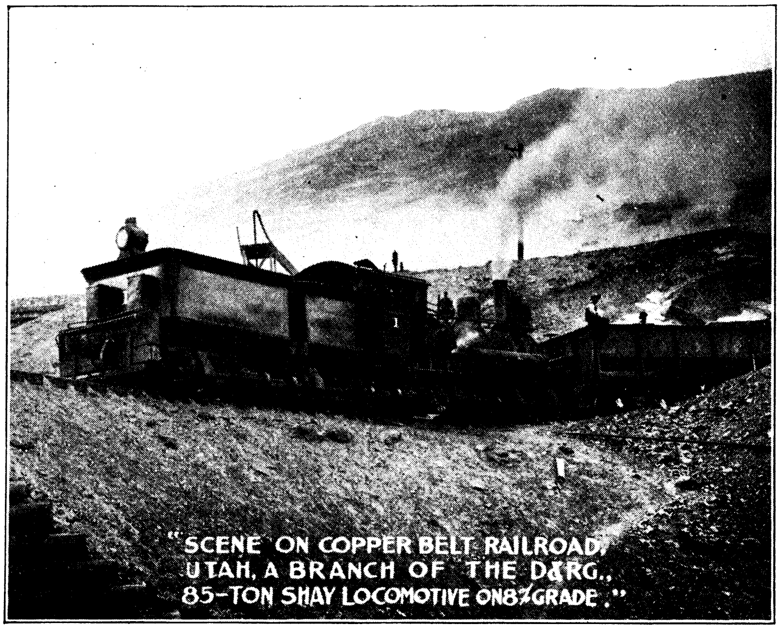
The LOCOMOTIVE WORLD



LOGGING PLANTATION MINING
INDUSTRIAL AND
STANDARD RAILROAD MOTIVE POWER

THE LOCOMOTIVE WORLD

Volume 5 JULY, 1912 NUMBER 3

THE LOCOMOTIVE WORLD.

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THE FRANKLIN TYPE AND PRINTING COMPANY

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NOTICE TO ADVERTISERS.

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THE FRANKLIN TYPE AND PRINTING COMPANY

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BAD STEAMING

A gratifying sign of the times among roundhouse men is the fact that we do not hear so much of bad steaming in locomotives as formerly. The amount of ignorance which existed in regard to the causes that led to the defect would be difficult to overestimate. Instead of making a systematic search for the trouble and, if possible, rectifying it, the locomotive became stamped with a character similar to that of certain individuals who are known to possess sundry moral and intellectual defects, but nobody takes time to make a sustained effort to set them right. In the case of the bad steaming locomotive, the experiments that have been so successfully carried on in regard to the appliances used in the smokebox or front end of the locomotive have brought about the improved condition in regard to steaming. As a general rule defects in steaming are now discovered and remedied with a degree of promptitude that leaves little to be desired. The careful experiments on the part of locomotive designers have brought the parts to a degree of symmetry and proportion so nearly perfect that a readjustment of any particular part is comparatively easy.

Defects in steaming may now be promptly looked for in the smokebox attachments. It should be seen that the exhaust pipe and nozzle are securely held in place, and that the exhaust nozzle is set exactly in line with the center of the smokestack. The diameter of the nozzle should be such as to allow the exhausted steam to fill the smokestack as completely as possible. An exhaust jet failing to fill the stack fails to produce the vacuum necessary for furnishing a strong and equable draught on the fire. A jet expanding beyond the limits of the smokestack, although less pernicious, has a disturbing effect on the fire, with a corresponding shortcoming in the generation of steam.

The same remarks apply with equal force to the adjustment of the petticoat pipe, if such forms part of the smokebox equipment, and it should be borne in mind that it does not follow that even with the most exact degree of careful designing in the original construction these appliances will continue to retain their just alignment for any considerable length of time. The variations in temperature and the incessant, though intermittent, blasts on the heated fastenings tend to distortion of the parts, and the divergence from their correct positions cannot be discovered by a mere casual glance. It does not take much time to level the engine and drop a plumb line in the smokestack, when any variation from the true adjustment will be readily revealed.

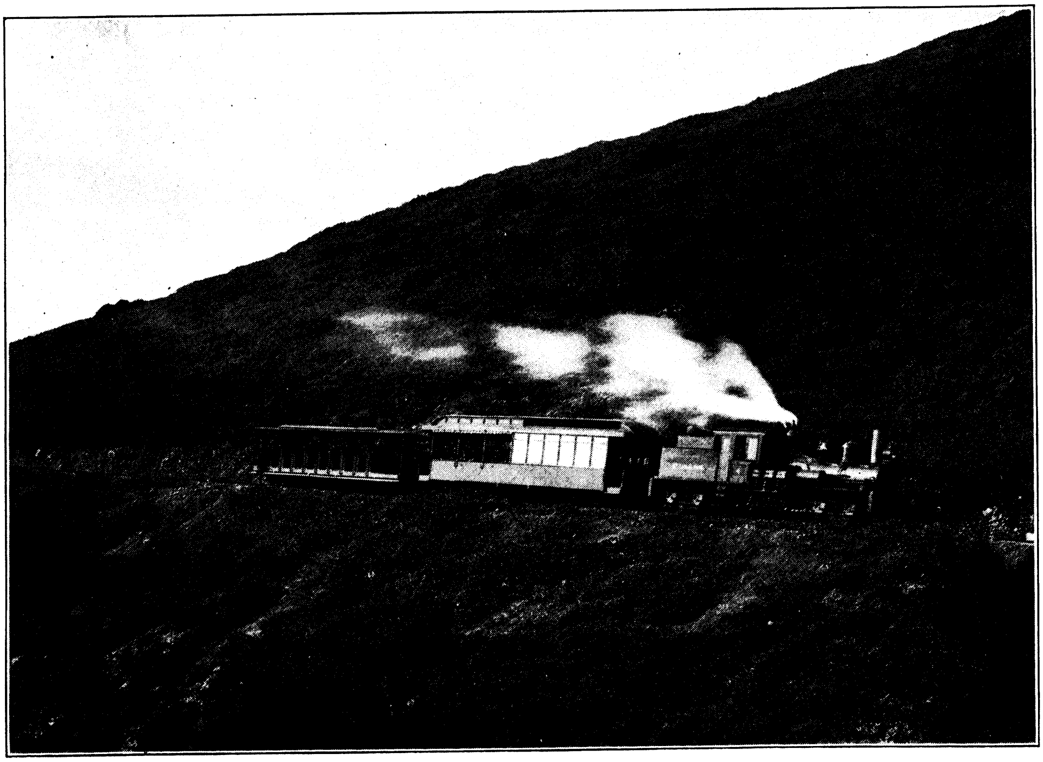
The diaphragm or deflector plate, although not so readily moved as the petticoat pipe, is also of great importance in its adjustment. If set too low the draught will be stronger in the lower flues, and if high set the draught will be more marked in the upper flues. It is desirable that the draught should be as equable as possible, and if the equalization of the draught is maintained and the parts in the smokebox kept in their proper position, little remains to be done other than that the netting should be kept free from obstructions, which accumulate rapidly.

It may be added that leaks either from the steam pipes or from the outer air, by reason of a defective joint in the front casting, or smokebox door, or smokestack base, all contribute in causing a marked defect in the steaming qualities of the locomotive, and when any of these are discovered they should not be set down as organic defects in construction, but should be looked upon sensibly as the natural results of the strenuous service which these parts of the complex mechanism are constantly called upon to bear, and the defects should be promptly and intelligently remedied.—*Railway & Locomotive Engineering.*

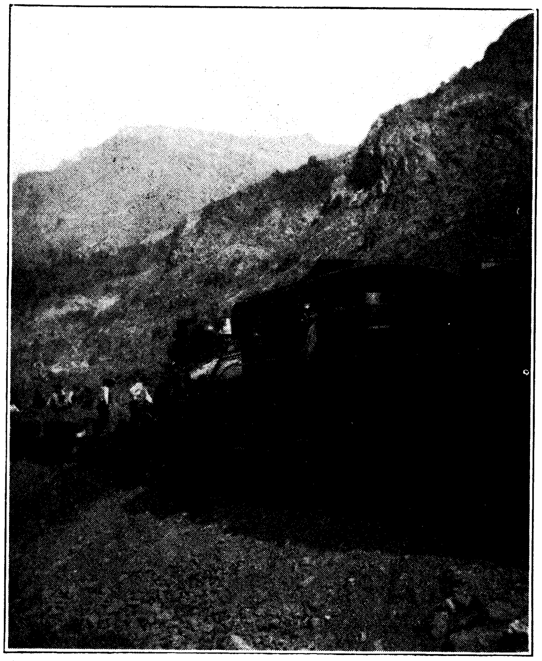
HERE, THERE AND EVERYWHERE

JUDGING from the title above, it would no doubt be difficult for our readers to form an idea of what is to follow. However, usually there is not so much in the name as there is in the goods which the name represents. Therefore, that is the case in this instance.

We are reproducing photographs of railroad operations taken in various parts of the country. These will show conditions encountered by some of our friends who are using the Shay Geared Locomotive.

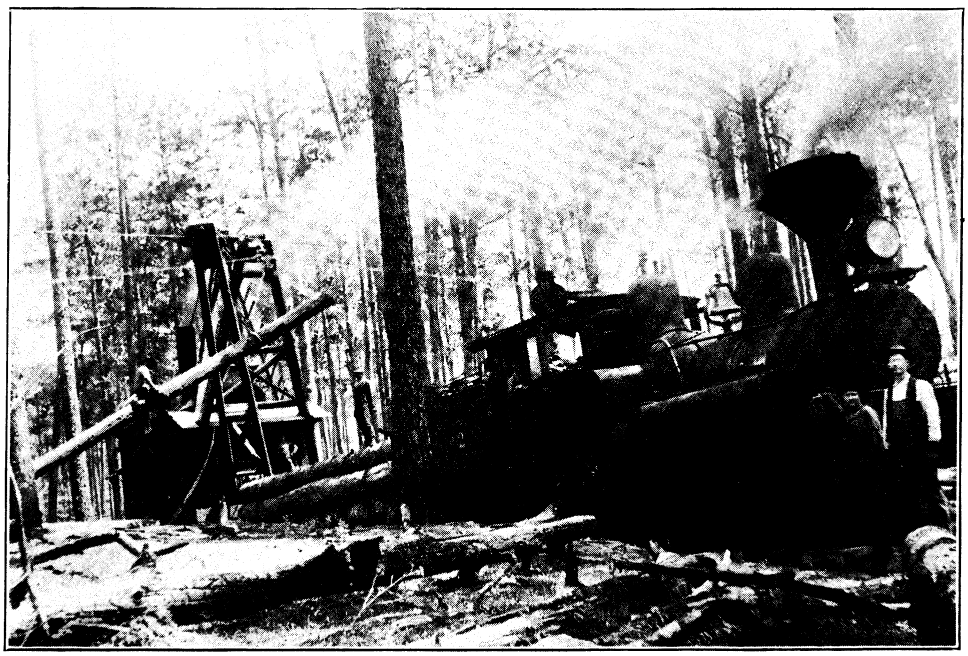


THE ABOVE VIEW SHOWS TRAIN ON THE MILL VALLEY AND MT. TAMALPAIS SCENIC RAILWAY. TRAIN IS ON 7% GRADE. THE AIR-LINE DISTANCE FROM MILL VALLEY, AT THE BASE OF THE MOUNTAIN TO THE SUMMIT IS ONLY THREE MILES, BUT AS AN ASCENT OF 2,500 FEET WAS TO BE MADE THE ROAD HAD TO TAKE A TORTUOUS COURSE AND IS 8.19 MILES IN LENGTH. THE GRADES AVERAGE 5% AND MAXIMUM 7%. THE ROADBED IS CUT IN THE SOLID ROCK OF THE MOUNTAIN-SIDE ALL THE WAY. THERE ARE NUMEROUS CURVES, IN FACT THIS ROAD IS KNOWN AS THE "CROOKEDEST RAILROAD IN THE WORLD." IT IS OPERATED BY SHAY GEARED LOCOMOTIVES, AND THEY NOW HAVE IN SERVICE FIVE OF 37 TON SIZE.

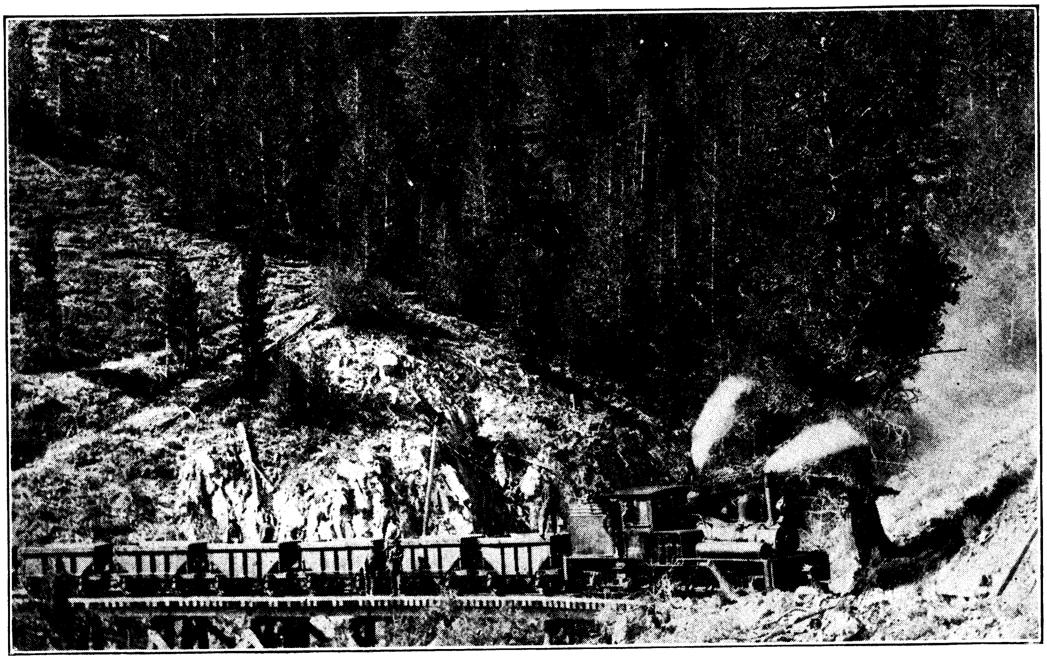


CONSTRUCTION WORK ON THE TRANSANDINE RAILWAY OF CHILE, ABOVE RIO BLANCO, 70 TON SHAY LOCOMOTIVE PUSHING 60 TONS ON 8% GRADIENT.

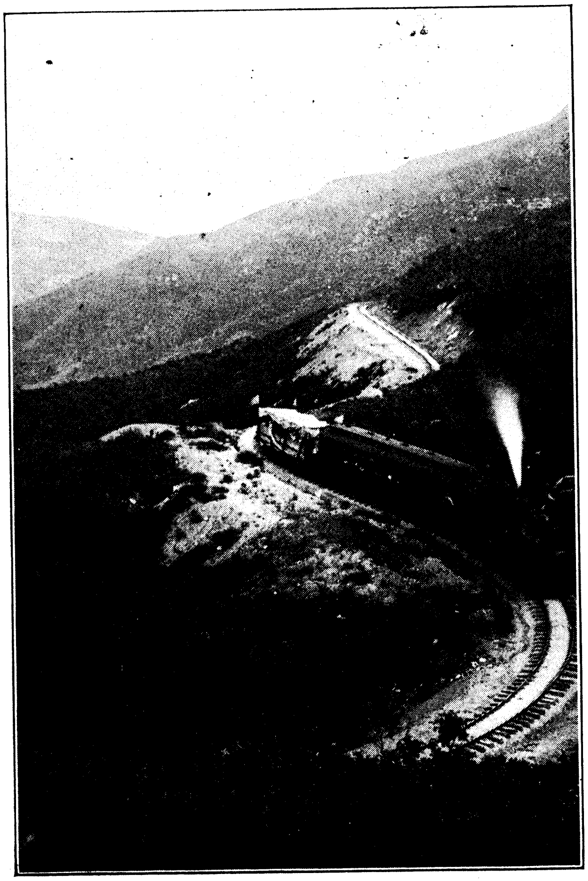
SCENE OF A 24 TON SHAY LOCOMOTIVE BUCKING SNOW ON THE EMPIRE COPPER COMPANY'S MINING RAILROAD IN IDAHO. NO MATTER HOW SEVERE THE STORM THE SHAY KEEPS THE ROAD OPEN.



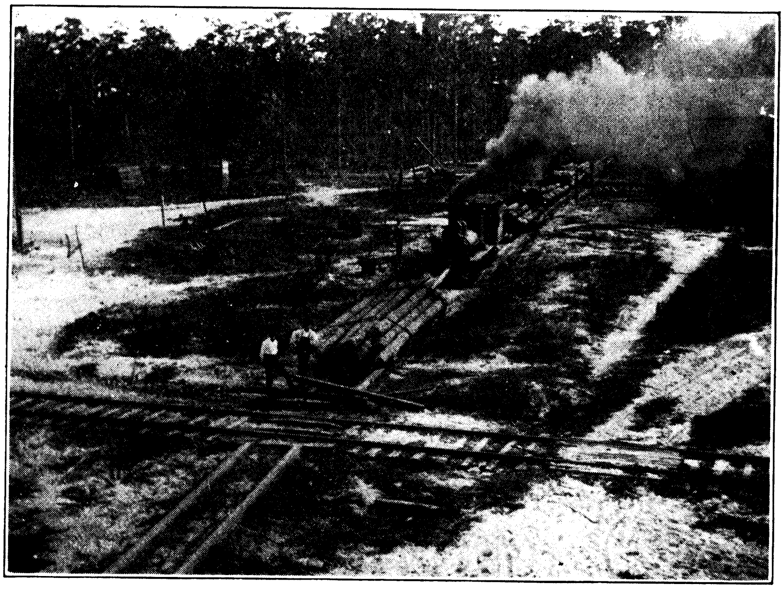
VIEW OF LOGGING OPERATIONS OF THE LOUISIANA LOGGING CO. SHAY LOCOMOTIVE NO. 724. THIS LOCOMOTIVE HANDLES 200,000 FT. PINE LOGS DAILY ON 60,000 POUND CAPACITY FLAT CARS 4 MILE HAUL, 8% GRADES, 15 DEGREE CURVES, ALSO DOES SWITCHING FOR STEAM SKIDDER.



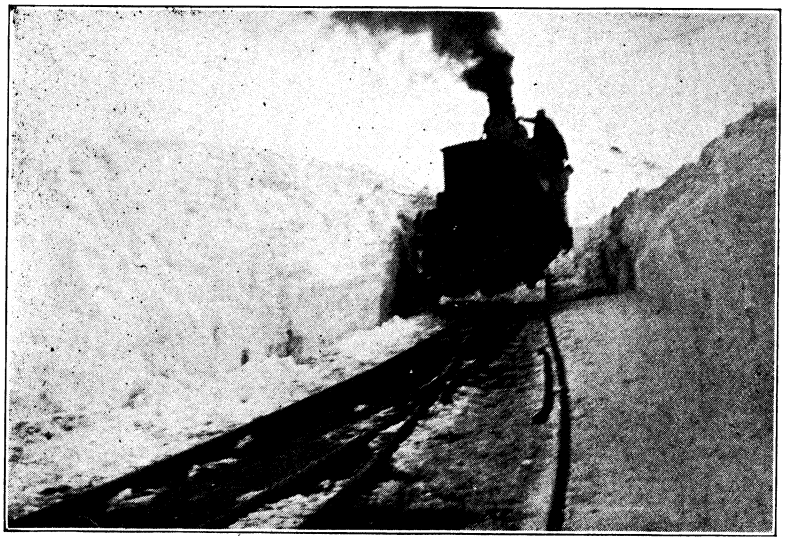
MINING TRAIN ON SHARP CURVE, EMPIRE COPPER CO'S MINING RAILROAD IN IDAHO.



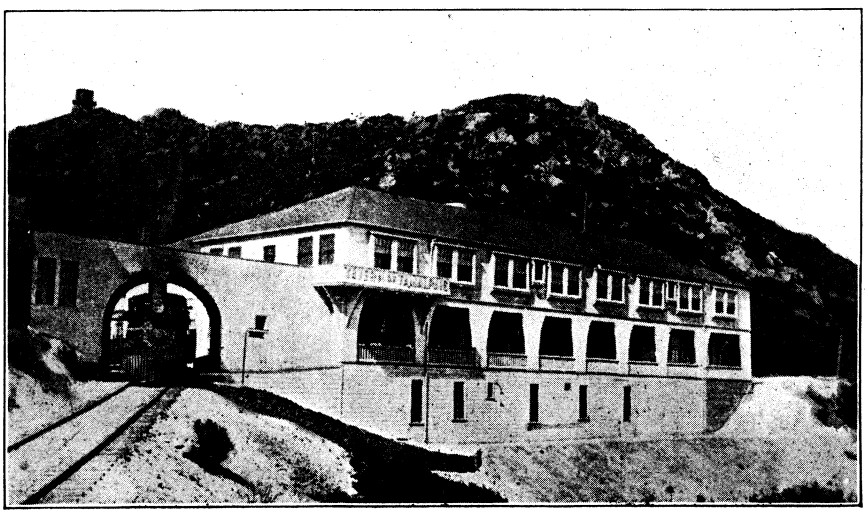
EXCURSION TRAIN ON THE UINTAH RAILWAY NEAR THE BOTTOM OF THE STEEP GRADE—7½%—ON THE EAST SIDE OF THE MOUNTAIN. TRAIN IS PULLED BY 50 TON SHAY LOCOMOTIVE. THIS RAILROAD IS KNOWN AS THE "LINE OF UNIMAGINABLE SCENES."



LOGGING OPERATION IN FLORIDA. A WOODEN TRACK BEING USED SUCCESSFULLY. AN 18 TON SHAY LOCOMOTIVE PULLING THE TRAIN.



A SHAY LOCOMOTIVE ON BAXTER PASS AFTER A STORM. AS THE NAME IMPLIES, BAXTER PASS IS THE GATEWAY OF THE UINTAH RAILROAD OVER THE SUMMIT OF THE BOOK CLIFF RANGE. THIS IS A PRECIPITOUS RANGE EXTENDING UPWARDS OF TWO HUNDRED MILES IN AN EAST AND WEST COURSE FROM THE MOUNTAINS OF COLORADO TO THE WASATCH AND UINTAH RANGES IN UTAH.



ONE OF THE SHAY LOCOMOTIVES OWNED BY THE MILL VALLEY AND MT. TAMALPAIS SCENIC RAILWAY IN THE ARCH OF TAVERN OF TAMALPAIS AT THE SUMMIT OF MT. TAMALPAIS.

Federal Boiler Inspection

Rules and Formulas Covering Calculations Necessary in Filling Out Boiler Form No. 4.

BY F. P. ROESCH IN LOCOMOTIVE FIREMEN AND ENGINEERS MAGAZINE.

(Continued from June issue.)

BACK HEAD BRACES

As the requirements of Boiler Form 4 call for the maximum stresses on the weakest brace, the back head braces, as well as the tube sheet braces, must be considered. Finding the area of the back head not supported by the stay bolts is a slightly different proposition from finding that of the tube sheet, as while in both cases we deal with segments of circles, yet in the back head calculations we have segments or semi ellipses within segments, depending on the type of firebox used. (These figures are called lunes.) Fig. 2 illustrates the back end of the boiler we are working with.

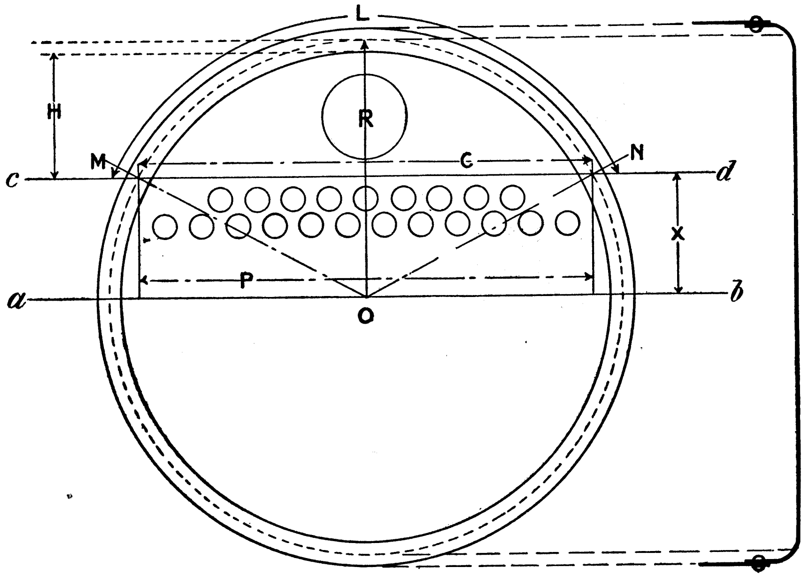


FIG. 3

It will be noticed that in this figure the crown sheet of the firebox is elliptical in form and a part of the ellipse projects above the chord of the segment of the circle representing the area to be braced. The area of this projection must therefore be subtracted from that of the segment. While this problem presents no difficulty to a mechanical engineer or to one who has had the benefit of a technical training, yet as the rules here given are intended solely for those employed on small roads that do not employ a mechanical engineer, men who perhaps are not versed in mensuration and higher arithmetic, we believe it best to explain each method and, where possible, suggest easier methods than those found in text books. In Fig. 2 the back head is flanged to a 4-inch radius. The radius, therefore, of the area to be braced would be the radius of the head less twice the radius of the flange. Eighty-four inches diameter =42 inches radius, which—8 inches, =34 inches =radius of segment. Draw the dotted line circle as shown in Fig. 2 to this radius. Now bisect this circle with the line *A A*. Parallel to this line and intersecting the line representing the top of the crown sheet draw the dotted line *B B*. From the intersection of the line *B B* at the dotted circle erect the dotted lines *M* and *N* at right angles to the lines *A A* and *B B*, and at the intersection of the lines *M* and *N* with the line of the crown sheet, draw the line *C C* parallel to lines *A A* and *B B*. This gives you a parallelogram whose height is equal to the distance between lines *A A* and *C C* and whose length is equal to the distance between lines M and N, or P, and above it the semi ellipse, whose height is equal to the distance between lines B B and C C, and whose length equals the distance between lines M and N, or P. The area of an ellipse is found by multiplying the two diameters by each other, and this by .7854. One-half of this would be the area of a semi ellipse.

In the boiler in question we find the diameter of the segment to be 68 inches.

The length of the chord of the arc = P =57½ inches.
The height of the arc =O =20 inches.
The height of the parallelogram =8½ inches.
The height of the ellipse =8½ inches.
The length of the ellipse and parallelogram =57½ inches.

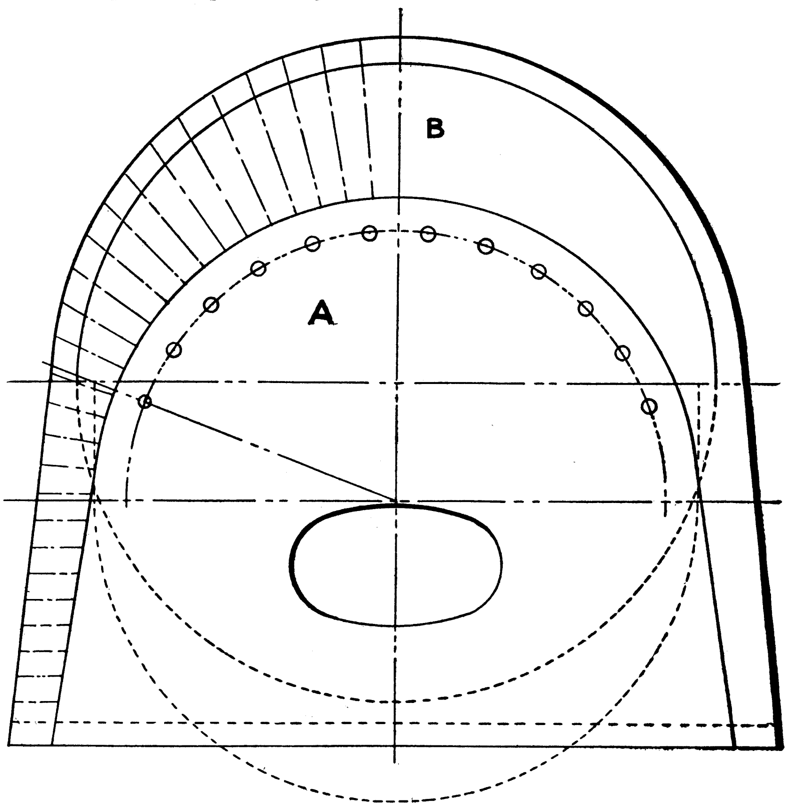


FIG. 3-A

If we now take half the area of the dotted semi-circle and subtract from this the areas of the parallelogram and the ellipse, the remainder will equal the area of the segment to be braced.

Dropping small fractions we have:

62x.7854 =3631.7, and 3631.7 ÷2 =1815.85 =area of half-circle.

Area of parrallelogram =57.5x8.5 =488.75 square inches.

$$\text{Area of semi ellipse} = \frac{57.5\text{x}8.5}{2} = 244.37 \text{ square inches.}$$

Then area of segment =1815.85—(488.75+244.37)=1815.85—733.12 =1082.7 square inches.
The total pressure to be supported =1082.7×175 =189,472 pounds.
We find the back head is braced with 11 round braces 2 inches in diameter.
Area of each brace =2x.7854 =3.1416 square inches.
Area of all braces =3.1416x11 =34,5576 square inches.
Stress on braces per square inch of area =

$$\frac{189{,}472}{34.5576} = 5{,}482.7 \text{ lbs., say } 5{,}483 \text{ lbs.}$$

GUSSET BRACES

The rule for finding the stress on gusset braces is the same as that for round or rectangular braces, viz.:

Area of surface supported in square inches multiplied by pressure, and this divided by the depth of the web at the narrowest part in inches multiplied by thickness of web in inches.

Thus if the web is 10 inches wide at the narrowest part and formed of ½-inch iron, the area would be 10x½ =5 square inches.

Fig. 3-A shows a section through furnace end of a radial stayed boiler of the wide firebox type. To find the area of the segment supported by braces, first reduce the radius of the outer circle by the radius of the flange, as before; then find the area of a half-circle (B) of this radius, and from this subtract the area of the segment (A) of the circle representing that part of the firebox projecting ahove the center line; the remainder gives the area of the segment supported by braces. The stress on braces can then be found in the usual manner.

SHEARING STRESS ON RIVETS PER SQUARE INCH

This applies to all rivets, both those fastening the braces and those in the seams; the maximum stress is always to be taken, as a boiler is only as strong as its weakest part.

To find the shearing stresses on brace rivets, divide the area supported by each brace (multiplied by the pressure) by the area of the rivets in each end of the brace. In calculating the area of a driven rivet the area of the hole is always taken, as the rivet, after driving. is supposed to fill the hole. The hole for all rivets above ¾-inch in diameter is $\frac{1}{16}$-inch larger than the rivet size; therefore, when figuring rivet areas, use $\frac{13}{16}$ for ¾, $\frac{15}{16}$for ⅞, etc. These sizes will be used in all following examples.

In the boiler in question we find the ends of the braces where rivets are in shear are fastened by four ⅞-inch rivets. The area of a ⅞-inch rivet hole is .690 square inch, and of four rivets is 2.76 square inches. In the previous example we found the greatest stress on any one brace to be on the front tube sheet braces and equal to $\frac{135975}{10} = 13{,}597$ pounds, and therefore the shearing stress on the rivets in the end of the brace is 13,597 ÷ 2.76 =4,926 pounds per square inch. (These rivets are in single shear.)

To find the shearing stress on rivets in the longitudinal seams (where rivets of different diameters are used in different seams the stress on each seam must be considered):

Rule:

$$S = \frac{DxPxa.r.}{A}$$

In which S =Shearing stress.
D =Diameter of course taken.
P =Boiler pressure.
a.r. =Area supported by each rivet.
A =Area of rivet.

Note.—Where cylindrical courses are not parallel, as in courses 2 and 3 (Fig. 1), the greatest diameter must be taken. Considering now the longitudinal seam in course 3:

In order to find the area supported by each rivet we must consult Fig. 4, and also Table 1. Fig. 4 shows greatest pitch of outside row of rivets =2 B; Table 1 shows B in seam 3 =3¼ inches; then pitch of outer row of rivets =3¼x2 =6½ inches. By dropping lines down through the centers of two rivets in outside row to the seam, we find these lines pass through the centers of two rivets in the second row above the seam, and enclose 5 rivets, 2 in the first, 1 in the second and 2 in the third row of rivets; therefore, we find the space enclosed by these lines, 6.5 inches, is supported by 5 full and 4 half-rivets, or a total of 7 rivets, and the area supported by each rivet =6.5 ÷7 =.928 square inch =value of a.r.

By consulting Table 1 we find that the rivets in seam 3 are $1\frac{1}{4}$ inches in diameter, and consequently have an area = 1.3125 x .7854 =1.35 square inches.

The diameter of the boiler at course 3 is 84 inches, and the pressure is 175 pounds; therefore, substituting values for letters in the formula we have for stress on rivets in seam 3,

$$S = \frac{84 \times 175 \times .928}{1.35} = 10{,}104 \text{ lbs. per square inch.}$$

Seam No. 2.—Referring to Table 1 we find the pitch of rivets for seam 2 the same as for seam 3 =$3\frac{1}{4}$ inches, diameter of rivets the same as for seam 3 =$1\frac{1}{4}$ inches, greatest diameter of course 80 inches. Therefore, substituting values we have for stress on rivets in seam 2,

$$S = \frac{80 \times 175 \times .928}{1.35} = 9{,}623 \text{ lbs. per square inch.}$$

Seam No. 1.—Referring to Table 1 and Fig. 4, we find the greatest pitch of the rivets in this seam to be 6 inches (2 B outside row), but as there are also 7 (5 whole and 4 half) rivets in this

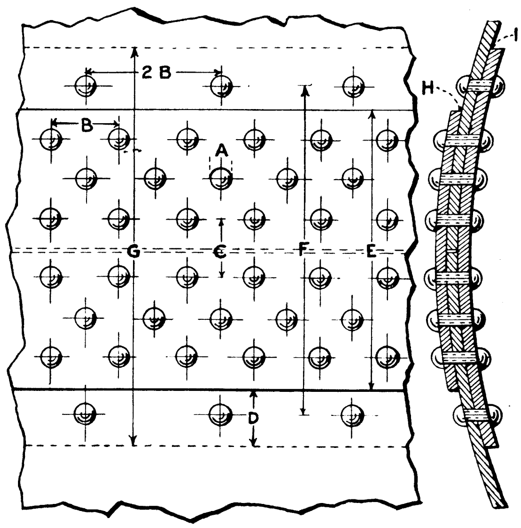


FIG. 4. Longitudinal Seams, Quadruple Riveted Butt Joint.

space we find the value of a.r. to be 6 ÷ 7 =.857. As the rivets are $1\frac{1}{8}$ inches in diameter the area ($1\frac{3}{16}$ hole), = 1.107 square inches. The greatest diameter of this course is 72 inches. Then, substituting values, we have, for seam No. 1,

$$S = \frac{72 \times 175 \times .857}{1.107} = 9{,}754 \text{ lbs. per square inch.}$$

CIRCUMFERENTIAL SEAMS, STRESS ON RIVETS.

As the greatest portion of the ends of the boiler is braced lengthwise by means of the tubes and braces, the stress on the rivets in the circumferential seams is equal to the unsupported area multiplied by the pressure, divided by the number of rivets in the seam divided by the area of the rivets. We will first consider seam No. 7 (Fig. 1). In previous calculations to determine the stress on braces, we found that the area represented by the circle having a diameter of 66 inches, was supported by the tubes and braces. This would then leave an area to be supported by the circumferential seam rivets equal to the area of boiler, 72 inches in diameter, minus area of circle, 66 inches in diameter.

72 x.7854 =4,071.5 square inches.
66 x.7854 =3,421.19 square inches.
4,071.5—3,421.19 =650.31 square inches =area to be supported.
650.31x175 =113,804.25 pounds, total pressure.

By consulting Table 1 we find this seam is single riveted with ⅞-inch rivets, spaced 2 inch pitch. Therefore, we would have 113 rivets in this seam. The area of a ⅞-inch rivet is .69 square inch; then 113,804 ÷ 113 = 1,007 pounds = stress on each rivet. 1,007 ÷ .69 = 1,459 pounds stress per square inch of area on rivets in seam No. 7.

Seam No. 4.—As shown in Table 1, this seam is a double riveted lap joint with zigzag spacing of rivets. As can also be seen by referring to Fig. 1, this joint must support in addition to the area supported by seam 7, one-half the area supported by the front tube sheet braces, for, as shown in Fig. 1, one-half of the front tube sheet braces are attached to the first boiler course ahead of seam 4, while the other five are attached to the next course, thus dividing the stress of supporting the area of the front tube sheet supported by braces, between seams 4 and 5.

In the previous calculations we found that seam 7 supported an area of 650.31 square inches, while the front tube sheet braces supported an area of 777 square inches. Then 777 ÷ 2 = 388.5, and 388.5 + 650.31 = 1,038.81 square inches = total area supported by rivets in seam 4. The boiler

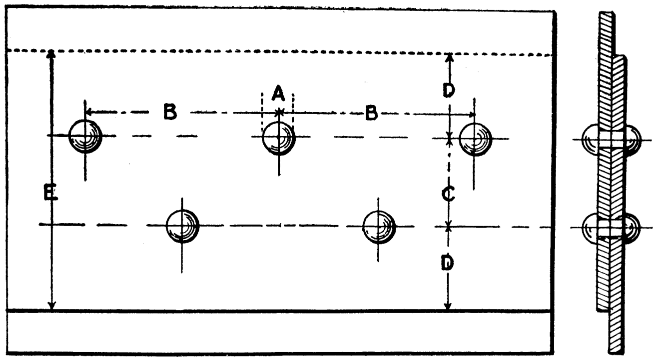


FIG. 5. Circumferential Seams, Double Zigzag Riveted

at seam 4 is 72 inches in diameter, and as the rivets are staggered and spaced 3-inch pitch, we find we have a total of 150 rivets in this joint. By consulting Table 1 we also find the rivets are 1⅛ inches in diameter. The area of a 1⅛-inch rivet is 1.107 square inches; then, 1,038.81x175 ÷ 150 = 1,212 pounds, = total stress on each rivet; 1,212 ÷ 1,107 = 1,094 pounds shearing stress on each rivet in seam 4.

Seam No. 5.—By referring to Fig. 1 and Table 1, we find this is also a double riveted lap joint with rivets 1¼ inches in diameter spaced 3½-inch pitch. We also find that the rivets in this seam must support the area supported by seam 7 plus the total area supported by the front tube sheet braces plus the area due to the difference in diameters of the first and second courses. Or, in other words, these rivets must support an area equal to the area of a circle whose diameter is equal to the greatest diameter of the course (80 inches), less the area supported by the tubes. We then have area of tube sheet minus area of segment supported by the front tube sheet braces = area supported by the tubes. As the tube sheet is 72 inches in diameter, then 72x72x.7854 = 4,071.50, and 4,071.50—777 = 3,294.5 square inches = total area of front tube sheet supported by the tubes. As the diameter of the boiler at this seam is 80 inches, then 80x80x.7854 = 5,026.56, and 5,026.56—3,294.5 = 1,732 square inches = area supported by the rivets in this joint. This area multiplied by the pressure and divided by the number of rivets in the joint (which we find to be 143) gives total stress on each rivet. This divided by area of rivet gives stress per square inch. Then, 1,732x175 = 303,100, and 303,100 ÷ 143 = 2,119 pounds stress on each rivet. The area of a 1¼-inch rivet is 1.35 square inches, and 2,119 ÷ 1.35 = 1,569 pounds stress on each rivet in seam 5 per square inch of area.

Seam No. 6.—These rivets are subject to the same stresses as those in 5 seam, plus the difference due to increased area of boiler at that seam. In calculating the stresses on this seam we must also take into consideration, however, the fact that the braces supporting that portion of the back head not supported by stay bolts are riveted to the roof sheet and back of the joint in question; therefore, we must take as the maximum stress the greater of these two, multiplied by the pressure, By consulting Form 4 we find the diameter of the boiler at seam 6 to be 84 inches. The area of a circle 84 inches in diameter is 84 x.7854 =5,541.78 square inches, and 5,541.78—3,294.5 (area supported by the tubes) =2,247.28 square inches supported by seam 6, due to pressure against front tube sheet and taper course of boiler. In calculating the stress on back head braces we found the total area supported by these braces to be 1,082.7 square inches. This latter area, however, does not include that portion of the back head supported by the flanging of the head. We found in former calculations that the diameter of the segment of the back head supported by braces was 68 inches. As the diameter of the boiler head is 84 inches, we have the half of an annular ring whose outside diameter is 84 inches, and whose inside diameter is 68 inches, as that part of the back head supported by the flanging of the head, the stress on which area, however, also comes on the rivets in seam 6;

then, $\frac{(84 \text{ x}.7854)-(68 \text{ x}.7854)}{2}$ =area supported by flange.

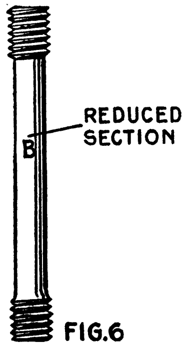


FIG.6
REDUCED STAY BOLT

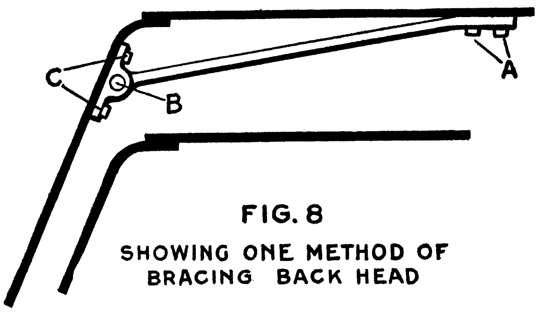


FIG. 8
SHOWING ONE METHOD OF BRACING BACK HEAD

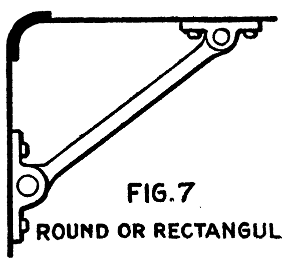


FIG.7
ROUND OR RECTANGULAR BRACE

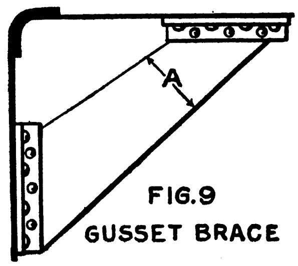


FIG.9
GUSSET BRACE

FIGS. 6, 7, 8 AND 9.

84 x.7854 =5,541.78 square inches. 68 x.7854 =3,631.69 square inches. 5,541.78—3,631.69 =1,910.09 square inches, and 1,910.09 ÷2 =955. square inches. Then 1,082.7+955 =2,037.7 square inches =total area of back head supported by seam 6. As we now find that this area is less than that of the forward part of the boiler as first found, we will take the first area, 2,247.28 square inches, to get the greatest stress on rivets in seam 6.

2,247.28x175 =393,274 pounds =maximum stress on rivets in seam 6. By consulting Table 1, we find this seam is double riveted with 1-inch rivets (staggered) and spaced $2\frac{3}{4}$-inch pitch. Then the maximum stress divided by the number of rivets in shear, divided by the area of one rivet, gives total stress on each rivet per square inch of area. We assume that the back head is circular in order to get area supported by each rivet, as it is plain that the greatest stress is on the crown and not the side rivets. Therefore, if this head was circular and 84 inches in diameter, we would have in the circumference 192 rivets. Then 393,274 ÷192 =2,048 pounds =maximum stress on each rivet. One-inch rivet =1 $\frac{1}{16}$-inch hole =.886, area of 1-inch rivet; and 2,048 ÷.886 =2,311 pounds =maximum stress per square inch of area on rivets in seam 6.

Seam No. 8.—By consulting the various figures illustrating the boiler in question, it can readily be seen that the only stress on the rivets in seam 8 is that imposed by the pressure acting on that portion of the back head not supported by the stay bolts and braces. As this is equal to twice the radius of the flange, then twice the radius of the flange multiplied by the pitch of the rivets, and this multiplied by the pressure equals the stress on each rivet. This divided by the area of the rivets equals the stress per square inch. By referring back we find the head is flanged to a 4-inch radius, and by referring to Table 1 we find seam 8 to be a single riveted joint with 1-inch rivets, spaced 2¼-inch pitch. Then $4 \times 2 \times 2.25 \times 175 = 3{,}150$ pounds = stress on each rivet, and $3{,}150 \div .886 = 3{,}555$ pounds stress on each rivet per square inch of area. This concludes the calculations necessary to obtain the shearing stress on the brace rivets and the rivets in all the major seams or

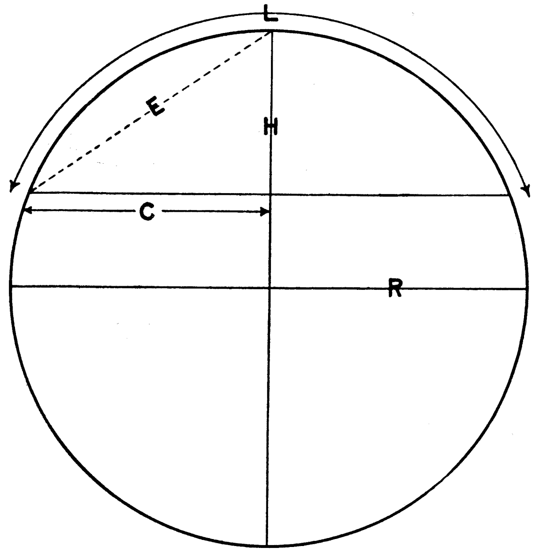


FIG. 10.

joints with the exception of the dome rivets. The stresses on these can be obtained by the usual formulas. Thus, for the longitudinal seam:

$$\text{Stress} = \frac{\text{Diameter of dome x pressure x pitch of rivets.}}{\text{Area of rivets.}}$$

$$\text{And stress on circumferential seam} = \frac{\text{Area of dome x pressure x pitch of rivets.}}{\text{Area of rivets}}$$

Where seams are double riveted zigzag, one-half the pitch to be taken.

The question has been raised as to whether the shearing stress on rivets in Form 4 refers only to brace rivets, or to rivets in the braces and also those in the shell seams, but taken in connection with foot note on first section of Form 4 the concensus of opinion is that all the above are to be considered.

Taken thus we obtain Table 11, reference to which is made in filling out item 7 in second section of Form 4.

TABLE II

Shearing stress on rivets, pounds per square inch of area:

Seam 1 = 9,754
Seam 2 = 9,623
Seam 3 = 10,104
Seam 4 = 1,094
Seam 5 = 1,569
Seam 6 = 2,311
Seam 7 = 1,459
Seam 8 = 3,555.
Front braces = 4,926.

TENSION ON NET SECTION OF PLATE IN LONGITUDINAL SEAM OF LOWEST EFFICIENCY, POUNDS PER SQUARE INCH

In order to determine the seam of least efficiency it will be necessary to calculate the efficiency of each longitudinal joint. The efficiency of a joint means its calculated strength as compared to the strength of a solid plate.

The following rules are used to determine the efficiency of a quadruple riveted joint, the first rule determining the efficiency per cent of the net section of plate, the second rule the efficiency of the riveting based on the extreme outer row of rivets, the third rule includes the next row of rivets, and the fourth rule all the rivets. It will be seen by following out all the calculations under the different rules that in all cases of riveting (it being understood, of course, that the correct sized rivets and proper pitch is used) the rivets are stronger than the net section of plate, and therefore it is seldom necessary to use any but the first rule, although it is always well to apply the second rule also, merely as a check, as in calculating the efficiency of any joint the lowest per cent is always to be taken.

In the rules below the strength of the rivet is to be taken as its strength in single shear, unless otherwise specified, and also in estimating the strength of a driven rivet the size of the hole is to be taken instead of the original size of the rivet.

Rules for estimating strength of quadruple riveted butt joints with welt straps:

Rule 1.

$$E = \frac{P—d}{P}$$

Rule 2.

$$E = \frac{(P—2d\ xTxts+sr}{PxTxts}$$

Rule 3.

$$E = \frac{(P—4d)xTxts3+sr}{PxTxts}$$

Rule 4.

$$E = \frac{arx8xds+3sr}{PxTxts}$$

In which

P = Greatest p tch of r vets (extreme outs de row
d = Diameter of rivet hole
T = Thickness of plate.
ts = Tensile strength of plate
sr = Shearing strength of rivet (single shear).
ar = Area of rivet hole.
ds = Double shear strength of rivet.
E = Efficiency of joint (per cent.)
Calculating efficiency of joints.

By referring to Table 1 and Fig. 4, we find that joint 1 is of the quadruple butt type, 11-16-inch plate, $1\frac{1}{8}$-inch rivets, whose greatest pitch is 6 inches. (2B,B = 3 inches.)

Then substituting values for letters in first equation (or Rule 1), we have:

$$E = \frac{6—1\frac{3}{16}}{6} \quad (1\frac{1}{8} \text{ rivet} = 1\frac{3}{16}$$

Reducing common fractions to decimals $1\frac{3}{16}$ = 1.1875; then, 6—1.1875 = 4.8125, and 4.8125 ÷ 6 = 80.2 per cent = E (efficiency of plate).

By Rule 2.

Thickness of $\frac{11}{16}$ plate =.6875-inch.

Tensile strength of plate =55,000 pounds.

Shearing strength of rivet in single shear per square inch of area =area x shearing strength.

Area =1.1875x.7854 =1.107 square inches.

Shearing strength of steel rivets, single shear (Government rule) =44,000 pounds; then

Shearing strength of rivet =44,000x1.107 =48,708 pounds. Substituting values, we have for second equation (or Rule 2),

$$E = \frac{(6-2.375)\text{x}.6875\text{x}55000+48708}{6\text{x}.6875\text{x}55000} = 92\%$$

And by Rule 3 we have

$$E = \frac{(6-4.75)\text{x}.6875\text{x}55000+146124}{6\text{x}.6875\text{x}55000} = 85.2\%$$

As stated before, in order to find the seam of least efficiency it is necessary to calculate the efficiency of all seams. We found the least efficiency per cent of seam 1 to be 80.2 per cent. We will now consider seam 2, using Rule 1, only as we have found in seam 1 that the net section of plate was the point of least efficiency in the seam, and as all other longitudinal seams are of the same type (quadruple butt joint), we can safely assume that the same proportions will obtain in seams 2 and 3 as in seam 1. Consulting Table 1 we find, plate 3-4-inch greatest pitch of rivets 6½ inches =6.5 inches. Rivet diameter 1¼ =1 5-16 hole =decimal 1.3125. Substituting values for letters in equation we have

$$E = \frac{6.5-1.3125}{6.5} = 79.8\%$$

As we find by consulting Table 1 that the rivet diameter and pitch are the same in both seams 2 and 3 we know that the joint efficiency must be the same in both; therefore, either seam 2 or 3 can be taken as the joint of least efficiency, but as both are on the taper course of the boiler, and as the stress on seams is as the diameter of the course, we take seam 3 as the seam of least efficiency on which to calculate the tension on net section of plate per square inch. The rule or formula for this is as follows:

$$T = \frac{P\text{x}r\text{x}p}{A} \text{ in which}$$

T =Tension on net section of plate.

P =Pressure in pounds per square inch.

r =Radius of greatest diameter of course.

p =Pitch of outside row of rivets.

A =Area of net section of plate.

The radius of the greatest diameter of course =84 ÷2 =42 inches.

Steam pressure =175 pounds.

Area of net section of plate =pitch of rivets—diameter of one rivet hole x thickness of plate; then area 6.5—1.3125x.75 =3.89 square inches.

Substituting values for letters in equation, we have

$$T = \frac{175\text{x}42\text{x}6.5}{3.89} = 12{,}281 \text{ lbs. per square inch.}$$

Before closing this article it might be well to call attention to a recent order issued by the chief inspector in regard to that part of Section 8 of the Boiler Inspection Law reading "And where the locomotive is disabled to the extent that it can not be run by its own steam, *the part or parts affected by the said accident shall be preserved by said carrier intact*, so far as possible," etc., advising that this portion of the rules must be absolutely complied with, and that the meaning of this clause is that these parts, steam gauge, safety valves, gauge cocks, water glass cocks, etc., must in all cases be left in place on the boiler, pending the inspection by the designated government inspector, instead of being removed and stored, as has been practiced on some railroads recently.

(N. B.—The italics are ours.)

QUESTIONS AND ANSWERS

Answers by F. P. Roesch, Locomotive Firemen and Enginemen's Magazine

LOCOMOTIVE RUNNING AND REPAIRS

Finding the Per Cent of a Grade

"Knowing the rise of a grade in feet per mile, how can I find the percentage of the grade?"—*Subscriber.*

Answer.—By dividing the rise in feet per mile by the number of feet (5,280) in the mile. As all are not familiar with handling percentage, however, two examples are given below, the first being "if the rise is 78 feet per mile, what is the per cent. of grade." In dealing with percentage we first add to the dividend as many ciphers as there are figures in the divisor, and commencing at the right, point off two ciphers; then the number of times that the divisor is contained in the dividend, ahead of the decimal point, would be represented as units, while back of the decimal point would represent tenths, hundredths, etc.

Taking the first example and adding four ciphers to 78, we have the number 7800.00; bringing down the decimal point ahead of the last two ciphers; we now find that 5,280 is contained in 7,800 one time, and we set the figure down in the quotient. After subtracting, we bring down the first cipher, back of the decimal point. We now find that we have a remainder of 25,200, and 5,280 is contained in this four times. As it was necessary to bring down one of the ciphers back of the decimal point, the figure 4 in the quotient falls into tenths place. The next remainder, after bringing down another cipher, is 40,800, and 5,280 is contained in this seven times, the 7 falling into hundredths place, giving us an answer 1 and 47 hundredths; or, in other words, a rise of 78 feet per mile is equal to a 1 and 47 hundredths per cent. grade. See solution:

5280)7800.00(1.47% 1 and 47 hundredths %.
5280
——
25200
21120
——
40800
36960

In the next example we wish to find what per cent. a rise of 36 feet per mile is equal to. Adding four ciphers to 36, as before, and pointing off the two right hand ciphers, we have the number 3600.00. We now find that 5,280 can not be divided into 3600; consequently, we bring down a cipher in units place and add the next cipher to our dividend; we now find that 5,280 is contained in this six times, and in the remainder, after adding the remaining cipher, eight times, showing that a rise of 36 feet per mile is equal to a .68 hundredths of 1 per cent. grade. Below is the solution:

5280) 3600.00(0.68% 68 hundredths %.

Handling Lubricators

"In your former answers to questions, you advocate wide open valves on lubricators. This is an old practice, due to ignorance of the true way of working these valves. I did it in this way until, by actual experience, I found that I got better results, used a little less oil, and have fewer gaskets blown out and less glasses broken, by using about three-fourths of a turn on globe and condensing valves and about one-fourth turn on other valves."—*J. C.*

Answer.—While we are always ready to acknowledge an error, admitting that we are as liable to make mistakes as any one else, yet in our answers to questions we try to be absolutely accurate, realizing that those asking questions are after reliable information and not guess work. While our correspondent's experience may have led him to believe that his manner of working a lubricator is correct, yet it is contrary to all theory and opposite to the experience of the majority of practical engineers. It is possible that the experience of our correspondent has been altogether with the old type of lubricator, where the choke plugs are located at the lubricator, and the oil pipes, together with the equalizing tubes, or equalizing passages, are exceptionally small. In this case, perhaps the steam pipe leading to the condensing chamber was considerably larger than the combined area of the two equalizing tubes, and a three fourths turn on the steam valve and condensing valve would be ample; in fact, a three-fourths turn on the condensing valve is usually sufficient in any case, and if our correspondent will read the answers carefully he will see that we only advocated a wide open steam valve.

The object of a lubricator is to deliver oil to the steam chest, and this can only be accomplished by maintaining a circulation of steam between the lubricator and the chest; consequently, if the pressure at the lubricator were reduced by wire-drawing the steam through partly open valves below the steam chest pressure, there would be no circulation through the oil pipes, with the result that a water seal would form at some exposed portion of the oil pipe, thereby preventing oil from getting to the steam chest until equilibrium had been restored.

OIL FUEL FOR LOCOMOTIVES

The cost of equipping a locomotive for burning oil, including the burner and regulating valves, the firebrick furnace and the tank with its pipes and valves on tenders is about $800, and the cost of steel storage tanks for roadway supply is approximately 25 cents per barrel. The oil stations cost only about 50 per cent of the cost of coaling stations, but the latter are in place and already provided, while the oil stations are clearly an extra expense.

One of the principal advantages of oil fuel for locomotives is the increased steaming capacity obtained from the boiler and the possibility of maintaining this at a maximum throughout the run. It is this which has made possible the successful operation of very large Mallet locomotives which develop a maximum tractive effort of 100,000 lbs. and nearly 2,000 h. p. at ordinary freight speeds. The fireman does not become fatigued, and even in hot weather the maximum output from the boiler can be depended on whenever needed for any part of the run. While the physical exertion of the fireman with oil fuel is not great, it requires more careful and close attention than in coal burning to secure economical combustion. The fireman and engineer must work in harmony, and with each change in the throttle or reverse lever a corresponding regulation must be given to the oil supply and atomizer. If the fireman is not careful and skillful in this way it is possible for him to waste large quantities of oil and a greater value in fuel than is wasted by a careless fireman on a coal-burning locomotive.

The capacity of the boiler is soon reached as speed increases in coal-burning locomotives, and the engineer reduces the cut-off to accommodate steam consumption to steam production; but with oil fuel the boiler can be so easily forced with increased oil consumption that there is a tendency on the part of enginemen to use a longer cut-off than is economical, and complete the trip in a shorter time than careful economy in oil consumption would require. This important feature in the operation of locomotives when fired with oil is referred to in a comprehensive paper on "Petroleum as Locomotive Fuel," by Eugene McAuliffe, general fuel agent of the Frisco Lines, presented at the recent meeting of the International Railway Fuel Association. The author gives carefully prepared rules for the guidance of engineers, firemen and others in handling oil-burning engines, including instructions for starting the fire, using the heater, preventing black smoke, and handling the atomizers, blowers and dampers. He exposes the careless habits of enginemen, and says it is easy to make schedule time or make up lost time with oil fuel, but it requires an excessive consumption of oil and results in a lack of discipline, and mechanical facilities are allowed to deteriorate on account of the comparative freedom from failure actually to deliver the train on time. It is possible to measure oil very accurately, and the consumption under proper regulation of speed and combustion could be easily established for each class of trains, so that it ought not be a difficult thing to determine which enginemen are careless in the use of oil fuel.

There is an economy in oil fuel as compared with coal which is seldom referred to, and that is the comparative loss when engines stand idle under steam at terminals or are delayed on side tracks. It is then possible so to regulate the oil jet that there is a small consumption of fuel, while with coal the large mass of incandescent fuel must be maintained over the grate to insure prompt movement when orders for departure are given. The saving in oil in this respect averages 50 per cent of the coal equivalent, and in freight traffic in winter it is an important item. It is fortunate that large deposits of petroleum are found in districts where there is a poor coal supply, and where coal for locomotives is expensive. This is particularly true of California and Mexico.

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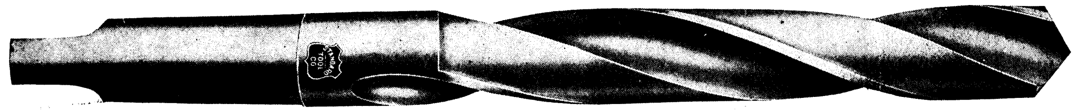

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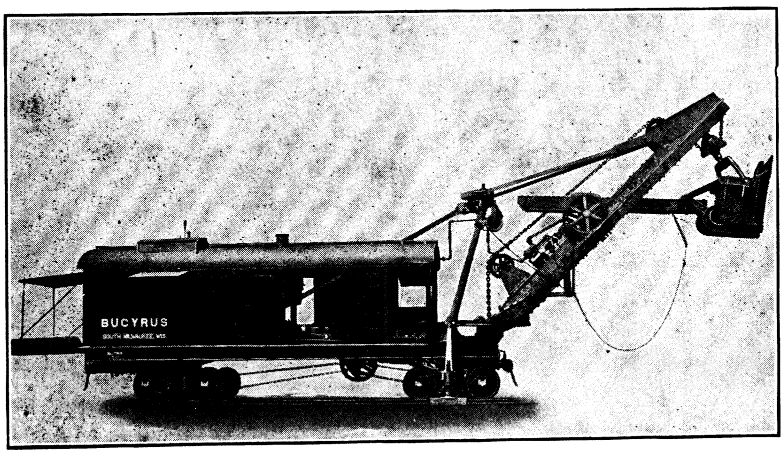

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Statistics gathered from all the world point with more or less certainty to the conclusion that although in the last fifteen years the rise in the cost of living abroad has been about 13 per cent., or a little more, the rise in the United States has been about 40 per cent. Reduced to plain figures this means that, while the European citizen's dollar has shrunk to about 83 cents the American dollar, measured by purchasing power of necessities, has dwindled to about 71 cents during this period.—*World's Work.*

CRAZY ON SISTOM

At a certain coal mine down in New Mexico the superintendent was greatly annoyed, from time to time, by employees moving into and out of the company's houses without due notification of their frequent changes of domicile. It became quite impossible to keep the rent accounts straight on the office books, and finally the superintendent, in his exasperation, resolved upon stringent measures. He therefore posted the following notice, which is given verbatim—orthography, syntax and all.

february the 11th.

Notice to all employes

aney Person or Persons that Mooves into A house Without My Consent shall be Put out Without anney Cemmony.

Dam it I Must and Will have some Sistom.

Hen Filster.—

Railway & Locomotive Engineering.

RESPONSIBLE FOR FULL VALUE OF BAGGAGE

Many transportation companies have a self comforting way of making laws of how far they will be responsible for the value of packages assigned to their care; but when a test case comes before the courts their rules go for naught. An action was recently brought against the New York Transportation Company for $1,200 for loss of a trunk and the company pleaded that it was liable for only $100. The case came before the Court of Appeals, which gave a decision that the transportation company was responsible for the full value of the trunk. A similar decision has been repeatedly made against hotel keepers who claim that they are not responsible for loss of baggage.

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"I try to do my duty," said the exceeding sincere person, "and I do not hesitate to remind others of their duty."

"Go ahead," replied the easy-going citizen. "You may prove to be a very useful member of society. But when you get through you'll have about as many sincere friends and admirers as an alarm clock."—*Washington Star.*

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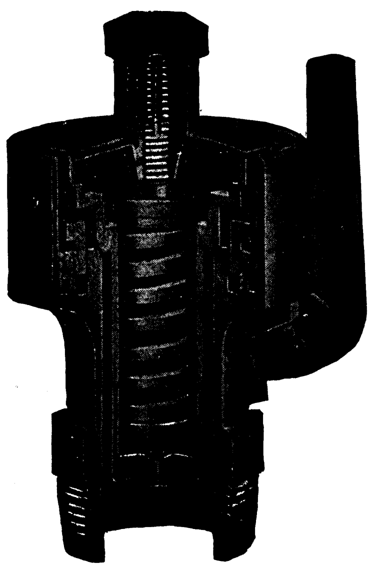

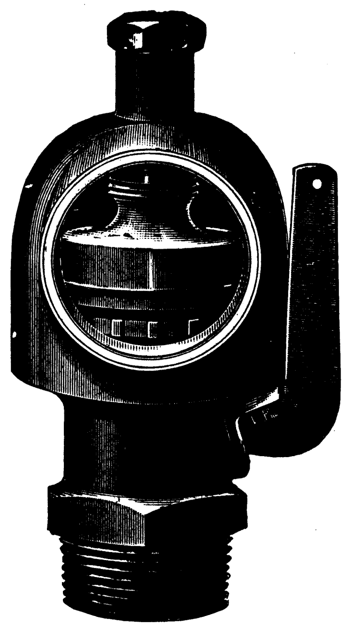

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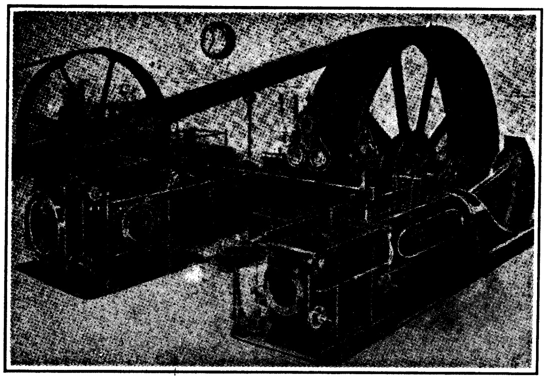

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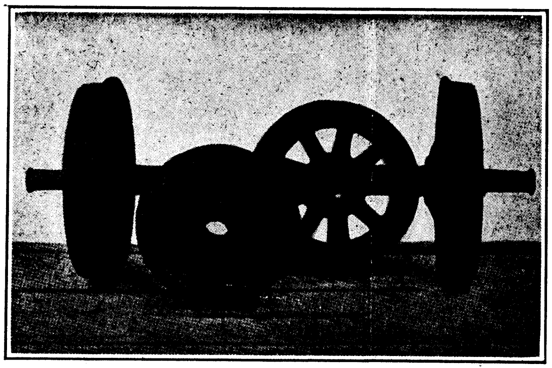

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