

JOURNAL
OF
THE FRANKLIN INSTITUTE
OF THE STATE OF PENNSYLVANIA
FOR THE
PROMOTION OF THE MECHANIC ARTS.

JANUARY, 1853.

CIVIL ENGINEERING.

On the Expansive Working of Steam in Locomotives. By DANIEL KINNEAR
CLARK, C. E.—(*With a Plate.*)*

[Abstract of a Paper read at the Institution of Mechanical Engineers.]

In locomotives, the adoption of a low standard of boiler pressure is the first obstacle in the way of carrying out the expansive working of steam, as the more expansively the steam is worked, the less is the work done by the engine. The second obstacle is, in many locomotives, the exposure of the cylinders, by which the steam within is partially condensed. Moreover, the proportion of steam so condensed increases with the degree of expansion, in a very formidable ratio, which will be afterwards submitted to examination.

The object of this paper is to show at what rate in practice the efficiency of steam is increased by expansive working in locomotives with the best existing arrangements of cylinders, valves, and valve-gear, and to point out the conditions on which expansive action may be most successfully carried out.

I.—*Of the Action and Capabilities of the Link-Motion.*—The action of the valves in the “distribution” of the steam (a term borrowed from the French) is regulated by three elements, the lap, the lead, and the travel.

* From the London Civil Engineer and Architect's Journal, September, 1852.

When these are given, the point of the stroke of the piston at which the steam is admitted to the cylinder, cut off, exhausted, and compressed or shut up, are all deducible by model, by diagram, or by calculation. This can be done, whether the valve derives its motion from a single eccentric, or from a link-motion, as the motion of the valve is virtually the same in both cases. The way in which the valve is caused to cut off or suppress the steam earlier by the link-motion, is by *shortening the travel of the valve*; this is accomplished by means of the reversing gear, in such a manner that whatever be the reduction of travel communicated to the valve, the lead is always at least the same as in full gear, and with the shifting-link is rather increased.

In working out the four changes in the distribution of the steam, already enumerated, which regulate the movements of the steam, the action of the link-motion is such that, 1st, the sooner the steam is cut off, the sooner it is exhausted, the sooner the port is closed for exhaustion, and the sooner the port is opened for the admission of steam.

2d, That though every change is made earlier—as measured in parts of the stroke—there is less difference in the position of the points of exhaust, compression, and admission, than in that of the cutting off. Consequently, the shorter the admission, the longer is the expansion, as the exhaust point does not recede so much as the point of cutting off.

3d, That by the shifting link-motion, the steam may be cut off at from $\frac{1}{6}$ th to $\frac{1}{4}$ th of the stroke.

4th, That though the exhaust takes place earlier for every increase of expansion, it does not in any case take place within the first half of the stroke. For mid-gear it occurs in fact at 54 per cent. of the stroke; and the steam is expanded into $3\frac{1}{2}$ times the length of stroke at which it is cut off.

5th, That the period of compression, increasing as the admission is reduced, amounts to about one-half stroke in mid-gear.

6th, That the pre-admission of the steam, not above 1 per cent. of the stroke in full gear, reaches about 10 per cent. in mid-gear.

These results prove that the link-motion is capable of cutting off steam as early in the course of the stroke as can ever be advisable in practice.

It has been seen that the earlier the steam is cut off, the *earlier also it is exhausted*; until in mid-gear it may be released at half stroke. This has been deemed a serious objection to the use of link-motions for high expansion, as it is supposed to lead to a serious loss of expansive action, by exhausting prematurely. This loss is, however, a mere trifle in practice. The escape of the steam is by no means instantaneous, as is easily proved by the diagrams in fig. 1 (Plate I.), taken by the writer from the *Caledonia*, passenger engine, by means of M'Naught's Indicator, at speeds of 1 and 2 miles per hour. The numbers in the diagrams indicate the number of the sector notches to which the reversing lever was placed, while the diagrams were described. Referring to No. 1, taken under full gear, the steam is shown to be admitted to the cylinder a little before the beginning of the stroke, at A. From B to C the steam is admitted, at C shut off, expanded to D, and thence exhausted to E, the end of the

stroke, whence it continues to be exhausted till the point F in the return stroke, where the exhaust port is closed. Now, the exhaust line, D E, shows that nearly all the period of exhaust for the steam stroke is employed for the complete evacuation of the steam. And if this be the case for speeds of 1 and 2 miles an hour, it is much more so for the regular working speeds of trains. To select from a very admirable series of Indicator Diagrams, with copies of which the writer has been favored by Mr. Daniel Gooch, by whom they were taken from the cylinder of the *Great Britain*, locomotive, on the Great Western Railway, the figs. 4 and 5 contain diagrams taken at 17 and 55 miles per hour respectively, under the 1st, 3d, and 5th notches of the sector. The following are the conditions of the valve motion of this engine, when the diagrams were taken:

State of the Valves of the "Great Britain" Locomotive, G. W. R.

Cylinder, 18 × 24 inches; wheel, 8 feet; lap, $1\frac{1}{4}$ inch; constant lead, $\frac{3}{8}$ -inch; travel in full gear, $4\frac{3}{4}$ inches; blast orifice, $5\frac{1}{2}$ inches diameter.

No. of Notch.	Position of Points of Distribution.			Period of exhaust during steam stroke.
	Cutting off.	Exhaust.	Compression.	
	Inches.	Inches.	Inches.	Inches.
1	16	$21\frac{3}{8}$	3	$2\frac{5}{8}$
3	$11\frac{1}{4}$	$19\frac{1}{4}$	5	$4\frac{1}{4}$
5	7	$17\frac{3}{8}$	$7\frac{1}{2}$	$6\frac{3}{8}$

On the diagrams the points of cutting off and exhaust are marked, and the steam line falls only very gradually during the period of exhaust, and especially at the high speeds. The expansion curves are shown by dotted lines, A, B, C, figs. 4 and 5, continued to the end of the stroke. These are easily calculated in terms of the relative volumes of steam, from the pressures indicated at the points of exhaust, and are such as would have been described had the exhaust been delayed till the end of the stroke. The shaded areas, A, B, C, inclosed between these dotted curves and the curves actually described, express the *power lost* by exhausting the steam *before the stroke is completed*. Averaging them for the whole stroke, they are as follows:

	Low Speeds.		High Speeds.
1st Notch,	$\frac{1}{8}$ lb. per in. loss.	.	1 lb. per in. loss.
3d " "	$2\frac{1}{4}$ lb. "	.	1 lb. "
5th " "	$3\frac{1}{8}$ lb. "	.	$\frac{3}{8}$ lb. "

The losses at high speeds are very small, merely nominal; and curiously enough, the loss by the *earlier* exhaust of the 5th notch is actually less than that under the 1st notch. The losses are of course greater at the low speeds; but even then, in the 1st notch, which is the only notch employed at very low speeds, the loss does not amount to 1 lb. per inch. The 3d and 5th notches are employed only at speeds much above 17 miles per hour, and the loss by them is of no practical moment.

Upon the whole, it follows that the possible *loss by the early exhaust* yielded by the link-motion is of *no importance*. On the contrary, it can

be proved to be *beneficial*, as an early exhaust is at high speeds essential to a perfect exhaust during the return-stroke. It plainly appears, therefore, that with the existing arrangements of locomotives, any attempts to eke out the power of the steam-line, by prolonging the expansion materially beyond what is accomplished by an ordinary valve and link-motion, are not only useless, but highly prejudicial.

Another objection to the link-motion is, that the steam is injuriously *wire-drawn* by it when under great expansion. Hence the numerous attempts to supersede the link by the employment of a separate expansion valve. The diagrams, fig. 5, may be referred to as examples of wire-drawing by the link. They were taken nearly consecutively with one opening of the regulator; and it is clear that the steam attained fully as high a pressure in the cylinder under the 5th notch as under the 1st. The pressure falls considerably towards the point of cutting off, but from the form of the steam-line, it is plain that very little additional steam is admitted for an inch or two before the cutting-off actually takes place. The most of the steam is admitted at the higher pressure, and in fact a partial expansion of the steam already admitted takes place for some distance before the expansion nominally begins. Thus the *wire-drawing* is, to a great extent, equivalent to an *earlier cutting-off*, and a greater degree of expansion. The whole possible loss by wire-drawing is comprised within the dotted line D, added to the diagram, which is merely an extension of the expansion curve to meet the steam line, drawn horizontally to represent a free admission up to an imaginary point, D, of cutting off, 5 inches from the beginning of the stroke. This shaded area, D, amounts exactly to a mean loss upon the whole stroke of *one pound per square inch*, by *wire-drawing*, under high expansion. For the 1st and 3d notches, the amount of loss by wire-drawing must obviously be still less; and, in short, the objection of *wire-drawing by the link-motion*, when of liberal proportions, is of *no practical weight*.

Another objection to the link-motion, and apparently the most formidable one, is the large fraction of power neutralized by the *compression* of the exhaust steam, and which increases with the degree of expansion. Compression, however, involves no loss of efficiency; for as by compression a quantity of steam is incidentally reserved and raised to a higher pressure, it gives out the power so expended in compressing it, during the next steam stroke, just as a compressed spring would do in the recoil. But, apart from this general argument, the actual efficiency of the steam in the cylinder, with and without compression, may be exactly estimated. The most direct method of doing so is, to find the quantities of water consumed as steam for one stroke, under the two conditions, and to compare them with the relative effective mean pressures. It will suffice to analyze, as an example, the high speed diagram, fig. 5, under the 5th notch, No. 5. The volume of steam admitted is measured by the product of the area of the piston, (254.47 in.,) and the period of admission, *plus* the total clearance in the cylinder and steam passage; the clearance being measured for simplicity in inches of stroke, we have $7 + 1.8 = 8.8$ inches, for the total volume admitted. The pressure of the steam when cut off is 65 lbs., for which the relative volume of water is 359. There-

fore, the volume of water as steam, or the water equivalent of the steam admitted, is

$$\frac{254.47 \times 8.8}{359} = 6.24 \text{ cubic inches.}$$

From this is to be deducted the quantity of steam reserved by compression; the volume so reserved is measured by the period of compression, *plus* the clearance ($7.5 + 1.8 = 9.3$), and the pressure at the point of compression is 8 lbs., for which the relative volume is 1125. Then the water equivalent of the reserved steam is—

$$\frac{254.47 \times 9.3}{1125} = 2.10 \text{ cubic inches;}$$

subtracting, there remains $6.24 - 2.10 = 4.14$ cubic inches of water as steam, actually expended for one stroke of the piston.

Were there to be no reservation of exhaust steam by foreclosing the exhaust port, the whole area of resistance by compression would be removed, and there would be a reserve of steam of atmospheric pressure equal in volume to the clearance only. The relative volume of atmospheric steam is 1669, and the water equivalent of the reserve would be

$$\frac{254.47 \times 1.8}{1669} = 0.27 \text{ cubic inches;}$$

the expenditure per stroke would be $6.24 - 0.27 = 5.97$ inches of water.

	Per Inch.
Now the positive mean pressure during the steam stroke, as indicated, is	40.9 lbs.
And the mean resistance by compression is	11.5 "
	<hr/>
Thus the effective mean pressure is	29.4 "

This effective mean pressure of 29.4 lbs. is maintained by a consumption of 4.14 inches of water per stroke; and it has just been found that with the compression removed, the positive mean pressure of 40.9 lbs. per inch would be maintained by a consumption of 5.97 inches of water per stroke. The effective pressure created per cubic inch of water is therefore,

$$\text{In actual practice} \quad \frac{29.4}{4.14} = 7.1 \text{ lbs.}$$

$$\text{And would be by removing compression} \quad \frac{40.9}{5.97} = 6.9 \text{ lbs.}$$

These quantities are expressions of the relative efficiency of steam employed with and without compression; they are virtually identical, and show that the resistance by compression in the cylinder, due to the action of the link-motion, does not in the slightest degree impair the efficiency of the steam.

The last objection to the use of the link, requiring notice, is that at *high speeds* considerable *back exhaust pressure* is created. The amount of this is very various, and it depends also on circumstances for which the link-motion is not responsible; such as a deficiency of inside lead (which is regulated by the lap), small ports, a small blast-orifice, and imperfect protection of the cylinder. It suffices on the present occasion to point to what can be done by superior arrangements, as exemplified in the diagram, fig. 5, taken from the *Great Britain*. The cylinders of this engine are in a manner suspended in the smoke box, and thoroughly protected; the steam ways are very large, 13×2 inches, being in area about $\frac{1}{10}$ th of the cylinder; the exhaust passage is very direct; and the blast-orifice is $5\frac{1}{2}$ inches diameter, or about $\frac{1}{11}$ th of the area of cylinder. As a whole, these proportions are superior to those of any other engines with which the writer is acquainted; and the diagrams prove that the per centages of back exhaust pressure, in terms of the positive mean pressure, at 55 miles per hour, are—

For the 1st notch,	8 $\frac{3}{4}$ per cent.
For the 3d notch,	5 $\frac{3}{4}$ per cent.
For the 5th notch,	nothing.

Better results than these should not in practice be required, for when locomotives are adapted to their work, and running at high speeds, they ought not to require an admission of steam above half stroke. However, the area of blast-orifice rules the back exhaust pressure; and, when the cylinder is duly proportioned to the boiler, it is quite practicable, by a few modifications in detail, still further to increase the orifice, sufficiently to banish all traces of back pressure of exhaust at all practicable speeds.

II.—*Of the Rate of Efficiency of Steam worked Expansively in the Locomotive, by the Link-Motion.*—To determine this ratio experimentally, under the actual circumstances of clearance, wire-drawing, and back pressure, the writer has analyzed twenty-six of the indicator diagrams from the *Great Britain*, already referred to, taken at speeds of 15 to 56 miles per hour, of which the figures are examples. The following table contains in the first nine columns an analysis of these diagrams; the effective horse powers, column 10, are estimated in terms of the diameter and stroke of cylinder, the diameter of the wheel, and the effective mean pressures in the 9th column. The water equivalents, columns 11, 12, and 13, are estimated from the indicated pressures and the period of the distribution for each notch, in the way already exemplified. The expenditure of steam per hour, column 14, is deduced from column 13, in terms of the speed, the cylinder, and the wheel; and, dividing that by the effective horse power, we have the contents of column 15 in inches, and of column 16 in pounds. Column 17 contains the coke consumed per horse power per hour, deduced for the several diagrams from the consumption of water, column 16, allowing 1 lb. of coke to evaporate 8 lbs. of water.

Referring to the contents of the last two columns of this table, it is obvious that the consumption of water as steam, or of coke, for a given amount of work done, becomes less the more expansively the steam is

Results from Indicator Diagrams, taken from the Great Britain, Locomotive, G. W. R., in 1850.

No. of Diagram.	Speed of Engine in miles per hour.	Indicated steam pressure in cylinder in lbs. per square inch.										Water equivalents.					Coke consumed per effective h. p. allowing 1 lb. for 8 lb. of water.
		Back pressures.					Effective horse power indicated.	Total Admitted for one stroke, from diagm.	Reserv- ed by Com- pression stroke.	Actually expended during 1 stroke.	Actually expended per effective horse power per hour.						
		Maxi- mum pressure during admis- sion.	Posi- tive mean pres- sure.	Ex- haust.	Com- pression.	Sum of back pressures.											
	Miles.	lbs.	lbs.	lbs.	lbs.	lbs.	h. p.	cu. in.	cu. in.	cu. in.	cu. ft.	cu. in.	lbs.	lbs.	lbs.		
1	15	70	63.8	1.6	2.4	4.0	59.8	190	1.32	1.00	12.53	89.83	817	29.5	3.09		
2	17	88	80.3	0.6	1.9	2.5	3.0	284	15.89	0.82	15.07	124.53	758	27.4	3.42		
3	21	95	86.2	1.2	3.0	4.2	4.7	372	16.71	1.09	15.62	159.45	741	26.8	3.35		
4	24	85	76.7	0.9	1.6	2.5	3.1	384	15.30	0.82	14.48	168.95	760	27.5	3.44		
5	27	80	70.6	1.5	2.2	3.7	5.3	389	13.89	0.91	12.98	170.37	757	27.4	3.42		
6	31	90	79.6	1.7	3.7	5.4	6.7	497	15.62	1.13	14.49	218.35	759	27.4	3.42		
7	31	80	73.2	2.9	2.2	5.1	6.9	456	14.76	1.00	13.76	207.35	786	28.4	3.56		
8	49	60	51.4	3.6	4.4	8.0	15.5	459	10.73	1.34	9.39	223.60	842	30.4	3.80		
9	54	89	80.4	6.8	6.0	12.8	15.8	763	15.62	1.56	14.06	369.07	846	30.2	3.77		
1st Notch—Means		82					48.2							28.3	3.54		
10	17	88	69.9	0.0	3.8	3.8	5.4	242	12.19	1.10	11.99	91.65	654	23.7	2.96		
11	18	70	55.3	0.8	4.5	5.3	9.4	104	10.33	1.35	8.98	78.57	700	25.3	3.16		
12	21	92	72.3	0.0	4.2	4.2	5.7	309	12.50	1.16	11.34	71.57	647	23.4	2.92		
13	26	72	57.1	0.0	4.9	4.9	8.5	293	10.55	1.41	9.14	115.52	681	24.6	3.07		
14	31	79	60.3	1.2	6.0	7.2	11.9	356	10.87	1.48	9.39	141.50	687	24.8	3.10		
15	32	86	64.4	0.8	4.9	5.7	8.4	407	11.33	1.29	10.04	156.20	663	24.0	3.60		
16	40	76	55.7	0.4	4.7	5.1	8.1	506	9.78	1.35	8.43	163.87	648	23.4	3.06		
17	51	70	49.1	2.0	6.2	8.2	16.6	450	8.65	1.54	7.11	176.30	677	24.5	2.92		
18	55	84	62.0	3.6	7.6	11.2	18.0	508	10.54	1.66	8.68	237.42	680	24.6	3.03		
3rd Notch—Means		80					34.5							24.3	3.03		
19	17	89	53.2	0.0	9.6	9.6	18.0	435	7.90	1.85	6.05	50.00	543	19.6	2.46		
20	18	70	42.1	0.5	6.6	7.1	16.7	350	6.59	1.76	4.83	42.26	537	19.4	2.42		
21	21	93	59.5	0.0	6.3	6.3	11.1	50.2	2.28	1.42	6.78	69.91	525	18.9	2.36		
22	23	74	41.8	0.4	6.2	6.6	15.7	35.2	2.13	1.68	5.19	70.64	537	20.7	2.69		
23	31	83	46.3	0.0	7.4	7.4	15.5	39.1	2.62	1.59	5.70	85.89	566	20.5	2.66		
24	36	80	39.0	0.0	8.5	8.5	21.1	30.5	2.37	1.85	4.23	74.02	540	20.6	2.44		
25	50	77	34.7	0.5	8.0	8.5	24.4	26.2	2.63	1.76	3.76	91.39	600	21.7	2.71		
26	66	90	40.9	0.0	11.5	11.5	28.1	28.4	3.53	2.10	4.14	113.20	654	20.1	2.51		
4th Notch—Means		82					36.1							10.1	2.51		

worked; and the means of the several quantities for the notches separately are as follows:

Consumption per Horse Power per Hour.

For the 1st notch,	28.3 lbs. water,	or 3.54 lbs. coke.
" 3d "	24.3 " "	or 3.03 " "
" 5th "	20.1 " "	or 2.51 " "

As the results under each notch vary very little, the means above stated may be adopted for all practical speeds without material error. From these mean quantities the following rule is derived:

RULE I.—*To find the consumption of Water as Steam per horse power per hour, for a given period of admission.* Multiply the part of the stroke in inches during which the steam is admitted by 22, and divide by the length of stroke in inches, and add 14 to the quotient. The sum is the required consumption in pounds. Let L =length of stroke, S =the period of admission of steam, and W =the consumption of water in pounds per horse power per hour; then

$$W = 22 \frac{S}{L} + 14 \quad (1.)$$

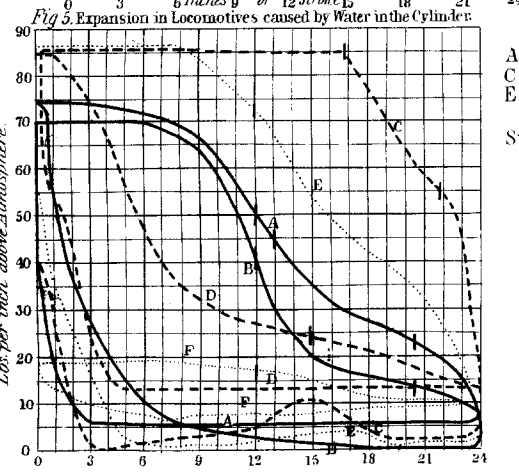
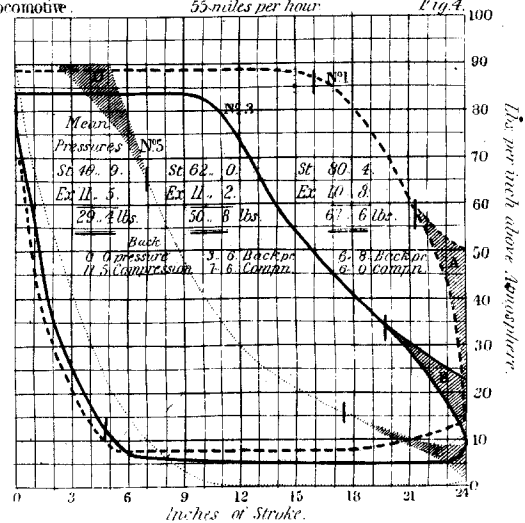
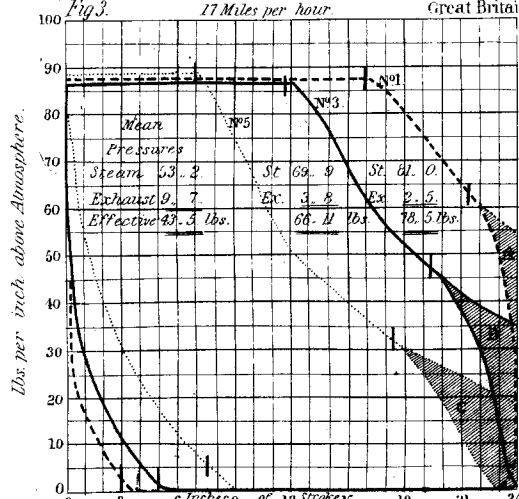
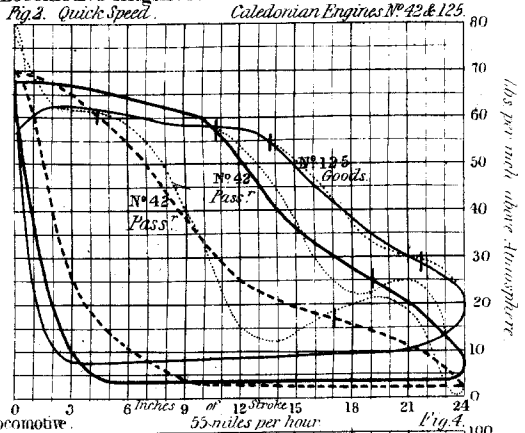
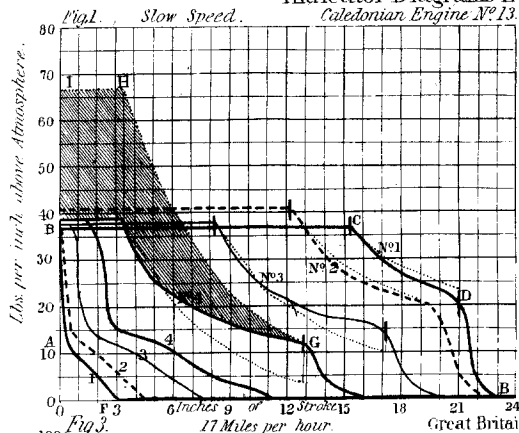
Allowing 8 lbs. of water to be evaporated by 1 lb. of coke, we have the following rule for the consumption of coke:

RULE II.—*To find the consumption of Coke per horse power per hour, for a given period of admission.* Multiply the period of admission in inches by 2.75, and divide by the length of stroke in inches, and add 1.75 to the quotient. The sum is the consumption in pounds per horse power per hour. Making C the consumption of coke, we have

$$C = 2.75 \frac{S}{L} + 1.75 \quad (2.)$$

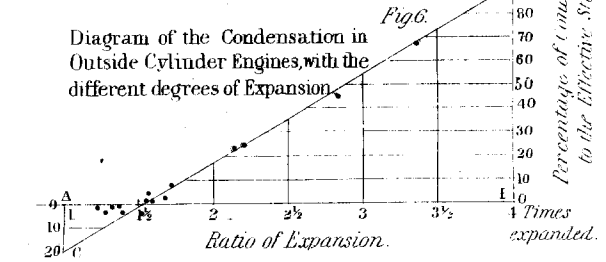
These rules may be employed with safety for all periods of admission between 10 and 75 per cent. of the stroke, which are the utmost limits worth regarding in the locomotive engine. They are applicable also for maximum pressures during admission, ranging between 60 lbs. and 120 lbs., though based on results from steam of 80 lbs. to 84 lbs. maximum pressure. For extreme pressures, the results by the rule are slightly too small in the case of lower pressures, and rather greater for the higher; these divergences being due to the constant deduction of 15 lbs. for atmospheric resistance from the total pressure. It is presumed that engineers will not return to the error of low pressures in locomotives, and that high pressures will be cultivated. For pressures above 80 lbs., the rules are perfectly safe, as they err rather by excess on the safe side. The following table is worked out by Rule I., to show the efficiency of steam by expansion in the locomotive cylinder, under good conditions in actual practice. The 4th column contains the theoretical maximum relative efficiency of steam, expanding to the end of the stroke, according to the law of Boyle, with a perfect vacuum behind the piston, and without clearance, back pressure, or compression; extracted from the ordinary tables on the

Indicator Diagrams from Locomotive Engines.



A & B Orion Pass' Engine 39 Miles per hour.
 C & D N^o 127 Goods Eng. 18 " "
 E & F D^o D^o 15 " "

Stroke of N^o 13. 20 Inches
 D^o N^o 42. 20 "
 D^o N^o 124 24 "



subject. In column 5 are given the relative amounts of work done by steam, under the admissions named in column 1, being directly as the effective mean pressures in the cylinder, which are found by a rule to be afterwards given.

Efficiency of Steam by Expansion in the Cylinder of the Locomotive in Actual Practice.

For Maximum Pressures during admission of 60 lbs. to 120 lbs.

Periods of admission or points at which the steam is cut off in parts of stroke.	Water as steam consumed in pounds per horse power per hour.	Relative efficiency of steam in actual practice.	Possible maximum efficiency.	Relative work done by steam of the same maximum pressure in the cylinder.
per cent.	lb.			
10	16.2	2.22	3.30	15
12.5	16.7	2.15	3.08	20
15	17.3	2.08	2.90	24
17.5	17.8	2.02	2.73	28
20	18.4	1.96	2.60	32
25	19.5	1.85	2.39	40
30	20.6	1.75	2.20	46
35	21.7	1.66	2.05	52
40	22.8	1.58	1.92	57
45	23.9	1.50	1.80	62
50	25.0	1.44	1.69	67
55	26.1	1.38	1.60	72
60	27.2	1.32	1.51	77
65	28.3	1.27	1.43	81
70	29.4	1.23	1.35	85
75	30.5	1.18	1.28	89
100	36.0	1.00	1.00	100

The periods of admission of steam to the cylinder may be varied by link-motion from 75, the greatest useful period, to 10 per cent. of the stroke. By the table, the relative efficiency varies within these limits from 1.18 to 2.22, the variation being as 1 to 2 nearly. It follows, that under the most favorable existing circumstances, *the utmost possible efficiency of steam worked expansively* in the locomotive by the link-motion, is *about twice* that of the steam when worked *under full gear*; that is, the same quantity of steam does twice the quantity of work.

By a consideration of the effective mean pressures in the first table, it appears that the average rate at which it increases with the period of admission is expressed by the following rule:

RULE III.—*To find the effective Mean Pressure in the Cylinder, in terms of the Maximum Pressure, for a given per centage of admission.* Multiply the square root of the per centage of admission by 13.5, and subtract 28 from the product. The remainder is the effective mean pressure in per cent. of the maximum pressure of steam admitted. By this rule the following table is composed:

Effective Mean Pressure in the Cylinder for Various Admissions.

For Maximum Pressures of 60 lbs. to 150 lbs.

Periods of admission in parts of the stroke.	Effective mean pressures in parts of maximum pressure.	Periods of admission in common fractions of stroke.	Effective mean pressure in common fractions of the maximum pressure.
per cent.	per cent.		
10	15	1-10th.	1-7th full.
12.5	20
15	24	1-8th.	1-5th.
17.5	28
20	32	1-6th.	1-4th.
25	40	1-5th.	1-3d.
30	46	1-4th.	1-25th.
35	52	1-3d.	1-2d.
40	57
45	62
50	67	1-2d.	2-3ds.
55	72
60	77
65	81	2-3ds.	4-5ths.
70	85
75	89	3-4ths.	9-10ths.

In all well protected cylinders, with blast orifices not less than $\frac{1}{8}$ th of the area of the cylinder, the foregoing rules and tables of data apply to the action of steam at speeds under 30 to 40 miles an hour, as the writer has fully shown in his work on *Railway Machinery*. For speeds amounting to 55 to 60 miles an hour, the loss by imperfect exhaust causes a large increase of consumption per horse power per hour, of from 33 to 12 per cent., according to the amount of admission. With steam ports of about $\frac{1}{4}$ th, and blast orifices $\frac{1}{4}$ th of the cylinder, the rules likewise apply, at speeds under 30 to 40 miles an hour. At the higher speeds, the useful power is considerably impaired by imperfect exhaust.

The proportions of the *Great Britain*, from the performance of which the foregoing results are deduced, may be repeated here as standard ratios for practice, until superior results are obtained.

Sectional area of cylinder,	1
“ “ steam port,	1-10th
“ “ blast orifice,	1-11th
Lap of valve, $1\frac{1}{4}$ in.; travel, $4\frac{1}{2}$ in. in full gear; lead, $\frac{1}{4}$ to $\frac{3}{8}$ in.	

In a second paper, the writer discusses the *conditions* necessary for the successful expansive working of steam in locomotives. The following is a comparison of the actual results of engines working with ordinary *gab-motions* and with *link-motions*. The engine *Europe*, on the Edinburgh and Glasgow Railway, cylinder 16×18 inches, wheel 6 feet, doing one week's work in 1849, with *gab-motion*, consumed an average of 19 cwt. of coke per day, and 2 cwt. of coal. As, in the locomotive boiler, coal is about two-thirds of the value of coke, 2 cwt. of coal is equivalent to 1.33 cwt. of coke; and the consumption per day may be stated at 20.33 cwt. coke.

The same engine, fitted with *link-motion*, used at the same season in 1851, and doing the same work, 12 cwt. of coke, and 3 cwt. of coal daily, equivalent to 14 cwt. coke. Over a run of 94 miles, the expenditure becomes

24.22 lbs. per mile with gab-motion.
16.70 " " link-motion.

7.50 " reduction, or 30 per cent. with link.

The periods of admission in the two cases would be about 70 and 45 per cent., and by the table of efficiency the consumption would be as 1.50 to 1.23, showing an economy of only 18 per cent., or barely two-thirds of what was actually made. The greater actual efficiency must in great part be due to the superior opportunity of working with high pressure, during the admissions offered by the link.

Again, the test may be applied by measuring the water consumed. The following are a selection of cases from the writer's own experience and observation :

Engine with Link-motion; cylinder, 15×20 inches; wheel, 6 feet. Edinburgh and Glasgow Railway.

Date.	Engine.	Mean speed. Miles per hour.	Average train of Carriages.	Consump- tion of water in feet, per mile.	Remarks.
1851 August 26.	"Orion," ordi- nary train.	19.6	16	2.97	Stiff wind ahead.
" 26.	Do. do.	24.4	7	2.01	Do. favorable.
August 27.	Do. do.	24.4	7	2.22	Do. ahead.
" 27.	Do. Express.	32.0	5	1.65	Do. favorable.
1850. Sept. 7.	Do. do.	32.7	5	1.65	Slight wind ahead.

Engines with fixed Gab-motion, cylinder 16 × 18 inches, wheel 6 feet. Edinburgh and Glasgow Railway.

1850. Sept. 3.	"America," ordi- nary.	21.5	13	3.01	Wind favorable.
October 10.	"Nile," Expr.	29.0	7½	3.00	Do. ahead
" 21.	"Niger,"	7	2.80	Calm.

Express engine, with fixed Gab-motion; cylinder, 16 × 18 inches; wheel, 6 feet. North British Railway.

1851.	Express.	38.5	5	2.70	Calm.
	"	38.5	5	2.70	Do.
	"	38.5	4	2.96	Wind ahead.
	Mail.	35.7	7	3.05	Calm.
	Ordinary.	22.0	12	3.45	Calm.

These results show, as before, that under similar circumstances, what has been deduced from an independent examination of indicator diagrams, taken under the link-motion, as to the *economy* of steam *worked expansively*, is fully borne out by a direct appeal to the relative consumption of coke and water.

(To be continued.)

For the Journal of the Franklin Institute.

Description of Lahaye's Patent Self-Acting Brake.

EXPLANATION OF THE FIGURES.

<i>a</i> Timbers of car body.	<i>k</i> Hangers suspending brake sockets from truck.
<i>b</i> " of trucks.	<i>l</i> Brake blocks, movable in their sockets.
<i>c</i> Bumper of pulling bar.	<i>m</i> Sockets, channeled to receive brake blocks.
<i>d</i> Pulling bar coupled to engine or train.	<i>n</i> Lifting bars, moving with " "
<i>e</i> Shaft communicating power of pulling bar, <i>d</i> , when train is checked to brake blocks.	<i>o</i> Bell crank levers, moving detaching rods <i>q</i> .
<i>f</i> Fixed pivot of shaft, <i>e</i> .	<i>p</i> " " " "
<i>g</i> Pulling rods, acting upon brake blocks from shaft, <i>e</i> .	<i>q</i> Detaching rods, acting upon cranks, <i>r</i> .
<i>h</i> Springs acting upon all brake blocks by <i>g</i> and <i>i</i> .	<i>r</i> Cranks, moving detaching shaft.
<i>i</i> Tie rods connecting brake blocks.	<i>s</i> Tongue, on detaching shaft, moving pulling bar, <i>d</i> , out of gear.
	<i>t</i> Rollers, between brake block and sockets.
	<i>u</i> Slets, for motion of rollers.
	<i>v</i> Pulling bar spring.

Operation of Brake, Fig. 1.—The car is supposed to be coupled to an engine, by the pulling bar *d* at *c*. Immediately on the steam being shut off, the momentum of the car and train behind it presses the pulling bar upon the engine at *c*; this pressure is communicated by a shoulder in *d*, through the shaft *e*, pulling rods *g*, springs *h*, and tie rods *i*, to all the brake blocks, causing them to act upon all the wheels with a force proportioned to the resistance at *c*, and ceasing only with the latter resistance.

Operation of Detaching Levers, Fig. 2.—The brake blocks are supposed to be pressed against the wheels, the engine reversed, the train stopped, and required to be backed. The wheel *B*, fig. 2, turning in the direction of the arrow, moves the brake block, *l*, upwards in the same direction, carrying with it the lifting bars, *n*, acting upon the bell cranks, *o p*, detaching rods, *q*, and crank, *r*, thus moving the tongue, *s*, upwards, and unengaging the pushing bar, *d*, allowing the car to be moved backwards by the bar, *d*, pressing against the ring bolt by a slot in *d*, through which said ring bolt passes.

General Remarks.—The detaching motion is instantaneous in its action. The ordinary hand brake can be applied to every car with the self-acting brake, and acts independently of the latter. When it is not required to back the train, the pressure of the brake blocks is relieved immediately by the forward motion of the pulling bar, *d*.

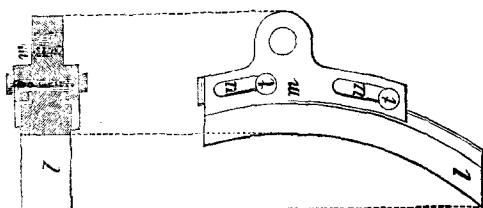
The chief advantages of this brake are as follows:

1st, It is self-acting, operating only when it is desired to stop the train or diminish its speed, and relieving the wheels of its pressure instantly, on the cars feeling the forward or backward motion of the engine.

2d, Its action is instantaneous on every car in the train, commencing the moment the steam is shut off the engine, and ceasing only with the motion of the cars.

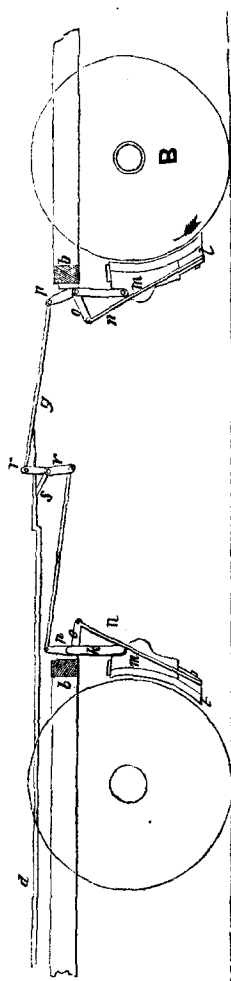
3d, Its power is exactly proportioned to the necessity for the gradual or sudden stoppage of the train, resulting from the judgment of the engineer in the management of his engine; the brakes acting with moderate force when the steam is shut off the engine, with increased intensity if

Fig. 3.



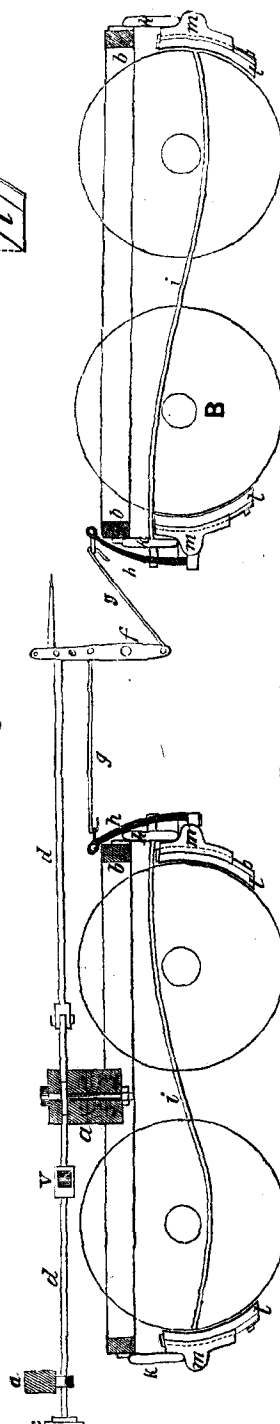
Brake Blocks.

Fig. 2.



Detaching Levers.

Fig. 1.



Braking Apparatus.

the tender brake be applied, and with sufficient force to back the wheels (if required) if the engine be reversed.

4th, It is simple in its construction and operation, not liable to derangement, and so constructed as to allow the use of the ordinary hand brake on every car, when the latter is moved separately.

CERTIFICATES.

Lahaye's Patent Brake has been in operation upon the passenger trains of the Philadelphia and Reading Railroad for several months past. It has worked to our satisfaction, has caused no expense for repairs, remaining in the same condition as when first put on, and proving on many occasions eminently useful in arresting the rapid motion of the train, when danger was observed and the engine reversed.

I consider it one of the most valuable inventions ever made to increase the safety of railroad traveling, as by its use a passenger train can be stopped in about half the space usually required.

G. A. NICOLLS,

Eng. & Sup. Philad. & Reading Railroad.

Reading, Pa., March 16th, 1852.

Mr. John Lahaye's Self-Acting Brake has been in operation on the passenger cars of the Philadelphia and Reading Railroad for several months, and its operation has been such as to give entire satisfaction. I think a fair trial of it by any railroad company will convince them that it is the only contrivance of the brake kind that can be relied on when there is a necessity for stopping a train in case of collisions.

I would recommend it particularly to railroads using but a single track; for them I think it indispensable.

JAMES MULHOLLAND,

Master Machinist, Philad. & Reading Railroad.

Reading, Pa., March 16th, 1852.

*Description of Pilbrow's Water Waste Preventer.**

This article, manufactured by Guest & Chrimes, of Rotherham, is introduced for the purpose of detecting and preventing waste of water supplied to the inhabitants of towns by water works, whether such waste be wilful (as is too often the case amongst the inconsiderate occupiers of cottage property), negligent, or accidental from leakage or bursting of pipes, leaving open taps, &c., and is especially applicable, and may be said to be almost indispensable, where water is supplied on the "high-pressure" and "constant-supply" system. When it is considered that under this system, at a pressure of, say 150 feet, the smallest tap in ordinary use—viz. $\frac{3}{8}$ -inch—will, if left open for only one night of ten hours, waste from four to five thousand gallons of water, it will be most manifest that any article which will prevent a waste of such serious magnitude is one that must fairly claim immediate attention, and insure extensive adoption.

The Water Waste Preventer is calculated and guaranteed not only to remove this difficulty, but also to obviate the necessity of stop-cocks, as these will be no longer required for the purpose of shutting off water whilst repairs are being made to taps and pipes in consequence of leakage, the removal, or repairs of broken taps or service pipes, bursting of pipes from frost, or any other ordinary cause.

* From the London Civil Engineer and Architect's Journal, May, 1852.