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The effect of inlet air temperature on the efficiency of a spark ignition engine

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THE EFFECT OF INLET AIR TEMPERATURE ON THE EFFICIENCY OF A SPARK IGNITION ENGINE

BY

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A

THESIS

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PART I

INTRODUCTION

The object of this thesis is to determine the effect of inlet air temperature on the efficiency of an internal combustion engine. In most text books the subject is dismissed with mentioning the negative effect of warm air on the maximum output of the engine. This fact is recognized but since most engines are run at maximum output only a small percentage of the time, it would be possible to control the air temperature as the situation demanded. With the assumption that cold air could be used when full power is desired, it was decided to determine the effect of using warm air at part throttle operation.

There are various devices on the market today that claim to increase the efficiency of automobile engines. Among them are water injection, screens in the intake manifold, air bleeds, and many other patented devices. Some engine manufacturers use warmer intake manifolds than others claiming to gain advantage of a better distribution made possible from a more complete vaporization of the fuel. Maximum output, of course, is sacrificed for the claimed better economy.

The air standard cycle may be defined as a theoretical Otto cycle using air as a medium compared with the fuel air cycle which uses a mixture of fuel and air as medium.

The less fuel per pound of air (fuel air ratio) taken in by an internal combustion engine, the closer it approaches the air standard cycle. As can be seen in Figure 1,[1] the closer the fuel air cycle approaches the air standard cycle the more efficient the engine becomes. In a rich mixture the efficiency is low because there is not enough oxygen to burn all the fuel, and therefore, some of the fuel goes out the exhaust without being burned.

(1) Taylor and Taylor, The Internal Combustion Engine, International Textbook Co., p. 45.

Taking $CH_{2.25}$ as the formula of a typical gasoline and balancing the equation for chemically correct burning we obtain:

$$CH_{2.25} + 1.56O_2 = 6.86N_2 \rightarrow CO_2 + 1.125H_2O + 5.86N_2 \qquad (1)$$

The chemically correct fuel air ratio will be:

$$\frac{12 \times 1 + 1 \times 2.25}{32 \times 1.56 + 5.86 \times 28} = \frac{14.25}{50.0 + 164} = .0665$$

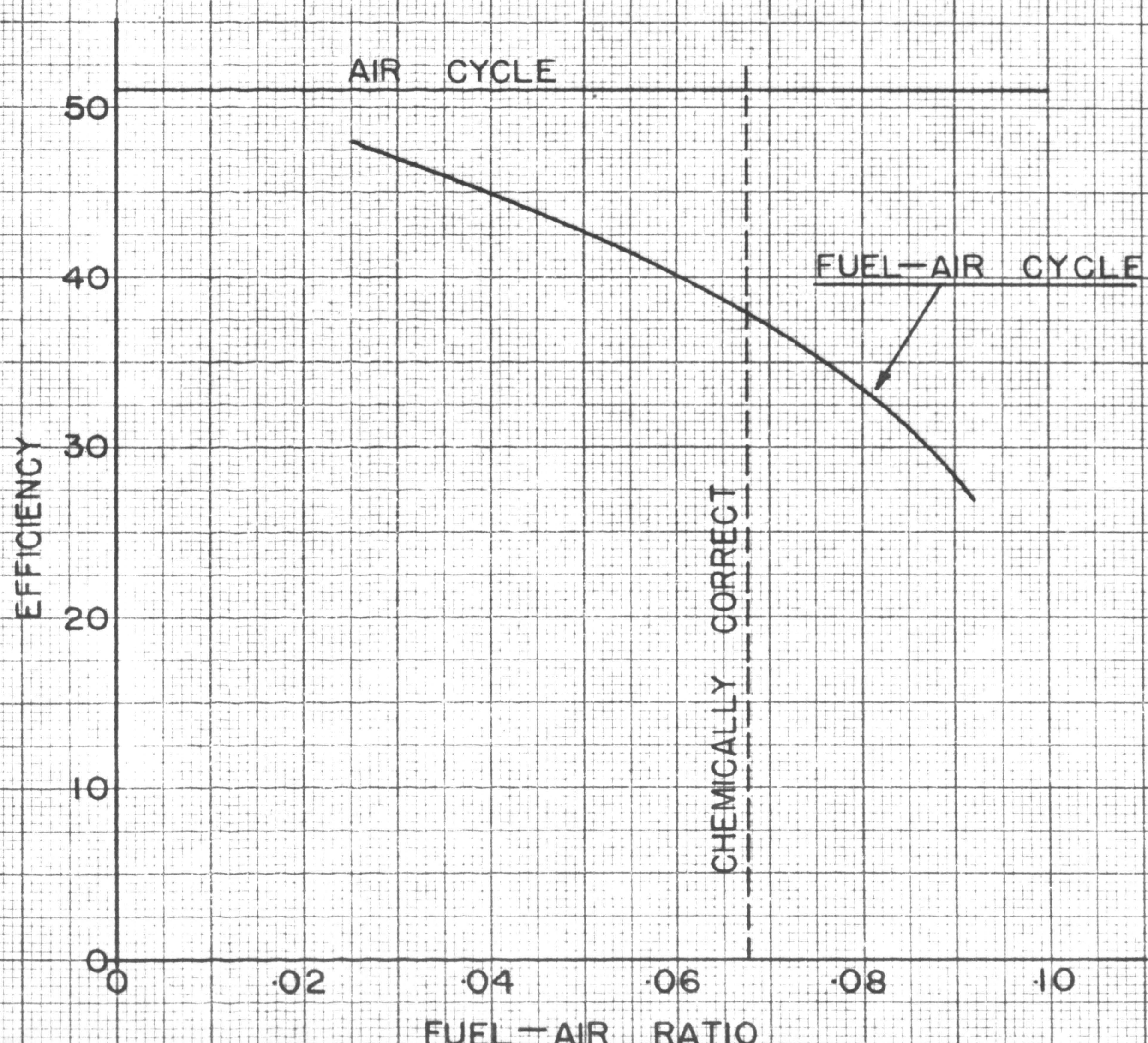
From this calculation it can be seen that all fuel in excess of .0665 pounds per pound of air will not find oxygen to burn and some of what does burn will burn to CO instead of CO_2 and, therefore, release less energy than would have been released if the carbon had burned to CO_2.

There are several reasons why a mixture leaner than chemically correct will burn more efficiently in an engine. With plenty of oxygen available, the possibility that some portion of the fuel may not be able to find sufficient oxygen to burn completely is reduced even though there is sufficient oxygen available within the cylinder. This would be the case when there is poor mixing of fuel and air. Another factor tending to increase efficiency as the air cycle is approached is that when the mixture consists predominantly of air it tends to act more nearly as a perfect gas due to the smaller temperature range resulting from less fuel being burned. The air cycle efficiency thus constitutes a limit which the efficiency of the fuel air cycle approaches as the fuel air ratio decreases. As seen in Figure 1, the air cycle is considerably more efficient over the normal operating range than the fuel air cycle. It is therefore apparent that any method that will enable an engine to operate at a leaner mixture will increase the fuel economy.

A spark ignition engine may be leaned only to the fuel air ratio which permits combustion of the fuel, due to the limited range of air fuel ratios which constitute a combustionable mixture. Mixtures outside this range will not ignite in the cylinder. The combustionable range is generally considered to be between the air fuel ratios of 8 and 20 to 1. In a single cylinder engine there is no problem in obtaining the leanest mixture that will burn. However, in a multi-cylinder engine, in order to take full advantage of the economy of a lean mixture, all cylinders must use the leanest possible mixture. Most multi-cylinder spark ignition engines use a single carburetor with manifolds from the carburetor to each of the cylinders. If the fuel air mixture is a homogeneous gas, each cylinder will receive a volume of this mixture containing part air and part fuel, and equal proportion of fuel and air going to each cylinder. In most cases, the mixture in the manifold is not homogeneous but consists not only of air and vaporized fuel but also atomized fuel suspended in the air along with a slight film of liquid which travels to the

FIG. I

EFFECT OF FUEL—AIR RATIO ON THE EFFICIENCY OF THE AIR CYCLE AND FUEL—AIR CYCLE AT A COMPRESSION RATIO OF 6



cylinder by clinging to the inside walls of the manifold. Under these conditions it is difficult to obtain an even distribution of fuel and air to all cylinders. Each cylinder may therefore receive a different fuel air ratio and while one may be lean the rest may be using an excessively rich mixture and thereby waste fuel.

If the air is heated more of the fuel will vaporize and produce a more homogeneous mixture. If the carburetor is compensated for the warmer air it is possible that the fuel consumption may be reduced by improving the distribution.

Heating the air to vaporize the fuel is not done in most engines because the greater specific volume of the warm air and because the added volume of the vaporized fuel reduces the breathing capacity and therefore the maximum output of the engine. For this reason a compromise,[2] credited to G. G. Brown, and generally accepted is that 60% of

(2) Taylor and Taylor, The International Combustion Engine, International Textbook Co., p. 155

the fuel be vaporized and 40% remain a liquid until vaporized on the compression stroke.

One of the most logical applications of heated air is in the unsupercharged aircraft engine. The airplane requires maximum power only during takeoff at which time cold air could be used. The airplane is one of the few uses of the spark ignition engine where the throttle and mixture can be left in the same position for a length of time. Most other uses, particularly the automobile, require that the throttle be continually changed. While it would not be impossible to use heated air in the automobile engine it would involve additional automatic equipment if maximum power was desired. Most standard airplane engines are equipped with alternate air intakes for use when carburetor icing is possible. This same device which takes warm air from behind the air cooled cylinders also might be used to advantage for economy.

According to Jennings and Obert[3] the weight of fuel used by a

(3) Jennings and Obert, Internal Combustion Engines, International Textbook Co., pp. 307 to 309

simple carburetor may be found by the following equation:

$$w_f = .79\ c_f d_f^{\,2}\ \sqrt{(sp\ gr)\ \Delta H} \qquad (2)$$

where

w_f = flow of fuel in lb. per sec;

c_f = coefficient of discharge;

d_f = diameter of orfice, in inches;

sp gr = specific gravity of fuel referred to water;

ΔH = manometer reading at the fuel nozzel tip, measured in inches of water.

The weight of air may be found by the following equation:

$$w_a = \frac{1.615\ c_a P_1 d^2}{\sqrt{T_1}} \sqrt{\frac{P_2}{P_1}^{1.43} - \frac{P_2}{P_1}^{1.71}} \qquad (3)$$

where

w_a = flow of air in lb. per sec;

P_1 = inlet pressure in psi

P_2 = venturi pressure in psi

d = venturi diameter, in inches;

T_1 = temperature of supply air at inlet, in deg R;

c_a = coefficient of discharge.

F/A = Fuel air ratio = $\frac{\text{weight of fuel}}{\text{weight of air}}$ =

$$\frac{.79 c_f\ \sqrt{(sp\ gr)\ \Delta H}\ d_f^{\,2}\ \sqrt{T_1}}{1.615\ c_a P_1 d^2 \sqrt{\frac{P_2}{P_1}^{1.43} - \frac{P_2}{P_1}^{1.71}}} = F/A \qquad (4)$$

With everything remaining constant except temperature it can be seen that as the temperature increases the mixture will be enriched. Therefore, if the air was heated before entering the carburetor the mixture would be enriched and the fuel consumption increased. In order to maintain a constant fuel air ratio with an increasing temperature it is necessary to reduce the orfice diameter d_f to compensate for the change in temperature.

$$F/A = K\, d_f \sqrt{T_1}$$

$$d_f = \frac{F/A}{K \sqrt{T_1}} = \frac{K}{\sqrt{T_1}} \qquad (5)$$

It can be seen from formula (5) that d_f varies inversly as the square root of inlet air temperature with everything else remaining constant.

d_f = diameter of orfice, in inches;
F/A = fuel air ratio;
K = constant;
T_1 = temperature of supply air at inlet, in deg R.

In addition to the reduced orfice size to compensate for the increased temperature, a further reduction is necessary to take advantage of any improvement in distribution due to the more homogeneous mixture.

The increase in inlet air temperature also enhances the possibilities of detonation as can be seen in Figure 2.[4] As the only time heating of the inlet air is advisable due to the high specific volume of heated air is at part throttle, hence, the possibility of detonation is slight at normal spark advance.

(4) Taylor and Taylor, The Internal Combustion Engine, International Textbook Co., p. 96.

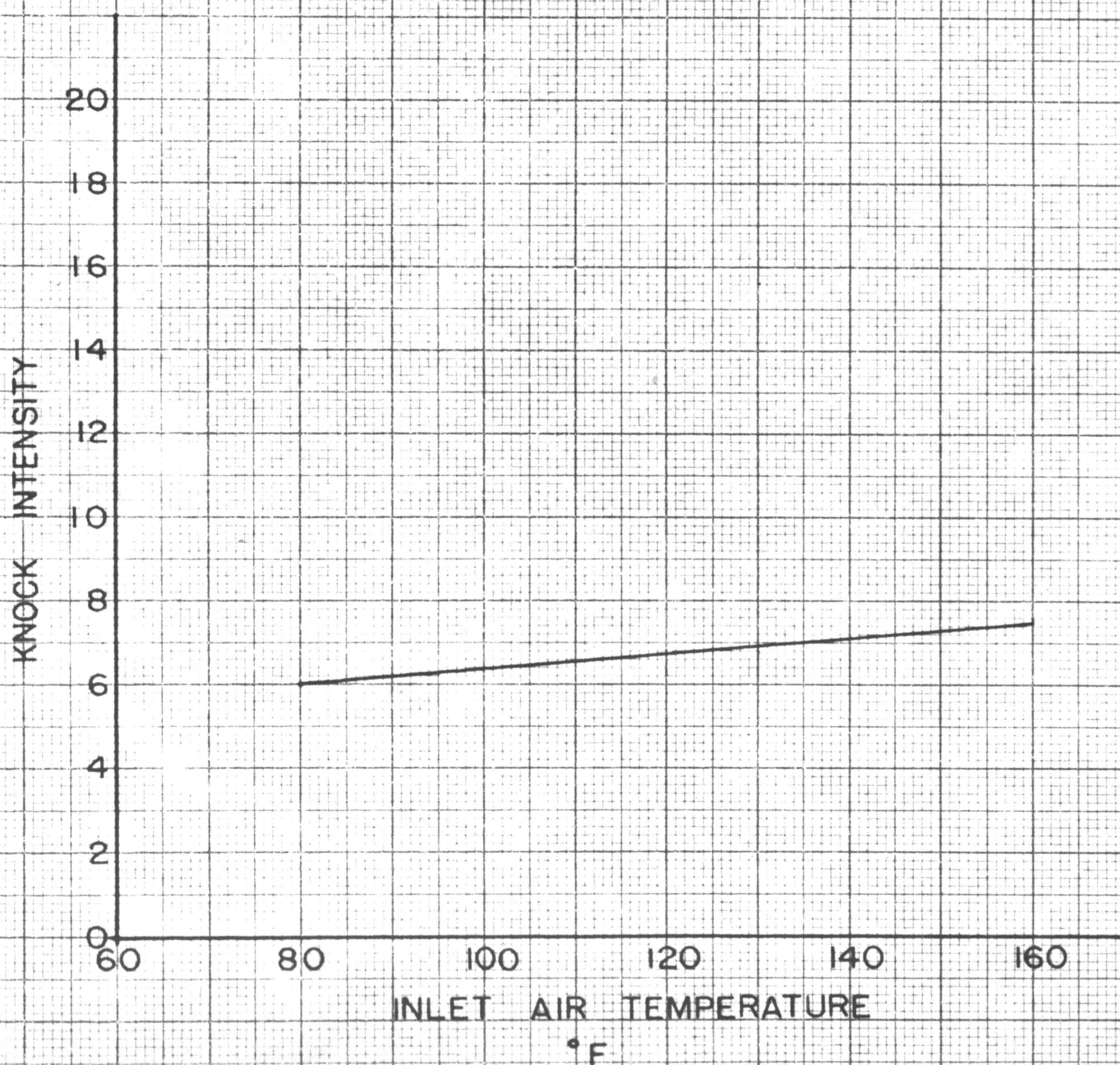
To partly balance the undesirable effect of warm air on detonation is the action of the heat as a catalyst[5] in the reaction of the tetra-ethyl lead in its action as a deterrent to detonation. This advantage, of course, is only realized when fuels containing tetraethyl lead are used, which is true of most fuels today.

(5) Ricardo and Glyde, The High Speed Internal-Combustion Engine, Blackie and Son Limited, p. 63.

One of the disadvantages of heating the inlet air is that the speed of the flame front is reduced as can be seen in Figure 3.[6]

(6) Taylor and Taylor, The Internal Combustion Engine, International Textbook Co., p. 71.

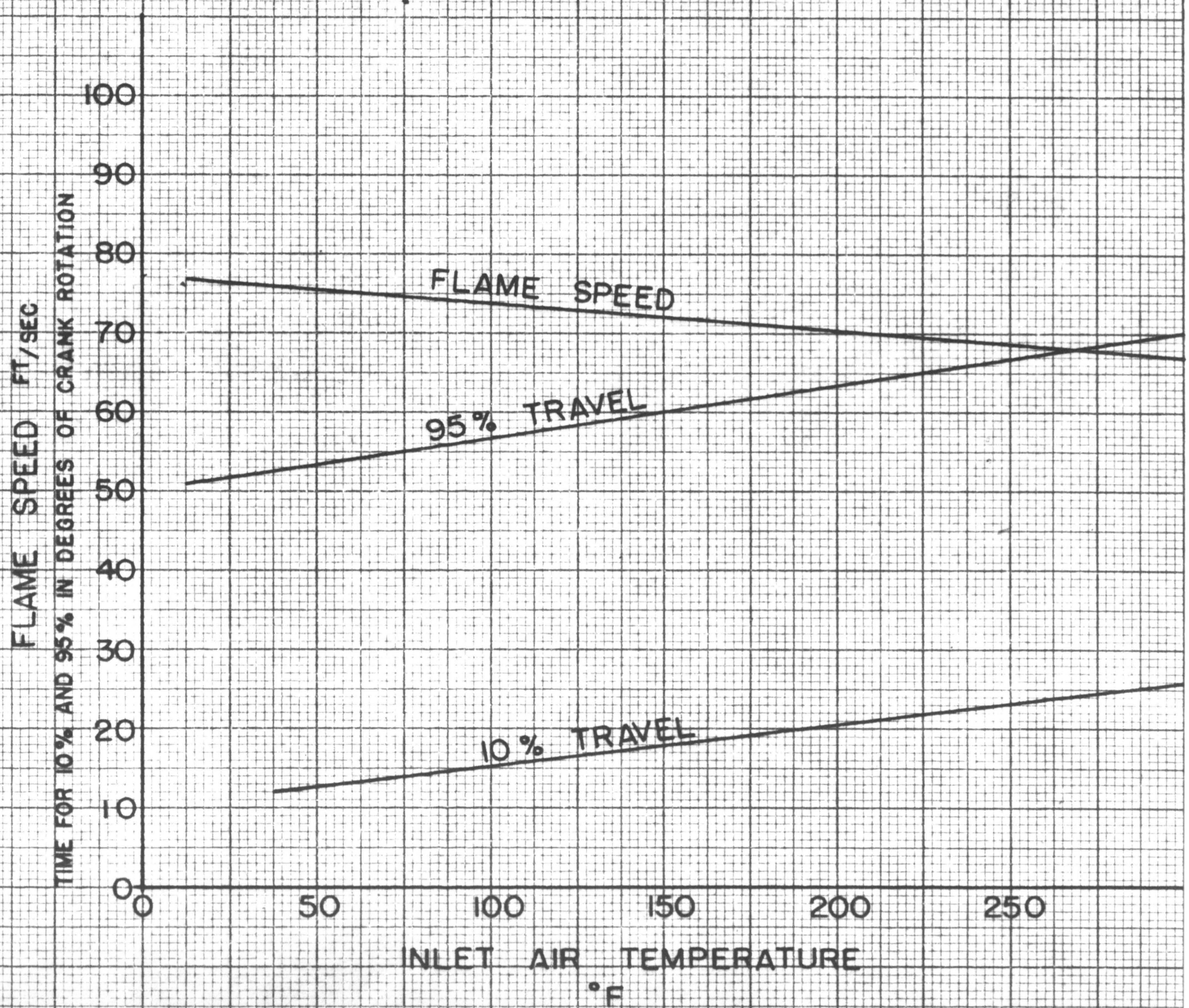
FIG. 2
EFFECT OF INLET AIR TEMPERATURE ON DETONATION



Some effect of this reduced speed can be compensated for by advancing the spark, but the loss due to the burning being spread over a greater number of degrees of crank angle can not be recovered.

With some factors indicating a reduction of the brake specific fuel consumption as the result of increasing the inlet air temperature it remains to be determined on the individual engine by actual test whether the advantages of heated air will outweigh the disadvantages.

FIG. 3
EFFECT OF INLET AIR TEMPERATURE ON FLAME SPEED



PART II

TEST EQUIPMENT

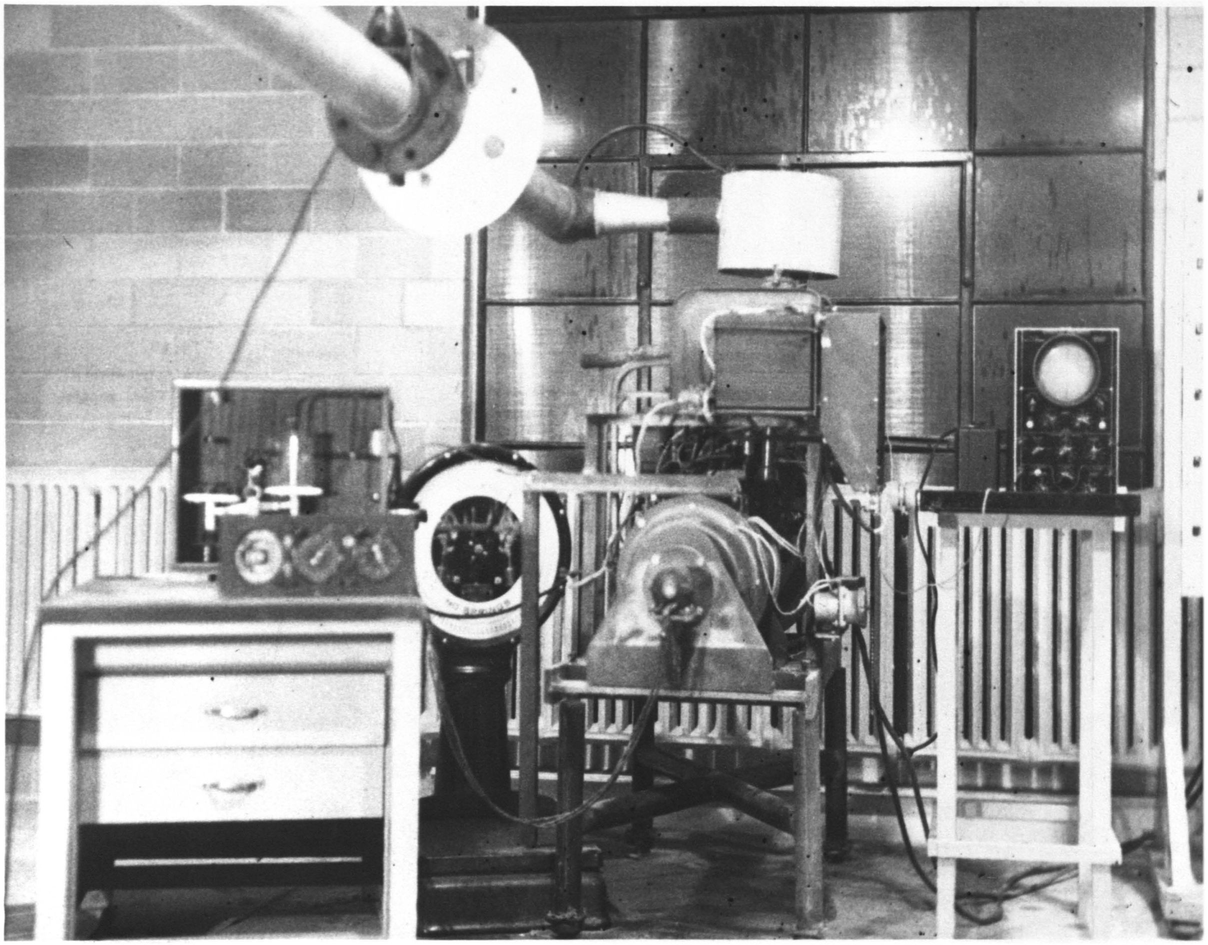


Illustration No. 1 Photograph of Test Equipment

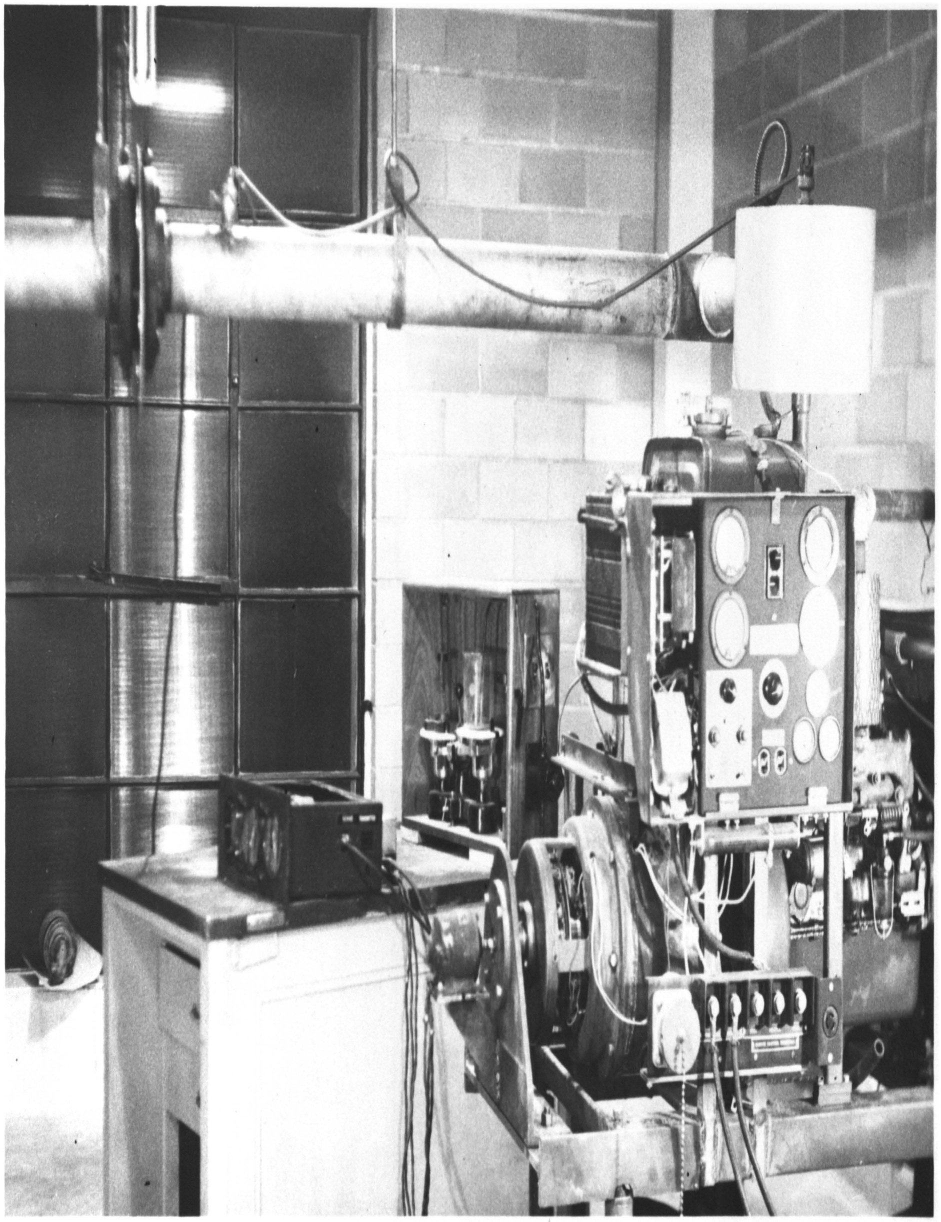


Illustration No. 2 Photograph of Test Equipment

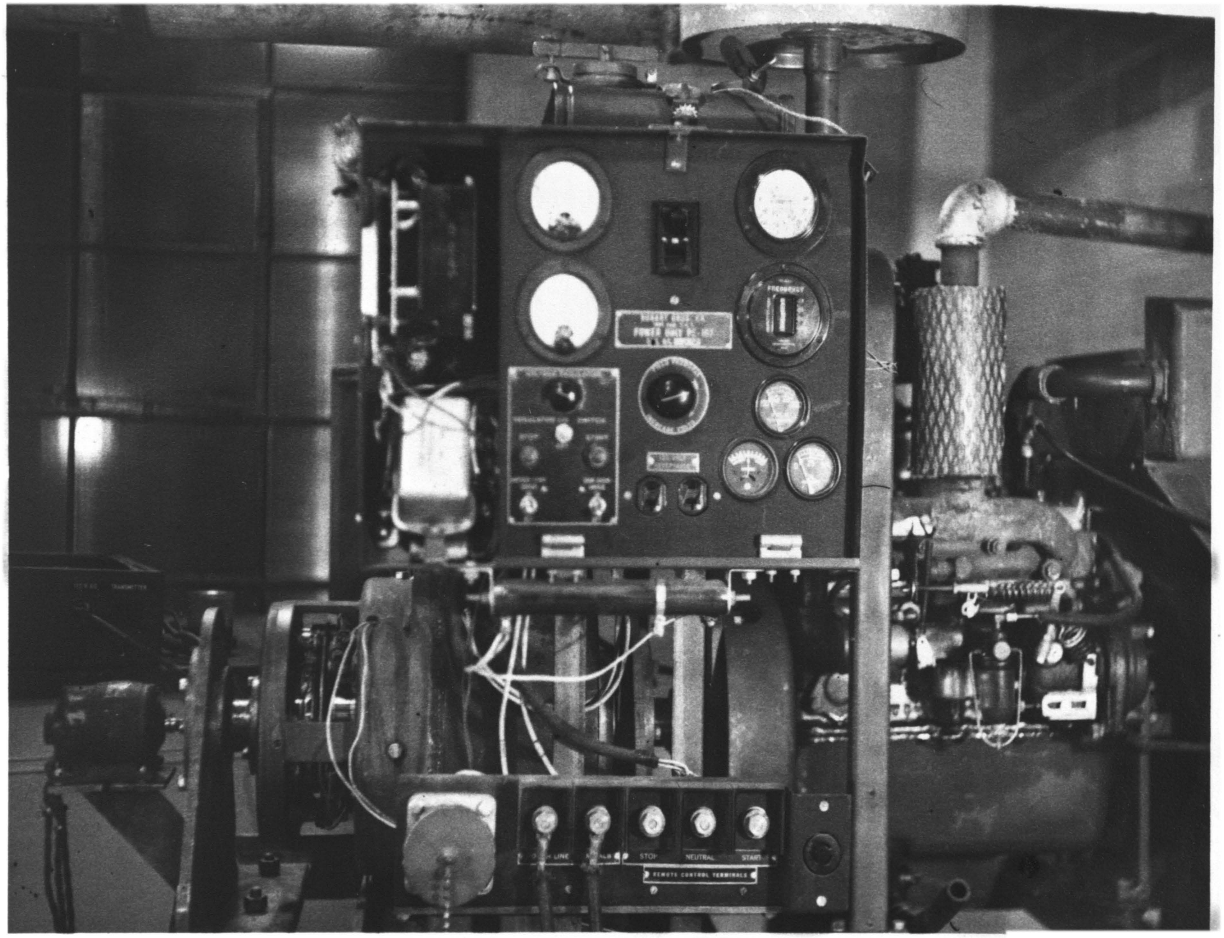


Illustration No. 3 Photograph of Test Equipment

The engine used for these tests was a four cylinder fourteen horsepower Hercules gasoline engine from a model PE-197 portable power unit.

Bore . . . . 2 5/8"

Stroke . . . . 3"

Rated Power . . . . 14 HP at 1800 rpm

The engine was originally designed to drive a 6.3 KVA alternator at 1800 rpm. The generator was converted to a dynamometer by mounting the frame on ball bearings at both ends so that the stator was free to turn with the armature which was connected directly to the engine. To determine the torque an arm was fixed to the frame of the generator with the free end of the arm resting on a platform scale. The brake horsepower could then be determined by the following equation:

$$BHP = \frac{2\pi LFN}{33000t} \tag{6}$$

where

L = length of brake arm in feet;

F = force on platform scale-tare weight, in pounds;

N = revolutions

t = time in minutes.

With constants grouped together;

$$BHP = \frac{CFN}{t} \tag{7}$$

where

C = brake constant = $\frac{2\pi L}{33000}$

Since the engine was designed to run at a constant rpm it was equipped with a fixed spark advance which could be changed by loosening the coupling at the distributor and tightening it in the desired position.

A small neon light capable of being lighted by leakage flux from a spark plug lead was attached to the engine shaft. A protractor scale was fixed along the path of the neon light such that the position of firing relative to top dead center could be determined by the position of the neon light when it lighted opposite the scale of the protractor as the spark plug received current.

The carburetor was originally equipped with a fixed jet which had to be enlarged and a needle valve placed in a position such that the size of the orfice could be varied to change the fuel air ratio.

A spring was attached to the choke so that it would remain open except when needed for starting. The throttle was also held in position by the tension of a spring and adjusted by turning a wing nut to open the throttle against the tension of the spring.

An automatic revolution counter and tachometer were driven by a selsyn motor which received current from a generator connected directly to the shaft of the dynamometer.

An electric clock and the revolution counter were set in operation by a mercury switch fixed to a balance scale. When one-quarter pound of fuel was used by the engine another mercury switch would stop the clock and counter by releasing an electric clutch.

The fuel consumption can be calculated by the following equation:

$$w_f = \frac{W \times 60}{t} = \frac{1/4 \times 60}{t} = \frac{15}{t} \qquad (8)$$

where

w_f = fuel consumption, in pounds per hour;

W = weight of fuel in pounds;

t = time in minutes.

The brake specific fuel consumption can then be calculated from the following equation:

$$BSFC = \frac{w_f}{BHP} = \frac{\frac{15}{t}}{\frac{(CFN)}{t}} = \frac{15}{CFN} \qquad (9)$$

where

$BSFC$ = brake specific fuel consumption in pounds/BHP per hour;

C = constant;

BHP = brake horsepower;

N = revolutions;

F = force on scale in pounds;

t = time in minutes;

As the counter was designed to be used on an auxiliary shaft of an aircraft engine the revolutions indicated by the counter had to be corrected by dividing by the factor 1.73:

$$BSFC = \frac{15 \times 1.73}{CNF} = \frac{C'}{NF}$$

where

C' = new constant.

To determine border line knock a vibration pickup was mounted on a stud bolt near number four cylinder. A lead from the pickup was connected to Du Mont 208 oscilloscope which was used in detecting the change of voltage from the vibrating pickup. Detonation being a severe vibration would show up as a large pip on the screen of the oscilloscope compared to the smaller pips of the ordinary operating vibrations.

To measure the weight of air used by the engine per unit time a calibrated orfice was placed in a 4 inch pipe connected to the intake manifold. A manometer was connected across the orfice so that the pressure drop could be observed, and the volume calculated using the following formula.(7)

(7) Meriam Catalog, No. 101, 1950

$$Q = C\sqrt{hp} \qquad (11)$$

where

Q = volume of air per unit time;

C = orfice constant = 44 for cubic ft. of air per hour;

h = drop in pressure across orfice;

p = absolute pressure

In measuring the flow of air it is desirable to have as small a pressure differential as can be accurately determined. For this reason manometer fluid was used instead of mercury in the manometer. As the manometer scale was calibrated in inches of water using mercury in the manometer it was necessary to correct the constant to take this into consideration.

The temperature of the inlet air was determined by a thermometer inserted in the inlet manifold just before the carburetor.

With the volume of air per unit time known the weight of air per unit time may be found by multiplying by the density.

The fuel air ratio may now be calculated using the definition:

$$\text{fuel air ratio} = \frac{\text{weight of fuel}}{\text{weight of air}}$$

The inlet air was heated by four different heating elements using 110 volt AC current. As these units were of several different sizes many inlet air temperatures could be obtained using combinations of the four elements.

PART III

TEST PROCEDURE

In general the procedure of this experiment was to run the engine at a constant load and speed at several different inlet air temperatures with the leanest possible fuel air ratio at all times, with the object of determining the most economical air temperature.

Since heating the inlet air changes the speed of the flame front as well as the fuel air ratio it is necessary that tests for BSFC be run at several different spark settings as well as various inlet air temperatures. While the method of determining the spark advance was very accurate the method of changing the spark timing was crude and therefore, it was difficult to obtain constant increments of spark advance. Several runs were made at each inlet air temperature and spark advance until enough values of BSFC were determined to fix an average for each temperature. After runs at various inlet air temperatures the spark advance was changed and the same procedure repeated.

Precautions were taken to be sure that the engine and air temperatures had reached equilibrium before any runs were made.

The temperature of the cooling water could not be held constant but varied from 150°F at low inlet air temperatures to 175°F at the higher temperatures of inlet air. This condition was not considered detremental to the accuracy of these tests as it is a natural condition of higher inlet temperatures and must therefore be considered when the effect of inlet air temperatures is to be determined.

After a run had been made at several different temperatures the engine was stopped and the spark advance changed. It was necessary that the engine be allowed to cool for several hours after each run before tests were begun at a different spark advance and lower temperature.

Each test was conducted in the following manner:

After the engine had reached equilibrium the mixture was leaned until the engine began to misfire. The mixture control was then turned back just enough to stop the misfiring. While the engine was being leaned the throttle was constantly being adjusted to maintain a force on the platform scale of 28 pounds and the rpm constant at 1800. With the torque and rpm constant the BHP was also constant.

It was decided to run all tests at 1800 rpm since that was the rated speed of the power unit. As the selsyn tachometer was inoperative, the rpm was observed during the tests by noting the frequency of the dynamometer output indicated on the control panel of the original power unit.

Observations of the following values were recorded for each run.

Time to burn one-fourth pound of fuel in minutes;

Pressure drop of the inlet air across the orfice;

Temperature of inlet air in degrees F;

Force on scale in pounds;

Revolutions during the burning of one-fourth pound of fuel;

Temperature of cooling water in degrees F.

The following information was recorded for each spark setting:

Spark advance in degrees before top dead center;

Room temperature in degrees F;

Wet bulb temperature in degrees F;

Atmospheric pressure in inches of mercury.

All runs at each spark setting were completed without stopping the engine and within two hours.

PART IV

INTERPRETATION OF TEST RESULTS

As shown in Figure 3, an increase in the inlet air temperature also increases the number of degrees necessary to complete the combustion process. The increase of inlet air temperature will place the peak of the cylinder pressure further past top dead center and thereby reduce the efficiency of the cycle. The peak can be brought back to the desired position of 12° after top dead center[8] by

(8) Pye, The Internal Combustion Engine, Vol. 1, p. 124, Oxford Press

advancing the spark which also tends to slow the speed of the flame front.[9] However, if the spark is advanced far enough the peak of

(9) Taylor and Taylor, The Internal Combustion Engine, International Textbook Co., p. 76.

cylinder pressure can of course be adjusted to its proper position. The position of peak cylinder pressure is not the only detremental effect of the longer burning process. Because combustion must be started sooner, the piston must work against a back pressure for a longer time during the compression stroke and because the combustion continues longer after the peak pressure, all the energy of combustion is not most efficiently utilized.

The slower burning speed of the flame front accompanied by the fact that the engine could not be run at a leaner fuel air ratio using warm air account for the fact that the brake specific fuel consumption increased with the increase of inlet air temperature.

The intake manifold and carburetor were very close to the exhaust manifold and therefore were warm enough to vaporize most of the fuel even with cold inlet air temperatures. Distribution therefore was good and no appreciable improvement could be made on this engine by heating the inlet air.

A more complete an analysis could be made if a method of determining the motoring friction at various jacket water temperature had been available.

TABLE I

SPARK 11° BTDC

Wet bulb temperature 80°F — Barometer 29.22" Hg

Dry bulb temperature 70°F — December 19, 1950

Air °F	95	140	160	190	225
Orfice differential	63	67	67	66.5	58
Torque, lbs.	18.3	18.9	19.0	19.0	19.0
Fuel, lbs.	1/4	1/4	1/4	1/4	1/4
Time. min.	2.400	2.401	2.400	2.385	2.356
Revolutions	7960	8058	8065	8035	7701
Water °F	160	162	163	165	167
RPM	1917	1940	1942	1945	1889
Air, lbs/hr.	95.0	97.6	97.6	97.2	91.0
Fuel, lbs/hr.	6.25	6.25	6.25	6.29	6.39
Fuel air ratio	.0658	.0581	.0581	.0588	.0638
BHP	8.37	8.70	8.85	8.88	8.65
BSFC, lbs/BHP-hr.	.741	.708	.705	.707	.738

FIG. 4
BSFC AND MINIMUM FUEL—AIR RATIOS AT VARIOUS INLET AIR TEMPERATURES WITH SPARK AT 11° B.T.D.C.

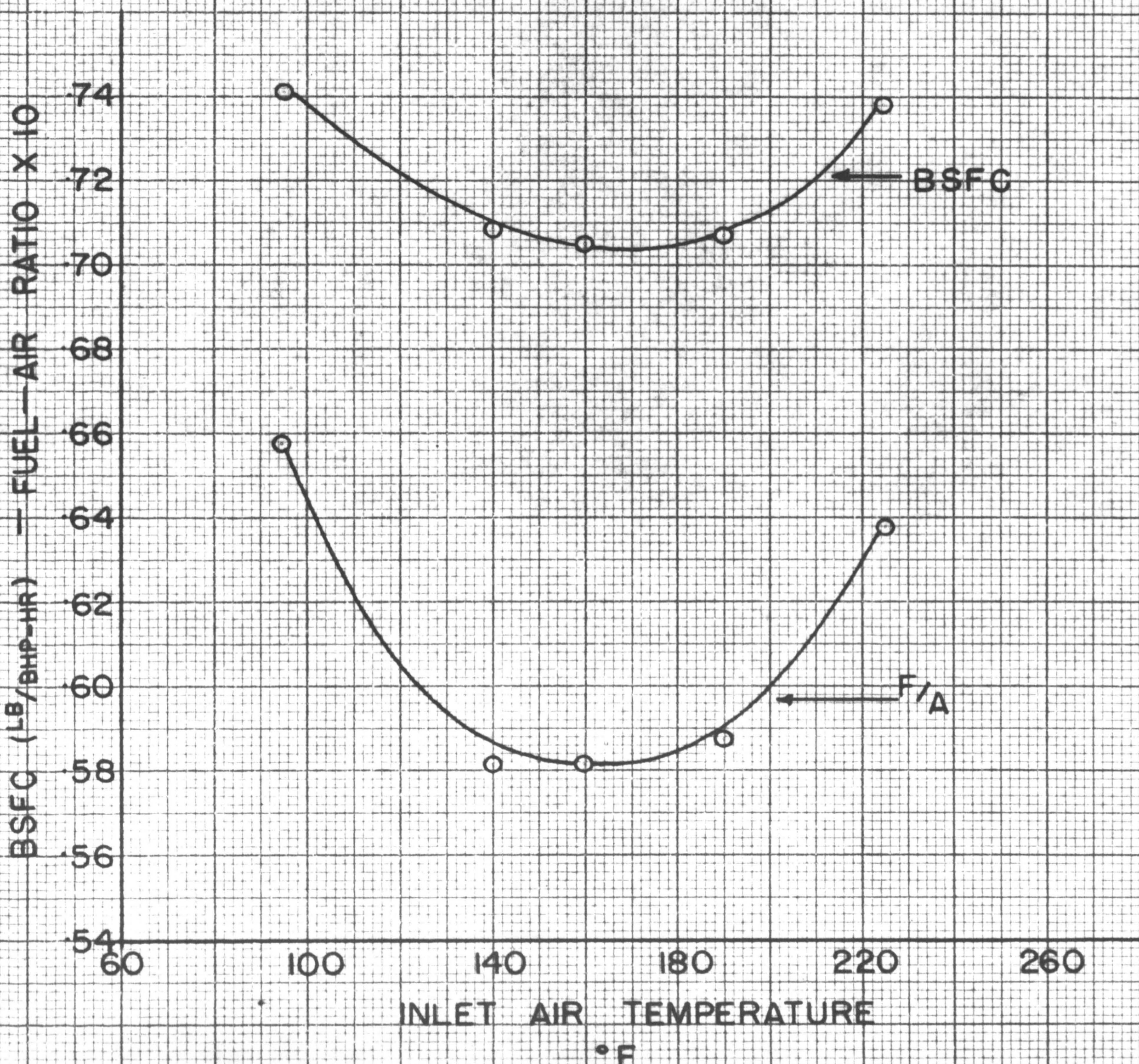


TABLE II

SPARK 17° BTDC

Wet bulb temperature 70°F Barometer 29.30" Hg

Dry bulb temperature 82°F December 21, 1950

Air, °F	100	150	170	190	235
Orfice differential	48	50	51	51	52
Torque, lbs.	18.9	19.0	19.0	19.2	19.2
Fuel, lbs.	1/4	1/4	1/4	1/4	1/4
Time, min.	2.640	2.601	2.591	2.581	2.567
Revolutions	8487	8421	8402	8378	8341
Water, °F	168	165	165	165	166
RPM	1856	1871	1950	1950	1890
Air, lbs/hr.	82.7	84.1	85.5	85.5	86.5
Fuel, lbs/hr.	5.68	5.77	5.80	5.82	5.85
Fuel air ratio	.0623	.0622	.0617	.0618	.0615
BHP	8.41	8.55	8.88	8.98	8.71
BSFC, lbs/BHP-hr.	.673	.675	.675	.672	.675

FIG. 5
BSFC AND MINIMUM FUEL—AIR RATIOS AT VARIOUS INLET AIR TEMPERATURES WITH SPARK AT 17° B.T.D.C.

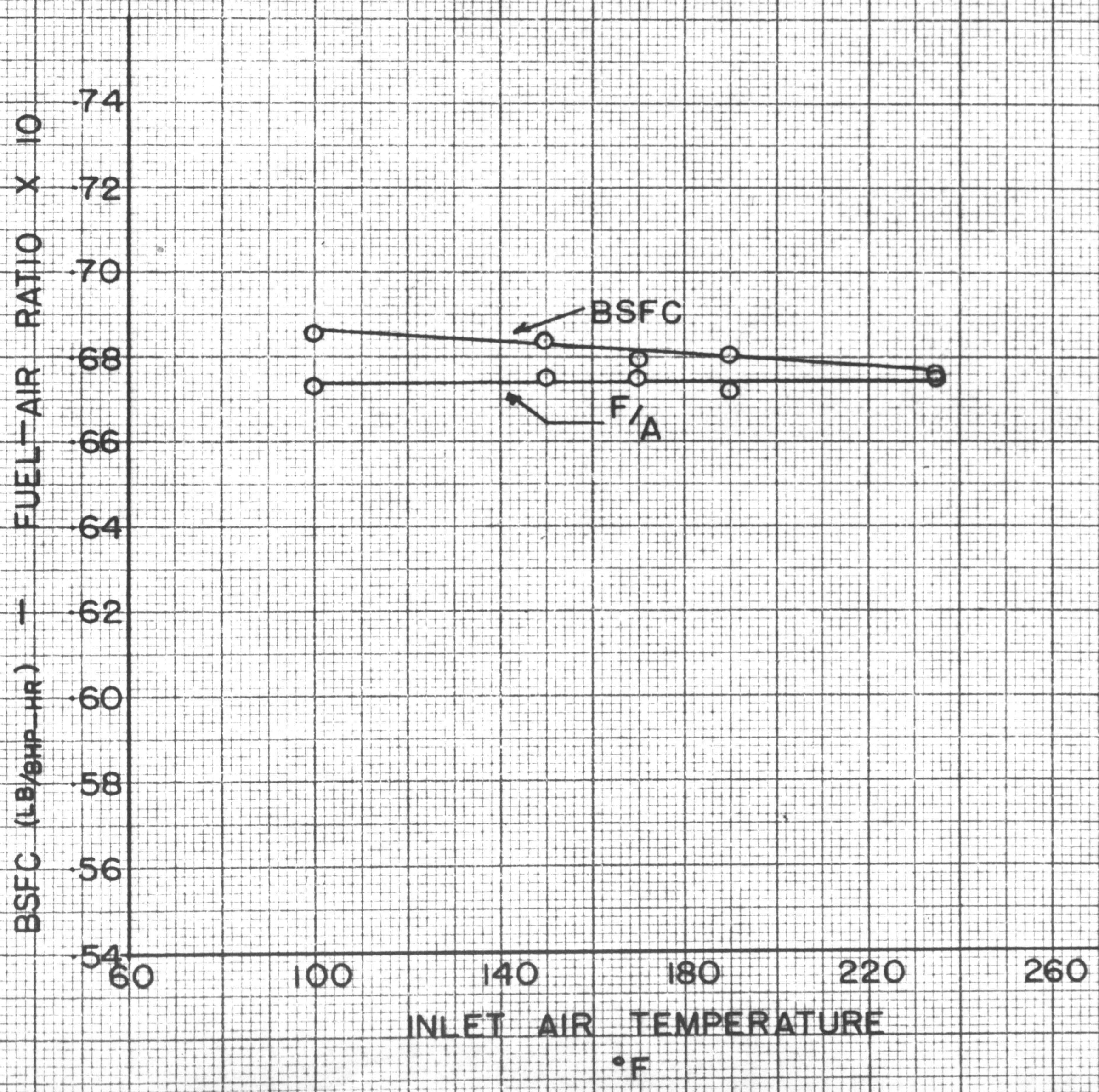


TABLE III

SPARK 22° BTDC

Wet bulb temperature 68°F | Barometer 29.22" Hg

Dry bulb temperature 82°F | December 19, 1950

Air, °F	100	158	180	200	240
Orfice differential	52.0	56.0	55.0	54.5	51.0
Torque, lbs.	19	19.6	19.3	19.1	19.1
Fuel, lbs.	1/4	1/4	1/4	1/4	1/4
Time, min.	2.699	2.581	2.550	2.534	2.599
Revolutions	8478	8104	8180	8230	8073
Water, °F	170	170	170	170	170
RPM	1814	1813	1850	1880	1800
Air, lbs/hr.	86.1	89.5	88.5	88.0	85.5
Fuel, lbs/hr.	5.56	5.81	5.88	5.92	5.77
Fuel air ratio	.0645	.0649	.0655	.0674	.0675
BHP	8.28	8.54	8.30	8.62	8.25
BSFC, lbs/BHP-hr.	.670	.678	.684	.692	.700

FIG. 6

BSFC AND MINIMUM FUEL-AIR RATIOS AT VARIOUS INLET AIR TEMPERATURES WITH SPARK AT 22° B.T.D.C.

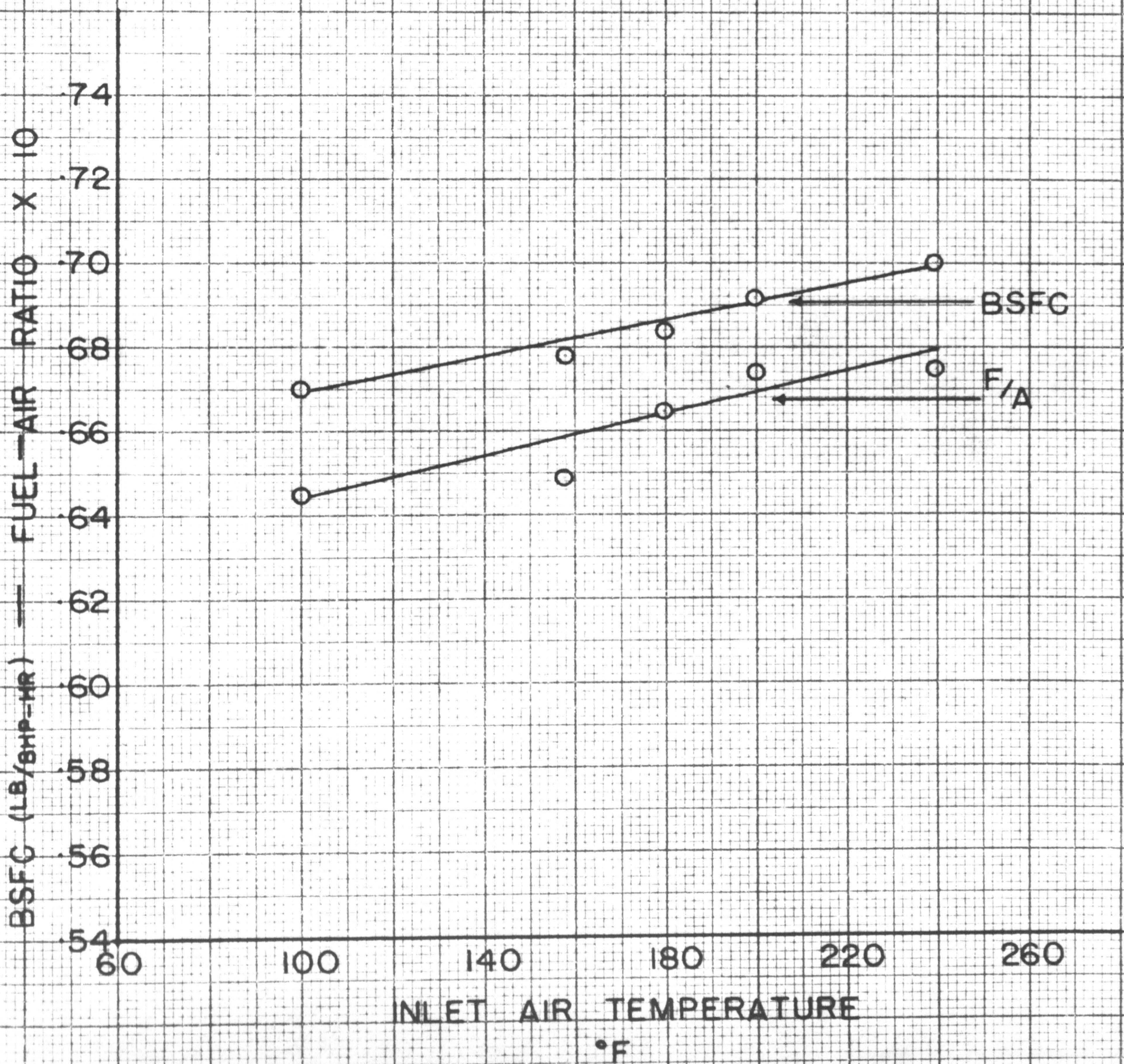


TABLE IV

SPARK 24° BTDC

Wet bulb temperature 65°F — Barometer 29.32" Hg

Dry bulb temperature 55°F — December 17, 1950

Air, °F	92	145	165	195	230
Orfice, differential	72	52	53	57	56.5
Torque, lbs.	19.3	18.6	18.6	19.3	19.0
Fuel, lbs.	1/4	1/4	1/4	1/4	1/4
Time, Min.	2.603	2.778	2.720	2.592	2.594
Revolutions	8575	8802	8690	8297	8272
Water, °F	150	155	157	160	160
RPM	1900	1831	1847	1850	1844
Air, lbs/hr.	99.2	84.7	85.5	88.5	88.1
Fuel, lbs/hr.	5.76	5.40	5.51	5.79	5.79
Fuel air ratio	.0570	.0594	.0606	.0636	.0636
BHP	8.80	8.18	8.25	8.57	8.41
BSFC, lbs/BHP-hr.	.653	.660	.668	.674	.687

FIG. 7

BSFC AND MINIMUM FUEL—AIR RATIOS AT VARIOUS INLET AIR TEMPERATURES WITH SPARK AT 24° B.T.D.C.

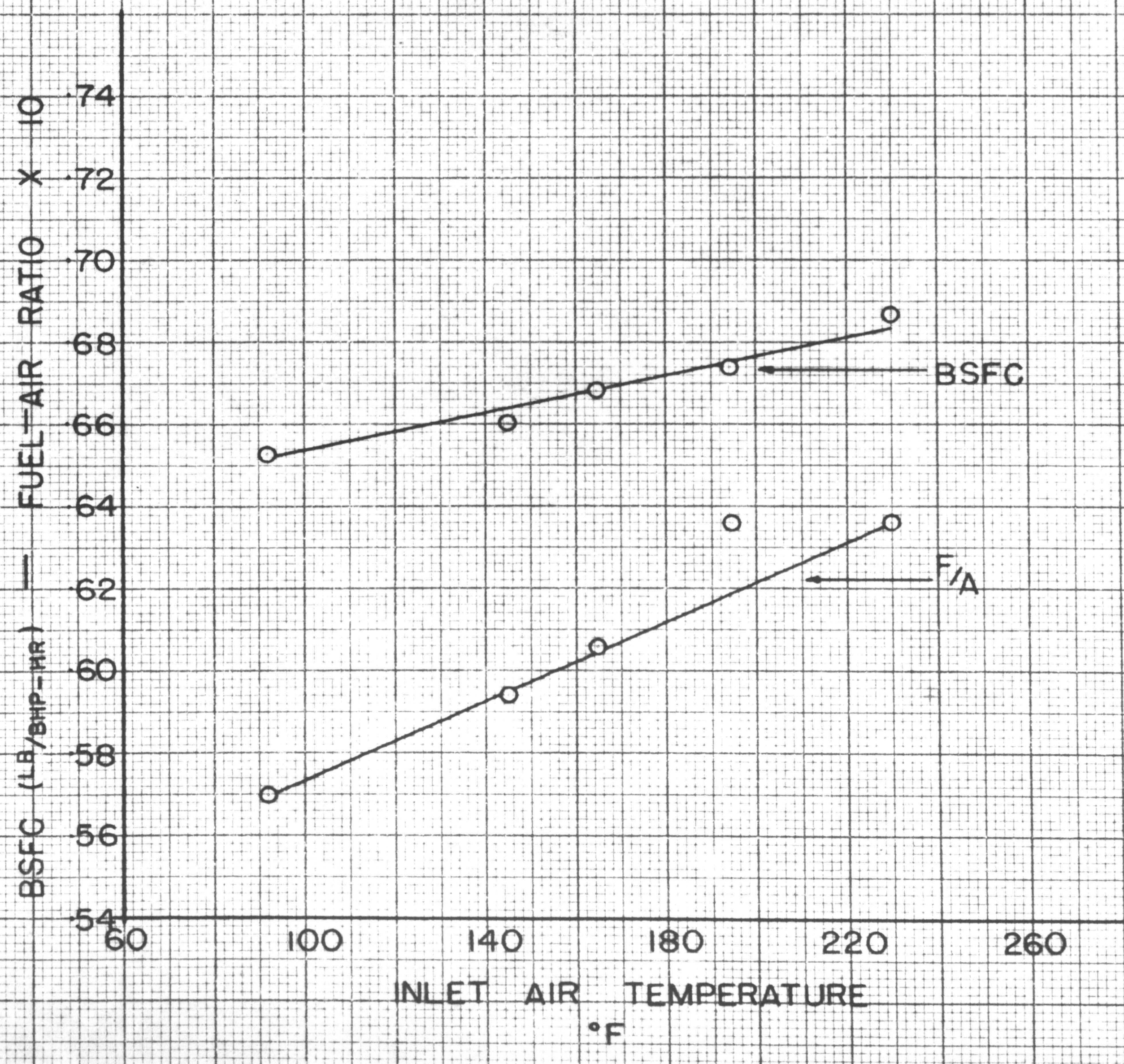


TABLE V

SPARK 28° BTDC

Wet bulb temperature 80°F Barometer 29.22" Hg

Dry bulb temperature 70°F December 19, 1950

Air °F	97	145	170	190
Orfice differential	63	58	60	58
Torque, lbs.	18.7	18.3	18.5	18.5
Fuel, lbs.	1/4	1/4	1/4	1/4
Time, min.	2.747	2.763	2.582	2.570
Revolutions	8984	8938	9632	8503
Water °F	159	161	170	170
RPM	1888	1865	1930	1911
Air, lbs/hr.	94.5	97.8	92.6	91.0
Fuel, lbs/hr.	5.45	5.43	5.82	5.84
Fuel air ratio	.0576	.0619	.0629	.0640
BHP	8.46	8.50	8.58	8.50
BSFC, lbs/BHP-hr.	.643	.660	.675	.686

FIG. 8
BSFC AND MINIMUM FUEL—AIR RATIOS AT VARIOUS INLET AIR TEMPERATURES WITH SPARK AT 28° B.T.D.C.

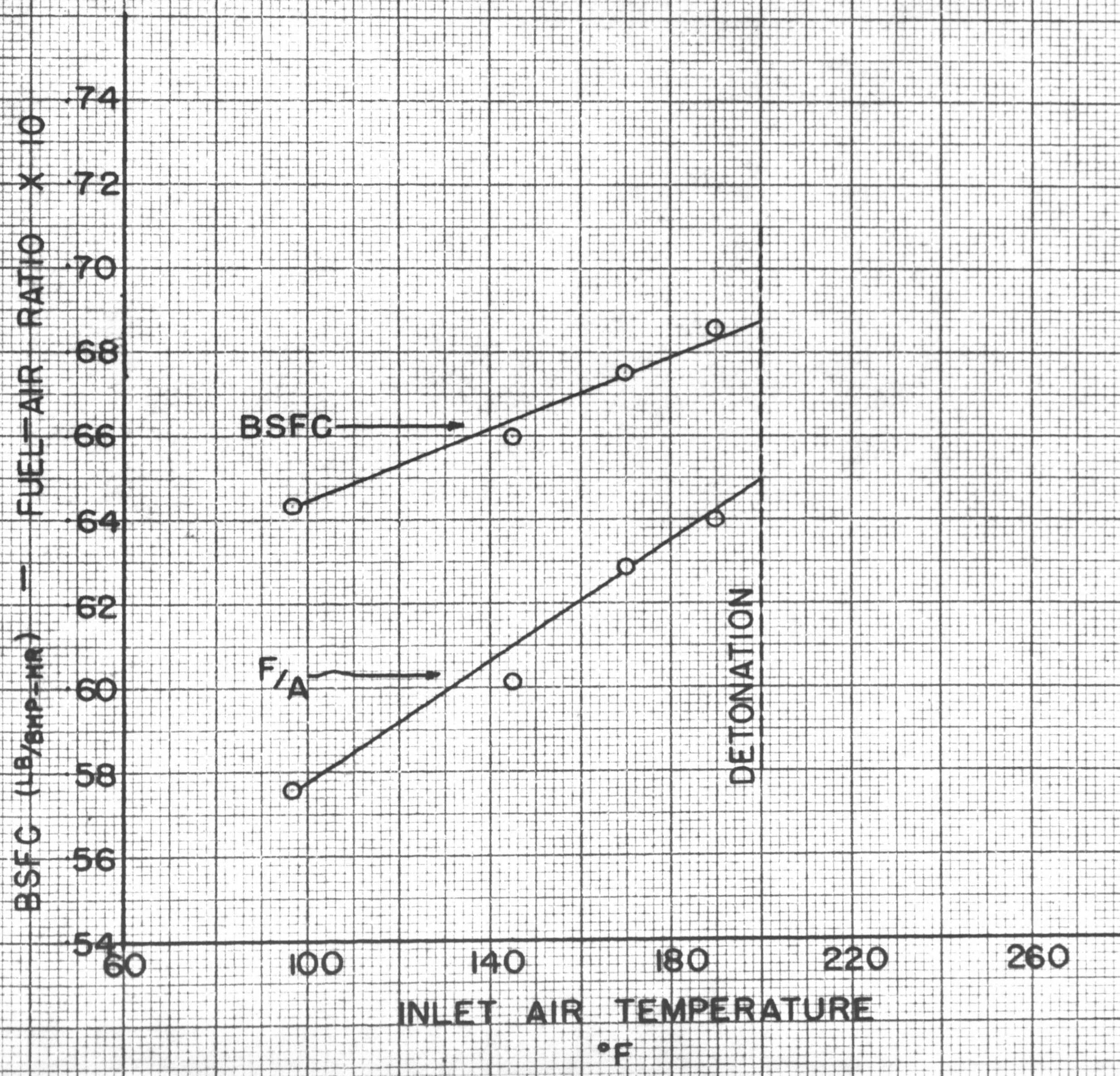


TABLE VI

SPARK 34° BTDC

Wet bulb temperature 45°F | Barometer 29.00" Hg

Dry bulb temperature 86°F | December 14, 1950

Air °F	98	152	175	200
Orfice differential	66.5	67.0	70	73
Torque, lbs.	18.1	17.4	17.8	18.1
Fuel, lbs.	1/4	1/4	1/4	1/4
Time, min.	2.863	2.820	2.710	2.608
Revolutions	9276	9334	8980	8667
Water, °F	160	170	175	180
RPM	1871	1909	1911	1918
Air, lbs/hr.	95.6	95.9	98.3	100.2
Fuel, lbs/hr.	5.23	5.31	5.54	5.75
Fuel air ratio	.0547	.0554	.0564	.0573
BHP	8.15	7.79	8.16	8.32
BSFC, lbs/BHP-hr.	.642	.665	.675	.688

FIG. 9

BSFC AND MINIMUM FUEL–AIR RATIOS AT VARIOUS INLET AIR TEMPERATURES WITH SPARK AT 34° B.T.D.C.

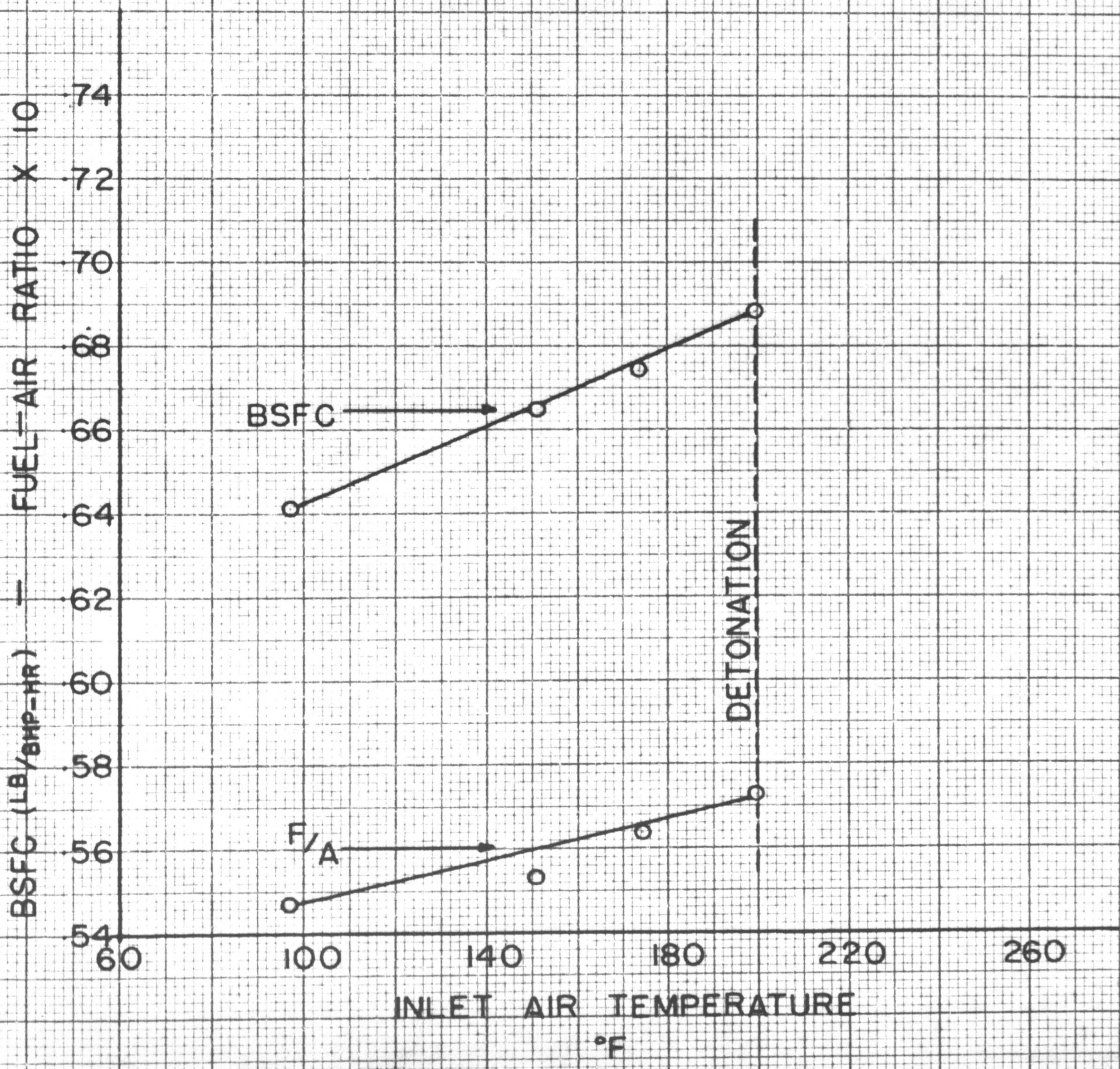
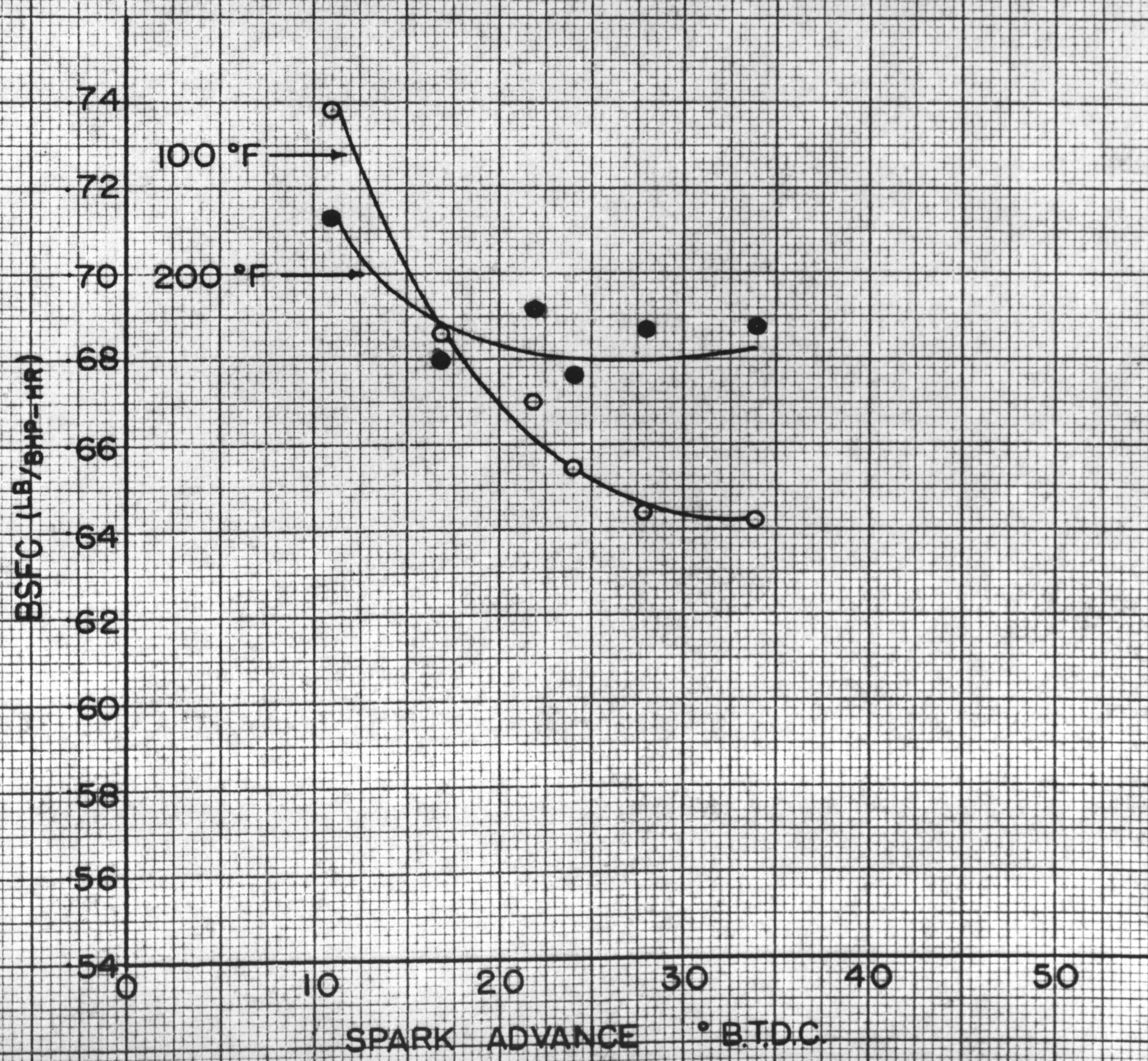


FIG. 10
BSFC AT VARIOUS SPARK ADVANCES FOR CONSTANT INLET AIR TEMPERATURES



PART V

CONCLUSIONS

The results of the tests on the engine used for this thesis show that heating the inlet air increase the brake specific fuel consumption at part throttle and at the most efficient spark advances. Besides being more efficient at the cooler temperatures of inlet air the warmer air limits the amount the spark can be advanced due to detonation which also increases the brake specific fuel consumption.

Because warm air did not prove economical on this engine does not mean that it would not be effective on other engines. It is suggested that further study be made using an engine that has difficulty operating at lean mixtures due to faulty distribution. This condition might be particularly true in the case of the air cooled aircraft engine due to the manifolds being exposed to the air stream.

It would also be desirable to test a cheaper less volitile fuel and possibly some reduction in the cost per brake horsepower hour might be achieved even though the brake specific fuel consumption increased.

Within the scope of this thesis it is apparent that the brake specific fuel consumption is increased as a result of heating the inlet air.

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VITA

The author was born July 5, 1924, at Fort Worth, Texas. He grew up in Greenwich, Connecticut and graduated from high school there in June of 1942. He enlisted in the Navy in August of that year and in November of 1943 was commissioned an ensign and sent to the Pacific as a pilot in a patrol bombing squadron. Following release to inactive duty in September of 1946 he entered the Missouri School of Mines that same month and was graduated in July of 1949. Since the fall of 1949 he has been a Graduate Assistant in the Mechanical Engineering Department of the Missouri School of Mines.