THE EFFECT OF MULTIPLE CARBURETORS AND A RACING TYPE CAMSHAFT ON THE PERFORMANCE OF A SPARK IGNITION ENGINE

by

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INTRODUCTION

During recent years, the automotive industry has been confronted with demands for increased power and improved engine performance. As a result of these demands, much emphasis has been placed upon the implements which bring about these objectives. Improved engine performance and increased output can be obtained from a given engine in the following ways:

1. Increasing the bore and stroke
2. Installation of a supercharger
3. Installation of high compression heads
4. Installation of a high lift camshaft
5. Installation of a dual intake manifold

For a given engine, increasing the bore and stroke is limited by the original design and construction of the engine; in this case fuel economy must be considered since increased displacement results in greater fuel consumption.

Supercharging an engine yields sizable improvements in power, but it seldom is used because of the very high fuel consumption. However, the Kaizer Corporation, in 1954, offered a variable speed supercharger as optional equipment. This supercharger, which is driven by variable speed pulleys and controlled by the manifold vacuum, increased the horsepower of their stock six cylinder engine from 128 to 140.

In 1949, R. E. Chestnutt and A. H. Bussey made a study of the effects of increased compression on engine performance. They found that improved fuel economy and a 11 per cent maximum power increase could be obtained by the use of 8.25:1 aluminum heads on an engine originally designed for 6.8:1 cast iron heads. They also found that an engine equipped with these heads and using regular gasoline (78-80 octane rating at that time) could be operated without serious detonation. However, they felt that serious detonation might have occurred had a higher compression ratio been used, especially during rapid acceleration. The use of high compression heads is limited by the design of the bearings, pistons, piston rings, connecting rods, valves, cooling system, and the fuel available. Each one of these items must be considered before a sizable increase in the compression ratio is made.

The increased popularity of automobile racing has brought about the use of devices which make a given engine more powerful. The high lift camshaft and the dual intake manifold along with high compression heads are among the most highly advertised and widely used equipment applicable to internal combustion engines for increasing speed and power. In almost every automotive publication, there are numerous advertisements for such devices. In 1951 and 1952, the demands for engines with increased horsepower became so great that several automobile manufacturers made available kits to increase engine output, as optional equipment. These kits contained high compression heads, dual exhaust systems, dual intake manifolds, and racing type camshafts, all of which are being used today by the more successful builders of racing engines.

The desire to know the effects of some of the more highly advertised devices for improving power and performance led to this investigation. This investigation is intended to show the effects of a dual intake manifold and a racing type camshaft installed on a stock engine in place of its conventional equipment.

II

REVIEW OF LITERATURE

1. Jennings, B. H. and E. F. Obert, Internal Combustion Engines, Second Edition, pp. 51, 165, 296-301, International Text Book Company, Scranton, Pennsylvania, 1952.

Jennings and Obert state that the power developed by a spark ignition engine is dependent upon the amount of charge inducted by the engine. The weight of the charge is not constant, but varies with speed. To raise the speed at which the maximum charge is inducted, one could use a large diameter smooth wall manifold that has long radius bends, and a cam that has a higher lift to reduce wire drawing and a dwell on the cam nose to reduce throttling affects.

The volumetric efficiency is affected by the temperature and the velocity within the manifold. A low manifold temperature will increase the density of the charge, thereby increasing the volumetric efficiency. However, a low temperature limit is reached at which the vaporized fuel will condense in the manifold, resulting in poor distribution to the cylinders and dilution of the lubricating oil.

2. Heldt, P. M., High-Speed Combustion Engines, Fifth Edition of the Gasoline Motor, Chaps. XII, XIV, pp. 351, 391, P. M. Heldt, Nyack, N. Y., 1951.

Heldt states that the volumetric efficiency of an engine is influenced in the following ways:

1. The volumetric efficiency decreases with an increase in engine speed.
2. It increases with an increase in the size of the inlet valve opening.
3. It decreases as heat is added to the incoming charge within the carburetor, the intake manifold, and the cylinder during the inlet period.

At low engine speeds the volumetric efficiency under full throttle conditions is somewhat higher than 85 per cent due to the fact that the inlet tract is at a lower temperature so that the incoming charge absorbs less heat. With an increase in engine speed the wire drawing effect of the inlet valve and the resistance to the flow of the mixture during the latter portion of the inlet tract results in a pressure lower than atmospheric within the combustion chamber during the inlet period. As a result of this low pressure the volumetric efficiency of the engine decreases. At the speed for maximum power the volumetric efficiency varies from approximately 65 to 70 per cent.

The use of a cam with a dwell in the full lift position tends to increase the volumetric efficiency because the throttling effect around the valves is reduced.

3. Faires, V. M., Applied Thermodynamics, Revised Edition, Chap. VIII, p. 119, The Macmillan Company, New York, 1949.

Faires states that the power obtained from an internal combustion engine for a given displacement depends upon the weight of the fresh combustible charge inducted into the cylinder. Any factor that reduces the density of the mixture decreases the output of the engine. In an engine the suction pressure is less than atmospheric pressure because of the throttling effect around the valves and within the inlet passages, hence, the density of the mixture is less than it would be had it been inducted without any loss in pressure. Furthermore, the internal surfaces of the engine are hot, so that the mixture is heated as it passes into the cylinder, resulting in a further decrease in the density of the incoming charge. Because of the lower density of the atmosphere, the weight of mixture drawn into an engine for a given throttle setting is less at high altitudes than that at sea level.

The products of combustion in the clearance space within the engine at the end of the exhaust stroke are at a pressure greater than atmospheric; before a new charge can enter the cylinder, these exhaust gases must expand to the intake pressure.

4. Ricardo, H. R., The High Speed Internal Combustion Engine, Fourth Edition, p. 211, Blackie and Sons, Limited, London and Glasgow, 1953.

Ricardo states that by increasing the inlet valve opening period, higher volumetric efficiencies are attained at high engine speeds, and lower volumetric efficiencies at low speeds.

5. Lichty, L. C., Internal Combustion Engines, Sixth Edition, pp. 271, 458, McGraw-Hill Book Company, Inc., New York, 1951.

Lichty states that in general the volumetric efficiency of an engine decreases with an increase in speed due to the increased velocity of the intake air and the increased pressure drop within the inlet system. However a maximum value is obtained at a specific engine speed due to the ramming effect of the inlet charge. Any change in design or operating conditions of an engine which increases its volumetric efficiency will increase the engine performance, if there is no combustion knock or overheating of the engine.

A small manifold passage will have high mixture velocities throughout the speed range of the engine. In opening the throttle for acceleration from a low speed to a high speed in a manifold of small cross-section compared to a manifold of large cross-section, there is less tendency for the fuel to condense and drop out of the air stream thereby leaning out the mixture.

The output of an engine with a small manifold is limited by the resistance to flow of the charge. Large manifolds are required to supply the necessary charge to produce a high output at high engine speeds. For a given output, acceleration at low speeds using a large manifold results in a greater consumption of fuel. The difficulties with large manifolds tend to disappear with lean mixtures.

6. Maleev, V. L., Internal Combustion Engines, First Edition, p. 143, McGraw-Hill Book Company, Inc., New York, 1933.

Maleev states that intake manifolds must be designed to give equal distribution to all cylinders. The area of the intake manifold cross-section should not be too large, because some of the particles of gasoline which are not vaporized, are kept in suspension, and may condense and drop out of the air stream when the velocity in the manifold is below 2400 ft per min.

7. Liston, Joseph, Aircraft Engine Design, First Edition, p. 223, McGraw-Hill Book Company, Inc., New York, 1942.

Liston states that the use of a cam which provides a quicker opening and closing of the valves, and a longer valve opening period (dwell) gives higher volumetric efficiencies.

8. Whitney, J. C. and Co., Catalog No. 106, p. 39, 1917-19 Archer Avenue, Chicago 16, Illinois, 1954.

In this catalog it is stated that the 3/4-grind racing cam is designed for road use and short track racing. It gives approximately 15 per cent more power, good lugging ability at low speeds, and good idling.

9. Sears, Roebuck and Company, Fall and Winter Catalog, p. 977, 2600 Lawndale Drive, Greensboro, North Carolina, 1954.

This catalog states that the manufacturers of Allstate dual intake manifolds (similar to the Wieand manifold tested in this thesis) found that an increase of as much as 15 horsepower could be obtained from an engine equipped with a dual manifold compared to the same engine equipped with a stock manifold.

10. Bussey, A. H. and R. E. Chestnutt, An Investigation of the Effects of Water Injection With Low Octane Gasoline on the Performance of a 100-Horsepower 1949 Ford V-8 Industrial Engine, Using Offenhauser High Compression Aluminum Heads (8.25:1), Thesis, Virginia Polytechnic Institute, 1950.

In their thesis, Chestnutt and Bussey concluded that an engine could be operated on regular gasoline (78-80 octane rating) without any detonation when using aluminum heads with an 8.25:1 compression ratio in place of the standard cast iron heads with a 6.8:1 compression ratio. Because of the high compression ratio afforded by the aluminum heads, they were able to increase the output of the test engine by 11 per cent.

11. Compressed Air and Gas Institute, Compressed Air Handbook, p. 257, Compressed Air and Gas Institute, 90 West Street, New York 6, New York, 1947.

This Handbook listed dimensions for standard flow nozzles and gave curves and tables for coefficients of discharge based on the nozzle throat dimensions and range of pressure drops through the nozzles. The coefficient of discharge for a 3 inch long radius flow nozzle having a 1 inch pressure drop at the throat was given as 0.986.

12. Sweeney, R. J., Measurement Techniques in Mechanical Engineering, p. 217, John Wiley and Sons, Inc., New York, 1953.

Sweeney defines a long radius flow nozzle as a nozzle having a smooth, gradual-contraction section to a throat followed by free, uncontrolled expansion back to the original pipe flow area. The upstream pressure drop to the throat, representing a gain in kinetic energy, is dissipated in vortex turbulence downstream from the nozzle. Hence, the throat pressure drop is lost by the flow nozzle.

The object of the approach curve is to avoid separation of the flow from the wall at the transition between the approach and the parallel throat sections. The shape of the approach curve is not critical as long as it is well streamlined. The long parallel throat section is intended to give time for refilling of the throat section should contraction at the throat entry occur. The variation of the discharge coefficient with Reynolds Number for an ASME long-radius flow nozzle is shown in Figure (9.14), page 218 of this reference.

An average discharge coefficient of 0.988 was calculated for the 3 inch nozzle used in this thesis. This value was based on Reynolds Numbers of

0.98×10^5 and 3.30×10^5 for engine speeds of 1000 and 4000 rpm respectively using volumetric efficiencies of 85 per cent and 70 per cent.*

13. Moss, S. A., ASME Transactions, pp. 775-785, 1916.

Moss states that a standard flow nozzle should have a smooth, rounded entrance tangent to a straight section. The curvature of the nozzle need not fit a templet as long as it is smooth. The parallel section at the end of the orifice should have a diameter approximately 1/3 the diameter of the pipe preceding the nozzle. It is possible that the discharge coefficient does not differ greatly from 0.99 even if the orifice is as much as 1/2 the diameter of the pipe. In all cases, the effect of varying velocities upstream from the nozzle is not transmitted through the orifice. The discharge jet has the same velocity at all points except within a narrow band close to the outer region of the jet.

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* Ibid, p. 8.

III

THE INVESTIGATION

A. The Object

The object of this investigation is to compare the effects of two twin-barrel carburetors, a specially designed dual intake manifold, and a racing type cam-shaft on engine performance with that of a regularly equipped spark ignition engine.

B. The Apparatus

1. Test Apparatus

    a. Test Engine
    1949 Ford V-8 Industrial Engine*
    Type 8 RNN
    Serial No. 15603-G18T
    Displacement - 239 cu inches
    Bore - 3.1875 inches
    Stroke - 3.750 inches
    Horsepower Rating - 100 @ 3800 rpm
    Maximum Torque - 181 in-lb @ 2000 rpm
    made by Ford in U. S. A.

    b. The intake manifold tested in this investigation, Figure No. 2, was a Weiand aluminum alloy dual intake manifold specially designed for installation with two twin-

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* The design of this Industrial engine is identical to the design of the automobile engine.

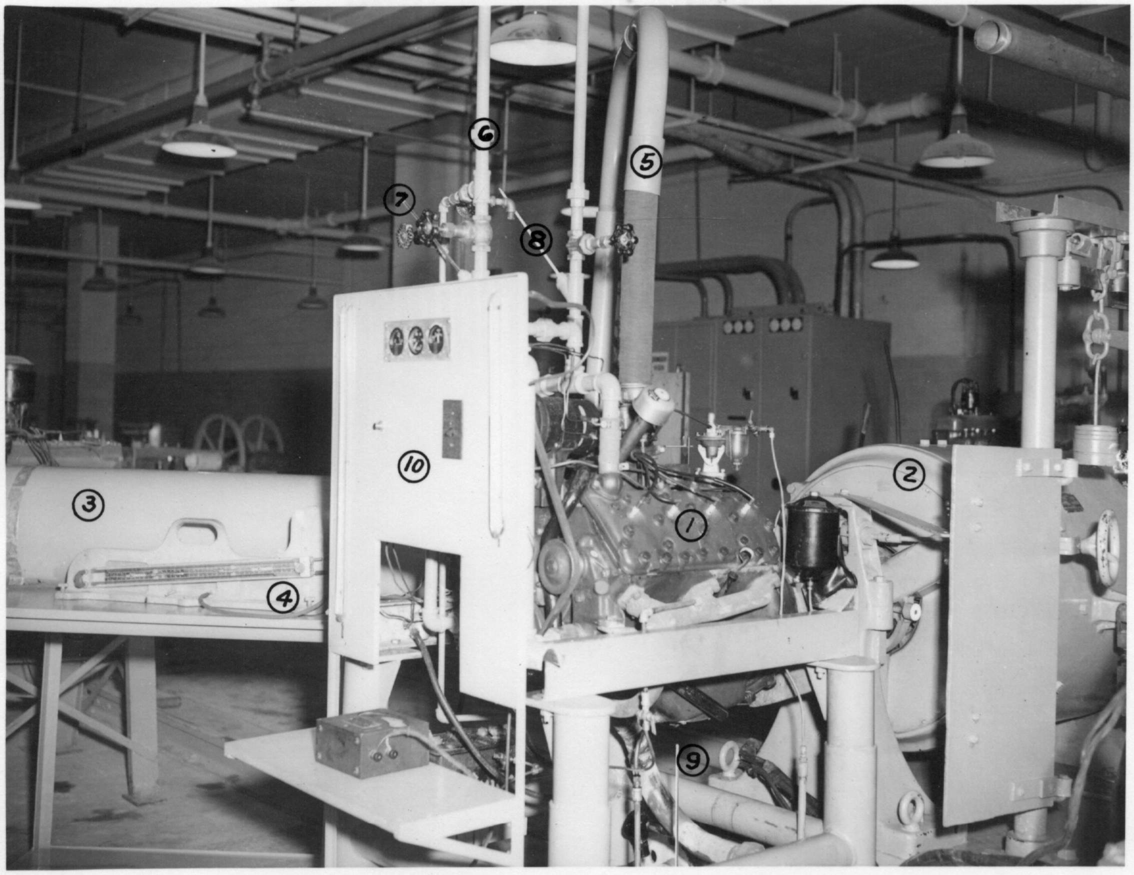


Figure 1. Photograph of General Laboratory Layout

1. 100-hp 1949 Ford V-8 Industrial Engine
2. G. E. cradle type dynamometer
3. Air-flow tank
4. Inclined manometer for determining the throat pressure drop through the flow nozzle
5. Air tank connection to the carburetor
6. Engine cooling water supply
7. Inlet water thermometer
8. Outlet water thermometer
9. Outlet oil cooling water thermometer
10. Engine instrument panel

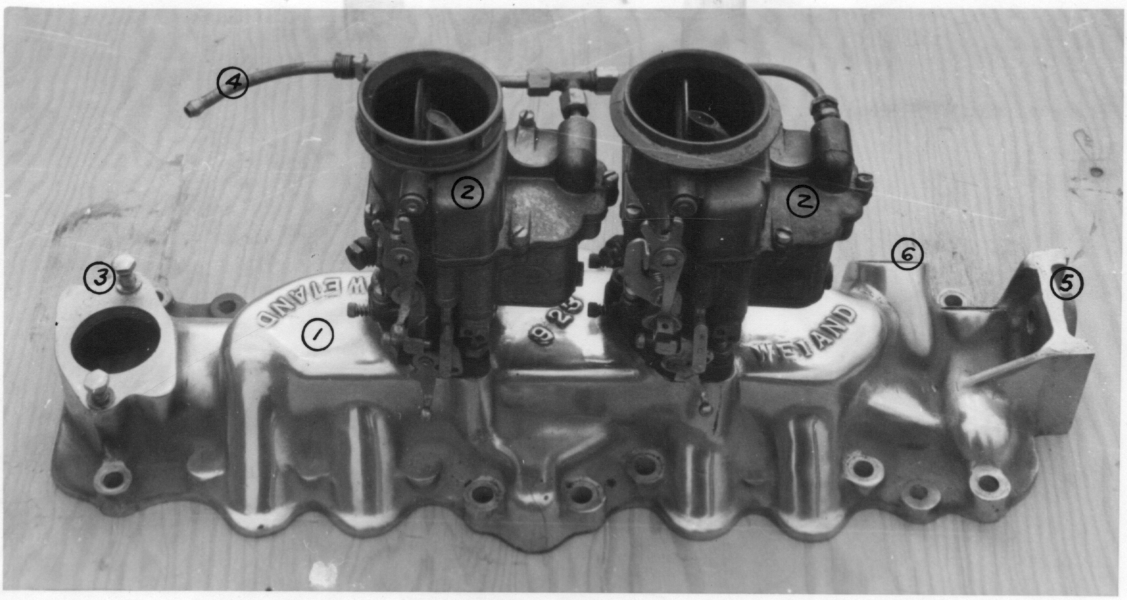


Figure 2. Photograph of Weiand Dual Intake Manifold

1. Weiand dual intake manifold
2. Two standard 1949 Ford carburetors
3. Fuel pump mount
4. Fuel line from the pump to the carburetors
5. Generator mount
6. Crankcase breather pipe connection

barrel carburetors. This intake manifold has short passages which are larger in diameter than the passages of the standard Ford manifold. These inlet passages are interconnected and have long radius bends. The purpose of this design is to reduce the flow resistance of the air-fuel mixture and to give more even distribution of the mixture to the cylinders, especially the cylinders fartherest away from the carburetor.

c. The camshaft tested in this investigation was an Iskenderian grind, 3/4-race camshaft, Figure No. 3, made of highly polished cast iron. This camshaft has a lift of 0.320 inches as compared with 0.296 inches for the standard Ford camshaft. The racing type camshaft was ground to provide a higher lift and an earlier opening and a later closing of the valves. The valve timing diagrams for both the standard Ford grind and the Iskenderian grind camshafts are shown in Figure No. 4.

Adjustable tappets were used to replace the standard non-adjustable tappets so that the clearance between the valve stem and the

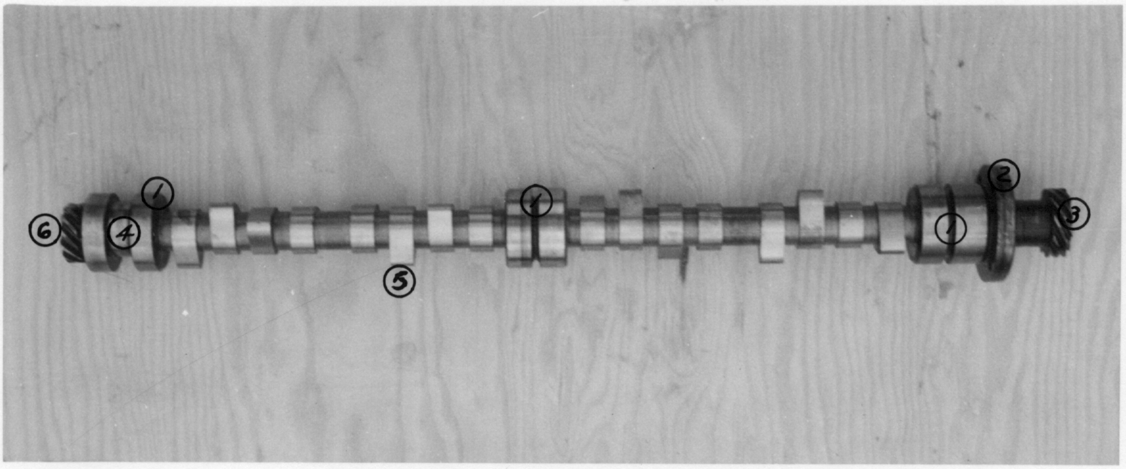
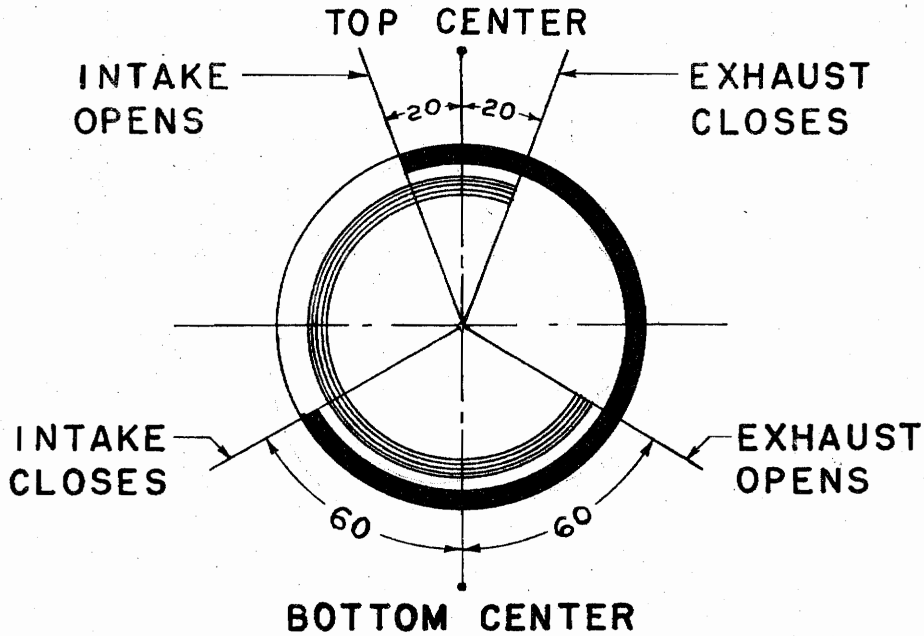
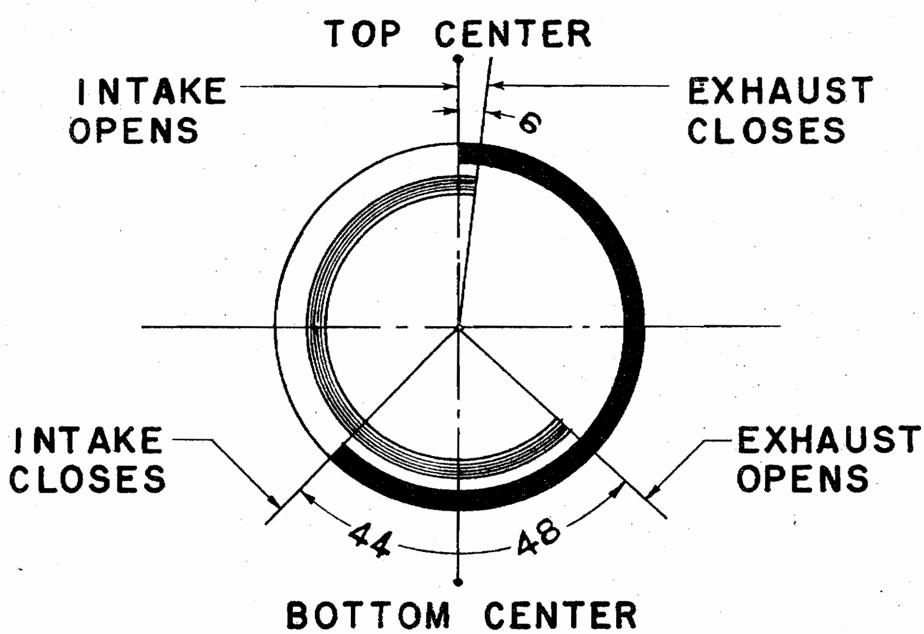


Figure 3. Photograph of the Iskenderian Grind 3/4-Race Camshaft

1. Camshaft bearings
2. Timing gear flange
3. Distributor drive gear
4. Fuel pump cam lobe
5. Valve cam lobe
6. Oil pump drive gear



ISKENDERIAN $\frac{3}{4}$-RACE CAM



STANDARD FORD CAM

FIGURE-4. VALVE TIMING DIAGRAMS

tappet could be set to specifications.*

The clearance specified for the Iskenderian grind cam is 0.014 inches for both the intake and the exhaust valves.

2. Accessory Apparatus

a. Electric Dynamometer
No. 1676510; Type TLC 50; Class 6-150-2200
Amperes 410; Volts 250
As a Generator: Absorbs 150 Hp at 2200-5000 rpm
As a Motor: Delivers 120 Hp at 1950-5000 rpm
Speed 4000-5000 rpm intermittent
Torque Arm 21.008 inches
General Electric Company, U. S. A.

b. Electric Dynamometer Control Panel
DL 2233087, CR1420; Volts 250, Hp 150
Inst. K-223541, K-2232993
General Electric Company
Schenectady, N. Y.

This Unit contains:

(1) Field Rheostat
(2) Field Potentiometer
(3) Load (Armature) Rheostat
(4) Generator Switch
CR 2960; SY 104 A Switch
Cat. 2881893

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* Note:
Racing camshafts are ground from standard camshaft cores. In order to provide a higher lift the base circle of the cam is ground to a smaller diameter; and the nose and flank are ground to the proper shape to give the desired valve timing and lift. Because the base circle is smaller, adjustable tappets must be used to lengthen the push rod so that the valve stem clearances can be set according to specifications.

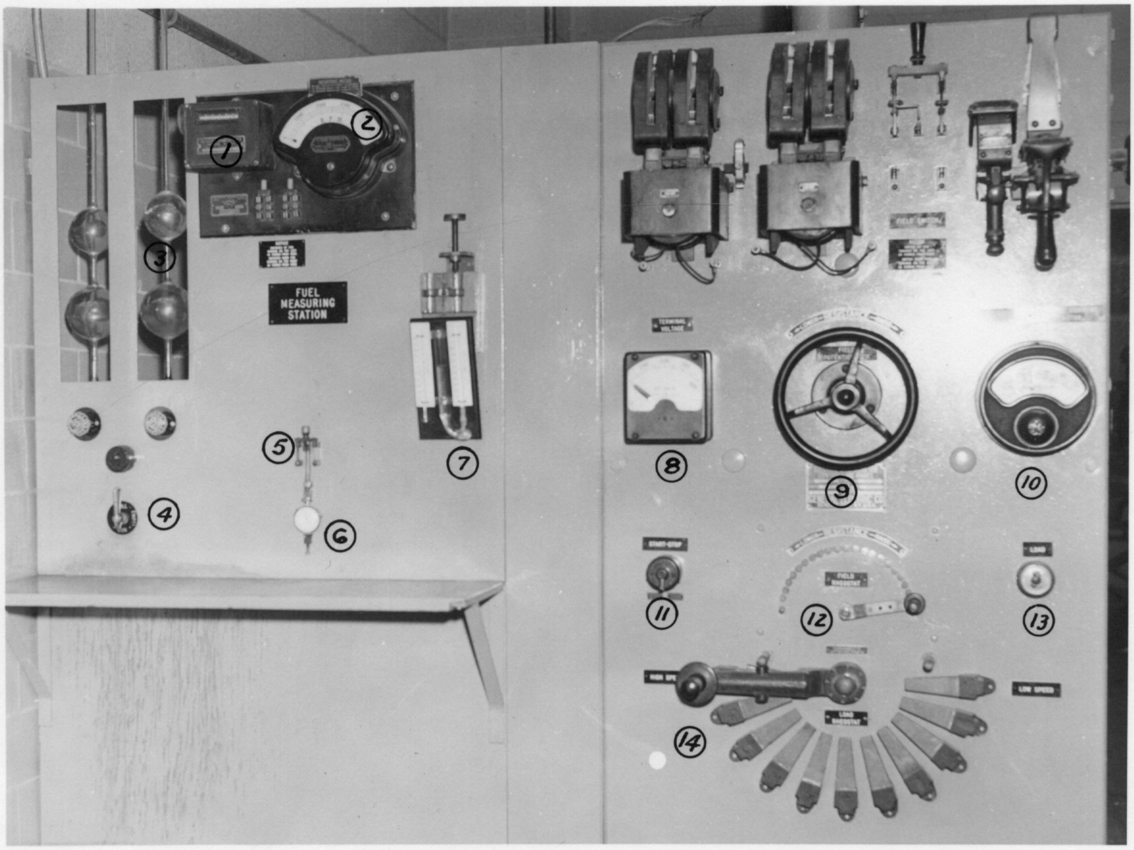

Figure 5. Photograph of Dynamometer Control Panel

1. Revolution counter
2. Tachometer
3. Fuel burettes
4. Three-way control valve for the measuring burettes
5. Stop watch and revolution counter switch
6. Stop watch
7. Hygrometer
8. D. C. Voltmeter
9. Field rheostat control
10. D. C. ammeter
11. Starting switch
12. Field rheostat vernier control
13. Loading switch
14. Load rheostat

Magnetic Switch
D-C Contactor
CR 2800-112152
Motor Load Switch
5A-250V; 10A-125V
Magnetic Switch
D-C Contactor
CR 2800-112152

(5) Direct Current Ammeter for
Armature Current
No. 921130, Type D-8, Shunt Ext.
Capacity: 400 amps as Generator
600 amps as Motor

(6) Direct Current Voltmeter for
Armature Voltage
Type DD-6
Model 8 DD 6VFL209 No. 2587846
Range 0-400 V. DC
General Electric, U. S. A.

(7) Overload Circuit Breaker
No. P-21590562, R. P. -2906
Poles 1
500 Amps, 650 Volts, DC
General Electric Company
Schenectady, N. Y.

(8) Field Switch to Reverse
Direction of Rotation of
Dynamometer 600 amps; 250 V. DC,
500 V. AC
Trumbull

c. Electric Tachometer for Approximate
Dynamometer Speed
Range 0-5000 rpm
Model 273, No. 13598, Model A
The Electric Tachometer Company
Philadelphia, Pennsylvania
Weston Electric Instrument Company
Newark, New Jersey

d. Revolution Counter
Range 0-9,999,999 Revolutions
Type 5-200, No. 1263
The Electric Tachometer Corporation
Philadelphia, Pennsylvania

e. Fuel Control Apparatus Consisting of:

(1) Two 300-500 cc glass burettes
(2) Three-way control valve
N. S. P. Co.
(3) Two Globe Valves
Lunkrenheimer - Fig. 2140
(4) Knife switch to start and stop the timing device and revolution counter

f. Engine Control Panel, including:

(1) Exhaust Manometer - Range 0 to 15" Hg
(2) Intake Manifold Manometer - Range 0 to 25" Hg
(3) Ignition Switch
(4) Starter Switch
(5) Engine Temperature Gage - C to H
(6) Oil Press Gage 0 to 200 psi
(7) Ammeter - 30 to 30 amps

g. Air Tank - 13-1/2" diameter by 56" long

(1) 3" long radius flow nozzle
(2) Inclined Manometer 0-20 "H_2O
(3) Piping for carburetor hook-up

h. Exhaust Pyrometer
Type A H, No. 0, Serial 21306
Range 0-400°C, 0-45 MV
Hoskins Manufacturing Company
Detroit, Michigan

i. Mix-Master Combustion Indicator
Combustion Efficiency 60 to 100%
Air/fuel ratio 10-16
E-3652
Electric Products Company
New York

j. Hygrometer
Range: Wet Bulb 16-124°F
Dry Bulb 18-122°F
Taylor Company
Rochester, New York

k. Battery for Ignition
6 volt auto battery

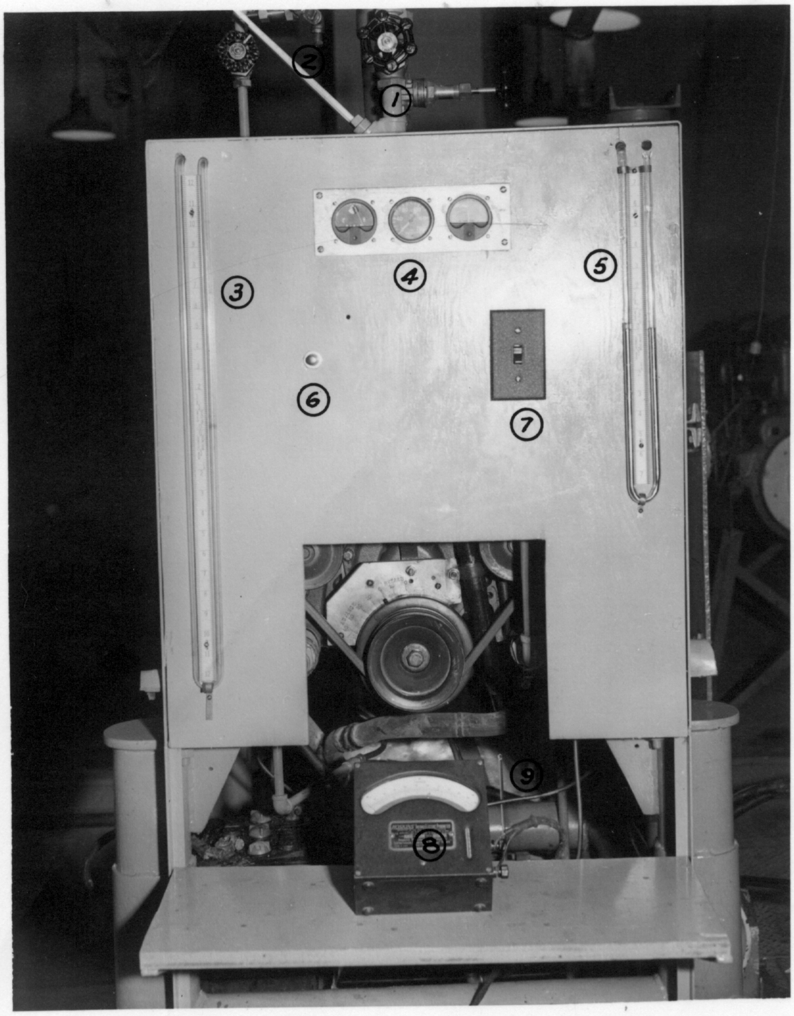

Figure 6. Photograph of Engine Instrument Panel

1. Inlet water supply valve
2. Thermometer in the inlet water supply line
3. Intake manifold manometer
4. Instrument panel consisting of an engine temperature gage, an oil pressure gage, and an ammeter
5. Exhaust manifold manometer
6. Starter button
7. Ignition switch
8. Exhaust pyrometer
9. Thermometer in the outlet oil cooling water line

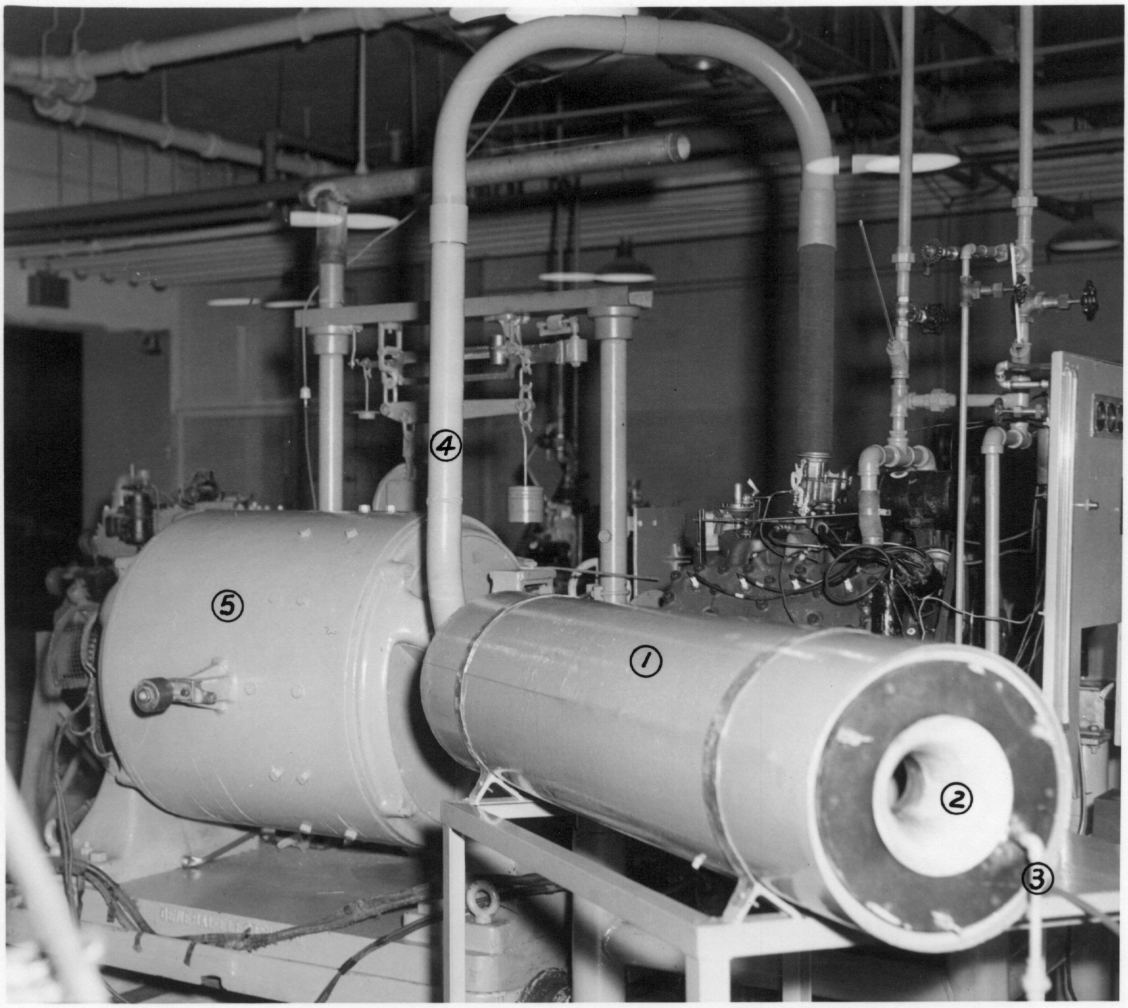

Figure 7. Photograph of Air Tank and Long Radius Flow Nozzle

1. Air tank
2. Three inch long radius flow nozzle
3. Throat pressure line to the inclined manometer
4. Pipe connecting the air tank to the single carburetor air intake
5. G. E. Cradle type dynamometer

l. Cooling Water from Supply Main
Jacket Water Lines:

(1) Globe valves and piping
Inlet - Powell Glob. Fig. 110
Outlet - Powell U. S. Gate, Fig. 500
(2) Two thermometer wells
(3) Two thermometers
Range - Inlet 0-120°F No identification
Outlet 0-400°F
Fisher E & A, U. S. A.
(4) Oil Pan Cooling Coil

(a) Inlet valve - Lunkenheimer Fig. 16
(b) One thermometer well
(c) One thermometer 0-400°F
Fisher E & A, U. S. A.
(d) Three-way Valve
1/4" Crane 125-A; 175 W

m. Stop Watch for Determining Length of Runs:
AR & J. E. Meyland
New York

n. Engine Lubricating Oil
Essolube SAE 30

o. Fuel Tank
approximately 40 gallons
M. E. Lab

p. Motor Generator Set for Supplying Armature Current to the Dynamometer:

(1) Motor
Westinghouse Type CS Induction Motor
Constant Speed, 75 Hp, Frame 635C
Volts 2300, Amperes Per Terminal 17.3
3 Phase, 60 cycle; rpm at full load -1750
Style 51M334, Serial 8151223
Continuous operation at full load 40°C rise
Service factor 1.115 at rated voltage and cycles
Westinghouse Electric and Manufacturing Corporation
East Pittsburg Works, Pittsburg, Pa.

(2) Generators:

(a) Westinghouse Compound Wound DC Generator
Type SK; 25 Kw; Frame; Frame 93, 125 v., 200 amps
Style 51M333, Serial 8151221, 1750 rpm
100% load, time 24 hours - temperature rise 40°C
Westinghouse Electric and Manufacturing Corporation
East Pittsburg, Pa.

(b) Same as (a)
Style 51M332, Serial 8151219

(3) Westinghouse Control Panel

(a) Voltmeter (Two), 0-150 V. DC

(1) Type Sx, Style 293402, Serial 1123051, Insul. Rating 750 v.
(2) Type Sx, Style 293402, Serial 11224046, Insul. Rating 750 v.

(b) Ammeters (two), 0-300 amps DC

(1) Type Sx, Style 304529, Serial 114606 Insul. Rating 750 v.
(2) Type Sx, Style 304529, Serial 1097718, Insul. Rating 750 v.

(c) Two Knife Switches for Generators 200 amp; 250 V. DC, 500 V. AC

(d) Two Rheostats for Controlling Voltage Output

(e) Two Carbon Circuit Breakers

200 amps. DC; 600 V.;
Style 501556-B

(1) Pole Unit #7, Type CL
Style 501556-B
(2) Pole Unit #10, Type CL
Style 501556-B

(4) Starting Box
Westinghouse Auto Starter Type A
Frame 11, 3 Phase, 2200 v.,
60 Cycle;
75 Hp
Style 402730
Starting Period -20 seconds

q. Motor Generator Set for Supplying Direct Field Current:

(1) General Electric Induction Motor,
Model 69A186
Type ET048-4-10-1800
Form CL; 3 amp, 60 cycle, 220 v.;
Speed full load 1740 rpm
No. EJ1162
10 Hp, continuous operation
50°C rise
General Electric Corporation
Schenectady, N. Y.

(2) Direct Current Generator,
Model 47A57
Type C065, Form AL, Compound Wound,
24 amps, 250/250 v.
Speed 1800 rpm; No. 1663447
6 Kw, continuous operation 50°C rise
General Electric Corporation
Schenectady, N. Y.

(3) Starting Mechanism

(a) Push Button Station
CR 2940 - 8579J;
Maximum Volts 600
General Electric Corporation
(b) Magnetic Switch
CR 7006-04, Cat. 1773589G2
220 v. at 60 cycles, 110 v. at
25 cycles
General Electric Corporation

C. Method of Investigation

The object of this investigation was to determine the effects of two twin-barrel carburetors, a specially designed dual intake manifold, and a racing type camshaft on engine performance. In order to obtain this objective, a series of performance tests were made on a 1949 Ford V-8 Industrial Engine.

All of the tests performed in this investigation were full throttle and general engine performance made in accordance with the recommendations of the power test code published by the American Society of Mechanical Engineers (PTC 17-1949).

Before the engine was started the spark plug gap, ignition timing, and distributor breaker points were set to Manufacturer's specifications; and the oil level, cooling water, and dynamometer coupling were checked. The engine was then started by the electric dynamometer which was supplied with 120 volt D. C. from a motor generator set. As soon as the engine began to fire, the dynamometer starting circuit was opened. The engine running at 500 rpm was then allowed to warm up for 10 minutes. The load circuit of the dynamometer was then closed, the engine speed increased to 1000 rpm, and a small load applied to the engine by decreasing the resistance in the dynamometer field circuit. The cooling water flow was regulated to maintain the jacket water temperature at 165 ± 5°F, and the oil cooling water temperature was held as near 110°F as possible. The throttle was gradually opened as the load was applied.

1. Power Tests

The full throttle power tests were made with the engine running at constant speed. Six full throttle performance tests were made on the engine as follows:

Test No. 1. The engine regularly equipped with the carburetor air intake open to the atmosphere.

Test No. 2. The engine equipped as in Test No. 1, except the carburetor was connected to an air tank with a long radius flow nozzle for the purpose of measuring the air flow to the carburetor.

Test No. 3. The engine equipped with a regular Ford camshaft, a dual intake manifold, and two twin-barrel carburetors with the air horns open to the atmosphere.

Test No. 4. The engine equipped as in Test No. 3, except with the two carburetors connected to the air tank.

Test No. 5. The engine equipped with a high lift camshaft, a standard intake manifold, and a single carburetor connected to the air tank.

Test No. 6. The engine equipped with a high lift camshaft, a dual intake manifold, and two twin-barrel carburetors connected to the air tank.

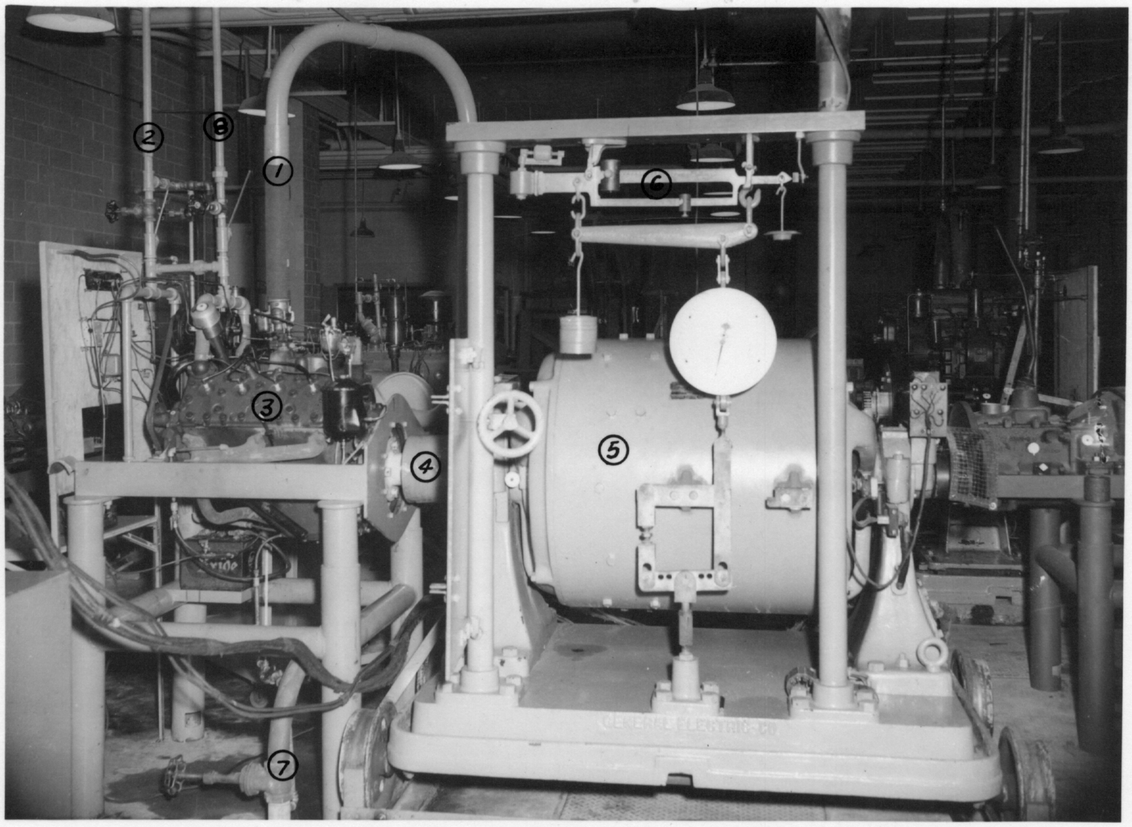

Figure 8. Photograph of the Single Carburetor Installation

1. Air tank connection to the single carburetor air intake
2. Engine cooling water supply line
3. 100-hp 1949 Ford V-8 Industrial Engine
4. Silent chain type dynamometer coupling
5. G. E. cradle type dynamometer
6. Dynamometer beam balance
7. Exhaust pipe
8. Engine cooling water discharge line

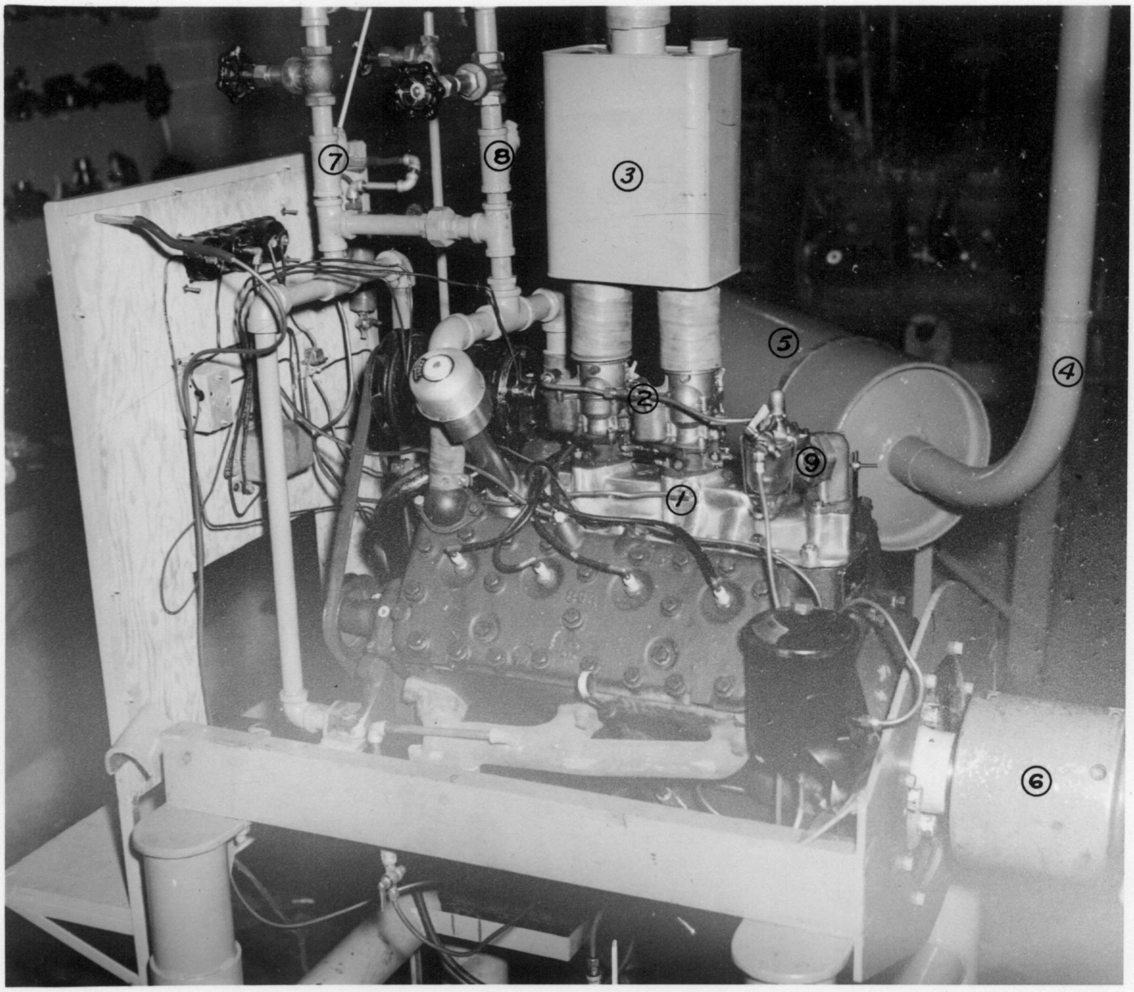

Figure 9. Photograph of Dual Carburetor Installation

1. Dual intake manifold
2. Two standard 1949 Ford carburetors
3. Auxiliary air tank
4. Air line to auxiliary air tank
5. Air tank
6. Dynamometer coupling
7. Cooling water supply line
8. Outlet water line
9. Fuel pump

The purpose of Tests No. 1 and No. 3 which were made with the carburetors open to the atmosphere was to compare the effect on power output of the engine with and without the air measuring tank.

A series of constant speed runs, ranging from approximately 1000 rpm to 4400 rpm, were made during each of the performance tests. Data (See Tables 1 through 6) were taken during each of the runs after operating conditions became steady as follows:

1. Tachometer Reading, rpm
2. Counter Reading, Revolutions
3. Time of Run, sec
4. Fuel consumed, cu cm
5. Dynamometer Brake Load, lb
6. Jacket Water Inlet and Outlet Temperatures, °F
7. Wet and Dry Bulb Temperatures, °F
8. Oil Cooling Water Inlet and Outlet Temperatures, °F
9. Exhaust Gas Temperature, °C
10. Oil Pressure, psig
11. Intake and Exhaust Manifold Pressures, in.Hg Gage
12. The Pressure Drop Through the 3 in. Flow Nozzle, in.H_2O
13. Barometric Pressure, in.Hg.

2. Constant Load Acceleration Tests

Constant load acceleration tests were made with the engine equipped as in Power Tests Nos. 2, 4, 5, and 6. For these tests the engine was operated with a 10 pound dynamometer brake load, and accelerated from 1000 rpm to 3000 rpm observing the time for the acceleration.

3. Friction Tests

Friction Horsepower Tests were made after each of the six power tests. The engine was driven by a cradle type electric dynamometer connected to a 240 volt D. C. supply. The friction tests were made with the fuel supply turned off, the ignition on, the throttle wide open, and with the water temperatures held as close to the temperatures of the power tests as possible. The data taken during the friction tests were time, revolutions, and brake load.

D. Method of Calculations

The following are sample calculations for Run 1 of Test No. 2:

a. Actual Speed, rpm = $\frac{60c}{t}$

where c = Counter Reading, revolutions
t = Time, sec

$$\text{rpm} = \frac{60(1470)}{92.2}$$

$$\underline{\text{rpm} = 955}$$

b. Brake Torque, T = FL

where T = Brake Torque, lb-ft
F = Dynamometer Load, lb
L = Length of Brake Arm, ft

$$T = (80.5)(1.75)$$

$$\underline{T = 141.0 \text{ lb-ft}}$$

c. Fuel Consumption,

$$W_f = \frac{(cc)(Sp.Gr.)(3600)(2.205)}{(t)\ 1000}$$

where W_f = Fuel Consumption, lb per hr
cc = Cubic Centimeters of Fuel
Sp.Gr. = Specific Gravity of Fuel
t = Time of Run, sec

$\frac{2.205}{1000}$ = Density of Water, lb per cc

$$W = \frac{(300)(.765)(3600)(2.205)}{(92.2)(1000)}$$

$$\underline{W = 19.8 \text{ lb/hr}}$$

d. Brake Horsepower, $Bhp = \frac{FN}{3000}$

where Bhp = Brake Horsepower
F = Dynamometer Load, lb
N = Actual Dynamometer rpm
3000 = Dynamometer Constant

$$Bhp = \frac{(80.5)(955)}{3000}$$

$$\underline{Bhp = 25.6}$$

e. Specific Fuel Consumption, $w_f = \frac{W_f}{Bhp}$

where w_f = Brake Specific Fuel Consumption, lb/Bhp-hr
W = Fuel Consumption, lb/hr
Bhp = Brake Horsepower

$$w_f = \frac{19.8}{25.6}$$

$$w_f = 0.775 \text{ lb/Bhp-hr}$$

f. Brake Mean Effective Pressure,

$$P_m = \frac{33{,}000 \text{ Bhp}}{L\ AN}$$

where P_m = Brake Mean Effective Pressure, psi
L = Length of Stroke, ft
A = Area of Piston, in.^{2}
N = Number Power Strokes per Minute

$$P_m = \frac{(33{,}000)(25.6)}{\frac{(3.75)}{12}(ft)\frac{(3.1875)^2}{(4)}(955)(.5)}$$

$\underline{P_m = 88.5 \text{ psi}}$

g. Higher Heating Value, HHV = 18,320 + 40(Deg. Be'-10)

where HHV = Higher Heating Value of Fuel, btu/lb
Deg. Be' = Gravity of Fuel
18,320 = Lower Heating Value of Gasoline as Listed by U. S. Bureau of Mines, Bulletin 43, p. 19

HHV = 18,320 + 40(53-10)

HHV = $\underline{20{,}040 \text{ btu/lb}}$

h. Brake Thermal Efficiency

$$e = \frac{2545}{W_f HHv)} \times 100$$

where e = Brake Thermal Efficiency, %
W_f = BSFC, lb/Bhp-hr
HHv = Higher Heating Value of Fuel, btu/lb
2545 = Constant, btu/hp-hr

$$e = \frac{(2545)(100)}{(0.775)(20,040)}$$

$$\underline{e = 16.4\%}$$

i. Correction Factor, $Nc = \dfrac{29.92}{P - xP_w} \times \sqrt{\dfrac{t + 460}{t_s + 460}}$

where Nc = Correction Factor
P = Barometric Pressure, in. hg.
Pw = Partial Pressure of Saturated Vapor at Temperature "t", in. hg
x = Relative Humidity, %
t = Dry Bulb Temperature, ^{o}F
t_s = Standard Temperature, $60^{o}F$

$$Nc = \frac{29.92}{27.92-(0.9352)(0.75)} \sqrt{\frac{77 + 460}{60 + 460}}$$

$$\underline{Nc = 1.116}$$

j. Corrected Brake Horsepower,

$$Bhp_c = Bhp \times N_c$$

$$Bhp_c = (25.6)(1.116)$$

$$\underline{Bhp_c = 28.6}$$

k. Friction Horsepower, $Fhp = \frac{FN}{3000}$

where
Fhp = Friction Horsepower
F = Dynamometer Brake Load, lb
N = Dynamometer rpm
3000 = Dynamometer Constant

$$Fhp = \frac{(16.3)(955)}{3000}$$

$$\underline{Fhp = 5.2}$$

l. Indicated Horsepower, Corrected,

$$Ihp_c = Fhp + Bhp_c$$

$$Ihp_c = 5.2 + 28.6$$

$$\underline{Ihp_c = 33.8}$$

m. Mechanical Efficiency, $Em = \frac{Bhp_c}{Ihp_c} \times 100$

$$Em = \frac{28.6}{33.8} \times 100$$

$$\underline{Em = 84.5\%}$$

n. Weight Flow of Air, $Wa = 18.3\ AC \sqrt{h_w P_a}$

where
W = Weight Flow of Air, lb/sec
A = Area of Orifice, ft^2, = 0.0491
C = Nozzle Discharge Coefficient, = 0.988
h_w = Pressure Drop at Nozzle Throat, in. H_2O
P_a = Density of Air, lb/ft^3, = 0.069

$$Wa = 18.3(.0491)(0.988)\sqrt{(.025)(.069)}$$

$$\underline{Wa = 0.0368 \text{ lb/sec}}$$

o. Weight of the Air Based on the Piston Displacement at Ambient Temperature and Pressure,

$$W_d = V_d\left(\frac{N}{120}\right)P_a$$

where W_d = Weight of Air, lb/sec
V_d = Displaced Volume, ft^3
N = Engine rpm
Pa = Density of the Air, lb/ft^3

$$W_d = \frac{(239\quad)}{1728}\frac{(1000)}{120}(0.069)$$

$$W_d = 0.0795 \text{ lb/sec}$$

p. Volumetric Efficiency, $E_v = \frac{Wa}{W_d} \times 100$

$$E_v = \frac{0.0368}{0.0761} \times 100$$

$$E_v = \underline{48.4\%}$$

q. Air-Fuel Ratio, $A/F = (3600)\left(\frac{Wa}{W_f}\right)$

where Wa = Weight Flow of Air, lb/sec
W_f = Fuel Consumption, lb/hr

$$A/F = 3600\frac{(.0368)}{(19.8)}$$

$$\underline{A/F = 6.70}$$

E. Results

The results of this thesis are in the form of Tables and Curves which may be found in Section X as follows:

1. Tables 1 through 6:

   These Tables contain data obtained during Test 1 through Test 6, inclusive.

2. Tables 7 through 12:

   These Tables contain the calculated results for the data in Table 1 through Table 6, inclusive.

3. Curves Sheet 1 through 6:

   These Curves are typical performance curves drawn from the results in Table 7 through Table 12, inclusive.

4. Curve Sheet 7:

   This Curve Sheet compares brake horsepower vs. engine speed for:

   (1) The stock engine.
   (2) The engine equipped with dual carburetors.
   (3) The engine equipped with a 3/4-race camshaft.
   (4) The engine equipped with dual carburetors and a 3/4-race camshaft.

5. Curve Sheet 8:

   These Curves compare the torque and brake specific fuel consumption vs. engine speed for:

   (1) The stock engine.
   (2) The engine equipped with dual carburetors.
   (3) The engine equipped with a 3/4-race camshaft.
   (4) The engine equipped with dual carburetors and a 3/4-race camshaft.

IV

DISCUSSION OF RESULTS

This thesis consists of a series of performance tests made on a 100 Hp, 1949 Ford V-8 Industrial Engine to determine the effects of a racing cam and dual carburetors on Engine Performance.

To the average automobile owner, the engine performance characteristics which are most important are the power output, the ability to accelerate, and the fuel economy. Hence the results of this thesis will be discussed in regard to these characteristics.

A. Power Output

Curve Sheet 7 shows the effects on engine performance of the 3/4-race cam and the dual manifold each tested separately and then compositely with an air tank attached to the engine air intake for measuring air flow.

The engine equipped with dual carburetors developed a maximum of 87.6 Hp at 3500 rpm which is a 9.5 per cent increase in power output over that of the stock engine which developed a maximum of 80.0 Hp at 3400 rpm. This power increase is accredited to reduced wiredrawing within the carburetors and the intake manifold resulting in a manifold pressure approaching that of the atmosphere and to a more even distribution of the charge to the cylinders brought about by the large diameter and long radius inlet ports which are more nearly equal in length.

The test engine equipped with the standard manifold and a 3/4-race cam delivered 89.0 Hp at 3650 rpm giving an increase of 11.25 per cent in maximum power at a speed 250 rpm higher than that of the stock engine. This power increase was brought about by the higher valve lift and the earlier opening of the valves which permitted a greater charge to enter the cylinder due to the ramming effect in the intake manifold at high speeds.

When the 3/4-race cam and dual carburetors were installed together on the test engine, it developed 94.0 Hp at 3850 rpm. This gave a 17.5 per cent increase in maximum power output over that of the stock engine at a 450 rpm higher speed.

These results were obtained from the engine when the air tank was in use. As shown by Test 1 for the single carburetor installation and Test 3 for the dual carburetor installation the pressure loss through the air tank caused a 10.85 per cent and a 6.42 per cent respective decrease in maximum power output. In each case maximum power occurred from 350 to 400 rpm below the speeds obtained without the use of an air tank. It is possible that the percentage increases in maximum power might have been slightly different had the engine been run without an air tank. Test 1 and Test 2 have little significance since an engine is usually equipped with an air filter which reduces the flow of air to the carburetor.

The purpose of the air tank was to furnish data for measuring the rate of air flow to the engine. This was done so that the expected increase in engine output could be accounted for in terms of the volumetric efficiency. However the results obtained for the volumetric efficiency had very little significance in that the volumetric efficiencies ranged from 34 to 60 per cent throughout the tests. The volumetric efficiency for an engine should range from 65 to 80 per cent.* The calculated values of volumetric efficiency in this thesis are undoubtedly incorrect since the air-fuel ratios, as calculated from the air flow data, were extremely small which indicated rich mixtures. Several of these air-fuel ratios were less than 7 to 1 which is the lower limit of combustion for gasoline.

One source of error which was discovered during Test 5 was a slight movement of the manometer leveling

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* Ibid, page 8.

screws which changed the zero reading of the manometer. This movement was caused by the vibrations from the engine being transmitted to the manometer table by way of the air pipe which was connected to the carburetor. In an attempt to counteract this disturbance the manometer was transferred to another table which was isolated from engine vibrations. However since the manometer readings were not materially changed, it was concluded that there were additional errors which could not be accounted for or perhaps it is possible that the manometer was not sufficiently sensitive to indicate the small pressure differentials.

B. Acceleration

The accelerating ability of an engine is dependent upon the engine torque. Curve Sheet 8 shows a comparison of the torque curves for the engine with various combinations of equipment. According to the acceleration tests made in this investigation, the stock engine required 12 seconds to accelerate from 1000 rpm to 3000 rpm at full throttle with a 10 lb initial brake load. The same engine equipped with dual carburetors required only 11.0 seconds to accelerate to this speed whereas 11.6 seconds were required for the engine using a 3/4-race cam. For a combination of equipment using a

3/4-race cam and dual carburetors it was found that the engine required only 10.95 seconds to attain 3000 rpm.

The acceleration time can be explained by comparing the torque curves plotted for the engine with various combinations of equipment. The stock engine developed a maximum torque of 161 lb-ft at 1500 rpm and decreased rapidly beyond 1500 rpm.

The engine with dual carburetors developed a maximum torque of 158 lb-ft at 2000 rpm. Although in this case the maximum torque was less than that for the stock engine, the faster acceleration time is explained by the fact that the torque is nearly constant from 1000 rpm to 2800 rpm which increases the accelerating ability of the engine.

The maximum torque for the engine equipped with a 3/4-race cam was 156.5 lb-ft at 2200 rpm. The torque for this installation was slightly lower than that for the dual carburetor installation below speeds of 3700 rpm. This fact indicates that the acceleration time in this case was greater than that for the engine equipped with dual carburetors.

For the engine using a 3/4-race cam and dual carburetors the maximum torque of 151.5 lb-ft occurred at 2400 rpm. Below engine speeds of 2900 rpm the torque for this installation was lower than that for the engine using a 3/4-race cam. This low torque indicates that the acceleration time would be greater than that for the engine equipped with a 3/4-race cam. However the time observed during the acceleration test made on the engine equipped with dual carburetors and a 3/4-race cam does not support this analysis. The variation in the acceleration time for this installation is explained by the fact that during the acceleration test a different person made the time reading which very probably introduced an error since the reaction time varies for different individuals.

The low values of torque for the 3/4-race cam installation occurring at low engine speeds were caused by the large overlap of the inlet and exhaust valves; whereas in the case of the dual carburetor installation, the low values of torque probably resulted from incomplete combustion due to rich mixtures afforded by improper carburetor jets.

C. Fuel Consumption

A minimum brake specific fuel consumption of 0.68 lb/Bhp-hr at 2500 rpm was obtained for the stock engine. The engine when equipped with dual carburetors should theoretically give a lower Bsfc, however, the minimum value obtained in this thesis was 0.795 lb/Bhp-hr at 2100 rpm a 16.9 per cent increase over that of the stock engine. The increased fuel consumption was probably caused by improper mixture ratios due to the use of standard carburetor jets which were designed for single carburetor installation. The flow rate through each carburetor in a dual carburetor installation is approximately one-half that for a single carburetor. A carburetor should be designed to give maximum economy for a particular range of flow rates; however, in this thesis the necessary jets were not available.

Improved fuel economy was obtained for the engine equipped with a 3/4-race cam. The Bsfc was equal to that for a stock engine at speeds of 1000 rpm to 1600 rpm and less than that of the stock engine at speeds from 1600 rpm to 4300 rpm. The minimum Bsfc obtained was 0.645 lb/Bhp-hr at 2800 rpm, 4.4 per cent less than that for the stock engine. This decrease in the Bsfc is the result of a large increase in power provided by the greater density of the mixture which was drawn into the cylinder.

A minimum Bsfc of 0.82 lb/Bhp-hr at 3400 rpm was obtained from the engine equipped with a 3/4-race cam and dual carburetors. This value was a 20.6 per cent increase over that of the stock engine. Again the increased fuel consumption is probably due to the use of improper carburetor jets.

At the speeds for maximum power, the Bsfc was decreased 4.4 per cent over that of a stock engine by the use of the 3/4-race cam, whereas it increased 15.88 per cent for the dual carburetor installation and for the combination of dual carburetors and the 3/4-race camshaft.

V

CONCLUSIONS

The following conclusions were drawn from the results of this investigation:

A. A 3/4-race camshaft when installed in a stock engine gave a 11.25 per cent increase in maximum power output and a 4.41 per cent decrease in Bsfc at a speed of 250 rpm higher than that for the standard engine.

B. By the use of the 3/4-race cam the minimum Bsfc was reduced by 4.4 per cent and the fuel economy was improved for all speeds above 1600 rpm.

C. Good idling and improved acceleration were afforded by the use of the 3/4-race cam.

D. The installation of dual carburetors with standard jets increased the maximum power output of the stock engine by 9.5 per cent and improved the accelerating ability.

E. The use of dual carburetors with standard jets increased the minimum Bsfc 16.9 per cent and the Bsfc at the speed for maximum power by 15.88 per cent over that of the stock engine. With this installation fuel consumption increased for all speeds.

F. It is possible that the fuel economy would have been improved had the proper jets been used with the dual carburetors.

G. A combination of dual carburetors and a 3/4-race cam increased the maximum power output by 17.5 per cent and the speed for maximum power increased approximately 400 to 450 rpm above that for maximum power of the stock engine.

H. With the use of a 3/4-race cam and dual carburetors the minimum Bsfc increased 20.6 per cent and the Bsfc at the speed for maximum power increased 15.88 per cent over that of the stock engine.

I. The equipment tested in this investigation proved beneficial at speeds above 3000 rpm, but the only advantage obtained below 3000 rpm was improved accelerating ability.

J. Both the 3/4-race cam and the dual carburetors increased the accelerating ability of the engine. However, the engine equipped with dual carburetors accelerated faster than the engine with the 3/4-race cam.

VI

FINAL SUMMARY

The 3/4-race cam and dual carburetors tested in this investigation provided moderate increases in maximum power output at speeds above 3000 rpm. The fuel consumption was reduced by the use of the racing type cam and increased with the use of two standard carburetors. However in the latter case it is possible that the economy would have been improved had the proper carburetor jets been installed.

The maximum power afforded by the dual carburetors occurred at about the same speed as that for the stock engine whereas the maximum power afforded by the 3/4-race cam occurred at 250 rpm higher than that of the stock engine.

Large increases in maximum power output and increase in the speed for maximum power were provided by a combination of a 3/4-race cam and dual carburetors. However the gain in speed and power was obtained at the expense of fuel consumption.

The above modifications in engine equipment were beneficial for high speed operation, the only advantage at low speeds being an increase in the accelerating ability of the engine.

VII

RECOMMENDATIONS

The following are recommendations suggested for future investigation:

1. To determine the effects of a 1/2-race, full-race, and a track-grind camshaft on engine performance.
2. To determine the optimum spark advance for an engine equipped with a 1/2-race, 3/4-race, full-race, and track-grind camshaft.
3. To determine the effects of different size carburetor jets on engine performance in a dual carburetor installation.
4. To determine effects on engine performance of a manifold designed for three carburetors.
5. To determine the effects on engine performance of enlarging the inlet and exhaust ports and valves.
6. To determine the effects of various types of superchargers on engine performance.

VIII

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IX

BIBLIOGRAPHY

A. Literature Cited

(1) Jennings, B. H. and E. F. Obert, Internal Combustion Engines, Second Edition, pp. 51, 165, 296-301, International Text Book Company, Scranton, Pennsylvania, 1952.

(2) Heldt, P. M., High-Speed Combustion Engines, Fifth Edition of "The Gasoline Motor," Chaps. XIII, XIV, pp. 351, P. M. Heldt, Nyack, N. Y., 1951.

(3) Faires, V. M., Applied Thermodynamics, Revised Edition, Chap. VIII, p. 119, The Macmillan Company, New York, 1949.

(4) Ricardo, Sir H. R., The High Speed Internal Combustion Engine, Fourth Edition, p. 211, Blackie and Sons, Limited, London and Glasgow, 1953.

(5) Lichty, L. C., Internal Combustion Engines, Sixth Edition, pp. 271, 458, McGraw-Hill Book Company, Inc., New York, 1951.

(6) Maleev, V. L., Internal Combustion Engines, First Edition, p. 143, McGraw-Hill Book Company, Inc., New York, 1933.

(7) Liston, Joseph, Aircraft Engine Design, First Edition, p. 223, McGraw-Hill Book Company, Inc., New York, 1942.

(8) Whitney, J. C. and Co., Catalog No. 106, p. 39, 1917-19 Archer Avenue, Chicago 16, Ill., 1954.

(9) Sears, Roebuck and Co., Fall and Winter Catalog, p. 977, 2600 Lawindale Drive, Greensboro, N. C., 1954.

(10) Bussey, A. H. and R. E. Chestnutt, An Investigation of the Effects of Water Injection With Low Octane Gasoline on the Performance of a 100-Horsepower 1949 Ford V-8 Industrial Engine, Using Offenhauser High Compression Aluminum Heads (8.25:1), Thesis, Virginia Polytechnic Institute, 1950.

(11) Compressed Air and Gas Institute, Compressed Air Handbook, p. 257, Compressed Air and Gas Institute, 90 West Street, New York 6, N. Y.

(12) Sweeney, R. J., Measurement Techniques in Mechanical Engineering, p. 217, John Wiley and Sons, Inc., New York, 1953.

(13) Moss, S. A., ASME Transactions, pp. 775-785, 1916.

B. General Literature

(1) Report of ASME Special Research Committee on Fluid Meters, Fluid-Meters-Their Theory and Applications, Part 1, 1932.

(2) Cousins, F. M., Analytical Design of High Speed Internal Combustion Engines, Pitman Publishing Corporation, New York, and Chicago, 1941.

(3) Rogowski, A. R., Elements of Internal Combustion Engines, McGraw-Hill Book Company, New York, 1953.

(4) Taylor, C. F., The Internal Combustion Engine, Revised Edition, International Text Book Company, Scranton, Pa., 1950.

(5) Obert, E. F., Internal Combustion Engines, Second Edition, International Text Book Company, Scranton, Pa. 1950.

(6) Vincent, E. T., Supercharging the Internal Combustion Engine, First Edition, McGraw-Hill Book Company, N. Y. - Toronto-London, 1948.

(7) Ricardo, H. R., The Internal Combustion Engine, Vol. II, High Speed Engines, D. Van Nostrand Company, Eight Warren Street, N. Y., 1923.

(8) Streeter, R. L., Internal Combustion Engines, Second Edition, McGraw-Hill Book Company, New York, 1923.

(9) Butler, Edward, Carburetors, Vaporizers, and Adjusting Valves, Second Edition, Charles Griffin and Company, Limited, Exeter Street, Strand, W. C. Z., London, 1919.

(10) Carpenter, R. C., and H. Diederichs, Internal Combustion Engines, Second Edition Revised, D. Van Nostrand Company, 23 Murray and 27 Warren Street, New York, 1909.

(11) Polson, J. A., Internal Combustion Engines, John Wiley and Sons, Inc., New York, 1931.

(12) Judge, A. W., The Testing of High Speed Internal Combustion Engines, D. Van Nostrand Company, 8 Warren Street, New York, 1925.

X. TABLES AND CURVES

TABLE 1

DATA FOR TEST 1

Engine: 100 Hp, 1949 Ford V-8
Engine Equipment:
Ford Camshaft
Standard Intake Manifold
Air Tank Disconnected
Timing and Point Setting:
Manufacturers Specification

Date: July 7, 1954
Observers: R. F. Stebar
B. B. Lindamood
Throttle Setting: Full Throttle
Gasoline: Esso Regular
(80.9 to 82 Octane)
Average Ambient Air Temperature: 80°F
Barometric Pressure: 27.94 in.Hg

Full Throttle Performance Test

Run No.	Approx. Rpm	Counter Reading	Brake Load lb	Fuel cc	Time sec	Pressures				Temperatures				Hygrometer	
						Exh. In.Hg	Intake In.Hg	Oil psi	Throat Drop In.H_2O	Jacket Water, °F In	Jacket Water, °F Out	Oil Out	Exh. °C	Wet Bulb °F	Dry Bulb °F
1	1000	1450	79.4	300	87.0	0.45	0.20	38	-	80	156	82	350	69	72
2	1500	1480	84.0	300	60.5	0.90	0.35	48	-	80	164	83	425	70	73
3	2000	1575	84.5	300	47.25	1.25	0.65	50	-	80	168	84	505	70	74
4	2500	1690	81.0	300	40.5	2.45	1.05	51	-	80	167	88	560	71	74
5	3000	1825	74.5	300	36.25	3.25	1.45	52	-	80	172	91	605	71	75
6	3200	1875	72.0	300	35.0	3.50	1.50	52	-	80	171	92	612	72	75
7	3400	1937	70.0	300	34.5	3.75	1.62	52	-	80	172	94	628	72	75
8	3600	2005	67.0	300	33.5	4.00	1.75	52	-	80	170	95	638	72	76
9	3800	3465	62.4	500	55.0	3.70	1.88	52	-	80	171	96	648	72	76
10	4000	3595	60.0	500	54.0	4.00	1.95	52	-	80	172	96	658	72	76

Friction Test

Run No.	Approx. Rpm	Counter Reading	Brake Load lb	Time sec
1	1000	985	14.5	60
2	1500	1470	18.0	60
3	2000	2048	21.9	60
4	2500	2530	24.5	60
5	3000	3040	26.5	60
6	3200	3185	27.8	60
7	3400	3325	28.4	60
8	3600	3660	28.5	60
9	3800	3940	30.3	60
10	4000	4120	32.2	60

TABLE 2

DATA FOR TEST 2

Engine: 100 Hp, 1949 Ford V-8
Engine Equipment:
Ford Camshaft
Standard Intake Manifold
Air Tank Connected to Carburetor Intake
Timing and Point Setting:
Manufacturers Specification

Date: August 7, 1954
Observers: R. F. Stebar
J. C. Bouldin
Throttle Setting: Full Throttle
Gasoline: Esso Regular
(80.9 to 82 Octane)
Average Ambient Air Temperature: 80°F
Barometric Pressure: 27.92 in.Hg

Full Throttle Performance Test

Run No.	Approx. Rpm	Counter Reading	Brake Load lb	Fuel cc	Time sec	Pressures Exh. In.Hg	Pressures Intake In.Hg	Pressures Oil psi	Pressures Throat Drop In.H_2O	Temperatures Jacket Water, °F In	Temperatures Jacket Water, °F Out	Temperatures Oil Out	Temperatures Exh. °C	Hygrometer Wet Bulb °F	Hygrometer Dry Bulb °F
1	1000	1470	80.5	300	92.2	0.24	0.22	32	0.025	85	159	98	370	71	77
2	1500	1474	79.5	300	58.0	0.62	0.40	46	0.085	85	164	97	440	71	77
3	2000	1595	81.0	300	47.6	1.08	0.72	50	0.105	85	165	101	535	72	77
4	2500	1769	78.3	300	44.2	1.90	1.14	51	0.139	85	163	102	610	72	77
5	3000	1847	71.1	300	36.8	2.63	1.58	52	0.175	85	166	100	642	72	78
6	3200	1949	67.3	300	36.8	2.88	1.61	52	0.184	85	164	102	655	72	78
7	3400	2008	62.8	300	35.7	3.25	1.80	52	0.206	85	167	103	670	72	78
8	3600	3437	58.9	500	57.2	3.70	1.96	52	0.219	87	167	110	685	73	79
9	3800	3568	54.8	500	56.3	3.95	2.00	52	0.234	88	168	120	700	73	79
10	4000	3680	47.8	500	54.9	4.20	2.15	52	0.246	89	169	124	710	73	80

Acceleration Test

Dynamometer Load, lb	Initial Speed Rpm	Final Speed Rpm	Time sec
10	1000	3000	12.0

Friction Test

Run No.	Approx. Rpm	Counter Reading	Brake Load, lb	Time sec
1	1000	468	16.1	30
2	1500	748	19.0	30
3	2000	1012	23.0	30
4	2500	1281	26.2	30
5	3000	1528	29.1	30
6	3500	1782	30.7	30
7	3800	1956	32.5	30
8	4100	2032	34.4	30

TABLE 3

DATA FOR TEST 3

Engine: 100 Hp, 1949 Ford V-8
Engine Equipment:
    Ford Camshaft
    Dual Intake Manifold
    Air Tank Disconnected
Timing and Point Setting:
    Manufacturers Specification

Date: July 27, 1954
Observers: R. F. Stebar
    B. B. Lindamood
Throttle Setting: Full Throttle
Gasoline: Esso Regular
    (80.9 to 82 Octane)
Average Ambient Air Temperature: 80°F
Barometric Pressure: 28.10 in.Hg

Full Throttle Performance Test

Run No.	Approx. Rpm	Counter Reading	Brake Load lb	Fuel cc	Time sec	Pressures				Temperatures				Hygrometer	
						Exh. In.Hg	Intake In.Hg	Oil psi	Throat Drop In.H_2O	Jacket Water, °F In	Jacket Water, °F Out	Oil Out	Exh. °C	Wet Bulb °F	Dry Bulb °F
1	1000	1600	79.5	300	98.1	0.30	0.05	35	-	75	166	99	365	68	74
2	1500	1465	81.5	300	59.3	0.90	0.10	48	-	75	160	97	435	68	74
3	2000	1442	83.0	300	43.5	1.60	0.25	50	-	75	164	100	495	68	75
4	2500	1490	82.6	300	35.9	2.40	0.36	50	-	75	168	102	560	69	76
5	3000	1550	78.7	300	31.9	3.40	0.50	52	-	75	168	99	600	69	76
6	3200	2747	76.3	500	51.5	3.70	0.54	52	-	75	168	99	612	69	77
7	3400	2852	74.0	500	50.7	4.10	0.60	52	-	75	168	103	628	69	77
8	3600	2930	71.7	500	49.2	4.40	0.64	52	-	75	166	102	645	69	77
9	3800	3002	67.7	500	47.6	4.70	0.65	52	-	75	169	100	656	69	77
10	4000	3085	64.3	500	46.2	5.00	0.73	52	-	75	167	99	665	69	78
11	4200	3115	59.8	500	45.0	5.20	0.77	52	-	75	167	100	670	69	78

Friction Test

Run No.	Approx. Rpm	Counter Reading	Brake Load lb	Time sec
1	1000	490	14.5	30
2	1500	750	19.0	30
3	2000	1010	22.1	30
4	2500	1220	25.0	30
5	3000	1472	27.3	30
6	3200	1545	28.2	30
7	3400	1642	29.7	30
8	3600	1692	30.6	30
9	3800	1720	31.1	30
10	4000	2135	34.3	30

TABLE 4

DATA FOR TEST 4

Engine: 100 Hp, 1949 Ford V-8
Engine Equipment:
Ford Camshaft
Dual Manifold
Air Tank Connected to Carburetor
Timing and Point Setting:
Manufacturers Specification

Date: August 6, 1954
Observers: R. F. Stebar
J. P. H. Mason
Gasoline: Esso Regular
(80.9 to 82 Octane)
Throttle Setting: Full Throttle
Average Ambient Air Temperature: 80°F
Barometric Pressure: 27.90 in.Hg

Full Throttle Performance Test

Run No.	Approx. Rpm	Counter Reading	Brake Load lb	Fuel cc	Time sec	Pressures				Temperatures				Hygrometer	
						Exh. In.Hg	Intake In.Hg	Oil psi	Throat Drop In.H_2O	Jacket Water, °F In	Jacket Water, °F Out	Oil Out	Exh. °C	Wet Bulb °F	Dry Bulb °F
1	1000	1420	79.1	300	90.3	0.30	0.05	38	0.030	79	162	100	352	66	74
2	1500	1410	80.9	300	57.1	0.90	0.12	48	0.080	79	159	100	440	66	74
3	2000	1380	80.9	300	41.1	1.50	0.32	50	0.170	79	167	102	500	67	75
4	2500	1400	80.3	300	34.0	2.40	0.58	52	0.190	79	163	98	548	66	74
5	3000	1500	76.3	300	29.8	3.60	0.78	52	0.285	79	170	104	585	67	75
6	3200	2530	74.2	500	47.7	4.00	0.82	52	0.316	79	169	105	605	68	76
7	3400	2620	69.1	500	46.6	4.40	0.85	52	0.341	79	168	105	618	68	76
8	3600	2620	65.6	500	44.9	4.80	0.95	52	0.379	79	165	112	632	68	76
9	3800	2760	62.1	500	43.5	5.10	1.00	52	0.398	79	166	118	640	68	77
10	4000	2840	58.5	500	43.5	5.40	1.10	52	0.410	79	170	120	650	68	77
11	4200	2920	51.0	500	41.9	5.60	1.15	52	0.424	79	170	126	650	68	77

Acceleration Test

Dynamometer Load, lb	Initial Speed Rpm	Final Speed Rpm	Time sec
10	1000	3000	11.0

Friction Test

Run No.	Approx. Rpm	Counter Reading	Brake Load, lb	Time sec
1	1000	490	15.8	30
2	1500	730	18.3	30
3	2000	1000	21.6	30
4	2500	1270	24.5	30
5	3000	1470	26.4	30
6	3200	1550	27.3	30
7	3500	1690	29.7	30
8	4000	1890	35.2	30
9	4200	2070	37.8	30

TABLE 5

DATA FOR TEST 5

Engine: 100 Hp, 1949 Ford V-8
Engine Equipment:
Iskenderian 3/4-Race Camshaft
Standard Intake Manifold
Air Tank Connected to Carburetor
Timing and Point Setting:
Manufacturers Specification

Date: August 12, 1954
Observers: R. F. Stebar
B. B. Lindamood
Throttle Setting: Full Throttle
Gasoline: Esso Regular
(80.9 to 82 Octane)
Average Ambient Air Temperature: 75°F
Barometric Pressure: 28.08 in.Hg

Full Throttle Performance Test

Run No.	Approx. Rpm	Counter Reading	Brake Load lb	Fuel cc	Time sec	Pressures				Temperatures				Hygrometer	
						Exh. In.Hg	Intake In.Hg	Oil psi	Throat Drop In.H_2O	Jacket Water, °F In	Jacket Water, °F Out	Oil Out	Exh. °C	Wet Bulb °F	Dry Bulb °F
1	1000	1562.5	75.5	300	96.7	0.30	0.18	32	0.036	79	159	98	360	61	69
2	1500	1537.5	79.0	300	93.2	0.90	0.44	48	0.054	79	165	98	442	61	70
3	2000	1635.0	81.5	300	49.4	1.20	0.72	50	0.130	79	170	106	515	61	70
4	2500	1697.5	81.5	300	41.2	2.30	1.14	52	0.190	79	169	100	590	61	70
5	3000	1815.0	76.5	300	36.5	3.00	1.50	52	0.225	79	167	105	625	61	70
6	3200	1915.0	74.7	300	36.2	3.40	1.70	52	0.235	79	164	100	645	62	72
7	3400	1925.0	72.5	300	34.6	3.70	1.85	52	0.255	79	164	101	655	62	72
8	3600	1857.5	69.7	300	32.6	4.15	2.00	52	0.275	79	168	100	662	62	72
9	3800	1982.0	65.4	300	33.4	4.40	2.15	52	0.288	79	168	100	670	62	72
10	4000	2137.5	62.5	300	32.1	4.75	2.20	52	0.307	79	166	100	680	62	72
11	4200	2165.0	58.0	300	31.4	5.00	2.35	52	0.317	79	166	106	680	62	72
12	4400	2240.0	53.3	300	30.8	5.15	2.40	52	0.328	79	166	110	690	62	72

Acceleration Test

Dynamometer Load, lb	Initial Speed Rpm	Final Speed Rpm	Time sec
10	1000	3000	11.6

Friction Test

Run No.	Approx. Rpm	Counter Reading	Brake Load, lb	Time sec
1	1500	740	16.5	30
2	1750	840	19.0	30
3	2000	1010	21.9	30
4	2500	1260	25.5	30
5	3000	1500	27.7	30
6	3500	1760	31.5	30
7	4000	2010	34.6	30
8	4200	2200	37.7	30

TABLE 6

DATA FOR TEST 6

Engine: 100 Hp, 1949 Ford V-8
Engine Equipment:
Iskenderian 3/4-Race Camshaft
Dual Intake Manifold
Air Tank Connected to Carburetors
Timing and Point Setting:
Manufacturers Specification

Date: August 15, 1954
Observers: R. F. Stebar
B. B. Lindamood
Gasoline: Esso Regular
(80.9 to 82 Octane)
Average Ambient Air Temperature: 79°F
Barometric Pressure: 28.16 in.Hg
Throttle Setting: Full Throttle

Full Throttle Performance Test

Run No.	Approx. Rpm	Counter Reading	Brake Load lb	Fuel cc	Time sec	Pressures				Temperatures				Hygrometer	
						Exh. In.Hg	Intake In.Hg	Oil psi	Throat Drop In.H_2O	Jacket Water, °F In	Jacket Water, °F Out	Oil Out	Exh. °C	Wet Bulb °F	Dry Bulb °F
1	1000	1440	73.0	300	92.1	0.33	0.08	33	0.012	79	165	98	320	70	75
2	1500	1425	76.3	300	56.4	0.85	0.10	48	0.038	79	170	99	420	70	75
3	2000	1372	77.5	300	40.7	1.55	0.40	51	0.066	79	165	98	480	70	76
4	2500	1395	78.1	300	34.0	2.25	0.55	52	0.122	79	166	100	540	70	76
5	3000	1445	76.4	300	29.0	3.40	0.80	52	0.200	79	165	104	575	71	76
6	3200	1485	74.9	300	28.1	3.80	0.83	52	0.224	79	165	102	590	71	76
7	3400	1532	72.0	300	27.7	4.15	0.95	52	0.246	79	163	104	605	71	76
8	3600	1505	70.0	300	26.5	4.62	1.00	52	0.278	79	164	100	618	71	77
9	3800	1610	68.5	300	26.0	4.95	1.05	52	0.300	79	166	100	630	71	77
10	4000	1670	64.7	300	25.4	5.35	1.10	52	0.320	79	165	102	645	72	77
11	4200	1550	60.3	300	22.4	5.65	1.25	52	0.339	79	167	104	652	72	77
12	4400	1745	56.4	300	24.5	5.95	1.35	52	0.355	79	168	103	655	72	77

Acceleration Test

Dynamometer Load, lb	Initial Speed Rpm	Final Speed Rpm	Time sec
10	1000	3000	10.95

Friction Test

Run No.	Approx. Rpm	Counter Reading	Brake Load, lb	Time sec
1	1500	720	19.3	30
2	1800	872	20.6	30
3	2000	1012	22.1	30
4	2500	1325	26.1	30
5	3000	1492	28.0	30
6	3250	1622	29.2	30
7	3500	1725	31.0	30
8	4000	1955	34.0	30
9	4200	2090	35.8	30

TABLE 7

RESULTS FOR TABLE 1

Run No.	Rpm	Torque lb/ft	Bhp	Fhp	Bmep psi	Fuel lb/hr	Bsfc	Correction Factor	Corrected			Ihp	Brake Thermal Eff.	Mech. Eff.
									Bhp	Torque	Bmep			
1	1000	139.0	26.4	4.8	87.1	20.9	0.795	1.112	29.4	154.8	97.0	34.2	16.0	86.0
2	1470	147.0	41.1	8.5	92.4	30.2	0.735	1.115	45.9	164.0	102.8	54.4	17.3	84.4
3	2000	148.0	56.4	14.4	93.0	38.6	0.685	1.115	62.8	165.0	103.6	77.2	18.6	81.4
4	2500	142.0	67.5	20.2	89.1	45.0	0.666	1.118	75.5	158.5	99.6	95.7	19.1	78.9
5	3020	130.5	75.0	26.7	82.0	50.3	0.670	1.118	83.8	145.8	91.6	110.5	19.0	75.8
6	3215	126.0	77.1	29.2	79.1	52.1	0.675	1.120	86.4	141.0	88.5	115.6	18.8	74.6
7	3370	122.5	78.6	31.4	77.0	52.9	0.673	1.120	88.0	137.1	86.2	119.4	18.9	73.7
8	3590	117.0	80.1	34.6	73.6	54.4	0.680	1.120	89.6	131.0	82.4	124.2	18.7	72.1
9	3780	109.0	78.6	37.6	68.6	55.2	0.702	1.120	88.1	122.0	76.8	125.7	18.1	70.1
10	4000	105.0	80.0	41.5	66.0	56.3	0.705	1.120	89.5	117.6	73.9	131.0	18.0	68.4

TABLE 8

RESULTS FOR TABLE 2

Run No.	Rpm	Torque lb-ft	Bhp	Fhp	Bmep psi	Fuel lb/hr	Bsfc	Correction Factor	Corrected Bhp	Corrected Torque	Corrected Bmep	Ihp	Brake Thermal Eff.	Mech Eff.	Air Flow lb/sec	Vol. Eff.	A/F
1	955	141.0	25.6	5.2	88.5	19.8	0.775	1.116	28.6	157.1	98.7	33.8	16.4	84.5	0.0368	48.4	6.70
2	1500	139.3	39.8	9.5	87.5	31.4	0.789	1.116	44.4	155.2	97.5	53.9	16.15	82.4	0.0684	57.3	7.84
3	2000	142.0	54.0	15.1	89.1	38.2	0.707	1.119	60.4	158.9	99.6	75.5	18.0	80.0	0.0757	48.6	7.14
4	2400	137.0	62.6	20.2	86.0	41.2	0.659	1.119	70.0	153.2	96.1	90.2	19.3	77.6	0.0883	46.3	7.71
5	3010	124.5	71.3	28.5	78.2	49.5	0.695	1.118	79.6	139.0	87.3	108.1	18.3	73.6	0.0980	41.1	7.12
6	3195	117.8	71.8	31.1	74.2	49.7	0.693	1.118	80.2	131.6	82.8	111.3	18.4	72.1	0.1006	39.6	7.30
7	3375	110.0	70.5	35.3	69.0	51.0	0.724	1.118	78.7	122.9	77.0	114.0	17.6	69.0	0.1063	39.7	7.31
8	3600	103.1	70.6	37.3	64.8	53.0	0.750	1.120	79.0	115.5	72.6	116.3	17.0	67.9	0.1094	38.3	7.45
9	3810	96.0	69.5	40.6	60.3	54.0	0.776	1.120	77.8	107.6	67.6	118.4	16.4	65.6	0.1132	37.4	7.56
10	4060	83.6	64.4	46.4	52.2	55.3	0.858	1.119	72.0	93.5	58.4	118.4	14.8	61.8	0.1160	36.0	7.56

TABLE 9

RESULTS FOR TABLE 3

Run No.	Rpm	Torque lb-ft	Bhp	Fhp	Bmep psi	Fuel lb/hr	Bsfc lb/Bhp-hr	Correction Factor	Corrected Bhp	Corrected Torque	Corrected Bmep	Ihp	Brake Thermal Eff.	Mech. Eff.
1	976	139.0	25.9	4.8	87.6	18.6	0.718	1.093	28.4	152.1	96.0	33.2	17.7	88.1
2	1482	142.5	40.4	9.6	90.0	30.6	0.758	1.093	44.3	156.0	98.5	53.9	16.7	82.2
3	1985	145.0	54.0	14.8	90.0	41.9	0.776	1.092	59.1	159.0	98.5	73.9	16.4	80.0
4	2490	144.6	68.5	20.8	90.9	50.8	0.742	1.093	75.0	158.1	99.5	95.8	17.1	78.3
5	2920	137.8	76.6	25.6	86.7	57.1	0.745	1.093	83.8	151.0	95.0	108.9	17.1	77.0
6	3200	133.2	81.3	31.0	84.0	59.0	0.725	1.090	88.6	145.1	91.5	119.6	17.5	74.1
7	3370	129.2	83.0	33.7	81.3	59.9	0.722	1.090	90.5	140.8	88.5	124.2	17.6	72.9
8	3570	125.4	85.4	37.3	79.0	61.6	0.722	1.090	93.0	136.8	86.0	130.3	17.6	71.4
9	3790	118.4	85.5	41.2	74.5	63.7	0.745	1.090	93.1	129.0	81.1	134.3	17.1	69.3
10	4000	112.3	85.6	45.3	70.8	65.6	0.766	1.090	93.4	122.5	77.1	138.7	16.6	67.5
11	4150	104.5	82.6	48.0	65.8	67.5	0.816	1.090	90.0	114.0	71.6	138.0	15.6	65.2

TABLE 10

RESULTS FOR TABLE 4

Run No.	Rpm	Torque lb-ft	Bhp	Fhp	Bmep psi	Fuel lb/hr	Bsfc lb/Bhp-hr	Correction Factor	Corrected Bhp	Corrected Torque	Corrected Bmep	Ihp	Brake Thermal Eff.	Mech. Eff.	Air Flow lb/sec	Vol. Eff.	A/F
1	944	138.5	24.9	4.8	87.1	20.2	0.812	1.111	27.7	154.0	97.0	32.5	15.6	85.8	0.0405	54.1	7.22
2	1480	141.6	39.9	9.1	89.0	31.9	0.800	1.111	44.4	157.6	99.0	53.5	15.9	83.0	0.0663	56.4	7.48
3	2010	141.6	54.2	12.2	89.0	44.4	0.820	1.111	60.3	157.6	99.0	72.5	15.5	83.1	0.0965	60.4	7.83
4	2470	140.7	66.1	19.3	88.3	53.6	0.811	1.111	73.5	156.6	98.2	92.8	15.6	79.1	0.1025	52.4	6.89
5	3020	133.5	76.8	26.9	83.9	61.2	0.796	1.111	85.4	148.5	93.4	112.3	15.9	75.9	0.1250	52.1	7.35
6	3180	130.0	78.6	31.6	81.6	63.7	0.810	1.111	87.5	144.7	90.8	119.1	15.7	73.5	0.1317	52.1	7.45
7	3370	121.0	77.6	32.9	76.0	65.1	0.838	1.111	86.4	134.7	84.5	119.3	15.1	72.4	0.1365	51.0	7.54
8	3510	115.0	76.8	35.8	72.2	67.6	0.880	1.111	85.5	128.0	80.4	121.3	14.4	70.5	0.1440	51.6	7.69
9	3810	109.0	79.0	40.4	68.5	69.9	0.885	1.111	89.0	121.2	76.2	129.4	14.3	68.9	0.1480	48.9	7.64
10	3960	102.5	77.1	46.9	64.4	70.6	0.915	1.111	85.6	114.0	71.6	132.5	13.9	64.6	0.1499	47.6	7.65
11	4190	89.3	71.2	53.8	56.2	72.5	1.020	1.111	79.3	99.4	62.5	133.1	12.4	59.5	0.1522	45.8	7.58

TABLE 11

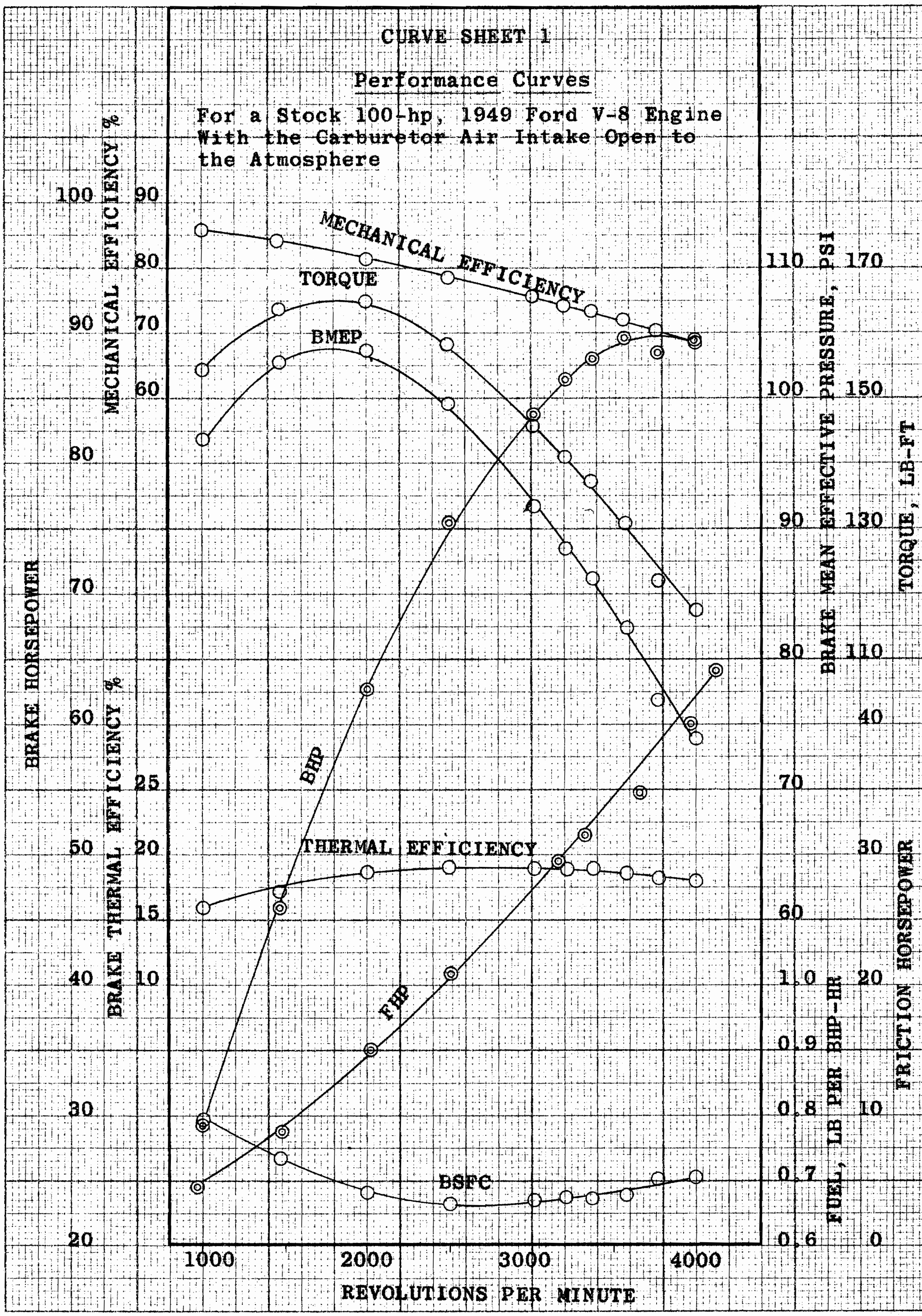
RESULTS FOR TABLE 5

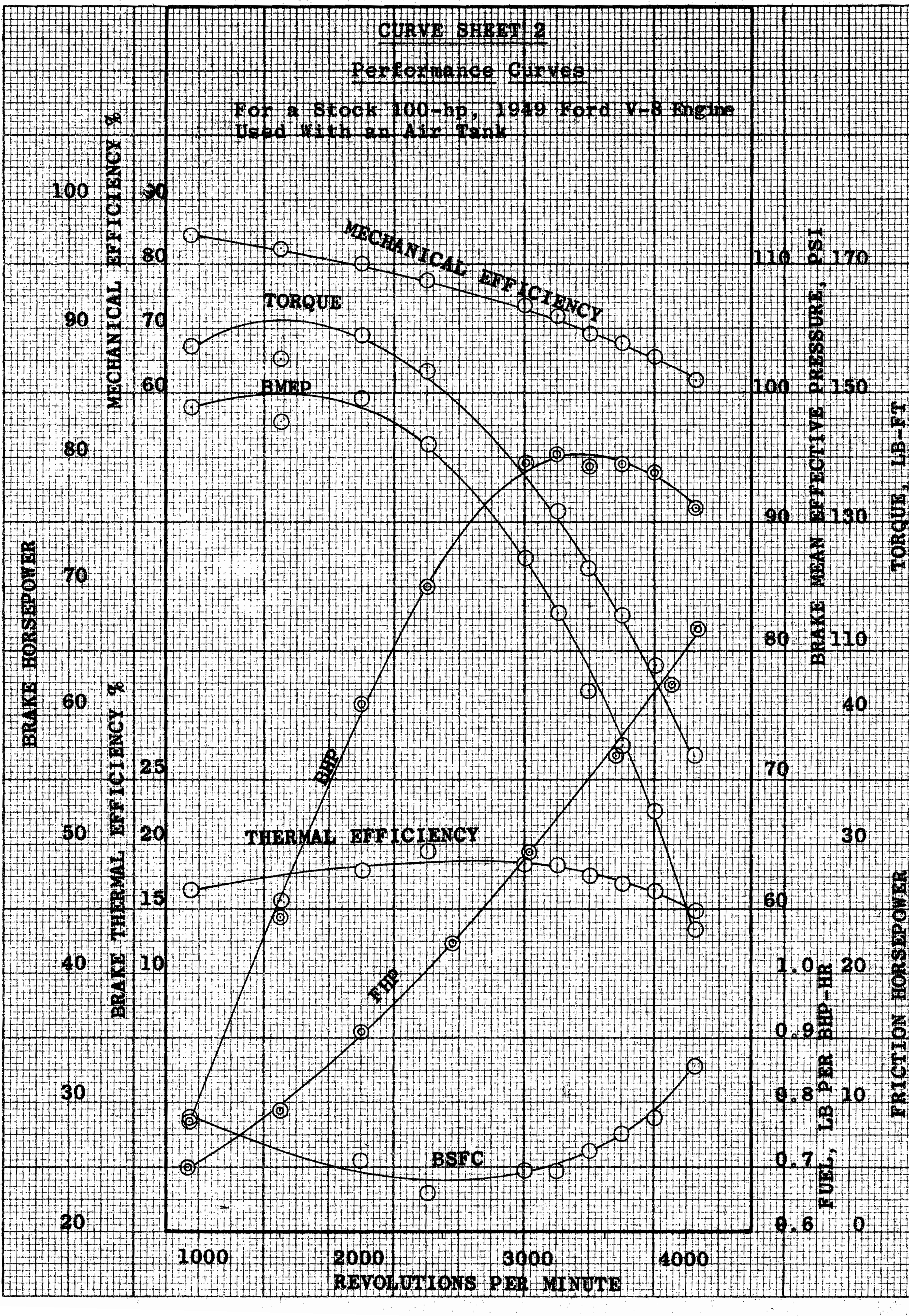
Run No.	Rpm	Torque lb-ft	Bhp	Fhp	Bmep psi	Fuel lb-hr	Bsfc lb/Bhp-hr	Correction Factor	Corrected Bhp	Corrected Torque	Corrected Bmep	Ihp	Brake Thermal Eff.	Mech. Eff.	Air Flow lb/sec	Vol. Eff.	A/F
1	970	132.0	24.2	4.8	82.4	18.8	0.776	1.091	26.4	144.0	89.9	31.2	16.4	84.6	0.0445	57.1	8.54
2	1460	138.2	38.5	8.6	87.0	28.8	0.748	1.092	42.1	151.1	95.1	50.7	17.0	83.0	0.0545	46.6	6.80
3	1985	142.7	54.0	14.3	89.8	36.9	0.684	1.092	59.0	156.0	98.2	73.3	18.6	80.4	0.0845	53.2	8.25
4	2470	142.7	67.0	20.6	89.5	44.1	0.659	1.092	73.2	156.0	97.9	93.8	19.3	78.1	0.1037	52.2	8.44
5	2985	134.0	76.2	27.9	84.2	49.8	0.654	1.092	83.3	146.5	92.0	111.2	19.5	75.0	0.1110	46.5	8.04
6	3170	130.8	79.0	30.6	82.3	50.3	0.636	1.097	86.6	143.2	90.2	117.2	20.0	73.9	0.1135	44.7	8.15
7	3340	127.0	80.6	33.4	79.6	52.6	0.654	1.097	88.4	139.1	87.3	121.8	19.5	72.8	0.1184	44.2	8.10
8	3420	122.1	79.0	34.9	76.1	55.8	0.706	1.097	86.5	134.0	83.5	121.4	18.0	71.2	0.1230	44.9	7.95
9	3560	114.5	77.6	37.5	72.0	54.5	0.702	1.097	85.0	125.5	78.9	122.5	18.1	69.4	0.1258	43.9	8.33
10	3990	109.3	83.0	46.0	68.7	56.7	0.684	1.097	91.0	119.8	75.4	137.0	18.6	66.5	0.1300	40.7	8.26
11	4140	101.6	80.0	49.1	64.0	58.0	0.725	1.097	87.6	111.2	70.2	136.7	17.5	64.0	0.1320	39.8	8.20
12	4360	94.3	77.6	54.1	58.6	59.1	0.763	1.097	85.0	103.4	64.3	139.1	16.7	61.1	0.1342	38.4	8.23

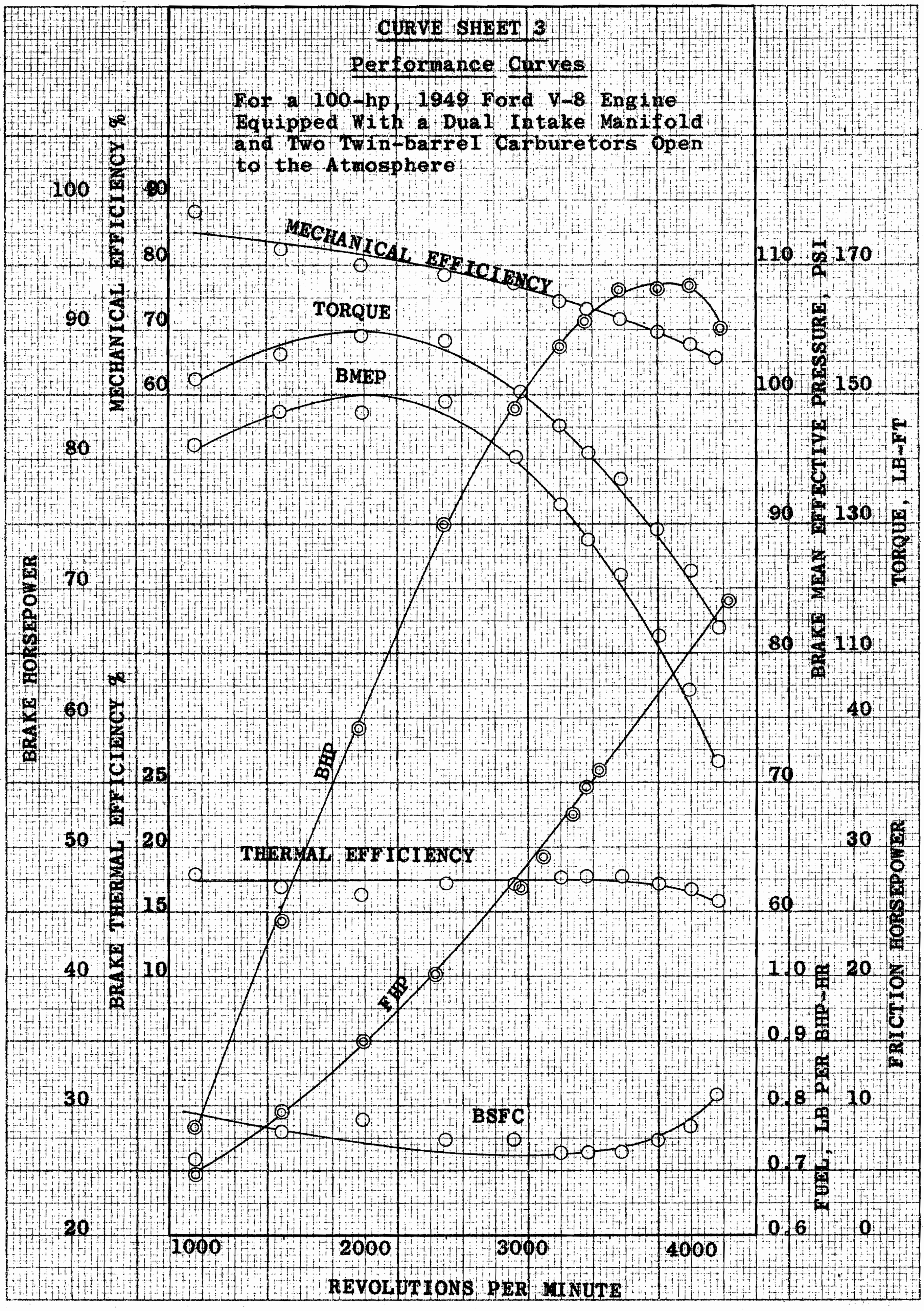
TABLE 12

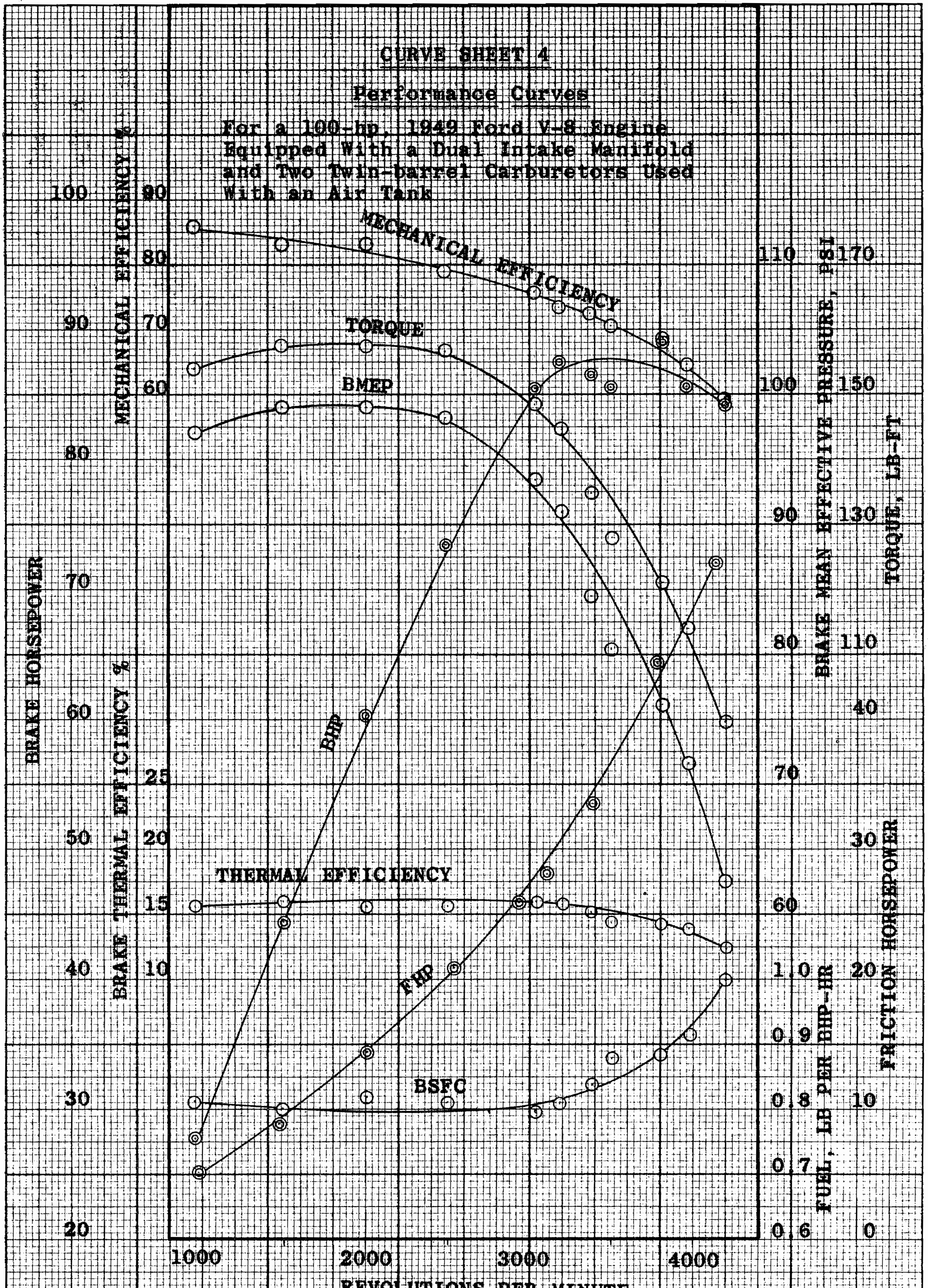
RESULTS FOR TABLE 6

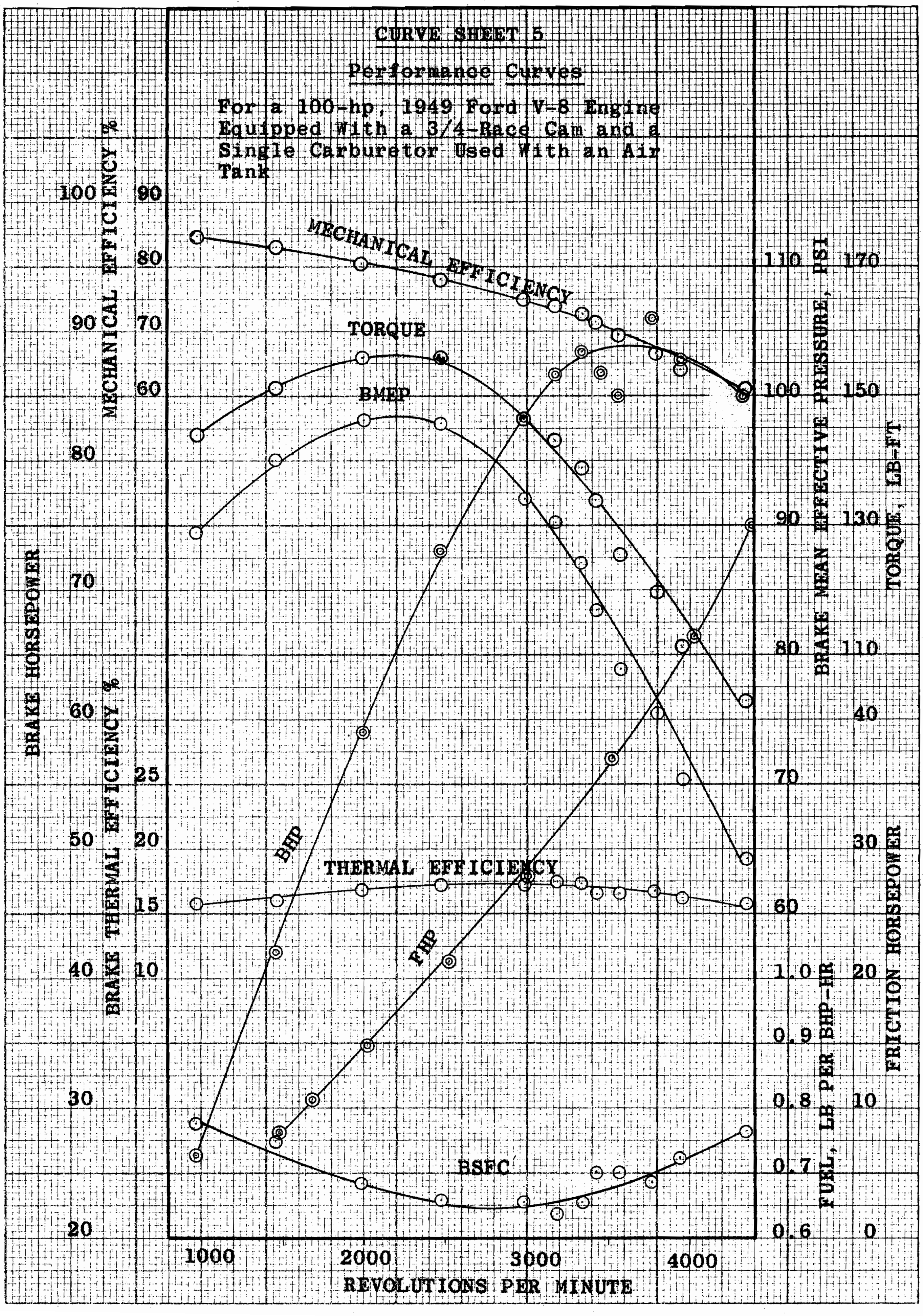
Run No.	Rpm	Torque lb-ft	Bhp	Fhp	Bmep psi	Fuel lb-hr	Bsfc lb/Bhp-hr	Correction Factor	Corrected			Ihp	Brake Thermal Eff.	Mech. Eff.	Air Flow lb/sec	Vol. Eff.	A/F
									Bhp	Torque	Bmep						
1	938	128.0	22.8	5.1	80.2	19.8	0.869	1.106	25.2	141.3	89.6	30.3	14.6	83.0	0.0256	34.4	4.65
2	1520	133.7	38.6	9.9	83.9	32.4	0.840	1.106	42.6	147.7	92.6	52.5	15.1	81.2	0.0456	37.7	5.06
3	2020	135.9	52.2	14.9	85.4	44.7	0.856	1.106	57.7	150.0	94.4	72.6	14.8	79.4	0.0601	37.5	4.85
4	2460	137.0	64.0	20.5	85.7	53.6	0.839	1.106	70.7	151.4	94.7	91.2	15.2	77.5	0.0818	41.8	5.50
5	2990	133.8	76.0	27.8	83.7	62.8	0.826	1.108	84.2	148.0	92.7	112.0	15.4	75.1	0.1047	44.1	6.00
6	3170	131.2	79.1	30.5	82.4	64.9	0.820	1.108	87.6	145.2	91.3	118.1	15.5	74.1	0.1118	44.5	6.20
7	3320	126.0	79.6	33.0	79.2	65.7	0.825	1.108	88.2	139.6	87.7	121.2	15.4	72.6	0.1160	44.0	6.37
8	3410	122.7	79.5	34.5	77.0	68.8	0.866	1.108	88.0	135.8	85.2	122.5	14.7	71.8	0.1235	45.6	6.46
9	3720	120.0	85.0	40.3	75.4	70.1	0.825	1.108	94.1	133.0	83.5	134.4	15.4	70.0	0.1280	43.2	6.60
10	3940	113.4	85.0	44.9	71.3	71.8	0.845	1.109	94.2	125.8	79.0	139.1	15.0	67.6	0.1324	42.2	6.65
11	4150	105.7	83.5	49.3	66.4	81.4	0.875	1.109	92.6	117.1	73.6	141.9	13.0	65.3	0.1362	41.4	6.06
12	4275	98.8	80.4	52.2	62.0	74.4	0.926	1.109	89.1	109.6	68.8	141.3	13.7	63.0	0.1395	41.1	6.76

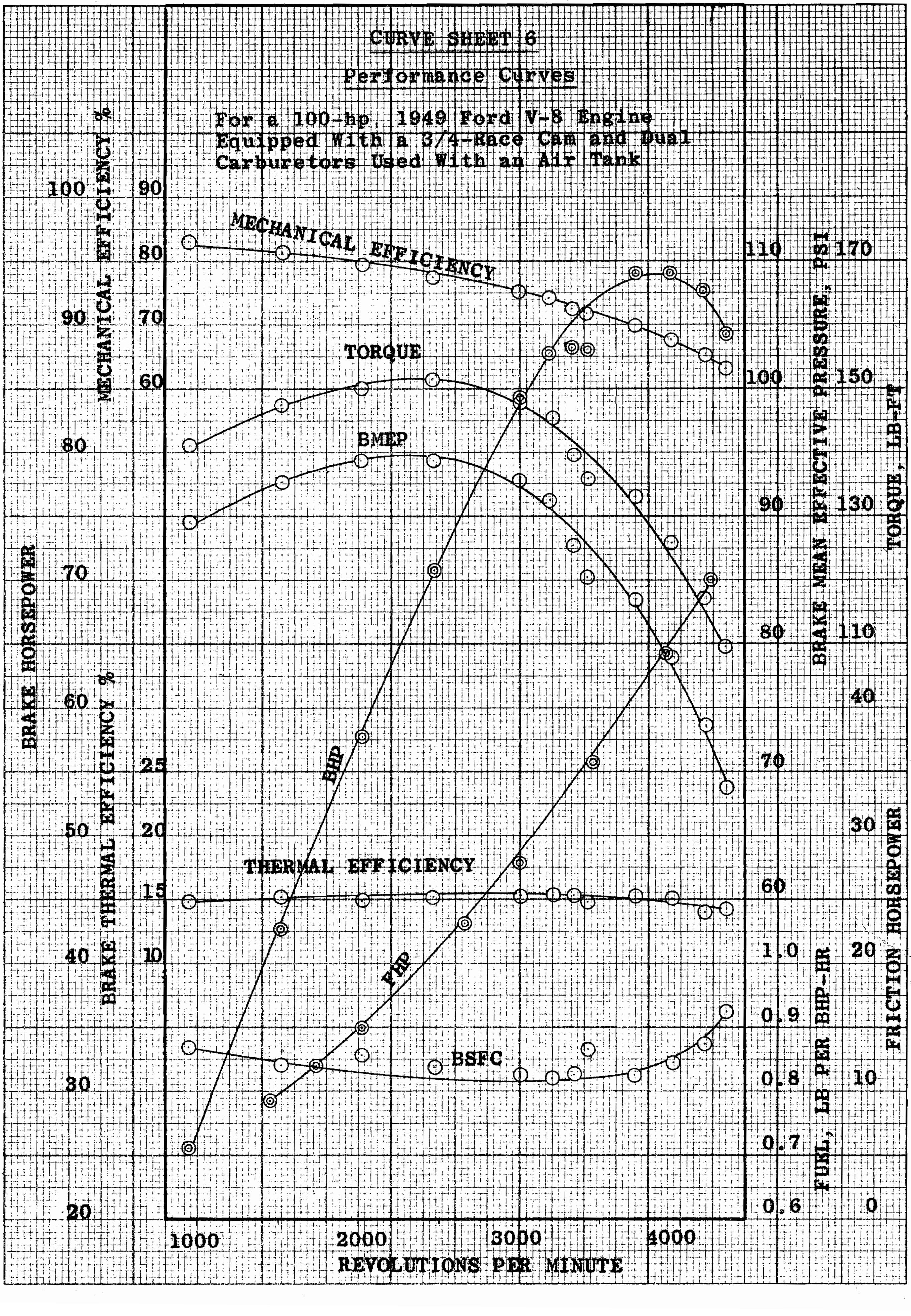

CURVE SHEET 1
Performance Curves
For a Stock 100-hp, 1949 Ford V-8 Engine With the Carburetor Air Intake Open to the Atmosphere
MECHANICAL EFFICIENCY
TORQUE
BMEP
BHP
THERMAL EFFICIENCY
FHP
BSFC
BRAKE HORSEPOWER
MECHANICAL EFFICIENCY %
BRAKE THERMAL EFFICIENCY %
BRAKE MEAN EFFECTIVE PRESSURE, PSI
TORQUE, LB-FT
FUEL, LB PER BHP-HR
FRICTION HORSEPOWER
REVOLUTIONS PER MINUTE
1000
2000
3000
4000
100
90
80
70
60
50
40
30
20
90
80
70
60
25
20
15
10
110
100
90
80
70
60
1.0
0.9
0.8
0.7
0.6
170
150
130
110
40
30
20
10
0

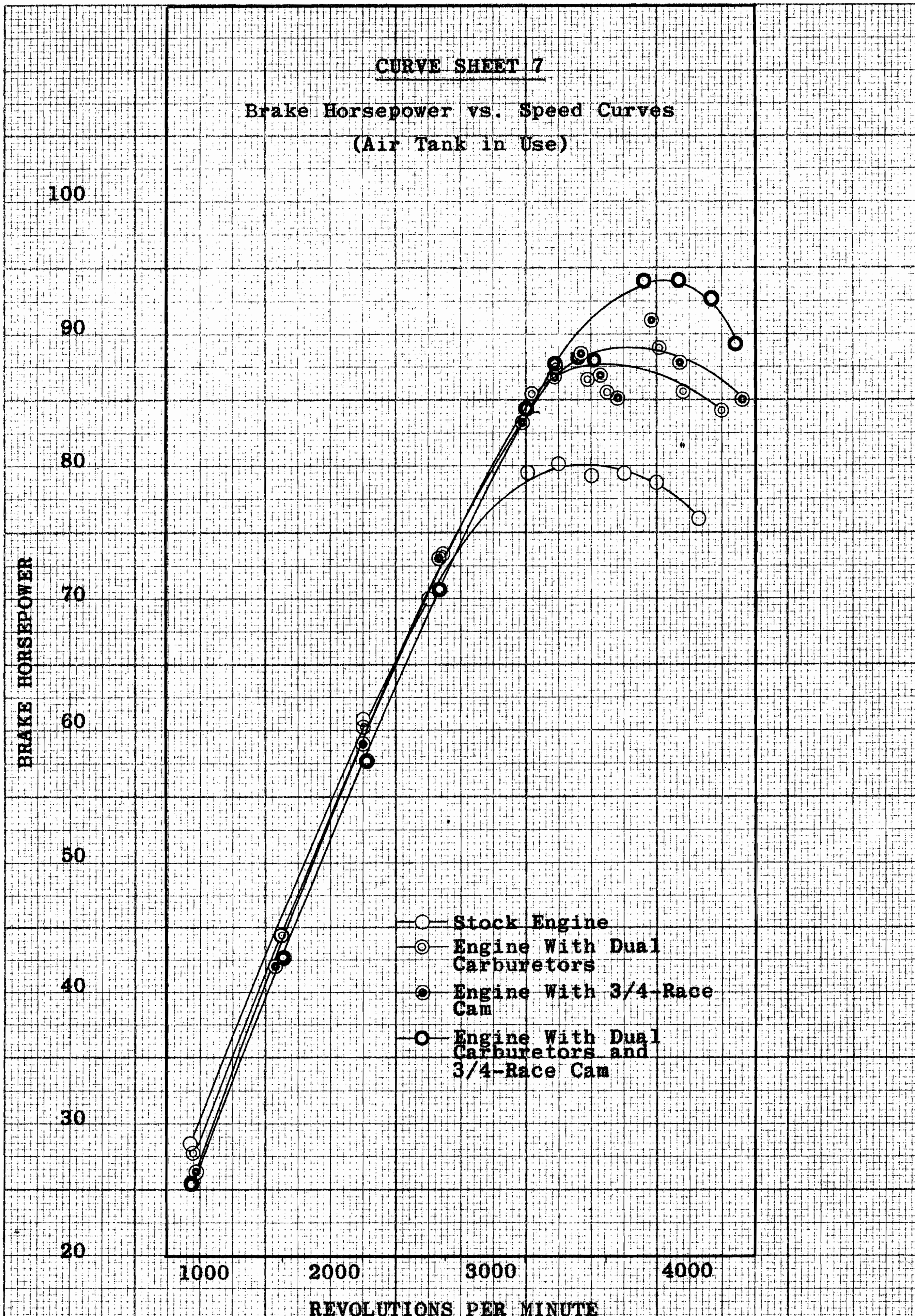

CURVE SHEET 2
Performance Curves
For a Stock 100-hp, 1949 Ford V-8 Engine
Used With an Air Tank
MECHANICAL EFFICIENCY
TORQUE
BMEP
BHP
THERMAL EFFICIENCY
FHP
BSFC
BRAKE HORSEPOWER
MECHANICAL EFFICIENCY %
BRAKE THERMAL EFFICIENCY %
BRAKE MEAN EFFECTIVE PRESSURE, PSI
TORQUE, LB-FT
FUEL, LB PER BHP-HR
FRICTION HORSEPOWER
REVOLUTIONS PER MINUTE
1000
2000
3000
4000
100
90
80
70
60
50
40
30
20
25
15
10
110
170
150
130
1.0
0.9
0.8
0.7
0.6
0

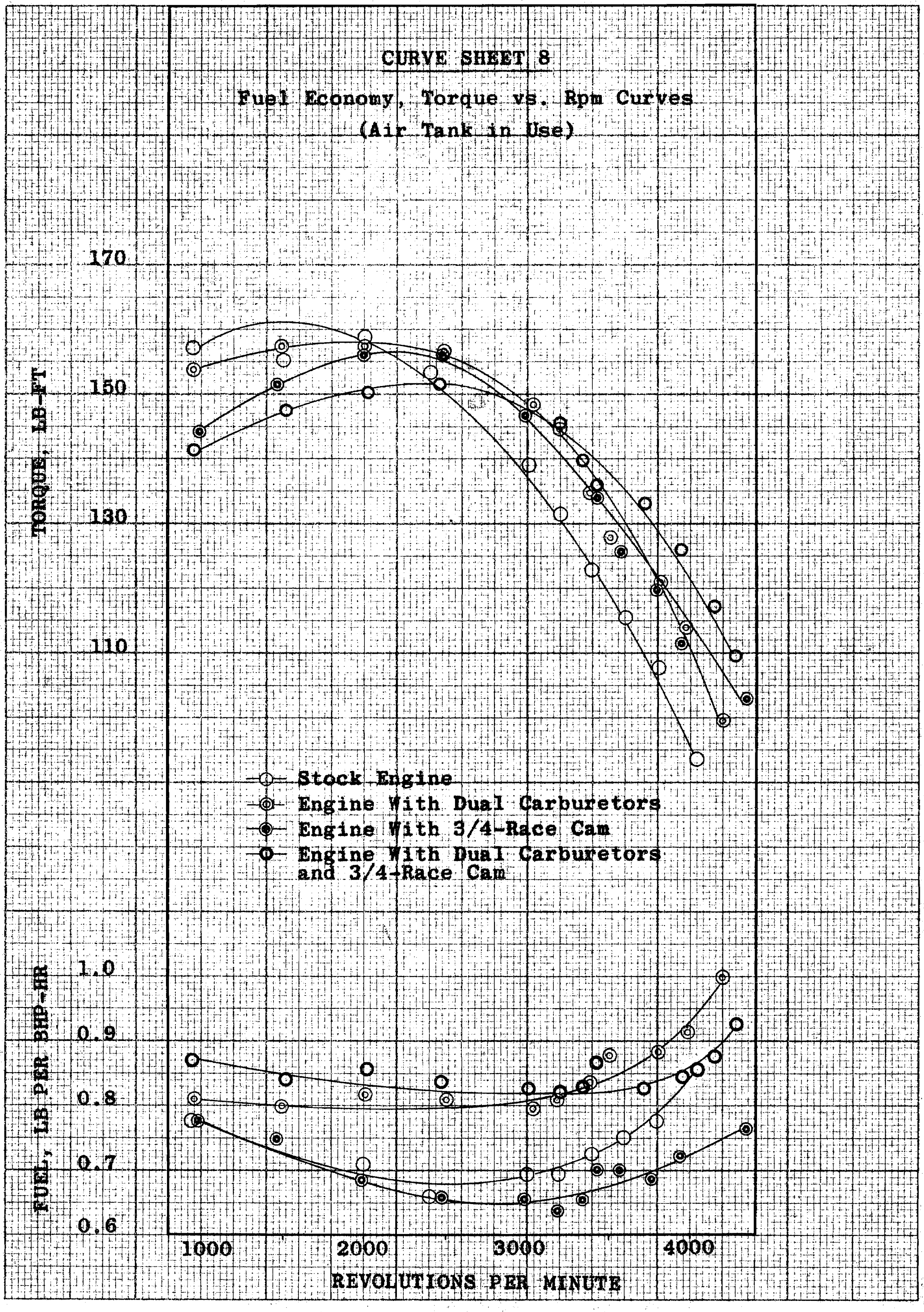

CURVE SHEET 3
Performance Curves
For a 100-hp, 1949 Ford V-8 Engine Equipped With a Dual Intake Manifold and Two Twin-barrel Carburetors Open to the Atmosphere
MECHANICAL EFFICIENCY
TORQUE
BMEP
BHP
THERMAL EFFICIENCY
FHP
BSFC
BRAKE HORSEPOWER
MECHANICAL EFFICIENCY %
BRAKE THERMAL EFFICIENCY %
BRAKE MEAN EFFECTIVE PRESSURE, PSI
TORQUE, LB-FT
FUEL, LB PER BHP-HR
FRICTION HORSEPOWER
REVOLUTIONS PER MINUTE
1000
2000
3000
4000
100
90
80
70
60
50
40
30
20
80
70
60
25
20
15
10
110
100
90
80
70
60
170
150
130
110
40
30
20
10
0
1.0
0.9
0.8
0.7
0.6


CURVE SHEET 4
Performance Curves
For a 100-hp, 1949 Ford V-8 Engine Equipped With a Dual Intake Manifold and Two Twin-barrel Carburetors Used With an Air Tank
MECHANICAL EFFICIENCY
TORQUE
BMEP
BHP
THERMAL EFFICIENCY
FHP
BSFC
BRAKE HORSEPOWER
MECHANICAL EFFICIENCY %
BRAKE THERMAL EFFICIENCY %
BRAKE MEAN EFFECTIVE PRESSURE, PSI
TORQUE, LB-FT
FUEL, LB PER BHP-HR
FRICTION HORSEPOWER
REVOLUTIONS PER MINUTE
1000
2000
3000
4000


CURVE SHEET 5
Performance Curves
For a 100-hp, 1949 Ford V-8 Engine Equipped With a 3/4-Race Cam and a Single Carburetor Used With an Air Tank
MECHANICAL EFFICIENCY
TORQUE
BMEP
BHP
THERMAL EFFICIENCY
FHP
BSFC
MECHANICAL EFFICIENCY %
BRAKE HORSEPOWER
BRAKE THERMAL EFFICIENCY %
BRAKE MEAN EFFECTIVE PRESSURE, PSI
TORQUE, LB-FT
FRICTION HORSEPOWER
FUEL, LB PER BHP-HR
REVOLUTIONS PER MINUTE
1000
2000
3000
4000


CURVE SHEET 6
Performance Curves
For a 100-hp 1949 Ford V-8 Engine Equipped With a 3/4-Race Cam and Dual Carburetors Used With an Air Tank
MECHANICAL EFFICIENCY
TORQUE
BMEP
BHP
THERMAL EFFICIENCY
FHP
BSFC
BRAKE HORSEPOWER
MECHANICAL EFFICIENCY %
BRAKE THERMAL EFFICIENCY %
BRAKE MEAN EFFECTIVE PRESSURE, PSI
TORQUE, LB-FT
FUEL, LB PER BHP-HR
FRICTION HORSEPOWER
REVOLUTIONS PER MINUTE
1000
2000
3000
4000
100
90
80
70
60
50
40
30
20
25
15
10
110
170
150
130
0.9
0.8
0.7
0.6
1.0
0


CURVE SHEET 7
Brake Horsepower vs. Speed Curves
(Air Tank in Use)
BRAKE HORSEPOWER
100
90
80
70
60
50
40
30
20
1000
2000
3000
4000
REVOLUTIONS PER MINUTE
Stock Engine
Engine With Dual Carburetors
Engine With 3/4-Race Cam
Engine With Dual Carburetors and 3/4-Race Cam


CURVE SHEET 8
Fuel Economy, Torque vs. Rpm Curves
(Air Tank in Use)
TORQUE, LB-FT
170
150
130
110
Stock Engine
Engine With Dual Carburetors
Engine With 3/4-Race Cam
Engine With Dual Carburetors and 3/4-Race Cam
FUEL, LB PER BHP-HR
1.0
0.9
0.8
0.7
0.6
1000
2000
3000
4000
REVOLUTIONS PER MINUTE

XI

VITA

The author was born in Newport, Virginia, September 27, 1932. After he had attended the Newport Elementary School for one year, his family moved to Christiansburg, Virginia, where he continued his grade school education at the Christiansburg Primary and Grammar Schools. In 1945, he entered Christiansburg High School where he later graduated in the spring of 1949.

In the fall of 1949, he was admitted to the Virginia Polytechnic Institute to pursue the study of Mechanical Engineering. As an undergraduate he was a member of Tau Beta Pi, Pi Tau Sigma, and the Student Branch of A. S. M. E. On June 7, 1953, the author was graduated with a Bachelor of Science Degree in Mechanical Engineering.

While in High School and during his study at the Virginia Polytechnic Institute he was employed on a part-time status by Mick or Mack Store, Christiansburg, Virginia. Following graduation from V. P. I. he was employed by the Radford Arsenal Branch of Hayes, Seay, Mattern & Mattern, Roanoke, Virginia.

In the fall of 1954, he enrolled as a graduate student at the Virginia Polytechnic Institute where he pursued the course of study leading to the degree of Master of Science in Mechanical Engineering.

Russell Ford Stetson