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JOSEPH A. HOLMES, DIRECTOR

COMPARATIVE FUEL VALUES OF GASOLINE AND DENATURED ALCOHOL IN INTERNAL-COMBUSTION ENGINES

BY

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By R. M. Strong and Lauson Stone.

INTRODUCTION.

Under the terms of the act establishing the Bureau of Mines, this bureau was authorized to carry on the work of testing and analyzing fuels which had been conducted by the technologic branch of the United States Geological Survey. That work included in its scope an investigation of the availability and uses of liquid as well as solid fuels, for the original outline of the fuel-testing investigations contemplated, as soon as the funds would be available, a study of the liquid-fuel resources of the country and the making of related researches to determine how these resources could be utilized with greatest efficiency.

Owing to the fact that many difficulties were being encountered in the adaptation of the heavier fuel oils for convenient use in internal-combustion engines, it was deemed best to begin the investigation of liquid fuels with tests of gasoline, a fuel in more or less general use.

When this investigation began, the extensive introduction, especially by foreign powers, of liquid fuels for small naval craft had awakened much interest. However, the quality of gasoline was reported to vary materially in different countries and the quantity available was said to be rapidly decreasing, with the probability of a prohibitive increase in price. At the same time the claim was made that denatured alcohol, of fairly uniform quality, could be procured in all parts of the world, that unlimited quantities could be readily produced at a low cost, and that this fuel could be used much more efficiently than gasoline in internal-combustion engines. Such statements naturally led to a widespread belief that the time was near at hand when denatured alcohol would entirely displace gasoline as engine fuel. Therefore, the first investigations of the liquid mineral fuels logically embraced a careful series of comparative tests of gasoline and denatured alcohol in engines. A series of over 2,000 such tests was conducted at the Government fuel-testing plants at St. Louis, Mo., and Norfolk, Va., details of which are given in the following pages. The report is published by the Bureau of Mines because of the transfer of the fuel-testing investigations to this bureau.

PERSONNEL.

The investigation was conducted under the supervision of R. H. Fernald, then engineer in charge of the producer-gas section of the technologic branch of the United States Geological Survey. The tests were made by R. M. Strong, assistant engineer in charge of the oil-fuel section, assisted by J. C. Barnaby, W. A. Wicks, H. A. Talbott, W. B. Loye, E. W. Gallenkamp, P. G. Weidner, G. H. Hopkins, and other junior engineers who were temporarily transferred for these particular investigations. The computations were made under the direction of Lauson Stone assisted by S. P. Howell. Mr. Stone also collaborated with Mr. Strong in the preparation of this report. The necessary chemical work was performed, under the general supervision of Prof. N. W. Lord, by the regular laboratory force of the technologic branch.

ACKNOWLEDGMENT.

The authors wish to express their thanks to Dr. Charles E. Lucke and Prof. S. M. Woodward for suggestions found in a report entitled "Tests of Internal-Combustion Engines on Alcohol Fuel," published by the Office of Experiment Stations, Department of Agriculture. The report was frequently consulted during the procedure of the investigations covered by this bulletin.

PRELIMINARY REPORT ON DENATURED ALCOHOL AND GASOLINE.

In 1909 Strong presented in a small bulletin[a] the general characteristics of the engines used and a general description of the methods of procedure and the properties of the fuels. The commercial deductions set forth in that preliminary report are given herewith.

HEATING VALUE.

The low heating value of completely denatured alcohol averages 10,500 British thermal units per pound, or 71,900 British thermal units per gallon.

[a] Strong, R. M., Commercial deductions from comparisons of gasoline and alcohol tests on internal-combustion engines. U. S. Geol. Survey Bull. 392, 1911, 38 pp. Reprinted as Bulletin 32, Bureau of Mines.

The low heating value of gasoline having a specific gravity of 0.71 to 0.73 averages 19,200 British thermal units per pound, or 115,800 British thermal units per gallon.

The low heating value of 1 pound of alcohol is approximately six-tenths of the low heating value of 1 pound of gasoline.

One pound of gasoline requires approximately twice the weight of air for complete combustion that is required by 1 pound of alcohol.

The heating value of 1 cubic foot of an explosive mixture of alcohol vapor and air having theoretically just sufficient air for complete combustion is approximately equal to that of 1 cubic foot of a similar explosive mixture of gasoline vapor and air—about 80 British thermal units per cubic foot.

EXPLOSIVE MIXTURES.

Explosive mixtures of alcohol vapor and air in an engine cylinder can be compressed to much higher pressures, without preigniting, than explosive mixtures of gasoline vapor and air. The maximum compression that can be used in an engine without causing preignition depends on the quality of the explosive mixture, the design of the engine, and the speed at which it is operated.

For 10 to 15 horsepower 4-cycle stationary engines of the usual type a pressure of about 70 pounds per square inch above atmospheric pressure was found to be the maximum that could be used for gasoline mixtures, and about 180 pounds the maximum that could be used for alcohol mixtures, without causing preignition.

The maximum compression that could be used without causing preignition was in each case found to be the most advantageous with regard to fuel economy.

MOST ECONOMICAL DEGREE OF COMPRESSION.

When the degree of compression in each engine is that best suited to the economical use of the fuel designated, some types of gasoline engines are better adapted to the service for which they are designed than similar alcohol engines, and vice versa. This is also true (the relative quantity of fuel consumed being disregarded) when the degree of compression is that ordinarily used for gasoline mixtures, as when denatured alcohol is used in gasoline engines. But, in general, the alcohol engine is or can be so designed and constructed as to be equal to the gasoline engine in adaptability to service.

A gasoline engine having a compression pressure of 70 pounds, but otherwise as well suited to the economical use of denatured alcohol as gasoline, will, when using alcohol, have an available horsepower about 10 per cent greater than when using gasoline.

When the fuels for which the respective engines are designed are used to the same advantage, the maximum available horsepower of an alcohol engine having a compression pressure of 180 pounds is about 30 per cent greater than that of a gasoline engine having a compression pressure of 70 pounds, but of the same cylinder diameter, stroke, and speed.

When denatured alcohol is used in 10 to 15 horsepower 4-cycle stationary engines having a compression pressure of approximately 180 pounds and the engines are operated at their maximum loads, the pressures during explosion or combustion reach 600 to 700 pounds. Stationary gasoline engines in which the compression pressure can be raised to 180 pounds are not usually built heavy enough to withstand such explosion pressures for any considerable length of time.

QUANTITY REQUIRED FOR ENGINE FUEL.

A gasoline engine working at the degree of compression ordinarily used for gasoline mixtures will, in general, require 50 per cent more denatured alcohol than gasoline per brake horsepower-hour.

Gasoline and alcohol engines of similar construction, working at degrees of compression best suited to the fuel supplied, in general require equal volumes of gasoline and denatured alcohol respectively per brake horsepower-hour.

Gasoline engines of the usual 4-cycle stationary type ordinarily consume about 1 pint of gasoline per brake horsepower-hour when operated at about rated load and with a reasonably favorable adjustment of the mixture quality and the time of ignition.

When carrying light loads or maximum loads gasoline and alcohol engines governed for constant speed require a greater quantity of fuel per brake horsepower-hour than when carrying their rated loads, if rated at about 75 to 80 per cent of their maximum loads; but unless the mixture quality and time of ignition are adjusted to suit each change of load the rate of consumption per brake horsepower-hour is in general least at the maximum load and increases with a decrease in load.

When any of the usual methods of governing are used to control the speed of gasoline or alcohol engines the rate of fuel consumption per brake horsepower-hour is ordinarily about twice as great at one-third load as at maximum load. At the same time an excessive rate of consumption of gasoline or denatured alcohol at any given load, if only due to the incorrect adjustment of the mixture quality and the time of ignition, may reach approximately twice the minimum rate required before it is noticeable from outward indications.

THERMAL EFFICIENCY OF ENGINES.

The thermal efficiency of alcohol and gasoline engines generally increases with the pressure to which the charge is compressed when ignited.

The maximum thermal efficiency of 10 to 15 horsepower 4-cycle stationary engines of the usual type when operated with a minimum amount of throttling was found to increase with the compression pressure according to the formula $E=1-\left(\frac{14.7}{P}\right)^{.17}$ for gasoline and the formula $E=1-\left(\frac{14.7}{P}\right)^{.19}$ for alcohol, in which E represents the thermal efficiency based on the indicated horsepower and low heating value of the fuel, and P represents the indicated absolute pressure of the charge at the end of the compression stroke in pounds per square inch.

A high thermal efficiency and a rate of consumption of less than 1 pint per brake horsepower-hour, both for gasoline and denatured alcohol, can often be obtained when the degree of compression, the load, the quality of the explosive mixture, and the time of ignition are carefully adjusted. A fair representation of the best economy values obtained, taken from the results of tests on 10 to 15 horsepower Nash and Otto stationary engines, and the corresponding thermal efficiencies are given in the following table:

Results from tests made on 10 to 15 horsepower Nash and Otto stationary engines.

Fuel.	Compression pressure.[a]	Fuel consumed per brake horsepower-hour.		Thermal efficiency.[b]
	Pounds.	*Pounds.*	*Gallons.*	*Per cent.*
Gasoline	70	0.60	0.100	26
	90	.58	.097	28
Alcohol	70	.96	.140	28
	180	.71	.104	39
	200	.68	.099	40

[a] Per square inch above atmosphere.
[b] Based on the indicated horsepower and the low heating value of the fuel.

PROPER COMPRESSION IN ENGINES.

When, by means of a double carburetor, gasoline and alcohol are used simultaneously, in varying proportions from practically all gasoline to practically all alcohol, the most advantageous degree of compression varies from that found to be the best for gasoline mixtures to that found to be the best for alcohol mixtures.

Tests that were made with such an adjustment of compression indicate that the total amount of fuel (gallons of gasoline + gallons of denatured alcohol) required for any given load is practically constant for the entire range of proportions, from all gasoline to all denatured alcohol.

MIXTURES WITH WATER.

When water is sprayed into an explosive mixture of gasoline vapor and air, as the mixture enters the cylinder, it may often be supplied in quantities up to as much water as gasoline, by weight, without affecting the performance of the engine, except as noted below:

(*a*) The capacity or maximum available horsepower of an engine decreases with an increase in the percentage, by weight, of water present in the explosive mixture of gasoline vapor and air.

(*b*) When used in an engine having a constant degree of compression, water in an explosive mixture of gasoline vapors and air in weights up to those of the gasoline does not change the quantity of gasoline required to carry a given load.

(*c*) The pressure to which an explosive mixture of gasoline vapor, water, and air can be compressed in an engine cylinder without preigniting increases with an increase in the percentage of water in the mixture, and can be raised to about 140 pounds when the weights of water and gasoline are equal.

(*d*) That the quantity of gasoline required is not affected by an increase in the compression pressure, when preignition is prevented only by the introduction of water as above stated, is indicated by the results of tests made on an engine having a compression pressure of 130 pounds. These tests are so few, however, that the results are not conclusive.

Denatured alcohol diluted with water in any proportion up to 50 per cent can be used in gasoline and alcohol engines, if the engines are properly equipped and adjusted.

In an engine having a constant degree of compression the quantity of pure alcohol required for any given load increases and the maximum available horsepower of the engine decreases with a diminution in the percentage of pure alcohol in the diluted alcohol supplied. The rate of increase in the quantity of pure alcohol required is such, however, that the use of 80 per cent instead of 90 per cent alcohol, (denatured alcohol is about 90 per cent pure) has little effect on the performance of the engine.

When an engine is supplied with diluted alcohol, the compression pressure that can be used without causing preignition increases with an increase in the percentage of water, but no tests were made to determine the effect of increased compression pressure on the economy with which diluted denatured alcohol could be used.

STORAGE.

The relative hazard involved in the storage and handling of gasoline and denatured alcohol is of particular importance in considering their use as fuels for marine, factory, and other engines, where a general fire would be likely to result from the accidental burning of the fuel stored or carried for immediate supply, or where the forming of explosive or inflammable mixtures of the fuel vapors and air in the immediate vicinity would be hazardous.

In accordance with the general consensus of opinion of those experienced in handling gasoline, kerosene, and alcohol, statistics indicate that the hazard involved in the use of denatured alcohol is much less than that in the use of gasoline and possibly less than that in the use of kerosene.

COMPARATIVE CLEANLINESS.

In regard to general cleanliness, such as absence of smoke and disagreeable odors, alcohol has many advantages over gasoline or kerosene as a fuel. The exhaust from an alcohol engine is never clouded with a black or grayish smoke as is the exhaust of a gasoline or kerosene engine when the combustion of the fuel is incomplete, and it is seldom, if ever, clouded with a bluish smoke when a cylinder oil of too low a fire test is used or an excessive quantity thereof is supplied, as so often happens with a gasoline engine. The odors of denatured alcohol and the exhaust gases from an alcohol engine are also not likely to be as obnoxious as the odor of gasoline and its products of combustion.

NUMBER OF ALCOHOL ENGINES USED.

Very few alcohol engines are being used in the United States, and little has been done toward making them as adaptable as gasoline engines to the requirements of the various classes of service. However, engines designed especially for using denatured alcohol for stationary, marine, and traction service and for automobiles, motor trucks, and motor railway cars have been tried with considerable success.

RELATIVE PRICE.

The price of denatured alcohol is greater than the price of gasoline, and the quantity of denatured alcohol consumed by an alcohol engine as ordinarily constructed and operated is, in general, relatively greater than the quantity of gasoline consumed by a gasoline engine of the same type. Considerable attention is being given to the development of processes for the manufacture of alcohol from cheap raw materials that are generally available, and the expectation seems reasonable that the price of denatured alcohol may eventually become as low as or lower than the price of gasoline, especially if the price of gasoline advances. It also seems reasonable to expect a greater general improvement in alcohol engines than in gasoline engines.

When used as a fuel, denatured alcohol is not always so classed as to be exempt from restrictions placed on the use of gasoline by the rules of insurance and transportation companies or by city ordinances. The restrictions that are placed on the use of denatured alcohol are, however, never greater than those placed on the use of gasoline. In some places they are such that the use of an alcohol engine is permitted where the use of a gasoline engine is prohibited. For instance, alcohol motor trucks and automobiles are admitted to many New York City steamer piers that are not open to gasoline machines.

Where the restrictions placed on the use of denatured alcohol are less than those placed on the use of gasoline, or where safety and cleanliness are important requisites, the advantages to be gained by the use of alcohol engines in place of gasoline engines may be such as to overbalance a considerable increase in the fuel expense, especially if the cost of fuel is only a small part of the total expense involved, as is often the case. Denatured alcohol will, however, probably not be used for power purposes to any great extent until its price and the price of gasoline become equal. Also, the equality of gasoline and alcohol engines in respect to adaptability to service required and quantity of fuel consumed per brake horsepower-hour must become more generally realized.

A further general development in the design and construction of engines that use kerosene, cheaper distillates, or crude petroleum may be reasonably expected and may delay the extensive use of denatured alcohol, but as yet comparatively few data pertaining to this phase of the general investigation are available.

SCOPE OF PRESENT REPORT.

In this report will be found a detailed description of the equipment used, the methods of procedure, and the complete logs and deductions from the various tests.

For convenience in analyzing the large quantity of material presented, the report has been divided into the following sections: (1) Gasoline and denatured alcohol as fuels; (2) apparatus used in tests; (3) procedure of tests; (4) deductions from tests. The material embraced in the last-mentioned section has been taken up under the following headings: (*a*) Mechanical efficiency; (*b*) mixture quality; (*c*) time of ignition; (*d*) character of ignition spark; (*e*) jacket-water temperature; (*f*) speed of engine; (*g*) load variation; (*h*) mixtures with water; (*i*) compression.

FUTURE INVESTIGATIONS.

Taking as a basis the results of the series of tests described in this report, the Bureau of Mines, as part of its fuel-testing investigations, proposes to carry on further researches relating to liquid fuels in internal-combustion engines. In this connection, the field of application of the heavier fuel oils, both to the internal-combustion engine and to the oil-gas producer, will undoubtedly demand early attention.

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GASOLINE AND DENATURED ALCOHOL AS ENGINE FUELS.

Liquid fuel is so well adapted for use in the smaller sizes of internal-combustion engines that it has been so used extensively for many years. The fuels of this character which are most commonly used for power purposes are petroleum distillates, to which many different trade names are applied.

Denatured alcohol, of such a composition as to relieve the manufacturers from the necessity of paying the usual internal-revenue tax on ordinary alcohol of $2.20 per gallon, has become available. Consequently, the belief seems warranted that eventually the cost of the mineral oils will become so much higher than that of denatured alcohol as to make their cost prohibitive.

Although there are several of the lighter distillates of petroleum that may be used in internal-combustion engines, gasoline is probably in more general use than all others combined. This being the case, the determination was early reached that the investigation resulting in this bulletin should take on the nature of a comparison between the fuel values of gasoline and denatured alcohol for power purposes. A brief prefatory discussion of some of the more important characteristics of the two fuels is therefore considered desirable.

PHYSICAL AND CHEMICAL CHARACTERISTICS.

GENERAL STATEMENT.

Gasoline may be somewhat roughly defined as the fractional distillate from petroleum oils up to a temperature of 300° F. at atmospheric pressure. In this discussion gasoline may be considered as having approximately the formula of hexane (C_6H_{14}). However, not only does gasoline from different districts differ in specific gravity, and hence in approximate formula, etc., but gasoline from different refineries varies in many characteristics.

Denatured alcohol consists of grain alcohol (C_2H_6O) made poisonous and repulsive by the addition of other ingredients according to an approved formula,[a] for instance:

100 parts grain alcohol
10 parts wood alcohol
½ part benzine

[a] Regulations No. 30, U. S. Internal Revenue Service, Sec. 26.

There are many other formulas used. The above is only one of several for completely denatured alcohol. Alcohol denatured according to any approved formula is exempt from the usual internal-revenue tax. The possibility of the manufacture of alcohol from otherwise waste material makes its cheap production a probability in the near future.

CHEMICAL COMPOSITION.

The most fundamental chemical difference in the characteristics of the two fuels lies in the fact that oxygen is a constituent of denatured alcohol, whereas none is present in gasoline. The average composition of 12 samples of gasoline and 20 samples of denatured alcohol used in this investigation (Table 1) is as follows:

Average composition of samples of gasoline and denatured alcohol.

Element.	Gasoline.	Denatured alcohol.
Hydrogen	14.84	12.68
Carbon	84.65	47.08
Oxygen	....	40.24
	99.49	100.00

Because of the large quantity of oxygen in denatured alcohol less air is necessary for combustion of the alcohol than is required for combustion of an equal amount of gasoline. By reference to Table 1 it will be seen that the average value obtained for the quantity of air necessary for theoretically complete combustion of gasoline was 14.96 pounds, whereas for denatured alcohol it was 8.13 pounds.

TABLE 1.—*Chemical analyses*

DENATURED ALCOHOL.[q]

Test number.	Specific gravity at 60° F.[a]	Degrees Baumé.[b]	Flash point.[c]	Per cent by weight. Water. From Sp. Gr.[d]	Hydrogen.[e] From formula.[f]	Hydrogen. From analysis.[g]	Carbon. From formula.[f]	Carbon. From analysis.[g]	Oxygen.[h] From formula.[f]	Oxygen. From analysis.[i]
15161	0.8181		Approximately 55° F.	8.3	12.88	12.64	47.85	47.17	39.27	40.19
15170	.8183			8.4	12.87		47.80		39.33	
15167	.8187			8.5	12.87		47.75		39.38	
5971	.8188			8.5	12.87	12.74	47.75	47.47	39.38	39.79
5973	.8191			8.7	12.88	12.73	47.64	47.28	39.48	39.99
15160	.8192			8.7	12.88		47.64		39.48	
15163	.8192			8.7	12.88		47.64		39.48	
5969	.8192			8.7	12.88	12.75	47.64	47.30	39.48	39.95
15158	.8196			8.9	12.87	12.75	47.54	47.02	39.59	40.23
15166	.8197			8.9	12.87		47.54		39.59	
15169	.8197			8.9	12.87		47.54		39.59	
15157	.8198			8.9	12.87	12.73	47.54	46.92	39.59	40.35
15159	.8198			8.9	12.87		47.54		39.59	
15168	.8201			9.0	12.87		47.48		39.65	
15162	.8202			9.1	12.86		47.43		39.71	
15164	.8202			9.1	12.86		47.43		39.71	
5972	.8206			9.2	12.86	12.71	47.38	47.72	39.76	39.57
15165	.8213			9.5	12.86		47.22		39.92	
5970	.8225			9.9	12.85	12.60	47.01	45.97	40.14	41.43
5974	.8241			10.5	12.84	12.47	46.70	46.87	40.46	40.66
Average	.8199			9.0	12.87	12.68	47.50	47.08	39.63	40.24

GASOLINE.[r]

Test number.	Specific gravity at 60° F.[a]	Degrees Baumé.[b]	Flash point.[c]	Water. From Sp. Gr.[d]	Hydrogen. From formula.[f]	Hydrogen. From analysis.[g]	Carbon. From formula.[f]	Carbon. From analysis.[g]	Oxygen. From formula.[f]	Oxygen. From analysis.[i]
5967	0.7122	66.6	Below 0° F.	2.09		14.83		83.31		1.86
5965	.7165	65.3		.09		14.94		84.98		.08
5966	.7168	65.2		.10		14.88		85.03		.09
5968	.7175	65.1		−.19		14.85		85.32		−.17
15176	.7285	62.2								
15172	.7285	62.2		.57		14.81		84.64		.51
15177	.7289	62.1								
15174	.7289	62.1								
15173	.7292	62.0		.46		14.75		84.84		.41
15178	.7292	62.0								
15171	.7294	62.0		.84		14.80		84.46		.75
15175	.7301	61.8								
Average	.7246	63.2		.57		14.84		84.65		.51

[a] For method of determining, see page 23.

[b] Calculated from the specific gravity by means of the formula $°B. = \frac{140-130 \text{ sp. gr.}}{\text{sp. gr.}}$

[c] Determined in a closed-cup flash-testing apparatus.

[d] For the denatured alcohol the percentage of water is determined from the specific gravity (see p. 24); for gasoline and kerosene it is determined from the percentage of oxygen in the ultimate analysis (per cent H_2O=9/8 of the percentage of O).

[e] Total hydrogen, including H in the H_2O.

[f] Formula for denatured alcohol assumed to be x per cent $H_2O+(100-x$ per cent $C_2H_6O)$.

[g] From ultimate analysis, for method of making which, see page 27.

[h] Total oxygen, including O in the H_2O.

[i] From ultimate analysis (by difference) 100−(H+C).

[j] Determined in a Mahler bomb calorimeter (see p. 26).

[k] Water vapors condensed.

[l] Water vapors not condensed. Low value = high value − the total water × its latent heat = high value 9H (from ultimate analysis) × 536.5 calories.

[m] 1.8 × the calories.

of liquid fuels.

DENATURED ALCOHOL.[q]

Heating value.								Per cent difference between calorimeter and calculated B. t. u.		Pounds of air per pound of fuel for theoretical complete combustion.	
Calorimeter.[j]					Calculated.[n]						
Calories.		B. t. u. per pound.[m]			B. t. u. per pound.						
High.[k]	Low.[l]	High.	Low.	Difference.	High.	Low.	Difference.	High.	Low.	From formula.[o]	From analysis.[p]
6,460	5,850	11,628	10,530	1,098	11,703	10,584	1,119	+0.64	+0.51	8.32	8.13
6,490	5,886	11,695	10,595	1,100	11,690	10,571	1,119	− .04	− .23	8.31	....
6,486	5,874	11,675	10,573	1,102	11,677	10,558	1,119	+ .02	− .14	8.30	....
6,549	5,934	11,788	10,681	1,107	11,677	10,558	1,119	− .09	− .22	8.30	8.21
6,528	5,913	11,750	10,643	1,107	11,652	10,533	1,119	− .83	−1.03	8.28	8.18
6,451	5,838	11,612	10,508	1,104	11,652	10,533	1,119	+ .34	+ .24	8.28	....
6,440	5,827	11,592	10,489	1,103	11,652	10,533	1,119	+ .52	+ .42	8.28	....
6,466	5,850	11,639	10,530	1,109	11,652	10,533	1,119	+ .11	+ .03	8.28	8.19
6,440	5,825	11,592	10,485	1,107	11,626	10,507	1,119	+ .29	+ .21	8.26	8.14
6,447	5,832	11,605	10,498	1,107	11,626	10,507	1,119	+ .18	+ .09	8.26	....
6,477	5,862	11,659	10,552	1,107	11,626	10,507	1,119	− .28	− .43	8.26	....
6,424	5,810	11,563	10,458	1,105	11,626	10,507	1,119	+ .54	+ .47	8.26	8.12
6,415	5,800	11,547	10,440	1,107	11,626	10,507	1,119	+ .68	+ .64	8.26	....
6,446	5,831	11,603	10,496	1,107	11,613	10,494	1,119	+ .09	− .02	8.25	....
6,420	5,804	11,556	10,447	1,109	11,601	10,483	1,118	+ .39	+ .34	8.24	....
6,468	5,852	11,642	10,534	1,108	11,601	10,483	1,118	− .35	− .48	8.24	....
6,479	5,865	11,662	10,557	1,105	11,588	10,470	1,118	− .63	− .82	8.24	8.24
6,423	5,804	11,561	10,447	1,114	11,550	10,432	1,118	− .10	− .14	8.21	....
6,374	5,766	11,473	10,379	1,094	11,499	10,382	1,117	+ .23	+ .03	8.17	7.92
6,377	5,775	11,479	10,395	1,084	11,422	10,306	1,116	− .50	− .86	8.12	8.01
6,453	5,840	11,616	10,512	1,104	11,618	10,499	1,119	+ .02	− .12	8.26	8.13

GASOLINE.[r]

11,434	10,718	20,581	19,292	1,289	20,580	19,290	1,290	−0.00	−0.01	....	14.74
11,376	10,655	20,477	19,179	1,298	20,532	19,242	1,290	+ .27	+ .33	....	15.04
11,404	10,686	20,527	19,235	1,292	20,528	19,238	1,290	+ .00	+ .02	....	15.04
11,433	10,716	20,579	19,289	1,290	20,524	19,234	1,290	− .27	− .29	....	15.09
11,334	10,625	20,401	19,125	1,281	20,408	19,118	1,290	+ .03	− .04	....	....
11,339	10,625	20,410	19,125	1,285	20,408	19,118	1,290	− .01	− .04	....	14.96
11,337	10,625	20,407	19,125	1,282	20,404	19,114	1,290	− .02	− .06	....	....
11,323	10,611	20,381	19,100	1,281	20,404	19,114	1,290	+ .11	+ .07	....	....
11,328	10,616	20,390	19,109	1,281	20,400	19,110	1,290	+ .05	+ .01	....	14.96
11,346	10,634	20,423	19,141	1,282	20,400	19,110	1,290	− .11	− .16	....	....
11,327	10,613	20,389	19,103	1,283	20,400	19,110	1,290	+ .05	+ .04	....	14.91
11,337	10,625	20,407	19,125	1,282	20,392	19,102	1,290	− .07	− .12	....	....
11,360	10,646	20,448	19,162	1,286	20,448	19,158	1,290	.00	− .02	....	14.96

n For denatured alcohol, calculated from the calorific value of absolute alcohol, which, as determined in a Mahler bomb calorimeter by A. C. Fieldner, assistant chemist, United States Geological Survey, from a sample of 98.5 per cent alcohol (sp. gr. at 60° F.=0.7984) computed to the water-free basis, is 7,090 small calories, or 12,762 B. t. u. per pound. High B. t. u.=12,762−per cent water (from sp. gr.). Low B. t. u.= high B. t. u−9H×965.7, in which H is the total hydrogen from the formula, 9H is the total water, and 965.7 is the latent heat of vaporization in B. t. u. per pound.

For gasoline, calculated from the formulas, high B. t. u.=18,320+40 (°B.−10); low B. t. u.=17,030+40 (°B.−10).

For kerosene, calculated from the formulas, high B. t. u.=18,440+40 (°B.−10); low B. t. u.=17,190+40 (°B.−10).

o Calculated from the formula: Pounds of air per pound of fuel $=\frac{8H+2.67C-O}{0.23}$, in which H = the total hydrogen, C = the total carbon, and O = the total oxygen obtained from the formulae; 8 = the ratio of oxygen to hydrogen and 2.67 = the ratio of oxygen to carbon by weight required for complete combustion; 0.23 = the proportion of oxygen in air by weight.

p Calculated from the above formula (note *m*) by substituting the values obtained from the ultimate analysis for total H, C, and O.

q Completely denatured alcohol manufactured by the U. S. Industrial Alcohol Company and denatured according to the formula: 100 parts ethyl alcohol (above 90 per cent by weight) + 10 parts methyl alcohol + ½ part approved benzine.

r Designated as 65° B. gasoline. Samples 5965 to 5968, inclusive, purchased from the Standard Oil Company, Norfolk, Va. Samples 15171 to 15178, inclusive, purchased from the Waters-Pierce Oil Company, St. Louis, Mo.

TABLE 1.—*Chemical analyses*

Test number.	Specific gravity at 60° F.	Degrees Baume.	Flash point.	Per cent by weight. Water. From Sp. Gr.	Hydrogen. From formula.	Hydrogen. From analysis.	Carbon. From formula.	Carbon. From analysis.	Oxygen. From formula.	Oxygen. From analysis.
NONANE.[a]										
	0.7230	63.6	....	....	15.62	....	84.38	....	....	....
KEROSENE.[b]										
7562..............	0.7912	46.9	113	0.57	....	14.33	....	85.16	....	0.51
7561..............	.7925	46.7	116	.18	....	14.56	....	85.28	....	.16
7534..............	.7930	46.5	118	.26	....	14.37	....	85.40	....	.23
7563..............	.7999	45.0	107	.70	....	14.22	....	85.16	....	.62
Average......	.7942	46.3	114	.43	....	14.37	....	85.25	....	.38
TETRADECANE.[c]										
	0.7960	45.9	....	....	15.15	....	84.85	....	....	....

[a] Nonane normal (C_9H_{20}). The hydrocarbon of the methane series for which the specific gravity and formulas are the nearest to those of the average gasoline given above.

[b] Samples 7561, 7562, and 7564 designated as 150° test "water-white" kerosene. Sample 7563 designated as 120° test "standard" kerosene, purchased from the Atlantic Refining Company, Pittsburgh, Pa.

of liquid fuels—Continued.

Heating value. Calorimeter. Calories. High.	Heating value. Calorimeter. Calories. Low.	Heating value. Calorimeter. B. t. u. per pound. High.	Heating value. Calorimeter. B. t. u. per pound. Low.	Heating value. Calorimeter. B. t. u. per pound. Difference.	Heating value. Calculated. B. t. u. per pound. High.	Heating value. Calculated. B. t. u. per pound. Low.	Heating value. Calculated. B. t. u. per pound. Difference.	Per cent difference between calorimeter and calculated B. t. u. High	Per cent difference between calorimeter and calculated B. t. u. Low.	Pounds of air per pound of fuel for theoretical complete combustion. From formula.	Pounds of air per pound of fuel for theoretical complete combustion. From analysis.
NONANE.[a]											
....	....	....	....	....	....	....	....	....	....	15.22	....
KEROSENE.[b]											
11,068	10,376	19,922	18,676	1,246	19,926	18,676	1,250	+0.02	0.00	....	14.87
11,052	10,349	19,894	18,629	1,265	19,918	18,668	1,250	+ .12	+ .21	....	14.96
11,045	10,351	19,881	18,632	1,249	19,910	18,660	1,250	+ .14	+ .15	....	14.91
11,040	10,353	19,872	18,636	1,236	19,850	18,600	1,250	− .11	− .19	....	14.83
11,051	10,357	19,892	18,643	1,249	19,901	18,651	1,250	+ .05	+ .04	....	14.89
TETRADECANE.[c]											
....	....	....	....	....	....	....	....	....	....	15.12	....

[c] $C_{14}H_{30}$, a hydrocarbon of the methane series for which the specific gravity and formula are the nearest to those of the average kerosene given above.

PHYSICAL CHARACTERISTICS.

Gasoline is a much lighter liquid than denatured alcohol. The average specific gravity of the samples used in compiling Table 1 was 0.7246 and 0.8199 respectively.

The flash point of gasoline is somewhat below 0° F., while that of completely denatured alcohol is approximately 55° F.

The average high heating values given for gasoline and denatured alcohol in the table are 20,448 and 11,616 British thermal units per pound respectively. If these heating values be corrected for the latent heat carried away by the steam resulting from the combustion of the hydrogen in the fuels, 19,162 and 10,512 British thermal units remain as their low heating values. These low heating values were used in determining all engine efficiency unless otherwise stated.

EXPLOSIVE MIXTURES OF VAPORS AND AIR.

Due to the fact that a less weight of air is necessary for the complete combustion of denatured alcohol than for the complete combustion of gasoline, the heating values for explosive mixtures of the two fuels are approximately equal, at 80 British thermal units per cubic foot of vapor.

An explosive mixture of alcohol vapors and air is capable of withstanding a much higher compression pressure without auto-ignition than is a similar explosive mixture of gasoline vapors. The importance of this fact is far-reaching for two reasons: (1) "Pressure directly accelerates the reactions of combustion."[a] (2) The theoretical efficiency which may be expected in the operation of any heat engine is a direct function of the compression to which the working medium is subjected.

An excess or a deficiency of fuel in mixture with air does not necessarily prevent combustion. There are, however, limits beyond which such excess or deficiency renders the mixture noncombustible, and these limits are termed limits of combustibility. Experiments by Sorel[b] to determine these superior and inferior limits by laboratory methods showed that the range of combustibility is slightly greater for gasoline than for 90 per cent alcohol. The figures given are 1.84 and 1.5 times the theoretical air necessary for combustion as the superior limits, and 0.45 and 0.40 times the theoretical air necessary for combustion for the inferior limits of gasoline and alcohol respectively. With 50 per cent excess of air, Sorel[c] calculates the temperature of combustion as being practically the same for gasoline as for denatured alcohol, which is about 1,870° C.

[a] Woodward and Preston's translation of "Carburetting and Combustion in Alcohol Engines," by Sorel, p. 44.

[b] Op. cit., p. 53.

[c] Op. cit., p. 80.

The velocity with which combustion is propagated in an engine cylinder has an important bearing on the efficiency of the combustion and it is therefore important to call attention to a few points in this connection which have been ably brought out by Sorel. He says that the rate of flame propagation depends on: (1) The composition of the mixture; (2) the initial temperature; and (3) the condition of external heat loss. According to Michelson's [a] experiments, the maximum velocity of flame propagation does not correspond to the theoretical chemical mixture of fuel and air.

Sorel's [b] experiments with laboratory conditions show the maximum rates of flame propagation of alcohol and gasoline to be 1.04 and 1.10 meters per second respectively, with a ratio of air supplied to air necessary of 1.35. As the mixture was enriched there was a more rapid falling off in the rate of propagation in the gasoline mixtures, and if the mixtures were impoverished there was a more rapid falling off in the rate of propagation in the alcohol mixtures.

From the foregoing somewhat elementary discussion of the physical and chemical characteristics of the fuels that are the subject of this investigation it will be seen that the characteristics mentioned have a most intimate bearing on the interpretation of the results of tests. Should further information be desired along these lines, reference to Woodward and Preston's translation of Sorel's work is recommended.

CHEMICAL INVESTIGATIONS.

Samples of each shipment of fuel were sent to the laboratory in glass bottles averaging about 1 pint in capacity. The following determinations were made upon the samples: Specific gravity, ultimate analysis giving the total carbon and hydrogen present, and calorific value in calories and British thermal units. There were also reported for each sample the percentage of water, the latent heat absorbed in the products of combustion, and the lower calorific value, or total calorific value less this latent heat. The laboratory work was done by E. E. Somermeier and A. C. Fieldner, under the general supervision of Prof. N. W. Lord.

DETERMINATION OF SPECIFIC GRAVITY.

The determination of the specific gravity was made in a pycnometer or specific-gravity bottle. This method gives accurate results if care be used to bring the temperature of the water or the alcohol, with which the bottle is filled, to exactly 15.6° C. when weighing. In using volatile liquids like alcohol or gasoline the work is necessarily performed in a room that is at a temperature of practically 15.6° C.

[a] Zeitschr. für physik. Chemie, vol. 3, p. 489.

[b] Woodward and Preston's translation of "Carburetting and Combustion in Alcohol Engines," by Sorel, pp. 63-66.

so that the liquid shall not be lost by evaporation during the weighing. The dew point of the atmosphere of the room should be below the weighing temperature, so as to avoid any condensation of moisture on the instrument. The specific gravities obtained in the pycnometer were checked in many cases by means of a standard hydrometer and also by the use of a Westphahl sinker suspended from the arm of an analytical balance, the sinker being weighed, first in water and then in the alcohol, both water and alcohol being maintained at a temperature of 15.6° C. All three methods gave closely agreeing results. The hydrometer was found to give results 0.0009 higher than those obtained from the pycnometer determination. The main objection to the hydrometer was the considerable volume of the material required.

ESTIMATION OF WATER IN DENATURED ALCOHOL.

The estimation of the water in denatured alcohol from the specific gravity was of prime importance in the engine tests. No direct accurate method of determining the water was available. Several experiments in this connection gave such discouraging results that it was concluded to investigate carefully how far the use of the specific-gravity method of estimating the water could be applied to the materials used in the tests and submitted to the laboratory. As the percentages of grain alcohol corresponding to the specific gravity of various dilutions with water are known to a high degree of accuracy, it is only necessary to take the specific gravity of a sample under standard conditions in order to determine the percentage of alcohol and of water in any aqueous alcohol.

The various denatured alcohols consisted almost wholly of mixtures of ethyl or grain alcohol and methyl or wood alcohol with small amounts of benzine and acetone, the latter being always present in the commercial wood alcohol used in denaturing. As the specific gravity of pure acetone, wood alcohol, and grain alcohol are nearly the same, it seemed probable that the use of the ethyl-alcohol tables with the specific gravity of the denatured mixture might be sufficiently accurate for the estimation of the water in the mixture. The percentage of wood alcohol estimated by using the ethyl-alcohol tables was found to vary only slightly from that obtained by using the methyl-alcohol tables. In other words, the methyl alcohol might be estimated from the specific gravity as ethyl alcohol and water obtained by difference, without appreciable error.

EFFECT OF BENZINE AND ACETONE.

In order to determine the effect of the one-half of 1 per cent by volume of benzine and acetone present in wood alcohol upon the determination of water from the specific gravity, the following experimental work was carried out by Mr. Fieldner:

"Colonial spirits" (methyl alcohol) was tested for acetone and found to be absolutely free. Its percentage of water by weight was determined from the specific gravity and reference to methyl alcohol tables. A sample of pure acetone of the specific gravity 0.8019 was found to contain 99.1 per cent acetone by weight by Messinger's method (for details of this method see United States Internal Revenue specifications for the wood alcohol used for denatured alcohols). An artificial wood alcohol was made up from these materials which contained the Government high limit for acetone, about 27 grams per 100 cubic centimeters. Another artificial wood alcohol was made up containing a lower limit of acetone, and also some samples with no acetone at all. These prepared wood alcohols being of a definite composition, their exact analysis by weight of water, methyl alcohol, and acetone could be accurately calculated. Using these synthetic wood alcohols, a 96 per cent grain alcohol and a kerosene with specific gravity 0.800, synthetic denatured alcohols were made of a known composition. The proportions used in making the synthetic denatured alcohols were 100 volumes ethyl alcohol, 10 volumes methyl alcohol, one-half volume kerosene, according to Government specifications. The specific gravity of the resulting alcohol was determined by the pycnometer. The equivalent percentage by weight of ethyl alcohol was obtained from tables. The difference between this percentage and 100 gave the percentage by weight of water, assuming the mixture to be a pure aqueous ethyl alcohol. The percentage of water so obtained experimentally in every case agreed within 0.2 per cent of the known amount of water present in the synthetic mixture. Usually the agreement was within 0.1 per cent. These results indicate that the percentage of water in denatured alcohol may be obtained with sufficient accuracy by the direct specific-gravity determination.

In accordance with these experiments the percentage of water was estimated for each sample from its specific gravity, using the ordinary ethyl alcohol tables.

It may be of interest to state in this connection that a series of experiments was made upon the separation of acetone and benzine from the ethyl and methyl alcohols as proposed by Herrick in "Denatured or Industrial Alcohol," with the intention of then determining the specific gravity of the mixture of ethyl and methyl alcohols freed from acetone and benzine, and from this calculating the alcohol and, by difference, the water. The net result of the experiments was to show that the separation was not satisfactory and that the danger of loss of alcohol in the various distillations was so great as to make the results of little value and far inferior in accuracy to those obtained by the direct determination of the specific gravity of the mixture.

DETERMINATION OF CALORIFIC VALUE.

The determination of the calorific value of the samples was made by the combustion in the Mahler bomb calorimeter. The liquids were weighed in gelatin capsules. The capsule containing the alcohol was then transferred to the bomb and burned with its contents in the ordinary way, using iron wire for ignition, the wire being coiled around the capsule. The size of the capsule used was 00 (Park, Davis & Co.). Sixteen to seventeen atmospheres of oxygen were found sufficient to give the alcohol complete combustion, but with gasoline more difficulty was experienced, as it was apt to burn with explosive violence, throwing unburned or carbonized gasoline against the walls of the bomb. By using a pressure of 25 atmospheres, more satisfactory results were obtained. In order to avoid incomplete combustion from the scattering of the material with explosive violence, it was important that in filling the capsules with alcohol or gasoline they should be carefully filled and no air bubbles inclosed as whenever air bubbles were formed the combustion occurred with explosive violence and was incomplete, as shown by carbonaceous residues in the bomb. The heat developed in the bomb was then corrected for the heat of combustion of the empty capsule.

PREPARATION OF SAMPLE.

The capsule contained ignited asbestos as an absorbent of the alcohol. The asbestos was washed and then ignited in a muffle for four hours to insure the removal of any combustible matter. As the gelatin capsules were very hygroscopic, before being used they were dried to constant weight over sulphuric acid in a desiccator. The capsules, after drying, were kept continuously in a desiccator over sulphuric acid and were thus always ready for use. The calorific value of the capsules was obtained by burning five capsules alone in the bomb. Two closely agreeing determinations made in this way gave 5,217 calories and 5,214 calories per gram, from which the average of 5,216 calories was taken as the heating value of the capsule material. In each determination the weight of the capsule in grams multiplied by this heating value was deducted from the heat developed in the bomb. The weighings were made rapidly, all capsules being taken directly from the desiccator.

METHOD OF DETERMINATION.

The actual method of proceeding in making a determination on an alcohol was as follows:

The capsule was taken from the desiccator and weighed rapidly, this weight being used for determining the heating value of the capsules in the tests. The capsule was then filled with ignited asbestos and allowed to stand in the air of the laboratory for 12 to 18 hours, so as to attain a condition of moisture equilibrium. It was then weighed again. This second weighing was necessary, as experiments showed that the dried capsule gained weight so rapidly as to render impossible the filling of the capsule with alcohol, weighing it again, and determining correctly the amount of alcohol taken. The capsule after standing and reweighing was filled with the alcohol or gasoline by means of a small pipette. It was then weighed again, the increase in weight being the weight of the alcohol taken. The details of the determination of heating value in the bomb calorimeter have been described in Technical Paper 8 [a] of the Bureau of Mines, and are not repeated here. The following figures are given as an illustration:

Specimen figures used in alcohol determinations.

Weight of the dry gelatin capsule	0.1277 gram.
Weight of the capsule+the asbestos	0.3770 gram.
Weight of capsule+asbestos+liquid	1.0706 grams.
Weight of liquid taken	0.6936 gram.
Heating value in the calorimeter	5,114.5 calories
Heating value of 0.1277 gram of capsule=(0.1277×5,216)	666.1 calories.
The heat of 0.6936 gram of liquid=(5,114−666)	4,448 calories.
Heat from 1 gram or calorific value	6,414 calories.

ULTIMATE ANALYSES.

Ultimate analyses were made in the combustion train regularly utilized in coal analysis. The important detail with these volatile liquids was to obtain some means of vaporizing with slowness, especially with regard to gasoline, as only a small part of gasoline vapor in mixture with oxygen (one-sixteenth by volume) is required to produce an explosive mixture.

The method followed in these experiments was to blow an elongated glass bulb or tapering tube, to a capillary opening and to fill this with the liquid by repeatedly heating the bulb and immersing the opening in the liquid until the necessary amount of about 0.2 gram was inclosed. Diffusion through this capillary opening was shown to be inappreciable, for by allowing it to remain on the balance pan 30 minutes, the change in weight was not over 0.2 milligram. The bulb containing the liquid was placed in the platinum boat in an inclined position with the opening elevated and directed toward the front of the combustion furnace. Gradual heating was necessary to prevent explosion caused by rapid vaporization of the liquid from the bulb. The bulb was necessarily at least 8 inches beyond the ignited burners, leaving the top of the furnace open above it. By careful observation of the water formed in the cold end of the furnace the rate of combustion could be regulated so as to prevent explosion. In all cases the part of the furnace containing the copper oxide was at a bright red heat before inserting the sample in the tube. The other details,

[a] Methods of analyzing coal and coke, by F. M. Stanton and A. C. Fieldner, 1912. 21 pp.

such as the quantity of air aspirated and oxygen used, were the same as those followed in ordinary coal combustion, and are described in the publication named above.

APPARATUS USED IN TESTS.

Throughout the discussion of the experiments described in this work, an understanding of the details of design of the engines and testing appliances used will be of value in connection with the interpretation of results. It is felt that if the discussion of each series of tests were encumbered with the details of description of the apparatus involved, much clearness would be sacrificed, and accordingly all such matter is here grouped in a logical manner either under the heading of "Engines" or of "Testing Appliances."

ENGINES.

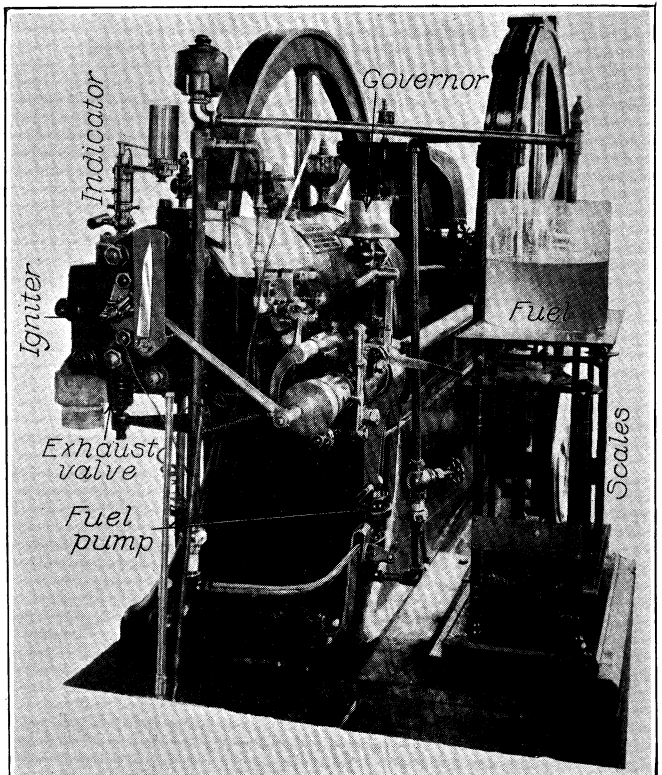
The engines used in the tests comprised two Otto gasoline engines, one Nash gasoline engine, and one Otto and one Nash alcohol engine.

OTTO 15-HORSEPOWER GASOLINE ENGINES NOS. 1 AND 2.

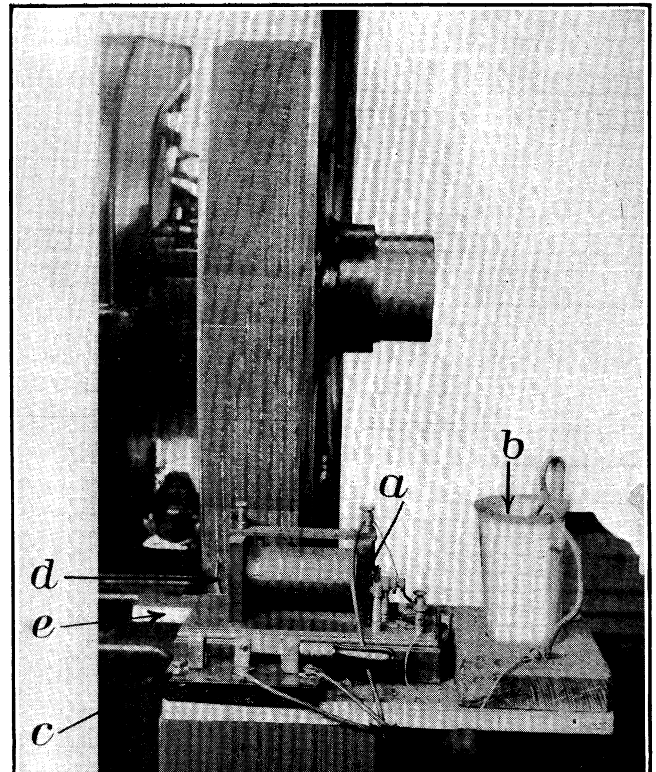
The two Otto gasoline engines were of the single-cylinder, horizontal, 4-cycle, stationary type and were alike in size, design, construction, and equipment. The engines were rated at 15 brake horsepower with a normal speed of 260 revolutions per minute. The piston diameters were $6\frac{3}{4}$ inches and the lengths of piston stroke were $15\frac{1}{2}$ inches.

An end view of the Otto gasoline engine No. 1 is shown in Plate I, *A*, and a general view of the Otto gasoline engine No. 2 in Plate II, *A*. Inasmuch as the engines are similar, each of the views may be considered as being of either engine.

The cylinders, cylinder heads, and exhaust-valve housings were water-cooled. The engines were each equipped with a fuel pump, a carburetor, and an electric igniter of the make-and-break type. The carburetor (fig. 1) which was built as a part of the cylinder head, was supplied with fuel issuing from a small nozzle, situated in the air passage near the inlet valve (fig. 1) under a static head of 16 inches. The carburetor and air passages were so constructed and connected that the fuel was atomized by the air taken in during the suction stroke and then mixed with it. The inlet and exhaust valves, which were on opposite sides of the cylinder head, were of the conical-seat poppet type and were 2 inches in diameter. The inlet valve, actuated automatically, was horizontal and opened directly into the cylinder. The exhaust valve, which was mechanically operated, was placed at the side of the cylinder head in a housing that was connected with the cylinder by a port. The engines were governed by the hit-or-miss



A. OTTO 15-HORSEPOWER GASOLINE ENGINE NO. 1. END VIEW.



B. DEVICE FOR CALIBRATING MAKE-AND-BREAK IGNITER ADJUSTMENTS.

method applied to a third poppet valve, which when opened admitted liquid fuel to the carburetor.

The cylinders of the engines were not counterbored and hence the compression pressure could be changed by varying the length of the connecting rods. The design of the connecting rods was such that they could not readily be shortened, but it was possible to lengthen them by placing a liner between the brasses and the crank end of the connecting rod. The normal clearance volume was 175 cubic inches. For some of the tests on the Otto gasoline engine No. 2 this volume was decreased to 152 cubic inches, which was the least that could be obtained without having the piston strike the cylinder head. The

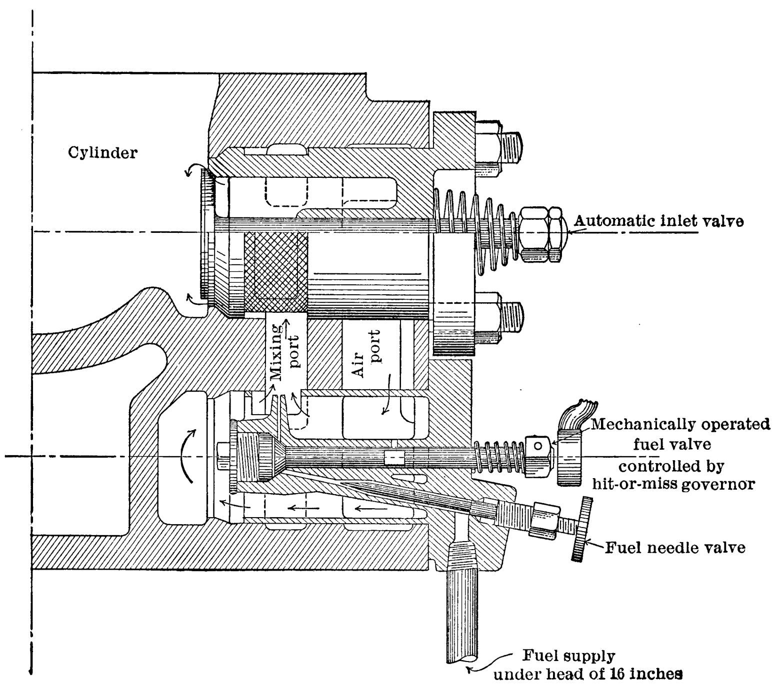


FIGURE 1.—Horizontal section through carburetor and inlet valve of Otto gasoline engine.

ratio of the clearance volumes to the total volume of the cylinder was 0.24 and 0.21, respectively, with the clearance volumes given above. The volume of the charge at atmospheric pressure in proportion to the volume displaced by a complete stroke of the piston, sometimes called the "volumetric efficiency" of the engine, was in each case 94 per cent and was kept constant at this figure by adjusting the timing of the inlet valves. The normal ratio of length of connecting rod to length of crank was $4\frac{1}{2}$: 1, and the total weight of the reciprocating parts of each engine was $96\frac{3}{4}$ pounds.

The arrangement of the parts and test appliances is shown in Plates I, *A*, and II, *A*. The sketches given in figures 1 and 2 present the horizontal section of the cylinder heads through the carburetor and inlet valve and the vertical section through the inlet and exhaust valves respectively.

FUEL SUPPLY.

A small "fuel pump" (Pl. I, *A*), which was driven by an eccentric attached to the cam shaft of the engine, pumped the fuel from the supply tank to a small "fuel reservoir" placed about 16 inches above the carburetor. The pump delivered more fuel to the reservoir than was at any time used from it by the carburetor, so that a constant head was maintained by allowing the excess to overflow and return to the supply tank by gravity. Fuel under a head of 16 inches entered the carburetor as shown in figure 1. During the suction stroke the fuel poppet valve was opened and a quantity of fuel, regulated by the fuel needle valve, passed through a small nozzle as shown in the mixing port. Here it was atomized and more or less vaporized by the air, and as the fuel and air impinged on the wire netting through which it passed before entering the engine cylinder by way of the automatic inlet valve their mixture was completed.

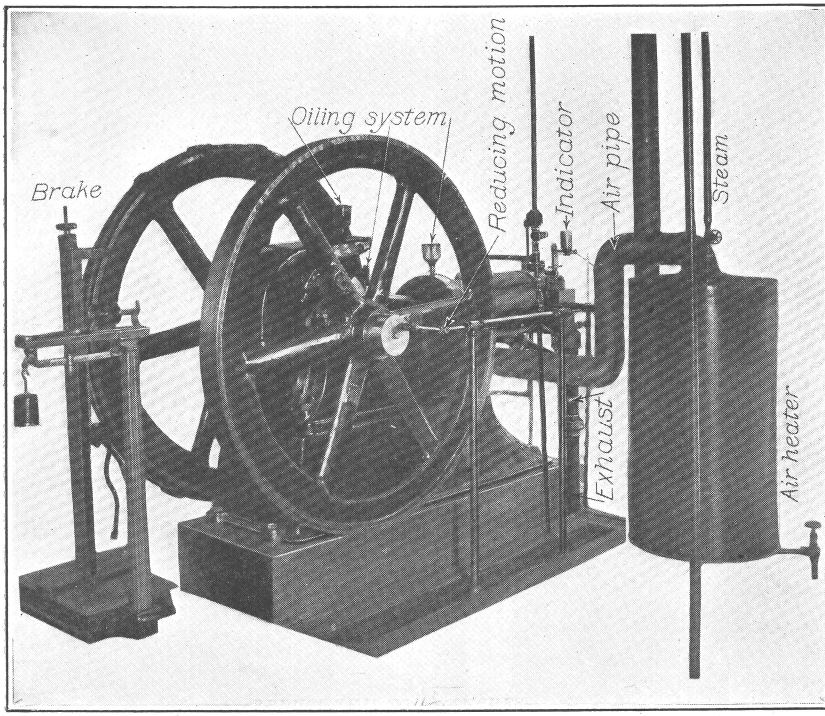
The fuel poppet valve was mechanically operated by a cam and rocker. One arm of the rocker shaft carried a small wheel, under which a cam on the cam shaft rolled and caused the other arm of the rocker to open and close the poppet valve at the desired time.

GOVERNOR.

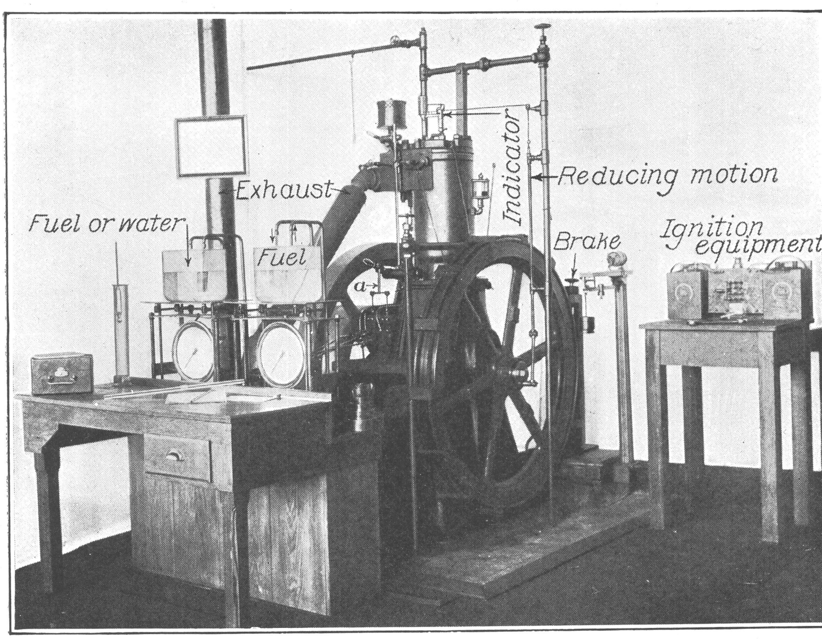
The governor of the Otto gasoline engines was of the weighted fly-ball type and connected with the small wheel or roller that engaged with the cam that actuated the fuel poppet valve. The roller was free to move laterally on the pin that carried it. A slight movement of the governor was sufficient to slide the roller either into or out of engagement with the cam, thereby causing the fuel to be admitted only when the speed was at or slightly below that for which the governor was adjusted.

VALVES.

The inlet and exhaust valves were on opposite sides of the cylinder head as shown in figure 2. The inlet valve was opened automatically by the vacuum produced in the cylinder by the outward movement of the piston during the suction stroke. The time of its opening and closing depended (1) upon the difference between the pressures in the cylinder and in the valve housing; (2) upon the tension of the spring, which tended to keep the valve closed, and (3) upon its lift, which was controlled by an adjustable stop. The valve admitted a charge either of mixture or of air during each suction stroke, and was in no way affected by the action of the governor, which, as previously stated, controlled the fuel supply only. The exhaust



A. OTTO 15-HORSEPOWER GASOLINE ENGINE NO. 2. GENERAL VIEW.



B. NASH 10-HORSEPOWER ALCOHOL ENGINE. GENERAL VIEW.

valve was opened by a lever arm, which was fulcrumed beneath the cylinder and actuated by a cam. The exhaust gases were conducted to a large cast-iron muffler pot directly below the exhaust-valve housing and from there to the outside atmosphere.

IGNITER.

The igniter, which was a little to one side of the center of the cylinder head, as shown in Plate I, *A* consisted of a brass flange, in which was housed a fixed electrode and a movable electrode. The fixed electrode was insulated from the surrounding parts by lava bushings and asbestos washers. The movable electrode was free to revolve between the fixed electrode and a stop. At the outer end it carried an interrupter hammer, which was free to revolve on the movable

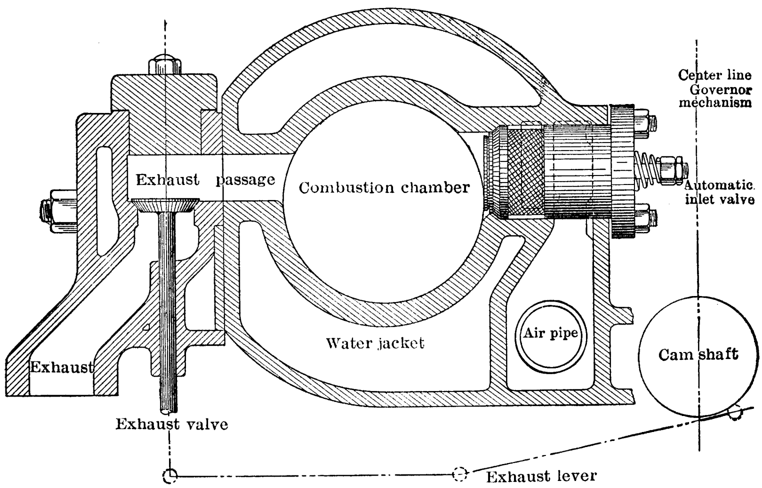


FIGURE 2.—Vertical section through inlet and exhaust valves of Otto gasoline engine.

electrode except as restricted by a lug on the movable electrode and by a spiral spring that tended to keep the interrupter hammer and lug in contact. A push rod, the end of which engaged with the hammer, first revolved the movable electrode until contact was made between it and the stationary electrode, after which the hammer revolved on the movable electrode against the tension of the spiral spring and was separated from the lug. The hammer was then released by tripping off the end of the push rod and struck the lug a blow, causing the movable electrode to revolve, with a sudden breaking of the contact that had been formed with the fixed electrode.

The electrodes of the igniter were connected in series with a spark coil and were supplied with current from an "Apple" storage battery, which was kept charged by a small generator driven from the fly wheel of the engine by a friction pulley. The spark which was caused by the sudden break of contact between the platinum points of the two electrodes occurred a fraction of a second after the interrupter hammer was tripped. The time of ignition could thus be adjusted by regulating the time at which the hammer was released or tripped from the end of the push rod. The adjustment was accomplished by raising or lowering a spool over which the push rod reciprocated. The spool was attached to an adjustable plate with dial at the end of the cylinder head as shown in Plate I, *A*, and the position of the crank in degrees before or after dead center at the time of ignition was indicated by a stationary pointer.

WATER CIRCULATION.

The cylinder, cylinder head, and exhaust-valve housing were well water jacketed. The jackets were connected by ports through which the water circulated. Water from the street main entered the cylinder jacket of engine No. 1 at the bottom, about midway between the crank and head ends, and was discharged at the top diametrically opposite the point of entry. There was no direct circulation through the jacket around the head of the cylinder or exhaust-valve box. The jackets of engine No. 2 were so connected that water from the mains entered the jacket of the exhaust-valve housing, passed through the cylinder-head jacket, and was discharged from the top of the cylinder jacket at a point about midway between its head and crank ends. A direct circulation of water around the cylinder head was thus obtained.

LUBRICATION.

The piston was lubricated by means of a sight-feed pressure oil cup placed near the crank end of the cylinder. Oil from this cup also lubricated the wrist pin. The crank pin was oiled by means of a wiper fed by a sight-feed oil cup, and the main bearings by ring oilers dipping into reservoirs beneath the bearings.

TESTING APPLIANCES.

The test appliances which were used with these engines and which are described hereafter were arranged as follows: The fuel-supply tank, filled with fuel, and the spring-balance scales on which it rested were placed near the engine as shown in Plate I, *A*. The rope brake was applied directly to the flywheel of the engine, its standard resting on the platform scales as shown in Plate II, *A*. Continuous speed counters were attached to the cam shaft and the rocker arm actuating the fuel poppet valve. The indicator was directly above the exhaust valve and was connected to the combustion chamber of the cylinder by a $\frac{1}{2}$-inch cock, as shown in Plate I, *A*. The indicator was driven by a positive reducing motion connected with the end of the crank shaft as shown in Plate II, *A*. Thermometers were placed in the fuel-supply tank and in the air pipe leading to the carburetor, but insulated from the pipe by a rubber cork. A thermometer was also placed in the inlet-valve housing. A hole was drilled through the cylinder head in such a way that the bulb of the thermometer could be placed in the mixture port between the wire gauze and the head of the inlet valve. This thermometer was also insulated from the surrounding metal by a rubber cork. Thermometer cups were placed in the water-cooling pipes near their points of entrance to the cylinder, so that the temperature of the incoming and outgoing water could be measured when desired. The cooling water after passing through the jackets was piped to a large weighing tank placed on platform scales. The air-heating device, when used, was attached as shown in Plate II, *A*.

The thermocouple that was used to measure the temperature of the exhaust gases for some of the tests made on engine No. 2 was placed in the exhaust pipe as near to the exhaust valve as possible. The wires of the couple were insulated from the pipe by small porcelain tubes cemented in a ¼-inch nipple. The millivoltmeter to which the thermocouple was attached was placed far enough away from the engine to be free from vibration.

NASH 10-HORSEPOWER GASOLINE AND ALCOHOL ENGINES.

The Nash engines, shown in Plate II, *B*, and Plate III, *A*, were of the single-cylinder, vertical, 4-cycle, stationary type, and were alike in size, design, construction, and equipment, except that the clearance volume of the alcohol engine was made smaller than that of the gasoline engine by filling a part of the combustion chamber with an extension on the cylinder head. These engines differed from the Otto engines in details of design, such as arrangement of valves, shape and volume of clearance space, type of carburetor used and in the way the method of governing was applied.

The cylinders were 7⅝ inches in diameter, the stroke of the piston was 10 inches and the brake horsepower was rated as 10 with a normal speed of 280 revolutions per minute. The cylinder, cylinder head, and main-valve housings were water-jacketed.

VALVES.

In each engine the main inlet and exhaust valves, which were placed side by side (fig. 3) in an extension of the combustion chamber, were of the conical seat poppet type and were mechanically operated. A third poppet valve, which admitted a mixture of air and fuel (fig. 4), was placed in one of the two passageways leading to the main inlet valve. The other passageway was provided with a hand-adjusted cock through which an auxiliary supply of air was admitted. The engines were governed by the hit-or-miss method applied to the poppet valve in the gas or mixture passageway, which has just been referred to. The air and fuel mixture when admitted by this valve was mixed with the auxiliary air supply in passing into the cylinder through the main inlet valve.

CARBURETORS.

The engines were each equipped with a fuel pump, a carburetor, and an electric igniter of the make-and-break type. The carburetor (fig. 5), which was attached to the gas or mixture intake passage, was of the aspirator or atomizer type having two constant-level overflow reservoirs, one for fuel and the other for water. By means of needle valves separate sprays of water and fuel were admitted and controlled as desired. The water compartment of the carburetor was originally supplied with water from the cylinder-head jacket; but in order that the water used might be measured or fuel substituted, the carburetor was connected with a pump exactly the same as that supplying the fuel compartment. These pumps were placed side by side and driven by the same rocker arm.

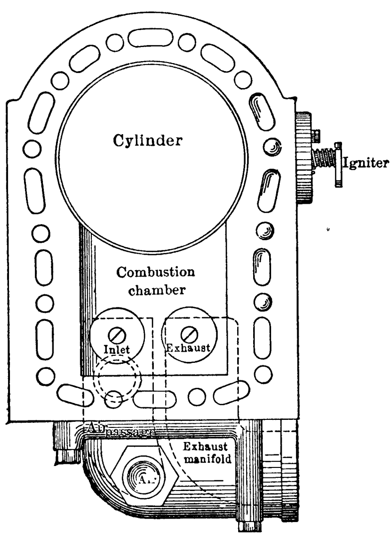


FIGURE 3.—Top view of cylinder with head removed, Nash engine.

For some of the tests with the Nash engines the regular carburetors that were supplied with them were replaced by carburetors of the same general principles but differing in size of the air passageway and in design and arrangement of the aspirator or spray nozzle. These special carburetors have been designated double-cone carburetors (fig. 6) on account of the shape of the aspirator tubes.

The ratio of the clearance volume to the total cylinder volume of the gasoline engine was 0.20 and of the alcohol engine was 0.16. The volume of the charge at atmospheric pressure in proportion to the volume displaced by a complete stroke of the piston, otherwise known as the "volumetric efficiency" of each engine, was variable and depended upon the carburetor used and upon the setting of the auxiliary air valve. The greatest volumetric efficiency was obtained in each engine with the double-cone carburetor, and was 96 per cent for the gasoline engine and 97 per cent for the alcohol engine. The compression pressure varied with the clearance ratio and the volumetric efficiencies of the engines. The greatest compression pressure for the gasoline engine was 97 pounds and for the alcohol engine 135 pounds (per square inch above atmosphere).

The fuel pumps were driven by the valve and igniter actuating mechanism and are shown in Plate II, *B*, at *a*. These pumps delivered denatured alcohol and water, denatured alcohol and gasoline,

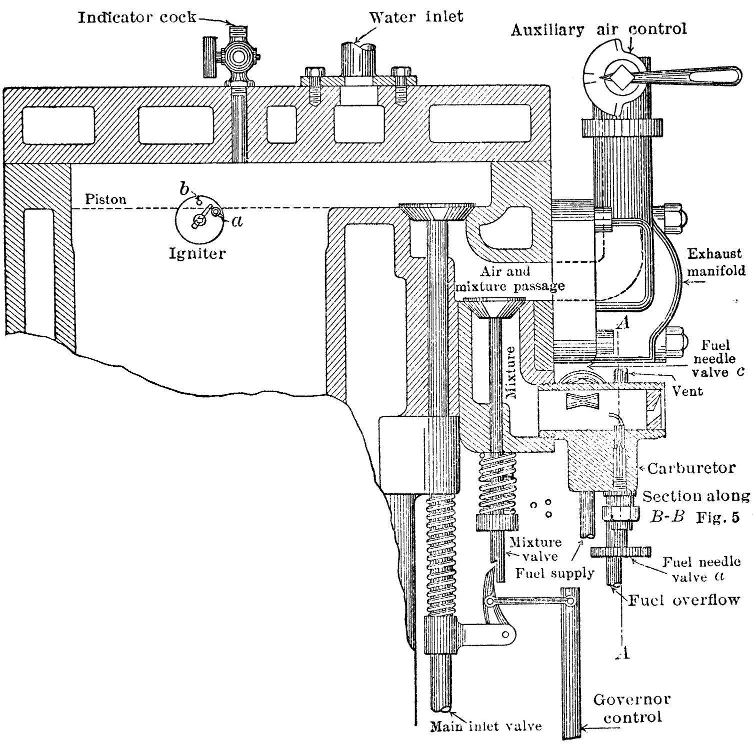


FIGURE 4.—Vertical section through inlet valves and carburetor, Nash engine.

or gasoline and water, as the case might be, to the reservoirs, shown at *a* and *b*, figure 5, always delivering more than was used, so that a constant level was maintained by allowing the excess to overflow and return to the supply tank by gravity. The mixture valve was connected to the cam-actuated main-inlet valve by a trigger, as shown in figure 4, and was lifted by it when the main-inlet valve was opened, except when the trigger was thrown out of engagement, which occurred when the speed of the engine exceeded that for which the governor was set.

During the suction stroke, when the main inlet and mixture valves were opened, part of the air taken in passed through the air passageway to the carburetor, in which the aspirator tubes or spray nozzles connected with the fuel and water reservoirs as shown (fig. 4). The air was drawn in through the orifice opening upward to the atmos-

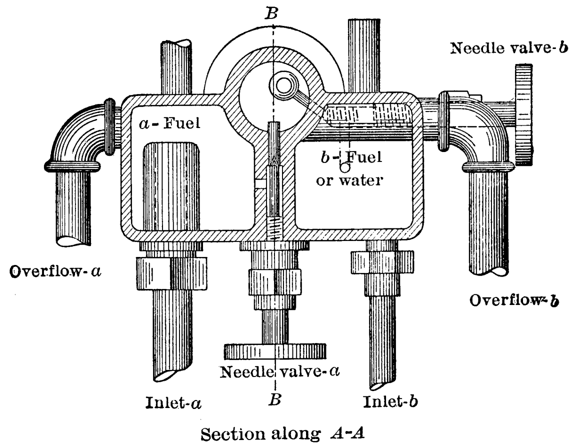


FIGURE 5.—Vertical section of carburetor, Nash engine.

phere, and the liquid fuel, which was lifted to the spray nozzles by the partial vacuum formed, was atomized and mixed with the air. This mixture of air, fuel vapors, and atomized fuel on passing through

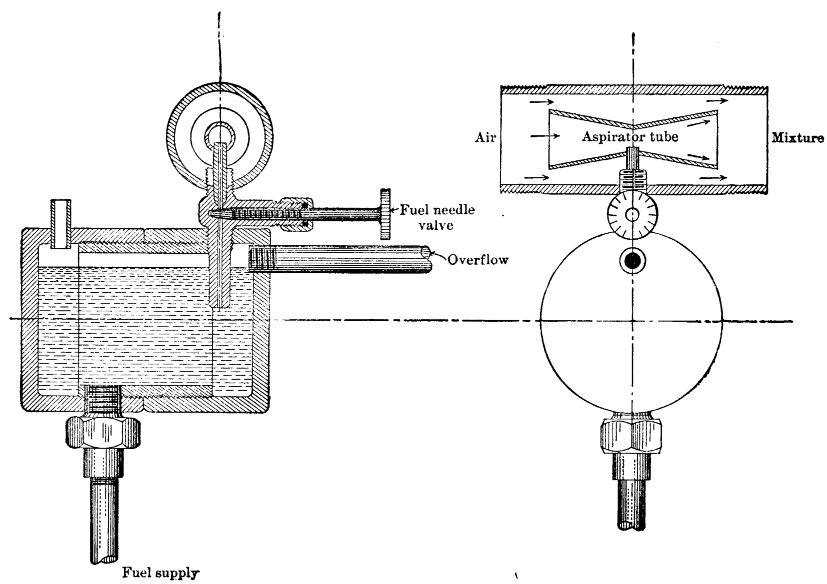


FIGURE 6.—Double-cone carburetor, Nash engine.

the fuel valve met and was mixed with fresh air that entered through the auxiliary air valve. The mixture then passed through the main inlet valve into the cylinder where it was more or less mixed with the hot products of combustion from the preceding cycle of operations. Here it was compressed, ignited, burned, and exhausted in the usual manner. When the trigger that connected the mixture valve to the main inlet valve (fig. 4) was thrown out of engagement by the governor, the mixture valve was held closed by a heavy spring, and air from the auxiliary air passage alone was admitted to the cylinder.

The double-cone carburetor that was used for some of the tests on these engines was attached to the engine at the entrance to the mixture passage, and replaced the regular carburetor. It was made from pipe fittings and is shown in section (fig. 6). Air entered through and around the aspirator during the suction stroke, as indicated. Fuel was lifted from the constant-level overflow reservoir by the partial vacuum produced at the spray nozzle and was atomized by and mixed with the air as it passed to the cylinder. The double-cone aspirator was introduced in order that the fuel might be lifted and thoroughly atomized with the least possible suction loss, and was placed concentric with and free from the air passage in order that the atomized fuel might be thoroughly mixed with the air before striking the sides of the intake passage where it had a tendency to collect and was thrown down. This carburetor was used for the sole purpose of obtaining an effective carburization with a maximum "volumetric efficiency," as it was found that satisfactory conditions in these details could not be obtained with the regular carburetor. In other words, the regular carburetor was replaced by the double-cone carburetor in order to obtain the greatest power for the given cylinder dimensions and the greatest compression pressure for the given clearance volumes.

GOVERNOR.

The governor, which was of the centrifugal type, was attached to the fly wheel near the hub and was protected by the drive pulley (Pl. III, *A*). Bell cranks which carried the governor weights at one end were attached at the other end to a sliding collar on the crank shaft between the fly wheel and the main bearing. One end of a vertical rod, which was pivoted in the middle, rested in an annular groove in this sliding collar. The other end of the rod was attached to the trigger which connected the mixture valve to the main inlet valve (fig. 4).

EXHAUST GASES.

The exhaust gases were conducted to a large iron muffler placed beneath the floor near the engine and leading thence to the outside atmosphere. The covered pipes (Pl. II, *B*, and Pl. III, *A*) are the exhaust pipes. A small stream of water from the cylinder jacket was piped in a $\frac{1}{8}$-inch pipe to the 45-degree exhaust elbow at the exhaust manifold (Pl. II, *B*, and Pl. III, *A*). Water was introduced at this point to cool the exhaust gases.

IGNITER.

The igniter at the side of the cylinder (figs. 3 and 4, Pl. III, *A*) consisted of a brass flange, in which were housed a fixed electrode and a movable electrode. The fixed electrode was insulated from the brass flange by mica washers. The movable electrode was free to revolve between the fixed electrode *a* and the stop pin *b* (fig. 4), except for the tension of a spiral spring which tended to hold it against the stop pin. The spiral spring was placed around the stem of the movable electrode between the cylinder and the small actuating lever (fig. 3). A small trip lever attached to the push rod engaged with and lifted the lever attached to the movable electrode. This revolved the movable electrode until contact was made in the cylinder, between it and the stationary electrode, and the circuit of the ignition system to which they were connected was closed. A further motion of the push rod caused the trip lever to release the lever of the movable electrode, and the spring to which it was attached caused the movable electrode to snap back against the stop pin. Contact between the two electrodes was thus broken and the circuit of the igniter system caused an electric spark in the cylinder between the two contact points.

The spark occurred a fraction of a second after the movable electrode was tripped. The igniter push rod or trip rod, which was driven by a rocker arm at one end and guided by a sleeve at the other end, was so arranged that the time of ignition could be adjusted. This was accomplished by changing the position of the guide sleeve, which was carried by a spindle, one end of which was threaded and screwed freely through a tapped hole in the supporting bracket from the cylinder. By revolving the spindle the guide sleeve, the igniter push rod, and the trip lever could be moved closer to or farther from the lever of the movable electrode with which the trip lever engaged and tripped. The spindle was graduated and the setting of the igniter, or time of ignition, was indicated by a small pointer attached to the bracket. The graduations on the spindle were calibrated and compared with the position of the crank in degrees before or after dead center. The spindle was held at any given setting by a thumb-lock nut at the bracket.

The "Apple" ignition equipment shown at the right of the engine in Plate II, *B*, was used to supply current to the igniters of the Nash engines. It consisted of a spark coil, switchboard, two storage batteries, and a small charging generator which was driven by friction from the flywheel of the engine. The spark coil was connected in series with the igniter electrodes. The switchboard was so arranged that when one of the storage batteries was thrown into the igniter circuit the other was being charged by the generator.

WATER CIRCULATION.

The water jackets around the cylinder and cylinder head were connected by ports. Cold water from the street main entered the cylinder jacket at the head end (fig. 4) and was discharged near the crank end of the cylinder. The circulation in these engines was in the opposite direction to that in the Otto gasoline engine No. 2.

LUBRICATION.

The cylinder, crank pin, and main bearings were lubricated by the splash system from the crank case. The piston and wrist pin were lubricated from a sight-feed pressure oil cup placed near the crank end of the cylinder.

TESTING APPLIANCES.

The test appliances were arranged as follows: The fuel-weighing device consisted of two accurately calibrated spring-balance scales and two fuel-supply tanks which rested freely on them. The apparatus was placed at the back of the operator's table, near the engine, as shown in Plate II, *B*. The rope brake was applied directly to the flywheel, and is also shown clearly in the illustration. The indicator, which was connected with the cylinder by the indicator cock as shown in Plate III, *A*, was driven by a positive reducing motion connected to the end of the crank shaft and supported by a pipe standard (Pl. II, *B*).

A continuous-ratchet speed counter was attached to the inlet-valve stem and another to the mixture-valve stem. The latter counter recorded the fuel admissions and the former recorded every two revolutions, thus making correct reading much easier than if the counter had been connected to the main shaft of the engine.

Thermometers were placed in the fuel reservoirs of the carburetor, at the opening of the auxiliary air valve, in the mixture passageway between the carburetor and the mixture valve, and between the mixture valve and main inlet valve. The two thermometers last mentioned were insulated from the surrounding metal by rubber corks and asbestos packing. Thermometer cups were placed at the inlet and outlet jacket-water connections so that the cooling water at these points could be measured when desired. The igniter calibrating device was attached as shown in Plate III, *A*.

OTTO 15-HORSEPOWER ALCOHOL ENGINE.

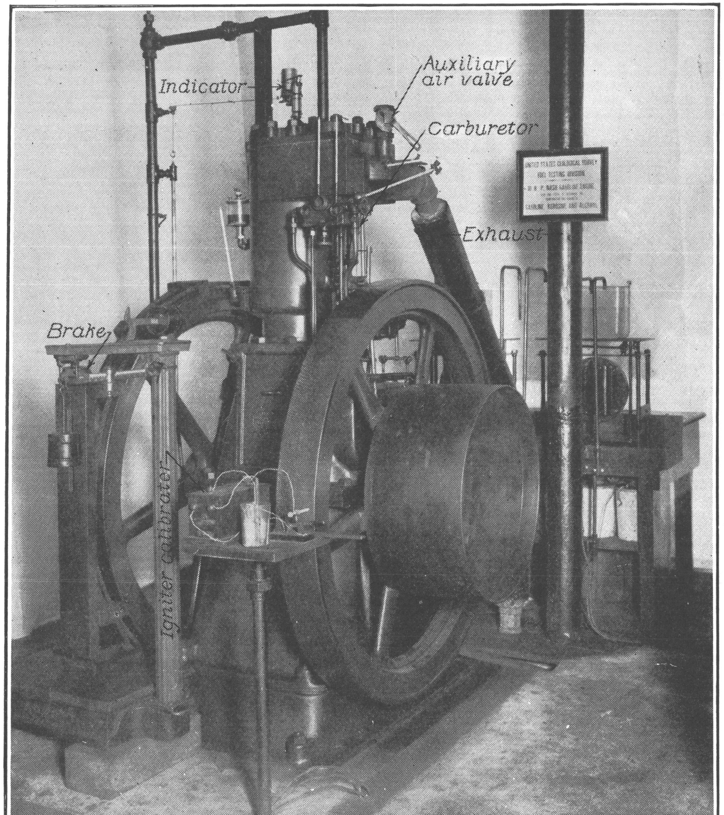
A general view of the Otto alcohol engine and its equipment is shown in Plate III, *B*. This engine, like the Otto gasoline engines, was of the single-cylinder, horizontal, 4-cycle, stationary type, rated at 15 brake horsepower, with a normal speed of 260 revolutions per minute. The cylinder, with a diameter of 6¾ inches, and a stroke of 15½ inches, had the same dimensions as in the Otto gasoline engines. The only differences to be found between the Otto gasoline and alcohol engines were in the shape and volume of the clearance space, the arrangement of the valves, the valve-actuating mechanism, the method of governing, and the carburetor.

The clearance volume was varied by placing liners of different thickness between the crank end of the connecting rod and its brasses. The alcohol engine had a much smaller combustion chamber in the cylinder head than had the gasoline engines, and clearance volumes of 78, 65, and 56 cubic inches were used for tests on the alcohol engine. The inlet and exhaust valves were placed one above the other, opened directly into the combustion chamber, and both were mechanically operated. The governor, which was connected with the inlet-valve actuating mechanism, controlled the lift and to a certain extent the timing of the valve. The method of governing was thus a combination of the throttle and cut-off methods, and the volumetric efficiency of the engine varied with the action of the governor. The greatest volumetric efficiency, which was not affected by the change in clearance volume, was 97 per cent.

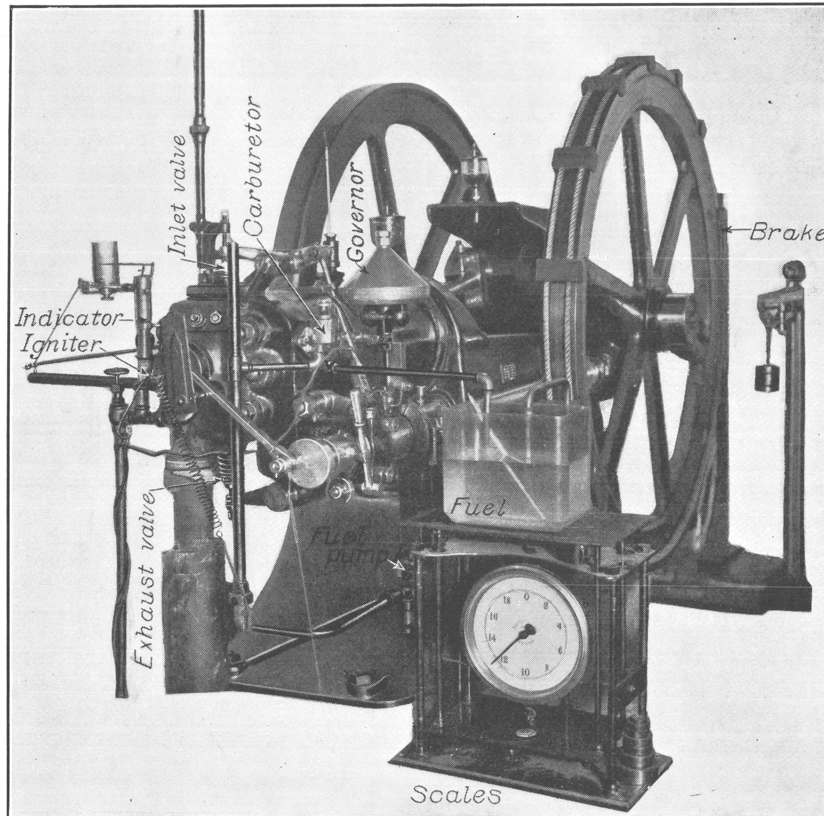
The clearance ratios corresponding to the different clearance volumes were 0.085, 0.075, and 0.063, respectively. The greatest compression pressures corresponding to the above clearance ratios and volumetric efficiencies were 140, 180, and 210 pounds per square inch, respectively, above 1 atmosphere.

CARBURETOR.

The carburetor was of the constant-level overflow, suction-lift, atomizer type, similar in principle to those with which the Nash engines were equipped. A liberal amount of fuel was pumped from the supply tank to the overflow reservoir of the carburetor, and the excess returned by gravity, as indicated in Plate III, *B*. During the suction stroke fuel from this constant-level reservoir was lifted by a partial vacuum in the air passage at the spray nozzle, where it was atomized by the air and mixed with it as the two passed into the cylinder. The flow of fuel was controlled in part by the needle valve shown in figure 7 and in part by the vacuum at the spray nozzle. This vacuum was controlled in part by a hand-adjusted butterfly valve in the air pipe leading to the air port or passageway and in part by the flow of air which varied with the opening of the inlet valve and was controlled by the governor.



A. NASH 10-HORSEPOWER GASOLINE ENGINE. GENERAL VIEW.



B. OTTO 15-HORSEPOWER ALCOHOL ENGINE. GENERAL VIEW.

INLET VALVE.

The inlet valve was operated by a cam *c* (fig. 7) on the side shaft of the engine. This cam rolled under a wheel or roller *d*, carried at one end of a short rocker arm, as indicated. To this rocker arm was attached a hardened steel plate which engaged with the end of a push rod attached to the lever which operated the inlet valve. The profile of this hardened steel plate, as shown in figure 7, was

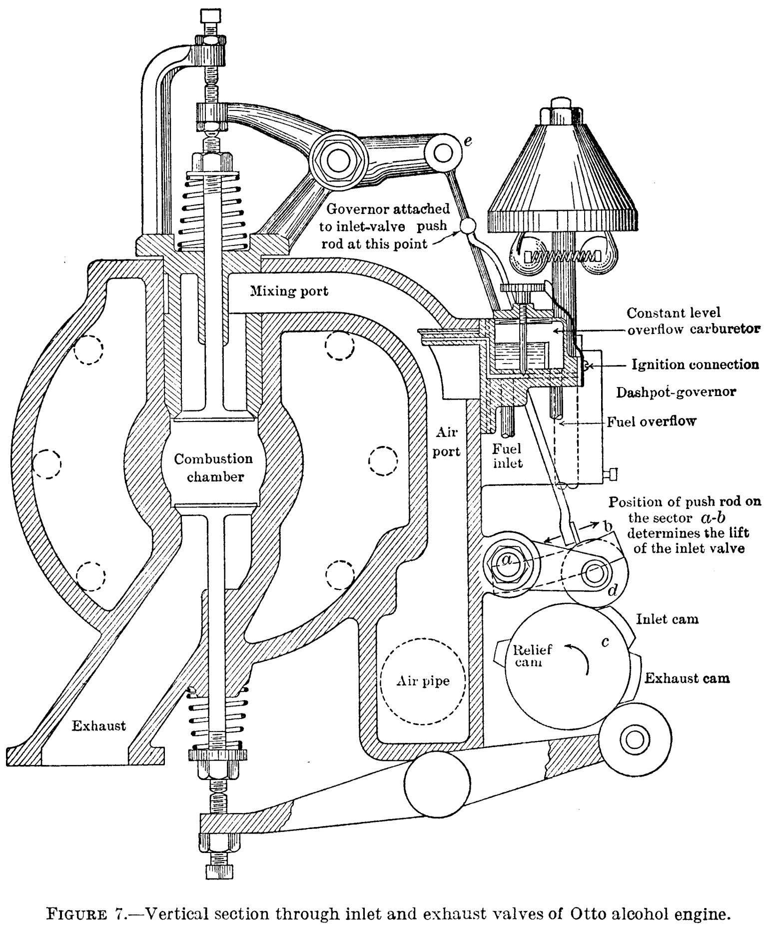


FIGURE 7.—Vertical section through inlet and exhaust valves of Otto alcohol engine.

the sector of a circle whose center was on the axis of the pin that connected the push rod with the inlet-valve lever at *e*. This sector passed through the center line of the rocker-arm pivot, and the position of the free end of the push rod with reference to the outer end of the sector determined the lift of the inlet valve. For example, the lift of the valve was the greatest when the free end of the push rod was at *b* and was zero when it was at *a*.

GOVERNOR.

The governor, which was of the fly-ball weighted type, was connected with the inlet-valve push rod by a small bell crank at the point indicated (fig. 7), and a small movement of the governor was sufficient to shift the push rod through its entire range of position. Due to a small back lash between the free end of the push rod and the sector plate with which it engaged, the lift of the inlet valve became zero before the push rod reached the point *a*, and, because of this clearance, not only the lift of the inlet valve, but also the length of time it was held open, was decreased when the position of the push

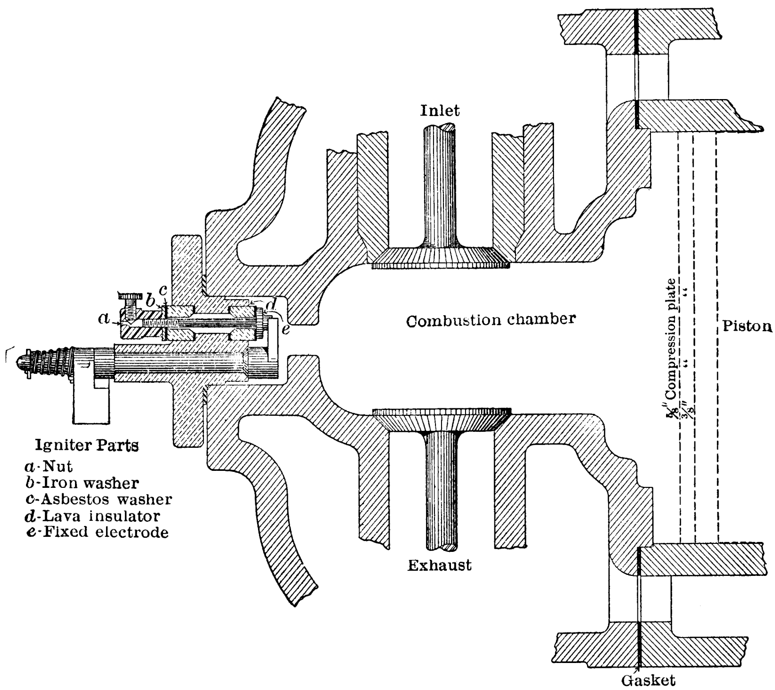


FIGURE 8.—Longitudinal section through combustion chamber and igniter of Otto alcohol engine.

rod was shifted from *b* to *a*. Governing was thus accomplished by a combination of the throttle and cut-off methods.

EXHAUST VALVE.

The exhaust valve was operated by a cam and a lever arm pivoted underneath the cylinder in exactly the same way as were those of the Otto gasoline engines. The relative position of the inlet and exhaust valves with respect to the combustion chamber and igniter is shown in figure 8.

IGNITER.

The igniter (fig. 8), the mechanism which actuated it, and its battery equipment were identical with those of the Otto gasoline engines. The adjustment of this igniter as indicated by the adjusting dial was also arranged to give the position of the crank at the time ignition took place when the engine was running.

WATER CIRCULATION.

The water jackets of the cylinder and cylinder head were connected by ports, and the water from the street main which entered at the end of the cylinder head near the bottom was discharged at the top near the crank end of the cylinder. The cooling system was similar to that of the Otto gasoline engine and was very effective. The circulation of the water was similar to that in the Otto gasoline engine No. 2.

LUBRICATION.

The method of oiling and the arrangement of all the test appliances, except the indicator and the thermometer, that were used to measure the mixture temperature were the same as those of the Otto gasoline engines.

INDICATOR.

The indicator was attached to the head of the cylinder of the alcohol engine, as shown in Plate III, *B*, and was connected with the igniter compartment of the combustion chamber by a ½-inch hole. The thermometer, used to determine the mixture temperature, was placed in a hole drilled into the mixture port near the inlet valve. It was insulated from the surrounding metal by asbestos packing, with the bulb alone exposed to the air-and-fuel mixture.

TESTING APPLIANCES.

The brakes and indicator reducing motions used on the different engines were of the same type and construction, the fuel weighing devices were of the same principle, and the indicators, platform scales, speed counters, and thermometers were identical in every respect. The igniter settings of each engine were calibrated with the same specially constructed device. An air heater and a weighing tank for cooling water were also constructed; a thermocouple was used for measuring certain high temperatures. These appliances are described below.

BRAKES.

The type of brake used is shown in figure 9. This type of brake was constructed for the 50-inch flywheel of an engine rated at 10 horsepower at a speed of 300 revolutions per minute. It consisted of a ⅝-inch manila rope which encircled the flywheel and was attached to an oak standard, as shown. The standard, one end of which rested on platform scales, was held a small distance from the face of the flywheel by two distance pieces of equal length, one placed at the top of the standard and the other an equal distance below the center of the flywheel. The ends of these distance pieces, which rested against the face of the flywheel, were notched in such a way as to hold the ropes in place, but not to bind them. A hole was drilled through the top of the standard at such an angle that the iron rod which passed through it was on a tangent to the rim of the flywheel, as shown in the figure (fig. 9). This rod was hook-shaped at one end and threaded to within an inch or so of the hook to allow for adjustment by the hand wheel, which bore on a steel washer, between which and the end of the stand-

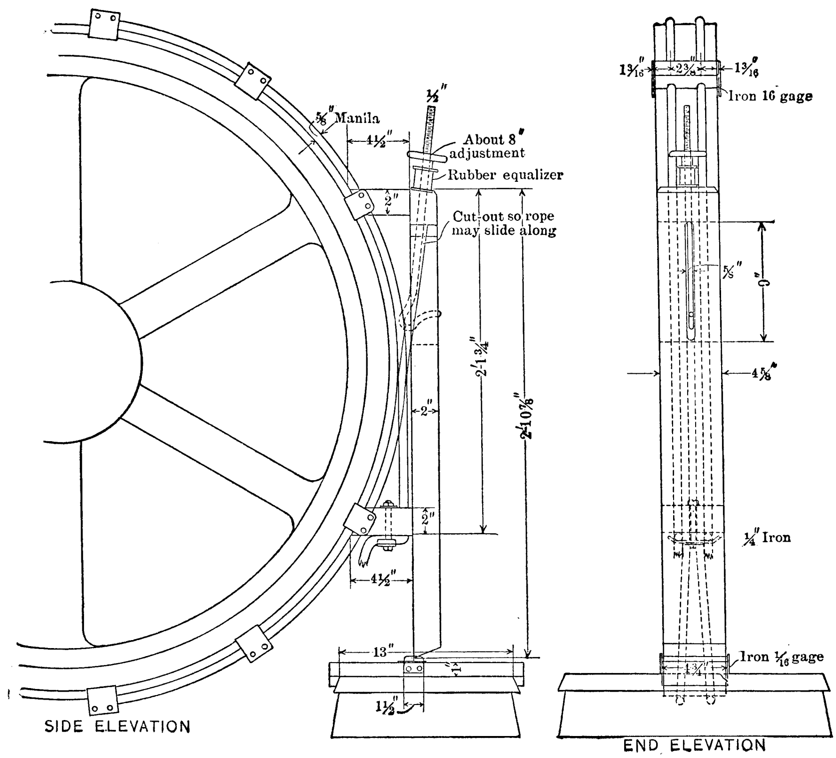


FIGURE 9.—Rope Prony brake.

ard was placed a rubber collar, as shown. The rod was kept from revolving by the point of the hook, which passed through a slot cut in the standard.

The back of the standard below the upper distance piece was channeled out in such a way that the rope which was looped over the hook did not prevent its being raised from the position shown in the figure. The rope passed down through the notches in the lower distance piece and around the wheel through the notches in the upper distance piece, and the ends were attached to the standard. The notches in the distance pieces were so spaced that the loop over the hook passed freely where it crossed between the ends that were fastened to the standard. The rope formed the main rubbing surface and was held in place on the rim of the flywheel by guide blocks attached to it and spaced about 1 foot apart. The flywheel revolved toward the brake standard and the rope tended to revolve with it, owing to the friction between the rope and the surface of the flywheel. This movement was prevented by the hook over which the rope was looped and by which it was attached to the upper end of the standard.

By revolving the handwheel, the hook could be raised or lowered, increasing or decreasing the initial tension in the rope and the pressure of the rope on the wheel. By means of this adjustment of pressure, the friction between the rope and the face of the wheel could be varied at will or kept constant for a given brake load. The downward pull of the rope on the hook was transmitted to the standard through the rubber collar, which acted as an equalizer. Any small or sudden change in friction caused the rubber to be compressed or allowed it to expand with a corresponding lowering or raising of the hook and adjustment of the pressure and friction between the rope and wheel.

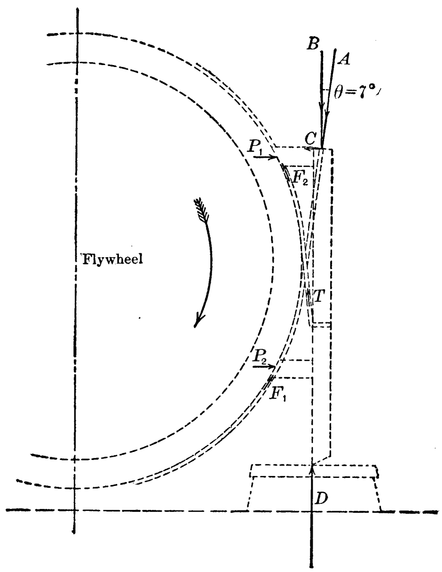


FIGURE 10.—Diagram showing forces acting on rope Prony brake.

A diagram showing the forces acting on the standard and their points of application is given in figure 10, from which it will be seen that both the initial tension of the rope and the tension due to the friction is transmitted to the upper end of the standard; but since the ends of the rope are fastened to the standard as indicated, only the force due to the friction of the rope and the distance pieces is transmitted by the standard to the scales, which also receive the weight of the standard.

The resistance to rotation of the flywheel, or the net brake load, may thus be determined by subtracting the weight of the brake standard and accessories from the total weight recorded by the scales. The length of the brake arm is the perpendicular distance between the center of the flywheel and a vertical line passing through the knife-edge at the foot of the standard.

Brakes of this design were constructed by the operators for each engine and found to be very satisfactory. With them the load was quickly and easily adjusted and could be held constant with little attention. When any difficulty was experienced in holding the load constant with these brakes, it could almost invariably be traced to the binding of the slack ends of the rope at the slots in the upper distance piece.

The flywheels to which the brakes were applied were cooled only by air. This was sufficient, however, to keep the temperature at the rim below that which would burn the rope and distance pieces. These were occasionally coated with graphite and oil. With continuous use the rope lasted about one month, while the distance pieces did not wear away an appreciable amount in three months. The brake standards were always plumbed, and since the wear on the distance pieces was negligible the lengths of the brake arms were practically constant.

PLATFORM SCALES.

The platform scales used in connection with the above-described brakes had a capacity of 300 pounds and were so adjusted that they balanced at zero when the brake standard and accessories, exclusive of rope and guide blocks around the wheel, were placed on them. The net brake loads were thus recorded directly.

FUEL-WEIGHING DEVICE.

The fuel-weighing device or apparatus used for measuring the fuel consumed by the engine during a test is shown plainly in the photographs of the engines and their equipment, Plates I, *A*, II, *B*, III, *A*, and III, *B*. Each consisted of platform spring-balance scales of the type shown and a small supply tank for the fuel, which rested freely on the scales. The scales and fuel tank each had a capacity of about 20 pounds. A siphon was rigidly fastened to the stand of the scales and held in such a way that it was entirely free from the supply tank and the moving parts attached to the spring balance. One leg of the siphon was immersed in the fuel. The other leg was connected to the suction pipe of the pump on the engine. The leg of the siphon which was immersed in the fuel was made of thin tubing of such dimensions that the error in the weight of fuel due to its displacement was negligible. A flexible metal pipe was used to connect the other leg of the siphon to the fuel pump. A globe valve was placed in this pipe line, and between the globe valve and the pump was an air chamber. By using the flexible pipe the scales were not affected by the vibration of the engine. The globe valve was used to throttle the suction of the pump, and its pulsations were cushioned by the air chamber, so that a steady flow of fuel from the supply tank was obtained and the vibration of the spring-balance needle reduced to a minimum.

The overflow from the carburetor was piped back to the supply tank. The overflow was also supported in such a way that it did not interfere with the balancing of the supply tank or the action of the scales. The spring-balance scales were accurately calibrated by means of standard weights, and were sensitive to a weight of 0.05 pound when in use.

INDICATORS.

Crosby combination gas and steam engine indicators were used. The cylinders of the indicators were constructed in such a way that two pistons of different diameters could be used. The areas of these pistons were $\frac{1}{4}$ and $\frac{1}{2}$ square inch, respectively. Indicator springs of eight different sizes, which were adapted to different pressures, were used. Depending upon whether the $\frac{1}{4}$ square inch or $\frac{1}{2}$ square inch piston was used, the scale of the springs was 10 or 20, 30 or 60, 50 or 100, 60 or 120, 80 or 160, 100 or 200, 120 or 240, 150 or 300 pounds per square inch.

For taking diagrams from which the mean effective pressure was to be determined the $\frac{1}{4}$ square inch piston and a 120 or 300 pound spring were almost invariably used. For taking diagrams of the pressures in the cylinder during the suction and exhaust strokes the $\frac{1}{2}$ square inch piston and a 10-pound spring were used. When taking these so-called light-spring or suction diagrams a small sleeve was placed over the piston rod of the indicator. The sleeve acted as a stop, preventing the parallel motion and spring from being injured by the high explosion pressures. Both the $\frac{1}{4}$ square inch and $\frac{1}{2}$ square inch pistons, with suitable springs, were used for measuring the compression pressure of the engines. All the indicator springs were calibrated when first received and again after the tests were completed, but no measurable error was found.

INDICATOR REDUCING MOTIONS.

Reducing motions of the type shown in Plate II, *A* and *B*, were constructed by the operators and fitted to each engine. The reducing-motion crank pin was attached to a cap over the end of the crank shaft of the engine. The cap, which revolved with the crank shaft, was held in place by a cap screw, and was set so that the axes of the reducing-motion crank pin, the engine crank pin, and the crank shaft were in the same plane. This adjustment was obtained by means of a trammel and the usual method of placing an engine on dead center. The throw of the reducing-motion crank pin was about $1\frac{5}{8}$ inches for each engine. The connecting rod of the reducing motion was made of such length that the ratio of the length of the crank to the length of the connecting rod was the same as that for the engine on which it was used. The sliding rod to which the connecting rod was attached was placed in perfect alignment with the cylinder of the engine and held in position by guides shown in the illustrations. The indicator drum received its rotary reciprocating motion from the sliding rod by means of a cord which passed through the guide pulleys. The connecting rod and the sliding rod of the reducing motion were made of tubing, so that the inertia would be small and any lost motion could be taken up by the tension of the spring in the indicator drum.

PLANIMETER.

An Amsler polar planimeter was used for measuring the areas of the indicator diagrams. By adjusting the sliding arm of this planimeter the area traced by the point and recorded by the wheels could be measured in different units.

The area of an indicator diagram in square inches, when divided by its length (projected on the atmospheric line) in inches, is the average height of the diagram in inches. When a number of diagrams of the same length were to be measured, the average mean effective pressure could be determined directly by first adjusting the sliding arm of the planimeter so that the steel points were a distance apart equal to the length of the diagrams measured on the atmospheric line, and then multiplying the area recorded on the dials of the wheels by a factor equal to the scale of the spring divided by 40. This latter procedure was followed in calculating the mean effective pressures for the tests given in this report.

SPEED COUNTERS.

Small ratchet counters with registering dials reading to 100,000 were used. These continuously recording counters were actuated by means of a cord or wire attached to a valve stem, rocker arm, or other convenient moving part of the engine. Hand-revolution counters were used to check readings of the continuous counters, which when found in error were replaced. Because the speed of each engine was very nearly constant an error in the reading of the continuous counters could be very easily detected by comparisons with previous readings, which were made at intervals of 5 minutes during the tests.

TIME RECORDERS.

Stop watches reading to $\frac{1}{5}$ second and watches of the usual type were employed for recording time during the tests. The watches were kept in good condition and regulated as required.

THERMOMETERS.

The chemical thermometers read to about 300° F. and were of standard make. They were compared with a standard thermometer and none of them was found to be more than 1° in error.

THERMOCOUPLE.

A platinum, platinum-rhodium thermocouple was used for reading the high temperatures of the exhaust gases. Its leads were passed through a cold junction of the usual construction and attached to a millivoltmeter reading in degrees centigrade. The thermocouple and millivoltmeter were calibrated and the correction factor was applied in determining each temperature used.

DEVICE FOR CALIBRATING MAKE-AND-BREAK IGNITER SETTINGS.

The device used for calibrating make-and-break igniter settings, shown in Plate I, *B*, consisted of an induction coil, *a*, a Weynault interrupter, *b*, a small knife-blade switch, *c*, and a pointed electrode, *d*, attached to a slide, *e*, all of which were mounted on a stand placed in front of the engine flywheel. The induction coil was of the familiar type used with a jump-spark igniter equipment. The vibrator of the coil was, however, disconnected and a Weynault interrupter substituted. This interrupter consisted of a glass beaker partly filled with a weak solution of sulphuric acid in water, in which were immersed two electrodes. One of the electrodes was a sheet of zinc and the other was a small platinum wire sealed in the end of a glass tube. The tube was bent in such a way that the end of the platinum wire pointed upward. Contact was made with the platinum electrode by filling the tube with mercury, which also kept it from becoming overheated when in use.

A diagram of the connections is shown in figure 11. The calibrating circuits are indicated by the solid lines, and the regular ignition circuit by the dotted lines. The electrodes of the interrupter were connected in series with the igniter electrodes and the primary of the induction coils. In order to obtain satisfactory results with the Weynault interrupter it was necessarily supplied with direct current at about 40 volts, and in such a way that the platinum electrode had a positive polarity as indicated. The interrupter would not operate with a low voltage, but with the voltage used the interruptions were far more rapid than those of the usual mechanical vibrator and gave good service because of this essential feature.

One side of the secondary of the induction coil was grounded to the engine, and the other side was connected to the electrode, *d* (fig. 11), the point of which was about $\frac{1}{16}$-inch from the face of the flywheel.

The engine was first placed on the upper or outer dead center. The face of the flywheel for some distance from the electrode, *d*, was then covered with smoked paper which was fastened at both ends with shellac. The primary circuit of the induction coil was then completed by closing the switch and by making contact between the igniter electrodes. This caused a stream of sparks to flow between the point of the electrode, *d*, and the face of the flywheel. The sparks punctured the smoked paper, and when the point of the electrode, *d*, was moved across the face of the flywheel by means of the slide to which it was attached a horizontal white line on the smoked paper was produced. This line indicated the position of the electrode when the crank of the engine was on the upper or outer dead center and was designated as the center or 0° line. It was used in determining the actual point or time of ignition when the engine was running, records of which were made on the same smoked paper according to the following method:

The time of the tripping of the igniter was adjusted and the position of the point of the adjusting dial noted. The point of the

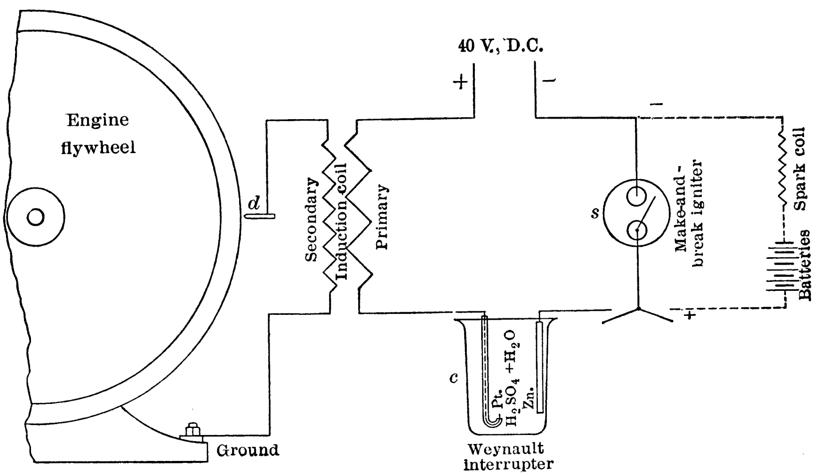


FIGURE 11.—Diagram of electrical connections of device for calibrating make-and-brake igniter adjustments.

electrode, *d*, was then placed near the edge of the smoked paper and the engine was revolved several times by hand. During this operation contact was made between the igniter electrodes and held until the movable igniter electrode was tripped. During the time the igniter electrodes were in contact the primary circuit of the induction coil was closed, but was open at all other times so that a white line on the smoked paper, made by the sparks from the electrode, *d*, in the secondary circuit, began at the time the contact was made and continued until the contact was broken by the tripping of the movable electrode. The distance of this end of the white line from the horizontal zero line indicated the position of the crank at the time the spark was produced in the cylinder and could be transposed to degrees by multiplying by 360 and dividing by the circumference of the wheel.

When the engine was revolved by hand the crank did not move a measurable distance between the time the igniter was tripped and contact was broken, but when the engine was running at its rated speed the fraction of a second that elapsed between the tripping of the igniter and the actual break of contact between the igniter points was often such that the crank would have traveled 10° before the spark occurred; in other words, for a given igniter setting, ignition occurred 10° later when the engine was running at its rated speed than when it was revolved slowly by hand.

A mechanical lag in the igniter mechanism had to be considered. The mechanical lag varied with the type and construction of the make-and-break igniter used, and in many instances varied also with the setting or adjustment of the time at which the movable electrode was tripped and with the speed at which the engine was operated. Hence the igniter adjustment dial of each engine was calibrated when the engine was running at its rated speed. The procedure of making the calibrations was as just described, except that in order to get a distinct and positive record the different igniter settings and the position of the electrode, *d* (fig. 11), were held constant for about 50 revolutions, because the punctures in the smoked paper caused by each successive spark were about one-half inch apart.

By shifting the horizontal position of the electrode, *d*, for each change in igniter setting a separate record on the same piece of smoked paper was obtained for each of the divisions of the igniter dial. While changes in the igniter settings were being made the switch in the calibrating circuit was opened and the engine was kept running by closing the switch in the regular igniter circuit.

In order to determine the accuracy of the above-described method of calibrating the igniter settings and to determine whether the variation in the recorded time of ignition for a given setting was due entirely to the mechanical lag of the igniter mechanism or due in part to a dying down of the current in the secondary circuit of the induction coil, the following experiment was made:

A contact plate was attached to the side of the rim of the flywheel, and a stationary brush was so placed that contact with the plate during part of the revolution of the flywheel was made and broken instantaneously. The contact plate and brush were substituted for the igniter electrodes in the primary circuit of the induction coil. The brush and contact plate were thus connected in series with the interrupter electrodes and primary of the induction coil. The engine was then revolved by hand at a speed of about 10 revolutions per minute, and as the secondary circuit was not altered the time at which contact between the brush and contact plate was broken was recorded on smoked paper. The engine was then started and brought up to its rated speed, and a second record was made of the time at which contact between the brush and plate was broken. This record was made on the same smoked paper as the previous one, but a little to one side of it, and the procedure was repeated several times.

The difference between the various records of the point at which contact was broken for speeds of 10 and 260 revolutions per minute was measured and found to be less than 1°. This difference is negligible so far as the calibration of the igniter-adjusting dials is concerned. The record also showed that practically the entire difference between the actual time of ignition when the engine is revolved by hand and when running at its rated speed is due to the mechanical lag of the igniter mechanism.

AIR HEATER.

The air heater consisted of a number of spiral coils of copper pipe inclosed in a sheet-iron drum. The drum was open at the bottom and closed at the top except for a 2½-inch pipe connection by which it was connected to the air-inlet pipe of the engine when desired. The coils, which were held in place by a strap-iron standard and such fastenings as were necessary, were connected by unions. The standard also supported the sheet-iron drum. The coils were connected to a pipe line carrying steam at a pressure of 125 pounds. The pressure of the steam in the coils and the flow of steam through them was controlled by globe valves at the entrance to and exit from the coils. The air which entered at the bottom of the drum passed up, around, and between the copper steam coils, and was thus heated before being taken into the cylinder through the carburetor. By adjusting the pressure and flow of steam in the copper coils their temperature was regulated and the temperature of the air was held constant at any point between the room temperature and about 260° F. The air passages in the heater and the pipes connecting it to the air-inlet pipe of the engine were of liberal size and did not measurably affect the suction loss of the engine.

WEIGHING TANK FOR COOLING WATER.

The weighing tank for cooling water was a large sheet-iron drum open at the top and fitted at the bottom with a large gate valve through which the water was discharged. The valve was so constructed that it could be opened or closed quickly and was so situated that when the tank was placed on the platform scales the water was discharged into an opening in the floor under which there was a drain. The tank thus rested freely on the scales which were placed so that the cooling water from any one of the engines could be piped to a point above the tank and discharged into it when desired.

PROCEDURE OF TESTS.

Throughout the conduct of the tests described in this report a certain regular procedure was adhered to as closely as the exigencies of testing, involving special conditions as they arose, would permit. The procedure involved three major subdivisions of the work of testing and preparing the results of the tests for presentation in this bulletin. For convenience the three subdivisions are designated "Adjustment of engines," "Data records," and "Observed and calculated results."

ADJUSTMENT OF ENGINES.

Prior to making any regular tests each engine was put in as nearly perfect running order as possible. Careful attention was given to the adjustment and lubrication of all bearings. The pistons of the engines were frequently cleaned with kerosene and the best cylinder oils were used. The pipe lines through which the fuel was conveyed to the carburetors were frequently inspected and leakage prevented. After each engine was prepared for test it was run at its rated load until a constant running condition, and one which could be maintained, was reached.

TESTING APPLIANCES.

The spring balances of the fuel-weighing devices were calibrated at frequent intervals. The indicator reducing motions were often adjusted and care was taken that the tension of the springs of the indicator drums was kept such that there was neither lost motion nor overtravel.

The distance blocks of the rope Prony brake were kept well lubricated, and it was the duty of one observer during each test to give the brake such attention as was required to keep the load constant.

FUEL PASSAGEWAYS.

Larger fuel passageways are necessary in the use of denatured alcohol than in the use of gasoline, and it was therefore necessary to enlarge the passageways in the carburetors of the Otto gasoline engines. Trial tests with gasoline were made before and after enlargement. They seemingly demonstrated that the change did not affect the fuel consumption with gasoline. Any possible effect could be remedied by proper adjustment of the fuel needle valves.

None of the carburetors of the other engines required alterations to suit them to the use of both gasoline and denatured alcohol.

VALVES.

The inlet and exhaust valves of all the engines were inspected at frequent intervals with a view to keeping them tight. When necessary they were removed and reground. The most advantageous

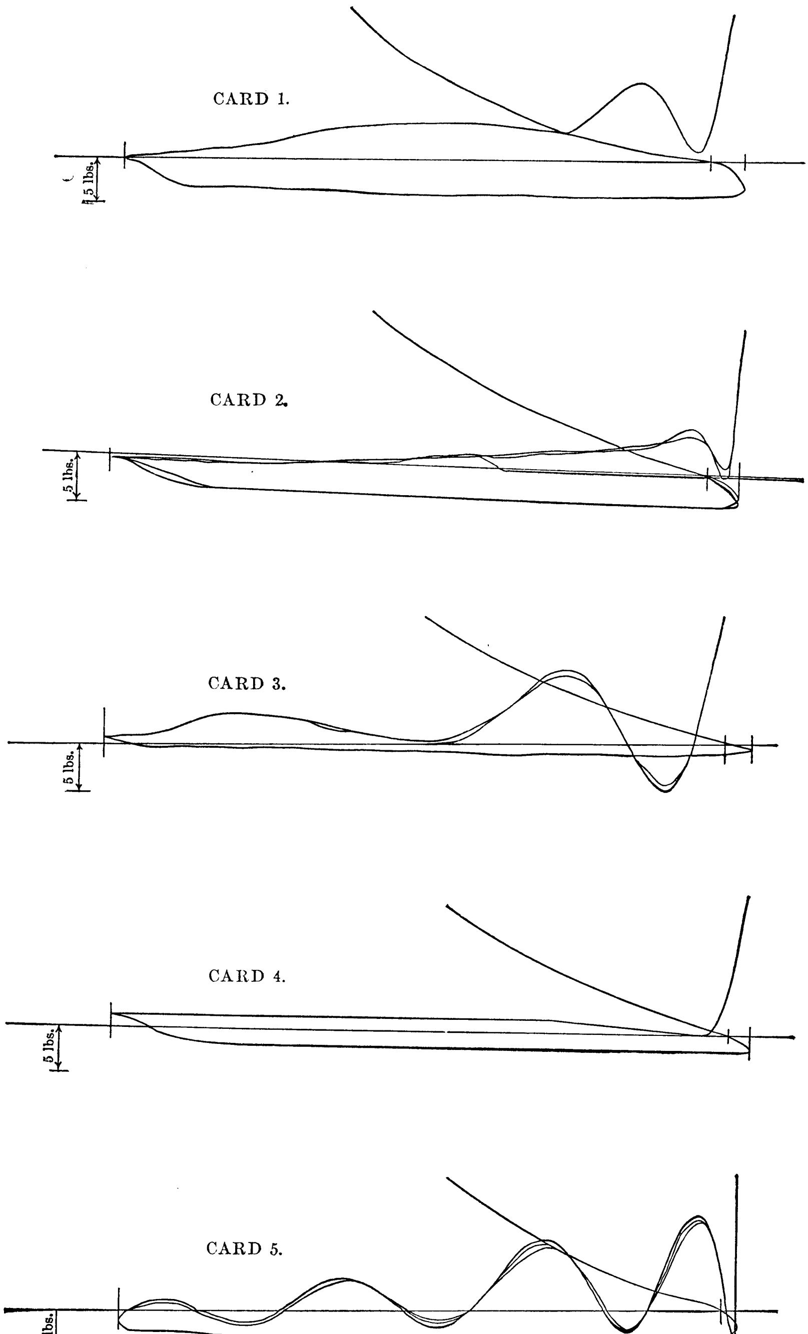


FIGURE 12.—Indicator cards showing pressure in engine cylinder during exhaust and suction strokes. Card 1 shows diagrams of Otto gasoline engine No. 1, volumetric efficiency, 91 per cent; card 2, Otto gasoline engine No. 2, volumetric efficiency, 94 per cent; card 3, Nash gasoline engine, volumetric efficiency, 96 per cent; card 4, Nash alcohol engine, volumetric efficiency, 97 per cent; card 5, Otto alcohol engine, volumetric efficiency, 97 per cent.

timing of the exhaust valves was determined by trials and the inspection of indicator cards. The lift and spring tension of the automatic inlet valves with which the Otto gasoline engines were equipped, and the mechanical actuating mechanism of the Nash engine and the Otto alcohol engine were so adjusted as to give the greatest volumetric efficiencies possible. The adjustment was obtained by means of light spring indicator cards, similar to those shown in figure 12.

In taking such diagrams as given in figure 12, a light spring and a stop to protect it from being strained by the explosion pressures were usually placed in the indicator and the pressures measured directly from the diagrams by means of suitably graduated scales. These scales were graduated in pounds per square inch and the measurements were made from the atmosphere line.

FUEL NEEDLE VALVES.

The heads of the fuel needle valves were divided into the greatest convenient number of divisions and they were also provided with stationary pointers attached in such a way that the needle valve could be held constant at any desired position or opening and that a definite numerical record of the setting could be made.

IGNITERS.

The igniter adjusting dials of each engine were graduated and the graduations calibrated as described on pages 38 and 58. The igniter electrodes of each engine were cleaned, their insulation and wiring inspected, and an electric current of sufficient strength to insure an effective spark was supplied. Each igniter with its actuating mechanism was kept in such adjustment that the time of ignition was constant during any one test.

Whenever the igniters of the Otto engines were removed for cleaning or any change made in the actuating mechanism such that the time of ignition for given settings of the adjusting dials was altered, the graduations were recalibrated or the position of the dials changed in such a way that the original calibration was correct.

The adjusting dials of the igniters of the Nash engines were also calibrated before regular tests were made on them. Frequent changes in the adjustment of the actuating mechanism of the igniters, which altered the relation between the position of the crank at the time of ignition and the graduations on the dials, made it impracticable to keep the dials accurately calibrated. Hence the time of ignition in degrees of the crank with respect to the inner dead center was ascertained for only a few of the first tests made on the Nash gasoline engine.

DATA RECORDS.

ORIGINAL DATA.

Data sheets and filing cards, 8 by 5 inches, were used for recording the test data and the calculated results. These data sheets and cards were numbered serially. The readings and observations of about four individual tests and all notes pertaining to them were recorded on each data sheet. Such of these readings as were taken at regular intervals during a test were averaged and all calculations that were necessary to determine the weight of fuel consumed per brake horsepower per hour or other fundamental calculated results were made and recorded on the data sheet by the operator.

PRELIMINARY CURVE SHEETS.

As the tests of each individual investigation progressed the operator platted preliminary curves of the fundamental results. These curves were platted on sheets of cross-section paper, 8 by 5 inches in size.

INDICATOR CARDS.

Duplicate sets of indicator cards were taken for all tests, and for each engine each set of cards was numbered serially. For each test the serial number of the indicator cards taken was recorded on the data sheet with the rest of the test data. This record of the indicator card numbers served, first, to identify them with the rest of the test data, and second, to designate or number the test. Where several indicator cards were taken during a test the test number was the number of the first indicator card. Each of the duplicate sets of indicator cards for the tests recorded on one data sheet was placed in a filing envelope, each of which was given the same number as the data sheet.

RESULT SHEETS.

At the completion of each experiment, the curve sheet, the data sheets, and one of the sets of indicator cards were sent to the computing department. The filing cards having on them carbon copies of the curve and the data sheets and the other set of indicator cards were retained by the operator for reference. All of the calculations given on the data sheets were checked by the computing department, the indicator cards integrated, and the final calculated results computed and entered on the "result sheet" which was numbered the same as the data sheet.

INDEX CARDS.

The data and result sheets were indexed according to the principal features of the experiment of which they were a part. As a result of the use of the above-described system of recording and filing data and notes, the compiling of the tables given in this bulletin was greatly facilitated.

OBSERVED AND CALCULATED RESULTS.

TEMPERATURES.

All temperature measurements given in the tables are in degrees Fahrenheit and are averages of readings taken at intervals of 5 or 10 minutes. All of the temperature readings except those of the exhaust gases were made directly with mercurial thermometers (p. 48). The exhaust-gas temperatures were read in degrees centigrade from the millivoltmeter of the thermocouple (p. 49). These readings were corrected for the temperature of the cold junction, and the corresponding Fahrenheit readings were calculated in the usual manner (°F. = 5/9 °C. + 32).

AIR.

The air temperature at the intake of the carburetor was practically constant for any one test and usually very nearly constant for any one series of tests.

FUEL.

The temperature of the fuel supplied was usually approximately the same as the temperature of the air and was very nearly constant for any one test or series of consecutive tests.

JACKET WATER.

The jacket-water temperature was measured on entering (inlet temperature) and leaving (exit temperature) the engine jacket through which it circulated. The jacket water was supplied from the street main, and its inlet temperature could not be controlled. The temperature was, however, practically constant and varied but little from the approximate values given. The exit temperature was kept constant during each test and practically constant during each series of tests except where otherwise noted.

MIXTURE.

The mixture temperature, or the temperature of the air-and-fuel mixture, as it passed into the cylinder probably fluctuated. The lowest reading of the thermometer, which was reached at about the time the piston reached the end of the suction stroke, was invariably taken. The readings were undoubtedly high, owing to the fact that the bulb of the thermometer, which was placed in the passageway near the inlet valve, was in the mixture current only during the suction stroke, or about one-fourth of the cycle. For the other three-fourths of the time the reading of the thermometer was affected by heat radiated from the inlet valve and the walls of the passageway, and the average values given for the different tests are therefore only relative.

EXHAUST GASES.

The temperature of the exhaust gases also varied during each cycle, because the temperature of the gases issuing from the exhaust valve was undoubtedly higher when the valve was first opened than toward the end of the exhaust stroke. The intermittent flow of the gases further complicated matters, and the measurements made with the thermocouple were, at best, integrations of the temperatures existing in the exhaust passage during the complete cycle. Hence the exhaust temperatures given in the tables are undoubtedly lower than the temperature of the gases when released by the exhaust valve, and higher than the temperature of the gases left in the clearance space. It is probable, however, that they are proportional to the true temperature in either case.

INDEX SETTINGS.

The engine settings comprising the adjustments of the fuel-needle valve, the air valve, and the igniter, all of which were adjusted by hand, were invariably kept constant during any one test and were read directly from the index dials with which they were fitted. The settings are tabulated in the same terms as those used when they were read.

The graduations on the dials of the fuel-needle valves were arbitrary and had no special significance except when all other conditions affecting the quality of the mixture were practically constant. Under constant conditions the quality of the mixture varied with the opening of the fuel needle valve, but not directly, as explained in connection with the discussion on effect of mixture quality. The same statement applies to the graduations of the hand-adjusted air valve or the valve controlling all or part of the air taken in during the suction stroke.

The igniter settings, except where otherwise noted, indicate directly the position of the crank in degrees before or after dead center at the time ignition took place, when the engine was running at its rated speed. Early ignition, or ignition before the crank had reached dead center, is indicated by a plus sign (+); late ignition, or ignition after the crank had passed dead center, is indicated by the minus sign (−).

SPEEDS.

During each test successive readings of the dials of the recording counters (p. 48) were made at intervals of 5 or 10 minutes depending on the duration of the test. The time intervals were measured with the second hand of a stop watch, which was started at the time the first reading was taken and stopped at the time of the last reading.

The total number of counts for every time interval was determined by subtracting the first reading from the last reading, and the number of counts per minute was determined by dividing the remainder by the number of minutes in the interval. The counts per minute for the different time intervals were recorded on the data sheet and averaged. As a check the average counts per minute for the entire time were determined by subtracting the first reading of the first interval from the last reading of the last interval and dividing the remainder by the total time interval in minutes.

The revolutions per minute as given are twice the average number of counts per minute recorded by the counter that was driven regularly by the 2-to-1 valve shaft of the engine (p. 48).

The average piston speed, when given, is in feet per minute and was calculated by multiplying twice the length of the stroke of the piston in feet by the revolutions of the engine per minute.

The fuel admissions per minute as given for the tests on the throttle-governed Otto alcohol engine are one-half of the revolutions per minute. For the tests on the other engines, which employed the hit-or-miss method of governing, the fuel admissions per minute are the average number of counts per minute recorded by the counters attached to the valve-actuating mechanism, which was moved only when a charge of fuel, or fuel and air, was admittedto the cylinder, of the engine (pp. 30 and 48). The explosions per minute were the same as the fuel admissions per minute except when the readings of the latter were corrected for misfiring or firing during cut-outs.

The cut-outs per minute, which are given only for the tests on the hit-or-miss governed engines, are obtained by substracting one-half of the revolutions per minute from the number of fuel admissions per minute.

PRESSURES.

All pressures were measured by means of the indicators (p. 47), and the values given are in all cases the average of several determinations.

The mean effective pressure, when measured from the indicator diagrams, is expressed by the equation

$$\text{M. E. P.}=\frac{a\times s}{l},$$

in which M. E. P.=the mean effective pressure in pounds per square inch;

a=the area of the indicator diagram in square inches;

s=the vertical scale of the diagram in pounds per square inch, or the scale of the indicator spring (p. 47).

l=the length of the diagram in inches.

In many cases s and l were constant, in which event the sliding arm of the planimeter was set so that the steel points on the upper side were a distance apart equal to the length of the diagram measured on a line parallel to the atmospheric line and the mean effective pressure was then calculated from the formula:

$$\text{M. E. P.} = \frac{r}{c},$$

in which r = the area of the diagram as indicated by the reading on the roller of the planimeter;

and (p. 40) $c = \frac{s}{40}$.

Since there is often a great difference in the area (a) of the indicator diagrams taken at different intervals during a test, it is quite evident that the mean effective pressure, calculated from an individual diagram or from a number of diagrams taken at random, may be far from the true average mean effective pressure, but since the variation in the mean effective pressure is due largely, if not entirely, to the action of the governor, the conclusion seems reasonable that if the load is constant and the governor is so constructed and adjusted that its action is not erratic the above variation will follow some fairly definite sequence. Hence the indicator diagrams were taken and the average mean effective pressure calculated as described below.

MEAN EFFECTIVE PRESSURE.

Shortly after a test was started two indicator cards or sets of indicator diagrams were taken. When the test was a trial test of about 15 minutes' duration, these were the only cards taken; but when the test was a check or final test of about 30 minutes' duration, four more cards were taken—two at about the time the test was half completed and two shortly before the end of the test. Five or ten consecutive diagrams were recorded on each card to form a set.

For tests on the two Otto gasoline engines and the two Nash engines, which employed the hit-or-miss method of governing (p. 28 and p. 34), the above-mentioned sets of diagrams usually began with the first explosion after the cut-outs. When the conditions under which the engine was running were such that the explosions between cut-outs occurred in sets of less than four, the pencil point was held against the card until diagrams for several sets of explosions were recorded and both the number of successive explosions between cut-outs and the total number of explosions or diagrams recorded were counted by the operator and noted on the card (card 1, fig. 13). When the operating conditions were such that the successive explosions between cut-outs were from 5 to 10 in number, diagrams for one

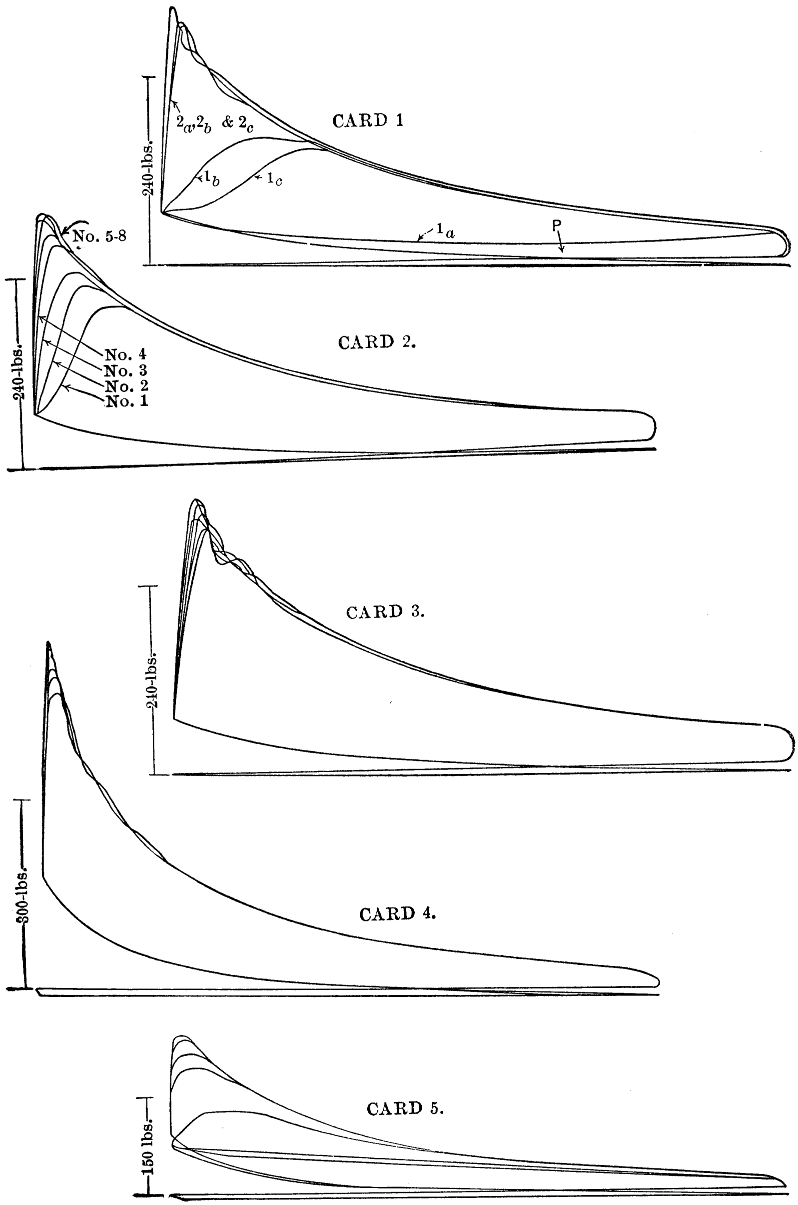


FIGURE 13.—Indicator cards illustrating method of determining mean effective pressure and explosion pressures. Card 1. Light load. Hit-or-miss governor. Explosions between cut-outs occur in sets of two. Diagrams taken of three sets, viz: 1_a and 2_a, 1_b and 2_b, etc. Card 2. Rated load. Hit-or-miss governor. Explosions between cut-outs occur in sets of 15. Diagram taken of first eight. Card 3. Rated load. Hit-or-miss governor. Explosions between cut-outs occur in sets of 15. Diagrams taken of last eight. Card 4. Rated load. Throttle governor. Diagrams taken of five consecutive explosions. Card 5. Light load. Throttle governor. Diagrams taken of eight consecutive explosions.

entire set were recorded on each indicator card taken and the number of explosions was noted as before. When the number of explosions between cut-outs was greater than about 10, and conditions were such that the first few diagrams only were irregular in size and shape—that is, irregular as compared with those for the remainder of the explosions before the next cut-out—diagrams for about the first eight explosions were taken, but both the number of diagrams recorded and the total number of explosions in the set between cut-outs were noted on the card (card 2, fig. 13).

In many cases no two diagrams of a set taken from cut-out to cut-out were of the same size and shape, but it was found that when the governor mechanisms of the two Otto gasoline and the two Nash engines were carefully adjusted, the diagrams for all but the first few explosions after cut-outs were generally of practically the same shape and size. To determine whether or not conditions were such that the diagrams for only the first few explosions of a set were irregular, diagrams of the last eight explosions of a set were sometimes taken on separate cards at intervals during the test (card 3, fig. 15).

On cards 1 and 2 (fig. 13) it may be noted that the smallest diagram is for the first explosion after cut-outs, the diagram for the second explosion being larger, etc. Such a sequence was usually obtained when the carburetor was adjusted for a relatively weak mixture, but when adjusted for a relatively rich mixture the diagram for the first explosion after cut-outs was usually larger than that for any other explosion in the set. The relative size and shape of the uniform and the irregular diagrams also depended to a certain extent on the time of ignition.

Except when a very light load was applied to the Otto alcohol engine, for which a combination of the throttle and cut-off method of governing was used (p. 40), the indicator diagrams for successive explosions were nearly uniform. Diagrams of five successive explosions were usually sufficient to show the entire variation, which in many cases was less than that shown by card 4 (fig. 13). When a light load was applied, adjustment of the governor so that it would not "hunt" was difficult. This condition caused irregular diagrams, as shown on card 5 (fig. 13).

In calculating the average mean effective pressure for such cases, as illustrated by cards 1 and 5 (fig. 13), the positive loop of each diagram was followed by the tracing point in the planimeter, and the total area recorded by the roller of the planimeter was noted. If any difficulty was experienced in tracing the diagrams, the procedure was repeated until a fair average was obtained. This average, when divided by the number of diagrams and multiplied by the constant, c, was the average mean effective pressure of the diagrams and was so noted on the card.

In cases such as illustrated by card 2 (fig. 13) the positive loops of the irregular diagrams (Nos. 1 to 4, inclusive) were traced consecutively and the total area noted. A diagram which was selected by inspection as representing the mean of the remaining diagrams (Nos. 5 to 8, inclusive) was then traced, and the area recorded was multiplied by the total remaining number of explosions (11) in the set of (15) consecutive explosions between cut-outs. The product was then added to the total area of the irregular diagrams (1 to 4, inclusive) and the sum divided by the total number of explosions in the set (15). In cases such as this the operation was usually repeated three times, and the average of the three average areas was multiplied by the constant *c*. This product represented the average mean effective pressure during the explosion period between cut-outs, and was so noted on the card.

In cases such as illustrated by cards 3 and 4 (fig. 13) a diagram representing the mean of the various diagrams was selected by inspection and the mean effective pressure calculated from this diagram was noted on the card.

When sets of two or more cards were taken during a test, the average mean effective pressure was calculated from each and an average of the average mean effective pressure given on the cards was made. This final value is the one given in the tables.

The indicator cards shown in figure 13 do not illustrate all the irregularities that had to be contended with, but for every test made the operators endeavored to obtain series of diagrams such that, with the aid of the notes relating to the relative number of uniform and irregular diagrams and their sequence, a logical procedure could be followed out in calculating the average mean effective pressure.

In a few cases where there was good reason to believe that the mean effective pressure could not be determined accurately from the indicator cards, it was calculated from the equation

$$\text{M. E. P.} = \frac{\text{B. H. P.} + k}{nC}$$

in which B. H. P. = the brake horsepower (p. 67);

k = the previously determined average friction loss or average frictional horsepower of the engine (p. 69);

n = the number of explosions per minute (p. 59);

C = the indicated horsepower constant of the engine (p. 68).

Tests for which this method of calculating the mean effective pressure was applied are as accurate as any of the other tests, and justification is established in the discussion of "Mechanical efficiency" (pp. 73 to 81) under "Deductions from tests."

EXPLOSION PRESSURE.

The average explosion pressure or the average of the greatest pressures produced by the individual explosions during a given test was determined from the indicator diagrams in a way similar to the procedure followed in determining the average mean effective pressure—that is, in cases such as illustrated by cards 1 and 5 (fig. 13); the greatest pressure recorded on each of the diagrams was measured directly by means of an appropriately graduated scale, and the sum of these pressure measurements, which were in pounds per square inch above atmosphere, was divided by the number of diagrams. In the case illustrated by card 2 (fig. 13), eleven times the average of the greatest pressures recorded on diagrams 5 to 8, inclusive, was added to the sum of the maximum pressures recorded on diagrams 1 to 4, inclusive, and the total sum was divided by 15. The average of the greatest pressures recorded on diagrams 5 to 8 (card 2) was ascertained with sufficient accuracy by inspection to be determined by one measurement. In such cases as illustrated by card 4 (fig. 13), the greatest explosion pressures recorded can also be averaged quite accurately by inspection and determined by one reading. When sets of two or more cards were taken during a test the average explosion pressure was determined in the above way for each card in the set and a second average made. These final averages are given in the tables.

COMPRESSION PRESSURE.

The compression pressures given at the head of the tables, or in the tables when variable, are the pressures in pounds per square inch above atmosphere of the charge when compressed to the end of the compression stroke. When this pressure could not be measured from the indicator diagram owing to early ignition, special compression diagrams were taken. These diagrams were usually taken either before or after making a test or series of tests, but the conditions of operation affecting the compression that existed during the test were invariably maintained while the compression diagrams were being taken.

The pressure to which the charge was compressed, when ignited, varied with the time of ignition and was appreciably less than the compression pressures given when the time of ignition was more than about 10° before or after dead center. These variations were, however, not taken into account and the three methods employed for determing the compression pressure were as follows:

1. The cord by which the drum of the indicator was revolved was connected to the reducing motion in the usual manner and the mixture charges for several successive cycles were then kept from being ignited by disconnecting one of the wires leading to the electric

igniter. Immediately on disconnecting the wire several indicator diagrams were taken. Owing to the momentum of the fly wheels the speed of the engine was not reduced appreciably while the dia-

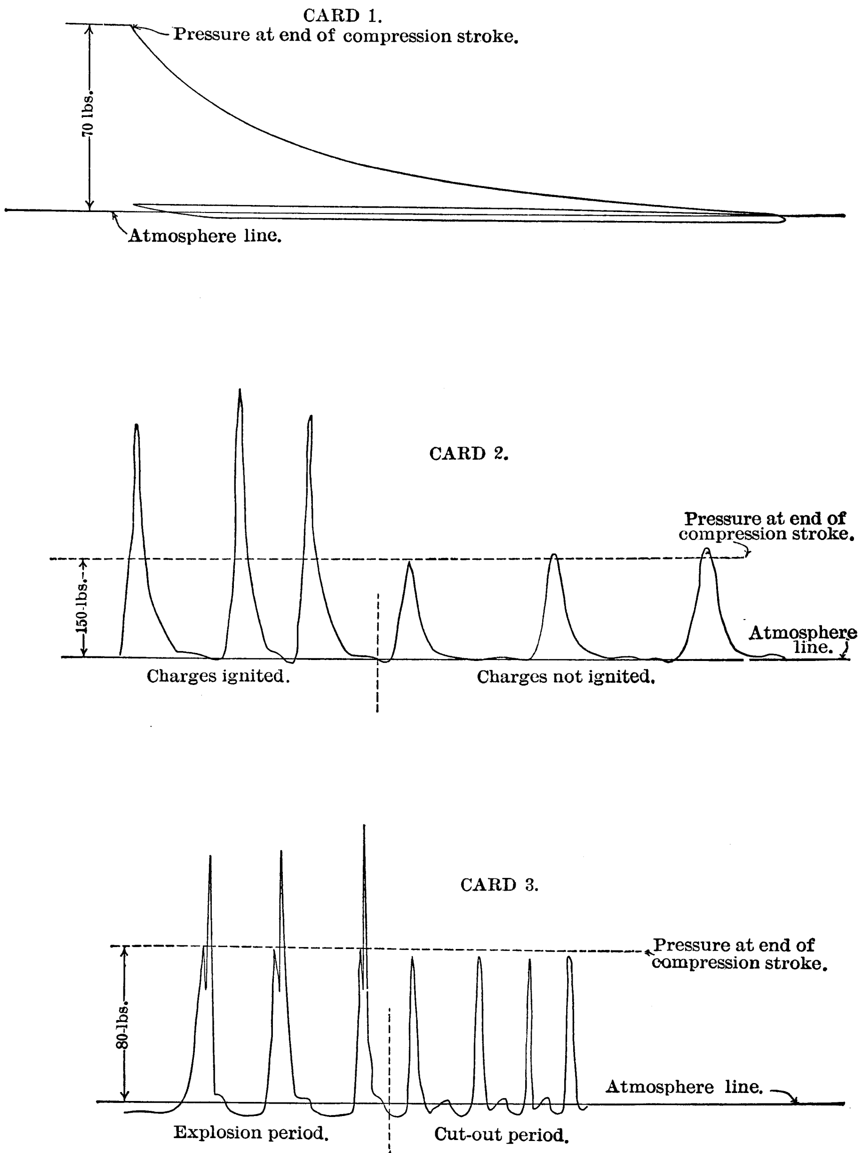


FIGURE 14.—Indicator cards illustrating methods of measuring compression pressure.

grams were being taken. This method (card 1, fig. 14) was used for the hit-or-miss governed engines only. The diagrams were invariably taken during what would have been a regular explosion period between cut-outs had not the circuit of the electric igniter been broken.

2. Instead of attaching the indicator drum to the reducing motion, it was revolved slowly by hand and the pressures developed in the cylinder for each different stroke were recorded separately and continuously on the card. While the pressures due to several consecutive explosions were being recorded in this way, the circuit of the electric igniter was opened as before, so that for several consecutive cycles the charge was not ignited and the compression pressures were recorded, as shown by the diagram (card 2, fig. 14).

3. The third method was the same as the second except that the igniter was set so that the charge was not ignited until the piston had passed the dead center. The circuit of the igniter was not broken. By this method both the compression pressure and the explosion pressure of each cycle of operations were recorded.

When used for a hit-or-miss governed engine, the compression pressure during the cut-out period could also be recorded as shown by the diagram (card 3, fig. 14).

The adjustment of valves affecting the compression (p. 30) was not altered during the time in which tests on the Otto gasoline engines No. 1 and No. 2 were being made, and as these engines were governed by the hit-or-miss method the compression was only measured when the clearance ratio was changed (p. 29) or when conditions of operation affecting the volume of the charge at atmospheric pressure, such as speed, were varied. In measuring the compression pressures of the Otto gasoline engines the first method described was employed, and the values obtained are given at the top of the tables in which the tests on them are reported.

The Nash engines were governed by the hit-or-miss method, and the conditions of operation for many of the tests were such that the compression pressure was constant. But as the clearance ratio of the two Nash engines was different and their volumetric efficiency was altered by the change of carburetors and also varied with the setting of the auxiliary air valve, it was necessary to determine the compression pressure of each engine for each setting of the air valve when each of the two carburetors was used.

VARIATION OF COMPRESSION PRESSURE.

The variation of compression pressure with change in air-valve setting from zero (or closed) to one (or wide open) for each engine and carburetor was determined as follows:

For several different air-valve settings compression diagrams, taken by the third method mentioned above, were measured and the compression pressure was platted with the corresponding air-valve setting as shown in figure 15. The compression pressure for each test on the Nash engines is given in the tables as determined from the curves given in figure 15.

During the time the tests were made on the Otto alcohol engine, three different clearance ratios were employed, and for each clearance ratio various settings of the air valve were used. As previously explained, the air valve supplemented the action of the governor, which was a combination of the throttle and cut-off types, and hence the compression pressure for each clearance ratio varied with the air-valve setting and with the action of the governor. The air-valve setting for any one test was constant, and as the load was also con-

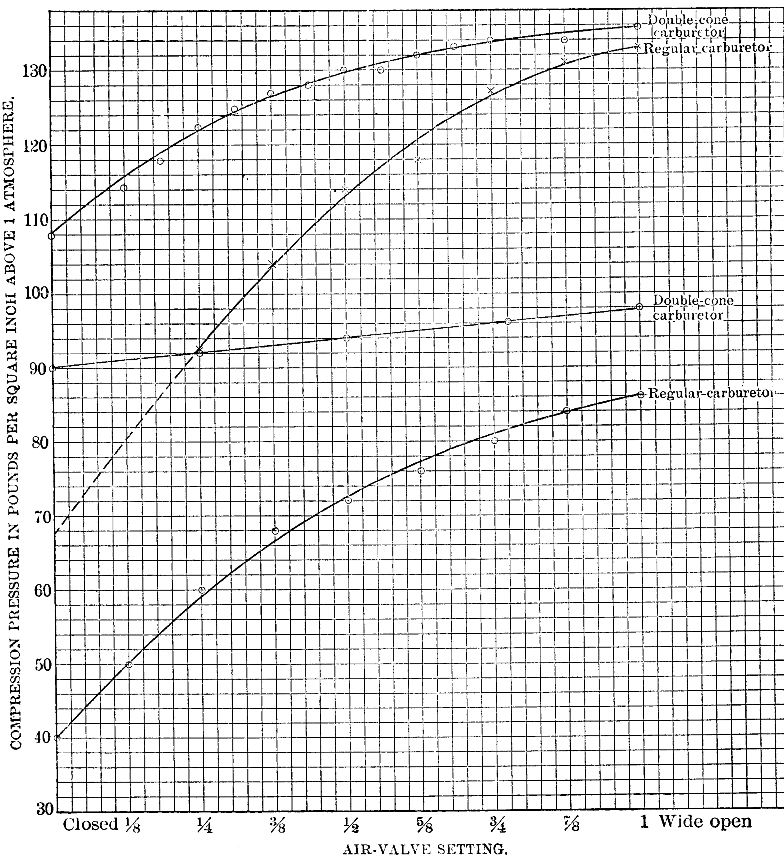


FIGURE 15.—Curves showing variation in compression with air-valve settings.

stant, the action of the governor was practically so in most cases, but owing to change of load and mixture quality the air-valve setting and governor action were seldom constant for two consecutive tests. Hence the compression pressure was determined for each test by the second method mentioned above, and the values measured from the diagrams are given in the tables in which the tests are reported.

HORSEPOWER MEASUREMENTS.

The horsepower measurements given are calculated values involving observations and measurements that have already been discussed and the usual unit of power, which is 33,000 foot-pounds per minute.

The brake horsepower, when measured by means of a brake, is the foot-pounds per minute developed at the rim of the brake wheel, divided by 33,000, and is expressed by the equation

$$\text{B. H. P.} = \frac{2\pi lnw}{33{,}000},$$

in which B. H. P. = the brake horsepower or available horsepower developed at the brake wheel;

l = the length of the brake arm in feet (p. 45);

n = the speed of the engine in revolutions per minute (p. 58);

w = the net brake load in pounds (p. 45).

Since the length of the brake arm for the type of brake used is constant for all the tests on each engine, the factor $\frac{2\pi l}{33{,}000}$ is reduced to a decimal fraction (c), which is called the brake horsepower constant, and the values given in the tables were calculated from the equation

B. H. P. = cnw (as defined above.)

In the notes at the bottom of each complete table of tests is given the value of the brake horsepower constant for each engine used, so that with the values of n and w that are given in the table the brake horsepower for each test can be checked.

For the tests reported, the speed (n) of the engine used was practically constant for any one test, but for tests at different loads and in some cases for different tests at the same loads, for which the time of ignition and quality of the explosive mixture were changed, the speed varied to such an extent as to require a separate measurement for each test. With the types of brakes used and with the care that was taken in keeping the scale beam balanced, the load (w) was practically constant during each test.

The indicated horsepower is the foot-pounds per minute developed behind the piston, divided by 33,000, and, when measured by means of indicator diagrams, is expressed by the equation

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

in which I. H. P. = the indicated horsepower, or horsepower developed in the cylinder;

P = the average mean effective pressure in pounds per square inch (p. 59);

L = the length of the stroke of the piston, in feet;

A = the area of cross section of the cylinder bore in square inches;

N = the number of explosions per minute (p. 59).

Since for each engine L and A are constant, the factor $\frac{LA}{33{,}000}$ was reduced to a decimal fraction (C), which is called the indicated horsepower constant, and, except where otherwise noted, the indicated horsepower values given in the tables were calculated from the equation

$$\text{I. H. P.} = CNP \text{ (as defined above).}$$

In the notes accompanying each complete table of tests are given the values of the indicated horsepower constants for the engine on which the tests were made, so that with the values of N and P that are given in the table the indicated horsepower for each test can be checked.[a]

The average values of P and N for individual tests vary with the conditions of operation, such as speed, load, time of ignition, and mixture quality, and hence are determined for each test.

In some cases the indicated horsepower was calculated from the brake horsepower by adding a constant which was equal to the average friction loss or frictional horsepower of the engine as previously determined (p. 73), in which case the indicated horsepower was calculated from the equation

$$\text{I. H. P.} = \text{B. H. P.} + k.$$

in which k = average frictional horsepower.

The average value of k for the five engines was 2.3 horsepower, but the value of k for the particular engine on which the tests were made was usually used (Table 2, pp. 76–77). The accuracy of the indicated horsepower determinations is estimated to be as great as when determined from the indicator diagrams.

DURATION OF TESTS.

The duration of the tests is given in round numbers and represents the approximate time interval between the making of the first and last observations involved in the fuel consumption determinations given. The time interval was made such that the accuracy of the fuel measurements was of as high a degree as could be obtained for the power measurements. The fuel measurements for trial tests, which were of about 15 minutes duration, were found to be sufficiently accurate for all practical purposes, but the duration of the final and check tests was extended to about 30 minutes in order to assure as high a degree of accuracy as possible, both in the fuel measurements and average power measurements. The duration of a test does not include the time required to adjust the engine and conditions of operation between tests or to make incidental measurements, such as compression pressure, nor does it include the time interval that was always allowed in order that the engine might reach a steady running condition between the time at which the above-mentioned adjustments were completed and the time at which the first regular observation of the test was made.

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[a] The dimensions A and L of the engines are given in round numbers which differ slightly from the exact measurements. The indicated horsepower constants given are calculated from the exact measurements.

FUEL CONSUMPTION.

Under fuel consumption is given the weight of fuel consumed in pounds per hour, pounds per charge, and pounds per brake horsepower and per indicated horsepower per hour, the volume of fuel consumed in gallons per brake horsepower and per indicated horsepower per hour, and also the heat value of the fuel consumed in British thermal units per brake horsepower and indicated horsepower per hour. The weight of fuel consumed per hour was calculated from observations made during the tests, and the remainder of the items were calculated from this determination and from other test data or calculated results that have already been discussed.

The quantity of fuel consumed per hour was determined in the following manner:

After the required adjustments for tests had been made and the engine had reached a steady running condition, the small fuel-supply tank on the spring balance scales (p. 46) was refilled. A sufficient number of small weights was then placed on the platform of the scales to cause the pointer to indicate a weight slightly in excess of the nearest even pound mark, which was noted and designated as w_1. The movement of the pointer was then watched, and at the instant sufficient fuel was used to cause it to rest squarely at the above even pound mark, the time was noted and designated as t_1. Toward the end of the test the movement of the pointer was again noted and when it reached a pound mark on the dial, which was designated as w_2, the time was again noted and designated as t_2. Hence a weight of fuel used during the time interval t_2-t_1 was w_1-w_2, and the weight of fuel consumed per hour was

$$W=\frac{w_1-w_2}{t_2-t_1}\times 60,$$

provided t_2-t_1 was expressed in minutes.

The pounds of fuel consumed per charge was determined by dividing the weight of fuel consumed per hour (W) by 60 times the number of fuel admissions per minute as recorded by the continuous counters (pp. 48 and 58).

The pounds of fuel consumed per brake horsepower per hour $=\dfrac{W}{\text{B. H. P.}}$

The pounds of fuel consumed per indicated horsepower per hour $=\dfrac{W}{\text{I. H. P.}}$

The gallons of fuel consumed per brake horsepower per hour refers to the gallons of fuel at 60° F. and was calculated from the equation:

Gallons per brake horsepower per hour $=\dfrac{W}{8.34 \times S \times B.\ H.\ P.}$,

in which W = weight of fuel consumed per hour.

8.34 = weight of 1 gallon of water at 60° F.;

S = specific gravity of the fuel at 60° F.;

B. H. P. = the brake horsepower (p. 68).

The specific gravity of the fuel used for each series of tests is given in the notes at the bottom of the table in which they appear.

The gallons of fuel consumed per indicated horsepower per hour $=\dfrac{W}{8.34 \times S \times I.\ H.\ P.}$, the terms being as defined above.

The British thermal units in the fuel consumed per brake horsepower per hour $=\dfrac{H \times W}{B.\ H.\ P.}$ in which H is the heating value of the fuel in British thermal units per pound as determined in a Mahler bomb calorimeter (p. 26). Except where otherwise noted, the low heating value (p. 8) was used.

The British thermal units in the fuel consumed per indicated horsepower per hour $=\dfrac{H \times W}{I.\ H.\ P.}$, the terms of which are defined above.

MECHANICAL AND THERMAL EFFICIENCIES.

In general two kinds of efficiencies are given for each test reported in the complete tables, namely, mechanical efficiency, or the efficiency of the mechanism in transmitting the power developed in the cylinder to the driving shaft or wheel of the engine, and the heat efficiency (thermal efficiency), or the efficiency of the engine in transforming the heat energy of the fuel into mechanical energy. The thermal efficiency is of two kinds: the percentage of the heat energy in the fuel that is converted into mechanical energy behind the piston (indicated thermal efficiency), and the percentage of the heat energy in the fuel that is converted into useful work at the end of the crank shaft (brake thermal efficiency). All the efficiencies given are calculated values and with few exceptions are calculated from observations or calculated results previously discussed. The exceptions are indicated thermal efficiencies, calculated from empirical formulas, as explained in the notes accompanying the table in which they appear.

The mechanical efficiency, except as noted below, was calculated from the equation

$$E=\frac{B.\ H.\ P.}{I.\ H.\ P.} \times 100,$$

in which E = the percentage of the horsepower developed in the cylinder that is available at the end of the crank shaft;

B. H. P. = the brake horsepower as measured by means of a brake (p. 68);

I. H. P. = the indicated horsepower as determined by the indicator diagrams (p. 68).

For a few tests, as noted in the tables in which they are reported, the mechanical efficiency was calculated from the equation

$$E=\frac{\text{B. H. P.}}{\text{B. H. P.}+k}\times 100,$$

in which k = the average friction loss or frictional horsepower of the engine as described under "Mechanical Efficiency" (pp. 73 to 81), or the mechanical efficiency is read directly from mechanical efficiency *vs.* load curve given under "Mechanical Efficiency" (p. 79).

The indicated thermal efficiency is expressed by the equation

$$E=\frac{2545}{W\times H}\times 100,$$

in which E = the indicated thermal efficiency in percentage as defined above and calculated from measurements given in the test data;

2545 = the heat equivalent of 1 horsepower in British thermal units;

W = the weight of fuel consumed per I. H. P. hour in pounds (p. 71);

H = the heating value of the fuel in British thermal units per pound (Table 1, pp. 18 to 21).

The brake thermal efficiency is expressed by the equation

$$E=\frac{2545}{W\times \text{H}}\times 100,$$

in which E = the brake thermal efficiency in percentage as defined above and calculated from the test data given; and

W = the pounds of fuel consumed per B. H. P. hour, the only factor that differs from those used to calculate the indicated thermal efficiency as given above.

In the notes at the bottom of each table are given both the high and the low heating value of the fuel used for the tests reported, so that efficiencies given may be checked or calculated using the high value if desired.

WEIGHT OF JACKET WATER USED PER HOUR.

While adjusting the engine for each regular test, the flow of cooling water in the jackets of the engine cylinder and cylinder head was regulated by means of a globe valve in the supply pipe, and was increased or decreased until the exit temperature T_w'' was constant at the desired degree. The water was then turned into the weighing tank (p. 52), the quantity of water used during the test was weighed, and the rate of flow in pounds per hour calculated was from the following equation:

$$W_w=\frac{w''-w'}{t''-t'}\times 60,$$

in which W_w = the weight of jacket water per hour;

w' = the total weight of the tank and water in pounds at the time t' taken at the beginning of the test, in hours, minutes, and seconds, reduced to minutes; and

w'' = the total weight of the tank and water in pounds at the time t'' taken at the end of the test, in hours, minutes, and seconds, reduced to minutes.

HEAT CARRIED AWAY BY JACKET WATER.

The percentage of the heat in the fuel supplied that is carried off by the cooling water (E_j) is obtained by dividing the weight of fuel supplied per hour (W, page 70), multiplied by its low heating value (H, Table 1), into the weight of cooling water per hour (W_w, page 72), multiplied by the difference between its outlet and inlet temperatures ($T_{w''} - T_{w'}$, page 72), and is expressed by the equation

$$E_j = W_w \frac{(T_{w''} - T_{w'})}{W_f \times H} \times 100.$$

DEDUCTIONS FROM TESTS.

MECHANICAL EFFICIENCY.

The mechanical efficiency of an engine is the ratio of the useful or available horsepower developed at the crank shaft or drive pulley to the horsepower developed in the cylinder. As given in this report, it is the brake horsepower (p. 68) divided by the indicated horsepower (p. 68), and is usually expressed by the equation $E = \frac{\text{B.H.P.}}{\text{I.H.P.}}$ The difference between these two horsepower measurements is the power required to overcome any loss, by leakage, of a portion of the energy which is received by the charge during compression, and also the power required to overcome the mechanical and fluid frictions. In other words, it is the power required to drive the engine itself, and is generally called the frictional horsepower (I. H. P. − B. H. P. = F. H. P.).

FACTORS AFFECTING MECHANICAL EFFICIENCY.

The loss of energy, during compression, is dependent upon the fit of the piston, the lubrication, the condition of the valve seats, and the pressures produced in the cylinder by the compression and combustion of the charge.

The mechanical or bearing friction of an engine, when running at constant speed, is affected largely by the design of the mechanism, by the condition and adjustment of the bearings, by the fit of the piston, and by the lubrication of the bearing surfaces. The mechanical friction may also depend to a certain extent on the load, but is usually considered practically constant for all loads or horsepowers developed.

By the fluid friction is meant the friction of the gases in the passageways during intake and exhaust, together with the windage. The friction of the gases in the passageways during intake causes a partial vacuum and during exhaust causes a positive pressure in the cylinder, so that during the suction and exhaust strokes the piston is moved against a small resistance. The amount of this resistance depends upon the relative size of the passageways and of the cylinder, upon the lift and timing of the valves, upon the speed of the engine, and upon any conditions of installation or operation that affect the amount of the throttling of the charge, or the amount of restriction of the exhaust gases in their escape to the atmosphere. Hence for a given engine that is governed for constant speed, the friction of the gases in the passageways varies only with the last-mentioned conditions of installation or operation. The windage is mainly friction of the air against the flywheels and hence varies with the number, the size, and the type of flywheels with which the engine is equipped and is dependent upon the speed at which the engine is run. The windage is usually small and is, of course, constant for an engine that is governed for constant speed.

As the two Otto gasoline engines were governed by the hit-or-miss method, which affects the supply of liquid fuel only, the fluid frictions of these engines can reasonably be considered to have been constant and equal.

The fluid friction of the Nash engines may have varied, however, with the carburetor used and with the setting of the auxiliary air valve, both of which affected the throttling of the charge. With these engines the action of the governor, which in general increases the number of cut-outs as the load decreases, may affect the fluid friction also because the air taken in during cut-outs is admitted through the auxiliary air valve only, and hence is more or less throttled by it. However, in many cases during this investigation, the fluid friction was made to vary for constant load or kept constant for varying loads by means of changes in the strength of the explosive mixture with which the engine was supplied, and therefore no uniform variation in the fluid friction of the engines could be expected.

With the throttling method of governing as employed by the 15-horsepower Otto alcohol engine, the fluid resistance caused during the suction stroke by the throttling or restricting action of the governor-controlled valve, will increase with a decrease in load until it may constitute a large percentage of the total engine friction at light load, provided the mixture quality remains constant. If, on the other hand, the construction or adjustment of the carburetor is such that increased suction resistance causes a resultant weakening of the mixture, it then requires little throttling to maintain constant speed even at no load, and the consequent fluid-friction loss may be correspondingly small. If the main inlet valve is used to throttle the charge and both its lift and timing are controlled by the governor, a combined throttling and cut-off effect may be obtained which will greatly reduce fluid resistance during the suction stroke for light loads.

A modification of the throttling method of governing, which combined the preceding method and an adjustment of the carburetor for weaker mixtures with reduced load was used in connection with many of the tests on the 15-horsepower Otto alcohol engine, with a further reduction of the fluid-friction losses. This modification gave surprisingly good fuel economies, as will be brought out later, but makes impossible any conclusion as to the effect of load on the mechanical efficiency of the engine.

METHODS OF TESTING.

The five engines used in connection with this investigation were each put in as nearly perfect running condition as possible, with reference to friction and leakage losses, before any of the regular tests were made on them. Furthermore, the lubrication, the bearing adjustments, and the condition of the valves, the valve seats, and the piston were given special attention during the entire time the engines were in use. Whenever there was any doubt as to the constancy of the mechanical condition of the engine, special determinations of the least rate of fuel consumption were often made immediately preceding and following a series of tests. When any cause for an undue variation in mechanical efficiency was found, it was remedied and all of the tests that were affected were repeated.

After three months of continuous running, it was found not only that the least rate of fuel consumption for any one of the engines was the same as when first adjusted for the tests, but also that this least rate of fuel consumption was obtained with practically the same engine settings. Hence disregarding changes in load, it is probable that for each engine the conditions affecting the mechanical friction were constant and the leakage loss was negligible.

RESULTS OBTAINED.

The general variation in the mechanical and fluid friction of the different engines with changes in load and other conditions of operation may be followed to some extent by a study of the fuel-economy tests which are given in full in the body of this bulletin. In order, however, to determine to what extent the total friction loss may be considered constant for the five different engines of approximately the same size and speed on which tests were made, and to afford a means of checking the indicated horsepower from the brake horsepower, as well as calculating the indicated horsepower when indicator diagrams were not taken, a compilation was made of the combined results of all the tests on each engine for which the indicated horsepower was calculated, and is presented in Table 2. The averages are weighed, in each case, according to the number of tests used.

TABLE 2.—*Results of tests of mechanical efficiency versus load.*

Number of tests averaged.	Speed (average per minute). Revolutions.	Explosions.	Fuel cut-outs.	Mean effective pressure (pounds per sq. in.).	Brake load (pounds).	Horsepower. Brake.	Indicated.	Frictional.[b]	Mechanical efficiency. $\frac{\text{B. H. P.}}{\text{I. H. P.}}\times 100$.	$\frac{\text{B. H. P.}}{\text{B. H. P.}+\text{Av. F. H. P.}}\times 100$.[c]
OTTO 15-HORSEPOWER GASOLINE ENGINE ($6\frac{3}{4}$ BY $15\frac{1}{2}$ INCH) NO. 1.[a]										
8	254.4	123.4	4	111.9	140	17.16	19.34	2.18	88.8	*88.3*
8	255.1	123.7	4	105.7	130	15.98	18.30	2.32	87.3	*87.6*
3	255.7	124.8	3	101.3	125	15.40	17.70	2.30	87.0	*87.1*
9	256.1	122.2	6	99.3	120	14.80	16.90	2.10	87.5	*86.7*
53	256.0	123.6	4	93.9	115	14.18	16.22	2.04	87.4	*86.2*
62	254.2	117.0	10	97.7	110	13.66	15.87	2.21	86.1	*85.8*
14	257.9	116.4	13	89.1	100	12.41	14.70	2.29	84.4	*84.6*
19	259.6	102.7	27	87.3	80	9.98	12.29	2.31	81.4	*81.5*
34	262.9	79.6	52	94.9	60	7.60	10.18	2.58	75.3	*77.0*
35	264.1	55.2	77	99.5	40	5.09	7.49	2.40	68.0	*69.2*
245				Average frictional horsepower for 245 tests=2.27						
OTTO 15-HORSEPOWER GASOLINE ENGINE ($6\frac{3}{4}$ BY $15\frac{1}{2}$ INCH) No. 2.[a]										
2	250.0	119.5	6	110.4	150	16.85	18.43	1.58	91.6	*87.5*
4	251.8	123.3	3	105.5	135	15.28	18.18	2.90	84.1	*86.4*
13	253.2	121.7	5	97.4	130	14.78	16.51	1.73	89.5	*86.0*
41	254.2	110.2	17	102.4	115	13.13	15.61	2.48	84.3	*84.5*
7	254.1	109.1	17	92.7	100	11.42	14.04	2.62	81.4	*82.6*
7	257.1	86.7	42	88.8	70	8.09	10.70	2.61	75.6	*77.1*
7	260.0	62.0	64	86.2	40	4.67	7.43	2.76	63.5	*66.0*
81				Average frictional horsepower for 81 tests=2.41						
NASH 10-HORSEPOWER GASOLINE ENGINE ($7\frac{5}{8}$ BY 10 INCH).[d]										
(*f*)	303.1	145.9	6	*119.1*	145	17.36	*19.75*	....	....	*88.0*
(*f*)	306.0	146.1	7	*114.5*	140	16.91	*19.30*	....	....	*87.6*
2	306.0	145.7	7	105.6	125	15.10	17.74	2.64	85.2	*86.4*
1	297.0	147.4	1	92.7	117.5	13.78	15.75	1.97	87.5	*85.2*
3	304.2	146.4	6	88.2	116.5	14.00	15.04	1.04	91.6	*85.4*
1	300.0	145.6	4	96.0	108	12.80	16.11	3.31	79.5	*84.3*
17	297.1	143.5	5	84.0	100	11.73	13.89	2.16	84.5	*83.1*
5	297.6	139.0	10	86.4	96	11.28	13.82	2.54	81.6	*82.6*
9	299.6	140.8	9	77.8	91	10.76	12.58	1.82	85.6	*81.9*
5	295.7	144.5	3	79.2	90	10.51	13.19	2.68	79.7	*81.5*
3	296.0	144.0	4	73.5	87	10.17	12.14	1.97	83.8	*81.0*
34	296.2	141.2	7	74.1	80	9.36	12.05	2.69	77.8	*79.7*
3	297.4	140.9	8	71.8	70	8.22	10.78	2.56	76.3	*77.5*
2	306.4	123.7	30	71.6	66	7.99	9.87	1.88	81.0	*77.0*
7	298.8	130.9	19	64.1	60	7.08	9.55	2.47	74.5	*74.8*
1	306.8	114.5	39	62.1	50	6.06	8.20	2.14	73.9	*71.7*
11	299.1	107.6	42	60.3	40	4.72	7.25	2.53	65.9	*66.4*
104				Average frictional horsepower for 104 tests=2.39						

TABLE 2.—*Results of tests of mechanical efficiency versus load*—Continued.

NASH 10-HORSEPOWER ALCOHOL ENGINE (7⅝ BY 10 INCH).[d]

Number of tests averaged.	Speed (average per minute).			Mean effective pressure (pounds per sq. in.).	Brake load (pounds).	Horsepower.			Mechanical efficiency.	
	Revolutions.	Explosions.	Fuel cut-outs.			Brake.	Indicated.	Frictional.[b]	$\frac{\text{B. H. P.}}{\text{I. H. P.}}\times 100$.	$\frac{\text{B. H. P.}}{\text{B. H. P.}+\text{Av. F. H. P.}}\times 100$.[c]
(f)	300.1	144.0	6	*111.9*	140	16.60	*18.60*	....	....	*89.3*
(f)	301.5	150.5	0	*104.1*	135	16.08	*18.08*	....	....	*89.0*
(f)	301.8	150.9	0	*97.8*	126	15.03	*17.03*	....	....	*88.3*
1	301.6	147.6	3	101.4	124	14.78	17.25	2.47	85.7	*88.1*
4	301.2	143.3	7	90.0	116.5	13.86	14.83	.97	93.6	*87.5*
4	300.9	147.0	3	88.5	110	13.08	15.00	1.92	87.3	*86.8*
1	303.1	146.1	6	86.0	100	11.98	14.50	2.52	82.7	*86.7*
4	302.6	133.0	19	78.7	85	10.16	11.95	1.79	85.2	*83.6*
1	305.9	135.9	17	81.4	83	10.02	12.75	2.73	78.6	*83.4*
3	303.0	142.2	9	70.4	75	8.94	11.53	2.59	77.9	*81.8*
1	306.7	124.4	29	73.2	66	8.00	10.50	2.50	76.2	*80.1*
3	303.9	111.1	41	72.8	59	7.09	8.87	1.78	80.0	*78.1*
3	307.0	123.1	32	61.2	50	6.06	8.64	2.58	70.1	*75.3*
25				Average frictional horsepower for 25 tests=2.00						

OTTO 15-HORSEPOWER ALCOHOL ENGINE (6¾ BY 15½ INCH).[e]

Number of tests averaged.	Revolutions.	Explosions.	Fuel cut-outs.	Mean effective pressure (pounds per sq. in.).	Brake load (pounds).	Brake.	Indicated.	Frictional.	$\frac{\text{B. H. P.}}{\text{I. H. P.}}\times 100$.	$\frac{\text{B. H. P.}}{\text{B. H. P.}+\text{Av. F. H. P.}}\times 100$.
(f)	260.0	130.0	0	*135.0*	190	22.12	*24.50*	....	....	*90.3*
1	258.4	129.0	0	120.0	170	19.09	21.70	2.01	90.9	*89.2*
2	258.6	129.5	0	105.0	148	17.15	18.95	1.80	90.5	*87.8*
4	258.6	129.3	0	95.1	130	15.06	17.18	2.12	87.7	*86.4*
8	259.9	130.0	0	85.9	115	13.39	15.59	2.20	85.9	*84.9*
6	259.5	128.5	1	79.3	100	11.62	14.27	2.65	81.6	*83.0*
4	260.9	130.8	0	59.7	70	8.19	10.85	2.66	75.4	*77.6*
4	261.8	122.8	8	44.0	40	4.69	7.40	2.71	63.3	*66.3*
29				Average frictional horsepower for 29 tests=2.38						

Combined results for the five engines (average frictional horsepower for 484 tests=2.31).

a Horizontal engine; hit-or-miss method of governing, acting on the supply of liquid fuel; governor controls a mechanically operated valve which admits or rejects the total charge of liquid fuel to the carburetor. The regular supply of air passes through the engine unaffected by the governor.

b Frictional horsepower or friction loss=indicated horsepower minus brake horsepower.

c Results calculated from average indicated horsepower values of the engine designated are printed in italics.

d Vertical engine, cylinder offset; hit-or-miss method of governing, acting on a supersaturated supply of air and fuel mixture; governor controls the mechanically operated valve, admitting or rejecting the portion of the air charge which passes through the carburetor and carries the total fuel charge with it, but does not affect the auxiliary air supply which is admitted uninterruptedly.

e Horizontal engine; combination of throttle and cut-off methods of governing; governor controls the lift and timing of the main inlet valve.

f Indicated horsepower could not be accurately determined from the indicator diagrams, which were obviously in error from coincident change in compression and leakage or effect of too long a passageway from the cylinder to the indicator. The indicated horsepower was calculated by adding the numerical value of the average frictional horsepower obtained from the remaining tests (as given in the table for the engine designated) to the values obtained for the brake horsepower. The mean effective pressure was also calculated by using the calculated indicated horsepower and the observed number of explosions per minute.

In connection with the table is also given a curve, figure 16, showing the relation between the brake horsepower developed and the average mechanical efficiency of the five engines on which the tests were made. The curve was platted from the equation

$$E = \frac{B.\,H.\,P.}{B.\,H.\,P. + 2.3}$$

in which E = mechanical efficiency; B. H. P. = brake horsepower, and the constant 2.3 = the average frictional horsepower or the difference between the average brake and indicated horsepower measurements for the 484 tests as given in Table 2.

CONCLUSIONS.

From a study of the table and curve it will be seen that the two mechanical efficiencies as obtained, first, from the indicated horsepower and brake horsepower, and second, from the frictional horsepower and brake horsepower, show that the original assumption regarding the constancy of the frictional horsepower must have been practically correct. In fact, such irregularities as are shown by the above comparison are well within the probable error of observation, which is less than 5 per cent for any individual test. It is therefore safe to say that the theoretical efficiencies obtained by using the curve (fig. 16) are more correct than the calculated efficiencies, because of the fact that the probable error of individual tests is reduced when a number of tests are averaged.

While there is a slight variation in the frictional horsepower for different loads on the same engine, it does not seem to bear any particular relation to the load, with the possible exception of the 15-horsepower Otto alcohol engine. This engine was run with exceedingly high compression, and this fact may have been responsible for errors of the indicator in recording the mean effective pressure; if so, the true friction loss of this engine is more nearly constant than is indicated by the results given in the table.

The difference between the average friction loss (Table 2) of the two Nash engines is not easy to account for. The engines were identical in size and construction with the exception of the degree of compression, which was made greater in the alcohol engine by changing the thickness of the cylinder head in such a way as partly to fill the original clearance space. In the case of each of these engines the pressure in the cylinder was transmitted to the indicator through a $\frac{1}{2}$-inch hole in the cylinder head. The increase in the thickness of the cylinder head of the alcohol engine thus lengthened the passageway to the indicator as well as increased the compression, but these changes were probably not sufficient to cause the difference noted. It is more probable that there were other influencing conditions

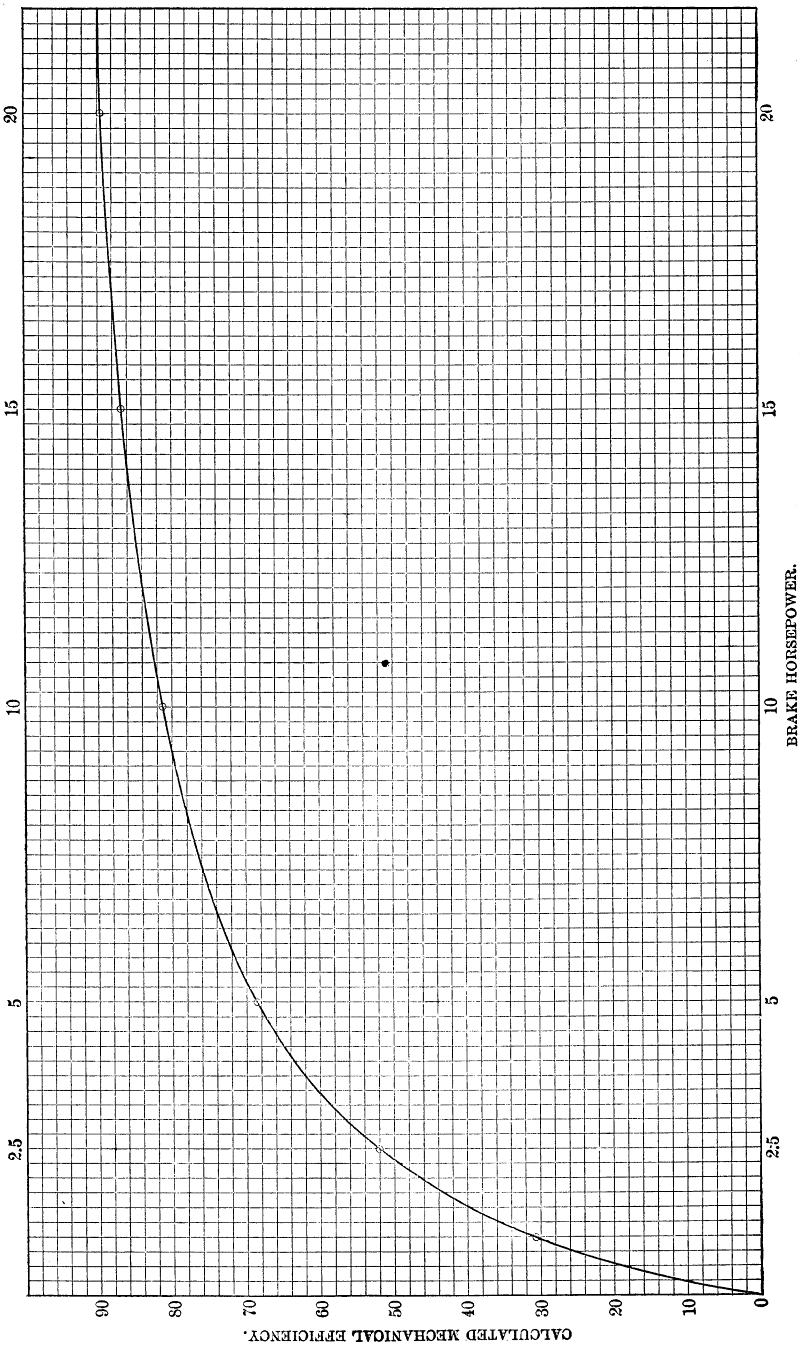

FIGURE 16.—Curve showing relation between brake horsepower and mechanical efficiency of the five engines on which tests were made.

which combined to cause these differences; for example, the operation of the Nash alcohol engine was very unstable with the settings that were used for most of the tests. Because of this unstable operation considerable trouble was experienced in obtaining representative indicator diagrams from which a fair average mean effective pressure could be calculated.

The greatest value of the table and the curve is that they afford an easy means by which the mean effective pressure, indicated horsepower, and dependent results of the tests presented in this report may be checked, corrected, or calculated when not given. This process is offered as a suggestion to those who may desire to work out theoretical deductions and carry the interpretation of certain series of tests into more specific fields than is possible in the scope of this bulletin.

Taken as a whole, the results may be of assistance in establishing an easier, and possibly more accurate, method than that in general use for determining the mechanical efficiency of engines on which an extended series of tests is desired.

It has frequently been suggested that for large engines, where it is difficult or impracticable to obtain the brake horsepower, an indicated-horsepower determination at no load is sufficient to establish the friction loss of the engine. Attention should be called, however, to the fact that under ordinary circumstances it is difficult to make an accurate determination of the indicated horsepower at no load, because the disturbing effect of the governor is frequently such as to produce a widely varying and irregular mean effective pressure for successive explosions. When the throttling method of governing is used, the uncertainty as to the extent to which increased fluid friction affects the total friction loss is also added to the difficulty in determining the true indicated horsepower of no load.

It is not necessary nor advisable to make a separate determination of the indicated horsepower for each test when a series of tests is to be made on the same engine. A mechanical-efficiency-load curve may be obtained for the engine by means of a special series of tests made under the most favorable conditions for obtaining uniform indicator diagrams. These tests need only be made of sufficient duration to determine accurately the brake and indicated horsepower for a few different loads from about one-quarter load to maximum load. All subsequent indicated horsepower values could then be calculated from this curve, and indicator cards would need to be taken only if desired as a guide for adjustment or some other purpose.

To what extent this method of obtaining the indicated horsepower and mechanical efficiency of an engine for different loads is applicable in general practice is largely a question of judgment. It is a suggestion, however, which may be the means of saving a great deal of time and troublesome work in taking and integrating indicator diagrams and it seems entirely feasible in the light of results obtained throughout this investigation.

MIXTURE QUALITY.

The ratio of the weight of air to the corresponding weight of gasoline or denatured-alcohol vapors which will form an explosive mixture or one that, when ignited, will burn by self-propagation of the flame, may be said for the purpose of this discussion to be the mixture quality. This so-called mixture quality may vary between wide limits, and the series of tests here outlined has aimed to determine to what extent and in what manner obtainable variations in mixture quality affect the efficiency with which the fuels can be used in engine cylinders.

PROBABLE EFFECT ON EFFICIENCY.

Fundamentally, the efficiency with which the fuels can be used may be expected to be affected by the mixture quality in two ways: (1) By changes in the completeness of combustion and (2) by changes in the rate of burning.

It must be admitted that both the completeness of combustion and the rate of burning are affected not only by mixture quality, but probably by other things, such as pressure, temperature, unscavenged exhaust gases remaining in the cylinder during the working stroke and hence becoming a part of the mixture, and the homogeneity of the whole explosive mixture. These conditions, however, can be maintained practically constant by proper manipulation of the engines, and the results obtained from the series of tests here given are thought to show only slight effects of these disturbing factors, and the conclusion drawn regarding the most advantageous adjustment of mixture quality is considered correct.

Obviously the completeness of combustion has a direct effect upon the efficiency with which the fuel is used, because if combustion is incomplete, the consequence is that a certain amount of the fuel goes on through the engine with the exhaust gases and does not become available for mechanical energy.

The changes in the rate of flame propagation coincident with changes in mixture quality, although affecting also the efficiency with which the fuel is used, do so in a more roundabout manner. If the mixture quality is of such a nature as to be slow-burning (p. 82), and such factors as piston speed and time of ignition combine to make the time interval allowed for the combustion of the explosive mixture within the cylinder comparatively short, then it may so happen that exhaust occurs before the fuel is entirely burned, with the result that much sensible heat of combustion is expended too late in the stroke to be transferred into mechanical energy.

In the case of the slow-burning mixtures, an indication of the loss is always made by the appearance of flame in the exhaust ports, but there is no way to determine, under ordinary operating conditions, whether combustion is complete or not.

METHODS OF TESTING.

In this series of tests it was impossible to equip the engines so that the mixture quality could be measured directly, nor could analyses of the exhaust gases be made in order to determine the change in completeness of combustion with change in mixture quality. Hence, only such tests were made as were necessary to enable the operators to proceed intelligently with other series of tests involving the determination of the most advantageous adjustments of the fuel needle valves.

Keeping all the previously mentioned difficulties constantly in mind, the following procedure of tests was planned and carried out with only such variations as the exigencies of testing demanded because of unstable governor action or other disturbing features that developed during the testing. A brake load was chosen arbitrarily and the time of ignition was so adjusted that a wide range of mixture quality could be obtained without interfering with the satisfactory operation of the engine. The mixture quality was then varied by changing the needle-valve settings, and all other variables, such as speed, load, compression, and time of ignition, were kept constant.

RESULTS OBTAINED.

The results of these series of tests at approximately the rated load of the engines are shown in Table 3, and the results of two series of tests at relatively light loads appear in Table 4. In Table 3 are tabulated two series of tests on the Otto gasoline engine No. 1, one with gasoline and one with denatured alcohol for fuel; also a series of tests with Otto gasoline engine No. 2 with gasoline for fuel. In Table 4 are tabulated two series of tests on the Otto gasoline engine No. 1 at light load, one with gasoline and one with denatured alcohol for fuel.

For convenience in comparison, the fuel consumption from each of the five series of tests is platted against the fuel needle-valve settings in figure 17.

TABLE 3.—*Results of tests of rated-load fuel*

[OTTO 15-HORSEPOWER GASOLINE ENGINES NOS. 1 AND 2.—Compression pressure constant at 70 pounds per square inch above atmosphere; diameter of cylinder, 6¾ inches; stroke, 15½ inches; method of governing, hit-or-miss fuel supply; jacket-water temperature: Inlet, 60° F.; outlet, 105° to 125° F. Curves platted in figures 17 and 19.]

ENGINE No. 1.—FUEL: GASOLINE.[i]

Test number.	Temperatures ° F.			Jacket water per hour (pounds).	Index settings.		Speed (average per minute).			Average pressures (pounds per sq. in.).		Brake load.	
	Air.	Fuel.	Mixture.[a]		Fuel needle valve.[b]	Igniter (degrees).[c]	Revolutions.	Explosions.	Fuel cut-outs.	Mean effective.	Maximum (above atmosphere).	Pounds.	Per cent of maximum.
A 806	81	80	102	....	30	+21	252.0	126	0	89.3	300	110	85
B 809	85	77	108	....	32	+21	256.6	123	5	95.6	440	110	85
C 812	83	85	107	....	34	+21	258.7	120	9	94.2	420	110	85
D 815	78	69	108	....	38	+21	259.3	112	18	101.6	465	110	85
E 818	77	73	108	....	42	+21	259.5	118	12	98.8	460	110	85
F 821	79	79	107	....	46	+21	260.3	118	13	97.0	450	110	85
G 824	80	67	103	....	50	+21	260.8	117	14	98.3	435	110	85
H 827	71	70	90	....	54	+21	260.7	113	18	101.8	440	110	85
I 830	77	76	91	....	58	+21	260.9	110	20	103.4	415	110	85
J 833	80	68	89	....	62	+21	261.3	107	24	107.6	385	110	85
K 836	83	73	88	....	66	+21	260.6	109	21	104.2	370	110	85
L 839	82	76	86	....	70	+21	258.9	112	17	103.0	315	110	85

ENGINE No. 1.—FUEL: DENATURED ALCOHOL.[j]

M 956	80	78	57	....	48	+17	256.8	116	6	100.4	325	110	79
N 959	73	69	55	....	52	+17	254.1	107	12	104.0	350	110	79
O 962	76	72	57	....	58	+17	257.4	104	25	108.2	365	110	79
P 977	81	77	62	....	64	+17	257.6	100	29	111.4	390	110	79
Q 965	77	72	59	....	70	+17	258.4	97	32	115.4	415	110	79
R 968	79	74	59	....	80	+17	258.7	96	33	117.2	430	110	79
S 971	77	76	64	....	90	+17	257.9	96	33	117.4	435	110	79
T 974	80	77	61	....	100	+17	257.8	96	33	117.0	430	110	79

ENGINE No. 2.—FUEL: GASOLINE.[k]

Test number.	Temperatures ° F.			Jacket water per hour (pounds).	Index settings.		Speed (average per minute).			Average pressures (pounds per sq. in.).		Brake load.	
	Exhaust.[h]	Jacket water.			Fuel needle valve.[b]	Igniter (degrees).[c]	Revolutions.	Explosions.	Fuel cut-outs.	Mean effective.	Maximum (above atmosphere).	Pounds.	Per cent of maximum.
		Inlet.	Outlet.										
U 2112	1,015	61	111	....	30	+20	254.2	109	18	98.6	292	115	85
V 2110	995	61	115	1,091	50	+20	255.1	99	29	108.8	330	115	85
W 2109	965	61	116	1,102	70	+20	256.0	94	35	110.3	350	115	85
X 2107	940	61	112	1,125	90	+20	256.1	94	34	111.0	345	115	85
Y 2105	915	61	107	1,161	110	+20	255.5	99	28	107.8	330	115	85
V′ 2087	930	62	106	1,317	50	+20	257.0	102	27	112.9	390	115	85
W′ 2085	900	62	107	1,309	70	+20	257.9	96	33	114.4	410	115	85
X′ 2089	845	62	106	1,302	90	+20	256.1	97	31	116.3	390	115	85
Y′ 2091	805	62	125	845	110	+20	255.2	104	24	112.4	370	115	85
Z′ 2093	800	62	108	919	130	+20	254.9	105	23	109.7	280	115	85

[a] Measured by a thermometer placed in the inlet-valve housing (p. 57).
[b] Opening of the fuel needle valve in hundredths of one revolution (p. 55).
[c] Point of ignition in degrees is given in terms of the crank position; += early or before dead center (p. 55).
[d] Horsepower constants: Engine No. 1, Brake=0.0004817, indicated=0.001400; Engine No. 2, Brake=0.0004493, indicated=0.001396 (p. 67).
[e] Calculated from the indicated horsepower and low heating value of the fuel (p. 72).
[f] Calculated from the brake horsepower and the low heating value of the fuel (p. 72).

economy versus fuel needle valve setting.

[OTTO 15-HORSEPOWER GASOLINE ENGINES NOS. 1 AND 2.—Compression pressure constant at 70 pounds per square inch above atmosphere; diameter of cylinder, 6¾ inches; stroke, 15½ inches; method of governing, hit-or-miss fuel supply; jacket-water temperature: Inlet, 60° F.; outlet, 105° to 125° F.]

ENGINE No. 1.—FUEL: GASOLINE.[i]

Horsepower.[d]		Duration (minutes).	Fuel consumption.								Efficiencies (per cent).			Heat carried away by jacket water (per cent).[g]
			Pounds per hour.	Pounds per charge.	Per brake horse-power-hour.			Per indicated horse-power per hour.			Mechanical.	Thermal.		
Brake.	Indicated.				Pounds.	Gallons.	B. t. u. (low).	Pounds.	Gallons.	B. t. u. (low).		Indicated.[e]	Brake.[f]	
13.35	15.6	32	8.33	0.00110	0.624	0.102	11,940	0.53	0.087	10,180	85.2	25.0	21.3	......
13.59	16.5	32	8.57	.00116	.631	.104	12,060	.52	.085	9,950	82.4	25.6	21.1	......
13.70	15.8	30	8.90	.00124	.650	.107	12,440	.56	.092	10,760	86.7	23.6	20.5	......
13.74	15.9	30	9.86	.00147	.718	.118	13,730	.62	.102	11,840	86.3	21.5	18.5	......
13.75	16.3	32	11.13	.00157	.809	.133	15,470	.68	.112	13,030	84.2	19.5	16.4	......
13.78	16.0	30	12.17	.00172	.883	.145	16,890	.76	.125	14,600	86.4	17.4	15.1	......
13.82	16.1	29	13.26	.00189	.960	.158	18,360	.83	.136	15,800	86.1	16.1	13.9	......
13.81	16.1	32	13.99	.00206	1.013	.166	19,370	.87	.143	16,670	86.0	15.3	13.1	......
13.82	16.0	28	14.78	.00224	1.070	.176	20,480	.93	.153	17,690	86.5	14.4	12.4	......
13.84	16.1	30	15.25	.00238	1.102	.181	21,090	.95	.156	18,150	86.1	14.0	12.1	......
13.80	15.9	29	16.37	.00250	1.186	.195	22,700	1.03	.169	19,640	86.5	13.0	11.2	......
13.17	16.2	28	17.95	.00267	1.309	.215	25,030	1.11	.182	21,220	84.8	12.0	10.2	......

ENGINE No. 1.—FUEL: DENATURED ALCOHOL.[j]

13.60	16.3	33	14.77	0.00200	1.086	0.159	11,340	0.91	0.133	9,500	83.7	26.8	22.4	......
13.46	15.8	32	14.97	.00217	1.112	.162	11,610	.95	.139	9,900	85.1	25.7	21.9	......
13.64	15.8	31	15.27	.00245	1.119	.163	11,680	.97	.142	10,100	86.4	25.2	21.8	......
13.64	15.5	30	16.19	.00270	1.186	.173	12,400	1.04	.152	10,900	87.8	23.4	20.5	......
13.69	15.6	30	16.95	.00291	1.238	.101	12,930	1.09	.159	11,400	87.5	22.4	19.7	......
13.71	15.7	33	18.14	.00315	1.324	.193	13,830	1.16	.169	12,100	87.3	21.1	18.4	......
13.66	15.8	31	19.57	.00340	1.432	.209	14,960	1.24	.181	12,900	86.2	19.7	17.0	......
13.65	15.7	28	21.30	.00370	1.561	.228	16,300	1.36	.199	14,200	86.8	18.0	15.6	......

ENGINE No. 2.—FUEL: GASOLINE.[k]

Horsepower.[d]		Duration (minutes).	Fuel consumption.								Efficiencies (per cent).			Heat carried away by jacket water (per cent).[g]
			Pounds per hour.	Pounds per charge.	Per brake horse-power-hour.			Per indicated horse-power per hour.			Mechanical.	Thermal.		
Brake.	Indicated.				Pounds.	Gallons.	B. t. u. (low).	Pounds.	Gallons.	B. t. u. (low).		Indicated.[e]	Brake.[f]	
13.14	15.0	29	8.37	0.00128	0.637	0.107	12,290	0.56	0.094	10,750	87.4	23.7	20.7	......
13.19	15.0	28	8.57	.00144	.650	.109	12,540	.57	.096	11,040	88.1	23.1	20.3	35.6
13.22	14.5	29	9.28	.00164	.702	.118	13,540	.64	.108	12,390	91.5	20.5	18.8	33.9
13.24	14.5	27	11.05	.00196	.835	.140	16,110	.76	.128	14,680	91.2	17.3	15.8	26.9
13.20	14.9	23	14.30	.00241	1.083	.182	20,890	.96	.161	18,440	88.2	13.8	12.2	19.4
13.28	16.0	28	8.55	.00140	.644	.108	12,420	.53	.090	10,300	82.9	24.7	20.5	34.6
13.32	15.3	28	8.68	.00151	.651	.109	12,560	.57	.095	10,940	87.1	23.3	20.3	35.7
13.24	15.7	26	10.52	.00181	.795	.134	15,340	.67	.112	12,890	84.0	19.7	16.6	28.2
13.19	16.3	27	13.37	.00214	1.014	.170	19,560	.82	.138	15,880	81.2	16.0	13.0	20.7
13.16	16.1	24	16.25	.00258	1.234	.207	23,810	1.01	.170	19,560	82.2	13.0	10.7	13.5

[g] The percentage of the heat in the fuel supplied that is carried away by the cooling water (p. 73).
[h] Measured by a thermocouple placed in the exhaust pipe near the exhaust valve (p. 57).
[i] Specific gravity at 60° F.=0.729. Heating value: High=20,381, low=19,100 B. t. u. per pound.
[j] Specific gravity at 60° F.=0.820. Percentage of alcohol by weight=91.5. Heating value: High=11,561, low=10,447 B. t. u. per pound.
[k] Specific gravity at 60° F.=0.712. Heating value: High=20,581, low=19,292 B. t. u. per pound.

OPERATOR'S NOTES (TABLE 3).

ENGINE NO. 1.—FUEL: GASOLINE.

Test A 806.—Load and time of ignition were so adjusted as to make possible a wide range of fuel needle-valve settings and corresponding mixture qualities. The mixture was the weakest that could be used with this load and time of ignition without reducing the speed and power of the engine. In test the first mixture charge after cutouts occasionally misfired, due seemingly to irregularity in mixture quality which was such that the above charges were sometimes too weak to burn. Ignition not at fault.

Test B 809.—The explosion or combustion of the charges caused a pound. Occasionally this explosion pound was very heavy.

Test C 812.—A heavy pound with each explosion was noted.

Test D 815.—Very heavy explosion pound noted.

Test E 818.—Very heavy explosion pound noted.

Test F 821.—Very heavy explosion pound noted.

Test G 824.—Very heavy explosion pound noted. Exhaust slightly clouded with black smoke.

Test H 827.—Heavy explosion pound. Exhaust slightly smoky.

Test I 830.—Heavy pound from first explosion after cut-outs. Much heavier than from successive explosions. Exhaust slightly smoky.

Test J 833.—Heavy pound from first explosion after cut-outs only. No smoke in exhaust.

Test K 836.—Heavy pound from first explosion after cut-outs. No smoke.

Test L 839.—Heavy pound from first explosion after cut-outs. Exhaust clouded with black smoke.

ENGINE NO. 1.—FUEL: DENATURED ALCOHOL.

Test M 956.—Load and time of ignition were so adjusted as to make it possible to obtain a wide range of fuel needle-valve settings and corresponding mixture qualities. The engine would not carry the load with a smaller supply of fuel than admitted by the fuel needle-valve setting used for this test. The first charge of fuel and air mixture after cut-outs always misfired. Other mixture charges occasionally misfired. See above note for test A 806.

Test N 959.—The first mixture charge after cut-outs misfired frequently, about six per minute.

Tests O 962, P 977, Q 965, and R 968.—Operation of engine seemingly normal in every respect.

Tests S 971 and T 974.—Heavy pounds from explosions were noted.

ENGINE NO. 2.—FUEL: GASOLINE.

Tests U 2112 to Y 2105, inclusive.—Load and time of ignition were so adjusted as to make possible a wide range of fuel needle-valve settings and corresponding mixture qualities. Operation of the engine seemingly normal in every respect.

Tests V′ 2087 to Z′ 2093, inclusive.—Repair to governor mechanism caused a change in the timing of the fuel inlet valve (p. 30). This changed the mixture quality for tests V′ to Y′, inclusive, though fuel needle-valve settings were the same as for the above tests V to Y, inclusive, which hence constitute a separate series as given. Operation of the engine seemingly normal in every respect.

TABLE 4.—*Results of tests of light-load*

[OTTO 15-HORSEPOWER GASOLINE ENGINE NO. 1.—Compression, constant at 70 pounds per square inch above atmosphere; diameter of cylinder, 6¾ inches; stroke, 15½ inches; method of governing, hit-or-miss fuel supply; jacket-water temperature: Inlet, 60° F.; outlet, 112° F. Curves platted in figures 17 and 19.]

Test number.	Temperatures (° F.).			Index settings.		Speeds (average per minute).			Average pressures (pounds per sq. in.).		Brake load.	
	Air.	Fuel.	Mixture.[a]	Fuel needle valve.[b]	Igniter (degrees).[c]	Revolutions.	Explosions.	Fuel cut-outs.	Mean effective.	Maximum (above atmosphere).	Pounds.	Per cent of maximum.
FUEL: GASOLINE.[g]												
A 262	82	87	88	26	+11	263.0	119	8	64.2	150	60	46
B 263	80	86	85	28	+11	261.7	80	26	84.0	215	60	46
C 264	78	85	86	30	+11	261.0	99	31	93.0	255	60	46
D 265	66	65	82	32	+11	263.0	93	38	108.0	350	60	46
E 266	69	74	82	34	+11	263.2	87	45	91.8	285	60	46
F 267	72	79	84	36	+11	263.6	81	51	90.6	270	60	46
G 268	74	82	85	38	+11	264.2	76	56	90.0	275	60	46
H 269	76	84	86	40	+11	264.5	72	60	98.4	310	60	46
I 352	79	75	88	42	+11	264.7	68	64	103.2	330	60	46
J 367	69	75	76	44	+11	263.7	67	65	108.0	315	60	46
K 272	80	83	88	46	+11	264.6	64	69	108.6	385	60	46
FUEL: DENATURED ALCOHOL.[h]												
L 1352	81	78	62	62	+21	263.6	65	67	91.8	225	40	29
M 1353	80	78	62	66	+21	263.5	61	71	94.2	240	40	29
N 1354	79	79	63	70	+21	263.9	56	76	102.0	270	40	29
O 1355	78	79	63	74	+21	264.5	52	80	108.6	320	40	29
P 1356	78	80	63	78	+21	264.9	49	84	106.8	320	40	29
Q 1357	78	80	64	82	+21	265.7	47	86	114.0	360	40	29
R 1361	87	81	65	84	+21	265.9	46	87	114.0	385	40	29
S 1364	79	81	64	86	+21	265.2	46	87	114.1	355	40	29
T 1367	80	82	65	88	+21	265.7	45	88	116.0	395	40	29
U 1359	79	81	66	90	+21	266.4	44	89	117.6	395	40	29
V 1360	79	80	65	94	+21	266.5	43	90	121.8	445	40	29

[a] Measured by a thermometer placed in the inlet-valve housing (p. 57).
[b] Opening of the fuel needle valve in hundredths of one revolution (p. 55).
[c] Point of ignition in degrees is given in terms of the crank position; + = early or before dead center (p. 55).
[d] Horsepower constants: Brake = 0.0004817; indicated = 0.001400 (p. 67).
[e] Calculated from the indicated horsepower and the low heating value of the fuel (p. 72).

OPERATOR'S NOTES (TABLE 4).

FUEL: GASOLINE.

Test A 262.—Many of the first mixture charges after cut-outs misfired, due seemingly to the action of the governor causing irregularities in the quality of the mixture, which sometimes became too weak to burn. Electric igniter not at fault.

Test B 263.—Most of the first mixture charges after cut-outs misfired.

Test C 264.—The first mixture charge after cut-outs misfired nearly every time.

Test D 265.—The first mixture charge after cut-outs misfired occasionally. A light explosion pound was also noted.

Test E 266.—The misfiring was less and the explosion pound greater than for the preceding test.

Test F 267.—The misfiring was very infrequent and the explosion pound was occasionally heavy.

fuel economy versus needle-valve setting.

[OTTO 15-HORSEPOWER GASOLINE ENGINE No. 1.—Compression, constant at 70 pounds per square inch above atmosphere; diameter of cylinder, 6¾ inches; stroke, 15½ inches; method of governing, hit-or-miss fuel supply; jacket-water temperature: Inlet, 60° F.; outlet, 112° F.]

FUEL: GASOLINE.[g]

Horsepower.[d]		Duration (minutes).	Fuel consumption.								Efficiencies (per cent).		
			Pounds per hour.	Pounds per charge.	Per brake horsepower-hour.			Per indicated horse-power hour.			Mechanical.	Thermal.	
Brake.	Indicated.				Pounds.	Gallons.	B. t. u. (low).	Pounds.	Gallons.	B. t. u. (low).		Indicated.[e]	Brake.[f]
7.60	10.7	8	7.35	0.00100	0.967	0.159	18,460	0.69	0.113	13,160	71.2	19.3	13.8
7.56	9.4	14	6.43	.00102	.851	.140	16,250	.68	.112	13,120	80.7	19.4	15.7
7.54	12.9	14	6.39	.00108	.847	.139	16,180	.50	.082	9,450	58.4	26.9	15.7
7.60	14.1	14	6.39	.00115	.840	.138	16,050	.45	.074	8,650	53.9	29.4	15.8
7.61	11.2	14	6.28	.00120	.825	.135	15,750	.56	.092	10,750	68.0	23.7	16.2
7.62	10.3	15	6.07	.00125	.797	.131	15,220	.59	.097	11,250	73.9	22.6	16.7
7.64	9.6	15	5.97	.00131	.781	.128	14,910	.62	.102	11,880	79.7	21.4	17.1
7.64	10.0	15	5.90	.00137	.772	.127	14,750	.59	.097	11,350	76.8	22.4	17.2
7.65	9.8	31	5.78	.00142	.756	.124	14,450	.59	.097	11,240	77.8	22.6	17.6
7.62	10.1	32	5.64	.00140	.740	.122	14,160	.56	.092	10,660	75.3	23.9	18.0
7.64	9.7	16	5.78	.00150	.757	.124	14,460	.60	.099	11,390	78.8	22.3	17.6
FUEL: DENATURED ALCOHOL.[h]													
5.08	8.4	14	10.52	0.00270	2.072	0.302	21,960	1.25	0.183	13,300	60.6	19.1	11.6
5.08	8.0	15	10.17	.00278	2.004	.293	21,220	1.26	.184	13,440	63.4	18.9	12.0
5.09	8.0	15	10.00	.00298	1.967	.287	20,820	1.25	.183	13,240	63.6	19.2	12.2
5.10	7.9	15	9.78	.00313	1.918	.280	20,310	1.24	.181	13,110	64.6	19.4	12.5
5.10	7.3	16	9.53	.00324	1.865	.272	19,790	1.31	.191	13,890	70.2	18.3	12.9
5.12	7.5	16	9.33	.00331	1.822	.266	19,300	1.25	.183	13,250	68.6	19.2	13.2
5.12	7.4	16	9.42	.00341	1.840	.269	19,500	1.28	.187	13,560	69.6	18.8	13.0
5.11	7.3	29	9.36	.00339	1.832	.267	19,400	1.29	.188	13,640	70.3	18.7	13.1
5.12	7.2	33	9.23	.00342	1.804	.263	19,100	1.28	.187	13,500	70.7	18.8	13.3
5.13	7.3	16	9.23	.00350	1.800	.263	19,070	1.30	.190	13,850	70.7	18.4	13.3
5.14	7.4	16	9.63	.00373	1.875	.274	19,860	1.30	.190	13,830	69.6	18.4	12.8

[f] Calculated from the brake horsepower and the low heating value of the fuel (p. 67).
[g] Specific gravity at 60° F.=0.729. Heating value: High=20,381, low=19,100 B. t. u. per pound.
[h] Specific gravity at 60° F.=0.818. Percentage of alcohol by weight, 91.6. Heating value: High=11,685, low=10,595 B. t. u. per pound.

Test G 268.—No misfiring. A heavy pound from each explosion noted.

Test H 269.—The explosion pound was heavier than for the preceding test.

Test I 352.—Heavy pound from all except the first explosion after cut-outs.

Test J 367.—Very heavy pound from second and third explosions after cut-outs.

Test K 272.—When the explosions occurred in sets of two between cut-outs the second explosion produced a heavy pound. When the explosions occurred singly there was no explosion pound.

FUEL: DENATURED ALCOHOL.

Tests L 1352 to V 1360, inclusive.—These tests were selected from a series of trial tests used to determine the least rate of consumption for the load given (p. 70) and cover only a small part of the possible range of mixture qualities. The operation of the engine for these tests was seemingly normal in every respect.

CONCLUSIONS.

The tests whose results are included in Tables 3 and 4 and the curves shown in figure 17, viewed from the standpoint of the purpose for which they were attempted, point to two obvious conclusions. It will be recalled that these tests were attempted solely with the idea of enabling the operators to proceed intelligently with other series of tests involving the determination of the most advantageous adjustments for the mixture quality. The following table shows the engine, method of governing, fuel, compression pressure, table number, and series of tests, for each curve:

Data for curves in figure 17.

Curve No.	Engine.	Method of governing.	Fuel.	Compression pressure.[a]	Table No.	Series of tests.
1	Otto 15-horsepower, gasoline, No. 1.	Hit-or-miss fuel supply.	Gasoline	Constant at 70 pounds.	3	First.
2	.....do	.....do	Denatured alcohol.	.....do	3	Second.
3	Otto 15-horsepower, gasoline No. 2.	.....do	Gasoline	.....do	3	Third.
4	Otto 15-horsepower, gasoline, No. 1.	.....do	.....do	.....do	4	First.
5	.....do	.....do	Denatured alcohol.	.....do	4	Second.

[a] Pounds per square inch above atmosphere. Compression pressure varies with the engine and air-valve setting used. See p. 66.

Generally speaking, reference to the data from tests at rated load shown in Table 3 will show that with either gasoline or alcohol as the fuel, the leanest mixture that can be used without reducing the speed of the engine may be expected to give the greatest thermal efficiency, provided all other variables, such as load, time of ignition, compression, and speed, are kept as nearly constant as operating conditions will permit. It will be observed that the "fuel per charge," which is the true measure of the mixture quality, varies inversely with the thermal efficiency, showing that the conclusion drawn seems to be unaffected by either the individuality of the engine or of the fuel used.

On the other hand, data given in Table 4, which are from tests run at light load, do not permit the application of the conclusion drawn above. The most efficient results were obtained with the mixture which gave the most uniform diagram between cut-outs.

Having arrived at the conclusion already stated, it is natural to look for the causes which combine to give the effects observed. It is known that the richest mixture qualities tend to cause highest temperatures, and it is therefore reasonable to suppose that with an increase in the richness of the mixture to the point at which there is an excess of fuel, there will be an increase in the exhaust and jacket-water losses. These losses, as will be shown later, can not be segregated. A first inspection of the series of tests with gasoline on the Otto gasoline engine No. 2 would indicate that the above conclusion is erroneous. It will be seen that with an increase in fuel per charge (which corresponds to an increase in the richness of the mixture) there is a noticeable dropping off in the temperature of the exhaust gases and in the amount of the jacket-water loss, and, strange to say, there is an accompanying reduction in the thermal efficiency.

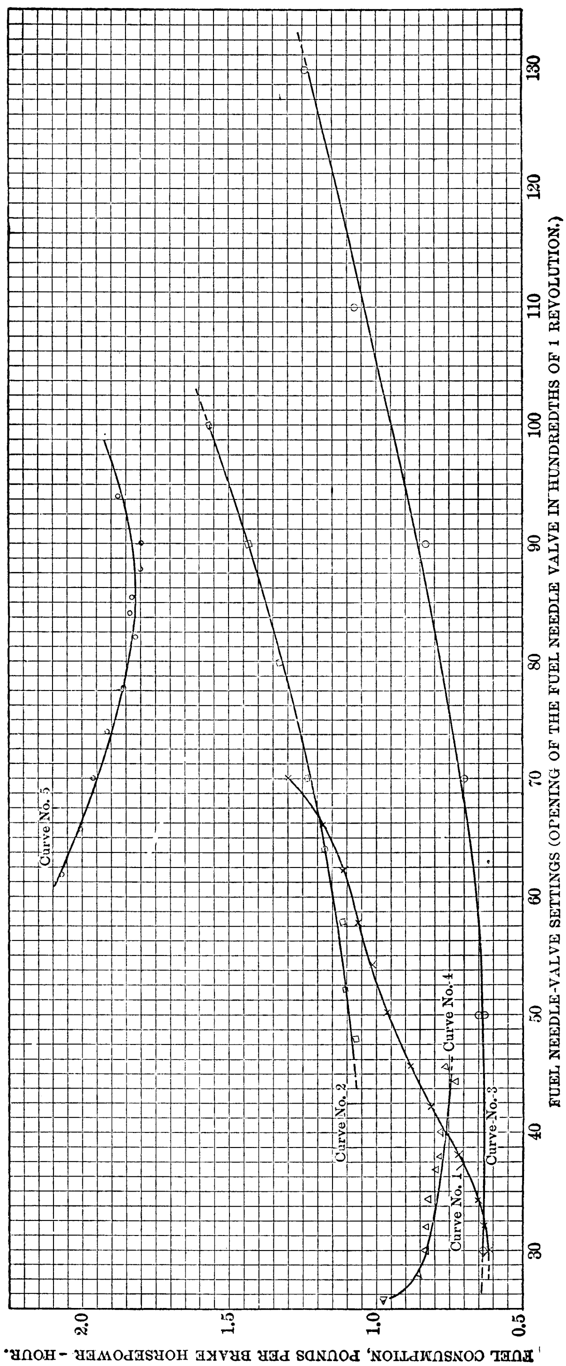

FIGURE 17.—Curves showing the variation of fuel consumption with needle-valve settings.

Unfortunately the only figures obtained relative to the effect of mixture quality upon jacket-water and exhaust losses were obtained on the Otto gasoline engine No. 2, and it is impossible to draw definite conclusions from the results because of changes that were made in the timing of the fuel poppet valve (see operator's notes V′ to Z′, Table 3) in the middle of the series of tests. In practically all the tests on this engine an excess of fuel was used, introducing a loss in the nature of incomplete combustion, which in these tests probably became large enough to account for the other paradoxical condition noted in the tests with richest mixtures; namely, a high efficiency occurring simultaneously with a high exhaust and jacket-water loss.

In the tests at light load it was impossible to keep the engine in operation with as great stability as when operating under normal-load conditions, and therefore the effect of irregular cutting-out became of greater magnitude than any effect which might be expected to occur because of a change in mixture quality.

In conclusion, it must be admitted, generally speaking, that although best economies may be expected from weak mixtures definite statement to that effect can not be made.

DETERMINATION OF TRUE MIXTURE QUALITY.

Under certain conditions, as stated on pages 22 and 57, the quality of the explosive mixture of fuel vapor and air (or ratio of air to fuel) with which the engines are supplied may vary with the setting of the fuel needle valve, but there is no fixed relation between the graduation of the index dial of the different needle valves and their opening. Furthermore, for any one of the fuel needle valves the setting as given in the tables refers to purely arbitrary positions of the valve, and, therefore, one tabular value that is twice as great as another in the abstract does not necessarily mean that for the greater value twice as much fuel is being used as for the lesser value. Again, the tabular values may be the same, but owing to the difference in the air-valve settings or other adjustments affecting the weight of air supplied, the ratio of air to fuel may vary greatly.

The problem then becomes one of obtaining a scheme for finding by means of recorded data a measure of the mixture quality, in order to determine how closely an exact chemical mixture is approached in the cylinder at any time—that is, whether there is an excess of air or an excess of fuel vapor, and the amount of this excess. The following method is employed:

By dividing the fuel consumed per hour by the number of fuel admissions per hour the average weight of fuel that may be said to be allotted to each explosion may be obtained directly, but as the weight of the fuel per charge (p. 70) affects the mixture ratio only through the weight of air with which it is mixed in the cylinder, and as the weight of air can not be readily measured directly, it becomes necessary to ascertain the volume, pressure, and temperature of the charge of air at some time during the compression stroke.

There was, however, found to be a dearth of information regarding the nature of the mixture of fuel vapor and air in the engine cylinder, and therefore figures were obtained (1) on the supposition that the vapor and the air were simply in the form of a mechanical mixture, and (2) on the supposition that they were in the nature of a solution. The results of the two assumptions showed such a slight variation that the more complicated of the two was discarded and all subsequent figures based on the assumption that there was a solution of the vapors in the air. There also arose the question as to whether to base the calculations upon a mixture ratio by volume or by weight, and this question was left unsettled until it developed that the derived equation resulting from the calculation on the weight basis was identical for both gasoline and alcohol and was a form of $xy=\mathrm{K}$, where x = fuel per charge, y = mixture ratio, and K = a constant solely dependent upon piston displacement, volumetric efficiency, and the temperature of the mixture of fuel and air in the cylinder.

The ability to use the same formula for either fuel so simplified matters that this arrangement was at once accepted as final. The computation necessary to the development of the formula was long and somewhat tedious and is omitted here.

As the volume of the charge at atmospheric pressure may be determined quite accurately by means of light-spring indicator diagrams (p. 47), the only remaining uncertainty now becomes the temperature at which the mixture of fuel and air takes place. That temperature depends on three variables as follows: (1) The temperature of the unscavenged exhaust gases remaining in the cylinder from the last working stroke; (2) the temperature of the air (bearing fuel vapors) which is taken into the cylinder and mixed with the above-mentioned exhaust gases; and (3) the latent heat absorbed by the vaporization of the fuel, which is in amount dependent upon the weight of the fuel and which tends to reduce the temperature of the mixture in the cylinder.

The first of these variables was an unknown quantity for most of the tests, but maximum and minimum figures were obtained experimentally and these were used in connection with maximum and minimum values for the other variables to determine the greatest possible and smallest possible values for the temperature of the resulting mixture. These temperatures were found to be approximately 160° F. and 80° F.

From the form of the equation it is seen that the curve is an equilateral hyperbola which is asymptotic to the axes, and hence, for the large values of x that occur when alcohol is the fuel, it makes but little difference in the mixture ratio whether one uses a curve obtained by using either the minimum temperature of 80° or the maximum temperature of 160°. For small values of x, however, which occur with gasoline as fuel, the possible error introduced by the uncertain temperature will not be as high as 5 per cent on either side of an average curve platted on the basis that an average temperature of 120° F. exists. It was found that the volumetric efficiencies of the Otto gasoline engines (which had the same cylinder dimensions) were constant at 94 per cent—that is, the volume of explosive mixture that

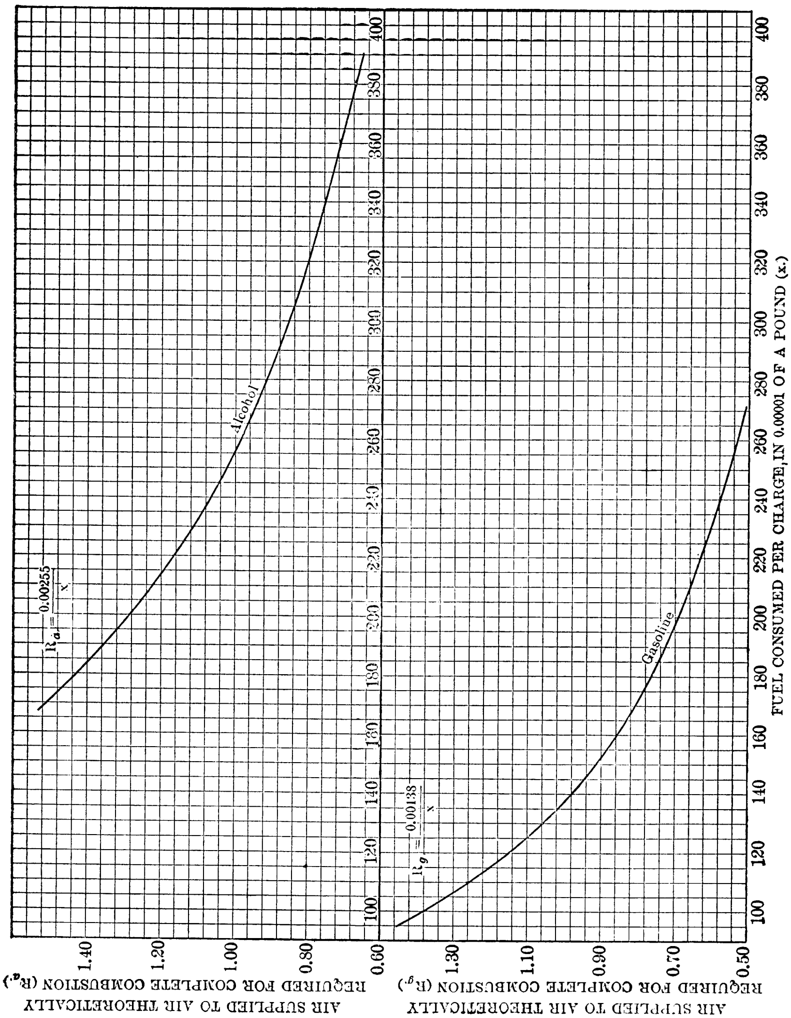

FIGURE 18.—Curves showing relation between air supply and fuel per charge for tests on Otto gasoline engines.

was entrained in the cylinder for each explosion was 94 per cent of the piston displacement. Consequently a value of K for these engines, when a temperature of 120° was assumed to exist, was used and gave the formula the form $xy=0.0207$. Solving for y, $y=\frac{0.0207}{x}$ for the Otto gasoline engines. Or, since the weight of air theoretically required for the complete combustion of 1 pound of gasoline and 1 pound of denatured alcohol (Table 1) is 14.94 and 8.13 pounds, respectively, the ratio of air supplied to air theoretically necessary for complete combustion (R) may be calculated from the equation

$$R_g = \frac{0.00138}{x}$$

for gasoline, and from the equation

$$R_a = \frac{0.00255}{x}$$

for denatured alcohol, if x, the weight of fuel per charge, is known.

Curves obtained by means of the above equations and from which the ratio of air supplied to the air theoretically necessary for complete combustion may be read directly for any of the tests on the Otto gasoline engines are given in figure 18. These curves were

FIGURE 19.—Curves showing variation of fuel consumption with mixture quality.

used in connection with Tables 3 and 4 to determine the data for the curves shown in figure 19.

Although the use of a curve of the form just described will introduce certain errors that seem unavoidable, it will still be of value in explaining engine efficiencies that are not made clear by a study of the needle-valve settings alone, because it will give a clear and true comparison of the mixture ratios used.

CONCLUSIONS.

A study of the curves shown in figure 19 brings out two interesting points:

1. Although no attempt was made to determine the limiting mixture qualities of gasoline and denatured alcohol that were com-

bustible nor the variation of these limiting mixtures in practice, the calculations show that explosive gasoline-and-air mixtures, varying in quality from approximately 1.4 to 0.5 of the theoretical, and denatured alcohol mixtures, varying in quality from approximately 1.3 to 0.7 of the theoretical, were explosive when used in the Otto gasoline engines, for which the compression pressure was about 70 pounds per square inch above atmosphere.

Emphasis should be laid on the fact that, between the limits given, with mixtures of either fuel, the engines were running under seemingly normal conditions as evidenced by the fact that there was no excessive misfiring when the weakest mixtures were used, nor was there smoke in the exhaust or other indication of flooding when the richest mixtures were used.

It should be borne in mind that the figures given do not represent the entire range of combustibility of gasoline and denatured alcohol mixtures, nor do they necessarily represent the entire practical range of combustibility that may be expected in engine practice. They do, however, serve to illustrate the relative variation which may be expected with the two fuels under consideration.

2. An interesting coincidence is observable by a study of curves 1 and 2 shown in figure 19. The curves at their extreme left show the lowest fuel consumptions obtained in this series of tests with gasoline and denatured alcohol, respectively. The curves as platted eliminate the effect of the igniter settings and the fuel needle-valve settings, which by reference to Table 3 are seen to differ greatly from the tests from which the two curves were platted. But, regardless of those differences, the curves show the best fuel consumption to be obtained when the air supply is approximately 1.25 of that theoretically necessary for chemical combination in each case.

The following table shows the engine, method of governing, fuel, compression pressure, table number, and series of tests, for each curve:

Data for curves in figure 19.

Curve No.	Engine.	Method of governing.	Fuel.	Compression pressure.[a]	Table No.	Series of tests.
1	Otto 15-horsepower gasoline No. 1.	Hit-or-miss fuel supply.	Gasoline	Constant at 70 pounds.	3	First.
2	do	do	Denatured alcohol,	do	3	Second.
3	Otto 15-horsepower gasoline No. 2.	do	Gasoline	do	3	Third.
4	Otto 15-horsepower gasoline No. 1.	do	do	do	4	First.
5	do	do	Denatured alcohol.	do	4	Second.

[a] Pounds per square inch above atmosphere. Compression pressure varies with the engine and air-valve setting used. See p. 66.

The results obtained by this method of determining the true mixture are felt to be reliable because of the range of combustibility which it gives, checking very closely for practical conditions with the experimental data given in the section "Gasoline and denatured alcohol as engine fuels" (pp. 16 to 28).

TIME OF IGNITION.

In the operation of an internal-combustion engine governed for constant speed it is entirely possible to ignite the explosive mixture at any point within a considerable range of time without any external indication as to whether the fuel is being used advantageously or not. Depending upon the point within this range at which ignition takes place, widely varying thermal efficiencies may be expected. It therefore seemed desirable to investigate the effect on the fuel economy of the engine of varying the time of ignition of the explosive mixture.

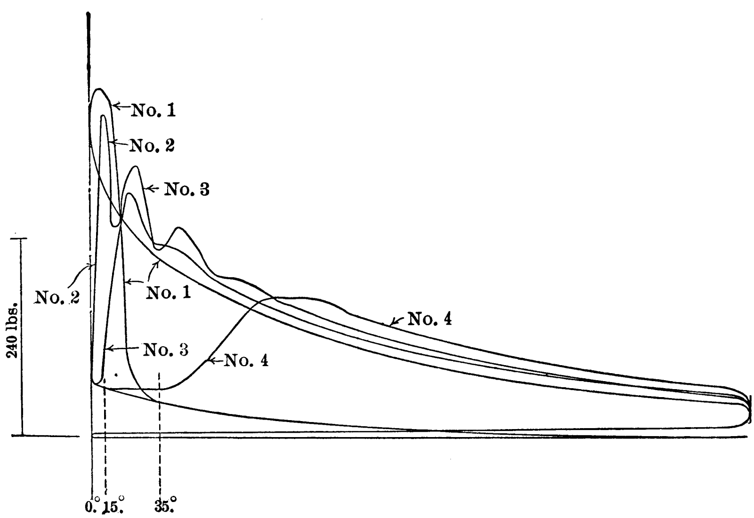

FIG. 20.—Indicator card showing effect of time of ignition on pressure and power developed.

EFFECT ON EFFICIENCY.

The general effect of varying the time of ignition may be studied by referring to the indicator diagrams shown in figure 20. The diagrams were all taken on the same engine, and the time of ignition was varied from 35° in advance of dead center to 15° after dead center, all other conditions being kept constant. Diagram No. 1 was taken with the igniter so adjusted that ignition occurred 35° in advance of dead center. The diagram shows the maximum pressure to be reached before the piston reaches dead center with a consequent negative loop and a corresponding loss in the available horsepower. The opposite extreme of late ignition is shown by diagram No. 4, which was taken with the igniter so adjusted that ignition occurred 15° after dead center. The latter diagram shows practically no available work developed during the first quarter of the expansion stroke. In addition, it can readily be conceived that if such a late ignition were used with a slow-burning mixture combustion would hardly be complete at the time the exhaust valve was opened; hence there would result a large loss in fuel economy, owing to the burning of the explosive mixture after it had left the cylinder. The intermediate diagrams, No. 2 and No. 3, represent much more stable and satisfactory conditions and permit the engine to operate at a considerably increased capacity, as shown by the mean effective pressures given in the table accompanying the diagram.

From these diagrams it is also evident that considerable variation in the thermal efficiency will result from a change in the time of ignition. This variation in thermal efficiency may be due to one or all of several contributing causes. The chief among such causes are: (1) A change in pressure at the time of ignition, depending on how far compression has advanced when combustion commences; (2) a change in the length of time available for the burning of the explosive mixture; (3) a variation in the rate and nature of volume change during combustion. The second of these causes is effective only when the mixture quality, speed, and time of ignition are such that the charge is still burning at the time the exhaust valve is opened. The third cause is probably the most important as the ideal condition of "heat added at constant volume" is never realized. If ignition takes place toward the end of the compression stroke some part, at least, of the explosive mixture is burned while the volume is being decreased, but if ignition takes place at some point in the expansion stroke the charge is expanded while burning. Depending on the variation in the time and duration of burning with respect to the compression and expansion strokes, there will be a variation in the thermal cycle and in the temperatures and pressures developed in the cylinder. These temperatures and pressures will influence the heat and leakage losses.

The "best time of ignition" is largely dependent on both the mixture quality and the speed of the engine. For example, when the time of ignition is constant, the speed of the engine determines the time allowed for combustion of the fuel after ignition takes place and before exhaust occurs; and for any given time of ignition and speed the mixture quality determines its rate of burning so that the amount of burning that takes place during the compression and expansion strokes of the piston is a correlated function of the three variables mentioned:

METHODS OF TESTING.

In order to determine to what extent and in what way varying the time of ignition will vary the thermal efficiency two series of tests, one with gasoline and one with denatured alcohol, were made on the Otto gasoline engine No. 1. Later, a similar series of tests with gasoline was made on the 15-horsepower Otto gasoline engine No. 2. With the information thus obtained, it was hoped that a comparison between the best time of ignition with gasoline and that with denatured alcohol could also be obtained and that some idea regarding the relation of best point of ignition with the same fuel in different engines would be forthcoming.

As usual before beginning a new series of tests the engines were thoroughly inspected to insure that they were in as good running condition as when first adjusted. The calibrations of the igniter adjusting dials were then checked and the engines were ready for the "fuel economy versus time of ignition" tests.

In making tests to determine the effect of the time of ignition upon thermal efficiency it is of course necessary to exclude as far as possible the effect of all other variables and still obtain a wide variation in the time of ignition. To this end, a brake load of a little less than the rated horsepower of the engine was applied and, by means of the fuel needle valve, the mixture quality was so adjusted that the engine cut out frequently when the time of ignition was normal. In this way it was possible to obtain a wide variation in the time of ignition without interfering with the steady running of the engine at its rated speed. The variations in fuel economy for different times of ignition were compensated for by the changes in the proportion of explosions to cut-outs. Having prepared the engines for the tests in the manner described under "Procedure of Tests" (pp. 53 to 55) the igniter was then set for the earliest time of ignition possible without any outward indication of an unstable condition, and a test of sufficient duration to determine the rate of fuel consumption was made. The time of ignition was then retarded in increments of 5 or 10 degrees until the latest time of ignition that could be used without causing the engine to slow down was reached. The rate of fuel consumption was determined for each setting.

RESULTS OBTAINED.

The observed and calculated results obtained from the last-mentioned tests are given in Table 5 and the pertaining operator's notes. The curves in figure 21 (p. 102) illustrate the variations and agreements obtained and are valuable in studying the results of the tests. The following table shows the engine, method of governing, fuel, compression pressure, table number, and series of tests, for each curve:

Data for curves in figure 21.

Curve No.	Engine.	Method of governing.	Fuel.	Compression pressure.[a]	Table No.	Series of tests.
1	Otto 15-horsepower gasoline No. 1.	Hit-or-miss fuel supply.	Gasoline......	Constant at 70 pounds.	5	First.
2	.....do..........	.....do........	Denatured alcohol	.....do..........	5	Second.
3	Otto 15-horsepower gasoline No. 2.	.....do........	Gasoline......	.....do..........	5	Third

[a] Pounds per square inch above atmosphere. Compression pressure varies with the engine and air valve setting used. See p. 66.

TABLE 5.—*Results of tests of fuel*

[OTTO 15-HORSEPOWER GASOLINE ENGINES NOS. 1 AND 2.—Compression pressure, 70 pounds per square inch above 1 atmosphere; cylinder diameter, 6¾ inches; stroke, 15½ inches; hit-or-miss fuel-supply governor; jacket-water temperature: Inlet, 60° F.; outlet, 100° to 125° F. Curves platted in figure 21.]

ENGINE NO. 1.—FUEL: GASOLINE.[h]

Test number.	Temperatures (°F.). Air.	Temperatures (°F.). Fuel.	Temperatures (°F.). Mixture.[a]	Temperatures (°F.). Exhaust.[b]	Index settings. Fuel needle valve.[c]	Index settings. Igniter (degrees).[d]	Speeds, etc. (average per minute). Revolutions.	Speeds, etc. (average per minute). Explosions.	Speeds, etc. (average per minute). Fuel cut-outs.	Average pressures (pounds per sq. in.). Mean effective.	Average pressures (pounds per sq. in.). Maximum (above atmosphere).	Brake load. Pounds.	Brake load. Per cent of maximum.
A 815	78	69	108	....	38	+21	259.3	112	18	101.6	465	110	85
B 842	78	70	101	....	38	+17	258.9	112	18	101.0	430	110	85
C 845	71	70	100	....	38	+13	259.8	107	23	103.7	395	110	85
D 848	77	80	102	....	38	+ 9	258.4	108	22	103.6	385	110	85
E 851	82	77	102	....	38	+ 5	258.1	111	18	102.4	335	110	85
F 854	84	76	101	....	38	0	258.2	113	16	99.6	285	110	85
G 857	84	78	105	....	38	− 5	258.2	116	13	99.6	275	110	85
H 860	83	87	102	....	38	−10	257.4	121	8	92.9	240	110	85
I 863	83	87	102	....	38	−15	257.0	127	1	86.0	195	110	85
ENGINE NO. 1.—FUEL: DENATURED ALCOHOL.[i]													
J 926	77	72	61	....	56	+30	260.9	112	18	102.4	380	110	79
K 929	79	74	61	....	56	+25	260.5	111	20	105.4	380	110	79
L 932	79	75	60	....	56	+21	259.8	111	20	102.5	350	110	79
M 935	64	62	54	....	56	+17	259.1	111	20	103.8	335	110	79
N 938	68	66	54	....	56	+13	258.5	111	19	105.8	325	110	79
O 941	71	69	55	....	56	+ 9	258.0	112	17	102.6	295	110	79
P 944	73	70	56	....	56	+ 5	257.0	114	15	98.7	280	110	79
Q 947	75	71	56	....	56	0	255.3	119	9	102.8	270	110	79
R 950	77	72	57	....	56	− 5	254.4	124	3	93.0	235	110	79
ENGINE NO. 2.—FUEL: GASOLINE.[j]													
S {2071	88	86	......	885	78	+20	253.9	106	21	107.6	420	115	85
S {2103	69	71	......	910	78	+20	256.5	98	31	111.4	405	115	85
T {2101	73	75	......	990	78	+10	256.8	107	21	117.8	335	115	85
T {2069	89	85	......	945	78	+10	253.6	104	23	111.4	385	115	85
U {2067	88	84	......	1,010	78	0	253.7	102	25	111.0	345	115	85
U {2099	74	74	......	1,100	78	0	256.5	103	25	111.1	290	115	85
V {2083	69	70	......	1,105	78	−10	255.1	107	21	100.6	235	115	85
V {2097	73	72	......	1,225	78	−10	256.3	109	19	99.8	230	115	85
W 2065	86	82	......	1,230	78	−15	254.9	118	9	93.0	220	115	85
X 2081	68	69	......	1,200	78	−20	253.0	115	12	96.2	215	115	85

[a] Recorded by a thermometer placed in the inlet-valve housing (p. 57).
[b] Measured by a thermocouple placed in the exhaust pipe near the exhaust valve (p. 58).
[c] Opening of the fuel needle valve in hundredths of one revolution.
[d] Point of ignition in degrees is given in terms of the crank position; + = early, or before dead center; − = late, or after dead center (p. 55).
[e] Brake horsepower constant for engine No. 1 = 0.0004817; for engine No. 2 = 0.0004493. Indicated horsepower constant for engine No. 1 = 0.0014; for engine No. 2 = 0.001396 (p. 67).

OPERATOR'S NOTES (TABLE 5).

The load and the fuel needle-valve setting were in each case so adjusted that a wide range in the time of ignition could be obtained without interfering with the satisfactory running of the engine.

ENGINE NO. 1. FUEL: GASOLINE.

Test A 815.—Very heavy pound with every explosion. Earlier ignition not attempted.

Test B 842.—Very heavy explosion pound.

Test C 845.—Heavy explosion pound.

Test D 848.—Heavy explosion pound.

Test E 851.—Light explosion pound.

Test F 854.—Light explosion pound.

Test G 857.—Very light explosion pound.

economy versus time of ignition.

[OTTO 15-HORSEPOWER GASOLINE ENGINES NOS. 1 AND 2.—Compression pressure, 70 pounds per square inch above 1 atmosphere; cylinder diameter, 6¾ inches; stroke, 15½ inches; hit-or-miss fuel-supply governor; jacket-water temperature: Inlet, 60° F.; outlet, 100° to 125° F.]

ENGINE NO. 1.—FUEL: GASOLINE.[h]

Horsepower.[e]		Duration (minutes).	Fuel consumption.								Efficiencies (per cent).		
			Pounds per hour.	Average per charge (pounds).	Per brake horsepower-hour.			Per indicated horsepower-hour.			Mechanical.	Thermal.	
Brake.	Indicated.				Pounds.	Gallons.	B. t. u. (low).	Pounds.	Gallons.	B. t. u. (low).		Indicated.[f]	Brake.[g]
13.74	15.9	30	9.86	0.00147	0.718	0.118	13,730	0.62	0.102	11,840	86.3	21.5	18.5
13.71	15.8	31	9.55	.00142	.696	.114	13,310	.60	.099	11,550	86.7	22.0	19.1
13.77	15.5	30	9.02	.00140	.656	.108	12,550	.58	.095	11,110	88.6	22.9	20.3
13.69	15.6	32	9.26	.00143	.676	.111	12,930	.59	.097	11,360	87.8	22.4	19.7
13.68	15.9	31	9.68	.00145	.708	.116	13,540	.61	.100	11,610	85.7	21.9	18.8
13.68	15.8	34	9.66	.00142	.706	.116	13,500	.61	.100	11,720	86.7	21.7	18.9
13.68	16.2	30	10.02	.00144	.734	.121	14,040	.62	.102	11,820	84.3	21.5	18.0
13.64	15.7	37	10.47	.00144	.768	.126	14,690	.67	.110	12,740	86.8	20.0	17.3
13.62	15.3	30	10.88	.00143	.799	.131	15,280	.71	.117	13,600	89.0	18.7	16.7
ENGINE NO. 1.—FUEL: DENATURED ALCOHOL.[i]													
13.82	16.1	31	15.57	0.00232	1.126	0.164	11,760	0.97	0.142	10,100	85.9	25.2	21.6
13.80	16.2	32	15.20	.00228	1.102	.161	11,510	.94	.137	9,800	85.1	26.0	22.1
13.77	15.8	31	15.33	.00230	1.113	.163	11,630	.97	.142	10,200	87.4	25.0	21.9
13.72	16.0	31	15.49	.00232	1.129	.165	11,790	.97	.142	10,100	86.6	25.1	21.6
13.70	16.3	31	15.53	.00233	1.133	.165	11,830	.96	.140	10,000	84.3	25.5	21.5
13.67	16.0	30	15.82	.00235	1.157	.169	12,090	.99	.145	10,300	85.3	24.7	21.0
13.62	15.9	30	16.19	.00237	1.189	.174	12,410	1.03	.150	10,800	86.6	23.7	20.5
13.51	17.1	30	16.60	.00232	1.229	.179	12,830	.97	.142	10,200	79.2	25.1	19.8
13.48	16.2	28	17.15	.00230	1.272	.186	13,290	1.06	.155	11,000	83.0	23.1	19.2
ENGINE NO. 2.—FUEL: GASOLINE.[j]													
13.11	15.9	24	9.96	0.00157	0.760	0.127	14,620	0.62	0.104	12,000	82.3	21.2	17.4
13.25	15.2	27	10.02	.00170	.756	.127	14,580	.66	.111	12,730	87.3	20.0	17.5
13.27	17.6	25	9.73	.00152	.733	.123	14,140	.55	.093	10,630	75.2	23.9	18.0
13.10	16.1	27	9.95	.00159	.760	.127	14,620	.62	.103	11,890	81.4	21.4	17.4
13.10	15.8	29	10.22	.00167	.780	.130	15,000	.65	.108	12,450	83.0	20.4	17.0
13.25	16.0	26	10.55	.00171	.796	.134	15,360	.66	.111	12,730	82.8	20.0	16.6
13.18	15.0	14	10.71	.00167	.812	.136	15,620	.71	.119	13,750	87.9	18.5	16.3
13.24	15.2	26	11.43	.00175	.863	.145	16,650	.75	.127	14,530	87.3	17.5	15.3
13.17	15.3	32	11.43	.00161	.868	.145	16,700	.75	.125	14,350	86.0	17.7	15.2
13.07	15.4	25	11.96	.00173	.915	.153	17,600	.78	.130	14,950	84.9	17.0	14.5

[f] Calculated from the indicated horsepower and low heating value of the fuel (p. 72).
[g] Calculated from the brake horsepower and low heating value of the fuel (p. 72).
[h] Specific gravity at 60° F.=0.729. Heating value: High=20,407 B. t. u. per pound; low=19,125 B. t. u. per pound.
[i] Specific gravity at 60° F.=0.820. Per cent alcohol by volume=92.2; by weight=91.5. Heating value: High=11,561 B. t. u. per pound; low=10,447 B. t. u. per pound.
[j] Specific gravity at 60° F.=0.712. Heating value: High=20,581 B. t. u. per pound; low=19,292 B. t. u. per pound.

Test H 860.—No explosion pound.

Test I 863.—The engine would not carry the load for later ignition timing than used for this test.

ENGINE NO. 1. FUEL: DENATURED ALCOHOL.

Test J 926.—Earlier ignition than used for this test was not attempted on account of the vibration caused by the explosion pound.

Tests K 929 to Q 947 inclusive.—Running of engine normal in every respect.

Test R 950.—The engine would not carry the load for later ignition than used for this test.

ENGINE NO. 2. FUEL: GASOLINE.

Tests S 2071 to X 2081 inclusive.—Running of engine seemingly normal in every respect.

CONCLUSIONS.

In studying the effect of time of ignition upon fuel economy, as shown in the foregoing tables and curves, there are some general relations which are obvious. It will be observed that the changes in economy are comparatively the same whether the fuel used be gasoline or denatured alcohol. In both cases the quantity of fuel used decreased as the time of ignition was advanced to some point considerably earlier than dead center, after which the quantity of fuel consumed increased.

A plausible explanation as to why the time of ignition giving the best economy differs with the different fuels or different engines may be found by comparing the two series of tests with gasoline, one on Otto gasoline engine No. 1 and the other on Otto gasoline engine No. 2. All the operating conditions were identical in these series with

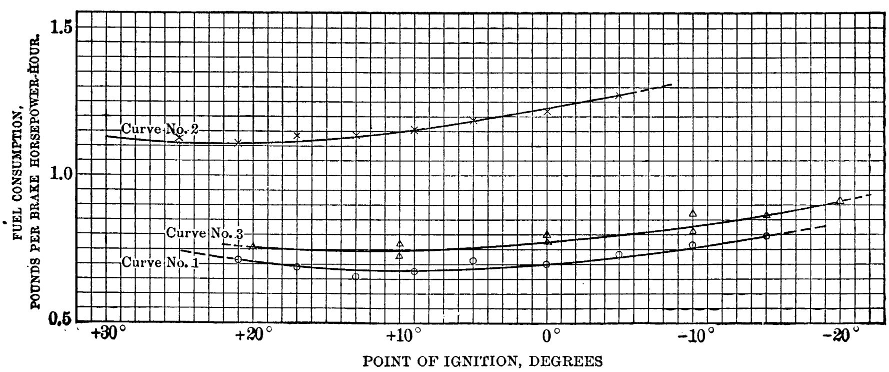

FIGURE 21.—Curves showing relation between point of ignition and fuel consumption.

the exception of the mixture quality, which by reference to figure 21 is seen to have been slightly richer in engine No. 2. By reference to the description of the engines under "Apparatus used in tests," it will be seen that they are similar in size and design and the general conclusion is therefore indicated that the best time of ignition is dependent upon the mixture quality (p. 22). This may explain the slight difference noted in all three series of tests, for by further reference to figure 21 it will be seen that not only was there a difference in the richness of the mixture for the two series of tests with gasoline, but also that the denatured-alcohol mixture which occurred at the best time of ignition (+25°) was weaker than either of the gasoline mixtures.

To recapitulate, disregarding both fuel and engines as affecting causes, the following relations exist:

Relations between mixture-quality ratio and best time of ignition.

Mixture-quality ratio.[a]	Best time of ignition.
	°
0.91	+10
0.94	+13
1.04	+25

[a] Ratio of the weight of air supplied to that theoretically necessary for complete combustion.

The above figures seem to indicate that the best time of ignition becomes earlier as the mixture-quality ratio approaches that which is the quickest burning.

In the discussion in the early part of this chapter it was pointed out that speed, compression, and load, as well as mixture quality, may affect the best time of ignition, and reference to later sections on these subjects will bring out this relation more in detail. The fact that both compression and speed were kept constant in these tests would seem further to strengthen the position taken, namely, that any differences indicated in the best time of ignition were solely due to differences in the richness of the mixture used.

Not only do all three series of tests show the same characteristic variations but the range through which the time of ignition may be varied, without giving any external evidences of inefficient use of fuel or unstable operating conditions, is shown to be practically the same for both engines and both fuels. Whatever slight differences there may be in this range may be explained as above by the difference in mixture quality.

From the data obtained it is difficult to explain the variations in the thermal efficiency which are shown, for the different causes discussed (p. 98) can not readily be segregated. It is evident, however, (1) that the variation in pressure at the time of ignition could not in itself account for the variation in the thermal efficiency, otherwise the highest thermal efficiency would have been obtained with the time of ignition nearest dead center; (2) that the entire variation could not have been due alone to the change in time between ignition and exhaust, for the highest thermal efficiency was not obtained at the earliest time of ignition.

No attempt was made to determine the variation of the best time of ignition with the size of the cylinder, the contour and shape of the clearance space, the position of the ignition spark in the clearance space, or the number of igniters used simultaneously. All of these factors may, to a certain extent, determine the best time of ignition to be used by changing the rate of burning of the explosive mixture.

The final conclusion to be drawn from these tests is that, regardless of the engine or fuel used, the same general effect may be expected as a result of the variation of the time of ignition. Further, this variation will in all cases show the greatest efficiency in operation when the igniter is so adjusted as to cause the ignition of the explosive mixture to take place at some point in advance of dead center. The amount of this advance which will give the best results is not always the same and is dependent upon "compression," "speed," "load," and the "mixture quality." The effects of these variables are discussed somewhat at length in the sections devoted to those subjects.

CHARACTER OF IGNITION SPARK.

A quantitative determination of the intensity and size of the electric ignition spark can not readily be made. However, the electrical energy, in watts supplied to the igniter, may be taken as a measure of the intensity or size (often called "fatness") of the spark. This energy may be varied by changing the current and the potential. One object of this series of tests was to determine the effect of a change in the character of the spark, brought about by this means, upon the efficiency of the engine.

There are two general types of electric igniters in common use, known as the "jump-spark" and the "make-and-break" types, respectively.

EFFECT ON EFFICIENCY.

If the method of ignition used is the jump spark in the high-tension secondary circuit of an induction coil there will be a certain amount of building up in the coils and lag of the vibrator, causing the spark at the igniter plug gap in the cylinder to lag after the primary circuit has been closed. It is altogether possible that a change in the current may appreciably affect the lag. If so, the ultimate effect on the fuel economy and the thermal efficiency of the engine may be solely that caused by the change in the time of ignition as discussed under "Time of ignition." At any rate, it is impossible to separate the effect of the change in time of ignition from the effect of a change in the character of the spark caused directly by changes in current, and it was therefore decided to conduct the tests on an engine equipped with the mechanically operated, low-tension, make-and-break method of ignition. With a system of this type there is little possibility for the time of ignition to be affected by changes in the current, because the spark is caused by a sudden breaking of the circuit after it has been closed for some little time. The effect of a change in the character of the spark, if any, may then be expected to be brought about by changes in the rate of burning and in the completeness of combustion.

METHOD OF TESTING.

For this series of tests the engine was put in good stable operating condition and the igniter mechanism was made positive in its action. The load, fuel needle-valve setting, and time of ignition selected were such that a mixture quality sufficiently uniform always to ignite was assured. This was necessary, as otherwise the results of the tests would have been difficult to interpret, as the cause of misfiring may be ineffective ignition, due either to a defective ignition system or to a varying mixture quality.

The tests made on the 15-horsepower Otto gasoline engine in connection with this investigation were run with operating conditions approximately those required for obtaining the best fuel economy. These conditions were kept constant, and the current and potential were reduced to the minimum that would ignite the poorest explosive mixture obtained. The current and potential were then successively increased, and the fuel consumption was determined for each of four different combinations as shown in Table 6.

Sets of Edison primary cells were used to supply the current of the igniter circuit for the first two tests. For the third test these batteries were replaced by an Apple igniter system consisting of storage batteries and charging generator attached to the engine. A combination of Edison primary cells and storage batteries was used for the fourth test. All the changes involved were made without stopping the engine or changing any of the settings.

TABLE 6.—*Results of tests of fuel economy versus ignition-spark intensity.*

[OTTO 15-HORSEPOWER GASOLINE ENGINE NO. 2 EQUIPPED WITH A MAKE-AND-BREAK IGNITER OF THE HAMMER-BREAK TYPE.[a]—Compression constant at 70 pounds per squar inch above 1 atmosphere; diameter of cylinder, 6¾ inches; stroke, 15½ inches; method of governing, hit-or-miss fuel supply; jacket-water temperature: Inlet, 60°; outlet, 112° F.]

Test number.	Intensity of spark.			Index settings.		Speed (average per minute).			Average pressures (pounds per sq. in.).		Brake load.		Horse-power.[d]		Duration (minutes).	Fuel consumption.									Efficiencies (per cent).		
																		Per brake horse-power-hour.			Per indicated horse-power-hour.				Thermal.	
	Volts.	Amperes.	Watts.	Fuel needle valve.[b]	Igniter (degrees).[c]	Revolutions.	Explosions.	Fuel cut-outs.	Mean effective.	Maximum (above atmosphere).	Pounds.	Per cent of maximum.	Brake.	Indicated.		Pounds per hour.	Pounds per charge.	Pounds.	Gallons.	B. t. u. (low).	Pounds.	Gallons.	B. t. u. (low).	Mechanical.	Indicated.[e]	Brake.[f]
A 2126	4.50	1.0	4.5	24	+30	252.7	118	8	96.8	310	115	85	13.05	15.9	28	8.70	0.00123	0.667	0.112	12,870	0.55	0.092	10,530	82.0	24.2	19.8
B 2128	4.25	2.4	10.2	24	+30	252.5	123	4	95.4	320	115	85	13.05	16.3	27	8.78	.00119	.673	.113	12,980	.54	.090	10,360	79.9	24.6	19.6
C 2130	14.25	3.6	51.3	24	+30	252.7	117	9	95.9	330	115	85	13.06	15.7	28	8.73	.00124	.668	.112	12,890	.56	.094	10,750	83.4	23.7	19.7
D 2132	19.05	4.4	83.8	24	+30	252.9	120	7	97.6	340	115	85	13.06	16.3	28	8.70	.00121	.666	.112	12,850	.53	.090	10,300	80.2	24.7	19.8

[a] Specific gravity of gasoline at 60° F.=0.712. Heating value: High=20,581 B. t. u. per pound; low=19,292 B. t. u. per pound.
[b] Opening of needle valve in hundredths of one revolution (p. 55).
[c] Point of ignition, in degrees, is given in terms of the crank position. +=*early*, or before dead center (p. 55).
[d] Brake horsepower constant=0.0004493; indicated horsepower constant=0.001396 (p. 67).
[e] Calculated from the indicated horsepower and the lower calorific value of the fuel (p. 72).
[f] Calculated from the brake horsepower and the lower calorific value of the fuel (p. 72).

OPERATOR'S NOTES (TABLE 6).

Test A 2126.—For the particular igniter with which the engine was equipped and under the operating conditions given in Table 6, the voltage and current are the least that could be used without causing the weaker mixtures following cut-outs to misfire.

Test B 2128.—For the voltage and current given for this test the weakest mixture that would enable the engine to carry the load satisfactorily never misfired.

Test C 2130.—The increase in voltage and current used for this test caused no apparent change in the running of the engine.

Test D 2132.—The voltage and current used for this test caused a rapid burning away of the igniter points.

RESULTS OBTAINED.

The results of this series of tests are given in detail in Table 6, and the particular points noted by the observers during the course of the tests appear in the operator's notes. The variation in efficiency shown by the tests is so small that no curves are given to show the relation between current and potential of igniter system and efficiency.

CONCLUSIONS.

The results of the tests given in Table 6 indicate that when the current or potential used in the ignition circuit of a make-and-break igniter is increased beyond that required to prevent misfiring, there is no effect upon the thermal efficiency of the engine being tested. The only variation observed in the efficiency and fuel-consumption figures shown in the table is well within the probable error of observation. Further, the similarity of the sets of indicator diagrams for the different tests showed that the current and potential used had no appreciable effect upon the rate of burning.

The significance of the points brought out above lies in the fact that variations in the character of the ignition spark may be disregarded in the consideration of all the results and conclusions of these investigations.

Inasmuch as a study of the possible effect of varying the size of the spark and the thermal efficiency with which a given fuel is used was the principal object in running these tests, no attempt was made to determine the relative efficiency of various types of make-and-break igniters in regard to the current and potential required by them. In the course of the experiments it was noted, however, that seemingly the more rapid or sudden the break of a make-and-break igniter the less potential was required to produce an effective spark. It was also observed that there was seemingly no difference in the effectiveness of the igniters for igniting either gasoline or alcohol mixtures.

The effect of the size or "fatness" of the spark upon the maximum power which an engine is capable of developing has been the subject of much popular discussion, but during these tests no attempt was made to obtain any data upon the subject.

However, the results obtained, indicating no effect upon thermal efficiency, lead to the conclusion that the power of an engine is independent of the current and potential used in the igniter system, provided the intensity is such as to preclude the possibility of excessive misfiring or any unintentional change in time of ignition.

JACKET-WATER TEMPERATURE.

It was desired to determine what effect, if any, the jacket-water temperature existing during the tests might have on the interpretation of the results obtained. To that end, a series of tests eliminating the other variables as much as possible was planned and carried out.

EFFECT UPON THERMAL EFFICIENCY.

The effect of a varying jacket-water temperature upon the thermal efficiency of internal-combustion engines is a question that has received considerable attention, but the literature on the subject gives widely varying opinions. The inconsistencies noted in the results of different investigators are possible to comprehend only when considered in the light of the failure of many investigators to differentiate between the direct effects of varying jacket-water temperature upon heat losses in the engine in one case, and in other cases upon variables, such as the lubrication, the fit of the piston and valves, carburetor adjustment, and other phenomena that often attend a variation of jacket-water temperature.

It is extremely difficult to make the differentiation, because a slight variation in the temperature of the outgoing jacket water is so closely associated with attendant changes in what might be termed the mechanical operating conditions of the engine that it is easy to confuse the secondary effects with the one under consideration.

METHOD OF TESTING.

With the above-stated conditions in mind a series of tests to determine the effect of the outgoing jacket-water temperature upon thermal efficiency was conducted, the greatest care being constantly taken to see that the best mechanical operating conditions of the engine were maintained.

Tests were made on four engines, three of which were the horizontal engines described under "Apparatus used in tests" as 15-horsepower Otto gasoline engines Nos. 1 and 2, 15-horsepower Otto alcohol engine, and 10-horsepower Nash gasoline engine. By an examination of the descriptions referred to, it will be seen that these engines gave wide variations as to position of carburetor, method of governing, and other features of construction that might possibly affect the ultimate results of the tests.

The cylinder jackets of the engines also differed in several details, such as the direction of circulation of the cooling water and the location of the points of inlet and outlet, but they were all practically alike in that the water was taken into the jacket in all cases at the temperature of the city mains and was discharged at some higher temperature, which, in this discussion, is termed the jacket-water temperature.

In order to make the conclusions drawn of as general a nature as possible both gasoline and alcohol were used, and different compressions were also used with the alcohol. Tests of two different natures were conducted, which, for the purpose of uniformity, have been called "normal fuel economy versus jacket-water temperature" tests (Table 7), and "best fuel economy versus jacket-water temperature" tests (Tables 8, 9, 10, 11, and 12), respectively.

As in other normal fuel-economy tests, the load, speed, fuel needle-valve, time of ignition, and other similar adjustments were maintained at the point giving, as found by previous tests, the best fuel economy for a jacket-water temperature of 120° F. but the temperature of the outgoing jacket-water varied from 75° F. to 200° F.

In the "best fuel economy" tests, the outgoing jacket-water temperature was changed as before and various combinations of the above variables were systematically tried in order to determine the least fuel consumption at each jacket-water temperature.

RESULTS OBTAINED.

The tabulated results of all the tests attempted in this investigation follow and the data and appended notes are complete and self-explanatory. For the purpose of easier interpretation, curves showing the important points brought out by the investigation accompany the tables.

Table 7 gives the results of normal fuel economy versus jacket-water temperature tests on the Otto gasoline engine No. 1 with gasoline and denatured alcohol as fuels.

Table 8 gives the fuel consumption only for the trial and check tests that were made to determine the best fuel economy as the jacket-water temperature was varied. The tests were run on the Otto gasoline engine No. 2, with gasoline and denatured alcohol as fuel, and the tabulated results of the check tests chosen are shown in Table 9.

Results of best fuel economy versus jacket-water temperature tests on the Nash gasoline engine with both fuels are shown on Table 10. The results of search and check tests on the Otto alcohol engine are given in Table 11, and the detailed figures from the check tests appear in Table 12.

Curves showing the relation between jacket-water temperature and fuel consumption with the different engines and fuels are platted on figure 22.

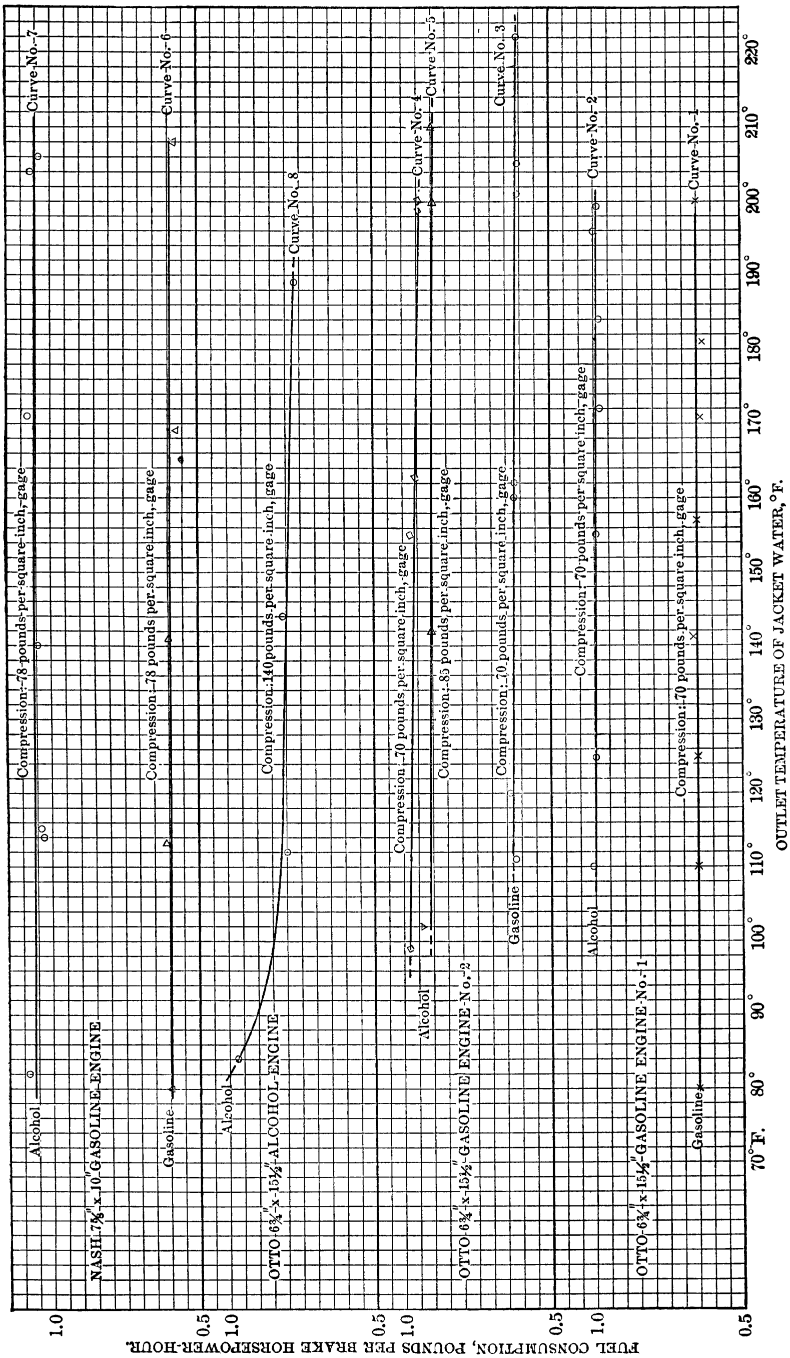

Figure 22.—Curves showing relation between jacket-water temperature and fuel consumption.

TABLE 7.—*Results of tests of normal fuel*

[OTTO 15-HORSEPOWER GASOLINE ENGINE No. 1.—Compression, constant at 70 pounds per square inch above atmosphere; diameter of cylinder, 6¾ inches; stroke, 15½ inches; method of governing, hit-or-miss fuel supply: Inlet temperature of jacket water, constant at 60° F.]

FUEL: GASOLINE.[g]

Test number.	Temperatures (°F.).				Index settings.		Speeds (average per minute).			Average pressures (pounds per sq. in.).		Brake load.	
	Air.	Fuel.	Mixture.[a]	Jacket-water outlet.	Fuel needle valve.[b]	Igniter (degrees).[c]	Revolutions.	Explosions.	Fuel cut-outs.	Mean effective.	Maximum (above atmosphere).	Pounds.	Per cent of maximum.
A 773	80	81	105	80	34	+ 9	256.0	127	1	94.5	320	115	88
B 776	85	86	108	96	34	+ 9	258.2	127	2	92.0	300	115	88
C 779	83	79	104	110	34	+ 9	256.9	126	2	94.8	320	115	88
D 797	85	82	103	125	34	+ 9	255.8	120	8	96.2	300	115	88
E 785	79	85	104	141	34	+ 9	256.9	120	8	96.1	320	115	88
F 788	82	80	108	157	34	+ 9	257.0	119	9	98.7	320	115	88
G 791	82	77	109	171	34	+ 9	256.5	119	9	98.0	330	115	88
H 794	83	80	114	181	34	+ 9	256.7	120	9	99.0	330	115	88
I 802	89	82	122	200	34	+ 9	256.9	121	8	97.0	330	115	88
FUEL: DENATURED ALCOHOL.[h]													
J 899	83	76	63	75	44	+30	254.7	121	2	92.2	300	115	82
K 902	85	82	61	94	44	+30	253.3	123	1	93.0	300	115	82
L 905	88	80	63	110	44	+30	255.7	126	1	91.2	300	115	82
M 908	89	80	62	125	44	+30	256.8	128	1	89.4	310	115	82
N 912	84	75	62	140	44	+30	258.6	125	1	91.5	300	115	82
O 914	87	77	64	155	44	+30	256.6	125	0	91.5	300	115	82
P 917	84	77	64	172	44	+30	257.6	125	1	90.8	300	115	82
Q 920	84	76	66	184	44	+30	260.6	127	0	90.8	300	115	82
R 923	81	75	67	196	44	+30	257.0	128	1	90.8	300	115	82

[a] Measured by a thermometer placed in the inlet-valve housing (p. 57).
[b] Opening of the fuel needle valve in hundredths of one revolution (p. 55).
[c] Point of ignition in degrees is given in terms of the crank position; += early or before dead center (p. 55).
[d] Horsepower constants: Brake=0.0004817; indicated=0.001400 (p. 67).
[e] Calculated from the indicated horsepower and low-heating value of the fuel (p. 72).

OPERATORS' NOTES (TABLE 7).

FUEL: GASOLINE.

Tests A 773 to C 779 inclusive.—The first fuel charge after cut-outs occasionally misfired, due seemingly to the action of the governor causing irregularities in the quality of the mixture which sometimes became too weak to burn. The electric igniter was not at fault.

Tests D 797 to I 802 inclusive.—Operation of the engine seemingly normal in every respect.

economy versus jacket-water temperature.

[OTTO 15-HORSEPOWER GASOLINE ENGINE No. 1.—Compression, constant at 70 pounds per square inch above atmosphere; diameter of cylinder, 6$\frac{3}{4}$ inches; stroke, 15$\frac{1}{2}$ inches; method of governing, hit-or-miss fuel supply: Inlet temperature of jacket water, constant at 60° F.]

FUEL: GASOLINE.[g]

Horsepower.[d]		Duration (minutes).	Fuel consumption.								Efficiencies (per cent).		
			Pounds per hour.	Pounds per charge.	Per brake horsepower hour.			Per indicated horsepower hour.			Mechanical.	Thermal.	
Brake.	Indicated.				Pounds.	Gallons.	B. t. u. (low).	Pounds.	Gallons.	B. t. u. (low).		Indicated.[e]	Brake.[f]
14.18	16.8	36	9.30	0.00122	0.656	0.108	12,550	0.55	0.090	10,600	84.6	24.0	20.3
14.31	16.4	32	9.40	.00123	.657	.108	12,560	.57	.094	10,960	87.3	23.2	20.3
14.24	16.7	32	9.35	.00124	.657	.108	12,560	.56	.092	10,670	85.0	23.9	20.3
14.17	16.2	33	9.19	.00128	.648	.106	12,400	.57	.094	10,850	87.4	23.5	20.5
14.22	16.2	35	9.38	.00130	.660	.108	12,630	.58	.095	11,070	87.9	23.0	20.2
14.24	16.4	29	9.31	.00130	.654	.107	12,510	.57	.094	10,830	86.5	23.5	20.3
14.21	16.4	33	9.19	.00129	.646	.106	12,360	.56	.092	10,740	87.1	23.7	20.6
14.22	16.6	33	9.05	.00126	.636	.104	12,160	.54	.089	10,410	85.7	24.4	20.9
14.24	16.4	29	9.34	.00129	.656	.108	12,550	.57	.094	10,900	87.0	23.3	20.3
FUEL: DENATURED ALCOHOL.[h]													
14.11	16.2	30	14.17	0.00189	1.004	0.147	10,530	0.87	0.127	9,180	87.2	27.7	24.2
14.03	16.0	30	14.00	.00185	.998	.146	10,470	.88	.128	9,200	87.8	27.7	24.3
14.16	16.1	29	14.27	.00187	1.008	.147	10,570	.89	.130	9,330	88.2	27.3	24.1
14.22	16.0	34	14.22	.00185	1.000	.126	10,500	.89	.130	9,340	89.0	27.2	24.2
14.32	16.0	30	14.24	.00185	.995	.145	10,390	.89	.130	9,270	89.2	27.5	24.5
14.21	16.0	30	14.00	.00182	.986	.144	10,300	.88	.128	9,140	88.8	27.8	24.7
14.27	15.9	30	13.97	.00182	.979	.143	10,230	.88	.128	9,180	89.6	27.7	24.9
14.44	16.1	30	14.03	.00180	.972	.142	10,160	.87	.127	9,090	89.5	28.0	25.0
14.23	16.3	30	14.17	.00184	.996	.145	10,410	.87	.127	9,110	87.6	27.9	24.4

[f] Calculated from the brake horsepower and low heating value of the fuel (p. 72).
[g] Specific gravity at 60° F.=0.729. Heating value: High=20,407, low=19,125 B. t. u. per pound.
[h] Specific gravity at 60° F.=0.820. Per cent alcohol by weight=91.0. Heating value: High=11,603, low=10,496 B. t. u. per pound.

FUEL: DENATURED ALCOHOL.

Test J 899.—The first fuel charge after cut-outs misfired about four times per minute, due seemingly to the same reason as given above.

Test K 902.—The first fuel charge after cut-outs misfired about three times per minute.

Test L 905.—The first fuel charge after cut-outs misfired about once per minute.

Test M 908.—None of the mixture charges misfired.

Test N 912 to Q 920.—The first mixture charge after cut-out misfired about three times per minute.

Test R 923.—No misfiring noted.

TABLE 8.—*Results of trial and check tests made to determine the best fuel economy for various jacket-water temperatures.*

[OTTO 15-HORSEPOWER GASOLINE ENGINE No. 2.—Diameter of cylinder, 6¾ inches; stroke, 15½ inches; method of governing, hit-or-miss fuel supply.]

FUEL: GASOLINE.[a]

[Speed, 245 to 255 r. p. m.; brake load, 85 per cent of greatest for gasoline; brake horsepower, 12.6 to 13.2; compression, 70 pounds per square inch above 1 atmosphere.]

Fuel needle-valve settings.[d]	Fuel consumption per brake horsepower-hour, pounds.[c]				
	Jacket temperature, 112° to 118° F.				
16	(f)	0.609	(f)		
18	0.639	0.601 *.634*	0.609 *.629*		
20		0.631	0.614		
22			[g] *0.621*		
	Jacket temperature, 144° to 165° F.				
16	(f)		0.602	(f)	
18		0.613	0.590 *.613* .574		
20		0.609	*0.620* [g].626 .603	[g]0.625 .614	
22			[g]0.639	[g]0.621	
24		[g]0.630	[g]0.630	[g]0.634	
Igniter settings.[e]	+30°	+25°	+20°	+15°	+10°

FUEL: DENATURED ALCOHOL.[b]

[Speed, 250 to 256 r. p. m.; brake load, 69 per cent of greatest for alcohol; brake horsepower, 14.6 to 14.9; compression, 70 pounds per square inch above 1 atmosphere.]

Fuel needle-valve settings.[d]	Fuel consumption per brake horsepower-hour, pounds.[c]		
	Jacket temperature, 99° F.		
94	(f)	*0.986*	(f)
	Jacket temperature, 146° to 163° F.		
90	0.996	0.966 *.960*	0.972
92		0.979 *.986*	0.986
	Jacket temperature, 194° to 205° F.		
90	0.950	0.941 .988	
92	0.969	0.951 .989	0.968
94	0.966	0.950 *.946*	0.956
	Compression, 85 pounds per square inch above atmosphere.		
	Jacket temperature, 100° F.		
88	(f)	*0.931*	(f)
Igniter settings.[e]	+35°	+30°	+25°

TABLE 8.—*Results of trial and check tests made to determine the best fuel economy for various jacket-water temperatures.*—Continued.

[OTTO 15-HORSEPOWER GASOLINE ENGINE NO. 2.—Diameter of cylinder, $6\frac{3}{4}$ inches; stroke, $15\frac{1}{2}$ inches; method of governing, hit-or-miss fuel supply.]

FUEL: GASOLINE.[a]

[Speed, 245 to 255 r. p. m.; brake load, 85 per cent of greatest for gasoline; brake horsepower, 12.6 to 13.2; compression, 70 pounds per square inch above 1 atmosphere.]

Fuel needle-valve settings.[d]	Fuel consumption per brake horsepower-hour, pounds.[c]				
	Jacket temperature, 182° to 209° F.				
18	(f)		0.612	0.608	(f)
20		0.628	0.593 *.603* *.609* [g].636	0.611	
22			[g]0.617 [g].629	[g]0.617 [g].633	
24			[g]0.621	[g]0.617 [g].615 [g].617 [g].631	[g]0.634
26				[g]0.629 [g].636	
56					[g]0.666
	Jacket temperature, 214° to 232° F.				
20			(f)	[g]0.626	[g]0.632
24				[g]*0.618* [g].613	
36				[g]0.647	
Igniter settings.[e]	+30°	+25°	+20°	+15°	+10°

FUEL: DENATURED ALCOHOL.[b]

[Speed, 250 to 256 r. p. m.; brake load, 69 per cent of greatest for alcohol; brake horsepower, 14.6 to 14.9; compression, 70 pounds per square inch above 1 atmosphere.]

Fuel needle-valve settings.[d]	Fuel consumption per brake horsepower-hour, pounds.[c]		
	Jacket temperature, 142° F.		
84	(f)	*0.912*	(f)
	Jacket temperature, 200° F.		
80	0.899		(f)
82	0.946	0.927 .934	0.942
84		0.943	
86	0.956	0.931 *.900*	0.923
Igniter settings.[e]	+35°	+30°	+25°

[a] Specific gravity at 60° F.=0.7168; heating value: High=20,527, low=19,235 B. t. u. per pound.
[b] Specific gravity at 60° F.=0.8188; per cent alcohol by weight, 91.5; heating value: High=11,768, low=10,681 B. t. u. per pound.
[c] The figures in the body of the table give the fuel consumption in pounds per brake horsepower-hour and are tabulated with respect to fuel needle-valve and igniter settings. The figures in italics are the results of check tests of about 30 minutes duration. The other figures in the body of the table are the results of trial tests of about 15 minutes duration.
[d] Opening of the fuel needle valve in hundredths of 1 revolution (p. 55).
[e] Point of ignition in degrees given in terms of the crank position += early, or before dead center (p. 55).
[f] Trial tests were attempted with this fuel needle-valve and igniter setting, but the engine would not carry the load satisfactorily.
[g] Fuel inlet valve was held open a little shorter time for these tests than for the others. Governor mechanism adjusted.

TABLE 9.—*Results of tests of best*

[OTTO 15-HORSEPOWER GASOLINE ENGINE No. 2.—Inlet temperature of jacket water, constant at 60° F.; diameter of cylinder, 6¾ inches; stroke, 15½ inches; method of governing, hit-or-miss fuel supply.]

FUEL: GASOLINE.[g]

[Compression, 70 pounds per square inch above 1 atmosphere.]

Test number.	Temperatures (° F.).				Index settings.		Speeds (average per minute).			Average pressures (pounds per sq. in.).		Brake load.	
	Air.	Fuel.	Mixture.[a]	Jacket-water outlet.	Fuel needle valve.[b]	Igniter (degrees).[c]	Revolutions.	Explosions.	Fuel cut-outs.	Mean effective.	Maximum above atmosphere.	Pounds.	Per cent of maximum.
A 1776	82	98	80	111	[h] 22	+20	253.5	123	4	90.0	350	115	85
A 1873	87	95	85	118	18	+20	250.4	121	4	92.3	340	115	85
B 1858	95	97	103	162	18	+20	252.1	121	5	89.9	320	115	85
B 1861	97	98	105	160	20	+20	251.9	121	5	89.4	330	115	85
C 1846	86	91	105	203	20	+20	254.7	121	7	89.3	350	115	85
C 1849	83	86	101	201	20	+20	254.8	121	7	88.1	340	115	85
D 1821	90	90	112	222	[h] 24	+15	250.9	124	2	87.5	360	115	85

FUEL: DENATURED ALCOHOL.[i]

[Compression, 70 pounds per square inch above 1 atmosphere.]

Test number.	Air.	Fuel.	Mixture.	Jacket-water outlet.	Fuel needle valve.	Igniter (degrees).	Revolutions.	Explosions.	Fuel cut-outs.	Mean effective.	Maximum above atmosphere.	Pounds.	Per cent of maximum.
E 1954	78	77	55	99	94	+30	253.2	122	4	98.2	320	130	87
F 1960	82	78	62	155	90	+30	253.9	123	4	95.3	320	130	87
F 1962	86	83	66	163	90	+30	254.5	124	3	96.6	320	130	87
G 1942	75	72	56	199	90	+30	255.3	124	4	96.1	320	130	87
G 1951	78	77	62	200	94	+30	254.6	124	4	95.3	320	130	87

FUEL: DENATURED ALCOHOL.[i]

[Compression, 85 pounds per square inch above 1 atmosphere.]

Test number.	Air.	Fuel.	Mixture.	Jacket-water outlet.	Fuel needle valve.	Igniter (degrees).	Revolutions.	Explosions.	Fuel cut-outs.	Mean effective.	Maximum above atmosphere.	Pounds.	Per cent of maximum.
H 1985	90	82	64	102	88	+30	250.7	119	7	97.0	345	130	87
I 1982	90	81	65	142	84	+30	250.2	123	2	97.0	345	130	87
J 1969	94	85	64	210	80	+35	252.4	123	3	93.0	365	130	87
J 1979	89	80	65	200	86	+30	251.4	119	7	97.2	335	130	87

[a] Measured by a thermometer placed in the inlet-valve housing (p. 57).
[b] Opening of the fuel needle valve in hundredths of one revolution (p. 55).
[c] Point of ignition in degrees is given in terms of the crank position; + = early or before dead center (p. 55).
[d] Horsepower constants: Brake = 0.0004493; indicated = 0.001396 (p. 67).
[e] Calculated from the indicated horsepower and low heating value of the fuel (p. 72).

fuel economy versus jacket-water temperature.

[Otto 15-Horsepower Gasoline Engine No. 2.—Inlet temperature of jacket water, constant at 60° F.; diameter of cylinder, 6¾ inches; stroke, 15½ inches; method of governing, hit-or-miss fuel supply.]

FUEL: GASOLINE.[g]

[Compression, 70 pounds per square inch above 1 atmosphere.]

Horsepower.[d]		Duration (minutes).	Fuel consumption.								Efficiencies (per cent).		
					Per brake horsepower-hour.			Per indicated horsepower-hour.				Thermal.	
Brake.	Indicated.		Pounds per hour.	Pounds per charge.	Pounds.	Gallons.	B. t. u. (low).	Pounds.	Gallons.	B. t. u. (low).	Mechanical.	Indicated.[e]	Brake.[f]
13.10	15.5	33	8.14	0.00110	0.621	0.104	11,940	0.52	0.088	10,140	84.8	25.1	21.3
12.94	15.6	30	8.14	.00112	.629	.105	12,100	.52	.087	10,020	82.9	25.4	21.0
13.04	15.2	30	8.00	.00110	.613	.102	11,790	.53	.088	10,160	86.0	25.0	21.6
13.01	15.1	30	8.07	.00111	.620	.103	11,930	.53	.089	10,270	86.0	24.8	21.3
13.16	15.0	30	7.93	.00109	.603	.101	11,600	.53	.088	10,140	87.5	25.1	21.9
13.16	14.9	30	8.02	.00110	.609	.102	11,710	.54	.090	10,370	88.5	24.5	21.7
12.96	15.1	30	8.02	.00108	.618	.103	11,890	.53	.089	10,210	85.8	24.9	21.4

FUEL: DENATURED ALCOHOL.[i]

[Compression, 70 pounds per square inch above 1 atmosphere.]

14.79	16.7	25	14.59	0.00199	0.986	0.144	10,530	0.87	0.127	9,300	88.3	27.4	24.2
14.82	16.4	15	14.31	.00194	.966	.141	10,320	.87	.128	9,340	90.5	27.3	24.7
14.86	16.8	25	14.26	.00192	.960	.140	10,250	.85	.124	9,090	88.7	28.0	24.8
14.91	16.6	13	14.03	.00189	.941	.138	10,050	.84	.123	9,010	89.7	28.2	25.3
14.87	16.4	26	14.07	.00189	.946	.138	10,100	.85	.125	9,150	90.5	27.8	25.2

FUEL: DENATURED ALCOHOL.[i]

[Compression, 85 pounds per square inch above 1 atmosphere.]

14.64	16.1	26	13.63	0.00191	0.931	0.136	9,940	0.85	0.124	9,060	91.0	28.1	25.6
14.61	16.7	27	13.34	.00181	.912	.133	9,740	.80	.117	8,540	87.6	29.8	26.1
14.75	16.0	14	13.25	.00180	.899	.131	9,600	.83	.122	8,880	92.5	28.7	26.5
14.68	16.2	27	13.21	.00185	.900	.132	9,610	.82	.120	8,740	90.9	29.1	26.5

f Calculated from the brake horsepower and low heating value of the fuel (p. 72).
g Specific gravity at 60° F.=0.717. Heating value: High=20,527, low=19,235 B. t. u. per pound.
h Governor mechanism adjusted. Fuel inlet valve was held open a little shorter time for this test than for the others.
i Specific gravity at 60° F.=0.819; per cent alcohol, by weight, 91.5. Heating value: High=11,788, low=10,681 B. t. u. per pound.

OPERATORS' NOTES (TABLE 9).

FUEL: GASOLINE.

Test A.—Of the 9 trial tests, 3 showed fuel consumption of less than 0.61 pound per brake horsepower per hour, but these values could not be checked. The range of trial settings was small and, with the resulting economies, is fairly represented by the figures given in the table. There was a light explosion pound in Test A 1776.

Test B.—Of the 16 trial tests, 5 showed fuel consumptions of less than 0.61 pound per brake horsepower per hour, but these values could not be checked. The trial tests ranged from needle-valve 24 and igniter +15°, with a corresponding fuel consumption of 0.63 pound per brake horsepower per hour, to the values given in the table.

Test C.—The 22 trial tests covered a range of settings from needle-valve 3 and igniter +10°, with a corresponding fuel consumption of 0.66 pound per brake horsepower per hour, to the values given in the table.

Test D.—The 5 trial tests covered a range of settings from needle-valve 36 and igniter +15°, with a corresponding fuel consumption of 0.64 pound per brake horsepower per hour, to the values given in the table.

FUEL: DENATURED ALCOHOL.

Test E.—This test was made with the needle-valve and igniter settings that had previously been found to give minimum fuel consumption for this engine.

Test F.—The 7 trial tests all show fuel consumptions of less than one pound per brake horsepower per hour. The needle-valve setting and fuel consumption given in the table were the smallest that could be obtained.

Test G.—The 11 trial tests show fuel consumptions ranging from 0.97 pound per brake horsepower per hour to the values given in the table, which also gives the limits of the trial needle-valve settings.

Test H.—This was the only test made with a low jacket temperature. The best settings were determined by observation.

Test I.—This was the only test made with a medium jacket temperature. The best settings were readily determined by observation.

Test J.—The 10 trial tests show fuel consumptions ranging from 0.96 pound per brake horsepower per hour to the values given in the table, but are somewhat inconsistent in needle-valve settings.

TABLE 10.—*Results of tests of best fuel*

[NASH 10-HORSEPOWER GASOLINE ENGINE.—Diameter of cylinder, 7⅝ inches; stroke, 10 inches; method of governing, hit-or-miss mixture supply; inlet temperature of jacket water, 60° F.; engine equipped with regular carburetor.]

Test No.	Temperatures (° F.).			Index settings.			Speeds (average per minute).			Average pressures (pounds per sq. in.).		
	Air.	Mixture near inlet-valve.[a]	Jacket-water outlet.	Air valve.[b]	Fuel needle valve.[c]	Igniter.[d]	Revolutions.	Explosions.	Mixture cut-outs.	Compression (above atmosphere).	Mean effective.	Maximum (above atmosphere).
						FUEL: GASOLINE.[h]						
A 3608	83	103	80	$\frac{1}{2}$	$1\frac{15}{16}$	$1\frac{1}{4}$	297.2	141	8	72	72.0	275
B 3328	102	......	113	$\frac{1}{2}$	$1\frac{15}{16}$	$6\frac{1}{2}$	299.2	146	4	72	70.8	255
C 3606	69	107	141	$\frac{1}{2}$	$1\frac{15}{16}$	2	297.6	144	5	72	70.2	200
D 3612	82	107	169	$\frac{5}{8}$	2	$\frac{1}{4}$	296.3	139	9	78	72.0	230
E 3610	71	112	208	$\frac{5}{8}$	2	2	297.6	143	6	78	71.7	240
						FUEL: DENATURED ALCOHOL.[i]						
F 3442	98	......	82	$\frac{5}{8}$	$3\frac{7}{16}$	+2=20°	296.7	147	1	78	82.2	300
G {3426	107	......	115	$\frac{5}{8}$	$3\frac{7}{16}$	+2=20°	294.7	144	3	78	82.2	300
G {3440	69	......	114	$\frac{5}{8}$	$3\frac{7}{16}$	+2=20°	296.1	146	2	78	83.4	320
H 3444	95	......	140	$\frac{5}{8}$	$3\frac{7}{16}$	+2=20°	300.0	142	8	78	83.4	315
I 3446	103	......	171	$\frac{5}{8}$	$3\frac{7}{16}$	+2=20°	296.0	143	6	78	84.3	330
J {3448	100	......	204	$\frac{5}{8}$	$3\frac{7}{16}$	+2=20°	297.4	140	9	78	81.6	325
J {3452	90	......	206	$\frac{5}{8}$	$3\frac{7}{16}$	+3=15°	296.9	140	9	78	84.3	290

[a] Measured by a thermometer placed in the inlet-valve housing (p. 57).
[b] Plug cock with dial, for regulating the auxiliary air supply which is introduced between the governor-controlled mixture valve and the main inlet valve.
[c] Scale arbitrary; numbers increase with opening of needle valve.
[d] Scale arbitrary; relation to crank position not constant; ignition before dead center is indicated by (+).
[e] Horsepower constants: Brake, 0.000395; indicated, 0.001153 (p. 72).

OPERATOR'S NOTES (TABLE 10).

Tests A to J, inclusive.—Operation of the engine seemingly normal in every respect. Required settings for best fuel economy determined from previous tests and from observation.

economy versus jacket-water temperature.

[NASH 10-HORSEPOWER GASOLINE ENGINE.—Diameter of cylinder, 7⅝ inches; stroke, 10 inches; method of governing, hit-or-miss mixture supply; inlet temperature of jacket water, 60° F.; engine equipped with regular carburetor.]

Brake load.		Horsepower.[e]			Fuel consumption.							Efficiencies (per cent).		
						Per brake horsepower-hour.			Per indicated horse-power-hour.				Thermal.	
Pounds.	Per cent of maximum.	Brake.	Indicated.	Duration (minutes).	Pounds per hour.	Pounds.	Gallons.	B. t. u. (low).	Pounds.	Gallons.	B. t. u. (low).	Mechanical.	Indicated.[f]	Brake.[g]
FUEL: GASOLINE.[h]														
80	64	9.39	11.7	32	5.71	0.608	0.102	11,730	0.49	0.082	9,450	80.5	26.9	21.7
80	64	9.46	11.9	31	5.79	.612	.102	11,760	.49	.081	9,340	79.4	27.3	21.6
80	64	9.39	11.6	32	5.68	.605	.102	11,660	.49	.082	9,430	80.9	26.9	21.8
80	64	9.36	11.5	33	5.44	.581	.098	11,200	.47	.079	9,080	81.0	28.0	22.7
80	64	9.40	11.8	32	5.57	.592	.099	11,410	.47	.079	9,080	79.5	28.0	22.3
FUEL: DENATURED ALCOHOL.[i]														
100	69	11.71	13.9	14	12.70	1.085	0.159	11,590	0.91	0.133	9,710	83.9	26.2	22.0
100	69	11.64	13.6	36	12.05	1.036	.151	11,066	.88	.129	9,420	85.2	27.0	23.0
100	69	11.69	14.0	30	12.08	1.034	.151	11,044	.86	.126	9,175	83.1	27.7	23.0
100	69	11.85	13.6	15	12.41	1.048	.153	11,200	.91	.133	9,710	86.7	26.2	22.7
100	69	11.69	13.9	14	12.49	1.069	.156	11,410	.90	.132	9,640	84.4	26.4	22.3
100	69	11.74	13.2	15	12.41	1.058	.155	11,300	.94	.139	10,080	89.1	25.3	22.5
100	69	11.71	13.6	15	12.27	1.048	.152	11,200	.90	.135	9,660	86.3	26.4	22.7

[f] Calculated from the indicated horsepower and the low heating value of the fuel (p. 72).
[g] Calculated from the brake horsepower and the low heating value of the fuel (p. 72).
[h] Specific gravity at 60° F.=0.7122; heating value: high 20,581, low 19,292 B. t. u. per pound.
[i] Specifiic gravity at 60° F.=0.8188; heating value: high 11,788, low 10,681 B. t. u. per pound. Per cent of alcohol, by weight, 91.5.

TABLE 11.—*Results of trial and check tests made to determine the best fuel economy for various jacket-water temperatures.*[a]

[OTTO 15-HORSEPOWER ALCOHOL ENGINE.—Diameter of cylinder, 6¾ inches; stroke, 15½ inches; method of governing, combination of cut-off and throttling; compression, variable; greatest, 140 pounds per square inch above 1 atmosphere.]

FUEL: DENATURED ALCOHOL.[b]

Fuel needle-valve setting.[c]	Fuel consumption per brake horsepower-hour (pounds).						Air-valve setting.[d]
	Jacket-water temperature, 84° F.						
23	(e)		*0.964*		(e)		10
22		(e)		0.865		(e)	15
22	(e)		.852 .841		(e)		18
25		(e)	.922	(e)			20
26					0.906		
29					.907		
	Jacket-water temperature, 112° F.						
18					(e)	0.895	0
19			(e)		0.844		10
23			*0.929*				
22			(e)		*.803* .792		20
24					.844		
30				(e)	.860	(e)	30
33			*.864*				
36				(e)		.889	40
56						.934	
60				(e)		.893	60
62						.905	
62				(e)		.863	70
	Jacket-water temperature, 144° F.						
22		(e)		*0.812*		(e)	20
	Jacket-water temperature, 189° F.						
22	0.795	0.787	*0.762*			(e)	20
Igniter setting.[f]	+40°	+35°	+30°	+25°	+20°	+15°	

[a] Figures in the body of the table give the fuel consumption in pounds per brake horsepower-hour and are tabulated with respect to fuel needle-valve, air-valve, and igniter settings. Figures in italics are the results of check tests of about 30 minutes' duration. The other figures in the body of the table are the results of trial tests of about 15 minutes' duration.

[b] Specific gravity at 60° F., 0.8206; per cent alcohol by weight, 90.8. Heating value: High, 11,662; low, 10,557 B. t. u. per pound.

[c] Opening of needle valve in hundredths of 1 revolution.

[d] Butterfly valve in air pipe to carburetor set so as to throttle the air and supplement the governor-controlled inlet valve (wide open at 80).

[e] Trial tests were attempted with these needle-valve, air-valve, and igniter settings, but the engine would not carry the load.

[f] Point of ignition, in degrees, is given in terms of the crank position; +=early, or before dead center.

TABLE 12.—*Results of tests of best fuel economy versus jacket-water temperature.*

OTTO 15-HORSEPOWER ALCOHOL ENGINE.—Diameter of cylinder, 6¾ inches; stroke, 15½ inches; method of governing, combination of cut-off and throttling; inlet temperature of jacket water 60° F.; compression, variable; greatest 140 pounds per square inch above 1 atmosphere.]

FUEL: DENATURED ALCOHOL.[a]

Test number.	Temperatures (°F.).			Index settings.			Speed (av. per minute).		Average pressures (pounds per sq. in.).			Brake load.		Horsepower.[f]		Duration (minutes).	Fuel consumption.							Efficiencies (per cent).		
																		Per brake horsepower hour.			Per indicated horsepower hour.				Thermal.	
	Air.	Mixture.[b]	Jacket-water outlet.	Air valve.[c]	Fuel needle valve.[d]	Igniter (degrees).[e]	Revolutions.	Explosions.	Compression (above atmosphere).	Mean effective.	Maximum (above atmosphere).	Pounds.	Per cent of maximum.	Brake.	Indicated.		Pounds per hour.	Pounds.	Gallons.	B. t. u. (low).	Pounds.	Gallons.	B. t. u. (low).	Mechanical.	Indicated.[g]	Brake.[h]
A 2234.	66	55	84	10	23	+30	259.7	129	120	87.8	465	115	68	13.38	15.8	23	12.90	0.964	0.141	10,180	0.82	0.120	8,560	84.7	29.5	25.0
B 2192.	95	69	112	20	22	+20	259.0	130	130	85.6	310	115	68	13.35	15.5	28	10.71	.803	.117	8,480	.68	.099	7,180	86.0	35.4	30.0
C 2238.	74	68	144	20	22	+25	259.6	130	130	87.0	295	115	68	13.38	15.8	28	10.68	.812	.119	8,570	.69	.100	7,260	84.7	35.0	29.7
D 2240.	73	75	189	20	22	+30	260.4	130	130	83.6	310	115	68	13.42	15.2	29	10.23	.762	.111	8,050	.67	.098	7,110	88.4	35.8	31.6

[a] Specific gravity at 60° F.=0.821. Per cent alcohol by weight, 90.8. Heating value: High 11,662; low 10,557 B. t. u. per pound.
[b] Measured by a thermometer placed in the inlet-valve housing (p. 57).
[c] Butterfly valve in air pipe to carburetor set so as to throttle the air and supplement the governor-controlled inlet valve (air valve wide open at 80) (p. 41).
[d] Opening of fuel needle valve in hundredths of one revolution (p. 55).
[e] Time of ignition, in degrees, is given in terms of the crank position; +=early, or before dead center (p. 55).
[f] Horsepower constants: Brake 0.0004478; indicated 0.001396 (p. 67).
[g] Calculated from the indicated horsepower and low heating value of the fuel (p. 72).
[h] Calculated from the brake horsepower and low heating value of the fuel (p. 72).

OPERATOR'S NOTES (TABLE 12).

Test A 2234.—For the low jacket temperature given the engine would not carry the load with the settings found to be best for the higher jacket temperatures and the conditions of operation were not stable except when a relatively rich mixture was used. The seven trial tests were made with air-valve settings of 15, 18, and 20, and the fuel needle-valve and ignition timings were varied accordingly from 29 and +20° to 22 and +30°, with corresponding fuel consumptions of 0.91 and 0.87 pound per brake horsepower per hour. With air valve set at 18, fuel needle valve at 22, and igniter at +30°, the fuel consumption was 0.84 pound per brake horsepower per hour, but these values could not be checked and the settings given in the table which partially throttled the engine were resorted to in order to obtain stable operating conditions.

Test B 2192.—The 27 trial tests were made with air-valve settings of from 0 to 70, the best of which was found to be 20. The fuel consumptions for these tests varied from 0.95 pound per brake horsepower per hour to the values given in the table.

Test C 2238.—The best settings for the jacket temperature given for this test were very nearly the same as for the preceding test (B 2192), the only apparent change that could be made tending to reduce the fuel consumption being an advance in the time of ignition.

Test D 2240.—As for the preceding test (C 2238), the only apparent advantageous change in setting for this jacket temperature was a still earlier ignition timing. The two trial tests that were made were different from the test given in the table only in having igniter settings of +35° and +40°, the fuel consumptions for these two tests being 0.797 and 0.795 pound per brake horsepower per hour, respectively.

CONCLUSIONS.

A study of the results tabulated in Tables 7 to 12 brings out the following important features:

1. When the jacket-water temperature is the only variable of moment the fuel consumptions and efficiencies of the individual engines using either alcohol or denatured alcohol are very nearly constant.

2. These same fuel consumptions and efficiencies when platted as curves do not show characteristics that warrant the conclusion that jacket-water temperature affects them appreciably.

3. The variation between individual tests is no greater, in most cases, than the probable error of observation.

When these three features are considered in connection with the fact that the tests were run on four different engines having different mechanical details entering into their construction, different types of governors, different arrangements of water jackets, and using denatured alcohol and gasoline as fuel, the conclusion seems warranted that in so far as the tests covered in the entire investigation are concerned the jacket-water temperatures are negligible, provided the operating conditions are kept up to standard and the cylinder is not allowed to get hot enough to cause preignition. In making comparison of various results obtained in this bulletin, the jacket-water temperature is therefore considered eliminated as a cause of variation of economy.

Whether it is safe to assume that this investigation proves that the thermal efficiency of all internal-combustion engines is independent of the temperature of the jacket water is quite another question and one which should lend itself to further investigation. Experience obtained in this series of tests, however, has verified the opinion of the authors that the effects upon thermal efficiency usually charged to the jacket-water temperature are really not a result of the jacket-water temperature itself, but rather of secondary mechanical operating conditions affected by the jacket-water temperature.

SPEED OF ENGINE.

In an attempt to study the relative economies of gasoline and denatured alcohol in internal-combustion engines there are many variables, incident upon the normal operation of the engine, which may possibly affect the accuracy of the comparison. It is therefore important that the isolated effects of all such variables be studied prior to the time when the final comparison is made in order to be assured that the comparison is based solely upon characteristics of the two fuels and not upon secondary variables whose elimination is desired.

Among the variables encountered in the operation of engines, such as used in this investigation, is the speed. The governors of the several engines maintained their speeds within 5 per cent of the normal under all ordinary circumstances, and there was a difference of about 15 per cent between the normal speed of the Otto and the Nash engines. In an investigation of this kind the question might easily arise as to whether these small variations do not affect the results upon which some of the conclusions are based.

EFFECT ON EFFICIENCY.

Theoretically there are five most important ways in which it is possible for the speed to affect the economy of internal-combustion engines. These are enumerated below.

1. If the passageways through which the fuel vapors, mixed with air, are drawn into the cylinder are small, an increase in speed might easily reduce the volume of each charge, thus reducing the compression and consequently causing the thermal efficiency to drop.

2. Both the fluid-friction and bearing-friction losses, as shown by the mechanical efficiency may change with the speed, the former varying directly with the velocity of the piston and flywheels, and the latter being a direct function of the distance through which the bearing surfaces travel.

3. The speed of the engine may affect the economy in precisely the same manner as may the time of ignition (discussed under "Time of Ignition"), that is, through the time interval allowed for the combustion of the fuel vapors between the moment of ignition and the moment of exhaust and through the loss of heat during the cycle.

4. With a variation in speed there may be a variation in the completeness with which the exhaust gases are expelled and a variation

in the homogeneity of the charge of fuel vapors and air, causing a variation in the efficiency of combustion.

5. An increase in speed may make possible the advancing of the time of ignition to such an extent as would otherwise be impossible, thus increasing the opportunity for obtaining combustion at the time that will produce the most perfect expansion of the hot gases.

The possible effect of a change in compression, coincident with the change in speed, can be eliminated by the use of an engine having induction passageways large enough so that the volumetric efficiency of the engine will be practically constant. All of the other results which might be expected to follow changes in speed are of such a nature as to be unavoidable and it is to these features that one must look for an explanation of the changes in economy which will be brought about by changes in speed.

The time interval necessary for the consummation of the physical and chemical reactions incident to the combustion of the fuel in the engine cylinder may be different for gasoline than for denatured alcohol. However, this difference should be compensated for by the igniter adjustment if the most favorable adjustment for each speed was maintained throughout the series of tests.

METHOD OF TESTING.

Having in view the points brought out, it was determined to run tests of such a character that the general effect of speed upon economy might be determined. The Otto gasoline engine No. 1 seemed the best suited to this purpose (1) because of the general features of its construction (for description see "Apparatus used in tests," p. 28) and (2), because it was used in more of the tests of this investigation than any other engine. It would thus give results permitting not alone a comparative but also an exact application to a large part of this work.

The series of tests outlined consisted of so-called "best economy versus speed" tests (Tables 13 and 14) in which the governor of the engine was so adjusted as to give speeds of approximately 200, 250, and 300 revolutions per minute, while the brake load was kept constant at that giving the best fuel economy. Then at each speed trial tests were run with many different combinations of "fuel needle-valve settings" and "ignition settings" until the best fuel economy obtainable was reached. These tests were followed with check tests of longer duration.

RESULTS OBTAINED.

The fuel consumptions obtained with both gasoline and denatured alcohol on the search and check tests are shown in Table 13, whereas Table 14 gives the detailed tabulated results of all the check tests.

The efficiencies and fuel consumptions are shown graphically in figure 23.

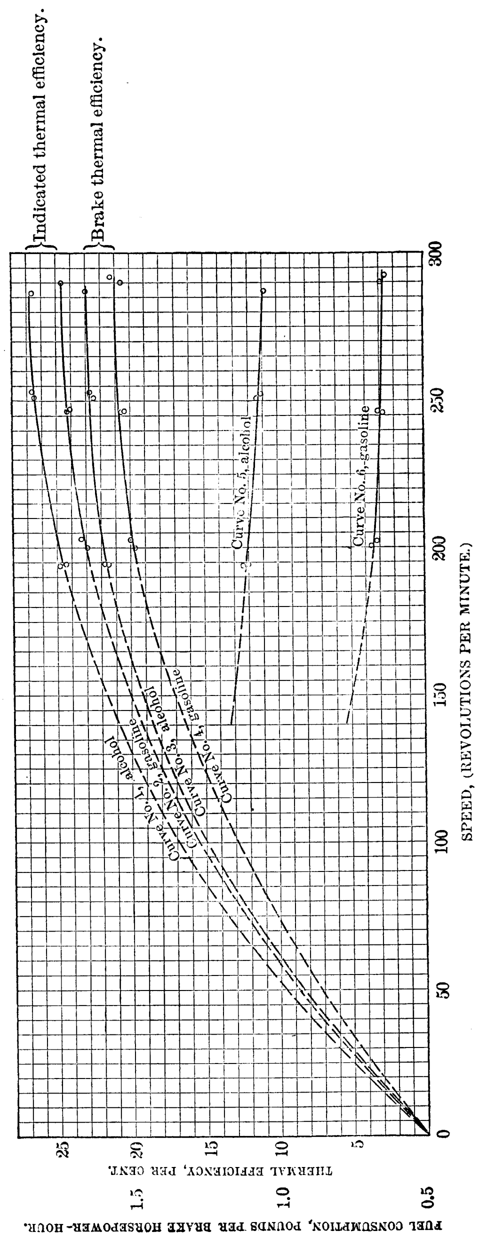

FIGURE 23.—Curves showing relations between speed, thermal efficiency, and fuel consumption with Otto gasoline engine No. 1.

TABLE 13.—*Results of trial and check tests made to determine the best fuel economy for various speeds.*[a]

[OTTO 15-HORSEPOWER GASOLINE ENGINE NO. 1.—Compression, 70 pounds per square inch above 1 atmosphere; diameter of cylinder, 6¾ inches; stroke, 15½ inches; method of governing, hit-or-miss fuel supply; jacket-water temperature: Inlet 60°, outlet 112° F.]

FUEL: GASOLINE.[b]
[Brake load, 85 per cent of greatest.]

Fuel needle-valve setting.[d]	Fuel consumption per brake horsepower-hour, pounds.					
	(Speed, 197 to 203 r. p. m.; brake horsepower, 10.4 to 10.8.)					
32		([e])	0.697	0.678 *.668*	0.676 *.689*	0.711
34			.695	.688	.698	
	(Speed: 246 to 248 r. p. m.; brake horsepower, 13.0 to 13.2.)					
34	([e])	0.655	.645 *.653*	.661		
36		.679	.674	.670		

FUEL: DENATURED ALCOHOL.[c]
[Brake load, 82 per cent of greatest.]

Fuel needle-valve setting.[e]	Fuel consumption per brake horsepower-hour, pounds.						
	(Speed, 194 to 199 r. p. m.; brake horsepower, 10.7 to 11.1.)						
42			1.124	*1.119* 1.106			
44				1.140			
46				1.131	1.147	1.182	1.193
	(Speed: 250 to 253 r. p. m.; brake horsepower, 13.9 to 14.0.)						
44		1.071	*1.069* *1.076*	1.071 *1.062*			

	(Speed: 288 to 293 r. p. m.; brake horsepower, 15.2 to 15.5.)					
34			.625			
36	0.630	.627 .639 *.631*				
38	.678	.651 *.644*	.668			
40	.629	.622 .652	.653			
44		.660				
Igniter settings.*f*	+21°	+17°	+13°	+9°	+5°	0°

	(Speed: 287 to 289 r. p. m.; brake horsepower, 15.9 to 16.0.)						
48	*1.047*						
50	1.077	1.087					
Igniter settings.*f*	+38°	+34°	+30°	+25°	+21°	+17°	+13°

a The figures in the body of the table give the fuel consumption in pounds per brake horsepower per hour and are tabulated with respect to the fuel needle-valve and igniter settings used. Results of tests of about 30 minutes' duration are indicated by figures in italics. Other tests are of about 15 minutes' duration.

b Specific gravity at 60° F.=0.729. Heating value: High=20,407, low=19,125 B. t. u. per pound.

c Specific gravity at 60° F.=0.819; per cent alcohol by weight=91.5; heating value: High=11,675, low=10,573 B. t. u. per pound.

d Opening of needle valve in hundredths of 1 revolution.

e Trial tests were attempted with these needle-valve and igniter settings, but the engine would not carry the load.

f Point of ignition in degrees, is given in terms of the crank position. (+=early, or before dead center.)

TABLE 14.—*Results of tests of best fuel economy versus speed.*

[OTTO 15-HORSEPOWER GASOLINE ENGINE No. 1.—Compression pressure, 70 pounds per square inch above 1 atmosphere; diameter of cylinder, 6¾ inches; stroke, 15½ inches; method of governing, hit-or-miss fuel supply; jacket-water temperature: Inlet 60°, outlet, 112° F.]

Test number.	Temperatures (° F.). Air.	Fuel.	Mixture.[a]	Index settings. Fuel needle valve.[b]	Igniter (degrees).[c]	Speed (average per minute). Revolutions.	Piston (feet).	Explosions.	Fuel cut-outs.	Average pressures (pounds per sq. in.). Mean (effective).	Maximum (above atmosphere).	Brake load. Pounds.	Per cent of maximum.	Horsepower.[d] Brake.	Indicated.	Duration (minutes).	Fuel consumption. Pounds per hour.	Pounds per charge.	Per brake horsepower-hour. Pounds.	Gallons.	B. t. u. (low).	Per indicated horsepower-hour. Pounds.	Gallons.	B. t. u. (low).	Efficiencies (per cent). Mechanical.	Thermal. Indicated.[e]	Brake.[f]
FUEL: GASOLINE.[g]																											
A 1094	86	89	102	32	+ 9	200.7	518	95	6	92.7	280	110	85	10.63	12.3	17	7.20	0.00126	0.678	0.111	12,960	0.59	0.097	11,220	86.6	22.7	19.6
A 1099	83	86	106	32	+ 9	203.2	523	93	9	96.2	320	110	85	10.77	12.5	33	7.20	.00129	.668	.110	12,780	.59	.097	11,000	86.0	23.1	19.9
B 1106	88	96	107	34	+13	246.5	636	116	7	94.2	350	110	85	13.05	15.3	14	8.42	.00121	.645	.106	12,340	.55	.090	10,510	85.3	24.2	20.6
B 1111	90	98	110	34	+13	246.9	637	116	8	95.2	340	110	85	13.08	15.5	28	8.55	.00123	.653	.107	12,490	.55	.090	10,600	84.8	24.0	20.4
C 1130	73	81	90	40	+21	292.0	754	135	11	95.3	340	110	85	15.47	18.0	15	9.73	.00120	.629	.103	12,030	.54	.089	10,330	86.0	24.6	21.2
C 1144	81	89	94	38	+17	290.1	750	140	5	92.7	290	110	85	15.38	18.2	30	9.89	.00118	.644	.106	12,320	.54	.089	10,390	84.5	24.5	20.7
FUEL: DENATURED ALCOHOL.[h]																											
D 1425	93	92	69	42	+25	194.7	503	93	5	93.8	280	115	82	10.78	12.2	15	11.93	0.00214	1.106	0.161	11,700	0.98	0.143	10,370	88.6	24.5	21.7
D 1427	92	90	68	42	+25	193.9	501	89	4	96.8	280	115	82	10.74	12.1	30	12.02	.00225	1.119	.163	11,830	.99	.145	10,540	88.8	24.2	21.5
E 1409	82	82	69	44	+30	251.3	650	118	4	99.0	330	115	82	13.92	16.4	32	14.88	.00203	1.069	.156	11,320	.91	.133	9,640	85.1	26.4	22.5
E 1417	88	92	70	44	+25	253.1	654	123	2	95.6	300	115	82	14.02	16.4	32	14.89	.00198	1.062	.155	11,230	.91	.133	600	85.5	26.5	22.7
F 1432	92	92	66	48	+38	287.1	742	139	2	96.4	320	115	82	15.91	18.8	29	16.65	.00197	1.047	.153	11,070	.89	.130	9,510	84.9	26.8	23.0

[a] Measured by a thermometer placed in the inlet-valve housing (p. 57).
[b] Opening of the fuel needle valve in hundredths of one revolution (p. 55).
[c] Point of ignition in degrees is given in terms of the crank position; +=early, or before dead center (p. 55).
[d] Horsepower constants: Brake=0.0004817; indicated=0.001400 (p. 67).
[e] Calculated from the indicated horsepower and the low heating value of the fuel (p. 72).
[f] Calculated from the brake horsepower and the low heating value of the fuel (p. 72).
[g] Specific gravity at 60° F.=0.729. Heating value: High=20,407; low=19,125 B. t. u per pound.
[h] Specific gravity at 60° F.=0.819. Per cent alcohol by weight=91.5. Heating value: High=11,675, low=10,573 B. t. u. per pound.

OPERATOR'S NOTES (TABLE 14).

Test A.—The 8 trial tests covered a range of settings from needle valve 34 and igniter +5°, with a corresponding fuel consumption of 0.70 pound per brake horsepower per hour, to the values given in the table. The operation of the engine was too unsteady at slower speeds for accurate tests.

Test B.—The 6 trial tests covered a range of settings from needle valve 36 and igniter +17°, with a corresponding fuel consumption of 0.68 pound per brake horsepower per hour, to the values given in the table. In test B 1111 a very light explosion pound was noted.

Test C.—The 15 trial tests covered a range of settings from needle valve 44 and igniter +17°, with a corresponding fuel consumption of 0.66 pound per brake horsepower per hour, to the values given in the table. The engine vibration was too great to allow for steady running at higher speeds. In test C 1144, a light explosion pound was noted.

Test D.—The 8 trial tests covered a range of settings from needle valve 46 and igniter +13° to needle valve 42 and igniter +25°, with corresponding fuel consumption figures from 1.19 pounds of fuel per brake horsepower per hour to the values given in the table.

Test E.—The 5 trial tests covered a range of settings from needle valve 44 and igniter +34° to needle valve 44 and igniter +25°, with corresponding fuel consumption figures from 1.08 pounds of fuel per brake horsepower per hour to the values given in the table. In test E 1409, the first fuel charge after cut-outs nearly always misfired, due seemingly to the action of the governor causing irregularities in the quality of the mixture, which sometimes became too lean to burn. Electric igniter not at fault. In test E 1417, the first fuel charge after cut-outs always misfired, due seemingly to the same cause.

Test F.—The 3 trial tests covered a range of settings from needle valve 50 and igniter + 34° to needle valve 48 and igniter +38°, with corresponding fuel consumption figures from 1.09 pounds of fuel per brake horsepower per hour to that given in the table. In test F 1432, the first fuel charge after cut-outs occasionally misfired. Igniter not at fault.

CONCLUSIONS.

By reference to the tables it can be seen that the mechanical efficiencies of the engine vary inversely with the speed. But in spite of this, the thermal efficiency of the engine increases with the speed sufficiently to more than counteract the mechanical losses shown. As a result, the fuel economy of the engine, relative to the available horsepower at the brake, shows an appreciable gain with increase of speed. The changes in economy effected by speed variation are alike with either gasoline or denatured alcohol as the fuel. This is shown by the exact similarity of the groups of curves (fig. 23).

The gain in efficiency, based on brake horsepower, varying 100 revolutions per minute, however, is somewhat less than 2 per cent. It is therefore obvious that the differences in speed that were encountered in the routine of tests involved in this investigation are negligible.

Reference to page 125, where five hypothetical ways in which speed might affect engine economy are given, will make clear that the cause of increased thermal efficiency with increased speed may have been due to the following variables: (1) decrease in heat losses due to a decrease in the time interval during which they take place; (2) more efficient expansion because of an advance in the time of ignition beyond that required to compensate for the tendency of an increase in speed to decrease the time interval between ignition and exhaust.

In the light of the conclusion above referred to, further investigation of the details of engine design and operation should lead to interesting and valuable information regarding speeds of internal-combustion engines.

LOAD VARIATION.

From an economical standpoint the load is perhaps the most important variable in the operation of internal-combustion engines. This class of engine is rarely called upon to sustain a constant or steady load, and therefore in the investigation a most exhaustive study of this variable in its effect upon engine efficiency was deemed essential.

This report has already treated in detail the effect of variables, such as mixture quality, time of ignition, current and potential used in electric-ignition circuit, jacket-water temperature, and speed of engine, all of which are subject to control by the operator.

Remaining to be studied is engine behavior as affected by the load, a factor which is controlled only by the conditions under which the engine works. The information already obtained relative to the effect of the many engine adjustments upon economy of operation becomes of great value in arranging a tentative outline of tests having as their object a determination of the effect of load on engine efficiency.

EFFECT ON EFFICIENCY.

A change in the brake load that an engine has to maintain is the variable which will cause perhaps the greatest variation in the economy with which the fuel is used. There are many contributing causes that may combine to bring about such a variation. There can be no doubt that the constant frictional horsepower (see "Mechanical efficiency") that at all times has to be overcome by a part of the energy supplied by the fuel will explain in a large part why the over-all efficiency of an engine is always low at low loads. The over-all efficiency of an engine may be considered as the product of the thermal efficiency into the mechanical efficiency. ("Procedure of tests," pp. 71 to 72.) Hence it appears that a dropping of the mechanical efficiency (fig. 16) coincident with a lowering of the load is in itself sufficient explanation for a considerable part of the lowering in over-all efficiency.

However, the thermal efficiency also falls off appreciably at low loads, thus increasing the cumulative effect on the economy relative to the load at the brake. It is of value, then, before outlining the method of procedure for the load-variation tests, to consider a few hypothetical causes which may contribute to the changes in thermal efficiency incident to change of load. The effect of mechanical efficiency is considered as definitely settled by previous experiment and will therefore be eliminated from further discussion.

The manner in which low load affects thermal efficiency will be materially different for throttle-governed engines than for hit-or-miss governed engines. With the former a reduction in load is attended with a reduction in the volume of the explosive mixture entrained in the cylinder prior to compression, and the resulting compression pressure is reduced. A subsequent section ("Compression") shows that efficiency varies directly with compression pressure. On the other hand, with a hit-or-miss governed engine, the number of cut-outs increases as the load is reduced, and this unstable condition of governing predisposes to misfiring and inefficient combustion, with a consequent lowering of efficiency.

Whether gasoline or denatured alcohol fuel may affect the efficiency differently at light loads with the hit-or-miss governed engine depends upon the different temperatures at which vaporization takes place. For example, light loads, with many and frequent cut-outs and consequent cooling of the cylinders and passageways, will effect a greater loss in efficiency with the low vaporization point of the denatured alcohol than with gasoline.

METHODS OF TESTING.

By reference to sections on "Mixture quality" (p. 81) and "Time of ignition" (p. 97) it may be seen that by proper manipulation of the fuel needle valve and of the igniter respectively it is possible to obtain widely varying efficiencies at a constant load. Plainly, therefore, a series of tests to determine the relation between load and efficiency can lead to no definite results unless certain restrictions be imposed as to the selection of tests. The Otto gasoline engine No. 1, using gasoline for fuel, shows a range in fuel consumed from 0.624 to 1.309 pounds per brake horsepower-hour for a change in needle-valve settings from 30 to 70 (Table 3), whereas the load remains constant. Similarly, the Otto gasoline engine No. 2, using gasoline for fuel, shows a range in fuel consumed from 0.760 to 0.915 pounds per brake horsepower-hour for a change in igniter settings from $+20°$ to $-20°$ (Table 5), and the load remains constant.

In view of the wide variation obtainable in economy at any stated load it was determined to divide the tests correlating economy and load into three separate and distinct series called "best economy versus load" tests, "normal economy versus load" tests, and "poorest economy versus load" tests. The "best economy" and "poorest economy" tests represent limiting conditions of efficiency between which all practical operations of the engine must fall. The "normal economy" tests represent arbitrary intermediate efficiencies which more nearly approximate results reasonably expected in practice.

In order to obtain the best fuel economy an exhaustive series of search tests was conducted at each load in the same manner as outlined in the discussions on the effect of "jacket-water temperature" and of "speed." From the knowledge already obtained relative to the effect of numerous engine settings it was possible to adjust conditions of operation at each load so as to approximate the best obtainable efficiencies at the start. Then by repeated readjustment and trial in short tests of 15 minutes' duration the lowest fuel consumption obtainable was reached with the engine maintaining its speed, and later verified by a check test of 30 minutes' duration in which the engine adjustments were identical with those existing in the trial test giving the best economy. In order that there might be no question as to the validity of the tests, practically no results are tabulated, except of the check tests. It is therefore safe to say that no figures are given in the table other than those which can be definitely repeated in practice.

Having determined the "best economy" as already outlined, the information becomes available for determining the "normal economy." Disregarding load, the test in the "best-economy" series giving the lowest fuel consumption was ascertained and the adjustments used in it were kept constant throughout a series of tests in which the load alone was the only variable. The above arrangement is considered as nearly approximating normal operating conditions where the operator adjusts his engine for its normal load and keeps his adjustments constant, or nearly so, during irregular fluctuations of load.

The lack of efficiency with which fuel can be used in internal-combustion engines is almost limitless as far as reasonable results are concerned, and it is therefore necessary to have some criterion which shall stand as an indication of a reasonable limit. In the "best fuel economy" tests the failure of the engine to maintain its rated speed may be considered as the criterion of the limiting condition of economical engine operation beyond which outward appearances of a failure of the engine to do its work become evident. Similarly, the appearance of a smoky exhaust showing incomplete combustion or a frequent back-firing is considered the criterion of the limiting condition of uneconomical engine operation, beyond which outward appearances of the failure of the engine to do economical work become evident. The practical operating efficiency of the engine may be found at any point between the limits as established by these two criteria. The personality of the operator is the only feature to determine whether the efficiency more nearly approaches the best or poorest condition.

Because of the different ways in which load may be expected to affect efficiency, depending upon the fuel used and the method of governing the engine, it was determined to make this series of tests most exhaustive. Tests were therefore run on all five engines, using both gasoline and denatured alcohol as the fuel with each except in the case of the Otto alcohol engine, which was not adapted to the use of gasoline because of its high compression. With the Otto gasoline engine No. 1 an additional series of tests, with denatured alcohol and the air preheated to 250° F., was run. With the Nash engines a special carburetor (pp. 34 and 37) was used in an additional series with denatured alcohol as fuel. The Otto alcohol engine which could not be used with gasoline was used in these series with denatured alcohol, each series being with a different compression pressure.

RESULTS OBTAINED.

The results of the series of tests conducted to determine the effects of load variation are shown in the following tabulations (Tables 15 to 31):

TABLE 15.—*Results of trial and check tests made to determine the best fuel economy for each load.*[a]

[OTTO 15-HORSEPOWER GASOLINE ENGINE NO. 1.—Compression pressure, 70 pounds per square inch above atmosphere; diameter of cylinder, 6¾ inches; stroke, 15½ inches; speed, 254 to 266 revolutions per minute; hit-or-miss fuel-supply governor; jacket-water temperature: Inlet, 60° F.; outlet, 112° F.]

FUEL: GASOLINE.[b]

Fuel needle-valve setting.[c]	Fuel consumption per brake horsepower-hour, pounds.													
	Brake load, 130 pounds; B. H. P., 15.8 to 16.0.													
36											(d)	(d)		(d)
38											(d)	0.675		0.673 *.655*
40											(d)	.715		.697
Igniter settings.[e]	+42°	+38°	+34°	+30°	+25°	+21°	+17°	+13°	+11°	+9°	+7°	+5°	+3°	0°

[a] Figures in the body of the table give the fuel consumption in pounds per brake horsepower-hour and are tabulated with respect to fuel needle-valve and igniter settings. The figures in italics are the result of check tests of about 30 minutes' duration. The other figures in the body of the table are the results of trial tests of about 15 minutes' duration.

[b] Specific gravity at 60° F.=0.729. Heating value: High=20,401; low=19,120 B. t. u. per pound.

[c] Opening of needle-valve in hundredths of one revolution.

[d] Trial tests were attempted with these needle-valve and igniter settings, but the engine would not carry the load.

[e] Point of ignition in degrees is given in terms of the crank position; +=early, or before dead center.

TABLE 15.—*Results of trial and check tests made to determine the best fuel economy for each load*—Continued.

[OTTO 15-HORSEPOWER GASOLINE ENGINE NO. 1.—Compression pressure, 70 pounds per square inch above atmosphere; diameter of cylinder, 6¾ inches; stroke, 15½ inches; speed, 254 to 266 revolutions per minute; hit-or-miss fuel-supply governor; jacket-water temperature: Inlet, 60° F.; outlet, 112° F.]

FUEL: GASOLINE—Continued.

Fuel needle-valve setting.	Fuel consumption per brake horsepower-hour, pounds.													
	Brake load, 115 pounds; B. H. P., 14.3 to 14.6.													
28									(a)	(a)				
30									(a)	0.605	(a)			
32								(a)	0.633		0.607	0.634	(a)	
34								(a)		.599 .608 *.613*	.634	.650		(a)
36									.606 *.627*	.646	.635		0.666	
38									.631					
	Brake load, 110 pounds; B. H. P., 13.6 to 13.9.													
26					(a)	(a)	(a)							
28					(a)	0.583 *.575*	0.595	(a)						
30				(a)	0.608	.593 *.590*	.601	0.606	0.613	0.625	(a)			
32						.633	.612		.633	.624	0.636			
34										.632	.641			
	Brake load, 100 pounds; B. H. P., 12.4 to 12.7.													
26					(a)									
28				(a)	0.609	0.609 *.598*	0.610 *.621*	0.626	(a)					
30						.630	.622	.624	0.636					
32						.632	.613 *.624*	.616 *.637*	.622	0.650				
34							.622	.630	.633					
Igniter settings.[b]	+42°	+38°	+34°	+30°	+25°	+21°	+17°	+13°	+11°	+9°	+7°	+5°	+3°	0°

[a] Trial tests were attempted with these needle-valve and igniter settings, but the engine would not carry the load.
[b] Point of ignition in degrees is given in terms of the crank position; + = early, or before dead center.

TABLE 15.—*Results of trial and check tests made to determine the best fuel economy for each load*—Continued.

[OTTO 15-HORSEPOWER GASOLINE ENGINE NO. 1.—Compression pressure, 70 pounds per square inch above atmosphere; diameter of cylinder, $6\frac{3}{4}$ inches; stroke, $15\frac{1}{2}$ inches; speed, 254 to 266 revolutions per minute; hit-or-miss fuel-supply governor; jacket-water temperature: Inlet, 60° F.; outlet, 112° F.]

FUEL: GASOLINE—Continued.

Fuel needle-valve setting.	Fuel consumption per brake horsepower-hour, pounds.													
	Brake load, 80 pounds; B. H. P., 10.0 to 10.2.													
24	0.724													
26	.667	0.730	0.722	0.672	0.687			0.750						
28			.679	.682	.667 *.677*	0.681	0.684		0.722					
30			.688	.698	.682	.669 *.692*	.682	.695	.692	0.722				
32						.686	.682	.690	.704					
34								.696	.699	.731				
36								.693	.697					
38									.690					
40									.709					
	Brake load, 60 pounds; B. H. P., 7.6 to 7.7.													
26				0.850	0.850	0.883	0.883	0.917	0.967					
28				.831	.802	.800	.800	.797	.851					
30				.812	.800	.812	.807	.832	.847					
32				.806	.826		.809	.837	.840					
34				.790			.807	.795	.825					
36									.797	0.797				
38									.781					
40								.765	.772	.786				
42								.742 *.746*	.741 *.756*					
44								.759	.757 .779	.759				
46									.757	.755	0.758			
48										.773				
Igniter settings.[a]	+42°	+38°	+34°	+30°	+25°	+21°	+17°	+13°	+11°	+9°	+7°	+5°	+3°	0°

[a] Point of ignition in degrees is given in terms of the crank position; + = early, or before dead center.

TABLE 15.—*Results of trial and check tests made to determine the best fuel economy for each load*—Continued.

[OTTO 15-HORSEPOWER GASOLINE ENGINE NO. 1.—Compression pressure, 70 pounds per square inch above atmosphere; diameter of cylinder, 6¾ inches; stroke, 15½ inches; speed, 254 to 266 revolutions per minute; hit-or-miss fuel-supply governor; jacket-water temperature: Inlet, 60° F.; outlet, 112° F.]

FUEL: GASOLINE—Continued.

Fuel needle-valve setting.	Fuel consumption per brake horsepower-hour, pounds.													
	Brake load, 40 pounds; B. H. P., 5.1 to 5.2.													
30				0.987										
40									0.961					
42									.875					
44								0.880	.874					
46							0.887	.880	.914 *.856* *.867* *.887*	0.895	0.908			
48							.861	.857 *.855*	.883 *.858* .873	.873				
50								.874	.868	.873				
Igniter settings.[a]	+42°	+38°	+34°	+30°	+25°	+21°	+17°	+13°	+11°	+9°	+7°	+5°	+3°	0°

FUEL: DENATURED ALCOHOL.[b]

Fuel needle-valve setting.	Fuel consumption per brake horsepower-hour, pounds.											
	Brake load, 140 pounds; B. H. P., 17.1 to 17.3.											
60			(c)	(c)	(c)	(c)		(c)				
62				(c)	1.116	1.094		1.192	(c)			
64				(c)		1.101	*1.127*	1.086 *1.119* *1.126* *1.130* *1.149*		1.115		(c)
66		(c)			1.130	1.088 1.091 1.140 1.134		1.096				(c)
Igniter settings.[c]	+34°	+30°	+25°	+21°	+17°	+13°	+11°	+9°	+7°	+5°	+3°	0°

[a] Point of ignition in degrees is given in terms of the crank position; +=early, or before dead center.
[b] Specific gravity at 60° F.=0.820. Per cent alcohol by volume=92.6; by weight=91.0. Heating value: High=11,604 B. t. u. per pound; low=10,500 B. t. u. per pound.
[c] Trial tests were attempted with these needle-valve and igniter settings, but the engine would not carry the load.

TABLE 15.—*Results of trial and check tests made to determine the best fuel economy for each load*—Continued.

[OTTO 15-HORSEPOWER GASOLINE ENGINE NO. 1.—Compression pressure, 70 pounds per square inch above atmosphere; diameter of cylinder, 6¾ inches; stroke, 15½ inches; speed, 254 to 266 revolutions per minute; hit-or-miss fuel-supply governor; jacket-water temperature: Inlet, 60° F.; outlet, 112° F.]

FUEL: DENATURED ALCOHOL—Continued.

Fuel needle-valve setting.	Fuel consumption per brake horsepower-hour, pounds.											
	Brake load, 130 pounds; B. H. P., 15.9 to 16.2.											
52			(a)	(a)	(a)	(a)	(a)	(a)	(a)			
54					(a)	1.089 *1.071* *1.060*	1.046 *1.076*	1.087	(a)			
56				(a)		1.093		1.090 *1.102* *1.082*		(a)		
58								1.093				
60								1.129				
62								1.134				
64								1.151				
66								1.187				
	Brake load, 125 pounds; B. H. P., 15.2 to 15.5.											
46			(a)	(a)	(a)	(a)	(a)	(a)				
48				(a)	1.038	1.042	1.048		(a)			
50			(a)		1.049 *1.049*	1.053	1.063	1.073				
52				*1.048*			1.076					
54				*1.082*			1.102					
	Brake load, 120 pounds; B. H. P., 14.6 to 14.8.											
44	(a)	(a)	(a)	(a)	(a)		(a)					
46	(a)	1.030	1.017 *1.045*	1.017 1.040	1.040 *1.056*		(a)					
48	(a)		1.057		1.065	1.070						
Igniter settings.[a]	+34°	+30°	+25°	+21°	+17°	+13°	+11°	+9°	+7°	+5°	+3°	0°

[a] Trial tests were attempted with these needle-valve and igniter settings, but the engine would not carry the load.

TABLE 15.—*Results of trial and check tests made to determine the best fuel economy for each load*—Continued.

[OTTO 15-HORSEPOWER GASOLINE ENGINE No. 1.—Compression pressure, 70 pounds per square inch above atmosphere; diameter of cylinder, 6¾ inches; stroke, 15½ inches; speed, 254 to 266 revolutions per minute; hit-or-miss fuel-supply governor; jacket-water temperature: Inlet, 60° F.; outlet, 112° F.]

FUEL: DENATURED ALCOHOL—Continued.

Fuel needle-valve setting.	Fuel consumption per brake horsepower-hour, pounds.											
	Brake load, 115 pounds; B. H. P., 14.1 to 14.5.											
42	(a)	(a)										
44	1.050	1.050 *0.993* *1.000* 1.008		(a)								
46		1.098	1.073 1.069 *1.040*	1.075 1.052 *1.040*								
48			*1.025* *1.064*	1.071								
52				1.122	1.097	1.098	1.102	1.111	1.122	1.143	1.178	1.186
58											1.192	1.120
64												1.310
70												1.430
	Brake load, 110 pounds; B. H. P., 13.7 to 14.0.											
42	(a)											
44	1.076	1.069 *1.105*	1.074 *1.075*									
48		1.123	1.093 *1.096*	1.092 *1.087*	1.097							
50			1.116	1.102	1.094	1.108						
52				1.187	1.181	1.199						
	Brake load, 100 pounds; B. H. P., 12.4 to 12.6.											
41	1.097	1.081	1.081	1.076								
42		1.102 *1.134*	1.097	1.086	1.094							
46		1.132 *1.148*	1.115	1.116	1.118	1.134						
50				1.236	1.235	1.233						
Igniter settings.a	+34°	+30°	+25°	+21°	+17°	+13°	+11°	+9°	+7°	+5°	+3°	0°

a Trial tests were attempted with these needle-valve and igniter settings, but the engine would not carry the load.

Table 15.—*Results of trial and check tests made to determine the best fuel economy for each load*—Continued.

FUEL: DENATURED ALCOHOL—Continued.

Fuel needle-valve setting.	Fuel consumption per brake horsepower-hour, pounds.											
	Brake load, 80 pounds; B. H. P., 10.0 to 10.2.											
40	1.208	1.205	1.188	1.351								
42	1.204	1.189 *1.311*	1.200	1.199 *1.414*								
48		1.256	1.228	1.305								
50		1.221	1.231 1.212 *1.258*	1.211 1.243 *1.243*								
52		1.283	1.201 *1.273* *1.337*	1.211								
54			1.299	1.274	1.279							
	Brake load, 60 pounds; B. H. P., 7.6 to 7.7.											
42	1.574	1.431 1.550	1.567									
44		1.407	1.406	1.365	1.432							
46				1.412	1.407	1.565						
54	1.338	1.338 *1.351*	1.326 *1.480*	1.338 *1.484*								
56	1.445	1.298 1.307 *1.354* *1.485*	1.356 1.347 *1.484*	1.311 *1.488*								
58			1.401	1.350	1.371							
60				1.407	1.380	1.392						
62		1.465	1.458	1.421	1.387	1.410						
	Brake load, 40 pounds; B. H. P., 5.0 to 5.2.											
44			1.775		1.898							
46		1.755	1.722	1.753	1.756							
48			1.737	1.752								
56		1.652	1.641	1.642	1.634 1.763	1.552 1.763		1.603 1.871				
58					1.667	1.663		1.790				
60			1.647	1.637	1.575 *1.623*	1.614 *1.687*		1.645				
62			1.647	1.572 *1.596*	1.617	1.724						
64			1.617	1.600	1.640							
Igniter settings.[a]	+34°	+30°	+25°	+21°	+17°	+13°	+11°	+9°	+7°	+5°	+3°	0°

TABLE 15.—*Results of trial and check tests made to determine the best fuel economy for each load*—Continued.

FUEL: DENATURED ALCOHOL—Continued.

(Air preheated to 250° F.)

Fuel needle-valve setting.	Fuel consumption per brake horsepower-hour, pounds.							
	Brake load, 120 pounds; B. H. P., 14.6 to 14.8.							
44			(a)	(a)	(a)			
46				1.130	1 118 *1.118*	1.136		
48					1.162 *1.143*	1.138 *1.158*	1.145	
50				1.230	1.216	1.205	1.217	
	Brake load, 115 pounds; B. H. P., 13.9 to 14.2.							
42				(a)	(a)			
44				(a)	*1.132* *1.110*	(a)		
46			(a)		1.141 *1.144*	1.127 *1.138*		1.160
48						1.162		
	Brake load, 80 pounds; B. H. P., 9.8 to 10.0.							
32	(a)							
34	1.206	(a)						
36	1.177 *1.199*	1.182 *1.193*						
38	1.221							
40	1.203	1.197	1.249	1.250	1.270			
	Brake load, 40 pounds; B. H. P., 5.0 to 5.1.							
44		1.864						
48		1.684						
50	1.578		1.574					
52	1.590	1.620 1.550 *1.580*	1.610 1.564 *1.605*					
54		1.607	1.592	1.609				
56			1.627	1.636				
Igniter settings.[b]	+38°	+34°	+30°	+25°	+21°	+17°	+13°	+9°

[a] Specific gravity at 60° F.=0.818. Per cent alcohol by volume=93.1; by weight=91.6. Heating value: High=11,695; low=10,595 B. t. u. per pound.

[b] Trial tests were attempted with these needle-valve and igniter settings, but the engine would not carry the load.

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TABLE 16.—*Results of tests of*

[OTTO 15-HORSEPOWER GASOLINE ENGINE No. 1.—Compression pressure, 70 pounds per square inch above atmosphere; diameter of cylinder, 6¾ inches; stroke, 15½ inches; hit-or-miss fuel-supply governor; jacket-water temperature: Inlet 60° F., outlet 112° F. For curves see figs. 24, 28, 29, and 30.]

Test number.	Temperatures (°F.). Air.	Fuel.	Mixture.[a]	Index settings. Fuel needle valve.[b]	Igniter (degrees).[c]	Speeds (av. per minute). Revolutions.	Explosions.	Fuel cut-outs.	Average pressures (pounds per sq. in.). Mean effective.	Maximum (above atmosphere).	Brake load. Pounds.	Per cent of maximum.
FUEL: GASOLINE.[g]												
A 730	83	81	101	38	0	254.9	126	2	101.8	320	130	100
A 734	75	74	98	38	0	255.4	125	3	105.0	325	130	100
B 183	64	64	84	34	+ 9	263.2	120	12	100.2	350	115	88
B 319	72	75	84	34	+ 9	258.6	120	9	99.1	330	115	88
C 305	76	79	92	30	+21	259.5	126	4	90.6	390	110	85
C 325	78	82	87	30	+21	257.6	127	2	90.7	380	110	85
D 317	79	86	90	28	+21	258.7	124	5	91.2	340	100	77
D 340	67	71	80	28	+21	258.5	120	3	87.4	340	100	77
E 224	79	86	88	28	+26	260.7	101	20	87.0	330	80	62
E 346	78	85	85	28	+26	260.5	97	16	91.1	300	80	62
E 226	69	70	83	26	+42	262.4	104	14	82.3	300	80	62
F 271	79	79	87	44	+11	264.5	65	67	108.2	340	60	46
F 367	69	75	76	44	+11	263.7	67	65	108.0	330	60	46
F 274	73	73	81	42	+13	264.2	67	66	106.8	330	60	46
G 297	70	74	81	48	+13	265.6	47	86	111.0	360	40	31
G 364	74	78	86	48	+13	265.8	48	85	107.6	300	40	31
FUEL: DENATURED ALCOHOL.[h]												
H 672	87	80	63	64	+ 8	255.7	125	3	112.2	330	140	100
H 678	79	75	62	64	+ 8	254.0	124	3	111.6	330	140	100
I 878	83	72	77	54	+13	254.3	124	3	105.6	340	130	94
I 884	73	69	58	54	+13	254.6	124	3	105.8	350	130	94
J 872	89	82	81	54	+21	255.6	121	6	103.6	370	125	89
J 869	86	77	79	52	+21	257.2	126	3	100.8	350	125	89
K 715	72	66	57	46	+21	255.1	125	2	94.2	340	120	86
K 722	73	69	61	46	+21	254.5	124	3	96.6	335	120	86
L 905	88	80	63	44	+30	255.7	126	1	91.2	300	115	82
M 642	90	85	64	44	+30	256.2	127	1	87.2	230	110	79
N 424	75	71	87	41	+25	258.6	112	8	87.6	340	100	71
N 425	79	81	90	41	+30	258.4	112	8	85.3	350	100	71
O 507	77	77	94	50	+25	263.3	94	23	93.4	350	80	57
O 621	81	81	64	50	+21	260.0	96	28	93.6	265	80	57
P 588	87	89	104	54	+30	264.7	73	59	92.9	365	60	43
P 591	88	88	105	56	+30	264.5	72	60	95.0	370	60	43
Q 473	76	77	91	62	+21	265.7	51	82	97.8	345	40	29
Q 579	87	85	103	62	+21	265.9	51	82	101.4	360	40	29

[a] Measured by a thermometer placed in the inlet-valve housing (p. 57).
[b] Opening of needle valve in hundredths of one revolution.
[c] Point of ignition, in degrees, is given in terms of the crank position; + = early or before dead center.
[d] Horsepower constants: brake, 0.0004817; indicated, 0.0014 (p. 67).
[e] Calculated from the indicated horsepower and the low heating value of the fuel (p. 72).

best fuel economy versus load.

[OTTO 15-HORSEPOWER GASOLINE ENGINE NO. 1.—Compression pressure, 70 pounds per square inch above atmosphere; diameter of cylinder, 6¾ inches; stroke, 15½ inches; hit-or-miss fuel-supply governor; jacket-water temperature: Inlet 60° F., outlet 112° F. For curves see figs. 24, 28, 29, and 30.]

FUEL: GASOLINE.[g]

Horsepower.[d]		Duration (minutes).	Fuel consumption.								Efficiencies (per cent).		
					Per brake horsepower-hour.			Per indicated horsepower-hour.				Thermal.	
Brake.	Indicated.		Pounds per hour.	Average per charge (pounds).	Pounds.	Gallons.	B. t. u. (low).	Pounds.	Gallons.	B. t. u. (low).	Mechanical.	Indicated.[e]	Brake.[f]
15.95	17.9	17	10.74	0.00142	0.673	0.111	12,860	0.56	0.092	11,450	89.0	22.2	19.8
16.00	18.3	32	10.47	.00140	.655	.108	12,520	.57	.094	10,910	87.3	23.3	20.3
14.59	16.8	17	8.74	.00121	.599	.098	11,450	.52	.085	9,920	86.6	25.7	22.2
14.33	16.7	34	8.78	.00122	.613	.101	11,710	.52	.085	10,050	85.9	25.3	21.7
13.75	16.0	18	8.15	.00108	.593	.097	11,330	.51	.084	9,740	86.1	26.1	22.5
13.65	16.1	31	8.06	.00106	.590	.097	11,270	.50	.082	9,590	85.0	26.5	22.6
12.45	15.8	16	7.58	.00102	.609	.100	11,640	.48	.079	9,150	78.7	27.4	21.9
12.45	14.7	32	7.44	.00098	.598	.098	11,420	.51	.084	9,680	84.8	26.3	22.3
10.04	12.3	18	6.70	.00101	.667	.110	12,750	.55	.090	10,410	81.6	24.4	19.9
10.04	12.4	31	6.79	.00099	.677	.111	12,890	.55	.090	10,490	81.1	24.3	19.7
10.11	12.0	13	6.75	.00095	.667	.110	12,750	.56	.092	10,760	84.4	23.6	19.9
7.65	9.9	16	5.68	.00146	.742	.122	14,170	.58	.095	11,000	77.6	23.1	17.9
7.62	10.1	32	5.64	.00140	.740	.122	14,160	.56	.092	10,660	75.3	23.9	18.0
7.64	10.0	16	5.68	.00141	.742	.122	14,180	.57	.094	10,910	76.7	23.3	17.9
5.12	7.3	14	4.39	.00156	.857	.141	16,360	.60	.099	11,550	70.5	22.0	15.5
5.12	7.3	34	4.38	.00152	.855	.140	16,350	.60	.099	11,480	70.1	22.2	15.6

FUEL: DENATURED ALCOHOL.[h]

17.25	19.7	31	19.30	0.00257	1.119	0.163	11,680	0.98	0.143	10,300	87.8	24.8	21.8
17.14	19.4	31	19.30	.00259	1.126	.164	11,750	1.00	.146	10,400	88.6	24.4	21.7
15.93	18.3	28	17.05	.00229	1.071	.156	11,240	.93	.136	9,800	87.2	26.0	22.6
15.93	18.4	28	16.89	.00227	1.060	.155	11,120	.92	.134	9,600	86.7	26.4	22.9
15.39	17.6	29	16.65	.00229	1.082	.158	11,360	.95	.139	9,900	87.4	25.7	22.4
15.49	17.8	30	16.23	.00215	1.048	.153	11,010	.91	.133	9,600	87.1	26.6	23.1
14.75	16.5	16	15.00	.00200	1.017	.148	10,660	.91	.133	9,500	89.4	26.7	23.8
14.71	16.8	31	15.31	.00206	1.040	.152	10,910	.91	.133	9,500	87.5	26.7	23.3
14.16	16.1	29	14.27	.00187	1.008	.147	10,570	.89	.130	9,300	88.2	27.3	24.1
13.58	15.5	32	14.28	.00187	1.051	.153	11,070	.92	.134	9,700	87.3	26.3	23.0
12.45	14.8	18	13.47	.00186	1.081	.158	11,310	.91	.133	9,500	84.1	26.7	22.5
12.45	14.4	18	13.47	.00186	1.081	.158	11,310	.93	.136	9,800	86.3	26.1	22.5
10.15	12.3	12	12.50	.00222	1.231	.180	13,000	1.02	.149	10,800	83.0	23.6	19.6
10.02	12.6	31	12.45	.00203	1.243	.181	13,090	.99	.145	10,500	79.9	24.3	19.4
7.65	9.5	32	10.34	.00236	1.352	.197	14,200	1.09	.159	11,500	80.8	22.2	17.9
7.64	9.5	32	10.34	.00239	1.354	.198	14,210	1.09	.159	11,400	80.3	22.3	17.9
5.12	7.0	15	8.05	.00263	1.572	.230	16,590	1.15	.168	12,200	73.3	20.9	15.3
5.12	7.3	30	8.18	.00267	1.598	.233	16,790	1.12	.164	11,800	70.3	21.5	15.2

[f] Calculated from the brake horsepower and the low heating value of the fuel (p. 72).
[g] Specific gravity at 60° F.=0.729; heating value: high=20,401, low=19,120 B. t. u. per pound.
[h] Specific gravity at 60° F.=0.820; per cent alcohol by weight=91.0; heating value: high=11,604, low=10,500 B. t. u. per pound.

TABLE 16.—*Results of tests of*

FUEL: DENATURED ALCOHOL.[i]

(Air preheated to 250° F.)

Test number.	Temperatures (°F.).			Index settings.		Speeds (av. per minute).			Average pressures (pounds per sq. in.).		Brake load.	
	Air.	Fuel.	Mixture.	Fuel needle valve.	Igniter (degrees).	Revolutions.	Explosions.	Fuel cut-outs.	Mean effective.	Maximum (above atmosphere).	Pounds.	Per cent of maximum.
R 1315	255	89	96	46	+21	255.6	127	1	92.6	330	120	100
R 1317	250	87	96	46	+21	256.2	127	1	94.5	330	120	100
S 1265	255	92	101	44	+21	256.5	126	2	94.5	320	115	96
S 1394	249	87	101	44	+21	254.5	123	5	96.0	320	115	96
T 1328	249	90	100	36	+38	256.0	107	12	80.4	290	80	67
T 1332	247	81	100	36	+34	255.5	112	15	78.0	245	80	67
U 1391	247	83	113	52	+34	263.4	57	75	96.0	300	40	33
U 1349	256	88	111	52	+34	263.7	58	74	91.0	290	40	33

[i] Specific gravity at 60° F.=0.818; per cent alcohol by weight=91.6; heating value: high=11,695, low=10,595 B. t. u. per pound.

OPERATOR'S NOTES (TABLE 16).

Test A.—The 5 trial tests covered a range of settings from needle valve 40 and igniter 0°, with a corresponding fuel consumption of 0.70 pound per brake horsepower per hour, to the settings and fuel-consumption figures given in the table. The smallest needle-valve setting that would enable the engine to carry the load satisfactorily was 38. In test A 730 there was a light explosion pound.

Test B.—The 15 trial tests covered a range of settings from needle valve 36 and igniter +4°, with a corresponding fuel consumption of 0.67 pound per brake horsepower per hour, to the settings and fuel-consumption figures given in the table. The smallest needle-valve setting that would enable the engine to carry the load satisfactorily was 34. In test B 183 occasional explosion pounds occurred toward the end of the test. In test B 319 a very light explosion pound was noticed.

Test C.—The 17 trial tests covered a range of settings from needle valve 34 and igniter +7°, with a corresponding fuel consumption of 0.64 pound per brake horsepower per hour, to the settings and fuel-consumption figures given in the table. The smallest needle-valve setting that would enable the engine to carry the load satisfactorily was 30, with which setting the best economy was obtained despite the heavy explosion pound, which was eliminated with a wider opening of the needle valve. In test C 305 there was a heavy explosion pound. The first fuel charge after cut-out occasionally misfired.[b] In test C 325 considerable explosion pound was noticed.

Test D.—The 20 trial tests covered a range of settings from needle valve 34 and igniter +11°, with a corresponding fuel consumption of 0.63 pound per brake horsepower per hour, to the settings and fuel-consumption figures given in the table. The best economy was obtained with the smallest needle-valve setting possible. In test

[b] Mixture too weak to burn; ignition not at fault.

best fuel economy versus load—Continued.

FUEL: DENATURED ALCOHOL.[i]

(Air preheated to 250° F.)

Horsepower.		Duration (minutes).	Fuel consumption.								Efficiencies (per cent).		
			Pounds per hour.	Average per charge (pounds).	Per brake horsepower-hour.			Per indicated horsepower-hour.			Mechanical.	Thermal.	
Brake.	Indicated.				Pounds.	Gallons.	B. t. u. (low).	Pounds.	Gallons.	B. t. u. (low).		Indicated.	Brake.
14.77	16.5	16	16.53	0.00217	1.118	0.163	11,840	1.00	0.146	10,660	89.9	23.9	21.5
14.82	16.9	29	16.56	.00217	1.118	.163	11,840	.98	.143	10,410	87.9	24.4	21.5
14.21	16.7	30	16.08	.00213	1.132	.165	11,920	.96	.140	10,150	85.2	25.1	21.3
14.10	16.5	31	15.66	.00212	1.110	.162	11,760	.95	.139	10,050	85.5	25.3	21.6
9.87	12.0	16	11.61	.00173	1.177	.172	12,470	.97	.142	10,210	81.9	24.9	20.4
9.85	12.2	36	11.91	.00174	1.199	.175	12,700	.97	.142	10,240	80.5	24.9	20.0
5.08	7.7	30	8.02	.00235	1.580	.231	16,740	1.04	.152	11,060	66.1	23.0	15.2
5.08	7.4	15	7.87	.00226	1.550	.226	16,420	1.06	.155	11,290	68.7	22.5	15.5

D 317 there was an occasional explosion pound. In test D 340 about 1 in 20 of the first fuel charges after cut-outs misfired.[a]

Test E.—The 34 trial tests covered a range of settings from needle valve 40 and igniter +10° to needle valve 24 and igniter +42°, with corresponding fuel consumptions of 0.71 and 0.72 pound per brake horsepower per hour. The fuel consumptions corresponding to the intermediate settings given in the table are very much less, showing that for this load too weak a mixture can be obtained with the small needle-valve settings, thus making the proportion of misfired and slow-burning charges too great. In test E 224 there was an occasional sharp explosion pound. The first fuel charge after cut-out misfired.[a] In test E 346 there was an occasional sharp explosion pound. The first fuel charge after cut-out misfired.[a] In test E 226 there was an occasional sharp explosion pound. The first fuel charge after cut-out misfired.[b]

Test F.—The 45 trial tests covered a range of settings from needle valve 48 and igniter +8° to needle valve 26 and igniter +30°, with corresponding fuel consumptions from 0.77 to 0.85 pound per brake horsepower per hour. The intermediate settings given in the table show better results. In test F 271 there was a heavy pound from second explosion after cut-outs. In test F 367 there was a single explosion between cut-outs, but no explosion pound. In test F 274 there was a heavy pound from second explosion after cut-outs.

Test G.—The 22 trial tests covered a range of settings from needle valve 50 and igniter +10° to needle valve 30 and igniter +30°, with corresponding fuel consumptions from 0.87 to 0.99 pound per brake horsepower per hour. The intermediate settings given in the table show better results. In test G 297 there was an occasional explosion pound. In test G 364 there was a single explosion between cut-outs, but no explosion pound.

Test H.—This was the maximum load that the engine would carry. The 17 trial tests cover a range of settings from needle valve 66 and igniter +8° to needle valve

[a] Mixture too weak to burn; ignition not at fault.

62 and igniter +17°, with corresponding fuel-consumption figures from 1.13 to 1.09 pounds of fuel per brake horsepower per hour. The latter test, however, is not given in the table on account of the unstable condition of operation.

Test I.—The 16 trial tests covered a range of settings from needle valve 66 and igniter +8° to needle valve 54 and igniter +17°, with corresponding fuel-consumption figures from 1.19 to 1.09 pounds of fuel per brake horsepower per hour and the values given in the table. In test I 884 just before the test was made the exhaust valve was reground and the piston cleaned.

Test J.—The 12 trial tests covered a range of settings from needle valve 54 and igniter +11° to needle valve 48 and igniter +17°, with corresponding fuel-consumption figures from 1.10 to 1.04 pounds of fuel per brake horsepower per hour. In test J 869 the first fuel charge after cut-out began to misfire.[a]

Test K.—The 9 trial tests covered a range of settings from needle valve 48 and igniter +13° to +25°, to needle valve 46 and igniter +8° to +30°, with corresponding fuel-consumption figuress from 1.07 pound of fuel per brake horsepower per hour to those given in the table. In test K 715 the first fuel charge after cut-out misfired.[a] In test K 722 the first fuel charge after cut-out occasionally misfired.[a]

Test L.—The 28 trial tests covered a range of settings from needle valve 70 and igniter 0° to needle valve 44 and igniter +21° to +34°, with corresponding fuel-consumption figures from 1.43 pounds of fuel per brake horsepower per hour to those given in the table. In test L 905 the first fuel charge after cut-out misfired.[a]

Test M.—The 19 trial tests covered a range of settings from needle valve 52 and igniter +13° to +25°, to needle valve 44 and igniter +25° to +34°, with corresponding fuel-consumption figures ranging from 1.20 pounds of fuel per brake horsepower per hour to that given in the table. In test M 642 the first fuel charge after cut-out occasionally misfired.[a]

Test N.—The 17 trial tests covered a range of settings from needle valve 52 and igniter +13° to +21°, to needle valve 41 and igniter +21° to +34°, with corresponding fuel-consumption figures from 1.23 pounds of fuel brake per horsepower hour to those given in the table. In tests N 424 and N 425 the first fuel charge after cut-out misfired.[a]

Test O.—The 28 trial tests covered a range of settings from needle valve 54 and igniter +17° to +25°, to needle valve 40 and igniter +21° to +34°, with corresponding fuel-consumption figures from 1.30 pounds of fuel per brake horsepower per hour to those given in the table, and back to 1.35 pounds of fuel per brake horsepower per hour. This load marked a change in method of judging the settings for minimum consumption. Even with the smallest settings of the needle valve the number of cut-outs was frequent, and the number of misfired fuel charges and the slow burning of the first few fuel charges following cut-outs overbalanced any gain that might otherwise have been obtained. For this and lighter loads a close inspection of the indicator cards was necessary in order to judge the best needle-valve and igniter settings. In test O 507 the fuel charge after cut-out misfired.[a] In test O 621 about one in five of the first fuel charges after cut-out misfired.[a]

Test P.—The 39 trial tests covered a range of settings from needle valve 62 and igniter +13° to +30°, to needle valve 42 and igniter +25° to +34°, with corresponding fuel-consumption figures from 1.45 pounds of fuel per brake horsepower per hour to those given in the table, and back to 1.57 pounds of fuel per brake horsepower per hour for the needle-valve setting 42. In tests P 588 and P 591 the first fuel admission after cut-out did not misfire.

Test Q.—The 34 trial tests covered a range of settings from needle valve 64 and igniter +17° to +25°, to needle valve 44 and igniter +17° and +24° with corresponding fuel-consumption figures from 1.64 pounds of fuel per brake horsepower per hour to

[a] Mixture too weak to burn; ignition not at fault.

those given in the table, and back to 1.90 pounds of fuel per brake horsepower per hour for needle-valve setting 44.

Test R.—The 13 trial tests covered a range of settings from needle valve 50 and igniter +13° to 25°, with a corresponding fuel-consumption figure of 1.22 pounds per brake horsepower per hour, to the values given in the table.

Test S.—The 8 trial tests covered a range of settings from needle valves 48, 46, and 44, and igniter settings from +9° to +21°, with corresponding fuel-consumption figures from 1.18 pounds of fuel per brake horsepower per hour to the values given in the table. The engine would not carry the load when the needle-valve setting was smaller than 44.

Test T.—The 11 trial tests covered a range of settings from needle valve 40 and igniter +21° to 38°, with a corresponding fuel-consumption figure of 1.27 pounds of fuel per brake horsepower per hour, to values given in the table. In test T 1328, the first fuel charge after cut-out misfired.[a] In test T 1332 the first fuel charge after cut-out misfired at times.

Test U.—The 16 trial tests covered a range of settings from needle valve 44 and igniter +34° to needle valve 56 and igniter +30°, with corresponding fuel-consumption figures of 1.86 and 1.64 pounds of fuel per brake horsepower per hour. Better economies were obtained with intermediate settings, as given in the table.

[a] Mixture too weak to burn; ignition not at fault.

TABLE 17.—*Results of trial and check tests made to determine the best fuel economy for each load.*[a]

[OTTO 15-HORSEPOWER GASOLINE ENGINE No. 2.—Compression pressure, 70 pounds per square inch above atmosphere; diameter of cylinder, 6¾ inches; stroke, 15½ inches; speed, 245 to 262 revolutions per minute; hit-or-miss fuel-supply governor; jacket-water temperature, inlet, 60° F.; outlet, 100° to 124° F.]

FUEL: DENATURED ALCOHOL.[b]

Fuel needle-valve setting.[c] / Igniter settings.[e]	0°	+3°	+5°	+10°	+15°	+20°	+25°	+30°	+35°	+40°
	Fuel consumption per brake horsepower-hour, pounds.									
	Brake load, 150 pounds; B. H. P., 16.5 to 17.2.									
122				(d)	(d)	(d)	(d)			
124				(d)	1.087 *1.115*	(d)				
126			(d)	1.082	1.079 1.096 1.155	[d]1.103	(d)			
128				1.146 1.157	1.115	1.212				
130				1.103	1.071 1.162	1.148	1.142			
132					1.230					
140			1.170	1.151	1.161					
	Brake load, 130 pounds; B. H. P., 14.6 to 14.9.									
90							(d)	(d)	(d)	
92						(d)		*1.021*		(d)
94								1.018 *1.026*		
96					(d)			*.986* 1.026 1.045	1.025	
98							1.018 1.018	1.016	1.035	
124					1.080 *1.023*					
160				1.386	1.386	1.417	1.507	(d)		

[a] Figures in the body of the table give the fuel consumption in pounds brake horsepower-hour and are tabulated with respect to the fuel needle-valve and igniter settings. The figures in italics are the results of check tests of about 30 minutes' duration. The other figures in the body of the table are the results of trial tests of about 15 minutes duration.

[b] Specific gravity at 60° F.=0.8192 to 0.8188. Per cent alcohol by volume=94.2 to 94.3; by weight=91.3 to 91.5. Heating value: High=11,639 to 11,178, low=10,530 to 10,681 B. t. u. per pound.

[c] Opening of the fuel needle-valve in hundredths of one revolution.

[d] Trial tests were made with these fuel needle-valve and igniter settings but the engine would not carry the load.

[e] Point of ignition, in degrees, is given in terms of the crank position. (+=early or before dead center.)

Table 17.—*Results of trial and check tests made to determine the best fuel economy for each load*—Continued.

FUEL: DENATURED ALCOHOL—Continued.

Fuel needle-valve setting.	Fuel consumption per brake horsepower-hour, pounds.									
	Brake load, 100 pounds; B. H. P., 11.3 to 11.5									
80								(a)	(a)	
84						(a)		1.189		(a)
86					(a)		1.181	1.172	1.166	
88								1.176		
90							1.175	1.172	1.152 *1.160*	1.176
92							1.200 1.188	1.176	1.191	
94							1.176 1.197		1.189 1.158	
96						1.212	1.194	1.188	1.210	
98						1.225	1.188	1.228		
100					1.240	1.210	1.208	1.214		
102						1.207	1.215	1.210		
124					*1.192*					
	Brake load, 70 pounds; B. H. P., 8.0 to 8.2.									
90							1.478	1.478	1.486	
96								1.525	1.484	
98							1.489		1.462	
100							1.430	1.473 1.413 1.450	1.415 1.389	
102							1.440	*1.435* 1.416 *1.440*	1.440	
104							1.435	*1.414* 1.436 1.480 *1.460* *1.448*	1.435	
106								1.464 1.451		
110						1.445	1.434	1.422		
Igniter settings.[b]	0°	+3°	+5°	+10°	+15°	+20°	+25°	+30°	+35°	+40°

[a] Trial tests were made with these fuel needle-valve and igniter settings but the engine would not carry the load.
[b] Point of ignition, in degrees, is given in terms of the crank position. (+ = early or before dead center.)

TABLE 17.—*Results of trial and check tests made to determine the best fuel economy for each load*—Continued.

FUEL: DENATURED ALCOHOL—Continued.

Fuel needle-valve setting.	Fuel consumption per brake horsepower-hour, pounds.									
	Brake load, 40 pounds; B. H. P., 4.6 to 4.7.									
100				2.500	2.300	2.390 2.220	2.440 2.220 2.230	2.240		
110						2.500	2.420			
116					2.410					
118				2.430	2.340	2.380				
120				2.450	2.510 2.340	2.410	2.150	2.080	2.090	
122				2.350	2.260	2.290				
124				2.320	2.290	2.300				
130					2.270	2.260	2.240			
140					2.320	2.300	2.200 2.030	1.980	2.060	
144								2.080	2.060 2.030	
146							2.040	1.984 2.020	2.090	
148						2.110	*1.984* 2.130 1.930 2.020	2.030		
150					2.230	2.150 2.080	2.220 1.960 2.110	2.060		
160						2.210 2.100	2.190 2.160	2.110 2.080 2.140 2.130 2.150		
170							2.180	2.150	*2.130*	
174						2.100	2.030	2.090		
176							2.150 1.990	1.970 2.130	2.050	
178							2.060	2.030	2.160	
180						2.270	2.070 2.190	2.050 2.090 2.010	2.060 2.130	
182		2.020					2.160		2.150	
184					2.180	2.040	2.030	2.090		
186						2.070	2.050	2.190		
Igniter sittings.[a]	0°	+3°	+5°	+10°	+15°	+20°	+25°	+30°	+35°	+40°

[a] Point of ignition, in degrees, is given in terms of the crank position. (+=early or before dead center.)

TABLE 17.—*Results of trial and check tests made to determine the best fuel economy for each load*—Continued.

FUEL: DENATURED ALCOHOL—Continued.

Fuel needle-valve setting.	Fuel consumption per brake horsepower-hour, pounds.									
	Brake load, 40 pounds, B. H. P., 4.6 to 4.7.									
188							2.080	2.020 2.050	2.480	
190						2.100	2.070 2.080	2.080 2.070	2.120	
200				2.350		2.160	2.120 2.150	2.190		
300	2.030		1.960	2.340 1.970	2.020	2.270	2.310			
400	1.990 2.400 2.340 2.560		2.320							
Igniter settings.[a]	0°	+3°	+5°	+10°	+15°	+20°	+25°	+30°	+35°	+40°

FUEL: GASOLINE.[b]

	Brake load, 135 pounds; B. H. P., 15.0 to 15.3.							
52						(c)	(c)	(c)
54					(c)	(c)	0.647	(c)
56					(c)	0.618 (c)	*.648* .617 .607 *.646* *.640*	0.633
58							.626 .665	.648
	Brake load, 115 pounds; B. H. P. 12.7 to 13.1.							
18		(c)		(c)	(c)	(c)	(c)	
22			(c)	0.623 .601	0.606 *.621*	.624	(c)	
24			(c)	.604 *.634*	.613 *.612*			
26			(c)	.606	.625			
Igniter settings.[a]	+40°	+35°	+30°	+25°	+20°	+15°	+10°	+5°

[a] Point of ignition in degrees is given in terms of the crank position; += early, or before dead center.
[b] Specific gravity at 60° F.=0.717. Heating value: high=20,527, low=19,235 B. t. u. per pound.
[c] Trial tests were made with these fuel needle-valve and igniter settings but the engines would not carry the load.

TABLE 17.—*Results of trial and check tests made to determine the best fuel economy for each load*—Continued.

FUEL: GASOLINE—Continued.

Fuel needle-valve setting.	Fuel consumption per brake horsepower-hour, pounds.							
	Brake load, 100 pounds; B. H. P., 11.3 to 11.6.							
12				(a)	(a)	(a)		
14			(a)		.653	.671	(a)	
16				.644 *.653*	.637 .654	.664 *.662*		
18				.660	.654	.663		
	Brake load, 70 pounds; B. H. P., 8.0 to 8.1.							
8	(a)	(a)	(a)					
10	0.835	0.816	0.840 *.821*	.854	(a)			
12	(a)	.816	.787 .805 *.794*	.821				
14		.779 .823 .810	.818 .794 .806	.807 .839				
16			.820					
	Brake load, 40 pounds; B. H. P., 4.6 to 4.7.							
30				1.197	1.211			
40				1.086	1.028 1.177 1.075	1.128		
42			1.053	1.056	1.086			
44			1.121 *1.189* 1.051 *1.092*	*1.098* 1.151				
46	1.087	1.060 *1.056*	1.086	1.120				
48		1.079 *1.007* *1.091*	1.076	1.081				
50				.990 1.066 1.073	1.086 1.059	1.105		
52			1.066 1.108	1.031 1.051	1.073			
54				1.008	1.080			
56							*1.120* *1.086*	
60						1.228		
Igniter settings.[b]	+40°	+35°	+30°	+25°	+20°	+15°	+10°	+5°

[a] Trial tests were made with these fuel needle-valve and igniter settings but the engine would not carry the load.

[b] Point of ignition in degrees is given in terms of the crank position; +=early, or before dead ecnter.

TABLE 18.—*Results of tests of*

[OTTO 15-HORSEPOWER GASOLINE ENGINE NO. 2.—Compression pressure, 70 pounds per square inch above atmosphere; diameter of cylinder, 6¾ inches; stroke, 15½ inches; hit-or-miss fuel-supply governor; inlet jacket-water temperature, 60° F. For curves see figs. 24, 28, and 29.]

FUEL: GASOLINE.[g]

Test number.	Temperatures (° F.).				Index settings.		Speed (av. per minute).			Average pressures (pounds per sq. in.).		Brake load.	
	Air.	Fuel.	Mixture.[a]	Jacket-water outlet.	Fuel needle valve.[b]	Igniter (degrees).[c]	Revolutions.	Explosions.	Fuel cut-outs.	Mean effective.	Maximum (above atmosphere).	Pounds.	Per cent of maximum.
A 1764	85	92	79	124	56	+10	252.4	123	3	108.0	420	135	100
A 1782	83	89	80	126	56	+10	252.1	124	2	105.8	410	135	100
B 1772	78	80	78	112	24	+25	253.1	125	2	87.8	370	115	85
B 1776	82	98	80	111	22	+20	253.5	123	4	90.0	350	115	85
B 1866	81	91	82	118	22	+25	250.8	116	10	91.5	350	115	85
C 1742	93	94	87	115	16	+20	255.3	120	8	86.5	320	100	74
C 1745	91	92	85	114	16	+25	255.0	120	7	87.7	320	100	74
D 1721	81	82	78	106	12	+30	256.7	94	20	86.8	300	70	52
D 1724	84	89	82	108	14	+30	256.7	93	21	84.4	280	70	52
E 1681	88	94	89	106	48	+35	260.7	66	64	85.5	300	40	30
E 1690	83	86	85	103	50	+25	260.9	65	66	91.0	275	40	30

FUEL: DENATURED ALCOHOL.[h]

Test number.	Air.	Fuel.	Mixture.[a]	Jacket-water outlet.	Fuel needle valve.[b]	Igniter (degrees).[c]	Revolutions.	Explosions.	Fuel cut-outs.	Mean effective.	Maximum (above atmosphere).	Pounds.	Per cent of maximum.
F 1892	81	85	54	112	124	+15	245.8	118	5	111.0	395	150	100
F 1903[i]	80	82	61	117	124	+15	254.1	121	6	109.7	390	150	100
G 1918[i]	76	78	53	111	94	+30	253.7	125	2	97.3	350	130	87
G 1954[i]	78	77	55	99	94	+30	253.2	122	4	98.2	320	130	87
G 1921[i]	77	80	52	108	92	+30	253.1	126	1	96.0	320	130	87
H 1635	93	80	62	112	90	+35	254.4	115	12	86.9	280	100	67
H 1641	97	88	62	111	90	+35	254.4	113	11	87.7	295	100	67
I 1597	85	81	65	108	104	+30	259.3	87	43	87.2	295	70	47
I 1604	88	82	66	105	102	+30	257.8	90	39	86.8	265	70	47
J 1563	94	92	70	106	148	+25	261.1	58	73	88.1	280	40	27
J 1566	94	89	70	106	146	+30	261.2	57	74	86.4	300	40	27

[a] Measured by a thermometer placed in the inlet-valve housing (p. 57).
[b] Opening of the fuel needle valve in hundredths of 1 revolution.
[c] Point of ignition, in degrees, is given in terms of the crank position, += early or before dead center (p. 55).
[d] Horsepower constants: Brake=0.000493, indicated=0.001396.
[e] Calculated from the indicated horsepower and the low heating value of the fuel (p. 72).
[f] Calculated from the brake horsepower and the low heating value of the fuel (p. 72).

OPERATOR'S NOTES (TABLE 18).

Test A.—The 11 trial tests were to a great extent in the nature of check tests, the possible range of settings with this load being very small. Needle valve 58, with igniter set at +5° and +10°, were the only other settings that could be used. The fuel consumptions for these settings were practically the same as those given in the table. In test A1782 there was an explosion pound.

Test B.—The 11 trial tests covered a range of settings from needle valve 56 and igniter +10°, with a corresponding fuel consumption of 0.67 pound per brake horsepower per hour, to the values given in the table. In test B1776 there was a light explosion pound.

best fuel economy versus load.

[OTTO 15-HORSEPOWER GASOLINE ENGINE NO. 2.—Compression pressure, 70 pounds per square inch above atmosphere; diameter of cylinder, 6¾ inches; stroke, 15½ inches, hit-or-miss fuel-supply governor; inlet jacket-water temperature, 60° F. For curves see figs. 24, 28, and 29.]

FUEL: GASOLINE.[g]

Horsepower.[d]		Duration (minutes).	Pounds per hour.	Average per charge (pounds).	Fuel consumption.						Efficiencies (per cent).		
					Per brake horse-power-hour.			Per indicated horse-power-hour.				Thermal.	
Brake.	Indicated.				Pounds.	Gallons.	B. t. u. (low).	Pounds.	Gallons.	B. t. u. (low).	Mechanical.	Indicated.[e]	Brake.[f]
15.31	18.6	36	9.89	0.00134	0.646	0.108	12,430	0.53	0.089	10,230	82.4	24.9	20.5
15.30	18.3	28	9.79	.00132	.640	.107	12,310	.53	.089	10,310	83.7	24.7	20.7
13.10	15.3	15	7.91	.00105	.604	.101	11,620	.52	.086	9,940	85.5	25.6	21.9
13.10	15.5	33	8.14	.00110	.621	.104	11,940	.52	.088	10,140	84.8	25.1	21.3
12.95	14.8	15	7.78	.00112	.601	.100	11,560	.52	.088	10,140	87.8	25.1	22.0
11.47	14.5	32	7.50	.00104	.654	.109	12,580	.52	.086	9,960	79.2	25.6	20.2
11.46	14.7	32	7.48	.00104	.653	.109	12,560	.51	.085	9,770	77.8	26.0	20.2
8.07	11.4	33	6.41	.00099	.794	.133	15,270	.56	.094	10,790	70.6	23.6	16.7
8.07	10.9	32	6.50	.00100	.806	.135	15,550	.59	.099	11,410	73.6	22.3	16.4
4.68	7.9	38	4.72	.00119	1.007	.168	19,310	.60	.100	11,510	59.5	22.1	13.2
4.69	8.2	32	4.64	.00119	.990	.165	19,040	.57	.094	10,890	57.3	23.4	13.4

FUEL: DENATURED ALCOHOL.[h]

16.56	18.3	13	18.00	0.00254	1.087	0.159	11,610	0.98	0.144	10,510	90.5	24.2	21.9
17.14	18.5	29	19.11	.00263	1.115	.163	11,910	1.03	.151	11,040	92.7	23.1	21.4
14.81	16.9	32	15.20	.00203	1.026	.150	10,960	.90	.132	9,610	87.6	26.5	23.2
14.79	16.7	25	14.59	.00199	.986	.144	10,530	.87	.127	9,300	88.3	27.4	24.2
14.78	16.8	28	15.09	.00200	1.021	.149	10,910	.90	.131	9,580	87.8	26.6	23.3
11.44	14.0	14	13.17	.00191	1.152	.168	12,130	.94	.138	9,960	82.1	25.6	21.0
11.41	13.8	32	13.23	.00195	1.160	.169	12,210	.96	.139	10,050	82.3	25.3	20.8
8.15	10.6	31	11.52	.00221	1.414	.207	14,890	1.09	.159	11,470	77.0	22.2	17.1
8.11	10.8	31	11.68	.00216	1.440	.210	15,160	1.07	.156	11,270	74.3	22.6	16.8
4.69	7.1	29	9.31	.00268	1.984	.290	20,890	1.31	.192	13,850	66.2	18.4	12.2
4.69	6.9	32	9.31	.00272	1.984	.290	20,890	1.36	.199	14,340	68.6	17.8	12.2

[g] Specific gravity at 60° F.=0.717. Heating value: High=20,527, low=19,235 B. t. u. per pound.
[h] Specific gravity at 60° F.=0.819; per cent alcohol by volume=94.2, by weight=91.3. Heating value: High=11,639, low=10,530 B. t. u. per pound.
[i] Repair to the governor mechanism caused a change in the timing of the fuel inlet valve, making the fuel consumption for these tests different from that for the others with the same needle-valve setting.

Test C.—The 11 trial tests were to a great extent in the nature of check tests, as the best settings were definitely determined by observation, and the fuel consumptions were all within 3 per cent of those given in the table.

Test D.—Of the 19 trial tests a number checked the values given in the table. It was found possible to use a smaller needle-valve setting of 10. With this needle-valve setting and igniter setting from +20° to +35°, the fuel consumption ranged from 0.86 to 0.82 pound per brake horsepower per hour. In tests D1721 and D1724 the first fuel charge after cut-out misfired.[a]

Test E.—The 41 trial tests covered a range of settings of from needle valve 33 and igniter +20° to needle valve 56 and igniter +60°, with corresponding fuel consump-

[a] Mixture too weak to burn; ignition not at fault.

tions of 1.21 and 1.12 pounds per brake horsepower per hour, whereas for the intermediate settings as given in the table the fuel consumptions were less.

Test F.—The 21 trial tests covered a range of settings from needle valve 140 and igniter +5°, with a corresponding fuel consumption of 1.17 pounds per brake horsepower per hour, to the values given in the table. In test F1892 the load was the maximum that the engine would carry. The engine would not carry the load with a smaller needle-valve setting, nor with an earlier time of ignition in combination with the given needle-valve setting.

Test G.—The 18 trial tests covered a range of settings from needle valve 160 and igniter +25°, with a corresponding fuel consumption of 1.51 pounds per brake horsepower per hour, to the values given in the table. With needle valve 160 and igniter +30° the engine would not carry the load. In test G1921 with the settings shown the engine would just carry the load.

Test H.—The 33 trial tests covered a range of settings from needle valve 124 and igniter +15°, with a corresponding fuel consumption of 1.19 pounds per brake horsepower per hour, to the values given in the table.

Test I.—The 30 trial tests covered a range of settings from needle valve 111 and igniter +20°, to needle valve 90 and igniter +35°, with corresponding fuel consumptions of 1.45 and 1.49 pounds per brake horsepower per hour, while for the intermediate settings as given in the table the consumption was less. In test I1604 the first fuel charge after cut-outs occasionally misfired.[a]

Test J.—The 128 trial tests covered a range of settings from needle valve 100 and igniter +30° to needle valve 40° and igniter 0°, with corresponding fuel consumptions of 2.24 and 2.56 pounds per brake horsepower per hour, while for the intermediate settings as given in the table the consumption was less.

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[a] Mixture too weak to burn; ignition not at fault.

TABLE 19.—*Results of trial and check tests made to determine the best fuel economy for each load.*

[NASH 10-HORSEPOWER GASOLINE ENGINE.—Compression pressure, greatest, 97 pounds per square inch above atmosphere; diameter of cylinder, $7\frac{5}{8}$ inches; stroke, 10 inches; speed, 290 to 310 revolutions per minute; hit-or-miss mixture-supply governor; jacket-water temperature: Inlet, 60° to 80° F.; outlet, 100° to 116° F.]

FUEL: GASOLINE;[a] REGULAR CARBURETOR.

Load (pounds)	90 (B. H. P. 10.3 to 10.8).																80 (B. H. P. 9.2 to 9.6).		
Air-valve settings [b]	$\frac{1}{2}$				$\frac{5}{8}$				$\frac{3}{4}$					$\frac{7}{8}$			$\frac{5}{16}$		
Compression [c]	72				78				80					84			64		
Fuel needle-valve settings [d]	$2\frac{1}{16}$	$2\frac{1}{8}$		$2\frac{1}{4}$	$2\frac{1}{4}$		$2\frac{5}{16}$	$2\frac{1}{2}$	$2\frac{3}{8}$	$2\frac{1}{2}$		$2\frac{3}{8}$	$2\frac{1}{2}$	$2\frac{5}{8}$	$2\frac{11}{16}$	$2\frac{3}{4}$	$1\frac{13}{16}$	$1\frac{7}{8}$	2
Igniter settings [e]	$8\frac{1}{2}$	8		8	8	$8\frac{1}{2}$	8	11	8		9	10		8	9	10	$7\frac{1}{2}$	9	10
Fuel consumption (pounds per B. H. P. hour)	0.620	0.623	0.623	0.660	0.628	0.622	0.621	0.689	0.600	0.631	0.636	0.642	0.638	0.627	0.648	0.644	0.644	0.667	0.772

Load (pounds)	80 (B. H. P. 9.2 to 9.6)—Continued.																
Air-valve settings [b]	$\frac{3}{8}$				$\frac{1}{2}$								$\frac{5}{8}$				
Compression [c]	68				72								78				
Fuel needle-valve settings [d]	$1\frac{15}{16}$	$1\frac{7}{8}$		2	$1\frac{15}{16}$						2	$2\frac{1}{8}$	$2\frac{1}{16}$		$2\frac{1}{8}$	$2\frac{3}{16}$	$2\frac{1}{2}$
Igniter settings [e]	5	7	8	9	$2\frac{1}{2}$	$3\frac{1}{2}$	5		$6\frac{1}{2}$		7	9	$2\frac{1}{2}$	5	$5\frac{1}{2}$	7	11
Fuel consumption (pounds per B. H. P. hour)	0.667	0.631	0.636	0.671	0.675	0.655	0.618	0.630	0.612	0.621	0.647	0.664	0.624	0.617	0.634	0.647	0.727

[a] Specific gravity at 60° F.=0.7168. Heating value: High=20,527; low=19,235 B. t. u. per pound.
[b] Plug cock with dial for regulating the auxiliary air supply, which was introduced between the governor-controlled mixture valve and the main inlet valve (p. 34).
[c] Per square inch above atmosphere.
[d] Scale arbitrary; numbers increase with opening of the valve.
[e] Scale arbitrary; relation to crank position not constant; ignition before dead center is indicated by (+).

TABLE 19.—*Results of trial and check tests made to determine the best fuel economy for each load*—Continued.

FUEL: GASOLINE; REGULAR CARBURETOR—Continued.

Load (pounds)	80 (B. H. P. 9.2 to 9.6)—Continued.								70 (B. H. P. 8.2 to 8.4).								
Air-valve settings [a]	$\frac{3}{4}$					$\frac{7}{8}$			$\frac{1}{4}$			$\frac{3}{8}$					
Compression [b]	80					84			60			68					
Fuel needle-valve settings [c]	$2\frac{5}{16}$			$2\frac{3}{8}$		$2\frac{9}{16}$	$2\frac{5}{8}$	$2\frac{3}{8}$	$1\frac{11}{16}$	$1\frac{3}{4}$	$1\frac{7}{8}$	$1\frac{3}{4}$		$1\frac{7}{8}$	2		$2\frac{1}{8}$
Igniter settings [d]	5		7	5	8	$7\frac{1}{2}$	8	11	$6\frac{1}{2}$	7	9	4	6		8		9
Fuel consumption (pounds per B. H. P. hour)	0.625	0.629	0.649	0.652	0.652	0.668	0.683	0.700	0.663	0.691	0.759	0.665	0.665	0.656	0.627	0.633	0.564

Load (pounds)	70 (B. H. P. 8.2 to 8.4)—Continued.													60 (B. H. P. 6.8 to 7.1).		
Air-valve settings [a]	$\frac{1}{2}$						$\frac{5}{8}$				$\frac{7}{8}$			$\frac{1}{8}$		
Compression [b]	72						78				84			50		
Fuel needle-valve settings [c]	$1\frac{13}{16}$		$1\frac{27}{32}$	$1\frac{7}{8}$	2	$1\frac{15}{16}$	$2\frac{1}{16}$	$2\frac{1}{8}$		$2\frac{1}{4}$	$2\frac{7}{16}$	$2\frac{1}{2}$	$2\frac{5}{8}$	$1\frac{1}{2}$	$1\frac{9}{16}$	$1\frac{1}{2}$
Igniter settings [d]	2			$3\frac{1}{2}$	$5\frac{1}{2}$	$6\frac{1}{2}$	3	4	5	6	$4\frac{1}{2}$	5	7	3	$4\frac{1}{2}$	5
Fuel consumption (pounds per B. H. P. hour)	0.624	0.618	0.620	0.633	0.644	0.679	0.656	0.642	0.703	0.646	0.717	0.717	0.725	0.684	0.709	0.742

Load (pounds)	60 (B. H. P. 6.8 to 7.1)—Continued.												40 (B. H. P. 4.6 to 4.8).					
Air-valve settings [a]	$\frac{1}{4}$					$\frac{3}{8}$				$\frac{1}{2}$		$\frac{5}{8}$	$\frac{1}{8}$					
Compression [b]	60					68				72		78	50					
Fuel needle-valve settings [c]	$1\frac{1}{2}$			$1\frac{17}{32}$	$1\frac{5}{8}$	$1\frac{21}{32}$	$1\frac{11}{16}$	$1\frac{3}{4}$		$1\frac{15}{16}$		2	$1\frac{1}{2}$	$1\frac{5}{8}$	$1\frac{3}{4}$			
Igniter settings [d]	0	1	$1\frac{1}{2}$		$3\frac{1}{2}$	0	$\frac{1}{2}$	$2\frac{1}{2}$	4	4	$6\frac{1}{2}$	2	3	2		3		4
Fuel consumption (pounds per B. H. P. hour)	0.642	0.648	0.679	0.710	0.698	0.713	0.646	0.700	0.744	0.747	0.760	0.759	0.896	0.957	0.968	1.039	0.881	0.978

Load (pounds)	40 (B. H. P. 4.6 to 4.8)—Continued.															
Air-valve settings [a]	$\frac{1}{4}$													$\frac{1}{2}$		
Compression [b]	60													72		
Fuel needle-valve settings [c]	$1\frac{1}{2}$	$1\frac{9}{16}$	$1\frac{5}{8}$	$1\frac{9}{16}$	$1\frac{5}{8}$					$1\frac{7}{8}$	$1\frac{15}{32}$	$1\frac{1}{2}$		2		$1\frac{15}{16}$
Igniter settings [d]	1		2	4			5		6		$8\frac{1}{2}$			3	4	$6\frac{1}{2}$
Fuel consumption (pounds per B. H. P. hour)	0.839	0.929	0.958	0.949	0.904	0.925	0.887	0.922	0.960	0.909	0.878	0.905	0.902	1.094	1.024	0.990

[a] Plug cock with dial for regulating the auxiliary air supply, which was introduced between the governor-controlled mixture valve and the main inlet valve (p. 34).
[b] Per square inch above atmosphere.
[c] Scale arbitrary; numbers increase with opening of the valve.
[d] Scale arbitrary; relation to crank position not constant; ignition before dead center is indicated by (+).

TABLE 19.—*Results of trial and check tests made to determine the best fuel economy for each load*—Continued.

FUEL: GASOLINE;[a] DOUBLE-CONE CARBURETOR.

Load (pounds)	125 (B. H. P. 15).		108 (B. H. P. 12.8 to 12.9).						100 (B. H. P. 11.6 to 12.0).									
Air-valve settings [b]	¾		⅜	½		⅝	¾	1	⅜	½					⅝	¾	½	15/16
Compression [c]	95		89	92		94	95	97	89	92					94	95	([d])	([d])
Fuel needle-valve settings [e]	2–5¼	2–5¾	1–0	1–8		1–9½	2–½	2–2¼	1–5⅛	1–6¼		1–6½	1–7¼	1–8	1–8	1–8	1–2¼	1–5¼
Igniter settings [f]	7¼	7½	6½	5	5½	6¾	6½	6¾	3	2¼	3¼	2¾	3¾	4¾	3¾	3½	3	2¼
Fuel consumption (pounds per B. H. P. hour)	0.839	0.832	0.630	0.595	0.624	0.635	0.631	0.775	0.593	0.601	0.610	0.597	0.595	0.649	0.645	0.612	0.622	0.602

Load (pounds)	100 (B. H. P. 11.6 to 12.0—Continued).			90 (B. H. P. 10.4 to 10.8).					80 (B. H. P. 9.5 to 9.6).								50 (B. H. P. 6.0).		
Air-valve settings [b]	½	¾	1	3/16	⅜	½		⅝	0	1/16	⅛		⅜	⅛	¼	⅜	0	0	
Compression [c]	([g])	([g])	([g])	84	89	92		94	79	81	83		89	([g])	([g])	([g])	79	([g])	
Fuel needle-valve settings [e]	1–4	1–5½	1–7¼	1–4¾	1–5	1–5	1–5⅛	1–6⅝	1–2	1–¼	1–3		1–4¾	0–8¾	0–9½	1–1	1–1⅜	4–6	4–7½
Igniter settings [f]	7½	7¼	6½	5	3	(–9)	(–9½)	(–9¼)	0	0	0	½	0	2¼	½	0	0	(–9)	(–9)
Fuel consumption (pounds per B. H. P. hour)	0.631	0.613	0.606	0.652	0.614	0.580	0.585	0.595	0.636	0.650	0.630	0.623	0.654	0.623	0.597	0.593	0.968	0.827	0.790

FUEL: DENATURED ALCOHOL;[h] REGULAR CARBURETOR.

Load (pounds)	117.5 (B. H. P. 13.7 to 14.0).						100 (B. H. P. 11.6 to 11.8).												80 (B. H. P. 9.2 to 9.5).	
Air-valve settings [b]	3/4	7/8				1	1/2			5/8				3/4			7/8		1/4	
Compression [c]	82	84				86	73			78				82			84		60	
Fuel needle-valve settings [e]	3-3/4	4-5/8		4-3/4	5-3/16	4-7/8	3-1/2		3-5/8	3-1/8	3-7/16	3-1/2		3-5/8	3-11/16	4	4-1/8	6-1/2	2-7/8	3-1/8
Igniter settings [f]	6	3	5	6½	7	6	2½			5	2		5½	4	2	4	4½	7	2	
Fuel consumption (pounds per B. H. P. hour)	1.320	1.068	1.094	1.060	1.198	1.099	1.140	1.170	1.169	1.039	1.036	1.078	1.080	1.023	1.070	1.027	1.057	1.271	1.366	1.534

Load (pounds)	80 (B. H. P. 9.2 to 9.5—Continued).											60 (B. H. P. 6.9 to 7.2).							
Air-valve settings [b]	3/8	1/2			5/8		3/4				7/8	3/16			1/4	3/8			
Compression [c]	68	73			78		82				84	56			60	68			
Fuel needle-valve settings [e]	2-5/8	2-3/4	2-13/16		3-3/16	3-7/16	3-1/4	3-3/8		6-1/2	3-7/8	2-1/8		2-3/16	2-1/4	2-5/16	2-3/8	2-7/16	2-3/8
Igniter settings [f]	1	0		1/2	+(-9½)	2	0	+(-9½)	0	7	0	1/2			+(-9½)	+(-8½)			0
Fuel consumption (pounds per B. H. P. hour)	1.100	1.063	1.115	1.067	1.135	1.143	1.109	1.080	1.101	1.614	1.150	1.178	1.186	1.205	1.199	1.191	1.134	1.221	1.245

[a] Specific gravity at 60° F.=0.7122. Heating value: High=20,581; low=19,292 B. t. u. per pound.
[b] Plug cock with dial for regulating the auxiliary air supply, which was introduced between the governor-controlled mixture valve and the main inlet valve (p. 34).
[c] Per square inch above atmosphere.
[d] Carburetor throttled by bushing the air opening to ½-inch pipe.
[e] Scale arbitrary; numbers increase with opening of the valve.
[f] Scale arbitrary; relation to crank position not constant; ignition before dead center is indicated by (+).
[g] Carburetor throttled by bushing the air opening to ¾-inch pipe.
[h] Specific gravity at 60° F.=0.8188 to 0.8225. Per cent alcohol: By volume, 93.3 to 94.3; by weight, 90.1 to 91.5. Heating value: High=11,788 to 14,373; low=10,379 to 10,681 B. t. u. per pound.

TABLE 19.—*Results of trial and check tests made to determine the best fuel economy for each load*—Continued.

FUEL: DENATURED ALCOHOL; REGULAR CARBURETOR—Continued.

Load (pounds)	60 (B. H. P. 6.9 to 7.2)—Continued.				40 (B. H. P. 4.5 to 4 8).														
Air-valve settings [a]	1/2		9/16	5/8	1/8		1/4	5/16					3/8						
Compression [b]	73		76	78	50		60	64					68						
Fuel needle-valve settings [c]	2-9/16	2-5/8	6-1/2	3-7/16	1-15/16		2-1/16	2-1/4	2-3/8		2-1/2		2-3/8	2-1/2	2-5/8				2-3/4
Igniter settings [d]	+(-8½)		7	2	+(-8½)		+(-8½)	1			1½		+(-8½)	1			1½	2	2½
Fuel consumption (pounds per B. H. P. hour)	1.221	1.226	1.827	1.347	1.476	1.490	1.510	1.705	1.706	1.787	1.730	1.724	1.670	1.675	1.737	1.750	1.733	1.875	1.818

Load (pounds)	40 (B. H. P. 4.5 to 4.8—Continued).																			
Air-valve settings [a]	1/2		5/8											3/4						
Compression [b]	73		78											82						
Fuel needle-valve settings [c]	3	4	3-1/8				3-1/4		3-1/2		4		4½	3-5/8			3-3/4		4	4½
Igniter settings [d]	2½	4	1		2	2½					3	4½		2½						
Fuel consumption (pounds per B. H. P. hour)	2.100	2.120	1.978	2.062	1.932	2.030	2.035	2.124	2.003	2.115	2.196	2.074	2.240	2.130	2.250	2.130	2.240	2.159	2.115	2.380

Load (pounds)	40 (B H. P. 4.5 to 4.8)—Continued.																
Air-valve settings[a]	¾					⅞			1								
Compression[b]	82					84			86								
Fuel needle-valve settings[c]	4		4½			4	4-½	6-½	4-⅛	4-½		5½		8-½	8-½	4-⅜	7
Igniter settings[d]	3	4			8	3	2½	7	2	2½	4	7				11	12
Fuel consumption (pounds per B. H. P. hour)	2.200	2.092	2.213	2.187	2.240	2.250	2.260	2.460	2.169	2.205	2.167	2.360	2.435	2.250	2.320	2.361	2.403

FUEL: DENATURED ALCOHOL;[e] DOUBLE-CONE CARBURETOR.

Load (pounds)	145 (B.H. P. 17.4).	140 (B. H. P. 16.8).		116.5 (B. H. P. 13.9 to 14.0).								
Air-valve settings[a]	1	1		⅜	½					⅝	¾	1
Compression[b]	97	97		89	92					94	95	97
Fuel needle-valve settings[c]	4-3	4-1	4-3	2-¼	2-2½	2-3	2-3½	2-4		2-4	2-7	2-7½
Igniter settings[d]	5½	5½	4¾	2½	2½		2¾	2½		3½	4	2¾
Fuel consumption (pounds per B. H. P. hour)	0.937	0.858	0.931	0.878	0.886	0.847	0.930	0.965	0.967	0.905	0.937	0.937

[a] Plug cock with dial for regulating the auxiliary air supply, which was introduced between the governor-controlled mixture valve and the main inlet valve (p. 34).
[b] Per square inch above atmosphere.
[c] Scale arbitrary; numbers increase with opening of the valve.
[d] Scale arbitrary; relation to crank position not constant; ignition before dead center is indicated by (+).
[e] Specific gravity at 60° F.=0.8206. Per cent alcohol: By volume, 93.8; by weight, 90.8. Heating value: High=11,750; low=10,643 B. t. u. per pound.

TABLE 19.—*Results of trial and check tests made to determine the best fuel economy for each load*—Continued.

FUEL: DENATURED ALCOHOL; DOUBLE-CONE CARBURETOR—Continued.

Load (pounds)	91 (B. H. P. 10.8 to 11.0).							66 (B. H. P. 7.8 to 8.0).							
Air-valve settings [a]	0	¼		½			1	0	⅛	¼	½		0		¾
Compression [b]	79	86		92			97	79	83	86	92		(c)		(d)
Fuel needle-valve settings [e]	1–7	1–9		2–1½	2–3	2–3½	2–3	1–6½	1–7¾	1–9	2–3	2–3½	1–5½		1–9¼
Igniter settings [f]	8¼	8¼		8¼	2½	2¾	8½	9½	9	9	2½	2¾	9¼		8¾
Fuel consumption (pounds per B. H. P. hour)	0.918	0.900	0.906	0.919	0.988	1.020	0.975	1.170	1.185	1.191	1.202	1.222	1.150	1.105	1.218

a Plug cock with dial for regulating the auxiliary air supply, which was introduced between the governor-controlled mixture valve and the main inlet valve (p. 34).
b Per square inch above atmosphere.
c Carburetor throttled by bushing the air opening to ½-inch pipe.
d Carburetor throttled by bushing the air opening to ¾-inch pipe.
e Scale arbitrary; numbers increase with opening of the valve.
f Scale arbitrary; relation to crank position not constant; ignition before dead center is indicated by (+).

TABLE 20.—*Results of tests of*

[NASH 10-HORSEPOWER GASOLINE ENGINE.—Compression pressure, greatest, 96 pounds per-square inch above atmosphere; diameter of cylinder, $7\frac{5}{8}$ inches; stroke, 10 inches; hit-or-miss mixture-supply governor. Jacket-water temperature: Inlet, 60° to 80° F.; outlet, 100° to 116° F. For curves see figs. 24 and 25.]

FUEL: GASOLINE;[h] DOUBLE-CONE CARBURETOR.[i]

Test number.	Temperatures (°F.).			Index settings.			Speeds (average per minute).			Average pressures (pounds per sq. in.).			Brake load.	
	Air.	Mixture near carburetor.[a]	Mixture near inlet valve.[a]	Air valve.[b]	Fuel needle valve.[c]	Igniter.	Revolutions.	Explosions.	Mixture cut-outs.	Compression (above atmosphere).	Mean effective.	Maximum (above atmosphere).	Pounds.	Per cent of maximum.
A 3657	72	38	80	$\frac{3}{4}$	$2\text{–}5\frac{3}{4}$	$g+17\frac{1}{2}$	306.3	145	9	96	105.6	370	125	100
A 3659	75	39	81	$\frac{3}{4}$	$2\text{–}5\frac{1}{4}$	$g+17\frac{1}{4}$	305.6	147	6	96	105.6	370	125	100
B 3655	74	36	88	$\frac{1}{2}$	1–8	$g+15$	300.0	146	4	92	96.0	335	108	85
C 3625	80	39	91	$\frac{1}{2}$	$1\text{–}6\frac{1}{4}$	$g+2\frac{1}{4}$	294.8	141	8	92	83.4	300	100	80
C 3641	91	45	103	$\frac{1}{2}$	$1\text{–}6\frac{1}{4}$	$g+3\frac{3}{4}$	301.7	147	4	92	64.6	270	100	80
D 3631	82	40	94	$\frac{1}{2}$	$1\text{–}5\frac{1}{8}$	$g+9\frac{1}{2}$	301.3	144	7	92	81.0	320	90	72
D 3635	77	40	92	$\frac{1}{2}$	1–5	$g+9\frac{1}{2}$	302.8	144	7	92	79.5	285	90	72
E 3681	73	36	91	$\frac{3}{8}$	1–1	$g+12\frac{1}{4}$	304.7	141	6	80	72.6	280	80	64
E 3679	70	35	91	$\frac{1}{4}$	$0\text{–}9\frac{1}{2}$	$g+12\frac{1}{2}$	304.6	144	5	80	75.0	265	80	64
F 3675	67	33	85	0	$0\text{–}4\frac{3}{4}$	$g+9\frac{3}{4}$	306.8	115	18	65	62.1	200	50	40
FUEL: GASOLINE;[j] REGULAR CARBURETOR.[k]														
G 3336	105	......	......	$\frac{3}{4}$	$2\frac{3}{8}$	+ 5°	294.0	144	3	81	80.4	280	90	100
H 3228	95	......	......	$\frac{5}{8}$	$2\frac{1}{16}$	+ 5°	298.1	148	1	78	69.9	175	80	89
H 3328	102	......	......	$\frac{1}{2}$	$1\frac{15}{16}$	+ 0°	299.2	146	4	72	70.8	255	80	89
I 3280	91	......	......	$\frac{1}{2}$	$1\frac{27}{32}$	+20°	297.6	137	12	72	76.2	250	70	78
I 3334	101	......	......	$\frac{1}{2}$	$1\frac{13}{16}$	+20°	297.6	143	5	72	65.4	250	70	78
J 3294	93	......	......	$\frac{3}{8}$	$1\frac{11}{16}$	+28°	298.7	118	15	68	67.8	210	60	67
J 3326	98	......	......	$\frac{1}{4}$	$1\frac{1}{2}$	+30°	297.3	147	1	60	57.9	190	60	67
K 3350	96	......	......	$\frac{1}{2}$	$1\frac{1}{2}$	+25°	299.8	110	18	60	55.2	215	40	44
K 3332	100	......	......	$\frac{1}{4}$	$1\frac{15}{32}$	+38°	298.4	107	14	60	57.6	190	40	44
FUEL: DENATURED ALCOHOL;[l] DOUBLE-CONE CARBURETOR.[i]														
L 3696	79	55	76	1	4–3	+ 3°	303.1	146	6	97	117.4	430	145.0	100
M 3694	78	56	75	1	4–1	+ 3°	306.0	146	7	97	114.6	420	140.0	97
N 3704	67	47	64	$\frac{1}{2}$	2–3	+18°	305.2	148	4	92	96.5	320	116.5	80
N 3763	59	......	......	$\frac{1}{2}$	$2\text{–}2\frac{1}{2}$	+20°	304.3	147	5	92	96.7	300	116.5	80
O 3715	78	50	65	$\frac{1}{4}$	1–9	+40°	303.6	146	5	86	79.3	260	91.0	63
O 3721	76	49	64	$\frac{1}{4}$	1–9	+40°	303.4	147	5	86	78.6	260	91.0	63
P 3733	77	50	71	0	$1\text{–}5\frac{1}{2}$	....	304.7	140	13	65	64.0	200	66.0	41

[a] Measured by a thermometer placed in the inlet-valve housing (p. 57).
[b] Plug cock, with dial, for regulating the auxiliary air supply, which is introduced between the governor-controlled mixture valve and the main inlet valve.
[c] Scale arbitrary; numbers increase with opening of needle valve.
[d] Horsepower constant: Brake=0.000395; indicated=0.001153 (p. 67).
[e] Calculated from the indicated horsepower and the low heating value of the fuel (p. 72).
[f] Calculated from the brake horsepower and the low heating value of the fuel (p. 72).

best fuel economy versus load.

[NASH 10-HORSEPOWER GASOLINE ENGINE.—Compression pressure, greatest, 96 pounds per square inch above atmosphere; diameter of cylinder, 7⅝ inches; stroke, 10 inches; hit-or-miss mixture-supply governor. Jacket-water temperature: Inlet, 60° to 80° F.; outlet, 100° to 116° F. For curves see figs. 24 and 25.]

FUEL: GASOLINE;[h] DOUBLE-CONE CARBURETOR.[i]

Horsepower.[d]		Duration (minutes).	Fuel consumption.							Efficiencies (per cent)		
			Pounds per hour.	Per brake horsepower-hour.			Per indicated horse-power-hour.			Mechanical.	Thermal.	
Brake.	Indicated.			Pounds.	Gallons.	B. t. u. (low).	Pounds.	Gallons.	B. t. u. (low).		Indicated.[e]	Brake.[f]
15.12	17.6	16	12.58	0.832	0.140	16,050	0.71	0.120	13,770	85.9	18.5	15.9
15.07	17.9	21	12.65	.839	.141	16,190	.71	.119	13,650	84.4	18.6	15.7
12.80	16.1	32	7.62	.595	.100	11,470	.47	.080	9,120	79.5	27.9	22.2
11.64	13.6	28	6.99	.601	.101	11,590	.51	.087	9,960	86.0	25.6	22.0
11.90	14.3	31	7.26	.610	.103	11,760	.51	.085	9,760	83.0	26.1	21.6
10.69	13.4	17	6.27	.585	.098	11,280	.47	.079	9,020	79.7	28.2	22.5
10.76	13.2	29	6.23	.580	.097	11,180	.47	.079	9,080	81.4	28.0	22.8
9.62	11.8	32	5.70	.593	.100	11,440	.48	.081	9,340	81.6	27.2	22.2
9.62	12.4	31	5.74	.597	.100	11,510	.46	.078	8,930	77.6	28.5	22.1
6.06	8.2	31	4.79	.790	.133	15,240	.58	.098	1,260	73.9	22.6	16.7

FUEL: GASOLINE;[j] REGULAR CARBURETOR.[k]

10.45	13.2	29	6.71	0.642	0.170	12,350	0.50	0.084	9,690	78.5	26.3	20.6
9.42	11.9	16	5.81	.617	.103	11,860	.49	.081	9,340	78.8	27.3	21.5
9.46	11.9	31	5.79	.612	.102	11,760	.49	.081	9,340	79.4	27.3	21.6
8.22	10.4	36	5.10	.620	.103	11,910	.49	.082	9,400	78.8	27.1	21.4
8.22	10.8	27	5.08	.618	.103	11,880	.47	.078	9,040	76.1	28.1	21.4
7.07	9.2	16	4.57	.646	.108	12,410	.50	.083	9,570	77.0	26.6	20.4
7.04	9.8	33	4.52	.642	.107	12,340	.46	.077	8,840	71.6	28.8	20.6
4.73	7.0	23	3.97	.839	.140	16,140	.57	.095	10,920	67.7	23.3	15.8
4.71	7.1	29	4.17	.878	.147	16,870	.58	.097	11,200	66.3	22.7	15.1

FUEL: DENATURED ALCOHOL;[l] DOUBLE-CONE CARBURETOR.[i]

17.36	19.7	17	16.28	0.937	0.137	9,900	0.82	0.120	8,710	88.0	29.2	25.7
16.91	19.3	31	14.51	.858	.125	9,060	.75	.109	7,930	87.6	32.1	28.1
14.05	16.5	30	11.90	.847	.124	8,942	.72	.105	7,640	85.4	33.3	28.4
14.00	16.4	31	12.41	.886	.129	9,354	.76	.110	7,990	85.4	31.9	27.2
10.91	13.3	15	9.83	.900	.131	9,501	.74	.107	7,790	81.9	32.7	26.8
10.90	13.3	30	9.88	.906	.132	9,565	.74	.107	7,830	81.9	32.5	26.6
7.94	10.3	41	8.76	1.105	.161	11,665	.85	.123	8,960	77.0	28.4	21.8

[g] Scale arbitrary; relation to crank position not constant; ignition before dead center is indicated by (+).
[h] Specific gravity at 60° F.=0.7122. Heating value: High=20,581; low=19,292 B. t. u. per pound.
[i] For description of double-cone carburetor see p. 34.
[j] Specific gravity at 60° F.=0.7168. Heating value: High=20,527; low=19,235 B. t. u. per pound.
[k] For description of regular carburetor see p. 34.
[l] Specific gravity at 60° F.=0.8206. Per cent alcohol: By volume=93.8; by weight=90.8. Heating value: High=11,750, low=10,643 B. t. u. per pound

TABLE 20.—*Results of tests of*

FUEL: DENATURED ALCOHOL; [m] REGULAR CARBURETOR. [k]

Test number.	Temperatures (°F.). Air.	Temperatures (°F.). Mixture near carburetor.	Temperatures (°F.). Mixture near inlet valve.	Index settings. Air valve.	Index settings. Fuel needle valve.	Index settings. Igniter.	Speeds (average per minute). Revolutions.	Speeds (average per minute). Explosions.	Speeds (average per minute). Mixture cut-outs.	Average pressures (pounds per sq. in.). Compression (above atmosphere).	Average pressures (pounds per sq. in.). Mean effective.	Average pressures (pounds per sq. in.). Maximum (above atmosphere).	Brake load. Pounds.	Brake load. Per cent of maximum.
Q 3428	108	......	......	7/8	4 5/8	+15°	297.0	147	1	84	92.7	360	117.5	100
R 3364	93	......	......	3/4	4	+24°	292.6	143	3	80	88.5	330	100.0	85
R 3366	92	......	......	3/4	3 5/8	+24°	295.7	146	2	80	86.6	335	100.0	85
S 3422	98	......	......	1/2	2 13/16	+29°	294.9	144	4	72	70.2	255	80.0	68
S 3382	87	......	......	1/2	2 3/4	+30°	295.0	146	1	72	72.0	230	80.0	68
T 3396	93	......	......	3/16	2 1/8	+27°	298.4	146	3	56	55.8	195	60.0	51
T 3420	97	......	......	3/16	2 1/8	+27°	296.6	145	2	56	55.8	205	60.0	51
U 3410	96	......	......	1/8	1 15/16	+38°	296.6	121	10	50	50.7	160	40.0	34
U 3416	89	......	......	1/8	1 15/16	+38°	295.1	119	6	50	49.2	175	40.0	34

[m] Specific gravity at 60° F.=0.8188. Per cent alcohol: By volume=94.3; by weight=91.5. Heating value: High=11,788, low=10,681 B. t. u. per pound.

OPERATOR'S NOTES (TABLE 20).

Test A.—The maximum load for this engine with gasoline was used. With the richer mixtures required for this maximum load, occasional preignitions and continual heavy explosion pound occurred. These were due to the high compression pressure obtained with the double-cone carburetor.

Test B.—The possible range of fuel needle-valve and igniter settings was very small and only two tests, which are given in Table 20, were necessary to determine the minimum fuel consumption for this load. The engine ran smoothly when the weaker mixtures for best economy with this and the following loads were used. The 6 trial tests were made with auxiliary air-valve settings of from three-eighths to wide open, and the best fuel needle-valve and igniter settings were obtained by observation.

Test C.—The 11 trial tests were made with auxiliary air-valve settings from three-eighths to wide open. With the best air-valve settings it was necessary to make 6 trial tests with various fuel needle-valve and igniter settings in order to determine the minimum fuel consumption for this load.

Test D.—The 6 trial tests were made with auxiliary air-valve settings from three-sixteenths to five-eighths open. With the best air-valve settings 2 trial tests only were made to determine the best fuel needle-valve and igniter settings.

Test E.—The 7 trial tests were made with auxiliary air-valve settings from 0 (or closed) to three-eighths open. The best needle-valve and igniter settings were obtained by observation.

Test F.—The minimum fuel consumption for this load was of little importance, so only 3 trial tests were made. The results given are, however, very close to the best that could have been obtained.

Test G.—Throttling effect of regular carburetor reduced the maximum load to 90 pounds, or 10.5 B. H. P., and compression, for wide-open air-valve settings, to 87 pounds per square inch. Compression for the other air-valve settings likewise affected. The *best fuel economy versus load*—Continued.

FUEL: DENATURED ALCOHOL;[m] REGULAR CARBURETOR.[k]

Horsepower.		Duration (minutes).	Fuel consumption.							Efficiencies (per cent).		
				Per brake horsepower-hour.			Per indicated horse-power-hour.				Thermal.	
Brake.	Indicated.		Pounds per hour.	Pounds.	Gallons.	B. t. u. (low).	Pounds.	Gallons.	B. t. u. (low).	Mechanical.	Indicated.	Brake.
13.78	15.7	37	14.71	1.068	0.156	11,407	0.94	0.137	9,980	87.5	25.5	22.3
11.55	14.6	15	11.86	1.027	.150	10,969	.81	.119	8,670	79.1	29.4	23.2
11.66	14.6	15	11.94	1.023	.150	10,927	.82	.120	8,750	80.0	29.1	23.3
9.31	11.7	27	9.94	1.067	.156	11,397	.85	.125	9,110	79.9	27.9	22.3
9.32	12.1	18	9.91	1.063	.155	11,353	.82	.119	8,720	76.7	29.2	22.4
7.08	9.4	16	8.40	1.186	.173	12,668	.89	.131	9,540	75.2	26.7	20.1
7.03	9.4	29	8.25	1.174	.172	12,539	.88	.129	9,390	74.9	27.1	20.3
4.68	7.1	15	6.97	1.490	.218	15,915	.99	.145	10,560	66.4	24.1	16.0
4.66	6.8	26	6.88	1.476	.216	15,765	1.02	.149	10,870	68.9	23.4	16.1

[k] For description of regular carburetor see p 34.

16 trial tests were made with auxiliary air-valve settings from one-half to seven-eighths open. Several trial tests with each air-valve setting were required to determine the best fuel needle-valve and igniter settings and the minimum fuel consumption for this load.

Test H.—The 28 trial tests were made with auxiliary air-valve settings from five-sixteenths to wide open and widely varying fuel needle-valve and igniter settings. A resulting variation in fuel consumption of from 0.73 pound per B. H. P. per hour to the values given in the table was obtained.

Test I.—The 22 trial tests were made with auxiliary air-valve settings from one-quarter to seven-eighths open. Several tests with different combinations of fuel needle-valve and igniter settings were required for each air-valve setting in order to determine the minimum fuel consumption for this load.

Test J.—The 15 trial tests were made with auxiliary air-valve settings from one-eighth to five-eighths open. The best fuel needle-valve and igniter settings could not be obtained by observation, so several tests for each of the air-valve settings were required.

Test K.—The 22 trial tests were made with auxiliary air-valve settings from one-eighth to one-half open. It was difficult to determine the best fuel needle-valve and igniter settings for any of the air-valve settings. With an air-valve setting of one-quarter, which was found to be the best, 12 trial tests with various fuel needle-valve and igniter settings were required to determine the minimum fuel consumption for this load.

Test L.—The maximum load and compression—auxiliary air-supply valve wide open—for this engine were obtained with the double-cone carburetor and are given for this test in Table 20. The range of possible settings for the maximum load was very small, the setting given being about the only one that could be used, and hence only one test was made.

Test M.—Only 2 trial tests were made with this load, as the range of possible settings was still very small. Besides the tests given in Table 20, one other was made for which practically the same settings and fuel consumption as for the maximum load were obtained.

Test N.—The 9 trial tests were made with auxiliary air-valve settings from three-eighths to wide open. The best fuel needle-valve and igniter settings were in each case determined by observation. In test N 3763 the engine back-filled twice during the test.

Test O.—The 7 trial tests were made with auxiliary air-valve settings from 0 (or closed) to wide open. The minimum fuel consumption for this load and best air-valve setting were determined in the usual way.

Test P.—The 8 trial tests were made with auxiliary air-valve settings of from 0 (or closed) to one-half open. By closing the air-valve and by bushing the air opening in the carburetor from 1½-inch pipe to 1-inch pipe, it was possible to run the engine in a more throttled condition, and this condition gave the best results.

Test Q.—Owing to the throttling effect of the regular carburetor, the maximum load was reduced to 117.5 pounds, or 13.8 B. H. P. The compression (for air valve wide open) was also reduced to 87 pounds per square inch. The compression for other air-valve settings was affected likewise. The 6 trial tests for this load were made with auxiliary air-valve settings of from three-quarters to wide open, but with the usual maximum load conditions of a very small range of possible fuel needle-valve and igniter settings.

Test R.—The 12 trial tests were made with auxiliary air-valve settings of from one-half to seven-eighths open. Several tests with different needle-valve and igniter settings for each air-valve setting were made in order to determine the minimum fuel consumption for this load.

Test S.—The 12 trial tests were made with auxiliary air-valve settings of from one-quarter to seven-eighths open. After determining the best air-valve setting, several trial tests were made to find the best fuel needle-valve and igniter setting and corresponding minimum fuel consumption. In test S 3382, the first fuel charge after cut-out misfired.

Test T.—The 12 trial tests were made with auxiliary air-valve settings of from three-sixteenths to five-eighths open, with the resulting best settings and minimum fuel consumption for this load given in the table.

Test U.—The 51 trial tests were made with auxiliary air-valve settings of from one-eighth to wide open. The smallest air-valve and fuel needle-valve settings that could be used were found to give the minimum fuel consumption for this load. These were among the first tests made on this engine and many trial tests were necessary before the operators could learn to judge the best settings by observation.

TABLE 21.—*Results of trial and check tests made to determine the best fuel economy for each load.*

[NASH 10-HORSEPOWER ALCOHOL ENGINE.—Compression pressure, greatest, 135 pounds per square inch above atmosphere; diameter of cylinder, $7\frac{5}{8}$ inches; stroke, 10 inches; speed, 293 to 308 revolutions per minute; hit-or-miss mixture-supply governor; jacket-water temperature: Inlet 60° to 80° F.; outlet, 100° to 134° F. For curves see figure 25.]

FUEL: GASOLINE[a] WITH A SPRAY OF WATER.[b] REGULAR CARBURETOR.[c]

Brake load (pounds)	124			100						87.5	
Per cent gasoline by weight [d]	43.5	42.9	42.0	78.9	49.0	42.0	40.6	40.5	38.0	42.4	42.3
Air-valve settings [e]	1	1	1	$\frac{3}{4}$	$\frac{3}{4}$	$\frac{5}{8}$	$\frac{7}{8}$	$\frac{5}{8}$	$\frac{3}{4}$	$\frac{5}{8}$	$\frac{1}{2}$
Compression [f]	133	133	133	127	127	(*g*)	130	121	127	121	114
Fuel needle-valve settings [h]	$10\frac{5}{8}$	$8\frac{7}{8}$	$8\frac{3}{4}$	$7\frac{5}{8}$	$6\frac{1}{2}$	$6\frac{1}{8}$	$6\frac{5}{8}$	$6\frac{1}{8}$	$6\frac{3}{8}$	$5\frac{3}{4}$	$5\frac{5}{8}$
Igniter settings [i]	6	$6\frac{1}{2}$	$6\frac{1}{2}$	7	6	4	....	5	$4\frac{3}{4}$	$1\frac{3}{4}$	$2\frac{1}{2}$
Fuel consumption (pounds per break horsepower-hour)	0.799	0.706	0.692	0.862	0.655	0.641	0.657	0.651	0.650	0.648	0.643
Brake horsepower	14.82	14.76	14.78	12.04	11.93	11.98	11.80	11.91	11.90	10.45	10.54

Brake load (pounds)	80		75					50				
Per cent gasoline by weight [d]	49.6	48.3	52.8	51.3	47.8	46.2	44.6	100	100	97.84	94.35	69.9
Air-valve settings [e]	$\frac{1}{2}$	$\frac{1}{2}$	$\frac{3}{8}$	$\frac{1}{2}$	$\frac{1}{2}$	$\frac{5}{8}$	$\frac{1}{2}$	$\frac{3}{8}$	$\frac{1}{2}$	$\frac{1}{2}$	$\frac{3}{8}$	$\frac{1}{4}$
Compression [f]	114	114	105	114	114	121	114	105	114	114	105	92
Fuel needle-valve settings [h]	2	$2\frac{3}{16}$	$5\frac{1}{8}$	$5\frac{1}{4}$	$5\frac{1}{4}$	$5\frac{5}{8}$	$5\frac{1}{4}$	$4\frac{7}{8}$	5	$5\frac{1}{8}$	$4\frac{7}{8}$	$4\frac{3}{4}$
Igniter settings [i]	$4\frac{1}{2}$	4	2	$\frac{3}{4}$	$\frac{3}{4}$	1	$\frac{3}{4}$	2	0	2	2	$1\frac{3}{4}$
Fuel consumption (pounds per brake horsepower-hour)	0.739	0.694	0.657	0.646	0.624	0.681	0.641	0.779	0.811	0.828	0.774	0.788
Brake horsepower	9.31	9.26	9.03	8.99	9.01	8.96	8.96	6.06	6.02	6.01	6.04	6.04

[a] Specific gravity of gasoline at 60° F.=0.7175; heating value: high=20,579, low=19,289 B. t. u. per pound.
[b] Just sufficient water introduced to prevent preignition from high compression.
[c] For description of regular carburetor see p. 34.
[d] Ratio of gasoline to gasoline + water, by weight.
[e] Plug cock with dial for regulating the auxiliary air supply which is introduced between the governor-controlled mixture valve and the main inlet valve.
[f] Per square inch above atmosphere.
[g] Carburetor throttled by bushing the air opening to $\frac{3}{4}$-inch pipe.
[h] Scale arbitrary; numbers increase with opening of the valve.
[i] Scale arbitrary; ignition before dead center is indicated by (+).

TABLE 21.—*Results of trial and check tests made to determine the best fuel economy for each load*—Continued.

[NASH 10-HORSEPOWER ALCOHOL ENGINE.—Compression pressure, greatest, 135 pounds per square inch above atmosphere; diameter of cylinder, $7\frac{5}{8}$ inches; stroke, 10 inches; speed, 293 to 308 revolutions per minute; hit-or-miss mixture-supply governor; jacket-water temperature: Inlet, 60° to 80° F.; outlet, 100° to 134° F. For curves see figure 25.]

FUEL: DENATURED ALCOHOL; *f* REGULAR CARBURETOR. *c*

Brake load (pounds)	126 (B. H. P. 15).			110 (B. H. P. 12.6 to 13.1).								100 (B. H. P. 11.7 to 12.0).			
Air-valve settings *e*	$\frac{7}{8}$	1		$\frac{5}{8}$	$\frac{3}{4}$			$\frac{7}{8}$				$\frac{3}{8}$	$\frac{1}{2}$		
Compression *f*	130	133		121	126			130				105	114		
Fuel needle-valve settings *h*	13	14		$8\frac{3}{8}$	$9\frac{3}{8}$		10	$9\frac{1}{2}$	$9\frac{1}{4}$	$10\frac{1}{8}$	$10\frac{3}{8}$	$7\frac{5}{8}$	$8\frac{1}{8}$	$8\frac{3}{8}$	$7\frac{5}{8}$
Igniter settings *i*	$8\frac{1}{2}$	$8\frac{1}{2}$	8	6	$6\frac{1}{4}$		$6\frac{1}{2}$	5		$6\frac{1}{4}$		$6\frac{1}{4}$	$3\frac{1}{4}$		$5\frac{1}{4}$
Fuel consumption (pounds per brake horsepower-hour)	1.046	0.982	0.986	1.010	1.011	1.005	0.995	0.944	0.934	1.002	1.028	1.121	1.144	1.186	1.036

Brake load (pounds)	100 (B. H. P. 11.7 to 12.0)—Continued.							90 (B. H. P. 10.7).			75 (B. H. P. 8.8 to 9.0).			
Air-valve settings *e*	$\frac{5}{8}$			$\frac{11}{16}$	1 or wide open.			$\frac{3}{8}$	$\frac{5}{8}$		$\frac{3}{8}$	$\frac{1}{2}$	$\frac{5}{8}$	$\frac{3}{4}$
Compression *f*	121			128	133			105	121		105	114	121	126
Fuel needle-valve settings *h*	$8\frac{1}{2}$		$8\frac{1}{8}$	$9\frac{5}{8}$	10	$9\frac{1}{8}$	$10\frac{1}{4}$	$7\frac{3}{8}$	8		$6\frac{7}{8}$	$7\frac{1}{8}$	$7\frac{5}{8}$	8
Igniter settings *i*	$2\frac{3}{4}$	3	$5\frac{1}{4}$	5	$2\frac{1}{2}$	$3\frac{1}{4}$		$5\frac{1}{2}$	5		5	$3\frac{1}{4}$	5	$3\frac{1}{4}$
Fuel consumption (pounds per brake horsepower-hour)	1.055	1.006	1.010	1.073	1.020	0.988	1.050	1.052	1.023	1.016	1.119	1.126	1.113	1.136

FUEL: DENATURED ALCOHOL;[k] DOUBLE-CONE CARBURETOR.[l]

Brake load (pounds)	140 (B. H. P. 16.8).			116 (B. H. P. 13.7 to 13.9).							
Air-valve settings [e]	9/16	7/8	1	1/4	3/8	9/16	5/8		7/8		
Compression [f]	130	(m)	(g)	121	126	131	134		135		
Fuel needle-valve settings [h]	2–8	2–5	3–3½	2–¼	2–1¼	2–8	2–3½		2–6	2–7½	3–2
Igniter settings [i]	1½	¾	1	1½	1½	1½	1½	1	1¾		0
Fuel consumption (pounds per brake horsepower-hour)	0.994	1.134	1.251	0.913	0.892	1.071	0.887	0.907	0.914	0.928	1.044

Brake load (pounds)	85 (B. H. P. 9.9 to 10.2).							59 (B. H. P. 6.9 to 7.2).						
Air-valve settings [e]	0	1/4	1/2	9/16	5/8	3/4	1	0	1/4	1/2	9/16	5/8	3/4	1
Compression [f]	108	121	129	131	132	134	136	108	121	129	131	132	134	136
Fuel needle-valve settings [h]	1–6	1–7¾	2–½	2–8	2–3½	2–2½	2–4¾	1–5¼	1–7	1–9½	2–8	2–3½	2–2	2–4¼
Igniter settings [i]	½	½	1½	½	1	0	1½	¼	+(–9½)	¼	1½	1	0	0
Fuel consumption (pounds per brake horsepower-hour)	0.973	0.971	1.010	1.294	0.909	1.054	1.084	1.230	1.320	1.435	1.588	1.451	1.422	1.520

c For description of regular carburetor see p. 34.
e Plug cock with dial for regulating the auxiliary air suply which is introduced between the governor-controlled mixture valve and the main inlet valve.
f Per square inch above atmosphere.
g Carbureter throttled by bushing the air opening to ¾-inch pipe.
h Scale arbitrary; numbers increase with opening of the valve.
i Scale arbitrary; ignition before dead center is indicated by (+).
j Specific gravity at 60° F.=0.8206; per cent alcohol by volume, 93.8; by weight=90.8; heating value, high=11,750; low=10,643 B. t. u. per pound.
k Specific gravity at 60° F.=0.8188; per cent alcohol by volume, 94.3; by weight=91.5; heating value, high=11,788; low=10,681 B. t. u. per pound.
l For description of double-cone carburetor see p. 34.
m Carburetor throttled by bushing the air opening to ½-inch pipe.

TABLE 22.—*Results of tests of*

[NASH 10-HORSEPOWER ALCOHOL ENGINE.—Compression pressure, greatest, 135 pounds per square inch above atmosphere; diameter of cylinder, 7⅝ inches; stroke, 10 inches; hit-or-miss mixture-supply governor; jacket-water temperature: Inlet 60° to 80° F., outlet 100° to 134° F. For curves see fig. 25.]

Test number.	Temperatures (° F.).		Index settings.			Speeds (average per minute).			Average pressures (pounds per sq. in.).			Brake load.	
	Air.	Mixture near carburetor.[a]	Air valve.[b]	Needle valve.[c]	Igniter (degrees).[d]	Revolutions.	Explosions.	Mixture cut-outs.	Compression (above atmosphere).	Mean effective.	Maximum (above atmosphere).	Pounds.	Per cent of maximum.
FUEL: GASOLINE[h] WITH SPRAY OF WATER[i]; REGULAR CARBURETOR.[l]													
A 4009	70	....	1	8¾	+6½	301.6	148	3	133	101.4	330	124	100
B 4021	72	....	⅝	6⅛	+4	303.1	146	5	130	86.0	300	100	81
C 4029	74	....	½	5¼	+ ¾	302.7	142	4	114	70.2	355	75	61
C 4031	75	....	½	5¼	+ ¾	303.2	141	5	114	73.4	320	75	61
D 4039	74	....	⅜	4⅞	+2	306.4	129	13	105	57.6	280	50	41
D 4045	78	....	⅜	4⅞	+2	306.8	128	12	105	57.6	300	50	41
FUEL: DENATURED ALCOHOL;[k] DOUBLE-CONE CARBURETOR.[l]													
E 4201	79	....	9/16	2-8	+1½	300.1	144	6	130	102.9	450	140.0	100
F 4191	71	....	⅝	2-9	+5¼	301.5	151	3	132	96.9	455	135.0	96
G 4163	71	....	⅝	2-3½	+1½	300.7	142	8	132	87.0	375	116.5	83
G 4169	61	....	⅝	2-3½	+1½	299.6	147	3	132	85.8	350	116.5	83
H 4187	78	....	¼	1-7¾	+ ½	302.0	144	7	121	72.0	270	85.0	61
H 4189	82	....	0	1-6	+ ½	301.8	142	5	108	69.9	280	85.0	61
I 4179	66	....	0	1-5¼	+ ¼	301.7	138	9	108	56.4	190	59.0	42
FUEL: DENATURED ALCOHOL;[m] REGULAR CARBURETOR.[j]													
J 4053	92	60	1	14	+8¼	300.0	150	6	133	90.0	465	126.0	100
J 4057	89	60	1	14	+8½	303.5	152	0	133	90.0	460	126.0	100
K 4119	73	56	⅞	9¼	....	297.4	148	3	131	85.5	360	110.0	88
L 4092	77	50	⅝	8	+5	299.7	145	5	121	73.9	370	90.0	71
M 4084	89	47	⅜	6⅞	+5	303.0	144	6	104	67.5	325	75.0	60
M 4086	86	46	⅝		+5	301.4	144	7	121	66.7	320	75.0	60

[a] Measured by a thermometer placed in the inlet valve housing (p. 57).
[b] Plug cock with dial, for regulating the auxiliary air supply which is introduced between the governor-controlled mixture valve and the main inlet valve.
[c] Scale arbitrary. Numbers increase with opening of needle valve.
[d] Scale arbitrary. Relation to crank position not constant. Ignition before dead center is indicated by (+).
[e] Horsepower constant: Brake=0.000395, indicated=0.001153.
[f] Calculated from the indicated horsepower and the low heating value of the fuel (p. 72).
[g] Calculated from the brake horsepower and the low heating valve of the fuel (p. 72).

OPERATOR'S NOTES (TABLE 22).

Test A.—Maximum load the engine would carry with gasoline and spray of water was required to prevent preignition. The possible settings and the amount of water used could be varied only within very small limits, and three trial tests were sufficient to determine the best settings.

best fuel economy versus load.

[NASH 10-HORSEPOWER ALCOHOL ENGINE.—Compression pressure, greatest, 135 pounds per square inch above atmosphere; diameter of cylinder, 7⅝ inches; stroke, 10 inches; hit-or-miss mixture-supply governor; jacket-water temperature: Inlet 60° to 80 ° F., outlet 100° to 134° F. For curves see fig. 25.]

FUEL: GASOLINE [h] WITH SPRAY OF WATER[i]; REGULAR CARBURETOR.[j]

Horsepower.[e]		Duration (minutes).	Fuel consumption.							Efficiencies. (per cent).			Ratio of gasoline to gasoline + water (per cent by weight).
			Pounds per hour.	Per brake horse-power-hour.			Per indicated horse-power-hour.			Mechanical.	Thermal.		
Brake.	Indicated.			Pounds.	Gallons.	B. t. u. (low).	Pounds.	Gallons.	B. t. u. (low).		Indicated.[f]	Brake.[g]	
14.78	17.3	30	10.22	0.692	0.15	12,410	0.59	0.99	10,640	85.7	23.9	20.5	42.0
11.98	14.5	31	7.68	.641	.107	11,510	.53	.088	9,520	82.7	26.7	22.1	42.0
8.96	11.5	31	5.74	.641	.107	11,600	.50	.083	9,060	78.0	28.1	21.9	44.6
8.99	11.9	33	5.81	.646	.108	11,860	.49	.081	8,960	75.5	28.4	21.4	51.3
6.04	8.5	13	4.68	.775	.129	14,930	.55	.091	10,500	70.4	24.3	17.1	94.4
6.06	8.5	32	4.72	.779	.130	15,010	.55	.092	10,650	70.9	23.9	16.9	100.0

FUEL: DENATURED ALCOHOL;[k] DOUBLE-CONE CARBURETOR.[l]

Brake.	Indicated.	Duration (minutes).	Pounds per hour.	Pounds.	Gallons.	B. t. u. (low).	Pounds.	Gallons.	B. t. u. (low).	Mechanical.	Indicated.	Brake.	Ratio.
16.60	16.9	25	16.50	0.994	0.145	10,580	0.97	0.142	10,360	98.0	24.6	24.1	......
16.08	16.9	29	15.42	.959	.140	10,200	.91	.133	9,700	95.0	26.2	24.9	......
13.84	14.3	15	12.27	.887	.130	9,440	.86	.126	9,140	96.9	27.9	27.0	......
13.78	14.6	50	12.50	.907	.132	9,650	.86	.125	9,130	94.5	27.9	26.4	......
10.14	12.0	15	9.84	.971	.142	10,340	.82	.120	8,740	84.8	29.1	24.6	......
10.13	11.5	30	9.86	.973	.142	10,350	.86	.126	9,160	88.5	27.8	24.6	......
7.04	8.9	31	8.66	1.230	.180	13,090	.97	.142	10,300	78.8	24.7	19.5	......

FUEL: DENATURED ALCOHOL;[m] REGULAR CARBURETOR.[j]

Brake.	Indicated.	Duration (minutes).	Pounds per hour.	Pounds.	Gallons.	B. t. u. (low).	Pounds.	Gallons.	B. t. u. (low).	Mechanical.	Indicated.	Brake.	Ratio.
14.94	15.6	18	14.72	.986	0.144	10,410	0.94	0.138	9,980	96.0	25.5	24.5	......
15.11	15.8	32	14.85	.982	.143	10,370	.94	.137	9,960	96.0	25.5	24.5	......
12.92	14.6	32	12.06	.934	.136	9,860	.83	.120	8,720	88.4	29.2	25.8	......
10.65	12.4	31	10.82	1.016	.148	10,730	.88	.128	9,260	86.4	27.5	23.7	......
8.98	11.2	15	10.05	1.119	.163	11,810	.90	.131	9,480	80.3	26.8	21.5	......
8.93	11.1	30	9.95	1.114	.163	11,750	.90	.131	9,470	80.5	26.9	21.6	......

[h] Specific gravity of the gasoline at 60° F.=0.7175. Heating value: High=20,579, low=19,289 B. t. u. per pound.

[i] Just sufficient water was introduced to prevent preignition from high compression.

[j] For description of regular carburetor see p. 34.

[k] Specific gravity at 60° F.=0.8206. Per cent alcohol by volume=93.8; by weight=70.8. Heating value: High=11,750, low=10,643 B. t. u. per pound.

[l] For description of double-cone carburetor see p. 34.

[m] Specific gravity at 60° F=0.8188. Per cent alcohol by volume+94.3; by weight=91.5. Heating value: High=11,788, low=10,681 B. t. u. per pound.

Test B.—For this and the following tests the minimum amount of water that would prevent preignition or excessive explosion pound was used in each case. Six trial tests were made for this load and the best fuel needle-valve and igniter settings determined in the usual way. The best air-valve setting was found to be five-eighths open, which reduced the compression and consequently the amount of water required.

Test C.—Five trial tests were made with air valve from three-eighths to five-eighths open. The best results were obtained with air valve one-half open, but considerable water was still required.

Test D.—Five trial tests were made with air valve one-half to five-eighths open. The best results were obtained with the air valve three-eighths open, which reduced the compression so much that little water was needed.

Test E.—Maximum load for this engine; obtained with the double-cone carburetor. The range of possible air-valve, fuel needle-valve, and igniter settings was small, and the settings for best fuel economy with this load, as given in the table, were determined by observation. Excessive back-firing prevented the use of wider open air-valve settings or smaller needle-valve settings; that is, back-firing prevented the use of weaker mixtures with which better fuel economy could have been obtained.

Test F.—Only one test was made for this load, as the range of possible auxiliary air-valve, fuel needle-valve, and igniter settings was still small and limited by excessive back-firing for the weaker mixtures, making it impossible to obtain relatively as good results for this engine as for the others.

Test G.—Nine trial tests were made with air-valve settings of from one-fourth to seven-eighths open, but much trouble, caused by back-firing, was experienced, making it impossible to obtain as good fuel economy as would otherwise be expected for the compression pressure of this engine. Every possible means of preventing back-firing was tried, and although the trouble was within the clearance space the exact cause could not be determined without eliminating pockets and providing for cooling of surfaces that might have become overheated. During test G 4163 two light back-fires occurred.

Test H.—Seven trial tests were made with auxiliary air-valve settings from 0 (closed) to 1 (wide open). The best results as given in the table were obtained with the auxiliary air valve closed, or nearly so, and the smallest fuel needle-valve settings (weakest mixtures) that could be used without excessive back-firing. For light-load tests, with frequent cut-outs, it was noted that the back-firing usually occurred following the first explosion after cut-outs; that is, as the second mixture charge after cut-outs was being taken in.

Test I.—Seven trial tests were made with auxiliary air-valve settings from 0 (closed) to 1 (wide open) and, as for the previous test, the best results were obtained when the air valve was closed, even though the compression was greatly reduced by consequent throttling of the charge.

Test J.—This was the heaviest load that could be carried with the regular carburetor, owing to its throttling effect. Only three trial tests were necessary, as the best results were obtained with the air valve wide open, the range of possible fuel needle-valve and igniter settings being very small.

Test K.—Eight trial tests were made with air-valve settings from five-eighths to seven-eighths open. The best results were obtained with the air valve seven-eighths open, but excessive back-firing made it impossible to get satisfactory results. During test K 4119 engine back-fired nine times. Eleven trial tests were made with a brake load of 100 pounds and air-valve settings of from three-eighths to wide open. Equally good results were obtained with air-valve settings of from five-eighths to wide open, but the back-firing was excessive in all cases.

Test L.—Three trial tests were made with air-valve settings of three-eighths and five-eighths open, but not much trouble was taken to find the best settings on account of back-firing. During test L 4092 engine back-fired twice.

Test M.—Five trial tests were made with air-valve settings of from three-eighths to three-fourths open. The best fuel needle-valve and igniter settings could be a little more definitely determined with this load, as the back-firing was not so frequent. During test M 4086 engine back-fired twice.

TABLE 23.—*Results of trial and check tests made to determine the best fuel economy for each load.*[a]

[OTTO 15-HORSEPOWER ALCOHOL ENGINE.—Diameter of cylinder, 6¾ inches; stroke, 15½ inches; speed, 251 to 265 revolutions per minute; throttle method of governing; governor controls lift and timing of mechanically operated inlet valve. Jacket-water temperature: Inlet, 60° F.; outlet, 100° to 200° F.]

FUEL: DENATURED ALCOHOL;[b] COMPRESSION PRESSURE, GREATEST, 140 POUNDS.[c]

Fuel needle-valve setting.[d]	Air-valve setting.[e]	Fuel consumption per brake horsepower-hour (pounds).						
		Brake load, 170 pounds; B. H. P., 19.7.						
70	35						(f)	*0.951*
		Brake load, 148 pounds; B. H. P., 17.1 to 17.3.						
40	30				(f)	(f)	(f)	(f)
41	30					(f)	0.857	(f)
42	30 30				(f)		*0.857* *.851*	
44							0.896	0.869
		Brake load, 130 pounds; B. H. P., 14.6 to 15.3.						
60	0						1.005	
46	10						1.257	
30	30				(f)		0.822	(f)
33	30 30 30			(f)		*0.819* .816 *.820*		(f)
34	30 30 30 30		(f)	0.854 .859		0.849 .820 .841 .841	0.844	
35	30			0.854				
36	30			0.850	0.873			
38	30				0.871			
56	30					0.968		
60	30 30						1.053 (f)	
64	35						0.961	
70	35						0.981	
54	40					(f)	(f)	(f)
60	40						1.001	
Igniter settings.[g]		+35°	+30°	+25°	+23°	+20°	+15°	+10°

[a] The figures in the body of the table give the fuel consumption, in pounds, per brake horsepower-hour, and are tabulated with respect to the fuel needle-valve and igniter settings. The figures in italics are the results of check tests of about 30 minutes' duration. The other figures in the body of the table are the results of trial tests of about 15 minutes' duration.

[b] Specific gravity at 60° F.=0.8188 to 0.8209. Per cent alcohol: By volume, 94.3 to 93.8; by weight, 91.5 to 90.8. Heating value: High=11,662 to 11,788; low=10,557 to 10,681 B. t. u. per pound.

[c] Pounds per square inch above atmosphere.

[d] Opening of fuel needle valve in hundredths of one revolution.

[e] Butterfly valve placed in the air pipe to the carburetor and set so as to throttle the air and supplement the governor-controlled inlet valve. Wide open at 80.

[f] Trial tests were made with these fuel needle-valve and igniter settings, but the engine would not carry the load.

[g] Point of ignition, in degrees, is given in terms of the crank position; +=early or before dead center.

TABLE 23.—*Results of trial and check tests made to determine the test fuel economy for each load*—Continued.

[OTTO 15-HORSEPOWER ALCOHOL ENGINE.—Diameter of cylinder, 6¾ inches; stroke, 15½ inches; speed, 251 to 265 revolutions per minute; throttle method of governing; governor controls lift and timing of mechanically operated inlet valve. Jacket-water temperature: Inlet, 60° F.; outlet 100° to 200° F.]

FUEL: DENATURED ALCOHOL; COMPRESSION PRESSURE, GREATEST, 140 POUNDS—Continued.

Fuel needle-valve setting.	Air-valve setting.	Fuel consumption per brake horsepower-hour (pounds).						
		Brake load, 130 pounds; B. H. P.., 14.6 to 15.3.						
74	40						0.969	
50	50					(a)	(a)	(a)
54	50						0.886	
60	50						0.916 .925	
70	65						0.917	
68	80 80					0.897 .886		
70						0.895		
		Brake load, 115 pounds; B. H. P., 13.3 to 13.6.						
18	0			(a)		0.895		
22	0		(a)					
19	10					0.844		
23	10 10		0.929 .964					
29	10		0.787					
22	15			0.865				
24	15		0.798					
26	15	(a)						
22	18 18		0.852 .841					
23	18		0.729	0.754				
24	18		(a)					
20	20	(a)	(a)	(a)				
22	20	0.787	0.762	0.812		0.803 .792		
23	20		(a)					
24	20					0.844		
25	20					0.730		
26	20		0.922			0.906		
27	20 20	0.751 .703						
29	20					0.907		
31	20		0.792					
30	25	0.807						
Igniter settings.[b]		+35°	+30°	+25°	+23°	+20°	+15°	+10°

[a] Trial tests were made with these fuel needle-valve and igniter settings, but the engine would not carry the load.
[b] Point of ignition, in degrees, is given in terms of the crank position; += early or before dead center.

Table 23.—*Results of trial and check tests made to determine the best fuel economy for each load*—Continued.

[Otto 15-Horsepower Alcohol Engine.—Diameter of cylinder, 6¾ inches; stroke, 15½ inches; speed, 251 to 265 revolutions per minute; throttle method of governing; governor controls lift and timing of mechanically operated inlet valve. Jacket-water temperature: Inlet, 60° F.; outlet, 100° to 200° F.]

FUEL: DENATURED ALCOHOL; COMPRESSION PRESSURE, GREATEST, 140 POUNDS—Continued.

Fuel needle-valve setting.	Air-valve setting.	Fuel consumption per brake horsepower-hour (pounds).						
		Brake load, 115 pounds, B. H. P.; 13.3 to 13.6.						
30	30					0.860		
36	30	(a)						
36	40						0.889	
56	40						0.934	
60	60 60						0.903 .893	
62	60						0.905	
62	70						0.863	
		Brake load, 100 pounds; B. H. P., 11.4 to 11.7.						
32	5				(a)	0.987	1.003 .989	(a)
28	10					(a)	0.985	(a)
30	10 10					1.024 .953	0.997	
32	10 10			(a)		1.043	0.984 .936	
34	10 10						1.000 .924	1.003
36	10						(a)	
38	10						1.118	
38	15					0.931		
48	15					0.984		
24	20					0.860		
48	25					0.885		
50	25					0.907		
52	25					0.924		
32	30					0.865		
56	35			0.910		0.903	0.877	
58	35					0.864		
37	40					*0.838*		
38	40			*0.842*		*0.839*		
60	40 40 40 40					0.869	0.969 .934 .898 .855	
60	50						1.040	
70	50					(a)		
Igniter settings.[b]		+35°	+30°	+25°	+23°	+20°	+15°	+10°

[a] Trial tests were made with these fuel needle-valve and igniter settings, but the engine would not carry the load.

[b] Point of ignition, in degrees, is given in terms of the crank position; += early or before dead center.

TABLE 23.—*Results of trial and check tests made to determine the best fuel economy for each load*—Continued.

[OTTO 15-HORSEPOWER ALCOHOL ENGINE.—Diameter of cylinder, 6¾ inches; stroke, 15¼ inches; speed, 251 to 265 revolutions per minute; throttle method of governing; governor controls lift and timing of mechanically operated inlet valve. Jacket-water temperature: Inlet, 60° F.; outlet, 100° to 200° F.]

FUEL: DENATURED ALCOHOL; COMPRESSION PRESSURE, GREATEST, 140 POUNDS—Continued.

Fuel needle-valve setting.	Air-valve setting.	Fuel consumption per brake horsepower-hour (pounds).						
		Brake load, 70 pounds; B. H. P., 8.0 to 8.2.						
15	0			1.064				
16	0 0			*0.960* *.966*				
20	0			1.041				
22	0			(a)				
24	10						1.007	
26	10 10					1.028	0.995 *1.010*	
28	10					1.061	1.055	
30	10						1.071	
32	10 10					1.117 1.043	1.075 .984	
34	10					1.168		
36	10					1.194		
38	10						1.118	
		Brake load, 40 pounds; B. H. P., 4.6 to 4.7.						
14	0			1.618				
15	0			1.575				
18	0			1.182				
20	0 0		1.216 1.230	1.165 *1.262*		1.502		
18	10			1.622				
28	10 10			1.488		1.488 1.330		
30	10 10 10 10		1.578	1.587 1.335		1.300 1.475 1.462 1.285	1.265 1.565 1.547 *1.271*	1.590
32	10			1.560		1.556	1.578	
40	10			1.748		1.537		
Igniter settings.[b]		+35°	+30°	+25°	+23°	+20°	+15°	+10°

[a] Trial tests were made with these fuel needle-valve and igniter settings, but the engine would not carry the load.

[b] Point of ignition, in degrees, is given in terms of the crank position; += early or before dead center.

TABLE 23.—*Results of trial and check tests made to determine the best fuel economy for each load*—Continued.

[OTTO 15-HORSEPOWER ALCOHOL ENGINE.—Diameter of cylinder, 6¾ inches; stroke, 15½ inches; speed, 251 to 265 revolutions per minute; throttle method of governing; governor controls lift and timing of mechanically operated inlet valve. Jacket-water temperature: Inlet, 60° F.; outlet, 100° to 200° F.]

FUEL: DENATURED ALCOHOL;[a] COMPRESSION PRESSURE, GREATEST, 180 POUNDS.[b]

Fuel needle-valve setting.	Air-valve setting.	Fuel consumption per brake horsepower-hour (pounds).						
		Brake load, 170 pounds; B. H. P., 19.81.						
48	40					(c)	(c) 0.702	(c)
		Brake load, 155.5 pounds; B. H. P., 18.0 to 18.1.						
54	60		(c)	(c)		(c)	(c)	
56	60				(c)	0.714	*0.705*	(c)
62	60				(c)		0.706	
		Brake load, 129 pounds; B. H. P., 14.9 to 15.1.						
20	10					(c)	0.735	
21	20			(c)	*0.714*	(c)		
21	25			(c)	(c)	(c)		
24	25 25 25			(c)	0.720 *.709* *.707*	(c)		
26	25 25		(c)		0.724	0.715 .750		
28	25						0.735	
29	30		(c)			0.754		
30	30				0.722	0.736		
40	40					0.783		
		Brake load, 111 pounds; B. H. P., 12.9 to 13.1.						
14	10		(c)	(c)	(c)			
15	10		(c)	*0.712*	(c)			
28	20			0.735				
24	25				0.727			
26	25				0.736			
42	50		0.739					
Igniter settings.[d]		+40°	+35°	+30°	+25°	+20°	+15°	+10°

[a] Specific gravity at 60° F.=0.8191. Per cent alcohol: By volume=94.2; by weight, 91.4. Heating value: High=11,750; low=10,643 B. t. u. per pound.
[b] Pounds per square inch above atmosphere.
[c] Trial tests were made with these fuel needle-valve and igniter settings, but the engine would not carry the load.
[d] Point of ignition, in degrees, is given in terms of the crank position; +=early or before dead center.

TABLE 23.—*Results of trial and check tests made to determine the best fuel economy for each load*—Continued.

[OTTO 15-HORSEPOWER ALCOHOL ENGINE.—Diameter of cylinder, 6¾ inches; stroke, 15½ inches; speed, 251 to 265 revolutions per minute; throttle method of governing; governor controls lift and timing of mechanically operated inlet valve. Jacket-water temperature: Inlet, 60° F.; outlet, 100° to 200° F.]

FUEL: DENATURED ALCOHOL; COMPRESSION PRESSURE, GREATEST, 180 POUNDS—Continued.

Fuel needle-valve setting.	Air-valve setting.	Fuel consumption per brake horsepower-hour (pounds).						
		Brake load, 85 pounds; B. H. P., 10.0.						
14	10	(a)	(a)	(a)				
16	10	(a)	0.808	(a)				
18	10					0.797		
19	10	0.802						
20	10			*0.778*		0.796		
21	10				*0.780*			
24	10			0.823				
28	10					0.823		
24	25				0.768			
		Brake load, 40 pounds; B. H. P., 4.7.						
10	0				1.510	1.455		
19	8		*1.133*					
24	25				1.188			
Igniter settings.[b]		+40°	+35°	+30°	+25°	+20°	+15°	+10°

FUEL: DENATURED ALCOHOL;[c] COMPRESSION PRESSURE, GREATEST, 210 POUNDS.[d]

Fuel needle-valve setting.[d]	Air-valve setting.[e]	Brake load, 140 pounds; B. H. P., 16.0 to 16.2.								
19	7						(a)	(a)	(a)	
21	7						(a)	0.820	(a)	
25	20						(a)	(a)	(a)	
27	20						(a)	0.849	(a)	
29	20						(a)	0.829	*0.853*	(a)
30	20									0.865
32	20									(a)
Igniter settings.[b]		+35°	+30°	+25°	+23°	+20°	+15°	+10°	+8°	+5°

[a] Trial tests were made with these fuel needle-valve and igniter settings, but the engine would not carry the load.
[b] Point of ignition, in degrees, is given in terms of the crank position; + =early or before dead center.
[c] Specific gravity at 60° F. =0.8191. Per cent alcohol: By volume =94.2; by weight =91.4. Heating value: High =11,750; low =10,643 B. t. u. per pound.
[d] Opening of fuel needle valve in hundredths of one revolution.
[e] Butterfly valve placed in the air pipe to the carburetor and set so as to throttle the air and supplement the governor-controlled inlet valve. Wide open at 80.

TABLE 23.—*Results of trial and check tests made to determine the best fuel economy for each load*—Continued.

[OTTO 15-HORSEPOWER ALCOHOL ENGINE.—Diameter of cylinder, 6¾ inches; stroke, 15½ inches; speed, 251 to 265 revolutions per minute; throttle method of governing; governor controls lift and timing of mechanically operated inlet valve. Jacket-water temperature: Inlet, 60° F.; outlet, 100° to 200° F.]

FUEL: DENATURED ALCOHOL, COMPRESSION PRESSURE, GREATEST, 210 POUNDS—Continued.

Fuel needle-valve setting.	Air-valve setting.	Fuel consumption per brake horsepower-hour (pounds).								
		Brake load, 130 pounds; B. H. P., 15.0 to 15.3.								
26	30	(a)								
28	30 30	0.662 .689	*0.677* .681	0.717 .683	*0.738* *.683*	*0.678*	(a)			
29	30 30	0.699	0.690 .685	0.683	0.704					
30	30						(a)			
36	40				(a)	0.730	(a)			
45	50			0.753						
		Brake load, 115 pounds; B. H. P., 13.2 to 13.6.								
18	20		(a)	*0.708*	(a)					
19	20	(a)		0.707	0.738					
24	20 20		0.743	0.710 *.710*		0.732				
25	20					0.730				
28	20 20	0.703 .751								
		Brake load, 100 pounds; B. H. P., 11.6 to 11.8.								
14	7				(a)	*0.811*				
15	7						0.819			
15	10 10	(a)	*0.753*	0.721 *.748*						
18	15					0.818				
		Brake load, 70 pounds; B. H. P., 8.2 to 8.4.								
13	5 5 5			0.945 .941 *.960*						
16				0.941						
Igniter settings.[b]		+35°	+30°	+25°	+23°	+20°	+15°	+10°	+8°	+5°

[a] Trial tests were made with these fuel needle-valve and igniter settings, but the engine would not carry the load.

[b] Point of ignition, in degrees, is given in terms of the crank position; + =early or before dead center.

TABLE 24.—*Results of tests of*

[OTTO 15-HORSEPOWER ALCOHOL ENGINE.—Diameter of cylinder, 6¾ inches; stroke, 15½ inches; throttle-governor; governor controls lift and timing of mechanically operated inlet valve; jacket-water temperature 60° F. For curves see figs. 24 and 26.]

FUEL: DENATURED ALCOHOL;[h] COMPRESSION PRESSURE, GREATEST, 140 POUNDS.[i]

Test number.	Temperatures (° F.).			Index settings.			Speed (average per minute).		Average pressures (pounds per sq. in.).			Brake load.	
	Air.	Mixture.[a]	Jacket-water outlet.	Air valve.[b]	Fuel needle valve.[c]	Igniter (degrees).[d]	Revolutions.	Explosions.	Compression (above atmosphere).	Mean effective.	Maximum (above atmosphere).	Pounds.	Per cent of maximum.
A 2164	70	58	Approximately 112.	35	70	+10	258.4	129	140	120.0	650	170	100
B 2169	74	67		30	42	+15	259.3	130	140	106.5	540	148	87
B 2171	89	71		30	42	+15	257.9	129	140	103.5	550	148	87
C 2183	88	68		30	33	+20	258.4	129	140	95.5	380	130	76
C 2184	91	69		30	33	+20	258.9	130	140	94.5	380	130	76
D 2191	93	68		20	22	+20	261.0	131	140	83.2	310	115	68
D 2192	95	69		20	22	+20	259.0	130	140	85.6	310	115	68
E 2197	88	71		40	37	+20	260.9	130	120	77.3	270	100	59
E 2198	90	73		40	38	+25	260.7	125	120	81.1	260	100	59
E 2196	88	70		40	38	+20	260.8	130	120	76.8	270	100	59
F 2202	82	60		0	16	+25	261.3	131	90	59.5	220	70	41
F 2204	86	56		0	16	+25	261.1	131	90	58.9	205	70	41
G 2213	62	59	200	0	20	+25	262.4	131	60	40.2	150	40	24
G 2008	94	80	103	0	30	+15	259.1	130	......	38.8	150	40	24
FUEL: DENATURED ALCOHOL;[j] COMPRESSION PRESSURE, GREATEST, 180 POUNDS.[i]													
H 2383	67	85	200	......	......	....	260.0	130	180	*[k]135.0*	720	190	100
I 2386	56	56	178	40	48	+15	260.0	130	180	*121.1*	680	170	89
J 2370	54	69	192	60	62	+17	260.0	130	180	*111.3*	560	155.5	82
J 2374	63	75	199	60	56	+15	259.4	130	180	*111.3*	545	155.5	82
K 2338	68	76	197	20	21	+25	258.7	129	160	*94.8*	470	129	68
K 2360	74	93	207	25	24	+25	259.1	130	180	*94.1*	460	129	68
K 2381	63	74	173	25	24	+25	259.2	130	180	*94.1*	460	129	68
L 2332	77	80	199	10	15	+30	259.5	130	160	*83.2*	370	111	58
M 2324	78	85	190	10	21	+25	263.2	131	120	*67.7*	330	85	45
M 2319	80	92	210	10	20	+30	264.0	132	120	*67.2*	300	85	45
N 2308	62	73	182	8	19	+35	262.6	131	60	*39.9*	....	40	21
FUEL: DENATURED ALCOHOL;[j] COMPRESSION PRESSURE, GREATEST, 210 POUNDS.[i]													
O 2268	59	82	......	20	29	+ 7	258.0	129	190	*102.1*	650	140	[l] 100
O 2271	61	90	......	20	29	+10	258.0	129	190	*102.1*	725	140	[l] 100
O 2297	51	70	......	7	21	+10	256.0	128	170	*102.4*	690	140	[l] 100
P 2265	71	86	150	30	28	+30	257.7	129	190	*95.5*	425	130	93
P 2277	71	88	171	30	28	+20	262.2	131	200	*95.1*	420	130	93
Q 2259	64	96	......	20	24	+25	258.8	129	190	*86.0*	455	115	82
Q 2283	85	95	174	20	18	+25	261.8	131	190	*85.8*	415	115	82
Q 2282	81	89	......	20	19	+25	263.4	132	200	*85.1*	390	115	82
R 2285	73	85	......	10	15	+25	262.3	131	170	*76.6*	285	100	71
R 2287	55	75	......	10	15	+25	262.8	131	170	*76.6*	330	100	71
S 2300	53	68	......	5	13	+25	262.4	131	130	*58.5*	210	70	50
S 2301	54	67	......	5	13	+25	265.6	133	130	*58.1*	230	70	50

[a] Measured by a thermometer placed in the inlet-valve housing (p. 58).
[b] Butterfly valve in air pipe to carburetor set so as to throttle the air and supplement the governor-controlled inlet valve (wide open at 80).
[c] Opening of the fuel needle valve in hundredths of one revolution.
[d] Point of ignition in degrees is given in terms of the crank position, += early, or before dead center; −= late, or after dead center (p. 55).
[e] Horsepower constants: Brake=0.0004478; indicated=0.001396 (p. 67).
[f] Calculated from the indicated horsepower and low heating value of the fuel (p. 72).
[g] Calculated from the brake horsepower and low heating value of the fuel (p. 72).

best fuel economy versus load.

[OTTO 15-HORSEPOWER ALCOHOL ENGINE.—Diameter of cylinder, 6¾ inches; stroke, 15½ inches; throttle governor; governor controls lift and timing of mechanically operated inlet valve; jacket-water temperature 60° F. For curves see figs. 24 and 26.]

FUEL: DENATURED ALCOHOL;[h] COMPRESSION PRESSURE, GREATEST, 140 POUNDS.[i]

Horsepower.[e]		Duration (minutes).	Fuel consumption.							Efficiencies (per cent).		
			Pounds per hour.	Per brake horsepower-hour.			Per indicated horse-power-hour.			Mechanical.	Thermal.	
Brake.	Indicated.			Pounds.	Gallons.	B. t. u. (low).	Pounds.	Gallons.	B. t. u. (low).		Indicated.[f]	Brake.[g]
19.69	21.7	8	18.72	0.951	0.139	10,040	0.86	0.126	9,120	90.9	27.9	25.4
17.20	19.3	14	14.74	.857	.125	9,050	.76	.111	8,070	89.2	31.5	28.1
17.10	18.6	25	14.60	.851	.124	8,980	.78	.114	8,270	91.7	30.8	28.3
15.05	17.2	15	12.34	.820	.120	8,660	.72	.104	7,560	87.4	33.7	29.4
15.07	17.1	29	12.34	.819	.119	8,650	.72	.105	7,620	88.2	33.4	29.4
13.45	15.8	14	10.65	.792	.116	8,360	.67	.098	7,070	84.9	35.8	30.4
13.35	15.7	28	10.71	.803	.117	8,480	.68	.099	7,180	86.0	35.4	30.0
11.69	14.1	12	9.80	.838	.122	8,850	.70	.102	7,350	82.9	34.6	28.7
11.67	14.2	24	9.82	.842	.123	8,890	.69	.101	7,330	82.5	34.7	28.6
11.68	14.0	12	9.80	.839	.122	8,860	.70	.102	7,400	83.6	34.4	28.7
8.20	10.8	15	7.87	.960	.140	10,130	.73	.106	7,650	75.5	33.3	25.1
8.19	10.7	30	7.91	.966	.141	10,200	.74	.107	7,770	76.2	32.8	24.9
4.70	7.3	30	5.93	1.262	.184	13,320	.81	.118	8,510	63.8	29.9	19.1
4.64	7.0	31	5.90	1.271	.186	13,580	.84	.123	8,970	66.1	28.4	18.7
FUEL: DENATURED ALCOHOL;[j] COMPRESSION PRESSURE, GREATEST, 180 POUNDS.[l]												
22.12	*24.5*	.....	.....	.....	.....	.....	.....	.....	.....	*90.3*	.....	.....
19.81	*22.0*	15	13.92	0.702	0.103	7,470	*0.63*	*0.092*	*6,720*	*90.0*	*37.9*	34.1
18.11	*20.2*	14	12.78	.706	.103	7,510	*.63*	*.092*	*6,720*	*89.5*	*37.9*	33.9
18.07	*20.2*	28	12.74	.705	.103	7,500	*.63*	*.092*	*6,720*	*89.4*	*37.9*	33.9
14.95	*17.1*	28	10.68	.714	.104	7,600	*.62*	*.091*	*6,640*	*87.4*	*38.3*	33.5
14.98	*17.1*	28	10.62	.709	.104	7,550	*.62*	*.091*	*6,610*	*87.5*	*38.5*	33.7
14.98	*17.1*	17	10.59	.706	.103	7,510	*.62*	*.091*	*6,580*	*87.5*	*38.7*	33.9
12.91	*15.1*	26	9.20	.712	.104	7,580	*.61*	*.089*	*6,460*	*85.3*	*39.4*	33.6
10.03	*12.4*	15	7.83	.780	.114	8,300	*.63*	*.092*	*6,730*	*81.2*	*37.8*	30.7
10.06	*12.4*	31	7.83	.778	.114	8,280	*.63*	*.092*	*6,750*	*81.3*	*37.7*	30.7
4.70	*7.3*	28	5.32	1.133	.166	12,060	*.73*	*.107*	*7,780*	*64.5*	*32.7*	21.1
FUEL: DENATURED ALCOHOL;[j] COMPRESSION PRESSURE, GREATEST, 210 POUNDS.[i]												
16.19	*18.4*	30	13.80	0.853	0.125	9,080	*0.75*	*0.110*	*7,980*	*88.1*	*31.8*	28.0
16.19	*18.4*	13	13.41	.829	.121	8,820	*.73*	*.107*	*7,770*	*88.1*	*32.8*	28.9
16.06	*18.3*	14	13.18	.820	.120	8,720	*.72*	*.105*	*7,660*	*88.0*	*33.2*	29.2
15.00	*17.2*	27	10.16	.677	.099	7,200	*.59*	*.086*	*6,280*	*87.4*	*40.4*	35.3
15.27	*17.4*	26	10.33	.678	.099	7,210	*.59*	*.086*	*6,280*	*87.6*	*40.3*	35.3
13.34	*15.5*	28	9.47	.710	.104	7,550	*.61*	*.089*	*6,490*	*86.0*	*39.2*	33.7
13.49	*15.7*	35	9.55	.708	.103	7,530	*.61*	*.089*	*6,490*	*86.2*	*39.2*	33.8
13.57	*15.7*	16	9.58	.707	.103	7,520	*.61*	*.089*	*6,490*	*86.3*	*39.2*	33.8
11.75	*14.0*	10	8.47	.721	.105	7,670	*.60*	*.088*	*6,440*	*84.0*	*39.5*	33.2
11.77	*14.0*	34	8.81	.748	.109	7,960	*.63*	*.092*	*6,700*	*84.0*	*88.1*	32.0
8.23	*10.7*	16	7.74	.941	.137	10,010	*.72*	*105*	*7,660*	*77.0*	*33.0*	25.4
8.33	*10.8*	23	8.00	.960	.140	10,210	*.74*	*.108*	*7,870*	*77.4*	*32.2*	24.9

[h] Specific gravity at 60° F.=0.821. Per cent alcohol by volume=93.8; by weight, 90.8. Heating value: High=11,662; low=10,557 B. t. u. per pound.

[i] Pounds per square inch above atmosphere.

[j] Specific gravity at 60° F.=0.819. Per cent alcohol by volume=94.2; by weight=91.4. Heating value: High=11,750 B. t. u. per pound; low=10,643 B. t. u. per pound.

[k] Italicized results indicate that the involved mean effective pressure was calculated from the mechanical efficiency. Load curve previously determined for this engine. Indicator diagrams were taken but not used (operator's note on test H).

[l] See operator's notes for test O.

OPERATOR'S NOTES (TABLE 24).

Test A.—Maximum load that the engine would carry with this compression. This load was only held with difficulty and the engine shut down after about 20 minutes. The vibration of the engine cylinder caused by the high-explosion pressures was such that it was feared the engine would be wrecked. No further attempt was made to carry the maximum load. The range of possible air-valve, fuel needle-valve, and igniter settings for this load was very small, and the best settings were readily determined by observation.

Test B.—The 5 trial tests were made with an air-valve setting of 30, which was found from observation to be the best. The range of fuel needle-valve and igniter settings was from 44 and +10° to 41 and + 15°, with corresponding fuel consumptions of 0.90 and 0.86 pound per brake horsepower per hour.

Test C.—The 36 trial tests were made with air-valve settings 0, 10, 20, 25, 30, 40, 50, 60, and 80. With air-valve settings from 0 to 20 the compression and maximum weight of mixture per charge, even with the governor-controlled inlet valve open wide, were so reduced as to require richer mixtures to carry the load than for the other air-valve settings. This was noted from the effect of the time of ignition on the combustion line of the indicator diagram, and was corroborated by the results of the tests. With air-valve settings from 40 to 80 the engine would not recover from a slight reduction in speed, caused either by fluctuation of load or hunting of governor, except when rich mixtures were used. With the air-valve setting at 30 the governor had the widest range of control and the engine operated satisfactorily for widely varying mixtures. The fuel needle-valve and igniter settings for the trial tests, with this air-valve position, ranged from the values given in the table to 72 and +20°, with a corresponding fuel consumption of 1.29 pounds per brake horsepower per hour.

Test D.—The 37 trial tests were made with air-valve settings from 0 to 70, as for tests C., and with similar results, although less in degree, as would be expected for the lighter load. The best air-valve setting was found to be 20, and the fuel consumption ranged from 0.95 pound per brake horsepower per hour to the values given in the table.

Test E.—The 36 trial tests were made with air-valve settings from 10 to 60. For this load the engine ran most steadily with the air valve set at 40, and correspondingly better results were obtained than for the other air-valve settings. Fuel consumptions varied from 1.12 pounds per brake horsepower per hour to the values given in the table. In test E 2198, hunting of governor caused an occasional misfire.

Test F.—The 19 trial tests were made with air-valve settings from 0 to 10, and the fuel consumptions for these tests ranged from 1.19 pounds per brake horsepower per hour to the values given in the table. With the air valve at 0 the passage was very much restricted, but not completely closed. The best results were obtained with this air-valve setting.

Test G.—The 29 trial tests were made with air-valve settings from 0 to 30 and the fuel consumptions for these tests varied from 1.75 pounds per brake horsepower per hour to the values given in the table. As for the previous load (Test F) an air-valve setting of 0 was found to be the best. In test G 2008, the mixture was throttled to such an extent at times as to cause misfiring.

Test H.—Maximum load that the engine would carry with this compression. No preignitions took place, but the engine was too light to withstand the extremely high explosion pressure, and no attempt was made to hold the load longer than was absolutely necessary to make an accurate determination of the brake horsepower. The range of possible settings was small. A liberal supply of fuel was used and the air valve opened wide. During the preliminary trials leading up to this load, it was found that the running of the engine was noticeably affected when the indicator cock was opened for taking diagrams, so that for these tests with high compression no reliance could be placed on the mean effective pressures determined in the usual way from the indicator diagrams. They were taken, however, as a guide to adjustment.

Test I.—It was possible to hold this load long enough to make an accurate determination of the fuel consumption, but it is not probable that the engine could have withstood the high explosion pressure for any great length of time. The range of possible settings was small and the best settings were readily determined by observation.

Test J.—This load was carried without difficulty and the best setting easily determined. Besides the two tests given in the table, a third was made with air valve at 60, fuel needle valve, at 56, and igniter +20°. The time of ignition was noted to be too early. The fuel consumption was 0.714 pound per brake horsepower per hour.

Test K.—The 16 trial tests were made with air-valve settings from 20 to 40, fuel needle-valve settings from 21 to 40, and ignition timing of from +15° to +35°. The fuel consumptions for these tests ranged from 0.78 pound per brake horsepower per hour to the values given in the table.

Test L.—The 5 trial tests made with this load were with widely varying air-valve settings of from 50 to 10. The corresponding fuel consumptions varied from 0.74 pound per brake horsepower per hour to the values given in the table.

Test M.—All of the 9 trial tests were made with an air-valve setting of 10. The fuel needle valve and ignition timing were varied from 16 and +35° to 28 and +20°, and the corresponding fuel consumptions were 0.81 and 0.82 pound per brake horsepower per hour. For the intermediate settings as given in the table, lower fuel consumptions were obtained.

Test N.—The 4 trial tests were made with settings of air valve 0, fuel needle valve 10, and ignition timings of +25° and 20°. The corresponding fuel consumptions for these tests were 1.51 and 1.46 pounds per brake horsepower per hour.

Test O.—Brake loads heavier than 140 pounds could not be carried on account of preignition. For this load preignition could only be prevented by a combination of settings tending to keep the compression low, or the mixture as weak as possible with the higher compressions. The weaker mixtures, with correspondingly earlier time of ignition than those given in the table, however, could not be obtained on account of the extremely high explosion pressures which threatened to wrench the unsupported cylinder from its fastenings and caused blowing out of gaskets. The 4 trial tests were made with the air valve set at 20 and the fuel needle-valve and ignition timing were varied from 30 and +5° to 27 and +10°. The fuel consumptions for these limiting settings were 0.87 and 0.85 pound per brake horsepower per hour. During the preliminary trials for these tests it was found that the running of the engine was appreciably affected when the indicator cock was opened for taking diagrams, so that no reliance could be placed on the mean effective pressures determined in the usual way from the indicator cards which were taken as a guide to adjustment.

Test P.—Preignition was prevented by a careful adjustment of the air valve and fuel needle valve. Under these conditions of preregulated mixture quality and compression, a steady load was carried without difficulty. The 17 trial tests were made with air-valve settings from 20 to 80. With an air-valve setting of 30, which was found to be the best, the fuel needle-valve and ignition timing were varied from 28 and +25° to 29 and +20°. Fuel consumptions for these tests, 10 of which are less than 0.70 pound per brake horsepower per hour, varied from 0.87 pound per brake horsepower per hour to the values given in the table.

Test Q.—For this and the lighter loads preignition was prevented by the automatic reduction of compression pressure caused by the increased throttling of the charge as the mixture was made richer, up to the point of greatest mean effective pressure. The 12 trial tests were made with air-valve settings of 15, 20, and 30. With the air-valve settings of 20, which was found to be the best, the range of fuel needle-valve and igniter settings was from 25 and +20° to 18 and +25°. The fuel consumptions for these tests varied from 0.75 pound per brake horsepower per hour to the values given in the table.

Test R.—The 6 trial tests were made with air-valve settings of 7, 10, and 15. Fuel consumptions for these tests ranged from 0.82 pound per brake horsepower per hour to the values given in the table. The best fuel needle-valve and igniter settings for each of the above air-valve settings were determined almost entirely by observation but with no great degree of certainty, their possible range being very wide.

Test S.—The 4 trial tests were made with practically the same settings as given in the table, the adjustment having been made almost entirely by observation. The fuel consumptions given are probably not as low as could have been obtained, had more time been given to the search for best settings.

TABLE 25.—*Results of tests of*

[OTTO 15-HORSEPOWER GASOLINE ENGINE No. 1.—Compression pressure, 70 pounds per square inch above atmosphere; diameter of cylinder, 6¾ inches; stroke, 15½ inches; hit-or-miss fuel-supply governor. For description, see page 28. Jacket-water temperature: Inlet, 60° F.; outlet, 112° F. For curves, see figs. 27, 28, and 29.]

Test number.	Temperatures (°F.).			Index settings.		Speed (average per minute).			Average pressures (pounds per sq. in.).		Brake load.	
	Air.	Fuel.	Mixture.[a]	Fuel needle valve.[b]	Igniter (degrees).[c]	Revolutions.	Explosions.	Fuel cut-outs.	Mean effective.	Maximum (above atmosphere).	Pounds.	Per cent of maximum.
FUEL: GASOLINE.[g]												
A 522	83	88	103	34	+ 9	260.5	129	2	91.4	300	115	88
B 558	82	88	102	34	+ 9	260.0	119	11	97.8	300	110	85
B 528	84	79	103	34	+ 9	261.3	123	8	95.8	310	110	85
C 555	98	100	108	34	+ 9	260.5	114	16	93.6	290	100	77
C 531	76	82	100	34	+ 9	260.1	117	13	89.6	260	100	77
D 534	80	87	101	34	+ 9	259.9	111	19	86.6	240	90	69
E 537	83	92	100	34	+ 9	260.3	112	18	79.1	200	80	62
E 552	96	95	107	34	+ 9	261.2	98	32	88.8	270	80	62
F 540	86	92	104	34	+ 9	262.8	86	45	83.4	240	60	46
F 549	93	96	107	34	+ 9	264.1	85	48	87.1	235	60	46
G 543	90	91	103	34	+ 9	264.3	68	59	79.8	245	40	31
G 546	92	94	105	34	+ 9	265.3	67	61	81.0	250	40	31
FUEL: DENATURED ALCOHOL.[h]												
H 642	90	85	64	44	+30	256.2	127	1	87.2	230	110	79
I 645	90	86	64	44	+30	256.1	127	1	80.6	220	100	71
J 648	91	88	64	44	+30	259.5	97	6	89.1	250	80	57
K 651	90	88	64	44	+30	260.2	77	15	86.1	240	60	43
L 654	88	86	65	44	+30	262.3	57	28	91.4	230	40	29
L 657	87	86	64	44	+30	261.1	55	24	88.4	210	40	29
FUEL: DENATURED ALCOHOL;[i] AIR PREHEATED TO 250° F.												
M 1394	249	87	101	44	+21	254.5	123	5	96.0	320	115	96
N 1406	248	88	102	44	+21	256.9	112	16	93.3	290	100	83
O 1397	251	89	107	44	+21	259.2	97	33	88.2	260	80	67
P 1400	253	91	107	44	+21	260.3	87	44	82.6	240	60	50
Q 1403	249	88	108	44	+21	261.1	78	52	96.8	290	40	33

[a] Measured by a thermometer placed in the inlet-valve housing (p. 57).
[b] Opening of the fuel needle valve in hundredths of one revolution.
[c] Point of ignition, in degrees, is given in terms of the crank position, + = early, or before dead center (p. 55).
[d] Horsepower constants: Brake=0.0004817; indicated=0.0014 (p. 67).
[e] Calculated from the indicated horsepower and the low heating value of the fuel (p. 72).
[f] Calculated from the brake horsepower and the low heating value of the fuel (p. 72).

normal fuel economy versus load.

[OTTO 15-HORSEPOWER GASOLINE ENGINE No. 1.—Compression pressure, 70 pounds per square inch above atmosphere; diameter of cylinder, 6¾ inches; stroke, 15½ inches; hit-or-miss fuel-supply governor. For description, see page 28. Jacket-water temperature: Inlet, 60° F.; outlet, 112° F. For curves, see figs. 27, 28, and 29.]

FUEL: GASOLINE.[g]

Horsepower.[d]		Duration (minutes).	Fuel consumption.								Efficiencies (per cent).		
			Pounds per hour.	Average per charge (pounds).	Per brake horse-power-hour.			Per indicated horse-power-hour.			Mechanical.	Thermal.	
Brake.	Indicated.				Pounds.	Gallons.	B. t. u. (low).	Pounds.	Gallons.	B. t. u. (low).		Indicated.[e]	Brake.[f]
14.44	16.5	33	9.07	0.00117	0.628	0.103	12,010	0.55	0.090	10,540	87.6	24.1	21.2
13.80	16.3	34	8.86	.00124	.642	.105	12,280	.54	.089	10,400	84.8	24.4	20.7
13.84	16.5	31	8.83	.00120	.638	.105	12,200	.53	.087	10,250	84.0	24.8	20.9
12.55	15.0	33	8.17	.00119	.650	.107	12,440	.54	.089	10,420	83.8	24.4	20.5
12.54	14.7	33	8.10	.00115	.646	.106	12,360	.55	.090	10,560	85.5	24.1	20.6
11.26	13.4	32	7.56	.00114	.671	.110	12,840	.56	.092	10,770	84.0	23.6	19.8
10.03	12.4	33	7.39	.00110	.735	.121	14,050	.59	.097	11,360	80.6	22.4	18.1
10.07	12.2	30	7.02	.00119	.697	.114	13,340	.57	.094	10,980	82.3	23.2	19.1
7.59	10.1	30	6.10	.00118	.804	.132	15,380	.60	.099	11,570	75.3	22.0	16.5
7.64	10.3	35	6.06	.00119	.793	.130	15,160	.59	.097	11,280	74.3	22.6	16.8
5.09	8.1	28	5.30	.00121	1.040	.171	19,900	.65	.107	12,450	62.5	20.4	12.8
5.10	8.1	35	5.17	.00129	1.014	.166	19,400	.64	.105	12,160	62.7	20.9	13.1

FUEL: DENATURED ALCOHOL.[h]

13.58	15.5	32	14.28	0.00187	1.051	0.153	11,070	0.92	0.134	9,700	87.3	26.3	23.0
12.34	14.3	32	13.92	.00183	1.128	.165	11,880	.97	.142	10,300	86.3	24.8	21.4
10.00	12.1	31	13.40	.00180	1.340	.196	14,110	1.11	.162	11,700	82.6	21.8	18.0
7.52	9.1	29	12.41	.00180	1.650	.241	17,390	1.34	.196	14,100	81.0	18.0	14.6
5.06	7.3	19	11.15	.00180	2.203	.322	23,210	1.53	.223	16,100	69.3	15.8	10.9
5.04	6.8	31	11.45	.00178	2.272	.332	23,930	1.68	.245	17,700	74.0	14.4	10.6

FUEL: DENATURED ALCOHOL;[i] AIR PREHEATED TO 250° F.

14.10	16.5	31	15.66	0.00212	1.110	0.162	11,760	0.95	0.139	10,050	85.5	25.3	21.6
12.37	14.7	29	14.36	.00214	1.161	.169	12,300	.98	.143	10,380	84.3	24.5	20.7
9.99	11.9	34	12.23	.00210	1.225	.179	12,980	1.03	.150	10,860	83.6	23.4	19.6
7.52	9.7	33	10.88	.00208	1.447	.211	15,340	1.12	.164	11,850	77.4	21.5	16.6
5.03	7.6	31	9.66	.00207	1.918	.280	20,310	1.27	.185	13,460	66.3	18.9	12.5

[g] Specific gravity at 60° F.=0.730. Heating value: High=20,407; low=19,125 B. t. u. per pound.

[h] Specific gravity at 60° F.=0.820. Per cent alcohol, by volume=92.5; by weight=90.9. Heating value: High=11,642, low=10,534 B. t. u. per pound.

[i] Specific gravity at 60° F.=0.818. Per cent alcohol, by volume=93.1; by weight=91.6. Heating value: High=11,695, low=10,595 B. t. u. per pound.

OPERATOR'S NOTES (TABLE 25).

Test A.—The adjustment of the load, fuel needle valve, and time of ignition for this test was that for which the engine consumed the least amount of gasoline per brake horsepower-hour. (Table 15, p. 135.) In test A522 there was a heavy explosion pound.

Test B.—No. 558, light explosion pound; No. 528, heavy explosion pound.

Test C.—No. 555, light explosion pound; No. 531, no explosion pound.

Tests D to F, inclusive.—Running of engine normal.

Test G.—No. 543, about 1 in every 7 of the first fuel charges after cut-outs misfired. Mixture apparently too weak to burn; ignition not at fault.

Test H.—The adjustment of the load, fuel needle valve, and time of ignition for this test was that for which the engine consumed the least amount of denatured alcohol per brake horsepower-hour. (Table 15, p. 135.) In test H642 first fuel charge after cutouts occasionally misfired.[a]

[a] Mixture seemingly irregular in quality and sometimes too weak to ignite. Electric igniter not at fault.

Test I.—No. 645, first fuel charge after cut-outs occasionally misfired.[a]

Test J.—No. 648, first fuel charge after cut-outs frequently misfired.[a]

Test K.—No. 651, first fuel charge after cut-outs misfired about every other time.[a]

Test L.—No. 654, first fuel charge after cut-outs misfired nearly every time.[a]

Test M.—A load which was very nearly the greatest the engine would carry when supplied with denatured alcohol and the air was preheated to 250°F. was applied; the adjustment of the fuel needle valve and time of ignition was that for which the engine consumed the least amount of denatured alcohol for this load and air temperature. (Table 15, p. 135.) Running of engine normal.

Tests N and O.—Running of engine normal.

Test P.—No. 1400, first fuel charge after cut-outs misfired at times.[a]

Test Q.—No. 1403, first fuel charge after cut-outs misfired at times.

[a] Mixture seemingly irregular in quality and sometimes too weak to ignite. Electric igniter not at fault.

TABLE 26.—*Results of tests of normal*

[OTTO 15-HORSEPOWER GASOLINE ENGINE No. 2.—Compression pressure, 70 pounds per square inch above atmosphere; diameter of cylinder, 6¾ inches; stroke, 15½ inches; hit-or-miss fuel-supply governor; inlet temperature of jacket-water, approximately 60° F. For curves, see figs. 27, 28, and 29.]

FUEL: GASOLINE.[g]

Test number.	Temperatures (°F.). Air.	Temperatures (°F.). Fuel.	Temperatures (°F.). Mixture.[a]	Temperatures (°F.). Jacket-water outlet.	Index settings. Fuel needle valve.[b]	Index settings. Igniter (degrees).[c]	Speed (av. per minute). Revolutions.	Speed (av. per minute). Explosions.	Speed (av. per minute). Fuel cut-outs.	Average pressures (pounds per sq. in.). Mean effective.	Average pressures (pounds per sq. in.). Maximum (above atmosphere.)	Brake load. Pounds.	Brake load. Per cent of maximum.
A 1798	83	85	79	121	56	+10	250.4	122	3	102.5	400	135	100
A 1782	83	89	80	126	56	+10	252.1	124	2	105.8	410	135	100
B 1785	84	83	81	114	56	+10	252.1	108	18	102.2	400	115	85
B 1801	85	86	82	116	56	+10	252.3	109	17	103.6	390	115	85
C 1789	84	88	80	114	56	+10	251.3	102	23	97.1	370	100	74
C 1804	86	86	83	114	56	+10	254.5	102	25	99.9	360	100	74
D 1791	84	88	82	105	56	+10	255.1	84	44	91.3	250	70	52
D 1807	87	88	87	112	56	+10	257.1	82	47	95.6	295	70	52
E 1794	83	89	85	114	56	+10	257.5	69	60	78.0	210	40	30
E 1809	87	92	89	100	56	+10	260.3	66	65	85.7	....	40	30

FUEL: DENATURED ALCOHOL.[h]

Test number.	Air.	Fuel.	Mixture.	Jacket-water outlet.	Fuel needle valve.	Igniter (degrees).	Revolutions.	Explosions.	Fuel cut-outs.	Mean effective.	Maximum.	Pounds.	Per cent of maximum.
F 1903	80	82	61	117	124	+15	254.1	121	6	109.7	390	150	100
G 1924	77	84	53	113	124	+15	255.2	109	19	108.5	360	130	87
H 1927	78	89	53	112	124	+15	254.8	92	36	102.8	300	100	67
I 1930	78	81	53	91	124	+15	257.3	77	43	89.1	220	70	47
J 1933	77	77	51	84	124	+15	258.6	53	46	88.7	215	40	27

[a] Measured by a thermometer placed in the inlet-valve housing. (p. 57.)
[b] Opening of the fuel needle valve in hundredths of one revolution.
[c] Point of ignition, in degrees, is given in terms of the crank position, +=early, or before dead center. (p. 55.)
[d] Horsepower constants: brake 0.0004493; indicated 0.001396. (p. 67.)

OPERATOR'S NOTES (TABLE 26).

Test A.—The greatest load the engine would carry when supplied with gasoline was applied, and the adjustment of the fuel needle valve and time of ignition was that for which the engine consumed the least amount of gasoline, that is, the least for the greatest load. (Table 17.) In test A1782 there was an explosion pound.

Test B.—No. 1785, explosion pound.

Test C.—No. 1789, light explosion pound.

Tests D and E.—No explosion pound.

fuel economy versus load.

[OTTO 15-HORSEPOWER GASOLINE ENGINE NO. 2.—Compression pressure, 70 pounds per square inch above atmosphere; diameter of cylinder, 6¾ inches; stroke, 15½ inches; hit-or-miss fuel-supply governor; inlet temperature of jacket-water, approximately 60° F. For curves, see figs. 27, 28, and 29.]

FUEL: GASOLINE.[g]

Horsepower.[d]			Fuel consumption.								Efficiencies (per cent).		
					Per brake horse-power-hour.			Per indicated horse-power-hour.				Thermal.	
Brake.	Indicated.	Duration (minutes).	Pounds per hour.	Average per charge (pounds).	Pounds.	Gallons.	B. t. u. (low).	Pounds.	Gallons.	B. t. u. (low).	Mechanical.	Indicated.[e]	Brake.[f]
15.19	17.5	34	9.85	0.00134	0.648	0.108	12,460	0.56	0.094	10,810	86.7	23.5	20.4
15.30	18.3	28	9.79	.00132	.640	.107	12,310	.54	.089	10,310	83.7	24.7	20.7
13.04	15.4	24	8.75	.00135	.672	.112	12,930	.57	.095	10,910	84.5	23.3	19.7
13.04	15.8	34	8.84	.00135	.678	.113	13,040	.56	.094	10,790	82.8	23.6	19.5
11.29	13.9	30	8.02	.00131	.711	.119	13,680	.58	.096	11,120	81.3	22.9	18.6
11.43	14.2	33	8.08	.00132	.707	.118	13,600	.57	.095	10,940	80.5	23.3	18.7
8.03	10.7	29	6.21	.00123	.774	.129	14,890	.58	.097	11,180	75.0	22.8	17.1
8.09	10.9	32	6.48	.00132	.801	.134	15,410	.59	.099	11,390	74.0	22.3	16.5
4.63	7.5	29	5.09	.00123	1.121	.187	21,560	.70	.116	13,370	62.0	19.0	11.8
4.68	7.9	35	5.08	.00128	1.086	.181	20,890	.65	.108	12,450	59.6	20.4	12.2
FUEL: DENATURED ALCOHOL.[h]													
17.14	18.5	29	19.11	0.00263	1.1 5	0.163	11,910	1.03	0.151	11,040	92.7	23.1	21.4
14.91	16.5	30	15.25	.00233	1.023	.150	10,930	.92	.135	9,870	90.3	25.8	23.3
11.45	13.2	31	13.66	.00248	1.192	.174	12,730	1.04	.152	11,080	86.9	23.0	20.0
8.09	9.6	32	12.11	.00235	1.498	.219	16,000	1.26	.185	13,500	84.5	18.9	15.9
4.65	6.5	30	12.16	.00244	2.617	.383	27,950	1.87	.274	19,980	71.5	12.7	9.1

[e] Calculated from the indicated horsepower and the low heating value of the fuel. (p. 72.)
[f] Calculated from the brake horsepower and the low heating value of the fuel. (p. 72.)
[g] Specific gravity at 60° F.=0.717. Heating value: high=20,527, low=19,235 B. t. u. per pound.
[h] Specific gravity at 60° F.=0.819. Per cent alcohol by volume=94.3; by weight=91.5. Heating value: High=11,788, low=10,681 B. t. u. per pound.

Test F.—The greatest load the engine would carry when supplied with denatured alcohol was applied and the adjustment of the fuel needle valve and time of ignition was that for which the engine consumed the least amount of denatured alcohol, that is, the least for the greatest load. (Table 17.)

Test I.—No. 1930, first fuel charge after cut-outs occasionally misfired. Mixture too weak to burn; ignition not at fault.

Test J.—No. 1933, the first fuel charge after cut-out misfired. Mixture too weak to burn; ignition not at fault.

TABLE 27.—*Results of tests of*

[NASH 10-HORSEPOWER GASOLINE ENGINE.—Compression pressure, greatest, 96 pounds per square inch above atmosphere; cylinder diameter, 7⅝ inches; stroke, 10 inches; hit-or-miss mixture-supply governor; inlet temperature of jacket water, 60° to 80° F.; outlet temperature, 109° to 115° F. For curves see fig. 27.]

Test number.	Temperatures (° F.).			Index settings.			Speed (average per minute).			Average pressures (pounds per square inch).			Brake load.	
	Air.	Mixture near carburetor.[a]	Mixture near inlet valve.[a]	Air valve.[b]	Fuel needle valve.[c]	Igniter (degrees).[d]	Revolutions.	Explosions.	Mixture cut-outs.	Compression (above atmosphere).	Mean effective.	Maximum (above atmosphere).	Pounds.	Per cent of maximum.
						FUEL: GASOLINE;[h] REGULAR CARBURETOR.[i]								
A 3338	....	......	......	$\frac{1}{2}$	$1\frac{15}{16}$	$+6\frac{1}{2}=0$	297.0	145	4	72	69.0	195	80	89
B 3340	....	......	......	$\frac{1}{2}$	$1\frac{15}{16}$	$+6\frac{1}{2}=0$	297.1	142	6	72	67.8	150	70	78
C 3342	....	......	......	$\frac{1}{2}$	$1\frac{15}{16}$	$+6\frac{1}{2}=0$	297.4	130	19	72	65.4	140	60	67
D 3344	....	......	......	$\frac{1}{2}$	$1\frac{15}{16}$	$+6\frac{1}{2}=0$	298.9	116	37	72	54.3	135	40	44
						FUEL: DENATURED ALCOHOL *j*; DOUBLE-CONE CARBURETOR.[k]								
E 3743	68	47	64	$\frac{1}{2}$	$2\text{–}3\frac{1}{2}$	$+2\frac{3}{4}=17$	303.1	144	7	92	98.2	350	116.5	80
F 3745	69	48	63	$\frac{1}{2}$	$2\text{–}3\frac{1}{2}$	$+2\frac{3}{4}=17$	306.4	122	31	92	95.6	270	91.0	63
G 3747	72	50	64	$\frac{1}{2}$	$2\text{–}3\frac{1}{2}$	$+2\frac{3}{4}=17$	308.1	108	51	92	84.2	210	66.0	41
						FUEL: DENATURED ALCOHOL *l*; REGULAR CARBURETOR.[i]								
H {3426	107	......	......	$\frac{5}{8}$	$3\frac{7}{16}$	$+2=20$	294.7	144	3	78	82.2	290	100.0	85
H {3440	96	......	......	$\frac{5}{8}$	$3\frac{7}{16}$	$+2=20$	296.1	146	2	78	83.4	310	100.0	85
I {3430	108	......	......	$\frac{5}{8}$	$3\frac{7}{16}$	$+2=20$	299.8	125	25	78	83.4	310	80.0	65
I {3438	95	......	......	$\frac{5}{8}$	$3\frac{7}{16}$	$+2=20$	298.9	130	19	78	80.4	280	80.0	65
J {3432	106	......	......	$\frac{5}{8}$	$3\frac{7}{16}$	$+2=20$	302.4	113	38	78	72.6	220	60.0	51
J {3436	92	......	......	$\frac{5}{8}$	$3\frac{7}{16}$	$+2=20$	300.9	116	35	78	73.2	225	60.0	51
K 3434	91	......	......	$\frac{5}{8}$	$3\frac{7}{16}$	$+2=20$	301.5	97	34	78	60.3	175	40.0	34

[a] Measured by a thermometer placed in the inlet-valve housing (p. 57).
[b] Plug cock with dial for regulating the auxiliary air supply which is introduced between the governor-controlled mixture valve and the main inlet valve.
[c] Scale arbitrary. Numbers increase with opening of needle valve.
[d] Scale arbitrary. Rotation to crank position not constant. Ignition before dead center is indicated by (+).
[e] Horsepower constant: brake 0.000395; indicated=0.001153 (p. 67).
[f] Calculated from the indicated horsepower and the low heating value of the fuel (p. 72).

OPERATOR'S NOTES (TABLE 27).

Test A.—The adjustment of the load, fuel needle valve, auxiliary air valve, and time of ignition for this test was that for which the engine consumed approximately the least amount of gasoline per brake horsepower-hour when equipped with the regular carburetor. (Table 19.)

Tests A to D, inclusive.—Running of the engine normal.

Test E.—The adjustment of the load, fuel needle valve, auxiliary air valve, and time of ignition for this test was that for which the engine consumed the least amount of denatured alcohol per brake horsepower-hour when equipped with the double-cone carburetor. (Table 19.)

normal fuel economy versus load.

[NASH 10-HORSEPOWER GASOLINE ENGINE.—Compression pressure, greatest, 96 pounds per square inch above atmosphere; cylinder diameter, 7⅝ inches; stroke, 10 inches; hit-or-miss mixture-supply governor; inlet temperature of jacket water, 60° to 80° F.; outlet temperature, 109° to 115° F. For curves see fig. 27.]

Horsepower.[e]		Duration (minutes).	Fuel consumption.							Efficiencies (per cent).		
				Per brake horsepower-hour.			Per indicated horsepower-hour.				Thermal.	
Brake.	Indicated.		Pounds per hour.	Pounds.	Gallons.	B. t. u. (low).	Pounds.	Gallons.	B. t. u. (low).	Mechanical.	Indicated.[f]	Brake.[g]
FUEL: GASOLINE;[h] REGULAR CARBURETOR.[i]												
9.38	11.6	31	5.82	0.621	0.104	11,940	0.50	0.084	9,660	80.9	26.3	21.3
8.22	11.1	32	5.58	.679	.113	13,050	.50	.084	9,660	74.0	26.4	19.5
7.04	9.8	31	5.35	.760	.127	14,600	.55	.091	10,480	71.7	24.3	17.4
4.72	7.3	30	4.67	.990	.165	19,040	.64	.107	12,360	65.0	20.6	13.4
FUEL: DENATURED ALCOHOL [j]; DOUBLE-CONE CARBURETOR.[k]												
13.94	16.3	28	12.97	0.930	0.136	9,818	0.80	0.115	8,390	85.4	30.5	25.9
11.02	13.4	32	11.23	1.020	.149	10,768	.84	.121	8,820	81.9	28.8	23.6
8.03	10.4	31	9.81	1.222	.178	12,901	.94	.136	9,940	77.0	25.6	19.7
FUEL: DENATURED ALCOHOL[l]; REGULAR CARBURETOR.[i]												
11.64	13.6	36	12.05	1.036	0.151	11,066	0.88	0.129	9,420	85.2	27.0	23.0
11.69	14.0	30	12.08	1.034	.151	11,044	.86	.126	9,180	83.1	27.7	23.0
9.47	12.0	31	10.82	1.143	.167	12,208	.90	.131	9,600	78.7	26.5	20.9
9.44	12.1	28	10.81	1.145	.167	12,230	.89	.131	9,560	78.1	26.6	20.8
7.16	9.5	31	9.65	1.347	.197	14,387	1.02	.149	10,870	75.5	23.4	17.7
7.13	9.8	28	9.68	1.356	.198	14,483	.99	.145	10,570	72.9	24.0	17.5
4.76	6.7	30	9.14	1.919	.281	20,497	1.35	.198	14,440	70.4	17.6	12.4

g Calculated from the brake horsepower and the low heating value of the fuel (p. 72).
h Specific gravity at 60° F.=0.7168. Heating value: high=20,527, low=19,235 B. t. u. per pound.
i For description of regular carburetor p. 34.
j Specific gravity at 60° F.=0.8206. Per cent alcohol: by volume=93.8; by weight=90.8. Heating value: high=11,750, low=10,643 B. t. u. per pound.
k For description of double-cone carburetor p. 34.
l Specific gravity at 60° F.=0.8188. Per cent alcohol: by volume, 94.3; by weight, 91.5. Heating value high=11,788, low=10,681 B. t. u. per pound.

Tests F and G.—Running of the engine normal.

Test H.—The adjustment of the load, fuel needle valve, auxiliary air valve, and time of ignition for this test was that for which the engine consumed the least amount of denatured alcohol per brake horsepower-hour when equipped with the regular carburetor.(Table 19.)

Tests H to K, inclusive.—Mixture quality seemingly became more and more irregular as the load was reduced, causing occasional misfiring of the first charges after cut-outs. Firing during cut-outs also became more noticeable as the load was reduced, but did not affect the running of the engine appreciably. This firing during cut-outs was caused by the auxiliary air supply picking up sufficient fuel that was left in the passageway to form a weak explosive mixture.

TABLE 28.—*Results of tests of*

[NASH 10-HORSEPOWER ALCOHOL ENGINE.—Compression pressure, greatest, 135 pounds per square inch above atmosphere; cylinder diameter, 7⅝ inches; stroke, 10 inches; hit-or-miss mixture-supply governor; inlet temperature of jacket water, 60° to 80° F.; outlet temperature, 100° to 200° F. For curves, see fig. 27.]

Test number.	Air temperature.	Index settings.			Speed (average per minute).			Average pressures (pounds per sq. in.).			Brake load.	
		Air valve.[a]	Fuel needle valve.[b]	Igniter (degrees).[c]	Revolutions.	Explosions.	Mixture cut-outs.	Compression (above atmosphere).	Mean effective.	Maximum (above atmosphere).	Pounds.	Per cent of maximum.
FUEL: GASOLINE WITH SPRAY OF WATER;[h] REGULAR CARBURETOR.[i]												
A 4021	72	5/8	6 1/8	4	303.1	146	5	130	86.0	300	100	81
B 4047	72	5/8	6 1/8	4	305.9	136	18	130	81.4	280	83	67
C 4049	78	5/8	6 1/8	4	306.7	124	29	130	73.2	265	66	53
D 4051	75	5/8	6 1/8	4	307.9	112	40	130	68.4	210	50	41
FUEL: DENATURED ALCOHOL;[j] DOUBLE-CONE CARBURETOR.[k] (CASE NO. 1.)												
E 4201	79	9/16	2 to 8	+1 1/2	300.1	144	6	130	102.9	455	140.0	100
F 4203	80	9/16	2 to 8	+1 1/2	302.5	137	30	130	99.6	490	116.5	83
G 4206	81	9/16	2 to 8	+1 1/2	303.0	117	57	130	93.6	500	85.0	61
H 4208	72	9/16	2 to 8	+1 1/2	306.5	81	74	130	96.6	290	59.0	42
FUEL: DENATURED ALCOHOL;[j] DOUBLE-CONE CARBURETOR.[k] (CASE NO. 2.)												
I 4193	60	5/8	2 to 3 1/2	+1	302.0	147	4	133	87.6	380	116.5	83
J 4195	70	5/8	2 to 3 1/2	+1	303.7	129	23	133	79.2	265	85.0	61
K 4197	73	5/8	2 to 3 1/2	+1	303.3	115	36	133	65.4	205	59.0	42

[a] Plug cock with dial for regulating the auxiliary air supply which is introduced between the governor-controlled mixture valve and the main inlet valve (p. 33).
[b] Scale arbitrary. Numbers increase with opening of needle valve.
[c] Scale arbitrary. Relation to crank position not constant. Ignition before dead center is indicated by (+).
[d] Horsepower constant: Brake=0.000395; indicated=0.001153 (p. 67).
[e] Calculated from the indicated horsepower and the lower heating value of the fuel (p. 72).
[f] Calculated from the brake horsepower and the lower heating value of the fuel (p. 72).

OPERATOR'S NOTES (TABLE 28).

Test A.—The adjustment of the load, fuel needle valve, auxiliary air valve, and time of ignition for this test was that for which the engine consumed the least amount of fuel per brake horsepower-hour. (Table 19.) Just sufficient water was introduced to prevent preignition from the high compression.

Tests A to D, inclusive.—Running of engine normal.

Test E.—The greatest load the engine would carry was applied for this test and the adjustment of the fuel needle valve, auxiliary air valve, and time of ignition was that for which the engine consumed the least fuel, that is, the least fuel for the greatest load. (Table 19.)

normal fuel economy versus load.

[NASH 10-HORSEPOWER ALCOHOL ENGINE.—Compression pressure, greatest, 135 pounds per square inch above atmosphere; cylinder diameter, 7⅝ inches; stroke, 10 inches; hit-or-miss mixture-supply governor; inlet temperature of jacket water, 60° to 80° F.; outlet temperature, 100° to 200° F. For curves, see fig. 27.]

FUEL: GASOLINE WITH SPRAY OF WATER;[h] REGULAR CARBURETOR.[i]

Horsepower.[d]		Duration (minutes).	Fuel consumption.							Efficiencies (per cent).			Per cent gasoline by weight.[g]
				Per brake horsepower-hour.			Per indicated horse-power-hour.				Thermal.		
Brake.	Indicated.		Pounds per hour.	Pounds.	Gallons.	B. t. u. (low).	Pounds.	Gallons.	B. t. u. (low).	Mechanical.	Indicated.[e]	Brake.[f]	
11.98	14.5	31	7.68	0.641	0.107	11,510	0.53	0.088	9,520	82.7	26.7	22.1	42.0
10.02	12.8	30	7.16	.714	.119	12,940	.56	.094	10,180	78.6	25.0	19.7	45.4
8.00	10.5	32	6.67	.834	.139	15,120	.64	.109	11,530	76.2	22.1	16.8	45.5
6.08	8.8	30	6.46	1.062	.177	19,210	.73	.122	13,270	69.1	19.2	13.3	44.4

FUEL: DENATURED ALCOHOL;[j] DOUBLE-CONE CARBURETOR.[k] (CASE NO. 1.)

16.60	16.9	25	16.50	0.994	0.145	10,580	0.98	0.142	10,360	98.0	24.6	24.1	......
13.92	15.7	32	14.92	1.071	.156	11,400	.95	.138	10,090	88.5	25.2	22.3	......
10.18	12.6	27	13.16	1.294	.189	13,760	1.04	.152	11,090	80.6	23.0	18.5	......
7.15	9.0	32	11.33	1.585	.231	16,860	1.26	.184	13,420	79.6	19.0	15.1	......

FUEL: DENATURED ALCOHOL;[j] DOUBLE-CONE CARBURETOR.[k] (CASE NO. 2.)

13.90	14.8	19	12.63	0.909	0.133	9,670	0.85	0.125	9,110	94.3	27.9	26.3	......
10.20	11.8	32	11.25	1.103	.161	11,730	.96	.140	10,190	86.8	25.0	21.7	......
7.07	8.7	30	10.26	1.451	.212	15,450	1.18	.173	12,570	81.6	20.2	16.5	......

g Ratio of gasoline to gasoline + water, by weight.
h Specific gravity of gasoline at 60° F.=0.7175. Heating value: High=20,579, low=19,289 B. t. u. per pound.
i For description of regular carburetor, see p. 34.
j Specific gravity at 60° F.=0.8191. Per cent alcohol: By volume, 94.2; by weight, 91.4. Heating value: High=11,750, low=10,643 B. t. u. per pound.
k For description of double-cone carburetor, see p. 34.

Tests E to H, inclusive.—Running of engine normal.

Test I.—The adjustment of the load, fuel needle valve, auxiliary air valve, and time of ignition for this test was that for which the engine consumed the least amount of fuel per brake horsepower-hour. (Table 19.)

Tests I to K, inclusive.—Running of engine normal.

TABLE 29.—*Results of tests of*

[OTTO 15-HORSEPOWER ALCOHOL ENGINE.—Cylinder diameter, 6¾ inches; stroke, 15½ inches; throttle governor; governor controls the lift and timing of the main inlet valve; inlet temperature of the jacket water, 60° to 80° F. For curves see fig. 27.]

FUEL: DENATURED ALCOHOL;[h] COMPRESSION PRESSURE, GREATEST 140 POUNDS.[i]

Test number.	Temperature (° F.). Air.	Temperature (° F.). Mixture.[a]	Temperature (° F.). Jacket-water outlet.	Index settings. Air valve.[b]	Index settings. Fuel needle valve.[c]	Index settings. Igniter (degrees).[d]	Speed (average per minute). Revolutions.	Speed (average per minute). Explosions.	Average pressures (pounds per sq. in.). Compression (above atmosphere).	Average pressures (pounds per sq. in.). Mean effective.	Average pressures (pounds per sq. in.). Maximum (above atmosphere).	Brake load. Pounds.	Brake load. Per cent of maximum.
A 2222	68	58	....	30	33	+30	258.7	129	140	96.0	430	130	76
B 2223	67	56	....	30	33	+30	260.9	130	120	87.0	375	115	68
C 2225	66	57	....	30	33	+30	260.9	130	120	78.0	310	100	59
D 2227	64	55	....	30	33	+30	263.1	132	90	58.5	190	70	41
E 2231	62	54	....	30	33	+30	263.6	119	....	55.5	....	55	32
F 2229	63	52	....	30	33	+30	263.8	99	....	53.1	....	40	24
FUEL: DENATURED ALCOHOL.[j] COMPRESSION PRESSURE: GREATEST, 180 POUNDS.[i]													
G 2360	74	93	207	25	24	+25	259.1	130	180	[k]*94.1*	460	129	68
H 2363	76	97	209	25	24	+25	262.1	131	160	*83.1*	400	111	58
I 2366	76	94	201	25	24	+25	263.2	132	130	*67.2*	295	85	45
J 2368	76	97	206	25	24	+25	263.8	125	85	*41.8*	....	40	21

[a] Measured by a thermometer placed in the inlet-valve housing (p. 57).
[b] Butterfly valve placed in the air pipe to the carburetor; set so as to throttle the air and supplement the governor-controlled inlet valve. Wide open at 80.
[c] Opening of the fuel needle-valve in hundredths of one revolution.
[d] Point of ignition, in degrees, is given in terms of the crank position: += early or before dead center.
[e] Horsepower constants: Brake 0.0004478, indicated=0.001396 (p. 67).
[f] Calculated from the indicated horsepower and the low heating value of the fuel (p. 72).
[g] Calculated from the brake horsepower and the low heating value of the fuel (p. 72).

OPERATOR'S NOTES (TABLE 29).

Test A.—The adjustment of the load, fuel needle valve, air valve, and time of ignition for this test was that for which the engine consumed the least amount of fuel per brake horsepower-hour. (Table 23).

Tests A to D inclusive.—Running of engine normal.

Test E.—Governor hunted a little, and after about every ten successive explosions cut-out one or two charges; that is, did not open the inlet valve.

Test F.—Owing to the action of the governor (Test E), explosions occurred singly and in series of two, with occasional misfires and cut-outs.

normal fuel economy versus load.

[Otto 15-Horsepower Alcohol Engine.—Cylinder diameter, 6¾ inches; stroke, 15½ inches; throttle governor; governor controls the lift and timing of the main inlet valve; inlet temperature of the jacket water, 60° to 80° F. For curves see fig. 27.]

FUEL: DENATURED ALCOHOL;[h] COMPRESSION PRESSURE, GREATEST, 140 POUNDS.[i]

Horsepower.[e]			Fuel consumption.							Efficiencies (per cent).		
				Per brake horsepower-hour.			Per indicated horse-power-hour.				Thermal.	
Brake.	Indicated.	Duration (minutes).	Pounds per hour.	Pounds.	Gallons.	B. t. u. (low).	Pounds.	Gallons.	B. t. u. (low).	Mechanical.	Indicated.[f]	Brake.[g]
15.06	17.3	14	12.48	0.829	0.121	8,750	0.72	0.105	7,600	87.0	33.5	29.1
13.44	15.8	16	11.61	.864	.126	9,120	.73	.107	7,740	84.8	32.9	27.9
11.69	14.2	26	10.45	.894	.130	9,440	.74	.107	7,760	82.2	32.8	27.0
8.27	10.8	21	8.51	1.029	.150	11,860	.79	.116	8,360	76.9	30.4	21.4
6.26	9.2	15	8.27	1.321	.193	13,950	.90	.131	9,470	67.9	26.9	18.2
4.73	7.3	15	7.83	1.656	.242	17,480	1.07	.156	11,300	64.6	22.5	14.6
FUEL: DENATURED ALCOHOL;[j] COMPRESSION PRESSURE: GREATEST, 180 POUNDS.[i]												
14.98	[k] *17.1*	28	10.62	0.709	0.104	7,550	[k] *0.62*	[k] *0.091*	[k] *6,610*	[k] *87.5*	[k] *38.5*	33.7
13.04	*15.2*	25	9.48	.727	.106	7,740	*.62*	*.091*	*6,610*	*85.5*	*38.5*	32.9
10.03	*12.4*	23	7.71	.768	.112	8,170	*.62*	*.091*	*6,640*	*81.2*	*38.3*	31.1
4.72	*7.3*	27	5.61	1.188	.174	12,640	*.77*	*.112*	*8,180*	*64.6*	*31.1*	20.1

[h] Specific gravity at 60° F.=0.821. Per cent alcohol by volume=93.8, by weight=90.8. Heating value: High=11,662, low=10,557 B. t. u. per pound.
[i] Pounds per square inch above atmosphere.
[j] Specific gravity at 60° F.=0.819. Per cent alcohol by volume=94.2, by weight=91.4. Heating value: High=11,750, low=10,643 B. t. u. per pound.
[k] Italicized results indicate that the involved mean effective pressure was calculated from the mechanical efficiency. Load curve previously determined for this engine. Indicator diagrams were taken, but not used (see operator's note G).

Test G.—The adjustment of the load, fuel needle valve, air valve, and time of ignition for this test was that for which the engine consumed the least amount of fuel per brake horsepower-hour. (Table 23). Running of the engine normal except in that with the change in the clearance volume that was made for this series of tests it was found that the engine slowed down appreciably when the indicator cock was opened for taking diagrams, and hence the mean effective pressure, if determined from the indicator diagrams, will not be correct.

Test H.—The governor cut-out occasionally; that is, occasionally the governor-controlled inlet valve was not opened.

TABLE 30.—*Results of trial and check tests made to*

[OTTO 15-HORSEPOWER GASOLINE ENGINE NO. 1.—Compression pressure, 70 pounds per square inch above atmosphere; diameter of cylinder, 6¾ inches; stroke, 15½ inches; speed, 253 to 263 revolutions per minute; hit-or-miss fuel-supply governor. Jacket-water temperature: Inlet, 60° F.; outlet, 112° F.]

FUEL: DENATURED ALCOHOL.[b]

Needle-valve settings.[c]	Fuel consumed per brake horsepower-hour (pounds).						
	Brake load, 140 pounds; B. H. P., 17.0 to 17.2.						
120	(d)						
86			1.632				
70					*1.335*		
66					(d)		
	Brake load, 130 pounds; B. H. P., 15.95.						
90				(d)	(d)	(d)	(d)
86					*1.513*	(d)	
76						*1.555*	(d)
	Brake load, 120 pounds; B. H. P., 14.8 to 15.1.						
100							(d)
98		*1.738*	(d)				
90							*1.705*
80							1.603
70							
	Brake load, 100 pounds; B. H. P., 12.2 to 12.6.						
120				1.850			
116				1.829			
90							
80							
78							
70							
Igniter settings.[e]	+30°	+25°	+21°	+17°	+13°	+9°	0°

[a] The figures in the body of the table give the fuel consumption in pounds per brake horsepower-hour and are tabulated with respect to the fuel needle valve and igniter settings used. The figures in italics are the results of check tests of about 30 minutes' duration. The other figures in the body of the table are the results of the trial tests of about 15 minutes' duration.

[b] Specific gravity at 60° F.=0.820. Per cent alcohol, by volume=92.7; by weight=91.1. **Heating value: High=11,605 B. t. u. per pound; low=10,498 B. t. u. per pound.**

determine the poorest fuel economy for each load.[a]

[OTTO 15-HORSEPOWER GASOLINE ENGINE NO. 1.—Compression pressure, 70 pounds per square inch above atmosphere; diameter of cylinder, $6\frac{3}{4}$ inches; stroke, $15\frac{1}{2}$ inches; speed, 253 to 263 revolutions per minute; hit-or-miss fuel-supply governor. Jacket-water temperature: Inlet, 60° F.; outlet, 112° F.]

FUEL: DENATURED ALCOHOL.[b]

Fuel consumed per brake horsepower-hour, pounds.						
Brake load, 140 pounds; B. H. P., 17.0 to 17.2.						
Brake load, 130 pounds; B. H. P., 15.95.						
Brake load, 120 pounds; B. H. P., 14.8 to 15.1.						
1.504						
Brake load, 100 pounds; B. H. P., 12.2 to 12.6.						
	(d)					
	1.805	*2.010*				
		2.030				
		1.802 1.765				
−6°	−15°	−19°	−24°	−29°	−33°	−38°

[c] Opening of needle valve in hundredths of one revolution.
[d] Trial tests were attempted with these needle-valve and igniter settings, but the engine would not carry the load.
[e] Point of ignition, in degrees, is given in terms of the crank position. (+=early, or before dead center; −=late, or after dead center.)

TABLE 30.—*Results of trial and check tests made to*

[OTTO 15-HORSEPOWER GASOLINE ENGINE NO. 1.—Compression pressure, 70 pounds per square inch above atmosphere; diameter of cylinder, 6¾ inches; stroke, 15½ inches; speed, 253 to 263 revolutions per minute; hit-or-miss fuel-supply governor. Jacket-water temperature: Inlet, 60° F.; outlet, 112° F.]

FUEL: DENATURED ALCOHOL—Continued.

Fuel needle-valve setting.	Fuel consumption per brake horsepower-hour (pounds).						
	Brake load, 80 pounds; B. H. P., 9.8 to 10.0.						
100							
96							
90							
82							
80							
26					1.895		
	Brake load, 60 pounds; B. H. P., 7.4 to 7.6.						
100							
96							
90							
82							
	Brake load, 40 pounds; B. H. P., 4.9 to 5.2.						
90							
80							
70							
60							
56							
Igniter settings.[b]	+30°	+25°	+21°	+17°	+13°	+9°	0°

[a] Trial tests were attempted with these needle-valve and igniter settings, but the engine would not carry the load.

determine the best fuel economy for each load—Continued.

[OTTO 15-HORSEPOWER GASOLINE ENGINE NO. 1.—Compression pressure, 70 pounds per square inch above atmosphere; diameter of cylinder, 6¾ inches; stroke, 15½ inches; speed, 253 to 263 revolutions per minute; hit-or-miss fuel-supply governor; jacket-water temperature: Inlet, 60° F.; outlet, 112° F.]

FUEL: DENATURED ALCOHOL—Continued.

Fuel needle-valve setting.	Fuel consumption per brake horsepower-hour, pounds.					
Brake load, 80 pounds; B. H. P., 9.8 to 10.0.						
	2.157	(a)				
		(a)				
		2.288	(a)			
			2.352			
				(a)		
Brake load, 60 pounds; B. H. P., 7.4 to 7.6.						
					(a)	(a)
					2.690	
					2.649	
					2.513	
Brake load, 40 pounds; B. H. P., 4.9 to 5.2.						
						3.061
						3.140
						3.336
						3.390
						3.510
−6°	−15°	−19°	−24°	−29°	−33°	−38°

b Point of ignition, in degrees, is given in terms of the crank position. (+ = early, or before dead center; − = late, or after dead center.)

TABLE 31.—*Results of tests of*

[OTTO 15-HORSEPOWER GASOLINE ENGINE No. 1.—Compression pressure, 70 pounds per square inch above atmosphere; diameter of cylinder, 6¾ inches; stroke, 15½ inches; hit-or-miss fuel-supply governor. Jacket-water temperature: Inlet, 60° F.; outlet, 112° F. For curves, see figs. 28 and 29.]

Test number.	Temperatures (° F.).			Index settings.		Speed (average per minute).			Average pressures (pounds per square inch).		Brake load.	
	Air.	Fuel.	Mixture.[a]	Fuel needle valve.[b]	Igniter (degrees).[c]	Revolutions.	Explosions.	Fuel cut-outs.	Mean effective.	Maximum (above atmosphere).	Pounds.	Per cent of maximum.
					FUEL: GASOLINE.[g]							
A 737	89	71	88	62	+21	258.1	126	3	103.0	335	130	100
B 746	85	72	83	64	0	257.7	124	5	99.2	270	120	92
C 755	83	77	90	64	−6	257.2	119	9	94.7	260	110	85
D 758	84	74	89	68	−6	258.3	107	22	98.3	220	100	77
E 761	87	83	91	70	−19	260.5	98	32	85.9	190	80	62
F 764	82	76	90	70	−24	260.9	81	49	85.4	190	60	46
G 767	78	69	89	72	−28	264.2	59	73	89.8	220	40	31
					FUEL: DENATURED ALCOHOL.[h]							
H 1086	74	72	64	70	+13	253.4	123	4	114.6	380	140	100
I 1282	86	86	.1	80	+13	254.7	119	8	112.0	315	130	94
I 1285	86	90	72	76	+9	254.7	123	5	107.2	290	130	94
J 1082	79	78	58	90	0	261.1	115	16	107.0	260	120	86
J 1018	83	81	64	98	+25	257.1	110	18	111.4	395	120	86
K 1065	71	77	57	80	−19	254.7	120	8	88.1	210	100	71
K 1076	76	77	58	78	−19	254.5	121	6	86.0	200	100	71
L 1033	91	87	64	90	−19	259.2	105	15	84.6	200	80	57
L 1036	90	88	66	82	−24	256.7	113	15	74.4	165	80	57
M 1043	80	85	65	90	−33	258.0	91	38	71.8	165	60	43
M 1046	81	85	65	96	−33	260.7	92	38	73.4	170	60	43
N 1053	65	72	54	90	−38	261.1	72	59	70.2	165	40	29

[a] Recorded by a thermometer placed in the inlet-valve housing (p. 57).
[b] Opening of the fuel needle valve in hundredths of one revolution.
[c] Point of ignition in degrees is given in terms of the crank position; +=early, or before dead center;− =late, or after dead center (p. 55).
[d] Brake horsepower constant=0.0004817. Indicated horsepower constant=0.0014 (p. 67).
[e] Calculated from the indicated horsepower and low heating value of the fuel (p. 72).

OPERATOR'S NOTES (TABLE 31).

Test A.—Maximum load that the engine would carry. The settings were easily determined by observation. Combustion of first fuel charge after cut-out produced an explosion pound.

Test B.—The two trial tests covered a range of settings from needle valve 72 and igniter +25°, with a corresponding fuel consumption of 1.28 pound per brake horsepower per hour, to the values given in the table. In test B 746 combustion of the first fuel charge after cut-out produced an explosion pound.

Test C.—The three trial tests covered a range of settings from needle valve 66 and igniter 0°, with a corresponding fuel consumption of 1.24 pound per brake horsepower per hour, to the values given in the table.

poorest fuel economy versus load.

[OTTO 15-HORSEPOWER GASOLINE ENGINE No. 1.—Compression pressure, 70 pounds per square inch above atmosphere; diameter of cylinder, 6¾ inches; stroke, 15½ inches; hit-or-miss fuel-supply governor. Jacket-water temperature: Inlet, 60° F.; outlet, 112° F. For curves, see figs. 28 and 29.]

FUEL: GASOLINE.[g]

Horsepower.[d]		Duration (minutes).	Fuel consumption.									Efficiencies (per cent).		
					Per brake horsepower-hour.			Per indicated horsepower-hour.				Thermal.	
Brake.	Indicated.		Pounds per hour.	Average per charge (pounds).	Pounds.	Gallons.	B. t. u. (low).	Pounds.	Gallons.	B. t. u. (low).	Mechanical.	Indicated.[e]	Brake.[f]
16.16	18.2	30	18.05	0.00239	1.117	0.183	21,350	0.99	0.163	19,020	89.1	13.4	11.9
14.90	17.2	34	17.65	.00237	1.185	.195	22,670	1.02	.167	19,590	86.5	13.0	11.2
13.63	15.8	32	17.14	.00240	1.258	.207	24,070	1.09	.179	20,730	85.2	12.3	10.6
12.44	14.8	29	16.41	.00256	1.320	.217	25,250	1.11	.182	21,250	84.2	12.0	10.1
10.04	11.8	32	14.88	.00253	1.483	.244	28,370	1.26	.207	24,110	85.1	10.6	9.0
7.54	9.7	33	12.63	.00260	1.675	.275	32,030	1.30	.213	24,850	77.6	10.2	7.9
5.09	7.4	28	9.58	.00271	1.882	.309	36,000	1.30	.213	24,810	69.0	10.3	7.1

FUEL: DENATURED ALCOHOL.[h]

17.09	19.8	25	22.80	0.00309	1.335	0.195	13,950	1.16	0.169	12,100	86.5	21.1	18.2
15.95	18.7	30	24.14	.00338	1.513	.221	15,940	1.29	.188	13,600	85.3	18.7	16.0
15.95	18.4	30	24.80	.00336	1.555	.227	16,380	1.35	.197	14,200	86.7	17.9	15.5
15.09	17.2	28	25.72	.00373	1.705	.249	17,810	1.50	.219	15,600	87.7	16.3	14.3
14.86	17.2	33	25.85	.00392	1.738	.254	18,240	1.50	.219	15,800	86.5	16.1	13.9
12.26	14.7	31	24.64	.00342	2.010	.293	21,000	1.67	.244	17,500	83.2	14.6	12.1
12.26	14.6	35	24.90	.00343	2.030	.296	21,200	1.71	.250	17,900	84.2	14.2	12.0
9.99	12.4	32	22.86	.00363	2.288	.334	24,000	1.84	.269	19,300	80.4	13.2	10.6
9.87	11.8	31	23.28	.00343	2.352	.343	24,700	1.98	.289	20,800	84.0	12.3	10.3
7.46	9.2	35	19.75	.00362	2.649	.387	27,800	2.16	.315	22,600	81.5	11.2	9.2
7.54	9.5	30	20.29	.00367	2.690	.393	28,230	2.15	.314	22,500	79.7	11.3	9.0
5.03	7.1	29	15.40	.00356	3.061	.447	32,120	2.17	.317	22,800	71.0	11.1	7.9

[f] Calculated from the brake horsepower and low heating value of the fuel (p. 72).
[g] Specific gravity at 60° F.=0.729. Heating value: High=20,401 B. t. u. per pound; low=19,120 B. t. u. per pound.
[h] Specific gravity at 60° F.=0.820. Per cent alcohol: By volume=92.7, by weight=91.1. Heating value: High=11,605 B. t. u. per pound; low=10,498 B. t. u. per pound.

Test D.—For a trial test with the same needle valve and igniter settings, a fuel consumption value of 1.40 pound per brake horsepower per hour was obtained. In test D 758 combustion of the first fuel charge after cut-out produced an explosion pound.

Test E.—Required needle valve and igniter settings were determined by observation. Combustion of the first fuel charge after cut-out produced an explosion pound.

Test F.—Required needle valve and igniter settings were determined by observation. Occasional explosion in muffler. No misfired fuel charges.

Test G.—Required needle valve and igniter settings were determined by observation. Every other first fuel charge after cut-out misfired. Mixture too weak to burn. Ignition not at fault.

Test H.—The two trial tests covered a range of settings from needle valve 120 and igniter +30° to needle valve 66 and igniter +13°, with corresponding fuel-consumption figures from 1.63 pound per brake horsepower per hour to those given in the table, the first of which were rejected on account of the unstable conditions of operation. Undersettings judged by inspection. The engine would not carry the load with needle valve set at 80 and igniter at +17°.

Test I.—The preliminary determination of settings was made by previous observations.

Test J.—The four trial tests ranged in settings from needle valve 70 and igniter −6° to needle-valve 98 and igniter +25° with corresponding fuel-consumption figures from 1.50 pound of fuel per brake horsepower per hour to those given in the table. Settings judged by inspection to a great extent. Engine would not carry load with needle valve set at 100 and igniter at 0°. In test J 1018, engine labored at times.

Test K.—The seven trial tests ranged in settings and fuel consumption from needle valve 120 and igniter +17°, with a fuel consumption of 1.8 pound of fuel per brake horsepower per hour, to the figures given in the table. In test K 1065, the brake load had to be slightly reduced for two short periods. During test K 1076, the engine labored slightly.

Test L.—The four trial tests ranged in settings and fuel consumption from needle valve 26 and igniter +13° and a consumption of 1.9 pound of fuel per brake horsepower per hour to the figures given in the table. Once during test L 1033, the engine labored. During test L 1036, the engine labored at times.

Test M.—A trial test with the same igniter setting and a needle-valve setting of 82 was made. The resulting fuel consumption was 2.31 pounds per brake horsepower per hour. Adjustments were determined principally by inspection. During test M 1043, engine back-fired once.

Test N.—The five trial tests covered a range of settings of from needle valve 56 and igniter −38° to needle valve 80 and igniter −38°, with fuel-consumption figures ranging from 3.51 pounds of fuel per brake horsepower per hour to those given in the table. All others were rejected on account of frequent misfiring and an unstable condition of operation.

CONCLUSIONS.

The effect of load upon the best fuel economy obtainable by manipulation of engine adjustments can best be studied by reference to the curves shown in figure 24. The following table shows the engine, the method of governing, the fuel, the compression pressure, the table number, and the series of tests for each curve:

Data for curves in figure 24.

Curve No.	Engine.	Method of governing.	Fuel.	Compression pressure.[a]	Table No.	Series of tests.
1	15-horsepower Otto gasoline No. 2.	Hit - or - miss fuel supply.	Denatured alcohol.	Constant at 70...	18	Second.
2	15-horsepower Otto gasoline No. 1.	.....do........	.....do........	.....do..........	16	Do.
3	10-horsepower Nash gasoline.	Hit - or - miss mixture supply.	.....do........	Varies; 72 at best load.	20	Fourth.
4	15-horsepower Otto alcohol..	Combination of throttle and cut-off.	.....do........	Varies; 180 at best load.	24	Second.
5	10-horsepower Nash gasoline.	Hit - or - miss mixture supply.	Gasoline......	Varies; 72 at best load.	20	Do.
6	15-horsepower Otto gasoline No. 2.	Hit - or - miss fuel supply.	.....do........	Constant at 70...	18	First.
7	15-horsepower Otto gasoline No. 1.	.....do........	.....do........	.....do..........	16	Do.

[a] Pounds per square inch above 1 atmosphere.

Here it is seen that the characteristics of all the economy load curves are similar regardless of the type of engine, the method of governing, the fuel used, or whether the air supply is preheated. In every case the fuel consumed per brake horsepower-hour falls off from the maximum load down to about 85 per cent of the maximum load, where the best economy is obtained. From this point the fuel consumption shows gradually increasing increments as the load is reduced.

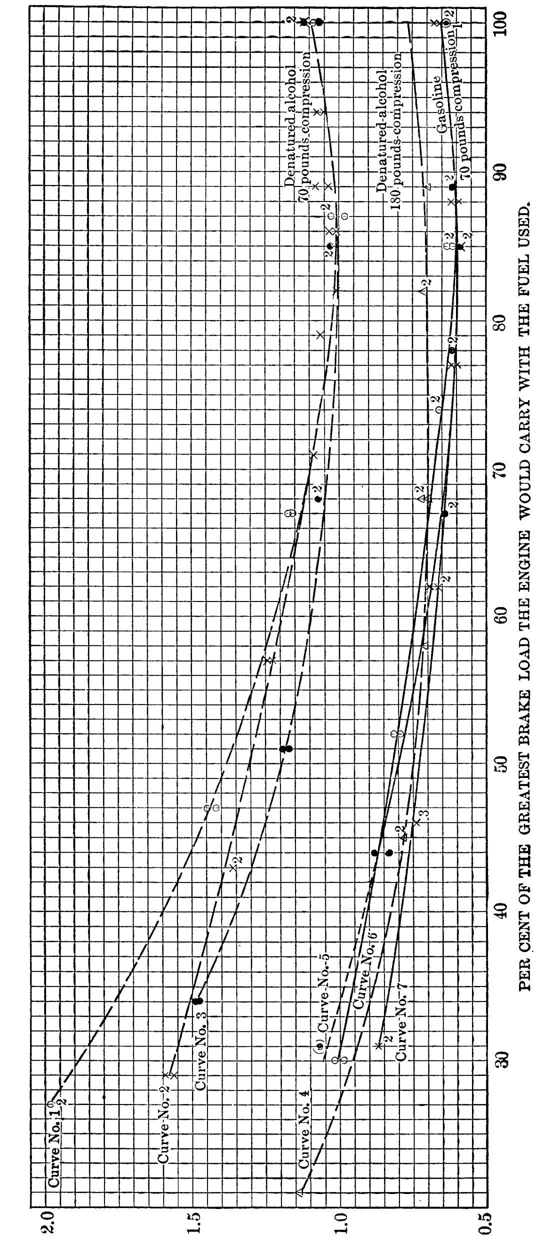

FIGURE 24.—Curves showing relation between load and least fuel consumption.

Best results with gasoline as fuel are shown by curves 5, 6, and 7 (fig. 25), representing economies obtained with the Nash gasoline engine, Otto gasoline engine No. 2, and Otto gasoline engine No. 1, respectively. The smallest fuel consumption per brake horsepower-hour for each of these engines is approximately 0.6 pound. For loads below that giving the best economy the engines showed a slight variation in fuel consumption, but for loads above that giving the best economy the curves are superimposed.

Best results with denatured alcohol as fuel are shown by curves 1, 2, 3, and 4, representing economies obtained with Otto gasoline engine No. 1, Nash gasoline engine, and Otto alcohol engine, respectively. Eliminating curve No. 4, a striking similarity is to be seen between the group of denatured alcohol and the group of gasoline curves, in both of which 70-pound compression is represented. The lowest alcohol consumption per brake horsepower-hour for each of the gasoline engines is approximately 1 pound. The variation in fuel consumption for loads below that giving the best economy show differences while for loads above the best the curves superimpose.

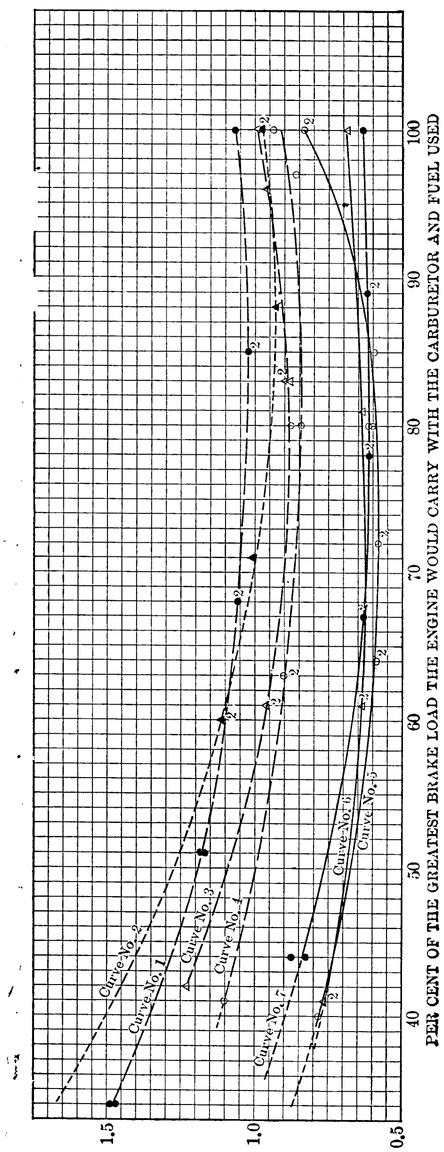

FIGURE 25.—Curves showing relation between load and least fuel consumption.

Curve No. 4 when compared with Nos. 1, 2, and 3 affords a striking example of the gain in efficiency which may be effected because of the greater compression to which it is possible to subject denatured alcohol. In this case, with a compression pressure of 180 pounds per square inch a best economy of 0.7 pound of fuel per brake horsepower is obtained as against 1 pound with the 70-pound compression.

Figure 25 shows a comparison between the best results obtained with the Nash gasoline engine and the Nash alcohol engine when using both gasoline and denatured alcohol in the regular carburetor, and also in the double-cone carburetor. Although these curves have the same general contour as the ones preceding, they do not show as consistent results. It will be recalled that in the description of the two engines attention was called to the lack of stability with which they governed, and this probably explains the failure of the results to check up to a greater refinement.

Attention was called to curve No. 4, figure 24, because of the unusual efficiency shown, and the explanation was offered that the high compression pressure was the cause. Unfortunately this was the only curve in that group showing results from the Otto alcohol engine. The following table shows the engine, method of governing, fuel, greatest compression pressure, table number, and series of tests for each curve:

Data for curves in figure 25.

Curve No.	Engine.	Method of governing.	Fuel.	Compression pressure.[a]	Table No.	Series of tests.
1	10-horsepower Nash gasoline.	Hit-or-miss mixture supply.	Denatured alcohol.	86	20	Fourth.[b]
2	10-horsepower Nash alcohol..	.....do........	.....do........	133	22	Third.[b]
3	.....do........	.....do........	.....do........	135	22	Second.[c]
4	10-horsepower Nash gasoline.	.....do........	.....do........	98	20	Third.[c]
5	.....do........	.....do........	Gasoline......	98	20	First.[c]
6	10-horsepower Nash alcohol..	.....do........	Gasoline and water.[d]	133	22	Do.[b]
7	10-horsepower Nash gasoline.	.....do........	Gasoline......	86	20	Second.[b]

[a] Greatest in pounds per square inch above atmosphere. Compression pressure varies with the engine, carburetor, and air-valve setting used. See p. 9.
[b] Regular carburetor. See p. 34.
[c] Double-cone carburetor. See p. 34.
[d] A spray of water is used to prevent preignition from high compression.

In order to show comparison between results obtained on one engine using different compressions, figure 26 was prepared. In this figure curve No. 2 is identical with curve No. 4, figure 24. In addition, however, are given two curves, No. 1 and No. 3, with compression pressures of 140 pounds and 210 pounds to 130 pounds, respectively. The curves with compression pressures of 140 pounds and 180 pounds are exactly similar, the latter showing uniformly more economical results. However, with the highest compression (210 pounds) better results than with a 180-pound compression were shown only in that small portion of the curve between 85 and 100 per cent of the maximum load.

The "normal economy load" curves shown in figure 27 are so similar in character that they might almost be considered as a group of parallel curves. They divide themselves into three groups. The following table shows the engine, method of governing, fuel, compression pressure, table number, and series of tests for each curve:

Data for curves in figure 27.

Curve No.	Engine.	Method of governing.	Fuel.	Compression pressure.[a]	Table No.	Series of tests.
1	15-horsepower Otto gasoline No. 2.	Hit-or-miss fuel supply.	Denatured alcohol.	70	26	Second.
2	15-horsepower Otto gasoline No. 1.	.....do........	.....do........	70	25	Do.
3	10-horsepower Nash gasoline...	Hit-or-miss mixture supply.	.....do........	78	27	Third.[b]

[a] Pounds per square inch above atmosphere. [b] Regular carburetor. See p. 34.

Data for curves in figure 27—Continued.

Curve No.	Engine.	Method of governing.	Fuel.	Compression pressure.	Table No.	Series of tests.
4	15-horsepower Otto gasoline No. 1.	Hit-or-miss fuel supply.	Denatured alcohol.	70	25	Third.[a]
5	10-horsepower Nash alcohol...	Hit-or-miss mixture supply.	.....do.........	130	28	Second.[b]
6	.....do.........	.....do.........	.....do.........	133	28	Third.[b]
7	10-horsepower Nash gasoline...	.....do.........	.....do.........	92	27	Second.[b]
8	15-horsepower Otto alcohol....	Combination of throttle and cut-off.	.....do.........	140 to 90	29	First.
9	10-horsepower Nash alcohol....	Hit-or-miss mixture supply.	Gasoline with water [c]......	130	28	Do.[d]
10	10-horsepower Nash gasoline...	.....do.........	Gasoline......	72	27	Do.[d]
11	15-horsepower Otto gasoline No. 2.	Hit-or-miss fuel supply.	.....do.........	70	26	Do.
12	15-horsepower Otto gasoline No. 1.	.....do.........	.....do.........	70	25	Do.
13	15-horsepower Otto alcohol....	Combination of throttle and cut-off.	Denatured alcohol.	180 to 85	29	Second.

[a] Air preheated to 250° F.
[b] Double-cone carburetor. See p. 34.
[c] A spray of water is used to prevent preignition from high compression.
[d] Regular carburetor. See p. 34.

The first group, consisting of curves 1, 2, 3, 4, and 5, all showing denatured-alcohol consumption, lie so near each other that their only difference may be explained by the possible error in determination and calculation of results. The second group, curves 6, 7, and 8, show somewhat better denatured-alcohol efficiencies, but the discrepancies are easily explained by special features involved in each series of tests, such as the double-cone carburetor in each of the Nash engines or the high-compression pressure in the Otto alcohol engine. Curves 9, 10, 11, 12, and 13 constitute the third group, made up of results from all normal economy load tests with gasoline as fuel.

It is interesting to note the relation between the best, normal, and poorest economy for any one engine. Curves 5, 4, and 1, respectively, figure 28, show this relation for the Otto gasoline engine No. 1, using gasoline for fuel. For purposes of comparison, curves 3 and 2 show the best and normal economies of Otto gasoline engine No. 2 with gasoline. The following table shows the engine, method of governing, fuel, compression pressure, table number, and series of tests for each curve:

Data for curves in figure 28.

Curve No.	Engine.	Method of governing.	Fuel.	Compression pressure.[a]	Table No.	Series of tests.
1	15-horsepower Otto gasoline No. 1.	Hit-or-miss fuel supply.	Gasoline......	Constant at 70 pounds.	31	First.
2	.....do.........	.....do.........	.....do.........	.....do.........	25	Do.
3	15-horsepower Otto gasoline No. 2.	.....do.........	.....do.........	.....do.........	26	Do.
4	15-horsepower Otto gasoline No. 1.	.....do.........	.....do.........	.....do.........	16	Do.
5	15-horsepower Otto gasoline No. 2.	.....do.........	.....do.........	.....do.........	18	Do.
6	15-horsepower Otto gasoline Nos. 1 and 2.	.....do.........	.....do.........	.....do.........	......	

[a] Pounds per square inch above atmosphere. Compression pressure varies with the engine and air-valve setting used. See p. 9.

Figure 29 shows the same relation with denatured alcohol for fuel that figure 28 does with gasoline as the fuel. In curve No. 1 (fig. 29) the weight of denatured alcohol consumed is the greatest that could be obtained by special adjustment of the time of ignition and quality of the mixture for each load, without causing some outward indication of its uneconomical use, according to the method of the second series of "poorest fuel economy versus load" tests, Table 31. In curves Nos. 2 and 3 the weight of denatured alcohol consumed is that obtained when the adjustments controlling the time of ignition and mixture quality are constant at those for the greatest brake load applied and the action of the governor is allowed to vary in the normal way, according to the method of the second series of "normal fuel economy versus load" tests, Tables 25 and 26. In curves Nos. 4 and 5 the weight of denatured alcohol consumed is the least that could be obtained by special adjustment of the time of ignition and quality of the mixture for each load, as shown in the second series of "best fuel economy versus load" tests, Tables 16 and 18. Curve No. 6 shows the variation of the least weight of denatured alcohol consumed for each load that probably would be obtained if the disturbing effects of the hit-or-miss method of governing employed were entirely eliminated,

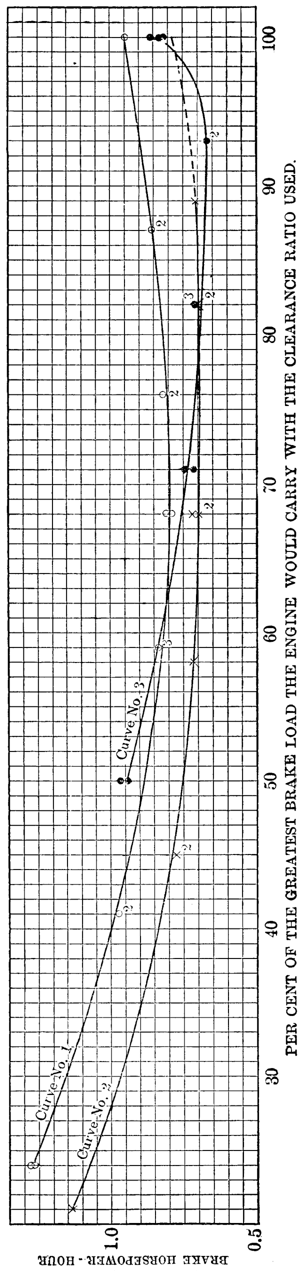

FIGURE 26.—Curves showing relation between load and least fuel consumption, Otto alcohol engine. Curve No. 1: Compression pressure constant at 140 pounds for load from greatest to 68 per cent of greatest, and varied from 140 to 60 pounds for loads from 68 to 24 per cent of the greatest. Least consumption of fuel per brake horsepower-hour was obtained with compression of 140 pounds. Curve No. 2: Compression pressure constant at 180 pounds for loads from greatest to 68 per cent of greatest, and varied from 180 to 85 pounds for loads from 68 to 21 per cent of greatest. Least consumption of fuel per brake horsepower-hour was obtained with a compression of 180 pounds. Curve No. 3: Compression pressure varied from 210 to 130 pounds. For compressions over 200 pounds mixtures of any quality between the limits of combustibility preignited; for compressions between 190 and 200 pounds all except those approaching the limit of dilution with air preignited; hence the greatest load is limited to that for which the action of the governor is sufficient to reduce the compression to a pressure below that which causes preignitions. The least consumption of fuel per brake horsepower-hour was obtained with a compression of 190 to 200 pounds.

that is, if the increase of fuel consumed per brake horsepower-hour with decrease in load below about 80 per cent of the greatest was due only to the decrease in the mechanical efficiency of the engine. The following table shows the engine, method of governing, fuel, compression pressure, table number, and series of tests, for each curve:

Data for curves in figure 29.

Curve No.	Engine.	Method of governing.	Fuel.	Compression pressure.[a]	Table No.	Series of tests.
1	15-horsepower Otto gasoline No. 1.	Hit-or-miss fuel supply.	Denatured alcohol.	Constant at 70 pounds.	31	Second.
2	.....do	.....do	.....do	.....do	25	Do.
3	15-horsepower Otto gasoline No. 2.	.....do	.....do	.....do	26	Do.
4	15-horsepower Otto gasoline No. 1.	.....do	.....do	.....do	16	Do.
5	15-horsepower Otto gasoline No. 2.	.....do	.....do	.....do	18	Do.
6	15-horsepower Otto gasoline Nos. 1 and 2.	.....do	.....do	.....do	......	

[a] Pounds per square inch above atmosphere. Compression pressure varies with the engine and air-valve setting used. See p. 9.

The efficiency of preheating the air supply is illustrated by figure 30. The following table shows the engine, method of governing, fuel, compression pressure, table number, and series of tests for each curve:

Data for curves in figure 30.

Curve No.	Engine.	Method of governing.	Fuel.	Compression pressure.[a]	Table No.	Series of tests.
1	15-horsepower Otto gasoline Nos. 1 and 2.	Hit-or-miss fuel supply.	Gasoline or denatured alcohol.	Constant at 70 pounds.	......	
2	.....do	.....do	.....do	.....do	......	

[a] Pounds per square inch above atmosphere. Compression pressure varies with the engine and air-valve setting used. See p. 9.

Curves 1 and 2 show the relation between unheated air supply and preheated air supply with denatured alcohol.

PREHEATING AIR SUPPLY.

It was demonstrated that the heating of the air supply to 250° F. reduced the maximum load the engine would carry to approximately 85 per cent of that which the engine would carry without the air preheated. The relative fuel consumptions showed an advantage with the preheated air at low loads only, probably because there is more opportunity under such conditions for an improvement in carburization. It follows, then, that when an engine is being used at a capacity very much below that at which it is rated, some advantage

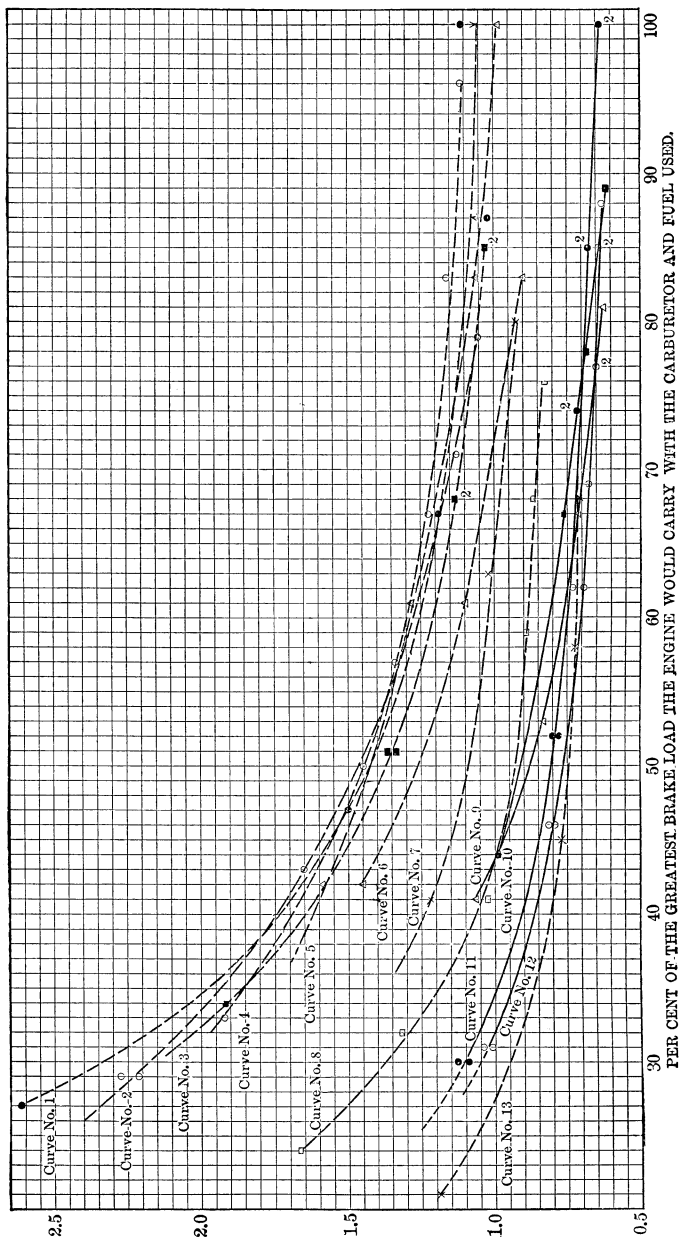

FIGURE 27.—Curves showing relation between load and normal fuel consumption.

might be gained in preheating the air supply; otherwise the limited capacity of the engine resulting from use of preheated air supply would operate against such use.

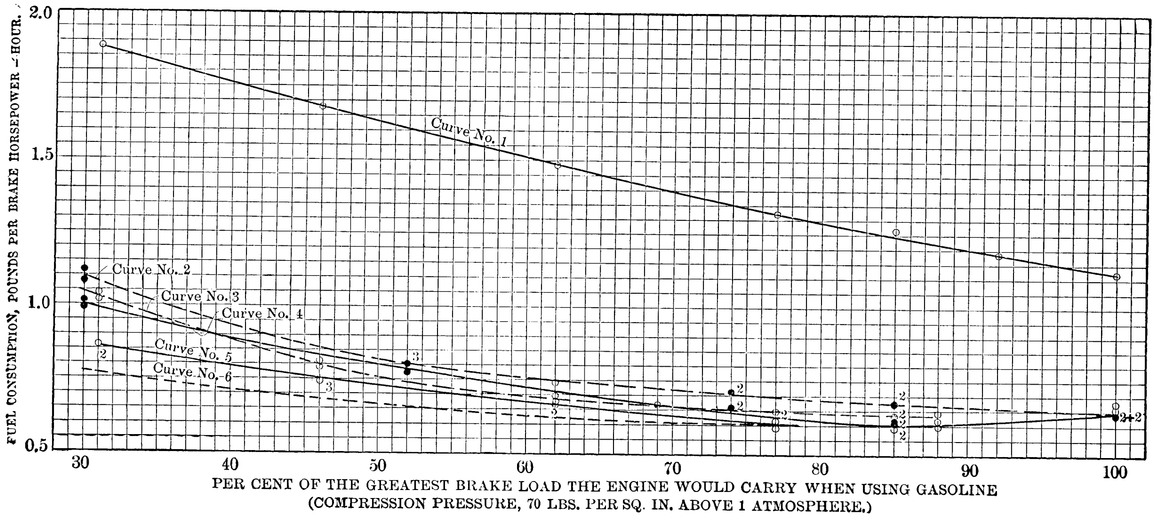

FIGURE 28.—Curves showing relation between load and least, normal, and greatest fuel consumption of gasoline. Curve No. 1: The weight of gasoline consumed is the greatest that could be obtained by special adjustment of the time of ignition and quality of the mixture for each load, without causing some outward indication of its uneconomical use, according to the method of the first series of "poorest fuel economy versus load" tests, Table 31. Curves Nos. 2 and 3: The weight of gasoline consumed is that obtained when the adjustments controlling the time of ignition and mixture quality are constant and are those for the greatest brake load applied, and when the action of the governor is allowed to vary in the normal way according to the method of the first series of "normal fuel economy versus load" tests, Tables 25 and 26. Curves Nos. 4 and 6: The weight of gasoline consumed is the least that could be obtained by special adjustment of the time of ignition and quality of the mixture for each load, according to the method of the first series of "best fuel economy versus load" tests, Tables 16 and 18. Curve No. 6 shows the variation of the least weight of gasoline consumed for each load that probably would be obtained if the disturbing effects of the hit-or-miss method of governing employed were entirely eliminated; that is, if the increase in the weight of fuel consumed per brake horsepower-hour with decrease in load below about 80 per cent of the greatest was due only to the decrease in the mechanical efficiency of the engine.

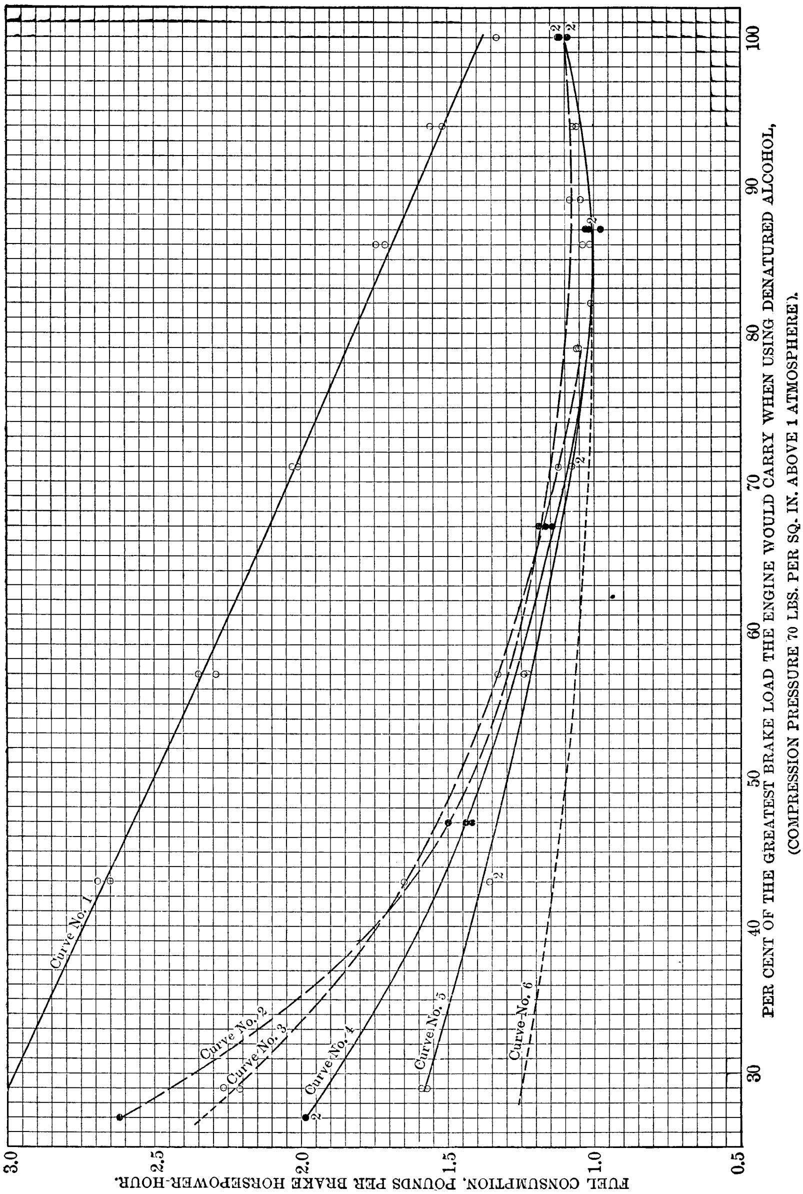

FIGURE 29.—Curves showing relation between load and least, normal, and greatest fuel consumption of denatured alcohol.

MIXTURES OF GASOLINE WITH DENATURED ALCOHOL, WATER WITH GASOLINE, AND WATER WITH DENATURED ALCOHOL.

One of the greatest objections which has been raised to the use of denatured alcohol as a fuel for internal-combustion engines is the difficulty ordinarily experienced in starting the engines because of the comparatively high vaporization point of the alcohol. On the other hand, the efficiency with which gasoline can be used is limited by the comparatively low compression with which it must be used. The combustion of gasoline also takes place more quickly than that of denatured alcohol and tends to cause a more rapid rise of pressures in the cylinder, and hence a less smooth running.

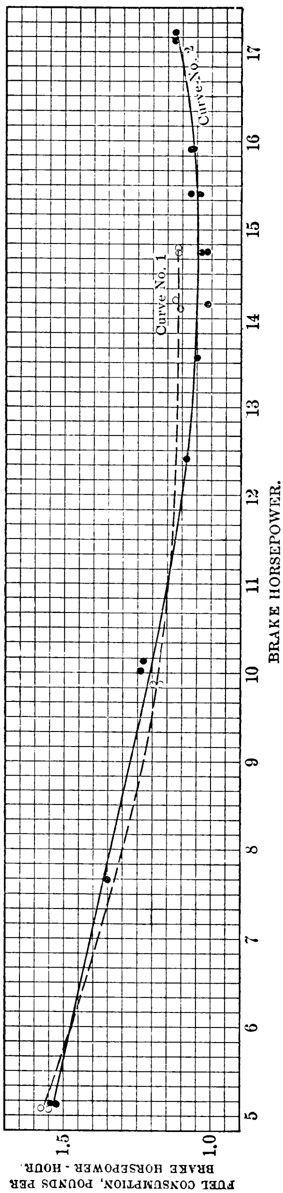

FIGURE 30.—Curves showing relation between load and least fuel consumption, air unheated and air preheated.

EFFECT ON EFFICIENCY.

Because of what might be termed the supplementing characteristics of the two fuels under consideration, it was decided to carry out a series of tests in which the two fuels were used together. In this way it was hoped to obtain the advantage of the best features of both fuels, namely, the ease in starting with gasoline and the smooth running conditions and higher efficiency of denatured alcohol.

After such a series of tests had been started, it seemed advisable, in order to have complete information on the use of diluents, to determine the effect of a dilution with water upon gasoline and denatured alcohol. It was hoped that water, when used with gasoline, would enable the use of higher compression and stop troublesome back-firing. Denatured alcohol is so easily miscible with water that the determination of a possible gain in efficiency effected by the use of the mixture was considered desirable.

The hypothetical effects that have been assumed in this discussion have all been considered to be advantageous ones. Lest this assumption be misleading, attention should be called to the deleterious effects that may be expected with the mixture of the two fuels or the introduction of water into either of them. When denatured alcohol is added to gasoline, the heating value of the mixture is decreased, but at the same time the quantity of air necessary for combustion is decreased, with the result that the capacity or available horsepower of the engine is not materially affected. However, when water is added to either of the fuels individually, a reduction in the high heating value, in exact proportion to the amount of water added, takes place, and with large percentages of water the capacity of the engine will undoubtedly be affected.

One important feature in connection with the use of water as a diluent is that all of the water which goes into the cylinder in the form of a vapor will emerge as steam and hence carry latent heat away with it. Whether this action will result in a definite loss, or perhaps only in the absorbing of some heat that would otherwise appear in the jacket and exhaust losses, is impossible to state.

METHODS OF TESTING.

Gasoline and denatured alcohol and gasoline and water are not miscible, and on tests involving the use of either of these combinations a special carburetor using two separate reservoirs and spray nozzles—one for each liquid—was used. In this way a complete mixture of the vapors of both liquids was always obtained and determination of the exact proportion of each by weight was possible.

The thermal efficiency tests of varying percentages of denatured alcohol and gasoline were the first undertaken. The proportion of gasoline to denatured alcohol was varied by changing the relative openings of the needle valves that controlled the quantity of each liquid supplied to its spray nozzle in the carburetor. For each proportion systematic trial tests were made, changing the load and all engine adjustments until the lowest fuel economies obtainable were reached. Unfortunately it was not feasible to increase the compression as the percentage of alcohol was increased. It is felt that if this could have been done in such a manner as to gain the advantage of the highest compression possible with each mixture, considerably better results might have been obtained.

RESULTS OBTAINED.

GASOLINE-ALCOHOL TESTS.

Two series of gasoline-alcohol (denatured) tests were run, the first being on the Nash gasoline engine and including mixtures ranging all the way from 0 to 100 per cent gasoline. The second series was run on the Nash alcohol engine and included mixtures ranging from 0 to 30.6 per cent gasoline. The results of trial and check tests from these series may be found in Table 32, section A, and the table of results of check tests on the Nash gasoline engine appear in Table 33, section A.

TABLE 32.—*Results of trial and check tests made to determine the least consumption of various proportions of gasoline and denatured alcohol, of gasoline and water, and of denatured alcohol and water.*

SECTION A.—FUEL: GASOLINE AND DENATURED ALCOHOL.[a]

[Gasoline and denatured alcohol supplied through separate spray nozzles in the regular carburetors of the Nash engines.[b]]

Brake load, in pounds.	Air-valve setting.[d]	Compression pressure (pounds)[e]	Gasoline needle-valve setting.[f]	Alcohol needle-valve setting.[f]	Igniter setting.[g]	Brake horse-power.	Per cent gasoline, by weight.	Pounds of gasoline and denatured alcohol consumed per brake horse-power-hour.	British thermal units per brake horse-power-hour (low).
NASH 10-HORSEPOWER GASOLINE ENGINE.[c]									
100	1/2	72	Closed.	3 1/2	2 1/2	11.62	0	1.170	12,500
100	5/8	78	Closed.	3 1/2	2	11.82	0	1.078	11,520
100	5/8	78	Closed.	3 7/16	2	11.67	0	1.034	11,040
100	3/4	81	Closed.	3 13/16	1 3/4	11.67	0	1.072	11,450
100	5/8	78	Closed.	8 1/8	1 3/4	11.67	0	.980	10,460
100	5/8	78	4 1/4	3 1/4	1 3/4	11.75	4.5	.983	10,750
100	5/8	78	5	2 5/16	2 1/4	11.70	36.6	.840	11,560
100	5/8	78	5 1/2	2	2 3/4	11.74	50.9	.832	12,480
100	5/8	78	5 3/4	1 1/2	4	11.69	70.8	.746	12,480
96	1/2	72	4 1/2	2 5/8	2 1/2	11.32	21.1	.967	11,980
96	5/8	78	4 9/16	2 3/4	1 1/2	11.17	21.5	.955	11,870
96	5/8	78	4 5/8	2 7/8	1 1/2	11.30	22.8	.994	12,400
96	1/2	72	4 1/2	2 5/8	2 3/4	11.41	22.7	1.000	12,540
96	5/8	78	4 11/16	3	2 1/2	11.29	23.2	1.028	12,940
96	5/8	78	4 21/32	2 15/16	1 3/4	11.24	23.7	.902	11,430
96	5/8	78	4 9/16	2 5/8	1 1/2	11.24	24.3	.834	10,900
96	5/8	78	4 21/32	2 15/16	1 3/4	11.35	25.7	1.014	12,980
96	5/8	78	4 9/16	2 1/2	1 1/4	11.25	25.9	.817	10,470
96	3/4	81	4 25/32	2 15/16	1 3/4	11.28	25.9	.964	12,350
96	3/4	81	4 3/4	2 3/4	1 1/2	11.20	27.7	.961	11,880
91	3/4	81	5 1/16	2 1/16	1/4	10.70	45.3	.765	11,160
91	5/8	78	5 1/32	2	1 1/2	10.70	47.3	.776	11,450
91	5/8	78	1 11/16	5 3/8	0	10.65	49.1	.727	10,850
91	5/8	78	5	1 7/8	1	10.70	49.5	.740	11,020
91	5/8	78	5	1 7/8	1	10.70	50.3	.743	11,160
91	3/4	81	5	1 15/16	9 1/4	10.54	51.9	.735	11,140
87	5/8	78	5 3/16	1 7/16	1/4	10.28	70.4	.631	10,550
87	5/8	78	5 3/16	1 3/8	1/2	10.15	72.0	.630	10,610
87	5/8	78	5 7/16	1 3/16	1 1/4	10.09	82.8	.587	10,510
80	1/2	72	5 3/8	Closed.	3	9.36	100.0	.608	11,730
80	3/4	81	5 17/32	Closed.	3	9.33	100.0	.600	11,560
NASH 10-HORSEPOWER ALCOHOL ENGINE.[h]									
110	5/8	120	Closed.	8 3/8	6	12.58	0	1.010	10,660
110	3/4	126	Closed.	9 3/8	6 1/4	13.05	0	1.011	10,675
110	3/4	126	Closed.	9 3/8	6 1/4	12.98	0	1.005	10,620
110	3/4	126	Closed.	10	6 1/2	13.13	0	.995	10,510
110	7/8	130	Closed.	9 1/4	5	12.92	0	.934	9,860
110	7/8	130	Closed.	9 1/2	5	13.09	0	.944	9,960
110	7/8	130	Closed.	10 1/8	6 1/4	13.02	0	1.002	10,580
110	7/8	130	Closed.	10 3/8	6 1/4	13.08	0	1.028	10,860
110	3/4	126	1 7/16	6 3/4	1 3/4	13.15	27.6	.840	10,940
110	1/2	113	1 3/8	6 1/4	3	13.14	28.4	.885	11,500
110	3/4	126	1 5/8	6 3/4	2 3/4	13.10	30.0	.831	10,950
110	3/4	126	1 5/8	6 3/4	2 3/4	13.14	31.0	.811	10,760
110	5/8	120	1 1/2	6 1/2	3 1/2	13.14	31.6	.830	10,930
110	7/8	130	1 3/4	6 7/8	2 1/2	13.06	32.3	.827	11,070
110	5/8	120	1 13/16	5 7/16	3 1/8	13.14	50.2	.772	11,540
110	7/8	130	2	5 7/8	2	13.08	50.5	.738	11,050
110	3/4	126	1 15/16	5 5/8	3	13.12	52.7	.744	11,280
100	5/8	120	1 3/8	6 3/8	2 1/4	12.01	26.2	.873	11,200
100	3/4	126	1 1/2	6 5/8	1 1/2	11.91	26.7	.862	11,210
100	3/4	126	1 1/2	6 5/8	1 3/4	11.84	27.3	.840	10,870
100	1/2	113	1 3/8	6 1/8	2 1/2	11.99	30.6	.852	11,280

[a] For tests on the gasoline engine: Specific gravity of the gasoline at 60° F. was 0.7122. Heating value: High, 20,581; low, 19,292 B. t. u. per pound. Specific gravity of the denatured alcohol at 60° F. is 0.8188. Heating value: High, 11,788; low, 10,681 B. t. u. per pound. Per cent alcohol by weight, 94.3. For the tests on the alcohol engine the specific gravity of the gasoline was the same as above. The specific gravity of the denatured alcohol at 60° F. was 0.8206. Heating value: High, 11,662; low, 10,557 B. t. u. per pound. Per cent alcohol by weight, 93.8.

[b] For description of the regular Nash carburetors see p. 34.

[c] Speed, 290 to 300 revolutions per minute. Horsepower constants: Brake, 0.000392; indicated, 0.001153 (p. 67.)

[d] Plug cock with dial for regulating the auxiliary air supply which is introduced between the governor-controlled mixture valve and the main inlet valve.

[e] Pounds per square inch above 1 atmosphere.

[f] Scale arbitrary. Numbers increase with the opening of the valve.

[g] Scale arbitrary. Ignition is always early or before dead center.

[h] Speed, 290 to 300 revolutions per minute. Horsepower constants: Brake, 0.000395; indicated, 0.001153 (p. 67).

TABLE 32.—*Results of trial and check tests made to determine the least consumption of various proportions of gasoline and denatured alcohol, of gasoline and water, and of denatured alcohol and water*—Continued.

SECTION B.—FUEL: GASOLINE AND WATER.[a]

[Gasoline and water supplied through separate spray nozzles in the regular carburetor.[b]]

NASH 10-HORSEPOWER GASOLINE ENGINE.

Brake load, in pounds.	Air-valve setting.	Compression pressure (pounds).	Gasoline needle-valve setting.	Water needle-valve setting.	Igniter setting.	Brake horsepower.	Per cent gasoline, by weight.	Pounds of gasoline consumed per brake horsepower-hour.	British thermal units per brake horsepower-hour (low).[c]
80	$\frac{3}{8}$	67	$1\frac{15}{16}$	Closed.	$3\frac{1}{4}$	9.39	100.00	0.667	12,830
80	$\frac{1}{2}$	72	$1\frac{15}{16}$	Closed.	$6\frac{1}{2}$	9.46	100.00	.612	11,760
80	$\frac{1}{2}$	72	$1\frac{15}{16}$	Closed.	$2\frac{1}{2}$	9.36	100.00	.675	12,980
80	$\frac{1}{2}$	72	$1\frac{15}{16}$	Closed.	$3\frac{1}{2}$	9.31	100.00	.655	12,600
80	$\frac{5}{8}$	78	$2\frac{1}{16}$	Closed.	....	9.27	100.00	.628	12,080
80	$\frac{1}{2}$	72	$1\frac{15}{16}$	4	3	9.29	99.82	.654	12,560
80	$\frac{1}{2}$	72	$1\frac{15}{16}$	$4\frac{1}{2}$	$\frac{3}{4}$	9.34	97.05	.619	11,650
80	$\frac{1}{2}$	72	$1\frac{15}{16}$	$4\frac{1}{16}$	3	9.34	94.30	.642	12,310
80	$\frac{1}{2}$	72	$1\frac{15}{16}$	$4\frac{1}{8}$	3	9.40	91.50	.642	12,300
80	$\frac{1}{2}$	72	2	$4\frac{1}{8}$	$3\frac{1}{2}$	9.40	90.64	.653	12,500
80	$\frac{7}{16}$	70	$1\frac{15}{16}$	$4\frac{1}{8}$	4	9.40	89.80	.666	12,840
80	$\frac{5}{8}$	78	$2\frac{1}{16}$	$4\frac{1}{4}$	$1\frac{1}{2}$	9.27	87.05	.633	12,090
80	$\frac{1}{2}$	72	$1\frac{15}{16}$	$4\frac{1}{8}$	$2\frac{11}{16}$	9.34	83.00	.643	12,250
80	$\frac{5}{8}$	78	$2\frac{1}{16}$	$4\frac{3}{8}$	2	9.20	80.65	.639	12,150
80	$\frac{5}{8}$	78	$2\frac{1}{16}$	$4\frac{7}{16}$	2	9.30	77.60	.678	12,860
80	$\frac{1}{2}$	72	$1\frac{15}{16}$	$4\frac{3}{8}$	$2\frac{1}{2}$	9.34	76.20	.643	12,180
80	$\frac{5}{8}$	78	$2\frac{1}{16}$	$4\frac{1}{2}$	$\frac{1}{2}$	9.30	73.10	.642	12,120
80	$\frac{1}{2}$	72	$1\frac{15}{16}$	$4\frac{1}{2}$	$1\frac{1}{2}$	9.40	71.40	.635	12,000
80	$\frac{3}{8}$	67	$1\frac{7}{8}$	$4\frac{1}{2}$	$3\frac{1}{2}$	9.27	69.40	.668	12,570
80	$\frac{1}{2}$	72	$1\frac{15}{16}$	$4\frac{9}{16}$	....	9.30	68.20	.636	12,040
80	$\frac{5}{8}$	78	$2\frac{1}{16}$	$4\frac{3}{4}$	$9\frac{1}{2}$	9.24	63.60	.641	11,980
80	$\frac{1}{2}$	72	2	$4\frac{3}{4}$	$1\frac{1}{4}$	9.25	62.70	.660	12,320
80	$\frac{1}{2}$	72	$1\frac{15}{16}$	$4\frac{3}{4}$	1	9.37	62.00	.630	11,750
80	$\frac{3}{8}$	67	$1\frac{7}{8}$	$4\frac{3}{4}$	$2\frac{1}{2}$	9.26	60.70	.644	11,950
80	$\frac{1}{2}$	72	$1\frac{15}{16}$	$4\frac{3}{4}$	$1\frac{1}{4}$	9.39	59.00	.628	11,650
80	$\frac{5}{8}$	78	2	5	$\frac{1}{4}$	9.24	57.50	.657	12,260
80	$\frac{1}{2}$	72	$1\frac{15}{16}$	5	0	9.32	54.40	.636	11,720
80	$\frac{3}{8}$	67	$1\frac{7}{8}$	$5\frac{1}{8}$	1	9.20	50.80	.661	12,090

[a] The specific gravity of the gasoline at 60° F. was 0.7175. Heating value: High, 20,579; low, 19,289 B. t. u. per pound.

[b] For description of the regular Nash carburetors see p. 34.

[c] Calculated from the low heating value of the gasoline minus the latent heat of the water added.

TABLE 32.—*Results of trial and check tests made to determine the least consumption of various proportions of gasoline and denatured alcohol, of gasoline and water, and of denatured alcohol and water*—Continued.

SECTION C.—FUEL: DENATURED ALCOHOL AND WATER.

NASH 10-HORSEPOWER GASOLINE ENGINE.

[Denatured alcohol diluted with water and used in the double-cone carburetor.[a]]

Brake load, in pounds.	Air-valve setting.	Compression pressure (pounds)	Fuel needle-valve setting.	Water needle-valve setting.	Igniter setting.	Brake horse-power.	Per cent denatured alcohol, by weight.	Pounds of denatured alcohol consumed per brake horsepower-hour.	British thermal units per brake horse-power-hour (low).[b]
110	⅜	89	2¼	....	2½	14.06	100	0.878	10,560
110	½	92	2-2½	....	2½	14.06	100	.886	9,354
110	½	92	2-3	....	2½	14.05	100	.847	8,942
110	½	92	2-3½	....	2¾	13.94	100	.930	9,270
110	½	92	2-4	....	2½	14.06	100	.965	10,180
110	½	92	2-4	....	2½	14.05	100	.967	10,200
110	⅝	94	2-4	....	3½	13.96	100	.905	9,550
110	¾	95	2-7	....	4	13.97	100	.937	9,890
110	Open.	97	2-7½	....	2¾	14.01	100	.937	9,210
110	¼	86	3-3	....	3¼	13.95	87½	.972	10,120
110	¼	86	3-3	....	3¼	13.95	87½	.996	10,370
110	⅜	89	3-3¾	....	3	13.96	87½	1.010	10,530
110	½	92	3-5½	....	2¾	13.90	87½	1.030	10,730
110	⅝	94	3-0	....	3¾	14.04	87½	.976	10,170
110	¾	95	3-9¼	....	2½	13.92	87½	1.017	10,610
110	Open.	97	4-3¼	....	2½	13.92	87½	1.028	10,710
110	Closed.	79	2-6	....	4½	14.10	75	1.130	11,520
110	¼	86	2-6¼	....	2½	14.05	75	1.046	10,710
110	¼	86	2-6	....	3	14.06	75	1.011	10,350
110	½	92	3-2½	....	2¼	14.02	75	1.082	11,070
110	½	92	2-3	....	3½	14.08	75	1.145	11,720
110	¾	95	3-8	....	2	13.99	75	1.060	10,870
110	¾	95	2-4	....	2¾	14.06	75	1.110	11,260
110	Open.	97	2-6¾	....	2¾	14.01	75	1.094	11,210
110	Closed.	79	2-8½	....	4	14.03	62½	1.168	11,670
110	¼	86	3-3	....	1¾	14.00	62½	1.154	11,520
110	½	92	2-5½	....	3½	14.06	62½	1.280	12,750
110	¾	95	2-7½	....	2¾	14.00	62½	1.180	11,790
110	Open.	97	3-0	....	1½	13.92	62½	1.152	11,500
110	Open.	97	3-3	....	¾	14.02	62½	1.200	12,000
110	½	92	3-8	....	1½	13.96	50	1.400	13,410

[Denatured alcohol and water supplied through separate spray nozzles in the regular carburetor.[c]]

Brake load, in pounds.	Air-valve setting.	Compression pressure (pounds)	Fuel needle-valve setting.	Water needle-valve setting.	Igniter setting.	Brake horse-power.	Per cent denatured alcohol, by weight.	Pounds of denatured alcohol consumed per brake horsepower-hour.	British thermal units per brake horse-power-hour (low).[b]
100	½	72	3½	Closed.	2½	11.62	100	1.170	12,500
100	⅝	78	3½	Closed.	2	11.82	100	1.078	11,520
100	⅝	78	$3\frac{7}{16}$	Closed.	2	11.70	100	1.034	11,040
100	¾	81	$3\frac{13}{16}$	Closed.	1¾	11.67	100	1.072	11,450
100	⅝	78	3⅝	4¼	1	11.71	94.2	1.131	12,010
100	⅝	78	$3\frac{11}{16}$	4⅜	1½	11.74	90.2	1.101	11,640
100	⅝	78	4	4¾	1¼	11.67	82.5	1.186	12,430
100	¾	81	4	4¾	1	11.75	81.6	1.115	11,680
100	½	72	$3\frac{9}{16}$	4¾	2	11.73	80.0	1.180	12,320
100	¾	81	4¼	5¾	¼	11.82	69.1	1.176	12,050
100	⅝	78	$4\frac{1}{16}$	5¾	¾	11.72	68.2	1.203	12,310
100	⅝	78	4⅜	6¾	9½	11.77	59.5	1.270	12,720
100	¾	81	4¼	7	9	11.74	59.4	1.288	12,930
100	¾	81	4⅞	10	8	11.59	49.6	1.390	13,490
100	⅝	78	5¼	Open.	....	11.73	43.2	1.535	14,460

[a] Specific gravity of the denatured alcohol at 60° F. was 0.8188. Heating value: High, 11,788; low, 10,681 B. t. u. per pound. Per cent alcohol by weight, 94.3. For description of the double-cone carburetor see p. 37.

[b] Calculated from the low heating value of the denatured alcohol minus latent heat of water added.

[c] Specific gravity of the denatured alcohol at 60° F. was 0.8206. Heating value: High, 11,662; low, 10,557 B. t. u. per pound. Per cent alcohol by weight, 93.8.

GASOLINE-WATER TESTS.

One series of gasoline-water tests was run on the Nash gasoline engine in which the percentage of diluent ranged from 0 to 50 per cent.

DENATURED ALCOHOL AND WATER TESTS.

Two series of denatured alcohol and water tests were run on the Nash gasoline engine. The first series was run with denatured alcohol, diluted by 0 to 50 per cent of water and used in the double-cone carburetor. In the second series the fuel-and-water mixture was supplied through separate spray nozzles and the percentage of diluent varied from 0 to 56.8. The results of the first series of tests are tabulated in Table 33.

Details of search tests and tabulated results of the check tests are given in Tables 32 and 33. In order to enable a more easy understanding of the results the more important ones are shown by curves as platted in figure 31. The following table shows the engine method of governing, fuel, compression pressure, table number, and series of tests, for each curve:

Data for curves in figure 31.

Curve No.	Engine.	Method of governing.	Fuel.	Compression pressure.[a]	Table No.	Series of tests.
1	10-horsepower Nash gasoline..	Hit-or-miss mixture supply.	Gasoline and denatured alcohol.	....	33	A.[b]
2	.....do......................	.....do........	Gasoline and water.	....	33	B.[b]
3	.....do......................	.....do........	Denatured alcohol and water.	....	33	C.[c]
4	.....do......................	.....do........	.....do........	....	33	C.

[a] Pounds per square inch above atmosphere. Compression pressure varies with the engine and air-valve setting used. See p. 9.
[b] Through separate spray nozzles in the regular carburetor.
[c] With the double-cone carburetor.

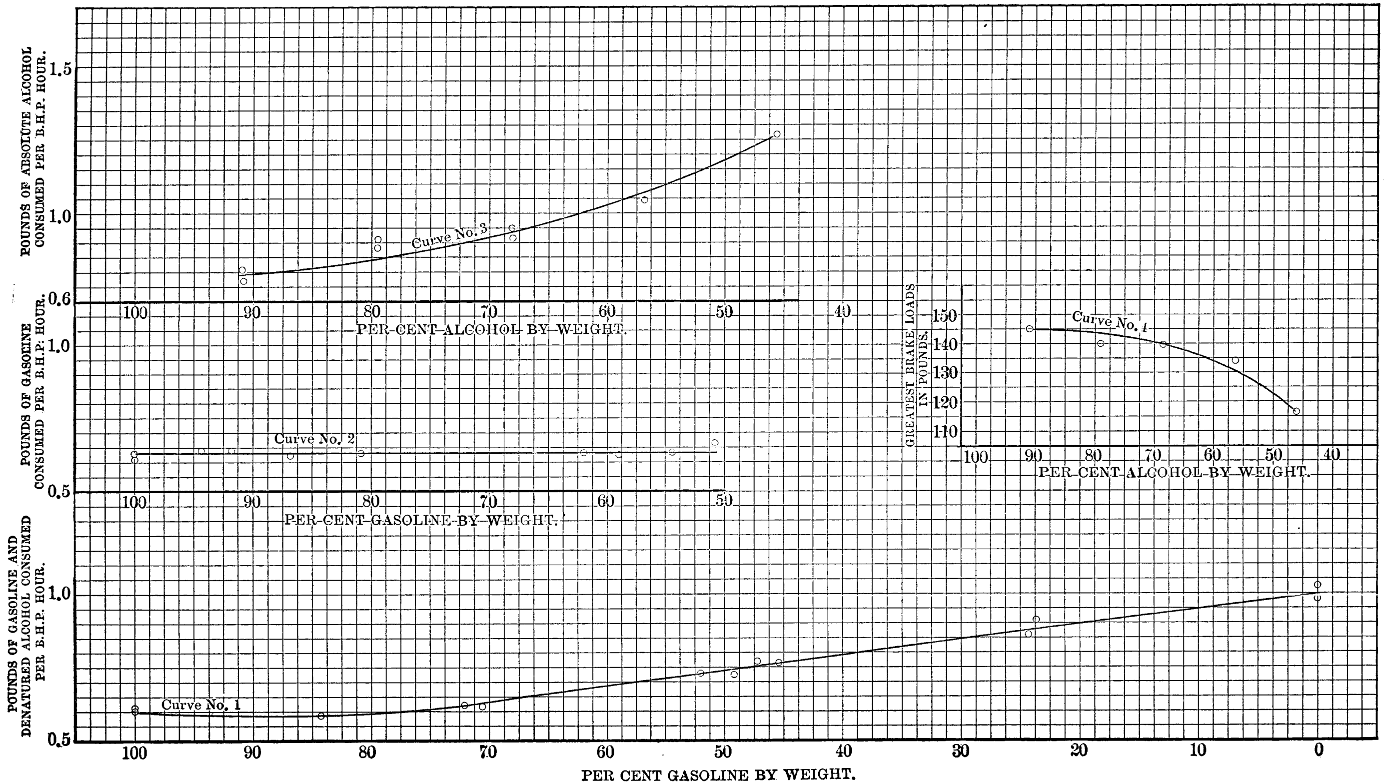

FIGURE 31.—Curves showing relation between fuel consumption and percentage of various solutions. Curve 1: Gasoline and denatured alcohol supplied through separate spray nozzles in the regular carburetor of the 10-horsepower Nash gasoline engine. Curve 2: Gasoline and water supplied through separate spray nozzles in the regular carburetor of the 10-horsepower Nash gasoline engine. Curve 3: Denatured alcohol diluted with water and used in the double-cone carburetor of the 10-horsepower Nash gasoline engine.

TABLE 33.—*Results of tests of the economy and thermal efficiency of mixtures of*

[NASH 10-HORSEPOWER GASOLINE ENGINE.—Diameter of cylinder, $7\frac{5}{8}$ inches; stroke, 10 inches; method of governing: Hit-or-miss mixture supply. Curves are platted in figure 31.]

(A) FUEL: GASOLINE *i* AND DENATURED ALCOHOL *j* EACH SUPPLIED THROUGH SEPARATE SPRAY NOZZLES IN THE REGULAR CARBURETOR.*k*

Test number.	Temperature (°F.).						Index settings.				Speed (av. per minute).			Average pressures (pounds per sq. in.).		
	Air.	Fuel.	Mixture.[a]		Jacket water.		Air valve.[b]	Needle valve.[c]		Igniter.[d]	Revolutions.	Explosions.	Mixture cut-outs.	Compression.	Mean effective.	Maximum (above atmosphere).
			Near carburetor.	Near inlet valve.	Inlet.	Outlet.		Alcohol.	Gasoline.							
A 3440	96	88	....	....	69	114	$\frac{5}{8}$	$3\frac{7}{16}$	Closed.	+2	296.1	146	2	78	83.4	310
A 3600	80	79	54	68	61	103	$\frac{5}{8}$	$8\frac{1}{8}$	Closed.	$+1\frac{3}{4}$	297.5	143	6	78	79.8	280
B 3559	89	85	52	94	63	112	$\frac{5}{8}$	$2\frac{15}{16}$	$4\frac{21}{32}$	$+1\frac{3}{4}$	296.8	133	15	78	88.2	345
B 3575	73	69	....	....	63	122	$\frac{5}{8}$	$2\frac{5}{8}$	$4\frac{9}{16}$	$+1\frac{1}{2}$	296.4	143	5	78	81.0	335
C 3581	73	69	40	80	61	......	$\frac{5}{8}$	2	$5\frac{1}{32}$	$+1\frac{1}{2}$	300.0	140	10	78	81.3	....
C 3583	74	71	40	82	61	......	$\frac{3}{4}$	$2\frac{1}{16}$	$5\frac{1}{16}$	$+\frac{1}{4}$	294.0	141	8	81	80.0	305
D 3596	100	83	41	93	61	111	$\frac{3}{4}$	$1\frac{15}{16}$	5	$+9\frac{1}{4}$	293.1	142	5	81	75.0	265
D 3598	78	71	41	84	61	104	$\frac{5}{8}$	$5\frac{3}{8}$	$1\frac{11}{16}$	0	296.3	141	10	78	77.4	315
E 3587	89	72	40	91	62	109	$\frac{5}{8}$	$1\frac{7}{16}$	$5\frac{3}{16}$	$+\frac{1}{4}$	299.0	146	7	78	73.5	280
E 3589	95	77	40	96	62	108	$\frac{5}{8}$	$1\frac{3}{8}$	$5\frac{3}{16}$	$+\frac{1}{2}$	295.4	144	4	78	72.6	290
F 3585	81	70	40	97	62	100	$\frac{5}{8}$	$1\frac{3}{16}$	$5\frac{7}{16}$	$+1\frac{1}{4}$	293.6	142	11	78	73.2	270
G 3591	93	84	42	113	62	109	$\frac{5}{8}$	Closed.	$5\frac{17}{32}$	+3	295.5	140	8	78	69.2	225
G 3608	83	70	40	103	60	80	$\frac{1}{2}$	Closed.	$1\frac{15}{16}$	$+1\frac{1}{4}$	297.2	141	8	72	72.0	280

(B) FUEL: GASOLINE *l* AND WATER EACH SUPPLIED THROUGH SEPARATE SPRAY NOZZLES IN THE REGULAR CARBURETOR.*k*

Test number.	Air.	Fuel.	Near carburetor.	Near inlet valve.	Inlet.	Outlet.	Air valve.	Water.	Gasoline.	Igniter.	Revolutions.	Explosions.	Mixture cut-outs.	Compression.	Mean effective.	Maximum.
H 3328	102	89	....	....	81	113	$\frac{1}{2}$	Closed.	$1\frac{15}{16}$	$+6\frac{1}{2}$	299.2	146	4	72	70.8	255
H 3497	96	85	....	....	66	111	$\frac{5}{8}$	Closed.	$2\frac{1}{16}$	......	293.5	140	7	78	76.8	250
I 3478	96	86	....	....	63	113	$\frac{1}{2}$	$4\frac{1}{16}$	$1\frac{15}{16}$	+3	295.6	144	4	72	75.0	265
I 3458	94	82	....	....	62	112	$\frac{1}{2}$	$4\frac{1}{8}$	$1\frac{15}{16}$	+3	297.6	143	6	72	70.8	250
J 3505	77	72	....	....	62	108	$\frac{5}{8}$	$4\frac{1}{4}$	$2\frac{1}{16}$	$+1\frac{1}{2}$	293.3	139	8	78	75.6	220
J 3507	84	78	....	....	62	105	$\frac{5}{8}$	$4\frac{3}{8}$	$2\frac{1}{16}$	+2	293.0	139	7	78	76.5	200
K 3454	86	78	....	....	75	117	$\frac{1}{2}$	$4\frac{3}{4}$	$1\frac{15}{16}$	+1	296.9	141	7	72	72.0	235
K 3468	90	79	....	....	62	109	$\frac{1}{2}$	$4\frac{3}{4}$	$1\frac{15}{16}$	$+1\frac{1}{4}$	297.2	144	4	72	72.0	230
L 3460	89	81	....	....	62	111	$\frac{1}{2}$	5	$1\frac{15}{16}$	0	295.0	145	2	72	72.0	220
L 3517	87	80	....	....	62	112	$\frac{3}{8}$	$5\frac{1}{8}$	$1\frac{7}{8}$	+1	291.2	141	4	67	75.6	285

a Measured by a thermometer placed in the inlet-valve housing (p. 57).
b Plug cock with dial for regulating the auxiliary air supply which is introduced between the governor-controlled mixture valve and the main inlet valve (p. 34).
c Scale arbitrary. Numbers increase with opening of needle valve.
d Scale arbitrary. Relation to crank position not constant. Ignition before dead center is indicated by (+) (p. 55).
e Horsepower constant: Brake, 0.000395; indicated, 0.001153 (p. 67).

gasoline and denatured alcohol, gasoline and water, and denatured alcohol and water.

[NASH 10-HORSEPOWER GASOLINE ENGINE.—Diameter of cylinder, 7⅝ inches; stroke, 10 inches; method of governing: Hit-or-miss mixture supply.]

(A) FUEL: GASOLINE [i] AND DENATURED ALCOHOL [j] EACH SUPPLIED THROUGH SEPARATE SPRAY NOZZLES IN THE REGULAR CARBURETOR.[k]

Brake load.			Horsepower.[e]			Fuel composition (per cent).		Heating value.[f]	Fuel consumption.							Efficiencies (per cent).		
										Per brake horsepower-hour.			Per indicated horsepower-hour.				Thermal.	
Greatest possible (pounds).	Amount carried (pounds).	Per cent of maximum.	Brake.	Indicated.	Duration (minutes).	By weight.	By volume.	B. t. u. per pound (low).	Pounds per hour.	Pounds.	Gallons.	B. t. u. (low).	Pounds.	Gallons.	B. t. u. (low).	Mechanical.	Indicated.[g]	Brake.[h]
						Gasoline.		Gasoline + denatured alcohol.										
117.5	100	85	11.69	14.0	30	0	0	10,681	12.08	1.034	0.151	11,040	0.86	0.126	9,170	83.2	27.7	23.0
117.5	100	85	11.74	13.1	29	0	0	10,681	11.50	.980	.143	10,460	.87	.128	9,350	89.4	27.2	24.3
111.0	96	87	11.24	13.5	16	23.7	26.3	12,720	10.14	.903	.137	11,480	.75	.113	9,510	82.9	26.8	22.2
111.0	96	87	11.24	13.3	32	24.3	27.1	12,770	9.59	.853	.129	10,900	.71	.109	9,180	84.3	27.7	23.3
105.0	91	87	10.77	13.1	19	47.3	51.0	14,750	8.36	.776	.122	11,450	.64	.100	9,420	82.4	27.0	22.2
105.0	91	87	10.70	13.0	31	45.3	48.7	14,580	8.19	.766	.119	11,160	.63	.098	9,170	82.2	27.8	22.8
104.0	91	88	10.54	12.3	32	51.9	55.2	15,150	7.75	.735	.116	11,140	.63	.099	9,570	86.0	26.6	22.9
104.0	91	88	10.65	12.6	32	49.1	52.6	14,910	7.75	.728	.114	10,850	.61	.097	9,190	84.7	27.7	23.5
98.0	87	89	10.28	12.4	16	70.4	73.5	16,730	6.49	.631	.102	10,550	.52	.085	8,770	83.1	29.0	24.1
98.0	87	89	10.15	12.1	29	72.0	74.5	16,880	6.39	.629	.102	10,610	.53	.086	8,960	84.2	28.4	24.0
94.0	87	92	10.09	12.0	17	84.0	85.6	17,910	5.92	.587	.097	10,510	.49	.082	8,850	84.2	28.8	24.2
90.0	80	89	9.33	11.1	32	100.0	100.0	19,292	5.90	.600	.101	11,560	.50	.085	9,700	83.8	26.2	22.0
90.0	80	89	9.39	11.7	32	100.0	100.0	19,292	5.71	.608	.102	11,730	.49	.082	9,450	80.5	26.9	21.7

(B) FUEL: GASOLINE [l] AND WATER EACH SUPPLIED THROUGH SEPARATE SPRAY NOZZLES IN THE REGULAR CARBURETOR.[k]

						Gasoline.		Gasoline + water.										
90.0	80	89	9.46	11.9	31	100.0	100.0	19,235	5.79	0.612	0.102	11,760	0.49	0.081	9,340	79.4	27.3	21.6
90.0	80	89	9.27	12.4	31	100.0	100.0	19,235	5.82	.628	.105	12,080	.47	.078	9,010	74.6	28.2	21.1
90.0	80	89	9.34	12.5	21	94.3	95.5	18,080	6.37	.681	.112	12,310	.61	.083	9,220	74.9	27.6	20.7
90.0	80	89	9.40	11.7	15	91.5	93.6	17,520	6.59	.702	.114	12,300	.57	.092	9,920	80.6	25.7	20.7
90.0	80	89	9.27	12.1	31	87.1	90.6	16,630	6.74	.727	.117	12,090	.55	.090	9,250	76.5	27.5	21.0
90.0	80	89	9.20	12.3	32	80.7	85.6	15,340	7.34	.792	.125	12,150	.60	.094	9,140	75.2	27.8	20.9
90.0	80	89	9.37	11.7	15	62.0	69.5	11,560	8.52	1.016	.151	11,750	.81	.121	9,380	79.9	27.1	21.7
90.0	80	89	9.39	11.9	16	58.9	66.7	10,930	10.01	1.066	.157	11,650	.83	.123	9,160	78.5	27.8	21.8
90.0	80	89	9.32	12.1	15	54.5	62.4	10,040	10.89	1.168	.170	11,720	.90	.131	9,050	77.2	28.1	21.7
90.0	80	89	9.20	12.3	30	50.8	58.8	9,300	11.96	1.300	.187	12,090	.97	.139	9,030	74.7	28.2	21.1

f Taken as the low heating value of the contained fuel minus the latent heat of the contained water.
g Calculated from the indicated horsepower and the low heating value of the fuel (p. 72).
h Calculated from the brake horsepower and the low heating value of the fuel (p. 72).
i Specific gravity at 60° F.=0.7122. Heating value: High=20,581, low=19,292 B. t. u. per pound.
j Specific gravity at 60° F.=0.8188. Heating value: High=11,788, low=10,681 B. t. u. per pound.
k For description of regular carburetor see p. 34.
l Specific gravity at 60° F.=0.7175. Heating value: High=20,579.

TABLE 33.—*Results of tests of the economy and thermal efficiency of mixtures of gasoline*

[NASH 10-HORSEPOWER GASOLINE ENGINE.—Diameter of cylinder, 7⅝ inches; stroke, 10 inches; method of governing: Hit-or-miss mixture supply. Curves are platted in figure 31.]

(C) FUEL: DENATURED ALCOHOL[a] DILUTED WITH WATER AND USED IN THE DOUBLE-CONE CARBURETOR.[b]

Test number.	Temperature (°F.).						Index settings.				Speeds (av. per minute).			Average pressures (pounds per sq. in.).		
	Air.	Fuel.	Mixture. Near carburetor.	Mixture. Near inlet valve.	Jacket water. Inlet.	Jacket water. Outlet.	Air valve.	Needle valve. Alcohol.	Needle valve. Water.	Igniter.	Revolutions.	Explosions.	Mixture cut-outs.	Compression.	Mean effective.	Maximum (above atmosphere).
M 3704	67	63	74	64	61	112	½	2–3	....	+2½	305.2	148	4	92	96.5	320
M 3763	59	57	....	....	52	118	½	2–2½	....	+2½	304.3	147	5	92	96.7	300
N 3774	60	60	....	....	52	117	¼	3–3	....	+3¼	303.3	150	2	86	94.6	380
N 3780	64	65	....	....	52	116	¼	3–3	....	+3¼	303.4	149	3	86	85.2	340
O 3786a	66	67	....	....	53	116	¼	2–6	....	+3	305.2	149	4	86	96.4	350
O 3786b	62	62	....	....	53	115	¼	2–6¼	....	+2½	305.6	152	1	86	94.2	355
P 3796	63	67	....	....	53	114	¼	3–3	....	+1¾	304.2	148	5	86	96.1	355
P 3798	60	62	....	....	54	120	¼	3	....	+1½	302.4	147	5	....	96.2	350
Q 3800	60	64	....	....	54	117	½	3–8	....	+1½	303.2	151	1	....	93.6	350

[a] Specific gravity at 60° F.=0.8188. Heating value: High=11,788.
[b] For description of double-cone carburetor see p. 37.

OPERATOR'S NOTES (TABLE 33).

Test A.—Five trial tests were made to check the best fuel economy that had previously been determined for this engine using alcohol.

Test B.—The percentage of gasoline and alcohol for the eleven trial tests was approximately the same, but the time of the ignition and total supply of fuel varied considerably. Best results obtained were selected and are given in the table (Table 33).

Test C.—Three trial tests were made with air valve three-fourths and five-eighths open. Best results for these settings are given in the tables.

Test D.—Three trial tests were made with air valve five-eighths and three-fourths open, two of which are given in the table.

Test E.—Only two tests were made and these are given in the table.

Test F.—Only one test was made and this is given in the table.

Test G.—Two tests were made. Both are given in the table and check the previously determined results for minimum fuel consumption for this engine.

Test H.—Five trial tests were made to check the minimum fuel consumption which had previously been determined for this engine.

and denatured alcohol, gasoline and water, and denatured alcohol and water—Contd.

[NASH 10-HORSEPOWER GASOLINE ENGINE.—Diameter of cylinder, 7⅝ inches; stroke 10 inches; method of governing: Hit-or-miss mixture supply.]

(C) FUEL: DENATURED ALCOHOL[a] DILUTED WITH WATER AND USED IN THE DOUBLE-CONE CARBURETOR.[b]

Brake load.			Horsepower.			Fuel composition (per cent).		Heating value.	Fuel consumption.							Efficiencies (per cent).		
										Per brake horsepower-hour.			Per indicated horsepower-hour.				Thermal.	
Greatest possible (pounds).	Amount carried (pounds).	Per cent of maximum.	Brake.	Indicated.	Duration (minutes).	By weight.	By volume.	B. t. u. per pound (low).	Pounds per hour.	Pounds.	Gallons.	B. t. u. (low).	Pounds.	Gallons.	B. t. u. (low).	Mechanical.	Indicated.	Brake.
						Absolute alcohol.[c]		Denatured alcohol + H_2O.										
145.0	116.5	80	14.05	16.5	31	90.8	93.8	10,557	11.90	0.847	0.124	8,940	0.72	0.105	7,640	85.4	33.3	28.4
145.0	116.5	80	14.00	16.4	31	90.8	93.8	10,557	12.41	.886	.129	9,350	.79	.110	7,990	85.4	31.9	27.2
140.0	116.5	83	13.95	16.3	17	79.5	85.0	9,120	15.88	1.138	.160	10,370	.97	.137	8,880	85.4	28.7	24.5
140.0	116.5	83	13.95	16.3	31	79.5	85.0	9,120	15.48	1.110	.156	10,120	.95	.133	8,660	85.4	29.4	25.1
140.0	116.5	83	14.05	16.5	14	68.1	75.1	7,680	18.94	1.348	.184	10,350	1.15	.157	8,820	85.4	28.9	24.6
140.0	116.5	83	14.05	16.5	32	68.1	75.1	7,680	19.60	1.395	.190	10,710	1.19	.162	9,120	85.4	27.9	23.8
134.0	116.5	86	14.00	16.4	15	56.8	64.5	6,240	25.85	1.846	.245	11,520	1.58	.209	9,830	85.4	25.9	22.1
134.0	116.5	86	13.92	16.3	30	56.8	64.5	6,240	25.65	1.842	.244	11,500	1.57	.208	9,820	.....	....	....
116.5	116.5	100	13.96	16.3	16	45.4	52.9	4,800	39.00	2.793	.360	13,410	2.39	.307	11,490	85.4	22.2	19.0

[c] Denatured alcohol taken as 90.8 per cent ethyl alcohol.

Tests I to L, inclusive.—Twenty-three trial tests were made with various proportions of gasoline and water, and the best results for the different proportions were selected and are given in the table. The best settings were judged as near as possible in each case by observations.

Test M.—Nine trial tests were made in order to check the minimum consumption, which had previously been determined on the same engine for this carburetor.

Tests N to P, inclusive.—Twenty trial tests were made, and those giving the minimum fuel consumption are given in the table. Best settings determined in the usual way.

Test Q.—A greater dilution of alcohol could not be made without decreasing the load. As the more dilute alcohol mixtures were used, increasing difficulty in starting the engine was experienced. The engine was started and brought up to speed by priming with gasoline. After running some time the engine would carry the load with a 50 per cent alcohol mixture.

CONCLUSIONS.

The gasoline and denatured-alcohol tests which were made on the Nash gasoline engine failed to show any change in the thermal efficiency sufficient to warrant the drawing of definite conclusions. The total quantity of fuel consumed ranges from the best obtainable with denatured alcohol alone to the best obtainable with gasoline alone. By referring to Table 32 it will be seen that a similar series of tests was run on a Nash alcohol engine using proportions of gasoline up to 30.6 per cent. The quantity of fuel consumed was never appreciably less than in the first series, despite the fact that this engine ran with a compression pressure of 130 pounds as compared with the compression pressure of 78 pounds in the series of similar tests on the Nash gasoline engine. However, no definite conclusions can be drawn from the comparison, because of these two engines the alcohol engine was throughout found to be relatively less efficient than the gasoline engine.

The second series of tests (water and gasoline), results of which are tabulated in Table 33, was run on the Nash gasoline engine. The water and gasoline were each supplied through separate spray nozzles in the regular carburetor. Amounts of water up to 50 per cent were used and no appreciable gain in efficiency is shown.

Tests with gasoline and water were run on the Nash alcohol engine also, and the results are tabulated in Table 22. In the tests only that amount of water required to prevent preignition was used. The compression of the engine was 130 pounds, or 52 pounds greater than that of the Nash gasoline engine, but no increase in efficiency over that obtained on the latter engine was found. Again, as in the gasoline and denatured alcohol tests mentioned above, this result is not a reasonable one and should not be considered conclusive.

It was impossible to raise the compression pressure of either engine so as to make tests for increased compression with the increase in water, and thus the sole advantage which was expected with the use of water remained undetermined.

Although no determinations were made of the quantity of cooling water required in these tests, report was made by the operators to the effect that much less was used as the quantity of water mixed with the fuel was increased.

It is probable that the effect of the varying humidity of the air upon the operator's adjustment of an internal-combustion engine must be considered in obtaining the best possible results, but it is manifest that the effect of such comparatively small variation in the natural humidity as may have been experienced during the procedure of the other tests is negligible so far as any of the results of this investigation is concerned.

The effect of diluting alcohol with water is shown by curve 3 in figure 31 to be detrimental. By the introduction of water in amounts up to 54.6 per cent by weight the efficiency of operation was reduced from 28.4 per cent to 19 per cent. Furthermore, this dilution with water caused an appreciable reduction in the capacity of the engine, as shown by curve 4, figure 31.

The trouble experienced in starting an engine with denatured alcohol was early mentioned as one of the drawbacks of its use. This trouble is augmented by the dilution with water, but did not become of sufficient importance to interfere with the use of considerable diluent.

The effect upon the quantity of cooling water of increasing the amount of diluent was the same as when gasoline was the fuel, except that it was somewhat more pronounced. With 51 per cent of water as a diluent scarcely any cooling water was required.

A supplementary series of tests with denatured alcohol and water introduced through separate spray nozzles was run on the Nash gasoline engine. The results were identical with those obtained when the fuel and water were mixed and used in the regular carburetor, and were therefore not tabulated in detail.

In conclusion, it does not appear that the use of diluents with denatured alcohol can be expected to effect an appreciable gain in engine efficiency. However, alcohol may be diluted with from 10 to 15 per cent of water without appreciably affecting the efficiency with which it is used, and the use of small quantities of water with gasoline, in order to prevent troublesome preignition, is efficacious.

COMPRESSION.

Throughout the discussion of the results of the tests performed during the investigation here reported, frequent references have been made to the higher efficiencies that were to be expected with increased compression pressures. The expectation was warranted to a certain extent, for the textbooks on thermodynamics give the standard formula for the thermal efficiency of the ideal air engine as follows:

$$E=1-\left(\frac{P_a}{P_b}\right)^{\frac{r-1}{r}}$$

in which E is the thermal efficiency, P_a the initial pressure, P_b the compression pressure, and r the ratio of specific heat at constant pressure to specific heat at constant volume.

EFFECT ON EFFICIENCY.

Inspection of the above empirical formula will show that with a constant specific heat of the exhaust gases the efficiency must increase with an increase in the compression pressure, P_b.

The results of tests have almost invariably shown increased efficiency whenever the design of the engine and the nature of the fuel would permit increasing the compression. But for some reason, with two equal compressions, denatured alcohol seems to have invariably given a higher thermal efficiency than has gasoline.

METHOD OF TESTING.

To determine the effect of compression on the thermal efficiency with which the two fuels under consideration can be used, a series of tests was unnecessary. Instead, tests which had given the best efficiencies at each different compression with each fuel during all of the tests, were carefully chosen and tabulated.

RESULTS OBTAINED.

Table 34 shows variation of best efficiencies with compression. The first part of the table shows this variation when denatured alcohol was the fuel, and the second part when gasoline was the fuel. Figure 32 shows graphically the relation found to exist between compression and efficiency with both fuels.

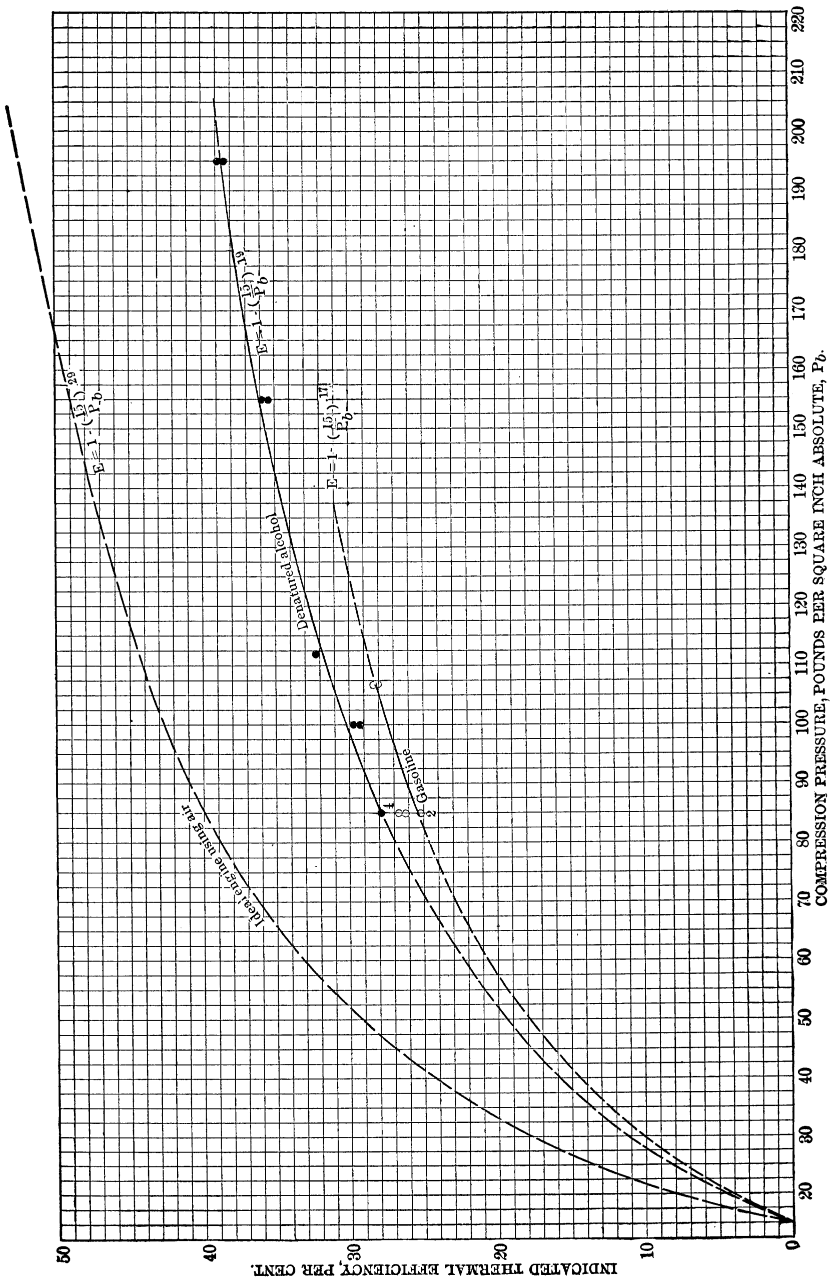

FIGURE 32.—Curves showing relation between compression and efficiency.

TABLE 34.—*Results of tests of best fuel*

FUEL: DENATURED ALCOHOL.[g]

Test number.	Engine number.[a]	Temperatures (°F.).				Time of ignition.[c]	Speed (average per minute).			Pressures (pounds per sq. in.).			Load in per cent of greatest.	Horsepower.		Duration, minutes.
		Air.	Fuel.	Mixture.[b]	Jacket-water outlet.		Revolutions.	Explosions.	Fuel cut-outs.	Compression (above atmosphere).	Average mean effective.	Average explosion (above atmosphere).		Brake.	Indicated.	
A 914	1	87	77	64	155	30	256.6	125	0	70	91.5	300	82	14.21	16.0	30
A 1962	2	86	83	66	163	30	254.5	124	3	70	96.6	320	87	14.86	16.8	25
A 923	1	81	75	67	196	30	257.0	128	1	70	90.8	300	82	14.23	16.3	30
A 1951	2	78	72	62	200	30	254.6	124	4	70	95.3	320	87	14.87	16.4	26
B 1979	2	89	80	65	200	30	251.4	119	7	85	97.2	345	87	14.68	16.2	27
B 1942	2	90	81	65	142	30	250.2	123	2	85	97.0	345	87	14.61	16.7	27
C 3694	4	78	72	75	110	30	306.0	146	7	97	114.6	420	97	16.91	19.3	31
D 2191	3	93	68	72	112	20	261.0	131	0	140	83.2	310	68	13.45	15.2	14
D 2192	3	95	69	69	110	20	259.0	130	0	140	85.5	310	68	13.35	15.5	28
E 2360	3	74	....	93	207	25	259.1	130	0	180	94.1	460	68	14.98	17.1	28
E 2381	3	63	....	74	173	25	259.2	130	0	180	94.1	460	68	14.98	17.1	17

FUEL: GASOLINE.[h]

Test number.	Engine number.[a]	Air.	Fuel.	Mixture.[b]	Jacket-water outlet.	Time of ignition.[c]	Revolutions.	Explosions.	Fuel cut-outs.	Compression (above atmosphere).	Average mean effective.	Average explosion (above atmosphere).	Load in per cent of greatest.	Brake.	Indicated.	Duration, minutes.
F 305	1	76	79	92	112	21	259.5	126	4	70	90.6	390	85	13.75	16.0	18
F 325	1	78	82	87	112	21	257.6	127	2	70	90.7	380	85	13.65	16.1	31
F 1866	2	81	91	82	118	25	250.8	116	10	70	91.5	350	85	12.95	14.8	15
F 1846	2	86	91	105	203	20	254.7	121	7	70	89.3	350	85	13.16	15.0	30
G 3631	4	82	....	94	112	....	301.3	144	7	92	81.0	320	72	10.69	13.4	17
G 3635	4	77	....	92	112	....	302.8	144	7	92	79.5	285	72	10.76	13.2	29

[a] Engine No. 1.—15-horsepower Otto gasoline engine: Cylinder dimensions, $6\frac{3}{4}$ by $15\frac{1}{2}$ inches; method of governing, hit-or-miss fuel supply. Horsepower constants: Brake, 0.0004817; indicated, 0.00140. For description see p. 28. Engine No. 2.—15-horsepower Otto gasoline engine: Same size and type as engine No. 1. Horsepower constants: Brake, 0.0004493; indicated, 0.001396. For description see p. 28. Engine No. 3.—15-horsepower Otto alcohol engine: Cylinder dimensions same as for No. 1 and No. 2; method of governing, combination of throttle and cut-off methods. Horsepower constants: Brake, 0.0004478; indicated, 0.001396. For description see p. 39. Engine No. 4.—10-horsepower Nash gasoline engine: Cylinder dimensions, $7\frac{5}{8}$ by 10 inches; method of governing, hit-or-miss mixture supply. Horsepower constants: Brake, 0.000395; indicated, 0.001153. For description see p. 33.

[b] Temperature of the air and fuel vapor mixture just before entering the cylinder.

[c] Time of ignition is given in degrees of the crank before dead center.

[d] Calculated from the cycle equation $E=1-\left(\frac{15}{P_b}\right)^x$, in which E is the measured indicated thermal efficiency given in the table, P_b is the absolute pressure corresponding to the compression pressure given in table, and x is the so-called indicated thermal efficiency exponent.

OPERATOR'S NOTES (TABLE 34).

Test A.—These tests were selected from the best fuel economy versus jacket-water temperature tests given in Tables 8 and 10. Results of other tests showing practically the same economies and thermal efficiencies for a compression pressure of 70 pounds will be found in Tables 16 and 18 of the best fuel economy versus load tests.

Test B.—These tests were also selected from those given in Table 10, and the results of the trial tests will be found in Table 9.

economy versus compression pressure.

FUEL: DENATURED ALCOHOL.[g]

Fuel consumption. Per brake horsepower per hour. Pounds.	Fuel consumption. Per brake horsepower per hour. Gallons.	Fuel consumption. Per brake horsepower per hour. B. t. u. (low).	Fuel consumption. Per indicated horsepower per hour. Pounds.	Fuel consumption. Per indicated horsepower per hour. Gallons.	Fuel consumption. Per indicated horsepower per hour. B. t. u. (low).	Measured efficiencies (per cent). Mechanical.	Measured efficiencies (per cent). Thermal. Brake. For low B. t. u. value.	Measured efficiencies (per cent). Thermal. Brake. For high B. t. u. value.	Measured efficiencies (per cent). Thermal. Indicated. For low B. t. u. value.	Measured efficiencies (per cent). Thermal. Indicated. For high B. t. u. value.	Values of x, the indicated thermal efficiency exponent.[d] For low B. t. u. value.	Values of x, the indicated thermal efficiency exponent.[d] For high B. t. u. value.	Calculated indicated thermal efficiencies (per cent). For denatured alcohol and for gasoline.[e]	Calculated indicated thermal efficiencies (per cent). For the standard reference or air diagram.[f]
0.980	0.144	10,300	0.88	0.128	9,140	88.8	24.7	22.5	27.8	25.1	0.190	0.169	28.0	39.6
.960	.140	10,250	.85	.124	9,090	87.7	24.8	22.5	28.0	25.8				
.996	.145	10,410	.87	.127	9,110	87.6	24.4	22.1	27.9	25.7				
.946	.138	10,100	.85	.125	9,150	90.5	25.2	22.8	27.8	25.2				
.900	.132	9,610	.82	.120	8,740	90.9	26.5	23.2	29.1	25.8	.185	.162	30.3	42.4
.912	.133	9,740	.80	.117	8,540	87.6	26.1	23.7	29.8	27.0				
.858	.125	9,060	.75	.109	7,930	87.6	28.1	25.5	32.1	29.1	.193	.171	31.8	44.2
.792	.116	8,360	.70	.102	7,390	88.4	30.4	27.6	35.8	31.4	.189	.164	35.8	49.3
.803	.117	8,480	.68	.099	7,180	86.0	30.0	27.2	35.4	32.2				
.709	.104	7,550	.62	.091	6,610	87.5	33.7	30.8	38.5	34.4	.190	.164	38.5	52.3
.706	.103	7,510	.62	.091	6,580	87.5	33.9	30.7	38.7	34.4				
									Average.....		.189	.166		

FUEL: GASOLINE.[h]

0.593	0.097	11,330	0.51	0.084	9,740	86.1	22.5	21.0	26.1	24.5	0.171	0.161	25.5	39.5
.590	.097	11,270	.50	.082	9,590	85.0	22.6	21.2	26.5	24.9				
.601	.100	11,560	.52	.088	10,140	87.8	22.0	20.5	25.1	23.4				
.603	.101	11,600	.53	.088	10,140	87.5	21.9	20.5	25.1	23.4				
.585	.098	11,280	.47	.079	9,020	79.7	22.5	21.1	28.2	26.6	.168	.157	28.4	43.4
.580	.097	11,180	.47	.079	9,080	81.4	22.8	21.3	28.0	26.6				
									Average.....		.170	.159		

[e] Calculated from the equation $E=1-\left(\frac{15}{P_b}\right)^{.19}$ for denatured alcohol, and from the equation $E=1-\left(\frac{15}{P_b}\right)^{.17}$ for gasoline, in which equations E is the indicated thermal efficiency, and P_b is the compression pressure in pounds per square inch above atmosphere.

[f] Calculated from the equation $E=1-\left(\frac{15}{P_b}\right)^{.29}$, in which E is the indicated thermal efficiency of the air diagram, and P_b is the absolute compression pressure.

[g] Specific gravity at 60° F.=0.8181 to 0.8241. Per cent alcohol by weight, 89.5 to 91.7. Heating value: High, 11,788 to 11,473; low, 10,681 to 10,379 B. t. u. per pound.

[h] Specific gravity at 60° F.=0.7122 to 0.7301. Heating value: High, 20,901 to 20,722; low, 19,289 to 19,100.

Test C.—This test was selected from the best fuel economy versus load tests given in Table 20. Results of trial tests will be found in Table 19.

Tests D and E.—These tests were selected from the best fuel economy versus load test given in Table 24. Results of trial tests will be found in Table 23.

Test F.—The first three of these tests were selected from the best fuel economy versus load tests given in Tables 16 and 18. Results of trial tests will be found in Tables 15 and 17. The fourth test, F 1846, was selected from the best fuel economy versus jacket-water temperature tests given in Table 10.

Test G.—These tests were selected from the best fuel economy versus load tests given in Table 20. Results of trial tests will be found in Table 19.

CONCLUSIONS.

Table 34 is arranged to show the increase in efficiency with increase in compression, regardless of the engine used. The figures obtained with four different engines are used, disregarding their type, method of governing, and all details of design. Nevertheless a remarkably good check is obtained in the value of $\frac{r-1}{r}$ calculated from results obtained with denatured alcohol and with gasoline. However, this value for gasoline averages 0.17, whereas for alcohol it averages 0.19, and for air it is given by most authorities as 0.29.

In figure 32, indicated thermal efficiency is correlated with compression pressure, and three curves are shown. The first curve shows the conditions which might be expected with the ideal engine with air as the working medium. The other two show to what degree this ideal condition has been approached with the gases obtained from the combustion of denatured alcohol and gasoline as the working medium.

Denatured alcohol more nearly approaches the ideal fuel than does gasoline, for at any one compression it shows a greater efficiency. Since the denatured alcohol curve and the gasoline curve were both platted by means of two similar formulas, differing only in the value of the exponent $\frac{r-1}{r}$, we must look to this value for an explanation of the reason why denatured alcohol seems to be a more ideal fuel than gasoline.

With increasing values of $\frac{r-1}{r}$ from 0 to 0.29 it is seen that increasing efficiency approaching the ideal may be obtained. Since this exponent has been determined as 0.19 for denatured alcohol and 0.17 for gasoline, it would appear that r for the gas resulting from the combustion of denatured alcohol must be greater than that of the gases resulting from the combustion of gasoline.

It seems, then, that some explanation for the greater efficiency with which denatured alcohol may be used at a given compression may be found in the character of the exhaust gases. However, because of the changing character of the gases as combustion progresses and the consequent changes in specific heat, a study of this feature will be surrounded with many difficulties. Nevertheless future work along this line could be profitably spent in analysis of exhaust gases with a view to learning something of their specific heats in the hope that knowledge thus gained may enable investigators to attack the problem in the future with greater intelligence and knowledge of the principles involved.

PUBLICATIONS ON FUEL TESTING.

The following publications can be obtained free of cost by applying to the Director of the Bureau of Mines, Washington, D. C :

Bulletin 1. The volatile matter of coal, by H. C. Porter and F. K. Ovitz. 1910. 56 pp., 1 pl.

Bulletin 2. North Dakota lignite as a fuel for power-plant boilers, by D. T. Randall and Henry Kreisinger. 1910. 42 pp., 1 pl.

Bulletin 3. The coke industry of the United States as related to the foundry, by Richard Moldenke. 1910. 32 pp.

Bulletin 4. Features of producer-gas power-plant development in Europe, by R. H. Fernald. 1910. 27 pp.

Bulletin 5. Washing and coking tests of coal at the fuel-testing plant, Denver, Colo., July 1, 1908, to June 30, 1909, by A. W. Belden, G. R. Delamater, J. W. Groves, and K. M. Way. 1910. 62 pp.

Bulletin 6. Coals available for the manufacture of illuminating gas, by A. H. White and Perry Barker. 1911. 77 pp., 4 pls.

Bulletin 7. Essential factors in the formation of producer gas, by J. K. Clement, L. H. Adams, and C. H. Haskins. 1911. 58 pp., 1 pl.

Bulletin 8. The flow of heat through furnace walls, by W. T. Ray and Henry Kreisinger. 1911. 32 pp.

Bulletin 9. Recent development of the producer-gas plant in the United States, by R. H. Fernald. 1909. 82 pp., 2 pls. Reprint of United States Geological Survey Bulletin 416.

Bulletin 11. The purchase of coal by the Government under specifications, by G. S. Pope. 1911. 80 pp. Reprint of United States Geological Survey Bulletin 428.

Bulletin 12. Apparatus and methods for the sampling and analysis of furnace gases, by J. C. W. Frazer and E. J. Hoffman. 1911. 22 pp.

Bulletin 13. Résumé of producer-gas investigations, October 1, 1904, to June 30, 1910, by R. H. Fernald and C. D. Smith. 1911. 393 pp., 12 pls.

Bulletin 14. Briquetting tests of lignite at Pittsburgh, Pa., 1908–9; with a chapter on sulphite-pitch binder, by C. L. Wright. 1911. 64 pp., 11 pls.

Bulletin 16. The uses and value of peat for fuel and other purposes, by C. A. Davis. 1911. 214 pp., 1 pl.

Bulletin 19. Physical and chemical properties of the petroleums of the San Joaquin Valley, Cal., by I. C. Allen and W. A. Jacobs; with a chapter on analyses of natural gas from the southern California oil fields, by G. A. Burrell. 1911. 60 pp.

Bulletin 21. The significance of drafts in steam-boiler practice, by W. T. Ray and Henry Kreisinger. 62 pp. Reprint of United States Geological Survey Bulletin 367.

Bulletin 24. Binders for coal briquets, by J. E. Mills. 56 pp. Reprint of United States Geological Survey Bulletin 343.

Bulletin 27. Tests of coal and briquets as fuel for house-heating boilers, by D. T. Randall. 45 pp., 3 pls. Reprint of United States Geological Survey Bulletin 366.

Bulletin 28. Experimental work conducted in the laboratory of the United States fuel-testing plant at St. Louis, Mo., Jaunary 1, 1905, to July 31, 1906, by N. W. Lord. 49 pp. Reprint of United States Geological Survey Bulletin 323.

Bulletin 29. The effect of oxygen in coal, by David White. 80 pp., 3 pls. Reprint of United States Geological Survey Bulletin 382.

Bulletin 30. Briquetting tests at the United States fuel-testing plant, Norfolk, Va., 1907–8, by C. L. Wright. 41 pp., 9 pls. Reprint of United States Geological Survey Bulletin 385.

Bulletin 31. Incidental problems in gas-producer tests, by R. H. Fernald, C. D. Smith, J. K. Clement, and H. A. Grine. 29 pp. Reprint of United States Geological Survey Bulletin 393.

Bulletin 32. Commercial deductions from comparisons of gasoline and alcohol tests on internal-combustion engines, by R. M. Strong. 38 pp. Reprint of United States Geological Survey Bulletin 392.

Bulletin 33. Comparative tests of run-of-mine and briquetted coal on the torpedo boat *Biddle*, by W. T. Ray and Henry Kreisinger. 49 pp. Reprint of United States Geological Survey Bulletin 403.

Bulletin 34. Tests of run-of-mine and briquetted coal in a locomotive boiler, by W. T. Ray and Henry Kreisinger. 32 pp. Reprint of United States Geological Survey Bulletin 412.

Bulletin 35. The utilization of fuel in locomotive practice, by W. F. M. Goss. 28 pp. Reprint of United States Geological Survey Bulletin 402.

Bulletin 36. Alaskan coal problems, by W. L. Fisher. 1911. 32 pp., 1 pl.

Bulletin 37. Comparative tests of run-of-mine and briquetted coal on locomotives, including torpedo-boat tests and some foreign specifications for briquetted fuel, by W. F. M. Goss. 57 pp., 4 pls. Reprint of United States Geological Survey Bulletin 363.

Bulletin 39. The smoke problem at boiler plants, a preliminary report, by D. T. Randall. 31 pp. Reprint of United States Geological Survey Bulletin 334, revised by S. B. Flagg.

Bulletin 40. The smokeless combustion of coal in boiler furnaces, by D. T. Randall and H. W. Weeks. 188 pp. Reprint of United States Geological Survey Bulletin 373, revised by Henry Kreisinger.

Bulletin 41. Government coal purchases under specifications, with analyses for the fiscal year 1909–10, by G. S. Pope, with a chapter on the fuel-inspection laboratory of the Bureau of Mines, by J. D. Davis. 1912. 97 pp., 3 pls.

Technical Paper 1. The sampling of coal in the mine, by J. A. Holmes. 1911. 18 pp.

Technical Paper 2. The escape of gas from coal, by H. C. Porter and F. K. Ovitz. 1911. 14 pp.

Technical Paper 3. Specifications for the purchase of fuel oil by the Government, with directions for sampling oil and natural gas, by I. C. Allen. 1911. 13 pp.

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BULLETIN 261. Preliminary report on the operations of the coal-testing plant of the United States Geological Survey at the Louisiana Purchase Exposition, in St. Louis, Mo., 1904; E. W. Parker, J. A. Holmes, M. R. Campbell, committee in charge, 1905. 172 pp. 10 cents.

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