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Technical Report

VARIABLE-COMPRESSION-RATIO PISTON AND ITS EFFECT ON IDEAL AND PRACTICAL ENGINE EFFICIENCY

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ABSTRACT

This report presents the results of calculations of the ideal performance of a compression-ignition engine over a range of compression ratios from 8 to 22:1. For these calculations it was assumed that the cylinder charge undergoes typical heat losses, but that combustion is perfect.

The results are compared with the results of a series of tests of the engine at three different compression ratios, for each of which injection characteristics were varied to achieve the best performance and specific fuel economy. Finally, a typical performance curve for an engine employing a normally operating variable-compression-ratio piston is compared with the above data.

The object is to establish the penalty, if any, of employing a fixed-injection system in which the chamber shape varies with the engine load but the maximum gas pressure remains roughly constant.

I. OBJECT

The object of this analysis is to examine the effect of varying compression ratio upon ideal and practical cycle efficiencies, and thus to determine how specific fuel consumption (SFC) varies as the compression ratio is changed over a wide range by means of a variable-compression-ratio (VCR) piston.

II. INTRODUCTION

The advent of variable-compression-ratio pistons has introduced another variable into the already formidable array of factors affecting the power output and specific fuel consumption of a compression-ignition engine. It has been established that the variable-compression-ratio piston does permit control of the maximum cylinder pressure as the degree of supercharging varies (Refs. 1 and 2). As a result, one can use high-ratio supercharging in conjunction with a large increase in†brake mean effective pressure without increasing the structural load on the engine mechanism, and yet at the same time retain high compression ratio at slow speeds and light loads; an engine built according to these principles should start well in cold weather, use little fuel while idling, and operate with any of a variety of fuels.

Currently available results of engine tests show no major effect of variable-compression-ratio on specific fuel consumption. True, fuel consumption at low ratios is somewhat higher, but the reason for this increase is not clear. It may be that a change of ratio should be accompanied by a change in the injection system, to allow for the rather large variation in the shape of the combustion chamber as the ratio varies, as well as some changes in the turbulence of the fuel-air mixture and other phenomena.

In this investigation we duplicated the range of compression ratios used in earlier tests and analyzed the ideal cycle to obtain the theoretical variation in efficiency, indicated mean effective pressure, specific fuel consumption, etc., as the piston position varies to maintain a constant maximum cylinder pressure. Then with the aid of test results taken with the compression ratio fixed, and injection tailored to that ratio, we predicted the effective ratio between the results of actual tests and ideal calculations. Using this material one can make theoretical comparisons between the performance of a variable-compression-ratio piston engine and the best performance of a fixed-ratio engine. Thus, one can work toward estimating how a compromise in injection characteristics will affect engine performance.

III. METHODS

To achieve the above objective, we began with an ideal cycle which would approximate engine conditions, taking into account the effects of heat losses. When boost ratio varies widely, heat losses change so greatly as to completely mask the effects of the variable ratio unless properly corrected for.

Some years ago E. T. Vincent charted the results of cycle calculations in which data on variable specific heat and on heat losses during compression, combustion, expansion, etc., were used in making close approximations of engine cycles in which supercharge ratio, heat loss, and compression ratio varied only moderately. With some extrapolation, these charts were extended to cover the range of values in question here. Errors introduced by the extrapolation are considered minor.

The ideal cycle was examined for two limiting cases: (1) maximum cylinder pressure of 2000 psi, and (2) maximum cylinder pressure of 3000 psi. For each maximum pressure a compression ratio of 22:1 was used for starting and low-load operation; as the load increased, the ratio was gradually reduced to 12:1. This is the range of variation used in current variable-compression-ratio engines. To provide for decrease in compression-ratio variation with further developments in engines, additional calculations down to a ratio of 8:1 were made. Also, to allow for increases in boost ratio above that at which a maximum pressure of 2000 or 3000 psi is reached, calculations were made in which the compression ratio was held at 12:1 while the boost ratio increased from 3.2, giving 2000 psi, to 5.0:1, giving a maximum pressure of 3200 psi for case (1). In the case of 3000 psi, the boost ratio was raised from 4.75:1, giving 3200 psi at 12:1 compression ratio, up to 8.0:1, giving 5200 psi at the same pressure ratio. Figure 1 shows the compression and boost ratio ranges employed, and the maximum pressures reached. To fix the cycle, it was necessary to select a value for the ratio P_{max}/P_{comp}. This was kept at 1.5 for all cases examined, since, in general, it is about the minimum at which good combustion can be secured. One other important factor is the fuel/air ratio; this was held at 0.038, giving a heat addition of 700 Btu per cycle per lb of air.

To include the effect of heat flow to the coolant, some assumptions were necessary. The literature contains little background for the boost ratios investigated, but on the basis of that at hand the schedule of heat losses in Table I was employed. It is based upon the established fact that as the boost ratio increases, the percentage of heat flow decreases, but of course the total Btu transferred increases because of the greater heat release per unit volume of cylinder. In the calculations performed, the percentage of heat flow to the coolant was assumed to vary as in Table I.

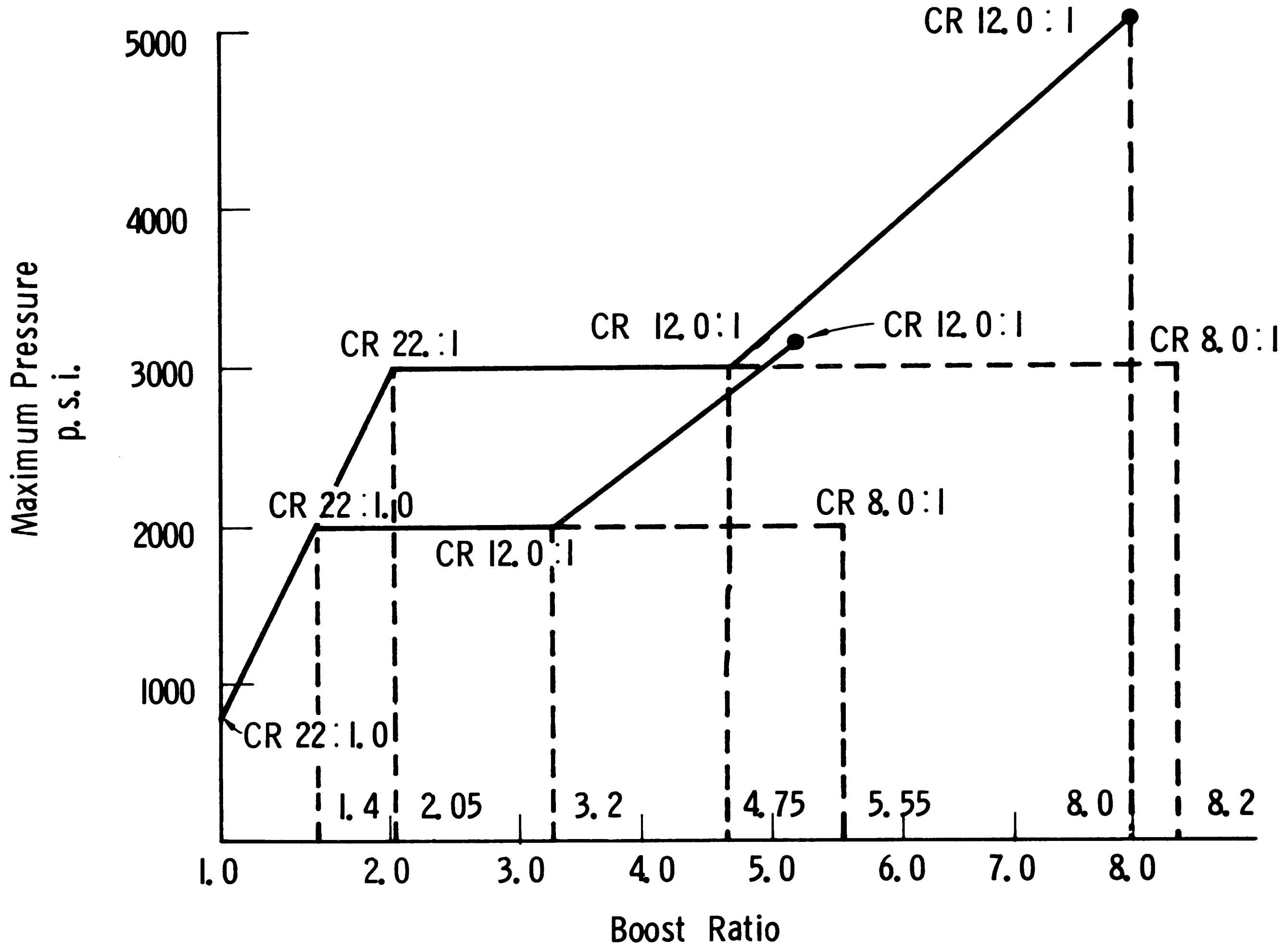

Figure 1. Maximum pressure, compression ratio, and boost ratio employed.

TABLE I

HEAT FLOW TO COOLANT

Phase of Cycle	Heat Transfer: Percentage of Btu of Fuel at Boost Ratio = 1.0 Compression Ratio 22:1	Boost Ratio = 5.0 Compression Ratio 12:1
Compression	0.5	0.35
Combustion	1.2	1.00
Incomplete Combustion	2.0	1.50
Expansion	4.0	3.50
Exhaust	5.0	4.00
Loss to Oil	1.2	1.00
Miscellaneous	4.1	2.65
Total Percentage of Loss	18.0%	14.0%

Of the heat losses listed, the only ones affecting the cycle efficiency are the first four, plus one quarter of the sum of the last two per stroke. This assumes that these last losses are distributed uniformly over the whole cycle. Losses at other ratios were assumed to be in linear proportion on the basis of the two conditions given in the table. This is only an assumption, but is considered reasonably close for the present purpose. Actually it is believed that as boost increases, these losses decrease fairly rapidly at first and then tend to level off.

By use of the results thus gained and of the charts already mentioned we determined the value of n in PV^n= constant. This is the compression for the heat loss assigned over compression ratios of 13 to 17:1, for which the charts were originally designed. The result is plotted as the solid line in Figure 2, in which straight lines were obtained; these were extrapolated over the range of ratios from 9:1 to 22:1, shown by the dotted lines. Examination of Figure 2 reveals that n changes from an average of 1.364 to 1.377 over the total range of ratios at the normal air inlet temperature of 600° and from 1.355 to 1.368 at about 700°, the gas temperature value employed for high boost with aftercooling. In view of this modest change, and its small effect on the cycle, a constant value of 1.363 was finally used for all cases. For expansion after combustion the value of n varied even less, so that it was taken as 1.265.

The heat addition during the cycle was calculated as a heat addition at constant volume but variable specific heat from the end of compression to the maximum cylinder pressure, followed by heating at constant pressure until the total 700 Btu per lb of air had been accounted for, either as state change or as the result of heat loss to coolant.

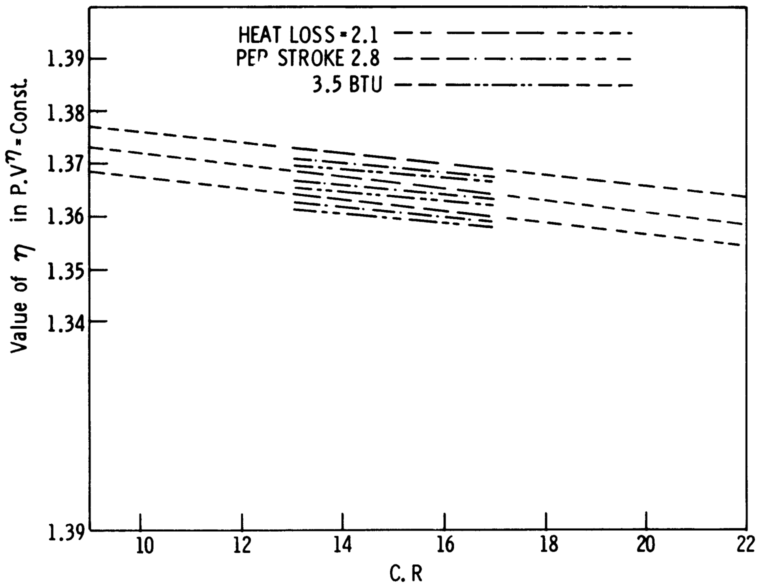

Figure 2. Index of compression.

With this information the work done, heat added, change of volume, etc., were obtained, and the ideal cycle efficiency, IMEP, IHP/lb of air, SFC/lb/IHP/hr, etc., were calculated for the complete range of boost and compression ratios desired. The results so obtained represent the practical ideal cycle without allowance for volumetric efficiency, mechanical losses, rounded corners of indicator diagram, etc.

IV. IDEAL PERFORMANCE

The results obtained are recorded in Table II for a limiting P_{max} of 2000 psi and in Table III for 3000 psi. They are plotted in Figures 3 and 4.

Using a naturally aspirated engine with 22:1 ratio, for easy starting, and a $P_{max}/P_{comp} = 1.5$, the boost was increased until $P_{max} = 2000$ rpm at a ratio of 22:1. After this point the compression was reduced as boost increased; as before, P_{max} was held at 2000 psi and P_{max}/P_{comp} at 1.5. Similar methods were employed for maximum pressure of 3000 psi. Note that the rpm of the engine enters into these relationships only indirectly, in the air flow. The data of Tables II and III, as far as IHP, etc., are concerned, are based upon an air flow of 1 lb/sec, which means that the rpm varies with cylinder displacement. To obtain the actual output per cubic inch of displacement for a given engine, the value given in the tables must be multiplied by the lb/sec of air flow and the ratio of total volume to displacement volume.

It must be remembered that the above results are for a volumetric efficiency of 100%, with no dilution by retained exhaust gas. In other words, we assumed that the clearance is scavenged perfectly, so that one pound of air completely fills the volume of displacement plus clearance.

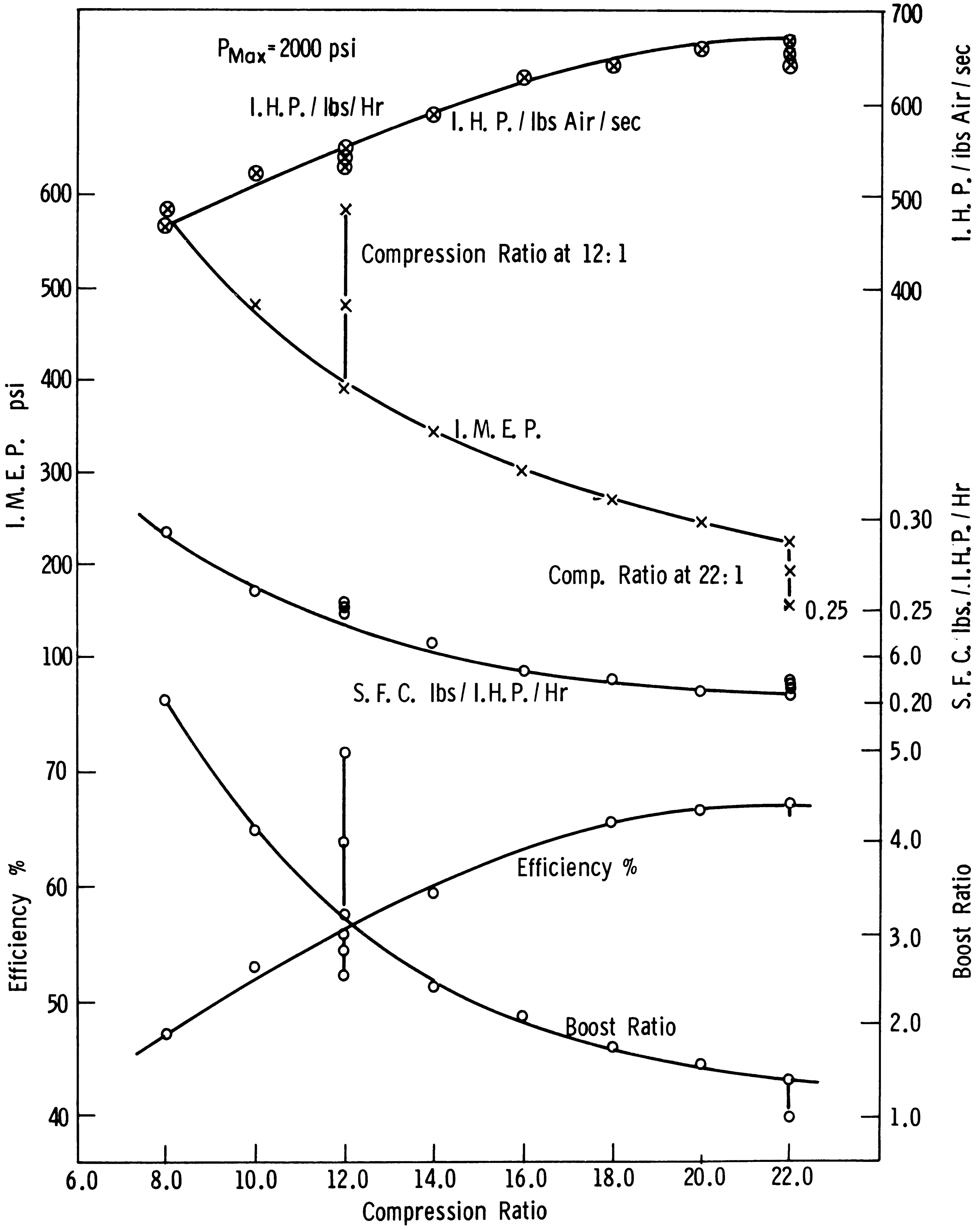

Figure 3. Ideal engine performance, P_{max} = 2000 psi.

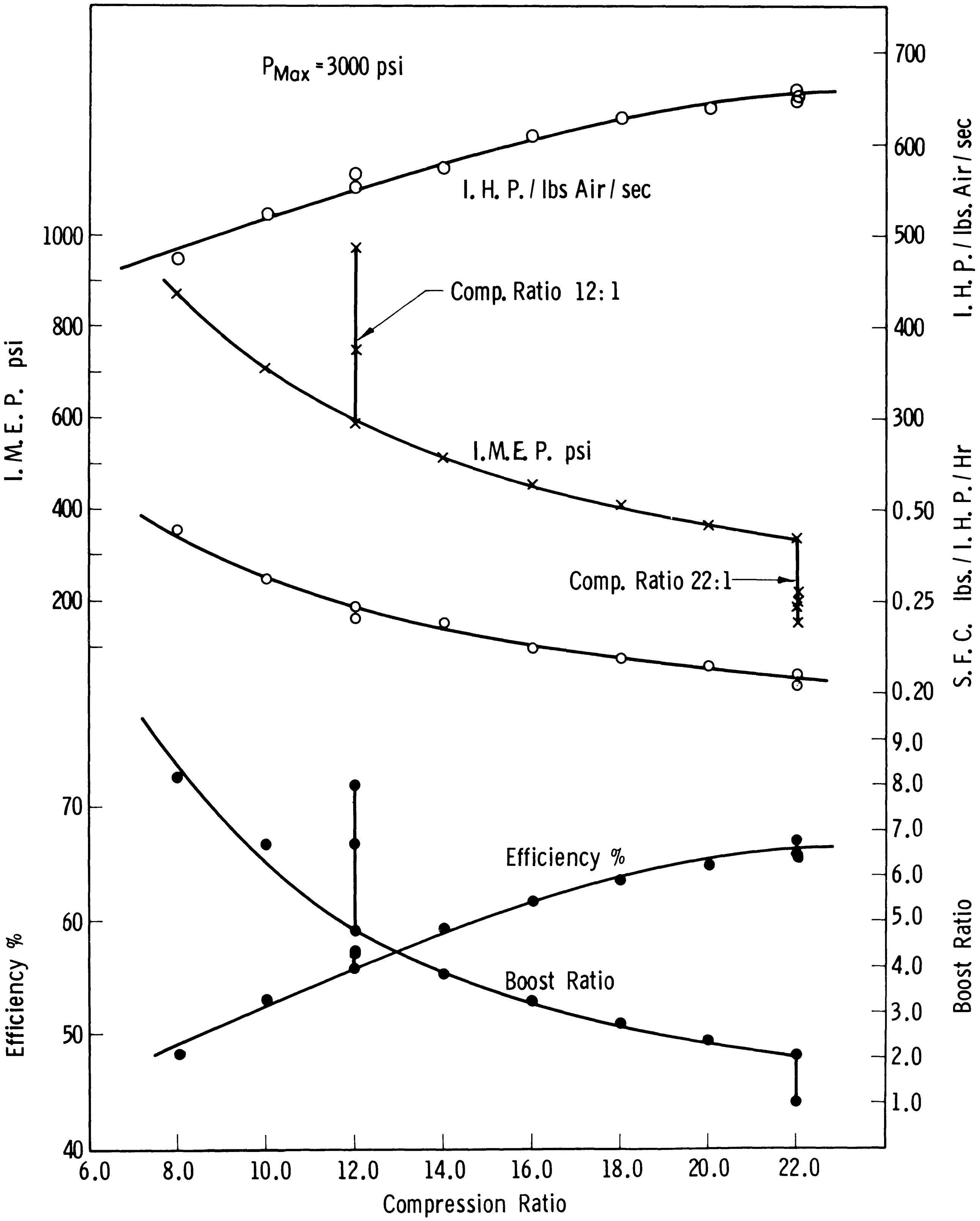

Figure 4. Ideal engine performance, P_{max} = 3000 psi.

TABLE II

PERFORMANCE WITH VCR PISTON SET FOR 2000 PSI

Compression ratio	22.0	22.0	22.0	22.0	20.0	18.0	16.0	14.0	12.0	12.0	12.0	10.0	8.0
Boost Ratio	1.0	1.2	1.3	1.4	1.58	1.84	2.11	2.43	3.20	4.0	5.0	4.12	5.55
P_{comp}	932	1156	1244	1330	1330	1330	1330	1330	1330	1695	2130	1330	1330
P_{max}	1400	1735	1868	2000	2000	2000	2000	2000	2000	2545	3198	2000	2000
Net Work (Btu)	458.0	461	459	474	469	455	446	417	389	382	374	371	331
Thermal efficiency (%)	65.6	65.8	65.6	67.6	67.0	64.9	63.6	59.6	55.6	54.6	52.4	52.9	47.2
IMEP (psi)	157.5	193.6	205	225	246	272	304	346	392	483	586	483	585
IHP (lb air/sec)	650	654	649	670	663	643	630	588	550	542	530	524	468
SFC (lb/IHP/hr)	0.210	0.209	0.211	0.204	0.207	0.213	0.218	0.233	0.249	0.252	0.258	0.261	0.292
Cylinder volume (cu ft/lb/sec)	16.51	13.52	12.67	11.92	10.86	9.57	8.46	6.99	5.85	4.66	3.76	4.60	3.49
Pressure at exhaust	52	62	67	71	82	95	111	134	166	203	248	218	298
HP (cu in of total vol/lb/sec)	0.273	0.336	0.356	0.390	0.424	0.467	0.518	0.582	0.653	0.808	0.98	0.792	0.932

TABLE III

PERFORMANCE WITH VCR PISTON SET FOR 3000 PSI

Compression ratio	22.0	22.0	22.0	22.0	22.0	20.0	18.0	16.0	14.0	12.0	10.0	8.0	12.0	12.0
Boost ratio	1.0	1.2	1.3	1.4	2.05	2.34	2.74	3.21	3.84	4.75	6.7	8.17	6.7	8.0
P_{comp}	932	1156	1244	1330	2000	2000	2000	2000	2000	2000	2000	2000	2570	3410
P_{max}	1400	1735	1868	2000	3000	3000	3000	3000	3000	3000	3000	3000	3855	5170
Net Work (Btu)	458	461	459	474	464	454	446	433	414	391	370.5	336	404	402
Thermal efficiency (%)	65.6	65.8	65.6	67.6	66.3	64.8	63.7	61.8	59.2	55.9	53.0	48.0	57.7	57.4
IMEP (psi)	157.5	193.6	205	225	343	371	414	459	516	593	716	878	752	979
IHP (lb air/sec)	650	654	649	670	657	640	629	611	574	552	523	475	569	567
SFC (lb/IHP/hr)	0.210	0.209	0.211	0.204	0.209	0.214	0.218	0.224	0.238	0.248	0.262	0.288	0.241	0.242
Cylinder volume (cu ft/lb/sec)	16.51	13.52	12.67	11.92	7.65	6.96	6.15	5.43	4.67	3.89	3.11	2.365	3.16	2.42
Pressure at exhaust	52	62	67	71	109	123	145	166	199	248	327	450	310	411
HP (cu in of total vol. per lb/sec.)	0.273	0.336	0.356	0.390	0.596	0.639	0.710	0.782	0.854	0.964	1.17	1.398	1.252	1.615

V. ENGINE PERFORMANCE

The above ideal performance characteristics include an approximation for the cyclic heat losses, as they affect power output and fuel economy. To obtain a description of an actual engine that can be compared with variable-compression-ratio tests, the results of dilution of the charge by exhaust left over from the previous cycle must be allowed for, as must the rounded (rather than square) corners of the indicator diagram, the effects of volumetric efficiency, and the mechanical efficiency of the engine. For convenience, the performance was based upon results reported here.

CHARGE DILUTION AND VOLUMETRIC EFFICIENCY

The actual cylinder charge is a mixture of air, fuel, and exhaust gases left in the clearance space from the preceding cycle. The amount of residual gas present is difficult to determine, for it varies greatly with valve overlap, pressure difference between inlet and exhaust manifold, dynamic effects of the inlet and exhaust system, etc. To approximate this value for an engine with well designed manifolds, valves, etc., we have assumed that scavenging of the clearance space is uniform, leaving 25% of the residual gas trapped. The maximum air charge in the cylinder, then, is given by Equation (1), for a total cylinder volume containing 1 lb of air:

$$\text{Air charge per cu ft of total volume} = 1.0 - \frac{1.0}{R - 1} \times 0.75 \text{ lb} \tag{1}$$

$$= 1.0 - \frac{0.75}{R - 1} \text{ lb}$$

where R = compression ratio.

It is recognized that the above equation includes no factor for the varying boost ratio, which tends to improve scavenging as it increases. However, as the boost increases, the compression ratio decreases; hence the clearance volume to be scavenged increases, and the air flow must be greater during valve overlap to scavenge the assumed 25% of the clearance. Thus it is hoped that between these two effects—boost ratio and clearance volume—a good approximation will be obtained.

The effect of volumetric efficiency is mainly one of pressure reduction and temperature increase between the atmospheric conditions and the charge in the cylinder when the engine is operating naturally aspirated. For the unboosted condition, atmospheric pressure in the cylinder was assumed to be 14.7 psi, and the temperature 100°F. If at BDC these values become 13.7 and 150°F, then without scavenging the maximum volumetric efficiency is

$$\eta_v = \frac{13.7 \times 560}{14.7 \times 610} \times 100 = 85.5\% \qquad (2)$$

$$= \text{weight trapped in cylinder}$$

Now engine tests indicate that η_v increases with boost, to over 100% at high ratios. We will assume that η_v reaches 98% at a boost ratio of 5:1, and that it changes linearly with ratio. Thus Figure 5 would represent an approximate plot of η_v vs. boost. The relation between η_v and x for this line is

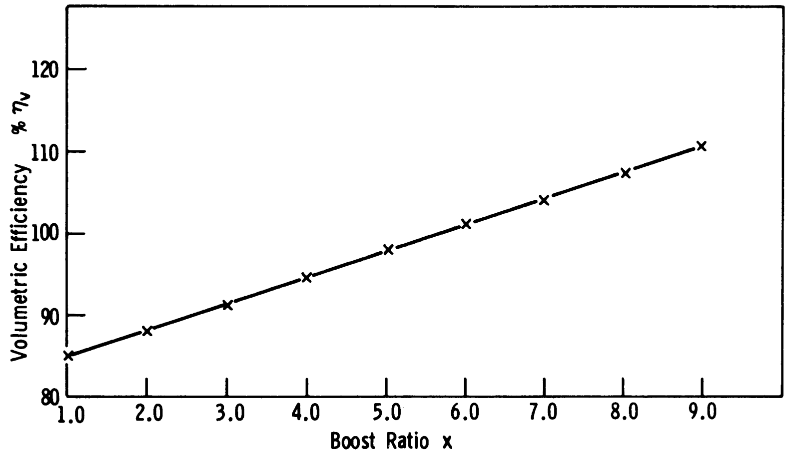

Figure 5. Volumetric efficiency and boost ratio.

given by

$$\eta_v = 3.25x + 81.75 \qquad (3)$$

It follows that the new air change in lb per cycle will be given approximately by

$$\text{air change} = (1.0 - \frac{0.75}{R-1})\ (3.25x + 81.75)\ \text{lb/cycle} \qquad (4)$$

Equation (4) applies to a cylinder which contains 1 lb of air per cycle when scavenging is complete.

The results of plotting Equation (3) against boost ratio are also shown in Figure 5. Note that when the boost ratio is 9:1, η_v, calculated normally, is 111%. This may seem high, but at such a high ratio the pressure at the inlet and exhaust manifolds will differ by about 20 psi when normal pressure drop for turbocharging is used. With valve overlap, good scavenging of the clearance space is then possible. If the compression ratio is 8:1 for a boost ratio of 9:1, a volumetric efficiency of 115% could be achieved.

From Figure 5, in conjunction with Equation (4) and the boost-compression relationship of Figures 3 and 4, we have calculated the expected mass air flow for a cylinder having a total capacity of 1 lb of air per cycle. The results are shown in Figure 6, for a P_{max} of 2000 psi, and in Figure 7 for 3000 psi.

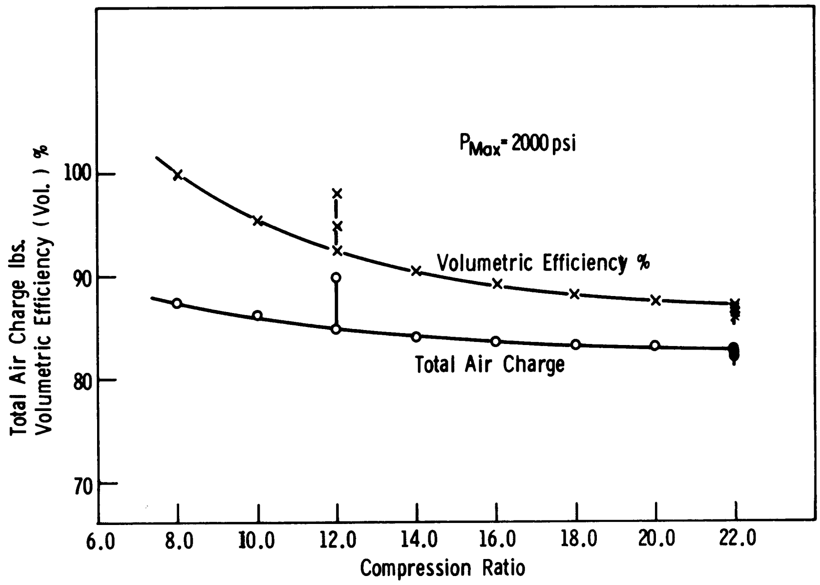

Figure 6. Cylinder air charge, P_{max} = 2000 psi.

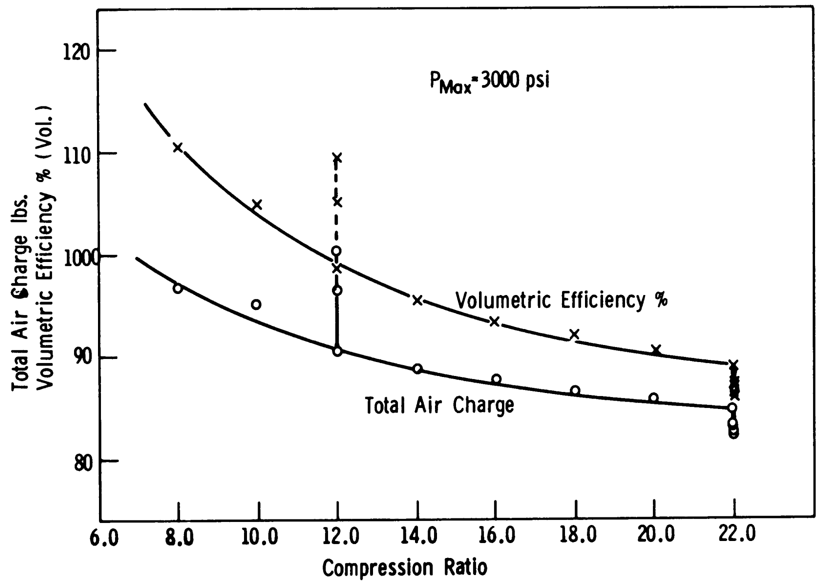

Figure 7. Cylinder air charge, P_{max} = 3000 psi.

These values were calculated as follows:

Example. Let boost ratio be 1.4:1 at a compression ratio of 22:1, calculate the mass of new air inducted per cycle.

From Figure 5 or Equation (3), volumetric efficiency = 86.2 = η_v. This is based upon the usual definition for η_v and thus refers to the engine displacement volume, not to total volume.

Translated to total cylinder volume, Equation (4) gives

$$\text{air change} = \left(1.0 - \frac{0.75}{R-1}\right)(3.25x + 81.75)$$

$$= \left(1.0 - \frac{0.75}{21}\right) 86.2 = 0.964 \times 86.2$$

$$= 83.2\%$$

For a total cylinder volume equal to 1 lb of air per cycle, then, the volumetric efficiency becomes (83.2 x 22)/21 = 87%, rather then 86.2% as given by Equation (3).

The correction for clearance volume scavenging is seen to be small in this case, where the boost ratio is low and the compression ratio high. From Figure 6 for P_{max} = 2000 psi the difference becomes (100.3 - 98.2) = 2.1% when the boost ratio is 5.55:1 and compression ratio is 8:1, whereas at P_{max} = 3000 psi with 8.17:1 boost and compression of 8:1, the difference between the values in 5 and 7 becomes (110.5 - 108) = 2.5%

It is now possible to obtain the effective air charge to the cylinder for all the supercharge ratios used over the range of compression ratios given by the VCR piston.

The results of these calculations are given on lines 6, 7, and 8 of Table IV. Line 9 gives a correction factor, necessary because up to this point the indicator diagram has been assumed to have square corners rather than the actual rounded ones. The correction factor has been set at 95% for a naturally aspirated engine, and increases, with the diagram area, up to 99% at high boost ratios. By this means results predicted for the engine were calculated; they are given on lines 10, 11, and 12 for P_{max} = 2000 psi, and on lines 22, 23, and 24 for P_{max} = 3000 psi. All these figures, as corrected, are diagrammed in Figures 8 and 9.

PERFORMANCE OF VCR ENGINE

Let us now compare the predicted performance characteristics with the results of engine tests. These data are given in two different ways:

1. Engine performance was obtained with the piston locked at a series

TABLE IV

CORRECTED PERFORMANCE
(P_{max} = 2000 psi, F/A = 0.038)

Compression ratio	22	22	22	22	20	18	16	14	12	12	12	10	8
Boost ratio	1.0	1.2	1.3	1.4	1.58	1.84	2.11	2.59	3.20	4.0	5.0	4.12	5.55
Charge density (lb/cu ft)	0.0605	0.074	0.079	0.0839	0.092	1.047	1.184	1.432	1.735	0.215	0.266	0.2175	0.286
Cylinder volume (1 lb of air)	16.51	13.52	12.67	11.92	10.86	9.57	8.46	6.99	5.85	4.66	3.76	4.60	3.49
Charge wt (lb)	0.82	0.824	0.828	0.830	0.832	0.833	0.838	0.840	0.850	0.872	0.900	0.862	0.876
IMEP (psi)	129.1	159.5	169.6	186.6	204.5	226.5	254.5	290.6	333.0	421.0	526.0	417.0	512.0
HP (lb air/sec)	532	538	568	556	552	535	528	494	468	472	476	453	409
SFC (lb/IHP/hr)	0.257	0.254	0.241	0.246	0.249	0.256	0.260	0.277	0.292	0.289	0.287	0.301	0.334
Correction factor	0.95	0.953	0.956	0.96	0.963	0.966	0.97	0.973	0.976	0.98	0.983	0.986	0.99
Corrected IMEP	123	152	162	179	196.8	219	247	283	325	412	517	411	507
HP (lb air/sec)	505	512	543	533	531	516	512	480	457	462	468	447	405
SFC	0.271	0.267	0.252	0.257	0.259	0.265	0.268	0.285	0.299	0.295	0.292	0.305	0.337

Compression ratio	22	20	18	16	14	12	12	12	10	8.0 P_{max} = 3000 psi, F/A = 0.038
Boost ratio	2.05	2.34	2.74	3.21	3.84	4.75	6.7	8.0	6.7	8.17
Charge density	.1307	.1437	.1625	.1842	.2141	.257	.3165	.4132	.3217	.422
Cylinder volume (1 lb of air)	7.65	6.96	6.15	5.43	4.67	3.89	3.16	2.42	3.11	2.37
Charge wt (lb	0.852	0.859	0.866	0.877	0.887	0.906	0.965	1.005	0.950	0.967
IMEP (psi)	292	319	358	402	458	537	725	984	681	848
HP (lb air/sec)	560	550	544	536	509	500	549	570	498	549
SFC (lb/ IHP/sec)	0.245	0.249	0.252	0.255	0.268	0.274	0.250	0.240	0.275	0.298
Correction factor	0.96	0.963	0.966	0.97	0.973	0.976	0.98	0.983	0.986	0.99
Corrected IMEP	280	307	346	390	445	524	710	967	671	840
HP (lb air/sec)	538	529	525	520	495	488	538	560	491	455
SFC	0.255	0.259	0.261	0.263	0.275	0.281	0.255	0.244	0.279	0.301

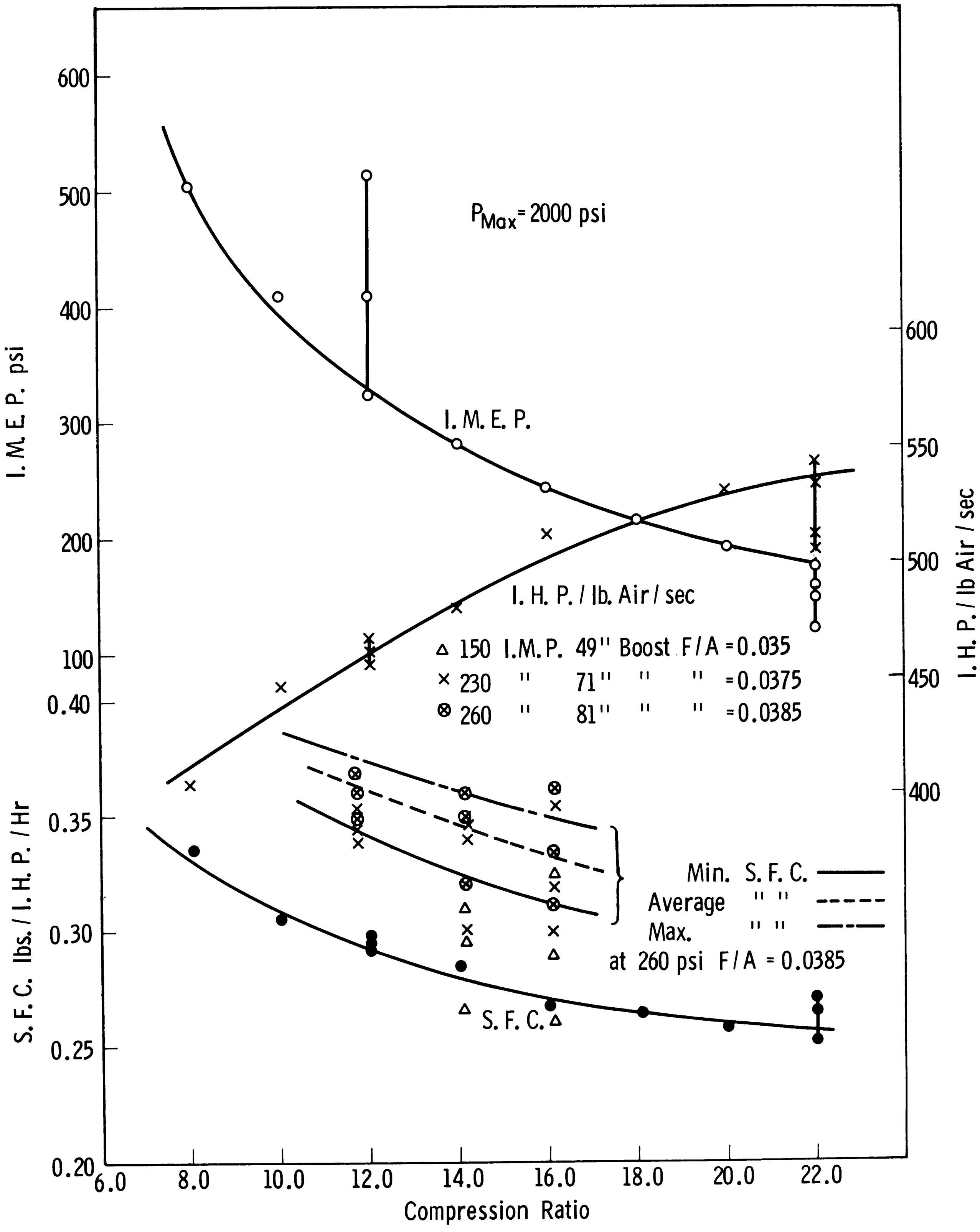

Figure 8. Corrected performance, P_{max} = 2000 psi.

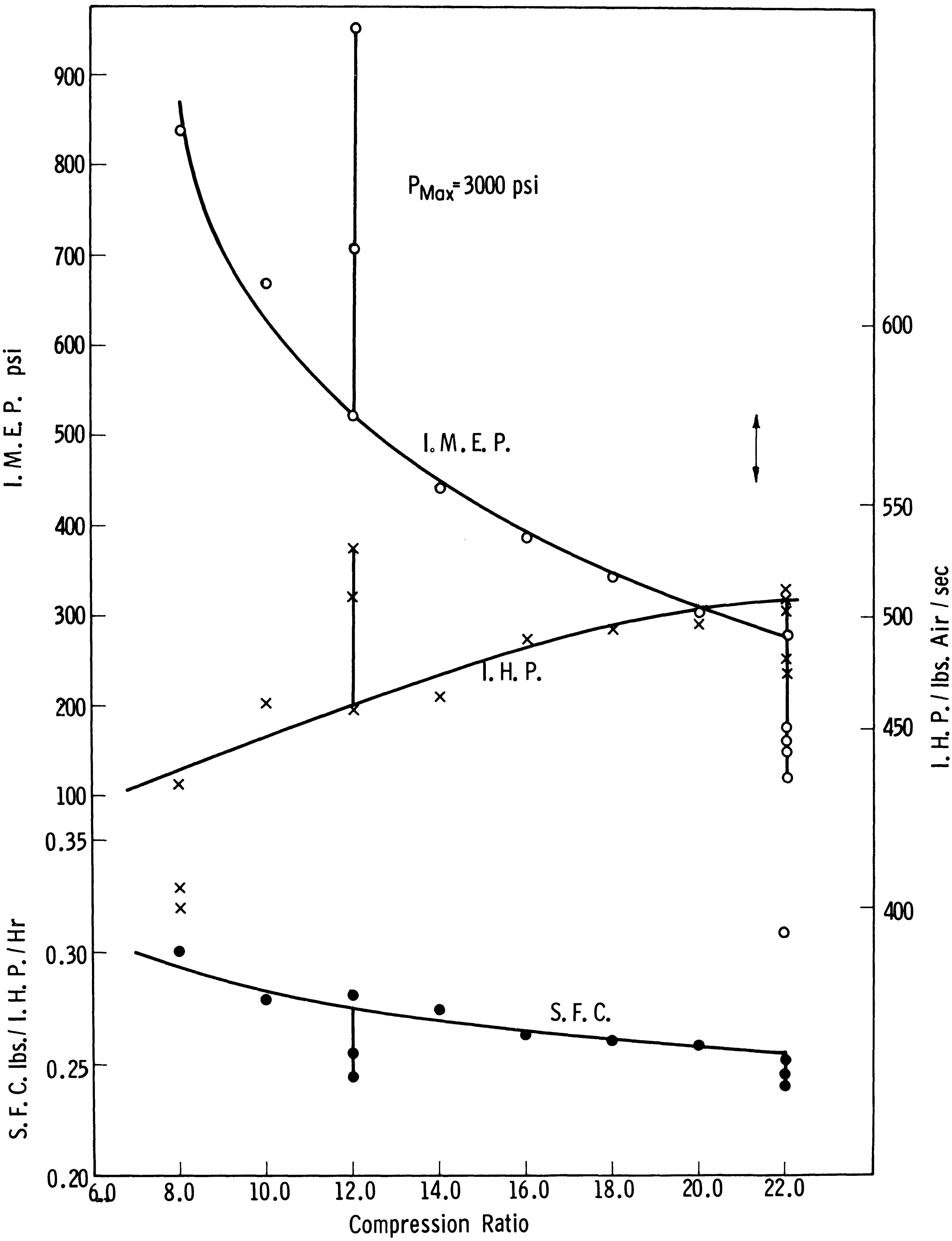

Figure 9. Corrected performance, P_{max} = 3000 psi.

of fixed compression ratios, for various injector adjustments at each position. The results thus represent the best performance, the average, and the worst, obtainable for the given combustion shape at each compression ratio.

2. Engine performance was obtained with the VCR piston in operation, so that the ratio was varied automatically with load; the same fixed-injection system characteristic was used for all ratios and loads. In other words, the second set of results is for normal operation of the VCR system.

Figure 10 shows the results of the first series of tests in which compression ratios of 11.7, 14.1, and 16.1 were used, with the engine set for a gas pressure of 2000 psi. Each graph consists of three lines, a solid one and two broken ones. The solid line is drawn through the points obtained by averaging all the data available for different injection characteristics, etc.; the lower broken one is drawn through the best specific consumption data, and the upper one through the worst.

Figure 10 is not exactly comparable with Figure 8, because of variations in F/A and turbocharger ratio in the VCR engine tests, but the points are shown in Figure 8. In general the F/A ratio is higher than the theoretical curves because the latter are obtained for complete combustion whereas the former is naturally affected by some incomplete combustion products. If follows that for any given turbocharger ratio the IMEP developed will be somewhat less then the calculated values even if the F/A ratio is same.

The next step was the combination of the data on the VCR engine in normal operation with the data of Figures 8 and 10. The results are shown in Figure 11, together with some material on a smaller engine. On this diagram the performance of the test engine has been plotted at the IMEP of the engine test; the mean pressure curve of the theoretical calculations was used to locate the point. This was necessary because the engine-compression ratio was not known exactly, being a variable affected by many factors. Hence the ratio scale of Figure 11 is not exactly true except for the ideal case, but it is considered approximately so. The results shown are thus a first indication of the effect of an automatically variable compression ratio on ideal engine performance. Note that Figure 11 gives the average F/A ratio for the test results; it is considerably higher at 410 IMEP than that obtained in the ideal calculations, approximately the same at 300 psi, and lower at 200 psi and 140 IMEP; these differences, and their effect upon MEP, account for the change of slope of the curves.

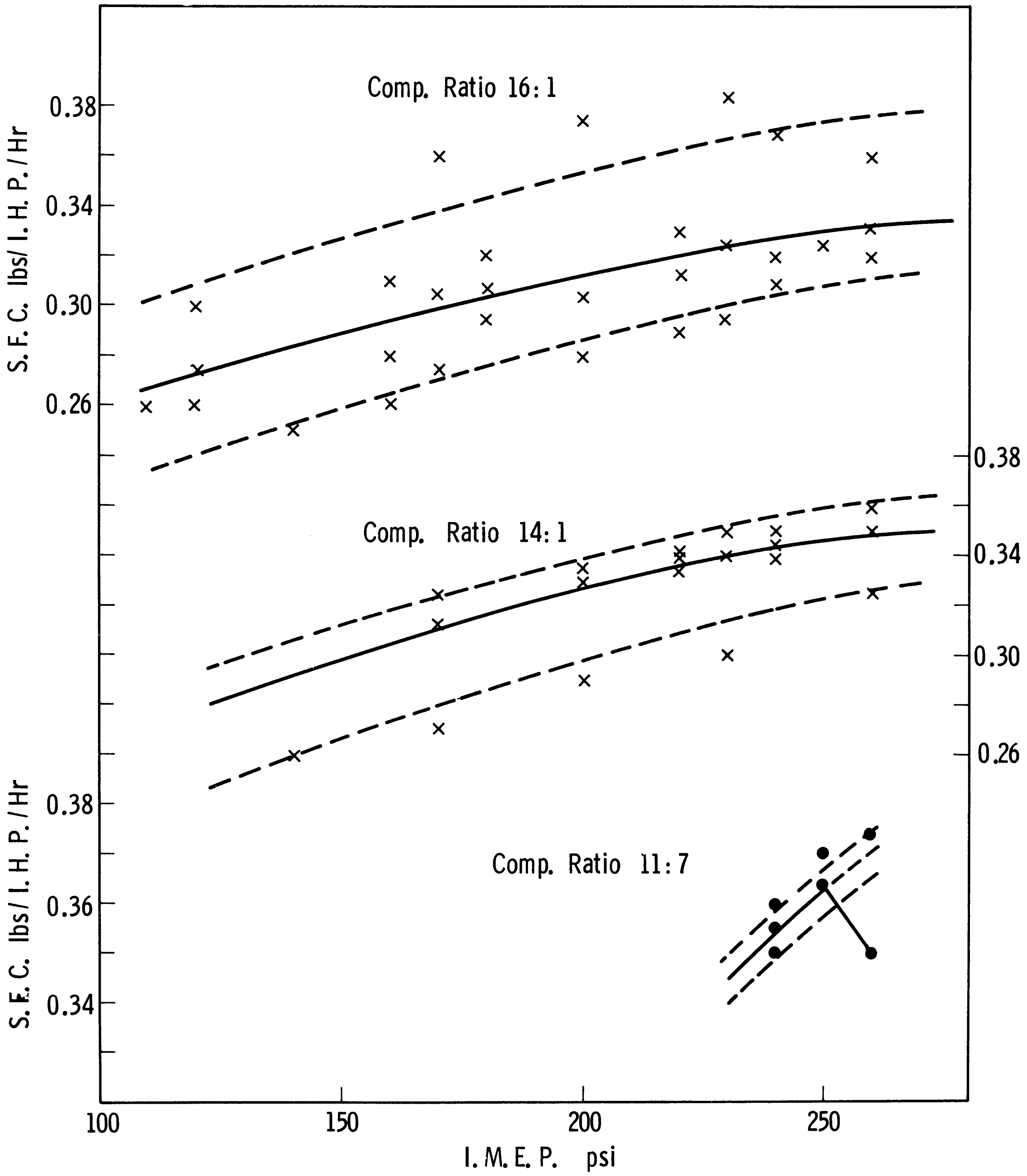

Figure 10. Performance at fixed compression ratios.

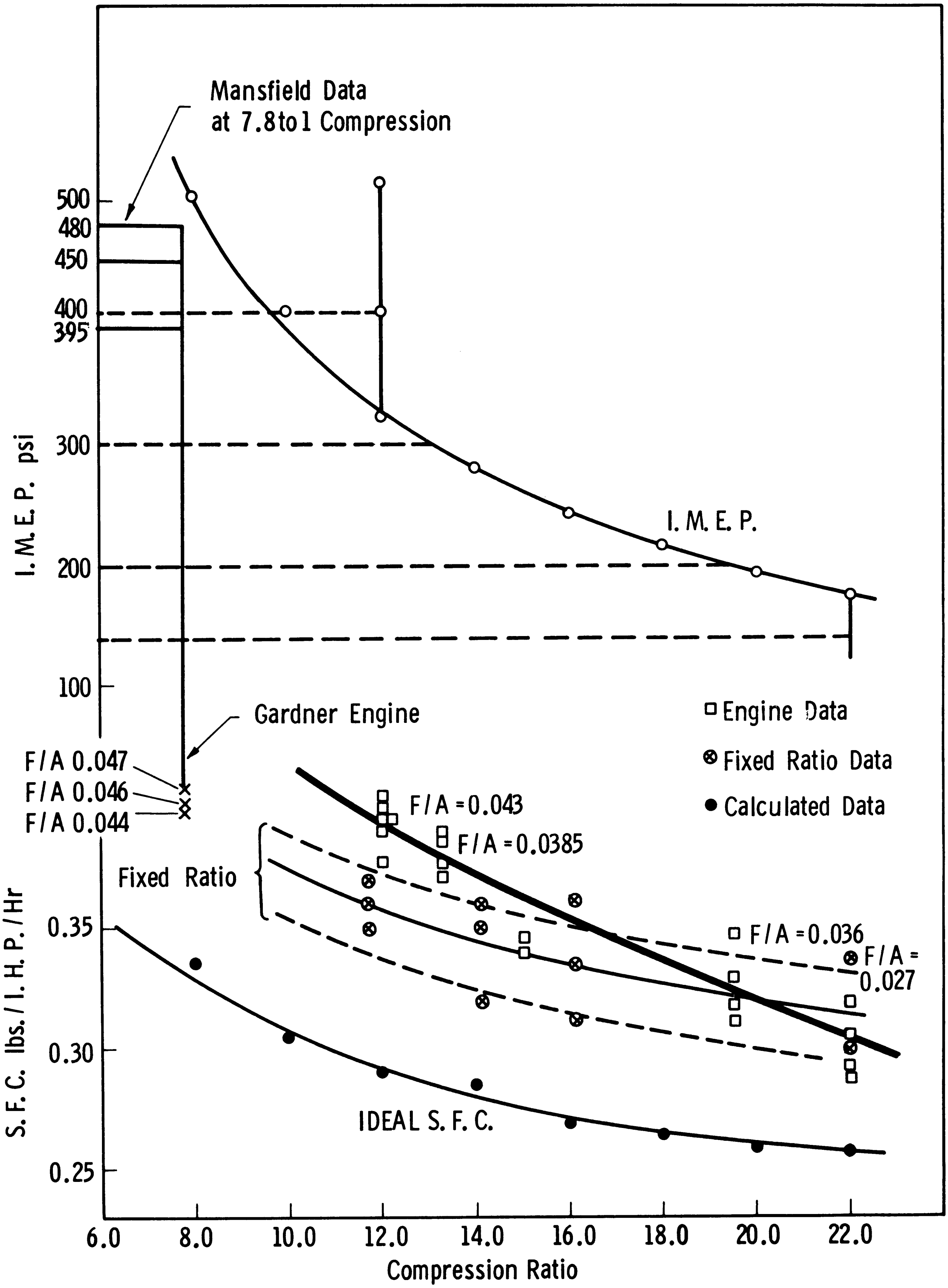

Figure 11. Comparison of ideal and engine results.

VI. DISCUSSION

This report is summarized in Figure 11, a comparison of the performance of the normally operating VCR engine, the fixed-compression-ratio engine, and the ideal one. To this diagram have been added Mansfield's results for the Gardner engine (Ref. 3), converted to IMEP as closely as possible. This engine was operated as a fixed-ratio one, with an injection characteristic carefully developed to suit the combustion chamber. It is seen that the results line up quite well with those for the assumed fixed-ratio engine, considering that the F/A ratio is different and that P_{max} = 1,600 psi rather than 2,000 psi.

The ideal data, at a F/A ratio of 0.038, naturally give the lowest SFC. The ratio between the ideal and fixed-ratio engines has an average of 0.80 when based upon the mean SFC curve for fixed ratios, and of 0.86 when evaluated from the minimum SFC curve. When the VCR engine curve is examined, this ratio ranges from about 75%, at F/A = 0.043, to 85%, at F/A = 0.027. Where the VCR engine F/A ratio is the same as the calculated results, the engine data roughly correspond with those for the fixed-ratio tests. Correction of the engine results at F/A = 0.043 to F/A = 0.038 gives a SFC of 0.36 lb/IHP/hr, which also gives substantial agreement with the fixed-ratio results.

VII. CONCLUSIONS

It can be concluded that:

1. As the compression ratio changes from 22:1 to 12:1, ideal thermal efficiency for the cycle decreases by about 11%, resulting in an increase of SFC from 0.258 to 0.292 lb/BHP/hr. In view of the great change in inlet air density, heat loss from cycle, volumetric efficiency, etc., accompanying this change of ratio and IMEP from 140 to 400 psi, the change in cycle efficiency is considered small.

2. The engine data for different fixed ratios shows the same general trend as the theoretical calculations. An efficiency ratio of about 0.86 is shown when based upon the performance of the best fixed-ratio engines.

3. The VCR engine results examined had a widely varying F/A ratio over the range of compression ratios employed, so that direct comparison with the theoretical data was difficult. However, correcting the VCR data in direct proportion to the F/A ratio in the high-power range resulted in substantial agreement with the results for the fixed-ratio engine.

4. In the present engines, it can be concluded, little performance (possibly 10%) is lost when the injection characteristics are correctly adjusted to allow for the great change in chamber shape that occurs as the compression ratio varies.

5. Eliminating the multi-fuel requirement, or even just gasoline starting, would permit a lowering of the maximum compression ratio from 22:1 down to 17 or 18:1; this would be a considerable improvement in the usable mean pressure, and smaller changes in SFC. Further, mean effective pressures of 500-600 psi could at least be considered. Probably no increase in peak cylinder pressure would occur in the process.

6. The increased SFC at full load and speed (about 10% because of the use of the VCR principle) has an extremely small effect upon the overall fuel consumption per 24 hr when the engine is operating on the present duty cycle since the VCR engine is more efficient at part load than is the standard one.

7. Increasing the maximum cylinder pressure from 2000 to 3000 psi would reduce the variation in SFC with compression ratio. It would also make possible an IMEP of 700-800 psi, if the necessary cooling and injection systems and turbochargers could be developed.

VIII. REFERENCES

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3. W. P. Mansfield and W. S. May, "Diesel Combustion at High MEP With Low Compression Ratio," SAE Paper 660343 (1966).