The effect of inlet valve throttling on the exhaust hydrocarbon emission of a spark ignition engine

by Raymond Robert Reid

A thesis submitted to the Graduate Faculty in partial fulfillment of the requirements for the degree of MASTER OF SCIENCE in Aerospace and Mechanical Engineering
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Abstract:
An experimental investigation was carried out to determine the effects of inlet-valve throttling on internal combustion engine exhaust gas hydrocarbon concentration as compared to conventional carburetor-valve throttling. The data were obtained from a single cylinder engine of 7.09 cubic-inch piston displacement which was specially equipped to allow operation with either inlet-valve or carburetor-valve throttling.

The concentration of unburned hydrocarbons was measured using an infrared absorption instrument of the nondispersive type which was developed. and constructed during this investigation. Details of the construction and calibration of this instrument are included.

Exhaust gas hydrocarbon concentrations were measured at engine speeds of 1500, 2000, 2500, and 3000 revolutions per minute (rpm), and at three different engine loads at each speed, for both methods of throttling. The same ignition timing, inlet-valve timing, and exhaust-valve timing, along with best-torque fuel air ratios, were utilized for all the above measurements. Additional measurements were made at an engine speed of 2000 rpm, and at four different engine loads, with the inlet-valve timing adjusted to two different specifications than those used in the above tests. All other variables remained the same.

With few exceptions, the data obtained show that even though inlet-valve throttling results in leaner best-torque air-fuel ratios than is the case with conventional carburetor throttling, the concentration of unburned hydrocarbon in the exhaust gas is higher with inlet-valve throttling. The increase in exhaust-gas hydrocarbon concentration' with inlet-valve throttling as compared to conventional throttling is attributed to the increased charge turbulence inherent with inlet-valve throttling which produces greater heat transfer to the combustion-chamber walls. This greater heat transfer results in a larger quench volume and, as a result, increased concentrations of unburned hydrocarbon in the exhaust gas.

In general, within the limits of this investigation, inlet-valve throttling results in a higher concentration of unburned hydrocarbon in the exhaust gas than does conventional carburetor-valve throttling.

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ABSTRACT

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I. INTRODUCTION

Early in the 1950's, the exhaust gas from spark-ignition internal combustion engines was identified as a major contributor to the air pollution problems being encountered in the highly populated areas of the United States (1).* The extensive use of the automobile for private transportation in metropolitan areas was increasing the atmospheric concentrations of such pollutants as carbon monoxide, oxides of nitrogen, and hydrocarbons at an alarming rate. Investigations during this period also disclosed that two of these pollutants, namely oxides of nitrogen and hydrocarbons, combined under the catalysis of sunlight to produce the photochemical smog which was plaguing Los Angeles as well as other areas of the country (1).

Since this time many investigators have studied the effects of engine operation and design variables on the concentration of atmospheric pollutants in automotive-engine exhaust. With the restrictions placed on automotive exhaust gas pollutant levels, first by the California Motor Vehicle Pollution Control Board effective for the 1966 production year and later by the federal government Clean Air Act effective for the 1968 production year, and the promise of more stringent restrictions to come (2), those variables found to reduce pollution levels in exhaust gas, without significantly reducing the engine performance characteristics, have been incorporated into the design of production automobile engines. These variables, however, have served to reduce the level of pollutants

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*Numbers in parentheses designate references in List of References.

to just under the present requirements. With increasing pressure being placed on the automotive industry, by both state and federal governments, to reduce the pollutant levels even further, new methods are being sought to accomplish this task.

Until recently it was common design practice to induce turbulence in the combustion chamber before ignition by squish (by the piston and cylinder head forming an area of thin cross section when the piston is at top dead center). Engine-design engineers are now working toward elimination of the squish area, however, since investigations by Daniel and Wentworth (3), and by Huls, Myers, and Uyehara (4) have shown that flame quenching (reduction of the reaction temperature below the self-sustaining limit) in this area results in increased exhaust hydrocarbon emissions.

The advantages of a turbulent charge in the combustion chamber of a spark-ignition engine at the time of ignition had been recognized as early as 1923 when Ricardo (5) patented his "turbulent chamber" for L-head engines. Two of these advantages are increased thermal efficiency and increases in lean air-fuel ratio tolerance. These advantages result from the higher flame speed obtainable with a turbulent charge.

In order to retain the advantages of a turbulent charge without the squish type combustion chamber, turbulence must be induced in the charge during the inlet process. Two methods which have been used to accomplish this result are reduction of inlet port cross sectional area and shrouding of the inlet valve. These methods result in increased pressure drop, velocity, and Reynolds number through the port or valve with resulting charge turbulence in the cylinder which persists through the compression process producing the same desirable results as squish-induced turbulence. These methods, however, are not suitable for automotive engines since the port dimensions or valve shrouding necessary for adequate turbulence at light loads and low speeds reduces the volumetric efficiency, and thus the power, to an unacceptable value at higher loads and speeds.

A third method of producing inlet-induced turbulence is inlet-valve throttling. In this method the required throttling pressure drop, from atmospheric to cylinder pressure, is provided at the inlet valve. Load control is obtained by modulating the maximum lift of the inlet valve. A small flow cross section is thus provided at the inlet valve with resulting high velocities and turbulences at light loads. At wide open throttle (full inlet-valve lift) there is no reduction of flow cross section as compared to conventionally throttled engines. High charge turbulence is therefore provided at light loads without sacrificing high speed performance.

A search of the available literature disclosed that no determination has been reported of the effect of inlet-valve throttling on the unburned hydrocarbon concentrations in spark-ignition engine exhaust gasses. Since inlet-valve throttling appears to be a possible method of eliminating the combustion chamber squish area while still retaining the advantages of a turbulent charge in automotive engines, an investigation was initiated to determine the effect of this mode of operation on exhaust gas hydrocarbon concentrations as compared to conventional carburetor-valve throttling.

II. STATEMENT OF PROBLEM

The object of this investigation is to determine the effects of inlet-valve throttling on the exhaust gas hydrocarbon concentrations of a spark-ignition engine as compared to conventional carburetor-valve throttling.

III. EXPERIMENTAL APPARATUS

A. Test Engine

The basic engine used in this investigation was a single-cylinder Continental model AU-7. This engine has a bore of 2.125 inches and a stroke of 2.000 inches resulting in a piston displacement of 7.09 inches.

Although this engine was originally of L-head design, the original air-cooled cylinder head has been replaced by a water-cooled overhead valve system designed and built by John H. Herber (6) at the Mechanical Engineering Laboratories of the University of Texas. This design incorporates a unique system of dual-overhead camshafts which allows independent variation of either inlet or exhaust valve timing (opening, closing, and duration) over a wide range, while the engine is in operation. A complete description of the variable-valve-timing mechanism will not be given here since its operation is not important to this investigation.

The combustion chamber in this cylinder head is cylindrical in shape with a diameter of 2.250 inches and a depth of 0.300 inches. The original flat-top piston was retained resulting in a combustion chamber which is free of any turbulence inducing mechanisms. The combustion chamber volume within the cylinder head combined with a 0.040 inch thick head gasket results in a compression ratio of 5.25 to 1.

Water passages are provided in the cylinder head to allow at least some degree of control over the cylinder head temperature. The effectiveness of this cooling system, as far as controlling the combustion chamber surface temperature, is probably far from ideal. It is, however, better than no control at all.

A new inlet-valve rocker arm assembly incorporating a movable fulcrum was designed, built, and installed on the engine by the author. This assembly allows modulation of the maximum inlet-valve lift by locating the fulcrum at a point on the rocker arm which provides the desired lift. Inlet-valve lifts of from zero (by placing the fulcrum at the centerline of the inlet valve) through greater than that obtained with the original system are possible. Guides are provided to prevent the rocker arm from rotating in the plane of the cylinder head surface or from being displaced linearly from its proper position with respect to the camshaft or valve. A return spring is provided at the camshaft end of the rocker arm to keep the cam follower in contact with the cam when the rocker arm fulcrum is positioned near the inlet-valve axis. A schematic cross section of the inlet-valve rocker arm assembly is shown in Fig. 1.

Initial attempts at operation of the engine with inlet-valve throttling revealed that the original carburetor (fixed venturi, butterfly throttle valve) would not be satisfactory since at light loads and low speeds the pressure drop through the fixed venturi was not sufficient to provide proper delivery of fuel to the air stream. This carburetor was replaced with a Kie-Hin carburetor (variable venturi area throttling) originally manufactured for a Honda Trail 90 motorcycle. The Kie-Hin carburetor allowed reduction of the venturi area for proper fuel delivery during light-load, low-speed, valve-throttled operation.

An auxiliary battery ignition system was installed on the engine to provide a greater flexibility of ignition timing than could be obtained with the original magneto ignition system. Advance or retard of more than a few degrees from the basic 14 degrees before top dead center magneto timing resulted in a sufficient change in the magneto "E" gap to cause ignition failure and engine misfire. Most tests were run at the basic magneto timing, however, and magneto ignition was used in these instances.

A degree wheel, accurate to within plus or minus 15 seconds of arc, was built and installed on the engine crankshaft to allow accurate determinations of ignition and valve timing. The methods used for these determinations are given in the experimental procedures section.

An iron-constantan thermocouple was installed in the exhaust system 4 inches downstream of the exhaust valve for the purpose of obtaining an indication of the changes in exhaust temperature with changes in operating conditions. An exhaust gas sample tap was installed in the exhaust system 5 inches downstream of the exhaust valve to supply a continuous flow of exhaust gas to the hydrocarbon analyzer. To provide sufficient pressure in the exhaust system to force the exhaust sample through the hydrocarbon analyzer, a valve was installed in the exhaust pipe 10 inches downstream of the exhaust valve. This valve was also used to maintain the exhaust back pressure at approximately the same value for all test runs.

In order to obtain an indication of the cylinder head temperature, an iron-constantan thermocouple was installed under a cylinder-head bolt near the exhaust port.

A fitting was installed in the intake manifold for the connection of a manometer to monitor intake manifold pressure.

B. Exhaust Gas Hydrocarbon Analyzer

The method originally proposed for the determination of hydrocarbon concentration in the exhaust gas was gas chromatography. The rather complicated sampling, preparation, and analysis techniques required for the application of this method, however, as pointed out by Hurn, Hughes, and Chase (7), resulted in the abandonment of this proposal.

The use of one of the commercially available instruments designed specifically for the purpose of determining the hydrocarbon concentration in gaseous mixtures was next considered. This idea had to be abandoned since an instrument of this type, which could be used on a loan basis, could not be located, and sufficient funds were not available for the purchase of a new instrument.

The only remaining alternative was to construct an instrument from components either on hand or available at low cost. The construction of an instrument using an infrared source, a flow-through sample cell, interference filters, and an infrared detector, as described in a paper by Neerman and Millar (8), was begun. Although all equipment necessary for the construction of this instrument was on hand, or readily available, except the interference filters (used to block all wavelengths of infrared radiation except that of 3.3 to 3.5 microns, which is absorbed by the carbon-hydrogen bonds in hydrocarbons), the cost of the required filters was prohibitive.

The description of the operating principles utilized by commercial nondispersive infrared analyzers, as given by Considine (9), suggested the possibility of employing the pressure change induced in a confined hydrocarbon gas by the absorption of infrared radiation as an infrared transmittance monitor. Since a differential-pressure transducer capable of detecting the small pressure changes expected was on hand, development proceeded on this principle. A diagram of the instrument, as finally used in this investigation, is shown in Fig. 2.

The infrared source was a one-inch length of nichrome wire coil which was originally made, in a longer length, for use in an electric space heater. The nichrome wire was electrically heated with power being supplied through a Variac to provide control over the temperature of the source and thus the intensity of infrared radiation emitted. The reflector was constructed from a 1-$\frac{1}{4}$ inch length of 2 inch diameter aluminum pipe which was cut in half axially and polished on the internal surface.

The flow-through sample cell was a 17 inch length of 1 inch electrical conduit. Conduit connectors, which were drilled and tapped for the attachment of sample inlet and exhaust lines, were installed on each end of the conduit after a liberal application of number-one Permatex to insure against leakage. Quartz windows of 1.250 inch diameter and 0.1875 inch thickness were installed at each end of the sample cell. The windows were held in place by the conduit-connector nut and were sealed with a neoprene O-ring placed between the connector and the window.

The detector cell was machined from a 1-$\frac{1}{4}$ inch length of 2 inch diameter mild-steel shaft. Details of the detector cell construction are given in Fig. 3.

Pressure changes in the detector cell were monitored by an Ultradyne pressure transducer model S-40-RP$\pm$.2-D which was manufactured by Ultradyne of Albuquerque, N.M. This transducer is of the metal-diaphram, variable-reluctance type with a differential pressure range of plus or minus 0.2 pounds per square inch. The inactive side of the transducer was sealed, except for an extremely small bleed hole, to minimize the effects of air-pressure disturbances (such as the opening or closing of the laboratory door).

Although the oscillator power supply, inductance bridge, and demodulator electronic components normally used with this type of transducer (9) were not available, an available Baldwin-Lima-Hamilton model 120C strain indicator was found to be a reasonable substitute since it combines essentially these same components in one unit (10). The output from the strain indicator (normally used for oscilloscope input) was connected to a Hewlett Packard model 412A vacuum tube voltmeter. Shielded cable was used for all instrument leads and all components of the system were connected to a common ground to minimize "noise" in the detector signal.

The operational theory of this detector is similar to that of commercial nondispersive infrared analyzers. Any molecule, or group of molecules, when subjected to electromagnetic radiation in the infrared region will absorb energy causing either vibration of the molecular bonds or rotation of the molecule as a whole. Any specific molecule will absorb electromagnetic radiation at a number of distinct wavelengths depending upon the elements involved in the molecular structure and the type of bonding between these elements. The carbon-hydrogen bond for instance, typical of all hydrocarbons, absorbs electromagnetic radiation in the region of 3.43 microns causing vibration of the bond (11).

If a gaseous sample is confined and then exposed to infrared radiation of the wavelengths absorbed by the molecules of that gas, the resulting increased molecular activity will produce an increase in pressure which is a function of the intensity of the incident radiation. Exposure of the gas to wavelengths other than those absorbed by the molecules causes no excitation and thus no significant pressure change. The detector cell in this instrument was filled with propane and was therefore sensitive only to those wavelengths of infrared radiation which are absorbed by propane, one of which is 3.43 microns absorbed by the carbon-hydrogen bond.

With the sample cell of the instrument filled with a gas which does not absorb infrared radiation at the same wavelengths as propane, essentially the entire energy spectrom emitted by the source, up to the maximum wavelengths of 4 microns transmitted by the quartz windows (9), reaches the detector cell resulting in a maximum detector-cell pressure. If, however, the gas in the sample cell contains a component which absorbs radiation at some wavelength which propane also absorbs, a portion of the energy at this wavelength is removed from the beam in the sample cell and is not available to heat the molecules in the detector cell. A decrease in detector-cell pressure results. Since the absorbance of a component is related to the concentration of that component in the sample by the Lambert-Beer Law, given by

$$A = abc = \log_{10}\left(\frac{1}{T}\right)$$

where:

A = absorbance of sample

a = absorptivity at normal temperature and pressure

b = sample length in infrared path

c = molecular concentration of absorbing component of sample

T = transmittance of sample,

the pressure in the detector cell is a function of the concentration of components within the sample cell which abosrb the same wavelengths of electromagnetic radiation as those absorbed by propane.

Unburned hydrocarbons in exhaust gas from a spark-ignition engine are the only components of that gas which absorb infrared radiation in the same wavelength regions as propane (12) up to the 4 micron limit of the quartz windows. The reading of the voltmeter, which indicates pressure within the detector cell, can therefore be calibrated in terms of the concentration of unburned hydrocarbons in the exhaust gas passing through the sample cell. Although the response of the analyzer is not the same for given concentrations of the various hydrocarbon molecules present in the exhaust gas, the analyzer was calibrated with n-hexane and all readings are expressed in units of mole per cent n-hexane.

The calibration of the analyzer followed the same procedure, except for the use of a wet test meter, as that used by Neerman and Millar (8) for calibration of their instrument. Normal hexane was chosen for the calibration since its absorptivity closely approximates that of a typical gasoline and variations in the hydrocarbon species passing through the detector with time, as may be the case if gasoline were used, are avoided.

A diagram of the calibration equipment is shown in Fig. 4. Adjustment of the hydrocarbon concentration in the mixture is provided by diverting more or less of the total air flow through the flask containing the n-hexane. The hydrocarbon concentration is determined by evaporating a 10 gram mass of n-hexane through the analyzer and measuring the total airflow passing through the analyzer during this same period. The mole per cent n-hexane may then be calculated.

The calibration procedure was as follows:

1. With air only flowing through the sample cell and the infrared beam blocked with the shutter, the voltmeter was zeroed using the strain indicator null-balance.
2. The beam was then opened and the voltmeter reading was adjusted to a 100 mv reading by using the Variac to adjust the intensity of infrared radiation emitted by the source.
3. The shutter was then closed since the detector cell itself will experience some heating from the infrared radiation causing a slow increase in cell pressure, and thus a drift in the voltmeter reading, if the beam is not blocked between readings.
4. Purge air flow through the sample cell was stopped and flow through gas meter and flask containing n-hexane was established.
5. After 5 minutes at this condition to establish equilibrium of

the hydrocarbon mixture, the hexane balance was zeroed and the reading of the gas meter was simultaneously taken.

6. The infrared shutter was then opened, the voltmeter reading was recorded and the shutter was again closed. This procedure was repeated several times during each run to detect sample concentration changes if they should occur. The run was restarted in these instances.
7. The reading of the gas meter was again recorded at the time the balance indicated that 10 grams of hexane had been evaporated.
8. Flow through the gas meter was stopped, the sample cell was purged with air and the voltmeter zero and 100 mv full scale readings were checked.
9. The needle valves were adjusted to give a different hydrocarbon concentration and steps 1 through 8 were repeated.

A plot of the response of the analyzer to different concentrations of n-hexane in air, expressed in units of mole per cent n-hexane, is given in Fig. 5.

Although this instrument was adequate for this investigation, considerable difficulty was experienced due to the temperature sensitivity of the detector cell. If the room temperature changed slightly or if the infrared shutter was left open for more than a few seconds, a zero shift would occur as a result of thermal expansion of the gas in the detector cell. This zero shift did not appear to have any effect on the pressure change in the detector cell due to infrared absorption by the detector gas.

A suggested modification of the analyzer, which would improve the stability of the instrument considerably, is conversion to a dual beam instruement. This would involve the construction of an identical infrared source (or the division of the radiation emitted by the present source into two equal beams), the construction of a second sample cell which would contain only air and would not require sealing or flow-through capabilities, and the construction of a new detector head incorporating two identical detector cells. If both detector cells were filled with an identical gas and one cell was connected to each side of the differential-pressure transducer, then any thermal expansion experienced by the gas in one cell would also be experienced by the gas in the other cell maintaining a pressure balance across the transducer. Introduction of an absorbing sample into the flow-through sample cell would produce a pressure change in the detector cell of that beam only, resulting in a differential pressure between the cells. This differential pressure could then be monitored and calibrated in terms of sample concentrations by the same procedure used for the single-beam instrument. Continuous monitoring of a flowing sample would then be possible since heating of the detector head would have no effect on the differential pressure between the cells and zero shift would be eliminated.

C. Air-Fuel Ratio Measurement Equipment

The fuel-consumption rate for each test run was determined by mounting a fuel tank on one arm of an equal-arm balance and clocking the time required for the consumption of 50 grams of fuel with a stop watch.

The air-consumption rate was determined by connecting a model 50MH10-1 Meriam laminar flow meter element to the engine carburetor inlet. Although the manufacturer states that pulsating flow, such as that experienced in a single cylinder engine induction system, will not cause error in the measurement of air-flow with these units (13), a surge tank 10 inches in diameter and 48 inches long was installed between the carburetor and the flow meter to minimize the possibility of pulsating flow measurement errors.

Early tests with the above equipment indicated extremely rich air-fuel ratio delivery to the engine. Since it had been a number of years since the flow meter had been calibrated by the factory, a check of the calibration curve was made. This check consisted of placing the laminar flow meter element in series with a sharp-edge orifice-plate flow meter, and checking the flow indication of one unit against the other. All orifice-plate flow meter calculations were made in accordance with the equations, coefficients, and correction factors recommended in the Flow Meter Engineering Handbook (14).

Flow measurements were made at flow rates near one cubic foot per minute and near three cubic feet per minute. In both cases the flow rate indicated by the orifice-plate flow meter and the laminar-element agreed within 3 per cent. The calibration of the laminar-element flow meter was thereafter considered accurate enough for the purposes of this investigation.

The discrepancies in air-fuel ratio measurement were subsequently found to be due to an air leak into the inlet manifold.

D. Dynamometer and Miscellaneous Equipment

A Go-Power model DY-6D test dynamometer was used to provide a load on the engine when engine loading was utilized. Fair load control was experienced with this unit at most loads and speeds. Some fluctuation in loading was experienced at certain torques, however, which resulted in slow oscillations of engine speed of about plus or minus 20 revolutions per minute (rpm).

Although the Go-Power dynamometer unit was equipped with a tachometer, this tachometer fluctuated badly at some speeds resulting in extreme difficulty in obtaining a correct reading. A type 1531 Strobotac, manufactured by the General Radio Company, was found to be a satisfactory method of engine speed determination and was subsequently used for all rpm readings.

All pressure measurements were made with Meriam manometers, and the thermocouple signals were recorded on Brown Electronik recorders.

IV. EXPERIMENTAL PROCEDURE

A. General

In order to determine the effects of inlet-valve throttling on the unburned hydrocarbon concentration in the exhaust gas, test runs were made at several engine speeds and at various loadings at each speed. Test runs were also made with three different inlet-valve timings, all at the same engine speed, with various engine loadings.

At any particular set of operating conditions (load, speed, and inlet-valve timing) a test run was first made utilizing the carburetor throttle for engine control. This test was immediately followed by a test run, at the same operating conditions, utilizing inlet-valve throttling for engine control.

All test runs were made with the same exhaust-valve timing and lift to reduce the number of variables involved. The exhaust-valve timing and lift used is similar to that used by standard engines of this size and type, i.e., exhaust opening at 45 degrees before bottom dead center, exhaust valve lift of 0.100 inch, and exhaust closure at 12 degrees after top dead center. The valve timing was determined, for both inlet and exhaust valves, by defining the valve opening (closing) timing as being that crankshaft position at which the valve is 1 per cent of the maximum valve lift off the valve seat during the opening (closing) motion. This valve position was determined with a dial indicator and the crankshaft position was then read from the degree wheel. The dial indicator was also used to determine the inlet-valve lift when inlet-valve throttling test runs were made and when the inlet-valve lift was returned to 0.100 inch, which was used for all carburetor throttled runs.

All test runs used for plotting of results were made with a set ignition timing of 14 degrees before top dead center, again to reduce the number of variables. Two test runs were made, however, in which the ignition advance was set to minimum best torque positions. The ignition timing was determined from the degree wheel using a standard automotive timing light.

All test runs used for plotting of results were made at the best-torque air-fuel ratio. In other words, this was the air-fuel-ratio which would maintain the desired engine speed at the desired load with the minimum carburetor throttle valve opening or minimum inlet-valve lift, depending upon the type of throttling used. The air-fuel ratio adjustment for carburetor-valve throttled operation was made with the idle system mixture needle valve. This was possible since all test runs were made at engine loads and speeds requiring less than one-quarter throttle (because the effects of valve throttling should be greatest at light loads), and since the idle system in this carburetor is at least partially utilized up to at least one-quarter throttle. The air-fuel-ratio adjustment for inlet-valve throttled operation was accomplished by positioning the carburetor-throttle valve (varying the venturi area) to a position which would produce the correct negative pressure at the fuel nozzle for the delivery of the required amount of fuel to the air stream. This method of air-fuel-ratio adjustment did not compromise the theory of inlet-valve throttling since the pressure drop through the carburetor was found to be only 5 inches of water or less for all tests using inlet-valve throttling.

The fuel used was commercial regular-grade gasoline. All fuel was obtained from the same source and was essentially of the same composition since no delivery of fuel to the source tank was made during this investigation.

B. Choice of Engine Loads and Speeds

The greatest effects of inlet-valve throttling will occur at light loads since this is where the greatest difference exists between the inlet-valve throttling and carburetor-valve throttling modes of operation, full throttle being the same for both modes by definition. Because of this, the test runs made during this investigation were confined to engine loadings of one-half maximum torque or less.

Since the dynamometer scale reading is a direct measure of the engine torque, although not the correct numerical value since the dynamometer lever arm is 6.3 inches long and the scale reading is in pounds, the scale reading was chosen as a measure of the engine loading. It is realized that this is not as good a measure as indicated mean effective pressure but no means were available for obtaining indicator diagrams from the engine. Scale readings of 0, 1, $2\frac{1}{2}$ and in some cases 4 were chosen as test loadings for each engine speed. The maximum (wide-open throttle) scale reading obtainable over the speed range utilized was 8.3 pounds.

The minimum engine speed for loaded engine operation was chosen as 1500 rpm since dynamometer scale readings were too erratic for good readability at lower engine speeds. The maximum engine speed was chosen as 3000 rpm since continuous operation of most spark-ignition engines (automobile engines in particular) above this speed is rarely encountered. Two intermediate speeds of 2000 and 2500 rpm were chosen to provide data between the maximum and minimum speeds.

C. Procedure for Carburetor-Valve Throttling

The engine was started and allowed to run for several minutes to warm up. It was then shut off and the inlet-valve timing and lift were checked and adjusted to the desired values if necessary. After restarting, the load, speed, and exhaust back pressure were adjusted to approximately the values desired for the test to be performed.

After a stable cylinder head operating temperature of approximately 155°F was established (by adjusting the flow of cooling water through the cylinder head), the load, throttle position, exhaust back pressure, and air-fuel ratio were simultaneously adjusted to give the desired test conditions at the best-torque air-fuel ratio.

The fuel balance was then set to a condition of slightly over-weight on the fuel-tank arm. At the time the balance reached the equal-weight condition (due to the consumption of fuel), a stop watch was started, and a 50 gram weight was removed from the ballast arm of the balance.

The exhaust-hydrocarbon analyzer was next purged with air, and the zero and full-scale adjustments were made. The purge air flow was shut off, and a minimum of 2 minutes were allowed for the sample cell to be saturated with exhaust sample (a response time of about 30 seconds between a change in engine conditions and a change in detector signal was experienced). The hydrocarbon-analyzer reading was then taken and recorded.

During the time required for the consumption of 50 grams of fuel, all other required data were taken and recorded on the data sheet. A sample data sheet is shown in Fig. 6. The hydrocarbon-analyzer reading was also checked at least once during this period.

When the fuel balance again reached the equal-weight condition (indicating the consumption of 50 grams of fuel), the stop watch was stopped, and the time taken for the consumption of 50 grams of fuel was recorded. Following this, with the engine still running at test conditions, the hydrocarbon-analyzer reading was again checked, the sample cell was purged with air, and the analyzer zero and full-scale points were checked.

If for any reason the hydrocarbon-analyzer readings were not the same during the test period or if the zero and full-scale points did not check, the test was immediately rerun. If all checks were satisfactory, the engine was stopped and preparations were made for the next run.

D. Procedure for Inlet-Valve Throttling

The test procedure for the runs utilizing inlet-valve throttling was the same as that followed for carburetor-valve throttling except for the modifications made necessary by the inlet-valve throttling mechanism.

Since the inlet-valve mechanism is not a zero lash system, throttling by modulating the maximum inlet-valve lift with the variable rocker arm fulcrum point results in a change in valve timing as well. This is a consequence of the greater motion of the camshaft end of the rocker arm (and thus further rotation of the camshaft) necessary to take up the valve system lash as greater inlet-valve lift modulation is used.

In order to obtain the correct valve timing for each test, the engine was started, allowed to warm up and then adjusted to the desired test conditions using inlet-valve throttling. The engine was then stopped and the inlet-valve timing was adjusted to the desired values, using the adjustable valve timing mechanism, at the lift required to produce the test conditions. This process was repeated until the desired test conditions were met with the correct valve timing. Only one readjustment was normally required.

The fuel-consumption-rate measurement was then begun with the remainder of the test procedure being the same as for carburetor throttling except for a final check of the inlet-valve timing and measurement of the inlet-valve lift after the engine was shut down at the end of each test run.

V. DISCUSSION OF RESULTS

The raw data obtained during this investigation are presented in the appendix. After converting these data to more useful forms, they were combined in the form of a series of plots (Figs. 7 to 17). Figs. 7 to 12 present the exhaust-gas hydrocarbon concentration as a function of dynamometer torque-meter readings at constant engine speeds and inlet-valve timing. Figs. 13 to 15 present the exhaust-gas hydrocarbon concentration as a function of engine speed at a constant dynamometer torque-meter reading and inlet-valve timing. The measured best-torque air-fuel ratio is recorded next to each data point on the above plots. Where the data points for both modes of operation fell at the same plot coordinates, the air-fuel ratio corresponding to the carburetor-throttled mode of operation is prefixed with a "C-" and that for inlet-valve throttling by a "V-" for identification.

Fig. 16 presents the required valve lift for inlet-valve throttled operation as a function of dynamometer torque-meter readings, for the three inlet-valve timing variations investigated, at a constant engine speed of 2000 rpm.

Fig. 17 presents the indicated exhaust-gas temperature as a function of engine speed for both inlet-valve and carburetor throttling, at dynamometer torque-meter readings of 0, 1.0 and 2.5, all at the same inlet-valve timing of 10°btdc and 51°abdc.

Figs. 7, 8, and 9 show curves of the same general shape for both carburetor and inlet-valve throttling at engine speeds of 1500, 2000, and 2500 rpm. The important point revealed by these curves is that, at every speed and load within the range covered, the exhaust-gas hydrocarbon concentration is greater for inlet-valve throttling even though, in most cases, a leaner air-fuel ratio is required for best-torque operation. This is of interest since previous investigations with conventional throttling have shown that, in general, exhaust-gas hydrocarbon concentrations are decreased by the use of a leaner air-fuel ratio (within the range of air-fuel ratios encountered in this investigation) at any given engine speed and load (4, 15, 16).

This apparent discrepancy can be explained by considering the difference in charge turbulence between carburetor and inlet-valve throttling as reported by Stivender (17), and the effect of this difference on the heat transfer between the burning mixture and the considerably cooler combustion-chamber walls. With conventional throttling the combustion reaction approaches to within a finite distance of the combustion-chamber walls at which point it is extinguished due to lowering of the reaction temperature to below the self-sustaining limit due to heat transfer to the combustion-chamber walls. With inlet-valve throttling the combustion reaction is halted a greater distance from the combustion-chamber walls due to the increased heat transfer resulting from the increased charge turbulence.

In each case a certain amount of the unburned mixture will be discharged with the exhaust contributing to the exhaust-gas hydrocarbon concentration. Since the volume of unburned mixture, following combustion, is greater with inlet-valve throttling, it is reasonable that a greater volume of unburned mixture would be included in the exhaust gas (not the entire amount because of the clearance volume). As a result, even though a specific volume of unburned mixture with inlet-valve throttling may contain a lower concentration of hydrocarbons, the overall concentration of hydrocarbons in the exhaust gas can be, and in this case is, greater than with carburetor throttling.

Another interesting point noted in the curves of Figs. 7, 8 and 9 is the reduction in exhaust-gas hydrocarbon concentration with a reduction in engine load (dynamometer torque-meter reading) for both methods of throttling. Huls, Myers, and Uyehara (4) reported a slight increase in hydrocarbon concentration with decreasing load. This trend was attributed to the decrease in exhaust temperature with a decrease in load (lowered reaction rate of hydrocarbons with oxygen in the exhaust system at lower temperatures) since their exhaust sample tap was located downstream of a 20 inch length of exhaust pipe and a 2000 cubic inch mixing tank. Location of the sample tap near the exhaust valve with rapid sample cooling, as was the case in this investigation, reduces the effect of exhaust temperature on hydrocarbon concentration in the exhaust-gas sample, and could explain the difference in results.

The curves of Fig. 10, which are plots of the same quantities as those in Figs. 7, 8, and 9, but at an engine speed of 3000 rpm, do not appear to fit the pattern set in Figs. 7, 8, and 9. These same data, however, when plotted, with data from Figs. 7, 8, and 9, in the form of exhaust-gas hydrocarbon concentration as a function of engine speed for constant dynamometer torque-meter readings, resulting in the curves of

Figs. 13, 14, and 15, do fit the curves established by the lower engine speed data. (An exception is the data for 3000 revolutions per minute at a dynamometer torque-meter reading of 2.5.) This leads to the conclusion that the shape of the curves in Fig. 10 is a result of changes in the exhaust-gas hydrocarbon concentration due to increased engine speed and not due to inconsistent data.

The experimental points for an engine speed of 3000 rpm, a dynamometer torque-meter reading of 2.5, and with inlet-valve throttling (see Figs. 10 and 15) are not consistent with the rest of the data. This inconsistency may be attributed to the characteristics of the system rather than to the valve-throttling process itself.

With inlet-valve throttling at all loads and speeds the pressure drop through the carburetor was maintained at a low value, resulting in poor atomization of the fuel and poor mixing of the fuel with the air at the carburetor. At light loads or low speeds this poorly atomized, nonhomogeneous mixture condition is corrected at the inlet valve since the sonic velocity encountered through the small inlet-valve aperture (17) produces large shear forces on the liquid fuel breaking it up into finely divided particles. The induced turbulence then thoroughly mixes the fuel particles with the air.

It is postulated that at the engine speed and load in question, the pressure drop through the carburetor (and thus the venturi velocity) is still low enough to result in poor atomization and mixing of the fuel with the air, and that the pressure drop through the inlet-valve aperture has decreased to below the critical ratio (found by Stivender (17) to be the case for an inlet-valve throttled automobile engine at about 50% load) resulting in a loss of at least a portion of the beneficial atomization and mixing of the fuel experienced at lower speeds or lighter loads. The poorly atomized, nonhomogeneous mixture in the combustion chamber at the time of ignition would then account for both the high concentration of unburned hydrocarbon in the exhaust gas and the rich air-fuel ratio necessary for best-torque operation. Note that two experimental points are included at these conditions since a second test was run to substantiate or discredit the results of the first test. Both tests resulted in nearly the same data.

The curves of Figs. 13, 14, and 15, except for the point on Fig. 15 just discussed and the carburetor throttled curve on Fig. 13, all show the increase in exhaust-gas hydrocarbon concentration with decrease in speed shown by Hagen and Holiday (16) in their investigation of hydrocarbon emissions. This trend was also predicted by Huls, Myers, and Uyehara (4) in the discussion presented at the end of their paper. Their explanation of this increase in exhaust-gas hydrocarbon concentration with a decrease in engine speed is that the quench thickness (thickness of unburned mixture at the combustion-chamber walls) should increase with a decrease in speed resulting in increased hydrocarbon emissions. This increase in quench thickness can be explained by considering the greater time available for heat transfer from the charge to the combustion-chamber walls at a lower engine speed. It seems reasonable that this would be true for either method of throttling considered in this investigation, as the results indicate.

The reason for the peculiar shape of the carburetor-throttled curve of Fig. 13 is not clear. It may be due to the changes in carburetor fuel-air mixing characteristics associated with the transition from 100% fuel delivery through the idle circuit to participation of both the idle and high-speed circuits in the fuel delivery to the air stream.

It should be noted that, as with the plots of Figs. 7, 8, 9, and 10, in almost every case, within the range of the plots of Figs. 13, 14, and 15, the inlet-valve throttling method resulted in a greater exhaust-gas hydrocarbon concentration than the conventional carburetor throttling method.

Comparison of the inlet-valve throttled curves of Figs. 8, 11, and 12 reveals that there is little change in the shape of the curves, with changes in inlet-valve timing, over the same dynamometer torque-meter reading range (note that the curves of Figs. 11 and 12 are extended to a higher load than those of Fig. 8).

A slight decrease in the exhaust-gas hydrocarbon concentration is evident with inlet-valve throttling at light loads when the inlet-valve-closed timing is changed from 51 degrees after bottom dead center to 35 degrees after bottom dead center (compare Figs. 8 and 11). This decrease may be due to the higher effective compression ratio obtained at light loads with earlier inlet-valve-closed timing.

Little change is noted at light loads with inlet-valve throttling when the inlet-valve-open timing is changed from 10 degrees before top dead center to top dead center (compare Figs. 11 and 12). The difference between the hydrocarbon concentrations in the exhaust gas with inlet-valve throttling at the highest load of Figs. 11 and 12 can be attributed to the difference in air-fuel ratios.

It is reasonable that inlet-valve timing would have little effect on the exhaust-gas hydrocarbon concentration when operating at light loads with inlet-valve throttling since the valve lifts are so small toward the ends of the valve-open duration that flow of a significant amount is allowed only during that part of the duration when the valve is near its maximum lift. Since maximum lift occurs during the middle and latter part of the intake stroke, where cylinder conditions do not change drastically with crankshaft rotation, a change in inlet-valve timing should not result in a large change in the exhaust-gas hydrocarbon concentration.

A plot of valve lift as a function of dynamometer torque-meter readings for the three inlet-valve timing conditions investigated (Fig. 16) indicates little change in the valve lift necessary for the same load with different valve timing. The small valve lift variations noted may be attributed to the change in duration of inlet-valve opening.

As an aside, the discussion above leads to speculation on the possible use of inlet-valve throttling with long-duration, high-lift valve timing specifications, resulting in good high-speed performance (full valve lift) along with good idle and low load characteristics (due to the factors considered above). This possibility was not investigated but could produce interesting results.

Changing the inlet-valve timing had a pronounced effect on the exhaust-gas hydrocarbon concentration during light-load, carburetor-throttled operation. This was particularly evident with the change from 51 degrees after bottom dead center to 35 degrees after bottom dead center closing (compare the no-load points on the carburetor-throttled curves of Figs. 8 and 11) which resulted in a substantial increase in exhaust-gas hydrocarbon concentration.

A similar increase was noted (within the range of air-fuel ratios encountered in this investigation) by Hagen and Holiday (16) during an investigation of the effects of valve overlap on exhaust-gas hydrocarbon concentration at engine idle. Their method of adjusting the valve overlap consisted of increasing or decreasing the valve system lash which, as they pointed out, also changes the total number of crankshaft degrees that each valve remains open. The data obtained by this method indicate an increase in exhaust-gas hydrocarbons with a decrease in valve overlap within the air-fuel ratio range of from 9.5 to as high as 14.5.

It is suggested here that this increase in exhaust-gas hydrocarbon concentration is not due to a decrease in valve overlap, but is instead due to the earlier closing of the inlet valve which results from the method used to reduce the valve overlap. This suggestion is substantiated by the lack of an increase, and in fact a decrease, in the exhaust-gas hydrocarbon concentration at light load, with carburetor-throttling, when the valve overlap was reduced while holding the valve-closed-timing constant during the present investigation. (Compare the 0 and 1.0 torquemeter reading points on the carburetor-throttled curve of Fig. 11 to the corresponding points in Fig. 12.) Further investigation into the effects of valve overlap and inlet-valve-closed timing may be worthwhile.

As a matter of interest, two test runs were made at an engine speed of 2000 rpm, a dynamometer torque-meter reading of 2.5, with best-torque air-fuel ratio, and with minimum-best-torque ignition advance. One run utilized inlet-valve throttling and the other carburetor throttling. The inlet-valve timing in both cases was valve open at 10 degrees before top dead center and closed at 51 degrees after bottom dead center.

The minimum-best-torque ignition timing was more advanced for both runs than the basic 14 degrees before top dead center ignition timing used for all other tests. The minimum-best-torque ignition timing was more advanced for carburetor throttling (23° btdc), however, than was the case for inlet-valve throttling (18° btdc). This is consistent with the data obtained by Stivender (17). As with most other tests, the best-torque air-fuel ratio is leaner with inlet-valve throttling than with carburetor throttling.

The interesting result of the comparison at minimum-best-torque ignition advance is the same concentration of hydrocarbons in the exhaust for both modes of operation. Although this single comparison is not sufficient to support any conclusions, it is possible that a comparison of the two methods of throttling, using minimum-best-torque ignition advance for all tests, would result in a better position for inlet-valve throttling in the hydrocarbon emission standings.

The data from this comparison are included as a single point on Figs. 8 and 15. The increased exhaust-gas hydrocarbon concentration for these test runs, as compared to those runs with the basic 14 degrees before top dead center ignition timing, is due to the more advanced ignition timing. This increase in hydrocarbon emission with an increase in ignition advance is well documented (4, 15, 16).

The plot of exhaust-gas indicated temperature as a function of engine speed (Fig. 17) is presented to illustrate the difference in indicated exhaust-gas temperatures between the two methods of throttling. In all cases investigated, the indicated exhaust temperature with carburetor throttling was higher than that with inlet-valve throttling at the same engine speed and load. This is consistent with the data reported by Stivender (17).

An observation made during the course of this investigation is that a very pronounced softening of the exhaust noise occurred when the method of throttling was switched from the carburetor to the inlet valve during light load operation. A possible explanation for this is that the faster burning characteristics of the mixture during inlet-valve throttled operation produce the same effect as advancing the ignition timing would have with conventionally throttled operation. This, however, would have to be determined in a separate investigation.

The conclusion reached as a result of this investigation is that, in general, even though inlet-valve throttling results in best-torque operation at leaner air-fuel ratios than does carburetor throttling, the concentration of unburned hydrocarbons in the exhaust gas is higher for inlet-valve throttling than it is for carburetor throttling.

VI. SUMMARY

The purpose of this investigation was to determine the effects of inlet-valve throttling on the concentration of unburned hydrocarbons in the exhaust gas of a spark-ignition engine.

The test engine was a single cylinder Continental with a piston displacement of 7.09 cubic inches and a compression ratio of 5.25 to 1. Engine modifications included a unique dual-overhead-camshaft cylinder head assembly and a variable fulcrum point inlet-valve rocker arm assembly which was designed and constructed to allow modulation of the inlet-valve lift for inlet-valve throttling.

Determinations of the concentration of unburned hydrocarbons in the exhaust gas were made with an instrument constructed by the author. This instrument is of the nondispersive infrared absorption type.

Tests were conducted for inlet-valve throttled operation and for conventional carburetor throttled operation over a load range of from no-load up to approximately one-quarter load and a speed range of from 1500 to 3000 rpm, all with a constant inlet-valve timing. Additional tests were conducted at 2000 rpm with different inlet-valve timing specifications.

All tests were performed, with one exception, at a constant ignition timing and at constant exhaust valve timing specifications. All plotted results were obtained utilizing best-torque air-fuel ratios.

The conclusion reached as a result of this investigation is that, in general, inlet-valve throttling will provide best-torque operation at leaner air-fuel ratios than conventional throttling but results in a higher concentration of unburned hydrocarbons in the exhaust gas.