

test of 6.63, while piston displacement computation showed 8.32.

The second approximate computation for the weight of the exhaust gases is based upon the exhaust gas analysis. The method of doing this has been explained in detail in Chapter VI. The trouble with this scheme lies in the difficulty of obtaining representative gas samples. But granting even that these are obtained, it is often found that computations based upon the analyses show an excess coefficient smaller than that really used. That is, the weight of exhaust gases so determined is less than the actual amount. Thus, Schöttler, in computation on some of Slaby's tests, shows in one case that, based upon analysis, the excess coefficient was 5.52, while in reality it was 6.2. Schöttler attributes this discrepancy to a change in the analysis due to a burning up of the lubricating oil, which is apt to increase the CO_2 content of the exhaust gas at the expense of the percentage of O. It has been shown that a variation in the supply of lubricating oil may change the fuel consumption under circumstances quite materially, and Schöttler's surmise is therefore probably correct. At the same time, however, any such effect must be more marked in the smaller machines, and the writer believes that, given representative samples of exhaust gas from anything but the small machines, in conjunction with accurate fuel analysis, a very close approximation to the actual weight of the exhaust gases can be obtained. At any rate the method should be more accurate than that based on piston displacement.

Finally, it should not be forgotten that even an accurate determination of the air supply still leaves open the question of the specific heat of the exhaust gases.

The heat balance of the four items above outlined is sometimes shortened to three by combining items 3 and 4 and determining their sum by difference. On the other hand, a balance of many more items can be drawn up. Thus each event of the cycle may be examined by itself and the heat and energy interchanges be determined. A very detailed balance of this kind is given in Schöttler's "Die Gasmachine," p. 321. How far this question needs to be entered into depends altogether upon the importance of the test, but an effort should be made in every case to so arrange

the apparatus that at least a heat balance of the kind first discussed can be drawn up from the results of the test.

8. Results of Tests of Engines and Gas Producers. — By this term is here meant the results shown by engines or producers when operated under normal conditions and at or near normal load. The number of tests from which this data can be obtained is at the present writing quite large, and some notable collections of test data have been made. The most extensive is perhaps that contained in Appendix A of Brian Donkin's "Gas, Oil and Air Engines." This collection consists of eleven tables, arranged according to fuel used, containing in all some 280 tests. Another large collection is that found appended to Witz's "Moteurs à Gas et à Pétrole." Such collections are valuable as showing what has been done, and serve as a guide as to what may be expected from an engine under design or construction. The greatest care, however, should be exercised to see that only reliable data is incorporated.

ENGINE TESTS. — Table I (pp. 544 and 545) contains a series of engine test data taken from various sources. The tests are arranged according to the kind of fuel used, this being the most logical way. In some cases not all the data is given in the original report. Wherever possible computations have been made to make the items complete. In many cases, however, not enough information is given to permit of this, and the record necessarily remains incomplete.

TESTS OF PRODUCERS AND PRODUCER PLANTS. — Table II, p. 546, gives some of the results obtained on tests of producers. In some cases the tests refer to producers only, giving no data regarding the engine used; in others the data is fairly complete for the entire plant.

In conclusion, Fig. 17-5 shows a set of curves drawn by C. H. Day in 1905 during an investigation on the economy of producer gas. The curves show the relation between cubic feet of gas used and brake horse-power for various kinds of gas. While it must be understood that they are approximate only, they represent a fair general average of what is accomplished to-day. Many plants show much better fuel consumption, but others show

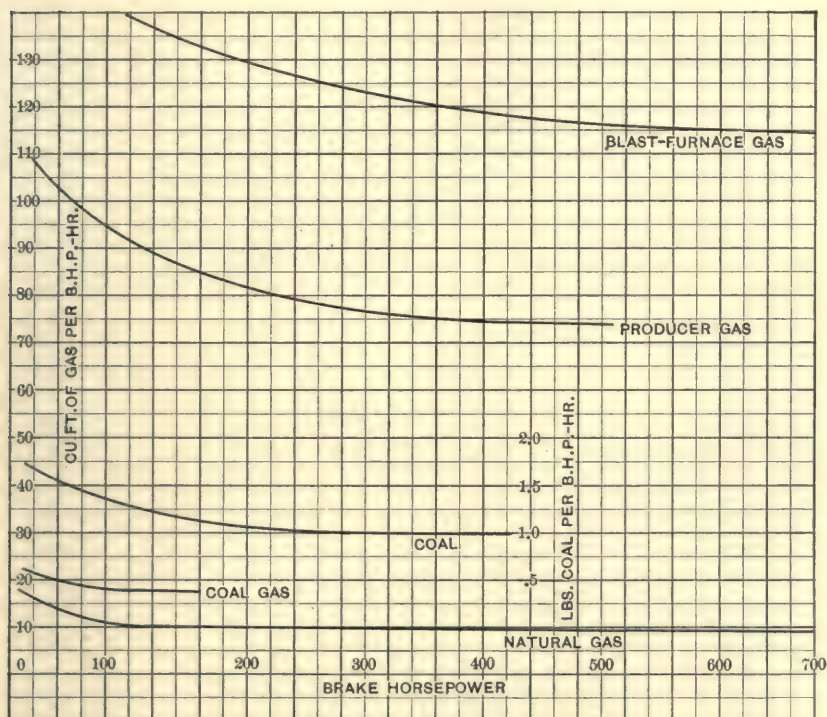


FIG. 17-5.

correspondingly worse. The curves also show in a measure the sizes up to which the various engines are built. Thus an illuminating gas engine of 150 horse-power is a large engine of its type. Producer plants in 1905 apparently were not built much larger than 500 horse-power, while natural gas and blast furnace gas engines were built exceeding 1000 horse-power, and for the latter gas 2000 and 3000 horse-power is not now out of ordinary.

TABLE I.

Kind of Fuel	Name of Engine	DIMENSIONS AND OTHER DATA					Rated B.H.P.	R.p.m.
		Dia. of Cyl. Inch.	Stroke Inch.	No. of Cyls.	2 or 4-cycle	Single or Double Act.		
Illuminating Gas	Koerting	—	—	—	4	S	—	161
Illuminating Gas	Deutz	4.96	5.92	1	4	S	2	260
Illuminating Gas	Güldner	—	—	1	4	S	15	—
Illuminating Gas	Güldner	10	15.7	1	4	S	20	210.7
Illuminating Gas	Bánki	—	—	1	4	S	16	255
Illuminating Gas	Tangye	10	19	1	4	S	—	193.6
Illuminating Gas	Crossley	7	15	1	4	S	—	200
Illuminating Gas	Westinghouse	—	—	3	4	S	—	235
Natural Gas	Westinghouse	25	30	3	4	S	550	150
Natural Gas	Snow	25	48	4	4	—	—	—
Natural Gas	Otto	11.25	18	1	4	S	36	220.4
Natural Gas	Walrath	13	14	3	4	S	75	253.3
Producer Gas	R. D. Wood	25	30	2 (Tandem)	4	S	300	149
Producer Gas	Westinghouse	—	—	3	4	S	—	235
Producer Gas	Crossley	26	36	—	4	S	—	152.4
Producer Gas	Koerting	21.6	37.7	1	2	D	—	101
Producer Gas	Deutz	—	—	—	4	—	—	161.6
Coke Oven Gas	Oechelhäuser (Borsig)	65	37.5	1	2	2 pistons in 1 cylinder	500	110.6
Mond Gas	Crossley	16.9	24	2 (Opposed)	4	S	—	162
Mond Gas	Crossley	26	36	2 (Opposed)	4	S	400	148.5
Blast Furnace Gas	Nürnberg	33.5	43.4	2 (Tandem)	4	D	—	105.6
Blast Furnace Gas	Berlin-Anhalt	16.95	27.5	1	4	S	60	160.6
Blast Furnace Gas	Cockerill	51.2	55.2	1	4	D	—	94.57
Gasoline	Fairbanks	6.5	9	1	4	S	7	300
Gasoline	Springfield	6.5	12	1	4	S	6	230.4
Gasoline	Lozier	5	6	1	2	S	5	513
Gasoline	Westinghouse	5.75	8	2	4	S	10	289.7
Gasoline	Bánki	—	—	1	4	S	—	209.1
Gasoline	Daimler	3.56	5.11	4	4	S	16-20	400
Gasoline	Daimler	3.56	5.11	4	4	S	16-20	600
Gasoline	Daimler	3.56	5.11	4	4	S	16-20	1000
Kerosene	Diesel	10.23	16.16	1	4	S	—	186.6
Kerosene, Russian	Diesel	15.75	23.65	1	4	S	70	158.8
Kerosene, Russian	Diesel	6.65	10.60	1	4	S	8	270.3
Kerosene	Priestman	10.9	14.1	1	4	S	—	172
Kerosene	Grob & Co.	9.07	9.07	1	4	S	8	266
Kerosene	Swiderski	10	10	1	4	S	—	249
Kerosene	Koerting	6.9	10.82	1	4	S	—	222
Kerosene	Blackstone	7	12	1	4	S	—	240
Kerosene	Stevenson	9.5	18	1	4	S	—	200
Kerosene	Hornsby	8.2	14	1	4	S	—	213.8
Kerosene	Hornsby	14.5	17	1	4	S	25	202.6
Crude Oil	Diesel	—	—	—	4	S	225	169.1
Alcohol, 86.1 vol. %	Deutz	8.35	11.8	1	4	S	12	276.9
Alcohol, 86.1 vol. %	Mariefelde	9.95	15.75	1	4	S	14	197.6
Alcohol, 86.1 vol. %	Koerting	6.16	9.95	1	4	S	6	307.3
Alcohol, 87 vol. %	Bánki	—	—	1	4	S	20	225

ENGINE TESTS

H.P. ON TEST		LOWER HEATING VALUE OF FUEL B.T.U. PER		HEAT DISTRIBUTION, %				Mech. Eff. %	Thermal Eff. on B.H.P. %	References and Remarks
I.H.P.	B.H.P.	Lb.	Cu. ft.	Work I.H.P.	Jacket	Exhaust	Rest			
—	108.1	—	500	—	—	—	—	—	28.22	De La Vergne Machine Co., Cat.
2.30	1.72	—	572	21.5	50.4	25.0	4.1	75	16.1	Wimplinger, Zeitschrift d. V. d. I. Sept. 8, 1906.
—	—	—	580	40.5	—	—	—	—	—	Dubbel, Z. d. V. d. I., Nov. 3, 1906.
35.9	—	—	—	42.7	33.2	24.1	—	—	—	Test by Schröter, Z. d. V. d. I., June, 1904.
—	17.03	—	—	—	27.97	—	—	—	31.0	Schimanek, Z. d. V. d. I., Jan. 17, 1903.
28.6	25.4	—	609	—	—	—	—	85	24.5	Witz, 1902.
14	12	—	680	—	—	—	—	86	22	Clerk, 1894.
196.5	147.5	—	597.4	35	—	—	—	76	26.7	Ballinger & Hunt, Sibley College Thesis, 1904.
618	550	—	1000	27.1	—	—	—	89	24.1	J. R. Bibbins, A.I.E.E., Dec. 1903.
736.7	594.5	—	1175(?)	29.4	—	—	—	80.7	23.7	Hastings & Parker, Sib. Coll. Thesis, 1901.
36.3	28.8	—	1086	20.2	—	—	—	79.5	16.1	Hunting, Sib. Coll. Thesis, 1902.
86.7	78.7	—	1041	27.1	49.5	23.4	—	78.7	21.3	Geer & Vanelain, Sib. Coll. Thesis, 1902.
242	169	—	145	24.4	25.0	50.6	—	70	17.1	Goldsmith & Hartwig, Sib. Coll. Thesis, 1906.
173.6	124.8	—	123.8	33	—	—	—	71.8	23.7	Ballinger & Hunt, Sib. Coll. Thesis, 1904.
377.9	313	—	90.13	—	—	—	—	83	21.78	Humphrey, Inst. Mech. Engs., 1900.
481	341	—	129.5	—	—	—	—	71	24.1	E. Meyer, Jour. für Gas Belenchtung, 1900.
81	70.4	—	113	—	—	—	—	87	29	E. Meyer, 1903.
765 (net)	628 (Blowing Cyl.)	—	376	33.4	—	—	—	82.1	27.5	E. Meyer, Z. d. V. d. I., Feb. 1905. Test No. VIIb.
141.6	—	—	135	29.1	—	—	—	—	—	} Humphrey's Test, Schöttler, Z. Jan. 1902.
440	364	—	138	28.6	—	—	—	83	23.7	
1427	1186	—	88	33.9	—	—	—	83.1	28.2	Riedler, Gross-gasmaschinen, p. 158.
79.5	54.5	—	102	—	—	—	—	—	26.2	E. Meyer, Z. 1899, p. 448.
786.16	575	—	99.5	—	—	—	—	—	20.4	Hubert, March, 1900.
11.28	7.23	18200	—	20.5	—	—	—	64	13.15	Elwood, Ford, Garrow, Sib. Coll. Thesis, 1907.
9.63	6.16	20078	—	15.8	—	—	—	64	10.2	Keeley & Spier, Sib. Coll. Thesis, 1900.
6.28	4.92	18520	—	19.8	—	—	—	78.9	15.7	Bayne & Speiden, Sib. Coll. Thesis, 1902.
9.19	5.58	—	—	28.5	—	—	—	60.7	17.3	Glasgow & Powley, Sib. Coll. Thesis, 1902.
—	26.4	—	—	—	—	—	—	—	28	Jonas & Taborsky, Z. Oest. Arch. & Ing. Verein, p. 512, 1900.
—	—	17500	—	19.3	—	—	—	86.1	16.6	} Prof. B. Hopkinson, Cambridge, 1906.
—	—	17500	—	22.0	—	—	—	85.4	18.8	
—	—	17500	—	24.2	—	—	—	79.8	19.3	
30.46	20.81	18604	—	37.7	—	—	—	68.3	25.8	Cat. American Diesel Engine Co.
88	69.6	18610	—	40.3	—	—	—	79.1	31.9	} E. Meyer, Z. d. V. d. I., May, 1903.
11.19	8.6	18610	—	35.8	—	—	—	77.0	27.6	
10.69	10.18	19800	—	—	—	—	—	95.2(?)	13.4	Hartman, Z. d. V. d. I., 1895, p. 586.
11.68	9.22	19800	—	—	—	—	—	79	13.6	Hartman, Z. d. V. d. I., 1895, p. 616.
—	10	19600	—	—	—	—	—	—	15.8	} Schöttler, die Gasmaschine, p. 207.
—	4.15	19699	—	—	—	—	—	—	9.9	
—	3.05	18000	—	—	—	—	—	—	19.7	} Engineering, Vol. 88, 1899.
—	12.44	18000	—	—	—	—	—	—	9.2	
6.07	4.95	18600	—	16	29	55	—	81.5	13	Robinson, 1893, Gas & Pet. Engines, p. 702.
31	26.74	18870	—	21	50	29	—	86.0	18	Robinson, 1898, Gas & Pet. Engines, p. 710.
—	249.7	19460	—	—	—	—	—	—	28.1	Kimberley & Clark Paper Co., Kimberley, Wis., Power, Oct. 1906.
—	16.8	9900	—	—	—	—	—	—	31.6	} E. Meyer, Z. d. V. d. I., April, 1903.
—	19.77	9900	—	—	—	—	—	—	32.7	
—	7.39	9900	—	—	—	—	—	—	21.8	
—	32.13	9700	—	—	32.1	—	—	—	30.18	Schimanek, Z. d. V. d. I., Jan. 17, 1903.

TABLE II. TESTS OF GAS PRODUCERS

Make of Engine	Make of Producer	Type of Producer	Kind of Fuel	Heating Value of Fuel B.T.U. per cu. ft.	ENGINE			HORSE-POWER		Producer Efficiency %	FUEL, LBS. PER HOUR	Heating Value of Gas B.T.U. per cu. ft.		References and Remarks
					Dia. Cyl. Inch.	Stroke Inch.	R. p.m.	Indicated	Brake			L.H.P.	H.H.P.	
R. D. Wood	Taylor	Pressure	Anthracite	10240	25	30	149	242	169	79.4 (cold gas)	1.28	1.83	145	Goldsmith & Hartwig, Sibley College Thesis, 1905, Zeitschrift des Vereins deutscher Ingenieure, June 25, 1904, } Z. d. V. d. I., June 17, 1905.
Guldner	Guldner	Suction	Anthracite	13630	—	—	220.8	44.8	35.6	—	.692	.875	—	
Mees	Fintsch	Suction	Coke	13270	11	15.7	220	25.76	20.2	—	.766	1.09	—	
Campbell	Fintsch	Suction	Anthracite	14620	11	15.7	220	26.30	21.48	—	.732	.895	—	
Campbell	Campbell	Suction	Scotch Anth. Pea Coal	—	9.5	18.0	192.4	23.73	20.44	—	—	.93	—	
Crossley	Crossley	Suction	"	—	7	12	232.5	9.66	8.25	—	—	1.22	—	
Ind. Eng. Co. Manchester	Ind. Eng. Co. Manchester	Suction	"	—	8.5	14	189.2	18.25	15.35	—	—	.77	—	
National	National	Suction	"	—	8.5	14	204.9	—	9.72	—	—	1.13	—	
National	National	Suction	"	—	7	15	219.80	11.88	9.74	—	—	.84	—	Trials of Suction Gas Plants, Highland and Agricultural Soc. of Scotland, Power, Feb., 1906.
Tangye	Tangye	Suction	"	—	10	18	193	25.54	20.57	—	—	.80	—	
Tangye	Tangye	Suction	"	—	7	16	224.4	10.86	8.34	—	—	1.25	—	
Nürnberg, Single Act.	Nürnberg, Single Act.	Suction	Anthracite	—	10	19	197.9	23.45	19.98	80	.78	.83	—	Test at Royal Foun- } Math- dry of Wurtemberg } at in Test in Post Office, } Power, Hamburg Feb. 07
Nürnberg, Single Act.	Nürnberg, Single Act.	Suction	Coke	—	—	—	—	134.4	107.4	75	.93	.975	—	
Tangye	Tangye	Suction	Scotch & Welsh Anth. Bit. & Pocahontas Slack	—	—	—	—	137.5	110.0	—	—	1.16	—	
Westinghouse, 500 H.P.	Loomis-Pettibone	Suction	Scotch & Welsh Anth. Bit. & Pocahontas Slack	—	11	20	186	36.34	29.46	—	—	.709	—	R. E. Mathot in Power, May, 1906. Includes stand-by loss, Norton Plant, Worcester, Mass., Power, July, '07.
Westinghouse, 250 H.P.	Taylor	Pressure	Colorado Bit.	—	—	—	—	—	—	—	—	1.42	—	
Stockport, 250 H.P.	Wilson	Pressure	Bituminous	9767	—	—	—	—	—	—	—	1.95	—	
Campbell, 280 H.P.	Dowson	Pressure	Bituminous	—	—	—	—	—	—	—	—	1.40	—	Data from Tests, collected by J. I. Wile, Power, April, 1906.
Westinghouse, 3000 H.P.	Dowson	Pressure	Anthracite	—	—	—	—	—	—	—	—	1.12	—	
Deutz, 250 H.P.	Deutz	Suction	Anthracite	14000	—	—	—	—	—	—	—	1.20	—	
Crossley, 90 H.P.	Crossley	Suction	Coke	—	—	—	—	—	—	—	—	.744	—	Test No. 15, U. S. Geological Survey Coal Testing Plant, St. Louis, Mo. Report by A. Wilz.
Crossley, 300 H.P.	Crossley	Suction	Anthracite	11370	—	—	—	—	—	—	—	.91	—	
Westinghouse, 225 H.P.	Taylor	Pressure	Colorado, No. 1 Lignite	—	19	22	200	—	235.4	64.2	—	1.55	—	
Letombe	Letombe	Suction	—	—	23.6	31.5	135	—	290	—	—	.81	—	

CHAPTER XVIII

COST OF INSTALLATION AND OF OPERATION

1. **Cost of Producers and Engines.** — With the approach to standardization of the design of the internal combustion engine there has also come a tendency toward standard prices. The figures available, however, for anything but small engines are not plentiful. This is no doubt due to the fact that large installations are as yet not numerous and information is correspondingly scarce. The writer has been able to glean the following data from current literature:

PRODUCERS. — The *Gas Engine*, June, 1906, gives the following figures for producers:

Pressure Producers, erected complete but not including freight, \$25 per horse-power for hard coal, and \$27 per horse-power for soft coal producers. These figures are for sizes in the neighborhood of 300 horse-power. The cost is distributed approximately as follows:

- 25 per cent in the producer.
- 15 per cent in the scrubber.
- 30 per cent in the gas holder.
- 30 per cent in piping and fittings.

The increased cost of the bituminous coal plant is mainly due to greater cost of washing apparatus.

Suction Gas Plants will, on a rough estimate, cost only about one-half the above. The reasons for this are greater capacity of producer and the elimination of the gas holder. One maker gives the following approximate figures for the price of suction gas plants from 25 to 250 horse-power.

$$\text{Price per horse-power} = \$11 + \frac{400}{\text{H.P.}}$$

Thus a 25 horse-power plant would cost \$37.00 per horse-power, and one of 250 horse-power \$12.60 per horse-power. In the writer's experience practice agrees fairly well with these figures.

ENGINES. — The cost of stationary gas engines at present seems to range from about \$70 per horse-power for very small engines to in the neighborhood of \$25 per horse-power for large ones. Recent quotations on producer-gas engines for instance showed the cost to be about \$55 per horse-power for a 10 horse-power engine, and \$36 per horse-power for a 50 horse-power machine. The latter is, however, a comparatively low price for an engine of this size. The price of launch engines is frequently lower than this, from \$50 per horse-power for the small engine to about \$20 per horse-power for an engine of 25 to 30 horse-power. The cost of Diesel engines runs somewhat higher, varying probably between \$50 and \$90 per horse-power, depending upon size. None of these figures include cost of erection.

Suction Gas Plants. — Recent quotations for plants of this type have been for a 10 horse-power plant, which is below the ordinary commercial size, \$90 per horse-power for a 50 horse-power plant \$51 per horse-power and again \$61 per horse-power, and for a 75 horse-power plant \$53 per horse-power.

2. Cost of Erection. — The cost of erection varies largely with the size of the foundations. Regarding the latter there seems to be no definite proportionality between it and the size of the engine. In general, of course, vertical engines do not require as large or as heavy a foundation as horizontal engines, but the condition of the ground has a great deal to do with the final cost. Güldner* estimates that from 15 to 20 cubic feet of foundation per brake horse-power of engine should be sufficient in the ordinary case if the ground is safe. For large machines, however, this may even be as low as 8 cu. ft. per B. H. P. The total cost of erection and starting should not exceed about 3 per cent of the purchase price in the case of gasoline and gas engines, and should not be over 5 per cent in the case of producer gas plants. These figures agree fairly well with the estimates made by H. A. Clark in 1903.†

3. Piping and Auxiliaries. — This item is apt to vary within wide limits, especially in producer gas plants, where much depends upon the location of the producer plant with reference to

* *Verbrennungsmotoren*, p. 433.

† H. Ade Clark, *The Diesel Engine*, *Proceedings of Mechanical Engineers*, 1903.

the engine. In a normal case, 5 per cent of the cost price of the engine or plant should cover the cost. This, however, does not include the cost of an air compressor system if this is required to start the plant.

4. Floor Space and Buildings.—The cost of buildings is intimately connected with the floor space required by the plant, but outside of this nothing very definite can be said of the cost of the building, since everything depends upon the construction used. The cost based on the square foot of floor space may vary from about \$1.50 for a plain brick building with steel roof, to \$2.75 for a modern steel concrete structure. Among the figures available are the following. A. H. Clark, in the article already mentioned, estimates the cost of building as follows:

Type of Engine	Size of Engine B. H. P.	Cost of Engine, Producer and Accessories	Cost of Building	Ratio Cost of Bldg. Cost of Plant
Producer Gas *	35	\$2,360	\$ 640	.27
Diesel	35	2,860	440	.16
Producer Gas ...	80	4,400	930	.21
Diesel	80	4,400	685	.16
Producer Gas ...	160	7,450	1,150	.16
Diesel	160	8,300	880	.11

Güldner, in making a similar estimate, combines the cost of building and of necessary foundations for the engine and producer. Expressing this combined cost in per cent of the cost of plant and accessories, the following table shows the results:

Type of Plant	Horse-power of Plant													
	5	10	15	20	25	30	40	50	100	125	150	175	200	
Illuminating Gas Engine	27	22	20	19	18	17	16	15	13	13	13	13	13	
Suction Gas Plant Complete	30	26	24	22.5	21	19.5	19	18	15.5	15	15	15	14.5	

Turning next to the floor space required, there is considerable data available in the literature published. Collaborating the

* Pressure producer.

figures given by manufacturers for the floor space required for horizontal single-cylinder gas, gasoline and oil engines, and for suction gas producers, we obtain Curves I and III of Fig. 18-1.

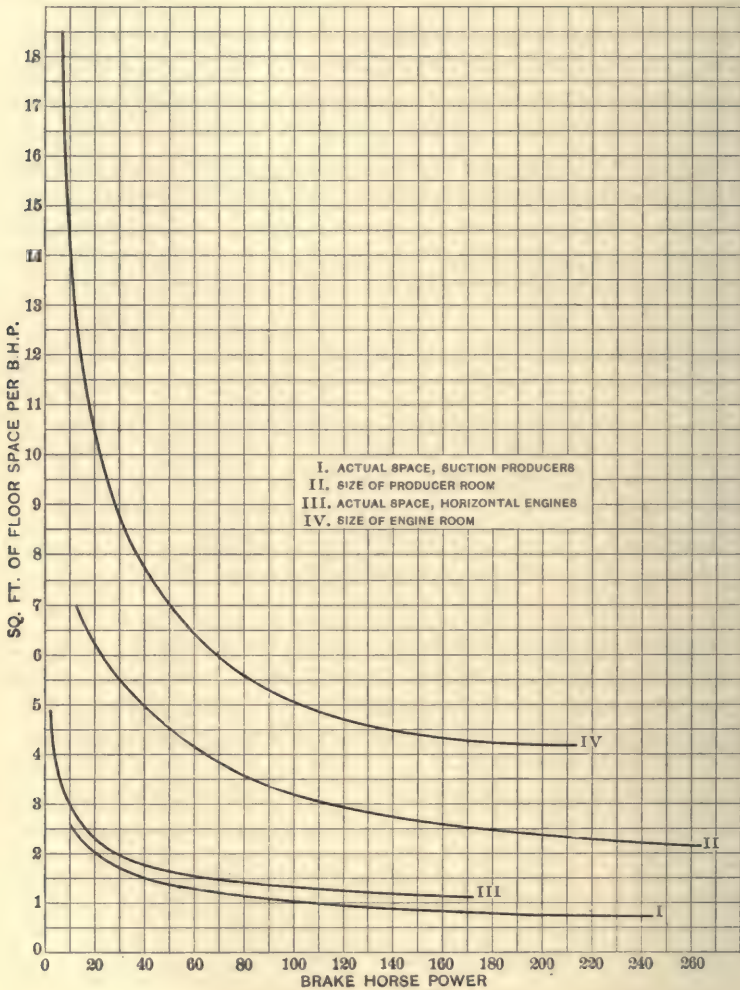


FIG. 18-1.

It should be remembered in connection with these curves, however, that they represent average figures only and they can therefore give only approximate results. For example, a 20 B. H. P.

suction gas producer will, according to Curve I, occupy approximately $20 \times 2 = 40$ sq. ft. of floor space, say a space $9 \times 4\frac{1}{2}$ ft.; similarly, a 200 B. H. P. installation would require $200 \times .8 = 160$ sq. ft.

In like manner Curve III gives the approximate floor space for horizontal engines. Thus a 5 B. H. P. gasoline engine of standard make would probably require $5 \times 4 = 20$ sq. ft., say a space 3 ft. by 6 ft.; while a 150 B. H. P. producer or illuminating gas engine calls for $150 \times 1.5 = 225$ sq. ft. The curve does not include multi-cylinder engines because the available figures for this type of engine are very erratic. For the same reason no curve could be drawn for vertical engines.

While Curves I and III represent the space actually occupied by the plant, they give no idea of the size of the producer or engine room required. In the producer room space must be left for the proper cleaning of the producer, etc., and in the engine room space for dismantling, etc. On this point the only information that seems available is that given by Güldner in his estimates, the results of which are plotted in Curves II and IV of Fig. 18-1. The former shows the approximate room allowance for suction gas apparatus, the latter for horizontal single-cylinder engines, based on one brake horse-power in each case.

The data available for *pressure producers* is not so extensive as that for suction producers. In the former plants much depends upon the size of the gas holder. Outside of this the space required for suction and pressure producers should not vary greatly. The figures at hand seem to indicate that pressure producer plants, including the gas holder, of from 30 to 40 horse-power occupy 75 per cent more space than suction plants of like capacity. As the power increases the difference grows less, a plant of 100 horse-power requiring apparently only about 50 per cent more space, and one of 175 horse-power only 20 per cent more.

The *minimum head room*, i.e., the height of the room required for either type of plant, seems to vary from 12 ft. in the small 20 horse-power plant to 18 ft. in the large 250 horse-power producer.

In Power for April, 1907, L. L. Brewer published the following table of "Practical Data for Modern Gas Engines." The table gives in column 13 figures for the floor space required by engines larger than those so far discussed. The letters in column 7 have

the following meaning: s.c., single cylinder; tw., twin; td., tandem; d. td., double tandem.

1	2	3	4	5	6	7	8	9	10	11	12	13
Brake Horse-Power	Revolutions per Minute	Builder	No. Cylinders	Strokes per Cycle	Single or Double Acting	Cylinder Arrangement	Weight of Engine without Fly-wheel, Lbs.	Weight of Fly-wheel for Blowing and Pumping Service, Lbs.	Weight of Fly-wheel for Dynamo Driving	Weight of Engine and Fly-wheel (9) per B. H. P.	Weight of Engine and Fly-wheel (10) per B. H. P.	Sq. ft. Floor Space per B. H. P.
100	150	Cockerill	1	4	s.	sc.	45,000	9,000	21,100	540	661	2.05
200	105	Cockerill	1	4	s.	sc.	83,000	25,000	58,500	540	706	1.81
250	150	Cockerill	2	4	s.	td.	65,000	10,000	23,400	300	353	1.24
300	120	Deutz	1	4	s.	sc.	83,500	35,000	81,800	295	551	2.07
300	120	Deutz	2	4	s.	tw.	101,000	14,000	32,800	383	447	1.52
300	140	Deutz	4	4	s.	d. tw.	110,000	3,500	8,200	484	394	1.32
600	80	Cockerill	1	4	s.	sc.	207,000	100,000	234,000	512	734	0.99
600	130	Cockerill	2	4	s.	td.	185,000	46,000	107,500	386	487	1.13
800	110	Oechelhäuser	1	2	s.	sc.	143,000	48,000	112,000	318	425	1.23
800	130	Deutz	2	4	s.	tw.	158,000	28,000	65,500	310	371	1.67
800	130	Deutz	4	4	s.	d. tw.	189,000	7,000	16,400	327	342	1.08
800	110	Koerting	1	2	d.	sc.	136,500	18,000	42,200	258	297	1.11
750	90	Nürnberg	1	4	s.	sc.	297,000	115,000	26,900	560	538	1.03
1200	80	Cockerill	2	4	s.	td.	365,000	95,000	222,000	383	488	0.68
1200	130	Deutz	4	4	s.	d. tw.	354,000	14,000	32,800	307	322	1.01
1200	120	Nürnberg	4	4	s.	d. tw.	280,000	16,000	37,400	246	264	0.94
1200	110	Oechelhäuser	2	2	s.	tw.	260,000	16,000	37,400	230	248	0.9
1200	110	Koerting	2	2	tw.	td.	250,000	4,500	10,500	212	218	0.9
1400	110	Cockerill	2	4	d.	td.	374,000	8,600	20,000	170	164	0.42

It will be noted from column 6 that there is only one double-acting four-cycle engine in the list. The large engines cited are all single-acting twin or tandem, or double engines. It would naturally be assumed that the later types of large four-cycle engines, which are nearly always double-acting tandem, or double-acting twin tandem machines, should show a decrease in the floor space required as compared with the figures in column 13. An indication of the saving in both weight and floor space that may be effected by adopting the double-acting principle is given by the figures for the 1400 B. H. P. double-acting tandem Cockerill engine in the last line of the table. The saving there shown is remarkable and the writer has not been able to check it in the case of other engines the design of which has been changed from the twin or double-twin single-acting to the double-acting tandem. The reason for this probably is that opposed single-acting engines usually work without a cross-head,

while double-acting cylinders demand the use of the same. Hence the total length of engine is not changed materially.

5. Cost of Operation. — The total cost of operation consists of the following items:

- (a) Interest on capital.
- (b) Depreciation of Plant and Buildings.
- (c) Insurance.
- (d) Fuel Cost.
- (e) Cost of Cooling Water.
- (f) Lubricating Oil and Waste.
- (g) Attendance.
- (h) Maintenance and Repairs.

Of these, the first three items are usually called the *fixed charges*, and the last five the *operating* or *works cost*. The sum of fixed charges and works cost is the *total operating cost*.

(a) **INTEREST ON CAPITAL.** The usual allowance in this country for interest on capital is 6 per cent.

(b) **DEPRECIATION OF PLANT AND BUILDINGS.** There is little doubt that as far as gas producers are concerned, the wear and tear on this part of the plant is much less than it is on a boiler plant of the same capacity. On the other hand, the stresses in a gas engine are generally higher throughout than those in a steam engine of the same power, and this naturally leads to a shorter life and consequently somewhat higher allowance for depreciation. Taking it altogether, an allowance of from 7 to 10 per cent of the capital outlay for the power plant should cover depreciation, the lower figure for a producer plant, the higher where engines alone are concerned.

Depreciation on the building should not exceed 2 to 3 per cent of the building cost. In case the building is rented, the rent instead of depreciation should be charged against the plant.

(c) **INSURANCE.** This item is usually very small as compared with the rest and is therefore in most cases neglected.

(d) **FUEL COSTS.** This item is very often used as the sole criterion of the economic status of a plant, but in many cases this is not true. Where the cost of fuel is very high, as for instance where illuminating gas or gasoline are used, the fuel cost usually forms the major part of the operating expenses. But where the cheaper fuels are used or the plant is of high efficiency, as is often

the case in producer plants, an analysis will show that what are usually considered the "incidental" expenses in many cases far outweigh the fuel cost.

The fuel cost varies with the load on the engine. The figures given by manufacturers usually represent the best figures obtained at maximum load, but such conditions rarely obtain in practice. At any rate it would not be safe to base computations upon such "parade" figures. Haeder * considers that the economy figure given by an engine at $\frac{7}{10}$ maximum load, which is equal to 85 per cent of rated load if an over-capacity of 20 per cent is assumed, is the best figure on which to base computations. The same writer gives the following table for the variation of the fuel consumption with load. The third line in this table gives the same information regarding a steam plant:

Gas Engine	VARIATION OF FUEL CONSUMPTION WITH LOAD									
	Max. Load	.9	.8	.7	.6	.5	.4	.3	.2	.1 of Max. Load
Hit-and-miss regulation	1	1.03	1.08	1.14	1.23	1.35	1.50	1.8	2.5	3.0
Throttling regulation	1	1.06	1.13	1.21	1.33	1.50	1.75	2.2	3.0	5.0
Steam plant	1.1	1.05	1.02	1.0	.96	1.02	1.07	1.15	1.30	1.60

The table is based upon the assumption that the gas engine operates normally on $\frac{7}{10}$ of its maximum load, and this is put equal to the normal load on the steam engine. It is interesting to note the superiority of hit-and-miss regulation, as far as economy is concerned, over the other method of governing, and the superiority of the steam engine over both as regards small variation in economy over a wide range of load.

Gulder † makes a similar estimate with the following results:

INCREASE OF FUEL CONSUMPTION WITH DECREASING LOAD

Approximate Load	.75	.66	.50	.33	.25	of Normal Load.
Illuminating Gas Engine.	10	20	35	60	90	} per cent higher fuel consumption than at full load.
Suction Gas Engine	20	30	50	75	100	
Diesel Oil Engine.....	10	20	30	55	80	

* Haeder, Die Gasmotoren.

† Güldner, Verbrennungsmotoren, p. 427.

Thus, for example, if an illuminating gas engine uses 20 cu. ft. of gas per B. H. P. hour at full load, we may except it to use $1.35 \times 20 = 27$ cu. ft. per B. H. P. hour at half load, and $1.9 \times 20 = 38$ cu. ft. per B. H. P. hour at quarter load.

The anthracite coal consumption of suction gas plants for various sizes of plants and for different loads is given by Haeder in the following table:

CONSUMPTION OF ANTHRACITE IN SUCTION GAS PLANTS IN POUNDS PER
B. H. P.-hour

Max. Capacity B. H. P. . . .	14	40	70	100	140	210	420
Normal Load B. H. P.	10	30	50	70	100	150	300
Consumption							
pounds per	B. H. P. max.	1.21	1.0	.93	.90	.88	.86
H. P.-hour	.7 B. H. P. max.	1.45	1.19	1.12	1.10	1.06	1.03
based on . .	.5 B. H. P. max.	1.80	1.47	1.38	1.34	1.32	1.28
						1.28	1.25

It is evident from a study of the figures in the above tables that the probable average load at which a prospective plant will operate plays an important part in the estimation of fuel costs, a point which should not be lost sight of.

We find in the engineering literature of the day considerable information regarding fuel consumption and fuel costs. The cost figures given belong to one of two classes: either they are based upon assumptions regarding both consumption of fuel and cost of unit weight of the fuel, or the figures are obtained in actual operation. It goes without saying that the latter class is much more valuable than the former. In the case of assumed costs and consumptions, comparisons are usually made between steam and gas power under various conditions of operation. Such computations are interesting because they show what might be realized in each case, but in attempting to practically apply the information the greatest attention should be paid to the conditions assumed. As regards actual consumption figures, the tables of the previous chapter give considerable information regarding efficiency of operation. From this data it should be easy to compute the fuel costs as soon as the local cost of unit weight is known.

Below are given a few hypothetical computations. In some cases both the fuel consumption and the fuel cost are assumed outright, in others an attempt has been made to determine the

fuel consumption in relation to the size of the engine.' These figures are followed by a few fuel cost figures from actual practice. Additional data for fuel cost from actual operation will be found in the data on total operating costs at the end of this chapter.

The following table shows an estimate taken from the catalogue of a well-known manufacturer.

RELATIVE COSTS OF FUEL WITH DIFFERENT TYPES OF ENGINES

Type of Engine	Class of Fuel	Price of Fuel	Fuel Consumed per B. H. P. per hour	Cost in cents per B. H. P. per hour	Cost per annum 3000 hours per 100 B. H. P.
Simple non-condensing slide valve	Bituminous Coal	\$3.00 per gross ton	5 lbs.	.669	\$2,007.00
Compound condensing	Bituminous Coal	\$3.00	3 lbs.	.402	1,205.00
Steam Turbine	Bituminous Coal	\$3.00	3 lbs.	.402	1,205.00
Oil Engine	Gasoline	14 cents per gal.	0.125 gal.	1.75	5,250.00
Oil Engine	Crude Oil	4 cents per gal.	0.10 gal.	.40	1,200.00
Gas Engine	Illuminating Gas	75 cents per 1000 cu. ft.	19 cu. ft.	1.425	4,275.00
Gas Engine	Natural Gas	20 cents per 1000 cu. ft.	13 cu. ft.	.26	780.00
Gas Engine with suction producer.	Anthracite Coal	\$3.00 per gross ton	1 lb.	.134	402.00
Gas Engine with suction producer.	Coke	\$3.00 per gross ton	1.25 lb.	.167	502.00

The next table, compiled by J. I. Wile, is similar to the above, except that the assumptions differ somewhat. A comparison of the corresponding items in the two tables serves to show the variation in the final results arrived at by computations of this kind.

**STATISTICS OF FUEL CONSUMPTION AND COST PER ANNUM OF PRODUCER
GAS POWER AND OTHER POWERS**

Type of Engine	Kind of Fuel	Price per Ton	Fuel consumption in lbs. per B. H. P. per hour	Cost in cents per B. H. P. per hour	Cost in dollars per 100 B. H. P. per hour	Cost per 100 B. H. P. per Annum 3000 hours	Gas and Oil Engines Steam Eng. Prod. Gas Eng.
Oil Engine	Gasoline	12 cents per Gallon	1 Pint	1.50	\$1.50	\$4,500	
Gas Engine	Illuminating Gas	75c per 1000 Cubic Feet	18 Cubic Feet per B. H. P.	1.35	1.35	4,050	
Gas Engine	Natural Gas	30c per 1000 Cubic Feet	13 Cubic Feet per B. H. P.	.39	.39	1,170	
Simple Steam Engine...	Bituminous Coal	\$3.00	8 lbs.	1.2	1.20	3,600	
Compound Steam Engine Non-condensing, ...	Bituminous Coal	3.00	5	.75	.75	2,250	
Triple Expanding Steam Engine, condensing...	Bituminous Coal	3.00	2	.3	.30	900	
Producer Gas Engine...	Anthracite Coal	5.00	1	.25	.25	750	
Producer Gas Engine...	Gas Coke	3.00	1.25	.1875	.1875	565	
Producer Gas Engine...	Bituminous Coal	2.50	1.25	.1565	.1565	470	
Producer Gas Engine...	Anthracite Coal	3.00	1	.15	.15	450	

C. H. Day,* in an investigation on the economy of gas producer engines, compiled the following fuel cost data for the various kinds of prime movers mentioned. The fuel consumption figures were obtained in each case by collaborating the results of a considerable number of tests. The costs per annum were then computed by assuming the cost of unit weight of the fuel. The total time of operation per year is taken at 3000 hours.

**ANNUAL FUEL COSTS FOR STEAM ENGINES. COST OF COAL ASSUMED AT
\$3.00 PER TON OF 2000 LBS.**

Type of Engine	B. H. P.	Lbs. Coal per B. H. P. per hour	Cost of Coal per B. H. P. per year
Simple, non-condensing	50	5.75	\$25.90
Simple, condensing	100	4.45	20.00
Compound, condensing	200	2.74	12.70
Compound, condensing	600	1.97	8.85
Compound, condensing	1000	1.90	8.55
Triple exp. condensing	2000	1.87	8.40

The data available for *steam turbines* is not as extensive as

*Sibley College Thesis, 1905.

that for steam engines, and the range covered is not so wide as regards capacity. The following figures, also compiled by Mr. Day, show some fuel costs for prime movers of this kind. The working time has again been assumed at 3000 hours per annum.

ANNUAL FUEL COSTS FOR STEAM TURBINES. COST OF COAL ASSUMED AT \$3.00 PER TON OF 2000 LBS.

Number of Tests	Average B. H. P.	Lbs. Coal per B. H. P.-hour	Cost of Coal per B. H. P. per year
27	616	1.740	\$7.83
10	1085	1.735	7.80
5	1359	1.725	7.75
25	1739	1.655	7.44

The computations made on *illuminating* and *natural gas* engines show the following results. Time of operation per year 3000 hours, cost of illuminating gas 75 cents per 1000 cu. ft., cost of natural gas 50 cents per 1000 cu. ft. The latter assumption is high, since in many localities natural gas is sold at 30 or even 20 cents per 1000 cu. ft. It is a simple matter, however, to reduce the figures in the table in the corresponding ratio.

ANNUAL FUEL COST FOR ILLUMINATING AND NATURAL GAS ENGINES. COST OF ILLUMINATING GAS ASSUMED AT 75 CENTS, THAT OF NATURAL GAS AT 50 CENTS, PER 1000 CU. FT.

B. H. P.	Cu. ft. per B. H. P.-hour		Annual Cost of Gas per B. H. P. per year of 3000 hours	
	Illuminating Gas	Natural Gas	Illuminating Gas Engines	Natural Gas Engines
10	22.0	—	\$49.50	\$24.00
20	21.5	16.0	48.30	23.50
30	21.0	15.5	47.20	23.30
40	20.5	15.0	46.00	22.60
50	20.0	14.5	44.80	21.80
75	19.1	13.8	42.80	20.80
100	18.3	13.0	41.10	19.60
200	—	11.4	—	17.10
300	—	10.2	—	15.40
400	—	9.7	—	14.60
500	—	9.4	—	14.10

For *producer gas* engines Day found the following figures for the consumption of gas and of coal, from which the fuel costs per year of 3000 hours are computed, assuming coal to cost \$4 per ton of 2000 pounds.

ANNUAL FUEL COST OF PRODUCER GAS ENGINES

B. H. P.	Cu. ft. of Gas per B. H. P.-hour	Lbs. of Coal per B. H. P.-hour	Annual Cost of Coal per B. H. P.
50	105	1.45	\$8.70
100	96	1.35	8.10
150	89	1.23	7.33
200	83	1.17	7.03
250	79	1.12	6.72
300	76	1.08	6.48
400	73	1.05	6.30
500	72	1.03	6.18

The cost of the gas for *blast furnace* gas engines has in many computations been neglected on the assumption that this gas, if not used in engines, is a mere waste product. Lately, however, it has come to be recognized that some value must be assigned to this fuel since money was expended on it in every case for cleaning it preparatory to making it fit for use in engines. The ordinary method of evaluating this gas is to compare its heat content with that of steam and to assign a value to the gas corresponding to the cost of steam. Thus the cost of the gas will in every locality vary with the cost of coal. As an example, H. Freyn * makes the following computation:

"Let us assume that the price of coal delivered into bins at the plant be \$2.75 per ton, that the coal have a heat value of 13,000 B. T. U. per pound, and, further, that steam of 150 pounds boiler pressure or about 165 pounds absolute pressure be raised by burning this coal under boilers. One pound of steam will then contain 1225 B. T. U. from zero degrees Fahrenheit. Assuming feed water at 70 degrees, there would be required 1155 B. T. U. to generate 1 pound of steam at 150 pounds boiler pressure. In a boiler plant having 65 per cent efficiency, 1000 pounds of coal

* H. Freyn, Available Power and Cost of Operation of a Power Station for Waste Gases from a Blast-furnace Plant. Journal Western Society of Engineers, February, 1906.

could raise $(0.65 \times 1000 \times 13,000) \div 1155 = 7300$ pounds of steam. The value of 1000 pounds of steam would be $2.75 \div (2 \times 7.3) = \$0.188$, or 18.8 c. To this must be added for labor and maintenance approximately 1 c. per 1000 pounds of steam, making the total value of 1000 pounds = 19.8 c. 1000 cu. ft. of blast furnace gas have a heat value of $1000 \times 90 = 90,000$ B. T. U. and are equivalent to $(0.65 \times 90,000) \div 1155 = 51$ pounds of steam, which in turn are worth $(51 \div 1000) \times 19.8 = 1$ c.

"The value of 1000 cu. ft. of blast furnace gas would, therefore, be 1 c."

L. Eberhardt, in an article in the *Zeitschrift d. V. d. I.*,* makes a similar computation, arriving at a somewhat different result. Starting with the assumption that 1000 pounds of steam will cost from 25 to 32 c., according to the price and quality of coal, he finds that 1000 cu. ft. of cleaned blast furnace gas should have a value of from 1.43 to 1.84 c., the gas having a heating value of 102 B. T. U. per cu. ft. This is considerably higher than the value found by Freyn, which is mainly due to the lower grade of coal (11,650 B. T. U. per pound) and the higher grade of gas assumed.

To get some idea of the fuel cost of blast furnace gas power as compared with steam power, Eberhardt, in the article mentioned, gives the following tables. The original tables give the cost per B. H. P. hour, but these have been recomputed to the basis of a year of 3000 working hours, to make them directly comparable with the figures in previous tables.

ANNUAL FUEL COST OF BLAST-FURNACE GAS ENGINES. HEATING VALUE OF COAL TAKEN AT 11650 B. T. U. PER LB. HEATING VALUE OF GAS TAKEN AT 102 B. T. U. PER CU. FT.

Cost of coal per ton of 2000 lbs.....dollars	2.34	2.75	3.18
Cost of 1000 lbs. of steam cents	24.8	28.4	31.8
Cost of 1000 cu. ft. of gas cents	1.43	1.64	1.84

A fair blast furnace gas engine will show the following fuel consumption:

At full load	99 cu. ft. of gas per B. H. P. hour.		
At $\frac{9}{10}$ load	106	"	"
At $\frac{8}{10}$ load	113	"	"
At $\frac{7}{10}$ load	122	"	"
At $\frac{1}{2}$ load	131	"	"

* June 3, 1905.

With these assumptions of gas consumption per B. H. P., and of local cost of coal, the fuel costs per B. H. P. per year of 3000 hours will then be as per the following table:

Cost of Coal per Ton	Per Cent full Load on Engine				
	100	90	80	66	50
	Annual Cost of Blast Furnace Gas per B. H. P. Dollars				
\$2.34	4.26	4.56	4.87	5.25	5.63
2.75	4.85	5.18	5.55	5.97	6.42
3.18	5.46	5.83	6.23	6.72	7.20

The fuel cost of operating on *gasoline* as fuel is of course considerably higher than any of the figures above quoted. The usual assumption made in regard to gasoline engines is that they will require 1 pint of gasoline per B. H. P. hour. This corresponds to a thermal efficiency of about 18 per cent on the brake, a figure which should be reached by a fair-sized engine in good condition. For the smaller machines, however, one pint is perhaps somewhat low, and for say a 2 horse-power machine a consumption of 2 pints per B. H. P. hour is probably a safer assumption. It should also be borne in mind that the purchase price of gasoline varies somewhat with the quantity bought. The following table of fuel costs takes these various points into account.

FUEL COST FOR GASOLINE ENGINES PER B. H. P. PER YEAR OF 3000 WORKING HOURS

Size of Engine, B. H. P.	2	4	6	10	20
Cost of gasoline per gal., cents	20	18	16	14	13
Consumption of gasoline per B. H. P.					
hr., gallons	.25	.20	.18	.15	.12
Fuel cost per B. H. P. per year, dollars	150.00	108.00	86.40	63.00	46.80

(e) COST OF WATER FOR COOLING AND WASHING. Many estimates of total operating costs totally neglect the cost of water for cooling and washing purposes required by the plant, although in many localities this may amount to a considerable item of expense. In general terms, where water has to be brought from city mains, it pays to install cooling apparatus of some kind. Of course for small installations this may be simply a tank from

the surface of which the heat is radiated. The tank is connected at top and bottom with the water jacket of the engine, and the water circulates by convection. Where such a cooling tank cannot be placed in the immediate vicinity of the engine, and connected to the jacket by short straight pipes, a circulating pump of some kind must be used. As the size of the installation grows, cooling towers or cooling ponds have to be resorted to, unless an abundant source of clean water is available without cost, except that of pumping. Under such conditions the cost of cooling water is very materially reduced. With any kind of cooling system the cost then consists of the cost of pumping plus the cost of any clean water that must be supplied from time to time to replace that lost by evaporation. Of course it is not possible to give any definite figures for this cost item, because it depends upon the cost of water in the particular locality, upon the total amount of water circulated, and upon the kind of pumps used. A few figures for the amount of water required or in circulation per B. H. P. hour are given below.

In a producer plant an additional amount of water is required for the producers and scrubbers. The water vaporized for the producers of course is not again available. That used for the washing plant is in general subject to the same considerations as that used for cooling the engines. But where a cooling system is used for both, they should never be combined, because the requirements for clean cooling water for the engines are much more strict. The scrubber water usually carries considerable quantities of sediment and is contaminated with ammonia and sulfur compounds, so strongly in some cases as to give it a very noxious odor. In such cases, settling tanks and ponds, together with a considerable addition of clean water, seem to be the only remedies to recover at least a part of the water.

The approximate quantity of cooling water required may be figured from the following considerations:

One horse-power, assuming an engine efficiency of say 25 per cent at the cylinder, requires the expenditure of $\frac{2545}{.25} = 10,200$ B. T. U. per I. H. P. hour. The jacket loss approximates about 40 per cent of the heat expended, that is, in this case the jacket water must carry off $.40 \times 10,200 = 4080$ B. T. U. per hour.

Assuming the inlet temperature at 70 degrees, the outlet at 150 degrees, we find the number of pounds of water required = approx. $\frac{4080}{150-70} = 51$ pounds = say $6\frac{1}{2}$ gallons per I. H. P. hour at full load.

The following figures from various sources show how this estimate agrees with others and with some data from actual practice.

W. Heym, in the *Gasmotorentechnik*,* states that a producer plant requires per horse-power hour 6.5 gallons for the engine, 4 gallons for the scrubber, and .13 gallons for the producer vaporizer.

Freyn, in the paper already mentioned, makes the following estimates for a blast furnace gas plant of 10,500 B. H. P. Theissen washers are used to clean the gas.

GALLONS OF COOLING WATER REQUIRED

Load on Engines	For the Engines per B. H. P.-hour	For the Washers per 1000 cu. ft. of gas cleaned per minute
Full	8.5	12.0
$\frac{3}{4}$	10.5	14.0
$\frac{1}{2}$	13.0	16.0

Freyn also estimates that if the plant is located near a stream of water, the cost of pumping would probably be in the neighborhood of 2 c. per 1000 gallons.

J. R. Bibbins† on a 51-hour test of a 500 horse-power Westinghouse horizontal engine, reported a water consumption for the engines only of 9.4 gallons per B. H. P. hour at full load. The same authority reports a figure of 5.65 gallons per B. H. P. hour‡ found on similar engines in another plant.

In conclusion it should be said that the water-consumption of gas engines depends somewhat upon the size of the cylinder. Thus a single-acting cylinder of very large diameter usually

* November, 1907.

† J. R. Bibbins, Proceedings A. S. M. E., Mid-November, 1907.

‡ J. R. Bibbins, Gas Driven Electric Power Station, Proc. of the Eng. Soc. of W. Pa.

requires more cooling water than two single-acting or one double-acting cylinder of the same capacity. Skilful attendance also is a considerable factor in the amount of cooling water used, a point which is very often neglected.

(f) OIL AND WASTE. The consumption of lubricating oil per B. H. P. hour depends upon the size of the engine and, as in the case of cooling water, largely also upon the care of attendants. It is also true that new machines may for a time require two or three times the ordinary amount of oil until all the parts have become adjusted to their service. In large and medium sized plants it is usual to employ a gravity or circulating system of oiling, in which case the oil is recovered, filtered, and again used. In such systems the cost of oil is a small item and consists mostly of the cost of oil necessary to replace that unavoidably lost. Of course none of the cylinder oil used is recovered.

Güldner estimates that the consumption of lubricating oil usually varies from .006 to .008 pints of oil per B. H. P.-hour, and that under favorable conditions .003 pints per B. H. P.-hour may be reached. L. L. Brewer finds that the consumption for the large engines quoted in the table, page 552, varies from .0045 to .0055 pints per B. H. P.-hour, and that in double-acting two-cycle engines the consumption may be as low as .0035 pints.

The following table compiled by J. R. Bibbins and published in the paper before the Society of Engineers of West Pennsylvania, already quoted, shows the actual oil consumption in a plant containing two horizontal Westinghouse engines of 500 horse-power each, direct connected to 300 K. W. generators. The figures cover a period of four months and the results should therefore be very reliable. The system of oiling used is of the continuous circulating and filtering type. The consumption amounts to .00202 pints of cylinder oil and .00286 pints of engine oil per horse-power hour, which agrees well with the figures given by Brewer.

OIL CONSUMPTION

4 months ending Sunday, May 23, 1906

	CYLINDER OIL *	ENGINE OIL †
Kind	Imperial Cylinder Oil	Special Gas Engine Oil
Maker	Union Petroleum Co., Philadelphia	
Price (barrel lots)	32 cents per Gal.	18 cents per Gal.
Quantity used	561 Gals.	765 Gals.
Quantity used per month	140½ Gals.	191½ Gals.
Quantity used per (opertg.) day ...	4.68 Gals.	6.38 Gals.
Quantity used per full day	6.07 Gals.	8.27 Gals.
Engine Hours per day	37	37
Engine H.P. Hours per day	18500 *	18500
Oil per Engine Hour	0.127 Gals.	0.172 Gals.
Oil per H.P. Hour	0.000253 Gals.	0.000345 Gals.
Cost per Engine Hour	4.05 cents	3.1 cents.
Cost per H.P. Hour	0.00809	0.00621
Total Cost		0.0143 cents per H.P. Hour

The cost of cotton waste or other cleaning material is an item very hard to estimate, and in any case of little influence on the final result. It is usually combined with the cost of oil, and some figures for this combined cost will be found in the estimates of total operating costs at the end of this chapter.

(g) ATTENDANCE. The statement very often carelessly made with regard to small gas engine installations, that they require no waiting on, is of course not quite the truth. The fact is that a small gas or gasoline engine, if in good order, does not require, after starting, much attending except the proper handling of the lubricators. In this respect the gas power installation has an advantage over the small steam power plant, where at least one man is generally required all the time. For small natural gas or illuminating gas engines, therefore, it is possible to employ the "engineer," to use a familiar term, somewhere else at least part of the time, but naturally, as the size of engine increases, this spare time decreases, and it is doubtful if suction gas plants, no matter how small, can get along with less than one man's entire time.

The general methods of taking care of medium sized and large

*Includes crank case oil for exciter engines. Drainage from glands of main engines collected, mixed with old engine oil and used in crank case. No crank case oil purchased.

†Includes oil consumption of auxiliaries. Sufficient quantity old engine oil drawn from circulating system to supply auxiliaries replaced by fresh oil.

gas engines are perhaps not far different from those used in steam engine practice. It cannot be denied, however, that gas engines have much more opportunity, so to speak, to go wrong. That is, jacket water, lubrication, igniters, valves, etc., all need careful looking after, and any of these things, if neglected, may cause a shut down. While, therefore, steam engine attendants generally accustom themselves to the new service very quickly, it would be wrong to assume that they can in general take care of a gas engine plant without some instruction and practice. Of course the larger the plant, the more important this point becomes.

There seems to be nothing in English engineering literature to give an approximate idea of the time actually required in attending gas engines. Güldner proposes the following equations, apparently based on practical experience:

For illuminating gas, natural gas and oil engines,

$$W = .25 \sqrt{N_n} \text{ hours.}$$

For suction gas plants $W = 1.25 \sqrt{N_n}$ hours

where W = time of attendance required in hours per day of 10 hours and N_n = rated capacity of plant.

Thus, for instance, a 100 horse-power natural gas engine would actually require about $.25 \sqrt{100} = 2.5$ hours in a 10-hour shift. Hence the attendant's time could be largely used somewhere else. On the other hand, a 100 horse-power suction gas plant would actually require $1.25 \sqrt{100} = 12.5$ hours, which means that two men will have to be employed.

Where a station is made up of several smaller units and producers, the cost of attendance naturally increases. Thus if there are n units in a plant of N_n horse-power total, Güldner states that the actual time required per operating day of the plant may be expressed by

$$W = 1.25 \sqrt{n N_n} \text{ hours.}$$

Thus if we take a 1200 horse-power suction gas plant made up of 4 units of 300 horse-power each,

$$W = 1.25 \sqrt{4 \times 1200} = 86.5 \text{ hours,}$$

which would mean the service of 9 men in the 10-hour shift. If

this capacity had been put in one unit, the time would have been

$$W = 1.25 \sqrt{1200} = 43.5 \text{ hours,}$$

which would have called for only 5 men per shift.

If in any of the above equations we introduce the hourly wage scale, we may express the cost of attendance per B. H. P. per hour. Thus if the scale should be 20 c. per hour, we would have for illuminating gas, natural gas, or oil engines, for the 10-hour shift,

$$\text{Cost per B. H. P.-hour} = \frac{20 \times .25 \sqrt{N_n}}{10 N_n} = \frac{.50}{\sqrt{N_n}} \text{ cents.}$$

For suction gas plants,

$$\text{Cost per B. H. P.-hour} = \frac{20 \times 1.25 \sqrt{N_n}}{10 N_n} = \frac{2.5}{\sqrt{N_n}} \text{ cents}$$

and for large suction gas plants of n units,

$$\text{Cost per B. H. P.-hour} = \frac{20 \times 1.25 \sqrt{n N_n}}{10 N_n} = \frac{2.5 \sqrt{n}}{\sqrt{N_n}} \text{ cents.}$$

There is little data available to check the accuracy of these formulæ, but the two or three instances cited by various authorities for the labor cost in suction gas plants check very well with the results of the formula.

(h) MAINTENANCE AND REPAIRS. The expenditures for maintenance and repairs of a gas engine installation should not exceed 3 per cent of the purchase price of the engines and producers. There is no doubt that this item may easily run to 10 per cent and over, especially if in a producer plant insufficient attention is paid to cleaning the gas. But such conditions are not normal and should not occur in good practice.

Freyer states that the following are average figures for repair accounts for a blast furnace gas station:

2½ per cent per year of the price of the engines and electric generators.

7 per cent per year of the price of the cleaning plant.

5 per cent per year of the price of the air compressor used for starting.

2 per cent per year of the price of piping, etc.

To this may be added from 1 to 2 per cent of the cost of the building as maintenance for the building.

Total Operating Costs and Cost as Compared with other Prime Movers — As in the case of fuel costs, the information available in engineering literature on total operating costs is of two kinds. In the first kind the estimates are based entirely upon hypothetical assumptions, in the second the results are those obtained in actual operation. Again, the latter information is of course of much greater value, but except for large installations little of this class has been made public. Especially as regards small installations of gasoline, oil and illuminating gas engines there seems to be an almost absolute lack of data referring to total operating costs. For that reason, in order to get some idea of costs, it will be necessary for the time being to depend upon cost computations based on assumptions. In any concrete case these assumptions should be carefully scanned to see how they agree with actually existing conditions before forming a final idea of operating cost.

The data available from actual practice is mainly due to Mr. J. R. Bibbins* and others, of the Westinghouse Machine Company, and of course relates to Westinghouse engines. This, however, does not preclude their applicability to other similar plants. This information, combined with that from a few English plants, is about all that can be cited at the present writing.

Hypothetical computations are nearly all made on a comparative basis. Of this kind are the tables published by W. O. Webber in the *Engineering News*, August 15, 1907. It is not easy to give a concise abstract of the various factors assumed in the computation of these tables, and for that reason they are given in full, except for electric power which does not especially interest us here.

* The following is a partial list of the valuable papers published:

J. R. Bibbins, Gas Driven Electric Power Systems, Warren & Jamestown Street Railway, Eng. Soc. of W. Pa.; Gas Power for Central Stations, Am. Inst. of E. E., New York, December 18, 1903; Producer Gas Power Plant, Proc. A. S. M. E., December, 1906; Application of Gas Power to Central Station Work, National Elec. Light Assoc., Washington, D. C., June, 1907; Duty Test on Gas Power Plant, Proc. A. S. M. E., Mid-November, 1907; and A. West, Gas Power in Electric Railway Work, American Street & Interurban Railway Assoc., Philadelphia Convention, 1905.

E. E. Arnold, The High Power Internal Combustion Engine and its Fitting for Central Station Work, New England Street Railway Club, March, 1904.

Reprint from Power, December, 1903, A Gas Engine Pumping Station.

COST OF INSTALLATION AND OF OPERATION 569

COST OF GASOLINE POWER

Size of plant in H.P.	2	6	10	20
Price of engine in place	\$150.00	\$325.00	\$500.00	\$750.00
Gasoline per B. H. P. per hour..	$\frac{1}{2}$ gal.	$\frac{1}{4}$ gal.	$\frac{1}{2}$ gal.	$\frac{1}{2}$ gal.
Cost per gallon	\$0.22	\$0.20	\$0.19	\$0.18
= cost per 3,080 hours	\$451.53	\$924.00	\$975.13	\$1,386.00
Attendance at \$1 per day	308.00	308.00	308.00	308.00
Interest, 5 per cent.	7.50	16.25	25.00	37.50
Depreciation, 5 per cent	7.50	16.25	25.00	37.50
Repairs, 10 per cent	15.00	32.50	50.00	75.00
Supplies, 20 per cent	30.00	65.00	100.00	150.00
Insurance, 2 per cent	3.00	6.50	10.00	15.00
Taxes, 1 per cent	1.50	3.25	5.00	7.50
Power cost	\$824.03	\$1,371.75	\$1,498.13	\$2,016.50

To these figures should be added charges on space occupied as follows:

Value of space occupied	\$100.00	\$150.00	\$200.00	\$300.00
Interest, 5 per cent.	\$5.00	\$7.50	\$10.00	\$15.00
Repairs, 2 per cent	2.00	3.00	4.00	6.00
Insurance, 1 per cent	1.00	1.50	2.00	3.00
Taxes, 1 per cent	1.00	1.50	2.00	3.00
Total annual charge for space	\$9.00	\$13.50	\$18.00	\$27.00
Total cost per annum.	\$833.03	\$1,385.25	\$1,516.13	\$2,043.30
Cost of 1 H.P. per annum, 10 hour basis.	416.51	239.87	151.61	102.17
Cost of 1 H.P. per hour	\$0.1352	\$0.0780	\$0.0492	\$0.0331

COST OF GAS POWER

\$1.50 per 1000 cubic feet of gas less 20 per cent, if paid in 10 days = \$1.20
net, gas 760 B. T. U.

Size of plant in H.P.	2	6	10	20
Engine cost in place	\$200.00	\$375.00	\$550.00	\$1,050.00
Gas per H.P.-hour in cubic feet.	30	25	22	20
Value of gas consumed, 3080 hours	\$221.76	\$554.40	\$843.12	\$1,478.00
Attendance, \$1 per day	308.00	308.00	308.00	308.00
Interest, 5 per cent.	10.00	18.75	27.50	52.50
Depreciation, 5 per cent.	10.00	18.75	27.50	52.50
Repairs, 10 per cent	20.00	37.50	55.00	105.00
Supplies, 20 per cent	40.00	75.00	110.00	210.00
Insurance, 2 per cent	4.00	7.50	11.00	21.00
Taxes, 1 per cent	2.00	3.75	5.50	10.50
Power cost	\$615.76	\$1,023.65	\$1,387.62	\$2,237.50
Annual charge for space	9.00	13.50	18.00	27.00
Total cost per annum.	\$624.76	\$1,037.15	\$1,405.62	\$2,264.50
Cost of 1 H.P. per annum, 10 hour basis.	312.38	172.86	110.56	143.22
Cost of 1 H.P. per hour	\$0.1014	\$0.0561	\$0.0456	\$0.0367

COST OF STEAM POWER

Size of plant in H.P.	6	10	20
Cost of plant per H.P.	\$250.00	\$220.00	\$200.00
Fixed charge, 14 per cent	\$35.00	\$30.80	\$28.00
Coal per H.P.-hour, in pounds	20	15	12
Cost of coal at \$5 per ton	\$154.00	\$103.00	\$82.50
Attendance, 3080 hours	75.00	50.00	30.00
Oil, waste and supplies	15.00	10.00	6.00
Cost 1 H.P. per annum, 10-hour basis	\$279.00	\$194.80	\$146.50
Cost of 1 H.P. per hour	\$0.0906	\$0.0832	\$0.0475

ANNUAL COST OF POWER PER BRAKE HORSE-POWER *

B. H. P. of Unit	Steam	Gas	Gasoline
1	\$600.00	\$380.00	\$487.50
2	500.00	312.50	416.00
3	437.50	260.00	350.00
4	375.00	220.00	300.00
5	320.00	192.50	262.50
6	280.00	172.50	240.00
7	250.00	160.00	210.00
8	230.00	152.50	182.50
9	210.00	145.00	165.00
10	195.00	140.00	152.00
12	175.00	132.50	137.50
14	165.00	126.00	122.00
16	157.50	120.00	112.50
18	150.00	116.50	107.50
20	146.00	113.00	102.00
22	140.00	110.00	98.00
24	137.50	107.50	95.00
26	133.00	105.00	92.50
28	130.00	102.50	90.00
30	127.50	102.00	87.50
35	124.00	100.00	85.00
40	120.00	98.00	82.50
50	112.50	96.00	80.00
60	105.00	94.00	78.00
70	100.00	92.00	76.00
80	95.00	90.00	74.00
90	90.50	88.00	72.00
100	86.40	86.00	70.00

Attention is called in these tables to the high allowance for repairs in the case of the gasoline and gas engine and the high coal consumption assumed for the steam engine.

Another interesting comparison is that made by H. A. Clark

* Unit costs: Coal, \$5 per ton; gas \$1.20 per 1000 cubic feet, at 760 B. T. U.; gasoline, \$0.20 per gallon.

in a paper on the Diesel Engine.* In this case the computations are carried out for three sizes of Diesel engine, Crossley Gas Engine and Dowson producer, and high-speed compound condensing steam engine. The conditions assumed are partly as follows, the remainder being given in the table itself.

Cost of fuel delivered:

Diesel engine, crude oil at \$10.90 per ton = appr. 3.8 c. per gallon.

Crossley engine, anthracite at \$5.70 per ton.

Steam engine, coal at \$3 per ton.

The fuel cost is based on the following assumption:

For the Diesel, the consumption has been taken in each case as the mean between full and half-load rates. These rates were actually determined by tests.

For the Crossley, 1.5 to 1.25 pounds of coal per B. H. P. hour plus an allowance for stand-by losses.

For the steam engine, 4 to 3.5 pounds of coal per B. H. P. hour for the lowest and highest powers, assuming an evaporation of 8 pounds of water per pound of coal.

The cost figures in the original table have all been transposed to dollars and cents. The computations are based on a year of 2700 working hours.

In the discussion on Clark's paper, the cost computations were rather severely criticised, mainly on account of the fuel costs assumed. The claim was made that both the coal consumption for the steam engine and the cost of the steam coal was assumed too high, and that the cost of oil at 3.8 c. per gallon could only apply to seaboard towns. As far as American conditions are concerned, the assumption regarding the steam engine would seem to be about right, while the cost of anthracite at \$5.70 per ton for the gas engine is certainly not too low. On the other hand, the price of 3.8 c. per gallon for the crude oil would seem to favor the Diesel engine. It all comes to the point that applies to all computations of this kind, and that is that the results are strictly applicable only to the locality for which they are computed. Allowing for possible difference in the labor costs, however, it seems to the writer that the comparison is quite fair as between the steam and the gas engine.

* Proc. of M. E., 1903, II.

TOTAL COSTS IN POWER PRODUCTION

	Diesel Engine 35	Gas Engine 35	Steam Engine 35	Diesel Engine 80	Gas Engine 80	Steam Engine 80	Diesel Engine 160	Gas Engine 160	Steam Engine 160
1. Means of production	12x20x15	12x20x15	12x20x15	16x24x20	16x24x20	14x24x20	20x27x20	20x27x20	17x27x20
2. Brake horse-power	—	10x20	—	10x18x15	11x24	15x46x20	12x26	12x26	12x26
3. Size of engine house, ft.	—	8x16x12	—	10x18x15	11x24	15x46x20	10x20x16	10x20x16	20x48x20
4. Ground for gas holder, ft.	—	\$ 107.00	\$ 146.00	\$ 73.00	\$ 141.00	\$ 194.00	\$ 104.00	\$ 199.00	\$ 272.00
5. Size of boiler house and gas producer, ft.	—	631.00	1115.00	681.00	925.00	1750.00	875.00	1140.00	2040.00
6. Cost of land at \$1.70 per square yard	49.00	73.00	291.00	98.00	136.00	389.00	170.00	194.00	487.00
7. Cost of buildings	437.00	—	—	—	—	—	—	—	—
8. Engine foundations, etc., boiler settings	49.00	—	—	—	—	—	—	—	—
9. Engines, tanks, piping and condenser, etc.	\$2890.00	1360.00	1165.00	4370.00	3080.00	2430.00	8170.00	5550.00	3890.00
10. Boiler or gas producer and accessories	—	974.00	1408.00	—	1260.00	1700.00	—	1845.00	2910.00
11. Total capital cost	3425.00	3145.00	4125.00	5222.00	5544.00	6463.00	9319.00	8928.00	9589.00
12. Capital cost per B. H. P.	97.90	90.00	117.70	65.25	69.30	80.80	58.20	55.80	59.90
ANNUAL CHARGES	\$	\$	\$	\$	\$	\$	\$	\$	\$
13. Interest on capital, 4 per cent	135.20	126.50	165.50	209.00	222.00	259.00	373.00	357.00	384.00
14. Building, maintenance and depreciation, 5 per cent	21.90	31.60	55.90	34.10	43.80	87.50	46.20	57.20	104.00
15. Engines and machinery, maintenance and depreciation, 10 per cent	284.50	136.20	116.80	438.00	309.00	243.00	817.00	554.00	389.00
16. Boilers, Maintenance and depreciation, 12 per cent	—	—	170.50	—	—	204.00	—	—	350.00
17. Gas producer and holder. Depreciation, 5 per cent	—	48.70	—	—	63.20	—	—	—	—
18. Fuel	231.00	438.00	632.00	477.00	1000.00	1265.00	904.00	1670.00	2140.00
19. Lubricant, waste, etc.	97.40	97.40	97.40	170.50	170.50	170.50	268.00	268.00	268.00
20. Wages, removal of ash, etc.	365.00	438.00	642.00	365.00	462.00	535.00	426.00	535.00	680.00
21. Total	1133.00	1316.40	1700.00	1693.60	270.50	2764.00	2834.20	3535.50	4315.00
22. Annual cost per B. H. P.	32.40	37.60	48.70	21.19	28.38	34.55	71.72	22.09	26.30
23. Cost per B. H. P. hour, cents	1.08	1.38	1.78	.78	1.04	1.26	.64	.80	.98

As a final example of this kind of hypothetical computation, the following table is given. The figures are due to Mr. L. G. Findlay of the De La Vergne Machine Company of New York. They were presented at the Dayton meeting of the Ohio Society of Mechanical, Electrical, and Steam Engineers, and were published in *Power*, July, 1907. The table is very complete and would repay careful study. Especially interesting is the effect of the load factor on operating costs. This factor is here defined as the average load on the plant divided by the rated capacity of the plant.

In Clark's tables above no mention is made of the load factor and it is evidently assumed to be 100 per cent. That, however, is a condition existing and maintained in very few plants. In fact, if it were, it would be poor engineering, as it gives the plant but very little overload capacity. The marked effect that a decrease of the load factor from 100 per cent to 50 per cent has upon operating costs is very clearly brought out in the table. In the tables given below for actual cost data, the load factor is in most cases even lower than this.

In the discussion on the results of the table following its presentation, the steam men objected to the comparison between the steam and gas plants at the rated load, on account of the much greater overload capacity of the steam engine plant. Lines 36, 37, 38, and 39 of the table are computed on the assumption that 80 per cent of the steam used by the engines is available for heating, but no standing charges are made against the heating plant.

Turning next to data from actual practice, the first of the following tables was given by Mr. J. E. Dowson in a paper in the *Journal of the Institute of Electrical Engineers* April, 1904. The table refers to results obtained in two English plants; the first is equipped with Dowson generators and Crossley engines, the second with the same type of producer and Westinghouse vertical engines of comparatively small size. In spite of the higher cost of coal, the greater capacity of the Walthamstow plants brings down the wage and repair items enough to make the total works costs not far different. The table only gives works cost, and it should be remembered that the results apply to English conditions.

19	Interest, 5 per cent. on total cost	1.17	0.85	0.85	0.92	0.60	0.70
20	Total of lines 17, 18, 19	3.75	2.72	2.72	2.95	1.92	2.24
21	Coal used in B. H. P.-hr. in lb.	2.00	2.20	4.75	1.10	0.90	0.90
22	Cost of coal per B. H. P. per day of 24 hours	4.80	5.28	11.40	2.64	4.86	4.86
23	Attendance of engine B. H. P. per day of 24 hours	1.20	2.40	3.00	6.00	1.05	2.10
24	Attendance of fuel plant per day of 24 hours	0.65	1.30	1.75	3.50	1.50	3.75
25	Oil, waste and supplies	0.50	0.80	0.50	0.80	0.50	0.70
26	Total daily expense per B. H. P. per 24 hour day	7.15	9.78	16.65	26.52	4.94	7.90
27	Yearly oper. expenses 309 days per B. H. P., dollars	22.09	30.22	51.45	81.95	15.26	24.41
28	Cost of coal per B. H. P. per day of 10½ hours	2.05	2.25	4.86	6.46	1.13	1.54
29	Attendance of engine B. H. P. per day of 10½ hours	0.70	1.40	1.50	3.00	0.40	0.80
30	Attendance of fuel plant per day of 10½ hours	0.28	0.56	0.88	1.75	0.45	0.90
31	Oil, waste and supplies per day of 10½ hours	0.27	0.49	0.26	0.49	0.27	0.46
32	Total daily expenses per day of 10½ hours	3.30	4.70	7.50	11.70	2.25	3.70
33	Yearly oper. expenses 309 days per B. H. P., dollars	10.20	14.52	23.17	36.15	6.95	11.43
34	Yearly expenses lines 10, 20, 27 = 24-hr. day	28.86	43.76	56.26	91.57	22.87	39.63
35	Yearly expenses lines 10, 20, 33 = 10½-hr. day	16.97	28.06	27.98	45.77	14.56	26.55
36	Cost of power, all exhaust steam used for heating	24 -hr. day.	37.61	61.25			
37	Cost of power, same as line 36 but short run	10½-hr. day.	19.07	30.68			
38	Cost of power, 50 per cent exhaust steam for heating	24 -hr. day.	42.41	69.56			
39	Cost of power, same as line 38 but short run	10½-hr. day.	21.11	34.90			
	Total cost, lines 6 + 16 for entire plant	\$51,000.00	\$7,200.00	\$61,500.00	\$13,400.00	\$56,500.00	\$12,500.00
	Water evaporated per lb. coal, at 212°	10.00	8.00				

The second and third tables following are due to Mr. J. R. Bibbins of the Westinghouse Machine Company and were published in a paper on the Application of Gas Power to Central Station Work, read before the National Electric Light Association at Washington, June, 1907. Both of the stations mentioned use natural gas for fuel, but the Bradford plant is equipped with rather small Westinghouse vertical engines, while the Warren and Jamestown station contains two large direct connected units of 500 horse-power each. In both cases the load factor is considerably less than 50 per cent; in the case of the Bradford plant it is only about 19 per cent. The figures for the latter plant are remarkable for the length of time covered, 8½ years.

WORKS COST OF OPERATING GAS PLANTS

	Plant	
	Midland Railway Co., Leicester	Urban District Station, Walthamstow
Engine:		
Capacity tested	300	1500
Number	6	7
Make	Crossley	Westinghouse
Length of run	5 months	12 months
Load factor	—	15.25
K. W.-hour generated	200497	659756
Remarks — service	Arc and incandescent lighting, Dowson gas.	Arc and incandescent lighting, Dowson gas.
Duty of plant: *		
Lbs. per K. W.-hour	3	2
Lbs. per B. H. P.	2.25	1.5
Cost:		
Coal per ton	\$3.75	\$6.50
Fuel per K. W.-hour	.123c.	.17c.
Oil, waste, water	.020c.	.095c.
Wages	.292c.	.015c.
Total works cost power, cents per K. W.-hour	.473c.	.445c.
Authority:	R. M. DEELY, Supt. Locomotives.	F. A. WILKINSON, Elec. Engineer in charge.

WORKS COST OF POWER — GAS DRIVEN CENTRAL STATION

BRADFORD ELECTRIC LIGHT & POWER CO., BRADFORD, PA.

Station capacity (five engines)	470 kilowatts
Number of years in operation	8.5
Average station load factor ('03-'06)	19.09 per cent
Average gas consumption per kilowatt-hour ('03-'06)	25.5 cu. ft.
Average heat efficiency at switchboard	14.14 per cent

* Including "stand-by" losses.

Works Cost of Power	Dollars per year. Average 8 years	Cents per Kilowatt-hour. Average 4 years
Fuel, gas, including heating	\$3,121	0.307
Station wages	3,068	0.410
Oil, waste and supplies	454	0.054
Running repairs (total)	848	0.085
Running repairs gas engines	285	0.034
Total operating cost	\$7,776	0.890

Engine repairs — \$0.36 per horse-power per year.

57.00 per engine per year.

.75 per cent on investment.

Fuel — Bradford natural gas.

WORKS COSTS — GAS POWER RAILWAY PLANT

WARREN & JAMESTOWN STREET RAILWAY COMPANY

Capacity of plant (two units)	600 kilowatts
Average time operated per day	18.5 hours
Average output per day	4115 kilowatt hours
Average load, per cent rating	37 per cent

Works Cost of Power	Dollars per day	Cents per Kilowatt-hour
Average First Four Months, 1906:		
Fuel gas	\$12.97	0.315
Wages	12.37	0.300
Oil	2.63	0.064
Repairs and miscellaneous supplies ..	3.29	0.080
Total	\$31.26	0.759

Fuel — "Bradford Sand" natural gas.

In conclusion, to make the figures in the above tables readily comparable among themselves, the cost data has been recomputed to the basis of cost in cents per B. H. P. hour in all cases.

Table I gives the total operating costs for small and medium sized plants as computed from the data of Webber & Clark. The load factor is in all cases 100 per cent. The difference in the cost of steam power arrived at by the two authorities is remarkable, Clark's figures being less than 50 per cent those of Webber. The reason for this is found in two directions, different assumptions as to cost of coal, \$5 per ton as against \$3, and different assumptions as to the coal consumption per horse-power. Webber

assumes as high as 20 pounds of coal per horse-power for a 6 horse-power plant, and charges a 20 horse-power plant up with 12 pounds per horse-power-hour. In fact Webber's figures are very liberal throughout for all three kinds of power.

In Table II, the works and total operating costs per B. H. P.-hour are shown side by side as computed from Findlay's table.

TABLE I
Total Operating Costs per B. H. P.-hour
Small and Medium Sized Plants

Kind of Power	Kind of Fuel	Load Factor	Working hours per year	Size of Plant												Authority
				5	10	16	20	24	30	35	50	80	100	160		
				Cost in cents per B. H. P.-hour												
Steam Engine.	Coal	100	3080	13.9	6.3	5.1	4.7	4.5	4.1	4.0	3.7	3.1	2.8	Webber		
Gas Engine ..	Ill. Gas	100	3080	6.2	4.6	3.9	3.6	3.5	3.3	3.2	3.1	2.9	2.8	Webber		
Gas Engine ..	Gasoline	100	3080	8.5	4.9	3.7	3.3	3.1	2.8	2.7	2.6	2.4	2.3	Webber		
Steam Engine.	Coal	100	2700							1.78		1.26	.98	Clark		
Gas Engine ..	Prod. Gas	100	2700							1.38		1.04	.80	Clark		
Oil Engine, Diesel	Crude Oil	100	2700							1.08		.78	.64	Clark		

TABLE II
Total Operating and Works Cost per B. H. P.-hour.
Computed from Table by Mr. L. G. Findlay.
Works costs in brackets.

Hours in Operation per day	Load Factor	Size of Plant B. H. P.	Kind of Power		
			Steam	Bituminous Producer Gas	Anthracite Producer Gas
			Cents per B. H. P.-hour		
24	100	1000	.39 (.30)	.31 (.21)	.39 (.30)
24	50	1000	.69 (.41)	.54 (.33)	.67 (.49)
10.25	100	1000	.53 (.33)	.46 (.22)	.55 (.33)
10.25	50	1000	.88 (.47)	.84 (.36)	.99 (.56)
24	100	200	.76 (.69)	.40 (.29)	.43 (.38)
24	50	200	1.23 (1.10)	.71 (.49)	.86 (.66)
10.25	100	200	.88 (.73)	.59 (.33)	.69 (.45)
10.25	50	200	1.44 (1.15)	1.11 (.59)	1.26 (.78)

The data from the Bradford and Jamestown plants and from the two English plants shows the following results for the works cost per B. H. P.-hour.

Fuel	Load Factor	B. H. P. rated	Works Cost per B. H. P.-hour cents	Plant
Natural gas	19.09	640	.67	Bradford
Natural gas	37.0	800	.57	Jamestown
Dowson gas	—	400	.35	Midland Railway
Dowson gas	15.25	2000	.33	Walthamstow

From these figures, and those of Table II, the conclusion seems justified that it should be easily possible to produce one brake horse-power in a fair sized plant, say not under 200 horse-power, for a works cost of from .5 to .75 cents per hour, depending upon the conditions involved. This excludes fixed charges and assumes a load factor in the neighborhood of 50 per cent.

INDEX

A

	PAGE
Abeille carbureter	189
Absolute temperature	6
Accumulators or storage batteries	411
Acetylene, constants for	211
Adiabatic and isothermal changes, graphical expressions for	55
Adiabatic change	16
expansion, work performed in	53
line, equation of	52
Admixture of benzol to alcohol	184
After-burning	220
Air gas, production of	147
Air required for combustion of alcohol	183
Air required for combustion	136
Air thermometer	8
Alcohol, commercial, table of specific gravities and heating values	183
Alcohol, air required for combustion of	183
Alcohol, composition and heating value of	181
Alcohol engine	390
vapor air mixtures	203
Alcohol, denatured	183
Alcohol vaporizer, Altman	198
Deutz	198
Dresden	201
Duerr	201
Swiderski-Longuemarre	199

PAGE

Alcohol, vaporizing devices for ..	196
Allis-Chalmers gas engine	345
Altman alcohol vaporizer	198
American Crossley suction producer	167
Atomizing or spraying carbureters	187
Atomizer, Hornsby-Akroyd	193
Attendance, cost of	565
Automatic cut-off engine, Jacobson	271
Automobile engine, Continental	373
Franklin	372, 374
Moore	374
Horch	373
horse-power rating of	483
Automobile gasoline engine	372
Auto-sparker	416
Auxiliaries and piping, cost of ..	548
Auxiliary spark gap	409

B

Barnett's engine	236
ignition cock	392
Barsanti and Matteucci free piston engine	239
Batteries, primary and secondary, method of connecting	420
Beau de Rochas	243
Beau de Rochas or Otto cycle, theoretical	65
Benzol	184
admixture of to alcohol ...	184
Blast furnace gas, composition and heating value of	209

	PAGE		PAGE
Blast furnace gas engines, fuel costs for	561	Centrifugal governors for hit-and-miss regulation	454
Blast furnace gas, preparation of	210	Charging of storage batteries ..	414
value of	560	Classification of fuels	146
Bomb calorimeter, Mahler's ..	129	heat engines ..	17
Brayton cycle, theoretical	69	internal combustion engines ..	27
engine	248	Classification of producers	158
engine diagram	250	Cleaning of blast furnace gas for use in engines	210
oil engine, test of	250	Clearances and compression pressures for various fuels, Otto cycle, table of	90
Brake horse-power, definition of ..	38	Clerk's engine	255
Brake, Prony	39	Clerk engine diagram	257
British Thermal Unit	14	engines, tests on	257
Brown, engine of	234	Clerk's experiments on specific heat	224
Bruce-Merriam-Abbott engine ..	275	Clerk-Lanchester starter	432
Buckeye two-cycle engine, governing of	467	Closed cycle, definition of	61
Buckeye two-cycle gas engine ..	286	Cockerill engines	336
Buffalo tandem engine	282	Code of German Society of Engineers, for testing gas producers and gas engines, ..	511
Buildings and floor-space, cost of ..	549	Coefficient, excess, definition of, ..	137
C			
Calorie	14	Coefficient of fly-wheel regulation	439
Calorific intensity	144	Coefficient of governor regulation ..	440
Calorimeter, Carpenter's coal ..	130	Coke oven gas, constants for ..	208
Junker's gas	131	Cold gas efficiency	151
Mahler's bomb	129	Cooling water conditions affecting economy	530
Calorimetric thermometers	11	Combining weights and volumes, for gases	127
Campbell oil-engine, governor for	456	Combination gasoline and alcohol vaporizer	202
Carbon, heating value of	132	Combination producers	173
Carbon monoxide, heating value of	132	Combination producer, Crossley ..	173
Carbureter, atomizing or spraying	187	Deutz double zone	173
Carbureter, bubbling, type of ..	185	Loomis-Pettibone	174
Daimler	188	Combination systems of governing	447
DeDion	190	Combustion, air required for ..	136
Gautier	191	Combustion line, Otto cycle ..	90
Sintz	188	Combustion of alcohol, air required	183
surface	186		
Carbureters	185		
Carnot cycle	62		
Carnot or reversible engine	54		
Cells, wet and dry	410		

	PAGE		PAGE
Combustion, pressure, due to ..	220	Constant volume, transfer of	
products of	136	heat at	50
Comparative cost of power for		Constants for acetylene	211
various prime movers, 569-579		coke oven gas ...	208
Comparison of theoretical and		Constants, gas, table of	126
actual heat engines	61	Constants for gas engine fuel	
Comparison of various theoret-		gases, table of	213
ical cycles	73	Constants for illuminating gas..	206
Composition and heating value		natural gas	212
of alcohol	181	water-gas	212
Composition and heating value		Conversion of solid fuels to gas	146
blast furnace gas	209	Coal calorimeter, Carpenter ...	130
Composition and heating value		Cost of attendance	565
of gasoline	179	Cost of erection	548
Composition and heating value		Cost of floor-space and build-	
of kerosene	179	ings	549
Composition of oil gas	207	Cost of fuel for illuminating gas	
Composition of most common		and natural gas engines	558
commercial gases, graphi-		Cost of power for various prime	
cal representation	214	movers	569-579
Composition of producer gas by		Cost of producers and engines .	547
volume per pound of car-		oil and waste	564
bon gasified	152	operation	533
Composition of producer gas by		piping and auxiliaries..	548
weight per pound of car-		Cost of water for cooling and	
bon gasified	151	washing	561
Compression, effect on economy	532	Costs, fuel	553
Compression pressures and tem-		operating	568
peratures for Otto cycle,		Crosby indicator	34
table of	88	Crossley combination producer .	173
Compression pressures and clear-		gas engine	324
ances for various fuels,		Crossley hit-and-miss governor.	452
Otto cycle, table of	90	vaporizer	194
Compression stroke, Otto cycle	86	Crouse-Hinds distributor	425
Compression, temperature due to,	220	double-ball timer.	408
Condensers in spark coils	404	Crude oil	178
Conditions required for the prop-		Crude oil distillates	180
er formation of alcohol		Crude oils, table of heating value	
vapor-air mixtures	203	and composition	178
Constant pressure engines	30	Current, sources of	410
cycle	104	Cycle, Brayton, theoretical	69
Constant pressure, transfer of		Carnot	62
heat at	50	choice of best	79
Constant temperature engines .	31	closed, definition of	61
Constant volume or explosion		constant pressure	104
engines	28	definition of	3

	PAGE		PAGE
Cycle, Diesel, theoretical	71	Design, general features of	263
Otto, combustion line	90	Determination of excess coefficient from exhaust gas-analyses	143
Otto, compression stroke ..	86	Deutz alcohol vaporizer	198
Otto, cyclic efficiency of ...	68	Deutz double-zone combination producer	173
Otto, exhaust stroke	100	Deutz gas engine	351
Otto, expansion line	97	pressure producer	162
Otto, pressure ratio in ...	92	suction producer	165
Cycle, Otto, requirements for best efficiency in	100	Development of the Diesel engine	259
Cycle, Otto, suction stroke	84	Development of the gas engine industry	258
Cycle, Otto, table of allowable compression pressures and clearances for various fuels ..	90	Diagram, Clerk engine	257
Cycle, Otto, table of compression pressures and temperatures	88	Diagram, entropy, graphical construction of	120
Cycle, Otto, table of cyclic efficiencies	69	Diagram, entropy, interpretation of	119
Cycle, Otto, table of volumetric efficiencies and suction pressures	86	Diagram, entropy, mathematical construction of	112
Cycle, Otto, typical lower loop diagrams	101	Diagram of Lenoir engine	242
Cycle, Theoretical Beau de Rochas or Otto	65	Diagram pressure volume, defined	3
Cycle, two-stroke	102	Diagrams, Diesel engine	106
Cycles, theoretical, comparison .	73	indicator, forms of	40
Cyclic efficiency, definition of ..	62	lower loop, Otto cycle	101
Cyclic efficiency of theoretical Otto cycle	68	Otto cycle	94
Cyclic efficiencies for Otto cycle, table of	69	Diesel cycle, theoretical	71
		engine	385
		Diesel engine diagrams	106
		Diesel engine, development of ..	259
		governor for	459
		Diesel engines, early, tests on ..	262
		Dissociation, theory of	98
		Distributor, Crouse-Hinds	425
		high tension	423
		Leavitt	426
		Dresden alcohol vaporizer	201
		Dürr alcohol vaporizer	201
		Dynamos and magnetos	415
		E	
		Economist crude oil vaporizer .	195

D

Daimler carbureter	188
Dashboard spark coil	406
DeDion carbureter	190
Delamar hit-and-miss governor ..	454
De La Vergne two-cycle oil engine	380
Denatured alcohol	183
Denaturizing agents for alcohol, table of	184
Depreciation of plant and buildings	553

E

Economist crude oil vaporizer .	195
---------------------------------	-----

	PAGE		PAGE
Efficiency, cyclic, definition of..	62	Engine economy, as affected by	
Efficiency, cyclic, of theoretical		cooling water conditions	530
Otto cycle	68	Engine economy as affected by	
Efficiency, hot gas	151	piston speed	530
cold gas	151	Engine economy as affected by	
Efficiencies for Otto cycle, table of	69	variation in fuel mix-	
Ehrhardt and Sehmer engine,		ture	533
governor for	461	Engine economy depending upon	
Ejector muffler	428	load	537
Electrical thermometers	8	Engine economy depending upon	
starters	436	point of ignition	534
Energy, kinetic	2	Engines, constant pressure	30
Engine, Barnett	236	constant temperature	31
Brayton	248	Engines, constant volume or ex-	
Brayton oil, test of	250	plosion	28
Brown	234	Engines, Diesel, tests on	262
Carnot or reversible	54	heat, classification of	17
Cayley's	26	Engines, heat, comparison of	
Clerk	255	theoretical and actual ..	61
Clerk, tests on	257	Engines, hot-air	22
Engine diagram, Clerk	257	Engines, internal combustion,	
Brayton	250	classification of	27
Engine diagrams, Diesel	106	Entropy	15, 107
Engine, Diesel, development of	259	Entropy diagram, graphical con-	
furnace gas	25	struction of	120
Engine, free piston, Otto and		Entropy diagram, interpretation	
Langen	244	of	119
Engine, gas, method of opera-		Entropy diagram, mathematical	
tion	26	construction of	112
Engines, gunpowder	233	Entropy relations, general	109
Engine, Hugon	242	Erection, cost of	548
Engine indicator	32	Ericsson hot-air engine	22
Engine, Lebon	234	Excess coefficient, definition of .	137
Lenoir	239	Excess coefficient from exhaust	
Lenoir, diagram of	242	analysis	143
Engine, Lenoir, gas consump-		Expansion line, Otto cycle	97
tion of	242	Experiments on specific heat by	
Engine, Otto	251	Clerk	224
Engines, Otto, tests on	254	Experiments on specific heat by	
Engine, Papin	232	Langen	220
Perry	238	Experiments on specific heat by	
Robert Street	233	Mallard and LeChatelier	220
Stirling hot-air	24	Exhaust gas analysis, excess co-	
Wright	235	efficient from	143
Engine economy as affected by		Exhaust gas, computations on .	138
compression	532	stroke, Otto cycle	100

	PAGE		PAGE
Explosibility of fuel mix- tures	215, 218	Fuel costs for illuminating gas and natural gas engines ..	558
Explosion or constant volume engines	28	Fuel costs for producer gas en- gines	559
Explosion recorder, Mathot	500	Fuel costs for steam engines ...	557
Explosion, time of	227	steam turbines ..	558
Explosive mixture	215	Fuel gases, constants for	213
		Fuel mixture, computations on.	138
		Fuel mixture, variation in, af- fecting economy	533
		Fuel mixtures, explosibility of ..	215
		Fuels, classification of	146
		liquid, heating value of ...	134
		solid, heating value of	135
		Furnace gas engine	25
		Fusion thermometers	11
		G	
		Gas and oil engines, tests of, Code of A.S.M.E.	487
		Gas calorimeter, Junker's	131
		Gas, coke oven, constants for ..	208
		Gas constants R for perfect gases, table of	48
		Gas constants, table of	126
		Gas consumption of Lenoir en- gine	242
		Gas, blast furnace, composition and heating value of	209
		Gas, illuminating, constants for,	206
		natural, constants for	212
		oil, composition of	207
		oil, heating value of	208
		perfect, specific heat of ...	12
		Gas producers and engines, tests of	542
		Gas producers in practice	156
		Gas, water, constants for	212
		Gases, combining weights and volumes	127
		Gases, commercial, composition of	214
		Gases, fuel, constants for	213
		Gases, illumination, table of composition	207
F			
Fairbanks engine	288		
Fairbanks, Morse & Co., engine.	277		
Fairbanks-Morse crude oil vapor- izer	389		
Fairbanks-Morse suction pro- ducer	168		
Fay and Bowen make-and-break igniter	400		
Felten and Guillaume electric starter	436		
Fixation of tar-forming gases in producer gas	157		
Flame propagation, velocity of .	227		
Floor-space and buildings, cost of	549		
Fly-wheel regulation, coefficient of	439		
Fly-wheel weights, table of	440		
Foos gasoline engine	358		
Forms of indicator diagrams ...	40		
Formula for mean effective pres- sure, Grover's	472		
Four-cycle gas engine, Koerting.	280		
Four-cycle gasoline engine, Stre- linger	362		
Four-terminal spark coil	405		
Franklin automobile engine, 372,	374		
Free-piston engine, Barsanti and Matteucci	239		
Free-piston engine, Otto and Langen	244		
Fuel costs	553		
Fuel costs for blast furnace gas engines	561		
Fuel costs for gasoline engines..	561		

	PAGE		PAGE
Gases, perfect, characteristics of .	45	Gas engines, blast furnace, fuel	
Gases, perfect, laws of	46	costs for	561
Gases, perfect, table of specific		Gas engines, illuminating and	
volumes	47	natural, fuel costs for . . .	558
Gases, specific heat of	48	Gas engines, method of opera-	
Gas engine, Allis-Chalmers	345	tion	26
Bruce-Merriam-Abbott	275	Gas engines, methods of testing	486
Buckeye two-cycle	286	producer, fuel costs for . . .	559
Buffalo tandem	282	small and medium size . . .	265
Crossley	324	Gasification in pressure produc-	
Deutz	351	ers, rate of	177
Diesel	385	Gasoline, composition and heat-	
Fairbanks, ...	288	ing value of	179
Fairbanks, Morse & Co.	277	Gasoline, mixing devices for . .	185
furnace	25	Gasoline engine, automobile . .	372
Hautefeuille	232	Foos	358
Gas engines, Jacobson	267	Lozier two-cycle	364
Gas engine, Koerting four-		Olds	360
cycle	280	Standard marine	367
Koerting two-cycle	313	Strelinger four-cycle	362
Nürnberg	339	Gasoline engines :	358
Gas engine, Nürnberg, table of		fuel costs for	561
standard sizes	346	marine	361
Gas engine, Oechelhäuser	355	Gautier carbureter	191
Olds	292	Generators, current, mechanical	
Philadelphia Otto	291	forms of	415
Premier	347	German Society of Engineers,	
Riverside	320	Code for testing gas pro-	
Sargent	310	ducers and gas engines	511
Gas engines, Snow	329	Gibbon kerosene vaporizer . . .	193
Gas engine, Tod	306	Governor, Campbell oil engine .	456
Warren hit-and-miss	295	Crossley hit-and-miss :	452
Westinghouse	266	Delamare hit-and-miss	454
Westinghouse horizontal	304	Diesel engine	459
Gas engine, Westinghouse verti-		Governor, Ehrhardt and Sehmer	
cal single-acting tandem, 306		engine	461
Gas engine governing	439	Governor, Hornsby-Akroyd en-	
Gas engine industry, develop-		gine	458
ment of	258	Governor, Koerting four-cycle	
Gas engine regulation	439	engine	460
tests, tables of	544, 545	Governor, Nürnberg engine . . .	458
Gas engines and producers, cost		Governor regulation, coefficient of	440
of	547	Governor, Robey	455
Gas engines and gas producers,		Springfield hit-and-miss . . .	453
results of tests	542	Governor, Westinghouse vertical	
Gas engines, Cockerill	336	engine	460

	PAGE		PAGE
Governing Buckeye two-cycle engine	467	Heat at constant volume, transfer of	50
Governing of gas engines	439	Heat balance	539
Governing, hit-and-miss system	444	Heat, definition of	3
Governing of Koerting engine ..	468	Heat engines, classification of ..	17
Governing of Oechelhäuser engine	467	Heat engines, theoretical and actual, comparison of ..	61
Governing of two-cycle engines ..	449	Heat, mechanical equivalent of ..	15
Governing, Letombe system of ..	465	relation of to entropy	54
Reinhardt's method	464	Heat unit	14
Reichenbach engine	463	Heating value and composition for true explosive mixtures from liquid fuels, table of	217
Governing small two-cycle engine	467	Heating value and composition of alcohol	181
Governing systems	441	Heating value and composition of blast furnace gas	209
Governing by varying time of ignition	449	Heating value, definition of ...	129
Governing by varying the quality of the fuel mixture ..	444	Heating value of carbon	132
Governing by varying the quantity of fuel mixture	446	Heating value of carbon monoxide	132
Governors, pendulum, for hit-and-miss regulation	450	Heating value of crude oils, table of	178
Governors, centrifugal, for hit-and-miss regulation	454	Heating value of gasoline	179
Governors, mechanical details of, ..	450	hydrogen	132
Graphical construction of the entropy diagram	120	kerosene	179
Graphical expressions for adiabatic and isothermal changes	55	liquid fuels ..	134
Graphical representation of composition of most common commercial gases	214	oil gas	208
Grover's formula for mean effective pressure	472	solid fuels ...	135
Güldner's method of determining horse-power	477	Heating values and specific gravities of commercial alcohol, table of	183
Gunpowder engines	233	Heating values of hydrocarbons, table of	132
H		Heating values of true explosive mixtures, table of	216
Hammer break ignition	398	High tension distributor	423
Hautefeuille, Abbé, gas engine of ..	232	jump-spark system	423
Hay vaporizer	190	Hit-and-miss engine, Jacobson ..	268
Heat at constant pressure, transfer of	50	Warren	295
		Hit-and-miss governor, Crossley ..	452
		Delamare	454
		Springfield	453
		Hit-and-miss system of governing	442

	PAGE
Horch automobile engines	373
Horizontal gas engine, Westing-house	304
Horsnby-Akroyd atomizer	193
Hornsby-Akroyd engine, governor for	458
Hornsby-Akroyd oil engine	376
Horse-power, brake, definition of	38
Horse-power defined	1
Horse-power, determination from mean effective pressure..	472
Horse-power from standard air reference diagram	476
Horse-power, Guldner's method	477
Horse-power, indicated, definition of	38
Horse-power rating of automobile engines	483
Hot-air engine, Ericsson	22
Stirling	24
Hot-air-engines	22
Hot gas efficiency	151
Hot tube igniter, Koerting	395
Newton's	238
Hot tube ignition	394
with timing valve	395
Hugon engine	242
Hydrocarbons, table of heating values of	132
Hydrogen, lower and higher heating value of	132
Hyperbola, methods of drawing	43

I

Ignition	392
Ignition by electric spark	397
heat of compression	396
hot tube	394
open flame	392
Ignition cock, Barnett's	392
Ignition, hammer break	398
jump-spark	401
make-and-break	397
Ignition, variation in, affecting economy	534

	PAGE
Igniter, Fay and Bowen make-and-break	400
Igniter, hot tube, Newton's	238
Koerting, hot tube	395
Koerting open flame	393
Illuminating gas, constants for.	206
Illuminating gases, table of composition of	207
Index for expansion and compression lines, method of finding	115
Indicated horse-power, definition of	38
Indicators, engine	32
Indicator, Crosby	34
optical	35
Tabor	34
Thompson	33
Indicator diagrams, forms of...	40
Inertia governors for hit-and-miss regulation	450
Internal combustion engines, classification of	27
Isothermal line, equation of...	52
Isothermal and adiabatic changes, graphical expressions for	55
Isothermal change	16
Isothermal expansion, work performed in	53

J

Jacobson automatic cut-off engine	271
Jacobson gas engines	267
hit-and-miss engine...	268
throttling engine	272
Jump-spark ignition	401
Jump-spark and make-and-break systems compared	410
Jump-spark system, high tension	423
Junker's gas calorimeter	131

K	PAGE	PAGE
Kerosene, composition and heating value of	179	Lower loop diagrams, Otto cycle 101
Kerosene vaporizer, Gibbon ...	193	Lozier two-cycle marine gasoline engine
Kinetic energy	2	Lunckenheimer mixing valves ..
Koerting engine, governing of ..	468	M
four-cycle gas engine ..	280	Magneto, action of
Koerting four-cycle engine, governor for	460	Magnetos and dynamos
Koerting hot tube igniter	395	Magnetos, systems of wiring employing
open flame igniter ...	393	Mahler's bomb calorimeter
suction producer	166	Maintenance and repairs
pressure producer	161	Make-and-break and jump-spark systems compared
Koerting suction producer for peat	172	Make-and-break ignition
Koerting two-cycle engine	313	Mallard and LeChatelier's experiments on specific heat ..
L		Manograph
Lacoste timer	407	Marine gasoline engine, Lozier two-cycle
Langen's experiments on specific heat	220	Marine gasoline engine, Standard ..
Laws of perfect gases	46	Marine gasoline engines
Leavitt distributor	426	Mathematical construction of the entropy diagram
Lencauchez double-zone suction producer	170	Mathot explosion recorder
Lencauchez suction producer ..	170	Matteucci and Barsanti free piston engine
Lenoir engine	239	Mean effective pressure, Grover's formula
diagram of	242	Mean effective pressure, tables for determining
gas consumption of	242	Mechanical details of governors ..
Letombe system of governing ..	465	equivalent of heat ..
Lebon's engine	234	forms of generators ..
Limits of explosibility for fuel mixtures made from different fuels	218	Method of connecting up primary and secondary batteries
Limit of piston speed	471	Method of drawing hyperbola ..
Liquid fuel engines	358	Method of finding index for expansion and compression lines
Liquid fuels, heating value of ...	134	Method of governing, Reinhardt's
Liquid fuels, mixing devices for ..	185	Method of operating gas engines
Loomis-Pettibone combination producer	174	26
Lowe system for making oil gas ..	196	
Low tension system of wiring ..	421	
Lower and higher heating value of hydrogen	132	

	PAGE
Method of test, Code of 1901, brake horse-power.....	497
Methods of test, Code of 1901, calibration of instru- ments	488
Methods of test, Code of 1901, computation of temper- atures	503
Methods of test, Code of 1901, duration of tests	492
Methods of test, Code of 1901, heat balance	502
Methods of test, Code of 1901, heat units consumed ...	493
Methods of test, Code of 1901, indicated horse-power...	494
Methods of test, Code of 1901, indicator diagrams	501
Methods of test, Code of 1901, measurement of fuel	493
Methods of test, Code of 1901, measurement of jacket water	494
Methods of test, Code of 1901, speed determination	498
Methods of test, Code of 1901, standards of economy...	501
Methods of test, Code of 1901, starting and stopping tests	493
Mietz and Weiss oil engine	383
Mixing devices for gasoline	185
liquid fuels ..	185
Mixing valves, Lunkenheimer ..	187
Mond pressure producer	163
Mond process for making pro- ducer gas	163
Moore automobile engine	374
Morgan pressure producer	160
Muffler, ejector	428
Powell	428
Mufflers	426

N

Natural gas, constants for	212
----------------------------------	-----

	PAGE
Newton's hot tube igniter	238
Non-trembler spark coil	402
Nürnberg engine	339
governor for	458
Nürnberg gas engine, table of standard sizes	346

O

Oechelhäuser engine, governing of	467
Oechelhäuser gas engine	355
Oil and gas engines, tests, of, Code of A.S.M.E.	488
Oil and waste, costs of	564
Oil, crude, distillates	180
crude	178
Oils, crude, table of heating value and composition ..	178
Oil engine, Brayton, test of	250
De La Vergne two-cycle ...	380
Hornsby-Akroyd	376
Mietz and Weiss	383
Priestman	388
Oil engines	375
Oil gas, composition of	207
heating value of	208
Lowe system for making ..	196
Olds gas engine	292
Olds gasoline engine	360
Open flame igniter, Koerting ..	393
ignition	392
Operating costs	568
Operation, cost of	533
Optical indicator	35
pyrometers	9
Otto or Beau de Rochas cycle, theoretical	65
Otto cycle, combustion line ...	90
compression stroke.....	86
diagrams, typical	94
exhaust stroke	100
expansion line	97
pressure ratio in	92

	PAGE		PAGE
Otto cycle, requirements for best efficiency in	100	Piston speed, effect on economy, 530	
Otto cycle, suction stroke.....	84	limit of.....	471
Otto cycle, table of allowable compression pressures and clearances for various fuels	90	Pittsfield timer	407
Otto cycle, table of compression pressures and temperatures	88	Poetter pressure producer	162
Otto cycle, table of cyclic efficiencies	69	Powell muffler	428
Otto cycle, table of volumetric efficiencies and suction pressures	86	Power of gas engines	471
Otto cycle, theoretical, cyclic efficiency of	68	Premier gas engine	347
Otto cycle, two-stroke.....	102	Pressure after combustion	220
Otto cycle, typical lower loop diagrams	101	producers.....	159
Otto engine	251	producer capacities ...	176
engines, early, tests on ...	254	Pressure producers, rate of gasification.....	177
Otto and Langen free piston engine	244	Pressure producer, Deutz	162
Otto-Langen free piston engine, table of tests	247	Koerting	161
Otto-Langen free piston engine, typical diagram	248	Mond	163
Otto gas engine, Philadelphia ..	291	Morgan	160
		Poetter.....	162
		Taylor	159
		Wile	160
		Pressure ratio in the Otto cycle ..	92
		volume diagram defined ..	3
		Priestman oil engine	388
		vaporizer	194
		Prime movers, comparative cost of power for	569-579
		Producers and gas engines, cost of	547
		Producers, classification of	158
		combination	173
		Producer, combination, Crossley ..	173
		Producer, combination, Deutz	173
		Producer, combination, Loomis-Pettibone	174
		Producer details	175
		gas	149
		Producer gas, composition of by volume per pound of carbon gasified	152
		Producer gas, determination of weight and volume per pound of carbon	151
		Producer gas engines, fuel costs for.....	559
		Producer gas, fixation of	157
		gas, Mond process ...	163
P			
Papin's engine	232		
Pendulum or inertia governors for hit-and-miss regulation.....	450		
Perfect gases, characteristics of ..	45		
laws of	46		
Perfect gas, specific heat of	12		
Perfect gases, table of specific volumes	47		
Perry's engine.....	238		
Petreano surface carbureter....	187		
Philadelphia Otto engine	291		
Piping and auxiliaries, cost of ..	548		

PAGE	PAGE
Table of heating values of hydrocarbons	132
Table of heating values of true explosive mixtures	216
Table of relative fly-wheel weights	440
Table of specific gravities and heating values of commercial alcohol	183
Table of specific heats	13
Table of specific heats for perfect gases	48
Table of specific volumes of perfect gases	47
Table of standard sizes Nürnberg gas engine	346
Table of tests on producer plants; thermometric scales . . .	5
Table of true explosive mixtures, for commercial gases	216
Table of vapor tension for alcohol and water	204
Table of various gas constants . .	126
Table of volumetric efficiencies and suction pressures for Otto cycle	86
Tabor indicator	34
Tandem gas engine, Buffalo . . .	282
Westinghouse	306
Tar-forming gases, fixation of . .	157
Taylor pressure producer	159
Temperature, absolute	6
Temperature after combustion . .	220
Temperature, definition of	3
Temperatures and compression pressures for Otto cycle, table of	88
Temperature scales	5
Test on Brayton oil engine	250
Tests of Clerk engines	257
Tests of engines and gas producers, results of	542
Tests of gas engines	544, 545
on early Diesel engines	262
on early Otto engines	254
Tests of gas and oil engines, Code of A.S.M.E. for 1901	487
Tests on producer plants, table .	546
Testing of gas engines, methods for	486
Testing of storage batteries . . .	414
Theoretical and actual heat engines, comparison of . . .	61
Theoretical Brayton cycle	69
cycles, comparison of	73
Diesel, cycle	71
Theoretical Otto cycle, cyclic efficiency of	68
Theoretical Otto or Beau de Rochas cycle	65
Theoretical yield of producer . .	153
Theory of dissociation	98
stratification	97
Thermal unit, British	14
Thermo dynamics, second law of	55
Thermo-element	8
Thermometer, air	8
Thermometers, ...	6
calorimetric	11
electrical	8
fusion	11
vapor	11
Thermometric scales, table of . .	5
Thermopyle	8
Thompson indicator	33
Three-port two-cycle engine . . .	366
Three-terminal spark coil	405
Throttling engine, Jacobson . . .	272
engines, Warren	295
Time of explosion	227
Timer, Crouse-Hinds double-ball .	408
Lacoste	407
Pittsfield	407
Sintz	406
Timing valve	395
Timers	405
Tod gas engine	306
Total operating costs	568
Transfer of heat at constant pressure	50

	PAGE		PAGE
Transfer of heat at constant volume	50	Vaporizer, Crossley	194
Trembler spark coil	403	crude oil, Economist	195
True explosive mixture	215	Vaporizer, crude oil, Fairbanks-Morse	389
True explosive mixtures, for commercial gases, table of	216	Vaporizer, Deutz alcohol	198
Two-cycle engine, governing of	449, 467	Dresden alcohol	201
Koerting	313	Gibbon kerosene	193
three-port	366	Priestman	194
Two-cycle gas-engine, Buckeye	286	W. Hay	190
Two-cycle marine gasoline engine, Lozier	364	Vaporizing devices for alcohol	196
Two-cycle oil engine, De La Vergne	380	Vaporizing devices for crude oil and kerosene	192
Two-stroke Otto cycle	102	Variation of fuel consumption with load	554
Typical computations on fuel mixtures and exhaust gases	138	Variation of specific heat with temperature	220
Typical diagram, Otto-Langen free piston engine	248	Velocity of flame propagation	227
Typical Diesel engine diagrams	106	Vertical gas engine, Westinghouse	306
Typical lower loop diagrams, Otto cycle	101	Volume of scrubber in suction plants	176
Typical Otto cycle diagrams	94	Volumetric efficiencies and suction pressures for Otto cycle, table of	86
U		W	
Unit, British thermal	14	Warren hit-and-miss engine	295
V		throttling engines	295
Value of blast furnace gas	560	Waste and oil, cost of	564
Vapor tension for alcohol and water, table of	204	Water gas, constants for	212
Vapor thermometers	11	production of	147
Vaporizer, alcohol, Dürr	201	Wiring, low tension system of	421
Vaporizer, alcohol, Swiderski-Longuemar	199	Weights, fly-wheel, table of	440
Vaporizer, Altman alcohol	198	Westinghouse gas engine	266
Vaporizer, combination alcohol and gasoline	202	Westinghouse horizontal gas engine	304
		Westinghouse vertical engine, governor for	460
		Westinghouse vertical single-acting tandem gas engine	306
		Wet and dry cells	410
		Wile pressure producer	160
		Wiring employing magnetos, systems of	422

INDEX

597

	PAGE		PAGE
Wiring, systems of	420	Wright's engine	235
Work performed in adiabatic expansion	53		
Work performed in isothermal expansion	53	Y	
Work, rate of	1	Yield of producer, theoretical ..	153

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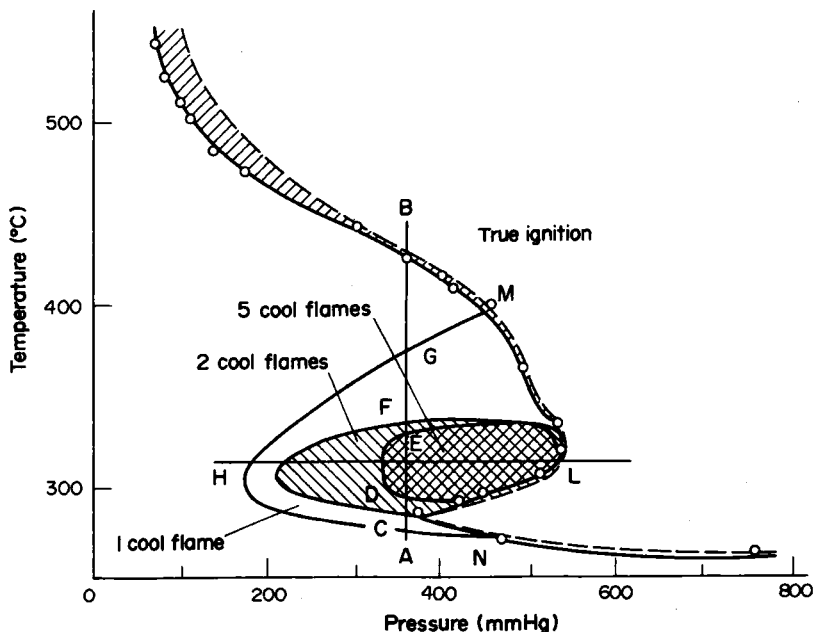


FIG. 5.6. Spontaneous ignition temperatures of a $C_3H_8 + O_2$ mixture. (From Hewitt and Thornes,⁽⁵⁾ by courtesy of the Chemical Society.)

These results were obtained by Hewitt and Thornes.⁽⁵⁾ We shall examine the conditions along a constant pressure line (AB) corresponding to a pressure of 360 mm. Below 275°C slow oxidation reactions are observed; between 275-285°C a faint luminosity develops after several minutes and remains until the reaction is nearly complete. At 290°C (point A) the luminosity is followed by a pale blue "cool" flame of low intensity which starts near the

centre of the vessel and spreads outwards giving rise to a pressure pulse. After the passage of the flame, the luminosity persists for several seconds. Between 300°C (D) and 340°C (E) the luminosity appears immediately on filling the vessel and is succeeded by three to five "cool" flames at intervals of several seconds, each of which traverses the whole vessel before its extinction gives rise to a pressure pulse. Between 340°C (E) and 345°C (F) only two cool flames, and between 345°C (F) and 385°C (G) only one cool flame, are observed. The intensity of the cool flames is increased as the number diminishes. Above 350°C the single cool flame becomes less intense as the reaction temperature is raised; on the other hand, the general luminosity increases in intensity until at 425°C (B) a bright blue flame appears. At a slightly higher temperature this changes to the characteristic yellow flame associated with true ignition.

Outside the cool flame and ignition region the chemical reactions proceed at a rate corresponding to the pressure and temperature.

In traversing the cool flame region to the true ignition point there may be several seconds delay between the appearance of a cool flame and true ignition at the point L, e.g. along an isothermal HL. The region NHM is called the cool flame peninsula. In this experiment the hydrocarbon-oxygen reaction exhibited two types of auto-ignition, a low temperature ignition-cool flames and a high temperature ignition-normal flames. To distinguish between the two types of ignition the region below 350-400°C is called the low temperature region and above 350-400°C the high temperature region. The chemical mechanism for the reactions in the two regions is different. Not all hydrocarbon-oxygen mixtures exhibit the two types of ignition and not all mixtures which exhibit cool flames generate five cool flames; the number varies from one to five. Hydrocarbon-oxygen mixtures with low temperature and high temperature ignition regions include the higher paraffinics, such as n-heptane and iso-octane, to which we shall refer to later.

When a hydrocarbon-oxygen mixture has only one ignition temperature at a given pressure this temperature is in the high temperature region. The ignition curves for these mixtures are monotonic functions of pressure. Typical hydrocarbons exhibiting high temperature ignition only are benzene and methane whose ignition curves are shown in Fig. 5.7, together with the ignition curves for typical hydrocarbons n-heptane and iso-octane. Cool flames are not observed for methane and benzene.

In hydrocarbon-oxygen mixtures the blue cool flames are identified with the fluorescence spectrum of formaldehyde. Aldehydes are always present in cool flame regions. The concentrations vary with time reaching a maximum on the appearance of the cool flame (Fig. 5.8).

Significant quantities of peroxide are also present in the reacting mixture reaching a maximum just before the appearance of the cool flame and then dropping as shown in Fig. 5.8. In some experiments⁽⁷⁾ the drop in peroxide concentration is not observed.

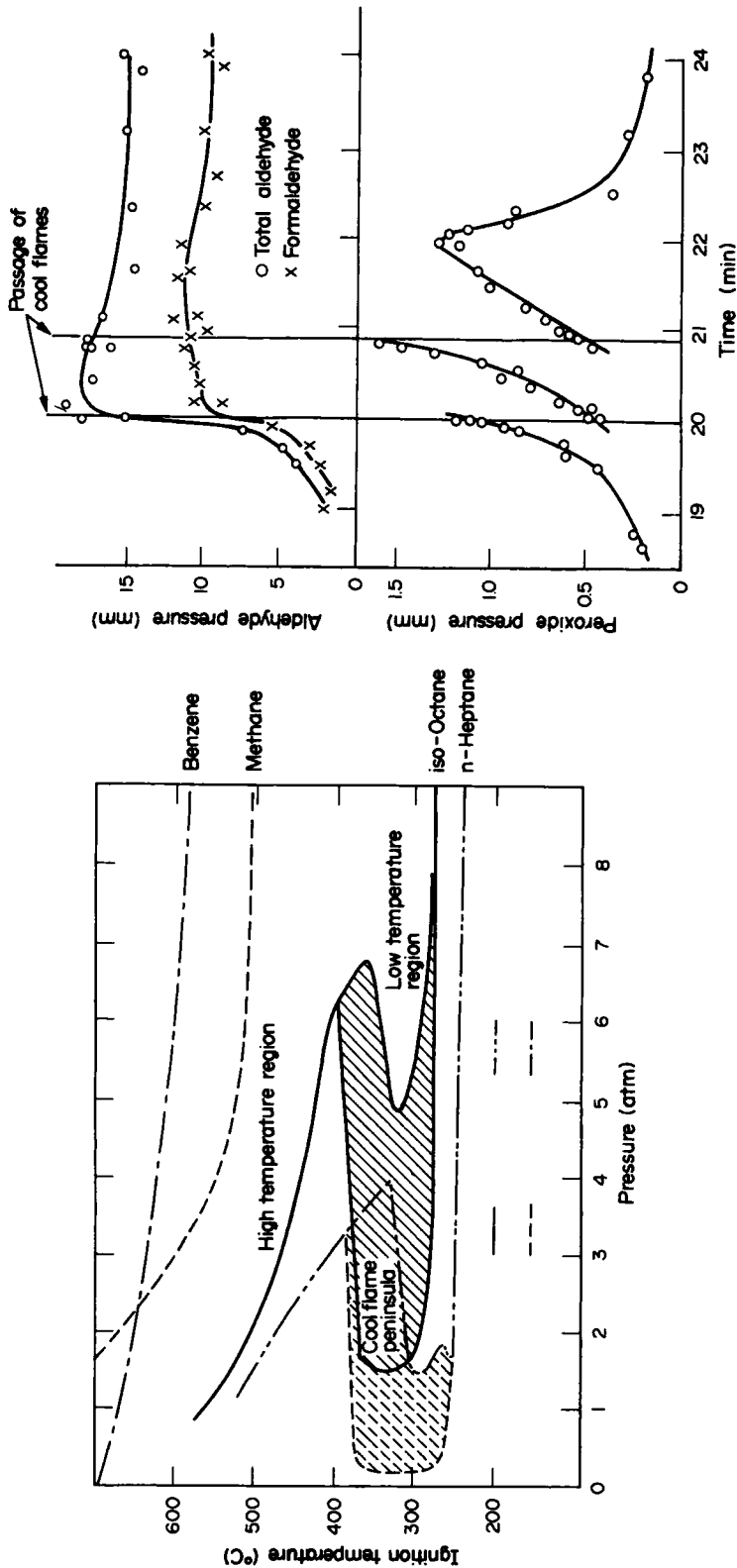


FIG. 5.7. Typical ignition temperature pressure curves.

FIG. 5.8. Variation of peroxide and aldehyde concentration during cool flame oxidation of butanone at 2830C. (From Bardwell and Hinshelwood⁽⁶⁾ by courtesy of the Royal Society.)

For hydrocarbon-oxygen mixtures not exhibiting cool flames the concentrations of aldehydes and peroxides are very much lower than in cool flame reactions.

The extinction of the cool flame is followed by a pressure pulse (Fig. 5.9).

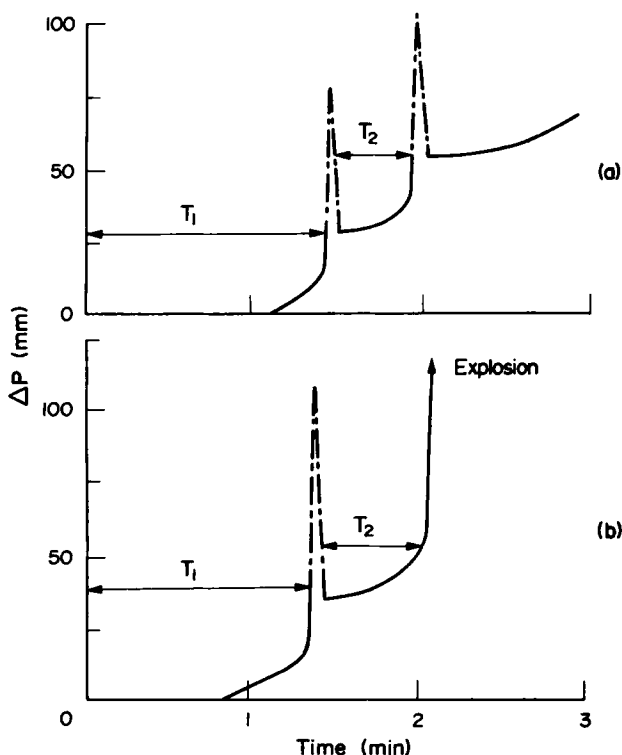


FIG. 5.9. Pressure time record during cool flames of butanone at 290°C. (a) Two cool flames; (b) one cool flame followed by ignition. (From Bardwell and Hinshelwood⁽⁶⁾, by courtesy of the Royal Society.

The time intervals between the start of a reaction and the appearance of cool flames and the period between cool flames and normal ignition are called induction periods. In Fig. 5.9(b) we can identify two periods—the first period T_1 up to the appearance of the cool flame and the second period T_2 from the appearance of the cool flame to the appearance of a normal flame explosion. This is a two-stage combustion process and the two separate periods have significance in our discussion later on knock. For high temperature ignition, with no cool flames, there is only one induction period.

We can therefore summarize the phenomena observed in laboratory experiments on the oxidation of hydrocarbon mixtures.

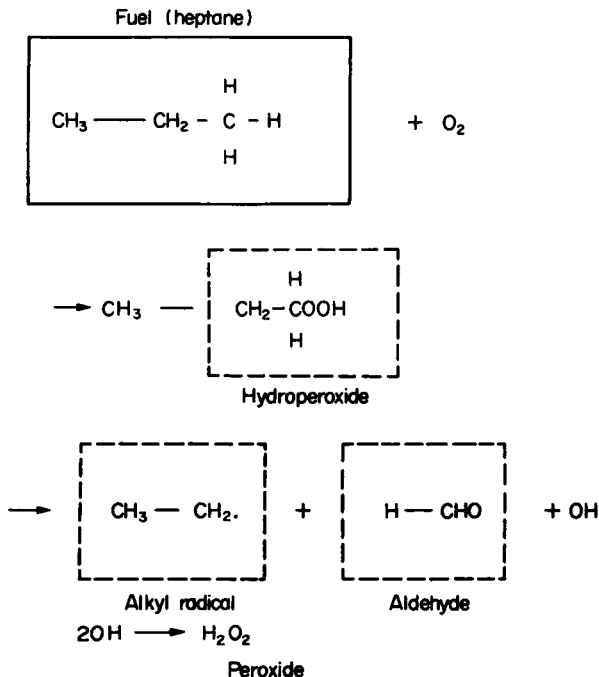
We have observed that there are two types of auto-ignition—a low temperature ignition and a high temperature ignition. Some hydrocarbon-oxygen mixtures exhibit both types of ignition; others only high temperature ignition. In the case of hydrocarbon-oxygen mixture with low and high temperature ignition there are significant concentrations of aldehydes and peroxides which peak just before the low temperature ignition (cool flame). For mixtures with high temperature ignition only, the concentrations of aldehydes and peroxides are small and almost insignificant in some cases.

The presence of the aldehydes and peroxides in the end gas in the engine cylinder are important clues in identifying the mechanism of knock, so we shall now review the possible mechanism of their appearance in a very simple manner. A typical hydrocarbon which knocks under certain conditions in an engine is heptane C_7H_{16} . The overall combustion reaction for a heptane-oxygen mixture is



We can picture this as a reaction between one molecule of heptane and eleven molecules of oxygen. Such a reaction is called a twelve-body reaction. The collision between the twelve molecules will not form the seven molecules of carbon dioxide and eight molecules of water vapour in one step, it must take place by successive collision of oxygen molecules with the heptane molecule degrading the heptane molecule to intermediate bodies before the formation of the final products carbon dioxide and water vapour. This is called a chain process.

Although the complete process is not known we can suggest possible intermediates. Downs⁽⁹⁾ and Wheeler⁽⁸⁾ have suggested the following reactions.



In this model we see that hydroperoxide is produced in the first stage of the reaction and aldehyde in the subsequent stages. Thus by this relatively simple model we can identify the presence of aldehydes and peroxides in the oxidation processes.

Our description of the oxidation processes has been based on laboratory tests in bench-type experiments under static conditions. Experiments under conditions more close to engine conditions such as in rapid compression machines confirm the same processes occurring in hydrocarbon-oxygen reactions. We are therefore in a position to use these results in assessing the processes occurring in engines. We shall now turn to engine research.

5.3.2 Engine Research

We have already noted that knock is due to spontaneous or auto-ignition of the end gas. The bench experiments showed that there are two types of auto-ignition—a low temperature ignition in which cool flames are observed with significant aldehyde and peroxide concentrations, and a high temperature ignition in which low aldehyde and peroxide concentrations are observed. The engine experiments have been devised to establish which form of ignition takes place in the end gas in the engine cylinder. It would be impossible to observe cool flames in the end gas under power conditions because the normal combustion process would generate an intense yellow flame. Therefore, under power conditions gas sampling only is used. To ascertain the relationship between the aldehyde and peroxide concentrations and cool flames, motoring tests are carried out. In these tests quartz windows are fitted and the cylinder contents are observed visually and photometrically.

Downs and Wheeler⁽⁸⁾ have reported extensive experiments. By sampling in the region of the end gas during the combustion period they were able to measure the variation of the aldehyde and peroxide concentrations with crankangle. Some typical results are shown in Figs. 5.10 and 5.11. In Fig. 5.10 the experiments were carried out under power with iso-octane with compression ratios well below the knock limit to a compression ratio giving heavy "knock". These curves show that a rise in pre-flame peroxide concentration before top dead centre is followed by a higher peak value just before the flame. With increase of compression ratio there is an earlier rise in peroxide concentration and a much higher peak value. There is a point of inflexion just after top dead centre. Downs suggests that the shape of the curves are two-stage in nature. In passing we might note that the addition of enough tetraethyl lead to suppress knock lowers the peroxide concentration considerably. We will refer to this result shortly.

The aldehyde concentration starts to increase at about top dead centre and the curves show a smooth rise up to the time of flame arrival, achieving a peak concentration some twenty times greater than the peak peroxide value. The effect of increasing the compression ratio on the aldehyde concentration parallels the effect on the peroxide curve. Tetraethyl lead decreases the aldehyde concentration.

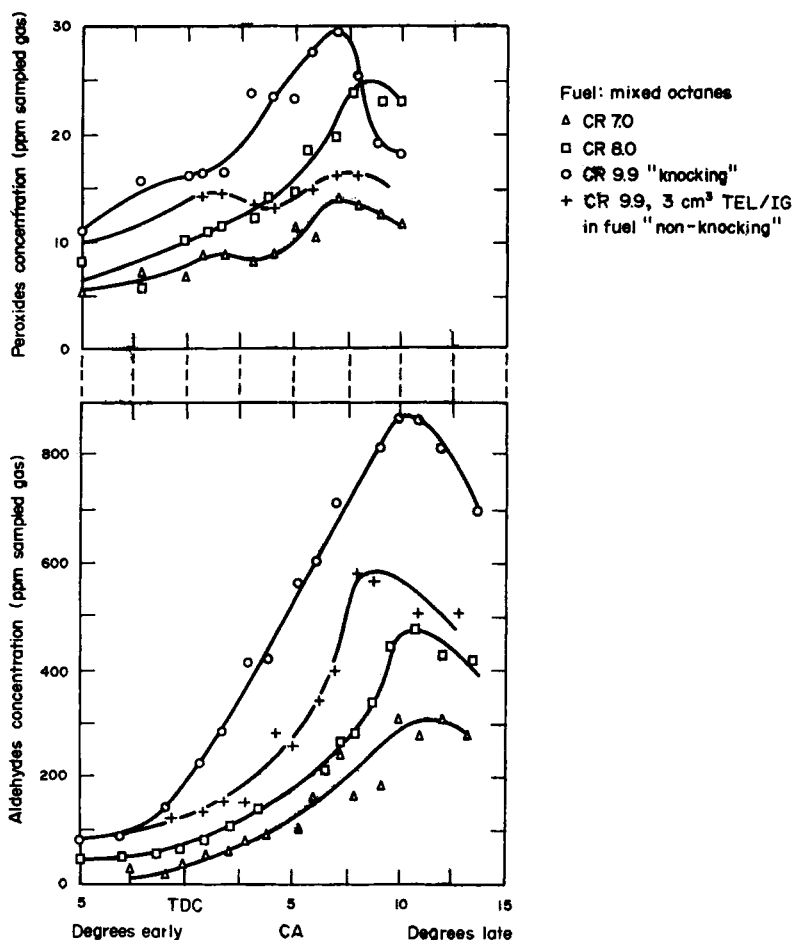


FIG. 5.10. Peroxide and aldehyde concentrations in gases sampled from the engine at various crankangle. (From Downs and Wheeler,⁽⁸⁾ by courtesy of the Institution of Mechanical Engineers.

In Fig. 5.11 the experiments were carried out under power with methane as a fuel. It will be seen that the results are somewhat different from the iso-octane results. Raising the compression ratio to cause knock results in a peak in the peroxide curve; this peak is eliminated by the addition of tetraethyl lead to the fuel. There is not a great deal of difference between the knocking and non-knocking aldehyde curves. Both the peroxide and aldehyde concentrations were well below those obtained with iso-octane under comparable conditions. Tests with benzene as a fuel gave negligible concentrations of both peroxide and aldehydes.

In the motoring tests with iso-octane, cool flames were observed when the compression ratio was raised. Photometric experiments showed that the cool flame passed in the region of top

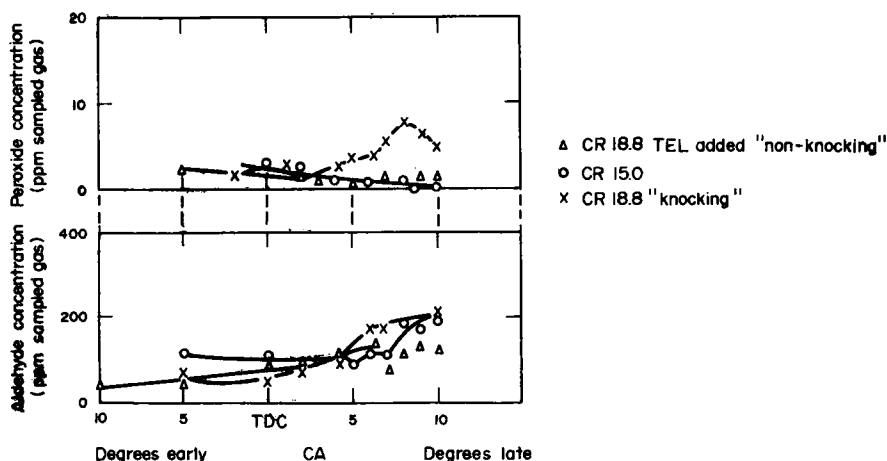


FIG. 5.11. Peroxide and aldehyde curves with methane as fuel.
(From Downs and Wheeler,⁽⁸⁾ by courtesy of the Institution of Mechanical Engineers.)

dead centre, the timing of the cool flame was in agreement with the point of inflexion of the sampling curves (Fig. 5.10). The hot flame started in the wake of the cool flame. No cool flames were observed with methane and benzene fuels.

From these experiments it is clear that the spontaneous ignition in the end gas with iso-octane fuel is a two-stage ignition phenomenon, whilst the spontaneous ignition in the end gas with methane is a single-stage ignition phenomenon. The results with iso-octane are typical of those obtained with the higher paraffinic components of commercial gasoline, and the knocking compression ratios are of the same order. Thus knock in commercial engines is of the two-stage low temperature type since this is associated with spontaneous ignition in the low temperature region of the ignition curve (Fig. 5.7). We normally call this type of knock low temperature knock.

The mechanism for low temperature knock can therefore be described in the light of the above experiments. As the normal combustion flame traverses the cylinder, the pressure and temperature in the end gas increase until they reach values when, firstly, there is low temperature auto-ignition followed after a short delay by high temperature auto-ignition which produces a rapid increase in pressure generating a shock wave.

Whether under normal combustion conditions the end gas will spontaneously ignite before the flame arrives at the combustion chamber walls will depend on:

- (1) the composition of the fuel;
- (2) the pressure and temperature in the end gas;
- (3) whether there is sufficient time for the two stages of ignition to take place.

We are now in a position to relate these conditions to the observed engine parameters which influence knock. Before doing so we shall very briefly refer to fuel composition. Fuels with ignition curves similar to the higher paraffinics show a greater tendency to knock than fuels with ignition curves similar to methane and benzene. We shall shortly refer to how the fuel quality is assessed. Table 5.1 summarizes the influence of a number of operating factors on the tendency to knock. We shall examine these.

TABLE 5.1. Summary of Factors Influencing Tendency to Knock

Factor	Effect	Pro-knock	Anti-knock
Compression ratio	Increase	✓	
Ignition timing	Advance	✓	
Throttle setting	Full open	✓	
	Partially open		✓
Mixture strength	Weak		✓
	Slightly rich	✓	
	Over rich		✓
Engine speed	Increase		✓
Inlet air temperature	Increase	✓	
Inlet air pressure	Decrease		✓
Cooling water temperature	Increase	✓	

Compression Ratio. The higher the compression ratio the higher the pressure and temperature in the end gas during normal conditions; this increases the tendency to knock. For a given engine setting and fuel there will be a critical compression ratio above which knock occurs. This compression ratio is called the highest useful compression ratio (HUCR).

Ignition Timing. Advancing the ignition timing raises the cylinder pressure and temperature which in turn raises the end gas pressure and temperature thus increasing the tendency to knock. If the ignition is advanced too much the cylinder pressures fall and the tendency to knock is reduced. If in a given engine the fuel quality is changed and knock takes place, retarding the ignition may eliminate the knock; it will also reduce the engine power.

Throttle Setting. This controls the cylinder inlet pressure and mixture strength. If we consider mixture strength under a separate heading the effect of opening the throttle is to raise the cylinder pressure, thus increasing the pressure in the end gas and increases the tendency to knock.

Mixture Strength. This controls the combustion temperature and pressure, hence influencing the end gas temperature and pressure. The tendency to knock follows the power variation with mixture strength. That is at maximum power there is a greater tendency to knock; this is reduced as the mixture is strengthened or weakened.

Engine Speed. The lower the engine speed the longer the absolute time for the flame to traverse the cylinder, the longer the time available for the end gas reaction to take place and the greater the tendency to knock.

Inlet Air Temperature. Raising the air inlet temperature raises the cylinder temperature, thus raising the end gas temperature and increasing the tendency to knock.

Cooling Water Temperature. Raising the cooling water temperature raises the overall cycle temperature, hence increasing the end gas temperature and increasing the tendency to knock.

As well as the engine operating conditions certain design factors also influence the tendency to knock; two of these are the locations of the sparking plug and the exhaust valve. The spark plug should be located to allow for the minimum distance for the normal flame to traverse the combustion chamber thus reducing the time for the end gas reactions. The exhaust valve should be located close to the spark plug so that it is not in the end gas region, otherwise there will be a tendency to knock.

Another important factor influencing the knock limit is the geometric form of the combustion chamber shape including the dimensions of the clearance space between the cylinder head and piston crown. For the interested reader an extensive review of the influence of engine design and operating conditions on the knock limits and performance of motor vehicle gasoline engines has been presented by Caris et al.⁽²³⁾

Although the designer has a wide range of variables at his disposal to eliminate knock, ultimately his final design will be dependent on the fuel quality. To assess the knock quality of a fuel, special engine tests are carried out under set engine conditions. The engine is a variable compression engine, i.e. an engine whose compression ratio can be varied whilst running. The fuel under test is supplied to the engine and the compression ratio increased until there is incipient knock as measured by an instrument sensitive either to rate of pressure rise, temperature or some other knock sensitive parameter. The fuel is then cut off and a mixture of n-heptane and iso-octane is supplied to the engine. The composition of the mixture is adjusted until incipient knock takes place at the same compression ratio. The fuel is rated by the

percentage of iso-octane in the n-heptane and iso-octane mixture. Iso-octane has a lower tendency to knock than n-heptane, so that the higher the octane rating of a fuel the lower the tendency to knock.

The standard engine used in these tests is the CFR engine; this is a variable compression ratio engine similar to the Ricardo E6 (Fig. 5.12). There are two basic tests called the motor method

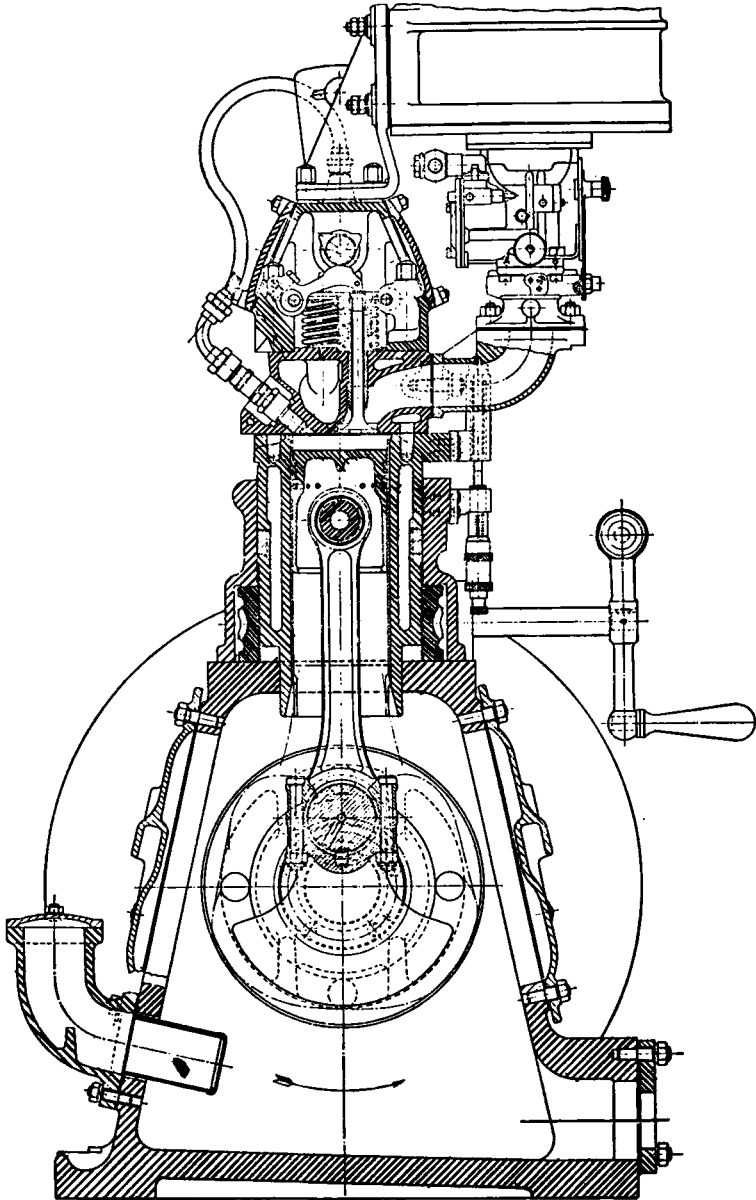


FIG. 5.12. Ricardo E6 cross-sectional arrangement of variable-compression ratio research engine of 3 in. bore by $4\frac{1}{8}$ in. stroke (petrol engine).

and the research method. The test conditions for the two tests are summarized below:

Method	Motor	Research
Engine speed (rev/min)	900	600
Mixture temperature ($^{\circ}\text{C}$)	149	51.7

The sensitivity of a fuel to knock is measured by the difference in the two knock ratings; the greater the difference the greater the fuel sensitivity. The octane rating of fuel is lower in the motor method than in the research method.

Although the numerical value of the sensitivity has no physical meaning, it does give an indication of the relative merits of fuels used in automobile gasoline engines on the road. It might be expected that a fuel with a high sensitivity would tend to knock with the wide range of engine operating conditions experienced in normal traffic, whilst a fuel with low sensitivity would be relatively insensitive to the same conditions. Other methods have been developed to define the fuel quality using motor vehicle tests; the reader is referred to Ricardo and Hempson⁽¹⁾ for a discussion of these methods.

We have mainly discussed low temperature knock. It was shown by Downs and Wheeler⁽⁸⁾ that certain fuels knock at high compression ratios. These fuels do not exhibit cool flames. The auto-ignition mechanism is therefore of the high temperature type. This is a single-stage auto-ignition and the phenomenon is called high temperature knock. Modern fuels contain components which exhibit both low temperature and high temperature knock. Experiments show that sensitive fuels knock more by a high temperature process than by low temperature knock.

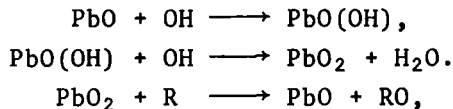
5.3.3 Influence of Fuel Additives on Knock

Up to this point we have examined the knock mechanism in engines and related these to the engine operating conditions and the fuel. There has been much research to develop fuels with high knock ratings (high octane number). These are mainly achieved by refining processes, but they are in general expensive fuels. Alternative methods have therefore been sought, the most promising of which are fuel additives. Research has shown that, in general, an anti-knock additive in a fuel which inhibits low temperature knock enhances knock in a fuel which exhibits high temperature knock, and vice versa. Hence the selection of the correct additive for a particular fuel is most important. The most effective additive for commercial fuels at the time of writing is tetraethyl lead (TEL). There has in recent years been criticism of this additive since environmentalists claim it pollutes the atmosphere, although at present there is no direct evidence of resulting damage. Nevertheless, an increasing number of legislatures throughout the world are specifying reduction in the lead content in fuels with its eventual elimination. Since TEL is so effective there has been much research on the mechanism of its action. It is probable from

this work that alternative, and less obnoxious, additives might be developed. The exact mechanism of the action of TEL is not clear and there is some controversy in scientific circles on the mechanisms, but it is possible to identify some of the probable facets of its action.

The chemical formula of TEL is $\text{Pb}(\text{C}_2\text{H}_5)_5$, i.e. it is a metallorganic salt. It is mixed with the fuel and is present in the cylinder in the fuel vapour. It is generally accepted that it is decomposed thermally during the compression stroke and reacts with oxygenated intermediates to give lead oxides. There are two opposing theories as to whether the lead oxide is in vapour form or in particulate form (fog). The lead oxide acts as a chain-breaking agent (this was observed in Figs. 5.10 and 5.11). It does not appear to influence the reaction in the first stage of combustion nor to have any influence on the cool flames, but it does have an inhibiting action in the second stage by increasing the induction time T_2 .

The suggested chain-breaking mechanism is:



where R is a radical.

The vital point, not clearly understood, is whether the chain termination is a surface or a gas phase effect, since if it is the former the anti-knock effectiveness of TEL might be increased by adding other organometallic compounds so that mixed oxides could be formed which would be more efficient than solid PbO itself in destroying the HO_2 radicals or peroxides. Downs *et al.*⁽⁹⁾ have claimed to have observed PbO_2 fog in the cylinder indicating possible surface reactions. In addition to the pollution effect of lead it also poisons catalytic reactors used to reduce emissions from gasoline engines, so that the intense research directed to improve fuel quality with alternative additions might, by the time this book is in print, have the effect of rendering TEL to the history book. Whatever new additive is used, however, its mechanism will have to have the same effect as TEL, i.e. to act as a chain terminator.

5.4 UNCONTROLLED COMBUSTION, PRE-IGNITION AND RUNNING-ON

Under certain conditions the fuel/air mixture is ignited by a hot spot in the cylinder. The hot surface might be the spark plug insulator or electrode, or combustion deposits, or when knock occurs the exposed surface, where the boundary layer is scored by the shock wave. When the ignition occurs before the spark is ignited the phenomenon is called pre-ignition. When the phenomenon occurs after the ignition is switched off it is called running-on. Combustion deposit ignition is called rumbling.

5.4.1 Pre-ignition

This is the most severe form of uncontrolled combustion. Under severe operating conditions some part of the cylinder surface may be hot enough to ignite the charge. If there is insufficient time for hot spot ignition to take place before the spark ignites the mixture, normal combustion will take place. As the temperature of the hot spot increases, a condition may be reached when the engine would run normally with the ignition switched off. This is often called auto-ignition. Further increase in hot spot temperature would cause the mixture to ignite before the spark ignites. This is equivalent to advancing the ignition, but since the hot spot surface is larger than the spark the combustion rate would be faster than normal combustion creating very high cylinder pressures and temperatures resulting in excessive negative compression work and increased heat loss to the walls, the overall effect being a loss in power. Under certain conditions the engine could stall and possibly cause mechanical failure.

Pre-ignition normally follows heavy knocking due to the breakdown of the cylinder wall boundary layer. The combustion mechanism for pre-ignition corresponds to high temperature ignition, and it is noteworthy that although benzene does not knock it has a strong tendency to pre-ignite. Methods have been developed for pre-ignition ratings for different fuels. Descriptions of these methods are given by Downs and Pigneguy.⁽¹⁰⁾

5.4.2 Running-on

When the ignition is switched off and the throttle closed the engine may continue to fire. This might be due to a hot surface in the cylinder. It is considered, however, that the major cause is spontaneous ignition of the fuel/air mixture. The tendency to spontaneous ignition is dependent on the temperature/pressure ignition curve (Fig. 5.7) for the fuel, reflected in the octane rating of the fuel. Physical factors influencing spontaneous ignition are an elevated inlet temperature of the mixture, poor cooling of the combustion chamber surface, duration of valve overlap and high compression ratio.⁽²⁹⁾

5.4.3 Rumble

This is ignition associated with the combustion deposits. This might occur with sudden power demand from the engine causing the deposit to glow and ignite the fuel/air mixture. The tendency of deposits to form and to cause ignition depends on the fuel composition. By adding phosphorus compounds the deposits become chemically inactive and do not cause ignition.

5.5 CHEMICAL THERMODYNAMIC MODELS FOR NORMAL COMBUSTION

In the previous discussions we have directed our attention to describing the combustion ~~process~~ from the point of view of the various modes of combustion and their influence on the engine performance. We now propose to describe a chemical-thermodynamic

model for the combustion process which will enable us to predict the engine performance. In Chapter 9 we shall develop these methods further from the view point of cycle analysis; in this section we shall outline the model and the assumptions leading on to the important new subject of combustion-generated emissions in the next section.

It is clear from the previous discussion that the chemical reactions during combustion are complex. For the predictions of the pressure and temperature in the cylinder and the overall composition of the products of combustion we can make fairly simple assumptions to obtain a broad picture of the overall process under normal combustion conditions. For this purpose we use equilibrium thermodynamics,[†] the major principles of which were reviewed in Chapter 2.

The combustion chamber may contain a heterogeneous mixture of burnt and unburnt products. For calculation purposes the combustion chamber is subdivided into zones. Lavoie et al.⁽¹⁸⁾ and Blumberg⁽²⁰⁾ assumed a multi-zone model for the combustion process with a uniform temperature in each zone giving a variable temperature distribution in the combustion chamber. A burning rate is assumed for the combustion process. This is usually in the form of fraction of fuel/air burnt as a function of crankangle.^{(20)††} Another model⁽¹²⁾, which will be outlined in Chapter 9, assumes that the combustion chamber is subdivided into two zones—a burnt zone and an unburnt zone. The two zones are separated by a flame front which is spherical in form and centred on the spark plug. With the second model a relationship is required between the flame speed and the cylinder conditions. The selection of an appropriate relationship is critical to the modelling of the combustion process. In practice the flame will be turbulent; unfortunately there is no clear model for a turbulent flame, so that attention has been drawn to computing the laminar flame speed with a suitable modification to allow for turbulence called the flame factor ff .

Benson et al.⁽¹²⁾ suggest the following expression for the turbulent flame speed u_t :

$$u_t = ff \times u_l,$$

where u_l is the laminar flame speed. This model is due to Annand.⁽¹⁹⁾

The laminar flame speed can be calculated by a number of methods which are reviewed by Phillips and Orme⁽¹¹⁾ and Benson et al.⁽¹²⁾

[†]For accurate predictions of the chemical composition reaction rate, equations must be included.

^{††}More recently a turbulent flame combustion model has been developed by the same research group to predict the burning rate. Full details are given by Blizard and Keck⁽²⁷⁾ and Tabaczynski et al.⁽²⁸⁾

The flame factor ff is adjusted to assume the combustion time is symmetrical about top dead centre. A calculation of the flame speed using the Annand model is shown in Fig. 5.13.

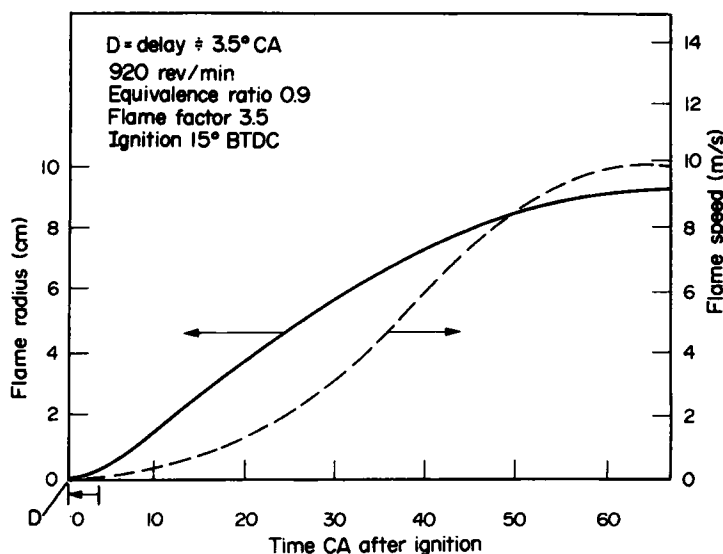
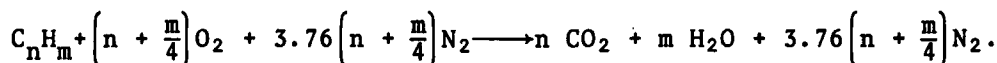


FIG. 5.13. Computed flame speed and radius

An outline of the calculation procedure is given in Chapter 9. In the Annand model it is assumed that in the unburnt zone there are no chemical reactions, whilst in the burnt zone reactions that satisfy chemical and thermodynamic equilibrium, including the appropriate reaction rate equations, take place.

For a correct mixture of a hydrocarbon C_nH_m and air the overall chemical reaction can be expressed as



At the elevated temperatures in the burnt gas zone the carbon dioxide (CO_2) and water vapour (H_2O) dissociate to form carbon monoxide (CO), oxygen (O_2) and hydrogen (H_2). Furthermore, the molecular oxygen (O_2), hydrogen (H_2) and nitrogen (N_2), dissociate to form atomic oxygen (O), hydrogen (H) and nitrogen (N). In addition there will be equilibrium concentrations of hydroxide (OH) and nitrous oxide (NO). Thus in the burnt gas zone we have the following species:



The concentration of these various species will depend on the air/fuel ratio and the prevailing pressure and temperature. By setting

up the appropriate equations and the equilibrium constants, the numerical concentrations can be computed by the methods outlined in Chapter 2 and Chapter 9 of Vol. II. The overall effect of the dissociation processes is to reduce the maximum temperature of the burnt gas below the value which would have been reached if dissociation had not been allowed for. This is illustrated in Fig. 5.14 for a benzene/air mixture. The maximum temperature occurs with a mixture about 10-20% on the rich side of stoichiometric.

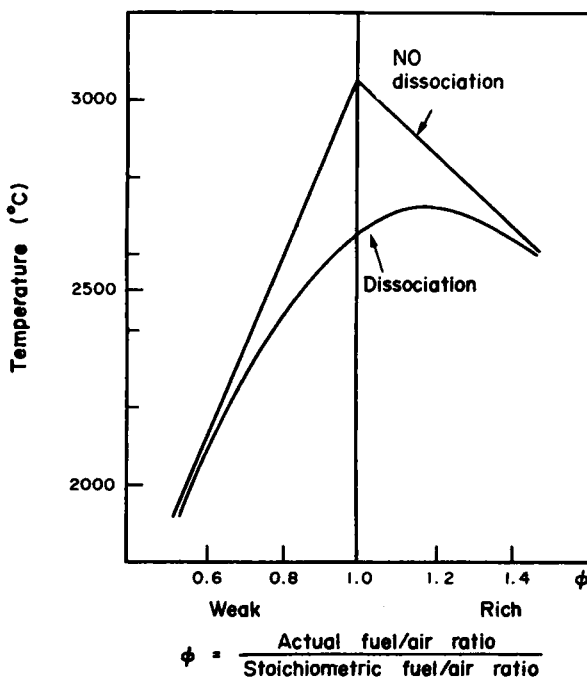


FIG. 5.14. Calculated maximum temperature for benzene-air mixtures.

It will be seen that dissociation effects are quite strong either side of the stoichiometric mixture. A full discussion on the influence of dissociation on the combustion temperature and the engine power is given in Chapter 9.

The gas composition in the cylinder changes during the expansion stroke, some of the dissociating species combine; others, however, remain in concentrations above the equilibrium values. We shall discuss this in the important topic of combustion-generated emissions in the next section.

5.6 COMBUSTION-GENERATED EMISSIONS

In recent years public attention has been directed to the problem of exhaust emissions from automobiles. The problem was first highlighted in the United States in the Los Angeles area where air pollution caused plant damage, eye and throat irritation, cracking of stressed rubber and decrease in visibility (smog). In the United States legislation has been implemented to limit the concentrations of hydrocarbon (HC), carbon monoxide (CO), nitric oxides (NOX) and particulates in the automobile exhaust gases. Although some of the emissions are due to the lubricating oil, the main source is the combustion process of the hydrocarbon/air mixture.

In Europe the attention of legislatures has been directed mainly to the reduction of lead in the exhaust gases by limiting the lead content of the fuel.

An excellent review of the current knowledge of pollutant formation and control in spark ignition engines has been given by Heywood.⁽²⁴⁾ In this section we will outline the possible mechanisms of the formation of the combustion-generated emissions and indicate the thermodynamic factors which influence the concentrations. We shall see that for carbon monoxide and nitric oxide the evidence indicates that during the expansion process recombination does not take place. In these circumstances the rate of the forward reaction is different from the rate of the backward reaction. The chemical reactions are therefore rate-controlled. A brief discussion of the thermodynamics of rate-controlled reactions is given in Chapter 2 (section 2.7).

5.6.1 Carbon Monoxide

Carbon monoxide (CO) is always present in the exhaust gases due to the dissociation processes. With rich mixtures the CO concentration is further increased due to the incomplete combustion to carbon dioxide. Some typical results are shown in Fig. 5.15 for a four-cylinder engine at full throttle. The measured concentrations of CO are greater than the equilibrium concentrations. This indicates that recombination is not complete in the expansion stroke. The reactions in the expansion stroke must therefore be rate-controlled. The full mechanism for the formation of CO is at the time of writing (1976) not fully known, although some of the rate-controlled reactions are known, so it is not possible at present to predict the CO concentrations. A review of the present position for CO predictions is given by Heywood⁽²⁴⁾ and Benson.⁽²⁵⁾

To reduce CO emissions the engine may run with a weak mixture with loss of power; or devices may be appended to the exhaust to oxidize the CO. One such group of devices are thermal reactors in which air is injected into the exhaust stream and a flame is produced which aids in the combustion of CO to CO₂. Another method is to oxidize the CO in the presence of a catalyst. A number of these methods are described in the symposium on Air Pollution in Transport Engines⁽⁴⁾ (Proc. Instn. Mech. Engrs. 1971) and in the review paper by Heywood.⁽²⁴⁾

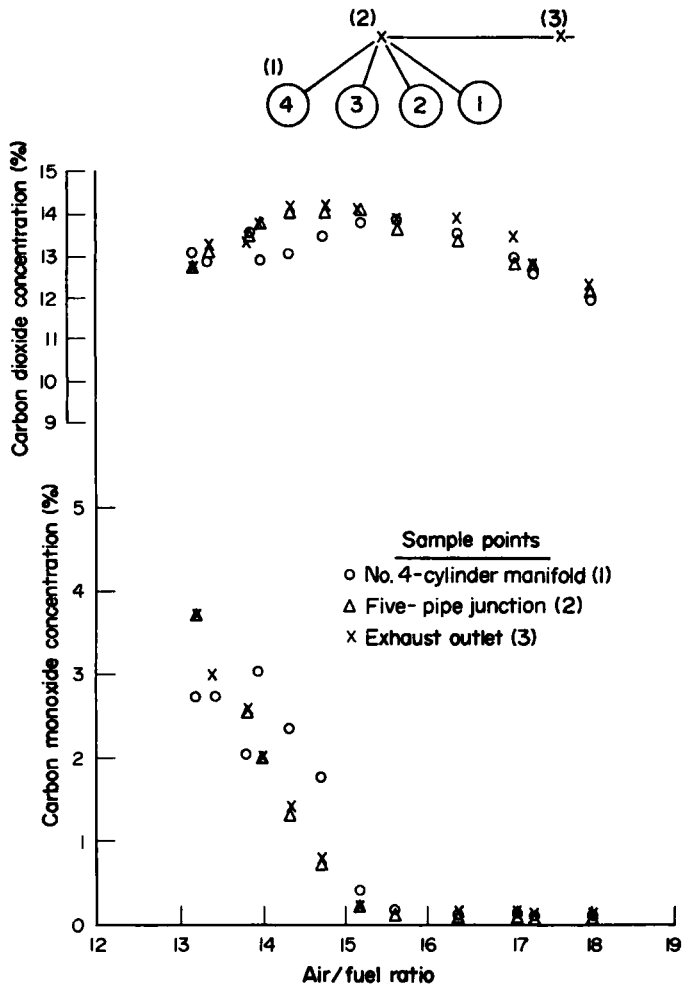


FIG. 5.15. Measured carbon monoxide and carbon dioxide in exhaust pipe of a four cylinder gasoline engine. Full throttle at 3000 rev/min.

5.6.2 Nitric Oxide

Whilst the obnoxious effects of CO are well known the significance of the nitric oxide (NO) emissions has only recently been discovered. It is the NO and hydrocarbon emissions which react in sunlight to cause photochemical smog which caused the peculiar pollution problem in the Los Angeles area. There has been intensive research to understand the source of the NO emissions and to develop methods to reduce them. The pioneering work was carried out by Starkman and Newhall.⁽¹⁵⁾ There have been numerous publications. We have already indicated that NO is present in the products of combustion due to dissociation of the molecular oxygen and nitrogen. Under equilibrium conditions one would expect that the concentrations of NO would fall with pressure and temperature

during the expansion stroke to negligible proportions at exhaust valve opening. All experiments on engines indicate that the NO concentrations in the exhaust gases are well above the equilibrium values. It is therefore deduced that the NO reactions are rate-controlled. In order to calculate the quantity of NO to be expected it is necessary to establish the chemical mechanism so that the appropriate rate equations can be used. There have been a number of models suggested. Experiments by Newhall and Shahed⁽¹⁶⁾ using spectroscopic techniques to record the time rate of formation of NO in a hydrogen-oxygen reaction in a closed vessel showed that NO formation occurred predominantly in the post-flame combustion products. Lavoie^(17,18) showed that substantial temperature and NO gradients occurred behind the flame in an engine combustion chamber. In Fig. 5.16 some of Lavoie's results are shown from spectrographic measurements. It will be seen that after the flame arrives at the window the NO concentration increases up to a maximum and then remains steady. Different concentration curves were observed at each window.

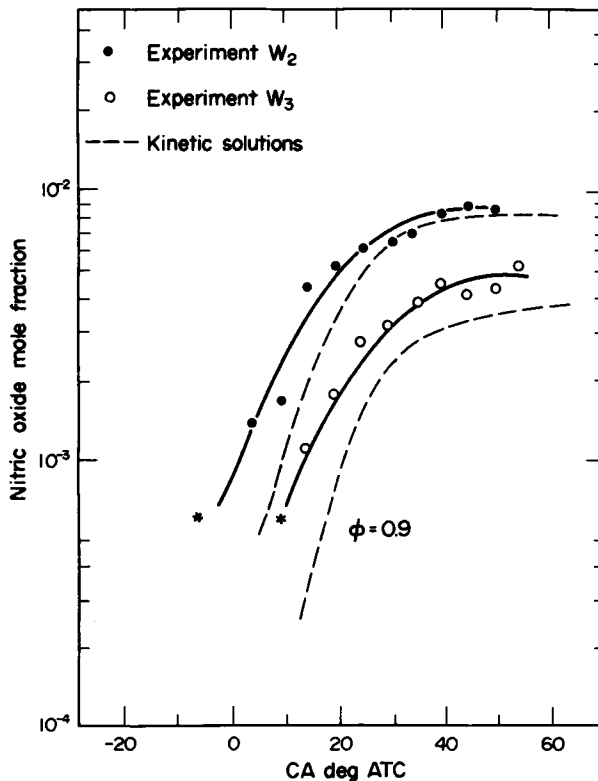
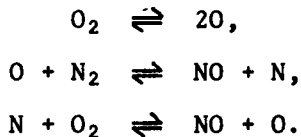


FIG. 5.16. Nitric oxide concentrations in engine cylinder. Experimental NO concentration versus crankangle for $\phi = 0.9$ for two windows, W₂ being closer to the spark than W₃. The asterisks mark estimated initial conditions and flame arrival times. The dotted lines are theoretical kinetic solutions for parts of the charge burning at the corresponding times and starting with zero NO concentrations. (From Lavoie,⁽¹⁷⁾ by courtesy of Combustion and Flame).

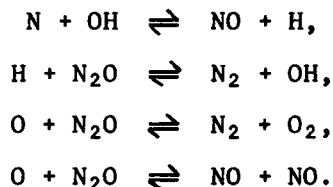
Because the detailed experiments on engines are difficult, computer models have been set up to study the effect of engine operating variables on the NO emissions and to identify the important parameters which influence the formation of NO.

Lavoie et al.⁽¹⁸⁾ subdivide the combustion chamber into a number of elements and assume that each element is burnt separately. This method enables the temperature gradient to be computed and the variation in NO concentration to be determined. The average value of NO concentration is then taken as the NO at exhaust valve opening. The rate of combustion of the elements is called the burning law, and this is taken from an analysis of the measured indicator diagram.[†] Annand,⁽¹⁹⁾ on the other hand, has suggested a simple two-zone model with an unburnt and burnt region with uniform temperatures in each region. In this method the indicator diagram is synthesized by assuming a spherical flame front separating the two zones and calculating the temperature, pressure and composition of the burnt zone from a combination of the equilibrium equations for a number of reactions and the rate-controlled equations for the oxides of nitrogen simultaneously.

Both methods require a mechanism for the formation of NO. In their original work, Newhall and Starkman⁽¹⁵⁾ suggested the so-called Zeldovich chain reaction:



It is suggested that at elevated temperatures in the cylinder the molecular oxygen (O_2) dissociates to atomic oxygen which reacts with the molecular nitrogen (N_2) to form NO and atomic nitrogen (N) the latter reacting with O_2 , forming NO and atomic oxygen (O). There have been suggestions that this model is not complete and Lavoie et al.⁽¹⁸⁾ have proposed a modified Zeldovich reaction with the hydroxide (OH) radical:



Using this model, Lavoie compared his experimental results with the predicted concentration of NO (Fig. 5.16).

[†]More recently Blizard and Keck⁽²⁷⁾ and Tabaczynski et al.⁽²⁸⁾ have developed turbulent flame combustion models to predict the burning law

Annand⁽¹⁹⁾ has suggested a further group of reactions to be added to the modified Zeldovich reaction, these are:



Benson et al.,⁽²⁰⁾ using Annand's method, predicted the NO emissions for a single-cylinder gas engine operating with propane/air as shown in Fig. 5.17. It will be seen that the predictions agree with the experiment. It should be pointed out that the flame factor ff was adjusted to give agreement between the experimental and predicted value of NO at the maximum emission. It was then kept constant and the predictions made over the whole air/fuel ratio range shown.

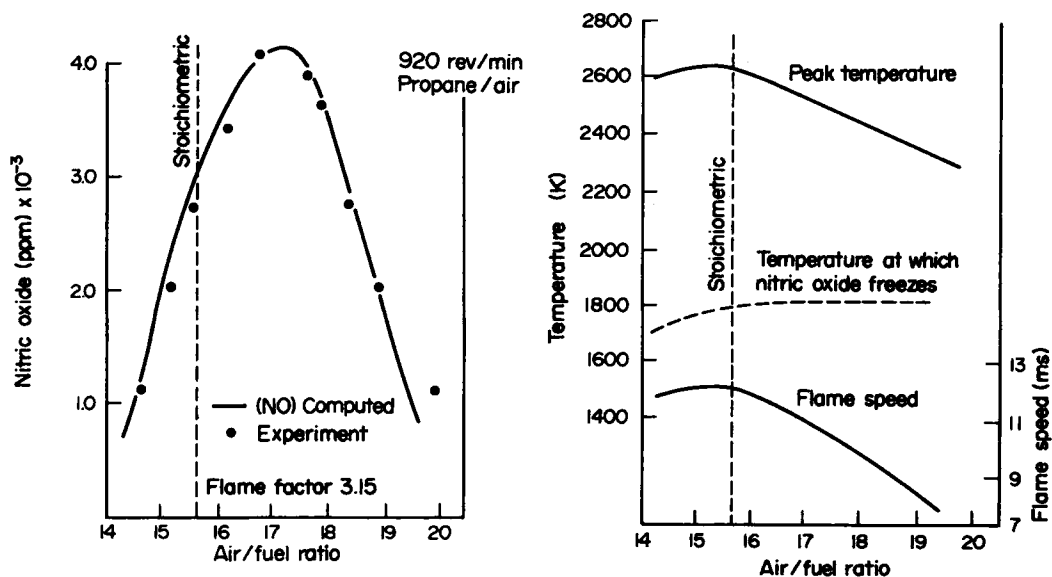


FIG. 5.17. Influence of mixture strength on nitric oxide and maximum and freezing temperatures. (From Benson et al.⁽¹²⁾.)

The variation in NO with crankangle is shown in Fig. 5.18. The same trends as noted by Lavoie (Fig. 5.16) will be observed.

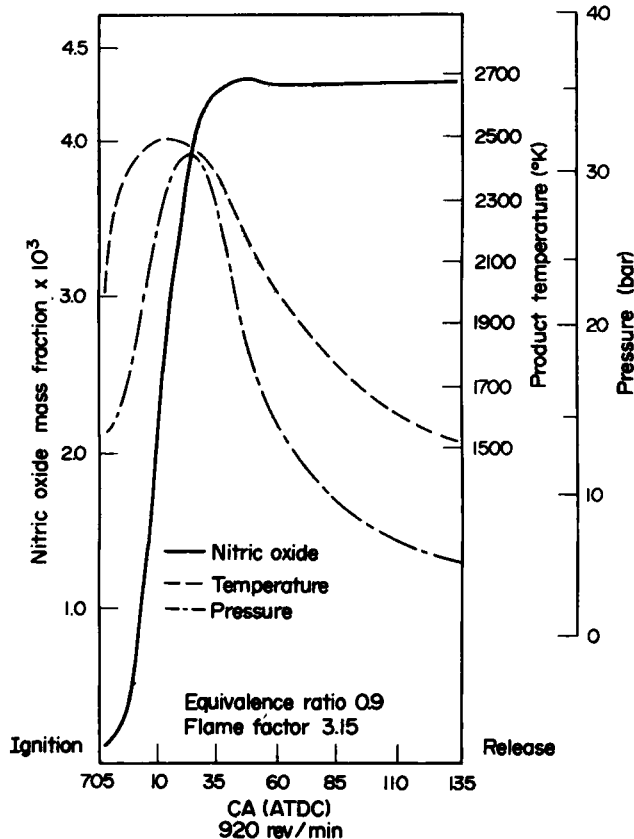


FIG. 5.18. Variation in pressure, temperature and nitric oxide concentration. (From Benson et al.⁽¹²⁾.)

From these calculations it could be observed that after ignition the NO concentration increases rapidly, then there is a slight fall and the NO concentration remains constant, even though the temperature and pressure are falling. This phenomenon is called "freezing" and is due to the slow rate of the backward reactions, i.e., the dissociation of NO to N₂ and O₂ through the chain given above.

Lavoie et al.⁽¹⁸⁾ have shown for their multi-element model and the modified Zeldovich reaction, that, depending on the time of burning of a particular element, even the forward rate of some of the reactions are not fast enough to achieve equilibrium conditions. This is illustrated in Fig. 5.19. Here we notice that the element burning 30° before top dead centre ($\theta_b = -30^\circ$) reaches the peak value at about the equilibrium concentrations, but the element burning at 10° after top dead centre ($\theta_b = 10^\circ$) does not reach the peak equilibrium value. In Fig. 5.19 we clearly see the difference

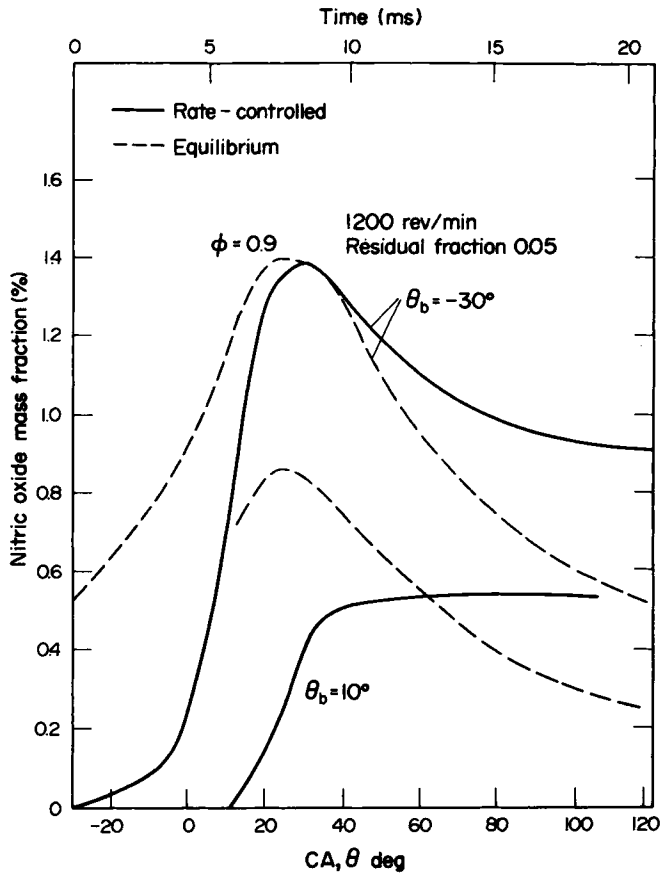


FIG. 5.19. Variation in nitric oxide concentration. Nitric oxide mass fractions as a function of time in the burnt gas for two elements which burn at different times: $\theta_b = -30^\circ$ is the first part of the charge to burn; at $\theta_b = 10^\circ$, one-third of charge is burnt. From Lavoie et al.,⁽¹⁸⁾ by courtesy of the Massachusetts Institute of Technology.)

between the equilibrium concentration and the predicted frozen concentration at the end of the expansion.

If we turn to Fig. 5.17 again we see that for weak mixtures the temperature at which NO freezes is almost constant; and even on the rich side there is only a slight drop in temperature. The peak NO concentration occurs on the weak side and falls off considerably over a few air/fuel ratios. This trend is observed in engines of all types. The NO concentration may vary in the exhaust pipe. Some results obtained by the authors in a four-cylinder engine are shown in Fig. 5.20.

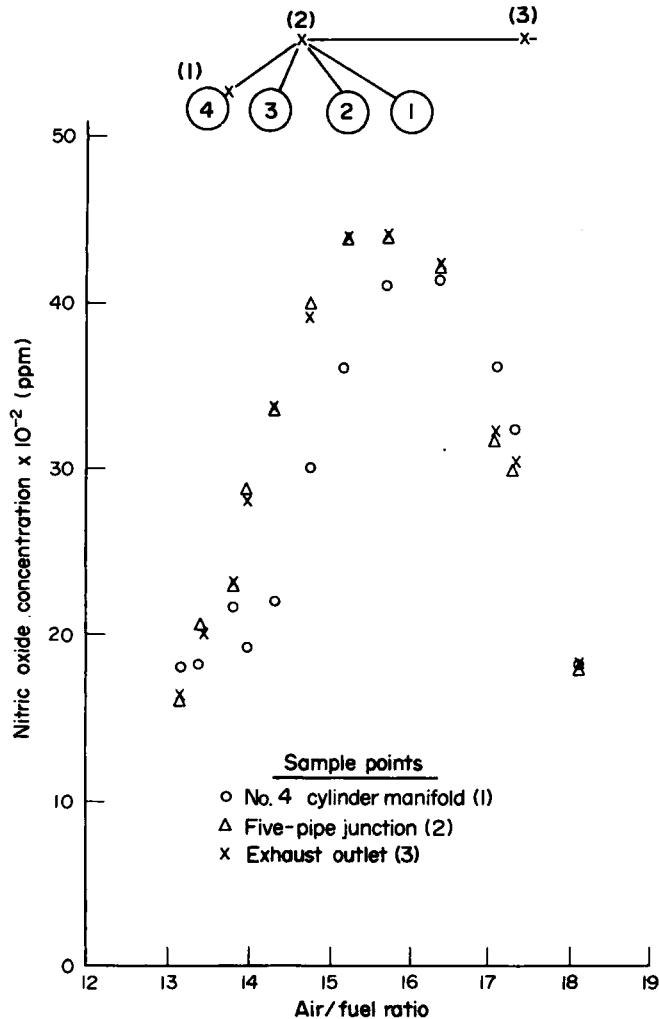


FIG. 5.20. Measured nitric oxide concentrations in exhaust pipe of a four-cylinder gasoline engine. Full throttle at 3000 rev/min.

Thus we may conclude that the NO formation in the cylinder is associated with the rate-controlled combustion processes. It is therefore related to the thermodynamic conditions in the cylinder at ignition. To minimize the NO concentration the problem has to be attacked by controlling the combustion process. It would seem from Fig. 5.19 that the NO concentration is related to the peak temperature and the flame speed. The higher the peak temperature the higher the concentration, and the lower the flame speed the longer the time for the NO to dissociate to molecular nitrogen and oxygen. This does not completely hold since we should expect the peak NO to be on the rich side, but the general trend is in these directions. Therefore

any parameter that can reduce the peak temperature or decrease the flame speed or both will reduce the NO emission. Blumberg and Kummer⁽²⁾ have carried out a large number of calculations based on the Lavoie *et al.*⁽¹⁸⁾ model to investigate a number of engine parameters. This work identified the leading factors which influenced the NO emissions. In Figs. 5.21-5.24 the results are given of calculations by the authors based on the methods developed by Benson *et al.*⁽¹²⁾ These calculations differ from those of Blumberg and Kummer not only in the combustion model, which is simpler, but also because allowance was made for the influence of the intake and exhaust systems on the trapped engine conditions. The same broad trends as Blumberg and Kummer were observed.

We have already noted that retarding the ignition timing reduces the maximum pressure (and hence peak temperature) (Fig. 5.5). It would therefore be expected that decreasing the spark advance would also reduce NO emissions. In Fig. 5.21 this effect is observed but the magnitude will depend on the air/fuel ratio. Reducing NO by this method, however, will reduce the engine power.

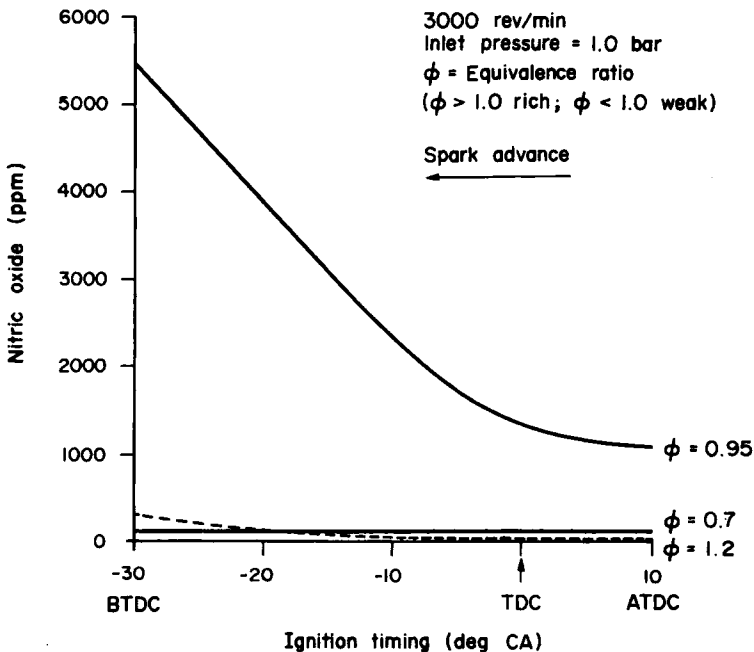


FIG. 5.21. Influence of ignition timing on nitric oxide emissions.

The effect of engine operating conditions such as speed and manifold pressure are shown in Figs. 5.22 and 5.23. The lower the engine speed, the longer the time available for the dissociation of NO, but the magnitude of the change in NO concentration will depend on the ignition timing, mixture strength and heat losses. Finally, running very lean will also reduce the NO emission (Fig. 5.7). Here we are faced with a practical problem of ensuring a uniform distribution of fuel between cylinders in multi-cylinder operation and ignition stability already referred to earlier in our discussion of ignition timing.

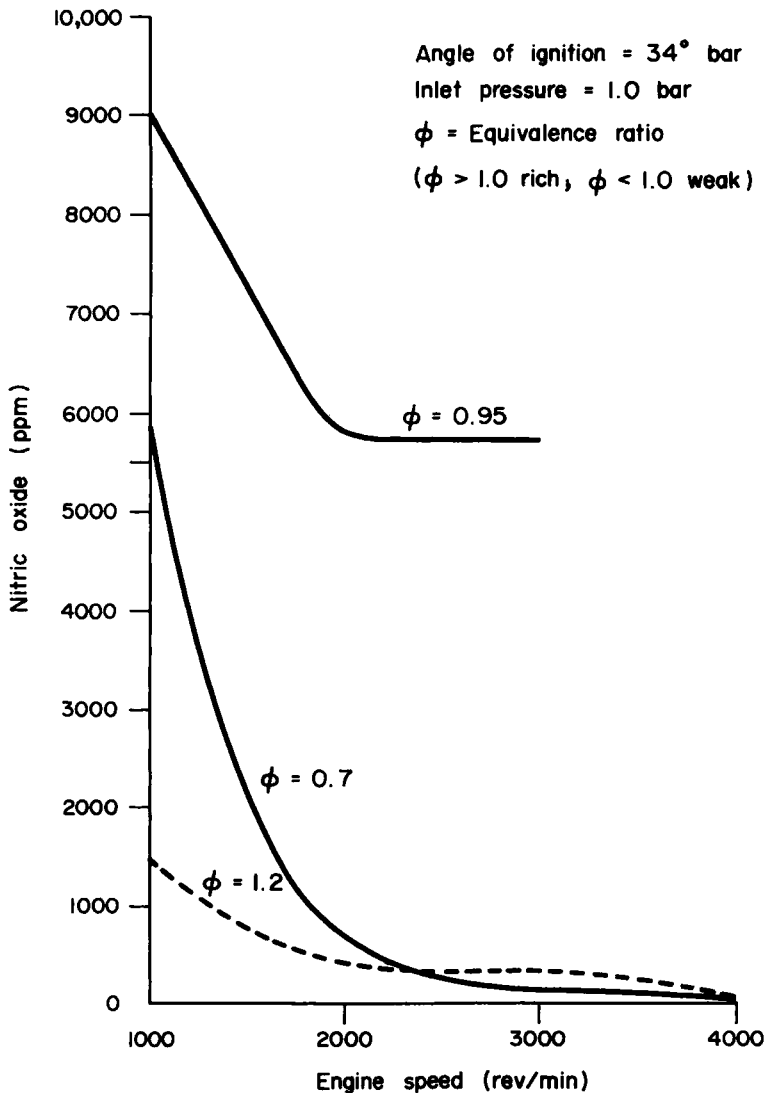


FIG. 5.22. Influence of engine speed on nitric oxide emissions.

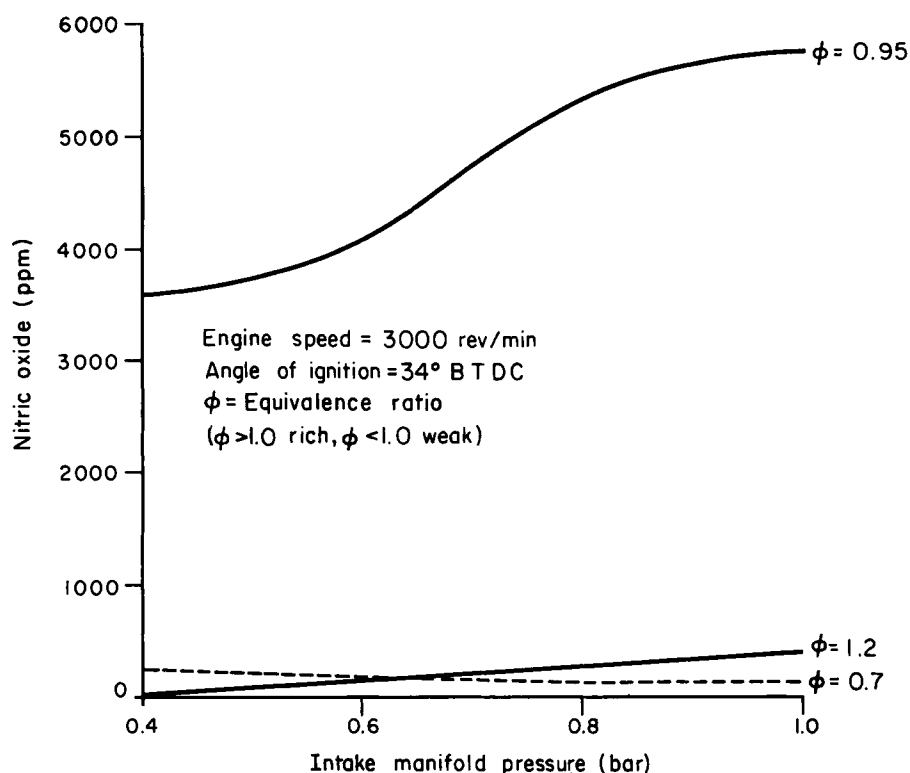


FIG. 5.23. Influence of intake manifold pressure on nitric oxide emissions.

Another method of reducing the maximum temperature and flame speed is to increase the residual fraction of the products of combustion from the previous cycle. In Fig. 5.24 it will be seen that a small change in the residual fraction has a marked significance on the peak NO.

A practical method of implementing this is to recirculate some of the exhaust gas back to the inlet manifold. This method is called exhaust gas recirculation (EGR). Some typical results obtained in the authors' laboratories on a four-cylinder engine with EGR are shown in Fig. 5.25. Here it will be seen that a reduction in peak NO concentration from about 4000 ppm to 2500 ppm was obtained with approximately 3.5% EGR. The percentage EGR decreases with increase in air/fuel ratio for a fixed valve setting. In practice, the EGR valve is shut off during idle and closed throttle.

All the above methods for reducing NO reduce the engine power and impair performance. Alternative methods have therefore been sought. One method⁽²²⁾ is to subdivide the combustion chamber into two parts—one with a rich mixture and the other with a weak mixture. The rich mixture is ignited first and it, in turn, ignites the weak mixture. It is claimed that by this method the NO emissions are reduced. These engines are called stratified charge engines.

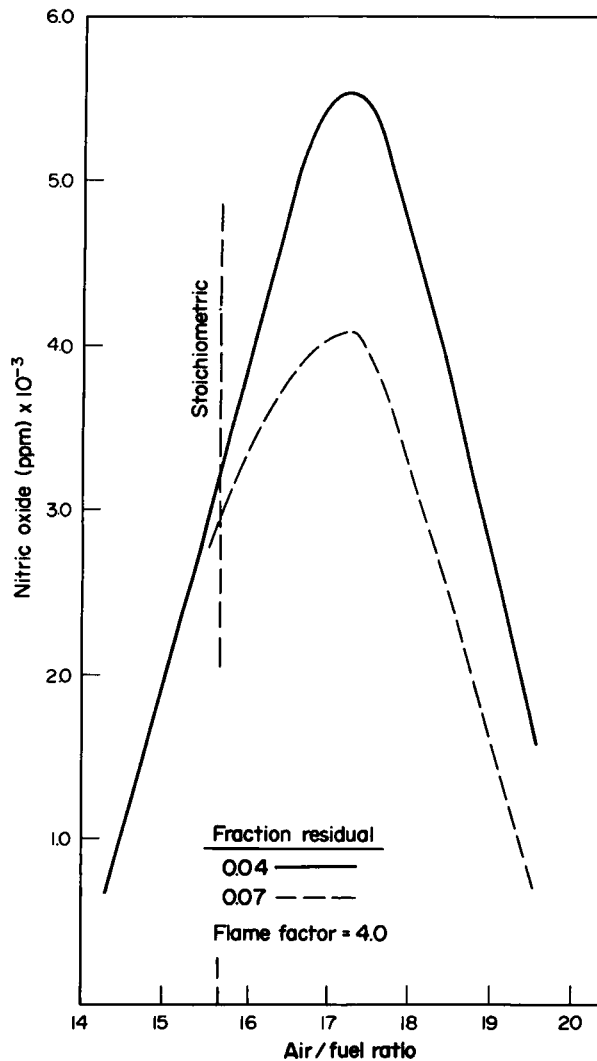


FIG. 5.24. Influence of residuals on nitric oxide concentration. (From Benson *et al.*⁽¹²⁾.)

Finally, devices have been developed for mounting on the exhaust pipe to assist in the reduction of NO to oxygen and nitrogen. These are called catalytic reactors. The reaction takes place in a reducing atmosphere in the presence of a catalyst. There is evidence to show that ammonia is formed in the reduction process.⁽¹⁴⁾ The catalysts are very susceptible to lead poisoning, and for their effective use lead-free fuels must be used.

5.6.3 Hydrocarbons

Hydrocarbons may appear in the exhaust gas from a number of sources; some of these can be directly related to unstable ignition

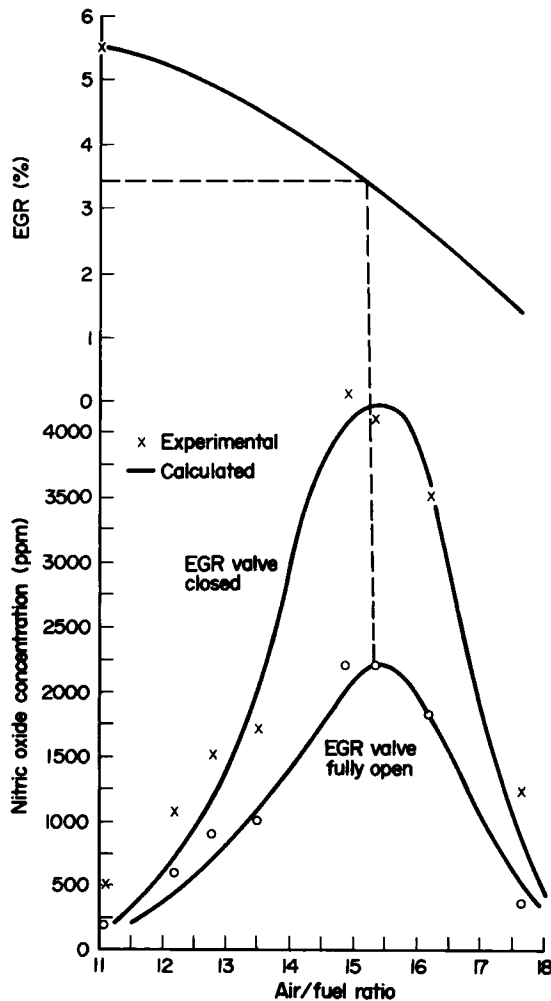


FIG. 5.25. Some typical results for a four-cylinder engine with exhaust gas recirculation.

(misfire), some to poor operation (excessive lubricating oil in exhaust due to blow-by, etc.) and some to crevices in the combustion chamber which are too narrow for the flame to enter; for example, around the valves and between the piston crown and cylinder head. The two main combustion-generated sources are the intermediate products and wall quenching.

We have already pointed out that the oxidation of hydrocarbons is a chain process with intermediate products such as aldehydes. Some of these aldehydes are stable and pass into the exhaust gas. The major source of hydrocarbon formation is wall quenching. When photographic measurements are made of the combustion process in

engines it is observed that as the flame propagates to the wall there is a thin layer adjacent to the wall where no chemical reaction except some fuel breakdown occurs. This thin layer, containing non-reacting hydrocarbons, is called the quench distance. The wall acts in two ways, firstly, as a chain breaker in the reacting mechanism, and secondly, as a heat sink which causes the temperature of the boundary layer to drop below the ignition temperature of the mixture.

The quench distance varies from 0.008 cm to 0.038 cm thickness.⁽²⁴⁾ The unburnt hydrocarbons are swept upwards by the piston ring during the exhaust stroke and the boundary layer rolls in the form of a vortex causing the hydrocarbons to enter the main stream and pass through the exhaust valve.

The quench distance is influenced by the mixture temperature, pressure, air/fuel ratio, the wall surface temperature, to the wall surface material and combustion deposits. The variation of hydrocarbon concentration with mixture strength is of similar form to the carbon monoxide curve (Fig. 5.16) with high concentration with rich mixture decreasing to a minimum at about the point of maximum temperature and then remaining almost constant, although if the mixture is too lean the concentration will increase due to misfiring.

Exhaust devices are used to reduce hydrocarbon in the gas mainly by oxidation processes in thermal or catalytic reactors. In order that the oxidation can be successfully carried out there are critical sizes for these reactors to allow for adequate residence time for the oxidation process to take place.

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Chapter 6

Heat Transfer in Engines

Notation

a	constant in Annand's eqn. (6.21) and Table 6.2	Pr	Prandtl number
A	area	q	heat flow per unit time
b	constant in Annand's eqn. (6.21) and Table 6.2	r	distance
c	constant in Annand's eqn. (6.21) and Table 6.2	Re	Reynolds number
C	particle mass per unit volume	s	particle surface area per unit mass
c_n	constants in Woschni's eqn. (6.24) ($n = 1, 2, 3$)	T	temperature, absolute
C_p	specific heat at constant pressure	T_g	gas temperature
D	cylinder bore	T_w	wall temperature
E_b	black body energy radiated per unit area per unit time	v_p	piston velocity
E_g	grey body energy radiated per unit area per unit time	V	volume
f	shape factor (Table 6.1)	V'	trapped volume (Woschni eqn. (6.24))
F	shape factor (eqn. (6.10))	w	local average velocity (Woschni eqn. (6.24))
$\bar{h}_c, h_c$	heat transfer coefficient	w_c	combustion induced velocity (Woschni eqn. (6.24))
I	radiant energy per unit solid angle	x	length
$I_{\lambda x}$	intensity of radiant energy at wavelength λ , distance x	$\alpha_{G\lambda}$	absorptivity of gas layer
k	thermal conductivity	ϵ	emissivity
k_f	thermal conductivity of gas	ϵ_g	grey body emissivity
k'	monochromatic absorption coefficient	$\epsilon_{G\lambda}$	emissivity of gas layer
L	linear length (Table 6.1)	λ	wavelength
Nu	Nusselt number	ψ	angle (Fig. 6.2)
p	pressure	ω	angle (eqn. (6.4))
p'	trapped pressure (Woschni eqn. (6.24))	ϕ	angle (Fig. 6.2)
p_o	motoring pressure (Woschni eqn. (6.24))	ρ	density
		ρ_r	reflectivity
		μ	absolute viscosity
		ν	kinematic viscosity
		σ	constant of proportionality in Stefan-Boltzmann law

6.1 BASIC PRINCIPLES

In Chapter 3 the limitations of the air standard cycles were discussed and heat loss to the walls of the combustion chamber from the burning gases was claimed to be the second most important reason for the deviation of such a calculation from an engine test result. Apart from performance calculations heat transfer is of importance because of the thermal loading of components—liner, piston, valves and cylinder head.

The reader is probably familiar with the principles of heat transfer, so only a brief description, of relevance to engine conditions, will be given here.

Heat transfer may be defined as the transmission of energy from one region to another as a result of a temperature difference between them. This may be accomplished by one or more of three methods—conduction, convection and radiation. In conduction, heat transfer is achieved by molecule to molecule contact without appreciable movement of the molecules. Although it may be claimed that heat transfer from a gas to a solid is dependent upon conduction, through a thin stationary boundary layer of gas molecules adhering to the wall of the solid, this concept is seldom used practically, and for our purposes conduction may be ignored.

Convection is the transport of energy by the movement of molecules over appreciable distances, the molecules acting as energy stores, picking up energy in high temperature regions and giving it up in low temperature regions. In internal combustion engines convection tends to reduce temperature gradients within the gases and also to transfer energy from the hot gases to cold walls. A major influence on the rate of convective heat transfer is the rate of movement of the gases.

Radiation is a process by which energy is transferred from one body to another without the use of molecular movement or contact. It may occur through a vacuum or any transparent medium, and is an electromagnetic phenomenon. It includes light, X-rays and radiowaves, etc.

In engines heat losses from the combustion chamber gases are due both to radiation and convection, but the losses have seldom been measured separately. An understanding of the two forms of heat transfer, as applied to engines, may be expected to become increasingly important.

6.1.1 Radiation

The essential facts about radiation for our purposes are summarized below. The reader is recommended to read elsewhere for greater depth and width of understanding.

Radiation consists of electromagnetic waves usually covering a variety of wavelengths. Solid bodies radiate at all wavelengths, producing a continuous spectrum. The proportion of the total energy radiated at any given wavelength is dependent upon the type of body and on its temperature. If a body at any specified temperature radiates the maximum possible amount of thermal radiation at all wavelengths, it is an ideal radiator, called a "black body". The energy radiated by a black body is proportional to the fourth power of its absolute temperature (Stefan-Boltzmann law). Thus

$$E_b = \sigma T^4, \quad (6.1)$$

where E_b is the energy radiated per unit area per unit time and σ is the constant of proportionality = $5.68 \times 10^{-8} \text{ J/s m}^2 \text{ K}^4$

Real surfaces radiate less than a black body at the same temperature. If at a given temperature the ratio of the radiative power at a given wavelength to that of a black body for the same wavelength and the same temperature is constant for all wavelengths, the body is called a grey body. For the purpose of heat transfer surfaces are usually considered to be grey. The ratio of energy radiated by a grey body to that radiated by a black body at the same temperature is called its emissivity, ϵ_g , so that

$$E_g = \epsilon_g \sigma T^4. \quad (6.2)$$

For a true grey body the emissivity ϵ_g will be a constant independent of temperature, but most real bodies are not grey and their emissivities vary with temperature. Also surfaces have directional qualities, but these may be neglected for our purposes.

Kirchhoff's law states that, at thermal equilibrium, the ratio of the emissive power of a surface to its absorptivity is the same for all bodies. For any body the absorptivity for black body radiation from a surface at temperature T is the same as the emissivity of the body at the same temperature T .

So far we have considered the total radiation emitted by a black body in all directions. This concept may be visualized by placing a hemispherical surface over a surface element dA_1 as shown in Fig. 6.1.

The hemisphere will intercept all of the radiation emitted from the surface element, but only from directly above it (taking dA_1 to be horizontal) will dA_1 be seen undistorted. When viewed from a point on the hemisphere displaced by an angle ϕ from the normal to dA_1 , the surface element will appear as the projected area $dA_1 \cos \phi$.

If we define the intensity of radiation from dA_1 , I , as the radiant energy propagated in a particular direction per unit solid angle, per unit of area of dA_1 as projected on a plane perpendicular to the direction of propagation, the rate of radiant heat flow from dA_1 to dA_2 is

$$dq_{1-2} = I \cos \phi \, dA_1 \, dA_2 / r^2, \quad (6.3)$$

$$= I \, dA_{1p} \, d\omega_1, \quad (6.4)$$

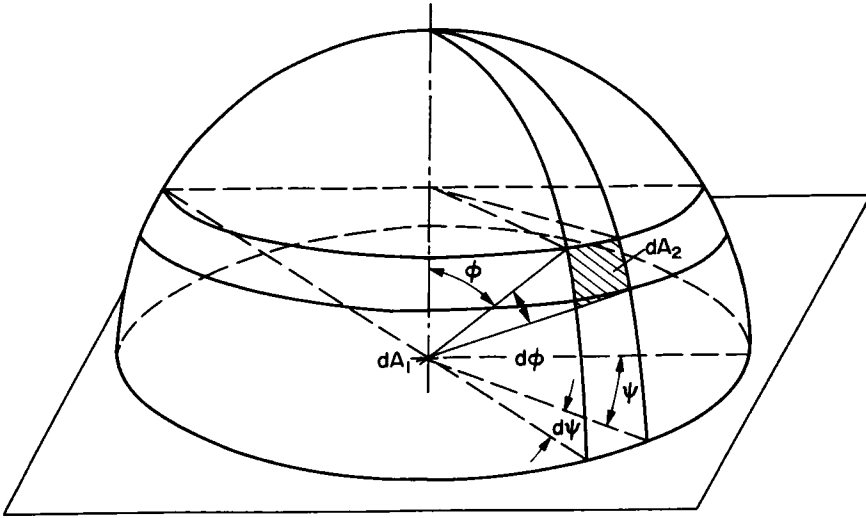


FIG. 6.1. Radiation from surface element to circumscribing hemisphere. (From Kreith⁽³⁹⁾ by courtesy of the International Text Book Co.)

where $d\omega_1 = (dA_2/r^2) =$ the solid angle subtended by dA_2 at dA_1 and $dA_{1p} = dA_1 \cos \phi =$ the projected area of dA_1 in the direction ϕ .

If the intensity of radiation from surface dA_1 does not vary with the direction ϕ , as for a diffuse surface, then integrating the radiation intercepted by all such elemental areas as dA_2 over the total surface of the hemisphere, the total rate of emission per unit area of A_1 , or emissive power will be

$$E = \int_{A_2} I \cos \phi \frac{dA_2}{r^2}, \quad (6.5)$$

but
$$\frac{dA_2}{r^2} = d\omega_1 = \frac{r \sin \phi \, d\psi \, r \, d\phi}{r^2} = \sin \phi \, d\psi \, d\phi.$$

Therefore
$$E = I \int_0^{2\pi} \int_0^{\pi/2} \sin \phi \cos \phi \, d\phi \, d\psi, \quad (6.6)$$

$$= \pi I. \quad (6.7)$$

So far we have considered emission and absorption of radiation by various surfaces but not the net flow between them. To do this we have to determine the fraction of the total emission from each of the radiating surfaces which reaches and is absorbed by the others. If only black body surfaces are involved, all incident radiation is absorbed and only the geometric relation between the surfaces has to be considered.

Figure 6.2 shows the surfaces of two black bodies separated by a transparent medium.

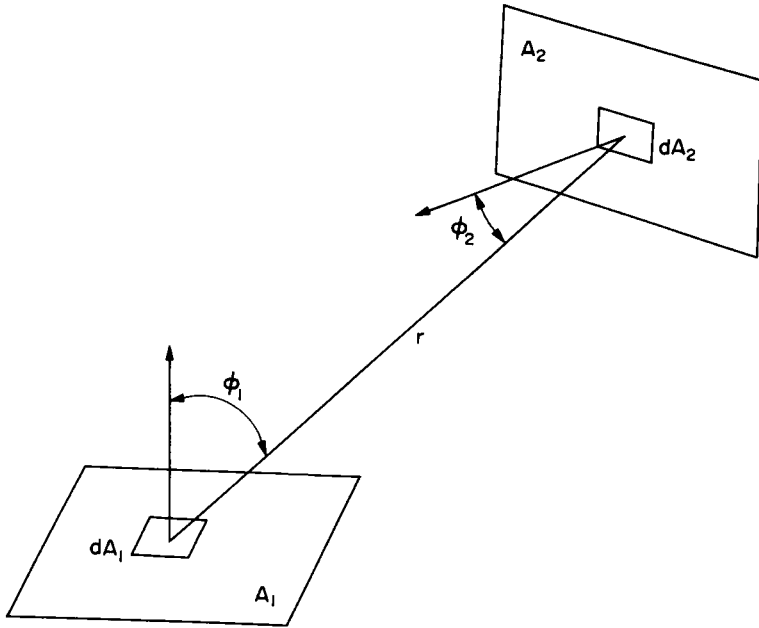


FIG. 6.2. Radiation between two elemental surface areas.

To determine the fraction of the energy radiated from surface A_1 that strikes surface A_2 , consider the two elements dA_1 and dA_2 , distance r apart:

$$dq_{12} = I \cos \phi_1 dA_1 d\omega_{12}, \quad (6.8)$$

where I_1 is the intensity of radiation from dA_1 , $dA_1 \cos \phi_1 =$ projection of area dA_1 as seen from dA_2 , $d\omega_{12}$ is the solid angle subtended by dA_2 with respect to centre point of dA_1 , but

$$d\omega_{12} = \cos \phi_2 dA_2 / r^2$$

$$\text{giving} \quad dq_{12} = E_{b1} dA_1 (\cos \phi_1 \cos \phi_2 dA_2 / \pi r^2), \quad (6.9)$$

and $\cos \phi_1 \cos \phi_2 dA_2 / \pi r^2$ is the fraction of the total radiation emitted from dA_1 that strikes dA_2 .

Similarly, $dq_{21} = E_{b2} dA_2 (\cos \phi_2 \cos \phi_1 dA_1 / \pi r^2)$. The net rate of radiation between the two surfaces A_1 and A_2 will be

$$q = (E_{b1} - E_{b2}) \int_{A_1} \int_{A_2} \frac{\cos \phi_1 \cos \phi_2 dA_1 dA_2}{\pi r^2}.$$

This shows that the net rate of heat flow between two black bodies may be determined by evaluating the radiation from either to the other but replacing its emissive power by the difference between the emissive powers of the two surfaces.

If we divide the double integral by the total area of one of the surfaces the result is called the shape factor for the system evaluated on the basis of that surface. Thus shape factor on basis of area A_1 is

$$F_{12} = \frac{1}{A_1} \int_{A_1} \int_{A_2} \frac{\cos \phi_1 \cos \phi_2 dA_1 dA_2}{\pi r^2} \quad (6.10)$$

F_{12} is the fraction of the total radiant energy leaving A_1 that is intercepted by A_2 .

Thus

$$q = (E_{b1} - E_{b2}) F_{12} A_1 = (E_{b1} - E_{b2}) F_{21} A_2. \quad (6.11)$$

If surface A_1 is totally enclosed by surface A_2 it can be seen that all of the net radiation that leaves A_1 will be intercepted by A_2 , so that the shape factor F_{12} should be unity. This will be true unless some of the radiation from A_1 is intercepted by another part of A_1 . In this latter case the true area A_1 may need to be corrected to become an effective area.

If the surfaces are opaque but not black the analysis becomes difficult because all of the incident radiation is not absorbed—some is reflected. The next simplest case is if the surfaces are grey. It is not intended to delve deeper into these fascinating complexities but of interest may be the simple case of radiation between two grey concentric spheres. Their net radiation becomes

$$q = (E_{b1} - E_{b2}) F_E F_{12} A_1, \quad (6.12)$$

where

$$F_E F_{12} A_1 = \frac{1}{\rho_{r1}/A_1 \epsilon_1 + 1/A_1 + \rho_{r2}/A_2 \epsilon_2} \quad (6.13)$$

and ρ_r = reflectivity and ϵ = emissivity (note, as surfaces are opaque, $\rho_r + \epsilon = 1$).

Radiation may also occur from gases. Many common gases and gas mixtures, such as nitrogen, oxygen and hydrogen, have symmetrical molecules and are practically transparent to thermal radiation, they neither emit nor absorb appreciable amounts of radiation at temperatures of practical interest. However, some gases and vapours of interest, such as carbon dioxide, water vapour, carbon monoxide and hydrocarbons are not so neutral.

Gases do not radiate at all wavelengths but emit and absorb only in narrow regions of wavelength, called bands. Thus the radiation bands for carbon dioxide lie between wavelengths of 2.36 and 3.02 μ ,

4.10 and 4.80μ and 12.5 and 16.5μ ; for water vapour between 2.24 and 3.27μ , 4.8 and 8.5μ and 12 and 25μ . Even if complete absorption/emission within these bands was achieved, i.e. the quantities of gas involved were sufficient to achieve black body conditions within these bands, the intensity of radiation (absorption) would only be a part of that obtained from an equivalent black body. At 473°K the parts would be 12% for carbon dioxide and 67% for water vapour, at 1273°K 10.5 and 46% respectively and at 2273°K 6 and 24%. These changes, which indicate that the intensity of gas radiation increases less rapidly with temperature than does black body radiation, are because the wavelength at which maximum intensity for black body radiation occurs become shorter as temperature rises, moving to shorter wavelengths than those at which the gas radiation occurs.

As is implicit in the statement above, gas radiation is not a surface phenomena as is usual for a solid body (apart from a few transparent substances and even these are relatively opaque). To calculate the radiation emitted or absorbed by a gas layer, its thickness, pressure and shape, as well as its surface area, must be taken into account.

If radiation of a single wavelength and intensity $I_{\lambda 0}$ passes through a gas layer of thickness L , the radiant energy absorbed in a distance dx , distance x into the layer is

$$dI_{\lambda x} = -k'_{\lambda} I_{\lambda x} dx, \quad (6.14)$$

where $I_{\lambda x}$ is the intensity at a distance x and k'_{λ} is the monochromatic absorption coefficient, a proportionality constant, value depending on pressure and temperature of the gas.

Integration between the limits $x = 0$ and $x = L$ gives

$$I_{\lambda L} = I_{\lambda 0} e^{-k'_{\lambda} L}. \quad (6.15)$$

So energy absorbed by gas layer $= I_{\lambda 0} - I_{\lambda L} = I_{\lambda 0}(1 - e^{-k'_{\lambda} L})$.

$(1 - e^{-k'_{\lambda} L}) = \text{absorptivity } \alpha_{G\lambda} = \text{emissivity } \epsilon_{G\lambda}$ of the gas layer. Such calculations would need to be done and summed over all the radiation bands. For large thicknesses, gas radiation approaches black body radiation within the wavelengths of the appropriate bands.

6.1.2 Radiation from Clouds of Solid Particles Such as Soot

A particle cloud may be treated as an absorbent medium similar to the gas layer described earlier, provided that the particles are large enough—compared with the radiation wavelength—not to produce scattering. The absorption coefficient of a suspension of black particles having a total particle surface per unit mass s and a total mass per unit volume C can be shown to be given by

$$k' = 0.25Cs. \quad (6.16)$$

An isothermal cloud of grey particles may be replaced by an equivalent grey body. To do this the fraction of the energy emitted by each particle in the cloud which emerges from each element of area of the cloud boundary must be determined, making allowance for absorption on the way, and then it must be summed over all particles and all surface elements. Analytical solution is possible only in very simple cases and numerical methods must be used in others. It is found convenient to express the results in the form

$$E = 1 - e^{-fk'L}, \quad (6.17)$$

where L is a chosen linear dimension and f a shape factor for the cloud. Some values are presented in Table 6.1 and Fig. 6.3 shows the resulting values of $E^{(1)}$

TABLE 6.1. Approximate Emissivity of a Uniform Isothermal Cloud
Values of L and f for use in equation $E = 1 - e^{-fk'L}$

Cloud shape	Linear dimension L	Values of f		
		$k'L=0$	$k'L=1$	$k'L=2$
Arbitrary convex cloud	Ratio of volume to surface	4	3.4	3.4
Sphere	Diameter	0.67	0.64	0.61
Cylinder	Diameter			
Height = dia.		0.67	0.61	0.55
Height = 5 dia.		0.91	0.82	0.75
Height $\gg$ dia.		1	0.91	0.84
Rectangular parallelepiped				
Cube	Edge	0.67	0.59	0.58
1:1:5	Shortest edge	0.91	0.77	0.68
1:1: ∞	Shortest edge	1	0.85	0.75
1: ∞ : ∞ (slab)	Thickness	2	1.5	1.4

Combustion of pulverised coal. (From Field et al.⁽¹⁾ by courtesy of the Institute of Fuel).

6.1.3 Convective Heat Transfer

When a fluid flows along a surface the fluid particles adjacent to the surface have zero velocity relative to the surface. Other fluid in the vicinity is retarded by viscous shear if the flow is laminar. If the flow is turbulent, retardation will be due to turbulent shear as well. A short distance from the surface the velocity of the fluid approaches that of the undisturbed free stream. The region of substantial velocity change is called the hydrodynamic layer. Even if the flow is turbulent there will be a very thin

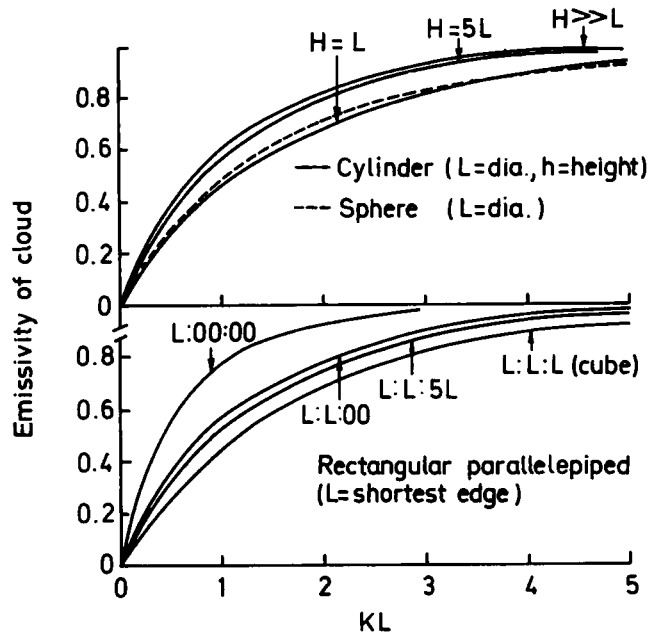


FIG. 6.3. Variation of the whole-body emissivity of a uniform cloud with shape. (From Field et al.⁽¹⁾ by courtesy of the Institute of Fuel.)

layer adjacent to the wall that is laminar—called the laminar sublayer.

From this brief description it may be deduced that very near to the surface heat can only flow by conduction through the stationary layer of fluid. Further from the wall the movement of the fluid helps heat transfer by transporting material and energy by turbulent motion. If the temperature gradient $\partial T/\partial y$ at the surface was known, the rate of heat transfer per unit area q/A would be $-k_f(\partial T/\partial y)_{(y=0)}$.

Usually this gradient is not known, and a heat transfer coefficient $\bar{h}_c$ is used so that, for a steady flow situation,

$$\frac{q}{A} = \bar{h}_c(T_s - T_\infty), \quad (6.18)$$

where T_s is surface temperature and T_∞ is the bulk temperature of the fluid (usually the two temperatures known). Equating these two concepts and introducing a length L representative of the system leads to the relationship

$$\frac{\bar{h}_c L}{k_f} = \frac{-\left(\frac{\partial T}{\partial y}\right)_{(y=0)}}{\frac{(T_s - T_\infty)}{L}} \quad (6.19)$$

$(\bar{h}_c L)/k_f$ is a dimensionless number, called the Nusselt number, and is the ratio of the temperature gradient at the surface to a representative temperature gradient of the system. If the value of the Nusselt number is known, then the convective heat transfer coefficient may be calculated from $\bar{h}_c = Nu(k_f/L)$. This shows that for similar systems the heat transfer coefficient will be inversely proportional to the significant length dimension.

There are a number of methods available for obtaining heat transfer coefficients, such as solving the boundary layer conditions, by analogy with mass and momentum transfer or by dimensional analysis combined with experiments. Unfortunately they tend to be applicable to much simpler systems than the internal combustion engine.

Nonetheless, it is of interest to note that such studies have been made and additional dimensionless groups have been found to be of value. One is the group $VL\rho/\mu$, the Reynolds number, and another is the Prandtl number $C_p\mu/k_f$, known as Re and Pr respectively. According to the principle of similarity the behaviour of two systems will be similar if the ratios of their linear dimensions, forces, velocities, etc., are the same. In geometrically similar systems with forced convection if the ratio of inertia forces to viscous forces, that is the Reynolds number, is the same the velocity fields will be similar. So similar flow conditions may be expected if the Reynolds number is the same for any two systems. Again, the ratio of kinematic viscosity μ/ρ , which affects the velocity distribution, to thermal diffusivity $k_f/\rho C_p$, which affects the temperature profile, gives the Prandtl number. So the Prandtl number may be regarded as a dimensionless group that relates temperature distribution to velocity distribution. The Nusselt number has already been shown to relate the temperature gradient at a fluid/surface interface to a reference temperature gradient. These three groups may therefore be expected to be interrelated, i.e. $Nu = f(Re, Pr)$.

The ideas developed above have been largely derived from steady flow considerations. Applying them to engine conditions may be thought to be a dubious procedure but in general has proved to be useful.

6.2 HEAT TRANSFER IN INTERNAL COMBUSTION ENGINES—A SURVEY

The processes inside the combustion chamber of an intermittent type engine are very complex, involving rapidly varying gas pressure and temperature and local fluid velocities which may also vary more or less rapidly. Heat flux into the walls varies throughout the engine cycle from a small negative value to a positive value reaching up to several megawatts per square metre, and also varies spatially due to differences in local gas temperatures and velocity and doubtless also due to differences in orientation of the wall to the radiating flames. As yet an adequate theoretical analysis of heat

transfer under these conditions has not been achieved although a number of studies aiming to improve this situation have taken place. Measurements of air movement in an engine cylinder have been made by Fitzgeorge,⁽²⁾ Dent⁽³⁾ and Tindal⁽⁴⁾ during the compression stroke but not during combustion and expansion. Some attempt has also been made to predict convective heat transfer coefficients from air movement using momentum concepts.⁽⁵⁾

These studies do not deal with any effect due to combustion itself. At one time it was widely held that combustion would in itself produce a major increase in turbulence but recent studies indicate that this is not so.⁽⁶⁾ However, local velocity changes can be caused by expansion of the hot burned products. In the spark ignition engine expansion of the charge produces a clearly defined difference between the rate of flame movement and the speed of propagation relative to the unburnt gas. Calculation for typical cases has given velocity changes due to this cause of the order of 10 m/s—quite large compared with fluid velocities normally found in this type of engine before the passage of the flame. This effect has also been shown by Summers⁽⁷⁾ in a single-cycle engine simulator. Figure 6.4 shows the effect observed in which combustion increased Nusselt number considerably. Similar effects are expected in compression ignition engines. In indirect injection engines, in particular, the air motion produced by expansion is deliberately channelled and consequently produces high local velocities, and so convective heat transfer. Henein⁽⁸⁾ presents some observations on an engine with an energy cell combustion system with analysis showing high values of heat transfer associated with charge velocities of the order of 500 m/s. In direct injection engines charge motion produced by combustion is much smaller and less well defined.

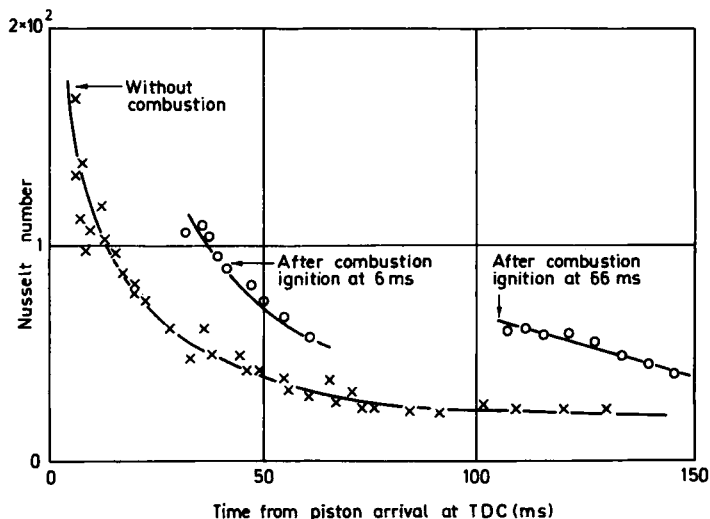


FIG. 6.4. Nusselt number variation with time, with and without combustion, in single-cycle simulator.

At one time convective heat transfer was considered to be by far the major form with radiation relatively small, perhaps negligible. The importance of radiation, at least in the compression ignition engine, is now firmly established, even if calculation on this basis is not yet usual.

Radiant heat transfer in the spark ignition engine has been investigated less than it has been in the compression ignition engine. Under the usual conditions of homogeneous deflagration no solid intermediate products are formed and only gas phase radiation has to be considered—effectively radiation from carbon dioxide and water vapour. Some assessments have given typical maximum effective emissivity about 0.1, with maximum radiant flux of half a megawatt per square metre and total maximum flux of 10 times this. As radiation diminishes much more rapidly than convection as temperature falls during expansion, the overall effect could be of negligible radiation relative to convective heat transfer.

These assessments have been quoted from Annand⁽⁶⁾ who adds that although this view may be correct it is not substantiated, and there are grounds for doubt, such as that radiation from the flame itself has not been calculated, radiation from carbon dioxide and water vapour was based upon data obtained at lower partial pressures than exist in engine cylinders and the non-uniformity of cylinder temperature and non-equilibrium cylinder conditions may affect the result. Nonetheless, radiation from premixed flames is normally much less than from diffusion flames, and this seems to be borne out by experience of premixed combustion in the compression ignition engine, Lyn and Valdamis.⁽⁹⁾

For the compression ignition engine there is a considerable amount of published evidence showing that radiation can account for an appreciable fraction of the total. Thus Oguri,⁽¹⁰⁾ Flynn⁽¹¹⁾ and Sitkei⁽¹²⁾ agree in showing measured radiant heat transfer amounting to between 20% and 30% of the total, averaged over the complete engine cycle. Belinkii⁽¹³⁾ made measurements on a small, open-chamber engine and concluded that the diesel engine flame radiated as a grey body within the rather narrow range of wavelengths covered by his experiments. He showed emissivities ranging from 1.0 at top dead centre down to about 0.6 at 50° after top dead centre. The similar but more extensive work by Flynn⁽¹¹⁾ gave a somewhat non-grey distribution. Typical observations are shown in Fig. 6.5.

So far calculation of radiation from consideration of soot cloud density and size and temperature has hardly begun although Annand⁽⁶⁾ has made some tentative suggestions towards this end, with the flame treated as a grey body in a black box, filling the box so that form factors are not needed. Even so, soot temperature and quantity has to be estimated, and neither is as yet readily calculable though knowledge is accumulating (see Khan⁽¹⁴⁾).

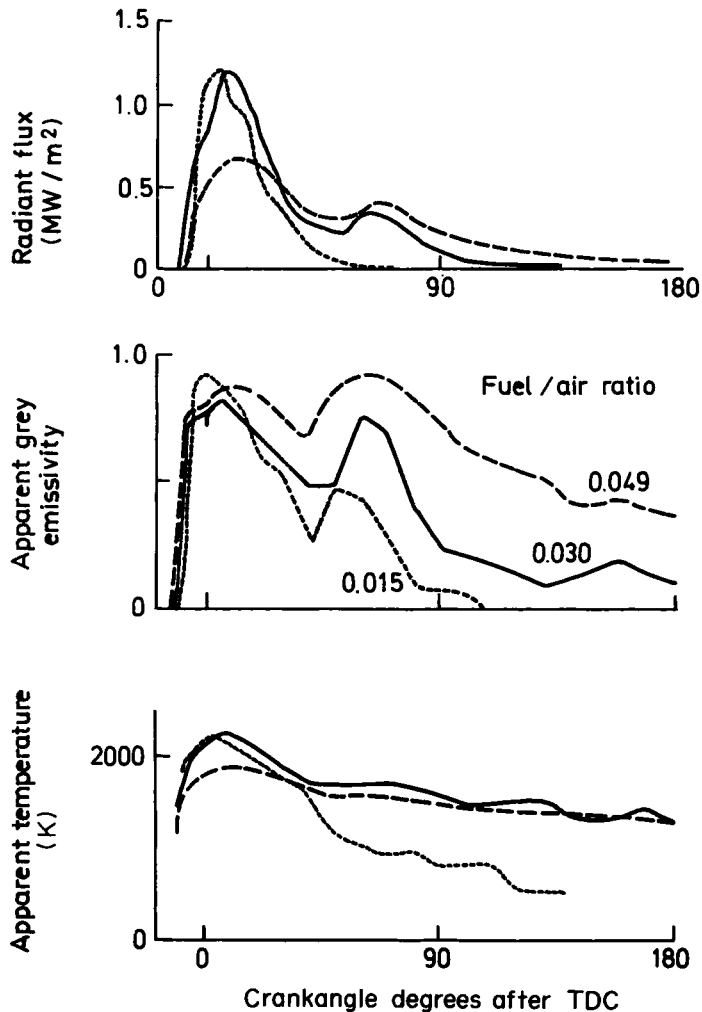


FIG. 6.5. Typical observations of radiant heat transfer in a diesel engine. (From Flynn *et al.*⁽¹¹⁾ by courtesy of the Society of Automotive Engineers.)

6.3 HEAT TRANSFER IN INTERNAL COMBUSTION ENGINES—SOME PRACTICAL CONSIDERATIONS

Heat transfer in engines may be considered in two ways:
 (1) the overall effect which provides, if the engine is running at a

steady load and speed, a nominally steady, or mean, rate of heat loss to the combustion chamber walls and hence to a coolant; (2) the instantaneous rates that occur during a cycle of events in the engine cylinder. The second way may be integrated with respect to time to give the first. This second way is of particular interest as it is involved in the thermodynamic cycle calculations considered in Chapters 8 and 9. It also allows for a more detailed and deeper study of the heat transfer process than does the overall approach, and so gives the hope of better predictive value in the future, even though there is doubt at present. As the attempts to tackle the problem fundamentally have not yet arrived at conclusions of practical value, as may be deduced from the previous section, semi-empirical studies that have been found useful will be dealt with hereafter.

Such studies have been based mainly on temperature measurements in components forming the combustion chamber walls. These may be steady values, obtained throughout a component, from which isothermals may be drawn as outlined in Chapter 11, and mean heat fluxes deduced; or surface transient temperatures may be obtained and instantaneous heat fluxes into the combustion chamber surfaces calculated. Figure 6.6 shows typical variations in surface temperature in a medium speed engine and Fig. 6.7 the resultant calculated instantaneous heat flux diagram.

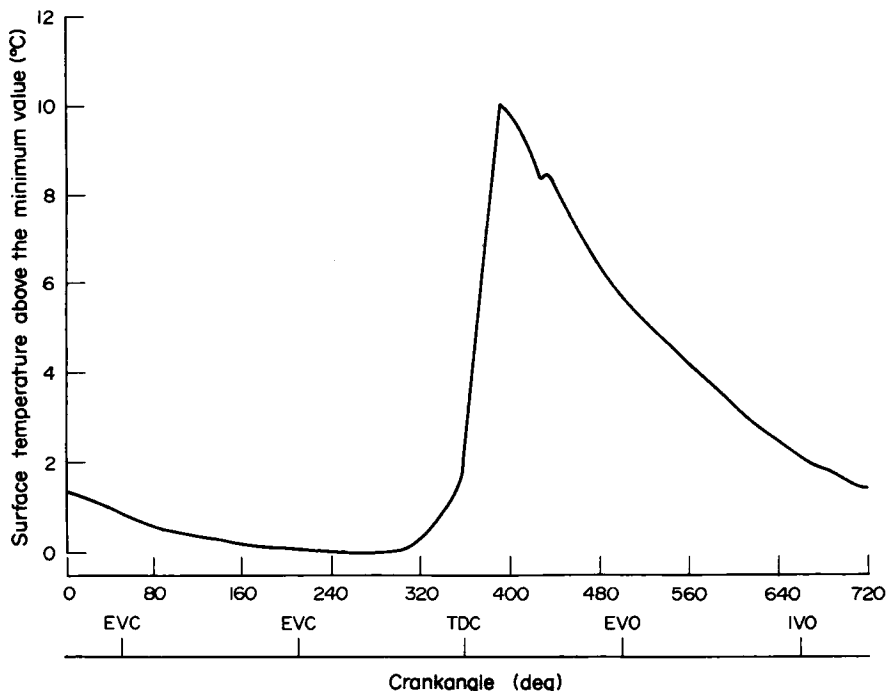


FIG. 6.6. Typical values of cyclic variation of surface temperature on cylinder head of diesel engine (b.m.e.p. 10 bar).

The fluctuation in temperature at the surface is rapidly reduced to a steady state value within a few millimetres of the surface. Typical calculated values of temperature are shown in Fig. 6.8 to illustrate this.

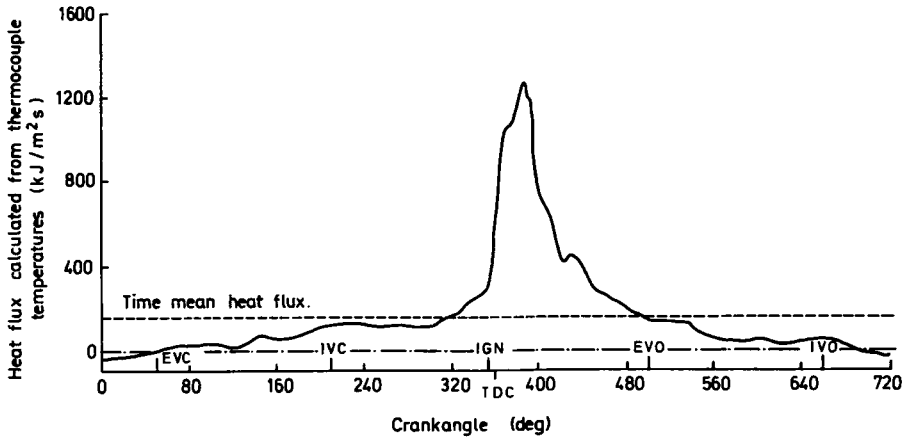


FIG. 6.7. Instantaneous heat flux at cylinder head thermocouple. (From Whitehouse,⁽²⁸⁾ by courtesy of the Institution of Mechanical Engineers.)

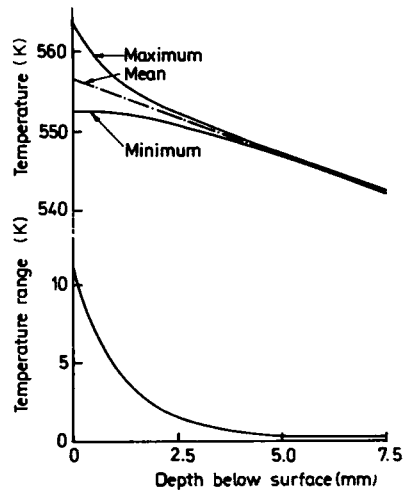


FIG. 6.8. Cyclic temperature variation in surface region of a piston crown. (From Whitehouse et al.,⁽¹⁵⁾ by courtesy of the Institution of Mechanical Engineers.)

For most practical purposes the component temperatures may be taken to be the steady state values. Only rarely have the surface thermal stresses been considered to be worthy of consideration, as in one case of piston thermal loading.⁽¹⁵⁾ These steady state temperatures vary widely within components, depending both upon the heat transfer into the component from the burning substances, which is our main concern, and upon the component design, particularly cooling arrangements, which is a major concern of the engine designer.

Figures 6.9-6.12 illustrate these points showing component temperatures in a medium speed two-stroke diesel engine with different piston-cooling arrangements and with two different cooling fluids (oil and water). These also show heat flux lines, normal to the isothermals, which show the direction of flow of heat through the component. The heat flux per unit area at the combustion chamber surfaces can be evaluated from such diagrams if the conductivity of the component is known.

The design engineer may use empirical information of this sort, but for prediction of heat transfer in a new engine, improvement on such direct empiricism is sought. Semi-empirical models in use are mainly based upon single-zone models of the combustion process in which it is assumed that the cylinder temperature and other factors are uniform. The method can be extended to multi-zone models. These semi-empirical models will now be examined.

6.4 INSTANTANEOUS HEAT TRANSFER CALCULATIONS

The methods outlined in the following sections show how calculations are usually performed—for practical engineering purposes—if estimates of instantaneous values are required. These methods are all dependent upon an estimate of the instantaneous conditions in the engine combustion chamber, and in practice usually form part of cycle calculations such as those shown in Chapters 8 and 9. It is also assumed that the ideas generated from the steady flow convective heat transfer concepts given in the first part of this chapter are valid for each instant of time, i.e. quasi-steady calculations are correct. This concept has been challenged by Overbye⁽¹⁶⁾ who has shown that the heat flux and the overall driving temperature difference will normally be out of phase. However, the shift in phase is unlikely to be of importance and the quasi-steady approximation is unlikely to be bettered in the foreseeable future. It may also be noted that other deficiencies of greater significance can arise from the assumptions used—such as mean gas temperatures when reality involves a complex mixture of gases at different temperatures.

6.4.1 Single-zone Heat Transfer Calculations

As a matter of historical interest reference may be made to some of the numerous investigators who have reported on work of this kind (refs. 17-22). Annand's paper⁽²³⁾ gives a critical examination of much of the earlier published work, using dimensional analysis to indicate fundamental weaknesses in various equations put forward for

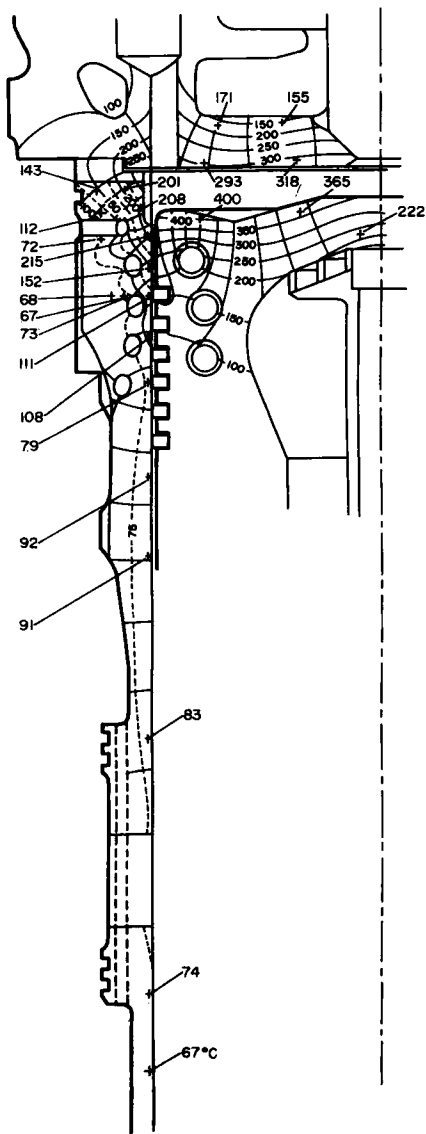


FIG. 6.10. Two-stroke engine : isothermal chart of cylinder head, liner and oil-cooled piston. (From Steiger and Aue,⁽⁴⁰⁾ by courtesy of the Institution of Mechanical Engineers.)

Engine operating data:

b.m.e.p.:	10.2 bar
Mean piston speed:	7.0 m/s
Supercharging pressure:	2.4 at a
Relative air weight:	6.3 kg/h.p.h.
Scavenging air temp.:	45°C
Cooling water temp.(outlet)	65°C
Cooling oil temp.(inlet)	40°C

Remark

Heat flow lines are spaced 2000 kcal/h apart

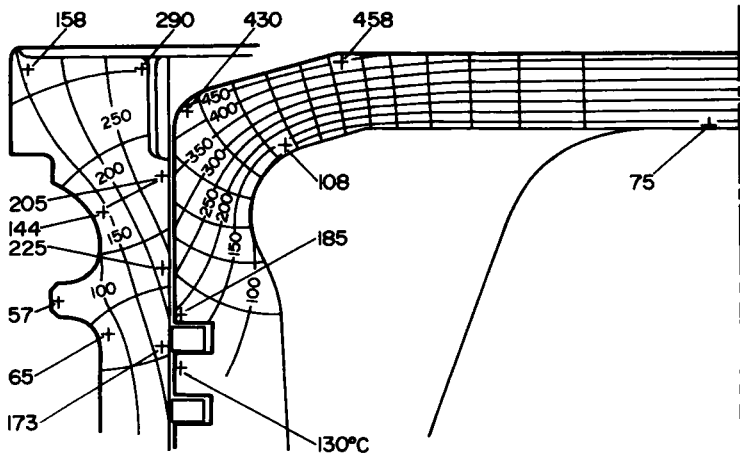


FIG. 6.11. Two-stroke engine : isothermal chart of water-cooled piston and liner. (From Steiger and Aue,⁽⁴⁰⁾ by courtesy of the Institution of Mechanical Engineers.)

Engine operating data:

Remark

b.m.e.p.:	9.7 bar	Heat flow lines are spaced 10,000 kcal/h apart
Mean piston speed:	6.1 m/s	
Supercharging pressure:	2.1 at a	
Relative air weight:	6.9 kg/h.p.h.	
Cooling water temp. (outlet)	57°C	
Piston cooling water temp. (outlet)	45°C	

heat transfer calculations and proposing a more satisfactory alternative.

Few of the multitude of formulae referred to are now of much more than historical interest. Three have been of use to the authors in cycle calculations at various times and are known to be still in use. These are by Eichelberg, by Annand and by Woschni, i.e.

$$\text{Eichelberg } \frac{q}{A} = \frac{2.43}{10.47} \times 10^{-3} v_p^{\frac{1}{3}} (pT)^{\frac{1}{2}} (T_g - T_w) \text{ kJ/m}^2\text{s} \quad (6.20)$$

$$\text{Annand } \frac{q}{A} = a \frac{k}{D} (Re)^b (T_g - T_w) + c (T_g^4 - T_w^4), \text{ where } Re = \frac{\rho v_p D}{\mu}. \quad (6.21)$$

$$\text{Woschni } \frac{q}{A} = C_3 D^{-0.2} p^{0.8} T_g^{-0.53} (C_1 v_p + C_2 \frac{VT'}{p'VT} (p - p_o))^{0.8} (T_g - T_w). \quad (6.22)$$

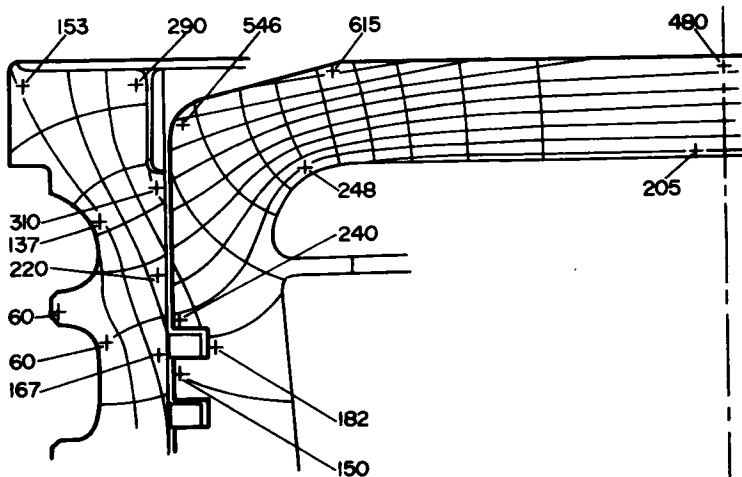


FIG. 6.12. Two-stroke engine : isothermal chart of oil-cooled piston and liner. (From Steiger and Aue,⁽⁴⁰⁾ by courtesy of the Institution of Mechanical Engineers.)

Engine operating data:

b.m.e.p.:	9.7 bar
Mean piston speed:	6.1 m/s
Supercharging pressure:	2.1 at a
Relative air weight:	6.9 kg/h.p.h.
Scavenging air temp.:	30°C
Cooling water temp.(outlet):	57°C
Piston cooling oil temp. (outlet):	45°C

It may be noted that Eichelberg's equation has the virtues of simplicity and explicitness. There are no unknown or variable constants so it is very easy to use. It is not dimensionally homogeneous and many of the relationships appear quite arbitrary. To use such a formula in conditions outside those where experience of its reliability has been gained is therefore questionable. Modified versions, with different values of the initial constant have been used.

The Annand and Woschni expressions although different in form are both based largely on turbulent convection considerations. Annand shows that the Prandtl number for the gases forming engine cylinder contents will be approximately constant at a value of 0.7, claims that the Reynolds number is the major parameter affecting convection and that the convective heat transfer relationship may be represented by $(Nu) = a(Re)^b$, where a and b are constants. This gives the first term in Annand's equation. The second is a straightforward radiation term assuming grey body radiation.

Woschni's equation is also based upon a similar forced convection model, with the index b specified as 0.8, as for turbulent flow in pipes. The constant C_3 is a number obtained from Hausen's equation⁽²⁵⁾ for turbulent flow in pipes. The Reynolds number is replaced by pressure and temperature relationships for the gases, so that the equation is in the parameters normally used. From the above considerations Woschni's expression for the heat transfer coefficient would be

$$h_c = C_3 D^{-0.2} p^{0.8} w^{0.8} T_g^{-0.53}, \quad (6.23)$$

where w is the local average gas velocity,

$$= C_1 v_p. \quad (6.24)$$

The value of the constant C_1 was obtained from experiments with a motored four-stroke diesel engine with two inlet valves and no swirl. During combustion and expansion, however, radiation and additional combustion-induced turbulence is allowed for by adding on a velocity $w_c = (C_2(VT'/p'V'))(p-p_0)$, where the pressure difference term is the difference between cylinder pressure with combustion and cylinder pressure at the same point in the cycle, for the same conditions except for the absence of any combustion.

Both the Annand and the Woschni formulae have constants which may be adjusted to give reasonable overall heat transfer values, and this can be useful if experience or available information is adequate. Some values of the constants that have been used are given at the end of the chapter.

It should be noted that the gas temperature is the mean value for the cylinder, whereas in reality the cylinder contents are not homogeneous. These formulae have been used with single-zone cycle calculation models, and that anomalies should become apparent between calculation and experience is not unexpected. One such anomaly is that heat transfer to the cylinder walls would apparently be evenly distributed both in time and place. This has been found not to be so. Reference may be made to papers by Seale and Taylor,⁽²⁶⁾ Annand and Ma⁽²⁷⁾ and Whitehouse⁽²⁸⁾ which illustrate these points.

Figure 6.13 from the last of these three papers shows another apparent anomaly whereby instantaneous heat fluxes obtained under a variety of operating conditions are found to have much less variation in maximum values than would be expected from single-zone calculation using Annand's equation.

Accepting that any of the heat transfer equations mentioned is best thought of as unreliable except in so far as experience justifies its use and in particular the use of given values of the constants involved, it may also be remarked that for performance calculations a high accuracy is not of very great significance.

The authors must acknowledge a prejudice, compounded partly of local patriotism and greater experience, in favour of the use of the Annand equation. It may also be defended on the grounds that it does separate, as logically one should, the convective and the

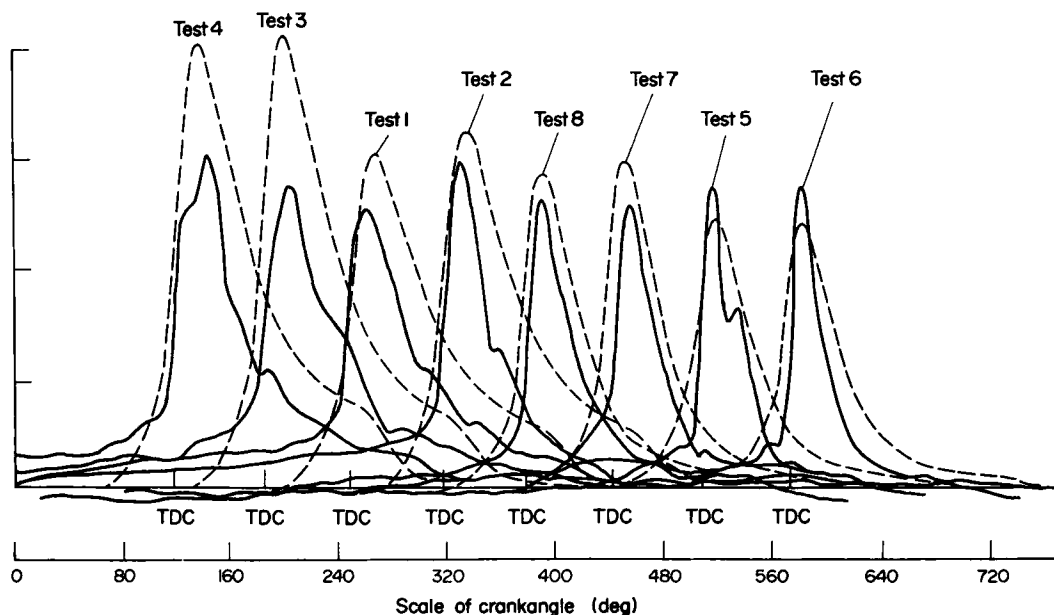


FIG. 6.13. Instantaneous heat fluxes for a range of conditions. (From Whitehouse,⁽²⁸⁾ by courtesy of the Institution of Mechanical Engineers.)

radiative components of heat transfer. Calculations with the single-zone model tend to suggest that radiation is small. The experience of some research workers suggest that it is not. Ebersole *et al.*⁽²⁹⁾ indicate that radiation may comprise up to 40% of the total heat transfer and mention is made by Borman *et al.*⁽³⁰⁾ of instantaneous values of perhaps 70% during combustion. Such quantities are thought due to radiation from hot soot particles at temperatures well above the mean gas temperature. This would also agree with comments by Alcock⁽³¹⁾ that changes in wall temperatures have much less effect on heat transfer than would be expected from mean gas temperature considerations.

These various effects are not unexpected if the heterogeneity of engine combustion and the fourth power effect of temperature on radiation is considered.

6.4.2 Multi-zone Heat Transfer

These considerations inevitably lead to the conclusion that only a multi-zone model of the happenings in the engine cylinder during combustion and expansion can hope to produce calculations valid

enough for heat transfer to be estimated at novel conditions. This conclusion reinforces the trend towards multi-zone models for performance and pollution calculation.

As radiative heat transfer is believed to be important and as the zone temperatures in a multi-zone model are relatively very high during combustion, the importance of a heat transfer formula that deals directly and explicitly with radiation, using the fourth power of temperature relationship, is obvious. Of the formulae considered earlier, that due to Annand, or some modification of it, is undoubtedly superior. It also provides, incidentally, for both spark ignition and diesel engines.

There is comparatively little published material on multi-zone modelling (two or more). There is even less that deals with any explicitness with the heat transfer aspects of such models. Thus Shahed et al. (32) mention that heat transfer (from the cylinder contents) is calculated based on a bulk average temperature, using an Annand type correlation, with total heat transfer apportioned to the various zones in relation to their mass and temperature. Heat transfer between zones has not been modelled. Bastress et al. (33) calculated total heat transfer to the walls using a simple formula $Q = C_4 A (T_h - T_w)$, where C_4 was a mean heat transfer coefficient, and then divided this quantity among the different zones in proportion to their volumes. Whitehouse et al., (34, 35) with a two-zone model, calculated radiation from the burning zone to the combustion chamber walls, assuming radiation from the total surface of the burning zone. Convective heat transfer was calculated from both zones where contact was made with the walls. It was assumed that heat transfer between the zones did not occur. Annand's equation was used for the heat transfer calculations.

This use of the total surface area of a number of burning jets for radiation calculations is excessive as each jet will have some of its radiation intercepted and absorbed by the other jets. This sort of problem has been mentioned earlier in this chapter and the theoretical method of determining a shape factor indicated. Unfortunately, determination of shape factors for such complex shapes is difficult. As heat transfer by radiation as at present calculated is anyway very approximate, and the surface area and shape factor change for every step of a cycle calculation, some simple approach is required.

Saadawi (36) has used a simple "effective area" approach for this purpose. Considering black body radiation, to avoid difficulties with reflection, etc., from a hot body totally enclosed in a container, it is obvious that all the radiation is absorbed by the container irrespective of its size, so the distance from body to walls may be made large, if desired, without change to the heat radiated. Consider the radiation reaching some small area A_s of the large container. In the limit the area receives the same radiation from the body as it would from an area equal to the projection of the surface of the body on the plane normal to the line joining body and area A_s and at the centre of the body.

The actual body may thus be imagined to be replaced by a number of such projected areas. It is then necessary to determine the

appropriate number and orientation of such areas that will approximate to the original body.

Should the body be a sphere the surface area of the body would be $4\pi r^2$ and the surface area of any projected area would be πr^2 . Thus any four projected areas would represent the sphere appropriately. Making the assumption that the same number would be appropriate for other shapes it is necessary to choose the most representative orientations. Saadawi chose projections from vertically above and vertically below the combustion chamber, horizontally in line with one jet and horizontally at right angles to that same jet. A three-jet model is shown in Figs. 6.14 and 6.15, before and after impingement, to illustrate the technique.

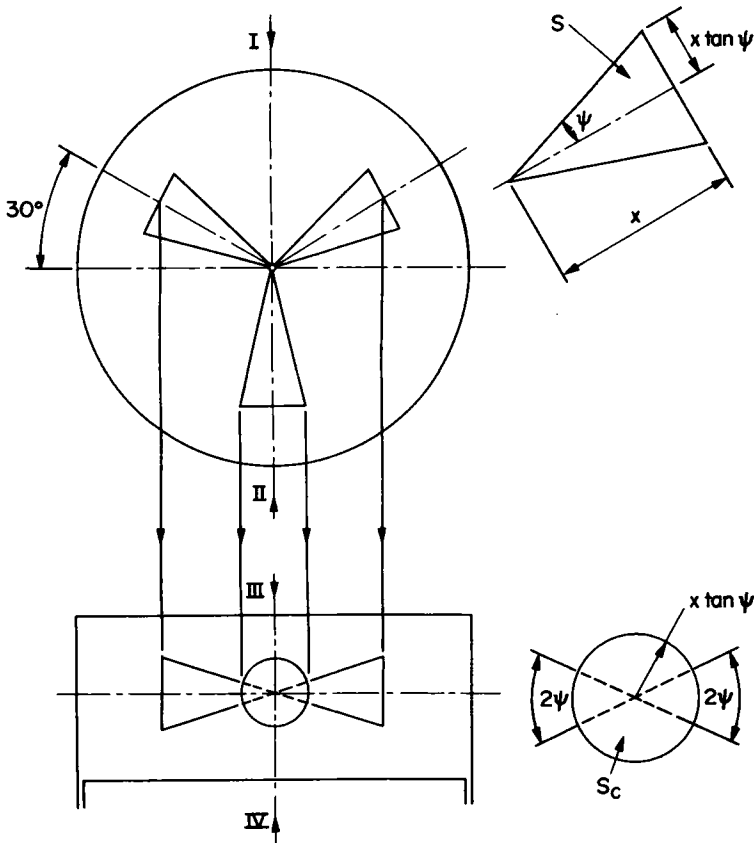


FIG. 6.14. Model for heat transfer calculations before impingement.

6.5 NUMERICAL VALUES

The value of the constants in the Annand equation for heat transfer, quoted earlier, have usually been chosen to suit empirical data, often to give overall percentage heat transfer of the correct order by comparison with estimates from steady state heat

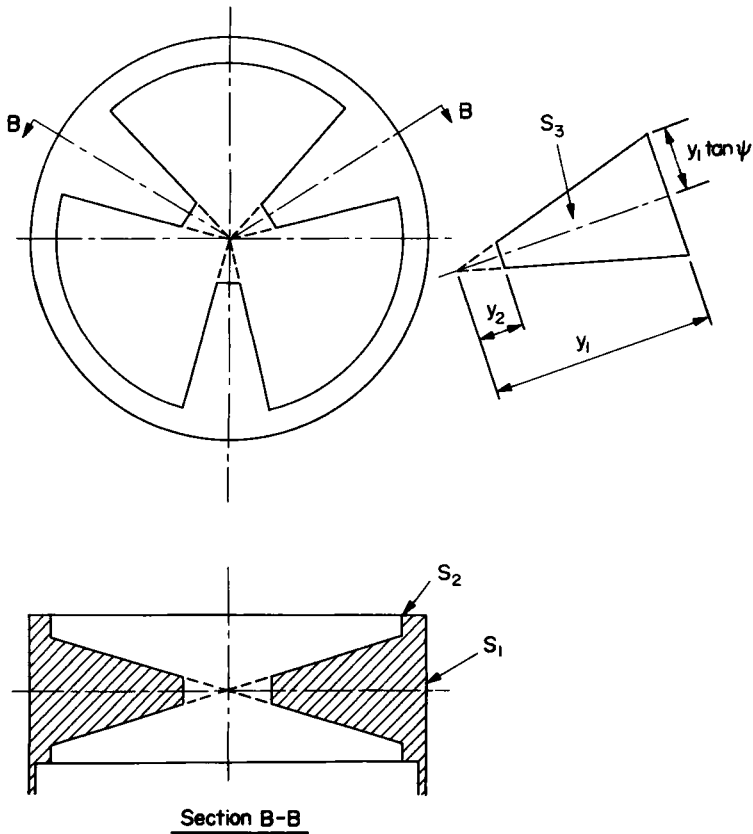


FIG. 6.15. Model for heat transfer calculations after impingement.

TABLE 6.2

Annand coefficients engine	Ref.	a	b	c
				$J/s \ m^2 \ K^4$
Two-stroke	(23)	0.76	0.64 ± 0.10	$3.03 \pm 1.06 \times 10^{-8}$
Four-stroke	(23)	0.26	$0.75 \ 0.15$	$3.88 \pm 1.39 \times 10^{-8}$
Foden FD6 (2-S)	(37)	0.3-0.5	0.7	3.28×10^{-8}
Rolls-Royce, K60	(37)	0.3	0.7	
Dorman 4LBT	(37)	0.2	0.7	
Mirrlees National FV2	(37)	0.06	0.85	
Mirrlees National FV2	(37)	0.4	0.7	

balances.(37) Annand himself gave estimates based upon an analysis of a limited amount of published data. Such data is summarized in Table 6.2.

In considering these values it must be remembered that there is a considerable interaction between a and b. Thus the two sets for the Mirrlees National FV2 engine are alternative ways of achieving almost the same effect.

The values of c used with the single-zone combustion model give very low values of heat transfer by radiation, in line with beliefs current when the formula was first proposed.

Awbi, (38) comparing Woschni's formula with Annand's for heat transfer in a four-stroke medium speed engine (the Mirrlees National FV2), found they gave similar results. Woschni had given values of $C_3=3.715 \times 10^{-5}$, $C_2=10.63 \times 10^{-3}$ ft/s °C and $C_1 = 2.28$ during compression and expansion and 6.18 during the gas exchange period. Awbi used the same ratio between C_1 and C_2 but varied C_1 to give the same mean heat transfer as did the Annand equation. This resulted in $C_1 = 4.83 \pm 0.3$. These units gave heat transfer in CHU/ft² s K with D in feet, p_{in} lbf/in and v_p in ft/s. Converting to J/m metres, bars and m/s we have, for the Awbi values

$$\frac{q}{A} = 7.54 D^{-0.2} p^{0.8} T_g^{-0.53} \left[(4.83 \pm 0.3) v_p + \left[(2.12 \pm 0.13) 10.63 \frac{VT'}{p'V'} (p-p_o) \right]^{0.8} (T_g - T_w) \right] \quad (6.25)$$

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Preface

The ever present energy crisis and the need for environmental controls has had a major impact on the development of the internal combustion engine. In this development a closer understanding of the thermodynamic processes occurring within the engine is necessary. Both authors have been continuously involved in industry and the universities over the past 30 years in the design, development, research and the teaching of internal combustion engines. The present text represents the fruits of some of their labours. Much of the material is original and some has not been published heretofore. The material has been used in the authors' department in the final year's BSc courses and in the MSc course.

The text has been written as a companion to one of the authors (R.S. Benson's) text in the same series entitled *Advanced Engineering Thermodynamics* (2nd edition). A novel feature in the text is the presentation of FORTRAN listings of two programs for simple cycle calculations—one for a compression ignition engine cycle and the other for a spark ignition engine cycle. Methods are also outlined for more complex cycle calculations of the type which are now normally carried out in design offices. The quantitative material for combustion processes in compression ignition engines and some of the data for spark ignition engines are based on the latest research carried out in the authors' laboratories.

The text is divided into two volumes to suit the convenience of students. The first volume contains material suitable for an undergraduate course in internal combustion engines, whilst the second volume is more relevant to postgraduate courses.

The book is primarily concerned with the thermodynamics of internal combustion engines but inevitably we have included hardware features. Since the successful understanding of the processes in which the engine operates is dependent on experimental work, a section is included on experimental methods which is appended to Volume I although some of the techniques are only used in advanced research establishments.

The authors wish to acknowledge with thanks the help of the numerous research students, research assistants and technical staff in producing the data used in the text. They wish to thank the various publishers and institutions for the reproductions of figures, due acknowledgement of which is given in the appropriate place. They also wish to thank Mrs. M. McDonnell and Mrs. P. Shepherd for typing the draft and Mrs. J.A. Munro for typing the camera ready copy of the text.

Finally, they wish to thank their respective wives and families for their patience and forbearance for the many evenings and weekends spent in preparing the text.

Appendix I

Experimental Methods

Notation

c specific heat
 G_n Fourier coefficient
h Planck constant
I intensity of light
k thermal conductivity
 K_n Fourier coefficient
n harmonic number
Q heat transfer rate
r coordinate
R resistance

T temperature
V electrical potential, voltage
x coordinate
y coordinate
z coordinate
 α thermal diffusivity
 λ wavelength
 ω angular frequency
 ρ density
 ν frequency

Our understanding of the thermodynamic processes in internal combustion engines depends on accurate experimental work. In this appendix we describe some methods currently in use to elucidate the phenomena occurring in engines. It is not intended to describe the range of equipment for engine testing (these are well reviewed by Ricardo and Hempson⁽¹⁾) or to discuss in detail the accuracy of electronic measurements in internal combustion engines unless they specifically refer to the influence of the accuracy on thermodynamic analysis. The proceedings of an excellent conference on the accuracy of electronic equipment has been published by the Institution of Mechanical Engineers.⁽²⁾ Nor will we describe electronic circuitry, since modern developments in this area would render description out of date. In this chapter we shall outline the principles of the instrumentation and some typical designs manufactured and tested in the authors' laboratories. We shall also refer to instrumentation developed by others and to the standard equipment now required for emissions, although for a comprehensive description of this equipment the reader is referred to the book by Patterson and Henein.⁽³⁾

The most common equipment is the pressure transducer since this enables us to study the pressure changes throughout the cycle. Special problems arise in recording the pressure accurately and analysing the subsequent record. Since these are critical to the successful analysis of the thermodynamic processes, we shall discuss the installation of transducers and the methods of recording in some detail. We shall follow with temperature measurement. This is extremely difficult, and complex equipment is normally required; we shall describe some methods successfully applied by recent workers. Studies of combustion processes involve high speed photography, gas sampling, flame speed measurement and spectrographic techniques—some of the methods in current use—will be described. Finally, we conclude with the special instrumentation used for gas analysis; this can be equally used with sampled gas from the cylinder or exhaust pipe emissions.

I.1 PRESSURE MEASUREMENT AND RECORDING

The measurement of cylinder pressures has been of interest to engineers since the birth of the steam engine for which the Watt indicator was developed. In the early days of the internal combustion engines the Watt indicator was used, but as the engine speeds increased the frequency response of this and similar types of indicator, as well as passage effects, militated against their use, and alternative methods have been developed. It is now widely recognized that the most satisfactory method of recording cylinder pressure is with an electrical transducer. In this type of transducer the deflection of a diaphragm subjected to the cylinder pressure is recorded electrically. For cylinder pressure

measurement the piezoelectric transducers are favoured since its frequency response and linearity hold over a wider range. Its major disadvantages are temperature drift causing a lack of pressure reference stability⁽⁴⁾ and low electrical output. Extreme care must be taken in the correct installation of piezoelectric transducers to avoid overheating which will cause the quartz elements to break up, producing loss of sensitivity, severe electrical leakage and drift. In all cases these transducers are water-cooled.

For accurate pressure recording the transducers should be located flush with the cylinder wall surface. Unfortunately under these conditions the impingement of fuel or a hot flame causes thermal shock which produces a distortion in the pressure record similar to that shown in Fig. I.1. The expansion line drops often

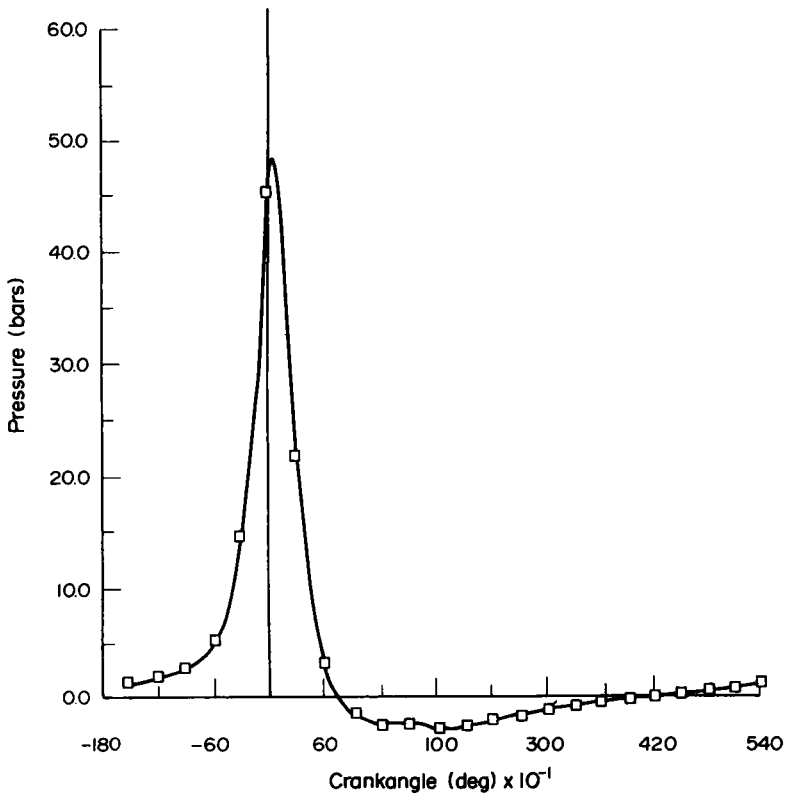


FIG. I.1. Effect of defective transducer or thermal shock.

to an apparent pressure below absolute zero, with recovery of the diaphragm during the gas exchange period. An analysis of a diagram taken with thermal shock may give an i.m.e.p. less than the b.m.e.p. To eliminate thermal shock a flame trap can be fitted in front of the diaphragm.⁽⁴⁾ To allow for the zero drift which causes instability in the pressure reference level the authors have developed a two- or three-pressure transducer system. In the

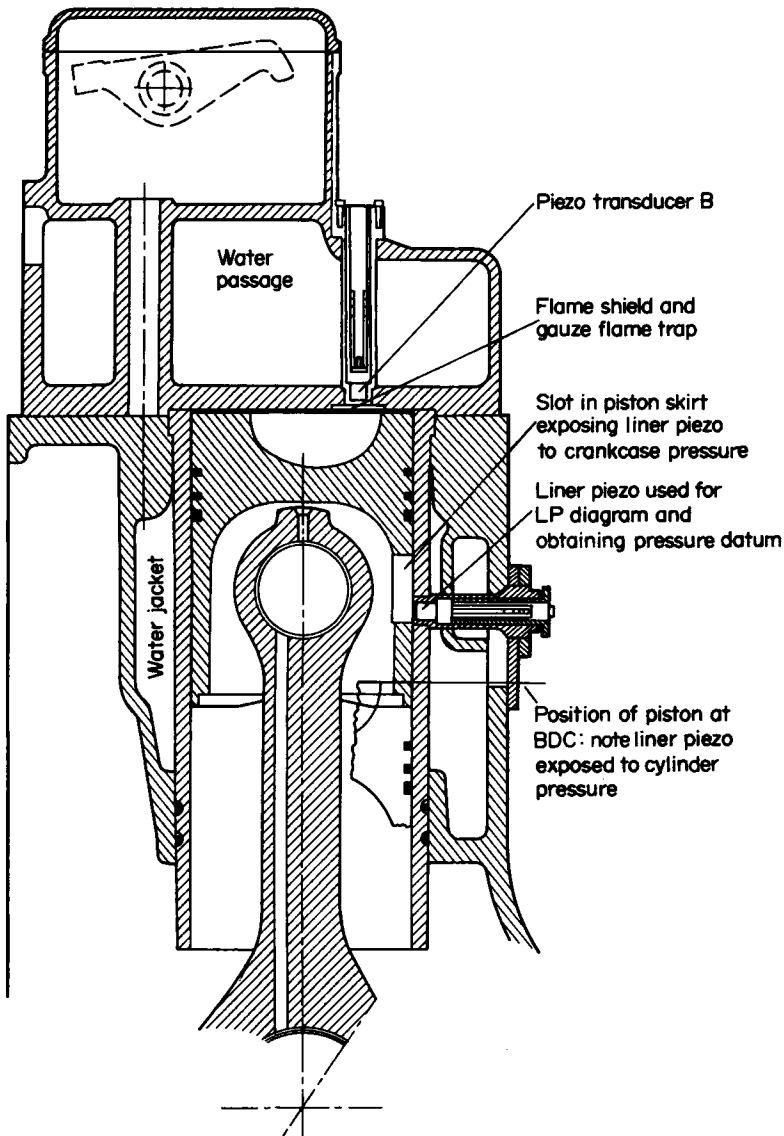


FIG. I.2. Location of pressure transducers.

example shown in Fig. I.2 one transducer (HP) is located in the combustion space and the second (LP) is located in the liner approximately mid-stroke position. The LP transducer is exposed for part of the stroke to the cylinder pressure and for part of the stroke it is in communication with the crankcase. A third transducer may be fitted in the crankcase. Since this transducer is not subject to zero drift, it can be used to set the baseline for the LP transducer when the latter is exposed to the crankcase pressure. The LP transducer is then used to calibrate the baseline

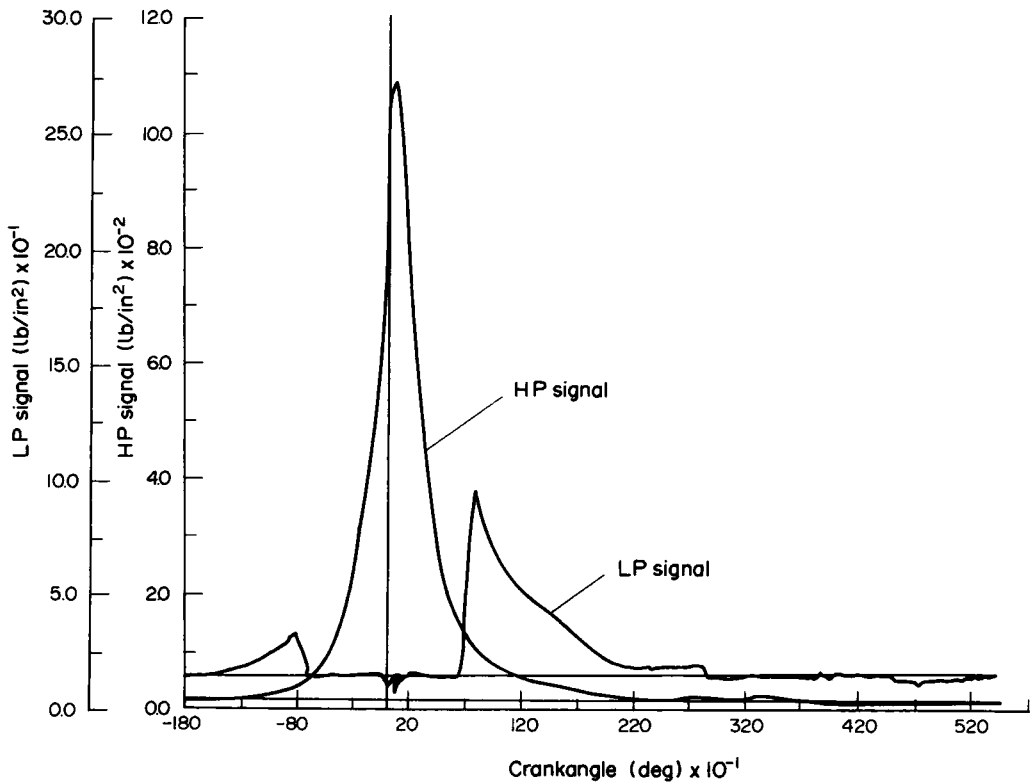


FIG. I.3. Typical cylinder pressure diagram direct output from on-line computer. (From Benson and Pick,⁽⁴⁾ by courtesy of the Society of Automotive Engineers.)

for the HP transducer. A typical record is shown in Fig. I.3. In addition to the accurate measurement of the pressure, the phasing of the diagram is also critical. In the authors' laboratories an optical encoder is used. This is essentially a perspex disc with some 720 apertures on the periphery for the passage of light. A photocell is located on one side of the disc and a lamp on the other. There is one reference aperture which is lined up to one cylinder bottom dead centre—the final adjustment being made with the engine running with the aid of a stroboscope.

The accurate recording of the pressure is most important if a successful understanding of the cylinder phenomena is to be obtained. The signal may be recorded on a cathode ray oscillograph and subsequently photographed or the signal directly processed on an on-line computer. In this system the analogue signal is converted into digital impulses at fixed crankangles in an analogue to digital

converter (ADC). The digital signal is then transmitted to the computer where it is stored. It is immediately processed as soon as all the data for one or any number of cycles have been completed as requested. The processing may be in the form of a pressure crankangle diagram or a pressure volume diagram or a heat release crankangle diagram. A full discussion of the techniques is given by Benson and Pick⁽⁴⁾ from which two figures are taken to illustrate the influence of zero drift (Fig. I.4) and incorrect timing (Fig. I.5) on the cumulative heat release diagrams for a compression ignition engine.

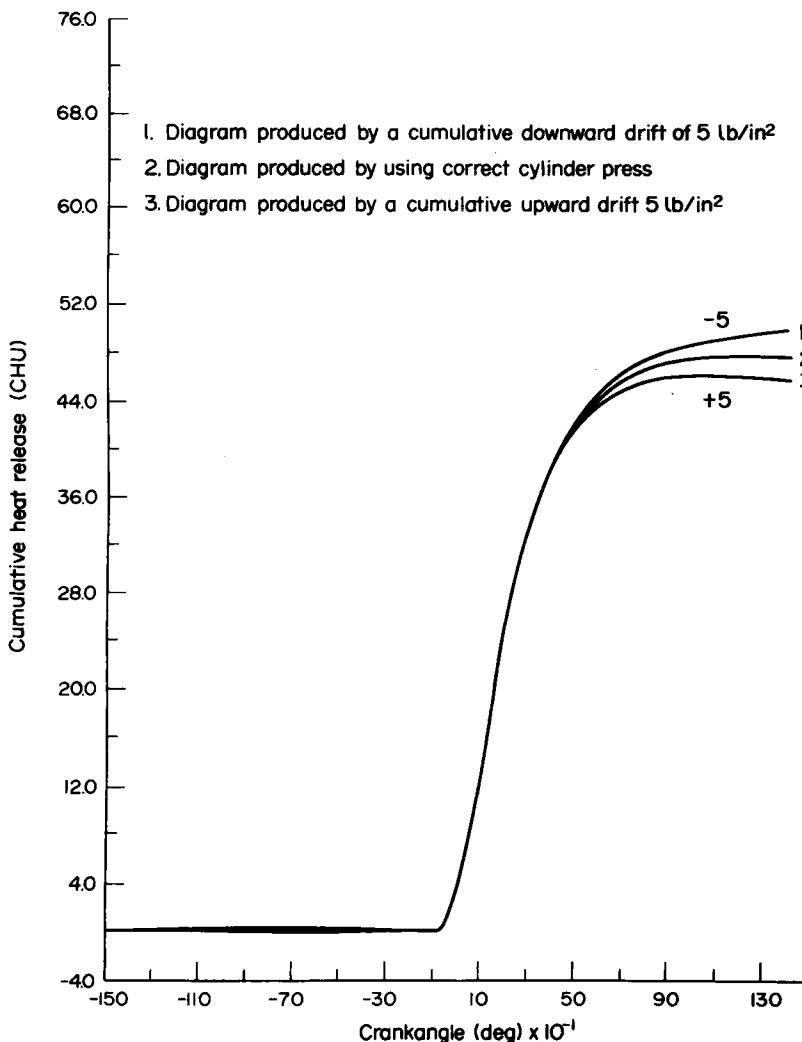


FIG. I.4. Effect of h.p. piezo drift on the cumulative heat release diagram. (From Benson and Pick⁽⁴⁾ by courtesy of the Society of Automotive Engineers.)

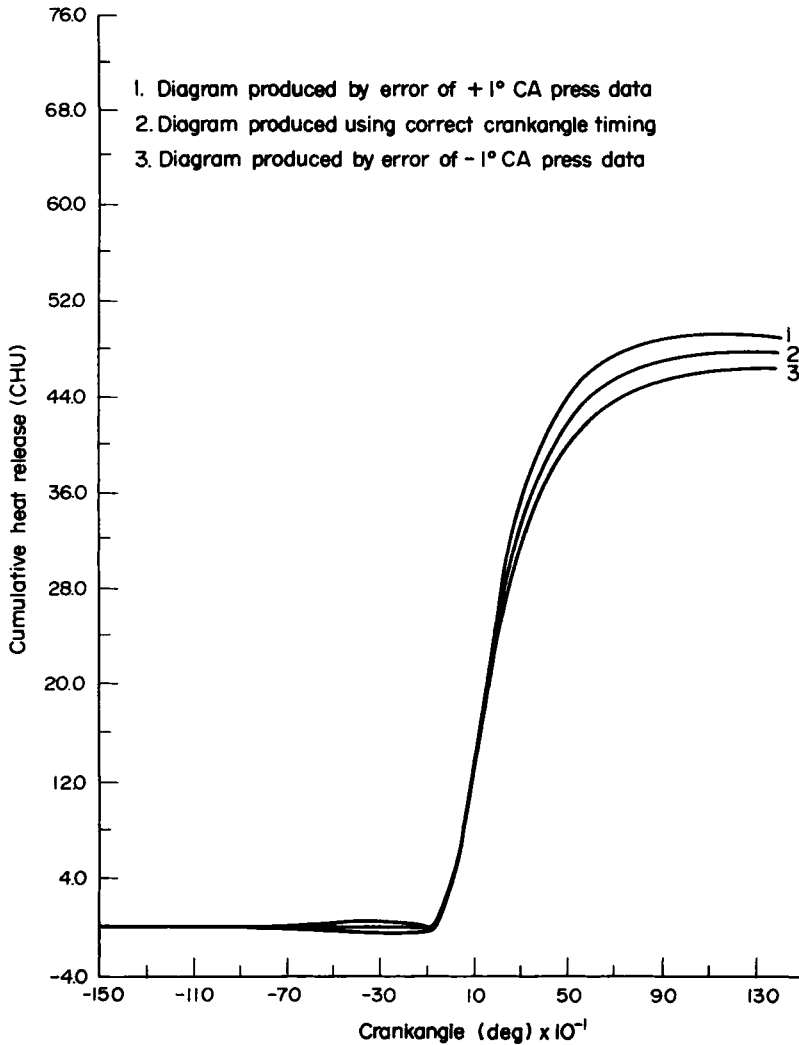


FIG. I.5. Effect of crankangle error on the cumulative heat release diagram. (From Benson and Pick,⁽⁴⁾ by courtesy of the Society of Automotive Engineers.)

For pressure measurements in the intake and exhaust systems it is usual to use inductive transducers or strain gauge transducers. Both types of transducer must be water-cooled not only to reduce temperature drift but also to preserve transducer life.

I.2 TEMPERATURE MEASUREMENT AND RECORDING

We are interested in both surface temperature measurements as well as gas temperatures. The former are required for heat transfer studies. Because of the rapidly fluctuating temperatures the measuring techniques are complex and different approaches have to be made if the cylinder temperature is required, or the exhaust gas, or the cylinder surface fluctuating temperature, or the component mean temperature. We shall first discuss the cylinder surface temperature measurement techniques, then follow with methods for measuring mean temperature at a point following by an analogue method for using these measurements to calculate heat flow. We shall then turn to a method for measuring the cylinder gas temperature and conclude with exhaust temperature measurements.

I.2.1 Component Temperature Measurement

At the cylinder surface—gas side—the temperature fluctuates with time, but a short distance from the surface the temperature is constant. Separate techniques are therefore required for the two locations. For surface measurement a thin film thermocouple may be used (Fig. I.6). A small plug is inserted in the cylinder wall or piston surface. The plug is bored at the axis and a coated nickel wire inserted. The nickel oxide coating acts as an insulator. A junction is then made between the nickel wire and the plug with a 5×10^{-5} mm nickel film deposited under a pressure of 1×10^{-5} mmHg absolute. Details of the method are given by Benson and Pick.⁽⁵⁾ The signals from the piston thermocouples are normally conveyed through a link mechanism of the type shown in Fig. I.7. Wires from the thermocouple pass via a C spring through a stainless steel tube as an oscillating link. Thence the wires pass a second C spring to an outside connector. A typical piston surface temperature record is shown in Fig. I.8.

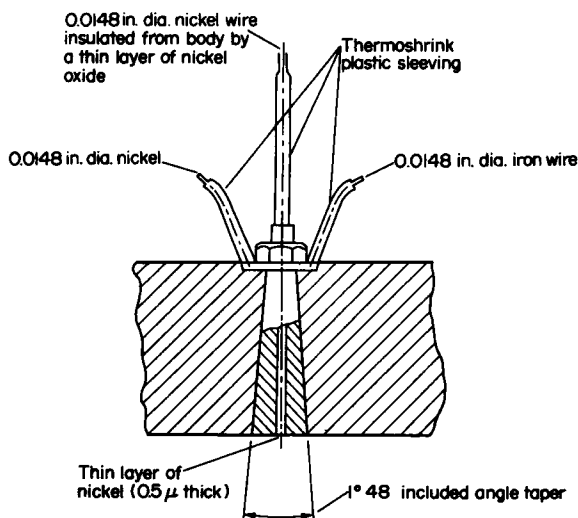


FIG. I.6. Piston and liner surface thermocouple.

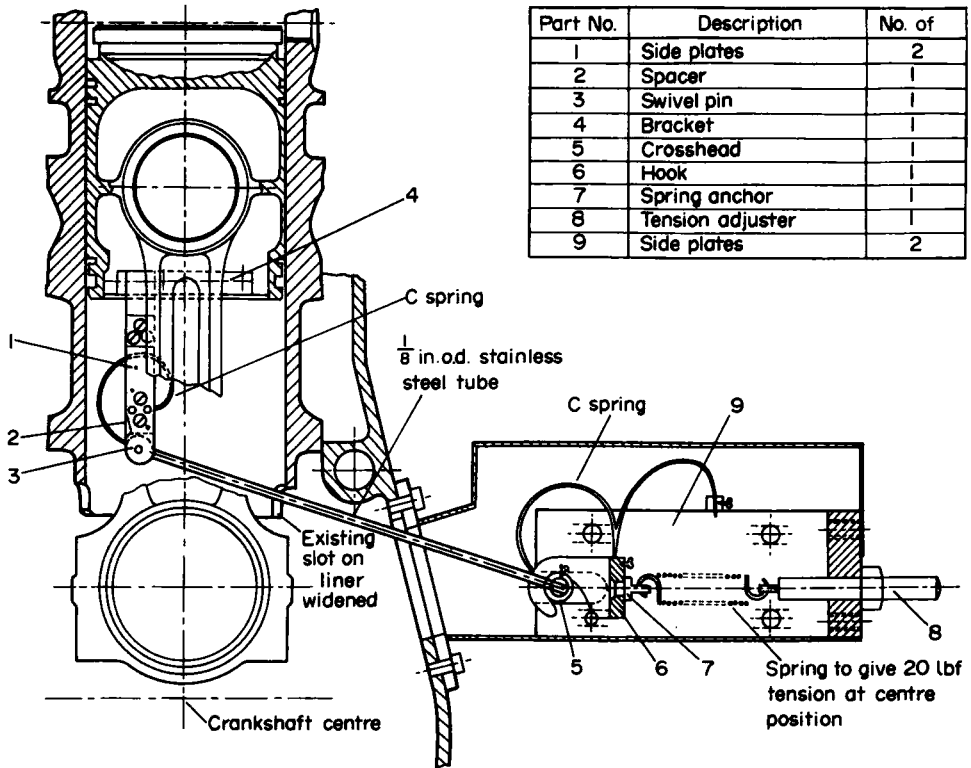


FIG. I.7. Link mechanism for recording piston surface temperatures.

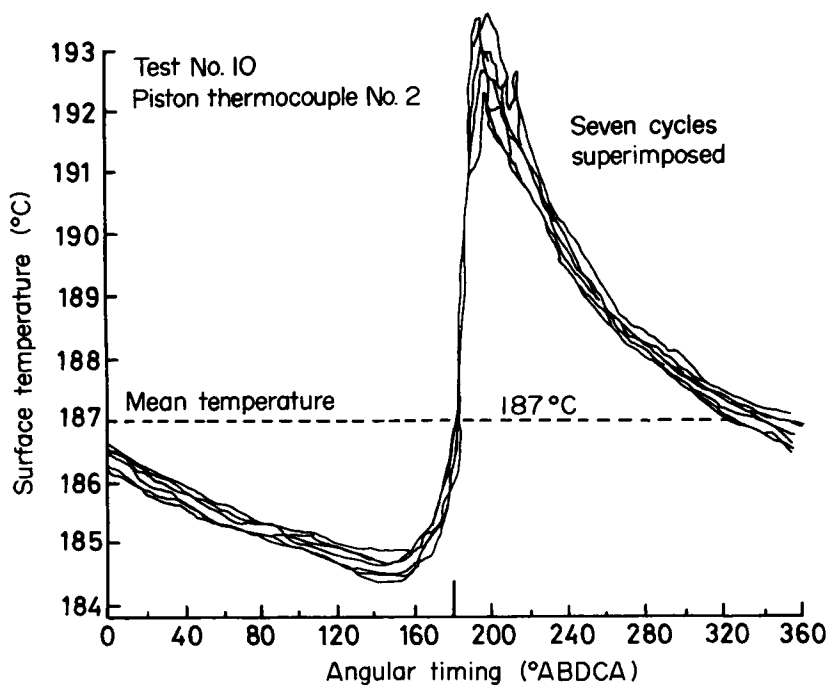
The instantaneous heat flux Q through the cylinder wall surface (liner, cylinder head or piston crown) can be computed from a diagram of the type shown in Fig. I.8(a) if we can assume that heat flux is normal to the surface.

The theoretical background is given in standard textbooks.⁽¹⁹⁾ It can be shown that the one-dimensional heat flow normal to the surface can be represented by the following two expressions:

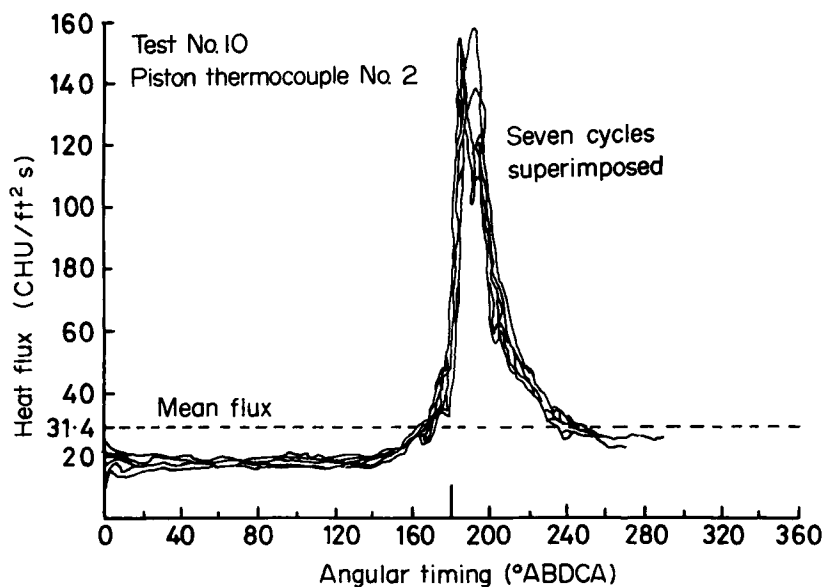
$$Q = -k \left(\frac{\partial T_x}{\partial x} \right)_{x=0} \quad \text{energy units/unit area/unit time, (a)}$$

$$\frac{\partial T_x}{\partial t} = \alpha \left(\frac{\partial^2 T_x}{\partial x^2} \right), \quad \text{(b)}$$

where T_x is the temperature of the wall material at a location x from the cylinder surface at time t , α is the thermal diffusivity of



(a) Cyclic piston surface temperature



(b) Cyclic heat flux through piston

FIG. I.8. Cyclic piston surface temperature and heat flux.
(From A.G. Law.⁽²⁹⁾)

the wall material given by $\alpha = k/\rho c$, and the density ρ , specific heat c and thermal conductivity k of the material are assumed to be constant.

The boundary conditions at the inner and outer wall can be represented by expressions of the form:

At $x = 0$, cylinder wall;

$$T_x = T_w = T_{wm} + \sum_{n=1}^{\infty} \left(K_n \cos n\omega t + G_n \sin n\omega t \right).$$

At $x = a$, coolant surface:

$$T_x = T_c = T_{cm} = \text{constant},$$

where T_{wm} is the time mean wall temperature in the cylinder at the point under consideration and T_{cm} is the time mean wall temperature at the coolant surface, K_n and G_n are Fourier coefficients, ω is the angular frequency equal to 2π times the cycle repetition rate, and n is the harmonic number.

Integration of the temperature differential equation (b) with the above boundary equations and substitution in the heat flux equation (b) gives the following expression for the heat flux:

$$Q = \frac{k}{a} (T_{wm} - T_{cm}) + k \sum_{n=1}^{\infty} r \left((G_n - K_n) \sin \omega t + (G_n + K_n) \cos \omega t \right),$$

where $r = n\omega/2\alpha$.

Law⁽²⁰⁾ has indicated that if a Fourier analysis is made of a temperature diagram of the type shown in Fig. 1.8(a) with up to $n = 90$ harmonics, taking sample points at two-degree intervals, an accurate representation of the wall temperature may be made. Typical results are shown for the heat flux in Fig. 1.8(b) taken from Law's work.⁽²⁰⁾ The heat flux and temperature curves correspond to the same test.

Two methods may be used to measure the steady temperature in the component. For a stationary component, Ricardos⁽¹⁾ have developed the traversing thermocouple. In Fig. 1.9 an adaptation of Ricardo's thermocouple is shown to measure the temperature gradient in the bottom deck of a cylinder head.⁽²¹⁾ It is important in this method that the plug on which the thermocouple is mounted is in contact with the surface. An alternative method used in the authors' laboratory was developed by the Shell Thornton Research Centre. This method makes use of the hardness temperature relationship for special alloy steels. Certain alloys undergo changes of hardness when subjected to elevated temperatures over a period of time. The hardness remains when the alloy returns to normal temperature so that by measuring the hardness the elevated temperature can be inferred. These alloys are used in the form of small screwed plugs called templugs. The plugs are screwed in the component whose temperature is required; the temperature of the component is measured at the point of contact of

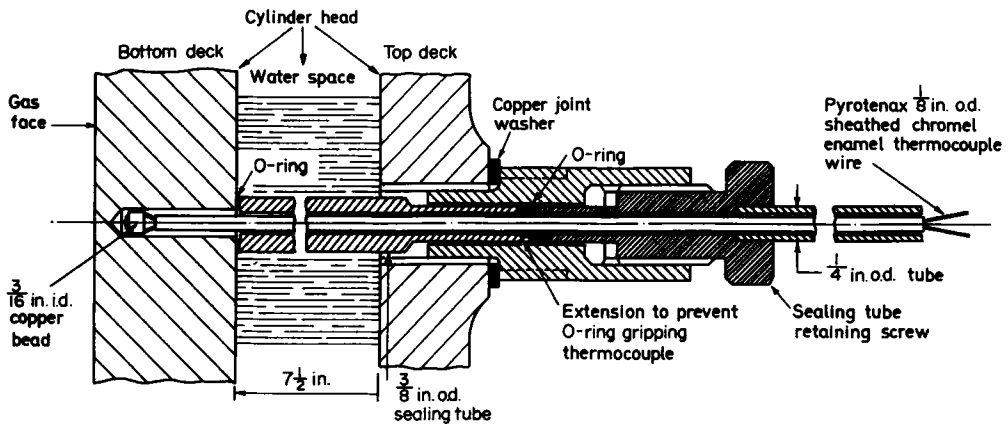


FIG. I.9. Modified Ricardo traversing thermocouple arrangement. (From Brock and Glasspoole,⁽²¹⁾ by courtesy of the Institution of Mechanical Engineers.)

the plug and the component. In Fig. I.10 templug positions are shown in a piston in the thrust section. The engine is run at a steady load and speed for a fixed period of time after which the plugs are removed and their hardness measured. If sufficient plugs are fitted the heat flow through the component can be computed either numerically or by using an electrical analogue. In both methods the isothermals in the component are constructed to agree with the measured temperature. We shall describe the analogue method.

The analogue method is based on the similarity of the heat conduction equation and the electrical potential equations. For steady state the conduction equation, using r , z , θ coordinates, is, for axisymmetry,

$$\frac{\partial^2 T}{\partial z^2} + \frac{1}{r} \frac{\partial^2 T}{\partial r} + \frac{\partial^2 T}{\partial r^2} = 0,$$

and, for electrical potential (V),