SECT. II.—OTHER SELECTED PAPERS.

(Paper No. 3058.)

“Failures in the Draw- and Buffing-Gear of Railway Wagons.”

By GEORGE TERTIUS GLOVER, Assoc. M. Inst. C.E.

THE most frequent, although perhaps not the most serious, casualties to goods or mineral trains are those occurring in the draw- and buffing-gear of the rolling stock. The arrangement of the gear used by many railway companies, and approved by the Clearing House for private owners' wagons, is shown in *Figs. 1*. The draw-gear is continuous and elastic, having laminated springs to take the buffing and drawing stresses, and a cradle with india-rubber (*a*) or spiral springs (*d*) to render the drawbar elastic. Cotter-boxes

Figs. 1.

Scale, $\frac{3}{16}$ inch = 1 foot.

ARRANGEMENT OF DRAW- AND BUFFING-GEAR FOR WAGONS.

(*b*) are used in a number of wagons (principally those belonging to private owners), and the “Bowles” method is also used, which consists in inserting a spring (*c*), thus avoiding the weight and the welding involved in the ordinary design of cradle. Any spiral or india-rubber spring may be employed.

With the exception of a few fish- and meat-vans for use in passenger trains, the system of loose coupling is adopted, partly for convenience in coupling and uncoupling, but chiefly to enable the engine to start a heavy train with ease, by starting each wagon

separately. The gear is thus liable to severe shocks during starting, stopping, and shunting. The failures in the buffing-gear are few and unimportant when compared with those in the draw-gear. Couplings are sometimes broken while running, as when a long goods train is turning the top of a bank, and the engine and the first few trucks commence to run down hill before the tail-end has crossed the top, thus throwing an additional stress on the couplings, causing a "break-away," when the rear portion either runs back or runs ahead, overtaking the front portion (which is being slowed up on the driver missing the remainder of his train), with serious damage to the rolling stock. Double coupling, *Fig. 2 (c)*, may save an occasional "break-away," but it undoubtedly causes much breakage of couplings and hooks.

CAUSES OF FAILURE.

Of the various causes of failure, rough usage may be considered the most frequent. The actual stresses induced by this cause cannot be reduced to exact figures, but the fact that couplings are in use, the links of which have collapsed between $\frac{1}{2}$ inch and $\frac{3}{4}$ inch, an amount which requires a total stress of 20 tons to 23 tons by a steady pull in a testing-machine, indicates the severity of the sudden shocks on the draw-gear. The pull on the draw-gear when starting a heavy train is approximately shown by the following example:—In a train of forty wagons with a load of 10 tons and tare of 6 tons, and with a starting resistance of 20 lbs. per ton, the weight of the train would be $40 \times (10 + 6) = 640$ tons, and the total resistance $\frac{640 \times 20}{2240} = 5.7$ tons. In tests

made with a dynamometer van, however, the sudden jerk in starting a heavy goods train with loose couplings has produced a stress in the draw-bar equivalent to a steady pull of 11.5 tons.

Wrought-iron buffer-heads are often found to be indented by continual striking on the socket, so that the force on the buffing-gear to produce such a result must frequently be far in excess of the 3 tons to 4 tons required to overcome the resistance of the spring. Flaws are largely the result of rough usage, and are not easily detected when the wagon is in use, as the outside gear is exposed to the weather, which rusts over any slight crack, and the internal gear is not readily accessible. In some cases, in which broken material has been examined, the fracture has been so dirtied and rusted that the existence of a flaw has been doubtful. As a rule, however, in the case of sound material, the crystalline texture can be seen after the dirt has been removed.

Wear is chiefly noted in the draw-gear pins and eyes, in the ends of coupling-links, and in buffer-sockets. Crushing occurs in cotters and cotter-boxes, some of the latter being very light in section. Fatigue in the spring-steel results in loss of camber of the spring, and causes the rubber to become hard and inelastic. Faulty design is not a common cause of failure, the officials of the various railway companies having given thought to the working out of the details, and the private owners being tied down to a standard design approved by the Railway Clearing House. A large number of old wagons are in use, containing unsuitable parts, such as draw-hooks of small section, small draw-bar eyes, and cotter-boxes. The great want of uniformity in these is shown by *Figs. 6*; while in use any one of these hooks was liable to have the maximum force transmitted to it, although the sections varied as much as 50 per cent. Draw-bar pins appear to be the weakest portion of the draw-gear, as in no other part is so much sound material found broken, but it is as well that, in case of overstrain, the gear should give way at a point such as this, where repairs can be easily and cheaply executed.

The relative frequency of failure of the various parts of the gear, out of a total of 2,900 failures, was as follows:—

Draw-Gear.				Buffing-Gear.
Couplings.	Hooks.	Draw-bars.	Springs.	
Per Cent.	Per Cent.	Per Cent.	Per Cent.	Per Cent.
48	18	28	4	2

DRAW-GEAR.

a. Couplings.—These receive the first shocks incidental to pulling the train, and therefore constitute the most fruitful source of failure. The following Table gives the number and causes of failures during a considerable period, as shown by examination of 1,374 couplings broken in working the traffic:—

TABLE I.

Owned by—	Rough Usage.	Flaw.	Unsound Weld.
	Per Cent.	Per Cent.	Per Cent.
Railway Company	42·7	26·9	30·5
Private Firms	22·4	19·2	58·4

The three-link coupling, $1\frac{3}{8}$ inch in diameter, is now chiefly in favour, the old five-link chain, as well as the shackle and pin, have been discarded, as the chain only contained more links liable to failure, and the shackle and pin, *Figs. 2 (f)* (which was never a satisfactory arrangement, the pins failing and getting lost, etc.), cost nearly 10 per cent. more, owing to the greater amount of work involved in its manufacture. The shackle, when subjected to an extra rough shunt, would close in and bind on the draw-hook shank, becoming so stiff as to cause considerable danger to the shunter, especially when coupling with the pole.

Many companies prefer Gedge's coupling, *Figs. 2 (a) and (b)*, in

Figs. 2.

METHODS OF COUPLING.

which a flat part on the last link slips through a slot in the draw-hook; the alternative plan, *Fig. 2 (e)*, consists in putting the last link unwelded through a hole in the draw-hook and then welding it up. This is rather more expensive, the price per ton for wages being higher, as the welding of the end link is more awkward to perform and less reliable; hence this link is made of iron $1\frac{7}{8}$ inch or $1\frac{1}{2}$ inch in diameter.

Side chains, $\frac{7}{8}$ inch in diameter, are being abandoned in modern wagon practice; they are found to be of slight use, as when the centre chain gives way the front portion of the train gains additional momentum to snap the side chains. The links, also, are of weak form, as they have to be "thrown wide" in order to admit the

hook, and the wider the links are made the weaker the chain becomes. Side chains are sometimes not of equal length, and the shock thus coming on each separately, breaks them at once. If these chains are used they should be specified to stand a total stress of at least 18 tons.

Couplings should be made of high-class chain-iron (giving about 35 per cent. elongation in 4 inches, and 45 per cent. reduction in area, with a tensile strength of 22 tons to 23 tons per square inch); when side-welded they should stand a pull of at least 50 tons before breaking, this being equal to a stress of 16·83 tons per square inch, and if end-welded they should stand 35 tons to 40 tons total stress, equal to 11·7 tons to 13·4 tons per square inch. Out of the last twenty couplings (representing 1,000) tested by the Author, the range was between 55·2 tons and 63·7 tons total stress, and the average was 59·2 tons total stress, equal to 19·5 tons per square inch. The best method to secure a good class of coupling is to test to destruction samples taken from bulk, say, one from every fifty, and to specify a minimum breaking load. No proof load (as in ordinary close-linked chain) can be specified, as the maximum load which could be applied without causing the links to collapse would be too small to be of any practical value. In close-linked chain, the ratio of minimum breaking load to proof load is 2·25 to 1, which gives, for $1\frac{3}{8}$ -inch coupling of 50 tons minimum breaking load, a proof load of 22 tons, which would collapse the links $\frac{5}{8}$ inch, as shown by the following experiments:—

TESTS SHOWING COLLAPSE OF CENTRE LINK IN 3-LINK COUPLINGS.

Load Applied.	Inside Width.	5 Tons.	10 Tons.	15 Tons.	20 Tons.	25 Tons.	30 Tons.	35 Tons.	40 Tons.	Breaking Load.
Collapse in inches.	$3\frac{1}{8}$	..	$\frac{1}{16}$	$\frac{3}{16}$	$\frac{5}{8}$	$\frac{15}{16}$	$1\frac{1}{4}$	$1\frac{1}{2}$	$1\frac{3}{4}$	Tons. 58
	3	..	$\frac{3}{32}$	$\frac{1}{8}$	$\frac{3}{16}$	$\frac{7}{8}$	$1\frac{1}{8}$	$1\frac{1}{2}$	$1\frac{9}{16}$	63·7
	3	..	..	$\frac{1}{8}$	$\frac{1}{8}$	$1\frac{1}{8}$	$1\frac{1}{8}$	$1\frac{1}{8}$	$1\frac{1}{2}$	not taken.

The private owners' specification requires chains to be made of $1\frac{3}{8}$ -inch best cable iron, Government chain quality, pins and shackles not to be used.

Steel couplings are rarely used; they are less reliable than those of iron owing to the difference in welding properties, the range of temperature at which sound welding takes place being so small. The gain (in the original bar) of 4 tons per square inch in tensile strength over that of wrought-iron, *i.e.*, from 22 tons per square

inch to 26 tons per square inch, is too small to counterbalance the unreliability in welding.

TABLE II.—BROKEN WAGON-COUPLING LINKS.

Railway Company.					Private Owners.				
Cause of Failure.	Elongation in 2 inches.	Reduction of Area.	Ultimate Stress.	Bending.	Cause of Failure.	Elongation in 2 inches.	Reduction of Area.	Ultimate Stress.	Bending.
R. S. Slight flaw	42·5	37	23·30	Close Cracked.	Rough Shunt	..	..	..	Close Cracked.
Rough Shunt	..	..	..	150° Cracked.	Unsound	44·0	41	22·7	Close Good.
Flaw	39·5	35	23·53	Close Cracked.	Unsound	6·5	40	22·9	Close Good.
Flaw	34·0	32	23·85	Close Good.	Unsound	..	..	..	40° Broke.
Flaw	41·5	47	23·75	Close Good.	Flaw and Unsound	..	..	..	Close Good.
Unsound	51·5	55	23·68	Close Good.					
Unsound	46·0	43	23·26	Close Good.					
Flaw and Unsound	..	..	..	170° Cracked.					

The term "best cable iron" is vague, and some of the material classed as such is not fit for use in wagon couplings. Table II shows the results of tests of the straight parts of broken couplings (tensile and bending test-pieces being taken from opposite sides);

Scale, $\frac{1}{4}$ inch = 1 foot.

TYPES OF FAILURES.

on the whole these show good material, one test-piece in particular being B. Y. chain quality, but the weld in this case was of the worst description.

The various causes of failure will now be dealt with in detail.

Rough Usage.—As will be seen from Table I, this is the cause of about 38 per cent. of all the coupling failures.

Failures due to this cause group themselves as three different types (*Figs. 3*). First a plain break (*a*) across the end of the link, which opens and allows the other link to be drawn out; the fracture usually occurs at the welded end, where pockets, due to unsoundness in the scarfs, no doubt weaken the links at the point of maximum bending moment,

but often the two ends appear to break simultaneously (*b*), the halves then falling on to the line. The third type of failure is shown at (*c*), the whole crown of the link coming away at the welded end; the fracture occurs first at one end of the scarf, the link then opening and snapping off at the other end. The fractures have a crystalline appearance, due to the sudden shock with which the links have been broken, but a coarsely crystalline structure, with large facets, is undoubtedly an indication of inferior iron. *Fig. 4 (k)* shows a piece broken right out of the end of a side-welded coupling, which, as the fracture proved, was made of an excellent quality of iron; no material can be expected to stand such usage as this indicates. In investigating the cause of a fracture, the section of the hook that the link was on at the time of failure must be considered, as sections such as that shown in *Fig. 6 (b)* and *Fig. 7 (c)*, now almost obsolete, tend to produce fractures of this sort. Many end links are bent, *Fig. 2 (c)*, and a large number of all three links are probably broken by double coupling, but the evidence of this is not so clear as in the case of draw-hooks, as no distinctive marks due to this appear on the links.

Flaws.—These account for about 25 per cent. of coupling failures, and a large number of them are caused by rough usage, which has strained the material and started slight cracks, which have gradually enlarged, as shown in the fracture by the rusting which has taken place in the flaw. Inferior material, laminated iron, overheating in the manufacture of the links, etc., soon start flaws, which lead to breakage. Some flaws are caused by the smith, in his anxiety to make a finished-looking weld, giving it, when at a dull red heat, needless dressing with the light hammer, which tends to crystallize the iron and causes it to give way under working stress in minute transverse cracks. Red-shortness, if excessive, also causes failures of a similar nature at both ends of side-welded, and at the unwelded ends of end-welded links.

Unsound Welding.—This class of failure may be due to (1) dirt; (2) insufficient heating; (3) overheating; (4) overworking by the smith. Dirt occurs from slag and scale and the cause of its presence in the finished weld is carelessness in clearing the scarfs. Overheating is indicated by a vitreous appearance of the surface of the link and by a sponge-like fracture. In reference to the over- and under-heating of the scarfs, it is found that the range of temperature at which iron can be properly welded is much smaller in the case of the best qualities of iron than for the common brands; the better qualities thus need greater care in heating; this is easily mastered by a good smith, who will avoid the last cause of weld

failure (also mentioned under "Flaws"), namely, overworking the weld, when cooling, with the light hammer, which opens the end of the scarf and causes crystallization.

Many railways use side-welded in preference to end-welded couplings, and the results amply justify the extra expense of 5s. to 10s. per ton, which is entailed in workmanship, on account of the ends having to be heated and bent separately to gauge, and adjustment for length not being so easy as in the end-welded links, since in the latter the scarf can be more or less bent as desired. Fewer side-welded links also can be kept in the smith's fire at one time, owing to the position of the weld.

End-welded links are in general not so sound as side-welded

Scale, $1\frac{1}{4}$ inch = 1 foot.

FAILURES IN COUPLINGS.

links, as the former require an awkwardly curved scarf, $2\frac{1}{2}$ inches to 3 inches long, which is more difficult to weld than the $1\frac{1}{2}$ inch to 2 inches plain scarf at the side. The ends of the link have to take all the wear, and there is also at these points a bending action in addition to the tension; the full strength of the original bar should therefore be maintained here, and the weld (always at least 10 per cent. weaker than the original bar) should be placed at the side, where it has to resist tensile stress only. Small "pockets" (Fig. 4 (f)) are sometimes found at the inner edge of the crown of the links, where the scarf is most difficult of access; these when small have but a slight weakening effect and are usually classed as "Rough shunt—slightly unsound"; 2·2 per cent. were

so reported. These pockets are found in large chains (2 inches in diameter and over) even of the best make. The jars and shocks to which the links are subjected during work tend to crystallize the iron and to open even the soundest welds.

Examples of unsound welds are shown in *Figs. 4, (b), (c), (d) and (e)*; some of these show hardly any sign of true welding. Several of these were at the sides of the links and in these cases all the advantages of side-welding (such as reliability in welding and full strength of iron at the ends, the weakest points of the links) have been lost on account of bad workmanship. The following Table (Table III) gives the results of breaking, in the testing machine, a large number of end-welded couplings which had been in use for at least 5 years; the poverty of the results must not be altogether ascribed to fatigue of the materials, unwelding by shocks during service, etc., as the majority gave way at the weld, showing an unsound fracture; also, a few years back, end-welded couplings often failed to stand a total load of more than 25 tons to 30 tons when under test, owing to inferior workmanship.

TABLE III.—OLD END-WELDED COUPLINGS 1½ INCH IN DIAMETER.

Annealed.			Unannealed.		
Total Load in Tons.	Tons per Square Inch.	Remarks.	Total Load in Tons.	Tons per Square Inch.	Remarks.
36·65	12·12	Slightly Unsound	22·78	7·85	
27·26	8·99	Slight Flaw	24·12	8·32	Slight Flaw
28·60	9·43	" "	22·78	7·51	" "
22·78	7·51	" "	22·78	7·51	Slightly Unsound
25·02	8·25		25·47	8·40	Flaw
28·15	9·50	Slightly Unsound	31·28	10·79	Unsound
32·63	10·78	Unsound	21·00	6·93	"
28·15	9·28	Slight Flaw	28·61	9·37	"
28·65	Average.		24·85	Average.	

Wear.—This cause of failure is simply loss of section at the ends, thus increasing the stress per square inch on the weakest point. Some couplings are kept too long in use. *Figs. 4, (g) and (h)*, show losses of 20 per cent. and 25 per cent. respectively. The continual hammering at the ends of the links tends to crystallize them; annealing, after a certain number of years' working, would relieve this, and would enable the chain to run for a longer period even when the section is reduced. In Table III the annealed links show 15 per cent. greater strength than the unannealed, under the steady pull of a testing machine, but in the case of

suddenly applied loads (such as starting a train) the annealed links would probably show even to greater advantage.

β. Draw-Hooks.—Taken over a period of 1½ year, 523 of these failed as against 1,374 couplings.

The number was made up as follows:—

TABLE IV.

Cause of Failure.	Per Cent.
Rough usage	72
Flaw	20
Light section	5
Bad material	3

The hooks are forged with a short length of draw-bar, which is attached by pin and eye to the central draw- and buffing-spring, or, in some cases, direct to a length of draw-bar. They differ chiefly in the method of attachment of the coupling. This may be "Gedge's" slotted hole with the link slipped in, *Fig. 2 (a)*, or a plain hole with the link welded in, *Fig. 2 (e)*, or a pin and shackle, *Figs. 2 (f)*.

Draw-hooks are usually forged to section, and then bent to a pattern either under the steam-hammer, or by a hydraulic machine. If stamped out, the grain of the iron should be kept running with the hook, an important point sometimes overlooked in second-rate work. The material should be best selected wrought-iron scrap, giving about 23 per cent. elongation in 4 inches, and 23 tons per square inch ultimate stress. When tested in the machine, the standard draw-hook shown in *Fig. 5 (a)* should stand about 45 tons before fracture.

In the following Table (Table V) are given the results of tests of such hooks, and of tensile tests of the forged draw-bar ends:—

TABLE V.—DRAW-HOOKS.

Total Breaking Load.	Tensile Test.			Calculated Ultimate Maximum Tensile Stress on Section AA, <i>Fig. 5 (a)</i> .
	Elongation in 4 Inches.	Reduction of Area.	Ultimate Stress.	
Tons.	Per Cent.	Per Cent.	Tons per Square Inch.	Tons per Square Inch.
47·35	30·25	..	24·64	23·64
45·00	..	..	..	22·47
41·07	22·50	32	23·02	20·53
39·73	20·50	23	20·89	19·85
37·94	24·75	31	23·80	18·97

Hooks are subjected to two stresses, namely, a direct tensile stress $\frac{W}{a}$, where W = load, and a = area of cross section, superposed on a bending stress $\frac{Wl}{Z}$, where Z = modulus of section = $\frac{1}{6} b h^2$, and l = distance of neutral axis of section from line of load.

Referring to *Fig. 5 (a)*, section AA:—

$$\text{maximum tensile stress (at inner edge)} = \frac{W}{a} + \frac{Wl}{Z},$$

$$\text{maximum compressive stress (at outer edge)} = \frac{W}{a} - \frac{Wl}{Z}.$$

This formula is only meant to apply to stresses within and up to the elastic limit, and has been constructed on the supposition that the beam is made up of a number of infinitely thin layers independent of each other, and no account has been taken of the adhesion which exists between the fibres of the iron; hence, when dealing with ultimate breaking stresses, it can only be successfully used by the introduction of a coefficient “ c ,” which varies (according to the section) between 1.5 and 2.5. For hooks of this section it is usually taken as 2.

Example.—Standard hook, *Fig. 5 (a)*, broken with a total load of 45.00 tons.

Maximum tensile stress on section A A, in tons per square inch—

$$\begin{aligned} &= \frac{W}{a} + \frac{Wl}{Zc} = \frac{W}{a} + \frac{Wl}{\frac{1}{6} b h^2 c} \\ &= \frac{45}{6.81} + \frac{45 \times 2\frac{7}{8} \times 6}{2 \times (3\frac{1}{2})^2 \times 2} \\ &= 6.6 + 15.875 \\ &= 22.47. \end{aligned}$$

In his “Applied Mechanics,”¹ Prof. Cotterill, in treating of the breaking strength of beams, says: “As soon as the elastic limit is passed, the stress, at points near the surface, no longer varies as the distance from the neutral axis. It does not increase so fast, because the extension or compression is not accompanied by a proportionate increase of stress. Hence, a partial equalization of stress is produced, and the maximum stress for a given moment of resistance is reduced.” Mentioning the adhesion of the layers, he says: “When the limit is passed, the connection between those layers which are most stretched and compressed with those layers

¹ 1895 edition, p. 428.

which have not yet lost their elasticity, prevents their contraction and expansion, and so raises the limit of elasticity. Thus the apparent elastic strength is greater in bending than in tension."

The various causes of failure will now be considered in detail.

Rough Usage.—This is the most frequent cause of failure, as it

Figs. 5.

Scale, $1\frac{1}{4}$ inch = 1 foot.

FAILURES IN DRAW-HOOKS.

comprises 72 per cent. of the total number. As in the case of couplings, the hooks have to bear heavy shocks both in starting and stopping a train. Under this head comes double coupling, *Fig. 2 (c)*, a bad practice that is so prevalent in some districts that the hooks show two points of wear, one for the normal position of the coupling link, and another higher up on the hook, *Figs. 5*

(*d*), (*e*), and (*k*). From the position of the link in double coupling, the bending moment on the hook is much increased, a working load is applied which the hook was not designed for, and an unknown side stress is introduced, owing to the twisting of the links. Some of the double marks are caused by the coupling having been hung up on its own hook, *Fig. 2 (d)*. This practice, of course, increases the bending moment in the same manner as double coupling, but does not produce the same bad effects (especially on the couplings), as it causes no twisting of the links, but no useful purpose is served by it, as the ends of couplings are usually 8 inches above rail-level, and there is no danger of them striking any object on the line. Breakages caused by rough shunting show a marked crystalline fracture, due to the suddenness with which the break has occurred. In *Fig. 7 (e)* is shown a form of damage which is noticed in the hooks of mineral wagons. This is caused by striking the "centre bar" of old chaldron wagons, many of which are still in use on the collieries and docks of the north country.

Flaws and Bad Material.—Next to rough usage, flaws are the most fertile source of failure. Many of them are doubtless caused by rough working, as they are found at the inner edge of the hook, the point where the maximum stress occurs. Very poor iron, as shown by the fracture, which consists of large, bright, crystalline facets, is worked into some hooks. It is of a class quite unsuited to stand the shocks to which draw-gear is subjected. Laminated draw-hooks occasionally occur, and are due to bad workmanship, *i.e.*, a want of care in piling the iron before forging.

Figs. 5 (g) and 7 (a) show fractures indicating a mixture of steel and iron. Although the different kinds of scrap are usually kept separate, still, with the amount of mild steel now in use, a small quantity of it sometimes gets into the pile. In these two cases it has probably been a small piece of plate. In both cases the welding was perfectly sound and the resultant material good, but if metal like this is used for parts requiring welding, the results are apt to be unreliable, owing to the different welding properties of iron and steel. Specially mixed metal, termed "Fibrous steel," is now on the market.

Light Section.—Under this heading are included all remarks on sections. The depth and area of section at the plane of maximum stress should be the same in all wagon hooks, as any wagon is liable to be placed next to the engine, and its hooks will then have to bear the whole force required to draw the train. The

great absence of uniformity in this respect is shown by *Figs. 6, a to k*; thus, comparing *a* and *h*, *Figs. 6*, the areas are in the ratio

of 1 to 2, but this does not represent the ratio of the strengths of the two hooks, since the strength varies as $\frac{b h^2}{l}$; so that in this case the

strengths are in the ratio 1 to 3. Section *a*, however, is the largest hook which has come under the Author's notice, and bears a ratio to

Scale, 1 inch = 1 foot.

SECTIONS OF HOOKS AND DRAW-BARS.

the standard hook, *c*, *Figs. 6*, of 1.27 to 1 in sectional area, and of 1.5 to 1 in strength. The hook, of which *b*, *Fig. 6*, represents a cross-section, and which belonged to a 10-ton wagon, should never have been made; the bearing edge, instead of being rounded, has a ridge capable of cutting open any coupling-link crown. This hook has probably been the cause of numerous broken couplings.

"Gedge" couplings occasionally fail through the metal beneath the slotted hole, as there is a bending moment at that point; thus, the hook shown in *Fig. 5 (g)* had not sufficient area at D D, for the ratio of the moment of resistance of section C C to the moment of resistance of section D D is 21 : 11, while the ratio of the bending moment on section C C to that on D D is 21 : 15; thus at section C C the hook is 36.5 per cent. stronger than at section D D. The hook shown in *Fig. 7 (k)* has been deficient in cross-sectional area at a critical point. This hook could have been improved by carrying the full depth $\frac{3}{4}$ inch further along before forming the shoulder, as shown by the dotted lines.

Smallness of section around the eye was the cause of three failures, almost exactly identical, which are represented by *Fig. 5 (b)*; the pull of the train, in these cases, acted in a line nearly 2 inches above the centre-line of the draw-bar; this defect is also shown in *Fig. 7 (k)*. Two new hooks (standard pattern), which, when tested in the machine, failed at 43.75 tons and 44.70 tons, are shown respectively in *Figs. 5 (f)* and *7 (b)*.

Fig. 7 (h) illustrates a clear case of faulty design in several respects. In the first place, the pulls on the hook and on the draw-bar could not have been in the same straight line even if the coupling had occupied the lower position, but the outline of the hook would not permit of this, and the link therefore took up the higher position, where it has 25 per cent. more leverage to

produce a bending moment on section A A, and in the case of section B B, instead of a straight pull, W (which could have been arranged for in the design), there has been a stress ($W + Wl$) where $l = 2$. This excessive stress has led to the flaw shown in the Figure, and subsequent fracture. The corners B B were also too sharp, although in this instance they did not lead to failure.

Figs. 7.

Scale, $1\frac{1}{2}$ inch = 1 foot.

FAILURES IN DRAW-HOOKS.

Figs. 7, (f) and (g), are examples of broken side chain hooks. It will be noticed how much the hook of circular section has opened out before fracture, an effect which does not occur with a flat section.

γ. Draw-bars.—The failures of these number 818, and comprise

failures of draw-bars and pins, draw-bar cotter-boxes and cotters, the number being made up as shown in the following Table:—

TABLE VII.

Cause of Failure.	Number of Failures.			
	Draw-bar.	Draw-bar Pins.	Draw-bar Cotter-Box.	Draw-bar Cotter.
Rough usage	232	131	8	5
Flaw	111	32	7	4
Unsoundness	77	..	..	..
Light section	64	10	..	..
Coarse and bad material .	23	2	21	..
Worn	..	9	20	39
Light section and flaw .	18	..	..	..
Miscellaneous	5	..	..	..
Totals	530	184	56	48

The draw-bars are now usually $1\frac{3}{4}$ inch in diameter, but those in use range between $1\frac{1}{4}$ inch in diameter and 2 inches square. They are between 3 feet and 6 feet in length, in the best practice being continuous and elastic, having an eye and pin at one end for attachment to the draw- and buffing-spring, and a screw at the other end to the cradle or mechanism for giving elasticity to the bar. In wagons in which the gear is continuous but not elastic the two ends are joined by cotters and box, *Figs. 9*. In some of the old wagons, and a few new ones for special purposes, the draw-bars are not continuous. This is apt to throw great stresses on the buffer beam and framework.

Draw-bars should be made from the same quality of iron as the hooks, giving 23 per cent. elongation in 4 inches and 23 tons per square inch ultimate stress. The draw-bar ends are forged and finished separately, and are afterwards welded on to the long plain portion. This method is adopted partly for convenience in handling, such as cutting the screw in the lathe, and also because it is found more satisfactory to forge the eye from scrap iron than to weld a collar on to the rod, "jumping up" the rod end being, of course, inadmissible.

The failures may be treated under the following headings:—

Rough Usage.—Table VII shows very clearly the extent to which draw-bars and pins suffer from this cause, the proportions being 45 per cent. and 71 per cent. respectively. The bars show a straight fracture across the fibres, with a clean crystalline surface and no reduction of area. The greater number of failures occur

in the eye, at the sides or the end, *Figs. 8, (a) to (e)*. Pins usually break right across, showing a wholly crystalline fracture, and as they are an easy fit even when new, they fail by a bending rather than a shearing action. Indeed, some pins which have been at work in a worn eye bend slightly, previous to failure. The Author is of opinion that a number of pins fail from being too light for their work rather than from rough usage, although a large majority have been classed under this head merely on account of the soundness of the material in them. Rough usage discloses the weak points of the V thread adopted, as most of the failures in draw-bars caused thus, other than those through the eye, occur at the bottom of the thread just outside the nut. Round threads (as used for carriage screw-couplings) would obviate this. Cotter-boxes give way by the metal being torn off at the end of the cotter-hole, sometimes by the complete end being torn off. Cotters show the planes of shearing, often being taken out in three pieces, one in each side of the box, and the centre piece remaining in the draw-bar.

Flaws.—Undoubtedly many of these are due to fatigue of the material. In the bars themselves they occur chiefly in the eyes, and have been caused in the first instance by rough usage, which produces a stress exceeding the elastic limit, and forms a crack at the weakest point, either the end or the side, and this crack afterwards rusts and extends under service. The fact that flaws frequently accompanied light sections serves to bear this out, as the sectional area was so small that shocks must have occasioned stresses in them in excess of the elastic limit, which only produced stresses well under this limit in the rest of the gear. The fractures of the eyes broken through the end, as in *Figs. 8, (a) and (d)*, are often found on examination to be old and rusty, showing that the end has been broken by a shock, and that the eye has been sufficiently strong to hold together under the ordinary stresses of daily use until it has experienced a second shock, which has bent open the sides and allowed the pin to escape. In the pins, flaws invariably show as a rusty crescent-shaped defect on one edge of the fracture, the rest of the area being finely crystalline in grain. Sometimes badly punched or square-cornered cotter-holes start cracks at their upper corners, causing fractures similar to those shown in *Figs. 8, (f) and (i)*. Cotter-box flaws are the result of poor material being used.

Unsound Welds.—Ordinary cases of bad welding at either end, to the eye or to the screwed portion, constituted 15 per cent. of the failures in draw-bars. In some cases the scarf had only been

holding by a small corner, *Figs. 8, (h) and (k)*. A common fault in all these draw-bar welds is the shortness of the scarf. They are sometimes little better than a butt weld.

Light Section.—In the draw-bar rods themselves this is simply a want of cross-sectional area. Thus two wagons may be next to one another, one having a draw-bar of $1\frac{1}{2}$ inch in diameter, and the other one of $1\frac{1}{8}$ inch square, a difference in area and strength of nearly 100 per cent. In draw-bar eyes (in which the greatest number of failures occur), design varies considerably, *Figs. 8, (a) to (e)*. The actual stresses in an eye are very complex. It is probable that the stress at the sides is not uniform, but is more intense at the inner edge near the hole. At the end of the eye there is a bending moment, due to the load on the pin, besides the

Figs. 8.

Scale, $1\frac{1}{2}$ inch = 1 foot.

FAILURES IN DRAW-BARS.

crushing action of the pin, which in time elongates the hole. The bending moment tends to burst open the back of the eye parallel to the fibres, and most failures of this kind show the fibrous texture very clearly. In designing the eye it should be noted that the sides have only to resist a tensile stress; the sectional area of the end should therefore be greater than the combined area of the sides, with an extra allowance for wear. In the standard eye the area of the end = (the area of the two sides) $\times 1.2$. Many cases of small pins are met with. The standard diameter is $1\frac{1}{8}$ inch, but pins of $1\frac{1}{16}$ inch in diameter are sometimes noticed. *Fig. 8, (i)*, shows one of $1\frac{1}{4}$ inch in diameter.

Although no failures in cotter-boxes are reported as "light section," there are numerous cases which are noted as "worn," of which *Fig. 9, (d)* is typical. In this case the bearing area on the

cotter is much too small, the combined area of the two sides being $\frac{1}{4}$ square inch, while the bearing area in the draw-bar is 1 square inch, a difference quite sufficient to explain the distortion of cotters into such outlines as those shown in *Figs. 9, (f) and (g)*. The material in the square boxes is usually so bad that no other cause is required to account for their failure.

Coarse and Bad Material.—Only a small percentage of draw-bars and pins fail from this cause, one case of a laminated bar having come under notice. In the others the fracture has shown the large crystalline facets characteristic of iron which has not had much work expended on it in manufacture. Some of these failures might be due to fatigue, as the shocks of working are severe and often repeated. Cotter-boxes are usually made of plate, the cotter-holes being first punched in them, and the plate then bent (to suit the draw-bar) until the edges touch. Sometimes the edges are welded and a ring is welded on at each end. The plate employed is of a common quality, often laminated, and showing a fracture made up of alternate fibrous and crystalline layers. Recently, however, a change for the better has been noted, boxes of $\frac{3}{8}$ -inch or $\frac{1}{2}$ -inch steel plate of good quality being used, which have only failed through rough usage. The total number of cotter-boxes in use is not very large, and they chiefly occur in private owners' wagons, but 37½ per cent. of failures of boxes is too large for a cause of

Scale, 1½ inch = 1 foot.

FAILURES IN COTTER-BOXES.

this sort. Of the fifty-six reported, only two belonged to railway companies' wagons.

Wear.—The percentage of failures due to this cause is largest in the case of cotters, of which 81 per cent. have failed in this way. As they are not fitted accurately at first, the play in the holes provides opportunities for crushing. The same applies to cotter-boxes, in which the area to resist crushing is often too small. Some cotter-holes in the boxes thus become so badly worn that the cotters would drop out altogether were it not that the hole in the draw-bar wears so little that the head of the cotter is unable to pass through. A case in which the cotter—of good material—has worn the hole right through by a crushing action until the metal has been insufficient to resist the shearing stress and has finally given way, is illustrated in *Fig. 9 (e)*. It is of little use to replace cotters broken as a result of wear in the holes unless the box is renewed at the same time, as a cotter which fits the hole in the draw-bar is usually so much smaller than the holes in the box that the jar, caused by taking up the $\frac{1}{2}$ inch or more of play, soon breaks any cotter, however good. Indeed, the cotter shown in *Fig. 9 (g)* was renewed twice in a journey of 80 miles, the holes being much worn. A few of the draw-bar eyes which failed through want of section at the end have been strong enough originally, but have become too slight after wearing $\frac{3}{8}$ inch or $\frac{1}{4}$ inch.

Miscellaneous.—One case each of "thread stripped" and "nut off" have been noticed, caused by the split pin at the end of the draw-bar dropping out or not having been put in at first, the nut then working back until it falls off or until it embraces so small a number of threads (usually three) that they are unable to resist the shearing stress. One case of a bent draw-bar pin was met with, caused by the nut dropping off and allowing the pin to jump up and get twisted between the draw-bar eye and the lug of the buffing spring buckle.

δ. *Springs, Buckles and Cradles.*—The Table (Table VIII)

TABLE VIII.

Cause of Failure.	Spring Buckles.	Spring Cradles.	Miscellaneous.
Rough usage	57	1	3 Spiral springs.
Flaw	14	..	2 Plates.
Unsoundness	18	3	
Light section	8	..	
Bad material; laminated . . .	4	..	
	101	4	

shows the number of various types of failure, which have been noticed and reported at the time of their occurrence; one common type of failure—broken plates, which are noticed in a large percentage of the springs when the wagons are under repair—is not included.

As will be seen from the general arrangement, *Figs. 1*, the most important springs are of the laminated type, about 5 feet 10 inches over the ends, and serve both for drawing and buffing purposes; the pull is transmitted through the hook or draw-bar to the buckle at the centre of the spring, while the buffing stresses are applied at the ends through the buffing-rods and shoes. The making of a spring involves several operations—shearing, nibbing, and tempering the plates, forging and putting on the buckles. As the majority of serious failures occur in the buckles, the best method of manufacture is indicated here. The standard buckle (*Fig. 10 (a)*) consists of two side-plates, $\frac{5}{8}$ inch thick, and two end pieces (each comprising an end and two lugs), $\frac{3}{4}$ inch thick. The end and lugs are bent and a slight scarf is left at each corner in order to ensure a sound weld; these are welded under the steam hammer, with special swage blocks. Given good material, which can be tested in the usual manner, the only other test that can be made on the buckles, as delivered from the makers, is to try the soundness of the welding, by selecting one or two buckles out of a batch and forcing a drift into the space to be

FAILURES IN BUCKLES.

occupied by the spring plates; this produces the results shown in *Figs. 11, (a) to (d)*. In *(a)*, the weld on one side has opened out $1\frac{7}{8}$ inch at one end and $1\frac{3}{8}$ inch at the other. A similar result is shown at *(b)*. As good a result as can be obtained is indicated by *(c)*, the weld being perfectly sound and the wrought-iron side piece tearing through the solid in two places. The opposite extreme is illustrated by *(d)*, the weld being very poor (about 40 per cent. unsound), the side plate leaving the end piece before sufficient stress has been put on even to bend the plate. The inner ends are the most difficult parts to weld soundly, and this test tries them first and most severely. Cradles are forged in

RESULTS OF BUCKLE TESTS.

two pieces (each consisting of one end washer and two half-sides), which are then welded together with a plain scarf at each side, or, as is sometimes preferred, with a plain scarf at one side and a "glut piece" welded in at the other. The causes of failure may be subdivided as before.

Rough Usage.—The majority of buckle failures due to this cause occur as shown in *Figs. 10 (d) and 11 (e)*, the fracture commencing at the inside corner and tearing first through it and then through the side piece. The pins in these cases are usually found broken, and there is little doubt that these failures are due to the breaking of the pins rather than to any fault in the design of the buckle, as, when the pin breaks, the draw-bar end causes the broken portions to force

the lugs apart, thus throwing a very severe stress on the corners of the buckle. Many spring plates are broken by rough usage; failures are assisted by burnt steel, too high carbon temper, or insufficient "letting down" after hardening. The cases of broken spiral springs were very complete, each being in about fifteen to twenty pieces, hardly a single turn remaining intact.

Flaws and Bad Material.—Flaws are found in numbers of spring plates, and are due either to the steel being too hard (either in its composition or due to careless tempering), or to faults in the manufacture of the spring, such as overheating, or water-cracking in the hardening process, etc. Springs taken out after being some years in use often display old flaws, rusted all over, in the

plates, the latter having evidently failed soon after being put into service. The flaws in buckles are found in the corners, and may have been caused by overheating preparatory to welding, with assistance, perhaps, from an extra rough shunt. Bad material, used in buckles, is shown by a laminated fracture (crystalline and gray bands alternating); this generally develops a flaw in the corner, as it will not withstand bending to form the end piece. Very few cases of poor material occur in the draw-bar cradles, as they are forged from scrap-iron, which must have a fair amount of work expended on it before the final shape is attained.

Light Section.—These cases occur in the U piece forming the sides and end; in the standard spring they are made $\frac{3}{4}$ inch thick, which gives strength to the corner, helping it to resist the bending action on it when a pin breaks; *Fig. 10 (d)* shows a case of failure in which the thickness was only $\frac{1}{2}$ inch, while the outside piece, which has chiefly a simple tensile stress to resist, was made $\frac{3}{4}$ inch thick. An extra large rivet ($\frac{1}{2}$ inch or over) through the centre of the buckle and spring plates seriously reduces the section at the point of maximum bending moment of the spring, and many plates fail at this point.

Unsoundness.—Many of the failures due to this cause might have been avoided by an alteration in the design, allowing greater facilities for thoroughly working the weld. Although very few cases of cradle failure are included in Table VIII, the Author finds from subsequent experience that the percentage is fairly correct, as out of eleven examined, seven, or 63 per cent. of the failures, were due to unsound welds. *Fig. 12 (a)* illustrates a

FAILURES IN SPRING CRADLES.

typical cradle failure; the welds were ordinary scarf-welds; the detail view of the weld shows that only about 20 per cent. was sound, and after this side has given way the pull on the train has attempted to set the uninjured side and the two end washers in one straight line, until this side has given way at the corner through the solid metal; the weld at this side was of the best to resist such usage without the least sign of opening. The cradle after being damaged had evidently run for some time before ultimate failure, as the draw-bar was bent and considerably worn, as also were the holes; it was probably broken by a shock, such as starting a train. In *Fig. 12 (b)* is shown a cradle which was so roughly used that it was bent $\frac{5}{8}$ inch at the centre; the weld is not in such a good position to resist the stresses as in the last example; neither is the method of welding so good. After the weld (60 per cent. sound) failed, the opposite side has broken through the solid; probably the whole breakage was caused by a severe shock, which bent the washer, opened the weld and broke the opposite side simultaneously. *Fig. 12 (c)* illustrates a form of buckle which does not lend itself to sound welding so well as the standard pattern (*Fig. 10 (a)*), and the surfaces of contact are only 3 inches by $1\frac{1}{2}$ inch at each corner, which could be doubled with advantage; only 45 per cent. of the weld that failed was sound; the weld has given way first, and has then broken the buckle through the solid metal at the opposite side. *Fig. 10 (b)* shows another method of making a similar buckle; this weld was hopelessly bad, only about 30 per cent. being sound; the rivet, $\frac{1}{2}$ inch in diameter, has evidently assisted to hold the fabric together, a purpose for which it was never intended. Any failures of buckles made by this method, which have come under the Author's notice, have been due to very poor welding.

Fatigue causes plate springs to weaken and lose camber; after the load is taken off, the points of each plate are found to be 1 inch or 2 inches away from the back of the next plate, which they should touch when new. Tensile tests fail to show any alteration in strength or elongation. Rubber springs (used in cradles) become hard, inelastic, and cracked after long service.

BUFFING-GEAR.

This does not fail so often while at work as the draw-gear, because the action of the stress is direct and acts on few and simple parts. The ratio of failures in draw-gear to those in buffing-gear is 46.7 to 1, leaving out of account in each case the

spring and buckle failures, as the same springs serve for both draw- and buffing-gears in the standard design, *Figs. 1*. There are various other arrangements of buffing springs (such as *Figs. 1 (e)*), but these necessitate a different type of draw-gear spring. The following Table shows the various causes of failure and the number of failures due to each:—

TABLE IX.

—	Heads.	Rods.	Sockets.
Rough usage	4	9	3
Flaw	1	8	1
Unsoundness	1	6	..
Light section	3	..	..
No account	8	11	2

Buffer heads and rods are made in one forging of hammered iron, from selected scrap; the sockets and shoes are castings; the springs are either plate or rubber, in the former case being the same as those mentioned in the draw-gear section; sometimes, as in the case of long trolley wagons, volute steel springs are enclosed in the buffer socket, a similar arrangement to that used in locomotive work. Failures in the heads and rods are often caused by the stress having been applied at the edge instead of the centre of the head, causing a bending moment at the point where it joins the rod and also on the rod itself just outside the buffer socket; this action soon leads to failure either at once, *Fig. 13 (a)*, or by first causing a flaw, *Fig. 13 (d)*. The heads may not meet fairly, owing to sharp curves, as in the case of shunting into sidings, where the outer edges of the heads on one side meet, or by running in the same train modern and old-fashioned low wagons, in which the sole timbers are prolonged to act as buffers, in which case the upper edges of the soles meet the buffer heads below the centre line of the buffing rods. The shoes and sockets were formerly made of cast iron, but are now being replaced by malleable iron castings, and stamped steel sockets in halves riveted together. The chief cause of breakage in sockets is rough usage, the heads being driven right home against them with violence; most buffer heads show a groove round the rod, *Fig. 13 (d)*, due to this cause, showing that it is of common occurrence. The wear of buffer sockets is never sufficient to be of any serious moment. The shoes, when made of cast iron, were continually breaking, either through the cotter holes or at the end where the spring abuts; this latter point has to bear considerable shocks,

especially when the spring loses camber, as there is then considerable play to be taken up between the end of the spring and the socket. In addition, the wear caused by the spring is very serious, *Fig. 13 (c)*, and reduces the section at a point where there are compressive and bending stresses; cases frequently occur in which a hole is worn completely through the end of the shoe, thus impeding the free movement of the end of the spring. This is a part of the wagon gear for which malleable cast iron is well suited, and its introduction has led to a considerable saving in

Figs. 13.

Scale, $1\frac{1}{2}$ inch = $1\frac{1}{2}$ foot.

FAILURES IN BUFFER-GEAR.

repairs. The method of testing these consists in forcing a drift, having a taper of 1 in 10, into them under the testing machine, the amount of force required to burst them being registered; this should be at least 15 tons, a tensile test piece from the same shoe giving about 25 tons per square inch ultimate stress. As the ends of the buffer rods abut on the bottom of the shoes, the cotters have to transmit no heavy pressure, and therefore do not give trouble. A typical shoe failure is represented in *Fig. 13 (c)*; this is caused by the pressure of the spring acting in a line not coincident with

the centre-line of the buffer-rod, and thus producing a bending action on the shoe. Some rods, as in *Fig. 13 (b)*, fail at the cotter hole, and it is probable that the rapid changes, both in intensity and kind of stress, cause the material of many rods to become fatigued; some of the fractures show a very coarsely crystalline structure, not accounted for by the quality of the iron.

In conclusion, the Author is indebted to Mr. Wilson Worsdell, M. Inst. C.E., Locomotive Superintendent of the North Eastern Railway, for having given him an opportunity of preparing this Paper, and to Mr. Arthur Collinson, Assoc. M. Inst. C.E., for help in its preparation.

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

APPENDIX.

TABLE OF TESTS OF MATERIAL FOR DRAW- AND BUFFING-GEAR.

I.—COUPLINGS $1\frac{1}{8}$ INCH IN DIAMETER.

Coupling. Fig. 3. Ultimate Stress.		Bar-Iron Test.		
Tons Total.	Tons per Square Inch.	Elongation per Cent. in 4 Inches.	Reduction of Area. Per Cent.	Ultimate Stress. Tons per Sq. Inch.
52·00	16·89	28·75	36	22·50
56·80	18·68	30·50	45	23·02
60·90	19·78	30·75	40	22·50
62·30	20·25	31·25	45	21·96
55·00	17·87	39·00	46	21·93
60·00	19·49	41·75	51	21·88
<i>50 tons (minimum).</i>		35·00	45	22·00

II.—DRAW-HOOKS.

Finished Hook, <i>Fig. 5 (a).</i>	Forged-Iron Test.		
Breaking Weight. Total Tons.	Elongation per Cent. in 4 Inches.	Reduction of Area. Per Cent.	Ultimate Stress. Tons per Square Inch.
44·68	26·00	37	23·30
44·19	24·75	35	24·37
45·53	31·25	41	22·23
46·71	25·50	27	22·85
47·35	30·25	37	23·02
48·66	33·00	41	22·30
Not broken			
<i>25·00</i>		40	23·00

III.—DRAW-BARS.

Elongation in 4 Inches.	Reduction of Area.	Ultimate Stress.
Per Cent.	Per Cent.	Tons per Sq. Inch.
35·00	40	23·02
33·25	43	22·33
34·00	42	23·02
26·75	32	21·96
31·00	43	22·83
32·00	39	22·76

30·00 40 22·00

The North Eastern Railway specified tests are shown in italics.

IV.—LAMINATED SPRINGS.

Iron for Buckles.			Spring Steel.		
Elongation in 4 Inches.	Reduction of Area.	Ultimate Stress.	Elongation in 4 Inches.	Reduction of Area.	Ultimate Stress.
Per Cent.	Per Cent.	Tons per Sq. In.	Per Cent.	Per Cent.	Tons per Sq. In.
31·75	36	21·54	21·25	42	47·35
29·75	37	21·00	20·75	49	47·40
32·75	43	21·53	22·25	47	49·70
25·75	23	23·95	14·00	25	53·57
30·00	35	23·82	13·00	28	53·60
22·00	27	24·56	10·75	17	55·83

Good tough fibrous iron.

15

.. 45 (minimum.)

Buffing Gear.

SHOES.—MALLEABLE CAST IRON.

Complete Shoe.		Test of Material.		
Bursting.		Tensile.		
Total Tons.		Elongation per Cent. in 2 Inches.	Reduction of Area. Per Cent.	Tons per Square Inch.
<i>Fig. 13 (c)</i>	13·39	..	..	26·40
	12·13	..	..	25·22
	16·65	..	..	..
	17·85	2	8	26·00
Heavy pattern	21·87	..	..	..
	27·05	..	..	..

Drift. Taper 1 in 10. Tensile from burst shoe.

CAST IRON.

Transverse Tests (2 Inches Deep by 1 Inch Broad).					
Tensile.	Supports = 18 Inches.		Supports = 36 Inches.		Crushing.
	Tons.	Deflection.	Tons.	Deflection.	
Tons per Sq. Inch.		Inch.		Inch.	Tons per Sq. Inch.
8·92	2·63	0·11	1·31	0·31	40·70
9·64	3·03	0·14	1·29	0·24	47·56
10·35	2·86	0·13	1·25	0·36	51·28
10·71	2·41	0·10	1·25	0·38	50·70
11·07	2·32	0·13	1·16	0·39	53·56
..	2·85	0·11	1·34	0·31	..
..	2·73	0·10	1·43	0·27	..
..	2·77	0·11	1·47	0·29	..
10·00	2·7	0·10	1·35	0·30	40·00

The North Eastern Railway specified tests are shown in italics.

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