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Final Report

STRATIFIED AND LEAN FUEL-AIR RATIO ENGINES AND THEIR POSSIBILITIES

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ABSTRACT

This report compares hybrid engines with conventional spark-ignition carbureted engines to determine any possible gains in fuel economy. Two combustion systems, the stratified-charge and the lean fuel-air ratio engines, are examined, and ideal cycle estimates for these engines are compared with those for the standard L-141 engine. These estimates are then converted to net BHP output for a practical engine comparison.

I. PURPOSE

In this study we investigated the possibilities of improving the cycle efficiency of gasoline engines operating at part loads by eliminating the throttling losses of the conventional engine. The fuel supply was reduced; the air supply remained about constant at all speeds.

For many years experimental work on gasoline engine efficiency has been done by various authorities and research organizations; however, to date no satisfactory method of increasing engine efficiency has been found. In the past few years one or two systems have shown promise, but developing them into acceptable engines requires much additional work and expense. In this study the theoretical possibilities of engine control systems were carefully analyzed to obtain an ideal yardstick of ultimate possibilities, and to justify the expenditure of additional funds where necessary.

II. INTRODUCTION

The overall combustion process in mixtures of gasoline and air in spark-ignition engines is well understood today. In this process the mixture ratio is maintained within well defined limits to produce the flame velocity needed to complete combustion in the time available for the desired cylinder size.

If the fuel supply in an engine is reduced while the air supply remains constant, the mixture becomes lean and the flame velocity is lowered. Eventually the mixture will become too lean to ignite. Therefore, in practical operation the load of the conventional spark-ignition engine is regulated by varying the mass of the mixture contained in the cylinder rather than by varying the quality of the mixture. Maintaining a relatively constant ratio of fuel to air keeps the combustion rate at acceptable levels.

Controlling an engine by varying the mass of the mixture produces considerable negative work at part load during the inlet phase of the cycle, and the more or less constant high temperature of the combustion process in the engine tends to keep heat losses high. Both of these factors reduce engine efficiency. Researchers have attempted to overcome these disadvantages by using lean fuel-air ratios and adjusting conditions in the cylinder for their successful combustion. The problem has received sporadic attention since about 1920, and there have been some notable advances, such as the well known Texaco combustion process.

Examination of the present work supported by A.T.A.C. together with other information reveals that two basic methods of approaching this problem are in use today. These methods, which blend slightly in some of the proposed designs, are described below.

A. LEAN FUEL-AIR RATIO ENGINE

A variable fuel-air ratio from rich to lean is used to regulate the load. The mixture is relatively uniform, filling the cylinder at all loads without throttling.

To date no method for achieving combustion as the mixture passes out of the normal ignitable ratios has proved satisfactory. One possible ignition method uses the combustion of a small quantity of rich mixture in a separate cell, or equivalent, connected to the main chamber by an orifice. This rich mixture is readily combustible, and the pressure resulting from its combustion discharges a multitude of ignition sources into the lean mixture in the main cylinder, producing a large number of combustion zones.

The speed of combustion is increased by having a number of sources from which combustion spreads. Normal combustion uses only one flame front.

B. STRATIFIED-CHARGE ENGINE

As in the first method, the cylinder is filled without using a throttle. Here the main charge is air. A pocket of fuel-air mixture of the normal combustible strength for rapid flame travel is produced in one of a number of ways, and the size of this pocket is varied according to the engine load. The aim is to produce a complete cylinder charge of the rich mixture at maximum load as in a normal spark-ignition engine, and then to reduce the amount of rich mixture as the load is reduced. Thus at half load the cylinder is filled with half combustible mixture and half air, and when the cylinder is idling, it contains a very small pocket of combustible mixture surrounded with a large volume of air. The Texaco combustion process uses this method of operation.

It is, of course, difficult to produce these two processes without some overlap, but we believe that these two simple methods represent the ideals for the various researchers.

We examined the engine performance possibilities for these two ideal cases:

(1) Lean Fuel-Air Ratio Engine: The cylinder is filled with a charge consisting of a uniform mixture of fuel and air, varying from $F/A = 0.00$, no fuel mixture at zero load, to $F/A = 0.0782$, a rich mixture at full load.

(2) Stratified-Charge Engine: The charge consists of a cylinder full of air in which a pocket of combustible mixture is produced, varying in size from a minimum of mixture, approaching zero at no load, to a completely filled cylinder of mixture, at full load.

III. METHODS OF CALCULATION

A. LEAN FUEL-AIR RATIO ENGINE

The conventional method of theoretical cycle prediction was used for all cases with a normal uniform mixture of fuel and air. Charts are available for the various combustion mixtures over the combustible range, from approximately F/A = 0.06 to F/A = 0.0782.[1] These charts provide for varying specific heat, chemical dissociation, etc. For leaner ratios such charts are not available, but the temperature of the cycle and dissociation are known to decrease rapidly. The varying specific heat then becomes the major factor.

Lean fuel-air ratio operation was therefore examined by using charts, whenever possible, and by using specific heat data,[2] at the leaner ratios.

B. STRATIFIED-CHARGE ENGINE

The stratified-charge engine produced another set of problems. In order to have the small combustible pocket in contact with the spark plug at the end of compression, the pocket must not mix with the main bulk of the air charge. Many inventions are concerned with the various processes for achieving this. An ideal engine cycle of this type would be one in which the combustible mixture was completely isolated from the remaining cylinder contents at all times and could be fired at will, with little or no mixing of the combustion products with the remaining air. Although this is an unattainable ideal, it would illustrate the optimum performance for the stratification system.

This cycle was examined by keeping the air and mixture separated but in pressure communication with one another at all times, producing uniform gas pressure throughout the engine cylinder at all points of the cycle. Thus two gases with differing properties were to be compressed and expanded simultaneously with pressure equilibrium in the same cylinder.

The stratified-charge engine can be visualized as an opposed piston engine in which the individual pistons can move as desired but the overall compression ratio remains constant. In such a cylinder a floating divider, leakproof but movable, maintains the same pressure throughout the gases, as represented in Figure 1, where

V_1 = volume of (1 + F/A) lb of mixture in cu ft

x = volume of mixture in the cylinder

$1 - x$ = volume of air in the cylinder

F/A = fuel-air ratio

Therefore, when $x = 1$, the cylinder volume is V_1 cu ft under maximum power conditions containing (1 + F/A) lb of mixture. Under these conditions the ideal cycle can be determined directly from the combustion charts.

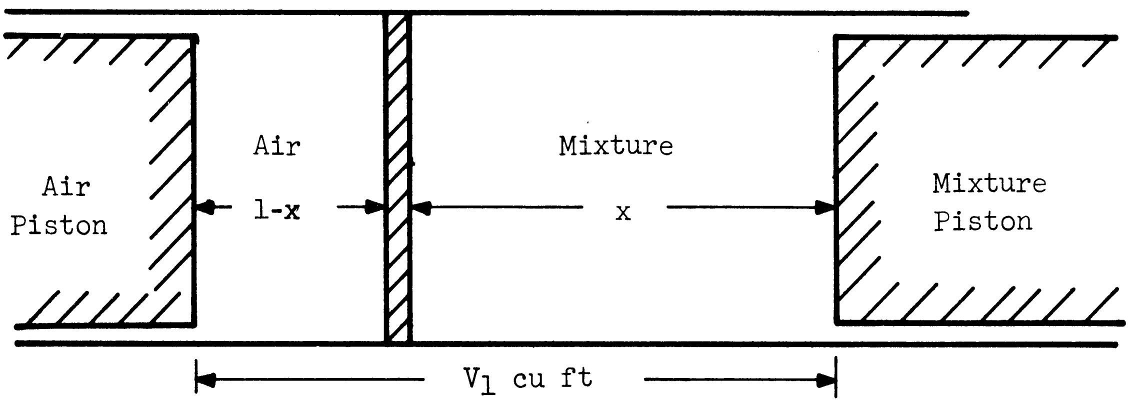

Figure 1. Diagrammatic sketch of imaginary cylinder for stratification.

At part load the cylinder will contain x (1 + F/A) lb of mixture occupying x (V_1) cu ft at a pressure and temperature that is the same as that for a full load. In addition the cylinder will contain a volume of (1 - x) V_1 cu ft of air at a density determined by the ambient inlet pressure and temperature. These two elements of the charge are divided by the partition.

C. COMPRESSION

It follows that when $0 < x < 1.0$, there are two separate gases present, each having its own thermodynamic properties. Air compression will follow the air tables[2] using variable specific heat, and fuel-air mixture compression will follow the compression chart[1] for that mixture ratio. At top dead center the pressure reached will be the same throughout the total clearance volume, but the compression ratio of each gas will be somewhat different, depending upon the individual gas properties. The condition to be fulfilled is

$$P_1 V_{a_1}^{n_a} = P_2 V_{a_2}^{n_a} \tag{1}$$

or

$$P_1 V_{g_1}^{n_g} = P_2 V_{g_2}^{n_g} \tag{2}$$

where

a = air

g = gas mixture

P_1 = pressure at end of inlet

P_2 = pressure at end of compression

n_a = index of compression (average) for air

n_g = index of compression (average) for gas

The compression ratio for air is

$$R_a = \frac{V_{a_1}}{V_{a_2}} \tag{3}$$

The compression ratio for gas is

$$R_g = \frac{V_{g_1}}{V_{g_2}} \tag{4}$$

The compression ratio of the L-141 engine, which is being used in the A.T.A.C. tests, is 7.5:1. This value will be employed in this study, though it can be changed. For this engine:

$$\begin{aligned} R &= \text{overall compression ratio for the engine} \\ &= 7.5 \\ &= \frac{V_{a_1} + V_{g_1}}{V_{a_2} + V_{g_2}} \end{aligned}$$

The average values to be employed for n_a and n_g for the compression stroke were determined from the air tables and combustion charts for a ratio of 7.5:1 with average ambient inlet pressure and temperature. The applicable values were

$$n_a = 1.38$$

$$n_g = 1.355$$

At the same time the gas constants for air and the various mixture ratios were obtained[2]; they are plotted in Figure 2.

From Eqs. (1) through (4) the following is obtained:

$$P_1 (R_a)^{1.38} = P_1 (R_g)^{1.355}$$

$$R_g = R_a^{1.02} \tag{5}$$

If the cylinder volume V_1 is that volume occupied by 1 lb of air plus fuel at inlet conditions, i.e., x = 1.0 or full load, the clearance volume is $V_1/7.5$.

For a rich mixture (F/A = 0.0782), an inlet pressure of 14.7 psia, and a temperature of 600°R, V_1 is 15.86 cu ft, and V_2, the total clearance volume, is 2.115 cu ft.

If both air and mixture are present, i.e., $0 < x < 1.0$, then:

$$\text{Total charge} = \text{volume of mixture} + \text{volume of air}$$

$$= 15.86\ x + 15.86\ (1 - x) \text{ cu ft}$$

$$= x\ (1 + F/A) + \frac{P_1 \times 144 \times 15.86\ (1 - x)}{53.34 \times T_1} \text{ lb}$$

where $P_1 = 14.7$ and $T_1 = 600°R$.

At the end of the compression stroke

$$\text{Volume of air} = \frac{15.86\ (1 - x)}{R_a}$$

$$\text{Volume of mixture} = \frac{15.86\ x}{R_g}$$

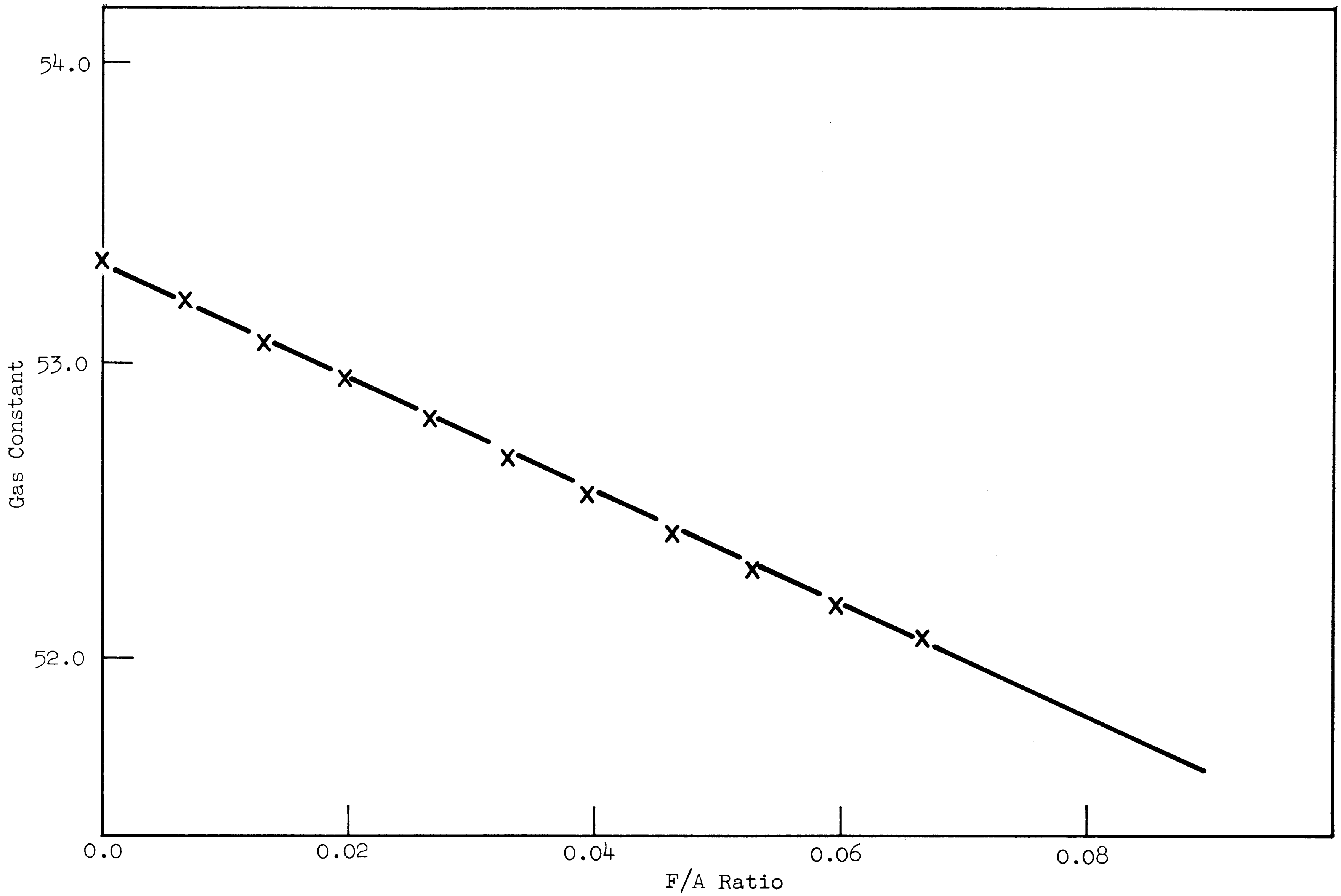

Figure 2. Gas constants for various mixture ratios.

Also,

$$V_{a_2} + V_{g_2} = 2.115$$

$$\frac{15.86\ (1 - x)}{R_a} + \frac{15.86\ x}{R_g} = 2.115 \qquad (6)$$

and

$$R_g = R_a^{1.02}$$

Substituting for R_g in Eq. (6) and simplifying, we get

$$7.5\ (1 - x)\ R_a^{0.02} + 7.5\ x = R_a^{1.02} = R_g \qquad (7)$$

Values satisfying Eq. (7) are shown in Table I and Figure 3.

TABLE I

VARIATION OF RATIO AS x VARIES

x	R_a	R_g
1.0	7.21	7.5
0.9	7.24	7.53
0.8	7.265	7.56
0.6	7.30	7.62
0.4	7.36	7.682
0.2	7.42	7.74
0.0	7.5	7.81

With the aid of Figure 3 it is now possible to obtain the change of state, work done, etc., on both the air and mixture contents of the cylinder, assuming they are separated at all times.

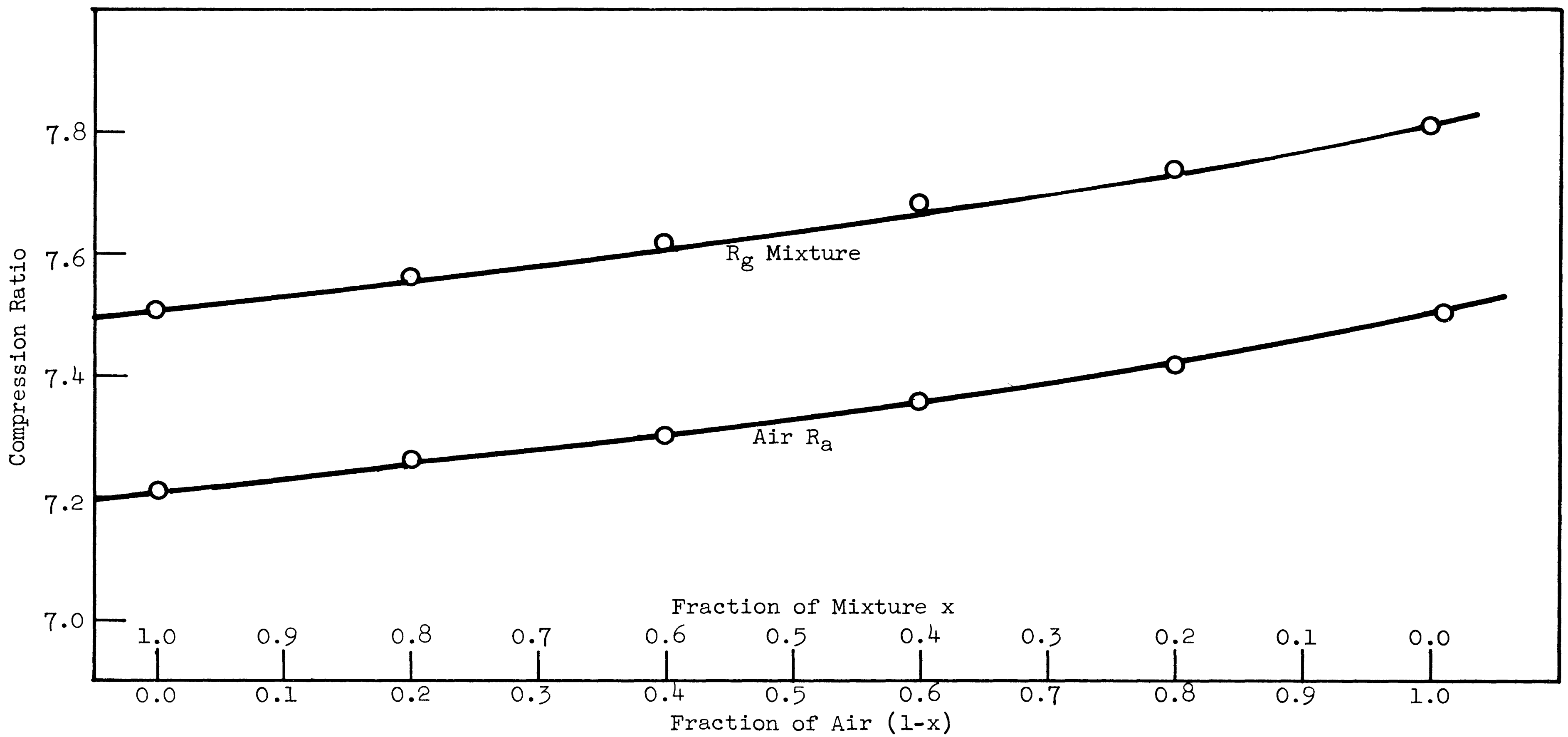

Figure 3. Compression ratios for air (R_a) and mixture (R_g), overall ratio = 7.5:1.

D. COMBUSTION

At the end of the compression stroke, combustion occurs in the x fraction of the mixture, which expands under the heat addition, compressing the air. The final pressure P_3 is reached at top dead center. This pressure P_3 can be obtained by a trial and error process using the combustion charts.[1] (See example below.) The values for P_3, T_3, etc., are then available for both the burnt mixture and the air.

E. EXPANSION

The gases now expand, forcing the pistons back to bottom dead center. The final state, with a uniform pressure P_4 psi, is shown in Figure 4. Expanding the air and mixture separately, in a manner similar to that of compression, yields

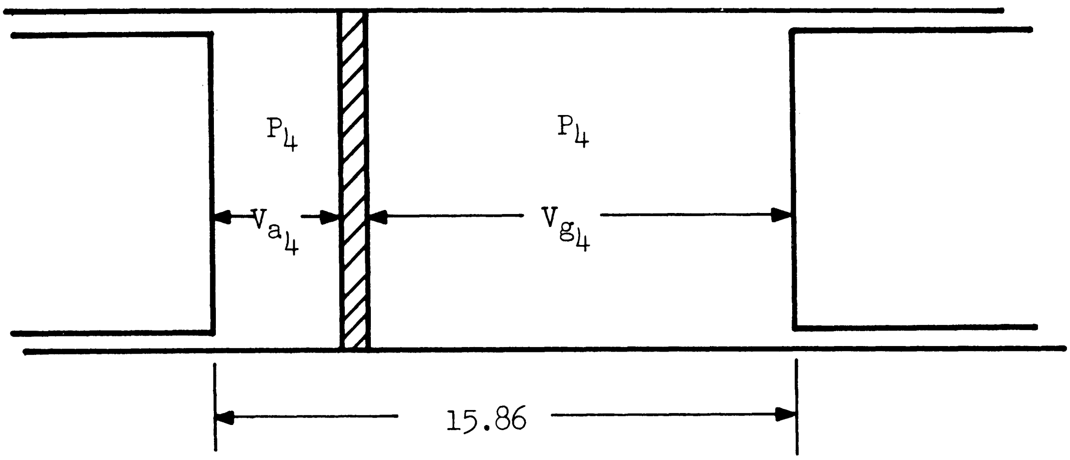

Figure 4. Diagram of conditions at BDC of expansion stroke.

$$P_3 V_{a_3}^{n_a} = P_4 V_{a_4}^{n_a} \tag{8}$$

and

$$P_3 V_{g_3}^{n_g} = P_4 V_{g_4}^{n_g} \tag{9}$$

where n_a and n_g are two new average values for the index, appropriate for the changed temperature and pressure conditions. As before, the relation between the ratio of mixture and air becomes

$$R_g = (R_a)^{n_a/n_g} \qquad (10)$$

Since the overall expansion ratio is still 7.5:1,

$$7.5 = \frac{V_{a_4} + V_{g_4}}{V_{a_3} + V_{g_3}}$$

and

$$15.86 = V_{a_3} R_a + V_{g_3} R_g$$

$$15.86 = V_{a_3} R_a + V_{g_3} (R_a)^{n_a/n_g} \qquad (11)$$

an equation in R_a and known values for V_{a_3}, etc. The magnitudes of n_a and n_g will differ from those of Eq. (5) because of the change in the temperature and composition of the gases. Calculations indicate that $n_a = 1.356$ and $n_g = 1.244$, making $n_a/n_g = 1.091$. These values cover the expected range with sufficient accuracy. It follows that Eq. (11) can be expressed as

$$15.86 = V_{a_3} R_a + V_{g_3} (R_a)^{1.091} \qquad (12)$$

Therefore, when V_{a_3} and V_{g_3} have been obtained after the combustion process, the value for the ratio of expansion of the air can be obtained, and the ratio of expansion of the gases can be obtained from Eq. (10).

The work of expansion can be obtained from the combustion charts for the gases and from calculations for the air by using the equations:

$$\text{Work of air during expansion} = \frac{P_3 V_{a_3} - P_4 V_{a_4}}{n_a - 1} \text{ ft lb} \qquad (13)$$

and

$$P_3 V_{a_3}^{n_a} = P_4 V_{a_4}^{n_a} \qquad (14)$$

IV. SAMPLE CALCULATION FOR THE STRATIFICATION SYSTEM

The following typical case represents the detailed calculations involved. The cycle will be examined for x = 0.9, with a manifold pressure of 14.7 psi, a temperature of 600°R, and a rich fuel-air ratio, F/A = 0.0782. The volume of the containing cylinder is maintained at 15.86 cu ft as required for x = 1. This case, therefore, represents a reduced engine load at rich mixture. The indicator diagram of Figure 5 represents the overall cycle.

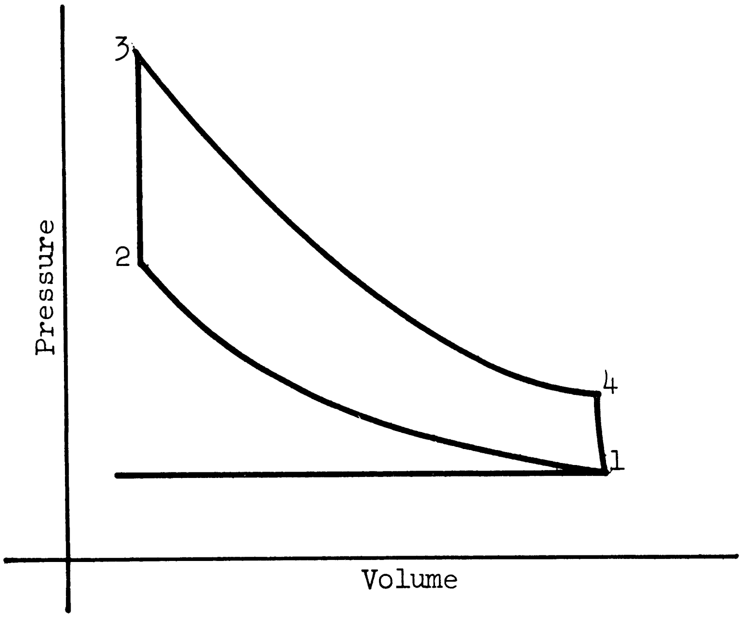

Figure 5. Indicator diagram when x = 0.9.

State 1

$P_1 = 14.7$, $T_1 = 600°R$, $x = 0.9$, $V_1 = 15.86$, $F/A = 0.0782$

Volume of mixture = 0.9 x 15.86 = 14.274 cu ft

Volume of air = 0.1 x 15.86 = 1.586 cu ft

Wt of mixture = 0.9 x 1.0782 = 0.9704 lb

$$\text{Wt of air} = \frac{14.7 \times 144 \times 1.586}{53.34 \times 600} = 0.1048 \text{ lb}$$

From Figure 3 $R_a = 7.24$ and $R_g = 7.53$.

Compression of Air

$$P_{a_2} = P_{a_1} (R_a)^{1.38} = 14.7 \times 15.40 = 226 \text{ psi}$$

$$V_{a_2} = V_{a_1}\left(\frac{P_{a_1}}{P_{a_2}}\right)^{1/1.38} = 1.586 \times \frac{1}{7.24}$$

$$= 0.219 \text{ cu ft}$$

$$V_{g_2} = \frac{V_{g_1}}{7.53} = 1.896 \text{ cu ft}$$

Total clearance volume = 1.896 + 0.219

= 2.115 cu ft

The calculated clearance volume must equal the designed clearance volume. Checking shows that

$$\text{Designed clearance volume} = \frac{15.86}{7.5} = 2.115 \text{ cu ft}$$

Compression of Mixture (by charts)

$P_1 = 14.7$, $V_1 = 15.86$/lb of air, $T_1 = 600°R$, $E = 15.0$

$$V_{g_2} = \frac{15.86}{7.53} = 2.108/\text{lb of air} = 1.896 \ (x = 0.9)$$

$S_2 = 0.088$, $E_2 = 1.37$, $P_2 = 224$ psi, $T_2 = 1146$

Pressure of air = 226

Pressure of mixture = 224

An average of the compression values for the air and the mixture of 225 psi is used for this operation.

Compression State 2

	Air		Mixture
P_{a_2}	= 225	P_{g_2}	= 225
V_{a_2}	= 0.219	V_{g_2}	= 1.896
T_{a_2}	= 1273	T_{g_2}	= 1146
W_a	= 0.1048 lb	W_g	= 0.9704 lb

Combustion State 3

Combustion now occurs and a pressure of P_3 is obtained. Then

$$P_{a_3} = P_{g_3} = P_3$$

It follows that for the air

$$P_{a_3} = P_{a_2}\left(\frac{V_{a_2}}{V_{a_3}}\right)^{1.356}$$

therefore

$$V_{a_3} = \left(\frac{225}{P_{a_3}}\right)^{0.738} \times 0.219$$

$$= \frac{11.83}{P_{a_3}^{0.737}} \qquad (15)$$

Various values for P_{a_3} are assumed until equilibrium with the gases at the end of combustion is obtained. The final value for P_{a_3} is 950 psi, which will be used to illustrate the method.

When P_{a_3} = 950 psi, V_{a_3} = 0.0759 cu ft, by Eq. (15). Therefore, V_{g_3} = 2.115 - 0.0759 = 2.0391 cu ft for the 0.9 lb of air in the combustible mixture. The chart volume (for 1 lb of air) then must be 2.267 cu ft/lb of air.

The volume of the mixture must match the above 2.267 cu ft to assure that the correct pressure has been assumed.

The mixture expands as combustion compresses the air and does work dW on it. The heat added by combustion is the calorific value of the fuel, 1507 Btu/lb of air.

By the first law of thermodynamics

$$dQ = dE + dW$$

$$dQ = dE + \frac{P_{a_3} V_{a_3} - P_{a_2} V_{a_2}}{n-1} \times \frac{144}{778} \times \frac{1}{x}$$

$$1507 = dE + \frac{950 \times 0.0759 - 225 \times 0.219}{0.356} \times \frac{144}{778} \times \frac{1}{0.9}$$

$$dE = 1507 - 13.0$$

$$= 1494.0 \text{ Btu/lb of air}$$

therefore

$$E_{g_3} = E_{g_2} + 1494$$

$$= 137 + 1494$$

$$= 1631 \text{ Btu}$$

The intersection of $P_{a_3} = 950$ and $E_{g_3} = 1631$ on the rich mixture charts shows that the volume of the mixture must be 2.27 cu ft, which checks closely with the desired 2.267 cu ft; 950 psi is accepted as the pressure at point (3) of the cycle.

Expansion

The gases and air now expand from P_3 to P_4. During this operation Eq. (12) must be satisfied.

$$15.86 = 0.0759\, R_a + 2.0391\, (R_a)^{1.091} \qquad (12)$$

Equation (10) shows that $R_a = 6.37$ and $R_g = 7.53$.

It follows that

$$V_{a_4} = 6.37 \times 0.0759 = 0.485 \text{ cu ft}$$

$$V_{g_4} = 2.0391 \times 7.53 = 15.365 = 17.06 \text{ cu ft/lb of air}$$

$$\text{Total volume} = 15.36 + 0.485 = 15.850 \text{ cu ft}$$

Checking the required 15.86 closely shows that

$$P_{a_4} = P_{a_3}\left(\frac{1}{R_a}\right)^{1.356} = \frac{950}{12.2} = 77.8 \text{ psi}$$

The mixture values on the chart are extended from state 3 to state 4 to obtain the properties:

$$P_{g_3} = 950,\ E_{g_3} = 1632,\ S_{g_3} = 0.571,\ V = 2.27$$

therefore,

$$S_4 = 0.571,\ V_{g_4} = 17.06,\ P_{g_4} = 77.0,\ E_{g_4} = 982$$

The average P_4 of 77.4 psi is used.

Net Work of Cycle

$$\text{Work of compression} = \text{work on air} + \text{work on mixture}$$

$$= \frac{225 \times 0.219 - 14.7 \times 1.586}{0.38} \times \frac{144}{778} + (137 - 15)\ 0.9$$

$$= 12.4 + 109.9$$

$$= 122.3 \text{ Btu}$$

Work of expansion $= \text{work on air} + \text{work on mixture}$

$$= \frac{950 \times 0.0759 - 77.4 \times 0.485}{0.356} \times \frac{144}{778} + (1632 - 982)\ 0.9$$

$$= 17.9 + 585$$

$$= 602.9 \text{ Btu}$$

Net work of cycle $= 602.9 - 122.3$

$$= 480.6 \text{ Btu}$$

Cycle efficiency $= \dfrac{480.6 \times 100}{1507 \times 0.9}$

$$= 35.4\%$$

IMEP $= \dfrac{\text{work}}{\text{change of volume}}$

$$= \frac{480.6 \times 778}{(15.86 - 2.115) \times 144}$$

$$= 189.5 \text{ psi}$$

IHP $= \dfrac{480.6 \times 778}{550}$

$$= 679.5 \text{ hp}/0.9 \text{ lb air/sec}$$

$$= 755 \text{ hp/lb air/sec}$$

SFC $= \dfrac{0.0782 \times 3600 \times 0.9}{679.5}$

$$= 0.373 \text{ lb/IHP/hr}$$

V. RESULTS

Repeating this calculation for all of the desired values of x produces the data in Table II, plotted in Figure 6.

TABLE II

PERFORMANCE OF IDEAL STRATIFIED-CHARGE ENGINE WITH RICH MIXTURE WHEN x = 1.0

(Compression Ratio 7.5:1, F/A = 0.0782)

x	Mixture, lb	Air, lb	P_{max}, psi	Net Work, Btu	IMEP, psi	IHP, lb air/sec	Thermal Eff., %	SFC, lb/IHP/hr
1.0	1.0782	0.0	1040	532.0	209.0	752.0	--	0.374
0.9	0.9704	0.1048	950	480.6	189.5	679.5	35.6	0.373
0.8	0.8625	0.2097	890	430.1	169.5	609.0	35.4	0.370
0.6	0.6469	0.412	730	418.2	125.2	450.0	35.6	0.375
0.4	0.431	0.617	575	214.0	84.2	303.0	35.2	0.373
0.2	0.2153	0.823	410	117.8	46.3	166.5	35.4	0.338
0.0	0.0	1.03	240	-	-	-	39.1	-

x	Compression Ratio		Expansion Ratio		Corrected Data			
	Mixture	Air	Mixture	Air	IMEP	IHP	Eff.	SFC
1.0	7.50	--	7.5	--	209.0	752.0	35.6	0.374
0.9	7.53	7.24	7.53	6.37	188.0	675.0	35.3	0.375
0.8	7.56	2.27	7.6	6.4	167.0	599.0	35.4	0.376
0.6	7.62	7.3	7.72	6.53	125.2	450.0	35.2	0.376
0.4	7.682	7.36	7.88	6.62	83.5	300.0	35.1	0.375
0.2	7.74	7.42	8.18	6.86	41.7	149.8	35.2	0.376
0.0	--	7.5	--	7.5	--	-	--	-

All of the data in Table II are for a constant F/A ratio of 0.0782, a rich mixture, ensuring positive ignition and high flame speeds at all conditions of operation,even if turbulence should cause some dilution of the charge.

Lean mixtures of 0.06 fuel to air also ignite readily in the spark-ignition engine, producing improved economy. Charts were available for fuel-air

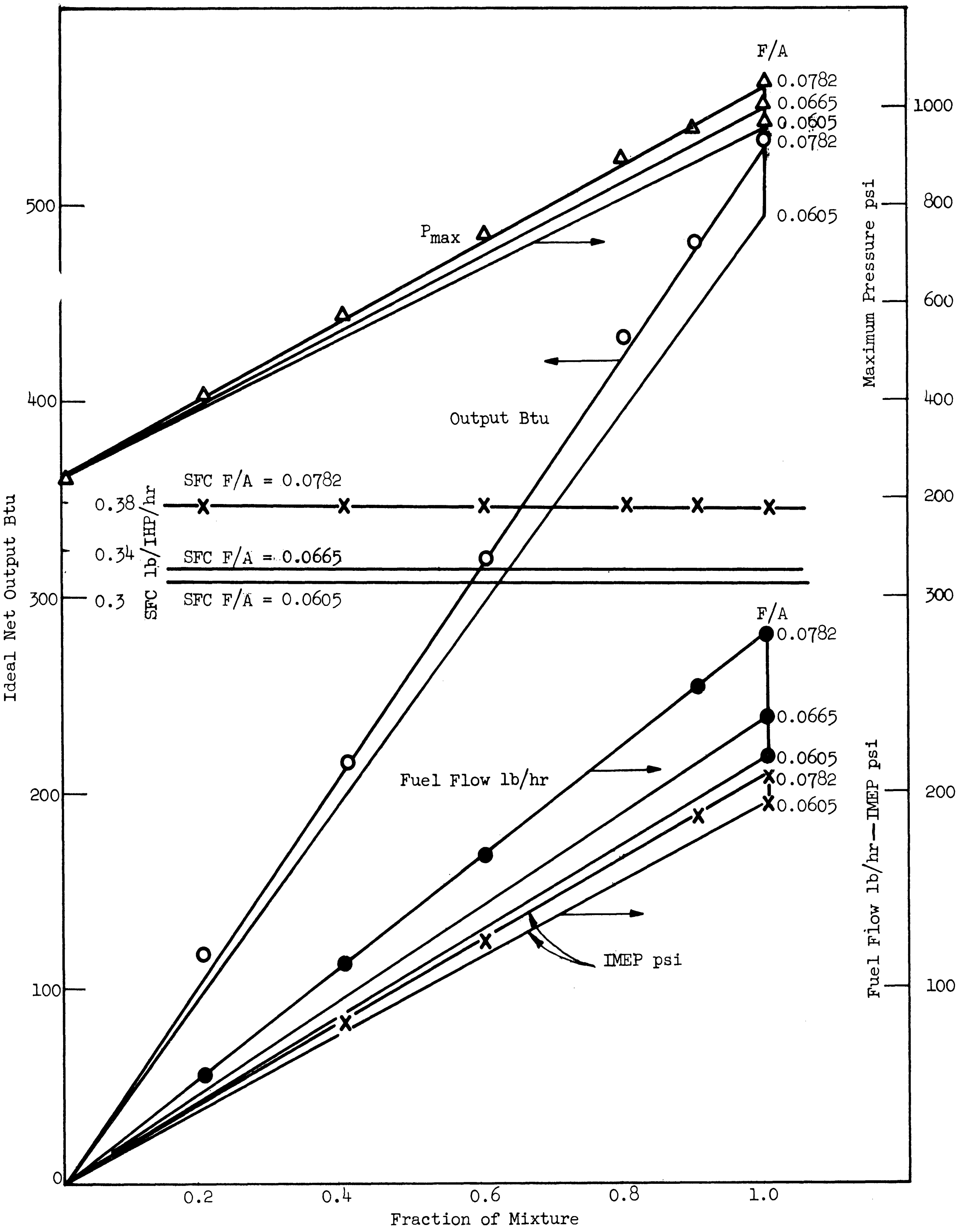

Figure 6. Ideal performance of a stratified-charge engine.

mixture ratios of 0.0665, the correct mixture, and 0.0605, a lean mixture. Calculations were performed when x = 1.0, using each of these mixtures, with the results given in Table III.

TABLE III

PERFORMANCE OF IDEAL STRATIFIED-CHARGE ENGINE WITH LEAN MIXTURE WHEN x = 1.0

	F/A = 0.0605	F/A = 0.0665
Output	495 Btu	527 Btu
IMEP	194.5 psi	207 psi
IHP	700/lb air/min	746/lb air/min
Fuel Flow	0.311 lb/IHP/hr	0.321 lb/IHP/hr
SFC	960 psi	1000 psi
P_{max}	218 lb/hr	239 lb/hr

The straight-line relationships obtained for the rich mixture as x varies are expected to apply to the lean mixture conditions also. Therefore, joining the points of Table III to zero by straight lines in Figure 6 described the performance to be expected from these lean mixtures for various values of x. A small triangular area is produced inside of which satisfactory engine performance can be expected. For a stoichiometric ratio of 0.0665, the differences in IMEP and output for the rich and the correct mixtures were too small to be shown on Figure 6 without greatly expanding the scale. For most practical purposes the F/A ratio values of 0.0782 and 0.0665 give identical data for these two parameters, also shown in Figure 7. If x = 1.0, 532 and 527 Btu are released in the form of work for F/A = 0.0782 and 0.0665, respectively. The principal change resulting from the reduced F/A is a reduction of SFC. Figure 6 thus records the ideal performance for a stratified-charge engine using conventional values for F/A.

The uniform mixture calculations for the lean fuel-air ratio engine were made by using the gas charts[1] for mixture ratios of 0.0782, 0.0665, and 0.0605. At lower values for F/A dissociation was neglected, and variable specific heat data[2] were employed. The results obtained are recorded in Table IV. The cylinder displacement employed was constant at the same volume as that used for the stratified-charge engine, i.e., 15.86 cu ft per lb of air, when F/A = 0.0782 with an inlet pressure of 14.7 psi and a temperature of 600°R. Tables II and IV contain corrected data obtained by using the values from the curve drawn through the plotted points to eliminate the effects of the trial-and-error process.

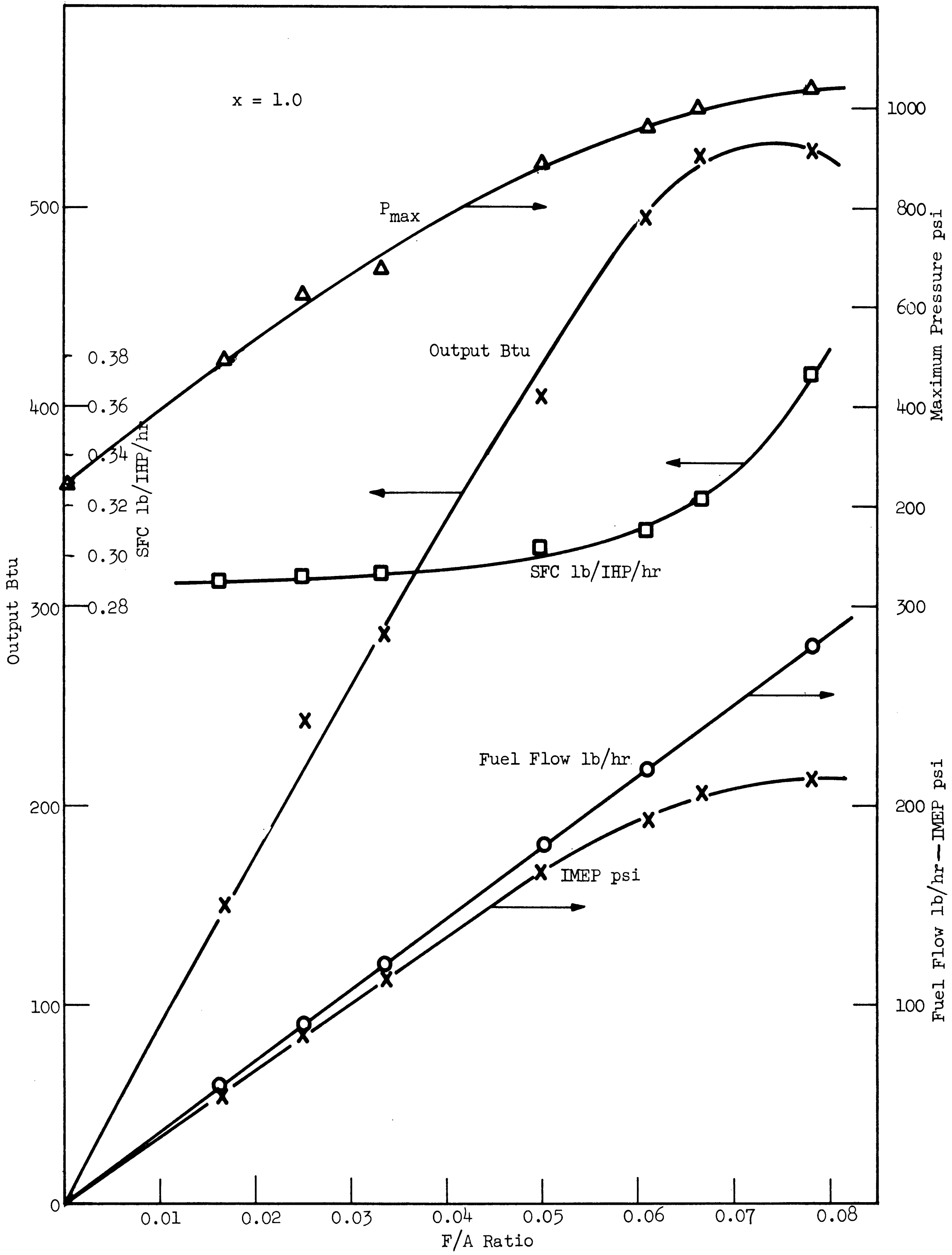

Figure 7. Ideal performance of a lean fuel-air ratio engine.

TABLE IV

IDEAL LEAN FUEL-AIR RATIO L-141 ENGINE

(Compression Ratio 7.5:1)

F/A Ratio	Net Work, Btu	IMEP, psi	IHP, lb air/sec	SFC, lb/IHP/hr	Thermal Eff., %	P_{max}, psi	Corrected Data			
							IMEP	IHP	Eff.	SFC
0.0782	532.0	209.0	752	0.374	35.6	1040	209.0	752	35.6	0.374
0.0665	527.0	207.0	746	0.321	41.2	1000	205.0	739	40.8	0.324
0.0605	495.0	194.0	700	0.311	42.5	960	194.6	703	42.7	0.310
0.05	403.8	159.0	570	0.316	42.0	890	165.5	592	43.7	0.304
0.0333	289.0	114.0	409	0.293	45.1	677	114.0	409	45.1	0.293
0.025	241.0	94.8	341	0.264	50.2	621	85.4	308	45.3	0.292
0.0167	150.0	59.0	212	0.284	46.9	496	57.8	208	45.9	0.290
0.0	--	--	-	-	--	240	-	-	--	-

VI. DISCUSSION

The data of Figures 6 and 7 show the relative effects of engine operation under the two systems examined. We will now discuss the implications for practical applications of these systems.

For the system for the stratified-charge engine we assume that a definite quantity of mixture, of a known combustible ratio, is available at the spark plug. The initiation of combustion is assured under all conditions, and the load is controlled by varying the quantity of this mixture in the pocket from a very small amount at idle to a full cylinder volume at full load and speed. The mixture ratio remains constant in the case examined; however, this ratio can be leaned out until ignition becomes critical.

In the lean fuel-air ratio engine the mixture ratio of the charge is changed continually from a very lean mixture at engine idle to a rich one at full load. It is well known that lean mixtures do not ignite and burn fast enough to be useful in a practical engine. Therefore, successful use of this system requires a means of accelerating the combustion process so that its time to completion is acceptable. We have assumed that such a process will be developed to a satisfactory degree. The torch ignition process is one example.

Figure 6 shows that for the stratified-charge engine the mixture ratio used when $x = 1.0$, i.e., at a full charge of combustible mixture, was varied over the well known combustible range from rich, F/A = 0.0782, to lean F/A = 0.0605. Actually fuel-air mixtures of 0.05 to 0.055 can be ignited readily, but they are not generally used in an engine because of the critical nature of their ignition and the unequal distribution of fuel between the individual cylinders of a multi-cylinder unit.

The data of Figure 6 indicate that although x, the proportion of mixture, varies, the SFC remains practically constant when the F/A is constant. The SFC lines for each F/A indicate that there is no gain in fuel economy on an IHP basis with a stratified-charge system. Less fuel is consumed in such a system, however, as the F/A is leaned out and $x = 1.0$, just as in a conventional engine. Since the power is controlled by varying the mixture and not by using a throttle, the pumping losses of a throttled engine do not exist, and an overall gain on a BHP basis is made by eliminating the negative work of engine charging, resulting from a considerable reduction in frictional losses at light loads. The magnitude of the gain would depend upon how lean the mixture could be made without sacrificing satisfactory combustion. This will be examined in detail later.

Similarly, the lean fuel-air ratio engine uses no throttle, and thus achieves a gain equivalent to that of the stratified-charge engine. Figure 7 also indicates a considerable reduction in SFC per IHP for the lean fuel-air ratio engine. This method of operation therefore seems superior to stratification. It must be remembered, however, that the results presented were based on the assumption that satisfactory combustion is maintained over the whole range of the mixture ratio employed. To achieve these results, lean fuel-air mixture ratios of about 0.02 to 0.025 must be made combustible. If this can be achieved, an SFC approaching that of the compression-ignition engine can be secured.

Comparing the data of Figures 6 and 7 shows little difference in engine performance, except in SFC, between the two systems. The stratified-charge engine at F/A = 0.0605 delivers 200, 300, and 400 Btu for a P_{max} of about 550, 680, and 820 psi; and the lean fuel-air ratio engine gives the same outputs for 600, 720, and 860 psi. However, the SFC is constant at 0.31 lb/IHP/hr for the stratified system and varies from 0.29 to 0.30 for the lean F/A system.

Combining Figures 6 and 7 into one allows both systems to be examined simultaneously. Such a plot is shown in Figure 8. This diagram is based upon the straight-line plot of Figure 6. When x is constant and equal to 1.0, power output is reduced and economy is improved as F/A is reduced. Therefore, in Figure 8 when x = 1.0, the data of Figure 7 can be plotted from F/A = 0.0782 to F/A = 0.0167. Joining these points for the lean ratios to the origin produces a plot equivalent to Figure 6 for each F/A. At the same time, by making the vertical axis one of relative power and IMEP, i.e., the ratio between the power or mean pressure at any desired F/A ratio to that power or pressure at stoichiometric ratio, this one diagram gives output, IMEP, and the SFC curve for both stratified and lean F/A systems.

A. STRATIFIED-CHARGE ENGINE

To use Figure 8 when x = 1.0 and F/A = 0.0605, locate the point of intersection of x = 1.0 with F/A = 0.0605. A horizontal line drawn from this point to the vertical axis (shown as dotted) gives the SFC at the intersection of this line with the fuel curve, i.e., 0.312 lb/IHP/hr. Proceeding to the vertical axis gives the relative hp and IMEP of 0.952. The values for the stoichiometric mixture, i.e., F/A = 0.0665, are listed on the diagram as basic hp = 739 and IMEP = 205 psi. Therefore the conditions for the ideal cycle specified are IHP = 703/lb air/sec, IMEP = 195, and SFC = 0.312 lb/IHP/hr.

For a stratified system when $x \neq 1.0$, say x = 0.8, and F/A = 0.0605, Figure 8 gives

Relative ratio = 0.762

SFC = 0.298

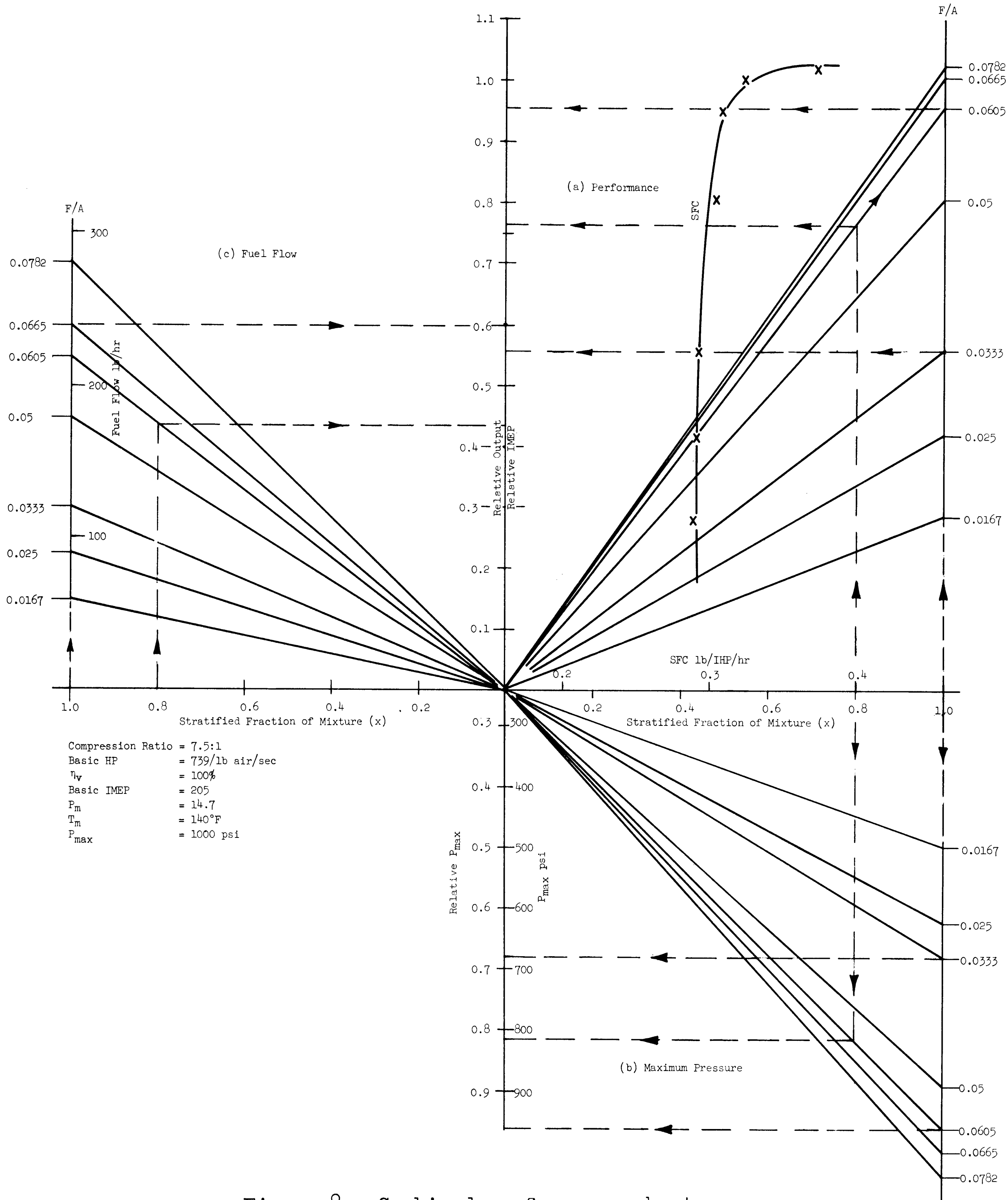

Figure 8. Combined performance chart.

Thus IHP = 563/lb air/sec, IMEP = 156.2, and SFC = 0.298 lb/IHP/hr.

B. LEAN FUEL-AIR RATIO ENGINE

In this case x = 1.0 at all times, since the cylinder is filled with a uniform charge. To find the performance when a lean F/A of 0.033 is used, proceed from the intersection of x = 1.0 and F/A = 0.033 to the intersection with the fuel curve at SFC = 0.293, and on to the vertical axis, where the relative performance is given as 0.555. Then

$$
\begin{aligned}
\text{IHP} &= 0.555 \times 739 \\
&= 410/\text{lb air/sec} \\
\text{SFC} &= 0.293 \text{ lb/IHP/hr} \\
\text{IMEP} &= 0.555 \times 205 \\
&= 114 \text{ psi}
\end{aligned}
$$

By combining two other graphs for the parameters P_{max} psi and fuel flow in lb/hr with Figure 8, these parameters also become available.

The essential performance factors for any ideal stratified or lean fuel-air ratio engine operating at a 7.5:1 compression ratio are available when there is 100% volumetric efficiency and when there are no heat losses or residual gas. However, the specific heat may vary and dissociation can occur.

It is interesting to compare these results with those obtained from the conventional ideal thermodynamic cycle efficiency equation.

$$
\begin{aligned}
\text{Ideal cycle efficiency} &= 1 - R^{1-k} \\
R &= \text{Compression ratio} \\
k &= \text{Ratio of specific heats}
\end{aligned}
$$

This equation uses variable values for k, already determined, based upon the average temperature for the cycle. As the mixture ratio varies, k_c for the compression stroke varies from 1.35 to 1.355, and k_e for the expansion stroke changes from 1.25 to 1.28. In the thermodynamic case for stratification, F/A is held constant at 0.0782, and hence k also remains constant for the mixture pocket. At a constant compression ratio of 7.5 the cycle efficiency was 45.7%. Figure 9 shows the efficiencies of Table III.

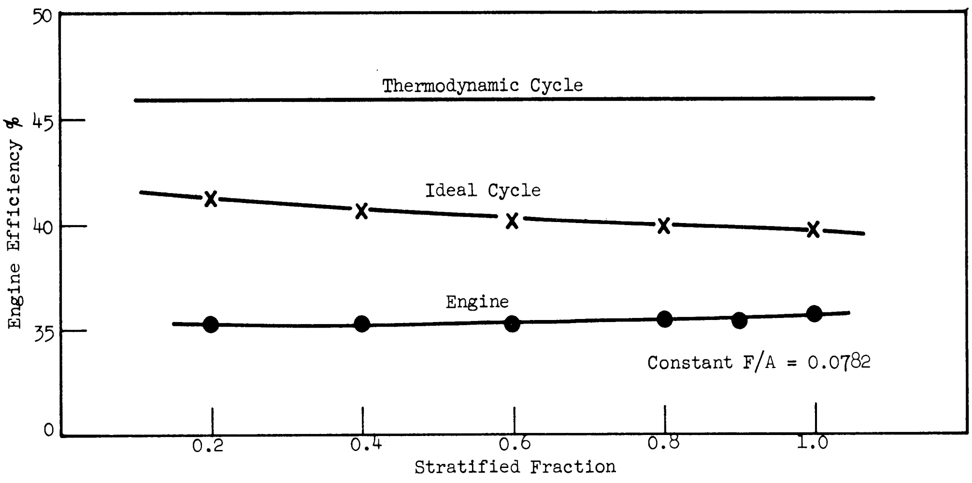

Figure 9. Engine efficiency for stratified-charge engine.

In addition Figure 9 shows an ideal cycle efficiency for the actual varying compression ratio, R_g, of the stratified cycle when combined with an average value of k for each cycle. The value for cycle efficiency shows only slight variation.

The third curve for a stratified-charge engine cycle illustrates a difference between the thermodynamic cycle and engine efficiencies that is also relatively constant. This is to be expected, since the F/A and the compression ratio are almost constant.

For lean F/A, as shown in Figure 10, the effects of dissociation, etc., at rich mixture settings are apparent, but the thermodynamic cycle and the practical engine cycle agree fairly closely when leaner mixtures are employed.

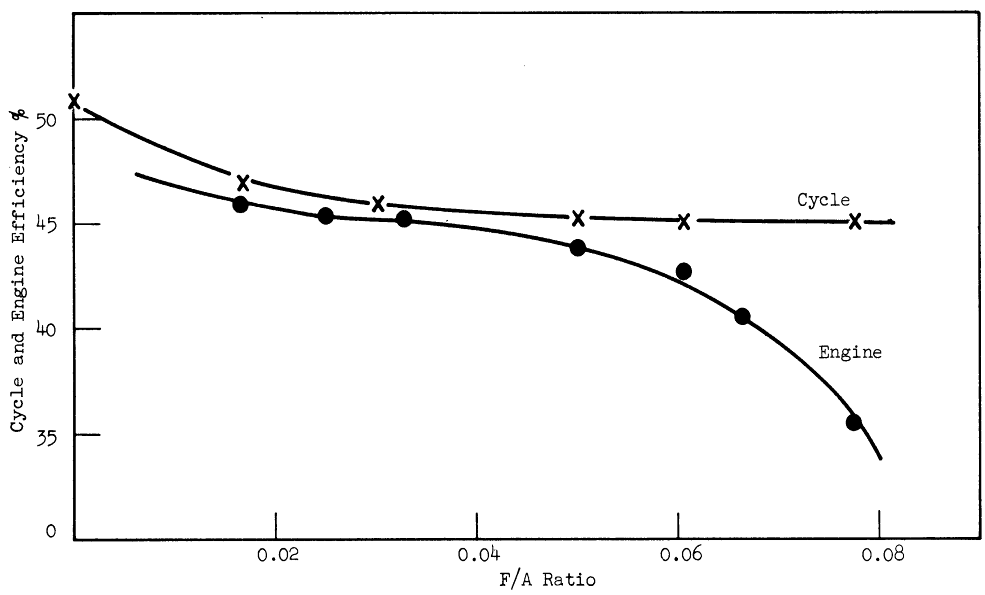

Figure 10. Engine efficiency for lean fuel-air ratio engine.

To compare these data with those for the L-141 engine, a best estimate of the frictional losses of the L-141 engine and of the SFC/IPH/hr were made. This estimate is shown in Figure 11, together with that for the two engines being examined. The solid-line L-141 plot is of an actual engine performance based upon gross IHP, and the solid-line curves for stratification and lean fuel are the ideal curves predicted for those engines. To make an approximate comparison the L-141 line was displaced to pass through the same SFC point, 752 hp/lb/sec, as the other two curves at full load. The dotted L-141 curve obtained is labeled ideal L-141. This curve follows the lean-fuel ideal curve fairly closely down to an F/A ratio of about 0.05, the leanest mixture usually employed. Below this point SFC increases rapidly as throttling occurs, and the mixture is richened somewhat for smooth idling. The difference between the dotted curve for the ideal L-141 and the lean F/A curve indicates the approximate penalty due to throttling and the need for stable operation. Generally speaking, the best way to compare cycles is to compare indicated data as we have done so far. However, the BHP data, which are of major interest to a purchaser, had to be calculated for the stratified-charge and lean-fuel engines from results based upon known losses in net output for the L-141 engine. Thus the net performances of the L-141, stratified-charge, and lean F/A engines are compared on a BHP basis. Losses from the air cleaner, generator, cooling fan, and exhaust system are included for a speed of 3600 rpm.

Figure 12 shows the BHP performance calculated for the ideal stratified-charge and lean-fuel cycles and the actual performance of the L-141 engine on the basis of the equivalent BHP for 1 lb air/sec. These results are shown by the solid-line curves. If the L-141 engine results are assumed to be for a relative cycle efficiency of 80%, a value usually exceeded in most engines, the dotted line marked 80% would represent the ideal L-141 engine performance. If the full-load SFC of the L-141 engine is reduced to equal that of the ideal cycles examined, a relative efficiency of 70.8% is obtained. The second dotted line then represents the ideal L-141 cycle on roughly the same scale as the other cycles under consideration.

A relative efficiency of 70.8%, therefore, must exist for the L-141, a rather low value for a modern engine; this low efficiency value could result from low volumetric efficiency, poor manifolding, or high manifold temperature.

The SFC of a stratified-charge engine with a constant F/A of 0.0782, then is inferior to that of the normal throttled engine over an appreciable portion of its output range. Not until 1/3 load is reached is any gain in economy made. However, the lean F/A engine has superior performance over the whole load range.

The F/A of 0.0782 was selected for the stratified-charge engine to give a mixture rich enough to assure ignition even with some dilution due to air turbulence. We assume that a leaner mixture could be used while maintaining ignition in the spark plug region. If a ratio of 0.05 could be used, the strati-

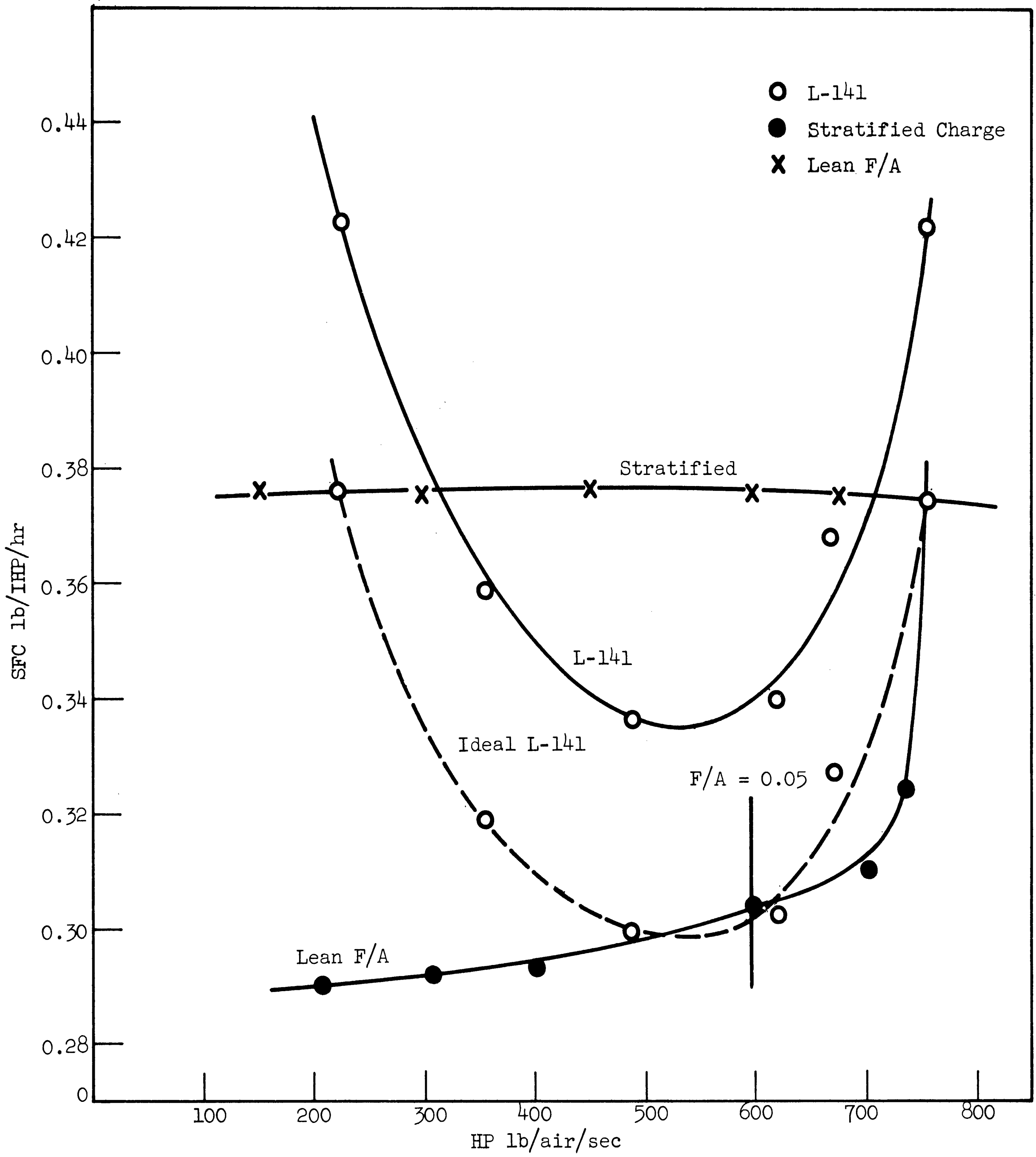

Figure 11. Comparison with L-141 engine.

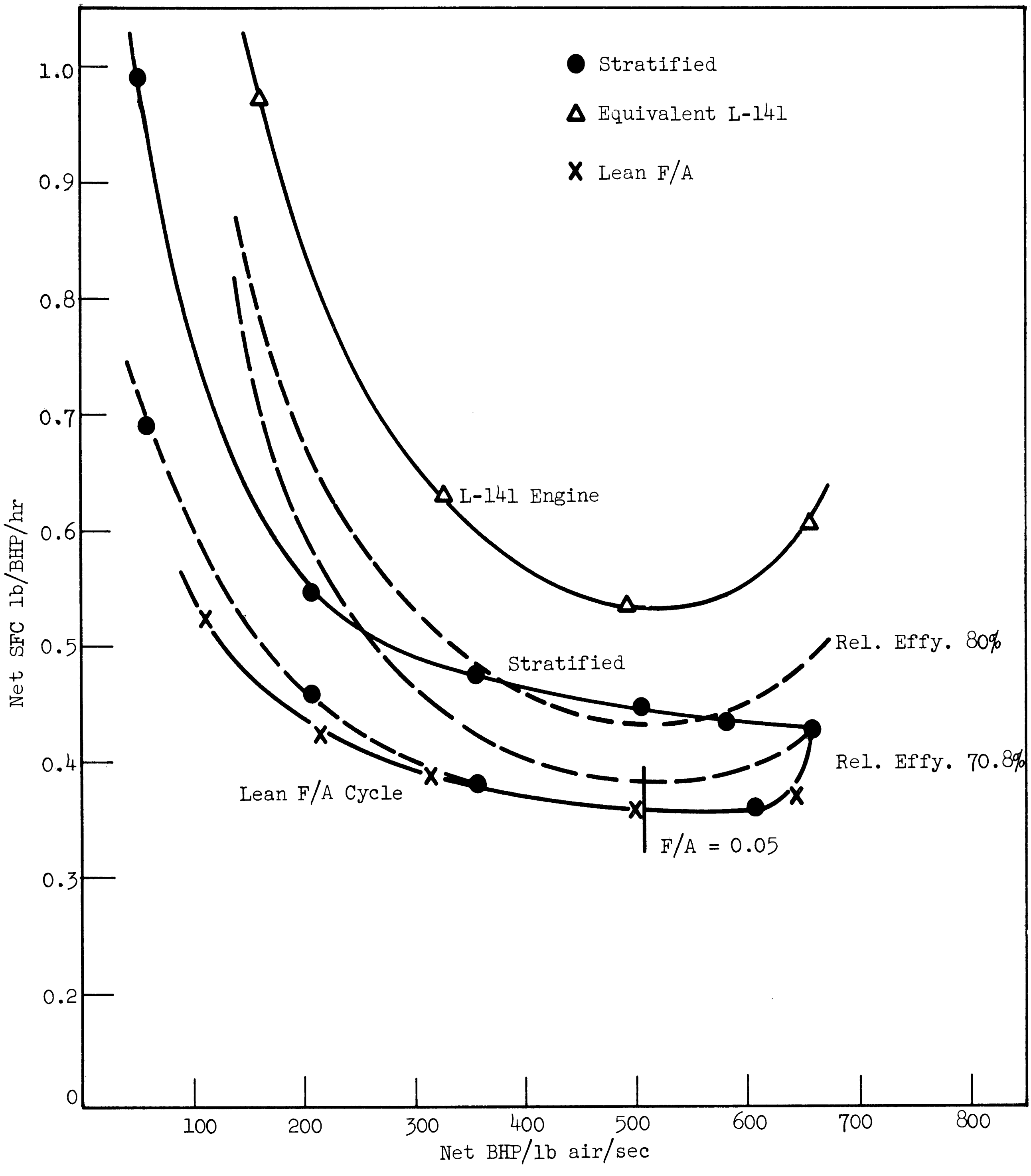

Figure 12. Net BHP performance at 3600 rpm, including air cleaner, generator, cooling fan, and exhaust system.

fied-charge engine efficiency would then follow the lean fuel-air ratio engine efficiency line down to F/A = 0.05, on to about half load, and along the dash-dot line of Figure 12 for lower engine powers. There would then be little difference between the two imaginary cycles, and it would be immaterial whether the stratification or the lean fuel-air ratio method was used.

The problem of igniting very lean fuel-air ratios has so far been ignored, since it is assumed that some method for this exists. The method in common use today is to inject a small pocket of rich mixture near the spark plug; combustion then produces many sources of ignition throughout the lean mixture, resulting in satisfactory combustion of all of the fuel. But this method has been only partially successful in practice, and some throttling at low loads is usually necessary. Some increase in SFC of the lean-mixture curve of Figure 12 is therefore expected in any practical engine. The difference between stratified and lean fuel-air conditions, then, will probably be less than this diagram shows; in fact, considerable increase in fuel flow for lean F/A engines will occur if much throttling is required.

A comparison of the net BHP per lb of air, as shown in Figure 12, with the actual net BHP output of the L-141 engine shows there is 656 hp/lb of air in the former and 62-63 BHP for the latter. The L-141 engine scale is, therefore, almost 1/10 that of the ideal, providing a convenient reference to both conditions.

The four important curves of Figure 12 are redrawn on a separate diagram in Figure 13 to summarize this investigation. Curve 1 shows the actual L-141 engine results for the assumed relative efficiency of 70.8% and for full-load conditions equal to those of the ideal cycles examined. Curve 2 shows the stratified-charge cycle for a rich mixture, 0.0782, throughout the whole power range. Curve 3 shows a stratified-charge cycle in which the load is reduced first by leaning the mixture from 0.0782 to 0.05, and then by stratification. For the cycle in curve 4 the load was controlled by using lean mixture ratio only.

The usable portions of each of these curves must now be determined. Since the data for curve 1 were obtained experimentally from an actual L-141 engine test, the whole curve represents possible engine operation.

The rich-mixture stratification results shown in curve 2 ensure ignition of the charge fraction containing the fuel at all times, unless turbulence should cause bad mixing of the air charge with the stratified charge. All of this curve could probably be used.

The stratified-charge engine is controlled by varying the fuel mixture down to the 0.05 ratio and then by varying the magnitude of x. This cycle as shown in curve 3, is also possible. Tests have already confirmed that mixtures of 0.04 are ignitable under such conditions, and the extra margin given by F/A = 0.05 should ensure regular ignition even if some mixing with the air occurs.

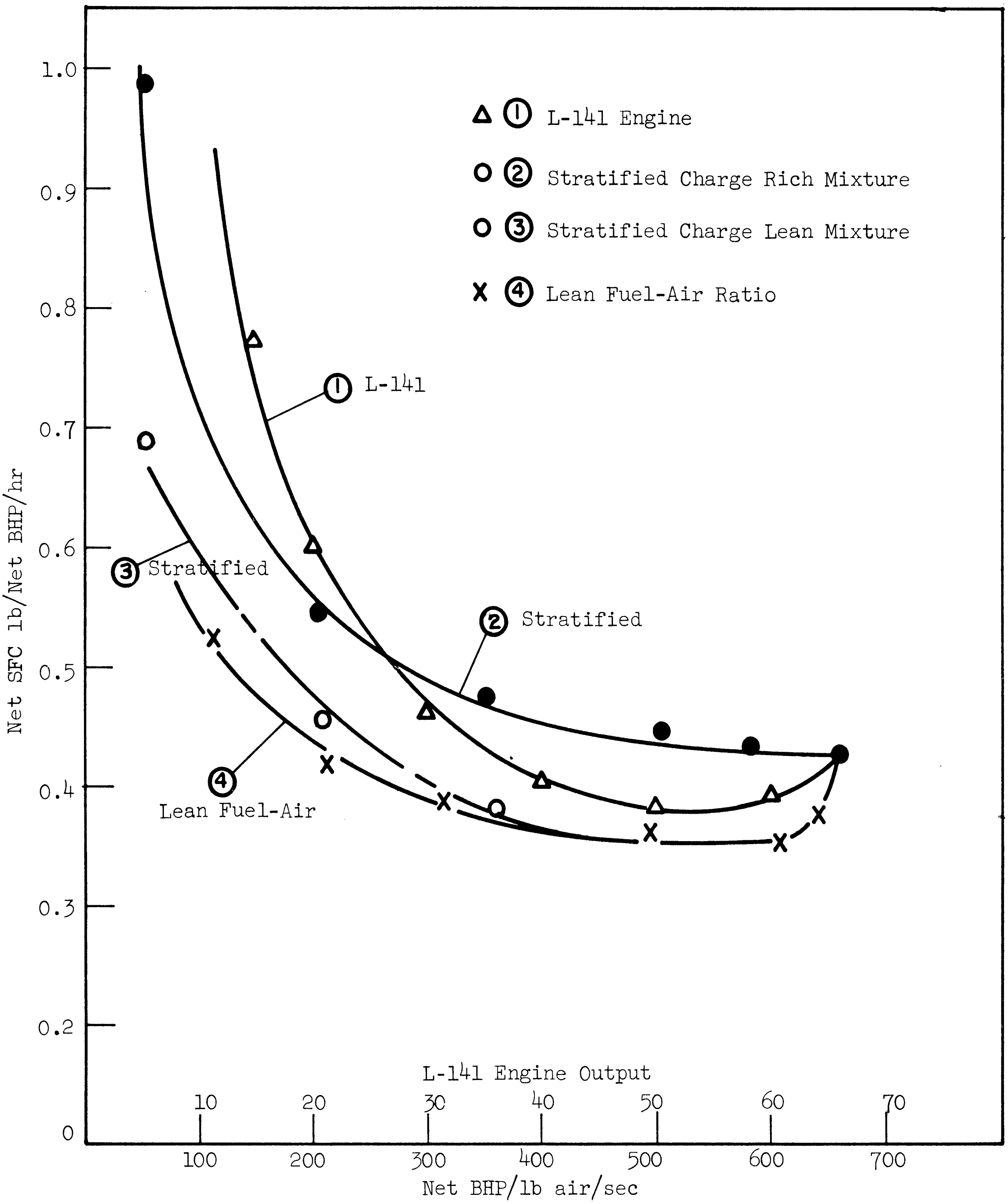

Figure 13. Comparison of cycles on BHP basis.

The lean-fuel mixture, as shown in curve 4, cannot be completely controlled without some throttling. Fuel-air ratios of 0.04 to 0.035 do appear to be possible, judging from some rather early tests, but at and below this point some throttling was necessary.

Therefore, curves 1, 2, and 3 represent possible operating conditions, and curve 4 can be employed down to about 300 net BHP per lb. We conclude that the area between curves 2 and 3 represents possible operation, with curve 3 being about the best we can expect.

From full load to about 75% load, the gains in fuel economy from these ideal engines are quite small, in fact negligible, since each ideal engine under these conditions operates with a full cylinder charge at a reduced F/A. Below 75% load the stratified-charge engine shows some advantage. At about 25% load the SFC of the stratified-charge engine has been reduced to 68% of that of the throttled engine, a saving of 42% in fuel consumption. This saving will increase at lower loads.

The engines considered in this report do not operate at constant load and speed; hence a complete map would have to be predicted to accurately determine the savings under typical operating conditions. When the low load predominates over any given time interval the savings could approach 25% to 30%; however, for heavy loads the savings would be smaller.

Some engines have been designed and tested using higher compression ratios than those of the standard engine. Any fuel gains from this type of engine design would add to the savings mentioned above.

VII. PRESENT STATUS OF RESEARCH

Each of the two methods of control examined has two systems that have shown some degree of practical success.

For the stratification method the Texaco combustion process, which forms a continually changing pocket of combustible mixture that burns with a minimum of mixing, appears to be the most advanced system and shows the best practical possibilities. Engines of this type have been run for a number of years. Costs are higher because of the need for both injection and ignition systems, but fuel savings, etc., have been demonstrated.

Another stratification engine, the Southwest Research engine, does not form such a clean cut pocket with its present design, and some mixing and lean-fuel burning seem unavoidable. The maximum load, speed range, and SFC of this engine are not well established.

The Walker system, a lean fuel-air ratio engine tested by Continental Aviation and Engineering Corporation, recently gave extremely good results developing about 85% of maximum load of the L-141 engine while giving a SFC at 20% load of 74% of that engine. A limited speed range was used in these tests. If similar results can be obtained at other speeds, a satisfactory engine should result. No injection system is involved.

The second lean fuel-air system, tested at the University of Rochester, has demonstrated the combustion of lean mixtures with some reservations.

More research on both of these systems is needed before a satisfactory engine will be available.

VIII. CONCLUSIONS

As a result of this investigation it can be concluded that:

1. A stratified-charge engine should possess the widest range of power outputs without throttling the air.

2. Theoretically a lean fuel-air ratio engine would give the lowest SFC provided that satisfactory ignition and combustion can be assured at all F/A ratios.

3. The SFC of a stratified-charge engine closely approaches that of the lean fuel-air ratio engine over most of the load range. Only at low loads does the SFC increase. Ignition of a lean F/A ratio engine in this low range, however, has not been definitely established.

4. Control of power by varying the mixture ratio only should permit a power output almost equal to that of a conventional engine.

5. Stratified-charge engines will probably give a maximum output slightly lower than that of present carbureted engines.

6. The two methods of control considered here could result in consuming as little fuel as the compression ignition engine when operating at low loads.

7. The main reasons for the high fuel consumption of conventional spark-ignition engines are the throttling losses and the need for a rich mixture for smooth idling.

8. The stratified-charge engine requires more research and development, eliminating turbulent mixing as much as possible, in order to increase cycle efficiency.

9. Lean F/A engines ignitied by torch should consume a very small quantity of rich mixture for ignition.

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