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“The Efficiency of the Shone System of Pumping.”

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THIS Paper was prepared with the object of adding to the data at present somewhat scarce¹ concerning the actual working of the Shone system of pumping, and embodies the results of careful tests carried out by the Author on the sewage-pumping plant of the Gosport and Alverstoke Urban District Council.

The district referred to has an area of 3,843 acres and a population of 30,000 persons. The difficulties of drainage were considerably increased by the flat conformation of the ground (the highest point drained being only 20 feet above, and the lowest point 2 feet below high-water level of spring-tides) and by the intersection of the town by creeks left practically dry at low water.

It was eventually decided to divide the district into areas and to adopt the Shone system of pumping by compressed air supplied from one central station; the scheme provides for the drainage of 2,200 acres and for an estimated population of 40,000 persons, storm-water being excluded as far as possible. As shown in *Fig. 1*, eighteen ejector-stations are provided, each station having ejectors in duplicate, varying in capacity between 50 gallons and 800 gallons. The total length of the sewers is approximately 30·3 miles, the rising-mains 5·5 miles, and the air-mains 11 miles. The rising-mains vary between 5 inches and 15 inches in diameter and the air-mains between 2½ inches and 12 inches. The compressor-plant consists of two twin sets of direct-acting compressors, each set of 160 I.H.P. and comprising two condensing steam-jacketed and covered steam-cylinders and two water-jacketed and covered double-acting air-cylinders with coupled fly-wheels, the safe maximum speed being 200 revolutions per minute. The inlet and delivery of the air-cylinder are governed by Corliss mechanically-controlled valves. Steam is supplied from two Galloway boilers, each 7 feet 6 inches in diameter and 28 feet in length, one boiler being connected to a two-cell back-feed Horsfall destructor

¹ See J. F. Brunton, “Notes on the Working of the Shone System of Sewerage at Karachi.” Minutes of Proceedings Inst. C.E., vol. clx, p. 211.

consuming 10 tons per cell per 24 hours. The sewage discharges from a cast-iron outfall-pipe 36 inches in diameter into the Solent, 600 feet beyond high-water mark, after passing through three

Fig. 1.

PLAN SHOWING EJECTORS, RISING-MAINS, AND MAIN SEWERS.

covered septic-tanks having a total capacity of 1,200,000 gallons. Mr. R. St. G. Moore, M. Inst. C.E., was the engineer for the scheme, and the total cost of the works amounted to £110,000.

Although the engines were started in June 1904, the district, at

the date of the tests (May 1906), was not wholly connected up, the average weekly flow being only about 3,000,000 gallons.

The air-mains were laid with every possible care and were tested in sections with a pressure of 40 lbs. per square inch, but there must always be some leakage of air in its transmission to the ejectors. This loss, together with the loss by leakage in the automatic, alternating, and stop-valves at the ejectors, was ascertained in the following manner:—On Sunday morning 29th April, 1906, the inlet- and delivery-valves at all the ejector-stations were shut down, preventing the ejectors from discharging, and at 8 a.m. the compressors were stopped. After allowing a few minutes for the air in the receiver to cool, the air-pressure was read from the gauge and recorded. At the expiration of 1 hour, the pressure then having fallen 5 lbs. per square inch, the engines were restarted and it was found that 362 revolutions of one set of engines were required to make good this loss. After pumping away the sewage accumulated during this test, all the stop-valves on the air-mains at the ejectors were shut down and the engines were again stopped; the fall of pressure during 1 hour was 2·5 lbs. per square inch and this loss was made good by 185 revolutions of one set of engines. In each case deducting, from the number of revolutions required to replace the loss of pressure, 2 revolutions assumed to be required to make good the loss in opening the air stop-valves on restarting the engines, and taking the average speed during all the tests to have been 48 revolutions per minute (Table I, Appendix), or 2,880 revolutions per hour, the revolutions required to make good constant loss by leakage amount to:—

$$\text{First test} \quad \frac{360 \times 100}{2,880} = 12\cdot5 \text{ per cent.}$$

$$\text{Second test} \quad \frac{183 \times 100}{2,880} = 6\cdot35 \text{ per cent.,}$$

the latter being the loss by leakage in the air-mains only and the former the loss in the air-mains and ejector-valves together.

By systematic search along the line of the air-mains in wet weather leaks may be discovered by the bubbles on the surface; and with careful supervision the loss from this cause should be reduced to an inconsiderable amount. The loss in the ejector-valves is more difficult to remedy, the free working of the valves, which are of the piston type, necessitating a slight leakage which is practically unavoidable.

The following particulars relating to the plant are common to all the trials :—

Diameter of steam-cylinders	12·50 inches.
" " air-cylinders	16·00 "
Length of stroke	18·00 "
Cut-off at	0·25 of stroke.
Effective area of steam-cylinders (front)	118·70 square inches.
" " " " (back)	119·60 " "
" " air- " (front)	198·00 " "
" " " " (back)	201·00 " "
" " " " (mean)	199·55 " "
Mean capacity of one air-cylinder	2·08 cubic feet.
Gauge-pressure of air on leaving compressors	22·50 lbs. per sq. in.

During the tests ejector-station No. 3 was completely shut down, there being no sewage flow, as Fort Elson, which discharges into the ejector, was unoccupied. This, however, in no way affects the results. In order to reduce the number of counters required, only one ejector at each station was in use, the other being shut off from the sewage inlet-pipe. At thirteen stations automatic counters were fixed to the levers actuating the air-valves, men being stationed at the remainder to record accurately the number of discharges during the tests. The 2nd, 3rd, and 16th May were the days allocated for the work, readings being taken continuously from 9 a.m. till 3 p.m. Owing to the breakdown of one of the automatic ejector counters the tests on the 2nd May proved abortive, and have not been included in this Paper. Readings of the steam- and air-pressures and temperatures were taken at the air-compressing station at intervals of 15 minutes, and indicator-diagrams were taken from the air- and steam-cylinders every 30 minutes. The air-supply temperature was taken in the fresh-air shaft and that of the compressed air by thermometers inserted in the air-chamber over the cylinders. These readings, together with the other readings taken, are given in Table I. The maximum variation of the air pressure-gauge reading was only 2 lbs. per square inch.

Particulars relating to the ejectors are given in Table II (Appendix), the figures given in column 6 being based upon a series of exhaustive tests made within a few days of the trials. These tests may be summarized as follows:—The air-pressure at the compressor-station was kept constant at 22·5 lbs. per square inch whilst a U tube with a pressure-gauge and thermometer attached was connected to the interior of each ejector in turn, and numerous readings were taken while the ejectors were discharging. The thermometer showed an average temperature in the ejector-bodies of 64° F. The average indicated pressure in the ejectors is shown in column 6 of Table II, and is converted into its equivalent dynamic head in column 7

(Table II). The work done at the ejectors is based upon this head and upon the quantity of sewage discharged.

Figs. 2.

INDICATOR-DIAGRAMS FROM AIR-COMPRESSOR.

The efficiency of the mechanism of the compressing-plant is shown in Table III (Appendix), and specimen indicator-diagrams are given in Figs. 2.

The actual quantity of sewage discharged at each ejector during the tests, based upon the capacity of the ejectors and the number of discharges recorded, is shown in Tables IV and V (Appendix). The average head obtained by dividing the sum of the products of head and discharge by the total quantity discharged, gives the average pressure per gallon in the ejectors. From the figures at the foot of Tables IV and V, it will be seen that the horse-power developed at the ejectors was 11·34 HP. on the 3rd May, and 11·94 HP. on the 16th May. The actual efficiency of the system from steam-cylinders to ejectors was therefore

$$\frac{11\cdot34}{29\cdot50} = 0\cdot384 \text{ on the 3rd May,}$$

and
$$\frac{11\cdot94}{30\cdot96} = 0\cdot385 \text{ on the 16th May.}$$

This loss of efficiency arises from many causes, the chief of which are, what may be termed the unavoidable losses due to the heating of the air in compression, and the practical impossibility of using the air at the ejectors expansively, together with the loss by leakage in the air-mains and ejector-valves.

In the formulas relating to the compression of the air at the compressors and its use at the ejectors, the following symbols are used :—

P_a = atmospheric pressure in lbs. per square foot.

p_a = " " " " " " inch.

p_1 = absolute pressure of compressed air in lbs. per square inch.

P_1 = " " " " " " " " foot.

v_a = volume of 1 lb. of air at pressure P_a and temperature T .

v_1 = " " " " " " P_1 " " "

T = absolute temperature, Fahrenheit scale.

c = 53·15, a constant.

γ = 1·41.

From the laws relating to gases

$$P_a v_a = c T.$$

In all practical methods of compression the air is heated, and as this heat is lost in transmission through the air-mains by conduction into the surrounding medium it is advisable that the cooling-apparatus should be as efficient as possible. In perfect adiabatic compression the final temperature of the compressed air would be

$$T_1 = T \left(\frac{p_1}{p_a} \right)^{0\cdot71}$$

which would give 234° F. for the 3rd May and 241° F. for the

16th May, whereas the actual temperatures on these dates were 165° F. and 167·2° F. respectively. The amount of work required to heat the air to these temperatures is the excess of work beyond that required to compress the air isothermally, and is the index of the efficiency of the compression-process.

The work done in isothermal compression is :—

$$P_a v_a \cdot \log^e \frac{p_1}{p_a} \text{ foot-lbs. per pound of air.}$$

Based upon the volume of 1 lb. of air at the temperatures in the air-supply and upon the mean capacity of the air-cylinder on 3rd May and 16th May the compression of 1 lb. of free air required 3·204 revolutions and 3·25 revolutions respectively of one set of engines.

The actual work done in compressing 1 lb. of free air on each day of the trials, the theoretical work required for isothermal compression of 1 lb. of air, the efficiency of the compressed air, and the total efficiency of the compression-plant, are shown in Table VI (Appendix).

The average dynamic head at the ejectors was 38·30 feet (Tables IV and V), equivalent to a gauge-pressure of 16·46 lbs. per square inch or 31·16 lbs. per square inch absolute pressure, and the average temperature of the air at the ejectors was 64° F. One pound of air at 64° F. and 31·16 lbs. per square inch absolute pressure has a volume of 6·22 cubic feet, and the possible amount of work it can do in an ejector without expansion is :—

$$\begin{aligned} V_2 \times 144 \times 16\cdot46 \text{ foot-lbs.} \\ = 6\cdot22 \times 144 \times 16\cdot46 = 14,743 \text{ foot-lbs.} \end{aligned}$$

The same amount of air working in adiabatic expansion from a gauge-pressure of 16·46 lbs. per square inch to atmospheric pressure will perform

$$\begin{aligned} \frac{\gamma}{\gamma - 1} \cdot P_1 v_1 \left[1 - \left(\frac{p_a}{p_1} \right)^{\frac{\gamma - 1}{\gamma}} \right] &= 3\cdot439 \times 27,903 \left[1 - \left(\frac{p_a}{p_1} \right)^{0\cdot29} \right] \\ &= 18,786 \text{ foot-lbs. of work.} \end{aligned}$$

Neglecting slip and clearance, the greatest possible theoretical efficiency of the air working non-expansively at the ejectors is

$$C_6 = \frac{14,743}{18,786} = 0\cdot784 \text{ (Table VII, Appendix).}$$

One pound of air at 70° F. and 22·5 lbs. per square inch gauge-pressure has a volume of 5·267 cubic feet.

One pound of air at 76° F. and 22·5 lbs. gauge-pressure has a volume of 5·33 cubic feet.

Taking 1 lb. of air at the gauge-pressure at which it leaves the compressors but at atmospheric temperature, the amount of work obtained non-expansively would be:—

$$5\cdot267 \times 22\cdot5 \times 144 = 17,067 \text{ foot-lbs. on the 3rd May,}$$

$$\text{and } 5\cdot33 \times 22\cdot5 \times 144 = 17,269 \text{ foot-lbs. on the 16th May.}$$

Comparing this with the amount of work obtained from 1 lb. of air working non-expansively at the ejectors, it will be seen that the loss of efficiency of the air as a working-medium during its transmission from the compressors to the ejectors was:—

$$\left. \begin{aligned} C_5 &= \frac{14,743}{17,067} = 0\cdot863, \text{ on 3rd May} \\ \text{and } C_5 &= \frac{14,743}{17,269} = 0\cdot853, \text{ on 16th May} \end{aligned} \right\} \text{(Table VII.)}$$

On the 3rd May the air supplied to the compressors, computed from the capacity of the air-cylinders and the number of revolutions, was 141,739·5 cubic feet, which at 70° F. and atmospheric pressure is equivalent to 10,633 lbs. On the 16th May it was 148,503 cubic feet, or 10,984 lbs. The air used at the ejectors, measured by the quantity of sewage displaced, was 56,208 cubic feet on the 3rd May, and 59,128 cubic feet on the 16th May, which at 64° F. and 31·16 lbs. per square inch absolute pressure is equivalent to 9,036 lbs. and 9,506 lbs. respectively.

The ratio of supply to discharge was therefore

$$\frac{9,036}{10,633} = 0\cdot85, \text{ on the 3rd May,}$$

$$\text{and } \frac{9,506}{10,984} = 0\cdot86, \text{ on the 16th May.}$$

These figures corroborate the results obtained on the 29th April—viz., a loss of 12·5 per cent. in the air-mains and ejector-valves, and it has been assumed that the coefficient of efficiency from this cause is $C_4 = 0\cdot875$ (Table VII).

A summary of the different coefficients obtained is given in Table VII, and it will be seen that there is no very great difference between the actual efficiency of the system and the computed efficiency. In comparison with the efficiency of other systems of pumping it must not be forgotten that where applied to a suitable

district the Shone system overcomes many difficulties which in other schemes would probably add materially to the cost.

In conclusion, the Author has not attempted to give any figures relating to the cost of pumping; the permanent automatic counters only show the actual amount pumped and not the number of gallons lifted at each station, and any estimate not based on the work done would be misleading and useless.

The Paper is accompanied by diagrams from which the Figures in the text have been prepared; and by the following Appendix.

APPENDIX.

TABLE I.—AVERAGE READINGS AT THE ENGINE-HOUSE DURING THE TRIALS;
9 a.m. to 3 p.m. on 3 May and 16 May, 1906.

	3 May.	16 May.
Revolutions per minute	47·30	49·58
Air-pressure in lbs. per square inch above atmosphere on leaving compressors	22·50	22·50
Steam gauge-pressure in lbs. per square inch	64·60	72·40
Vacuum in inches	25·50	24·50
Temperature of air in air-supply duct °F.	70	76
„ „ leaving compressors	165	167·2
„ of cooling water	88	85·8
„ „ hot-well	117	122·1
„ „ air at ejectors	64	64

TABLE II.—DETAILS OF EJECTORS, ETC.

1	2	3	4	5	6	7
Ejector Number.	Capacity of Ejector.	Diameter of Rising- Main.	Time taken to Discharge.	Initial Velocity of Sewage in Rising-Main.	Average Gauge- Pressure of Air in Ejector- Bodies.	Dynamic Head Equivalent to Gauge- Pressure.
	Gallons.	Inches.	Seconds.	Feet per Second.	Lbs. per Square Inch.	Feet.
1	100	6	32	2·54	19·00	44·19
2	250	10	25	2·93	17·00	39·53
3	50	5	..	..	..	..
4	400	10	25	4·70	17·00	39·53
5	50	5	20	2·93	17·50	40·07
6	150	7	23	3·91	16·30	37·90
7	250	8	23	4·89	18·50	43·02
8	250	9	25	3·62	18·60	43·25
9	800	15	27	3·80	17·00	39·53
10	200	8	34	2·69	19·00	44·19
11	250	9	25	3·62	17·30	40·23
12	700	15	25	3·72	8·00	18·60
13	200	8	26	3·52	16·70	38·83
14	50	5	25	2·34	14·50	33·72
15	500	12	25	4·07	17·60	40·93
16	50	6	21	4·76	16·30	37·90
17	100	7	14	4·28	15·30	35·58
18	50	5	18	3·25	14·60	33·95

TABLE III.—EFFICIENCY OF MECHANISM OF COMPRESSOR.

Date.	Steam.		Air.		Efficiency. C ₁
	Average Pressure by Indicator-Diagram.	HP.	Average Pressure by Indicator-Diagram.	HP.	
3 May . .	Lbs. per Square Inch. 28·80	29·50	Lbs. per Square Inch. 15·30	26·25	0·890
16 May . .	28·80	30·96	15·00	26·97	0·871

TABLE IV.—TEST OF 3 MAY, 1906, 9 a.m. TO 3 p.m.

Ejector Number.	Discharge during Test.	Dynamic Head (Table II, Col. 7).	Head × Discharge.
	Gallons.	Feet.	
1	7,800	44·19	344,682
2	13,500	39·53	533,655
3	..	..	..
4	17,200	39·53	679,916
5	5,550	40·07	222,389
6	16,650	37·90	631,035
7	30,250	43·02	1,301,350
8	50,000	43·25	2,162,500
9	80,000	39·53	3,162,400
10	12,600	44·19	556,794
11	10,250	40·23	412,357
12	35,000	18·60	651,000
13	18,600	38·83	722,238
14	6,800	33·72	229,296
15	34,500	40·93	1,412,090
16	3,250	37·90	123,175
17	4,000	35·58	143,175
18	5,350	33·95	181,632
	351,300	..	13,469,869

$$\text{Average head} = \frac{13,469,869}{351,300} = 38·34 \text{ feet.}$$

38·34 feet is equivalent to an average pressure of 16·43 lbs. per square inch.

$$\text{HP. developed at ejectors} = \frac{13,469,869 \times 10}{33,000 \times 6 \times 60} = 11·34 \text{ HP.}$$

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TABLE V.—TESTS OF 16 MAY, 1906. 9 a.m. to 3 p.m.

Ejector Number.	Discharge during Test.	Dynamic Head. Table II, Col. 7.	Discharge $\times$ Head.
1	Gallons. 6,500	Feet. 44.19	287,235
2	11,000	39.53	434,830
3	..	..	..
4	12,800	39.53	505,984
5	4,350	40.07	174,304
6	13,150	37.90	687,885
7	29,000	43.02	1,247,580
8	53,250	43.25	2,303,060
9	84,000	39.53	3,320,520
10	12,600	44.19	556,794
11	9,750	40.23	392,242
12	35,000	18.60	651,000
13	34,000	33.83	1,320,220
14	2,050	33.72	69,126
15	37,500	40.93	1,534,880
16	8,050	37.90	305,095
17	6,800	35.58	243,712
18	4,750	33.95	161,262
	369,550		14,195,729

$$\text{Average head} = \frac{14,195,729}{369,550} = 38.25 \text{ feet.}$$

38.25 feet is equivalent to an average pressure of 16.44 lbs. per square inch.

$$\text{HP. developed at ejectors} = \frac{14,195,729 \times 10}{33,000 \times 6 \times 60} = 11.94 \text{ HP.}$$