CORRESPONDENCE

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"Combustion-Efficiencies of Gas and Oil Engines." †

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Captain R. W. A. Brewer observed that the Paper was practically confined to a consideration of engines of the four-stroke-cycle type, and that the statement was made that no advance had been made in combustion-efficiency since the Reports of Professors Bertram Hopkinson and F. W. Burstall in 1908 ; *Fig. 6* (p. 181 §) was given as a proof that that statement was correct. The improvement in two-stroke-cycle engines had, however, been quite marked, and his Paper read before the Society of Engineers,* and discussed by Mr. Tookey, gave data on some developments in the United States under his direction. His results fell in the general group of points shown in *Fig. 6* for an engine-speed of 1,200 revolutions per minute, showing that the modern two-stroke-cycle commercial type of high-speed gas engine was comparable with the four-stroke engines referred to by the Author. Further information would be found in Captain Brewer's Paper.* In single-cylinder two-stroke test engines, better results had been attained, and indicator mean pressures of between 100 and 108 lb. per square inch had been recorded for mixture-strengths of between 35 and 39 B.Th.U. per cubic foot of total cylinder volume. An eight-cylinder two-stroke radial power-unit of the type described in his Paper * naturally developed less specific power, and its general performance was not so good, but it was well worth further development. Since then other engines had been made to his designs with cylinders having a bore and stroke of 8 inches. Curves relating mean indicated pressure to mixture-strength per cubic foot of total cylinder volume showed that the results from four-stroke-cycle and from two-stroke-cycle engines were in general agreement.

† Journal Inst. C.E., vol. 7 (1937–38), p. 166 (December 1937).

§ Page numbers so marked refer to the Paper. (Footnote (†) above.)—Sec. Inst. C.E.

* "The Development of a Two-Stroke Cycle Gas Engine." Trans. Soc. E., 1934, p. 90.

In the Brewer engine, the cylinder was filled with fresh air at a pressure of approximately 2·5 lb. per square inch at the instant of port-closing, and gas was then injected under a pressure of up to 40 lb. per square inch. It was thought that a method of computation based on the actual volumes of gas and air supplied prior to combustion should have been adopted; the Author had pointed out, however, that his method was more satisfactory in showing a true comparison between various trials, and that had proved to be the case.

Mr. G. H. Paulin, referring to p. 167 §, offered a more satisfactory explanation of the exhaust-scavenging process. When the exhaust-valve opened there was a discharge of exhaust-products at comparatively high pressure into the exhaust-pipe. That caused a wave of pressure which travelled towards the silencer end of the pipe with the same speed as sound would travel in an atmosphere similar to that in the pipe. Due to the elasticity and inertia of the atmosphere in the exhaust-pipe, that initial wave of pressure was followed by a rarefaction and again by a pressure-wave, and so on. The waves of pressure and rarefaction travelled on, until at the silencer end of the pipe they were reflected successively and travelled back to the engine, where they were again reflected. Thus there was set up in the exhaust-pipe a series of waves of pressure and rarefaction which travelled back and forth. If the pipe were made of such a length that a wave of rarefaction arrived at the exhaust-valve when it was on the point of closing, the pressure, and thus the weight of the residual gas in the clearance space, would be a minimum. It would be seen that there was, however, a possibility of there being a pressure-wave at the exhaust-valve at the instant of its closure, depending on the length of the exhaust-pipe and on the speed of the engine, so that for most efficient scavenging it would be necessary to "tune" the exhaust-pipe length to the engine-speed, and not just to make it 12 or more feet long.

The values of P_m in Table III (p. 171 §) for throttle-governed engines did not altogether agree with *Fig. 1* or Table I (p. 169 §). On bringing them into agreement, as shown in Table XVI, more grounds were found

TABLE XVI.

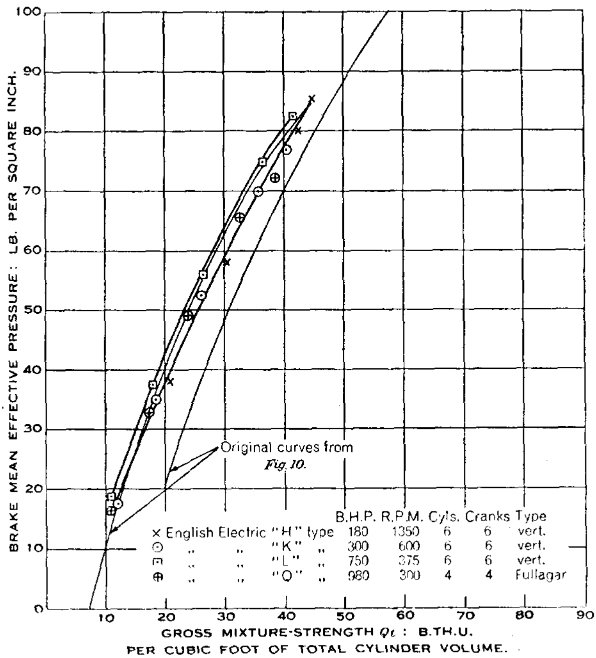
Q_t: B.Th.U. per cubic foot.	Indicator mean pressure, P_m: lb. per square inch.	Difference-factor.	Hit-and-miss governing difference-factor.
20–25	42·5–52·5	2	—
25–30	52·5–62·5	2	—
30–35	62·5–70·5	1·6	—
35–40	70·5–78	1·5	1·6
40–45	78 –85	1·4	1·5
45–50	85 –89	0·8	1·4

§ *Ibid.*

for the remark in the first paragraph on p. 171 § to the effect that the hit-and-miss governed engine was more efficient in converting heat into work than the throttle-governed engine over the range of mixture-strengths from 35 B.Th.U. per cubic foot to 50 B.Th.U. per cubic foot.

To prove that the high-speed engine need not be at so great a disadvantage as *Fig. 10* (p. 185 §) showed, the curves in *Fig. 10* had been replotted in *Fig. 25*, and the particulars of a number of English Electric engines had been added. All those engines were of the direct-injection type. Table XI (p. 186 §) had been extended, as shown in Table XVII

Fig. 25.



(p. 404), over the portion common to the industrial and high-speed types of engines, and to include the Tookey factors.

The values of n–1 in Table XIV (p. 189 §) which were based on *Fig. 11* (p. 187 §), were incorrect, because n–1 could only be based on indicated pressures, and not on brake pressures; further, it was thought that the value of n–1 could not be evaluated from the "difference of T_m per B.Th.U. per cubic foot of swept and clearance volumes." It could be calculated from the T_m-value, but surely it could not be obtained from

§ *Ibid.*

TABLE XVII.

Q_t : B.Th.U. per cubic foot.	P_n : lb. per square inch (industrial type).	P_n : lb. per square inch (high-speed type).	T_n (industrial type).	T_n (high-speed type—Paxman-Ricardo).
20	39	23	1·95	1·15
25	51	36	2·04	1·44
30	62	48	2·07	1·6
35	72	60	2·06	1·71
40	82	70	2·05	1·75

the $\triangle T_m$-values, where $\triangle T_m$ denoted the change in value of T_m for unit change in B.Th.U. per cubic foot of swept and clearance volumes ?

The Author, *in reply, observed that the further confirmation that* Mr. Brewer had adduced of combustion-efficiency being independent of the actual cycle of operations was very welcome. Mr. Brewer's engine, as described in his Paper †, possessed many novel features, and was designed primarily to use high-pressure natural gas which could be injected into an already assembled air-charge. In the Erren engine also, compressed coal-gas was injected after the completion of the charging period, whether on either the two- or the four-stroke cycle, and in that respect it was similar. Mr. Brewer's confirmation that truer comparisons were obtained on the basis of total cylinder volume than on piston-displacement volume was also valuable. With regard to the exhaust scavenging process, the Author agreed that Mr. Paulin's more lengthy explanation had merits over his own condensed phrases, but it was thought to be unnecessary to reiterate a generally-accepted statement as to pressure-waves in an exhaust-pipe system.

Mr. Paulin had evidently failed to realize that once the engine frictional mean pressures had been taken into account in $P_m - P_f = P_n$, the values of $\triangle T_m$ and of $\triangle T_n$ were bound to be one and the same. It was surely well worth while drawing attention to the changing values of $\triangle T_m$ or of $\triangle T_n$ with mixture-strength variations. It might be that the evaluation of $n - 1$ from $\triangle T_m$ or $\triangle T_n$ was unorthodox, but if excuse be needed material progress had often resulted from such departures. In the present instance the whole tenor of the Paper was to co-ordinate practice with theory, and the mathematical evaluation of $n - 1$ was deemed by the Author to be sufficiently interesting to warrant its inclusion in Table XIV (p. 189 §), as with Table XV, p. 190 § (calculated on a different basis), it was possible to realize the gradual deterioration of performance with increase of loading, whether by increment as in Table XIV or as a whole in Table XV, and by hypothesis to indicate apparent specific-heat ratios at the higher temperatures.

† Footnote (*), p. 401. § *Ibid.*