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“Permanent Way on Mountain Railways.”

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Introductory.—To those who consider the commercial aspect of the engineering profession as of paramount importance, the questions involved in the first part of this Paper may seem to be of little interest, mountain railways not being among the big enterprises of the world. There are, however, many who regard the principles and scientific problems involved as the real interest of the profession, the foundation from which commercial and financial interests spring.

In submitting this Paper, therefore, the Author feels that no apology is needed for the introduction of a subject which chiefly concerns mechanical problems, and which throws many sidelights on railway work in general, necessitating the reconsideration of principles that are only too apt to be taken for granted.

The second part of the Paper, dealing with a proposed new form of rail-head, arises naturally from the first, for it is the extreme severity of flange action on hill railways that compels attention to the necessity of an improvement in the standard sections for mountain railways. It would be idle, of course, to suggest its application to railways in general.

Although the capital involved in mountain railways in various parts of the world is considerable, it is not comparable with the vast sums sunk in railways in general, nor is the ton-mileage of traffic to be compared. But the interest in these railways arises rather out of the fact that they involve a reconsideration of all long-established methods, and of almost every item of practical working, building up again on fundamental principles. There is, perhaps, almost more need for elasticity of method in the care of mountain railways than in any other railway work that may be named.

It is also true that many of the lessons learnt on these railways are applicable in a modified form, if not to trunk lines, at any rate to branch lines in hilly and rough country. The difference between hill railways and ordinary railways is only one of degree, and indeed it is not always possible to say whether a given line is a hill railway or not. For the purposes of this Paper, however, we are considering a railway of a narrower gauge than the parent line, isolated in the sense that it cannot exchange rolling stock.

Problems Affecting Mountain Railways.—There are, perhaps, two outstanding problems on mountain railways that affect, more definitely than any others, the working of the line, namely:—

- (a) How to get sufficient hauling power on a train, in the case where a narrow gauge and short wheel bases are essential.
- (b) How to design the permanent way so as to make safe the passage of trains round exceptionally severe curves, without involving a ruinous expenditure of capital and revenue.

Where steam traction is employed, the first of these problems has led to many designs of flexible types of locomotives, but, at the best, the size of the train has always been limited to far less than that required, for efficient working, by the operating (or traffic) department. Steam traction also involves a limitation of the steepness of the grades, probably necessitating a lengthening of the line and an increase in the capital cost of construction.

For this reason electric traction is usually much to be preferred, for the hauling power obtainable on a single train can be greatly increased by the installation of motors on the axles of the vehicles until, if necessary, every wheel becomes a driving wheel, and grades as steep as the angle of repose could, in theory, be worked. Nor would there be any limitation to the size of the trains on account of the hauling power of the locomotive.

Flexibility for traversing the sharp curves is, of course, assured in electric traction, and it is also very noticeable that flexibility of power is so much greater. Additional electric power can easily be applied to overcome a sticking point (such as a sharp curve) over which a steam locomotive would visibly labour.

The object of this Paper is not, however, to advocate any special form of traction, but rather to indicate the difficulty of providing a sufficiently powerful, large unit train to make the operating of the line economical and effective.

Coming now to the question of the permanent way, which more immediately concerns the purpose of this Paper, no one can have, even casually, inspected a mountain line without being struck by the extraordinary amount of flange wear on the wheels and sides of the rails on curves. Yet it may be doubted whether the full meaning and cost of this is realized.

On a typical line of this sort, it is not too much to say that the extra cost due to the wear on the sides of the rails, beyond that which may be considered the ordinary wear of the permanent way on railways in general, is about 10 per cent. of the gross earnings of the line. The cost to the locomotive and carriage departments of the extra wear on the flanges of the wheels is probably no less, making a total of some 20 per cent. of the gross earnings of the line. These, of course, are general figures and will doubtless vary a good deal in different cases, but they were substantially true for the Kalka-Simla Railway, when the Author was connected with that line.

Further, the cost which this wear entails is not the only cause of anxiety, as, owing to the sharpness of the curves, the lightness of the loads holding the wheels down on the rails, and still more to the peculiarities in the phenomenon of the wear itself, there is an undoubted tendency to derailments, and vehicles are much more prone to leave the rails than on ordinary railways. The Author, on taking over charge of the Kalka-Simla Railway, well remembers his first impression in regard to this, but the anxiety was soon alleviated, it is true, upon realizing how quickly the vehicles came to rest when once the 18-inch and 20-inch wheels of the 2-foot 6-inch gauge rolling stock got on to the ballast.

There were many cases of derailments, two of runaway trains, and one of a curious smash of an empty stock train on the uphill journey, owing to excess speed on a short level stretch caused by the rash driving of a new and powerful locomotive.

None of the derailments led to anything more serious than damage to the rolling stock and the permanent way, but their frequency was one of the anxieties of the early days of that interesting and difficult railway, which runs through what is perhaps the most mountainous country in the world.

In regard to the danger of derailments arising out of flange wear, it cannot be too clearly emphasized that this danger is chiefly due, not to the tendency of light vehicles to jump off the line when passing rapidly round a sharp curve, but rather to the nature of the wear itself, and nothing could be more fatal than the general attitude of mind which regards rail and flange wear on sinuous

railways as a price which must inevitably be paid for the sinuosity, and which therefore neglects any endeavour to reduce it. It cannot, indeed, be too clearly asserted that the wear which is taking place is, in fact, far in excess of what it need be, and is productive of the most dangerous results.

In the first place the trouble arises out of the extraordinary obliquity of the positions which vehicles can, and do, assume in passing round curves when flanges and rails are badly worn. A worn flange crowding hard on to the outer rail of a curve with the usual $\frac{3}{4}$ -inch slack to gauge is shown in *Fig. 1* (the dotted line representing a new flange on a new rail when the wheels are resting centrally on the track which is true to gauge), and it will be seen that the inner flange lies about $2\frac{3}{4}$ inches away from the rail. This, indeed, is not an exaggerated state of affairs, and was common experi-

Fig. 1.

ence on the Kalka-Simla Railway. Now if the position of a vehicle with 6-foot 3-inch rigid wheel base be shown on the sharpest curve—namely, 120 feet radius—it will be found that the leading outer flange is cutting the rail at an angle of obliquity of 4° , when the flange of the inner wheel of the trailing axle is in the usual running position—that is, just touching the rail (*Fig. 2*). Careful estimates show that this angle of obliquity is increased to slightly over 5° by the play of the brasses on the journals, which results in the two axles not lying squarely in front of, and behind, one another, but slightly in echelon, the amount of the displacement being the sum of the play of the brasses on the journals of the leading and trailing axles.

This obliquity is probably the cause of most of the excessive wear, as well as of the unpleasant motion of the vehicle, for an oscillation of 10° round a vertical axis through the centre of gravity can and occasionally does take place.

It may be interesting to point out here a singular difference between British and Continental practice in endeavouring to over-

come this difficulty. The former endeavours to combat the trouble of obliquity by shortening the rigid wheel bases, and the latter by lengthening them. In the first case it is argued that the shorter the wheel base the more nearly both axles can take up a radial position, for, as the trailing axle automatically runs in a position approximately radial to the curve, the leading axle will be the less oblique the shorter the base. On the other hand, the Continental engineers argue that as obliquity is inevitable, it is better to face it and avoid the worse evil of oscillation by lengthening the base, as such lengthening does not always increase the obliquity in proportion.¹

Fig. 2.

Showing four wheel vehicle with rigid wheel base of 6' 3" passing round a 120 ft. curve, giving an obliquity of 4° when flanges are badly worn, which in practice is increased to 5° owing to the play in the journal brasses.

Note: Section of flanges at level of rail head is shown hatched.

Scale, 3 Feet = 1 Inch.
 Ins 12 6 0 1 2 3 4 5 Feet.

This difference in practice is remarkable, and Continental rigid wheel-bases are, *ceteris paribus*, 50 to 100 per cent. greater than ours.

The first Chief Engineer of the Kalka-Simla Railway shortened the wheel base of some of the four-wheeled stock from 6 feet 3 inches to 5 feet 3 inches, whereas the Chief Engineer of the Montreux-Oberland-Berne Railway (a railway of remarkable similarity to the former, with metre instead of 2-foot 6-inch gauge, and with 130-foot radius curves instead of 120 feet) lengthened the rigid wheel bases of their four-wheeled vehicles from 11 feet 9 inches to no less than 17 feet. The result was that on the former line the

¹ This is true in some cases. As an example, an increase in the length of the Kalka-Simla Railway, 6-foot 3-inch rigid wheel-bases to anything short of 8 feet 4 inches would rather decrease than increase the 4° angle of obliquity.

vehicles were reported to run so badly that they were altered back to 6 feet 3 inches, whereas on the latter the running of the vehicles was reported to be much improved. Astounding as these results may seem to be to British engineers, there is undoubtedly a good deal to be said for the Continental practice. The supersession of four-wheeled vehicles by bogies on all hill railways, however, renders the point of less practical interest.

Fig. 3 shows the section of a rail, on the worn surface of which is inscribed a 5° ellipse, which, it will be noticed, fits the worn surface with remarkable accuracy. This represents the cutting effect of the edge of the flange on the side of the rail, for the circle, which constitutes that cutting edge, projected through an angle of obliquity of 5° , presents the ellipse in question, and forms a most perfect tool for cutting away the side of the rail head to fit that ellipse. This is not a question of fair wear and tear, it is deliberate cutting, which may be largely eliminated by remedial measures, the first of which is the adoption of bogie under-carriages instead of the four-wheeled rolling stock. The bogie truck lies up against the outer rail, with both outer wheel flanges touching it, and, though the leading wheel flange presses a good deal harder than the trailer, the angle of obliquity either disappears altogether or is considerably reduced. In the locomotives adopted on the Kalka-Simla Railway, an excellent design of Bissel Truck for the leading and trailing ends produced a somewhat similar result.

In the next place the locomotive and carriage department might endeavour to prevent the cutting edge itself. It is, of course, not to be expected that they should put all the wheels on the lathes and turn off this edge at frequent intervals, but, on the other hand, the brake blocks might be made to engage with this edge as well as with the tread of the wheel without touching the throat of the flange. All wheels on such railways are braked, and a very reasonable alteration in the shape of the blocks would provide an excellent means of rounding off the edge which does so much harm. Finally it is desirable to adopt a shape for the flanges of the wheels and for the rail-heads that will prevent this cutting for any reasonable angle of obliquity.

The points of contact between flange and rail-head when the

obliquity is 5° and 2° respectively may be seen in the cases of a worn flange passing over a worn rail, and of a new flange passing over a new rail, in *Fig. 4a* and *b*. In this Figure it should be noticed that, as a matter of convenience, points of contact for these two angles are shown crosshatched on the opposite sides of the centre line. The position of the contacts were ascertained from sectional plans on lines AA, BB, CC, and DD, *Fig. 4a*.

Fig. 4c shows the result of adopting British Standard rail-heads and flange-sections on such a railway, a most undesirable section, it is true, on account of the vertical sides of the rail-head. With such a section of rail a groove would be worn within a month or two in the fillet of the flange, with the very bad results indicated. It should be especially observed that the position of contact is as bad for the 2° obliquity as for 5° , and this, of course, is the direct result of a vertical sided rail-head, the advantages of which it is hard to conjecture and the disadvantages of which are obvious. Indeed, the vertical sided head and the extreme dissimilarity between the radii of the corner of the rail and the fillet of the flange, adopted by our standardization committee and so warmly supported by the American writer, A. M. Wellington, are only consistent with the conception that no wear whatever occurs. This assumption is the more astonishing, inasmuch as the obvious endeavour of the design is to restrict the area of flange contact as nearly as possible to a point, thus involving an intensity of pressure as enormous as the area of contact is small.

A design of rail-head and flange is now submitted, in which the side of the rail-head and the flange are made to the contour of a tractory curve (*Figs. 4d* and *e*). From these it will be noticed that the position of contact is not so ready to leave the centre line, and also that the area of contact is not so unduly constricted when wear has taken place.

The British standard rail, shown in *Fig. 4c*, with its vertical sided head, square corners and the large radius fillet of the wheel flange, was designed, it is believed, in accordance with the principles propounded by the American writer, A. M. Wellington. Briefly the explanation is as follows :—

So long as the radii of the flange fillet and corner of the rail are made quite dissimilar and contact is thereby limited approximately to a point, the outer wheel lifts off the tread when passing round a curve, and contact only remains between the corner of the rail and the fillet of the flange, thereby virtually increasing the radius of the outer wheels, and so negotiating the curve without slipping.

Figs. 4.

In reply to this argument it may be said—

(a) If the materials of which rails and tires are composed could safely withstand pressures of 50 or 100 tons per square inch without detriment, there would be a good deal in favour of the arrangement. The best commercial steel, however, from which the rails and tires are made, cannot stand this pressure, and the fibres will inevitably be crushed. The result is that so far from Wellington's statement being correct, that "it will require four times the tonnage to account for a given unit of wear before that corner is worn off than it will afterwards," it is neither in accordance with reason nor with known facts.

(b) Inasmuch as it has been shown by Wolley Dod and other writers (including Wellington himself) that the horizontal pressure of the flange against the rail is independent of the radius of curvature of the track, being about 0.6 of the weight on the wheel for normal values of the coefficient of dry friction, the fillet of the flange will run up against the rail to the same extent for all curves. In other words the point of contact is the same for all radii of curvature, consequently the increase in the virtual diameter of the outer wheel, although it will suit one particular radius of curvature, will not suit any others. Hence sliding must, and does, take place, under a pressure of 50 tons or more per square inch, which the Author calculates is by no means an over-estimate of the pressure at the point of contact with a new flange and rail. It can hardly be wondered that initial wear in such cases is tremendous.

(c) Great dissimilarity of radii is not necessary to cause the wheel to lift off the tread, and this advantage, such as it is, can be obtained without paying such a high price as one-point contact involves. It must also be remembered that this one-point contact affects rails and tires adversely on the miles of straight line as well as on curves, whereas the hypothetical advantage is only applicable to the outer rails on curves.

New Section for Rail-head.—From all this it would surely seem that the principle on which the British Standard section rails are designed is wrong; that a revised and rational section of rail and wheel flange should be produced; and that this design, though it is of course impossible to hope to introduce it on trunk lines, should be adopted on mountain lines and other isolated railways that do not exchange rolling-stock.

The object to be aimed at in designing such a new rail head is to obtain a section that will give the least wear for a given tonnage carried by wheels with flanges correctly shaped to suit the rail. Flange action is without doubt the crux of the situation, and

the top of the rail must be merely more or less flat to take tread-wear.

It is submitted that the section which will comply with this stipulation, and give the least wear is that section which will tend to retain its shape during wear—in other words the shape which the wear itself is tending to produce. If such a section be adopted the action under a new tire is the same as that under an old and worn one, whether the rail be new or old, and there will be a good fit between rail and flange at the corner of the rail. This fit is essential, for it is impossible that wear can take place at points where there is no contact, just as it is impossible to believe that we have arrived at the natural curve which wear is tending to produce if the various flanges, new and old, are bearing at different points and producing irregular wear. Moreover, it is only when a good fit has been obtained that pressure is evenly distributed, instead of, as at present, being excessively high at various points—a state of affairs that must produce rapid wear. It may, of course, be readily admitted that in the case of a very much worn flange the contact will extend far lower than is contemplated in a new section, and will in consequence produce imperfect action, but this will be dealt with later, as a second consideration in the design.

Application of Tractrix Curve.—It is clear that any reasonable sort of curve for the flange and the rail corner which makes a fit, will not fulfil these conditions, as the wear is taking place under regular mathematical laws and the curve will therefore have to be made some definite mathematical shape.

The wear at each point of any two surfaces in contact will be, provided the material is homogeneous, proportional to the intensity of pressure at that point and to the speed of rubbing of the surfaces on one another.

If, therefore, the surfaces of flange and rail fit correctly and give an even distribution of pressure, the wear at each point will be proportional to the rubbing. By wear is meant the amount of metal worn off each element of surface, and that in turn means depth of wear measured normal to the surface.

If the curve is designed, therefore, so that the horizontal projection of the varying depth of wear is, by virtue of the varying angle of inclination of the surface, a constant quantity, the wear will cause only a horizontal displacement of the curve without distortion, and the object sought will have been obtained, namely, a section which will retain its shape during wear. Thus in *Fig. 5*, if *CDB* be the curve required and *C₁D₁B₁* shows the position of the same after wear, then the two curves are the same if *CC₁ = DD₁*

and is constant at all points on the curve. This will be so, if $Dn \operatorname{cosec} \theta$ is constant, that is, $Dn \operatorname{cosec} \theta = DD_1 = CC_1$; where θ is the varying angle of inclination of the curve, and Dn is the varying depth of wear.

Now the relative motion between the flange and the rail is that arising out of the fact that, at each instant the wheel and flange

are rotating round an instantaneous centre or axis, which is a line joining the points of tread contact of the two wheels of one axle on the tops of the rails. The rubbing at any point is therefore pro-

portional to the distance of that point from the instantaneous axis. If in *Fig. 5* AB is the instantaneous axis the rubbing at D is proportional to De , that is, $Dn = K \times De$, where K is a constant.

The two right angle triangles Def and DnD_1 being similar, it follows that $DD_1 = K, Df$, or $\frac{DD_1}{Df} = K$, and if DD_1 is to be constant, so also must Df .

The curve must, therefore, be a tractrix, and the line AB its asymptote. This will be more clearly understood when it is remembered that a tractrix is, by definition, the curve traced by a point D when taken in tow by an object passing along a line AB , and the constant length of the tangent Df is the length of the tow line.

DIAGRAM OF A PEDESTAL WITH THE FOOT CUT TO A TRACTRIX.

Consequently, if the rails are rolled so that the corner is a tractrix, of which the asymptote is the horizontal line passing through the highest point of the rail-head, and if the flanges of the wheels are cut to fit this curve, we shall have the shape that the wear itself is tending to produce, and the rail and flange will retain their form throughout the process of wear.

The case is, indeed, precisely analogous to that of a pedestal which, if the bottom of the shaft is turned square, will quickly wear to a rounded surface which bears unevenly. If, on the other hand, it is cut to the shape of a tractrix the wear will be even (*Fig. 6*).

It should be noted here that a slight error occurs in the action on the outside rails of curves, for the motion round such curves is a compound motion of rotation round the instantaneous axis as for the straight portions of the track, together with a sliding forward of the wheel on the outer rail. It is this latter motion which constitutes the irregularity. It need not be anticipated, however, that the momentary sliding forward of a tire on the rail, which it fits exactly, will cause a wear that will distort the surface; it will cause an even wear all over the surface of contact, and, indeed, it may be confidently expected that rails and wheel tires, shaped as above, will retain the shape with fair accuracy during wear.

Selection of the Tractrix Curve.—It now only remains to select the particular tractrix that will best meet the requirements, and the first question is how far down the flange, below the tread, must contact extend in order to provide for flange action.. It is, of course, a question of the intensity of pressure permissible. In view of the fact that the further the extension down the flange the greater the rubbing, it would be very desirable to limit this area, but a considerable area of flange contact is necessary as the flange pressure, which has to be transmitted, is considerable; and it is not feasible to limit the maximum intensities of pressure to a lower figure than that obtained for tread pressure.

Now the maximum intensity of tread pressure on a metre-gauge railway, loaded up to the government of India rule, which allows 3 tons load per foot of wheel diameter, was calculated by the Author in a Government memorandum to be about 8 tons per square inch. That was, taking the modulus of elasticity for rail and tire at 15,000 tons per square inch, and assuming the length of the surface of contact across the rail-head to be 1 inch. It will, therefore, be wise to aim at about the same maximum intensity of pressure for flange contact.

As already mentioned, the maximum flange pressure is the same for all radii of curvature of track, and assuming the coefficient of dry friction to be $\frac{1}{4}$, is about 0.6 of the weight on the wheel. Thus, in order to get the same intensity of pressure, the area of flange contact should be about 0.6 that of head contact.

It is not possible to make an exact computation of the area of flange contact, however. *Fig. 4d* shows that area for stated angles of obliquity, but the exact extent of this will be affected by the elastic deformation of the fibres in contact, and a drawing, however accurate, cannot deal with such minute measurements.

The area of tread contact on the other hand is susceptible of computation, as also is its length along the rail, and it is relevant to

point out that this length is greater in the case of flange contact than in that of tread contact, for the former is more glancing than the latter. The first is the case of a wide angle cone coming against a straight rail and the latter is merely a cylinder resting on a plane surface. In the case previously referred to, the maximum length of the contact along the rail, which varies directly with the radius of the wheel, is about 0.9 inch on a 20-inch diameter wheel, or 0.8 inch on an 18-inch diameter wheel, when the maximum pressure is 8 tons per square inch.

It follows that the length of flange contact will be greater than this. The actual average radius of curvature of the face of the flange in horizontal sections through various points in the area of contact is about 18 or 20 inches, and is thus about twice that of the wheel which produces the above-mentioned tread contact. The curvature of the rail also tends to lengthen the contact, for the pressure is against the inside or concave edge of the rail on a curved track.

The length of flange contact may, therefore, be regarded as about double that of tread contact with the same intensity of pressure, and thus the width of flange contact may be made half of that on the tread. Only 0.6 of the area is required, however, for flange contact to give the same intensity of pressure, as the total horizontal force is only 0.6 of the vertical force on the rail, as previously stated. Hence the width of flange contact should be $\frac{1}{2} \times 0.6 \times$ width of tread contact.

The latter has been taken at about 1 inch when calculating the maximum pressure mentioned above, so that the depth down the flange to which contact is to extend should be 0.3 inch to give the same maximum pressure. Three-eighths of an inch has been taken in the designs for the 41 $\frac{1}{4}$ -lb. rail in the diagrams. It is possible, however, that in the case of rail-heads for hill railways $\frac{1}{2}$ inch would have been better, on account of the heavy flange wear, whereas for standard gauge rails $\frac{5}{8}$ inch or $\frac{3}{4}$ inch will be ample.

Plotting the Tractrix Curve.—In the class-room the tractrix is drawn as the involute to a common catenary, but the latter not being an easy curve to draw the following is suggested as the simplest way to plot a tractrix, *Fig. 7*.

AB and AC, two lines at right angles, are taken as the axes of X and Y, and AC ($\frac{3}{8}$ inch as above for the 41 $\frac{1}{4}$ -lb. rail) as the constant tangent, the length of the tow line. The whole curve is a quadruple arc with cusps on the axis of Y where the tangent is vertical, but, of course, only one quarter is needed.

Take points 1, 2, 3, etc., equally spaced along AB, A to 1 being

only a half length. C1 represents the mean position of the tow rope whilst the towing body travels its first unit distance along AB. Set off on C1 from 1 the constant length and we get point *b* on the curve; join this to point 2 on AB and the mean position

Fig. 7.

of the tow line during the second unit distance travelled by the towing body is obtained, and so on.

This gives a set of tangents to the curve, but the method is liable to cumulative errors, so that a true point on the curve should be calculated from the equation. If the curve does not pass through the true point (a small error will be observed in this case), it must be moved till it does, when for practical purposes it is correct.

It will be observed that the line AB is the asymptote to the curve, and it will be necessary to stop the tractrix at about one-third the width of the rail-head from the inner edge and run on in the tangent in the line AB, for that line must form the top of the rail-head, as it was on this assumption that the investigation was carried out. The action will not be affected appreciably, for the deviation will be small and so high up as to be clear of flange action, and it is only for flange action that a tractrix is necessary; tread action will be satisfied with any flat curve.

With this curve drawn, the inside edge of the rail may be completed by drawing a line at a slope of 1 in 6 in case of the rail-head, and at about 50° in case of the flange tip, downwards to complete the contour. This slope will keep the flange tip well away from the side of the rail and the tongue rails of switches, whilst the slope

Fig. 8.

of 1 in 6 on the rail side will prevent the worn part of a badly worn flange actually exhibiting the worst features of the cutting already referred to, when it has worn to a cutting edge, and when serious obliquity of motion occurs.

Fig. 8 shows the proposed design of rail-head for the $41\frac{1}{4}$ lb. and the 75 lb. rail.

Further advantage may be expected from the adoption of this design, namely, that for a given amount of wear the quantity of material that has to be cut off the tire, on turning up in order to restore the correct gauge, is slightly less than half of what is necessary with British Standard Section rails, thus more than doubling the life of the tire, *Fig. 9*.

Fig. 9.

It should be noted that these rails should be laid flat on the sleeper or bearing-plate, and not canted, as the reason for the canting now disappears. This, of course, will much simplify the laying.

Conclusion.—Summing up, it may be claimed for this form of rail that:—

(a) A rational shape has been obtained that will be maintained throughout wear, giving an even bearing throughout its life.

(b) The intensities of pressure will, therefore, be much less than in present practice, and the wear much less rapid in consequence.

(c) That owing to the fact that the gauge may be restored on turning up, by only half the present depth of cut, the life of the tires for this reason alone will be doubled.

(d) That the cutting action on sharp curves and facing points will be, if not entirely eliminated, very much reduced and the danger of derailments will practically cease.

(e) That for the first time a rational design is introduced which is consistent with flange action on the straight portions of the track, with the consequence that many of the frequent signs of bad running will be eliminated, as well as serious, unnecessary wear at present obtaining on such sections.

The Paper is accompanied by a tracing, from which the Figures in the text have been prepared.