THE DEVELOPMENT OF AN ENGINE WITH A HIGHER COMPRESSION RATIO

THESIS

by

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In all internal combustion engines working on a cycle in which a combustible mixture is compressed prior to the ignition and expansion strokes, the efficiency is limited to the efficiency at which the maximum compression temperature is equal to the ignition temperature of the fuel used. For the internal combustion engines, such as are used in the automotive and airplane field the fuel is gasoline. This has a flash point at about 600° F., depending on the gravity of the fuel. Compression temperatures higher than this, cause preignition. In fact compression temperatures approaching this limit are difficult of realization because of detonation or knocking.

It is evident on analysis that the higher is the expansion ratio the higher will be the efficiency of any engine; for the larger is this ratio, the more will the heat of combustion be utilized in doing useful work. Due to mechanical reasons it is customary in engine work to keep the expansion ratio practically equal to the compression ratio, so that the efficiency is limited to that determined by the compression ratio, which is in turn limited by the final temperature of compression.

It has been the purpose of the investigation to develop a method which has been proposed for lowering the maximum compression temperature so that an increase in the expansion ratio could be realized. In this method cooling of the charge during the

PLATE I

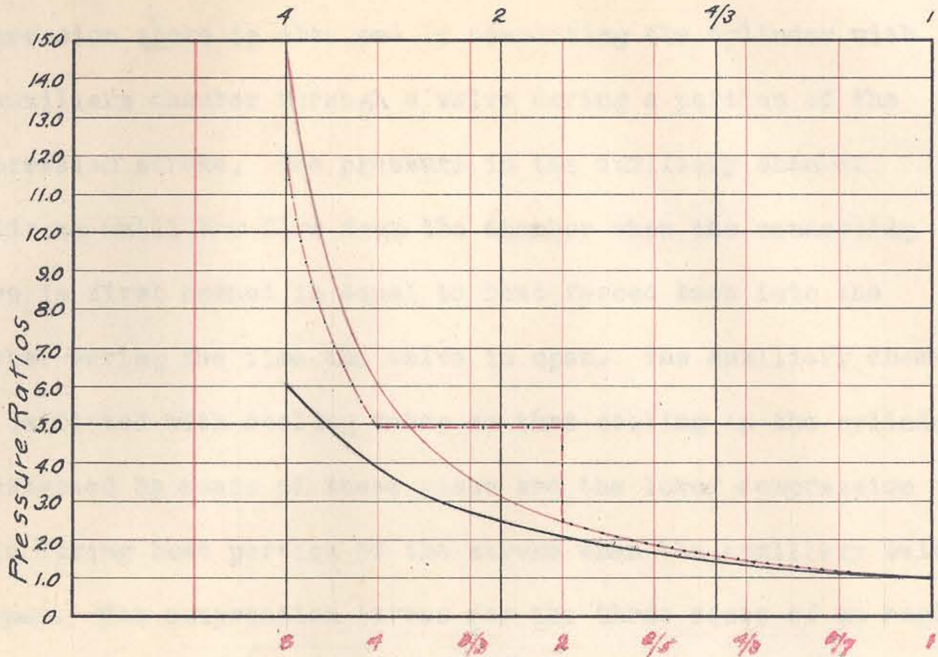


Fig. 1

Ordinary Cycle with 4:1 Compression Ratio

" " " 8:1 " "

Chamber Cycle " " " "

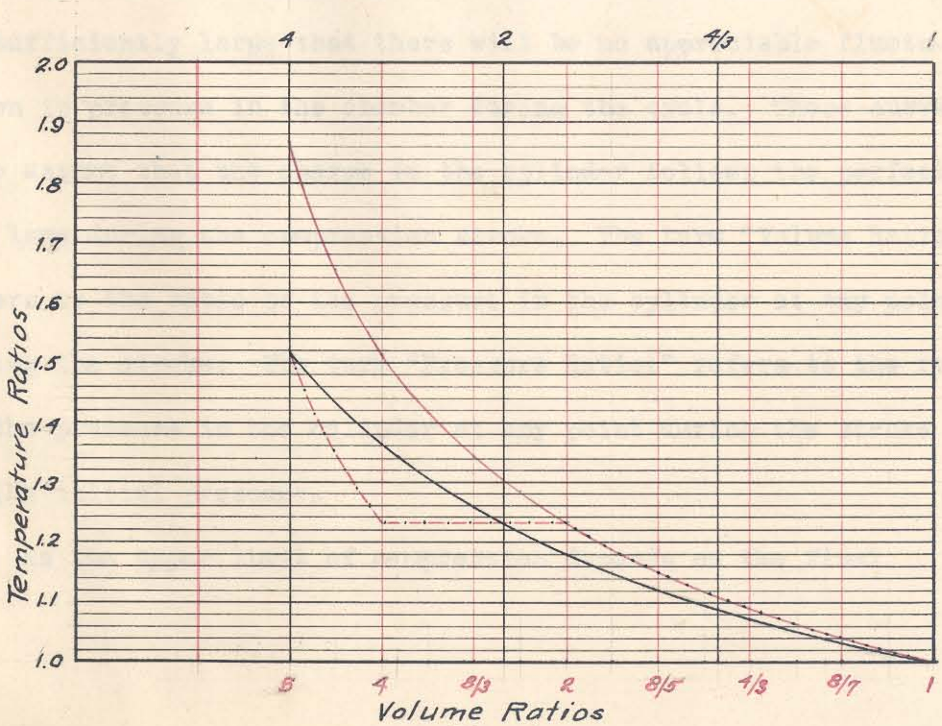


Fig. 2.

compression stroke is obtained by connecting the cylinder with an auxiliary chamber through a valve during a portion of the compression stroke. The pressure in the auxiliary chamber builds up until the flow from the chamber when the connecting valve is first opened is equal to that forced back into the chamber during the time the valve is open. The auxiliary chamber is fitted with cooling tubes so that cooling in the cylinder is affected by means of these gases and the lower compression ratio during that portion of the stroke when the auxiliary valve is open. The compression curves for the three cases of an engine with compression ratios of 4:1, and 8:1 with and without intermediate cooling are illustrated in fig. 1, Plate I.

For illustrative purposes these curves are computed, assuming that the ratio of the chamber volume to the cylinder volume is sufficiently large that there will be no appreciable fluctuation in pressure in the chamber during the cycle. These curves also assume that the charge in the cylinder follows the perfect gas laws during the compression stroke. The term "Volume Ratios" refers to the ratio of the pressure in the cylinder at any point during the stroke. The term "Pressure Ratios" refers to the ratio of the pressure in the cylinder at any point during the stroke to the initial pressure.

As the upper limit of compression depends on the final temperature, the ratio of the absolute temperature of the charge is shown in fig. 2, Plate I. As before the ratio of the chamber volume to the cylinder volume is assumed to be large so that no appreciable pressure fluctuations occur. In addition, the temperature of the mixture in the auxiliary chamber is assumed to be sufficiently low that the rise in temperature of the gas entering the cylinder neutralizes the rise in temperature of the cylinder charge as it is compressed by the chamber gas. Based on these assumptions the temperature of the charge in the cylinder will remain constant during that part of the stroke in which a portion of the charge is forced back into the chamber.

The losses of this cycle consist in this recompression of the chamber gas, but this loss is actually small compared to the total gain in efficiency.

These curves demonstrate the use of this cycle in increasing the compression ratio and the resulting efficiency. The actual computations involved will be found at the end of this discussion.

As has been previously mentioned, the greatest difficulty in the way of bringing the temperature of compression up to the flash point of the fuel used is detonation or knocking. The harmful effects caused by this difficulty can be decreased by a smaller spark advance. The result is a drop in power below the maximum that could be attained by means of a proper spark advance.

It has long been known that exhaust gas introduced as a diluent into the mixture decreases the tendency to detonate. In all the cases coming to my attention, the exhaust was cooled and introduced at the carburetor or intake manfold, thus replacing a portion of the fuel mixture. This effect seems to be due to two causes; (1) the decrease in total fuel used per stroke and; (2) a decrease in the rate of flame propagation. The first is obvious, as the dilution of the charge is at the expense of some of the fuel. This results in a decrease in power for a given displacement and compression ratio. The second is confirmed by experiments made at the University of Illinois Esperiment Station, by Ricardo and other investigators.

The difficulty in this method of dilution is the loss in power resulting from the replacement or fuel by exhaust. By the method of operation used in our cycle the dilution of the charge can be attained without any decrease in the quality of fuel taken into the cylinder; thus making three factors combine to increase the efficiency. (1) The charge is cooled by the incoming cooled exhaust, (2) the final temperature of compression is lowered due to the decrease in compression ratio during a portion of this stroke; and, (3) the spark advance for maximum power and efficiency can be attained by the proper volume ratios. An average set of results obtained by means of the standard head, and the high compression head using the two methods of operation are shown in tables I, II, III.

These tests were made on a four cylinder, Model R, Hupmobile engine using standard intake and exhaust valves and timing. This engine has a 5.5 inch stroke and $3\frac{1}{4}$ inch bore. Comparison tests for fuel consumption, power, and related values were made to determine the timing of the chamber valves to give maximum efficiency. Power tests were made on an electric dynamometer, in which tests, the quantities shown in the table of results were measured. The fuel consumption was determined by measuring the load on the dynamometer scales and the number of revolutions required to consume 155.6 cc of gasoline, the density of which was known. The dynamometer torque arm was arranged so that three pounds on the scale represented one HP output at 1000 revolutions per minute. Friction tests were made by motoring the engine and noting the load on the dynamometer.

The accompaning tables show a distinct gain in power and efficiency for the high compression cycles over the standard cycle. The mechanical difficulties, however, in the way of this cycle are several. The large number of valves required and the consequent increase in the number of cams and camshafts, and the noise resulting from the short duration of lift is probably the worst. Two valves are necessary in the operation of the chamber because (1) experiment has shown that one valve does not give sufficient exchange at the higher speeds, probably because of the inertia of the gas column. (2) The gas first introduced into the cylinder at the beginning of compression would be the last to have left the cylinder in the previous cycle and consequently, the least cooled.

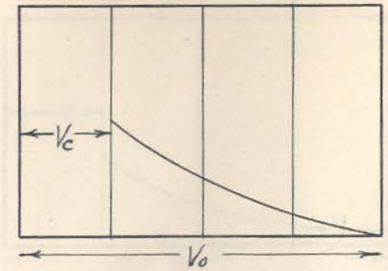
The conclusions to be drawn from the work to date establish the fact that the proposed cycle does permit higher compression ratios than are in common use for Otto Cycle engines at present. Although the gain in efficiency obtained is substantial, it is doubtful, at the present price of gasoline, whether this gain is sufficient to offset the complications involved.

Table IV shows the averages of the runs, and the percentage drop in fuel consumption below the consumption of the standard 4.55:1 engine. It is of some interest to note that the saving in fuel for throttle 5, which is only part throttle, (giving a compression pressure for the high compression engine approximately equal to the maximum compression pressure of the standard engine), is consistently equal to or higher than the saving at full throttle, as most passenger cars run at part throttle.

The tests were hardly fair to the cycle. The valves were operated by a loose chain-driven camshaft, giving the rather erractic results shown. Also the duration of each valve opening could be varied from 60° of crankshaft rotation only by decreasing the clearance of the cam follower. At present the engine is about ready to be tested using a specially designed head for all four cylinders. The camshaft is driven by the same Morse Silent Chain that drives the regular intake and exhaust camshaft. It is also made so that the timing can be easily and accurately changed, although the present duration of opening is still 60° on the crankshaft. This head is expected to give better and more consistent results.

It is evident that more must be done in determining the periods of opening and closing of the valves, duration of opening, and the relative value of the variations in the cycle. Further work must also be done to simplify the mechanical realization of the cycle. When this is accomplished the engine will have a chance in the commercial world.

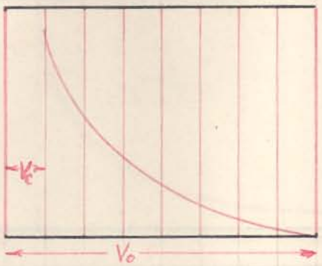
Computations for the Temperature Curve Fig. 2



4:1 ratio - ordinary cycle.

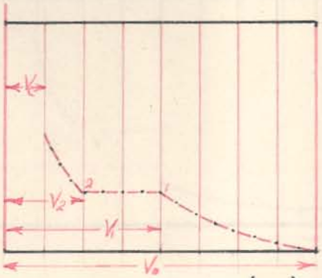
$$\left(\frac{T}{T_o}\right) = \left(\frac{V_o}{V}\right)^{n-1}$$

V_o/V =	1	4/3	2	4
T/T_o =	1	1.090	1.231	1.516



8:1 ratio - ordinary cycle

V_o/V =	1	4/3	2	4	8
T/T_o =	1	1.090	1.231	1.516	1.866



8:1 ratio - chamber cycle

From V_o to V_1 $\left(\frac{T}{T_o}\right) = \left(\frac{V_o}{V}\right)^{n-1}$

From V_1 to V_2 T is constant

From V_2 to V_c $\left(\frac{T}{T_2}\right) = \left(\frac{V_2}{V}\right)^{n-1}$

but $T_1 = T_2 = T_o\left(\frac{V_o}{V_1}\right)^{n-1}$ $\therefore \frac{T}{T_o} = \left(\frac{V_o}{V_1}\cdot\frac{V_2}{V}\right)^{n-1}$

For T_c to be equal to the temperature at the end of the 4:1 compression ratio

$$\left(\frac{V_o}{V_1}\cdot\frac{V_2}{V_c}\right) = 4, \quad \text{but } \frac{V_o}{V_c} = 8 \text{ ; so } V_1 = 2V_2$$

Also $\left(\frac{V_o}{V_1}\cdot\frac{V_1}{V_2}\cdot\frac{V_2}{V_c}\right) = 8$; take $\frac{V_o}{V_1} = \frac{V_1}{V_2} = \frac{V_2}{V_c} = 2$

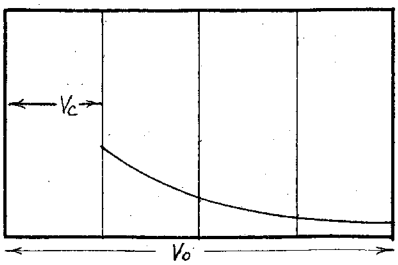
Then for the second part of the compression curve

$$\frac{T}{T_o} = \left(\frac{V_o}{V_1}\cdot\frac{V_2}{V}\right)^{n-1} = \left(\frac{V_o}{2V}\right)^{n-1}$$

$\frac{V_o}{V}$	1	8/7	4/3	8/5	2	8/3	4	8
$\frac{T}{T_o}$	1	1.038	1.090	1.151	1.231	------	1.231	1.516

For these computations, the value of the exponent "n" is taken as 1.3.

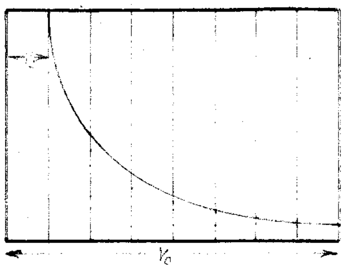
Computations for the Compression Curve Fig. 1



For 4:1 ratio - ordinary cycle

$$\frac{P}{P_1} = \left(\frac{V_1}{V}\right)^n$$

V_1/V =	1	4/3	2	4	8
P/P_1 =	1	1.45	2.46	6.06	14.9



For 8:1 ratio - ordinary cycle

V_1/V =	1	4/3	2	4	8
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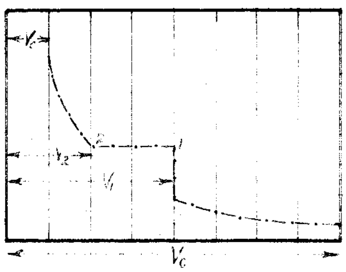

For 8:1 ratio - chamber cycle

From V_o to V_1; $\left(\frac{P}{P_o}\right) = \left(\frac{V_o}{V}\right)^n$

From V_2 to V_c; $\left(\frac{P}{P_2}\right) = \left(\frac{V_2}{V}\right)^n$

Wt. of gas in cylinder at volume V_2 = wt. at volume V_o when equilibrium between the chamber and the cylinder has been established.

$$\frac{P_2V_2}{T_2} = \frac{P_oV_o}{T_o} \quad \text{or} \quad \frac{P_2}{P_o} = \left(\frac{V_o}{V_2}\cdot\frac{T_2}{T_o}\right)$$

Then for the compression between V_2 and V_c

$$\frac{P}{P_o} = \left(\frac{V_o}{V_2}\cdot\frac{T_2}{T_o}\right)\left(\frac{V_2}{V}\right)^n \;;\; \frac{V_o}{V_2} = 4 \;;\; \frac{T_2}{T_o} = \frac{T_1}{T_o} = 1.231; \; \frac{V_o}{V_2} = 4$$

so $$\frac{P}{P_o} = 4.92\left(\frac{V}{4V}\right)^n$$

V_o/V =	1	8/7	4/3	8/5	2	8/3	4	8
P/P_o =	1	1.19	1.43	1.84	2.46 4.92	-------	4.92	12.12

For these computations the value of the exponent "n" is taken as 1.3.

Computations for Fuel Consumption

d = Density of fuel

N = Total number of revolutions

L = Load on dynamometer arm (sum of Brake and Resistance Loads)

$$\text{Fuel consumed per HP Hr} = \frac{\text{weight of fuel used} \times 1000 \times 60}{N \times L/3}$$

Weight of fuel used = 155.6xd gms = 155.6xdx0.002205 lbs.

$$\text{Fuel consumed per HP Hr.} = \frac{61,700d}{LN}$$

Table I

Standard Cycle 4.55:1 Compression Ratio

Jet	Load Lbs.	Throttle	Revs.	Temp. °F.	Spark Advance Deg.	Resis. Load Lbs.	Speed RPM	Fuel Consumption Lbs. Per B.HP.Hr.	I.HP.Hr.
16	6.24	5	6193	140	30	5.7	1000	1.30	.63
16	11.85	F	4532	127	22	5.6	"	.86	.59
15	11.80	F	5038	150	24	5.45	"	.78	.53
15	6.88	5	6696	135	29	5.6	"	1.00	.56
14	6.04	5	7621	156	34	5.8	"	1.01	.51
14	10.52	F	5723	132	30	5.5	"	.77	.51
15	7.05	5	6505	137	28	5.1	"	1.01	.58
15	12.03	F	4993	145	23	4.8	"	.77	.55
16	7.10	5	5963	143	26	5.2	"	1.10	.63
16	12.35	F	4413	140	22	5.1	"	.86	.60

Specific Gravity of Fuel 0.752

Note: The compression pressure at throttle 5 is approximately equal to the Compression pressure of the standard 4.55:1 engine used.

Table II

Original Chamber Cycle

Jet	Brake Load Lbs.	Throttle	Revs.	Jacket Temp. ^{o}F.	Spark Advance Deg.	Resis. Press. Lbs.	Speed RPM	Chamber Timing Deg. past L.D.C.	Fuel Consumption Lbs. Per B.HP.Hr.	Fuel Consumption Lbs. Per I.HP.Hr.	Chamber Press. Lbs.
14	8.3	5	6333	110	18	7.0	750	I 78-130	.88	.48	21
14	10.5	F	5767	130	20	7.2	975	II 57-117	.75	.43	28-30
14	10.6	F	5947	155	18	5.9	900		.73	.44	24-27
15	11.7	F	4740	120	10	6.8	750	I 80-139	.84	.53	30-32
15	12.8	F	4719	145	7.5	5.9	750	II 60-125	.77	.53	28-30
15	13.1	F	4708	160	7	5.3	750		.75	.53	28-30
15	9.2	F	5656	160	12	7.3	1050	I 100-160	.75	.50	28-40
15	9.5	F	5539	145	12.5	7.3	1050	II 60-125	.79	.50	27-39
15	11.5	F	5201	155	11	6.3	1000	I 90-150 II 60-125	.77	.50	30-36

Specific gravity of fuel = .748

Note: (1) The compression pressure at throttle 5 is approximately equal to the Maximum compression pressure of the standard 4.55:1 engine.

(2) I In the timing column refers to the timing of the first valve to open. II Refers to the second valve timing.

Table III

Supercharging by Exhaust

Jet	Brake Load Lbs.	Throttle	Revs.	Jacket Temp. °F.	Spark Advance Deg.	Resis. Load Lbs.	Chamber Press. Lbs.	Speed RPM	Chamber Timing Deg. past L. D. C.	Fuel Consumption Lbs. Per B.HP.Hr.	Fuel Consumption Lbs. Per I.HP.Hr.
16	8.35	5	6034	150	20°	6.1	14-22	1000	I 54-99	.92	.53
"	8.50	"	6094	"	"	"	"	"	II 270-340	.89	.52
"	8.80	"	6061	165	16	"	"	"		.87	.51
"	12.4	F	4524	140	6	6.5	20-32	"		.83	.54
"	12.5	"	4501	150	12	"	20-35	"		.82	.54
16	13.5	F	4457	160	16	6.1	20-35	1000	I 45-90	.77	.53
"	14.0	"	4439	"	22	"	20-33	"	II 291-320	.74	.52
16	9.2	5	5890	160	31	5.7	12-23			.85	.53
"	9.45	"	5902	165	28	"	"			.83	.52
15	9.3	5	6670	156	34	5.8	13-23		I 58-103	.74	.46
"	8.6	"	6748	160	17	5.85	9		II 280-340	.79	.47
"	11.6	F	5120	165	10	5.7	15-30			.77	.52
"	12.0	"	5156	160	9	"	18-30			.75	.51
14	10.8	F	5871	155	24	5.7	24-31		I 58-103	.73	.48
"	11.1	"	5906	160	24	"	24-31		II 280-340	.71	.47
"	11.6	"	5835	155	27	5.9	24-38			.68	.45
"	8.0	5	7419	140	30	5.7	19-24			.78	.46
"	8.2	"	7414	155	30	"	14-24			.76	.45
"	8.45	"	7336	158	33	6.2	14-24			.75	.43

Specific gravity of fuel = 0.745

Note: The compression pressure at throttle 5 is approximately equal to the compression pressure of the standard 4:1 engine used.

Table IV

Averages

Fuel Consumptions Per

Jet	Table	Throttle	B.HP.Hr.	% decrease below standard	I.HP.Hr.	% decrease below standard
16	I	F	.860		.595	
16	III	F	.755	13	.525	12
16	I	5	1.20		.63	
16	III	5	0.84	30	.525	16.7
15	I	F	.775		.54	
	II	F	.770	1.0	.50	7.4
	III	F	.760	1.9	.515	4.7
15	I	5	1.005		.565	
	III	5	0.765	24	.465	17.7
14	I	F	0.77		.51	
	II	F	0.74	3.9	.435	14.7
	III	F	0.703	8.7	.467	8.4
	I	5	1.010		.510	
	II	5	.880	12.9	.480	5.9
	III	5	.763	24.5	.447	12.4