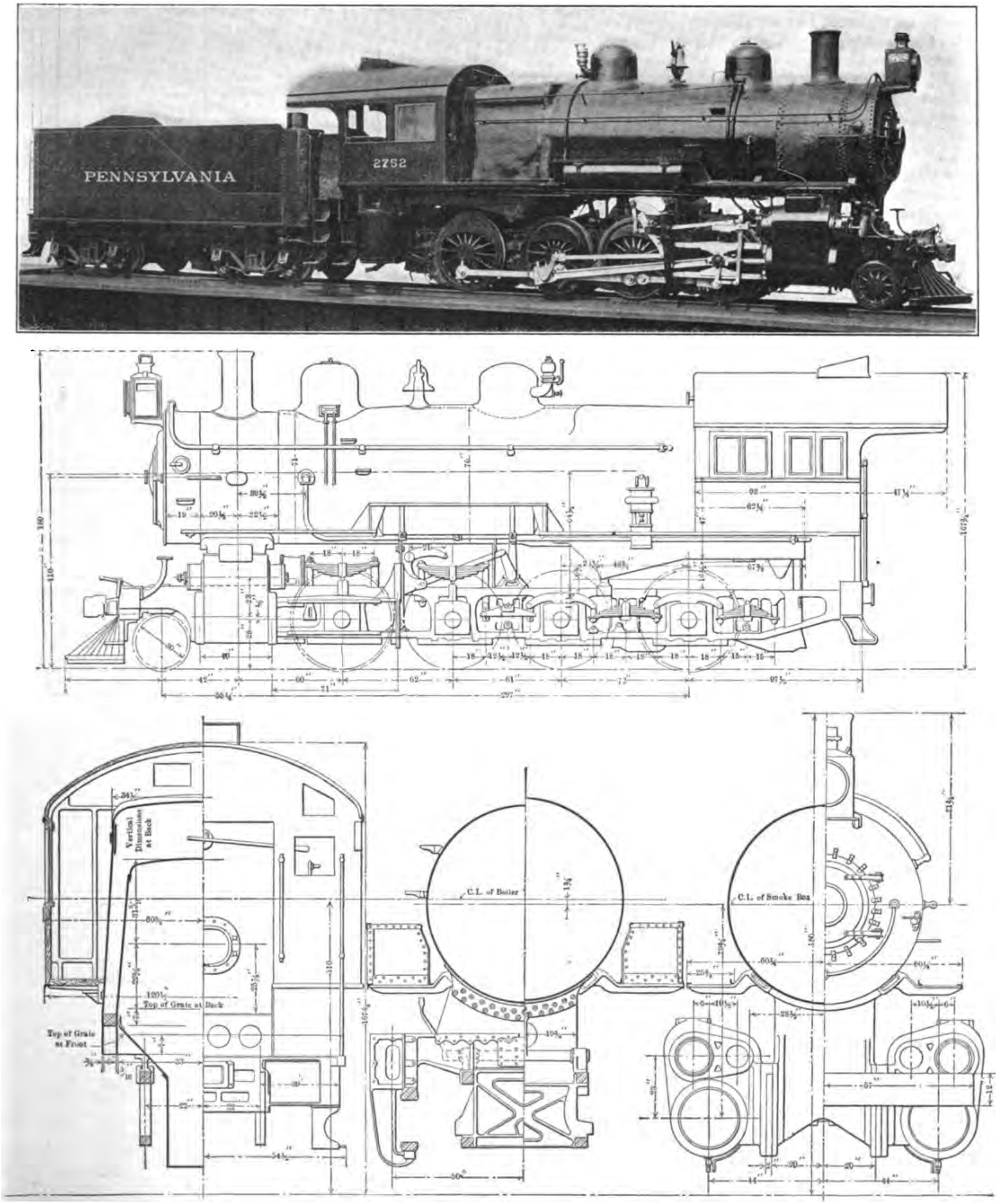
SIMPLE CONSOLIDATION LOCOMOTIVE WITH WALSCHAERT VALVE GEAR.

PENNSYLVANIA RAILROAD.

For a number of years the standard consolidation locomotive in use on the Pennsylvania Railroad has been a 22- by 28-in. simple engine with slide valves, 56-in. wheels, 70-in. Belpaire boiler, and weighing 194,200 lbs. This engine was titled Class H6A in the railroad company's classification, and one of this type was the first locomotive tested on the Pennsylvania Railroad's testing plant at St. Louis, where it gave a most satisfactory account of itself, as shown by the results published in the report of these tests issued by the company.

Recently, in considering an increase of this type of power, it was decided to apply the Walschaert valve gear and piston valves to the new engines, but in other respects to stick very closely to the dimensions and parts of H6A. An order of

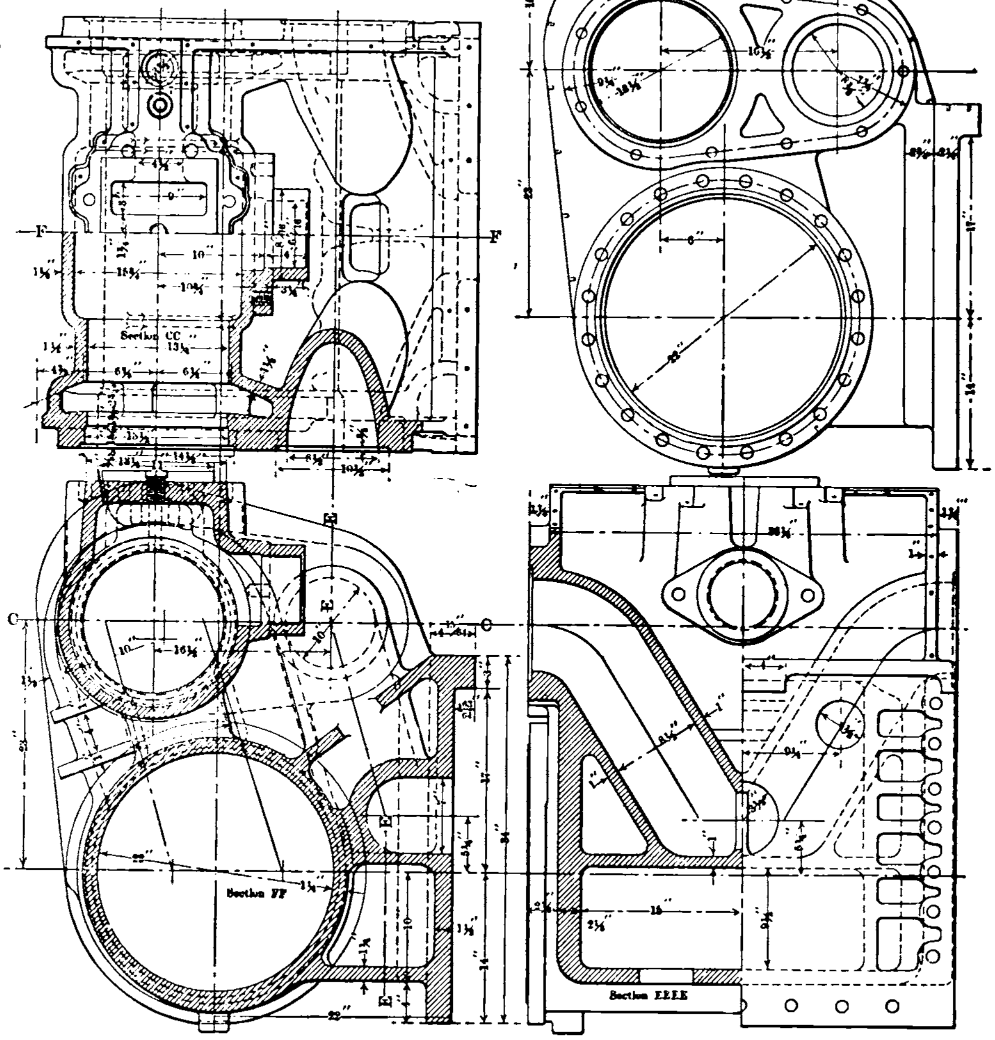


CONSOLIDATION LOCOMOTIVE, WITH WALSCHAERT VALVE GEAR.—PENNSYLVANIA RAILROAD.

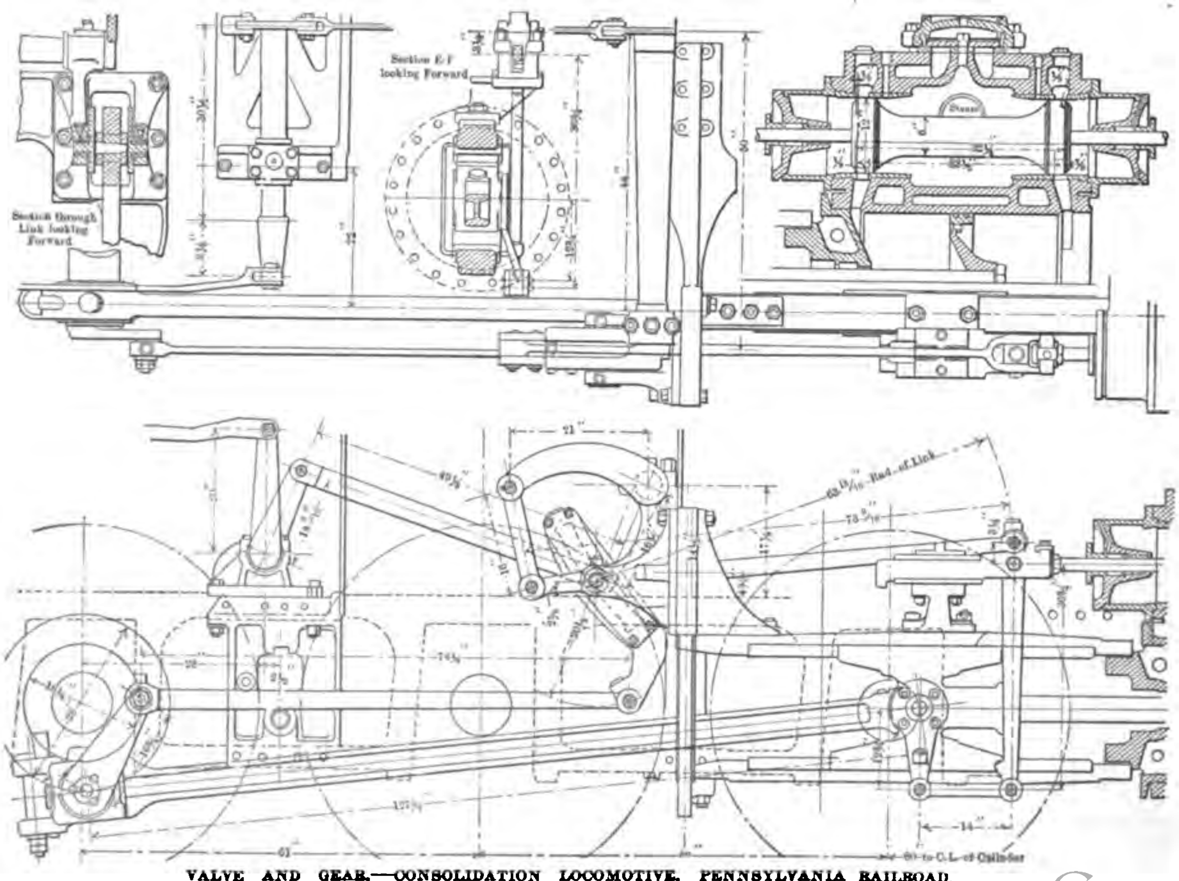
this newer type, which is known as Class H6B, has recently been built and delivered by the Baldwin Locomotive Works, and is illustrated herewith.

The change in design has increased the total weight of the engine somewhat, making this latter class weigh over 200,000 lbs., of which over 177,000 lbs. is on drivers. The tractive power figured at 85 per cent. boiler pressure is 42,200 lbs., which gives an adhesive ratio of 4.2. The engine as a whole is a simple and straightforward consolidation engine with a Belpaire boiler, and contains nothing particularly unusual outside the new features as applied to the H6B. The report of the locomotive tests at St. Louis contains a thorough description of the Class H6A, to which reference can be made for most of the details of this engine.

The new design of cylinders using piston valves contains a number of new and interesting features. As has been the custom for this type of power on this road they are cast with a separate saddle, a plate frame passing between the cylinders and the saddle, the whole construction being securely bolted together. The passage for live steam in the saddle opens above the frame connection and is continued by a short pipe with ground joints extending directly to the valve chamber. The exhaust passage, however, is in its usual place in the saddle casting, and an opening is cut in the plate frame connecting the passage from the cylinder casting. The cylinders are cast with a chamber for a 12-in.



CYLINDERS.—CONSOLIDATION LOCOMOTIVE, PENNSYLVANIA RAILROAD.



VALVE AND GEAR.—CONSOLIDATION LOCOMOTIVE, PENNSYLVANIA RAILROAD.

piston valve located above and 6 ins. outside the centers of the cylinders. The construction of this chamber is such that the steam ports into the cylinder are almost vertical and in itself it has no exhaust passage, this passage being formed in the heads, which are elongated and connect the end of the valve chamber with the opening forming the end of the cored exhaust passages, which is just inside of and on a line with the valve chamber.

The piston valve, which is somewhat longer than the stroke of the engine, has an extended valve rod which passes through the front head and is fastened at the rear to a small crosshead running in a guide bolted to the top guide bar. This crosshead has a connection to the combination lever of the Walschaert valve gear below the connection to the radius arm.

One of the governing features which led to the use of the Walschaert valve gear on this class was the fact that the removal of the eccentrics and motion work between the frames allowed space for the introduction of a more substantial and satisfactory frame bracing. In making the application of this gear it was necessary to considerably strengthen the guide yoke for carrying the large overhanging weight of the link and connections, and this has been done by making it of cast-steel in two sections, which are fastened to the frame and boiler brace as well as a heavy steel frame brace of open section which is placed between and stiffens all four bars of the frame. The reverse shaft has been left in its old location and has an upward extension arm in its centre which connects to the downwardly extending arm of the reverse shaft extending across beneath the boiler back of the guide yoke, to which the radius arms of the Walschaert gear are connected through hangers in the manner shown in the illustration.

The other features of this very powerful and well arranged locomotive will be made clear by reference to the illustrations and following table of dimensions:

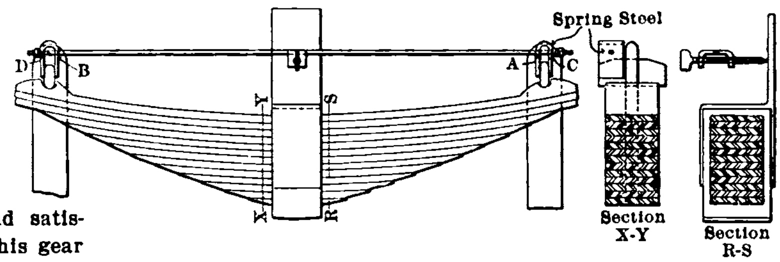
CONSOLIDATION LOCOMOTIVE, WALSCHAERT VALVE GEAR.

GENERAL DATA.	
Gauge	4 ft. 8½ ins.
Service	Freight
Fuel	Bit. coal.
Tractive power	42,200 lbs.
Weight in working order	200,380 lbs.
Weight on drivers	177,320 lbs.
Weight on leading truck	23,060 lbs.
Weight of engine and tender in working order	332,000 lbs.
Wheel base, driving	16 ft. 3 ins.
Wheel base, total	24 ft. 9 ins.
Wheel base, engine and tender	55 ft. 2¾ ins.
RATIOS.	
Weight on drivers ÷ tractive effort	4.2
Total weight ÷ tractive effort	4.75
Tractive effort x diam. drivers ÷ heating surface	823
Total heating surface ÷ grate area	58.1
Firebox heating surface ÷ total heating surface	6.4%
Weight on drivers ÷ total heating surface	62
Total weight ÷ total heating surface	70
Volume both cylinders	12.3 cu. ft.
Total heating surface ÷ vol. cylinders	233
Grate area ÷ vol. cylinders	4
CYLINDERS.	
Kind	Simple
Diameter and stroke	22 x 28 ins.
Valves	Piston
Diameter	12 ins.
WHEELS.	
Driving, diameter over tires	56 ins.
Driving, thickness of tires	3 ins.
Driving journals, main, diameter and length	9 x 13 ins.
Driving journals, others, diameter and length	9 x 13 ins.
Engine truck wheels, diameter	30 ins.
Engine truck, journals	5½ x 10 ins.
BOILER.	
Style	Belpaire
Working pressure	205 ins.
Outside diameter of first ring	71 ins.
Firebox, length and width	107⅛ x 66 ins.
Firebox plates, thickness	5-16, ⅝, ½ in.
Firebox, water space	5 ins.
Tubes, number and outside diameter	373, 2-in.
Tubes, length	13 ft. 9½ ins.
Heating surface, tubes	2,677 sq. ft.
Heating surface, firebox	182 sq. ft.
Heating surface, total	2,859 sq. ft.
Grate area	49.11 sq. ft.
TENDER.	
Tank	Waterbottom
Frame	Steel
Wheels, diameter	33 ins.
Journals, diameter and length	5½ x 10 ins.
Water capacity	7,000 gals.
Coal capacity	14 tons

SEMI-ELLIPTIC SPRINGS FOR LOCOMOTIVES AND TENDERS.

BY WILLIAM H. MUSSEY.

The proper design of locomotive and tender springs is a very important factor in securing the best results from the heavy motive power which has come into use. They are in many cases made to fit the design of frame, boiler, etc., and even then are not always given the attention they should have. Several years ago the writer made a number of tests to determine just what loads driving springs were subjected to in service. A recorder was made, as shown in the illustration, to register the deflections produced by service conditions. Ample play was allowed at C and D and a moving fit at A and B. The pointer was threaded and the tension on the plate was regulated, as desired, by hand. At times a check nut was placed on the pointer, bearing against the inverted U-shaped section of the rod, to guard against its screwing in or out.



With the recorder in place and the engine on a level track, a horizontal line was drawn to designate the static load. The springs had all been tested, before being applied, both for free height (set), static working load and a test load, so that the loads corresponding to the various heights were known. The horizontal line, which was used as a basis for the test, was found to check closely with the height obtained for the static working load by the manufacturer. The marks made on the plate, which was chalked, by the pointer gave the maximum and minimum deflections due to service conditions. From these deflections, which were measured from the base line, previously established, the corresponding actual loads were determined. The greatest value for the live load was found to be about 65 per cent. above the static working load, and the minimum 45 per cent. less than the static working load. These figures were obtained from a number of engines on which the spring rigging was considered satisfactory, and were obtained at cross-overs, switches and moving on and off turn-tables, thus representing very severe conditions.

Having gained an insight into the service demands, it was possible to more readily decide on the necessary requirements for a satisfactory spring. Introducing an arbitrary factor of safety, it was decided that in designing springs the plates should come within 3-16 in. of the horizontal for a load equivalent to twice the static working load, whenever it was possible with the conditions imposed by the locomotive construction; also that the fibre stress at this point should not exceed 130,000 lbs. per sq. in. The 3-16 in. is the construction variation allowed the manufacturers by many specifications, which state that heights for given loads must not vary more than 3-16 in. above or below those specified. With the utmost permissible variation, therefore, the plates will not pass the horizontal for a load equal to twice the static working load. The 3-16-in. allowance is entirely arbitrary, and has no relation to the total deflection of the spring, as it should. However, it is satisfactory to the manufacturer and also the railroad, and in no case in a well-designed locomotive spring have we found this allowance excessive.

By watching closely the life and service conditions of springs it was found that a good figure for the fibre stress per square inch under the static working load was between 60,000 and 65,000 lbs.; 70,000 lbs. is permissible, but 75,000 lbs. is too high, and limits the life of the spring. This latter figure means that for a possible live load the stress goes above 120,000 lbs. per sq. in., and repeated strains at that figure should be avoided with the average open-hearth steel. Our experience with springs has proved this conclusively. We specify a test load height, and fix this load so it produces a fibre stress of about 120,000 lbs. per sq. in. This is a precaution to insure a high-grade spring. The stress of 130,000 lbs. per sq. in. we might call our ultimate figure; we don't expect springs (open-hearth steel) to stand loads exceeding this even at rare intervals.

These springs are designed by the Reauleaux formulæ for semi-elliptic springs:

$$P = (\text{static load on one end}) = \frac{Snbh^2}{6L}$$

L = ½ span in inches less ¼ width of band.
S = fibre stress per sq. in.
b = width of plate in inches.
h = thickness of plate in inches.
n = number of plates.

FOR DEFLECTIONS.

$$D = \frac{6PL^3}{Enbh^3}$$

Equating for value of P.

$$D = \frac{SL^2}{Eh^3}$$

E = modulus of elasticity = 29,400,000.

There may be some question about the deduction from L of one-quarter the width of the band, but this is theoretically correct as a study of the action of the leaves of a spring will show. Springs designed on this basis will be guaranteed by the spring manufacturers for one year's service, and in practice they far exceed it. Springs so designed are still in service at the end of two years and show no signs of failure, and we confidently expect a continued satisfactory service for some time to come. The lower the fibre stress the longer the life of the spring within reasonable limits. To indicate the severe service that these springs are subjected to, it might be added that they are used on a road which has very little stone ballast.

In designing springs it should be the aim as nearly as possible to have equal deflections under the working loads for those springs which are equalized together. When possible short springs should be avoided and also narrow ones. Long springs with wide plates give the most even deflections, taking up shocks more effectively. On short springs, 24, 26 or 28-in. spans, it is better to use thin plates ¼ to 5-16 in. thick. As the length of span increases the thickness of plate should also be increased; for a 48-in. span ⅝-in. or even 11-16-in. leaves are not excessive. In one case a spring with a span of 26 ins. and ⅜-in. leaves was equalized with a spring having a 38-in. span and 7-16-in. plates and gave poor service, although the fibre stress was comparatively low. By reducing the thickness of the plates and with about the same fibre stress much better results were obtained. The aim is to increase the deflections for given loads on short springs, where it is naturally small, and to decrease them on long springs, where it is, on the other hand, excessive. By examining the formula for deflections $D = \frac{SL^2}{Eh^2}$ we find that deflections for a given load vary directly as the square of L, and inversely as the square of the thickness. From this the relation that thickness bears to the length is apparent.

Good practice is to keep the deflection from the free height to the static working load between 1½ and 2¼ ins. where possible. The former figure covers short springs, the latter longer ones; conditions, however, do not always permit this. The width of the bands should be about one-tenth the span of the spring. All of the foregoing applies more especially to driving springs. For engine truck and tender truck springs the service conditions are not as exacting. The springs may be stiffer, the fibre stress decreased and the life of the spring thus increased; in no case, however, is the use of leaves over 11-16 in. thick recommended. To reduce the fibre stress it is better to use wider plates, if possible, even though clearances require them to be reduced in width at the hanger.

FORGING AT THE COLLINWOOD SHOPS.

LAKE SHORE & MICHIGAN SOUTHERN RAILWAY.

On page 143 of our April issue we illustrated and described a number of forgings which are being manufactured in the forging and bulldozing machines and under a Bradley hammer at the Collinwood shops of the Lake Shore & Michigan Southern Railway. The value of these machines depends entirely on their being properly equipped with special tools and dies for the different parts to be manufactured, and if these are provided the rate at which forgings can be turned out is usually limited only by the facilities for heating the iron. Not only is it thus possible to greatly increase the output of the shop, but the grade of work turned out is superior to that done by other methods. In this article, which supplements the earlier one, the dies and formers for making several of the more intricate forgings are illustrated.

FIG. 1.—GENERAL VIEW OF THE MANUFACTURING SECTION OF THE COLLINWOOD SMITH SHOP.

One of the illustrations shows a general view of the manufacturing section of the smith shop. In the foreground, to the right, is a No. 3½ Ajax forging machine, the largest forging machine used in the shop. Just to the rear of it is a No. 6 and also a No. 8 Williams and White bulldozer. The No. 8 machine is served by a jib crane, so that the heavy cast-iron formers can readily be transferred from the storage platform, indistinctly shown in the background, to the machine. At the right and just opposite the No. 6 bulldozer is a large punch and shear. A 200-lb. Bradley hammer, to the left and