WORKING DETAILS OF A GAS ENGINE TEST

INCLUDING

A METHOD OF DETERMINING THE TEMPERATURES OF EXHAUST GASES

BY

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INTRODUCTION.

The work embodied in the following pages is but a small beginning toward the development of an investigation of some importance to the engineering and commercial world—the standardization of methods of comparison of gas engines.

With the heterogeneous collection of gas and oil engines thrown upon the market to-day, some method of standardization is very essential, but the problem is one of too great magnitude to be completed in the brief period of one college year, and it is the hope of the writer that some one may take up the work where it has been dropped.

The general scheme, as roughly outlined, is

1. The history and development of the explosive engine.

2. The gas engine of to-day, and a comparison of existing engines, resulting in a classification into types and a study of these, including:

(*a*) A complete bibliography of the subject, together with a list of patents granted, both in this country and abroad.

3. An investigation of the physical and mathematical problems involved, both quantitative and qualitative.

4. A complete and detailed method of testing explosive engines, leading to

(*a*) A criticism of points of design and construction and suggested improvements, based upon information obtained through 2 and 3.

5. General conclusions derived from the tests and observation, coupled with a comparison of the explosive engine with certain types of non-explosive engines.

The historical section has been fairly well developed by previous investigators. It is seen at once that sections 2, 3 and 4 are to a certain extent interdependent and that a thorough investigation of one involves certain information obtained from the others. A brief inspection of the outline of procedure indicates that the main portion of section 4 is at the present time of greatest moment, and the complete and detailed method of testing explosive engines has, therefore, received the entire attention of the writer for several months past.

To avoid the possibilities of duplicating work already well under way, or completed, a special trip was made to three technical institutions known to be especially interested in this subject, namely: the Massachusetts Institute of Technology, Boston, Mass.; Worcester Polytechnic Institute, Worcester, Mass.; and Cornell University, Ithaca, N. Y.

Although more or less work upon the heat engine has been carried on, or is now in progress at the laboratories of these institutions, yet the lines followed are quite different from those undertaken by the writer, and the field for thorough research was found practically clear, as the greater part of the work done is of the nature of ordinary commercial tests for the benefit of students taking the regular laboratory courses.

Correspondence and inquiry led to the belief that nowhere had research been undertaken on the plan outlined above, and preparations were at once made for carrying on the investigations. The engines in the laboratory at Columbia University in New York City were put into commission and used for experimental purposes, the greater part of the data being obtained from an eight horse-power Otto engine and a five horse-power Nash engine.

The results of the investigation have been prepared for presentation before the American Society of Mechanical Engineers at their meeting in Boston, in May 1902, which accounts for the form in which they are presented.

The volumes of Transactions and numbers of papers to which references are made in the text and footnotes are those issued by the American Society of Mechanical Engineers.

The writer desires to express his appreciation of courtesies extended and valuable assistance rendered, to Prof. F. R. Hutton, Prof. R. S. Woodward, Prof. W. L. Cathcart, Prof. W. Hallock, Mr. F. A. Goetze, and Mr. C. E. Lucke.

New York City, March 15, 1902.

PART I.

Working Details of a Gas Engine Test.

WORKING DETAILS

OF A

GAS ENGINE TEST.*

1. Considering the rapid advance of the gas † engine during the past few years, it is surprising to find how little has been done toward standardizing such engines, or at least toward adopting some form of test whereby a definite idea of the relative merits of different engines can be obtained. Of late years the market has been crowded with various types and modifications of heat engines, especially of the explosive type, many of which are mere freaks and of little or no value.

2. Although a very few types have received the careful attention of the expert, many engines thrown upon the market are the results of a desire to invent, with the possible chance of "hitting the right thing," or the product of dull times in machine shops.

Although in the case of the steam engine occasional forms are produced having entirely new features, yet the tendency is to conform closely to a general standard which usage and careful investigation have shown to be desirable.

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* For further references on this subject see *Transactions* as follows:

No. 0943, to be presented at this (Boston, 1902) meeting, "Final Report of Committee Appointed to Standardize a System of Testing Steam Engines."

Vol. xxii., p. 152: "Efficiency of a Gas Engine as Modified by Point of Ignition." C. V. Kerr.

Vol. xxii., p. 612: "Efficiency Tests of a One-hundred and Twenty-five Horsepower Gas Engine." C. H. Robertson.

† The term "gas engine" is used throughout this paper to include engines commonly termed oil engines.

3. Similar standards should be adopted for the gas engine as rapidly as possible. This does not prohibit or discourage invention or modification, which are very essential in the present condition of these engines, but something should be done to classify and standardize those already on the market, thus determining what experience and study have thus far shown to be good form.

The problem in its entirety is a large one, and will require many months or perhaps years of continued investigation and study. With this fact in mind, the writer has undertaken a few months' study of the subject, and although unable more than to begin the work, the results so far obtained may prove of interest to this Society.

4. Before the engines can be standardized some definite method of determining the relative merits of different engines must be settled upon, and this leads at once to the necessity of a standard method of testing gas engines, which will form the special part of the problem herein discussed.

It is of interest, before taking up in detail the method of conducting the tests, to note some of the items that require careful investigation and study for a proper classification of the engines. Much of the data desired is simply for general information, and to obtain the views of the manufacturer on certain points and to determine what he considers good practice.

Other information is needed because it enters into any test of the engine that may be made, while other questions lead toward a classification of such engines into general divisions. Although time has not permitted the testing of such data sheets practically, yet the following general form serves as a basis for beginning the investigation and experience alone can determine the necessary modifications and additions:

GAS ENGINE DATA.

No.......... Test No..........

Date..............................

1. Name of Engine..............................
2. Manufactured by ....
3. Is it two or four cycle?..............................
4. Kind of fuel used..............................
5. Assumed heat of combustion of fuel =.........B. T. U. per ....
6. Actual horse-power ....
7. Floor space occupied ....
8. Height, wheels to clear floor..............................
9. Weight ....
10. Number of cylinders ....
11. How are cylinders arranged if more than one?..............................
12. Diameter and weight of flywheel. Diam.=......ins. Wgt.=.......lbs.

13. Diameter and width of brake pulley. Diam.=.....ins. Width=.....ins.
14. Diameter of piston=.....ins. Length of barrel=.....ins. No. rings.....
15. Piston displacement=.............cu. ft. Clearance=...... ......cu. ft.
16. Length of stroke=..........ins. Length of connecting-rod=.......... ins.
17. Revolutions per minute=..
18. Kind of governor...
19. Method of governing..
20. Special governor features ....
21. Kind of ignition : (*A*), hot tube ; (*B*), flame ; (*C*), electric.
    - If (*A*) { (*a*) location.................(*b*) dimensions................ ; (*c*) how heated..................................
    - If (*B*) { (*a*) kind.....................(*b*) size of flame port............ ; (*c*) special features..................................
    - If (*C*) {
      - (*a*) contact spark { (1) current required.........(2) voltage... ; (3) battery recommended ...(4) number... ; (5) specifications of coil..................
      - (*b*) jump spark { (1) current required........ (2) voltage... ; (3) kind of plug............(4) location .. ; (5) battery recommended....(6) number .. ; (7) specifications of coil.................
22. Kind of valves ....
23. Special features of valve mechanism ....
    ....
24. Diameter of (*a*) gas valve......ins.; (*b*) air valve......ins.; (*c*) mixture valve ......ins.; (*d*) exhaust valve......ins.
25. Lift of (*a*) gas valve........ins.; (*b*) air valve ....ins.; (*c*) mixture valve ......ins.; (*d*) exhaust valve......ins.
26. Kind of { (*a*) carburettor.................................. ; (*b*) vaporizer .... ; (*c*) mixer ....
27. Means of maintaining constant proportions............................
    ....
28. Method of fuel feed...
29. Number of water inlets ....diameters..................ins.
30. Number of water discharges..............diameters..................ins.
31. Best rate of water feed ....
32. Kind of muffler...
33. Dimensions of muffler...
34. Does muffler use water?...
35. Means for preventing noise at air inlet...............................
    ....
36. Kind of gas bag ....
37. Dimensions of gas bag ....
38. Means of clearing engine of exhaust gases.............................
    ....
39. Devices for starting or aiding starting...............................

5. The investigations which have developed the details of conducting tests have been carried on at the mechanical laboratory of Columbia University during the present college year. The various experiments have been developed largely from work on a 6 x 12½ Otto engine, and a 6 x 9 Nash engine. Much time has been devoted to working out many details and to following incidental suggestions that offered themselves as the work progressed, and it is proposed to present rather fully many important points.

This may seem unnecessary to those already familiar with such work, but it is believed that this part of the paper may prove of interest and value to those who propose undertaking such tests for the first time. For this reason some of the difficulties and errors that are likely to occur are especially emphasized.

6. It is hardly necessary to offer any explanation of the items called for in the "log" of the test, as information may be obtained regarding each of these points from the details of the corresponding items that appear in the final report. The form of "log" appended is found to be convenient for making the preliminary records:

Date..................................

Log of..................Gas engine test............ Test No............

By..

Object ....

Length of brake lever..................................ft.

Weight of brake lever ....lbs.

Cubic feet of vapor per pound of..................................

Weight of gallon of ....lbs.

1. Number of run ....
2. Time ....
3. Speed and explosions, revolutions per minute, hand indicator ....
4. " " reading of speed counter ....
5. " " explosions per minute, special count..................
6. " " reading of explosion counter..................................
7. Total load on scales..................................lbs.
8. Temperatures, degrees Fahr. or Cent., gas..................................
9. " " " " air ....
10. " " " " jacket water, entering ....
11. " " " " " " leaving ....
12. " " " " " " barrel ....
13. " " " " exhaust, observed ....
14. " " " " " at pressure of atmosphere...
15. Valve index reading, gas ....
16. " " " air ....
17. Areas of valve openings, gas or vapor..............................sq. ins.
18. " " " oil.................................. "
19. " " " air .... "
20. " " " mixture.................................. "
21. " " " exhaust .... "
22. Weights and volumes, weight of jacket water ....lbs.
23. " " reading of gas meter, cu. ft.......or gals. of........
24. " " reading of air meter..................................
25. " " gas for igniter ....cu. ft.
26. Indicator springs used, power card..................................
27. " " " compression card ....
28. Pressures, gas, inches of ....
29. " air, inches of..................................
30. " jacket water, lbs. per sq. in. ....
31. " exhaust, lbs. per sq. in..................................

Remarks..

7. Before entering upon a discussion of a complete test of a gas engine it is necessary to establish certain standard units. Without question the proper unit for the energy derived from the fuel used is the "British Thermal Unit," which has been adopted throughout this country and England, and is designated in this

paper by the usual symbols, B. T. U. In like manner the term horse-power is used to designate the rate of work, and I. H. P. and B. H. P. have their usual significance, meaning indicated horse-power and brake horse-power respectively.

Further explanation of the units used will develop during the examination of the final report. It is hardly necessary to touch upon the proper methods of calibration of instruments to be used, with the exception of special instruments or instruments used under special conditions, as this subject has been so fully treated in many previous publications. These cases will be treated under their respective heads.

PRELIMINARIES OF THE TEST.

8. Before beginning the test it is very essential to see that the engine is in good running order, thoroughly oiled, and properly adjusted. All valves should be carefully examined and adjusted. All connections to the engine, whether fuel, air, or water, should be tested for leakage.

Special note should be made of any points out of the ordinary, and if they are of such a nature as to affect the results of the test, the difficulties should be set right as far as possible, and, when this cannot be done, careful estimates should be made of the changes in the results due to such conditions.

It is of the greatest importance that each observer know exactly what is expected of him, and that he at least be made thoroughly familiar with the details of all apparatus and machinery bearing upon his portion of the work and he should be shown exactly what to do in case of emergency, in order to rectify difficulties as quickly as possible without interrupting or possibly destroying an entire test.

9. In the series of tests from which the results for this paper were obtained, more than one test was entirely lost after hours of work through the hasty action or lack of judgment of some one observer. Where possible, a picked and trained crew should be retained, and even then as far as practicable the director of the test should attempt nothing but the oversight of the observers, and should stand ready for all emergencies, allowing the individual observer to touch nothing but the instrument he is observing, and then only as directed. Unfortunate experience has shown this to be the only possible way of obtaining reliability in results.

It is necessary that the engine be run a sufficient time before the real start of the test, to enable all parts to become properly adjusted to the desired running conditions, or to get the engine "warmed up," and to determine accurately the proper working of all instruments and recording devices, and especially if the engine is to carry the maximum load which it can carry continually. Much difficulty is often experienced in determining this maximum load, especially if the fuel used is city gas, for then the load carried yesterday may be far from the possible load of to-day. If a brake is used without cooling water, then the run previous to the start of the test must be of sufficient length thoroughly to warm the brake pulley, to permit further expansion and a sudden increase in the load. A brief preliminary trial should in every case precede the regular run, to make certain that every observer understands his work, and it is found advantageous not to inform the observer of any distinction between the preliminary run and the true test.

The reading of an ordinary meter used for measuring air supply is not a difficult matter, but it has been found that few men can obtain reliable readings without previous experience.

OBJECT OF THE TEST.

10. It is absolutely necessary that the object of the test be definitely determined, and this object should be kept constantly in mind during the run, whether the test be a general efficiency test, a test for proportions of air to gas, changes in conditions of ignition, temperature of jacket water, throttling of exhaust, or what not.

FORM OF REPORT BLANK.

11. Careful study coupled with experience has developed the accompanying form for the final report of the test. At first sight it may appear to some as too comprehensive, consequently cumbersome, but the desire has been to secure a form that will serve for all purposes, whether for an efficiency test only or for a complete laboratory test in which much data is desired for further study that might of itself be of little value to the manufacturer directly, but of great value scientifically, especially along lines which will aid in further development of the heat-engine problem.

It is deemed wise to take up each item of the report in detail, and, to assist in making the explanations clear, run number 6 has been selected from each of the two tests chosen for this paper.

The engine from which the results recorded as test "A" were obtained was run under full load, while in making test "B" the engine carried only half load.

12. It is not intended to draw any comparisons between the two runs, and they are both submitted simply to assist in further explanations of the method of working out a complete test if conducted on the plan outlined.

Date, *January* 15, 1902.

Report of................"A" ....Gas Engine Test.

Test No...

By...

Object................ ... ...*Efficiency, Proportions of Air to Gas* ....

1.		Number.............	1	2	3	4	5	6	7	8	9	10	Totals.	Average.
2.		Time intervals, minutes..................	10	10	10	10	10	10	10	10	10	10	100	10
3.	Revolut's and Explosions.	Total revolutions	2,979	2,959	2.884	2,775	2,812	2,679	2,728	2,758	2,675	2,856	28,105	2,810.5
4.		Revolutions per minute (mean)...	297.9	295.9	288.4	277.5	281.2	267.9	272.8	275.8	267.5	285.6	2,810.5	281.05
5.		Total explosions.................	1,369	1,329	1,393	1,293	1,400	1,339	1,344	1,379	1,335	1,393	13,574	1,357.4
6.		Explosions per minute (mean)....	136.9	132.9	139.3	129.3	140.0	133.9	134.4	137.9	133.5	139.3	1,357.4	135.74
7.		Ratio of revolutions to explos's...	2.18	2.22	2.07	2.14	2.01	2.00	2.03	2.00	2.00	2.05	20.70	2.070
8.	Jacket Water.	Weight in lbs....................	97	97	97	97	97	97	97	97	97	97	970	97
9.		Weight in lbs. per hour.........	582	582	582	582	582	582	582	582	582	582	5,820	582
10.		Temperature range..............	72	72	76	78	74.5	72	72	74	74	77	741.5	74.15
11.		Max. velocity, feet per second....	....	....	....	....	....	....	....	....	....	....	....	....
12.		Heat absorbed, B. T. Units.......	6,980	6,980	7,370	7,760	7,220	6,980	6,980	7,180	7.180	7,460	72,090	7,290
13.	Air.	Cubic feet.......................	198.0	196.2	193.2	190.6	193.0	194.0	195.5	197.0	195.5	196.0	1,949.0	194.9
14.		Cubic feet per hour	1,188	1,177	1,159	1,144	1,158	1,164	1,173	1,192	1,173	1,176	11,704	1,170.4
15.		Temperature, Fahr. deg..........	86.0	86.0	86.0	86.0	86.0	86.0	86.0	86.0	86.0	86.0	....	....
16.		Weight per cubic foot...........	.0727	.0727	.0727	.0727	.0727	.0727	.0727	.0727	.0727	.0727	....	....
17.		Max. velocity, feet per second....	....	....	....	....	....	....	....	....	....	....	....	....
18.		Specific heat, c_v	.1691	.1691	.1691	.1691	.1691	.1691	.1691	.1691	.1691	.1691	....	....
19.	Fuel Gas.	Cubic feet or ~~gals~~...............	29.00	28.25	28.50	27.50	27.50	26.25	24.50	25.75	26.00	27.00	270.25	27.025
20.		Cubic feet or ~~gals~~ per hour......	174.0	169.5	171.0	165.0	165.0	157.5	147.0	154.5	156.0	162.0	1,621.5	162.15
21.		Temperature, Fahr. deg..........	86	86	86	86	86	86	86	86	86	86	....	....
22.		Weight per cubic foot...........	....	....	....	....	....	....	....	....	....	....	....	....
		" " gallon	....	....	....	....	....	....	....	....	....	....	....	....
23.		Max. velocity, feet per second....	....	....	....	....	....	....	....	....	....	....	....	....
24.		Specific heat, c_v	.1691	.1691	.1691	.1691	.1691	.1691	.1691	.1691	.1691	.1691	....	....
25.		Cubic feet Standard Gas per hour (60° Fahr., 14.7 lbs. press.)...	166.5	162.5	163.5	158.0	158.0	151.0	141.0	148.0	149.5	155.0	1,553.0	155.3
26.	Valve Index Reading.	Gas.............................	5	5	5	5	5	5	5	5	5	5	....	....
27.		Air	....	....	....	....	....	....	....	....	....	....	....	....

28.	Values of n in Eq. $PV^n = P_1V_1^n$ for expansion curve.	....	1.14	1.14	1.14	1.14	1.14	1.14	1.14	1.14	1.14	1.14	11.40	1.14
29.	Mixture.	Temperature, Fahr. deg.	108	108	108	107	108	106	106	106	106	107	1,070	107
30.		Weight, per cubic foot	.0698	.0698	.0698	.0700	.0698	.0701	.0701	.0701	.0701	.0700	.6996	.06996
31.		Max. velocity, feet per second	....	....	....	....	....	....	....	....	....	....	....	....
32.	Pressures from Indicator Cards.	Pressure at end of Comp.	91	91	91	91	91	91	91	91	91	91	910	91
33.		Max. pressure or pressure at beginning of expansion	312	310	307	307	307	298	298	302	302	302	3,045	304.5
34.		Pressure at end of expansion	55	55	55	55	53	53	50.5	53	55	55	539.5	53.95
35.		Pressure if expansion be carried to end of stroke	51.0	51.0	49.5	49.5	49.5	48.0	47.0	48.5	49.5	49.5	493.0	49.3
36.		Mean effective pressure	68.0	70.6	67.9	66.0	62.0	66.3	68.5	71.5	67.7	67.7	676.2	67.62
37.	Exhaust.	Temperature Fahr. deg.	1,510	1,510	1,450	1,445	1,450	1,387	1,350	1,410	1,445	1,450	14,407	1,440.7
38.		Max. velocity, feet per second	....	....	....	....	....	....	....	....	....	....	....	....
39.		Specific heat, c_v	.1691	.1691	.1691	.1691	.1691	.1691	.1691	.1691	.1691	.1691	....	....
40.	Ratios.	Air to gas to	6.2	6.1	6.5	6.4	7.0	7.4	7.9	7.6	7.5	7.1	69.7	6.97
		Neutrals	1	1	1	1	1	1	1	1	1	1	10	1
			2.2	2.2	2.3	2.2	2.4	2.4	2.6	2.5	2.4	2.4	23.6	2.36
41.		Stroke to Expansion	1.15	1.15	1.15	1.15	1.15	1.14	1.14	1.14	1.14	1.16	11.47	1.147
42.		Vols. v_2 to v_1	4.68	4.68	4.68	4.68	4.68	4.68	4.68	4.68	4.68	4.68	46.80	4.68
43.		Max. pressure to M. E. P.	4.59	4.39	4.52	4.65	4.95	4.50	4.35	4.22	4.46	4.46	45.09	4.509
44.		Max. pressure to comp. pressure.	3.43	3.41	3.38	3.38	3.38	3.28	3.28	3.32	3.32	3.32	33.50	3.35
45.	Value of R		.970	.970	.970	.972	.970	.973	.973	.973	.973	.972	9.716	.9716
46.	Temperature, Fahr. deg. at comp.		285	285	285	283	285	282	282	282	282	283	2,834	283.4
47.	Max. Temperature, Fahr. deg.		2,090	2,075	2,050	2,045	2,050	1,970	1,970	2,000	2,000	2,000	20,250	2,025
48.	Energies.	Brake work, foot lbs.	2,460,000	2,440,000	2,380,000	2,290,000	2,320,000	2,210,000	2,250,000	2,280,000	2,210,000	2,360,000	23,200,000	2,320,000
49.		Brake work, foot lbs. per hour	14,750,000	14,650,000	14,300,000	13,750,000	13,900,000	13,250,000	13,500,000	13,700,000	13,250,000	14,150,000	139,200,000	13,920,000
50.		Brake horse-power	7.45	7.39	7.21	6.95	7.03	6.70	6.82	6.90	6.70	7.15	70.30	7.03
51.		B. T. U. equiv. to B. H. P.	3,160	3,130	3,060	2,950	2,980	2,840	2,890	2,930	2,840	3,030	29,810	2,981
52.		Indicated horse-power	8.25	8.35	8.45	7.60	7.75	7.93	8.22	8.80	8.06	8.44	81.85	8.185
53.		B. T. U. equiv. to I. H. P.	3,500	3,540	3,580	3,220	3,280	3,360	3,480	3,730	3,420	3,580	34,690	3,469
54.		Gas horse-power	42.5	41.2	41.7	40.3	40.3	38.6	35.8	37.8	38.0	39.4	395.6	39.56
55.		B. T. U. equiv. to Gas H. P. = H_1	18,000	17,550	17,700	17,100	17,100	16,400	15,200	16,000	16,100	16,700	167,850	16,785
56.		Heat supplied B. T. U. from indicator card = H'_1	6,230	5,960	6,250	5,860	6,350	6,040	5,990	6,250	6,160	6,180	61,270	6,127
57.		Heat extracted B. T. U. by observation = H_2	....	....	....	....	....	....	....	....	....	....	....	....
58.		Heat extracted B. T. U. from indicator card = H'_2	4,900	4,720	4,750	4,450	4,830	4,580	4,410	4,740	4,800	4,840	47,020	4,702
59.		I. H. P. — B. H. P.	0.80	0.96	1.24	0.65	0.72	1.23	1.40	1.90	1.36	1.29	11.55	1.155
60.	Throttling, cubic feet		.047	.048	.045	.042	.042	.034	.038	.037	.033	.042	.408	.0408
61.	Throttling, per cent.		18.1	18.5	17.3	16.2	16.2	13.1	14.6	14.2	12.7	16.2	157.1	15.71
62.	Work gained by complete expansion		....	....	....	....	....	....	....	....	....	....	....	....

		Number.	1	2	3	4	5	6	7	8	9	10	Totals.	Average.
63.	Efficiencies.	Mech. eff. $= \frac{\text{Item 50}}{\text{Item 52}}$	.904	.885	.854	.915	.907	.846	.830	.785	.831	.848	8.605	.8605
64.		Thermal for B. H. P. $= \frac{\text{B. H. P.}}{H_1} = \frac{\text{Item 51}}{\text{Item 55}}$	.176	.178	.173	.173	.174	.173	.190	.183	.176	.181	1.777	.1777
65.		Thermal for B. H. P. $= \frac{\text{B. H. P.}}{H'_1} = \frac{\text{Item 51}}{\text{Item 56}}$	.507	.525	.490	.503	.469	.471	.483	.469	.461	.490	4.868	.4868
66.		Thermal for I. H. P. $= \frac{\text{I. H. P.}}{H_1} = \frac{\text{Item 53}}{\text{Item 55}}$	.194	.202	.202	.182	.192	.205	.228	.223	.212	.214	2.064	.2064
67.		Thermal for I. H. P. $= \frac{\text{I. H. P.}}{H'_1} = \frac{\text{Item 52}}{\text{Item 56}}$	.562	.594	.572	.549	.501	.557	.581	.566	.554	.579	5.645	.5645
68.		Thermal $= \frac{H_1 - H_2}{H_1}$												
69.		Thermal $= \frac{H'_1 - H'_2}{H'_1}$	.214	.208	.240	.240	.240	.242	.264	.241	.220	.217	2.326	.2326
70.		Thermal $= \frac{H'_1 - H_2}{H'_1}$												
71.	Stand'd Gas per hour.	Fuel per I. H. P.	20.2	19.5	19.3	20.8	20.4	19.0	17.2	16.8	18.5	18.4	190.1	19.01
72.		Fuel and igniter per I. H. P.												
73.		Fuel per B. H. P.	22.4	22.0	22.6	22.7	22.5	22.5	20.6	19.1	22.3	21.6	218.3	21.83
74.		Fuel and igniter per B. H. P.												
75.		Cost per I. H. P. per hour, cents, gas at $1.00 per 1,000 cubic feet	2.02	1.95	1.93	2.08	2.04	1.90	1.72	1.68	1.85	1.84	19.01	1.901

Heat Balance.	B. T. U.	Per cent.
Heat turned into work	3,469	20.6
Heat rejected into jacket water	7,209	43.0
Heat rejected in exhaust	4,772	28.5
Heat rejected in condensation and radiation	1,325	7.9

Analysis of fuel by ....

Analysis of exhaust by ....

Heat equivalent by analysis, B. T. U., to fuel ....

" " " " " to exhaust ....

Remarks.—

The titles following paragraphs are numbered to correspond to the items as they appear in the report.

Especial attention has been given to the arrangement of this report, the items appearing in the order in which they are most readily deduced; *i.e.*, with one or two possible minor exceptions, no solution is called for for which the data have not already been supplied by the "log" or by a previous solution.

1. *Number.* 2. *Time Intervals.*

13. It is necessary to make frequent readings during the test, and item 1 corresponds to the numbers of these readings. What the interval between readings shall be is not material so long as the period is sufficiently long to eliminate errors that might creep in from too brief intervals. Ten-minute intervals are recommended in case the total run is not over two or three hours. When the nature of the test is such that several hours are necessary for the determination of average results, or quantities of fuel used, the time intervals may be lengthened as desired, although thirty minutes should probably be the maximum time between readings. When more convenient the time interval will be designated by "int."

REVOLUTIONS AND EXPLOSIONS.

3. *Total Revolutions.*

14. Whenever possible a continuous speed counter should be used, and it is advantageous to obtain simultaneous records from two such counters.

It is best to check the readings during each interval by a hand indicator, in order that some record may be had in case of accident to the continuous recorders—especially if one recorder only be used.

The fluctuations in speed of a gas engine of the single cylinder type are so great and so varied that the readings of a hand speed counter taken for one minute are apt to be far from reliable, and dependence should be put upon them only in case of absolute necessity.

4. *Revolutions per Minute (Mean).*

15. The mean number of revolutions per minute is simply the total number of revolutions for the time interval divided by the number of minutes in that interval. The mean speed thus ob-

tained is far more reliable than that obtained by the hand speed counter.

5. *Total Explosions.*

16. The general remarks regarding 3 apply. A continuous counter should be used whenever possible for determining the number of explosions. This can usually be done by connections to some stem or arm of the inlet valve, so that the counter is operated by the movements of this valve. Care should be taken to see that the ignitions are reliable and that every ignition gives an explosion.

Any method of simply counting the misses by sound or feeling, when they are frequent, is found to be absolutely unreliable.

6. *Explosions per Minute (Mean).*

17. These figures are obtained by dividing the total number of explosions for the time interval (item 5) by the number of minutes in that interval (item 2).

7. *Ratio of Revolutions to Explosions.*

18. This item gives a clear idea of the regularity of the explosions. In the single cylinder four-cycle engine making no misses, this ratio of the number of revolutions to the number of explosions would be 2.00. Any decrease in the number of explosions per minute would cause this ratio to increase, and a hasty glance at this item indicates at once the regularity of the explosions. In a single cylinder two-cycle engine the corresponding ratio is 1.00.

Date, *March* 1, 1902.

Report of..............."B"..Gas Engine Test.
Test No...
By..
Object...........*Efficiency, Proportions of Air to Gas*.......................................

1.		Number	1	2	3	4	5	6	7	8	Totals.	Average.
2.		Time intervals, minutes	10	10	10	10	10	10	10	10	80	10
3.	Revolutions and Exps.	Total revolutions	3,006	3,006	3,002	3,012	3,015	3,012	3,007	3,005	24,065	3,008
4.		R. P. M. (mean)	300.6	300.6	300.2	301.2	301.5	301.2	300.7	300.5	2,406.5	300.8
5.		Total explosions	981	952	966	951	926	934	918	940	7,568	946
6.		Exps. P. M. (mean)	98.1	95.2	96.6	95.1	92.6	93.4	91.8	94.0	756.8	94.6
7.		Ratio of revs. to explosions	3.06	3.16	3.11	3.17	3.25	3.22	3.28	3.20	25.45	3.18
8.	Jacket Water.	Weight in lbs.	93.3	93.3	93.3	93.3	93.3	93.3	93.3	93.3	746.4	93.3
9.		Weight in lbs. per hour	560	560	560	560	560	560	560	560	4.480	560
10.		Temperature range	39.0	39.0	40.0	39.0	39.9	39.0	40.0	39.0	314.0	39.3
11.		Max. veloc., feet per second	....	....	....	....	....	....	....	....	....	....
12.		Heat absorbed, B. T. U.	3,640	3,640	3,730	3,640	3,640	3,640	3,730	3,640	29,300	3,666
13.	Air.	Cubic feet	125.0	125.0	125.0	125.0	122.5	125.0	127.5	127.5	1,002.5	125.3
14.		Cubic feet per hour	750	750	750	750	735	750	765	765	6,015	752
15.		Temperature Fahr. deg.	87	87	87	87	87	87	87	87	....	87
16.		Weight per cubic feet	.0725	.0725	.0725	.0725	.7025	.0725	.0725	.0725	....	.0725
17.		Max. veloc. per second	....	....	....	....	....	....	....	....	....	....
18.		Specific heat, c_v	.1691	.1691	.1691	.1691	.1691	.1691	.1691	.1691	....	.1691
19.	Fuel *Gas*.	Cubic feet or ~~gals~~	15.25	15.25	15.25	15.50	15.00	15.25	15.75	15.00	122.25	15.28
20.		Cubic feet per hour	91.5	91.5	91.5	93.0	90.0	91.5	94.5	90.0	733.5	91.7
		Gallons per hour	....	....	....	....	....	....	....	....	....	....
21.		Temperature Fahr, deg.	87	87	87	87	87	87	87	87	....	87
22.		Weight per cubic feet	....	....	....	....	....	....	....	....	....	....
		Weight per gallon	....	....	....	....	....	....	....	....	....	....
23.		Max. veloc., ft. per second	....	....	....	....	....	....	....	....	....	....
24.		Specific heat, c_v	.1691	.1691	.1691	.1691	.1691	.1691	.1691	.1691	....	.1691
25.		Cu. ft. Standard Gas per hour (60° Fahr., 14.7 lbs. pres.)	87.5	87.5	87.5	89.0	86.2	87.5	90.5	86.2	698.9	87.36
26.	Valve Index Reading.	Gas	....	....	....	....	....	....	....	....	....	....
27.		Air	.75	.75	.75	.75	.75	.75	.75	.75	....	.75
28.		Values of n in Eq. $PV^n = P_1V_1^n$ for expansion curve	1.21	1.21	1.21	1.20	1.20	1.21	1.21	1.20	....	1.21

		Number	1	2	3	4	5	6	7	8	Totals.	Average.
29.	Mixture.	Temperature Fahr. deg.	113	113	113	113	113	113	113	112	....	113
30.		Weight per cu. ft.	.0692	.0692	.0692	.0692	.0692	.0692	.0692	.0693	....	.0692
31.		Max. veloc., ft. per sec.	....	....	....	....	....	....	....	....	....	....
32.	Pressure from Indicator Cards.	Press. at end of comp.	86	91	89	89	91	91	91	91	719	89.89
33.		Max. pressure or pressure at beginning of expansion	230	254	242	252	238	238	254	233	1,941	242.6
34.		Press. at end of expansion	40	43	42	40	42	43	43	43	336	42
35.		Pressure, if expansion be carried to end of stroke	39	40	40	39	40	40.5	40.5	41.0	330.0	41.3
36.		M. E. P.	58.0	58.5	57.0	57.0	57.0	56.5	61.0	61.0	466.0	58.25
37.	Exhaust.	Temperature, Fahr. deg.	1,060	1,100	1,100	1,060	1,100	1,120	1,120	1,140	8,800	1,100
38.		Max. veloc., ft. per sec.	....	....	....	....	....	....	....	....		
39.		Specific heat, c_v	.1691	.1691	.1691	.1691	.1691	.1691	.1691	.1691	....	.1691
40.	Ratios.	Air to gas, to	5.00	4.80	4.90	4.70	4.65	4.70	4.70	4.90	38.35	4.79
		Neutrals	1	1	1	1	1	1	1	1	8	1
			2.00	1.95	2.00	1.90	1.90	1.90	1.90	1.90	15.48	1.93
41.		Stroke to expansion	1.09	1.10	1.09	1.09	1.12	1.10	1.12	1.11	8.82	1.10
42.		Vols. v_2 to v_1	5.00	5.00	5.00	5.00	5.00	5.00	5.00	5.00	....	5.00
43.		Max. press. to M. E. P.	3.97	4.34	4.25	4.41	4.18	4.21	4.17	3.82	33.35	4.17
44.		Max. press. to comp. press.	2.68	2.79	2.72	2.83	2.62	2.64	2.79	2.56	21.63	2.70
45.		Value of R	.681	.681	.681	.681	.681	.681	.681	.682	....	.681
46.		Temperature Fahr. deg. at comp.	211	252	236	236	252	252	252	250	1,941	243
47.		Max. temperature, Fahr. deg.	1,610	1,600	1,470	1,550	1,580	1,700	1,600	1,600	12,710	1,589
48.	Energies.	Brake work, ft. lbs.	920,000	920,000	919,000	921,000	921,000	921,000	920,000	920,000	7,362,000	920,000
49.		Brake work, ft. lbs. per hr.	5,520,000	5,520,000	5,510,000	5,530,000	5,530,000	5,530,000	5,520,000	5,520,000	44,180,000	5,520,000
50.		B. H. P.	2.79	2.79	2.79	2.79	2.79	2.79	2.79	2.79	22.32	2.79
51.		B. T. U. equiv. to B. H. P.	1,180	1,180	1,180	1,180	1,180	1,180	1,180	1,180	9,440	1,180
52.		I. H. P.	3.66	3.59	3.54	3.49	3.40	3.40	3.60	3.69	28.37	3.55
53.		B. T. U. equiv. to I. H. P.	1,550	1,520	1,500	1,480	1,440	1,440	1,530	1,560	12,020	1,503
54.		Gas H. P.	22.4	22.4	22.4	22.8	22.1	22.4	23.1	22.1	179.7	22.5
55.		B. T. U. equiv. to Gas H. P. $= H_1$	9,500	9,500	9,500	9,650	9,350	9,500	9,800	9,350	76,150	9,519
56.		Heat supplied B. T. U. from Indicator Card $= H_1'$	2,280	2,200	2,020	2,160	1,970	2,190	2,480	2,400	17,700	2,213
57.		Heat extracted B. T. U. by observation $= H_2$	....	....	....	....	....	....	....	....	....	....
58.		Heat extracted B. T. U. from Indicator Card $= H_2'$	1,460	1,480	1,480	1,420	1,420	1,480	1,680	1,770	12,190	1,524
59.		I. H. P. − B. H. P.	0.87	0.80	0.75	0.70	0.61	0.61	0.81	0.90	6.05	0.76
60.		Throttling, cubic feet	.0498	.0496	.0494	.0498	.0524	.0498	.0474	.0479	.3961	.0495
61.		Throttling, per cent	27.0	26.9	26.8	27.0	27.2	27.0	25.7	25.9	213.5	26.7

62.	Work gained by complete expansion										
63.	Efficiencies. Mech. eff. $= \frac{\text{Item 50}}{\text{Item 52}}$	.763	.777	.788	.806	.821	.822	.775	.756	6.302	.788
64.	Thermal for B. H. P. $= \frac{\text{B. H. B.}}{H_1} = \frac{\text{Item 51}}{\text{Item 55}}$	.124	.124	.124	.122	.126	.124	.120	.126	.990	.124
65.	Thermal for B. H. P. $= \frac{\text{B. H. P.}}{H_1'} = \frac{\text{Item 51}}{\text{Item 56}}$	.517	.537	.584	.545	.599	.539	.476	.491	4.288	.536
66.	Thermal for I. H. P. $= \frac{\text{I. H. P.}}{H_1} = \frac{\text{Item 53}}{\text{Item 55}}$	.163	.160	.158	.153	.154	.152	.156	.167	1.263	.158
67.	Thermal for I. H. P. $= \frac{\text{I. H. P.}}{H_1'} = \frac{\text{Item 53}}{\text{Item 56}}$	.679	.691	.742	.685	.731	.657	.617	.650	5.452	.642
68.	Thermal $= \frac{H_1 - H_2}{H_1}$										
69.	Thermal $= \frac{H_1' - H_2'}{H_1'}$	.359	.328	.268	.342	.279	.324	.322	.262	2.484	.311
70.	Thermal $= \frac{H_1' - H_2}{H_1'}$										
71.	Stand. Gas per hour. Fuel per I. H. P.	23.9	24.4	24.7	25.5	25.3	25.7	25.7	23.4	198.6	24.8
72.	Fuel and Igniter per I. H. P.										
73.	Fuel per B. H. P.	31.4	31.4	31.4	31.9	30.9	31.4	32.4	30.9	251.7	31.5
74.	Fuel and Igniter per B. H P.										
75.	Cost per I. H. P. per hour, cents, gas at $1.00 per 1,000 cubic feet	2.39	2.44	2.47	2.55	2.53	2.57	2.57	2.34	19.86	2.48

Heat Balance.	B. T. U.	Per cent.
Heat turned into work	1,503	16.0
Heat rejected into jacket water	3,666	38.5
Heat rejected in exhaust	1,524	16.0
Heat rejected in condensation and radiation	2,826	29.5

Analysis of fuel by ....

Analysis of exhaust by ....

Heat equivalent by analysis, British Thermal Units, to fuel ....

" " " " " " " to exhaust ....

Remarks.—

JACKET WATER.

8. *Weight in Pounds.* 9. *Weight in Pounds per Hour.*

19. The simplest method of weighing the jacket water is by means of two oil barrels, with proper outlet pipes, set upon platform scales. The piping from the engine should be so constructed that the water may be fed into either barrel at will. It is hardly necessary to take the weight of the jacket water for each of the time intervals, provided the water flow has been carefully regulated before the test actually starts and the temperature of the feed water is found to be constant.

If the flow is maintained so nearly uniform that the fluctuations in temperature of the outlet water are not large—say for extreme range not over 15 degrees Fahr.—the total weight of water for the entire test may be recorded, and this result divided into equal proportions for the different runs.

20. Should the feed-water temperature show marked variations, which is likely to be the case if the water from the main passes through several buildings before reaching the engine, it should be weighed in small amounts and the weight and temperature entered on the "log."

The weight per hour is calculated directly from the values in item 8. In case of long runs it is well to determine the weight for each hour and to assign to each time interval its proportionate part.

10. *Temperature Range.*

21. Referring to items 10, 11, 12 of the "log," it is noticed that they are marked temperature of water entering, leaving, and barrel respectively. If the water used comes through a system of pipes running through warmed buildings, it is best, if possible. to allow the water to flow long enough to show fairly constant temperature before starting the test. This is especially necessary in winter when the water is taken from the city main—a difference of 45 degrees Fahr. is often observed under these conditions.

22. In cases where the weighing barrels are near the engine the temperature of the discharged water may be taken as it enters the barrel, which is often more convenient than to obtain the temperature just as the water leaves the engine. It has been found advantageous to keep a record of both when the conditions admit. In the tests shown no special attention was paid to the best tem-

perature of the leaving jacket water. In general it seems to be the opinion that the hotter the water is allowed to get without injuring lubrication or giving premature ignition the better. Some authorities give 160 degrees Fahr. as the best temperature for the discharge. Experiments seem to indicate that pressure in the water jacket is of little moment.

The range of temperature is the difference between the temperature of the leaving water and that of the entering.

11. *Maximum Velocity, Feet per Second.*

23. This has no direct bearing upon the test proper, but is inserted for use in making comparative tests of different engines, and for the determination of points of design of water inlets and outlets for the most effective results. The data collected would undoubtedly show variations depending upon the method of delivery, whether from natural circulation or forced.

12. *Heat Absorbed, B. T. U.*

24. The heat absorbed by the jacket water is a very large percentage of the entire heat supplied—often about 50 per cent. Since the specific heat of water is taken as unity, the calculation consists only in multiplying the number of pounds of water used during the interval by the range of temperature, this range of temperature being equal to the number of heat units absorbed per pound of water.

Data Given.	*Solution.*
s = specific heat of water = 1.	
t_r = temperature range.	
W = weight water for time interval.	$h_w = s \times t_r \times W.$
To Find—	
h_w = B. T. U. for the time interval.	
Examples.	
"*A*" Run No. 6:	
$s = 1.$	
$t_r = 72$ degrees.	
$W = 97$ pounds.	$h_w = 1 \times 72 \times 97 = 6{,}984$ B. T. U.
"*B*" Run No. 6:	
$s = 1.$	
$t_r = 39$ degrees.	
$W = 93.3$ pounds.	$h_w = 1 \times 39 \times 93.3 = 3{,}640$ B. T. U.

AIR.

25. It is very essential that the quantity of air used be accurately measured, and the simplest, and usually most accessible, method is by means of a gas meter of sufficient size. The accuracy of this meter should be carefully determined. (For methods of calibration, see "Report of the Committee Appointed to Standardize a System of Testing Steam Engines," paper No. 0943.

26. Even if the meter is carefully calibrated when working under suction, the readings will not be the same when the air enters under pressure, or vice versa. With the setting of the air valve as used in making test "B," the meter readings averaged for a given number of admissions 109.5 cubic feet when working under a pressure indicated by 2 inches of water, and 97.25 cubic feet when working under suction. The flow recorded by the meter was then 1.13 times as much when working under light pressure as when working under suction.

The meter through which the gas is measured works at all times under slight pressure, and for accurate determination of the proportions of air to gas the meter used for air measurement for the experiments upon which this paper is based was put under equal conditions by supplying air under pressure by means of an Ingersoll air compressor. In the absence of a compressor a blower may easily be arranged to accomplish the same result. The method of piping used is best explained by reference to the cut, Fig. 1.

27. *A* is the main compressed air line leading from the compressor to the air meter *B*. *C* is a relief tank filled with water to a depth corresponding to the head shown by the manometer attached to the gas pipe, and causing the flow of air through the meter to remain fairly constant. From *B* the air passes through the pipe *D*, directly to the engine save for the interposition of air bags *E* and *F* and an old muffler, used for the same purpose as the bags—namely, to reduce the variation in pressure of the air. At *H* is shown a manometer for determining the air pressure, and by means of the inlet valve *J* this pressure can be maintained the same as that shown by the gas manometer. By this method an accuracy in determining the proportions of air to gas was assured that otherwise could not be guaranteed.

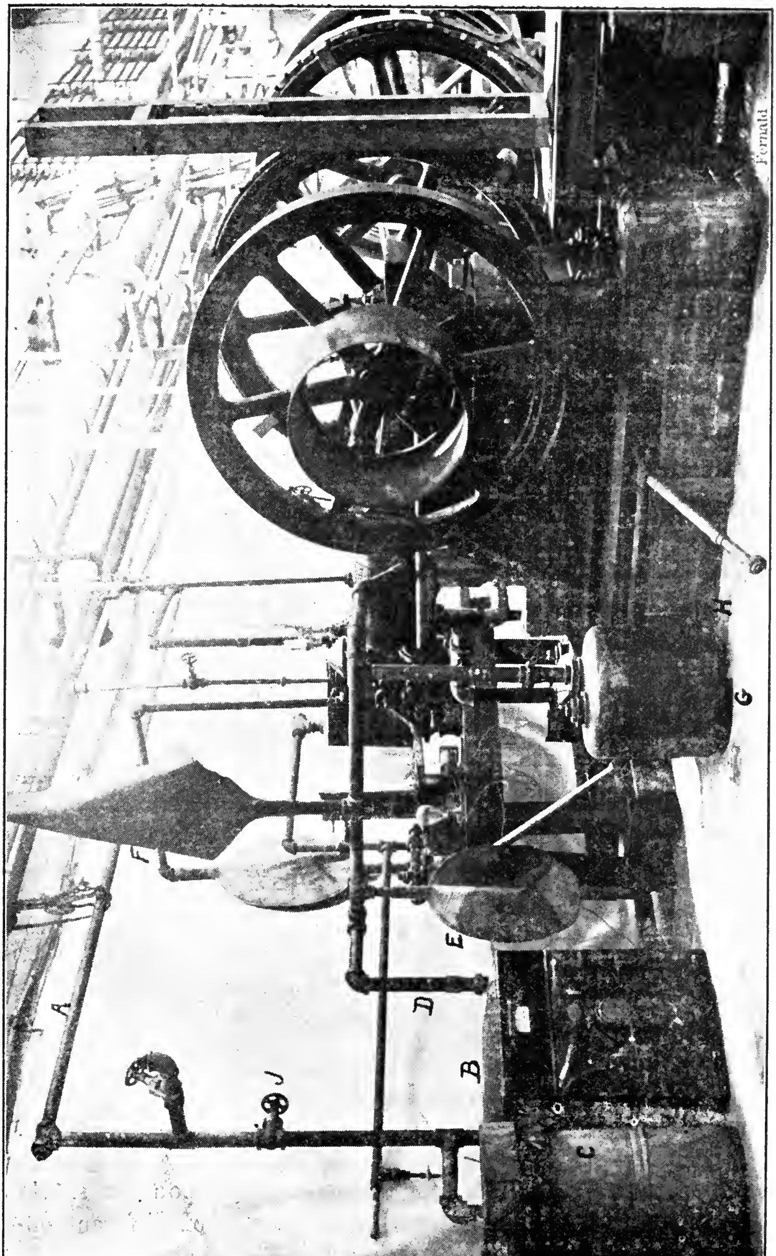

FIG. 1.

13. *Cubic Feet.* 14. *Cubic Feet per Hour.*

28. In reading the ordinary meter it is not sufficiently accurate to catch the readings by noting the positions of the index hands at the beginning and close of the time interval, but it is necessary to keep an observer at the meter and require the readings to be taken from the hand which indicates the single cubic feet, and whose complete revolution records 10 cubic feet.

The cubic feet per hour are readily calculated from the data for the given time interval.

15. *Temperature, Fahr.*

29. Under ordinary conditions it is sufficient to take the temperature of the air at the beginning and at the close of the test, but if the conditions are variable more frequent readings will be necessary.

16. *Weight per Cubic Foot.*

30. This weight is that of a cubic foot of air at the temperature given in item 15, and is found as follows:

Data Given.	*Solution.*
w_0 = weight cubic foot air at 32 degrees Fahr. = .0807 pound.	
$T_0 = 32° + 459° = 491°$ (absolute).	$\frac{w_1}{w_0} = \frac{T_0}{T_1}$ or $w_1 = .0807 \frac{491}{T_1}$.
T_1 = absolute temperature of air. = item 15 + 459°.	
To Find—	
w_1 = wgt. per cu. ft. at given temp.	
Examples.	
"*A*" Run No. 6: $T_1 = 86° + 459° = 545°$.	$w_1 = .0807 \frac{491}{545} = .0727$ pound.
"*B*" Run No. 6: $T_1 = 87° + 459° = 546°$.	$w_1 = .0807 \frac{491}{546} = .0725$ pound.

17. *Maximum Velocity in Feet per Second.*

31. Like item 11, this has no direct bearing upon the single test, but is for use in making comparative tests of different engines and for determination of points of design of air-inlet valves for the most effective results.

18. *Specific Heat, C_v.*

32. The specific heat of air at both constant pressure and constant volume may be found in any books on thermodynamics. The specific heat at constant volume, denoted by C_v, is the only value needed in this work—$C_v = .1691$.

FUEL.

33. The question of quantity of fuel used is the first to present itself, and the methods of determining this will depend entirely upon the kind of fuel. If coal is required the method of determining the amount is that usual in case of boiler tests, and is fully described in the *Transactions* of this Society, Vol. XXI., p. 34. The method used for ascertaining the amount of gas is by means of the standard gas meter. Methods of calibrating these meters are given in the "Report of the Committee appointed to Standardize a System of Testing Steam Engines," paper No. .0943. If oil is used it can easily be measured by means of calibrated tanks. For small engines using little oil, the tanks should be small in diameter, that the errors in measurement may be reduced as much as possible.

19. *Cubic Feet or Gallons.* 20. *Cubic Feet or Gallons per Hour.*

34. The method of measuring the fuel has already been explained. The quantities designated in item 19 refer to the amounts used for the given time interval, and if the fuel be gas, this would be the number of cubic feet as read directly from the meter.

If the fuel be oil the number of gallons for the period should be recorded.

21. *Temperature, Fahr.*

35. As in the case of air, it is usually sufficient to take the temperature of the gas at the beginning and at the close of the test, but if the conditions be variable more frequent readings will be necessary.

In case of engines that vaporize oil fuel before it enters the cylinder, this determination of temperatures is very difficult, if, indeed, possible. There are many forms of carburetors, and the temperatures of vaporization are found to vary greatly.

Further developments may determine the desired method of making reliable observations of the temperature of this type of fuel.

22. *Weight per Cubic Foot or Gallon.*

36. In case the fuel be gas, it is not always convenient to obtain the weight per cubic foot. If the temperatures of the air and gas be the same, this is hardly necessary, and even when these temperatures be different the necessity of knowing this weight is not sufficient to warrant great inconvenience or expense in obtaining it.

23. *Maximum Velocity in Feet per Second.*

37. As in items 11 and 17, this has no direct bearing upon the single test, but is for use in making comparative tests of different engines and for determination of points of design.

24. *Specific Heat, C_v.*

38. Unless it is convenient to ascertain accurately the desired specific heat of the gas used, no serious error results from taking the specific heat at constant volume the same as for air; namely, $C_v = .1691$.

25. *Cubic Feet of Standard Gas per Hour* (60 *Degrees Fahr.*, 14.7 *Pounds Pressure*).

39. The general standard recommended seems to indicate for "Standard Gas" the conditions given; namely, a temperature of 60 degrees Fahr. and a pressure corresponding to the usual atmospheric pressure. It is hardly necessary to make corrections for barometric readings, as the total possible variation is slight, and considering all other sources of error it is a question whether this supposed degree of refinement adds to the accuracy of the results.

Atmospheric pressure will then be understood to mean as indicated, 14.7 pounds per square inch.

40. The pressure shown by the manometer attached to the gas pipe might also be neglected in making the computations, as its effect upon the result is hardly perceptible. It has, however, been considered in the illustrative problems, the value having been recorded in the log of the test, thus adding no labor to the computations. It may be of interest to note that the total effect of neglect-

ing both the changes in barometric conditions and in the pressures shown by gas mains, when these changes are taken at a maximum, is less than one-half of one per cent. of the number of cubic feet per hour.

Data Given.

v_g = cu. ft. of gas per hour at t_g ° F.
T_g = absolute temp. of gas = t_g ° + 459°.
p_g = pressure under which gas is flowing.
= 14.7 lbs. + pressure shown by manometer.
T_s = 60° + 459° = 519° F., absolute temp. of standard gas.
p_0 = atmospheric pressure.
= 14.7 lbs. per sq. in.

To Find—

v_s = cu. ft. standard gas per hour.

Examples.

"A" Run No. 6:
v_g = 157.5 cu. ft.
T_g = 86° + 459° = 545°.
p_g = 14.7 + .1 = 14.8 lbs. per sq. in.

"B" Run No. 6:
v_g = 91.5 cu. ft.
T_g = 87° + 459° = 546°.
p_g = 14.7 + .1 = 14.8 lbs. per sq. in.

Solution.

$$\frac{v_s}{v_g} = \frac{p_g\,T_s}{p_0\,T_g}$$

$$v_s = v_g\,\frac{p_g\,T_s}{p_0\,T_g}$$

$$v_s = 157.5\,\frac{14.8 \times 519}{14.7 \times 545} = 151 \text{ cu. ft.}$$

$$v_s = 91.5\,\frac{14.8 \times 519}{14.7 \times 546} = 87.5 \text{ cu. ft.}$$

VALVE INDEX READING.

26. *Gas.* 27. *Air.*

41. This refers to the setting of the graduated inlet valves, and has no direct bearing on the single test, but is of value in making comparative tests.

28. *Value of n in* $P V^n = P_1 V_1^n$ *for Expansion Curve.*

42. At this point in the analysis, more or less difficulty is likely to be encountered. The necessity of carefully working out this value of n for each card may not at first seem apparent, but slight investigation shows it to be of great importance.

It is not unnatural to assume that the expansion curve and the compression curve, as given by the indicator card, follow very closely the adiabatic law.

Working upon this supposition leads in many cases to a network of difficulties, from which it is not easy to free one's self. For example, a case that came to the attention of the writer was as follows: In making a preliminary test, owing to the fact that it was inconvenient to determine the exact clearance volume of the engine, instructions were given to work out the clearance from the indicator card. The method is quickly shown by the following deductions and reference to the card shown in Fig. 2:

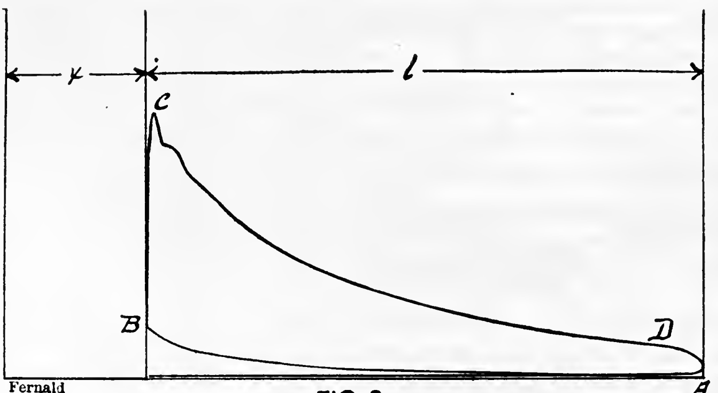

FIG. 2

43. Let v_1 = clearance volume of cylinder.
v_2 = total volume of cylinder.
p_0 = atmospheric pressure = 14.7 pounds per square in.
p_b = pressure at compression, from card.
l = length of stroke in feet or inches, as desired.
x = length of clearance in feet or inches, as desired.

It appeared a simple matter to solve for x in the following formulas: $\frac{p_b}{p_0} = \left(\frac{v_2}{v_1}\right)^n$ or $\left(\frac{p_b}{p_0}\right)^{\frac{1}{n}} = \frac{v_2}{v_1}$ which readily reduces to $\left(\frac{p_b}{14.7}\right)^{\frac{1}{n}} = \frac{x+l}{x}$, on the assumption that n for the compression curve was 1.41 or $\frac{1}{n} = .71$.

This gave an excessively large clearance volume, and the values of n worked out for the expansion curve varied greatly, but the best value seemed to be about 1.7, although the necessity of carefully determining the value of this exponent for each card was at once apparent.

Maximum temperatures computed on the basis of the value of n above proved to be in the neighborhood of 4,000 degrees Fahr. This surprisingly large value of n, together with the excessively high temperatures, led at once to a careful study of the problem, for it was apparent that no assumptions regarding the laws of expansion could safely be made. The clearance volume was at once determined by the usual method of filling the space with water. It may be well here to emphasize the necessity of great care in releasing all air from this clearance space as the water is poured in.

44. The values of n now deduced from the expansion curve for

FIG. 3

the diagrams taken from this particular engine showed for portions of the curve values as surprisingly low as those first determined were high. It was found that this exponent varied greatly in different parts of the curve for the diagrams taken from this engine, but no satisfactory law of variation could be determined. With the high pressure spring used in the indicator it was very difficult to obtain accurate values from the cards secured.

45. With a 240-lb. spring, which was the one used, each .01 of an inch in vertical measurement corresponds to 2.4 lbs. pressure, and errors resulting from irregularities in the curve, variations in the width of line, and mistakes in observation render it almost impossible to work with the degree of accuracy desired. In Fig. 3 is shown one of the cards taken in making test "A." The upper dotted curve represents isothermal expansion and the lower dotted curve adiabatic expansion.

It is seen at a glance that early in the expansion the full line follows very closely the adiabatic curve and then approaches nearer and nearer the isothermal as the expansion continues. The value of n that has been chosen for recording in item 28 is the value found for the latter part of the expansion, as this value is the one most needed in computations that follow. Owing to the fluctuations in many curves near the beginning of expansion, it has been

FIG. 4.

found extremely difficult to obtain values that could be regarded as reliable for the pressures desired.

46. The card shown in Fig. 4 was taken during test "B," and although the expansion curve seems to follow the adiabatic more closely than in Fig. 3, yet there is considerable variation even in this case.

Repeated calculations showed the best value of n for the expan-

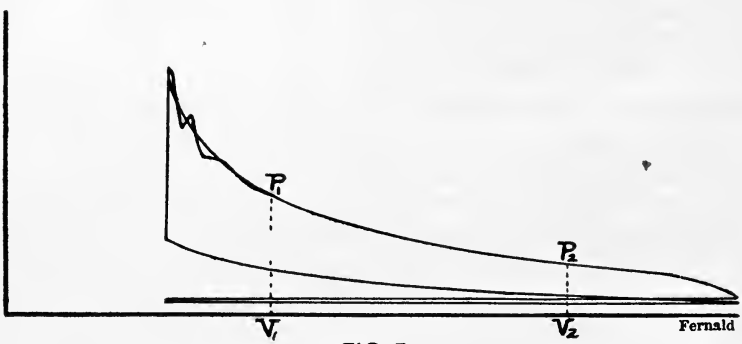

FIG. 5

sion curve for the cards of test "A" to be 1.14 and for test "B" 1.21 or 1.20.

The mathematical work involved in obtaining n is explained below:

Data Given.

Any corresponding values of pressure and volume obtained from the indicator card. Great care should be exercised in the selection of these points if the equation of the expansion curve is found to vary for different portions of the curve. All measurements must, of course, be made from the zero of volumes and zero of pressures.

V_1 = some assumed volume.
P_1 = pressure corresponding to V_1.
V_2 = another assumed volume.
P_2 = pressure corresponding to V_2.

Examples.

"A" Run No. 6 (For this problem the values of P and V are selected as shown in Fig. 5):

$V_1 = 6.2 \qquad P_1 = 141$
$V_2 = 12.2 \qquad P_2 = 65$

"B" Run No. 6:

$V_1 = 5 \qquad P_1 = 108$
$V_2 = 8 \qquad P_2 = 61$

Solution.

$$P_1 V_1^n = P_2 V_2^n$$

$$\frac{P_1}{P_2} = \left(\frac{V_2}{V_1}\right)^n$$

$$\log. \frac{P_1}{P_2} = n \log. \frac{V_2}{V_1}$$

$$n = \frac{\log. \dfrac{P_1}{P_2}}{\log. \dfrac{V_2}{V_1}}$$

or, $$n = \frac{\log. P_1 - \log. P_2}{\log. V_2 - \log. V_1}$$

$\log. P_1 = 2.14922 \qquad \log. V_2 = 1.08636$
$\log. P_2 = 1.81291 \qquad \log. V_1 = 0.79239$
$\quad .33631 \qquad\qquad .29397$

$$n = \frac{.33631}{.29397} = 1.14$$

Solving by the first formula,

$$n = \frac{\log. \dfrac{P_1}{P_2}}{\log. \dfrac{V_2}{V_1}},$$

shortens the work if a slide rule is used

$$\frac{P_1}{P_2} = \frac{108}{61} = 1.77, \text{ log.} = 0.24797$$

$$\frac{V_2}{V_1} = \frac{8}{5} = 1.6, \text{ log.} = 0.20412$$

$$n = \frac{.24797}{.20412} = 1.21$$

47. The following values of n were worked out with great care from a series of volumes and corresponding pressures, and show the fluctuations to which this calculation is subject. The values are arranged as derived in regular order from left to right, but do not include values for very small volumes, as the variations in pressure were too great to enable any reliable readings. As derived from the card selected n = 1.21, 1.12, 1.14, 1.16, 1.13, 1.135, 1.12, 1.10, 1.10 1.12, 1.15, 1.15, 1.15 1.14, 1.16, 1.13.

A variation of a single pound in the pressure, or, in some cases, of one-half pound, determines the accuracy of the deduction, but with such high pressure springs this degree of refinement is impossible. It was noted in several cases that the value of n for the compression curve was slightly higher than for the expansion curve.

A simple method of ascertaining whether this exponent is the same for corresponding compression and expansion curves is as follows:

48. Locate a series of different volumes upon the card in question and determine the corresponding pressures for both the compression and expansion curves. If the ratio of the corresponding pressures of the two curves remains constant, the exponents for the equations of the two curves will be found to be identical.

Let V_1 and V_2 = two volumes chosen, as in Fig. 5. For the expansion curve the two pressures = P_1 and P_2 and for the compression curve the pressures corresponding to the same volumes = P_3 and P_4. Let x be the exponent in the equation of the expansion curve, viz: $P_1\ V_1^x = P_2\ V_2^x$ and let y be the corresponding exponent for the compression curve, or $P_3\ V_1^y = P_4\ V_2^y$

By division $\frac{P_1\ V_1^x}{P_3\ V_1^y} = \frac{P_2\ V_2^x}{P_4\ V_2^y}$. If now the exponents for the two curves be the same, *i. e.*, if $y = x$ then $\frac{P_1\ V_1^x}{P_3\ V_1^x} = \frac{P_2\ V_2^x}{P_4\ V_2^x}$ or $\frac{P_1}{P_3} = \frac{P_2}{P_4}$.

MIXTURE.

29. *Temperature, Degrees Fahr.*

49. One of the longest deductions, as well as one of the most difficult, but at the same time of great importance, is that of obtaining the temperature of the mixture in the cylinder, composed of air, gas, and exhaust gases, or neutrals, as the last are sometimes called. The problem involves the larger portion of all the readings made during the test, as well as the careful determination of the temperature of the exhaust gases.

The mixture passes through a wide range of temperatures, and the only one that can be determined is that of the exhaust. This has proved to be of such great importance and so little seems to be known regarding the determination of this factor, that much of the time devoted to the subject of gas engine testing was given to the solution of this problem.

Very few methods for measuring this temperature seem to exist, and these are either too inaccurate to be of any value, or in cases where the results seem to indicate accuracy, the method is too complicated or the apparatus too expensive for use under ordinary circumstances.

50. Even the committee appointed by the Society to "Standardize a System of Testing Steam Engines" avoids the question entirely. This committee speaks of the "observed temperature of the exhaust gases," but in no way is any intimation given of a method of observation.

A simple device for determining these temperatures seems so necessary that it is deemed wise to give a complete description of the apparatus used by the writer, in a separate paper entitled, "A Method of Determining the Temperatures of Exhaust Gases," and to be presented as paper No. 0932 at this meeting.

To avoid confusion, the computations for determining the temperature of the mixture will be divided into two parts.

PART I.

51. To determine the combined volumes of air and gas per stroke and the temperature of the same:

Case 1. When the temperatures of the incoming air and gas are equal:

Data Given.	*Solution.*
$Mins.$ = No. minutes in time interval.	
g = gas per min. $= \dfrac{\text{item 19}}{\text{mins.}}$	
a = air per min. $= \dfrac{\text{item 13}}{\text{mins.}}$	$v' = \dfrac{g}{Ex.\ P.\ M.}$
T_2 = absolute temp. gas. $= T_1$ absolute temp. air. = item. 21 or 15 + 459°.	$v'' = \dfrac{a - v'\,[Ms.\ P.\ M.]}{\frac{1}{2}\,[R.\ P.\ M.]}$
$Ex.\ P.\ M.$ = explosions p. min. = item 6.	Since $T_{ag} = T_1 = T_2$,
$R.\ P.\ M.$ = revolutions p. min. = item 4.	$v_{ag} = v' + v''$.
$Ms.\ P.\ M.$ = explosions missed per min. $= \frac{1}{2}\,R.\ P.\ M. - Ex.\ P.\ M.$ for single cylinder four-cycle engine.	
To Find—	
v' = cu. ft. gas per explosion at T_1.	
v'' = cu. ft. air per explosion at T_2.	
T_{ag} = absolute temperature in F.° resulting from combining air and gas $= T_1 = T_2$.	
v_{ag} = combined vol. in cu. ft. of air and gas per explosion at T_{ag}.	

Examples.

"A" Run No. 6:

$$g = \frac{26.25}{10} = 2.625 \text{ cu. ft.}$$

$$a = \frac{194}{10} = 19.4 \text{ cu. ft.}$$

$T_1 = T_2 = 86° + 459° = 545°.$
Ex. P. M. = 133.9.
R. P. M. = 267.9.
Ms. P. M. = 0.

$$v' = \frac{2.625}{133.9} = .0196 \text{ cu. ft.}$$

$$v'' = \frac{19.4 - .0196 \times 0}{133.9} = .145 \text{ cu. ft.}$$

And since $T_{ag} = T_1 = T_2$,
$v_{ag} = v' + v'' =$ 1646 cu. ft.

"B" Run No. 6:

$$g = \frac{15.25}{10} = 1.525 \text{ cu. ft.}$$

$$a = \frac{125.0}{10} = 12.5 \text{ cu. ft.}$$

$T_1 = T_2 = 87° + 459° = 546°.$
Ex. P. M. = 93.4.
R. P. M. = 301.2.
Ms. P. M. = 57.2.

$$v' = \frac{1.525}{93.4} = .0163 \text{ cu. ft.}$$

$$v'' = \frac{12.5 - .0163 \times 57.2}{150.6} = .0769 \text{ cu. ft.}$$

And since $T_{ag} = T_1 = T_2 = 546°$,
$v_{ag} = v' + v'' = .0932$ cu. ft.

Case 2. When the temperatures of the incoming air and gas are different:

Data Given.

Mins. = No. minutes in time interval.

g = gas per min. $= \dfrac{\text{item 19}}{\text{mins.}}$

a = air per min. $= \dfrac{\text{item 13}}{\text{mins.}}$

T_1 = absolute temp. air in F.°
= item 15 + 459°.

T_2 = absolute temp. gas in F.°
= item 21 + 459°.

Ex. P. M. = explosions per minute = item 6.

R. P. M. = revolutions per minute = item 4.

Ms. P. M. = explosions missed per min.
= ½ *R. P. M.* − *Ex. P. M.* for single cylinder four-cycle engine.

w_1 = wgt. per cu. ft. air at T_1 = item 16.

w_0 = wgt. per cu. ft. air at 32° F. = .0807 lb.

T_0 = absolute temp. corresponding to 32° F. = 491°.

v' = cu. ft. gas per explosion at T_2 } as found under Case 1.
v'' = cu. ft. air per explosion at T_1 }

Solution.

The general equation from which T_{ag} can be computed is

$$c_p(T_1 - T_{ag})\,w_a = c_p(T_{ag} - T_2)\,w_g.$$

It is now necessary to find w_a and w_g.

If it is not convenient to obtain the weight of gas per cubic foot, the best that can be done is to take the weight of gas the same as that of air at the same temperature. The error involved by so doing is not serious.

Then $$w_2 = w_0\,\frac{T_0}{T_2}$$

$$w_a = w_1 \times v''$$
$$w_g = w_2 \times v'$$

T_{ag} can now be computed from the equation above.

$$v''' = v'\,\frac{T_{ag}}{T_2}$$

$$v'''' = v''\,\frac{T_{ag}}{T_1}$$

$$v_{ag} = v''' + v''''.$$

C_p = specific heat of air at constant pressure.

To Find—

w_2 = wgt. cu. ft. gas at T_2.
w_a = wgt. of air per explosion at T_1.
w_g = wgt. of gas per explosion at T_2.
T_{ag} = absolute temp. in F.° resulting from combining air at T_1 and gas at T_2.
v''' = cu. ft. gas per explosion at T_{ag}.
v'''' = cu. ft. air per explosion at T_{ag}.
v_{ag} = combined vol. in cu. ft. of air and gas per explosion at T_{ag}.

PART II.

52. The combined volumes of air and gas used per explosion, together with the temperature of the same, having been computed as indicated in Part I. of this section, the problem is now to determine the temperature of the final mixture in the cylinder after the air and gas have united with the exhaust gases in the clearance space—unless the engine is of the scavenging type.

It is to be noticed that if the governor is of the hit or miss type, the exhaust stroke following a miss corresponds to a scavenging stroke.

Data Given.

Assume the weight of the final mixture equal to the weight of air at the same temperature. Assume the specific heats of the different mixtures the same as for air.

w_0 = weight cu. ft. air at 32° F. = .0807 lb.
T_0 = absolute temp. corresponding to 32° F. = 491°.
C_p = specific heat air at constant pressure.
= specific heat air and gas at constant pressure.
= specific heat final mixture at constant pressure.
T_{ag} = absolute temp. of air and gas entering cylinder as computed in Part I.

Solution.

$$w_3 = w_0 \frac{T_0}{T_{ag}}$$
$$w_4 = w_0 \frac{T_0}{T_e}$$

$$w_{ag} = w_3 \times v_{ag}$$
$$w_e = w_4 \times v_b$$

T_m can now be computed from the equation

$$c_p (T_m - T_{ag}) w_{ag} = c_p (T_e - T_m) w_e$$
$$T_m = \frac{T_{ag} w_{ag} + T_e w_e}{w_{ag} + w_e}$$

v_{ag} = cu. ft. air and gas at T_{ag} as computed in Part I.

T_e = absolute temp. exhaust gases at atmospheric pressure.

= temp. observed and recorded in item 14 of log + 459°.

v_b = vol. of clearance in cu. ft.

To Find—

w_3 = weight cu. ft. of air and gas at T_{ag}.

w_4 = weight cu. ft. of exhaust gases at T_e.

w_{ag} = weight of air and gas per explosion at T_{ag}.

w_e = weight of exhaust gases per explosion at T_e.

T_m = absolute temp. of final mixture in cylinder.

$T_m - 459° =$ F.° as in item 29.

Examples.

"A" Run No. 6:

$w_0 = .0807$ lb.

$T_0 = 32° + 459° = 491°$.

$T' = T_1 = T_2 = 86° + 459° = 545°$.

$v_{ag} = .1646$ cu. ft. as computed in Part I.

$T = 180° + 459° = 639°$ by observation.

$v_b = .055$ cu. ft. by measurement.

$$w_3 = .0807 \frac{491}{545} = .0727 \text{ lb.}$$

$$w_4 = .0807 \frac{491}{639} = .0621 \text{ lb.}$$

$$w_{ag} = .0727 \times .1646 = .0120 \text{ lb.}$$

$$w_e = .0621 \times .055 = .00342 \text{ lb.}$$

$$(T_m - 545)\ .012 = (639 - T_m)\ .00342$$

$$.01542\ T_m = 8.72$$

$$T_m = 565°$$

$$565° - 459° = 106° \text{ F.}$$

"B" Run No. 6:

$w_0 = .0807$ lb.

$T_0 = 32° + 459° = 491°$.

$T_{ag} = 87° + 459° = 546°$.

$v_{ag} = .0932$ cu. ft. as computed in Part I.

$T' = 188° + 459° = 647°$ by observation.

$v_b = .037$ cu. ft. by measurement.

$$w_3 = .0807 \frac{491}{546} = .0725 \text{ lb.}$$

$$w_4 = .0807 \frac{491}{647} = .0612 \text{ lb.}$$

$$w_{ag} = .0725 \times .0932 = .00676 \text{ lb.}$$

$$w_e = .0612 \times .037 = .00226 \text{ lb.}$$

$$(T_m - 546)\ .00676 = (647 - T_m)\ .00226$$

$$.00902\ T_m = 5.15$$

$$T_m = 572°$$

$$572° - 459° = 113° \text{ F.}$$

30. *Weight per Cubic Foot.*

53. *Data Given.*

w_0 = weight cu. ft. air at 32° F. = .0807.

T_0 = absolute temp. corresponding to 32° F. = 491°.

T_m = absolute temp. of mixture as computed in 29.

Solution.

$$w_5 = w_0 \frac{T_0}{T_m}$$

$$= .0807 \frac{491}{T_m}$$

To Find—

w_5 = weight cu. ft. of mixture at T_m.

Examples.

"A" Run No. 6:
$T_m = 565°$ from 29.

$$w_5 = .0807 \frac{491}{565} = .0701 \text{ lb.}$$

"B" Run No. 6:
$T_m = 572°$ from 29.

$$w_5 = .0807 \frac{491}{572} = .0692 \text{ lb.}$$

31. *Maximum Velocity, Feet per Second.*

54. This velocity refers strictly to the rate of flow of the entering mixture of air and gas and not to the final mixture in the cylinder. As in columns 11, 17, 23, this maximum velocity is for use in making comparative tests only, and for determining points of design.

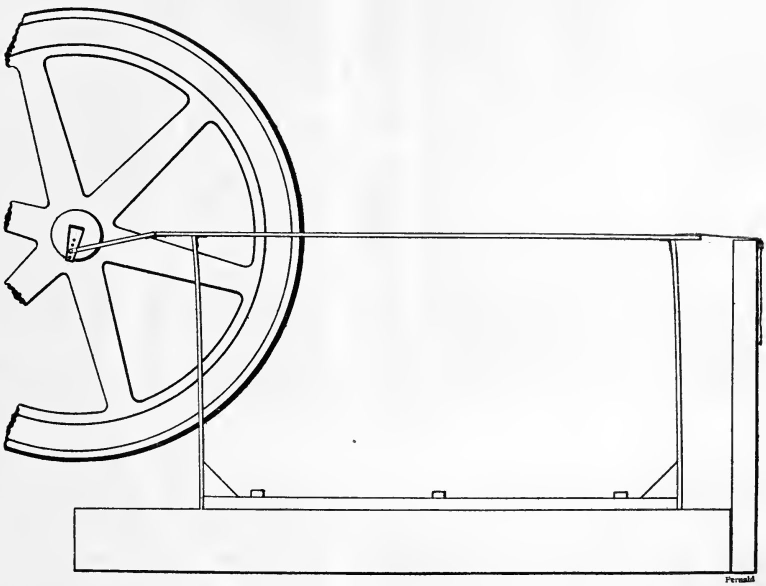

FIG. 6.

PRESSURE FROM INDICATOR CARDS.

55. The indicator card bears such an important relation to the test that great care should be taken to have the reducing motion, indicator, and all connections properly adjusted. Owing to the ex-

treme maximum pressure resulting from the explosion, the ordinary steam engine indicator is far too delicate for service. A

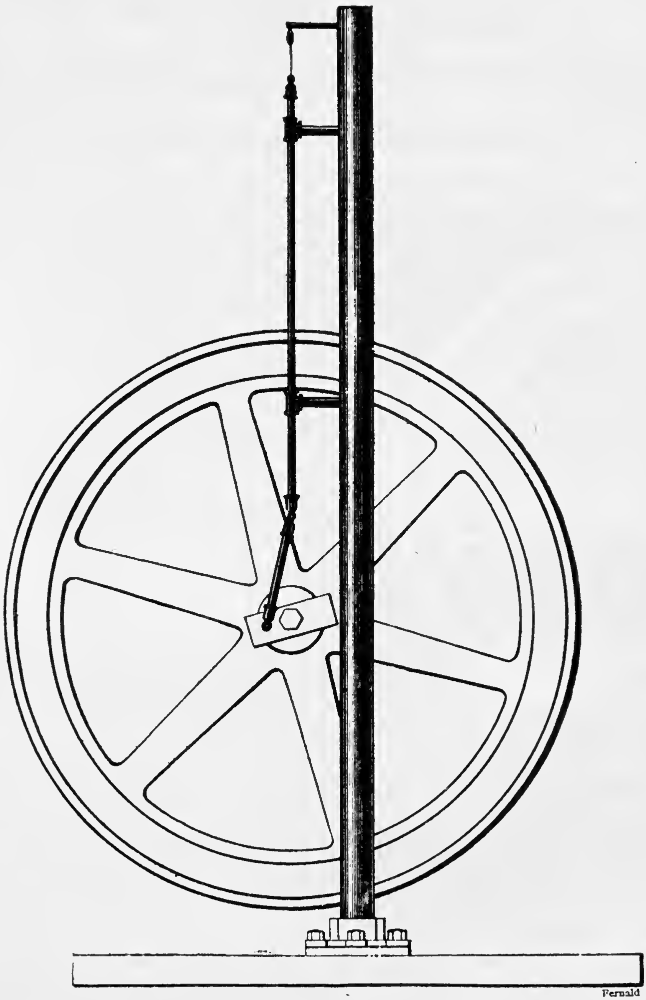

Fig. 7.

special gas engine indicator with a small piston, strong spring, strengthened pencil arm and carefully adjusted pin connections is very necessary. It is well at all times to keep the pipe leading to the indicator packed with cotton waste saturated frequently with water, to prevent the temperature of the indicator from becoming excessive and thus rendering it unreliable. When it can be easily done it is an excellent plan to jacket this connection.

Care should always be taken to record on the log sheet the scale of the spring used. Never trust to memory for this or any other fact.

When the compression cylinder is independent, the scale of the spring used for the cards taken from this cylinder should also be recorded on the log sheet, and the card so taken should be reduced to the same scale and combined with the power card.

For convenience in making this combination care should be exercised in adjusting the reducing motion to insure the same length of card in both cases.

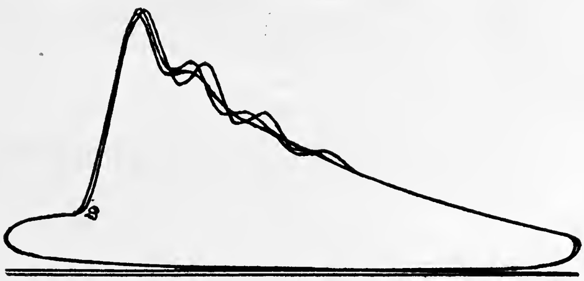

FIG. 8 Fernald

56. The reducing motion may be any one of the forms usually constructed for such purposes, provided it can be readily attached.

The lack of a cross-head and the enclosed crank case on the modern gas engine prevent the use of some reducing motions. The forms shown have been used with perfect satisfaction upon the engines tested at Columbia, that shown in Fig. 6, for horizontal engines, and that in Fig. 7 for vertical. They are especially recommended for cheapness and for the ease with which they can be constructed on the grounds.

The utmost care should be exercised in adjusting the reducing motion. It is not sufficiently accurate to regulate merely by the eye, but sample cards should be taken and adjustments made until the motion is correct.

57. Fig. 8 shows the result when the true position of the reducing motion is seriously disturbed. The engine appears to continue compressing long after the piston has started toward the crank

end of the cylinder, the maximum compression pressure being shown at S, at which point the ignition takes place. The series of explosions which follow in rapid succession when the ignition is

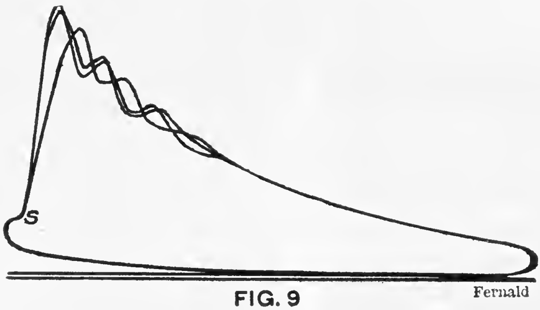

FIG. 9

early, as is the case in this instance, appear seriously distorted, continuing apparently through the greater part of the stroke, an effect produced by the relatively high speed of the indicator drum. With

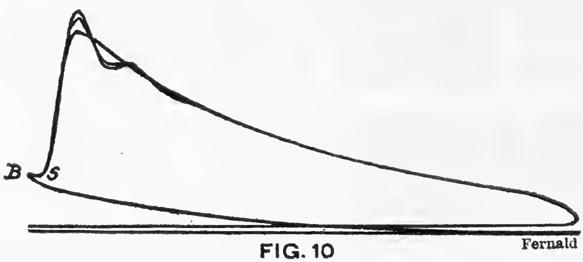

FIG. 10

a card so seriously distorted, the error is quickly perceived, but when the error is as shown in Fig. 9, the difficulty is not quite as apparent.

A casual glance might lead one to suppose this form of card

FIG. 11

due to late ignition, but closer examination will show the compression pressure to be steadily rising to the point S. This indicates that the reducing motion is still incorrectly adjusted.

When a similar effect is produced by late ignition only and the reducing motion is properly set, the line from B to S is horizontal or even depressed, as shown in Figs. 10 and 11.

58. Fig. 12 shows a case that is approaching the limiting position of the reducing motion, but close scrutiny shows the compression to continue slightly after the engine has passed its dead

FIG. 12

centre. The early ignition shown in Fig. 12 tends to conceal the fault due to the reducing motion, and in cases where the ignition is premature great care should be taken to secure the proper regulation before ignition is set too early.

In Fig. 13 is shown the typical card with early ignition and the reducing motion properly set so that the maximum compression as shown by the card corresponds to the position of the inner dead-centre of the engine.

FIG. 13

Errors of the kind just described, although apparently slight in the limiting cases, should be guarded against, as the deductions of the entire test may be seriously affected by such oversights.

To insure accuracy it is better to establish the line $L\ C$, Fig. 14, by allowing the indicator drum to remain stationary while the indicator cock is opened and the pencil moved up and down, than simply to draw $L\ C$ at right angles to the atmospheric line. The lines of zero volume and zero pressure can then be drawn parallel to $L\ C$ and the atmospheric line respectively.

32. *Pressure at End of Compression.*

59. Referring to the diagram (Fig. 14), the absolute compression pressure is represented by the distance LB, measured from the line of zero pressures.

33. *Maximum Pressure, or Pressure at the Beginning of Expansion.*

60. Some care is necessary in determining this value in cases where the diagram shows a series of explosive waves. In Fig. 14

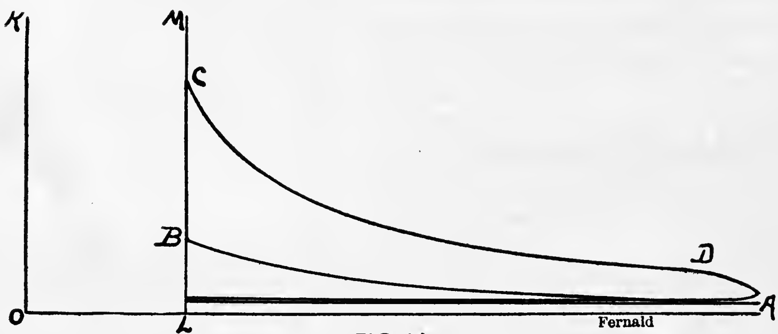

FIG. 14

the distance measured from the zero pressure line to the point C is easily determined as the maximum pressure, but in cases like Fig. 15 the possibility of error is much greater.

By marking the centre points of the series of explosion peaks and continuing the expansion line from some lower point in the curve through these points, a fairly accurate determination of the maximum pressure may be made.

If the value of n determined in 28 be regarded as accurate, a very good check may be had upon the maximum pressure obtained graphically, by computing the pressure from the equation of the curve.

34. *Pressure at End of Expansion.*

61. The determination of the end of expansion, or point of release, is often attended by some difficulty, as the point is not marked by any sudden change in the direction of the expansion curve, but is at the point of inflexion, D (Fig. 15). As in the other cases the pressure is measured from the zero lines of pressures.

35. *Pressure if Expansion were Carried to End of Stroke.*

62. This value is readily obtained from the equation of the expansion curve, the value of the exponent n having been computed in 28. If the expansion were thus continued it would give the point H, as shown in Fig. 15. The pressure corresponding to the point H is deduced as follows:

Data Given.

V_1 = volume at some point of the card.
P_1 = pressure corresponding to V_1.
n = value deduced in 28.
V_2 = total volume of cylinder.

To Find—

P_2 = pressure corresponding to V_2; *i. e.*, if expansion continued to H.

Solution.

$$P_1 \; V_1^n = P_2 \; V_2^n$$

$$P_2 = P_1 \left(\frac{V_1}{V_2}\right)^n$$

$$\log. P_2 = \log. P_1 + n \log. \frac{V_1}{V_2}$$

The volumes being used as a ratio, the piston area may be omitted, the ratio of lengths being the same as the ratio of volumes, as is customary in working with indicator cards.

Examples.

"A" Run No. 6:
V_1 = 12.2 for one point of card.
P_1 = 65 pounds pressure corresponding to V_1.
n = 1.14 from item 28.
V_2 = 15.9 for total volume of cylinder.

$$P_2 = 65 \left(\frac{12.2}{15.9}\right)^{1.14}$$

$$1.14 \log. \frac{12.2}{15.9} = 9.86885 - 10$$
$$\log. \; 65 = 1.81291$$
$$\log. \; P_2 = 1.68176$$
$$P_2 = 48$$

"B" Run No. 6:
V_1 = 8 for one point of card.
P_1 = 61 pounds pressure corresponding to V_1.
n = 1.21 from item 28.
V_2 = 11.25 for total volume of cylinder.

$$P_2 = 61 \left(\frac{8}{11.25}\right)^{1.21}$$

$$1.21 \log. \frac{8}{11.25} = 9.82076 - 10$$
$$\log. \; 61 = 1.78533$$
$$\log. \; P_2 = 1.60609$$
$$P_2 = 40.5$$

36. *Mean Effective Pressure.*

63. It is necessary to determine the area of the diagram, and for this purpose the planimeter should be used, although other methods will answer, but are very tedious and not as accurate. The length of the diagram is determined by the lines LM and RW (Fig. 15), and is equal to LR.

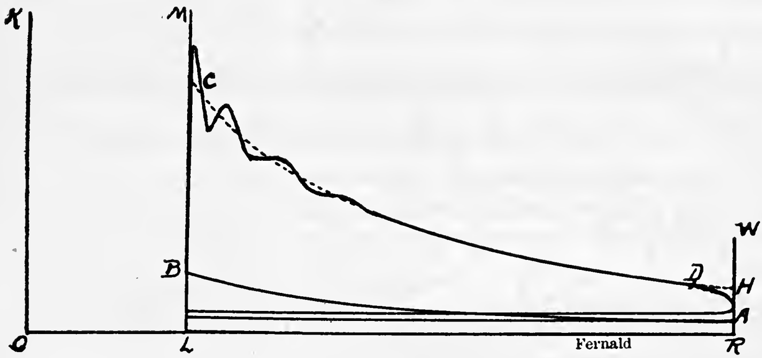

FIG. 15

The area in square inches divided by the length of the diagram in inches, multiplied by the scale of the spring used, will give the mean effective pressure in pounds per square inch.

Data Given.

A = area diagram, sq. ins.
L = length diagram, ins.
S = scale of spring used.

To Find—

$M.\ E.\ P.$ = mean effective pressure.

Solution.

$$M.\ E.\ P. = \frac{A}{L} S$$

Examples.

"A" Run No. 6:
$A = .84$ sq. ins.
$L = 3.04$ ins.
$S = 240$ lbs. per inch height.

$$M.\ E.\ P. = \frac{.84}{3.04} 240 = 66.3 \text{ lbs. per sq. in.}$$

"B" Run No. 6:
$A = .65$ sq. ins.
$L = 2.76$ ins.
$S = 240$ lbs. per inch height.

$$M.\ E.\ P. = \frac{.65}{2.76} 240 = 56.5 \text{ lbs. per sq. in.}$$

EXHAUST.

37. *Temperature, Degrees Fahr.*

64. The subject of exhaust temperatures has occasioned much discussion since the advent of gas engines, but, as far as the writer

is informed, very few methods have been devised for securing even an approximation to the correct temperatures, and even these devices seem too complicated or too expensive for ordinary use.

Until recently there seems to have been little or no comprehension of the real temperatures of these exhaust gases, and to-day, even when an appreciation of the high range of temperatures is had, there seems to be a feeling that some slight modification of the design of the engine will enable the inventor to save this excessive waste, and thus to secure for himself the fortune that would result from the invention of engines showing a higher efficiency. In the writer's opinion this cannot be done with engines working on the

FIG. 16

Otto or Beau de Rochas cycle, save by a reduction of the final or exhaust pressure. It can be easily shown that with high pressure of release, high exhaust temperatures must follow. Some attempts have been made to reduce this terminal pressure by compounding or other means, and while some degree of success has been attained, yet the results have not been of a nature to create any great enthusiasm, and this leads at once to the query, "Is this the best cycle to use for the most effective results?" It is not within the scope of this paper to treat this problem, but experimental results lead to renewed interest in the paper read by Mr. Charles E. Lucke at the last meeting of the Society, entitled, "The Heat-Engine Problem," in which the possibilities of the different cycles are treated in detail, leading to strong recommendations for non-explosive engines.

65. The details of apparatus for determining the temperature of the exhaust gases are given in the paper previously referred to in paragraph 29, but the method of computation will be outlined under the present heading.

Having determined the temperature of the mixture in the cylinder at atmospheric pressure, as recorded in item 29, the method employed for the present deduction is very simple:

Data Given.	*Solution.*
(See Fig. 16.)	The volumes at A and H being equal,
p_a = pressure at A = atmosph. pressure = 14.7 lbs.	$\frac{T_h}{T_m} = \frac{p_h}{p_a}$ or $T_h = T_m \frac{p_h}{p_a}$
p_h = pressure at H = value of item 35.	$T_h - 459° = $ F.°
T_m = absolute temp. of mixture at atmosph. pres. = item 29 + 459°.	
To Find—	
T_h = absolute temp. of exhaust.	
Examples.	
"A" Run No. 6:	
$p_a = 14.7$ lbs.	
$p_h = 48$ lbs.	$T_h = 565 \frac{48}{14.7} = 1{,}846°$
$T_m = 106° + 459° = 565°$.	$1{,}846° - 459° = 1{,}387°$ F.
"B" Run No. 6:	
$p_a = 14.7$ lbs.	
$p_h = 40.5$ lbs.	$T_h = 572 \frac{40.5}{14.7} = 1{,}579°$
$T_m = 113° + 459° = 572°$.	$1{,}579° - 459° = 1{,}120°$ F.

66. Owing to the throttling of the entering gases the mixture in the cylinder is slightly below atmospheric pressure at full cylinder volume, but the amount is not large and is not worth considering when compared with the possible errors that may occur in obtaining the various pressures from the indicator card when a high-pressure spring is used. The temperature at release would, of course, be considerably higher than that given for the point H. That the exhaust temperatures cannot be as low as has been frequently supposed, even by recognized authorities, is readily shown by a very brief calculation:

For an extreme case give both the temperature of the mixture and the pressure H very low values, thus reducing the exhaust temperature to a minimum.

Suppose the temperature of the mixture to be as cool as the atmosphere on a cool day, say 60 degrees Fahr., and take the pressure at H as low as 35 pounds. The resulting exhaust temperature even under these extreme conditions, must be

$$T_h = 519 \frac{35}{14.7} = 1{,}234 \text{ degrees or } 775 \text{ degrees Fahr.}$$

38. *Maximum Velocity, Feet per Second.*

67. As in items 11, 17, 23, 31, this maximum velocity is for use in making comparative tests only, and for determining points of design.

39. *Specific Heat, C_v.*

68. Unless the specific heat of the exhaust gases can be readily obtained, the specific heat at constant volume may be taken the same as for air, namely, $C_v = .1691$.

RATIOS.

40. *Air to Gas to Neutrals.*

69. By "neutrals" is meant the products of combustion left in the cylinder of a non-scavenging engine after exhaust—an amount equal in volume to that of the clearance space.

In determining the proportions called for, the number of cubic feet of gas is taken as unity, and the temperature of the gas is taken as the basis for the computation. The quantities of air and neutrals must be reduced to corresponding amounts at this temperature.

The cubic feet of air used in ten minutes or an hour cannot be taken as a basis of comparison, without modification, owing to the misses of explosions, in which case air is taken into the cylinder without gas.

Data Given.

T_1 = absolute temperature of air = item 15 + 459°.

T_2 = absolute temperature of gas = item 21 + 459°.

T_e = absolute temp. of exhaust gases = item 14 of log + 459°. at atmospheric pressure.

v' = cu. ft. gas per explosion at T_2 as computed in 29.

v'' = cu. ft. air per explosion at T_1 as computed in 29.

v_e = cu. ft. neutrals per explosion. = volume of clearance = v_b.

To Find—

v_x = cu. ft. air per explosion at T_2.

v_z = cu. ft. neutrals per explosion at T_2.

$v_x : v' : v_z = ?$

Solution.

$$\frac{v_x}{v''} = \frac{T_2}{T_1} \qquad v_x = v''\frac{T_2}{T_1}$$

$$\frac{v_z}{v_e} = \frac{T_2}{T_e} \qquad v_z = v_e\frac{T_2}{T_e}$$

Taking v' as the basis, *i.e.* calling v' unity, then

$$v_x : v' : v_z = v''\frac{T_2}{T_1} : 1 : v_e\frac{T_2}{T_e}$$

Examples.

"A" Run No. 6 :

$T_1 = 86° + 459° = 545°.$

$T_2 = 86° + 459° = 545°$

$T_e = 180° + 459° = 639°.$

$v' = .0196$ cu. ft.

$v'' = .1450$ cu. ft.

$v_e = .0550$ cu. ft.

$$v_x = .145 \frac{545}{545} = .145 \text{ cu. ft.}$$

$$v_z = .055 \frac{545}{639} = .0469 \text{ cu. ft.}$$

$$v_x : v' : v_z = 7.4 : 1 : 2.4.$$

"B" Run No. 6 :

$T_1 = 87° + 459° = 546°.$

$T_2 = 87° + 459° = 546°.$

$T_e = 188° + 459° = 647°.$

$v' = .0163$ cu. ft.

$v'' = .0769$ cu. ft.

$v_e = .0370$ cu. ft.

$$v_x = .0769 \frac{545}{545} = .0769 \text{ cu. ft.}$$

$$v_z = .037 \frac{545}{647} = .0312 \text{ cu. ft.}$$

$$v_x : v' : v_z = 4.7 : 1 : 1.9.$$

41. *Stroke to Expansion.*

70. This ratio shows the regularity of the opening of the exhaust valve, *i.e.*, the position in the stroke of the point of release, and is equal to the ratio of the full stroke to the horizontal projection of the expansion curve taken to the release point. In this diagram shown in Fig. 17, it is the ratio of FA to FJ.

FIG. 17

For "A" run No. 6, this ratio was 1.14, and for "B" run No. 6, 1.10.

The point of release does not seem always to correspond to the point of opening of the exhaust valve. The piston is moving rapidly and the drop in pressure is not always shown at once, especially if the exhaust valve motion is not rigid, or if the exhaust valve is of insufficient area.

42. *Volumes v_2 to v_1.*

71. In the diagram (Fig. 18), v_1 is proportional to the length OL, and represents the clearance volume of the cylinder. v_2 is

proportional to the length OR and represents the full cylinder volume—the clearance plus the stroke. The ratio $\frac{v_2}{v_1}$ is therefore equal to $\frac{OR}{OL}$. For "A" run No. 6 this ratio was 4.68 and for "B" run No. 6, it was 5.00.

43. *Maximum Pressure to Mean Effective Pressure.*

This ratio shows the relation between the pressure produced by explosion and the true working pressure, and gives an especially good idea of the possibilities of different engines in comparative tests, as well as determining the possible degree of constancy in this relation for any single engine. It is stated by some writers

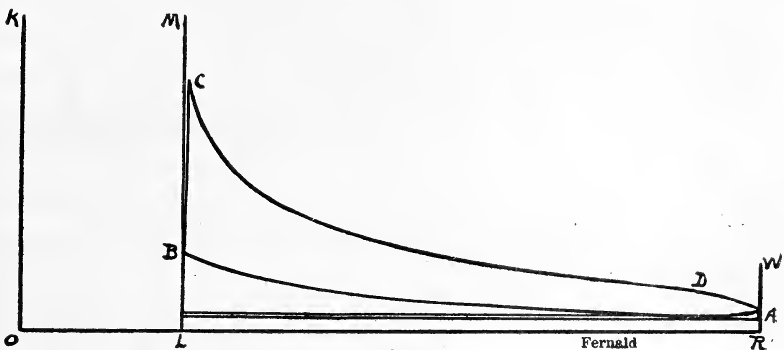

FIG. 18

that in practice it is found advantageous to proportion the amount of metal in the moving parts to this ratio. For "A" run No. 6, the maximum pressure was 298 lbs., and the mean effective pressure as given in item 36 was 66.3 lbs., thus giving a ratio of 4.50. For "B" run No. 6, the maximum pressure was 238 lbs., and the mean effective pressure 56.5 lbs., giving the ratio of 4.21.

44. *Maximum Pressure to Compression Pressure.*

72. Not only will variations in this relation tend to show changes in the quality of the mixture, due either to fluctuations in the quality of gas used, or to differences in the proportions of air to gas; but it also gives insight into possible changes in the form

of the combustion chamber for securing the best possible flame propagation after explosion, thus insuring the most complete combustion of the mixture, or maximum measure of heat that becomes effective.

It has been claimed that conditions have been attained under which this ratio has reached a figure as high as ten, but in general it is found to be about three or three and one-half, and occasionally as high as five. Cases have been reported of six and seven, but investigation revealed the fact that the pressures were measured from the atmospheric line. "A" run No. (= 3.28. "B" run No. 6 = 2.62.

45. *Value of R.*

73. In order to compute the temperatures corresponding to different points in the indicated diagram, it is necessary that the temperature of some one point be known, and it is for this reason that the temperature of the exhaust gases proves of so much interest.

In engines of the ordinary size it is hardly possible to perceive by means of the indicator diagram when a high-pressure spring is used any considerable reduction in the pressure of the mixture just after entering the cylinder. In most cases the reduction in pressure is slight and has very little effect upon the temperatures, and it is sufficiently accurate to regard the pressure of the mixture at full cylinder volume as that of the atmosphere, or 14.7 pounds per square inch. If the temperature of a perfect gas varies during expansion, the product of the pressure and volume is in proportion to the absolute temperature.

"R" is the constant which enters into the mathematical statement of the above law, viz.; $PV = RT$.

Data Given.

P_0 = atmospheric pressure per sq. ft.
= 2,117 lbs. per sq. ft.
v_2 = total vol. of cylinder in cu. ft.
T_m = absolute temperature of mixture filling cylinder before compression begins.
= item 29 + 459°.

To Find—

R = a constant.

Solution.

By the above law :

$$P_0 v_2 = RT_m$$

$$R = \frac{P_0 v_2}{T_m}$$

Examples.

"A" Run No. 6:

$P_0 = 2{,}117$ lbs. per sq. ft.

$v_2 = .260$ cu. ft.

$T_m = 106° + 459° = 565°$.

$$R = \frac{2{,}117 \times .260}{565} = .973$$

"B" Run No. 6:

$P_0 = 2{,}117$ lbs. per sq. ft.

$v_2 = .1844$ cu. ft.

$T_m = 113° + 459° = 572°$.

$$R = \frac{2{,}117 \times .1844}{572} = .681$$

46. *Temperature, Degrees Fahr., at Compression.*

74. Having determined "R," as in 45, the temperatures corresponding to any point in the diagram are readily determined by the general formula used in obtaining "R" after solving for T.

Data Given.

In general.

P = pressure in lbs. per sq. ft.

V = corresponding vol. in cu. ft.

R = constant determined in 45.

To Find—

T = absolute temperature corresponding to the point of the diagram selected.

Solution.

$$T = \frac{P\,V}{R}$$

$$T - 459° = F°$$

For the temperatures of compression.

Examples.

"A" Run No. 6:

P = pressure at compression per sq. ft. $= 91 \times 144$ = item 35×144.

$V = v_b$ = clearance vol. $= .055$ cu. ft.

$R = .973$ from 45.

$$T_b = \frac{91 \times 144 \times .055}{.973} = 741°$$

$$741° - 459° = 282° \text{ F.}$$

"B" Run No. 6.

P = pressure at compression per sq. ft. $= 91 \times 144$ = item 35×144.

$V = v_b$ = clearance vol. $= .037$.

$R = .681$ from 45.

$$T_b = \frac{91 \times 144 \times .037}{.681} = 711°$$

$$711° - 459° = 252° \text{ F.}$$

47. *Maximum Temperature, Degrees Fahr.*

75. Since the maximum temperature does not necessarily correspond to the maximum pressure, but depends upon the maximum value of the product of pressure and volume, it would in general be determined by the formula used in 46.

In cases where the ignition line rises vertically from the point of maximum compression and where the expansion curve drops at once from the maximum pressure, it is apparent that the maximum temperature corresponds to the maximum pressure. Under these conditions, the volumes being equal, the absolute temperatures will be proportional to the pressures.

FIG. 19

The diagrams from the two engines considered present the different conditions very clearly in the runs selected.

Consider first the diagram for run No. 6, test " A."

Data Given.

p_c = maximum pressure, lbs. per sq. in. = item 36.

p_b = compression pressure, lbs. per sq. in. = item 35.2

T_b = absolute temp. at compression = item 46 + 459°.

To Find—

T_c = absolute maximum temperature.

Solution.

$$T_c = T_b \frac{p_c}{p_b}$$

Example.

p_c = 298 lbs. per sq. in.

p_b = 91 lbs. per sq. in.

T_b = 282° + 459° = 741°.

$$T_c = 741 \frac{298}{91} = 2{,}429°$$

$$2{,}429° - 459° = 1{,}970° \text{ F.}$$

If the same computation be made by the formula of 46, the clearance volume v_b, is necessary.

v_b = .055 cu. ft.

By the method of 46

$$T_c = \frac{298 \times 144 \times .055}{.973} = 2{,}429°$$

$$2{,}429° - 459° = 1{,}970° \text{ F.}$$

Consider the diagram, Fig. 20, for run No. 6 of test " B."

76. In this case it is not apparent without some calculation at which point the maximum temperature will occur. Since $T = \frac{PV}{R}$ it is only necessary to determine the point for which the product

of the pressure and volume is a maximum. The most direct and simplest method seems to be by direct trial by measurement, as follows:

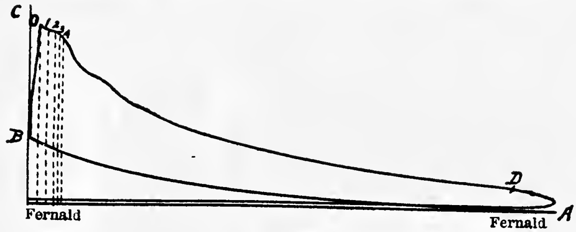

FIG. 20.

For the points 0, 1, 2, 3, 4, the volumes in cubic feet are respectively

.0395	.0417	.0438	.0453	.0460

and the pressures in pounds per square inches are

238	233	228	225	221

The products of the corresponding pressures and volumes are

9.42	9.74	10.00	10.20	10.15

showing the point marked 3 to have the highest temperature. Using the value of R deduced in 45, the temperature for the point 3 is found to be

$$T_3 = \frac{225 \times 144 \times .0453}{.681} = 2{,}159^\circ = 1{,}700^\circ \text{ F.}$$

ENERGIES.

48. *Brake Work in Foot Pounds.*

77. The usual method of measuring the output of an engine is by means of some form of friction brake. At times special dynamometers are used, but usually some form of the simple prony brake.

For moderate powers the form of brake shown in Fig. 21 has been found very satisfactory.

It consists of two or more cotton or hemp ropes one-half or five-eighths inch in diameter encircling the wheel and held in place by five or six blocks of wood fitting loosely over the rim of the wheel, and to which the ropes are fastened. The bottom tie-bar of the standards is fitted with a knife edge which rests on ordinary platform scales. The upper ends of the ropes are attached to a movable block, which can be adjusted by means of the hand-wheels to produce any desired friction upon the wheel.

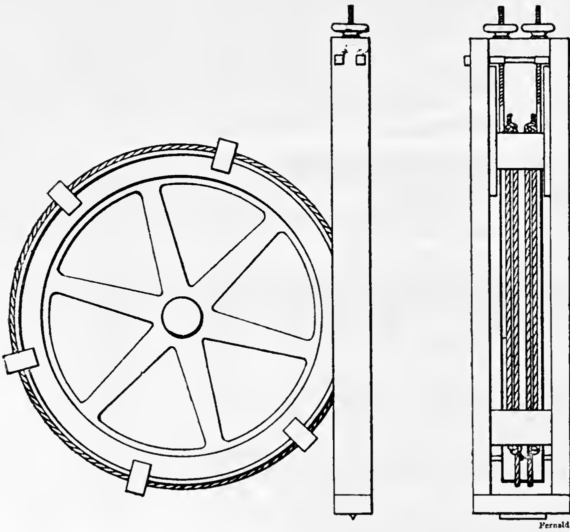

Fig. 21.

78. The downward pull of the ropes produced by this friction is transmitted through the frame of the brake to the scales.

If the wheel face be fairly wide and the diameter of the wheel large, the surface exposed to the air will allow sufficient radiation to keep the temperature of the wheel low enough to avoid the necessity of the use of water. This form of brake has been found very steady, requiring little regulation.

While the writer was at the Case School of Applied Science, such a brake was used upon the ten-foot fly-wheel of a Corliss engine from which seventy-five horse-power was readily taken, and no water was found necessary for a continuous run of three hours. Six ropes one-half inch in diameter were used, the wheel rim being fifteen inches in width. The same ropes were used for several seasons, and showed no signs of wear or burning. In many cases other forms of brake would undoubtedly prove more convenient or desirable.

The determination of the brake work in foot pounds is as follows:

Data Given.	*Solution.*
P = net pressure on scales, in lbs.	Ft lbs. per min. $= P \cdot 2\pi l \cdot$ (R. P. M.)
l = effective lever arm, in ft.	In case of the special rope brake, l =
N = No. revs. per time interval.	radius of wheel, or $2l = D$, then
To Find—	ft. lbs. per int. $= P\pi DN$.
Foot pounds per time interval.	
Examples.	
"A" Run No. 6:	Ft. lbs. per int. =
$P = 70$ lbs.	$70 \times 3.14 \times 3.75 \times 2{,}679$
$2l = D = 3.75$ ft.	= 2,210,000 ft. lbs.
$N = 2{,}679$ revs. for 10 mins.	
$\pi = 3.14$.	
"B" Run No. 6:	Ft. lbs. per int. =
$P = 30$ lbs.	$30 \times 3.14 \times 3.25 \times 3{,}012$
$2l = D = 3.25$ ft.	= 921,000 ft. lbs.
$N = 3{,}012$ revs. for 10 mins.	
$\pi = 3.14$.	

49. *Brake Work, Foot Pounds per Hour.*

79. This is readily deduced from the last deduction:

For "A" Run No. 6 = 2,210,000 × 6 = 13,260,000 ft. lbs.
For "B" Run No. 6 = 921,000 × 6 = 5,526,000 ft. lbs.

50. *Brake Horse-power, B. H. P.*

80. Horse-power being

$$\frac{\text{ft. lbs. per min.}}{33{,}000}, \text{ the B. H. P.} = \frac{\text{ft. lbs. per int. from 48}}{33{,}000 \times \text{int.}}.$$

For "A" Run No. 6: $\text{B. H. P.} = \frac{2{,}210{,}000}{33{,}000 \times 10} = 6.7.$

For "B" Run No. 6: $\text{B. H. P.} = \frac{921{,}000}{33{,}000 \times 10} = 2.79.$

51. *British Thermal Units Equivalent to B. H. P.*

81. The heat unit being taken as the consumption standard, it is necessary, in order to determine thermal efficiencies, to express

the horse-powers, or ratio of doing work, in terms of this unit. Since one thermal unit is equivalent to 778 foot pounds, to convert horse-power to thermal units proceed as follows:

Data Given.	*Solution.*
1 H. P. = 33,000. ft. lbs. per min. 1 B. T. U. = 778 ft. lbs.	B. T. U. per min. for 1 H. P. $= \frac{33{,}000}{778} = 42.4.$
B. H. P. = brake horse-power of 50. int. = time interval of item 2.	
To Find— B. T. U. per int. equivalent to B. H. P.	B. T. U. per int. $= 42.4 \times$ B. H. P $\times$ int.
Examples. "A" Run No. 6: B. H. P. = 6.70 int. = 10 mins.	B. T. U. per int. $= 42.4 \times 6.70 \times 10 = 2{,}840.$
"B" Run No. 6: B. H. P. = 2.79. int. = 10 mins.	B. T. U. per int. $= 42.4 \times 2.79 \times 10 = 1{,}180.$

52. *Indicated Horse-power, I. H. P.*

82. The indicated horse-power is determined by means of the mean effective pressure obtained from the indicator diagram. When the load is variable these diagrams should be taken at frequent intervals, and it is often advisable to allow the pencil to trace three or four diagrams on the same card and use the average mean effective pressure obtained from these. Under uniform conditions these successive tracings will show little variation. When the test is continued for many hours, so that the time intervals are long, it is well to take cards at stated periods during the regular interval. These periods should seldom exceed fifteen minutes. When the regular time interval for all readings is ten or fifteen minutes one card for each interval is sufficient, provided slight variations only are found in the different cards as taken (see remarks under Pressures for Indicator Cards, following paragraph 31 of this paper, and for the calibration of indicator springs, see section xiv. of paper No. 0943, the "Report of the Committee of this Society for Standardizing a System of Testing Steam Engines."

TO DETERMINE THE INDICATED HORSE-POWER.

P = M. E. P. from diagram = item 36.
L = length of strokes in ft.
A = area piston in sq. ins.
N = No. explosions per min.

$$\text{I. H. P.} = \frac{PLAN}{33{,}000}$$

Examples.

"A" Run No. 6:

P = 66.3 lbs.
L = 1.04 ft.
A = 28.3 sq. ins.
N = 133.9.

$$\text{I.H.P.} = \frac{66.3 \times 1.04 \times 28\ 3 \times 133.9}{33{,}000} = 7.93$$

"B" Run No. 6:

P = 56.5 lbs.
L = .75 ft.
A = 28.3 sq. ins.
N = 93.4.

$$\text{I.H.P.} = \frac{56.5 \times .75 \times 28.3 \times 93.4}{33{,}000} = 3.40$$

53. *British Thermal Units Equivalent to I. H. P.*

83. The determination is the same as that of 51, with the exception that indicated horse-power is substitued for brake horse-power.

"A" Run No. 6:	I. H. P. = 7.93	B. T. U. = 42.4 × 7.93 × 10 = 3,360
"B" Run No. 6:	I. H. P. = 3.40	B. T. U. = 42.4 × 3.4 × 10 = 1,440

54. *Gas Horse-power.* 55. *B. T. U. Equivalent to Gas H. P.* = H_1.

84. In order to determine the theoretically possible power, it is necessary to know the heat equivalent of the fuel used. This heat of combustion may be determined by chemical analysis or by means of a calorimeter. The calorimeter generally recommended seems to be the Mahler, for solid fuels and oils, and Junker for gas. (See "Report of Committee on Standardizing a System of Testing Steam Engines," paper No. 0943, and "Efficiency Test of a One Hundred and Twenty-five Horse-power Gas Engine," by C. H. Robertson, paper No. 907, American Society of Mechanical Engineers.)

Since the mixtures considered are explosive mixtures and are used under such conditions, a calorimeter designed and calibrated for such conditions should be used for accurate determinations. Such a calorimeter is now in operation at Columbia. When it is inconvenient to secure either a chemical or calorimeter test, it is usually possible to learn through the company supplying the fuel its approximate heat equivalent.

The number of heat units is equal to the heat equivalent of one pound of coal or oil, or one cubic foot of gas, multiplied by the quantity of fuel in corresponding units.

Data Given.	*Solution.*
H_f = heat of combustion of fuel determined by analysis or calorimeter.	
F = lbs. of coal or oil, or cu. ft. of standard gas per interval.	
To Find—	
H_1 = heat equivalent to G. H. P.	
$Gas\ H.\ P. = \dfrac{H_1 \times 778}{33{,}000 \times \text{int.}}$	$H_1 = H_f \times F$
Examples.	
"A" Run No. 6 :	
H_f = 650 B. T. U. per cu. ft. gas at standard temp. 60° F.	$H_1 = 650 \times 25.2 = 16{,}400$ B. T. U.
F = 25.2 cu. ft. gas for 10 mins. = item 25 ÷ 6.	$Gas\ H.\ P. = \dfrac{16.400 \times 778}{33{,}000 \times 10} = 38.6.$
int. = 10 mins. = item 2.	
"B" Run No. 6 :	
H_f = 650 B. T. U. per cu. ft. gas at standard temp. 60° F.	$H_1 = 650 \times 14.6 = 9{,}500$ B. T. U.
F = 14.6 cu. ft. gas for 10 mins. = item 25 ÷ 10.	$Gas\ H.\ P. = \dfrac{9{,}500 \times 778}{33{,}000 \times 10} = 22.4.$
int. = 10 mins. = item 2.	

56. *Heat Supplied, B. T. U., from Indicator Card* = H_1'.

85. There is always a wide discrepancy between the supply of heat shown by the indicator—by the pressure rise line BC of the diagrams—and that shown by the calorimeter or chemical test of the fuel.

The great difference in the values of these two quantities is readily seen by a glance at items 66 and 67, which show the efficiencies based on these different values.

The computation for the heat supplied, as shown by the indicator diagram, involves not only the temperature of compression and the corresponding temperature from the expansion curve, but strictly the specific heat of the gases before and after explosion, and during the process of explosion—a value which is indeterminate—and the total weight of these gases. This tends to seriously complicate the problem, but the specific heat will be assumed the same before and after explosion and can in both cases be regarded, without serious error, as the value given for air.

Data Given.

T_m = absolute temperature of mixture in cylinder before compression = item 29 + 459°.

T_{ag} = absolute temp. of entering air and gas as computed in paragraph 29.

v_{ag} = vol. of entering air and gas per explosion at T_{ag} as computed in Part I. of 29.

v_b = clearance vol. of cylinder.

w_5 = wgt. per cu. ft. of mixture at T_m as computed in 30.

C_v = specific heat at constant vol = .1691.

T_c = absolute temp. of point C of diagram.

T_b = absolute temp. of compression = item 46 + 459°.

$Exps$ = total explosions per time interval.

To Find—

v_t = vol. entering air and gas at T_m.

v_s = total vol. per explosion of mixture before compressing.

w_m = total wgt. of mixture in cylinder.

H_1' = heat supplied in B. T. U. per time interval.

Solution.

$$v_t = v_{ag}\frac{T_m}{T_{ag}}$$

$$v_s = v_t + v_b$$

$$w_m = v_s \times w_5$$

$$H_1' = C_v(T_c - T_b)\, w_m \times Exps.$$

Examples.

"A" Run No. 6:

$T_m = 106° + 459° = 565°$.

$T_{ag} = 545°$.

$v_{ag} = .1646$ cu. ft.

$v_b = .055$ cu. ft.

$w_5 = .0701$ lb.

$C_v = .1691$.

$T' = 1{,}970° + 459° = 2{,}429°$.

$T_b = 282° + 459° = 741°$.

$Exps. = 1{,}339$ for 10 mins.

$$v_t = .1646\frac{565}{545} = .1710 \text{ cu. ft.}$$

$$v_s = .171 + .055 = .226 \text{ cu. ft.}$$

$$w_m = .226 \times .0701 = .158 \text{ lb.}$$

$$H_1' = .1691\,(2{,}429 - 741)\,.158 \times 1{,}339 = 6{,}040 \text{ B. T. U.}$$

86. The determination of the temperature (T_c), corresponding to the point C of the diagrams is not always as readily made as in the case just cited, as may be observed by inspecting Figs. 19 and 20. In Fig. 19, which corresponds to run No. 6, of test "A," the line BC of the diagram coincides with the line drawn through B parallel to the line of zero volumes, thus making T_c and the maximum temperature, deduced in 47, identical. When these fortunate conditions do not exist, as in Fig. 20, or run No.

6 of test "B," it is necessary to continue the expansion curve to the point C, as in the computation for H'_1 it is specified that the temperatures must be taken for constant volume points.

When any question exists regarding the true value of n for the equation of the expansion curve near the point C, the curve may be continued from points below by the eye and the value thus obtained for C checked with that found by the equation. In the present instance the pressure at C was found to be about 282 pounds.

Having determined the pressure at C as just indicated, the temperature may be readily computed by the formula $T = \frac{PV}{R}$ or more easily by direct proportion, since the temperature at B is already known. By the latter method

$$T_c = T_b \frac{p_c}{p_b} \text{ or } T_c = 711 \frac{282}{91} = 2{,}203^\circ \text{ or } 1{,}744^\circ \text{ F.}$$

This result is quickly checked by the other method, or

$$T_c = \frac{288 \times 144 \times .037}{.681} = 2{,}203^\circ.$$

Continuing with the original problem,

"B" Run No. 6:

$T_m = 113^\circ + 459^\circ = 572^\circ$

$T_{ag} = 546^\circ$

$v_{ag} = .0932$ cu. ft.

$v_b = .037$ cu. ft.

$w_5 = .0692$ lb.

$C_v = .1691$

$T_c = 2{,}203^\circ$

$T_b = 711^\circ$

$Exps. = 934$ per 10 min.

$v_t = .0932 \frac{572}{546} = .0976$ cu. ft.

$v_s = .0976 + .037 = .1346$ cu. ft.

$w_m = .1346 \times .0692 = .0093$ lb.

$H_1' = .1691\ (2{,}203 - 711)\ .0093 \times 934$

$= 2{,}190$ B. T. U.

87. In the solution of Heat Extracted, B. T. U. from indicator cards, $= H_2'$, item 58, the same values of C_v, w_m, and $Exps.$ enter the calculations. It is, therefore, well in working the above to take the product C_v, w_m, $Exps.$ for each run and tabulate the results under a heading "K" for further use.

Thus for "A" No. 6, $K = .1691 \times .158 \times .1339 = 3.58$,
and for "B" No. 6, $K = .1691 \times .0093 \times 934 = 1.47$.

57. *Heat Extracted, B. T. U., by Observation* $= H_2$.

88. This value is determined by an analysis of the exhaust gases from which the heat equivalent of a cubic foot of these gases is determined.

Data Given.

T_e = absolute temp. exhaust at atmospheric pressure.
= item 14 of log + 459°.
T_m = absolute temp. of entering mixture = item 29 + 459°.
C_p = specific heat at constant pressure.
T_{ag} = absolute temp. of combined air and gas as found in 29.
v_{ag} = combined vol. of air and gas per explosion at T_{ag}.
T_k = absolute temp. of exhaust as analyzed.
h = B. T. U. per cu. ft. exhaust gases at T_k as found by analysis.
Exps. = explosions per time interval.
w_6 = weight per cu. ft. by analysis.

To Find—

v_k = vol. in cu. ft. at T_k per explosion.
w_k = total weight per explosion.
H_2 = total heat exhausted, B. T. U., per time interval.

Solution.

$$v_k = v_{ag}\frac{T_k}{T_{ag}}$$

$$w_k = w_6 \times v_k$$

$$H_2 = C_p\,(T_e - T_m)\,w_k \times Exps. + h \times v_k \times Exps.$$

58. *Heat Extracted, B. T. U., from Indicator Card* $= H_2'$.

89. This is the heat thrown off in the exhaust as derived from the pressures shown by the indicator card.

Data Given.

T_h = absolute temp. of exhaust = item 37 + 459°.
T_m = absolute temp. of mixture at atmospheric pressure = item 29 + 459°.
$K = C_v \times w_m \times Exps.$ as found in 57.

To Find—

H_2' = heat rejected, B.T.U., per time interval.

Solution.

$$H_2' = K(T_h - T_m)$$

Examples.

"A" Run No. 6:

$T_h = 1{,}387° + 459° = 1{,}846°.$
$T_m = 106° + 459° = 565°.$
$K = 3.58$ from 57.

$H_2' = 3.58\ (1{,}846–565) = 4{,}580$ B. T. U.

"B" Run No. 6:

$T_h = 1{,}120° + 459° = 1{,}579°.$
$T_m = 113° + 459° = 572°.$
$K = 1.47$ from 57.

$H_2' = 1.47\ (1{,}579–572) = 1{,}480$ B. T. U.

Both in this solution, and in that of 57, since the difference of temperatures is involved, the common factor may be omitted if desired, and the temperatures taken directly from the report blank in Fahrenheit degrees. Thus for "A" run No. 6.

$t_h = 1{,}120°.$
$t_m = 113°.$ $\quad H_2' \quad 1.47\ (1{,}120–113) = 1{,}480$ B.T.U.
$K = 1.47.$

59. *Indicated Horse-power Minus Brake Horse-power.*

90. The difference between these powers is frequently called the friction horse-power. It is not alone the power required of the engine to drive its own mechanism, but includes the error due to inability to take indicator cards every explosion. This is especially important with two cycle engines with throttling governors.

The calculation consists simply in subtracting the value of item 50 from item 52.

For "A" run No. 6 this is $7.93 - 6.70 = 1.23$,
and for "B" run No. 6, $3.40 - 2.79 = 0.61$.

60. *Throttling of the Entering Mixture, Cu. Ft. per Explosion.*

91. It is found, as might be expected, that the final volume of the mixture in the cylinder before compression is not equal to the full cylinder volume if the mixture is taken at atmospheric pressure. This is caused in part by the brief time allowed for entering, and in part by valve friction. In large engines this throttling may become serious, and all valves should be positively moved and not moved by suction.

Owing to many conditions that affect this result, the calculation of the amount is approximate only, but gives an idea of results of this throttling action.

In computing 56 the value v_s obtained is equal to the total volume for explosion of mixture at atmospheric pressure before compression, expressed in cubic feet. The total volume of the cylinder minus this value gives the effect of throttling.

"A" Run No. 6.

$$\begin{aligned} \text{Total cylinder vol.} &= .260 \text{ cu. ft.} \\ \text{From 57, } v_s &= .226 \text{ cu. ft.} \\ \text{Throttling}\ & \ .034 \text{ cu. ft.} \end{aligned}$$

"B" Run No. 6.

$$\begin{aligned} \text{Total cylinder vol.} &= .1844 \text{ cu. ft.} \\ \text{From 57, } v_s &= .1346 \text{ cu. ft.} \\ \text{Throttling} &= .0498 \text{ cu. ft.} \end{aligned}$$

61. *Percentage of Throttling.*

This is the ratio of the cubic feet throttled to the total cylinder volume in cubic feet.

$$\text{"A" Run No. 6:}\quad \frac{.034}{.260} = 13.1 \text{ per cent.}$$

$$\text{"B" Run No. 6:}\quad \frac{.0498}{.1844} = 27.0 \text{ per cent.}$$

62. *Work that would be Added if Expansion were Complete.*

92. A point of considerable interest was presented in attempting to make the computations involved in this column. The mathematical work required is given below:

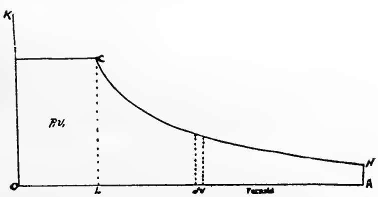

FIG. 22.

To find the area under the expansion curve proceed as follows:

$$pv^n = p_1 v_1{}^n$$
$$p = p_1 v_1{}^n v^{-n}$$

The general expression for the area is

$$A = \int_{v_1}^{v_2} p\, dv,$$

Substituting the value of p found above

$$A = p_1 v_1^n \int_{v_1}^{v_2} v^{-n} dv$$

$$A = p_1 v_1^n \frac{v_2^{1-n} - v_1^{1-n}}{1-n}$$

$$A = \frac{p_1 v_1^n}{n-1} (v_1^{1-n} - v_2^{1-n}),$$

which reduces to

$$A = \frac{p_1 v_1 - p_2 v_2}{n-1}$$

A glance at this formula reveals the fact that the value of A, which corresponds to the work done, depends upon the value of n, and in carrying out the solution any variation in n makes such serious variation in the value of A that no dependence can be placed upon the values obtained unless the values of n can be guaranteed correct. The time at command does not allow further investigation at present, but the trial solutions revealed a possible method of making more accurate determinations of the value of n than can possibly be made by the method employed in paragraph 28. The difficulty arising from slight variations in n prevented any attempt to supply the results called for in this column of the report.

EFFICIENCIES.

93. The efficiency may be expressed in many different forms, as indicated, and in general it is very essential, when referring to the efficiency of a gas engine, to designate clearly to which efficiency reference is made, in order to avoid serious misunderstanding.

The mathematical deductions are so fully indicated in the headings of the various columns that further explanation about the details seems unnecessary, and only general remarks will be made under each paragraph.

63. *Mechanical* $\frac{item\ 50}{item\ 52}$.

94. This efficiency is the ratio of the power which can be taken from the engine to the power shown by the indicator card. It therefore depends much upon the smooth, easy running of the engine itself.

$$\text{"A" Run No. 6} = \frac{6.70}{7.93} = .846$$

$$\text{"B" Run No. 6} = \frac{2.79}{3.40} = .822$$

64 and 66. Thermal for B. H. P. and I. H. P.

95. In these cases the efficiency is based upon the total heat put into the engine as shown by the calorimeter or chemical analysis of the fuel. Item 64 shows what percentage of the total heat in the fuel is converted into useful work and item 66 shows the percentage of the total heat converted for both useful and useless work.

"A" Run No. 6:

$$\text{For B. H. P.} = \frac{2,840}{16,400} = .173$$

$$\text{For I. H. P.} = \frac{3,360}{16,400} = .205$$

"B" Run No. 6:

$$\text{For B. H. P.} = \frac{1,180}{9,500} = .124$$

$$\text{For I. H. P.} = \frac{1,440}{9,500} = .152$$

65 *and* 67.

If the basis for computing the efficiency be that of the heat actually shown to be supplied by the indicator, then the thermal efficiency is much higher. This basis has been used in computing items 65 and 67.

"A" Run No. 6:

$$\text{For B. H. P.} = \frac{2,840}{6,040} = .471$$

$$\text{For I. H. P.} = \frac{3,360}{6,040} = .557$$

"B" Run No. 6:

$$\text{For B. H. P.} = \frac{1,180}{2,190} = .539$$

$$\text{For I. H. P.} = \frac{1,440}{2,190} = .657$$

68, 69 *and* 70.

96. In these columns are given other methods of estimating the efficiency, based as before on both the heat supplied and extracted

as determined directly from the gases, and as determined from the indicator diagram.

For item 69, $\frac{H_1' - H_2'}{H_1'}$, the efficiencies were

$$\text{"A" Run No. 6: } \frac{6{,}040 - 4{,}580}{6{,}040} = .242$$

$$\text{"B" Run No. 6: } \frac{2{,}190 - 1{,}480}{2{,}190} = .324$$

The necessity of clearly designating the efficiency selected is made very apparent by a comparison of the values given in the preceding paragraph.

STANDARD GAS PER HOUR.

97. In order for comparison to be made it is very essential that some standard be adopted for estimating the quantity of gas used. The basis of reckoning item 25, which has previously been recommended by a few writers, seems to be the proper one, and therefore standard gas is interpreted to mean gas under atmospheric pressure and at a temperature of 60 degrees Fahr.

71. *Fuel for Indicated Horse-power per Hour.*

98. This is readily obtained by dividing the values in column 25 by the corresponding values in column 52.

$$\text{"A" Run No. 6: } \frac{151}{7.93} = 19 \text{ cu. ft. per hr.}$$

$$\text{"B" Run No. 6: } \frac{87.5}{3.4} = 25.7 \text{ cu. ft. per hr.}$$

72. *Fuel and Igniter for Indicated Horse-power.*

In case of flame or hot tube ignition the gas used should be passed through a separate meter and for the data in this column the quantity of standard gas used for fuel combined with that used for the igniter will give the total standard gas per hour. This quantity, divided by the indicated horse-power, will give the desired value.

73 *and* 74. *Fuel, and Fuel and Igniter, per Brake Horse-power.*

99. The method of procedure is the same as in 71 and 72, save that the brake horse-power is now used instead of the indicated horse-power.

FUEL PER B. H. P.

"A" Run No. 6: $\frac{151}{6.70} = 22.5$ cu. ft.

"B" Run No. 6: $\frac{87.5}{2.79} = 31.4$ cu. ft.

75. *Cost per Indicated Horse-power per Hour, Cents.*

100. The value $1.00 per thousand cubic feet of standard gas is taken as a basis for determining the comparative cost of different engines. A similar deduction would be made for other fuels, based on an average cost of that fuel. In case the gas used for the igniter is reckoned in this cost, then when electric ignition is used the cost of maintaining the battery should also be considered, in order to make a proper comparison. For relative cost it is as well to disregard the igniter and base the computation alone on the quantity of fuel alone.

Since the fuel used in run No. 6 was 19 cubic feet for "A" and 25.7 cubic feet for "B," it is readily seen that the cost per indicated horse-power per hour was 1.9 cents and 2.57 cents respectively.

TOTALS AND AVERAGES.

101. In order to get a general series of values for the engine in question it is often well to average the results obtained under the same conditions. For this purpose space has been reserved on the report blank for the necessary totals and averages.

HEAT BALANCE.

102. By means of the heat balance a general idea is formed of the distribution and uses of the heat. As is apparent, the first result is the average of the results in item 53; the second of those in item 12; the third of those in item 58. To the last result is charged all the heat otherwise unaccounted for. The average of the British thermal units obtained from the gas as determined from item 55 is used as the basis in determining the percentages.

PART II.

A Method of Determining the Temperatures of Exhaust Gases.

A METHOD

OF

DETERMINING THE TEMPERATURE

OF

EXHAUST GASES.

---

1. In preparing detailed specifications and instructions for conducting tests of gas engines (paper No. 0933 presented before this Society at this meeting) the attention of the writer was directed to the necessity of an accurate and inexpensive method of determining the temperatures of exhaust gases. This problem had apparently received little attention, at least not sufficient to develop any simple means of making accurate determinations. In cases in which the results seemed to be reasonable, the apparatus used was far too expensive or too delicate for ordinary conditions, and efforts were at once centred upon the desired solution.

2. In the books at hand on engine tests, no mention is made of any method, and even in the very recent report of the committee appointed by this Society "To Codify and Standardize the Methods of Making Engine Tests," the committee says (paragraph XX.): "The computation of temperatures corresponding to various points in the indicator diagram is, at best, approximate. It is possible only when the temperature of one point is known or assumed, or when the amount of air entering the cylinder along with the charge of gas or oil, *and the temperature of the exhaust gases are determined*." In the fine-print detailed instructions under the same paragraph the report states, "T′ may be taken as the temperature of the exhaust gases leaving the engine, provided the engine is not of the 'scavenging' type."

Again, in referring to the value represented by T in formula

B of the same paragraph is found the expression, "If T be the observed temperature of the exhaust gases."

3. While references thus made indicate the necessity of obtaining these temperatures, yet nowhere in the report is there any suggestion of a method of making these determinations. Even if pyrometers and thermometers are regarded as sufficiently accurate, yet no measurements made at or near the muffler can give the true temperature of the exhaust gases unless proper corrections be made for the fluctuations in pressure. This at first appears to present little difficulty, but more careful thought shows the fallacy of the first impression.

Consider for a moment the action taking place at the opening

FIG. 1.

of the exhaust. Although the gases in the cylinder have undergone rapid expansion after explosion, yet the expansion is far from complete, and at the point of release the pressure is still relatively high, as shown by a glance at Fig. 1.

4. At the instant the exhaust opens there is an outrush of gases at this high pressure and correspondingly high temperature. Then follows, upon the forward stroke of the engine, a flow of a large mass of gases through the exhaust port, at a pressure little above that of the atmosphere and at a temperature necessarily less than that of the first discharge. In order to make the succession of events more apparent an indicator was attached directly to the exhaust pipe, and the cards obtained, as shown in Figs. 2 and 3, verified the statements just made in regard to the action taking place within the pipe. The diagrams represent the same conditions, the spring used for Fig. 2 being only one-half as heavy as that used in Fig. 3.

5. There is, as it were, a mixture of pressures in the exhaust pipe and muffler, and also a corresponding mixture of tempera-

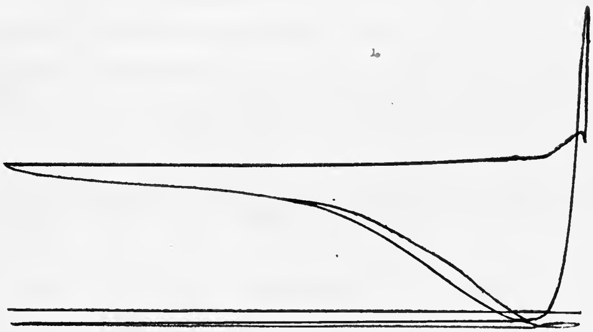

FIG. 2.

tures, which must rapidly adjust themselves to an equilibrium of pressures and of temperatures.

Temperatures determined at the muffler are, therefore, not the temperatures corresponding to the pressure at exhaust, but those corresponding to a much lower pressure, which is not determined. This accounts, no doubt, for the very low values of exhaust temperatures which have been reported even by recognized authorities upon gas-engine problems.

The problem, therefore, resolves itself into the determination of the temperatures of the exhaust at some *known* pressure.

6. As a matter of simplicity atmospheric pressure was the natu-

FIG. 3.

ral selection, and a method was at once sought for reducing the exhaust gases to atmospheric pressure without losing any of the heat to which they are entitled.

It was at once decided that a receiver, of a form to be determined, should be placed close to the exhaust outlet of the engine, and some means devised for admitting the exhaust gases and allowing their pressure to fall to that of the atmosphere. The desired temperature could then be ascertained.

7. What the form of the receiver should be was not at first apparent, and, as practically no information could be found bearing upon the subject, the problem was reduced to one of experiment.

The first steps in the development were necessarily very crude and the results were of value only as furnishing a basis upon which to judge future determinations, and as such were of great value.

The first experiments were conducted as follows: Two cylinders of sheet iron were made, one ten inches in diameter and the other fourteen. They were sixteen inches high. The ten-inch cylinder, after being generously perforated near the bottom, was placed inside the fourteen-inch cylinder, the latter having several deep notches cut out at the top. A cover of the same material, and fifteen inches in diameter was made, with a central hole about two and one-half inches in diameter, through which the exhaust pipe from the engine could be passed. A *T* was placed as close to the exhaust outlet of the engine as possible, and from it one line of pipe was run directly to the exhaust muffler, as usual, and the other line brought out horizontally and at right angles to the first, and then directed downward to the receiver just described, the end of the pipe projecting about two inches through the cover, into the inside cylinder. Fig. 4 gives a rough idea of the arrangement.

8. Valves were inserted in both pipes, so that the passage of the gases to the receiver or to the muffler or to both at once could be regulated at will.

In the receiver the gases were passed downward through the inside cylinder, out through the perforations and upward between the two cylinders, finally passing out through the notches in the top of the outside cylinder. The object of this arrangement was to prevent any direct draught or chimney action, which would cause an inrush of cold air at the bottom as soon as the inlet valve was closed. Two holes, large enough to receive the thermometers, were punched in the cover, one midway between the centre and the inner wall and the other about one-half inch from the inner wall. There was no expectation that temperatures even approaching correctness could be obtained by this arrangement, but such a preliminary step was essential in order to have values with which future results could be compared.

9. The exhaust was directed to the receiver, and after being allowed to flow until the cold air in the receiver was entirely expelled, was shut off and the thermometers quickly inserted. The gases were now held in the receiver at atmospheric pressure. The pressure before closing the inlet was much greater, but dropped almost instantly to that of the atmosphere when the

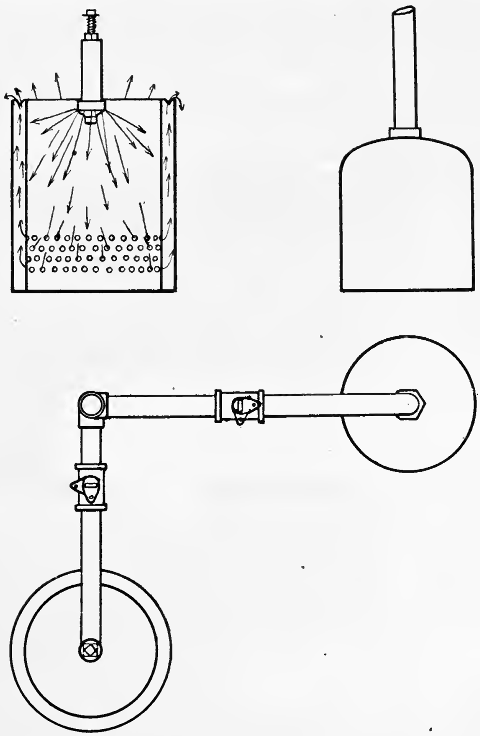

FIG. 4.

valve was closed. The difference in the readings of the two thermometers, one near the wall of the cylinder and the other about two inches from the wall, was not sufficiently marked to be of importance. The temperatures observed were:

370 degrees Fahr.	394 degrees Fahr.	398 degrees Fahr.
395 " "	372 " "	400 " "
	370 " "	

10. Various problems were now to be considered. Radiation and conduction from the cylinders to the outside air might be so

rapid as to vitiate the results. The thermometers might be affected by radiation from the inside walls. Radiation from the iron exhaust pipe, which projected into the cylinder, might make the readings too high. Longer runs might tend to cause the cylinders to store up heat.

Many such difficulties had to be considered, and, no data being found bearing directly upon the subject, the experiments were continued.

An asbestos lining was now placed in the outside cylinder, and the temperatures obtained were:

474 degrees Fahr.	511 degrees Fahr.	525 degrees Fahr.
540 " "	552 " "	517 " "

Evidently the gases were retained at a much higher temperature than before, but were these higher temperatures due to excessive storing up of heat, or simply to the prevention of excessive radiation? A similar lining was also placed in the inside cylinder and the temperatures immediately shot up to over 600 degrees Fahr., which were unquestionably far too high. In all of the above experiments radiation through the cover was prevented by an asbestos lining.

11. After considering cylinders of various materials, a clay fire flue 10 inches in diameter was secured and used in place of the inner iron cylinder, the outside iron cylinder with its asbestos lining being retained. The temperatures now recorded were:

360 degrees Fahr.	484 degrees Fahr.	585 degrees Fahr.
388 " "	498 " "	560 " "
405 " "	512 " "	564 " "
412 " "	541 " "	554 " "
465 " "	552 " "	569 " "
469 " "	554 " "	576 " "
	567 " "	

These figures seemed to indicate a gradual storing of heat from the first to about 560 degrees Fahr., when the readings became more constant, but not sufficiently so to warrant the conclusion that the apparatus was nearly correct.

The conclusions at once reached were that the volume for the gases was too small and the thermometers too near the walls. The absorbing of heat by the receiver must also be prevented. In studying this problem it was seen to be undesirable to allow the gases to enter the receiver at such high pressures and temperatures, and that it would be of great advantage to admit the gases at a pressure and temperature little above those desired at the time of reading the thermometers. This would do away to a large extent with the tendency of the receiver walls to store up heat, as at no time would they be excessively heated.

12. The proper throttling was secured by the device shown in Fig. 5.

It consists of a 2-inch T with a plug in the top, through which passes a $\frac{1}{2}$-inch bolt, and a nipple and cap in the bottom; the bolt

FIG. 5.

is supported by a helical spring, and in turn supports at the bottom a flat iron disk which rests against the 2-inch cap—the bottom of the cap being freely perforated.

By screwing down the nut at the top, against the spring, any desired resistance to the passage of the gases can be obtained, and at the same time any tendency to wire-drawing is obviated, as would not be the case if the throttling were done by means of the valve in the pipe alone.

Before experimenting to any extent with this device for reducing the pressure, a new clay fire flue, 18 inches in internal diameter, with cover of the same material, was obtained. In order as far as possible to prevent any radiation to the thermometer from the iron pipe delivering the gases to the receiver, the lower end of the device was allowed to pass barely through the cover, extending not over an inch below the inner face.

13. The bottom of the receiver was notched in several places to allow free passage of the gases from within. The gases were thus received at the top, passing downward and out at the bottom.

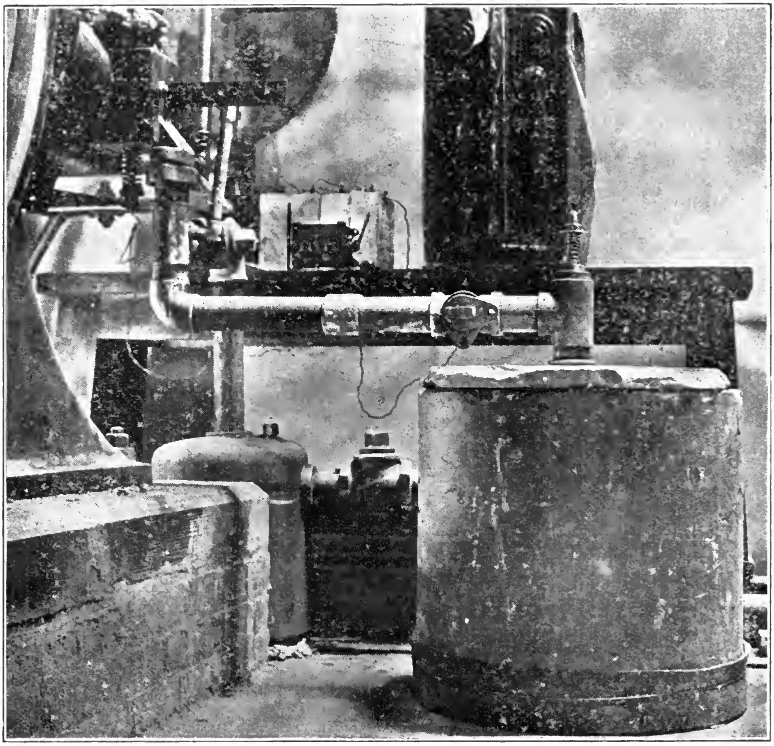

FIG. 6.

A rubber band 3 inches wide was bound about the bottom of the receiver, thus acting as a flap valve, the pressure of the gases entering the pot forcing the band outward, but the falling of the band preventing any inrush of cold air when the admission of the exhaust to the receiver was stopped. In Fig. 6 is shown the completed receiver as attached to the engine.

Many series of experiments have been carried on by means of this device, with most satisfactory results. With the feed so adjusted that the exhaust entered the receiver at a pressure but little above that of the atmosphere, the storing of heat in the walls was practically prevented.

14. In case the exhaust was delivered to the receiver for an hour or more without ceasing there was a slight rise in temperature, but as in practice the exhaust is cut off about every 10 minutes no difficulty is experienced from this cause.

The pressure under which the exhaust enters the receiver does not have to be closely regulated, as a slight difference does not affect the resulting temperatures, when taken at atmospheric pressure.

Especially satisfactory results have been obtained when the pressure was so regulated that the temperature of the gases in

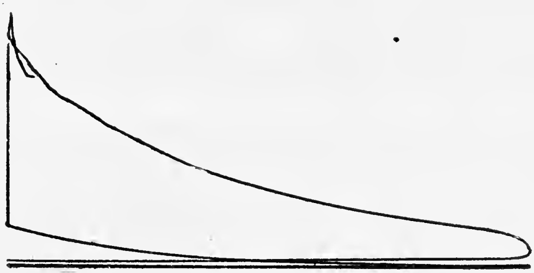

Fig. 7.

the receiver, while under pressure, was between 50 degrees and 100 degrees above the temperature at atmospheric pressure.

15. One or two preliminary tests will quickly determine what this temperature should be. The temperatures recorded from the new receiver did not range from 400 degrees Fahr. to over 600 degrees Fahr. but were:

200 degrees Fahr.	202 degrees Fahr.	203 degrees Fahr.
195 " "	201 " "	200 " "
195 " "	203 " "	200 " "
	203 " "	

The radiation from the receiver was not large. The walls were 1 inch thick and the hand could at all times be held on the outside.

Any change in the conditions tending to change the temperatures of the exhaust is quickly noticed by a corresponding change within the receiver. For example, if the point of ignition be changed, a corresponding change in temperatures is immediately observed. With the point of ignition as shown by the diagram, Fig. 7, the following temperatures were recorded:

199 degrees Fahr.	197 degrees Fahr.	197 degrees Fahr.
198 " "	196 " "	198 " "
198 " "	196 " "	198 " "
	199 " "	

FIG. 8.

16. With the ignition retarded as shown by Fig. 8, the temperatures were:

210 degrees Fahr.	210 degrees Fahr.	207 degrees Fahr.
216 " "	212 " "	212 " "
214 " "	211 " "	209 " "
	210 " "	

While making this particular series of experiments on points of ignition, the engine became stalled when set for a certain point of ignition and refused to run. The igniter was removed and

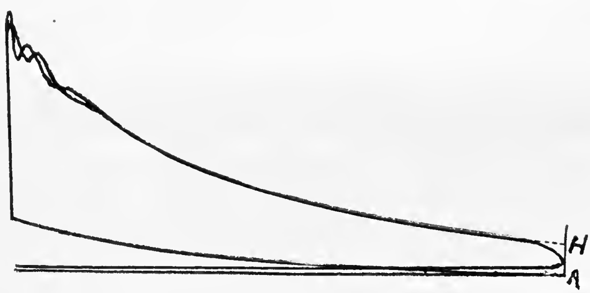

FIG. 9.

found to be badly burned. It was adjusted for sharper contact and a new series of readings taken.

An immediate drop of about 50 degrees in the temperature resulted, showing at once the quick response of the receiver to changed conditions. The spark was now sharp and short, while previously the condition of the sparker was such that it was "holding fire."

The new series on the variation of the point of ignition resulted

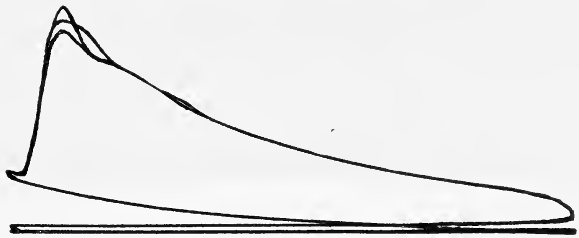

Fig. 10.

as follows, the temperatures in each column corresponding to the point of ignition shown in the diagram having the same number as the column of temperatures:

For Fig. 9.	For Fig. 10.	For Fig. 11.
151 degrees Fahr.	166 degrees Fahr.	172 degrees Fahr.
149 " "	168 " "	176 " "
151 " "	168 " "	178 " "
152 " "	167 " "	175 " "
152 " "	167 " "	178 " "

17. Owing to the size of the receiver and to the necessity of having other connections made to the engine, the exhaust pipe

Fig. 11.

leading to the receiver was longer than desired, having a drop of about 1 foot and a horizontal length of 27 inches.

The question of the necessity of covering the pipe, to prevent excessive radiation in conducting the hot gases to the receiver, was quickly settled by the following temperatures obtained:

Covered.	Not Covered.
155 degrees Fahr.	160 degrees Fahr.
155 " "	158 " "
154 " "	158 " "
156 " "	155 " "
158 " "	156 " "
158 " "	157 " "
158 " "	156 " "
160 " "	157 " "

Having become satisfied that the apparatus as outlined was working with a reasonable degree of accuracy, the next step was to devise a receiver which can be erected quickly and cheaply, as it

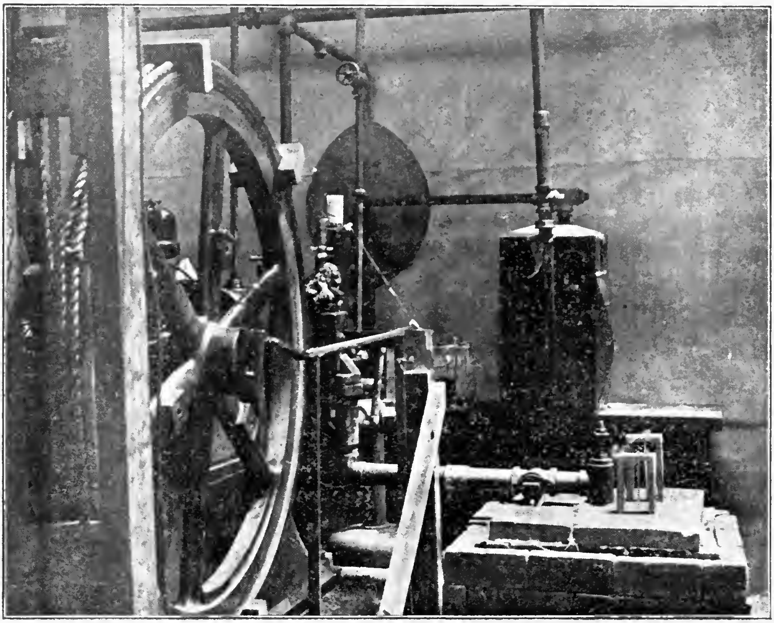

Fig. 12.

is not always convenient to procure a fire flue of the right size. Accordingly, a receiver was built of ordinary brick laid together loosely, as shown in the cut, Fig. 12.

18. This receiver was made of two layers of brick, the inside layer being laid on the face and breaking joints, the outside layer being laid on edge, thus breaking joints both vertically and horizontally with the inner layer. While the cracks furnished sufficient passages to prevent any overcharging of the receiver, yet the break-

ing of the joints prevented direct draughts, or inrush of cold air. As first constructed, the volume of the new receiver was made equal to that of the 18-inch fire flue previously used, and the fire flue cover was fitted to the new receiver. The results were highly satisfactory, the temperatures being the same as before.

The interior was then partially filled with brick until the volume was reduced from 16 x 16 x 23 to 16 x 16 x 13, and again tested, with the same results.

In all of these tests with the two larger receivers the thermometer bulbs were kept about 4 to 7 inches away from the walls and from the entering pipe. In the last receiver, when partially filled with brick, the thermometer bulbs were placed about the same distance from the cover and bottom.

19. It seems unnecessary, therefore, to have excessive volume, but there must be sufficient space so that the thermometer bulbs shall not be too near the retaining walls. Experiment indicates that the figures suggested for these distances are about right.

During the test the valve leading to the muffler in the main exhaust is kept wide open, unless the exhaust pipe is larger than is needed for the quantity of exhaust, and the valve in the pipe leading to the receiver is opened as desired. It is necessary to expel fully the cold air in the receiver before any readings can be taken.

With the larger receiver and an 8 horse-power engine this required about 20 minutes. The time can be much reduced when desired, by careful manipulation of the valves after one becomes familiar with the apparatus.

20. It is of great value to keep suspended within the receiver at all times a thermometer of sufficient range not to be broken by accidental increase of temperature. In the initial warming of the receiver this thermometer will readily show when the temperature has become constant. It also serves as a very efficient guide in adjusting the pressure as controlled by the inlet. For most of the readings taken the pressure was so regulated that this permanent thermometer recorded, while the gases were still under pressure, temperatures from 50 degrees to 70 degrees Fahr. higher than the final temperatures at atmospheric pressure.

The clay cover not being obtainable in all cases, gave way to one made of 2-inch plank, chinked with cotton waste, and this has proved entirely satisfactory.

21. Previous methods of measuring the temperatures of the

exhaust have in most cases furnished results which were surprisingly low, and not until Professor Robertson's paper on "An Efficiency Test of a One-hundred-and-twenty-five Horse-power Gas Engine" (Vol. XXI. *Transactions* A. S. M. E., p. 396), has the writer seen a series of temperatures for the exhaust, which seemed reasonably accurate. Professor Robertson secured by means of a copper-ball calorimeter values above 1,000 degrees Fahr., and in one instance records 1,209 degrees Fahr., occasioning this remark by Professor Thurston: "The possibilities of still further thermodynamic gain are indicated by the temperature of the exhaust gases, above 1,000 degrees Fahr., and far above that of the prime steam of our steam engines."

In Professor Robertson's second paper upon the same subject (*Transactions*, Vol. XXII., p. 612) he reports temperatures of the exhaust, as found by a La Chatelier pyrometer, ranging from 1,410 degrees Fahr. to 1,805 degrees Fahr., and states: "These temperatures appear to be rather high—so high in fact that the author has examined other data at hand to see if any confirmation of the above figures could be found."

It is the opinion of the writer that the last figures quoted by Professor Robertson are not far from correct for the engine tested, but the pyrometer used is far too expensive and requires too much special apparatus, as well as special calibration, to make its use possible in most tests.

22. It takes but a moment's consideration of the problem to show that very little if any thermodynamic gain is possible by any attempted reduction of the temperatures of exhaust, in the average gas engine of good modern design, unless accompanied by a reduction of the terminal pressure as shown by the expansion curve.

23. With terminal pressures ranging near 50 pounds it is quickly shown that the exhaust temperatures must of necessity be much higher than generally recorded. Suppose, for example, that the expansion line of the card shown in Fig 9 be continued to full cylinder volume, as shown by the point H.

The pressure at H is found to be 50 pounds absolute. Suppose the temperature of the mixture in the cylinder, composed of air and gas, taken in during the suction stroke and mixed with the neutral gases filling the clearance space, to be only as high as that of the room, say 70 degrees Fahr., or 529 degrees absolute at the point A, Fig. 9.

Since the volume at H is the same as that at A, the absolute temperatures will be proportioned to the absolute pressures, or if

T_a = absolute temperature at A = 529 degrees.

P_a = absolute pressure at A = 14.7 pounds.

P_h = absolute pressure at H = 50 pounds,

then T_h, the absolute temperature corresponding to the pressure at H will be derived from $T_h = T_a \frac{P_h}{P_a}$; $T_h = 529\frac{50}{14.7} = 1{,}800$ degrees absolute or 1,341 degrees Fahr. with the temperature of the mixture taken at 70 degrees Fahr. only. Actually the temperature of the mixture is much higher than this, and by means of the new apparatus described in this paper this temperature is readily deduced. The temperature secured at atmospheric pressure by means of the exhaust receiver is the temperature of the neutral gases that fill the clearance space of the cylinder. The method for determining the temperature of the mixture after having obtained the temperature of the exhaust gases is described in detail in paragraph 29 of paper No. 0933, to be presented at this meeting.

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