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UTILIZATION OF NONPETROLEUM FUELS IN AUTOMOTIVE ENGINES

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ABSTRACT

A number of substitute fuels and blends were tested to determine their relative efficiency in the operation of common types of engines. The tests showed that the maximum power developed with alcohol and with some of the other fuels was slightly greater than with gasoline. The specific fuel consumption with the various fuels was approximately in inverse proportion to the heat of combustion of the fuel used. Analysis showed that the mixture distribution was less uniform with the substitute fuels than with gasoline. Tests made with low-proof alcohols showed than an engine can be operated on blends as low as 70 proof, but it is ordinarily impractical to use a blend much below 190 proof because of the excessive volume required.

CONTENTS

I. INTRODUCTION

The tests reported herein are part of an extensive study [1][1] of substitute motor fuels conducted by the National Bureau of Standards for the Foreign Economic Administration. The fuels used in this

[1] Figures in brackets indicate the literature references at the end of this paper.

phase of the study were ethyl alcohol and related compounds, either in their relatively pure state or in blends with water or with each other. Particular attention was given to 190-proof and 200-proof alcohol. The principal objectives of this part of the investigation were to show (1) the comparative performance of various engines and fuels with respect to power and fuel economy, (2) the relation of combustion temperatures to air-fuel mixture ratios, (3) the relative efficiency of mixture distribution with various fuels, and (4) the relation of air-fuel ratio to exhaust-gas composition.

II. FUELS USED IN THE TESTS

The fuels used in the tests were (1) 190-proof alcohol, (2) 200-proof alcohol, (3) acetone, (4) *N*-butanol (normal butyl alcohol), (5) "No. 1 Blend"—a blend containing 50 percent of acetone and 50 percent by volume of butanol, (6) "No. 2 Blend"—a blend of 67 percent of butanol, 27 percent of acetone, and 6 percent of ethanol, (7) "No. 3 Blend"—a blend of 71.5 percent of butanol and 28.5 percent of acetone, (8) a blend of 20 percent of diethyl ether and 80 percent of 190-proof alcohol, (9) various low-proof alcohol blends, and (10) gasoline.

III. TEST EQUIPMENT

1. ENGINES AND ACCESSORIES

Two 1942 Plymouth engines, a 1942 Chevrolet, and a 1940 Ford V–8 engine were used. The engines were coupled to electric dynamometers, equipped with dynamometer scales. Special test equipment included instruments for measuring oil pressure and cooling water temperatures, thermometers for measuring the temperatures of the fuel and air, a volumetric fuel-measuring device, and thermocouples for measuring various engine temperatures.

2. THERMOCOUPLE SPARK PLUGS

The spark plugs used in the tests were of the type devised by Rabezzana and Kalmar [2]—a special-type plug with tubular center electrodes into which iron-constantan thermocouples are inserted and welded to the electrode near its tip. The purpose of using these spark plugs was to study the mixture distribution. Rabezzana and Kalmar found that the temperature measured at the tip of the electrode is strictly proportional to the mean combustion temperature and very sensitive to changes in mixture ratio. This being true, this temperature can be used for determining mixture conditions by correlation of temperature changes with changes made in mixture ratios or other operating conditions.

The temperatures indicated by spark-plug thermocouples in different cylinders should not be regarded as directly comparable, because it is unlikely that two thermocouples can be located in the same positions relative to the tip and walls of the electrodes. The temperature drop along the electrode is very sharp so that a very small shift in the position of the thermocouple in the electrode might make a difference of 50 to 100 degrees in its indicated temperature.

3. CARBURETION

To facilitate changing the mixture ratio, standard carburetors were modified to permit a greatly increased rate of flow, with the leaner mixtures being supplied by lowering the pressure in a sealed carburetor float bowl by suction from the venturi. A needle valve in the suction line was used to regulate the pressure drop in the bowl. Another needle valve in the top of the bowl was used as an air bleed for further control. The engine could then be operated over a wide range of mixture ratios.

4. EXHAUST-GAS ANALYZER

The Hays Orsatomat, used in analyses of the exhaust gas, removes the CO_2 from the sample by absorption. The resulting reduction in volume actuates a needle that indicates the percentage of CO_2. The instrument was frequently calibrated against known mixtures of CO_2 and air. The corrected readings are believed to be accurate within 0.5 percent.

5. AIR-FLOW INDICATOR

A calibrated flow indicator was used in the air measurements. The air is forced through it by a constant-speed fan whose output is controlled by a valve. The output of the instrument may be varied from about 5 ft^3/min to somewhat above 75 ft^3/min. The pressure at the carburetor air intake was kept balanced with the atmosphere by controlling the air flow. To guard against errors due to engine pulsations, a collapsible rubber sleeve was placed in a section of the air line.

IV. OUTLINE OF TEST PROCEDURE

In the tests at full throttle the engine speed was kept constant at 1,500 rpm by making the necessary changes in the electric load of the dynamometer. The operating range of mixture ratios was estalished for each of the fuels by making a number of test runs in which the mixture ratio was varied in steps from lean to rich by use of the back-suction control of the carburetor. The power and specific fuel consumption were then plotted against the measured fuel consumption, the latter being an index of the mixture ratio in operation at any constant throttle setting. This is shown in figure 1.

In the road-load tests (part throttle) the same procedure was followed, except that the engine speed and dynamometer load were kept constant at the value calculated as equivalent to road operation at a given speed, and the throttle was varied to keep the desired speed and load as the mixture ratio was changed.

For each test run a set of data was recorded, including temperatures observed for each of the spark-plug thermocouples, exhaust, intake riser, oil, and water; also intake-manifold depression, float-bowl depression (back suction from the venturi), the time required to use a measured amount of fuel, and the speed and torque developed by the engine.

The spark advance was set for maximum engine power with the engine operating at 1,500 rpm on gasoline. With the first Plymouth

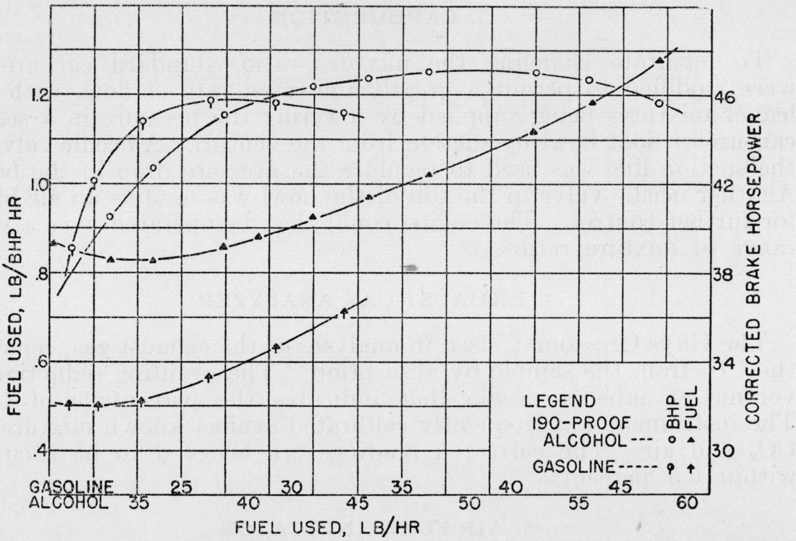

Figure 1.—*Power and specific fuel consumption for gasoline and 190-proof alcohol, 1942 Plymouth, test engine 4.*

The engine was operated at 1,500 rpm, full throttle, with the spark setting for maximum power.

test engine these settings were made by shifting the spark advance until the highest engine power was observed. Regular-grade gasoline of about 72 octane number was used. It was later discovered that this engine lost power because of excessive detonation when regular gasoline was used, and that the optimum spark advance was several degrees less than for fuels that did not cause detonation.

The optimum spark advance for the Chevrolet engine was also determined by observation. With the Ford V-8 engine and the second Plymouth engine (herein designated as test engine 4), the optimum spark advance was taken as the peak of a smooth curve joining the points when the observed power was plotted against the spark advance (see fig. 2). The optimum spark advance determined by this method for the various fuels ranged from 16.5 to 18.5 degrees with the Ford V–8 engine and from 16.4 to 19.0 degrees with the Plymouth engine. This variation was probably within the limits that might be accounted for by experimental error and by differences in atmospheric conditions. Therefore, the same spark advance was used for all the fuels except the low-proof alcohols, which required a definitely greater advance than the other fuels.

The optimum spark advance for low-proof alcohols of various proofs increased progressively from about 17 degrees with 200-proof alcohol to about 50 degrees with 70-proof alcohol, as shown in figure 3. The spark setting used in the part-throttle runs was that for maximum power with full throttle, as in road operation there is ordinarily no practical way to find the optimum spark advance at part throttle.

The air flow was measured in the road-load tests of the 1942 Plymouth engine and in some of the tests of the same engine at full throttle. The air-flow indicator was not available for use in the other tests.

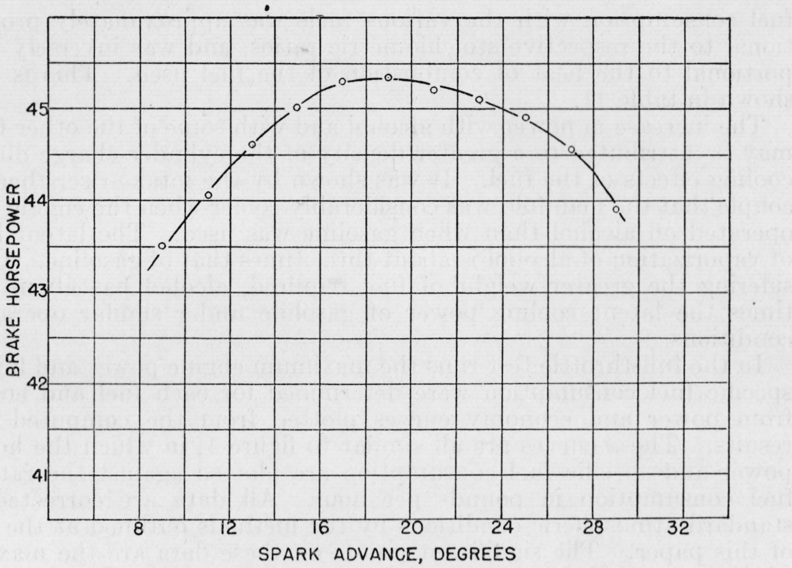

FIGURE 2.—*Spark advance and maximum engine power, 1942 Plymouth, test engine 4.*
The engine was operated at 1,500 rpm, full throttle on 190-proof alcohol.

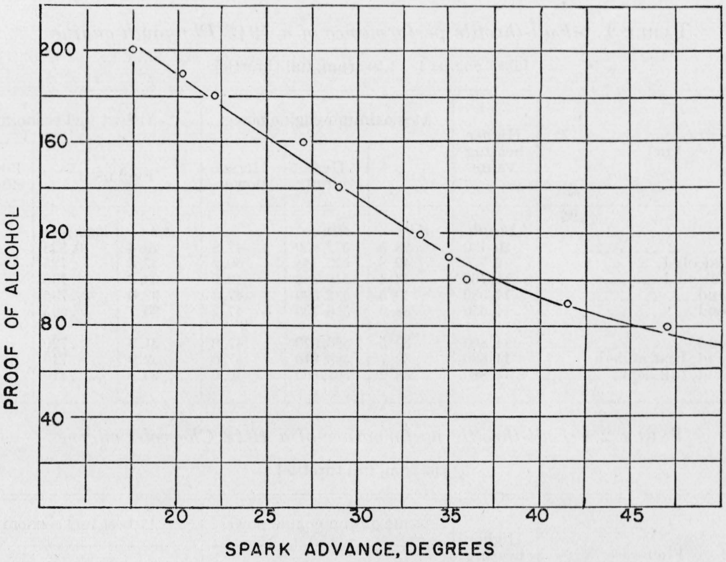

FIGURE 3.—*Optimum spark advance for alcohols of various proofs; 1942 Plymouth.*
The engine was operated at 1,500 rpm with full throttle.

V. TEST RESULTS

1. FULL-THROTTLE POWER AND FUEL CONSUMPTION

In general the alcohol fuels and blends produced slightly greater engine power (up to about 4 percent), but required considerably more fuel than was used when the engine was operated on gasoline. The fuel consumption with the various fuels was approximately proportional to the respective stoichiometric ratios, and was inversely proportional to the heat of combustion of the fuel used. This is best shown in table 11.

The increase in power with alcohol and with some of the other fuels may be attributed to a greater density of the cylinder charge due to cooling effects of the fuel. It was shown by the intake-riser thermocouple that the manifold was considerably cooler when the engine was operated on alcohol than when gasoline was used. The latent heat of vaporization of alcohol is about three times that of gasoline. Considering the greater weight of fuel required, alcohol has about five times the latent cooling power of gasoline under similar operating conditions.

In the full-throttle test runs the maximum engine power and lowest specific fuel consumption were determined for each fuel and engine from power and economy curves plotted from the computed test results. These curves are all similar to figure 1, in which the horsepower and specific fuel consumption are plotted against the rate of fuel consumption in pounds per hour. All data are corrected to standard atmospheric conditions by the methods outlined at the end of this paper. The significant points on these data are the maxima of the power and the minima of the economy curves. The values for maximum power and lowest specific fuel consumption from these plots are listed in tables 1, 2, 3, and 4.

TABLE 1.—*Full-throttle performance of a 1942 Plymouth engine*

[Test engine 1. 1,500 rpm, full throttle]

Fuel	Higher heating value	At maximum engine power			At best fuel economy		
		Fuel	Heat input	Horsepower	Fuel used		Thermal efficiency
	Btu/lb	*lb/hr*	*Btu/hr*		*lb/hr*	*lb/bhp hr*	%
Gasoline	20, 120	28. 5	572, 850	47. 8	23. 4	0. 525	24. 0
200-proof alcohol	12, 720	42. 8	543, 144	50. 0	32. 8	. 725	27. 6
190-proof alcohol	11, 760	43. 4	510, 380	49. 0	33. 3	. 735	29. 4
No. 1 Blend	14, 380	40. 5	582, 390	47. 3	32. 0	. 735	23. 9
No. 2 Blend	14, 650	38. 0	556, 700	47. 2	30. 7	. 695	25. 0
No. 3 Blend	14, 860	39. 5	586, 970	47. 0	31. 3	. 720	23. 8
No. 3 Blend, heat added	14, 860	39. 7	589, 940	47. 0	32. 0	. 740	23. 1
No. 3 Blend, full heat	14, 860	38. 2	567, 650	46. 0	29. 5	. 710	24. 1

TABLE 2.—*Full-throttle performance of a 1942 Chevrolet engine*

[1,500 rpm, full throttle]

Fuel	Higher heating value	At maximum engine power			At best fuel economy		
		Fuel	Heat input	Horsepower	Fuel used		Thermal efficiency
	Btu/lb	*lb/hr*	*Btu/hr*		*lb/hr*	*lb/bhp hr*	%
Gasoline	20, 120	27. 4	551, 300	46. 5	22. 3	0. 515	24. 6
190-proof alcohol	11, 760	46. 0	541, 000	46. 8	36. 5	. 853	25. 4
200-proof alcohol	12, 720	44. 0	559, 700	47. 8	35. 5	. 807	24. 8
No. 1 Blend	14, 380	39. 4	569, 450	47. 5	31. 5	. 710	24. 9
No. 2 Blend	14, 650	38. 6	565, 490	47. 4	31. 6	. 700	24. 8
No. 3 Blend	14, 860	38. 5	572, 000	47. 8	31. 3	. 700	24. 5
Acetone	13, 180	42. 0	553, 560	46. 8	34. 7	. 800	24. 1
Butanol	15, 460	37. 2	572, 110	46. 2	28. 6	. 675	24. 4

TABLE 3.—*Full-throttle performance of a 1940 Ford V–8 engine*

[1,500 rpm]

Fuel	At maximum engine power				At best fuel economy		
	Higher heating value	Fuel	Heat input	Horse-power	Fuel used		Thermal efficiency
	Btu/lb	*lb/hr*	*Btu/hr*		*lb/hr*	*lb/bhp hr*	%
Gasoline	20, 120	29. 0	551, 014	43. 7	21. 7	0. 538	23. 5
No. 1 Blend	14, 380	40. 5	553, 630	44. 0	30. 4	. 757	23. 4
No. 2 Blend	14, 650	39. 0	556, 700	43. 9	30. 2	. 738	23. 5
No. 3 Blend	14, 860	39. 0	557, 250	44. 1	29. 5	. 723	23. 7
Acetone	13, 180	43. 5	546, 970	44. 3	33. 5	. 835	23. 1
Butanol	15, 460	36. 0	541, 100	43. 3	28. 6	. 725	22. 7
190-proof alcohol	11, 760	49. 0	536, 256	44. 0	37. 0	. 915	23. 7
200-proof alcohol	12, 720	46. 0	566, 040	44. 2	33. 8	. 850	23. 5

TABLE 4.—*Full-throttle performance of a 1942 Plymouth engine*

[Test engine 4. 1,500 rpm]

Fuel	At maximum engine power				At best fuel economy		
	Higher heating value	Fuel	Heat input	Brake horse-power	Fuel used		Thermal efficiency
	Btu/lb	*lb/hr*	*Btu/hr*		*lb/hr*	*lb/bhp hr*	%
No. 1 Blend	14, 380	38. 4	552, 000	46. 9	29. 5	0. 692	25. 58
No. 2 Blend	14, 650	38. 0	556, 700	46. 9	29. 0	. 680	25. 55
No. 3 Blend	14, 860	36. 7	545, 362	47. 3	28. 5	. 660	25. 95
Acetone	13, 180	43. 4	572, 000	47. 4	33. 6	. 770	25. 08
Butanol	15, 460	36. 0	556, 560	47. 1	28. 0	. 645	25. 52
Gasoline	20, 120	26. 9	541, 228	45.8	21. 0	. 500	25. 30
190-proof alcohol	11, 760	50. 8	597, 400	47. 2	35. 0	. 830	26. 07
200-proof alcohol	12, 720	46. 6	592, 752	47. 6	33. 0	. 770	26. 0

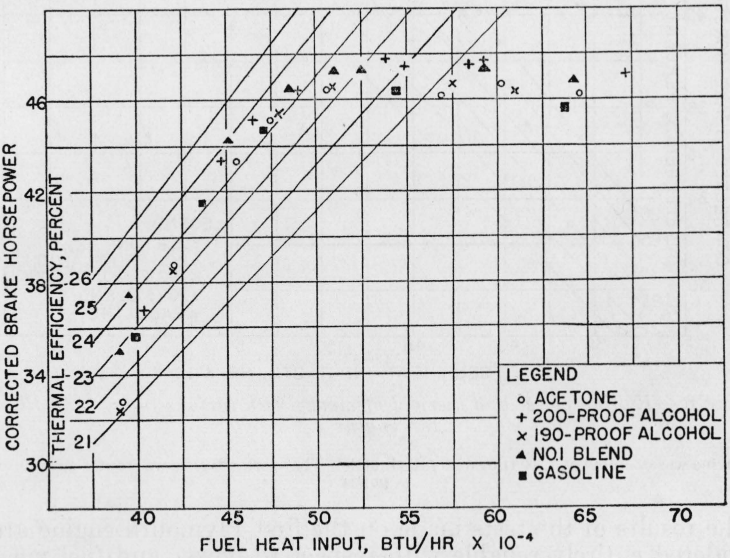

FIGURE 4.—*Engine power and thermal efficiency with various fuels, 1942 Chevrolet.*

The engine was operated at 1,500 rpm with full throttle. The spark advance was set for maximum engine power.

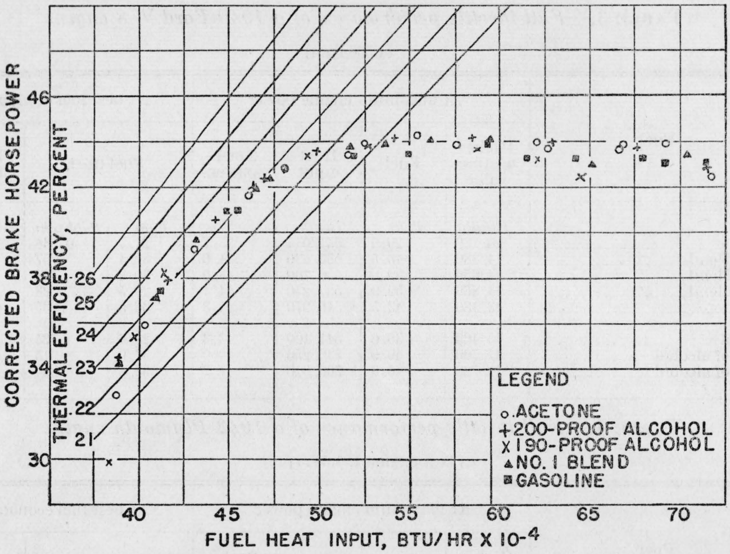

FIGURE 5.—*Engine power and thermal efficiency with various fuels, 1940 Ford V–8.*

The engine was operated at 1,500 rpm with full throttle. The spark advance was set for maximum engine power.

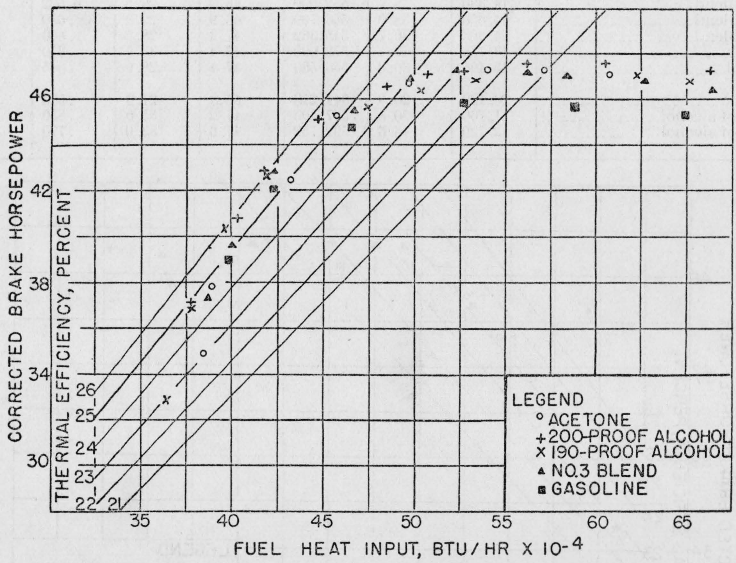

FIGURE 6.—*Engine power and thermal efficiency with various fuels, 1942 Plymouth, test engine 4.*

The engine was operated at 1,500 rpm with full throttle. The spark advance was set for maximum engine power.

The results of the tests made on the first Plymouth engine are not considered entirely reliable with respect to power and fuel measurements. The spark was retarded several degrees from that for best performance because of knock on regular gasoline, and the results of some of the tests are inconsistent. The power developed with 190- and 200-proof alcohol appeared to be too high, but check runs were not possible because of an engine breakdown, which made major repairs necessary. These data are included in the report because some of the tests, not repeated on the other engines, contain useful information.

Figures 4, 5, and 6 show the power of three of the engines tested as related to fuel heat input and thermal efficiency.

2. ROAD-LOAD POWER AND ECONOMY TESTS

The results of the road-load power and economy tests of the Plymouth engine are shown in table 5. The data were taken from curves plotted from the test results. These curves show the relation of the intake-manifold depression and specific fuel consumption to the air-fuel ratio when the engine is operated with a constant speed and load equivalent to level-road operation at a given car speed. The highest manifold depression indicates the point on the mixture-ratio curve at which the desired load and speed can be maintained with a minimum throttle opening. This condition corresponds to that of maximum power for a fixed throttle opening and constant speed. Figures 7, 8, 9, and 10 show the manifold depression, specific fuel consumption, and percentage of CO_2 with the engine operating at 40 mph with varying air-fuel ratios.

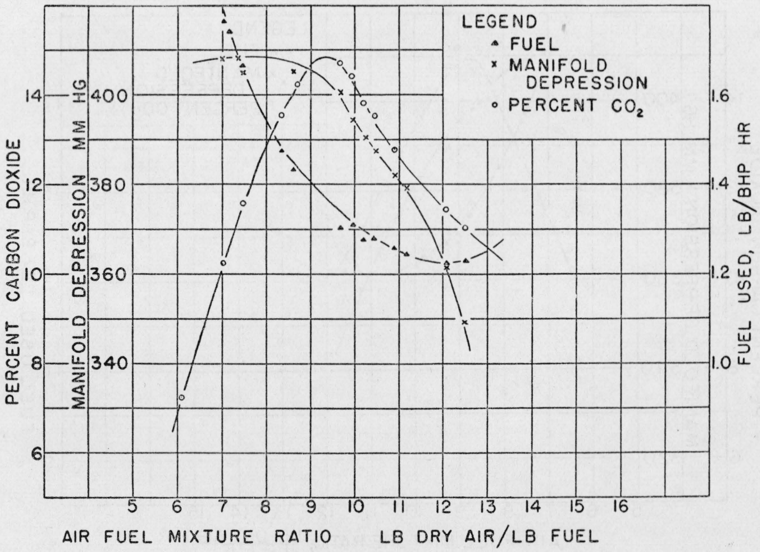

FIGURE 7.—*Manifold depression, percentage of* CO_2, *specific fuel consumption, and mixture ratio for 200-proof alcohol, 1942 Plymouth, test engine 4.*

The engine was operated at speed and load calculated as equivalent to road operation at 40 mph. The spark was set for maximum power at 1,500 rmp, full throttle.

Sixteen curves were plotted to show the road-load performance of four fuels at 20, 30, 40, and 50 mph with varying mixture ratios. The significant points on these curves were (1) the peaks of the manifold-depression curves which show the mixture ratio at which

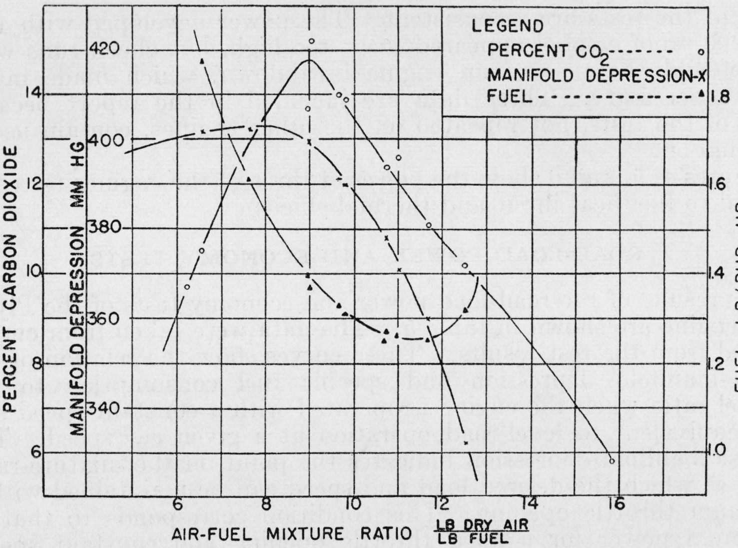

FIGURE 8.—*Manifold depression, percentage of* CO_2, *specific fuel consumption, and air-fuel ratio with 20-percent-ether blend; 1942 Plymouth, test engine 4.*

The engine was operated at speed and load calculated as equivalent to road operation at 40 mph. The spark was set for maximum power at 1,500 rmp, full throttle.

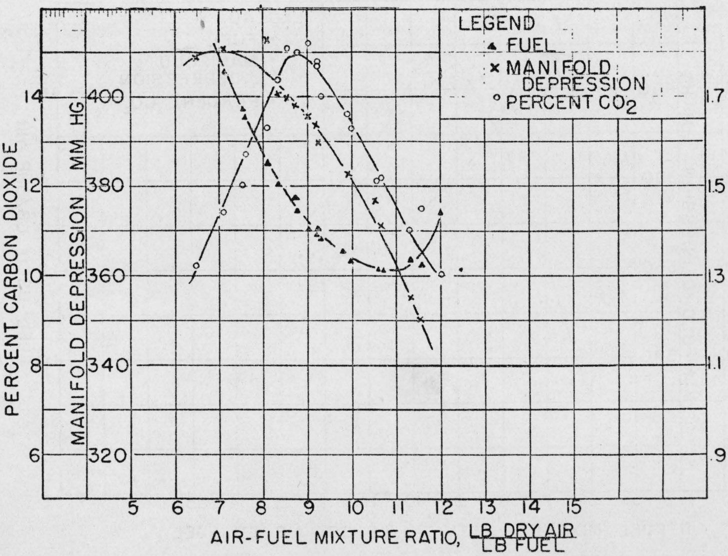

FIGURE 9.—*Manifold depression, percentage of* CO_2, *specific fuel consumption, and air-fuel ratio with 190-proof alcohol; 1942 Plymouth, test engine 4.*

The engine was operated at speed and load calculated as equivalent to road operation at 40 mph. The spark was set for maximum power at 1,500 rpm, full throttle.

the engine will hold the required speed and load with minimum throttle opening, and (2) the lowest point on the fuel-economy curve which shows the mixture ratio at which the fuel gives its maximum miles per gallon.

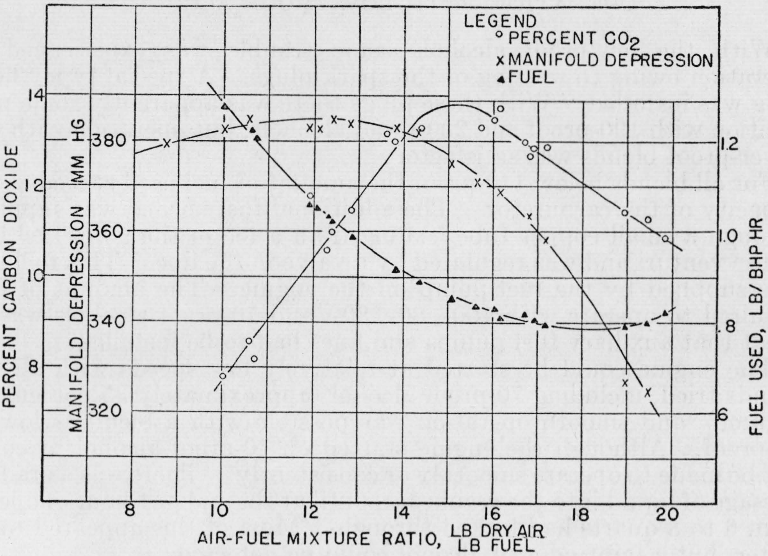

FIGURE 10.—*Manifold depression, percentage of* CO_2*, specific fuel consumption, and air-fuel ratio with gasoline; 1942 Plymouth, test engine 4.*

The engine was operated at speed and load calculated as equivalent to road operation at 40 mph. The spark was set for maximum power at 1,500 rpm, full throttle.

The fuel mileages shown in table 5 are computed from the fuel consumption for best economy and minimum throttle, as shown by the curves, assuming a fuel temperature of 77° F.

As in the full-throttle runs, the weight of fuel consumed is in inverse proportion to the heating value of fuel used and is approximately in the same direct proportion as the combining weights of the fuels with oxygen in complete combustion.

TABLE 5.—*Road-load power and economy of 1942 Plymouth engine*

[Test engine 4]

Fuel	Car speed	At best fuel economy			At minimum throttle		
		Mixture ratio	Fuel used		Mixture ratio	Fuel used	
	mph		*lb/bhp hr*	*mpg*		*lb/bhp hr*	*mpg*
Gasoline	20	[a] 16.5	1.135	26.3	12.6	1.390	21.5
Do	30	16.4	0.940	23.8	12.2	1.195	18.8
Do	40	18.0	.795	21.4	12.0	1.063	15.8
Do	50	17.8	.700	18.2	12.0	0.960	13.3
190-proof alcohol	20	10.25	1.880	18.0	7.7	2.250	15.0
Do	30	10.6	1.540	16.6	8.0	1.785	14.2
Do	40	10.65	1.320	14.5	7.4	1.685	11.4
Do	50	10.5	1.185	12.2	7.4	1.490	9.8
200-proof alcohol	20	10.5	1.730	19.2	8.2	2.040	16.4
Do	30	11.2	1.460	17.1	8.0	1.750	14.3
Do	40	[a] 12.0	1.220	15.3	7.9	1.550	12.1
Do	50	11.1	1.120	12.7	7.8	1.400	10.1
20% ether, 80% 190-proof alcohol	20	10.4	1.800	18.6	8.1	2.050	16.4
Do	30	11.35	1.465	17.2	8.3	1.730	14.5
Do	40	11.4	1.255	15.1	[a] 7.7	1.600	11.8
Do	50	11.0	1.150	12.5	8.1	1.360	10.5

[a] Curve not definite at this point.

3. OPERATION WITH LOW-PROOF ALCOHOLS

With the low-proof alcohols some trouble was experienced in operation owing to wetting of the spark plugs. A special type "hot" plug was installed. With these plugs there was apparently some preignition with 190-proof and 200-proof alcohols, but operation with the lower-proof blends was satisfactory.

For all blends below 140-proof the amount of fuel used exceeded the capacity of the carburetor. The additional fuel needed was supplied through a small copper tube leading from a tee in the main fuel line to the venturi and was regulated by a valve in the line. This fuel line was supplied by the fuel pump on the engine. The amount of fuel required to operate with 100-, 90-, 80-, and 70-proof alcohols was so great that auxiliary fuel pumps and lines had to be installed.

The engine could be started at relatively low speeds with all the blends tried, including 70-proof alcohol (approximately 35 percent by volume), and smooth operation was possible with a blend as low as 80-proof. Although the engine started on 70-proof alcohol, it could not be made to operate smoothly or consistently. There was a gradual passage of liquid into the crankcase, until at the end of 1 hour of operation, 6 to 8 quarts had leaked through. Most of this appeared to be water, but a faint odor of alcohol could be detected.

The thermocouple spark plugs could not be used with the low-proof alcohols, so the mixture distribution could not be analyzed by combustion-temperature relationships, as in the case of the other fuels. It appeared, however, that distribution was very poor with the lower-proof alcohols, possibly because of the method of supplying the additional fuel required. In some cases there was a definite increase in power and smoother engine operation when the auxiliary fuel-supply tube was moved to a slightly different position in the venturi. Table 6 is a summary of the test results with respect to power, fuel consumption, and spark advance.

TABLE 6.—*Summary of tests of low-proof alcohols of various proofs*

[Test engine 4. 1942 Plymouth. 1,500 rpm, full throttle]

Approximate proof of alcohol	Maximum power conditions			Gallons per hour of blend	Thermal efficiency	Optimum spark advance
	Brake horsepower	Ethanol	Ethanol			
		lb/hr	*lb/bhp hr*			*Degrees*
200	47.67	45.0	0.944	6.85	21.2	17.5
190	46.18	47.5	1.029	7.53	19.4	20.4
180	45.67	47.5	1.042	7.94	19.2	21.3
160	45.07	50.8	1.127	9.52	17.8	27
140	45.48	52.83	1.162	11.33	17.2	29
120	43.60	52.63	1.207	12.90	16.6	33
110	42.0	53.66	1.278	14.53	15.6	35
100	42.94	63.98	1.490	19.0	13.4	36
90	42.35	54.06	1.277	17.9	15.6	41.5
80	41.40	55.5	1.341	20.87	14.9	47
70	34.1	63.0	1.853	26.7	10.8	50(?)

The volumes of lower-proof alcohols used by an engine are so large that the fuel pump and carburetor would have to be completely redesigned for normal operation. Fuel tanks would have to be made larger. For example, a tank that would carry a vehicle 200 miles on gasoline would carry it about 135 miles on anhydrous alcohol (200-proof). The mileage on 120-proof would be 60, that on 90-proof, 40, whereas on 70 proof, the mileage would be less than 25. It is probably that there would be considerable trouble in operation due to wetting of the spark plugs.

There is progressive loss in thermal efficiency as the proof is lowered. Some of this undoubtedly is due to poor distribution, which could be corrected, but a considerable part of it is due to latent heat of vaporization of the water in the blend.

The use of low-proof alcohols as motor fuels is not considered practical whenever facilities for redistillation are available.

4. EXHAUST-GAS ANALYSIS IN PART-THROTTLE OPERATION

The percentage of CO_2 reached its maximum with all the fuels used in the part-throttle (road-load) tests near the stoichiometric air-fuel ratio. A slight excess of air was indicated at the peak of the curve, but this was less than the range of error in reading of the instruments.

The maximum percentage of CO_2 was slightly lower with gasoline than it was when the engine was operated on the alcohol fuels. This may be attributed to at least two possible causes. The theoretical maximum percentage of CO_2 is 15.0 with alcohol and 14.9 with gasoline. The alcohol molecule is smaller than that of most of the hydrocarbons that make up gasoline, and it may be possible that there is more complete combustion because the chemical changes that take place are less complex.

D'Alleva and Lovell [3] found that the maximum percentage of CO_2 with gasoline generally occurred with a mixture ratio of about 14.7, which is approximately the stoichiometric mixture. However, even with leaner mixtures, there was a small amount of carbon monoxide, and with richer mixtures there was a small amount of oxygen. This may be partly attributed to imperfect distribution and partly to the mechanics of combustion.

According to Gerrish and Voss [4], the carbon monoxide and oxygen may vary from 0 to 9.0 percent. They cite one reference [5] that question the presence of CO in lean mixtures and of oxygen in rich mixtures. Other references [3 and 6] note that there are at least small amounts of both present with any mixture ratio. A study of various types of exhaust-gas analyzers was made by Dilworth in 1941 [7].

Although the theoretical maximum percentage of CO_2 is about 14.9 with gasoline, the curve flattens out at a lower point at the top, probably because different cylinders have different individual mixture ratios at any given time, and the engine as a whole never operates at the stoichiometric ratio.

The actual content of CO_2 varies from about 9 to about 14 percent, and seems to be independent of the engine speed except in cases accounted for by poor distribution. The maximum percentage of CO_2 with 190-proof alcohol when operating at 20 mph was 12.7 and at 50 mph was 14.7. With gasoline the maximum percentage of CO_2 was about the same in both cases—13.7 percent. The distribution with gasoline was good at both speeds; with alcohol the distribution was good at 50 mph but was very poor at 20 mph.

It was indicated by the literature mentioned that at lean or rich mixtures the air-fuel ratio may ordinarily be estimated with reasonable accuracy, if the percentage of one or more of the constituents is known. This view is supported by the results of the CO_2 analysis in the current series of tests.

Table 7 shows computed data pertinent to the exhaust-gas analysis of the fuels used in these tests. Table 8 is a summary of the results of the CO_2 analysis.

TABLE 7.—*Computed mixture ratios, percentage of* CO_2 *for various fuels*

Fuel	Stoichiometric mixture ratio	Theoretical maximum percentage of CO_2			
		Excess air 0%	Excess air 10%	Excess air 20%	Excess air 33%
190-proof alcohol	8.32	15.0	13.5	12.3	11.0
200-proof alcohol	9.00	15.0	13.5	12.3	11.0
20% ether blend	8.90	15.0	13.5	12.3	11.0
Gasoline	14.7 to 14.8	14.9	13.4	12.2	11.0

TABLE 8.—*Maximum percentage of* CO_2 *in part-throttle operation of 1942 Plymouth*

[Test engine 4]

Speed	190-proof alcohol	200-proof alcohol	20-percent ether blend	Gasoline
mph				
20	12.7	13.5	13.5	13.7
30	14.4	14.4	14.6	14.0
40	15.0	14.7	14.7	13.7
50	14.7	14.3	14.0	13.6

5. COMBUSTION TEMPERATURES AND DISTRIBUTION

The general trends shown by the spark-plug thermocouple temperatures indicate that the mixture distribution is not as good with alcohol as with gasoline. The highest average temperatures occurred when the mixture ratio was near the point at which the engine develops its maximum power. However, there was a considerable spread in the mixture ratios (considering the engine as a whole) at which the highest temperatures occurred in different cylinders. This varied with different fuels and engines, but was always less with gasoline than with any of the other fuels used. Rabezzana and Kalmar in summarizing the results of a series of tests made with this type of spark plug draw the following conclusions:

"1. The spark-plug temperature is directly proportional to the mean combustion temperature and is a function of the mixture ratio.

"2. Variations in temperature are large enough to warrant its use as an index of the mixture ratio.

"3. The location of the peak of the temperature curve has a definite relation to the peak of the power curve."

The results of the current series of tests tend to verify these conclusions in every respect. The analysis of the combustion-temperature-distribution relationship is based on the assumption that the maximum thermocouple temperatures occur at equal values of the mixture ratio in each of the cylinders. A cylinder that is running richer than the

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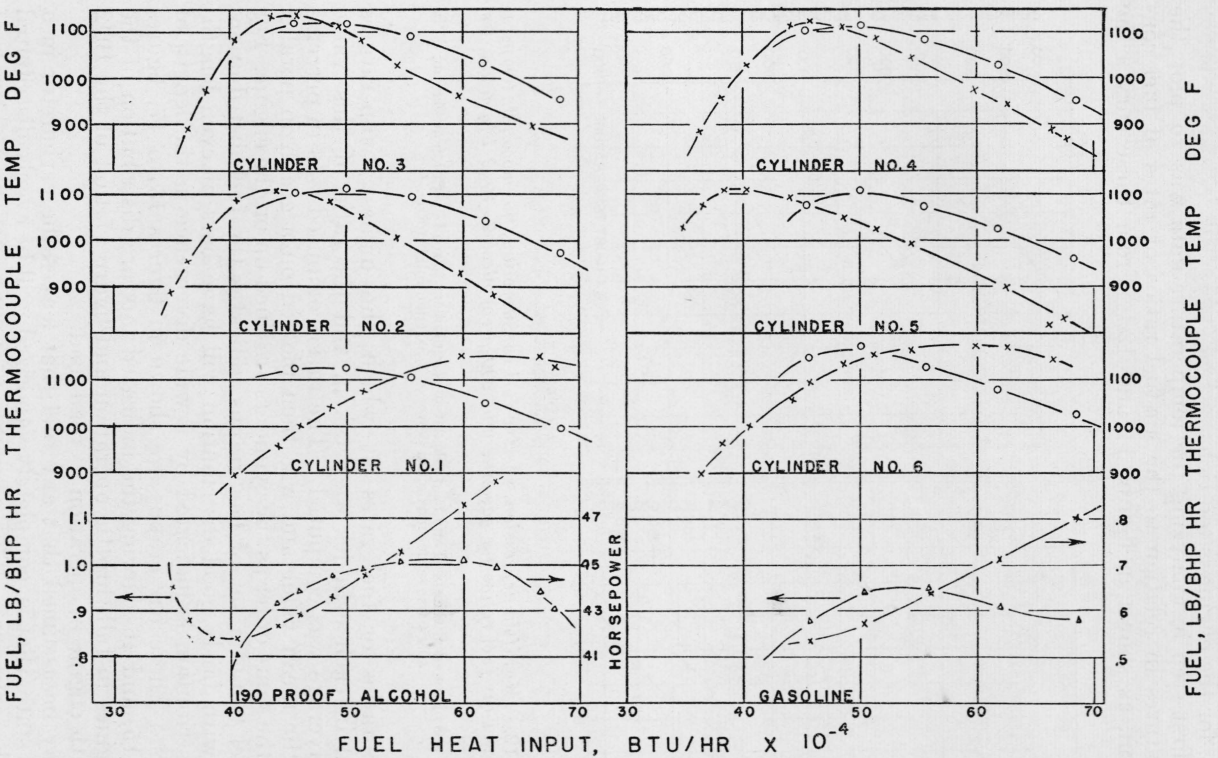

FIGURE 11.—*Power, specific fuel consumption, thermocouple temperatures, and fuel heat input for gasoline and 190-proof alcohol, 1942 Plymouth, test engine 4.*

The engine was operated at 1,500 rpm, full throttle, with spark setting for maximum power. Circles in graphs of cylinder temperatures indicate gasoline, crosses indicate alcohol.

others will reach its temperature peak when the mixture ratio for the engine as a whole is too lean. Cylinders that receive less than their fair proportion of fuel will reach their highest temperatures when the mixture for the engine is too rich.

The first step in analyzing the temperature data was to plot the temperatures in relation to the air-fuel ratios or rates of fuel consumption, as shown in figures 11 and 12. From these curves the

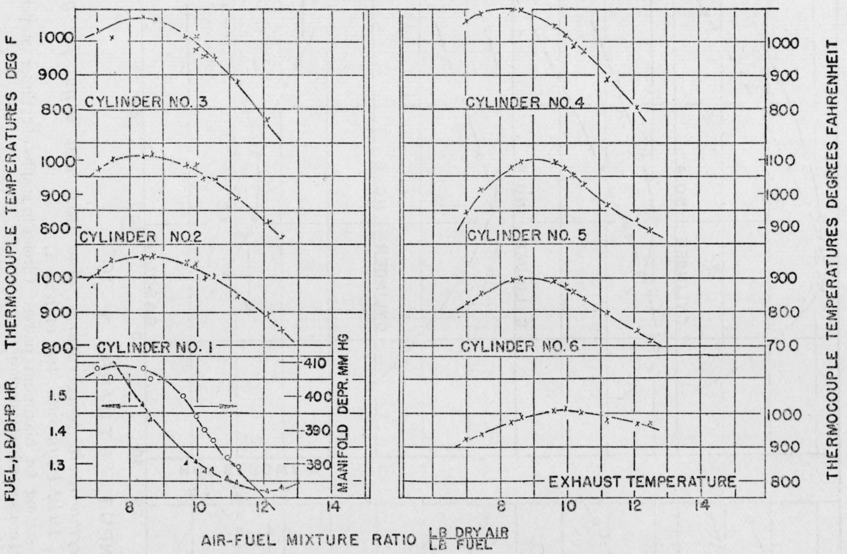

FIGURE 12.—*Manifold depression, specific fuel consumption, thermocouple temperatures, and air-fuel ratios for gasoline and 200-proof alcohol, 1942 Plymouth, test engine 4.*

The engine was operated at the speed and load calculated as equivalent to road operation at 40 mph. The spark was set for maximum power at 1,500 rpm, full throttle.

mixture ratios or fuel rates at which the highest temperatures occurred in each cylinder were taken, and the ratio of this value to the average was computed. The ratio computed was in percentage of the mean value and was then plotted on a graph in parallel bars in the same order as the cylinders are lined up on the engine. For each fuel tested a set of bar graphs indicates the fuel distribution. A fuel with fairly good distribution will have bars of even length; poor distribution is indicated by a wide divergence in the length of the bars. These bar graphs are shown in figures 13 to 17, figures 13, 14, 15, and 16 showing the indicated mixture distribution of the four engines in full-throttle operation and figure 17 that of the 1942 Plymouth engine in operation at road load.

It may be assumed that any fuel that leaves the carburetor in a gaseous state, that is, completely vaporized, will be evenly distributed to all the cylinders. This assumption is supported by the work of Rabezzana and Kalmar [2] on gaseous fuels, using the same type of spark plugs as were used in these tests. With liquid fuels this is true only when the manifold is so designed that each cylinder receives the same amount of unvaporized fuel. Gasoline may be no more than

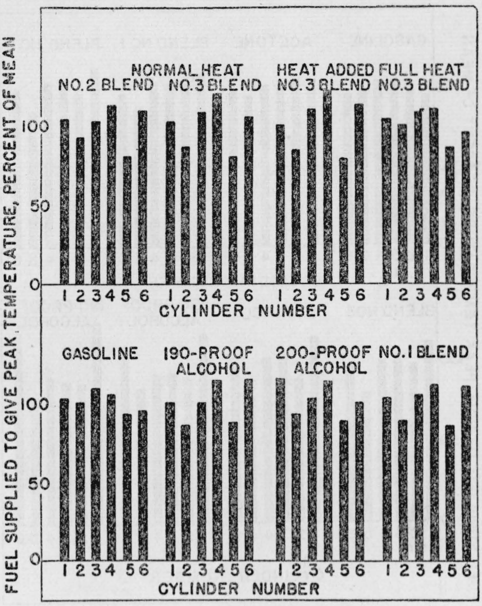

FIGURE 13.—*Mixture distribution with various fuels, 1942 Plymouth, test engine 1*

The engine was operated at 1,500 rpm with full throttle. The spark advance was set for maximum power.

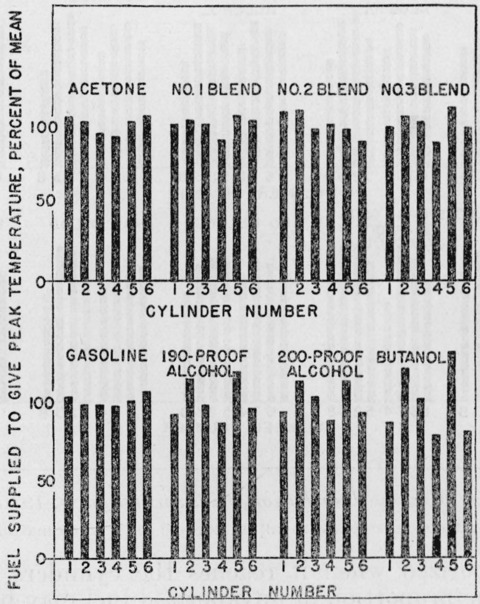

FIGURE 14.—*Mixture distribution with various fuels, 1942 Chevrolet.*

The engine was operated at 1,500 rpm with full throttle. The spark advance was set for maximum power.

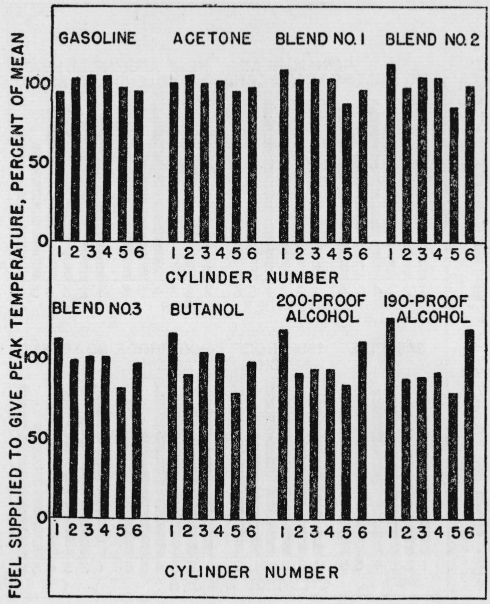

FIGURE 15.—*Mixture distribution with various fuels, 1942 Plymouth, test engine 4.*

The engine was operated at 1,500 rpm with full throttle and spark set for maximum engine power.

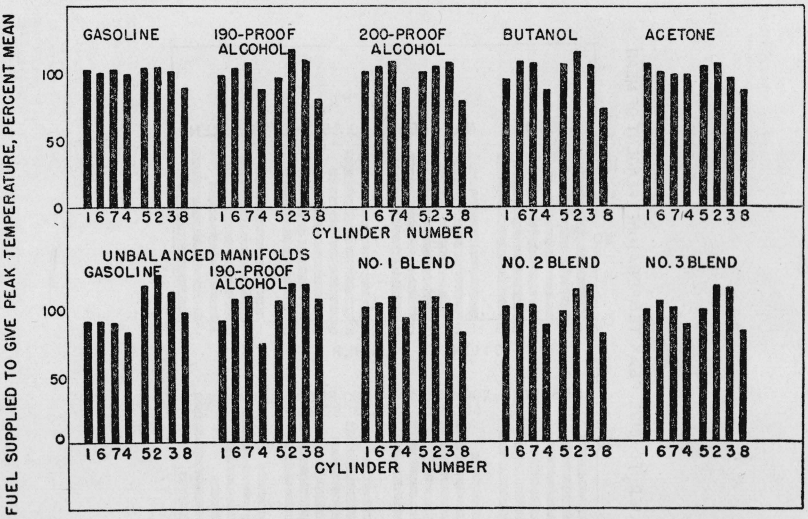

FIGURE 16.—*Mixture distribution, with various fuels, 1940 Ford V-8.*

The engine was operated at 1,500 rpm with full throttle and spark set for maximum engine power.

50 percent vaporized when it reaches the cylinders, and with less volatile fuels the proportion of unvaporized fuel may be even greater. The vaporization takes place throughout the intake manifold and in the cylinder itself during the intake and compression strokes. The rate at which a fuel becomes vaporized depends on its volatility,

boiling point, latent heat of vaporization, and the operating conditions of the engine.

The viscosity and surface tension of a fuel help to determine the path it travels in the manifold. Some of the temperature curves indicate that some of the cylinders received too great a proportion of the fuel added in enrichening a mixture.

The manifold of an automotive engine is built for operation with gasoline. It is to be expected that the other fuels with different physical characteristics would be distributed less uniformly than gasoline. However, there is no reason to believe that it is not possible to build a manifold for any one of these fuels that would distribute the fuel about as well as gasoline is distributed by the present manifold.

It is apparent that several factors may affect fuel distribution.

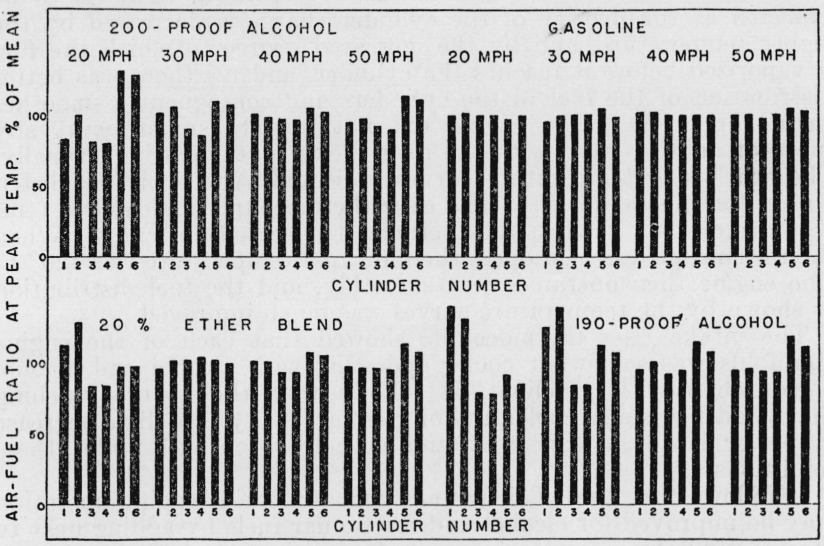

FIGURE 17.—*Mixture ratio distribution with various fuels, 1942 Plymouth, test engine 4.*

The engine was operated at speeds and loads calculated as equivalent to road operation at 20, 30, 40, or 50 mph. The spark was set for maximum power at 1,500 rpm full throttle.

Volatility, viscosity, latent heat of vaporization, and the amount of fuel used are the important factors, but the manner in which they operate is not always consistent. With the Chevrolet engine the end cylinders run slightly leaner than those nearer the carburetor when gasoline is used. With 190- and 200-proof alcohol the end cylinders run richer than the others. With the 1942 Plymouth engine, No. 4, the opposite is true, the middle cylinders being richer with alcohol and leaner with gasoline than are the end cylinders.

Usually, though not always, faults in distribution with gasoline are accentuated when a less volatile fuel is used. In figure 16 it is shown that cylinders No. 4 and No. 8 are somewhat too rich when the Ford engine was operated on gasoline. When the alcohol fuels and blends were used this tendency increased.

With the Ford engine some tests were made in which the dual carburetion system was purposely unbalanced to determine the effect this would have on the thermocouple temperatures. The cylinders in one manifold group received up to 20 percent more fuel than those in the other group. The temperature peaks of the cylinders in the rich manifold occurred when the other manifold was much too lean for its best performance (see fig. 16).

If vaporization of a fuel improves its distribution by the manifold, any change in operating conditions that increases the rate of vaporization should improve the distribution. The effect of increasing the manifold temperature was studied in tests made with the first Plymouth engine (No. 1).

In some tests of the No. 3 Blend the intake riser thermostat was allowed to operate normally, while in others the thermostat was held open for added heat. The addition of heat had two definite effects: (1) the power of the engine was slightly reduced. This is to be expected as the density of the cylinder charge is decreased by the higher temperature, and by the increased degree to which the fuel is vaporized before it reaches the cylinder, and (2) there was better distribution of the fuel to the cylinders and consequently smoother engine operation. The engine did not operate smoothly at any speed or mixture ratio with the riser thermostat operating normally. The spark-plug temperature curves indicated very poor fuel distribution for this condition. The riser thermocouple indicated a temperature of 193° F when the thermostat operated normally. When the thermostat was held open, the indicated temperature was 420° F. The engine then operated very smoothly, and the fuel distribution as shown by the temperature curves was much improved.

The intake riser thermocouple showed that each of the engine manifolds was somewhat cooler operating with alcohol and alcohol blends than with gasoline. It follows that the charge reaching the cylinder was also cooler and of greater density. A slight increase in power is to be expected (and actually occurred) under these conditions.

It follows that the efficiency and smoothness of engine operation may be improved for alcohol and for similar fuels by adding heat to the manifold for a greater degree of vaporization, but this cannot be accomplished without reducing the maximum power of the engine.

The effect of poor distribution on power and fuel consumption may best be analyzed if the cylinders are considered as independent units, each using its share of the fuel and developing part of the total power of the engine. Tests made on a single-cylinder engine have shown that when the fuel supplied is varied by 10 percent or less from the amount required for maximum power, the power varies 1.1 percent or less from the maximum. Within these limits the effect on engine power of imperfect distribution is less pronounced than that on fuel consumption. A change of 20 percent in the mixture ratio of a cylinder (10 percent rich to 10 percent lean) may be accompanied by a power change of 1 percent or less. With leaner mixtures there may be a sharp drop in engine power without a substantial change in the specific fuel consumption (see fig. 1). The power falls away more sharply with lean than with rich mixtures. For this reason the maximum-power air-fuel ratio always is richer for a multicylinder engine than for a single-cylinder engine operating under similar conditions (see fig. 18). The cylinders that run lean gain more when the mixture is enriched than the other cylinders lose.

A detailed mathematical analysis of the relationship of fuel distribution to power and fuel consumption has been given by Sparrow [8].

The apparent distribution of fuel shown by figures 13 to 16 of this paper may be used to calculate the approximate loss in power due to imperfect distribution, if it is assumed that each cylinder receives its same proportion of fuel throughout the range of air-fuel ratios. It is probable that there is some change in this proportion with changes in air-fuel ratios, but the change is not believed to be large enough to prevent the use of the apparent distribution of the fuel in making estimates of the effects of poor manifolding.

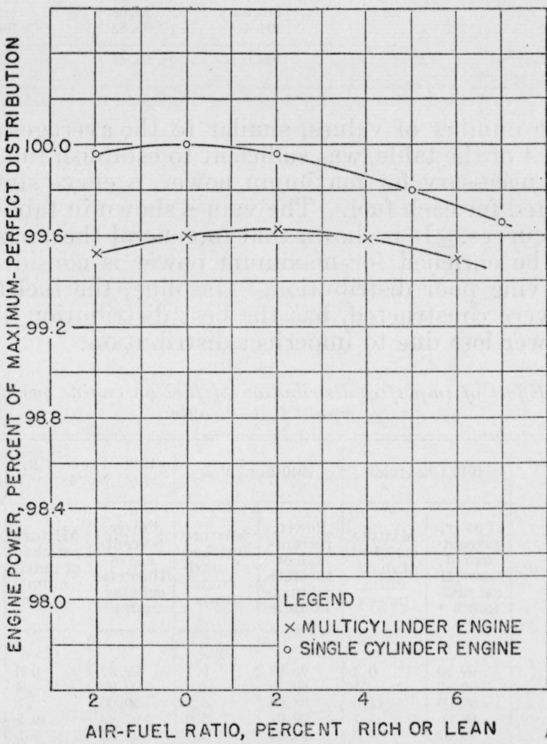

FIGURE 18.—*Effect of imperfect distribution on engine power, 1942 Chevrolet engine, No. 3 Blend.*

The graph is based on an average of the computed power of each of the cylinders in percentage of the theoretical maximum, as shown by tests of a single-cylinder engine. Distribution of fuel is based on figure 14.

A curve was plotted showing the relation of power to fuel consumption in tests of a single-cylinder engine, as shown in figure 18 of reference [1]. This curve was used in estimating the power loss due to imperfect distribution of fuel, as shown by the charts, figures 13 to 16. The percentage of the average amount of fuel received by each cylinder was tabulated, and the power of each cylinder was estimated in percentage of the theoretical maximum, first for the operating condition in which the air-fuel ratio for the whole engine was the same at which a single cylinder developed maximum power, then for various percentages rich or lean, as shown in table 9.

TABLE 9.—*Relation of power to distribution of fuel*

Cylinder number	Maximum-power mixture ratio: Fuel supplied, percentage of average	Maximum-power mixture ratio: Single-cylinder power, percentage of potential maximum	Fuel-supply increased 2 percent: Fuel supplied, percentage of requirement for maximum power	Fuel-supply increased 2 percent: Single-cylinder power, percentage of potential maximum
1	96.2	99.88	98.1	99.97
2	100.1	100.00	102.1	99.95
3	92.7	99.48	94.6	99.73
4	90.1	98.93	91.9	99.37
5	116.6	98.68	118.9	98.35
6	104.4	99.83	106.5	99.68
Average	100.0	99.47	102.0	99.51

When the number of values, similar to the averages shown in columns 2 and 4 of the table, was sufficient to establish the percentage of enrichment necessary for maximum power, a curve similar to figure 18 was plotted for each fuel. The values shown in table 10 are taken from these curves. It is shown that in general the amount the mixture must be enriched for maximum power is considerably greater for fuels having poor distribution. Gasoline, the fuel for which the manifolds were constructed, has the best distribution and shows the smallest power loss due to imperfect distribution.

TABLE 10.—*Effect of imperfect distribution of fuel on engine performance at 1,500 rpm, full throttle*

Fuel	1942 Chevrolet: Power, percentage of theoretical maximum [a]	1942 Chevrolet: Mixture enriched at maximum power	1940 Ford V-8: Power, percentage of theoretical maximum [a]	1940 Ford V-8: Mixture enriched at maximum power	1942 Plymouth, No. 4: Power, percentage of theoretical maximum [a]	1942 Plymouth, No. 4: Mixture enriched at maximum power	1942 Plymouth, No. 1: Power, percentage of theoretical maximum [a]	1942 Plymouth, No. 1: Mixture enriched at maximum power
		%[b]		%[b]		%[b]		%[b]
Gasoline	99.90	0.2	99.83	0.7	99.82	0.6	99.84	0.6
Acetone	99.80	.4	99.70	1.1	99.92	.4	----	----
Butanol	97.00	11.0	98.60	6.7	99.00	6.7	----	----
190-proof alcohol	98.86	5.7	99.07	5.0	97.83	10.7	99.08	5.1
200-proof alcohol	99.02	4.0	99.34	3.7	98.86	5.5	99.29	4.2
No. 1 Blend	99.79	1.0	99.51	3.1	99.69	2.1	99.26	3.7
No. 2 Blend	99.62	2.3	99.08	5.1	99.52	2.4	99.19	4.6
No. 3 Blend, intake riser normal	99.63	1.9	99.14	5.0	99.39	2.8	98.69	6.5
No. 3 Blend, heat added	----	----	----	----	----	----	98.55	7.8
No. 3 Blend, full heat on manifold	----	----	----	----	----	----	99.52	2.5

[a] Maximum power based on perfect distribution and tests of a single-cylinder engine.
[b] Percentage enriched above maximum power mixture for single-cylinder engine.

6. FULL-THROTTLE AIR-FLOW MEASUREMENTS AND MIXTURE RATIOS

The air-flow indicator was not available for use in the full-throttle power and economy tests. However, for the 1942 Plymouth, test engine 4, the air flow could be estimated with reasonable accuracy from measurements made in later tests. With gasoline, 190-proof alcohol, 200-proof alcohol, and the 20-percent ether blend enough measurements were made to plot curves over the whole range of mixture ratios. For the other fuels similar curves were plotted from air flow estimated by use of available data on gasoline and alcohol under comparable conditions.

The volume of air used by an engine operating at a constant speed and throttle opening varied with different fuels and mixture ratios. The volume reached a minimum when the engine was developing its maximum power and increased up to 3 percent with lean- or rich-mixture ratios. This variation presumably results from the fact that the residual gas and cylinder surfaces are at a maximum temperature when the mixture ratio is that for maximum power. Dilution of the incoming charge with hot residual gas, and heating from the combustion chamber surfaces reduces the amount of charge that can be inducted. With alcohol about 2 percent more air was used than with

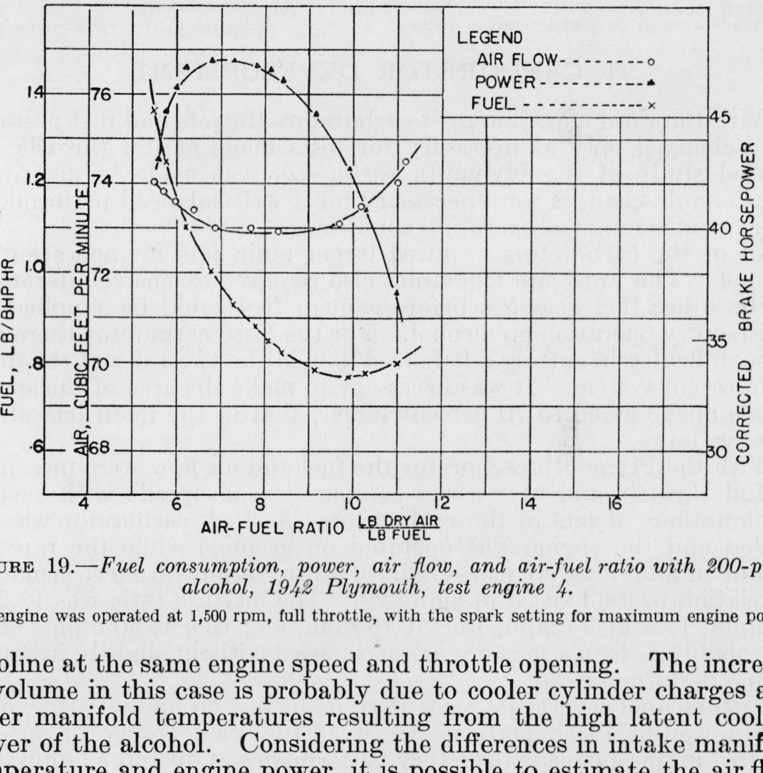

FIGURE 19.—*Fuel consumption, power, air flow, and air-fuel ratio with 200-proof alcohol, 1942 Plymouth, test engine 4.*

The engine was operated at 1,500 rpm, full throttle, with the spark setting for maximum engine power

gasoline at the same engine speed and throttle opening. The increase in volume in this case is probably due to cooler cylinder charges and lower manifold temperatures resulting from the high latent cooling power of the alcohol. Considering the differences in intake manifold temperature and engine power, it is possible to estimate the air flow in full-throttle operation, by comparison of available data on a fuel with data on fuels where the air flow was measured. Figure 19 shows the engine power, fuel consumption, and air flow plotted in relation to the air-fuel ratios when 190-proof alcohol was used at full throttle. Table 11 is a comparison of the performance of the Plymouth engine with various fuels, as shown by curves based on estimated air flow and the measured fuel consumption at 1,500 rpm with full throttle.

TABLE 11.—*Relation of heating values of fuels to mixture ratios and fuel consumption*

[1942 Plymouth. Test engine 4. 1,500 rpm, full-throttle]

Fuel	Higher heat of combustion	Percentage of heat for gasoline	Mixture ratio at maximum power [b]	Percentage of gasoline maximum power ratio	Mixture ratio at best economy	Percentage of gasoline best economy ratio	Specific fuel consumption		
							lb/bhp hr	Inverse ratio to gasoline used	Direct ratio to gasoline used
	Btu/lb [a]								
No. 1 blend	14,380	70.8	8.4	71.2	11.0	72.3	0.692	72.3	138
No. 2 blend	14,650	72.2	8.8	74.6	11.3	74.3	.680	73.5	136
No. 3 blend	14,860	73.2	9.0	76.3	11.5	75.7	.660	75.8	132
Acetone	13,180	64.9	7.8	66.1	9.7	63.8	.770	64.9	154
Butanol	15,460	76.2	9.0	76.3	11.8	77.6	.645	77.5	129
190-proof alcohol	11,760	57.9	6.6	55.9	9.3	61.2	.830	60.2	166
200-proof alcohol	12,720	62.6	7.0	59.3	9.8	64.5	.770	64.9	154
Gasoline	20,300	100	11.8	100	15.2	100	.500	100	100

[a] Higher value as determined by Jessups, National Bureau of Standards, 1943–44.
[b] Pounds of dry air per pound of fuel.

VI. CARBURETOR DEVELOPMENT

With the Ford and Chevrolet carburetors the jets and fuel passages were changed only as necessary for tests made at full throttle. A limited study of the Plymouth carburetor was made to determine what simple changes were necessary for a well-balanced performance over a wider range of operating conditions.

All of the carburetors required larger main jets for operation on alcohol. The Ford and Chevrolet also required extensive alterations in the other fuel passages before enough fuel could be supplied for satisfactory operation on alcohol. For the Ford carburetor there was some difficulty in getting a balanced flow in the two sides of the dual-carburetion system. It was necessary to make the area of the spray-nozzle openings 60 to 70 percent larger, that of the main jets about twice as large.

With the Plymouth carburetor the fuel and air flow were measured at full throttle and at various computed road speeds with several combinations of jets in the carburetor. A stock carburetor was installed and the engine was operated on gasoline while the rates of flow of air and fuel were measured. The mixture-ratio curve is plotted in relation to road speed in figure 20. The mixture ratio was 12.2 at 20 mph, 14.6 at 30 mph, 15.5 at 40 mph, and 15.6 at 50 mph. This is probably as lean a mixture as can be used without slightly irregular engine performance.

With alcohol several jet sizes were used, but no one jet size would give a well-balanced performance over the whole range of speeds. A 0.075-in. jet gave a satisfactory performance at 40 and 50 mph, but was so lean at lower speeds that it was difficult to keep the engine running. An 0.089-in. jet was satisfactory at full throttle, but gave too rich a mixture at all part-throttle settings, except those at which the idle jet furnished the major part of the fuel supplied. An 0.083-in. jet operated fairly well at all speeds, but was rich at high road speeds, part load, and lean at low speeds and at full throttle. When the idle jet was enlarged in proportion to the main jet, the fuel flow was better balanced but was rich, except at or near full load when the 0.083-in. jet was used. It was indicated that a 0.075-in. jet with an idle tube

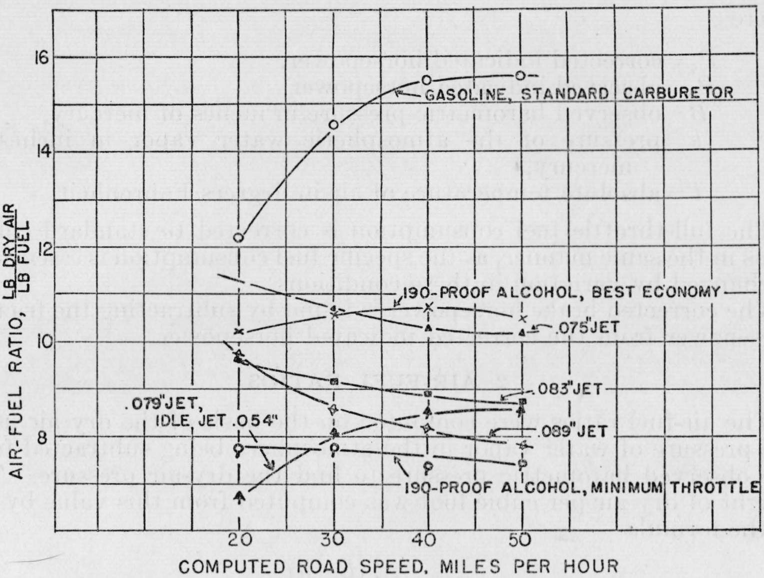

FIGURE 20.—*Carburetor performance and jet sizes with gasoline and 190-proof alcohol, 1942 Plymouth, test engine No. 4.*

The engine was operated at speeds and loads calculated as equivalent to road operation at 20, 30, 40 and 50 mph. The spark was set for maximum power at 1,500 rpm, full throttle.

about 50 percent larger would give an acceptable performance with 190-proof alcohol.

It is clear that the use of a gasoline carburetor for operation with alcohol involves redesign of all basic parts of the carburetor to produce a fuel flow that is proportionately increased in all the basic parts. Enlarging the main jets will not help materially at speeds at which most of the fuel is supplied through the idle passages. Changes of the idle passages will not affect high-speed operation, and the economizer assembly must be altered before the full-load performance will be satisfactory.

VII. METHODS OF COMPUTING DATA

1. FULL-THROTTLE POWER AND FUEL CONSUMPTION

Extensive studies previously made at the National Bureau of Standards [9, 10] have shown that the fuel consumption and indicated horsepower (sum of observed brake and friction horsepower) of an engine operating at constant speed with full throttle vary inversely as the square root of the absolute temperature of the air, and vary directly as the dry-air pressure. In correcting the data, 59° F and 29.53 in. of mercury are taken as standard conditions, and the full-throttle power and fuel consumption are corrected to these conditions by multiplying the observed values by the atmospheric correction factor, as shown by the formula

$$P_c = P_0 \frac{29.53}{B-h}\sqrt{\frac{T}{486.4}},$$

where

P_c=corrected indicated horsepower,
P_0=observed indicated horsepower,
B=observed barometric pressure in inches of mercury,
h=pressure of the atmospheric water vapor in inches of mercury,
T=absolute temperature of air in degrees Fahrenheit.

The full-throttle fuel consumption is corrected to standard conditions in the same manner, as the specific fuel consumption is essentially unchanged by variation in these conditions.

The corrected brake horsepower is found by subtracting the friction horsepower from the corrected indicated horsepower.

2. AIR-FUEL RATIOS

The air-fuel ratios were computed on the basis of the dry air used, the pressure of water vapor in the atmosphere being subtracted from the observed barometric pressure to find the dry-air pressure. The weight of dry air per cubic foot was computed from this value by use of the formula

$$W_1 = W_2 \frac{(B-h)}{29.92},$$

where

W_1=dry air, lb/ft ^{3}, at existing atmospheric conditions,
W_2=dry air, lb/ft ^{3}, at 29.92-in. pressure, existing temperature.

With the weight of air computed on this basis, the mixture ratio was computed as follows:

$$\text{Air, lb/hr} = \text{ft}^3/\text{hr} \times W_1.$$

$$\text{Air-fuel mixture ratio} = \frac{\text{Dry air, lb/hr}}{\text{Fuel, lb/hr}}.$$

3. DYNAMOMETER LOAD FOR ROAD-LOAD OPERATION

The load imposed on an automobile engine when propelling a car on a level road is composed of frictional resistance in the power-transmission system, including the tires, and of air resistance to the motion of the car and the rotation of its wheels.

An unpublished study made by Donald B. Brooks, of this Bureau, of the frictional resistance of an automobile shows that the total resistance in level-road operation can be approximated closely by the formula

$$P = 0.00000427 As^3 + 0.000000333 Ws^2 = 0.0000333 Ws,$$

in which

P=horsepower required to propel car,
A=projected frontal area of car, ft^2,
W=weight of car, lb,
s=car speed, mph.

Although no formula can represent all cars accurately, because of the differences in the coefficients of air resistance, this formula is a sufficiently good approximation to forecast maximum road speeds within 2 miles per hour in the majority of cases when used in conjunction with the engine-power curve.

In computing the road load for the 1942 Plymouth engine, a total loaded car weight of 3,860 lb was used. The computed engine speeds used for road speeds of 20, 30, 40, and 50 mph were respectively 935, 1,405, 1,870, and 2,340 rpm.

VIII. CONCLUSIONS

Engines operating on substitute fuels may develop from 2 to 4 percent more power than with gasoline, but require from 60 to 70 percent more fuel. The amount of any substitute fuel required is inversely proportional to its heat of combustion. The performance of automobile engines on alcohol fuels can be improved by use of a manifold designed specifically for better distribution of these fuels.

The mixture distribution is somewhat less uniform with substitute fuels containing ethyl or normal butyl alcohol than with gasoline. Distribution of highly volatile fuels is generally better than that of those of low volatility, such as butanol or low-proof ethyl alcohol.

The use of low-proof alcohols as motor fuels is impractical because of the excessive volumes of fuel required and other factors. Extensive changes in the fuel supply system would be necessary for normal operation.

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