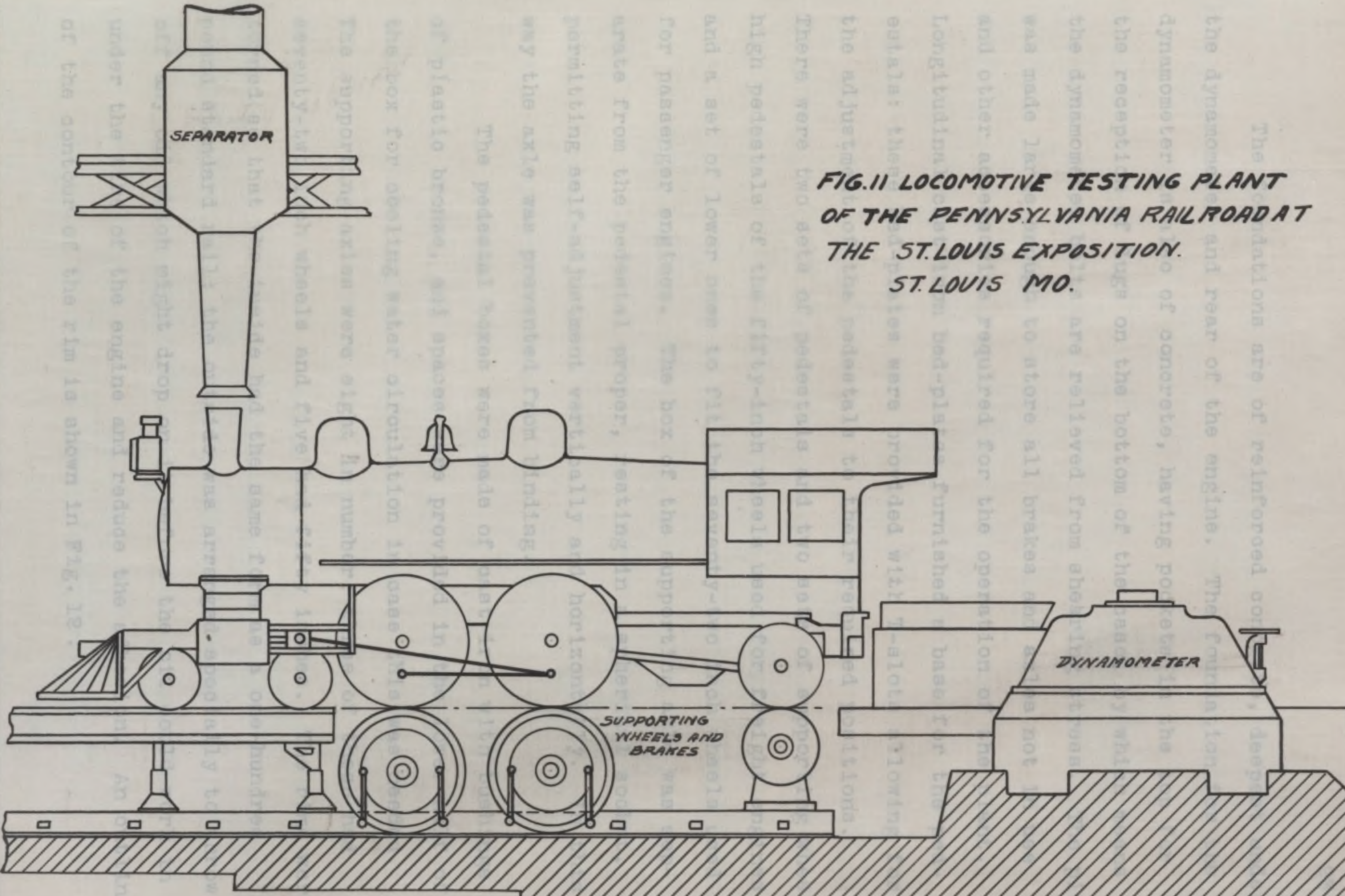

FIG. 11 LOCOMOTIVE TESTING PLANT OF THE PENNSYLVANIA RAILROAD AT THE ST. LOUIS EXPOSITION. ST. LOUIS MO.

The foundations are of reinforced concrete, deepest under the dynamometer and rear of the engine. The foundation for the dynamometer is also of concrete, having pockets in the top for the reception of lugs on the bottom of the case, by which means the dynamometer bolts are relieved from shearing stress. The pit was made large enough to store all brakes and axles not in use and other accessories required for the operation of the plant. Longitudinal cast iron bed-plates furnished a base for the pedestals: these bed-plates were provided with T-slots allowing for the adjustment of the pedestals to their required positions. There were two sets of pedestals and two sets of supporting wheels: high pedestals of the fifty-inch wheels used for freight engines and a set of lower ones to fit the seventy-two inch wheels used for passenger engines. The box of the supporting axle was separate from the pedestal proper, resting in a spherical socket, permitting self-adjustment vertically and horizontally. In this way the axle was prevented from binding.

The pedestal boxes were made of cast iron with bushings of plastic bronze, and spaces were provided in the lower half of the box for cooling water circulation in case this was needed. The supporting axles were eight in number; three of these had seventy-two inch wheels and five had fifty inches. The rims were turned so that the inside had the same form as a one-hundred pound standard rail; the outside was arranged specially to throw off any oil which might drop on it before the oil could work in under the tire of the engine and reduce the adhesion. An outline of the contour of the rim is shown in Fig. 12 .

The front of the engine on the plant was supported by a heavily braced extension track, and the trailing wheels of such engines as had them were carried by a supporting axle and pair of wheels. The locomotive was run on the plant by means of a temporary removable

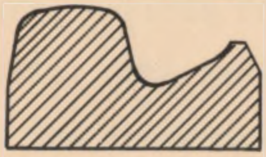

Fig.12

track, the level of which was one-fourth of an inch below the tops of the supporting wheels. The temporary track carried the locomotive on its flanges, as shown in Fig. 13. Therefore when the tire proper passed over one of the supporting wheels a portion of the weight was lifted from the track, and when all the drivers were in place over their respective supporting wheels all weight was removed from the track, and the latter could be lowered.

The engine is held in its place on the plant by the draw-bar only, and to guard against possible accident a pair of safety bars attached to the housing of the dynamometer are added: these connect with the safety bars of the locomotive. It was soon found necessary to provide these bars

Fig. 13.

with dash-pots to damp the fore and aft vibration of the engine, transmitting to the weighing mechanism the pull from the steam pressure but protecting it from the sudden reversals of stress brought about by the vibration at the higher speeds. Provision was made for raising or lowering the drawbar by means of a hand-wheel and bevel gears.

The traction dynamometer was built by William Sellers & Co.

and had a capacity of 80,000 pounds, either for pull or for push. By changing springs this capacity could be reduced to 40,000 or 20,000 pounds, in order to get more satisfactory records from the autographic attachment on lighter draw-bar pulls. The recording pen took its motion from one of the levers of the dynamometer and the paper was driven by a shaft and bevel gears actuated by the rear supporting axle, the recording table being attached to the dynamometer case.

Counterbalance tests were made by the method developed by Dr. Goss: a fine wire was run between the driver of the engine and its supporting wheel, and the points of maximum and minimum pressure determined by the changes in the cross-section of the wire. A means was also provided to record the amount of "nosing"; a pencil was attached to the left side of the pilot recording the motion from side to side at that point on a travelling roll of paper.

The plant as set up at Altoona after the exposition does not differ in any essential detail from the arrangement as described. Small changes have been made in the dynamometer, and the study of sparking and front end performance have been facilitated by the use of an attic above as a cinder separator.

The Swindon Plant.

The only locomotive testing plant in Great Britain is in the shops of the Great Western Railway at Swindon. This plant was designed and has been operated under the direction of Mr. G. W. Churchward, Chief Mechanical Engineer of the road. It has been used with a dual purpose: the actual testing of engines, and the "breaking in" of new or newly repaired engines. To this end all of the wheels are supported on carrying wheels, and the truck carriers are belted to those of the drivers, so when the drivers are started all the wheels rotate. The power developed at the drivers is used to drive a large air-compressor.

There are five pairs of supporting wheels, with room for the addition of a sixth pair if needed at any time. The bed-plates are of cast iron, set on heavy timbers in order to decrease as much as possible the vibration. Adjustable idler pulleys keep the tension in the belt at the required value, and the load on the brakes can be increased by means of a set of band brakes, one on each driver axle. The brakes are applied by hydraulic pressure, and they are boxed in with sheet iron to hold cooling water. The supporting axles may be adjusted to any desired position by means of a rack and pinion which is driven by a motor, a very great convenience, since the engines are frequently changed.

The locomotive runs on to the plant on its flanges; when the proper position above the wheels is reached, it is blocked from forward or backward motion by means of a set of hand screws, and the temporary rails are removed, bringing the weight on the

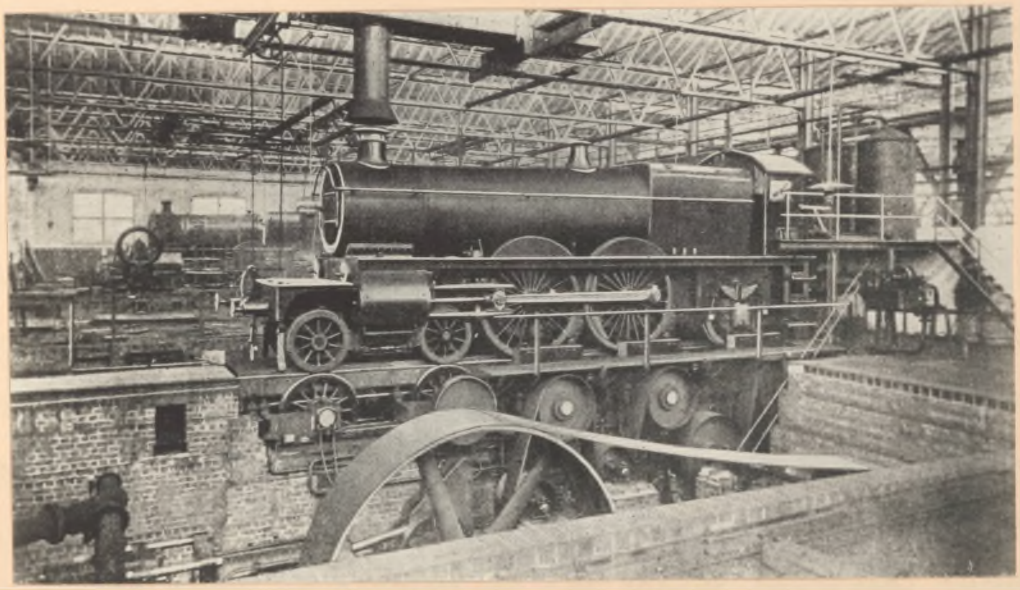

Fig. 14. The Testing Plant of the Great Western Railway, Swindon, England.

wheels. Hydraulic jacks raise and lower the flange rails. All the supporting wheels have a diameter of four feet.

An adjustable steel stack is provided, which is so boxed in that a rudimentary but effective cinder catcher is provided at the same time as smoke disposal. The usual facilities are furnished for measuring water consumed and weighing the coal burned. The pull at the drawbar is measured by a system of levers, the last of which is a greduated steelyard.

Very interesting is the report of a visitor who watched the progress of a test on this plant. (In 1905). The engine under test was of the "Atlantic" or 4-4-2 type, and it was running at speeds up to seventy miles an hour. The visitor remarks on the great amount of noise, on the lack of motion in the engine's spring rigging, due to the fact that the "track" was practically perfect; also on the "nosing" tendency at the high speeds and extreme fore-and-aft oscillation at all speeds over about forty miles an hour. He concluded that no one would wish to run a test at any higher speed on account of the danger involved, though this has since been done.

"The Testing Plant at Swindon" -- (The Engineer, London, Dec. 22, 1905).

The Illinois Plant.

Since the establishment of the Railway Department of the University of Illinois many comprehensive road tests of locomotives have been made. As early as 1903 the need of a more satisfactory means of investigation of some of the performance problems presented itself, but not until 1912 did the funds for the erection of a locomotive laboratory become available to the department. In July of 1912 construction was started, from the designs of Professor E. C. Schmidt; the plans give the plant the capacity to accommodate not only the largest and most powerful engines that have yet been built, but provide for future expansion in size and capacity to the extent of fifteen per cent beyond the limits of current practice.

The plant is located in a new building erected for the purpose, forty feet wide, 115 feet long, and twenty-two feet high under the roof trusses. The walls are of red faced brick; the floors throughout of reinforced concrete, and the roof of concrete covered with slate.

The supporting machinery of the plant is carried on a concrete foundation -- a slab ninety-three feet long and twelve feet wide, and varying in thickness from five feet at the rear to three feet at the front. At the rear is a pyramidal base which supports the dynamometer and anchors it. On this foundation are longitudinal bed-plates, each consisting of three sections placed end to end, 18" in height and 36" wide over all; the total length of the bed-plate being forty-two feet, allowing a maximum wheel-base of

Fig. 15. Foundation for the Supporting Mechanism and Dynamometer Pier, Illinois Plant.

Fig. 16. Completed Foundations for Supporting Mechanism and Dynamometer, Illinois Plant.

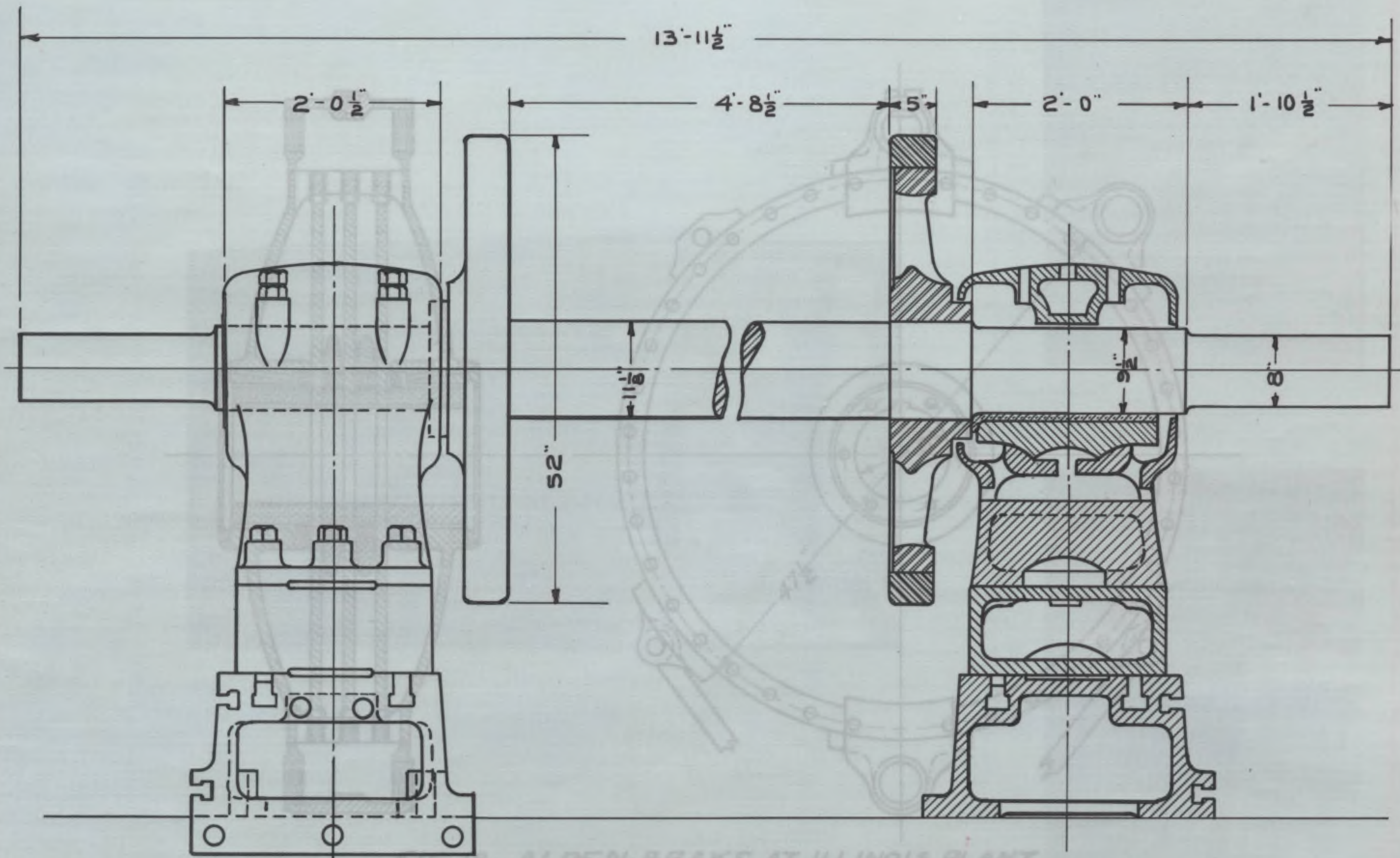

FIG. 17 SUPPORTING ELEMENT AT ILLINOIS PLANT

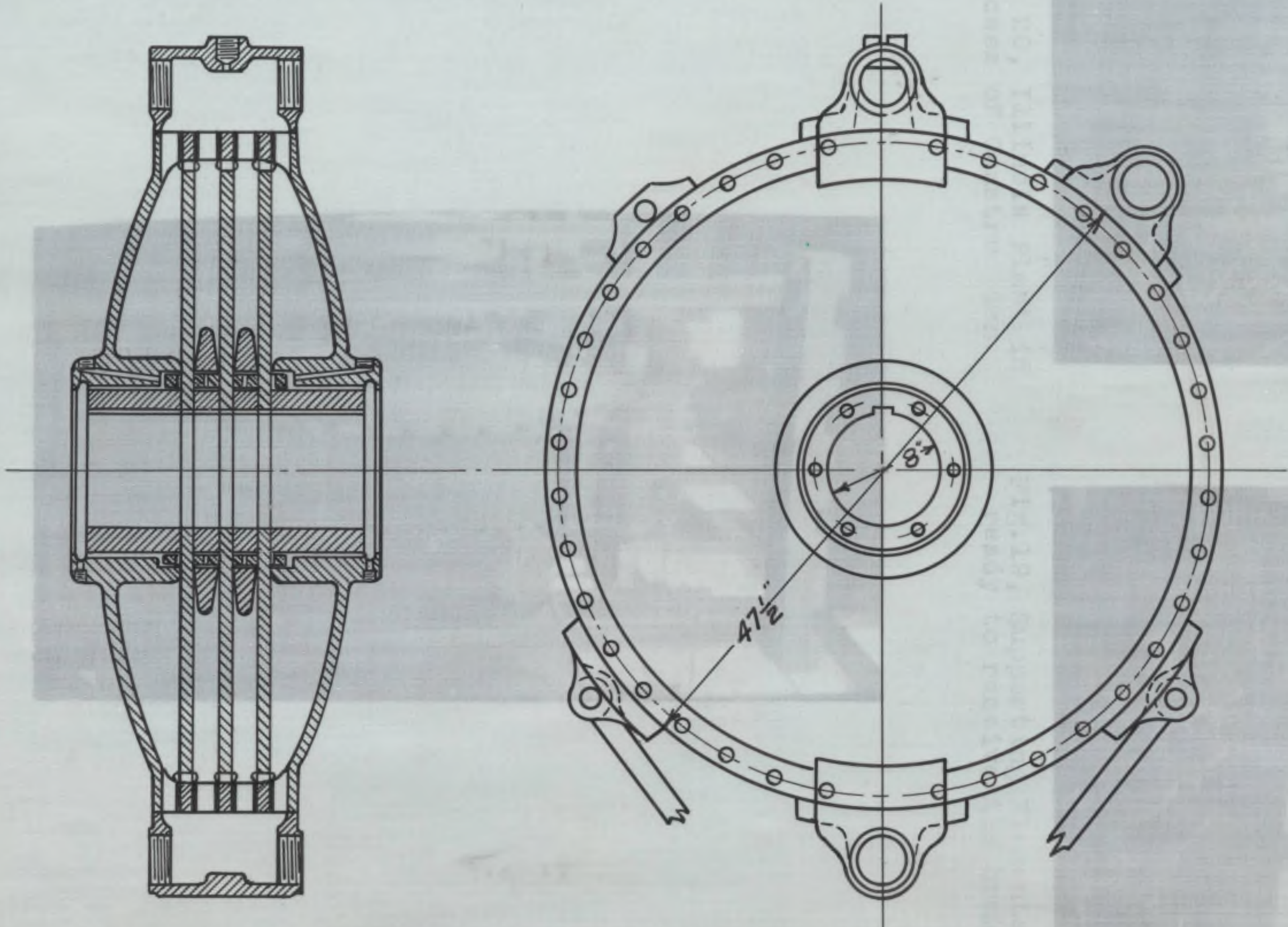

FIG. 18 ALDEN BRAKE AT ILLINOIS PLANT

Fig. 20, Illinois Plant in Process of Construction.

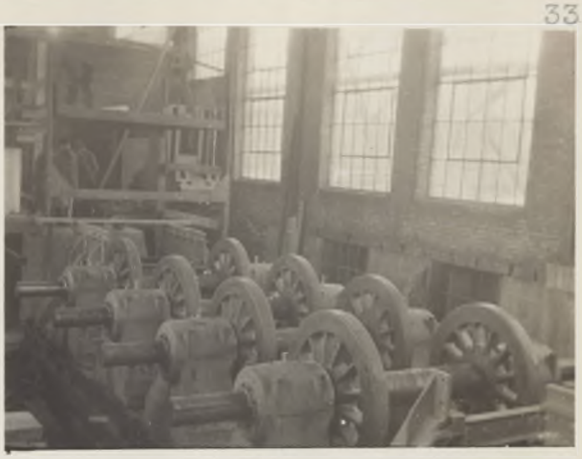

Fig.19, Supporting Elements ready to receive the brakes.

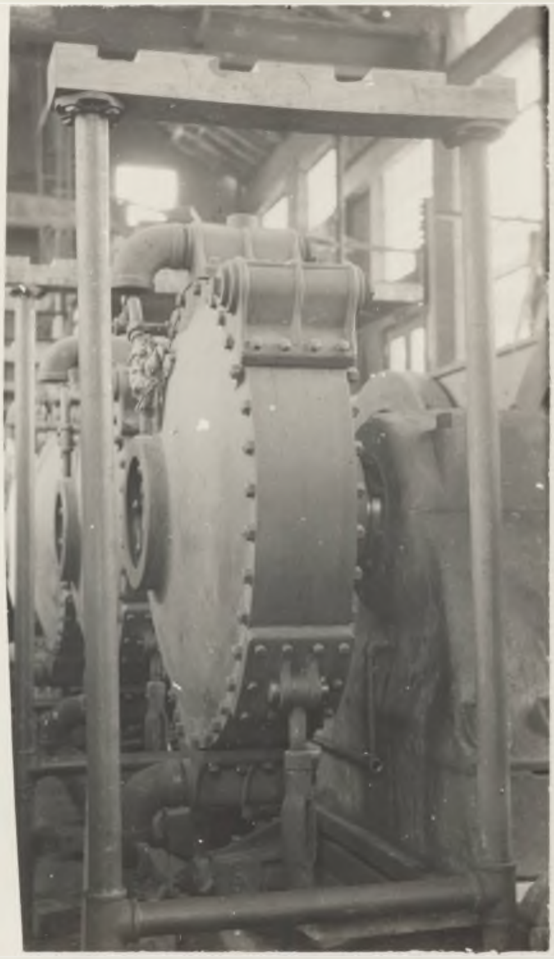

Fig. 21. Alden Brake, Illinois Plant.

thirty-six feet for the drivers. The foundation is built to receive two more 14-foot sections of bedplate. The top surface of the bed-plates is provided with two sets of T-slots, and also with grooves transversely placed to give a "bite" for the point of a bar, to be used in moving the pedestals to correspond to different wheel spacings. Figs. 15 and 16 show the work on the foundation at different stages, and the arrangement may be very plainly seen. In Figs. 15 and 16 the general shape and size of the bearing-pedestals is shown, and also in detail in the drawings, Fig. 17 Each pedestal consists of three parts: the pedestal proper which is made in two parts so that the height may be varied if in the future it becomes desirable to make use of a larger supporting wheel, and the bearing cap. The bearing itself is self-aligning. The journal is 9-1/2" x 20", the brass being provided for the under side of the journal only. Oil is fed in at two points in the bearing cap, under head from an elevated tank.

The supporting-wheels are made of heat-treated carbon steel. They have plain five-inch treads, are fifty-two inches in diameter, and are mounted on eleven-inch axles. In Fig. 17 is shown the general design of a complete supporting element, consisting of an axle, two wheels, and two bearing pedestals; in Fig.20 their general appearance, before the brakes were put in place. The brakes are of the Alden type, which has demonstrated its adaptability to the service required. (For description of the Alden brake, see under the Purdue plant, page 6). These brakes are eadh of 450 horse-power capacity; they are designed to receive and dissipate the energy of a torque of 18,000 pounds at one foot radius, at speeds as high as 130 revolutions per minute. This represents a greater duty than could be imposed by the most heavily loaded locomotive driver of the present day, and allows for considerable increase in the wheel loads. This set of brakes has six pairs of friction surfaces; there are three rotating iron discs, and three pairs of diaphragms pressed against them. At present there are provided four complete supporting units: pedestals, axles, wheels, and brakes. The present length of the bed-plates allows for the addition of four more units.

Fig. 22 shows the completed mounting apparatus, arranged for the reception of an eight driver locomotive, with the temporary track in place between the supporting wheels. The locomotive to be tested is backed on to the temporary track, on which it is carried by its flanges. It is run backward until the tops of the supporting wheels take the weight off of the track, when this latter is moved down out of the way. The locomotive is secured in its place by means of the heavy dynamometer draw-bar, supplemented by two safety bars which come into play in case of the failure of the draw-bar. Each of the three bars (draw-bar and two safety bars) is provided with a turn-buckle to vary its length. The front truck is carried by a section of track which is not removed, and immediately in front of the dynamometer there is provided also a short section of track for the trailing truck wheels.

The dynamometer is of the Emery type, and was built by William Sellers & Co., of Philadelphia. It consists essentially of three parts: the receiving apparatus, the weighing head, and the measuring and recording devices. The housing and receiving

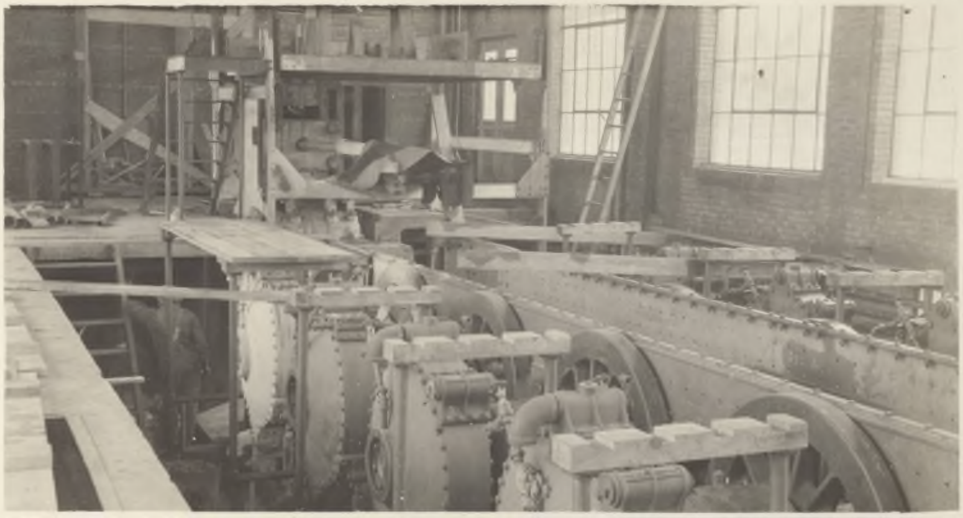

Fig. 22. Supporting Mechanism Complete, showing Wheels, Brakes, and the Removable Track in place.

Fig. 23. Interior View of Plant, arranged to test a Consolidation Locomotive.

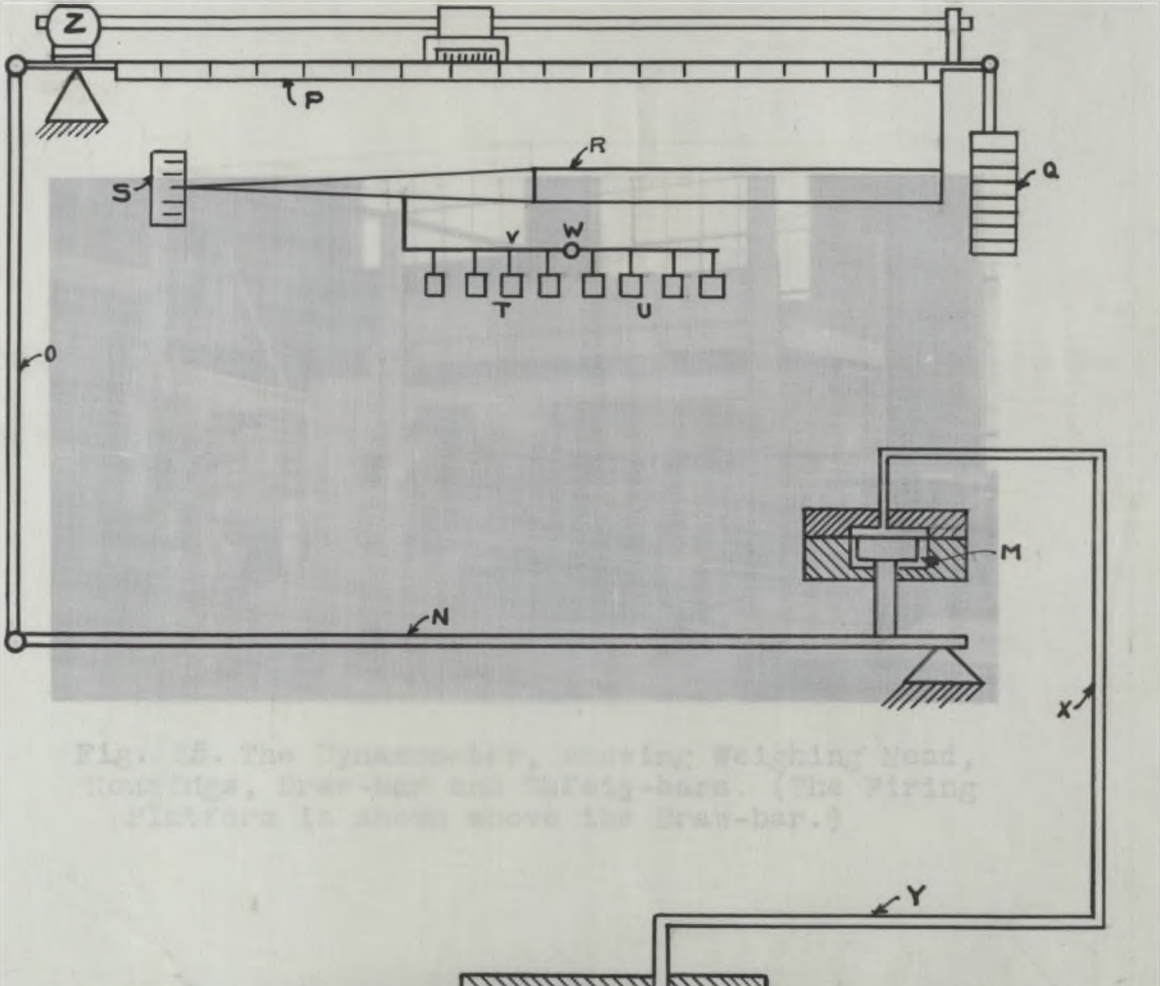

FIG. 24 DIAGRAMATIC SKETCH OF DYNAMOMETER AND SCALES-ILLINOIS PLANT.

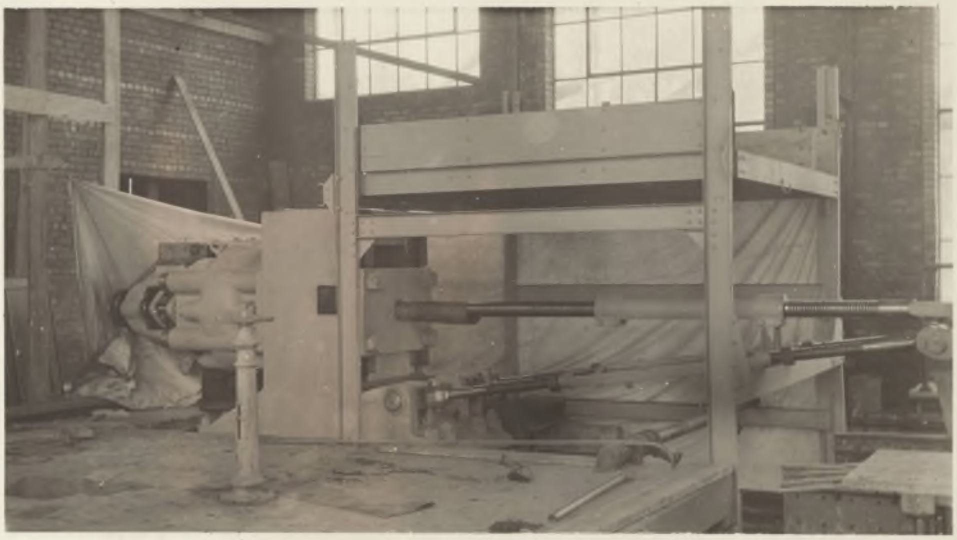

Fig. 25. The Dynamometer, showing Weighing Head, Housings, Draw-bar and Safety-bars. (The Firing Platform is shown above the Draw-bar.)

Fig. 26. Showing the Dynamometer Scale Case, and the Regulating Valves for the Brake Water Supply. (Main Valve in Foreground, Individual Brake-valves on the Rack below the gage-board)

bars are shown in Fig. 25. Within the housing is an oil chamber with a flexible wall or diaphragm, which receives the force of the pull or push of the locomotive. The pressure thus produced in the oil chamber is transmitted along a brass tube of small bore to the measuring apparatus; pressure is generated thus in a smaller oil chamber, and this moves the scale beam. The scale is of such design that balance is automatically maintained, and a graphical recording apparatus is to be provided in order that an unbroken record of the whole of each test may be had. The capacity of the dynamometer is 125,000 pounds, which is 45,000 pounds in excess of the largest previously built for this service, and also a clear excess of 10,000 pounds above the maximum tractive effort of the most powerful locomotive which has been built at this date, unless it work high pressure steam in its L. P. cylinders. An important specification for the dynamometer, if it is to be reliable in its measurement of the draw-bar pull is in relation to the allowable amount of forward and backward motion of the locomotive on the supporting wheels; as soon as the locomotive leaves the exact top point of the wheel, the draw-bar must support some of its dead weight. In the case of this instrument, the permissable range of movement is three one-thousandths of an inch.

In Fig. 24, which is to be considered only as a schematic sketch, are shown the essential feature and operation of the dynamometer. The lower section of the drawing shows the portion of the apparatus that is contained in the receiving head behind the draw-bar; the upper section shows that which is contained in the scale case, shown in Fig. 26. They are connected by the tube

XY which transmits the pressure.

In the lower half of the drawing (Fig. 24), AA is the continuation of the draw-bar, and on it are cut threads as shown to secure the heavy sleeves B and B'. The two sleeves are cast with fingers (shown by the dotted lines) which lay between the corresponding fingers in the casing C. Now let a pull be exerted on the draw-bar: the sleeve B moves in the direction of the arrow; the piston D (the weight of which is carried by the fulcrum plates E and E') is forced forward by the fingers on B. The sleeve B' is also carried forward, but the piston G can move only as far as the set of fingers on the casing at that side will allow. The action of piston D in conjunction with the diaphram H now generates a pressure in the oil chamber I, proportional to the amount of pull along the drawbar. Now suppose the draw-bar stress is a push: the backward movement of D is stopped by the fingers of the casing as was the movement of G before and the motion of G continues till the pressure generated in the oil balances the draw-bar stress. KKK are the springs which apply the preliminary load on being compressed by the capstan nut L. This preliminary load protects the dynamometer from reversals of stress. The amount of this preliminary stress is 50,000 pounds. It is evident that if there is 50,000 pounds compression in the oil before any tractive force is applied, that the pistons can not change their direction of motion unless there is a reversal of over 50,000 pounds, which is quite unlikely to occur.

The transmission tube is carried out of the oil chamber and along the outer casing of the dynamometer under a protective covering. The slack in this tube is enclosed in a heavy cast-iron cylinder attached to the support; the slack is necessary on account of the possibility of adjusting the height of the draw-bar. The pressure in the tube is transmitted to the small oil chamber shown diagrammatically at M, whence the force is applied to the scale beam P through the reducing lever N and the vertical link O. The beam P reads directly to 20,000 pounds in one hundred pound increments, with a vernier which may be read to ten pounds. Higher tractive forces will be read by the addition of weights at Q. The long pointer R is attached rigidly to the beam, and plays over a scale at S, reading zero in its mid-position. The reduction in motion from this scale to the draw-bar is 300,000 to 1.

The poise weight on P is arranged to be balanced automatically as follows: Any motion of R away from its mid-position causes a movement of the light bar V which is fulcrumed at W. In case R moves down, (caused by a decrease in the tractive force exerted) a contact will be made in one of the mercury cups TTTT; this will cause the motor Z to start in such a direction that the poise weight will be moved toward the fulcrum until the load is balanced. The amount of motion which R receives governs the coming into contact of the successive mercury cups: the largest drop of R making the contact in the cup furthest to the right and giving the motor its highest speed in returning the poise to balance. In case of an increase in the amount of tractive force, R will be lifted and the contact will be made in one of the tubes UUUU, with identical results except that the motor of course will be started in the opposite direction. The action of the counterpoise may also be controlled by means of a hand-switch of the double throw type.

The greatest difference between the Illinois Plant and others that are generally similar is in the arrangements made for disposing of the exhaust gases and collecting the cinders. Means have been provided for the entrapping of all of the solid matter which passes out of the stack. A separator of sufficient size for this purpose was found to be too large to be put inside of the building, and it was therefore decided to combine the stack with the separator into one structure, placed outside of the walls. This structure is shown in cross section in Fig. 29. The gases from the stack are discharged into a steel exhaust elbow, and led thence into the exhaust duct, and drawn through this by a 160-inch fan, which is located at the rear end of the building just under the roof. The fan forces the gases through the breeching into the separator. (See Fig. 27 and 28). They then pass downward and around the sleeve A, and in so doing receive a whirling motion; the cinders move toward the wall and drop to the bottom of the hopper, from which they can be drawn off and weighed. The sulphurous nature of the gas renders it necessary to construct as much of this passage as possible of non-corrosive materials: the duct is therefore built of asbestos board. Traps in the bottom facilitate the removal for weighing of such of the heavier cinders as may fall to the bottom of the duct. The final element of the exhaust passage is a forty-five foot radial brick stack, through which the gas is finally discharged eighty feet above the ground. The arrangement of the exhaust passage is best shown in the sectional views of the plant, Figs. 27 and 28.

The plan for the water supply for the plant embraces the construction of a one hundred thousand gallon storage reservoir at the southeast corner of the building, but this is as yet unrealized, and for the present the University mains must be depended on for water. On account of the large amount of cooling water required for the brakes, this must be recirculated by means of pumps,-pumped into the brake casings, then returned to the reservoir, and so again to the brakes. For the present, the water is not used a second time, but runs into a sump in the basement, thence to the sewer. There must be also a generous water supply for the boiler of the locomotive, which will, of course, be irrecoverable. This will for the present be drawn directly from the mains of the University, and later from the supply tank, which will be filled from the mains between tests. On an elevated platform at one side of the room are provided weighing tanks to measure the water used.

The plans of the laboratory show the coal room at the rear of the building, and a firing platform just behind the cab of the engine, where in actual service the deck of the tender would be. The coal is trucked over a scale platform in the coal-room, run onto a hydraulic elevator, and raised to the firing platform for use. This same elevator, dropping to the level of the pit floor provides a means for removing the ashes.

Other details of the installation that are not essentially involved in test work may be mentioned. All parts of the building except the coal-room are served by a ten-ton Whiting travelling crane. The hydraulic elevator mentioned in connection with the coal supply has a capacity of 2000 pounds, and was built by the Otis Elevator Co. Completely equipped locker- and wash-rooms are provided above the coal-room. The engine for running the fan is of one-hundred horse-power, and was built by the Buffalo Forge Company.

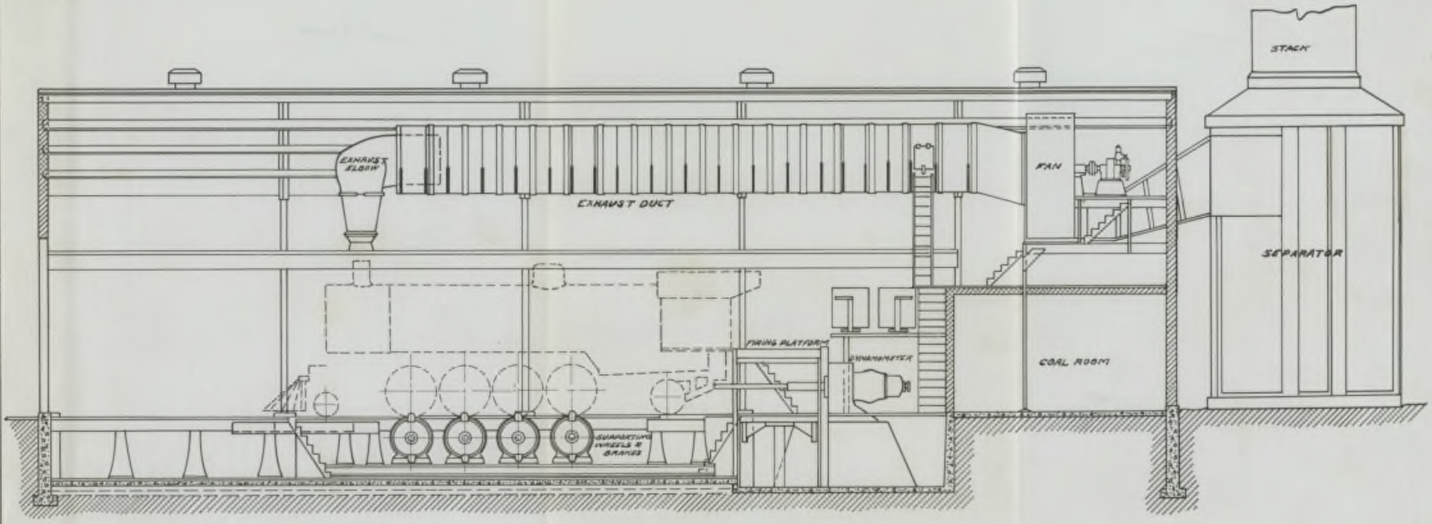

FIG. 27 ELEVATION THROUGH
ILLINOIS PLANT

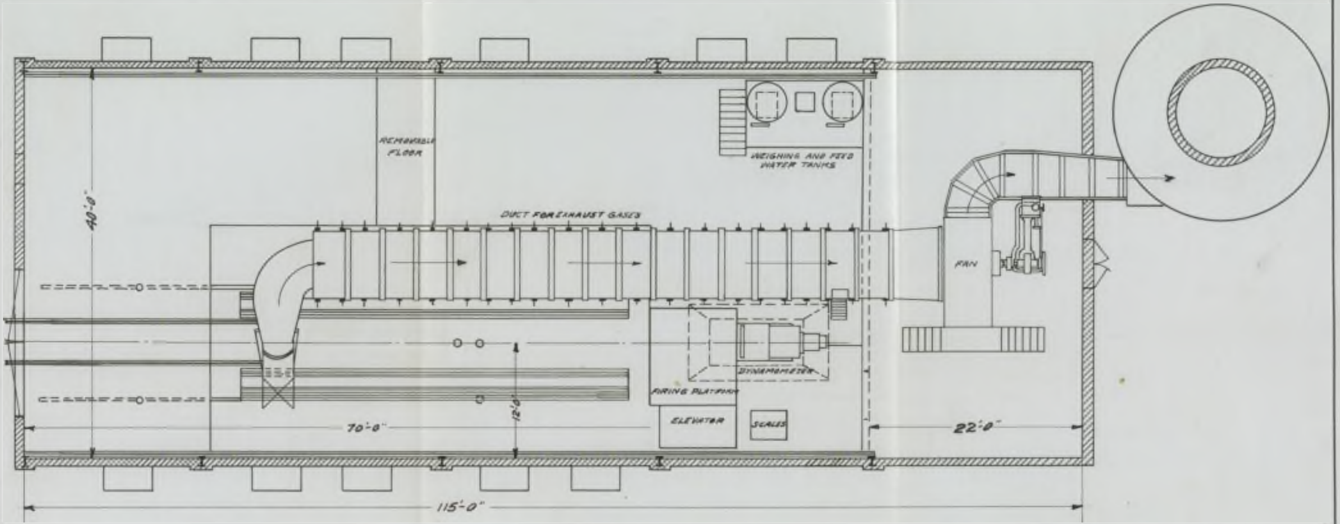

FIG. 28 LONGITUDINAL SECTION THROUGH ILLINOIS PLANT

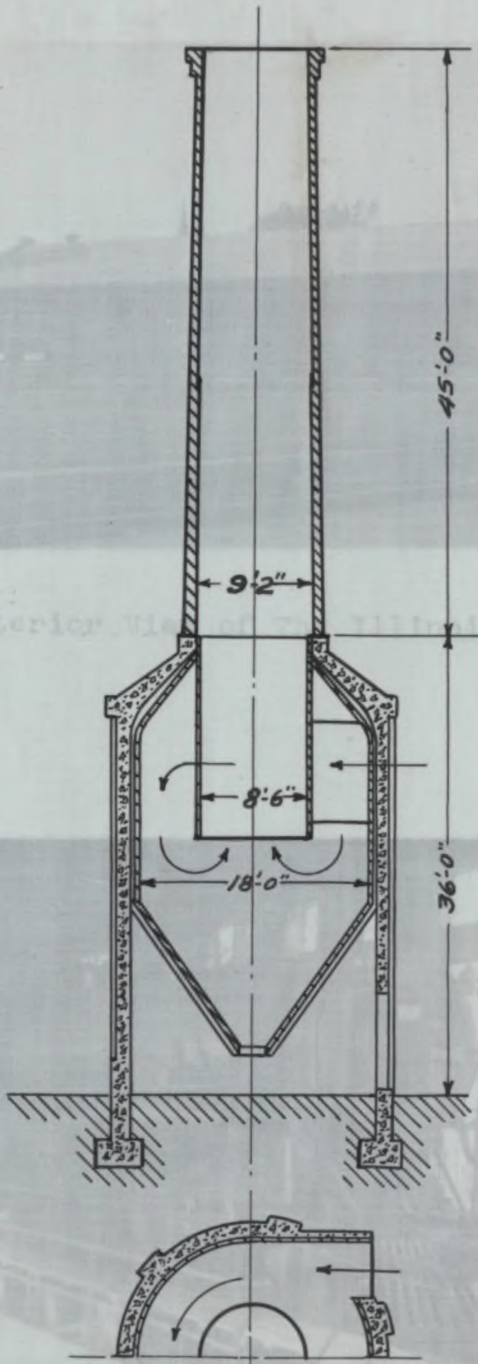

FIG. 29 SEPARATOR AND STACK ILLINOIS PLANT

Fig. 30. Exterior View of The Illinois Plant.

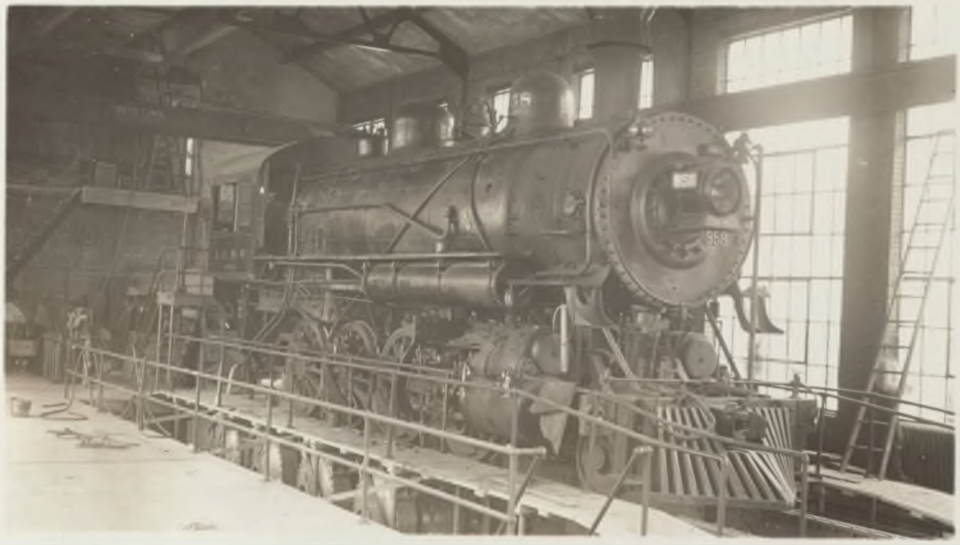

Fig. 31. Interior of the Building showing a
. Consolidation Locomotive mounted on the
Supporting Mechanism.

PART II.

INVESTIGATION OF THE DESIGN OF THE PLANT.*

The Supporting Elements.

The Wheels. One of the first considerations in the design is the size of the supporting wheels that are to be used. The size of the wheels is governed by the sizes of drivers of the locomotives that are to be tested, the peripheral speed of the treads (as affecting the rotative speed of the supporting wheels), and the spacing of the locomotive drivers.

The minimum driver diameter that is to be found on engines in regular service will be in the case of the heaviest freight engines, or on switchers. The smallest drivers that have been recently put on any road locomotive (broad gauge only considered) is fifty-two inches on a decapod type. On eight-wheel switchers it is fifty-one inches, and on the six-wheel type drivers as small as fifty inches have been applied. The minimum distance between the adjacent treads is four inches, corresponding to a spacing of fifty-six inches on a locomotive having fifty-two inch wheels. If supporting wheels of fifty-two inch diameter are used, with the assumption that they can be used on a wheel base of fifty-two and one-half inches, it will be possible to test a locomotive whose

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*Developed from design notes of Professor E. C. Schmidt.

drivers are as small as forty-eight and one-half inches. That is:

$(52\ 1/2 - 4) = 48\ 1/2"$ (With 4" between treads).

This allows the use of a driver smaller than is found on any standard-gauge freight engine, or switcher.

With this minimum wheel base of fifty-two and one-half inches, the brake-casing must be kept down to an outside diameter of fifty inches at the most, and forty-eight would be preferable. There will, however, be no trouble on this score, for an Alden Brake has been built for a capacity of six hundred horse-power with an outside diameter of forty-eight inches.

The use of fifty-two inch supporting wheels will involve very high rotational speed at the higher rates: a fifty-two inch wheel travelling at the rate of eighty miles an hour revolves

$$\frac{80 \times 5280 \times 12}{52 \times \pi \times 60} = 517.13 \text{ times per minute}$$

This may be briefly expressed:

$$R.P.M. = V \div 0.1547, \text{ if } V \text{ is in miles per hour.}$$

While this is a very high speed for so large a bearing, the bearing pressures are not high and forced lubrication is provided so that there is not likely to be trouble from this source. In case a large amount of testing of high speed engines is done, it will be found advantageous to provide a larger set of wheels, and the pedestals are designed accordingly.

The Brakes. For this type of service there are available two types of brakes, the Prony or band-brake, and the Alden absorption dynamometer. The band-brake has been applied in the locomotive plants of the Northwestern, but the Alden type has been more used, and has given entire satisfaction in its twenty years of operation at Purdue. Any design of band- or block-brake, successful as it might prove in the end, would naturally be an experimental matter in the early operation of the plant, and the desire not to be hampered in the start with the less-tried apparatus and the trouble which might be thereby incurred contributed largely to the choice of the Alden brake.

The capacity of the brakes will be governed by the weight on the drivers, the adhesion, and the driver diameter, these three factors giving the torque; and also there must be considered the speed at which this amount of turning effort must be absorbed. An investigation of the wheel loads of various locomotives of late design shows the following values of weights per driver:

Type	Road	Weight on Drivers	Per Driver
2-10-10-2	A.T. & S.F.	550,000	27,500
0-8-8-0	B. & O.	461,000	28,800
2-6-8-0	G. N.	359,600	25,700
0-6-6-0	B. & O.	334,500	27,900
2-8-0	Baldwin '07	232,700	29,100
2-8-2	C.R.I. & P.	238,000	29,600
0-6-0	P. & L.E.	178,200	29,700
4-6-2	Pennsylvania	197,800	32,900
4-4-2	Pennsylvania	133,300	33,325
2-6-2	A.T. & S.F.	174,700	29,100
4-6-0	S. P.	160,000	26,670
4-4-0	C.R.R.N.J.	111,300	27,800

It is seen that the heaviest wheel weights are found on the Pennsylvania's heavy passenger engines, the maximum load being

33,300 pounds. The maximum torque to be absorbed by the brakes will occur at minimum speed*, and it can not exceed the amount of adhesion of the drivers. In view of the present tendency to improve road-bed and increase the wheel loads, it seems reasonable to allow for a further increase in this maximum load; in line with the idea of making the capacity fifteen percent larger than the maximum at present required, the wheel load is taken at 36,657 pounds.** The Purdue experiments demonstrated that the maximum factor of adhesion that could be depended upon was about twenty percent, but for safety's sake, this was increased to twenty-two and one-half percent. The maximum torque is therefore:

$$36,657 \times 22\ 1/2\% \times \frac{52}{2 \times 12} = 17,900 \text{ pound-feet.}$$

In the case of a passenger engine the maximum torque (or tractive effort) will continue to be exerted up to a speed of twenty or twenty-five miles per hour, depending on the amount of heating surface in the boiler and on the amount of evaporation per square foot. For usual heating surface-cylinder volume ratios, if the evaporation is taken at twelve pounds per square foot per hour, twenty miles per hour will mark the limit of maximum torque; if the evaporation is fifteen pounds, the limit will be nearly twenty-five miles. Freight locomotives will have their maximum torque at a lower speed, generally from fifteen to twenty miles per hour.

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* Torque and tractive effort are identical; therefore the maximum torque will occur when the steam supply is the largest, i.e., at the lowest speeds.

** This value is an increase of ten percent in place of fifteen, difficulty in brake design warranting the decrease. (See note, p. 54).

If the brakes are designed to absorb the torque developed by the locomotive having the greatest weight on the drivers, at any speed up to 20 miles per hour (amounting to 129 r.p.m. of the supporting wheels), they will be capable of absorbing the power of any other locomotive working under any condition. The specification of a torque of 17,900 pound-feet at 130 revolutions per minute is equivalent to specifying a brake of

$$\frac{17,900 \times 2 \times \pi \times 130}{33,000} = 443 \text{ horsepower.}$$

Inspection of the design of the brakes doing a like duty shows that the diameters of the inner and outer circles of the brake diaphragm are 16 and 42 inches respectively. With this information the coefficient of friction that will be required may be found. Assuming that there are four rubbing surfaces,

$$M = \frac{4\pi}{18} p f (r_1^3 - r_2^3) *$$

where M is the moment of friction, p the water pressure, f the coefficient of friction, and r_1 and r_2 the outer and inner radii. If the available water pressure is forty pounds, and M is 17,900,

$$f = \frac{17,900 \times 18}{4 \times \pi \times 40(21^3 - 8^3)} = 0.0735$$

This indicates that an apparent coefficient of friction** of 7 1/2% will be needed in case a four-contact-surface brake is used.

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* See "Locomotive Performance", p. 17.

** The "Apparent Coefficient of Friction" includes all of the resistances encountered in the brake -- journal and hub friction, etc.

This is a higher coefficient than will be realized, so it becomes necessary to use a six-surface brake; that is, one having three revolving discs. If, under conditions of maximum output, under which this torque might be maintained at speeds as high as 27 or 28 miles per hour (175 R.P.M. of the supporting wheels) the oil between the discs becomes too fluid to produce the required coefficient of friction, the means of increasing the water pressure temporarily must be resorted to.*

Two methods of holding the brake casing from revolving present themselves: by means of a lug on the bottom, or by the use of two links, which may act either in compression or tension, on the sides of the casing. From the point of view of operation there is little difference, but examination proves that the stresses produced with the two-link scheme are slightly lower, and having no compensating disadvantages, this system was adopted.

First, consider the action on the casing bearing. The casing will be in equilibrium under the action of the forces shown in Fig. 32, g_1 and g_2, the rotational couple, k_1 and k_2, the stresses in the links; w_1, the weight of the casing, and w_2, the reaction of the shaft.

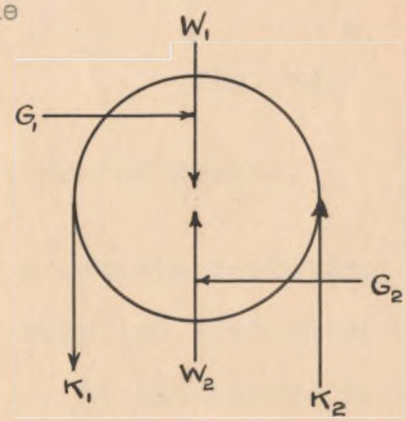

Fig. 32.

The only force on the bearing is the weight of the casing, about 2,000 pounds, and is negligeable, being only about 18 pounds

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* Assuming 18,500 as the torque, the coefficient required is 0.0758.

per square inch of bearing.* Considering the forces acting on the axle, wheels, disc, and casing as a body in equilibrium, the forces act as is shown in Fig. 33. Here $f_1 = f_2 = 8250$, the force of adhesion at the rim of the driver;

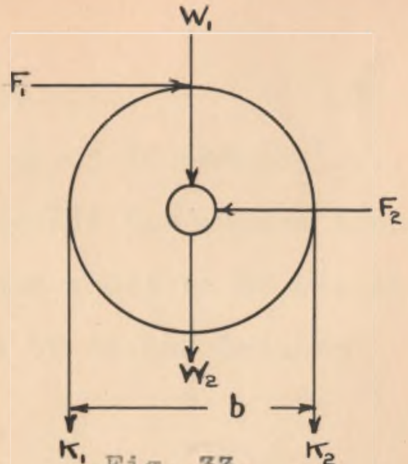

Fig. 33.

$K_1 = K_2$, as before, the forces in the holding links; w_1 is the negligeable weight of the casing, and w_2 the weight of the locomotive on the driver, the weight of the axle, and the weight of the supporting wheel, approximately equal to 45,000 pounds. Since $\Sigma M = 0$ and $\Sigma F = 0$, the only forces which affect the main bearing are the total weight and the adhesion, w_2 and f_2, giving the resultant R_3 as shown in Fig. 34.

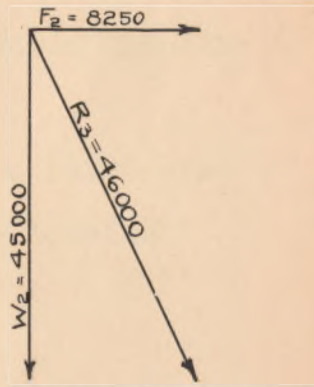

Fig. 34.

Extracts from Final Specifications for Brakes.

Maximum Torque. Each Brake shall develop a maximum resisting torque of 18,000 pound-feet when supplied with water at the pressure specified below....... At all speeds from 35 to 130 r.p.m.

Speed Range.....From 35 to 485 r.p.m.

Horse Power. The maximum brake horsepower will be developed at speeds between 35 and 130 r.p.m., and between these speeds

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* For size of casing bearing see specification for brakes, p. 56

the maximum power will be constant......

Water Pressure......Varying up to a maximum of 30 pounds; ...... casing designed to withstand 45 pounds per square inch.

Outside Diameter...... No part of the brake shall be so set as to prevent adjacent brakes being set 52 inches between centers.

Size of Hub......The end of the axle will be designed to conform to the design of the brake. It must not exceed 9" in diameter.

Method of Holding. The brake casing will be prevented from turning by two links attached to lugs provided on opposite edges of the brake casing.

Oil Supply. It is expected that the brakes will be lubricated with a low grade of cylinder oil, such as "capitol" brand.

<u>The</u> <u>Axles</u>. The proportions of the axles will depend on the following:

Weight per driver of the locomotive.
Weight of each brake.
Weight of each supporting wheel.
Torque transmitted from driver to brakes.

The loads are as follows:

Weight of each supporting wheel,	2000#
Weight on each driver (say,)	38000#
Weight on each brake,	2000#
Torque in the axle,	18000 lb-ft.
Weight of axle (about)	4000#

(Based on an estimate of 11" required, and found to be correct).

The loads are distributed as follows:

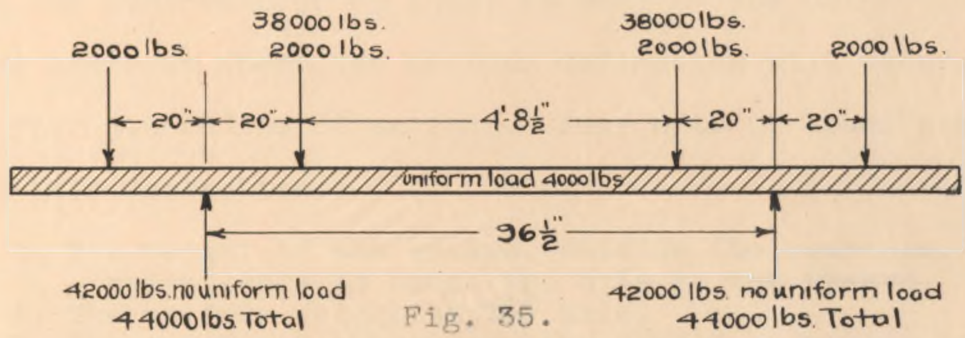

Fig. 35.

The bending moments: (midway between supports)

$(2000 + 38000) \times 29" = 1,360,000$
$2000 \times 68\ 1/4 = \underline{\ \ 136,000}$

1,496,000 inch-pounds

$42000 \times 48\ 1/4 =$ 2,030,000 " "

Net moment $2,030,000 - 1,496,000 = 634,000$ inch-pounds

$$M = \frac{S\ I}{c} = 0.1\ d^3 \therefore \frac{634,000}{S} = 0.1\ d^3$$

If $S = 8,000$ pound per square inch, the diameter is found to be nine inches. The maximum moment is at the center of the axle, or midway between the supports: the maximum moment under the bearing is 2000 x 20" = 40,000 inch-pounds. The torque in the axle is only between the supporting wheel and its corresponding brake, or through the bearing section, but the combined torque and bending moment stresses amount to only 44,000 pound-inches (equivalent moment) and are therefore inconsiderable. In these calculations the moment of the weight of the axle, which is a uniform load of 23 pounds per linear inch, is disregarded.

The size of the axle is not, however, governed in this design by the bending moment, but by the allowable deflection. In the bearings, the allowable deflection under the maximum load is about one hundredth of an inch; reduced to the point midway between the bearings where the maximum deflection will occur, this becomes four hundredths of an inch. Four sets of loads combine to cause this deflection:

1. The weight of the brakes, outside the bearings, which tends to cause the axle to bow upward.
2. The uniform weight of the axle,
3. The weight of the supporting wheels and their loads.
4. The horizontal effect of the torque. (See p. 55).

For a beam with two equal loads equidistant from the supports, the deflection is given by the formula

$$f = \frac{W a}{48 E I} \left(3L^2 - 4 a^2 \right)$$

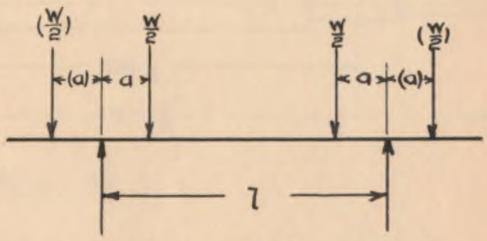

Fig. 36.

where f is the deflection, I the moment of inertia, E the elastic modulus, and other letters as shown. The loads in Nos. 1 and 4 being applied at the same point, they may be combined. The deflection being proportional to "W", the angle at which the resultant deflection acts may be thus found. The deflection amounts to

$$f = \frac{(2000^2 + 8250^2)^{\frac{1}{2}} \times 2 \times 20\ (3 \times 96\ 1/2^2 - 4 \times 20^2)}{48 \times 29{,}000{,}000 \times 730}$$

= .0085 inch (for an 11" axle)

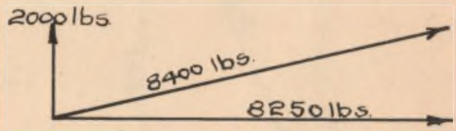

Fig. 37.

The deflection produced by the supporting wheels and their loads is

$$f = \frac{80{,}000 \times 20 \times (3 \times 96\ 1/2^2 - 4 \times 20^2)}{48 \times 29{,}000{,}000 \times 730}$$

$= 0.041$ inch

There is also the deflection caused by the uniform load to be considered. The deflection caused by the weight outside of the supports will balance that caused by the sixteen inches adjacent to the bearing inside, since the sixty-two inches at each end of the axle may be regarded as balanced about the center line of the bearing, which is the center of gravity of this sixty-two inches. The load may thus be considered as 13'11 1/2"-2x62"=31 1/2" in length, and since it weighs 4000 ÷ 167 1/2" per inch or 23# per linear inch, the load on the central 31 1/2 inches equals 725#. This will have approximately the effect of a load of this amount concentrated at the center, and the corresponding deflection is

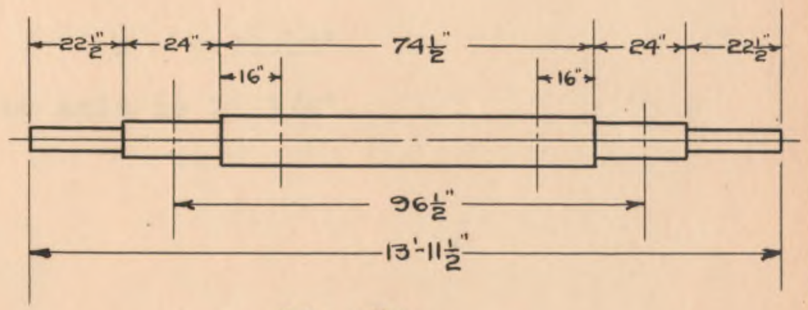

Fig. 38.

$$f = \frac{Wl^3}{8\ E\ I} = \frac{725 \times 96\ 1/2^3}{8 \times 29{,}000{,}000 \times 730} = 0.004 \text{ inch}$$

These deflections may now be combined as shown in Fig. 39, to obtain the resultant deflection, which is found to be 0.044", acting as shown. Angle a is about 11 degrees. This being slightly more than the allowed deflection the diameter of the shaft should be increased a little. This can be done on the basis for the calculations for the deflection caused by the wheel loads, as this is so much the largest factor entering into the deflection that the calculations need not be repeated for the smaller loads. For an "I" of 730, the diameter being 11", the deflection is .044; hence if the deflection is limited to .04",

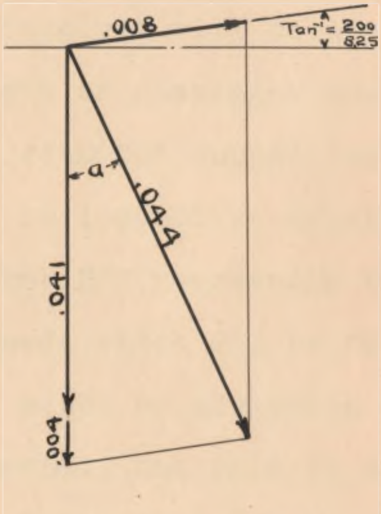

Fig. 39.

$$I = 730 \times \frac{44}{40} = 800,$$

$$d = 11.15"$$

The actual diameter of the axle is 11 1/8".

The Journals, Bearings, and Pedestals. The problem of the design of the journals was mentioned in the discussion of the wheels, page 49. The principal difficulty involved will be the very high speeds that may be involved in tests of passenger engines; as high as 530 R.P.M. may be reached with 52" supporting wheels. The maximum bearing pressure found in locomotive driving-boxes is about 230 pounds per square inch, but 200 represents the average value. On account of the higher speeds which may be required of the supporting-wheel journals, it might be advisable to reduce this figure for bearing pressure somewhat, but this is counteracted by the type of bearing, it being more easily and thoroly oiled, and better ventilated. The maximum load on each bearing is 45,000 pounds (considering 38,000 load on each driver of the locomotive.) In view of the conditions mentioned the allowable pressure was taken at 235 pounds, and the required area of 190 square inches is obtained in a journal 9 1/2" x 20".

The bearings are made self-aligning by mounting the shell in a spherical socket, as shown in Fig. 17. The oil for the journals is drawn from a tank on the side wall, elevated to give it some head. There are two oil-holes in each bearing-cap.

In each pedestal is a removable section ten inches in height, to adapt the plant to the use of 72" wheels in place of those at present installed. The top of the new wheels will then, as at present, be level with the main floor of the building. On account of the height of the bed-plate, the bottoms of the new wheels will clear the floor about an inch.

The Exhaust System.

The plan for the exhausting of the waste gases and steam involves the passing of the mixture successively through the exhaust duct, the spark chamber, and the stack. The fundamental considerations governing the design of a spark trap or chamber to collect all of the solid matter from the largest modern locomotives show that more space will be required than was possible to give within the building. There was also to be considered the necessity of discharging the gas mixture at such a height that it would be unobjectionable to the residents in the neighborhood and to the occupants of nearby University buildings. The further consideration of these things brought out the plan of combining the stack and separator chamber in one structure, outside and at the rear of the building.

The size and capacity of the whole exhaust system depends on the amount of gas which must be passed through it in a given time, and it is therefore necessary to determine the amounts which it will be called upon to handle. The first thing to be considered is the amount of coal that may be burned, and this involves the available heating surface. The largest heating surface employed in ordinary locomotives at present is 5,000 square feet, and in locomotives of the Mallet type, 6,630 square feet. If an increase of ten percent be allowed for each of these, they become 5500 and 7300 square feet respectively. Under the severest test conditions the evaporation may be run as high as fifteen pounds per square foot of heating surface per hour in the case of the ordinary locomotives, but there will be very great difficulty in driving the Mallet harder than evaporating twelve pounds. The St. Louis tests show boiler efficiency ranging from 4.5 to 7.7 pounds of water per pound of coal under these severest operating conditions; the highest value was only obtained in two of the eight engines tested. In view of this, six pounds of water seems to be a fair average for the ordinary types of locomotives. The performance of the Mallets in various tests has been slightly better, so that an average of 6.3 pounds of water per pound of coal may be assumed for them. This average may also be adopted for the ordinary types as nearer usual maximum working conditions. With these values, the maximum possible coal consumption per hour is for the ordinary locomotive,

$$5500 \times 15 \times 1/6.3 = 13,000 \text{ pounds.}$$

For the Mallet,

$$7300 \times 12 \times 1/6.3 \doteq 17,300 \text{ pounds.}$$

These values may be recognized as the ordinary maximum and a 33 1/3% overload maximum capacity, respectively. They will not under any circumstances be reached with the present amounts of heating surfaces, unless it be the result of the use of a very poor coal; with average coal and the present rate of increase in locomotive proportions, the former may be anticipated as an occasional demand and the latter as a very exceptional demand in the future, or as a result of the use of mechanical stokers. However, on account of the cost and permanent character of this installation it seems wise to make this provision for the extreme calls of at least the coming decade. Appendix i shows a table gathered from all available sources on which the assumptions made herein are based. The rates of combustion which will result from the foregoing are as follows:

Conditions	Sq. Ft. H. S.	Grate Area Corresponding Current design plus 10%	Coal per Hour	Coal per sq. ft. Grate A. per Hour.
Ordinary Maximum, (12#)	5500	80	13000	162#
Highest Limit, (15#)	5500	80	17300	216#
Mallet Maximum, (12#)	7300	110	17300	157#

All evaporations considered are equivalent.

Appendix ii is a table of the ratio of the air supplied to that which is theoretically needed, the study of which leads to the conclusion that for the high rates of evaporation chosen as the bases for this analysis, the proper ratio is

$$\text{Allowable Ratio, } \frac{\text{Air Supplied}}{\text{Air Needed}} = 1.40$$

The theoretical amount of air needed to burn one pound of coal varies chiefly with the carbon content of the coal. From the ultimate analyses and the formula

$$A = 12\ C + 35\ (H - \frac{O}{8})$$

the amounts obtained are as follows:

For various Illinois coals (forty different analyses) -- 8.02 to 10.80#

Scalp Level, (as used in St. Louis Tests, Pennsylvania Bituminous) -- 11.08#

Pocahontas ------------ 10.83#

The use of poorer grades of coal will involve less air per pound but a balancing increase of the amount of coal burned. Appendix iii gives further figures on the air requirements. The consideration of the foregoing has led to the adoption of 10.50 pounds of air as the theoretical need to burn each pound of coal.

Coal of the nature that will be used in the work of this plant will in general have about ten percent of ash: also a spark loss of about fifteen percent may be depended upon. Variations within reasonable limits of either of these values will have only the most trifling effect on the results. Taking all of these things into consideration the net amount of flue-gas that will be produced by one pound of coal is

$$(1.00 - 0.10 - 0.15) + (1.00 \times 10.50 \times 1.4) = 15.45 \text{ lbs.}$$

A chemical composition of this flue-gas must now be assumed, in order that the properties of the gas at various temperatures and pressures may be determined. Variations from any assumed analysis will not materially affect the conclusions. Three analyses were assumed, giving the ordinary range of the CO_2 content, as follows:

Constituent	Percent by Vol.			Percent by Weight.		
	#1	#2	#3	#1	#2	#3
CO_2	12.0	10.0	15.0	16.65	14.00	20.62
CO	1.5	1.0	2.0	1.45	.88	1.76
O	5.5	8.0	2.0	5.24	8.16	2.00
N	81.0	81.0	81.0	71.90	72.20	70.86
H_2O	----	----	----	4.76	4.76	4.76
	100.00	100.00	100.00	100.00	100.00	100.00

(H_2O is considered = 5% by weight of the dry gases.)

A consideration of all the available information on the subject of the temperature of the gases in the front end of the locomotive, in cases where the equivalent evaporation was ten pounds per sq. ft. of heating surface per hour, shows that the average temperature is about 690 degrees F., before mixing with the exhaust steam. (See Appendix v for basis.) The three different flue-gases above assumed have at 690°F. and atmospheric pressure (14.7# absolute) volumes of 28. 62, 28.78, and 28.29 cubic feet per pound. (See Appendix iv for calculations). In view of the very slight difference in volume between the gases with the lowest and highest CO_2 content (Samples #2 and #3, having only 1.6% difference in volume) further examination may be concerned only with the average sample, No. 1; the mean specific heat of this particular gas is 0.246, as shown in Appendix vi.

It will now be necessary to consider the amount of the exhaust steam which must be cared for in addition to the flue gases. Suppose that the water enters the boiler at an average temperature of 70, and must be evaporated at 200 pounds boiler pressure. The evaporation factor for these conditions,

$$\frac{(361.3 - 38.02) + 838.9}{965.7} = 1.195$$

An ordinary locomotive, having the 5500 square feet of heating surface assumed previously and evaporating at the rate of fifteen pounds will have an equivalent evaporation of 82,500 pounds per hour. A Mallet of 7300 square feet of heating surface, evaporating at the same rate, will have 109,500 pounds per hour; applying the evaporation factor, these values become 69,000 and 91,700 pounds of steam per hour, respectively. The ratio of the weight of the exhaust steam to the weight of coal used is

$$\frac{69000}{13000} = \frac{91700}{17300} = 5.30$$

and since this ratio is constant, further investigation may be based on the first condition only, namely, the performance of the ordinary locomotive, and increased by the overload factor of 33 1/3% when needful to consider the second condition.

Following the lines laid down by this Condition "A", the performance of the locomotive of the usual design with ten percent heating surface above that used on the largest current proportions, we have

Coal Burned per minute , $\frac{13000}{60} = 216.66$ pounds.

Water per minute = exhaust steam = $\frac{69000}{60} = 1150$ pounds.

And for each pound of coal there will be 15.45 pounds of flue-gas, or 216.66 x 15.45 = 3350 pounds per minute. Hence the exhaust system must be able to accommodate

3350 + 1150 = 4500 pounds of gas mixture per minute.

This mixture is 25.5% exhaust steam and 74.5% flue-gas, of the properties previously assumed. In order to find the volumes of the mixture at various temperatures, it will be necessary to assume the quality of the exhaust steam before the mixing takes place.

The maximum power of the engine will not be developed below a speed of twenty miles per hour, at which speed the cut-off will be about 80%. At higher speeds the maximum power will be obtained at cut-offs varying from thirty to forty percent. From an examination of the source material (see Appendix vii) it is found that for forty percent cut-off the quality at the instant of exhaust period will be very close to 100%, and thus the mean qualities throughout the exhaust period are 91 and 96 percent, respectively. A working mean of 92% quality may be taken as a basis, though this may result in some crowding of the capacity of the fan at high-power tests; these, however, will be rare. (If a higher quality be chosen, less heat must pass for the evaporation of the suspended water, and more will remain to maintain a high temperature and hence large volume in the mixture. Particularly will this be true if superheated steam is used: the quality will be high enough to cause considerable increase in the specific volume of the gas: but there will be a reduction in the consumption of coal, so that the gross increase will be small if any.)

The back pressure or pressure in the exhaust nozzle may be assumed to be about thirty pounds per square inch. (Slight variations in the amount of back pressure will not materially affect the quality of the steam after its expansion into the front end.) The maximum draft under the highest power is ten inches of water, which corresponds to 0.36 pounds per square inch vacuum; the net pressure is therefore

$$14.7 - 0.36 = 14.34\# \text{ absolute pressure} \approx 29.25'' \text{ Hg.}$$

The steam expands from the nozzle into the front end practically adiabatically: the quality* after expansion is 88%, - the steam

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* See Goodenough, "Thermodynamics", p. 186.

$$s_1' + \frac{x_1 r_1}{T_1} = s_2 - \frac{x_2 r_2}{T_2}$$

30# pr., 92%; 14.3#, x_2=?

s_1, s_2, entropies of the liquid.
x_1, x_2, qualities.
T_1, T_2, Temperatures, absolute.
r_1, r_2, latent heats of vaporization.
(All before and after the expansion).

(See Steam Tables)

$$0.3681 + 0.92 \times 1.3307 = 0.3101 + x_2^2 \times 1.4485$$

$$x_2 = 0.885$$

Say 0.88 or 88%

is therefore 12% suspended water after it enters the front end and before it mixes with the flue-gas. The mixture takes place at atmospheric pressure; the gas and steam will reduce to a common temperature, at which point their volumes will add.

As previously stated the proportions by weight of the mixture of gas and steam are 25.5% steam and 74.5% gas; if the weight of gas is taken as unity, the nature of the constituents before the actual mixing takes place is as follows:

In 1.3423 pounds, before mixing, there are

1.00# gas, at 690° F., and 14.7# pressure.
(This gas occupies 28.62 cu. ft., and has a mean specific heat of 0.25)
0.3423# steam and suspended water at 212° F. and 14.7# pressure.
(Containing 0.88 x 0.3423 = 0.3012# dry steam
0.12 x 0.3423 = 0.0411# water.)

The water will evaporate on coming into contact with the gas, taking from the latter

$$0.0411 \times 970.4 = 39.87 \text{ B.T.U.},$$

since its latent heat is 970.4 B.T.U. per pound. This loss of say 40 B.T.U. will cause a drop of $40 \div 0.25 = 160°$ reducing the temperature of the flue-gas to $690° - 160° = 530°$ F. In 1.3423 pounds of mixture there are now

1.0000# gas at 530° F and atmospheric pressure, and
0.3423# dry steam at 212° F and atmospheric pressure.

The resulting temperature may be found as follows:

Let t° be the resulting temperature; take as the specific heat of the steam 0.469. (Goodenough, p. 211)

Then $$1 \times 0.250 \times (530° - t°) = 0.3423 \times 0.469(t° - 212°)$$

$$t = 405.6 \quad (\text{say } 405°)$$

We have therefore on complete mixing, 1.3423 pounds of mixture at a temperature of 405° F., the constituents of which will be

CO_2	---------- 16.65 ÷ 1.3423	= 12.40	12.40	
CO	---------- 1.45 ÷ "	= 1.09	1.09	
O	---------- 5.24 ÷ "	= 3.90	3.90	(Percentages
N	---------- 71.90 ÷ "	= 53.56	53.56	by weight)
H_2O from coal	-- 4.76 ÷ "	= 3.55		
H_2O from steam	-- 34.23 ÷ "	= 25.50	29.05	
			100.00	

After the gas mixture has passed out of the front end of the engine into the exhaust duct it will begin to cool and this will continue as it traverses the length of the duct. The temperature at the fan will probably be from 300° to 250° F. The volumes of the first four constituents of the mixture may be calculated for any temperature from the relation of their absolute temperatures, and for the steam it may be readily obtained from tables.

At 405°, 29.05 lbs. of steam occupies 29.05 x 34.94 c.f.=1015.0 c.f.
At 350°, " " " " " 29.05 x 32.67 c.f.= 949.0 c.f.
At 300°, " " " " " 29.05 x 30.57 c.f.= 888.0 c.f.
(Atmospheric pressure.)

To correct the volumes of the other gases to the desired temperatures, use the following factors:-

For a temperature of 405° --- $\frac{405 + 460}{32 + 460} = 1.757$

For a temperature of 350° --- $\frac{350 + 460}{32 + 460} = 1.645$

For a temperature of 300° --- $\frac{300 + 460}{32 + 460} = 1.544$

Volumes of the Mixture at Atmospheric Pressure.

Constituents in 100# of Mixture	Vol. per lb. 32 F.	Cu. Ft. volume of mixture at various temperatures 32 F	405 F	350 F	300 F
Co_2 = 12.40#	8.151	101.07			
Co = 1.09#	12.810	13.96			
O = 3.90#	11.208	43.71			
N = 53.56#	12.773	684.12	1480.90	1386.50	1301.37
		842.96			
H_2O = 29.05#			1015.00	949.06	888.06
100.00#					
Volume of 100 pounds			2495.9	2335.56	2189.43
Volume of 1 pound			24.96	23.36	21.89

When firing 13000 pounds of coal per hour, under Condition A, the exhaust system must take care of 4500 pounds of mixture each minute (See p. 67); the volume of mixture as it leaves the stack is thus 112,320 cubic feet per minute; if the gas cools to 350° in the passage of the duct the volume will be 105120 cu. ft., and if it cools to 300°, the fan must care for 98,505 cubic feet per minute. To fit this to the condition of the high-power test of a Mallet engine it is necessary to multiply it by the "overload factor" of 33% (see p. 67.) With the gas cooled to 350 F., the fan must then be able to remove

105,120 x 1.33 = 140,160 cubic feet per minute

which sets the capacity of the fan and engine.

This value of the total exhaust-duct discharge varies obviously with the amount of coal burned per hour, and as this figure is set higher than will at present be realized, the volumes are too high in the same ratio. This does not, however, signify that the fan will be ten percent or thereabout oversize for all tests which might be run under the present conditions and practices in locomotive heating-surface design, for there are several other factors which may cause the total discharge to approach an amount more nearly equal to the maximum above set. The coal may be of very much poorer quality, requiring a greater increase in the amount of coal burned than the corresponding decrease in the air demand will counteract. The ratio of air supplied to air required may be larger than the assumed average for some tests. The front end temperature may be higher at times: likewise the quality at exhaust. It is further concievable that all of these effects might be cumulative, in which case the capacity of the fan might be crowded in the test of a Mallet of large proportions as at present designed. The accumulation of the effects mentioned is highly improbable, and any of them might vary in the other direction equally well; with these considerations in view it is evident that the capacity of the blower has been made great enough to meet any emergency which may be met within the next decade at least.

The exhaust passage, as shown on pp.46 seq., consists of the following sections: the elbow, the duct proper, the fan chamber, the breeching and the stack. It was originally intended to make all of the surface of the duct of material which would not be affected by the corrosive action of the gas, but this was found to be impracticable. The elbow is of sheet steel, and will be replaced from time to time as it is cut through by the sulphurous gases. The duct is built of asbestos board sold under the trade name "Transite", made by the Johns-Manville Co. A standard size was selected, seven feet diameter inside, the material being 5/8 inch thick. The breeching is 7'6" high and 4 ft. wide, and is made of the same material. The material of the separator is entirely concrete. The fan is the Buffalo Forge Company's 160" "Conoidal", run by a horizontal direct connected engine, with cylinders 12 x 12". To remove the maximum volume of gas the fan requires 95 horse-power.

Sources and Authorities on Locomotive Performance.

Goss, Locomotive Performance.
Goss, High Pressure Steam in Locomotive Service.
Goss, Superheated Steam in Locomotive Service.
P.R.R. Reports of St. Louis Tests.
U. of I. Railway Department Test Reports.

The Water Supply.

Water must be supplied to the plant for two purposes -- for feeding the boiler and for cooling the brakes. It will therefore be necessary to find the amounts that will be required for each use and also the rate of consumption.

The feed-water may be assumed to enter the boiler at a temperature of 70°F., and it is evaporated at 200# pressure in the case of most locomotives. The factor of evaporation is

$$\frac{1200.2 - 38.0}{965.7} = 1.2 = 1/0.832$$

The water rates of some representative locomotives, both Mallets and those of the ordinary designs are shown in the following table:

Type.	Heating Surface	Equivalent Evaporation per hour, at rate of 12#	15#	Actual hourly evaporation, same equiv. rates 12#	15#
4-6-2	5017	60200	75250	50000	62500
4-6-2	4448	53400	66750	44400	55500
4-8-2	4231	50800	63500	42300	52800
0-8-8-0	6629	79600	99450	66400	82600
2-6-6-2	6555	78650	98300	65400	81700

The performance of the first engine in this list at 12 pounds evaporation and at 15 pounds may be considered to represent the ordinary maximum demands which will be made on the plant. The performance of the 0-8-8-0 Mallet at a rate of 12 pounds is the "emergency maximum", since a rate of 15 pounds is very doubtful of attainment, even for very short periods. The hourly water supplies thus called for are 50000, 62500, and 66400 pounds respectively. To this may be added ten percent, in consideration of the future development which may be expected, calling on the sources of supply for 55000, 68750, and 73040 pounds per hour respectively.

Condition	Pounds per hr.	Gal. per hr.	Test period, hrs.	Total Gal.	Gal. per Minute.
Ordinary Maximum	55000	6600	3	19800	110.0
Ordinary Maximum	68750	8260	1	8260	137.6
Emergency Maximum	73040	8760	3	26280	146.0

The last column gives the supply in gallons per minute which will be needed under extreme conditions to feed the boiler.

The water that will be needed to cool the brakes depends directly on the amount of energy which the brakes must absorb. In this case again the full normal demand on the plant will be made by a high-speed test of a heavy passenger engine, and the emergency conditions in the testing of a Mallet.

The draw-bar horse-power of a locomotive is

$$DHP = \frac{P \times V}{375} = .002666 \ PV,$$

where P is the draw-bar pull and V the speed in miles per hour. The tractive effort for any locomotive at varying speeds may be obtained from the formula

$P = 161\ H/V$, H being the heating surface.

This maximum pull will be reduced by the amount necessary to overcome the machine friction. This amount may be represented by the expression $pd^2 l/D$; p is a portion of the effective pressure which may vary for the type of locomotive, d the cylinder diameter, D, the driver diameter, and l the stroke of the piston. (See Goss, "Performance", p. 415). In Appendix viii are characteristic curves for six representative engines giving results as follows:

Type	Heating Surface	Speed	Tractive Effort	Brake H.P.
4-4-2	(1) 3582	27	29300	2110
4-6-2	(2) 4448	24.3	40500	2620
2-8-2	(3) 5017	21.3	48200	2730
4-8-2	(4) 4231	18	52700	2530
0-8-8-0	(5) 6629	14.5	97800	3780
2-6-6-2	(6) 6555	20	71200	3800

The assumptions made in the calculations are

> Evaporation, 15# per sq. ft. Heating Surface per hour.
> Steam per hour, per I.H.P., 25# for engines 1, 2, 5, and 6; 27# for No. 3, 24# for No. 4.

The evaporation figure of 15# is however too high to be maintained, and the above figures represent a theoretical absolute maximum horse-power. In the case of the ordinary locomotives this might be approached on tests of very short duration -- say a half hour, but for longer periods the horse-power values should be reduced in the ratio of 12:15. The maximum power that will be realized is then as follows:

	Present Designs	Allowing for 10% increase.
Ordinary Locomotives, (three hour test)	2000-2160	2200-2376
Ordinary Locomotives (half hour tests)	2500-2700	2750-2970
Mallet Locomotives (three hour tests)	2400-3040	2640-3344

These last figures are the maximum brake horse-power for any locomotive that is likely to be built within the next ten years; during three-fourths or more of the time of operation of the plant such values will not be realized, but the water supply must be so provided for that when the occasion demands the water will be available. Under the conditions of the testing of a heavy Mallet provision may be made for a supplement to the water supply of the plant from outside for the brief periods at which the maximum demands will be made, but for the regular running of the plant at high capacity, there must be sufficient cooling water available to absorb the heat generated by the expenditure of from 2200 to 2400 horse-power through a period of three hours, or for as much as 3000 horse-power for a half hour test.

One horse-power-hour is equivalent to 2545 B.T.U., hence the thermal units which must be absorbed by the water are as follows:

Three hours test, of ordinary locomotive,
2370 x 2545 = 6,035,000 B.T.U. per hour.
Three hour test, Mallet Locomotive,
3340 x 2545 = 8,505,000 B.T.U. per hour.
Half hour or one hour test,
2970 x 2545 = 7,560,000 B.T.U. per hour.

If the cooling water is available at a temperature of 70°F., and can be heated to about 130°F. without impairing its efficiency as a cooling agent, each pound of water will absorb 60 B.T.U., and for the first condition above, 100,600 pounds of water will be needed per hour, and for the third condition the need will be 126,000 pounds per hour. The emergency maximum, as represented by the second condition, will be taken care of with help from an outside source.

Horse-power	B.T.U.	Water per hour.	Gals. per hour.	Time of Test	Total Gal.	Gal. per Min.
2370	6035000	100600#	12060	3	36180	201
2970	7560000	126000	15120	1	15120	252
3340	8505000	141750	17010	3	51030	284

The total water consumption of the plant will be therefore as follows:

Condition	Gallons of water required per minute		
	For the Brakes.	For the Boiler.	Total
Ordinary maximum (12#)	201	110	311
Ordinary maximum (15#)	252	137	389
Emergency	(284)	(146)	(430)

and the total requirements during the test periods are

311 x 180 = 56000 gallons

389 x 60 = 23340 gallons

430 x 180 = 77500 gallons

If it is found that the cooling water can be used satisfactorily up to a temperature of 150°F., in place of the 130° previously assumed, the consumption of cooling water by the brakes will be reduced 25% since each pound of cooling water will absorb 80 in place of 60 thermal units. The total requirements for the tests as before then become:

(201 x 75%) + 110 = 261; x 180 = 45700 gallons
(252 x 75%) + 137 = 326; x 60 = 19800 gallons
(284 x 75% + 146 = 357; x 180 = 64400 gallons.

Under these conditions the plant may be made independent of outside sources for water during the time of a test by providing a storage tank of a capacity of about 60,000 gallons. Only at the very highest rates of evaporation and the hardest firing in the testing of Mallet engines will it be necessary to supplement the supply from without.

The supply of water for the boiler is pumped out of the storage tank and weighed into a calibrated tank for measuring the amount used. (This measuring tank has a capacity of 16000 gallons, being large enough so that one filling will provide the water required in a three-hour test of an ordinary locomotive at all moderate powers). The cooling water after its passage through the brake-casings is returned to the storage tank. No account is here taken of the cooling of this water by radiation, which will decrease the amount of water actually required.

Note on the Water Supply.

For the present the plant will be run without the storage tank, the university and city mains being depended on to furnish the required water. This will without doubt limit the scope of the plant at first, but for the testing of the first engine on the plant (I.C. Consolidation, No. 958), there will be no such power requirements as those herein laid down. Its heating surface being about 3700 square feet a brake horse-power of 1450 may be expected, requiring an absorption of 61,600 B.T.U. per minute, which will be accomplished by supplying 125 gallons of water per minute, the water being heated to 130°.

The Dynamometer.

The most elaborate and complicated piece of apparatus about the whole plant is the dynamometer. This was built by the William Sellers Co., of Philadelphia, after their own plans, and from the following specifications:

1. The Dynamometer shall be in principle and general design like that used for the same purpose at Purdue University.

2. Capacity. The Maximum Tractive Force to be measured will be 125,000 pounds.

3. Initial Load. The Initial Load on the dynamometer shall be 40,000 pounds.

4. Piston Movement. The forward or axial movement of the piston from no load to full load shall be as small as possible, and it shall not be more than one hundredth of an inch, (0.01").

5. Vertical Adjustment. The height of the center line of the draw-bars of locomotives to be tested will vary from 27 to 47 inches above the rail. The dynamometer should be so mounted as to accommodate itself to this range of draw-bar height.

The first specification calls for a dynamometer of substantially the same type as the one used in the plant at Purdue University, and which was also built by the Sellers Company. (In 1894). The principle of the Emery Hydraulic Testing Machine is used, the head of the machine which receives the draw-bar pull being an adaptation of the head of a thirty thousand pound testing machine. The construction principle involves two hydraulic (oil) cylinders connected by a small-bore tube, which transmits to the second chamber pressure generated in the first by the pull of the locomotive on a piston acting within the chamber. The Purdue machine had a capacity of 30,000 pounds of draw-bar pull. This capacity is far below that required for the present plant as the largest locomotives at present have a maximum tractive effort of 115,000 pounds, and over and above this a percentage must be allowed for future increase. The capacity therefore determined on is 125,000 pounds.

The initial load is placed on the dynamometer by means of a set of helical springs provided for the purpose, their casing being shown in Fig. 25 at the extreme left of the dynamometer housing. This preliminary load serves to protect the remainder of the mechanism against shocks which would otherwise result in reversing stresses. This was afterward raised from 40,000 to 50,000 pounds; this with the tractive capacity of 125,000 pounds made it advisable to use the receiving head of a testing machine of a standard size, that for the 200,000 pound machine being used.

Regarding the fourth specification, that of the limit of piston travel, the Sellers Company reduced the one-hundredth inch allowed to about three thousandths of an inch as the greatest possible travel each way from the mid-position of the piston. This includes the travel produced by the preliminary load, for which the position of the engine may be adjusted, so that the extreme travel in test with moderate tractive powers will be probably not above one thousandth of an inch.

The action of the dynamometer is shown in the schematic sketch on page 37. No attempt is here made to show the parts in their relations of size or location. The action of the machine is described in Part I, pp.35-40. Fig.24 shows the general appearance of the casing and support, and fig.27 its location relative to the locomotive.

Supplementary Note.

The first locomotive to be tested on the Illinois Plant is a simple consolidation, No. 958, furnished by the Illinois Central Railroad Company. The engine was mounted April 11, 1913. After some days of preliminary running, testing was started on May 5th. The work was under the direction of Prof. E. C. Schmidt and Mr. F. W. Marquis as Director of Tests.

The test force was organized as follows:

Director	one
General Log	two
Indicators	two
Flue gas analysis.	two
Coal weighing	one
Feed Water	one
Dynamometer	one
Brake valves	one
Cab Log	one

In addition, the fireman, (an employee of the railroad,) two mechanics, and two laborers, were used.

The dimensions of the locomotive are as follows (see Fig. 31.)

Total Weight of Engine	223,000 pounds
Weight on the drivers	200,900 pounds
Cylinders, 22 inches in diameter by 30" stroke	
Drivers	63 inches
Working pressure	200 pounds
Heating surface, tubes, ...	3514 sq. ft.
Fire box, ...	177 " "
Total, ...	3691 " "
Grate Area	49.5 " "
Rated Tractive Effort	39,200 pounds.

The locomotive was built by the Baldwin Locomotive Works in September 1909.

APPENDIX 1.

Coal Consumption.

Conditions of running.	Heating Surface 10% over Present Design	Evaporation from and at 212°	Total Evap.	Evap. per lb. Coal.	Total Coal per hour	Time of Test hrs.	Conditions
1	2	3	4	5	6	7	8
A	5500	15	82500	4.5	18300	1	Ordinary
"	"	15	"	6.0	13750	1	"
"	"	15	"	6.3	13000	1	"
B	7300	12	87600	7.5	11670	3	Mallets
"	"	13-1/2	98600	6.7	14700	3	"
"	"	15	109500	6.0	18250	1	"
"	"	15	"	6.3	17300	1	"

APPENDIX ii.

Air Supply.

Source of Information.	Equivalent Evaporation from and at 212°. Lbs. - A.	Lbs. - B.	Ratio of air supplied to air theoretically. Need for range (A)	for range (B)
Goss, "High press. steam in Locomotive Service".	10-15	12-15	1.02-1.16	1.02-1.16
Goss, "Superheated Steam in Locomotive Service".	10-15	12-15	1.16-1.46	1.16-1.46
St. Louis Tests by Penn. R.R.	10-15	12-15	1.01-1.93	1.09-1.56

APPENDIX iii.

Air Needed to Burn one Pound of Coal.

Name of Coal.	Carbon	Hydro-gen	Oxygen	Nitrogen	Sulphur	Moisture	Ash	Air per lb.
	Ultimate Analysis of Coal.							
Christian Co.	58.34	3.90	7.91	--	6.06	--	23.79	8.02
Williamson Co.	75.22	4.81	0.14	1.69	1.47	--	7.67	9.91
Penn. Tests at St. Louis	84.20	4.28	1.44	2.94	0.80	--	6.34	11.08
I.C. Locomotive Tests U. of I. (Sangamon Co.)	69.52	4.75	9.04	1.07	4.17	--	12.47	9.44
I.C. Locomotive Tests U. of I. (Williamson Co.)	73.49	4.99	9.45	1.71	1.41	--	8.95	9.86

APPENDIX iv.

Volumes of Gases.

Sample (1)

Gas	Percent by Weight	Volume per lb. at 14.7# and 32°	Total vol. of 100 lbs. at 14.7#, 32°	Factor to correct to 14.7#, 690°	Total volume cu. ft. 100# gas at 14.7#, 690°	Total volume cu. ft. 1# gas at 14.7#, 690°
Co_2	16.65	8.151	135.714	2.3374	317.218	
Co	1.45	12.810	18.375	"	43.417	
O	5.24	11.208	58.730	"	137.276	28.268
N	71.90	12.773	918.370	"	2146.019	
H_2O	4.76	Vol. at 14.7#, 690°--45.71			217.579	
	100.00	45.71 x 4.76 = 217.579			2862.109	

Sample (2)

Gas	Percent by Weight	Volume per lb. at 14.7# and 32°	Total vol. of 100 lbs. at 14.7#, 32°	Factor to correct to 14.7#, 690°	Total volume cu. ft. 100# gas at 14.7#, 690°	Total volume cu. ft. 1# gas at 14.7#, 690°
Co_2	14.00	8.151	114.000	2.3374	266.800	
Co	.85	12.810	10.880	"	26.300	
O	8.16	11.208	91.500	"	213.600	28.78
N	72.20	12.773	916.000	"	2154.000	
H_2O	4.76				217.600	
	100.00				2878.300	

Sample (3)

Gas	Percent by Weight	Volume per lb. at 14.7# and 32°	Total vol. of 100 lbs. at 14.7#, 32°	Factor to correct to 14.7#, 690°	Total volume cu. ft. 100# gas at 14.7#, 690°	Total volume cu. ft. 1# gas at 14.7#, 690°
Co_2	20.62	8.151	162.800	2.3374	392.800	
Co	1.76	12.810	22.550	"	52.700	
O	2.00	11.208	22.416	"	52.400	28.29
N	70.86	12.773	895.000	"	2114.000	
H_2O	4.76				217.600	
	100.00				2829.500	

APPENDIX v.

Temperature of Gases in the Front End.

Sources	Values	Temperature range F.	Appx. Average Temp.
Goss "Locomotive Performance"	7	655° - 752°	690 - 700
Goss "High Press. Steam"	12	762 - 842	800
Goss "Superheated Steam"	15	715 - 824	770
St. Louis Tests	39	625 - 787	680 - 700
Ry. Dept. U. of I. engine 940	5	606 - 620	615 (high air excess)

APPENDIX vi.

Mean Specific Heat of Flue Gas.

(Sample 1).

Gas	Percent by weight		Mean specific heat between 0°C and 200°C	Heat absorbed per degree rise for each constituent.	
CO_2	16.65		0.2012	3.34998	Mean S.H. for gas
Co	1.45	(or weight	.2426	.35177	$\frac{246}{100} = .246$
O	5.24	in lbs. in	.2175	1.13970	
N	71.90	100# of	.2438	17.52922	
H_2O	4.76	mixture)	.4750	2.26100	say .250
	100.00			24.63167	

APPENDIX vii.

Quality of Steam in the Exhaust.

Goss "High Pressure Steam" Test No.	Average Cut-off %	Quality of Steam at Release %	Goss "Superheated Steam" Test No.	Av. Cutoff %	Quality of Steam at Release %
1	14.18	68.7	101	16.0	82.76
3	26.05	79.0	103	25.	83.13
5	16.04	75.0	105	23.	85.83
9	20.75	79.3	109	27.	89.70
11	15.52	78.40	111	22.	88.35
13	14.44	64.50	113	23.	91.74
15	26.98	72.8	115	21.	82.58
17	13.45	70.6	117	36.	84.65
19	26.41	78.2	119	20.	87.42
21	14.38	75.2	121	34.	90.86
25	21.15	78.3	125	37.	93.50
29	13.35	68.7	129	19.	89.85
31	26.95	75.5	131	43.	93.45
33	14.19	72.9	133	20.	90.60
35	26.75	75.5	135	51.	(104.04)

APPENDIX viii.

Horse-power Developed by Locomotives.

The horsepower developed by a locomotive at any speed is $H = \frac{T \times V}{375}$, where T is the tractive force and v the speed in miles per hour. The horse-power for any speed may be found by means of plotting the Speed-Tractive Effort curve. The method used by Dr. Goss, in "Locomotive Performance", chapter XXIV is used. The net pull on a testing plant will be the maximum pull less only the force of friction between the cylinders and the driver-rims. The equation of the curve is

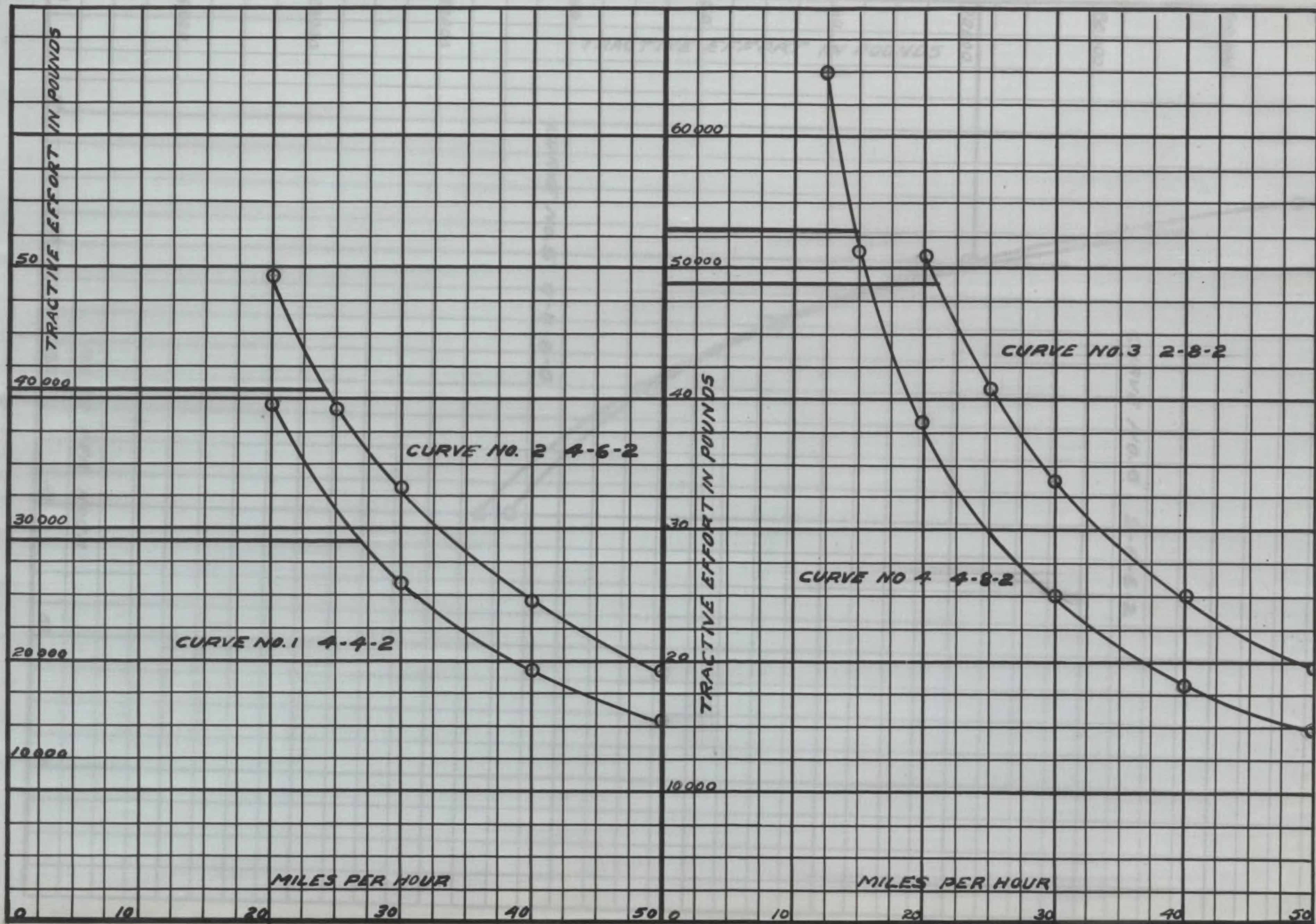
$$T = a \frac{H}{V} - b$$

where $a = \frac{E \times 375}{S}$, and $b = \frac{p\, d^2 L}{D}$

H, heating surface
E, equivalent evaporation per sq. ft. H. per hour
S, steam per I.H.P. Hour
V, speed in miles per hr.
d, diameter of cylinders
L, stroke of cylinders
D, diameter of drivers
b, machine friction loss
p, steam pressure to overcome machine friction

No.	Type.	Total Wt.	On Drivers.	Cylds.	Drivers.	H.	b.	E.	S.
1	4-4-2	231500	133,300	22x26	80	3582	629	15#	25#
2	4-6-2	272500	183,900	24x26	80	4448	750	15#	25#
3	2-8-2	274600	219,000	24x32	64	5017	1152	15#	27#
4	4-8-2	330000	239,000	29x28	62	4231	1518	12#	24#
5	0-8-8-0	445000	445,000	26,38x34	51	6629	3282	15#	25#
6	2-6-6-2	390000	323,500	23½,37x30	57	6555	2425	15#	25#

For all of these the adhesion is taken as 22% and the pressure required to overcome the machine friction is 4 pounds.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30 000
20 000
10 000
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MILES PER HOUR
0
10
20
30
40
50
60 000
50000
40
30
20
10 000
TRACTIVE EFFORT IN POUNDS
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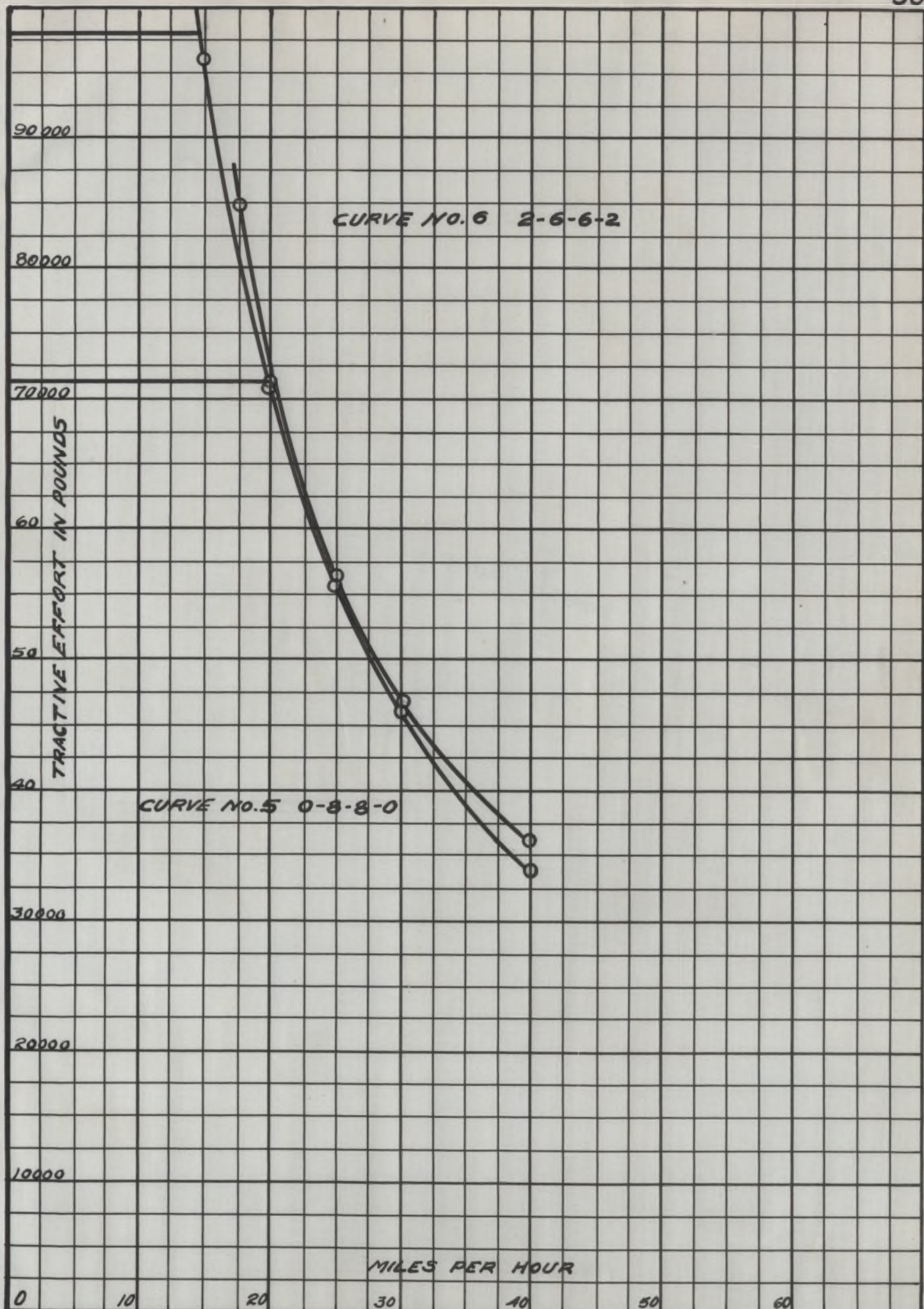
40
50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CURVE NO. 5 0-8-8-0
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MILES PER HOUR
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60
50
40
30000
20000
10000
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10
20
30
40
50
60