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Final Report

TWO-STROKE GASOLINE ENGINE EXHAUST EMISSIONS

David E. Cole
Jay A. Bolt

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I. INTRODUCTION

The purpose of this research program was to obtain a measurement and to develop an understanding of the exhaust emissions from a 2-stroke spark ignited outboard engine. Furthermore it was hoped that technology could be formulated that would lead to improvements in the design of the 2-stroke engine to lower undesirable exhaust emissions.

BACKGROUND

During the past few years interest in air pollution on both the local and national level has been increasing rapidly. The internal combustion engine has been receiving much criticism for its suspected contribution to the problem. In some special cases, notably Los Angeles with its photochemical smog, the automotive spark-ignition engine has been blamed for the greatest proportion of the total problem. California,¹ in fact, was the first governmental body to legislate for control of engine exhaust emissions, doing such for the automobile in 1961. As part of this legislation a 7-mode chassis dynamometer driving schedule was developed which was designed to simulate typical automotive driving conditions. Also specified was a nondispersive infrared analysis technique for measurement of the various exhaust pollutants. The concentration of the unburned hydrocarbons and carbon monoxide are recorded throughout the several modes of the driving cycle which is repeated a number of times. Through the use of appropriate multiplying or weighting factors a composite emission standard or limit was formulated for both the hydrocarbon and carbon monoxide emissions.

In 1967 California modified their earlier specifications and established a composite limit of 275 ppm (parts per million) for the unburned hydrocarbons and 1.5% by volume for carbon monoxide. These standards applied to automotive engines with greater than 140 cu in. displacement. The federal government has since followed California's lead and has adopted the Clean Air Act² which specifies automotive emission standards in the continental United States. The federal standards currently are established at 275 ppm measured as normal hexane for the unburned hydrocarbons and 1.5% carbon monoxide and also specifies the use of the California chassis dynamometer driving cycle. These standards are certainly subject to modification in the future; in fact a new set of standards has been adopted for 1970. Initially the 1970 standards were established on a concentration base at 170 ppm for the unburned hydrocarbons and 1% for carbon monoxide. Recently, however, there has been an important modification in this standard. The concentration base has been replaced with a mass base which limits the unburned hydrocarbons to 2.2 grams per vehicle mile and carbon monoxide to 23 grams per vehicle mile. It can reliably be projected that all future standards or regulations will be formulated with such a mass base. This mass

standard is actually very realistic when we consider that the total mass of pollutants entering the atmosphere is really of prime importance not just their concentration per unit mass or volume of exhaust gas.

The increased interest in the pollution aspects of our fossile fueled powerplants has thrust a new design parameter at the engine designer. In the past he was primarily concerned with obtaining maximum performance and economy together with acceptable durability and manufacturability from his powerplants. To satisfy these often diverse requirements, many compromises were necessary in order to approach an optimum system. Now, the new parameter of low undesirable exhaust emissions has assumed equal if not greater importance when compared with the others. This additional consideration has greatly complicated the already formidable task which faces our transportation industry and particularly the engine designers in that industry. Furthermore, the very existence of certain industries now appears to be a function of government standards and the ability of the engineer to satisfy them in his product design.

As control is accomplished for the more significant of our current pollution sources, such as the automobile and industrial stack effluence, those that were initially of lesser stature will most certainly increase in importance on a relative scale. We sincerely believe that federal regulations governing 2-stroke spark ignition engines are imminent and will appear in the not too distant future. This undoubtedly will require, as a minimum, substantial design modifications of the conventional 2-stroke gasoline engine. Because of this it is definitely desirable to have positive and valid technical information available at the time the outboard engine industry is requested or required to participate in the formulation of regulations or standards governing their products. It is imperative that fair and equitable laws be written that are cognizant of both the requirements of the general public for clean air and reasonably priced and efficient marine transportation and the manufacturer's problems of designing and building their engines.

Early in our test program we became aware of the seriousness of the possible pollution problem facing the 2-stroke engine. The unburned hydrocarbons in the exhaust were selected as a measure of the undesirable emissions because they are probably the most serious of the 2-stroke engine pollutants.* In the data

* Early in the test program it was decided to concentrate on the unburned hydrocarbon emissions since it is reasonable to expect that the carbon monoxide (CO) and oxides of nitrogen (NO_x) exhaust concentrations are similar to those from a 4-stroke engine and therefore not of immediate concern.

The concentration of carbon monoxide is basically a function of the stoichiometry of the fuel and air, and the quality of the combustion process. With the use of rich mixture ratios carbon monoxide is always present; the concentration being directly related to the degree of richness. On the other hand the oxides of nitrogen are formed near or in the high temperature flame front in the combustion chamber, the higher the flame temperature the higher the concentration of NO_x. In the 2-stroke engine with its lower compression ratio and greater exhaust dilution of the fresh charge, the flame temperatures are undoubtedly much lower than in the 4-stroke engine, and therefore, the oxides of nitrogen concentration should be substantially lower.

we observed boat load concentration emission levels from 3,000 to 7,000 ppm measured as normal hexane. This compares with typical road load emission levels of 150 to 250 ppm for the average 1968 passenger car. Stated another way in terms of mass hydrocarbon emission, a modern 16-ft boat driven by a 40 hp outboard engine and pulling a water skier emits about 20 times the mass of unburned hydrocarbons as the typical car travelling at 70 miles per hour on an expressway. Furthermore, the average 3 hp 2-stroke lawnmower engine, assuming comparable concentration emission levels with the outboard, will have equivalent mass emissions to the car mentioned above.

Before initiating the test program we made a thorough survey of the literature. Unfortunately, very little pertinent information was found. In fact, we were unable to find any previous emission work on spark-ignition 2-cycle engines which led us to base our investigation on automotive engine practice.

In our experimental study we have been primarily concerned with the 2-stroke outboard engine air pollution problem and have given limited consideration to the water pollution problem. The program as it is currently defined is intended to provide a first order evaluation of the hydrocarbon emissions from a 100 hp Johnson outboard engine. Emissions from the standard, as received engine, were measured under simulated boat load conditions to determine an appropriate base line for future comparison. Subsequently we initiated a parameter study to determine the effect of a number of operating and design variables on the unburned hydrocarbons. Because of unforeseen experimental problems particularly with poor base line reproducibility, random misfiring, and difficulties in measuring individual cylinder fuel/air ratios, we have only partially completed this phase of the test program. An experimental technique developed for automotive hydrocarbon emission measurement has been used for quantitative analysis. This technique relies on a nondispersive infrared, NDIR, analyzer³ for determining the concentration of the unburned hydrocarbons in the exhaust measured as ppm of n-hexane (C_6H_{14}).

Several exhaust sampling techniques were developed to provide a meaningful and appropriate sample for the analyzer. One system is a simulation of the underwater exhaust and the other, a condenser arrangement which permits controlled condensation of the low boiling point compounds in the exhaust and still allows analysis of the vapor.

As the project now stands, we are preparing a new hydrocarbon analyzer (Flame Ionization Detector, F.I.D.^{4,5} which permits much more accurate analysis of the total unburned hydrocarbons in the exhaust than can be obtained with the NDIR analyzer) and are developing instrumentation to quantify the engine misfiring.

II. OBSERVATIONS AND CONCLUSIONS

1. For the base engine the mean exhaust pipe hydrocarbon emissions ranged from 2800 to 5200 ppm with the highest emissions occurring at the lowest speeds and loads. These results are summarized in Table I.

TABLE I
SUMMARY OF BASE LINE HYDROCARBON EMISSION RESULTS
(NDIR Hydrocarbon Emissions—ppm n-Hexane)

Test Conditions	Exhaust Port Number				Mean Avg of the 4 Cyl Measured at the Exh Port	Exhaust Pipe
	1	2	3	4		
1000 rpm 1.5 BHP	7000	4900	6300	4800	5800	5200
2000 rpm 9.0 BHP	3600	3200	2900	3200	3200	3100
3000 rpm 24.0 BHP	3300	3300	3100	3200	3200	2900
4000 rpm 50.0 BHP	3800	3700	3200	3300	3500	2800

2. The mean average (of the four cylinders) exhaust port emissions were higher than the exhaust pipe emissions and are also tabulated in Table I.

3. Wide cylinder to cylinder exhaust port emission variation was observed, particularly at low speed (25% variation was observed at 1000 rpm).

4. Poor base line reproducibility was observed at the 1000 and 2000 rpm test conditions. From port no.1, for example, the hydrocarbon emissions varied from 6100 ppm to 7900 ppm.

5. The exhaust pipe mass rate of unburned hydrocarbon emissions increased from 1.3 lb/hr at 1000 rpm, boat load, to 4.8 lb/hr at 4000 rpm, boat load.

6. The mass rate of hydrocarbon emissions from the outboard engine is

approximately 15 times greater than those from an automotive engine at an equivalent speed and load.

7. Random engine misfiring is a significant problem particularly at the low loads.

8. The principal source of the hydrocarbon emissions is the unburned mixture which passes through the engine during the scavenging process.

9. The emissions related to the combustion process are of minor significance (400-800 ppm).

10. Variation of the fuel/air ratio causes significant variation in the emission level due primarily to changes in the misfiring frequency and the hydrocarbon concentration in the unburned mixture. Minimum emissions were found with slightly rich mixtures (13/1 air-fuel ratio).

11. The sampling system used appeared to have little effect on the measured gaseous emissions. The samples obtained with the underwater simulation system, the condenser system and the quench probe exhibited approximately the same hydrocarbon concentration.

12. The percentage of the original, unburned fuel-air mixture remaining in the exhaust can be determined for rich mixtures by measurements of the exhaust gas oxygen concentration.

13. If the engine misfiring frequency is low, the oxygen tracer method will provide a direct quantitative measurement of engine trapping efficiency.

14. The maximum engine trapping efficiency observed was slightly greater than 70%.

III. RECOMMENDATIONS

1. Work should be initiated "in house" to develop the analytical capability for measuring undesirable exhaust emissions.
2. Design modifications should be vigorously explored which could minimize the scavenging loss of raw fuel. These might include:
 - a. cylinder fuel-injection after exhaust port closing; and
 - b. dual scavenging, initially using air only and then the addition of the fuel/air mixture.
3. The carburetion appears to be a weak element in the engine package. Redesign to obtain better control over the overall fuel/air ratio and to improve cylinder to cylinder distribution would be appropriate. The band width of high speed and idle fuel-air ratios could be decreased to minimize misfiring and "through scavenging" of ultra-rich mixtures.
4. A comprehensive simulated boat load driving cycle should be developed from evaluation of typical "in service" operation.
5. Since the unburned hydrocarbons formed during the combustion process are of minor importance, development to decrease them would have little effect on the overall emission level.
6. Alternative designs to the 2-stroke cycle should be seriously considered.

IV. TEST PROGRAM

The original test program was defined in terms of automotive engine practice^{6,7} and was thus subject to modification as more was learned of the 2-stroke engine emission characteristics. Before meaningful data could be collected a simulated boat load driving schedule was established to define the engine test operating points. Fortunately, the average boat-motor combination is subject to a minimum of transient operation and therefore only steady-state conditions were selected. Performance characteristics were assumed for a displacement hull at low speeds and a planing hull at moderate and high speeds. The curve selected is shown in Fig. 1 and a summary of the boat-load test points are shown in Table II. The values for scale load and dynamometer speed were obtained using the dynamometer constant for our test stand and the lower unit gear ratio, respectively.

TABLE II

TEST SCHEDULE—ASSUMED BOAT LOAD OPERATING CONDITIONS

Engine, rpm	Dynamometer, rpm	Brake, hp	Torque, ft-lbf	Dynamometer Scale Load	Approx. Boat Speed V ft/sec
1000	575	1.5	7.88	10.4	5.0
2000	1150	9.0	23.6	31.3	8.6
3000	1725	24.0	42.0	55.6	17.5
4000	2300	50.0	56.6	87.0	32.0
5000	2875	86.0	90.3	120.0	54.0

A. BASE LINE

Initially the base line hydrocarbon emissions were measured at the simulated boat-load test conditions for the standard engine with the spark advance and carburetor adjusted to factory specifications. This step in any test program is vitally important as it provides knowledge of the statistical reproducibility of the data and a basis for future comparison.

To facilitate reduction of the large quantity of important experimental data, a computer program, shown in Appendix A, was written by Mr. D. Lebut and Mr. P. Van DeWalker. This program greatly simplified the data analysis.

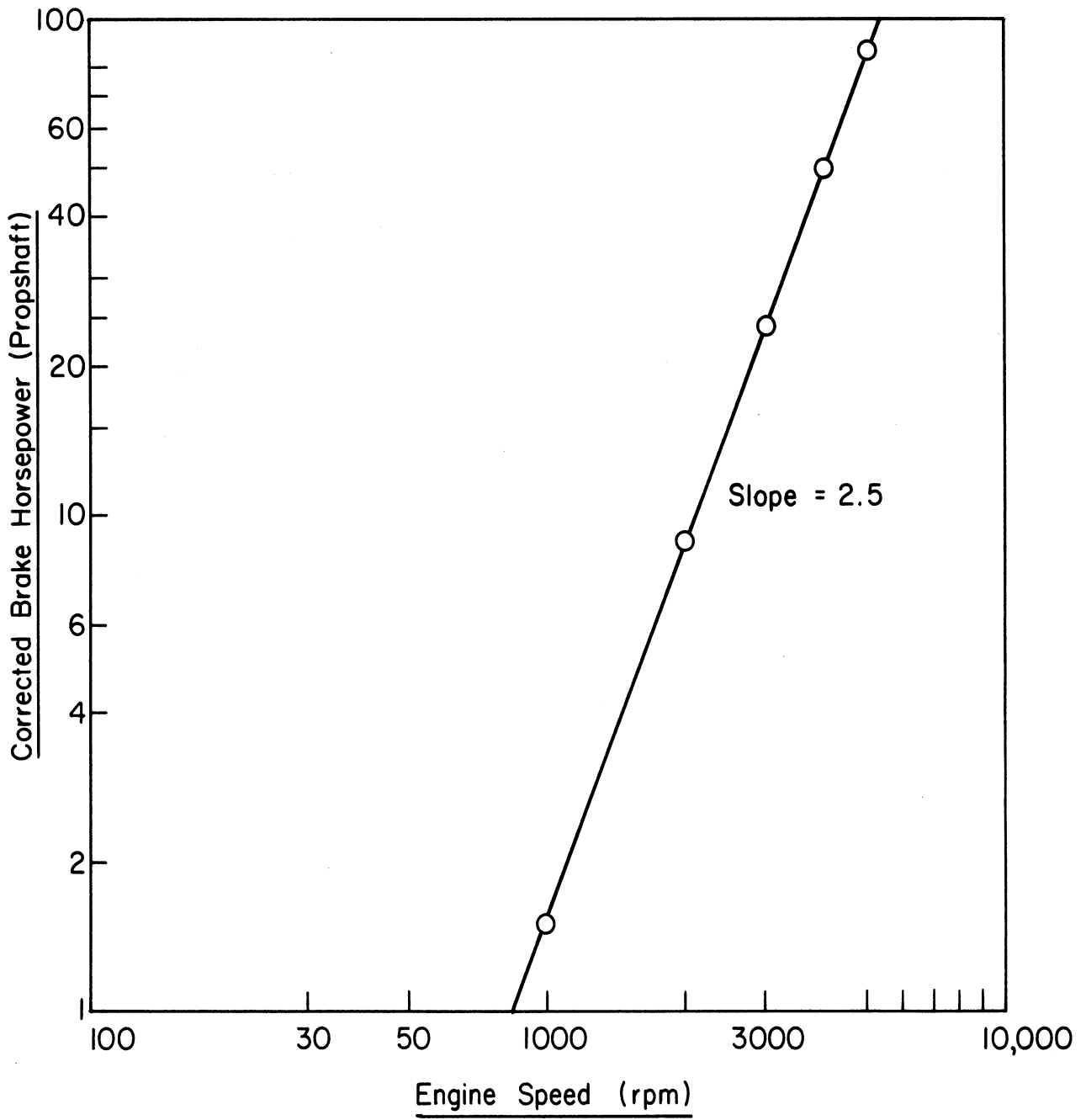


Fig. 1. Assumed boat load curve.

B. PARAMETER STUDY

The second stage of the program was the determination of the effect of certain operating variables on the unburned hydrocarbons. A number of important engine parameters were selected, the variation of which was expected to have significant effect on the undesirable emissions. The parameters selected were:

1. Fuel/air ratio
2. Spark advance
3. Fuel/air ratio and spark advance considered together
4. Engine coolant temperatures
5. Fuel—regular vs. white gasoline
6. Oil—a function of:
 - a. concentration of oil in fuel
 - b. composition (high paraffinic vs. low paraffinic)
7. Pressure ratio across the engine cylinder ($P_{\text{exhaust}}/P_{\text{inlet}}$)
8. Combustion chamber deposits

Due to unexpected problems and a significant amount of time spent developing new techniques and instrumentation, only a good start was made on this test program.

It should be mentioned that much work was done which was incidental to the primary objective of the program and included:

1. Development of underwater exhaust simulation.
2. Construction of an exhaust sample condensing system to permit collection of most of the high boiling point compounds.
3. Construction of stagnation quench probe.
4. Development of individual cylinder fuel/air ratio measurement system.

Two important instrumentation developments were brought about by findings in the early part of the testing and became part of the total program.

1. Very early it was deemed desirable to obtain a quantitative measure for the engine trapping efficiency. This instrumentation will be discussed in detail on page 42.

2. The data from light load engine testing strongly suggested that random misfiring was of major importance as a possible emission source. The ongoing development of possible techniques to measure misfiring is discussed in detail on pages 48-52.

V. TEST EQUIPMENT

A. DYNAMOMETER INSTALLATION

The Johnson 100 hp outboard engine was installed in room 244 of The University of Michigan Automotive Laboratory and was connected to a 125 hp General Electric motoring-absorption dynamometer with speed control. A special engine test stand including a water bath for the lower unit was furnished by Outboard Marine Corporation. The installation is shown in Figs. 2 and 3. Note in Fig. 2 that several components of the capacitor discharge ignition system were mounted on the side of the test stand to facilitate access to the exhaust plenum cover. Also, a heavy-duty aluminum shield was installed over the flywheel as a safety precaution.

Another modification which is not visible in the photographs is the electric fuel-pump used in place of the standard engine fuel pump.

B. EXHAUST GAS SAMPLING AND PREPARATION SYSTEMS

1. Sample Probes

Two different sampling probes were devised to collect the hot exhaust gases from the engine exhaust system. One of these was a static, stainless steel probe which was designed to collect a "hot" sample from each exhaust port and the exhaust pipe. The design is shown in Fig. 4 and the location of five of these probes on the exhaust plenum cover may be observed in Fig. 5. The probes used were joined in a manifold and valve assembly before entering the main sample line. By systematically shorting out the spark plugs and measuring the emission level of adjacent cylinders it was determined that there was very little "cross-talk" between cylinders, that is, the sample probes obtained practically all of their sample from the proper cylinder. In fact for cylinders nos. 1 and 2 the "cross-talk" was not measurable. Both the sample probes and the sample line were heated outside of the engine to minimize "hang up" or retention of the heavier hydrocarbons in the plumbing.

The other type of probe, shown schematically in Fig. 6, was designed to quench the exhaust sample and minimize the possibility of chemical reaction in the sample line. Partial oxygenation (to aldehydes) can be a problem with the heavier fuel and lubricating oil molecules in the exhaust and could cause erroneous measurement of the exhaust port oxygen and hydrocarbon concentration. To enable coolant flow very close to the sample inlet it was necessary to use only a single stagnation inlet hole at the probe tip. The sample was then ducted, at room temperature, to the measurement system.

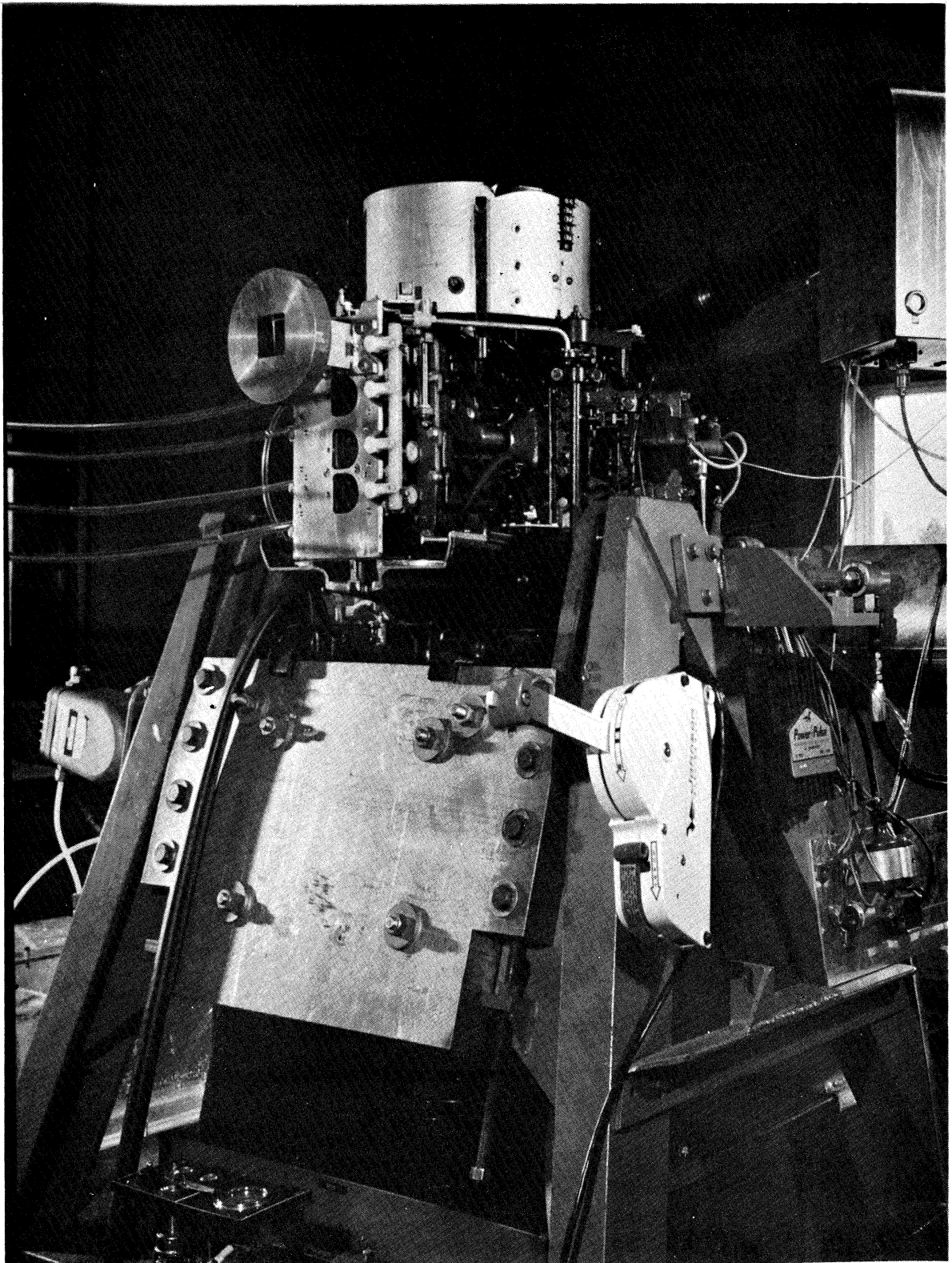


Fig. 2. Johnson outboard engine dynamometer installation—front quartering view.

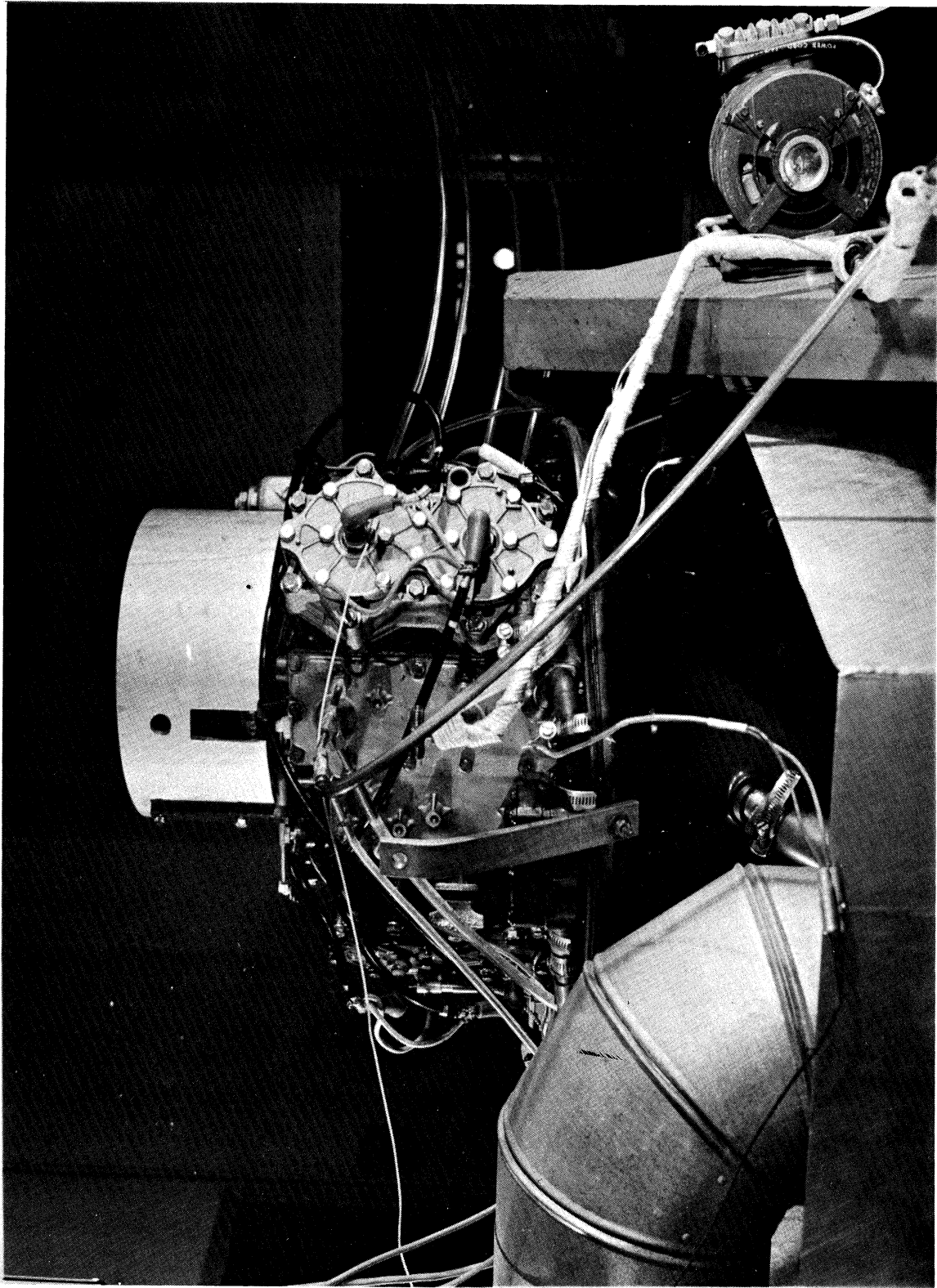


Fig. 3. Johnson outboard engine dynamometer installation--rear quartering view.

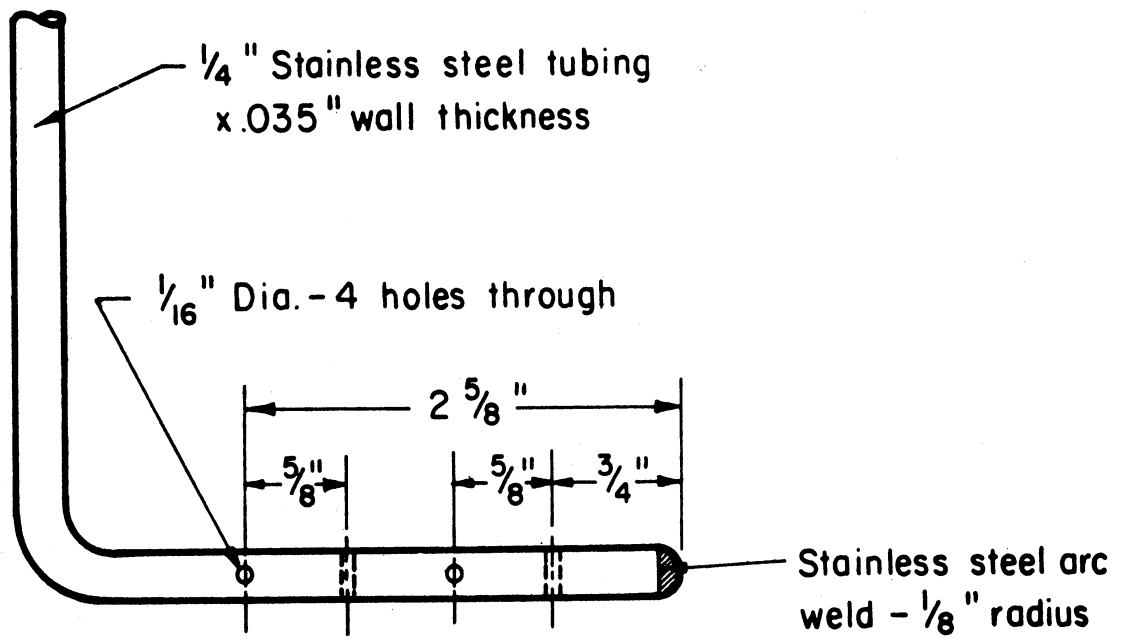


Fig. 4. Exhaust gas sampling probe.

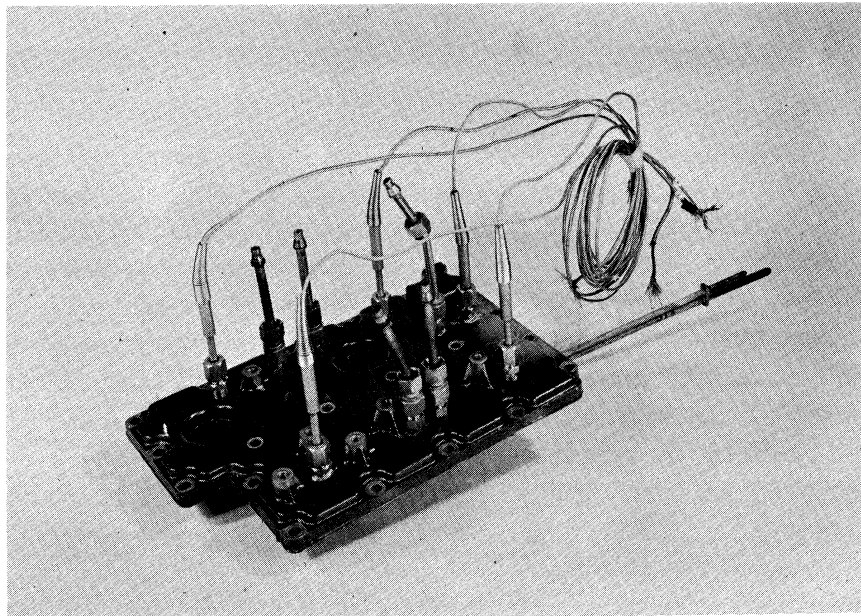


Fig. 5. Exhaust plenum cover showing the sampling probe and thermocouple locations.

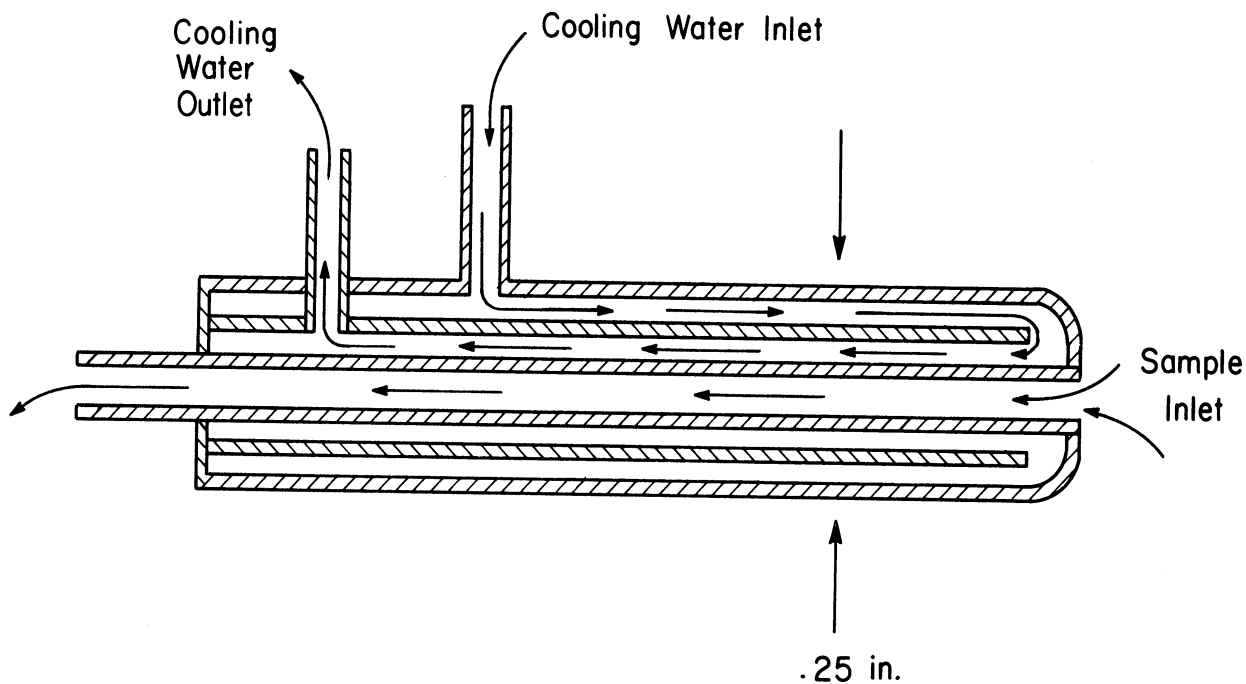


Fig. 6. Schematic diagram of the exhaust gas sampling "quench" probe.

2. Sample Preparation

a. Underwater exhaust simulation

Since the primary objective of this research program was the quantitative evaluation of 2-stroke engine air pollutants an underwater exhaust simulation was built to facilitate conditioning and collection of a realistic sample. In this system the hot exhaust sample was bubbled through a controlled flow of water and the "measurement" sample collected above the water. The assumptions and calculations involved in this development are given in Appendix B. A schematic diagram of the simulation device is shown in Fig. 7 and the installation is shown next to the engine test stand in Fig. 8. A viton-diaphragm Air Shield pump was used to force the sample into the device at the same rate the "measurement" sample was being drawn to the analyzer.

A wide range of water flow rate to exhaust gas flow rate ratios ($\dot{M}_{H_2O}/\dot{M}_{\text{exhaust}}$) was investigated to obtain a qualitative measure of the exhaust pollutant absorption by the water. It was found that the unburned hydrocarbons in the "bubbled" exhaust were not measurably affected by increasing the simulator water flow rate relative to the exhaust flow.

It had been planned to investigate the effects of various types of water in the simulation including pure water and typical lake water. Unfortunately, the logistics problem of providing a reasonable flow rate of these water types was a great problem with the present test facility.

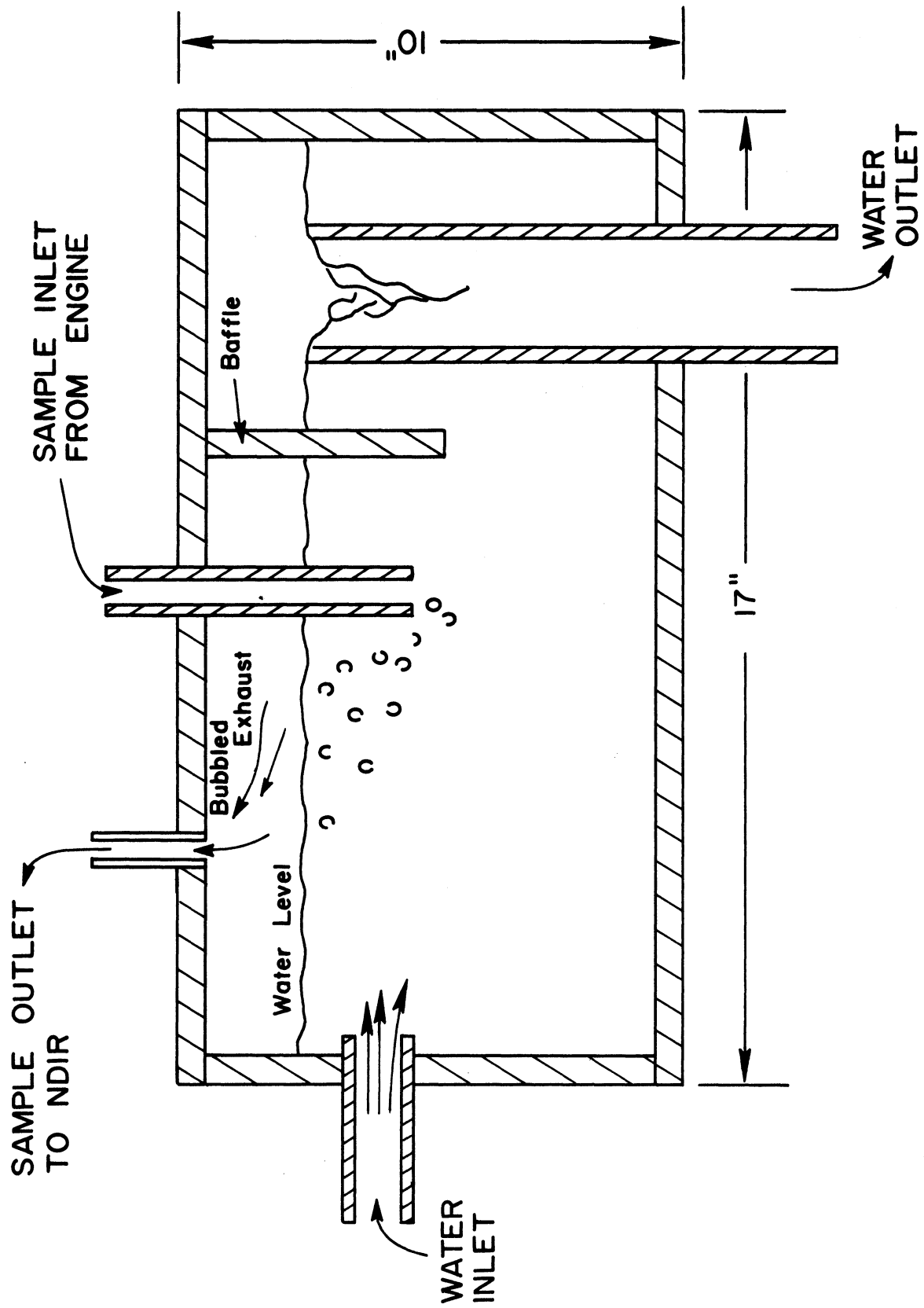


Fig. 7. Schematic diagram of the underwater exhaust simulation system.

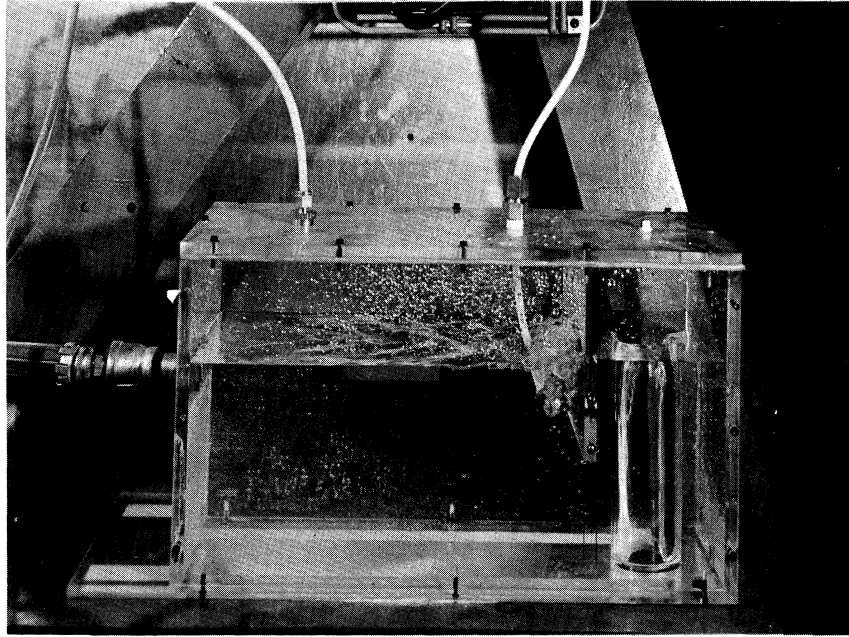


Fig. 8. Underwater exhaust simulation system.

A very important requirement of the simulation and exhaust sampling system, which was operated at a slight vacuum, was that it be leak free, i.e., it had to be sealed from the ambient to prevent dilution of the exhaust gas sample.

We checked this periodically by introducing a standard calibration gas of known hydrocarbon concentration into the sample system near the engine. A leak could be quickly detected if the analyzer showed a decrease in the reference gas concentration.

b. Sample condensation system

The second sample preparation system was designed to collect, in one location, the high boiling point compounds (heavy hydrocarbons and most of the water). Due to internal temperature limitations within the NDIR analyzer, these compounds would not be detected even if they could be sampled properly. They would be trapped or "hung-up" in various segments of the total system. It was hoped that the condensate as well as the gas could be quantitatively analyzed for its effect on environmental pollution.

The condenser consisted of a coil of glass in a water jacket. Water at approximately 36°F was circulated from the ice bath on the NDIR analyzer cart to the water jacket. The condensate was trapped in a test tube at the condenser outlet. To prevent any premature condensation before the exhaust sample reached the condenser, resistance heating tape was wrapped around the sample line and the temperature controlled at 600°-700°F. The condenser and associated equipment is shown in the photograph of Fig. 9. The device in the upper right corner of Fig. 9 is a pump used to circulate the cooling water.

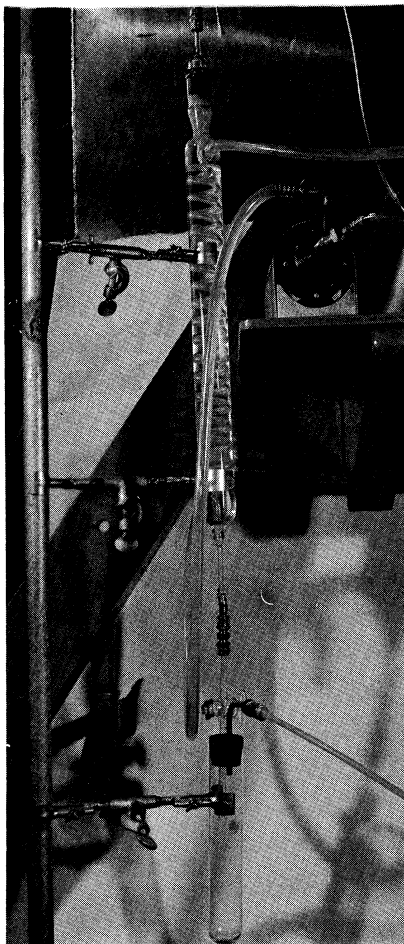


Fig. 9. Condenser exhaust sample conditioning system.

C. EXHAUST GAS ANALYSIS

Two constituents in the exhaust gases were analyzed in this study.

a. Unburned hydrocarbons; this constituent was the primary pollutant considered.

b. Oxygen; this component, while not a pollutant, was believed to be strongly related to the engine trapping efficiency and therefore related to an important internal engine source of unburned hydrocarbons. Only the analysis device will be discussed here, the details of the correlation with trapping efficiency will be discussed later.

1. Exhaust Gas Unburned Hydrocarbons

The primary instrument in the hydrocarbon analysis system was a nondispersive infrared (NDIR) analyzer which automatically and continuously determines the concentration of a particular component of interest in a complex gas mix-

ture. The measurement is based on a differential measurement of the absorption of infrared energy. In our study n-hexane, C_6H_{14} , was selected as the hydrocarbon component* of interest because it is specified in current government standards as being representative of the average composition of the unburned hydrocarbons in the exhaust gas from an automotive engine.**

The total analysis system is shown schematically in Fig. 10 and installed in a bench in Fig. 11. The normal path of the exhaust gas sample is shown by the heavy black line in Fig. 10. A viton-diaphragm, Air Shield pump was used to draw the sample from the sample system into an ice bath which was used to condense most of the water vapor. The condensation step was necessary because the NDIR analyzer is responsive to water and would indicate an erroneous hydrocarbon concentration if the water vapor were not removed. Following the trap, a 2-in. paper element filter made by Gelman Instrument Company removed particulate matter from the sample. The flow rate through the system was regulated by a Hoke needle valve and measured with a Fischer and Porter rotameter. A Beckman model 315 nondispersive infrared (NDIR) analyzer was used to measure the hydrocarbon concentration. Initially the Beckman instrument was equipped with a 5-1/4 in. sample cell which provided accurate hydrocarbon measurement in the range from 100 to 1500 ppm. Due to consistently high hydrocarbon readings in the early testing the 5-1/4 in. sample cell was replaced with a much less sensitive 2-1/2 in. cell which permitted accurate measurement of hydrocarbon concentration in the range from 500-15,000 ppm.

A stainless steel sample tank was incorporated in the system to collect an exhaust gas sample which could, if required, be analyzed by a Perkin-Elmer flame ionization analyzer available in the Combustion Laboratory of the Autom-

*It is not practical nor wise to use, as a reference gas, a mixture of unburned hydrocarbons because the composition of the exhaust is not constant.

**The unburned hydrocarbons in the 2-stroke engine exhaust should have a higher average molecular weight than in the 4-stroke engine exhaust because more raw fuel and lubricating oil are present. However, the average composition should be relatively close to n-hexane and should cause no serious problems in interpretation of the results. It is interesting to note that over 200 different species of hydrocarbons have been identified in the 4-stroke engine exhaust. These range from methane (CH_4) to molecules with 10-12 carbon atoms. Significantly more compounds should be present in the outboard exhaust. The various species are distributed primarily in the paraffin, olefin, diolefin, acetylene, cyclo-paraffins, and aromatic families. Unfortunately the NDIR analyzer, sensitized to n-hexane, does not respond linearly to the carbon atoms of all species of the various families because their infrared absorption spectra only partially overlap. Thus, it is found that the NDIR analyzer indicates only about one half of the carbon atoms actually present. This causes no serious problems in interpreting and using the data unless the distribution of the hydrocarbon carbon families change with changing test conditions.

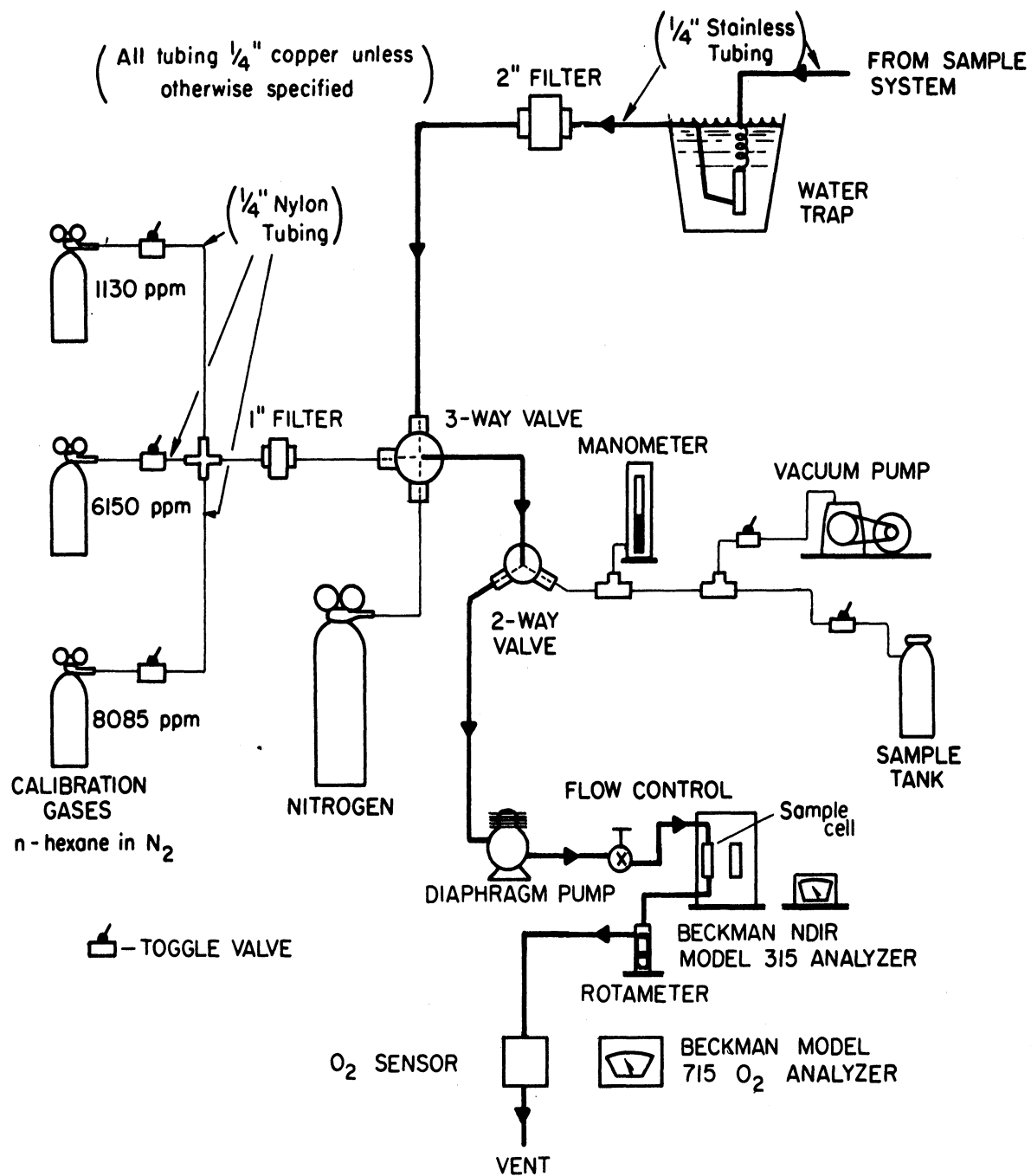


Fig. 10. Schematic diagram of the unburned hydrocarbon and oxygen analysis system.

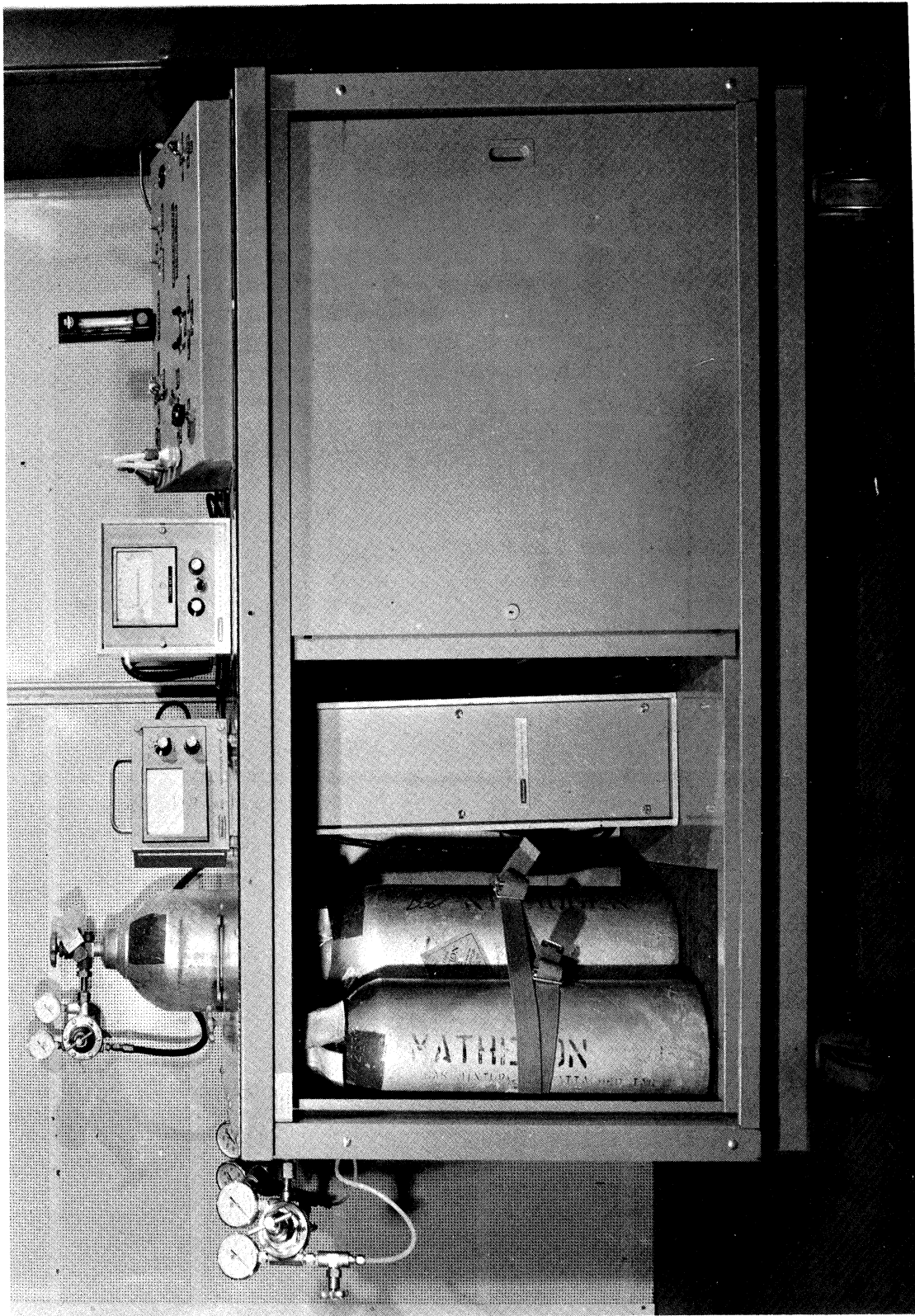


Fig. 11. Front view of the exhaust gas analysis system.

tive Laboratory. This was intended to provide a periodic check on the continuous sampling Beckman analyzer. The sampling tank and copper manifolding were evacuated by a Cenco vacuum pump prior to obtaining a sample.

To insure accurate analyses with the Beckman instrument periodic calibration was required. Four different concentrations of normal hexane (n-hexane) in nitrogen from 1000 to 8000 ppm were used as the standard reference gases. A sample calibration curve for the Beckman analyzer is shown in Appendix C.

Nitrogen was used to purge the system before and after exhaust sampling, and to establish a zero reference for the Beckman analyzer.

2. Exhaust Gas Oxygen Analysis

The oxygen concentration in the exhaust was measured with a Beckman model 715 amperometric oxygen analyzer.⁸ The transducer of the analyzer was placed at the outlet of the NDIR analyzer in a special fixture. This was permissible because the composition of the dry exhaust gas sample was unaltered during passage through prior equipment. The sensor location is shown in the schematic diagram of Fig. 10, and the analyzer amplifier can be seen in the photograph of Fig. 11 at the left of the NDIR analyzer amplifier. Both the amplifier and sensor are shown in Fig. 12.

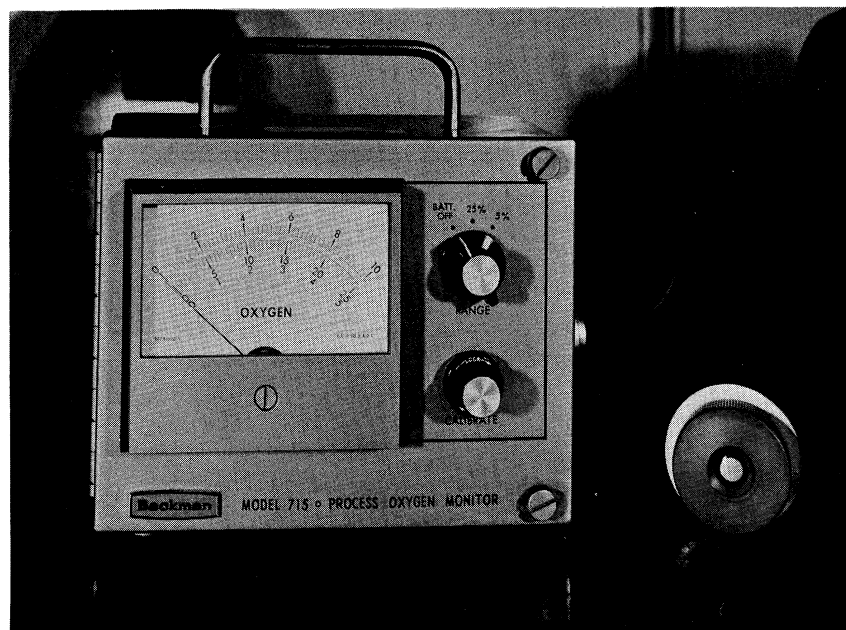


Fig. 12. Beckman model 715 process oxygen analyzer—amplifier and sensor.

Since the instrument has a linear response to oxygen concentration, calibration was accomplished merely by using pure nitrogen for the zero reference and air for the maximum scale reading.

D. ENGINE COOLING

It was found early in the test program that it was very difficult to control the engine water temperature because of the inability to provide precisely controlled low pressure water to the engine water pump with a conventional regulator. If the supply pressure was too low the engine overheated, and if too high the thermostat did not function properly and the engine ran too cold. Precise water pressure control was obtained by using a variable head water supply tank shown in Fig. 13. By adjusting the height of the plastic tank, any required pressure from 0 to 10 ft of water could be developed at the engine water inlet. In this study a constant pressure of 53 in. H₂O was used and permitted control of the engine water inlet temperature at 150°F.

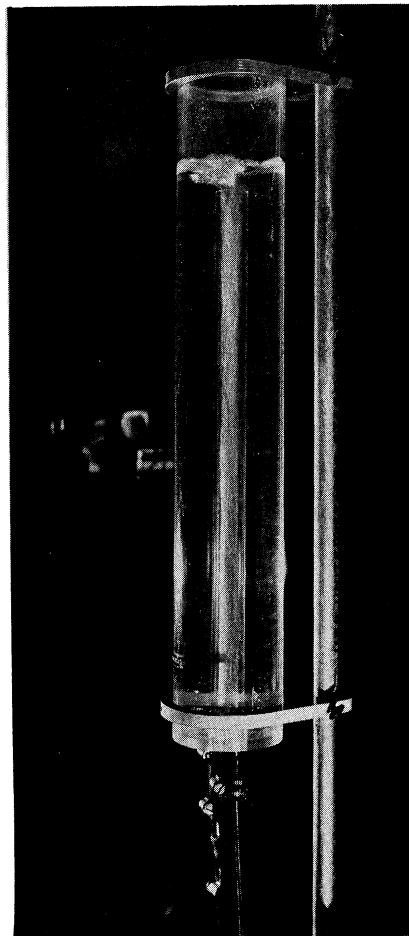


Fig. 13. Engine cooling water pressure control device.

E. TEMPERATURE MEASUREMENT

All temperatures of less than 1000°F were measured with copper-constantan thermocouples used in conjunction with a Brown continuous indicating potentiometer. The higher temperatures, those above 1000°F, were measured with the aid of chromel-alumel thermocouples also used in conjunction with a Brown potentiometer.

The important temperatures measured were:

1. The exhaust port and exhaust pipe temperatures; the locations of the temperature probes are shown in Fig. 5 where they enter the exhaust plenum cover.

2. Thermocouples were installed in the head of each cylinder. The heads were drilled through a solid metal region (not through the water jacket) to within 1/16 in. of the inner combustion chamber wall. The position of these in the cylinder heads is shown in Fig. 3.

3. Engine coolant temperatures (inlet and outlet) were measured in or near the thermostat body.

F. FLOW MEASUREMENT

1. Engine Air Flow

A number of techniques were used to measure the engine carburetor air-flow. In the early part of the program a Meriam model 50 MC2-6PF laminar flow element was used. The flow measurement data was corroborated with a rounded approach orifice system used in conjunction with a high volume centrifugal air pump. It was believed that little error in fuel metering and scavenging was introduced by the slight pressure drop at the carburetor inlet caused by the laminar flow element (1 in. of water at 4000 rpm, boat load). Problems developed in the latter stages of testing when the air cleaner of the laminar flow device became contaminated with engine oil as a result of blowback through the carburetor. This did not, however, cause a significant change in the air flow or carburetor inlet pressure.

Circumstances in recent testing suggested that the engine air-flow should be measured for each cylinder individually as part of the determination of individual cylinder air/fuel ratio. To accomplish this a special fixture was designed and built which plumbed an air cart with a rounded approach orifice measuring element to the carburetor inlet. The carburetor inlet for the proper cylinder was adjusted to atmospheric pressure and thus was balanced with inlets of the three other cylinders.

Currently a critical flow, air flow meter is being built to replace the rounded orifice air cart. It is believed that this device will provide more accurate and precise control of the individual cylinder air flow and will occupy much less space.

2. Fuel Flow

a. Overall fuel rate

Overall fuel-flow to the engine was measured using a weight scale and timer system. This device is shown in Fig. 14.

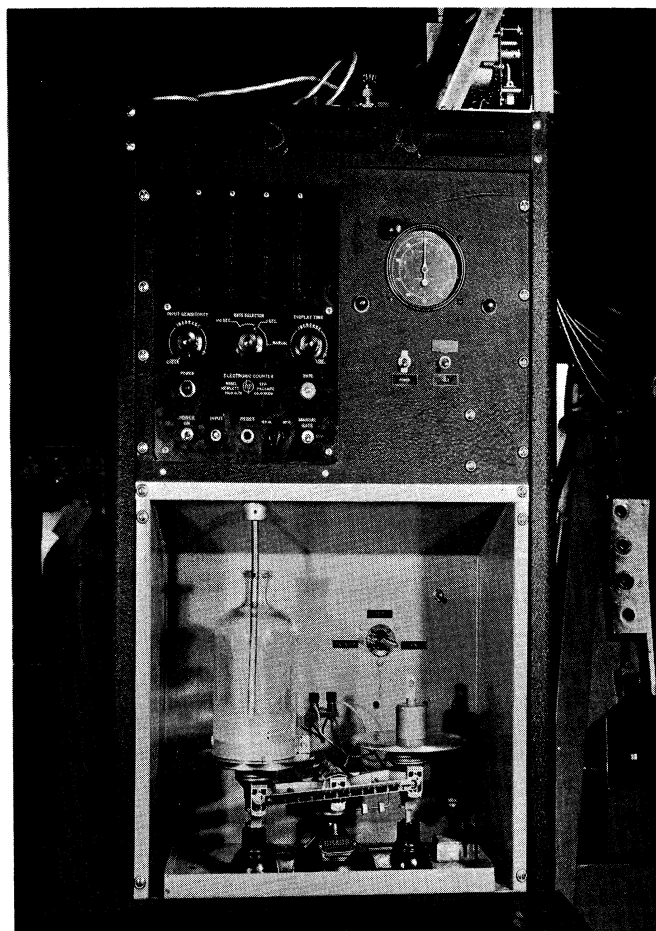


Fig. 14. Total engine fuel rate measurement system.

b. Individual cylinder fuel rate

Fuel rate to the individual cylinders was determined with the aid of a gravity feed volumetric burette device. During a fuel rate test the gasoline-oil mixture was directed to the individual float bowls of each carburetor barrel from the measurement burettes. An electronic timer interconnected with solenoid valves permitted accurate determination of the fuel test duration. The system is shown in the photograph of Fig. 15 and schematically in Fig. 16.

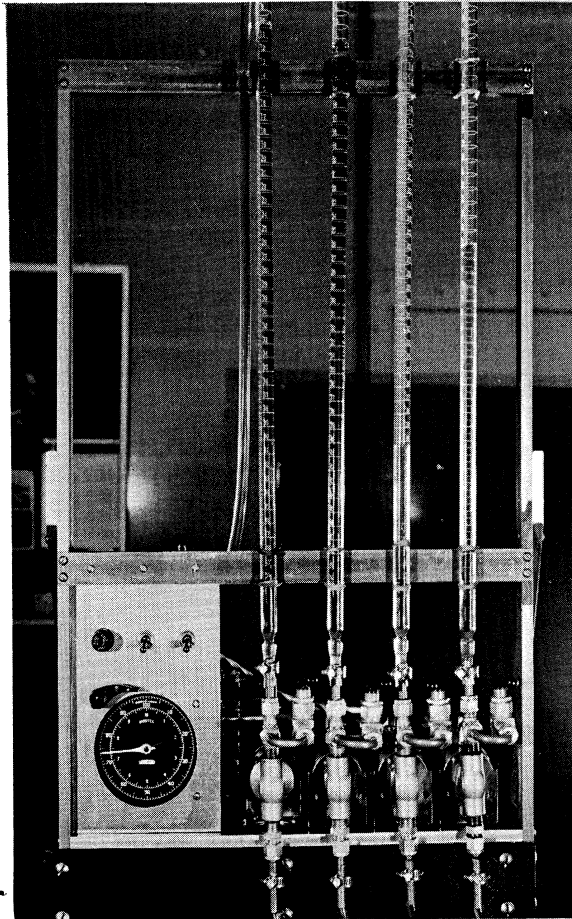


Fig. 15. Individual cylinder fuel rate measurement device.

G. FUEL/AIR RATIO CONTROL

The fuel/air ratio for the base line testing was obtained from the standard carburetor using .060 in. fixed orifice jets in the main metering system with the idle system adjusted to factory specifications.

For part of the test program, it was necessary to vary the fuel/air ratio. At moderate and high loads this was accomplished by controlling the air pressure in the float bowls. Some problems, however, were noticed with this technique because of the numerous air bleed holes. The best solution to this problem may be the use of lean limit jets with positive float bowl pressure to richen the mixture.

The low speed and load fuel/air ratio was varied by controlling the needle valves in the standard idle system.

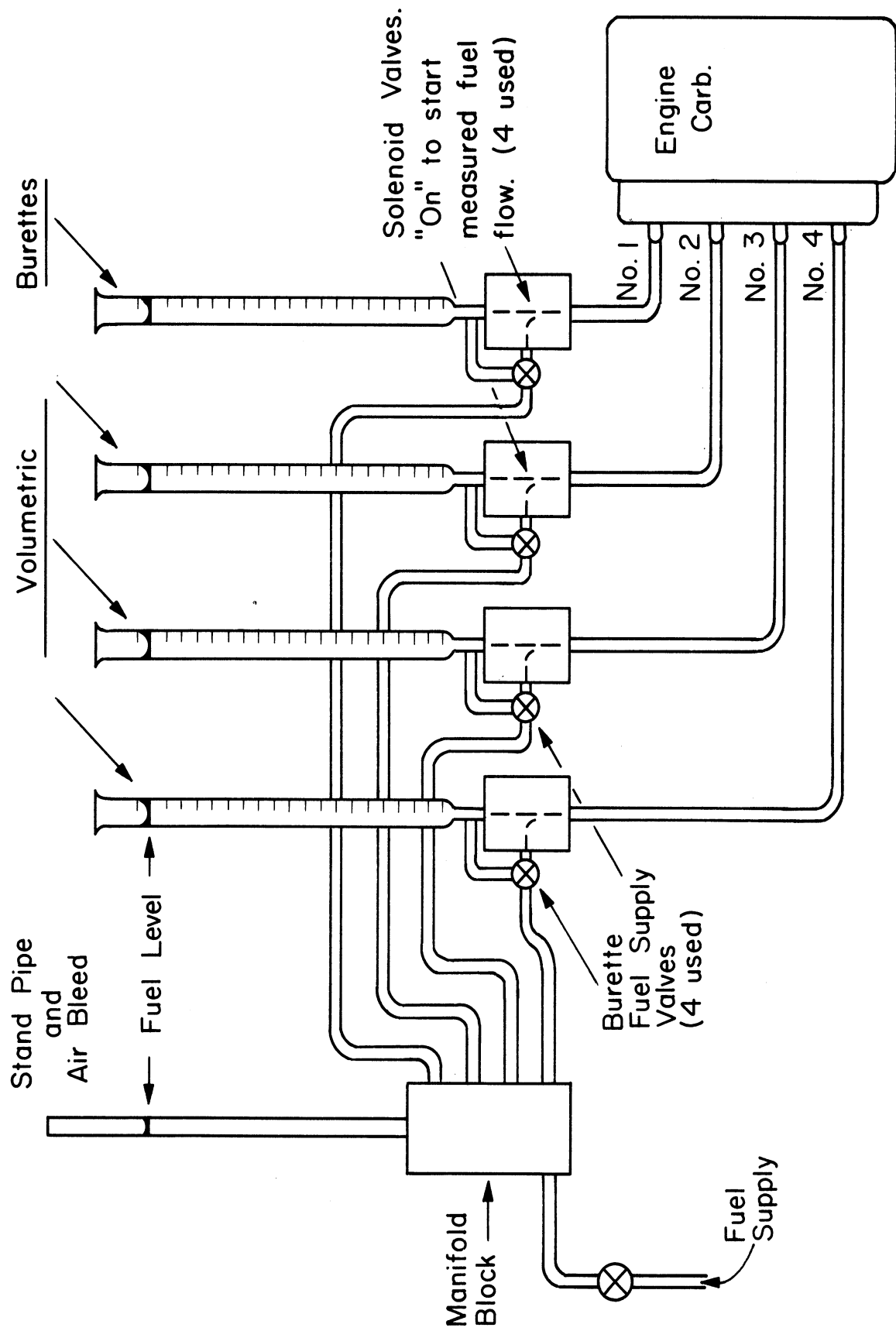


Fig. 16. Schematic diagram of the individual cylinder fuel measurement system.

VI. RESULTS AND ANALYSIS

A. BASE LINE HYDROCARBON EMISSION TESTS

1. Concentration Emissions

It was expected that the base line testing would occupy only a small portion of the total test program. However, it soon became apparent that this idea was much too optimistic because of the inability to obtain reproducible data. The base line exhaust gas (unburned hydrocarbon) data is shown in Figs. 17 a-d, as a band for the samples taken from each cylinder exhaust port and the exhaust pipe at the various boat load conditions. It is obvious, particularly at light loads, that the results exhibited a large standard deviation. This is a most unfortunate circumstance when trying to determine the effect of certain operating variables on emissions because one cannot be sure if a perturbation of the data is caused by random variation or the change in a selected parameter.

At 1000 rpm boat load the data showed the least reproducibility and highest variance both from test to test and from cylinder to cylinder. Cylinder no. 1 was significantly worse than the others (mean hydrocarbon emission level of 7000 ppm compared to an average mean port emission level of about 5800 ppm) both with respect to the emission level and variance. It was determined later in the program when the capability of measuring individual cylinder fuel rates was developed that cylinder no. 1 was operating with an excessively lean mixture which caused misfiring and hence a very high emission level. This imbalance in mixture ratio distribution was caused by improper adjustment of the idle system needle valves in the control rack. The relationship between engine misfiring and hydrocarbon emissions will be discussed in detail later in the report.

It should also be noted that the emissions from the other cylinder in the 1-3 cylinder bank had a higher than average (of the four cylinders) mean emission level which may be partially due to some "cross talk" between the cylinders, i.e., sample probe no. 3 may have collected some exhaust gas from port no. 1 as well as its own. It was found in a separate study that cylinder port samples from cylinders nos. 3 and 4 were slightly affected by the exhaust from cylinders nos. 1 and 2, respectively. However, the reverse was not true. Apparently this is related to the geometry of the engine, the flow path to the exhaust outlet and the design of the sample probes.

It was also observed that the mean exhaust pipe emission level was less than the average mean emission level from the four cylinders, the difference increasing with speed and load. This is understandable in light of the fact that the exhaust gases were exposed to high temperatures for a longer period of time at the pipe sample probe than at the cylinder ports at any given speed. The

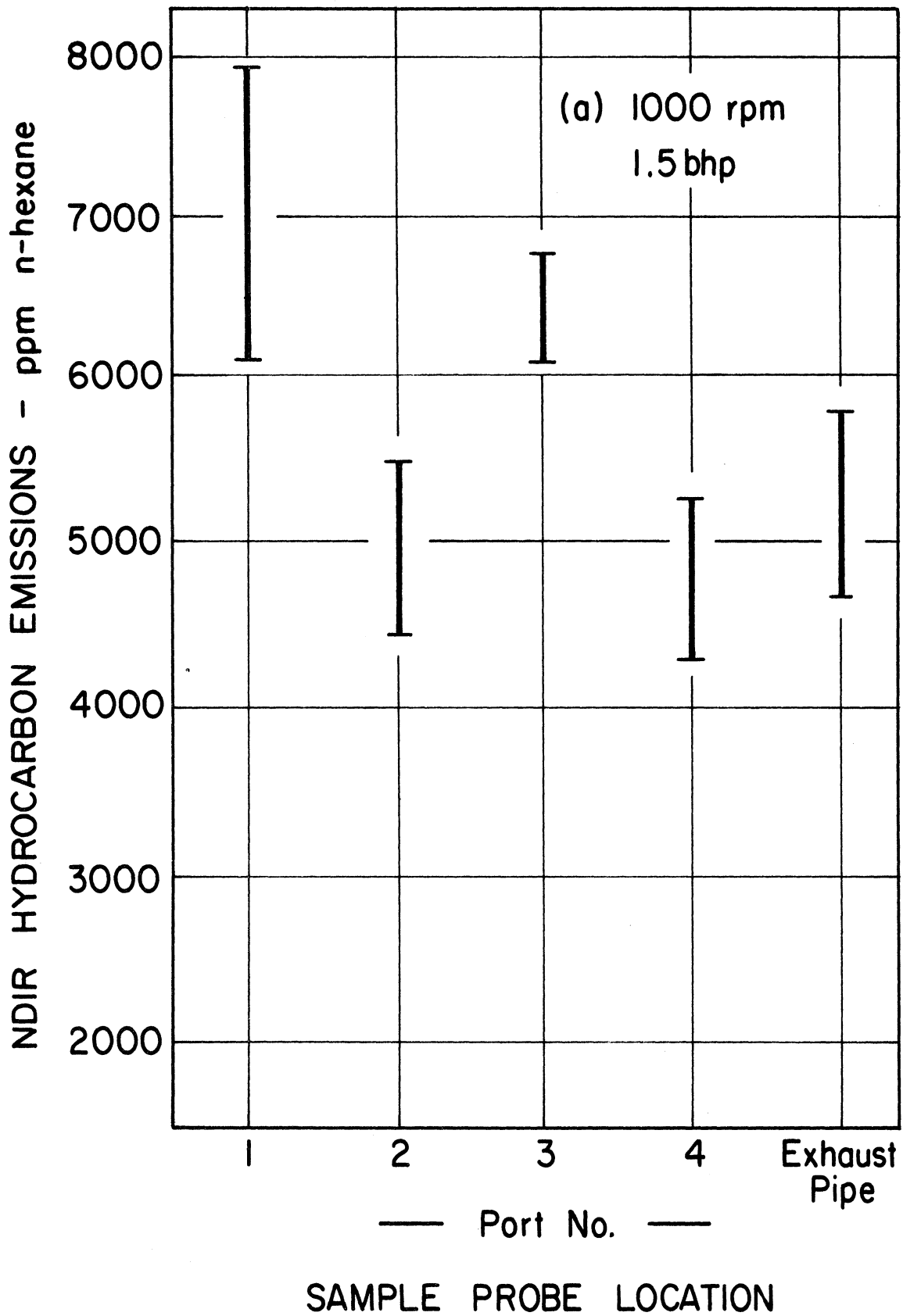


Fig. 17. Base line hydrocarbon concentration emission, ppm n-hexane, boat load.

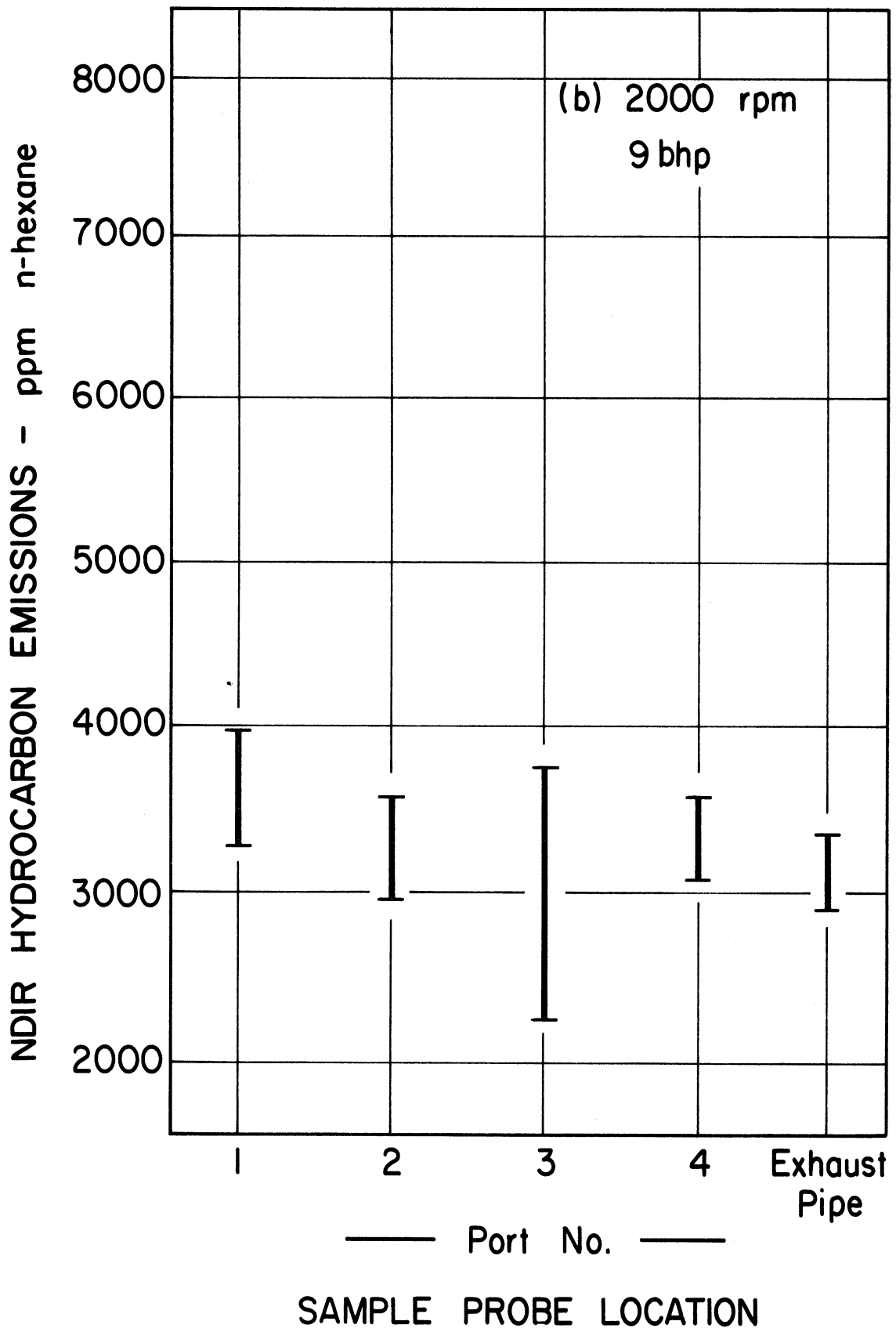


Fig. 17. (Continued).

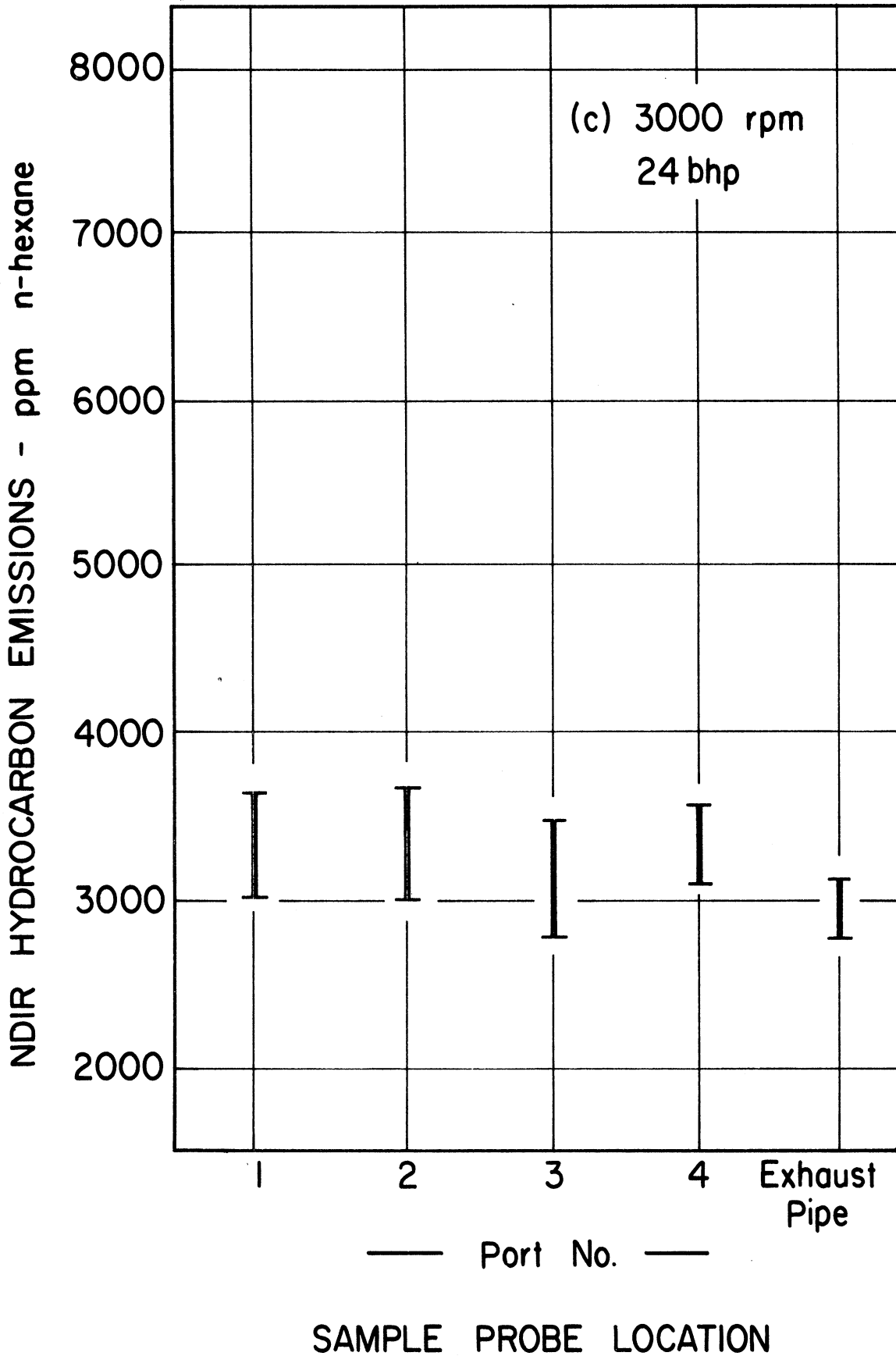


Fig. 17. (Continued).

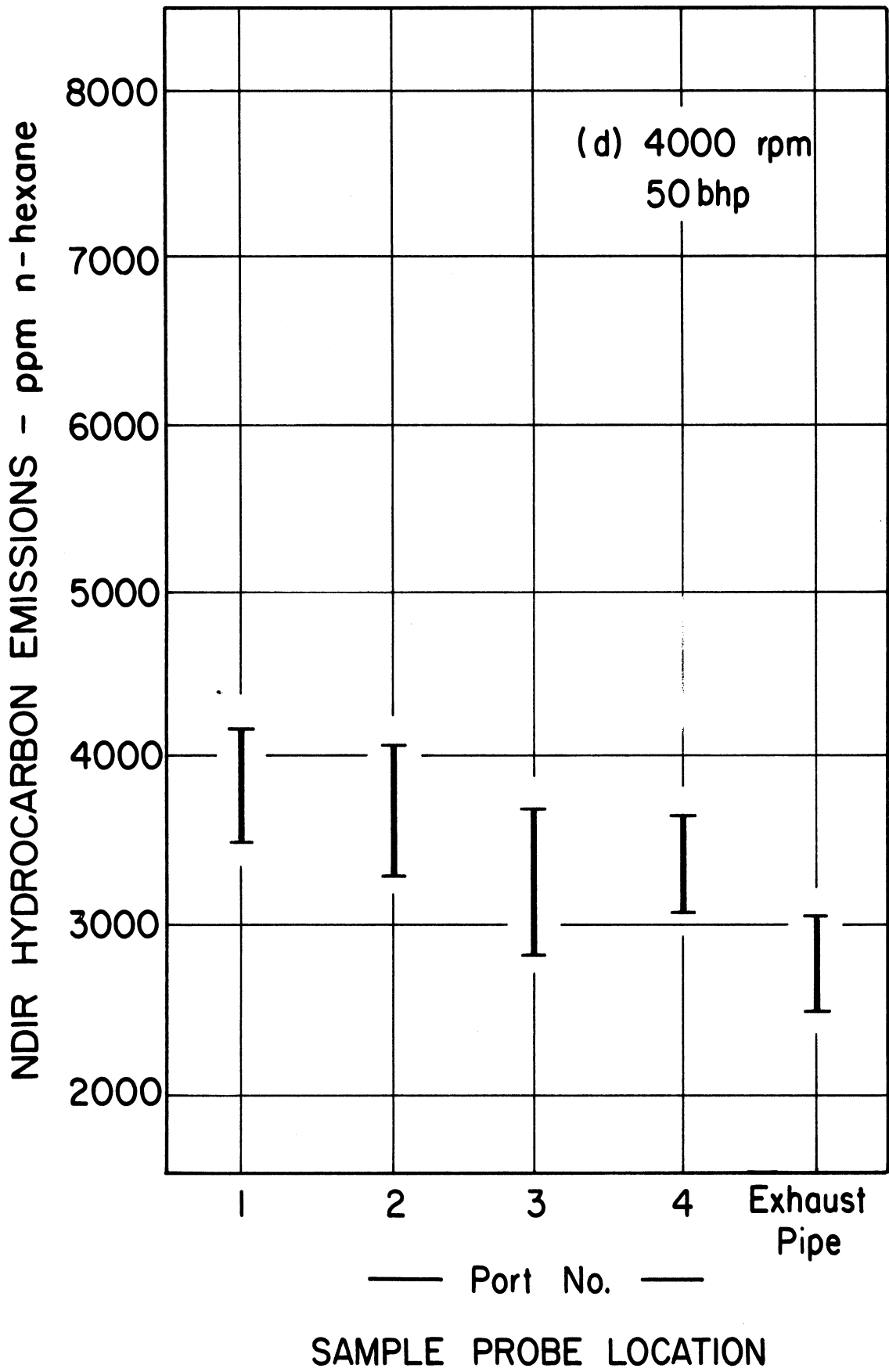


Fig. 17. (Concluded).

high temperature apparently causes a partial oxidation reaction of the unburned hydrocarbons in the exhaust plenum with the concentration of the hydrocarbons decreasing in proportion to length of time in the high temperature environment. A number of other factors affect this "after cylinder" reaction and will be discussed in a later section.

At 2000 rpm boat load the average hydrocarbon emissions were significantly lower than at 1000 rpm due undoubtedly to more efficient scavenging and decreased misfiring frequency. It is interesting to note that the variance in the port no. 3 data was greater than the others. Some of the port no. 3 data was significantly less than the average for some unexplained reason which may be of great importance when considering design for minimum emissions.

At 3000 and 4000 rpm boat load the concentration emissions were approximately the same as at 2000 rpm. The 4000 rpm emissions, however, were slightly higher from ports nos. 1 and 2 but slightly lower from ports nos. 3 and 4 and significantly lower in the exhaust pipe. No explanation is obvious for the variance in port data other than that there may be cylinder to cylinder variations in trapping efficiency, misfiring frequency or fuel-air ratio. The diminished pipe emission level at 4000 rpm was most probably caused by a more significant after reaction in the exhaust plenum as a result of a higher average plenum temperature. .

Base line data at 5000 rpm boat load is incomplete at the present time. Qualitatively, though, the data shows about the same concentration as at 4000 rpm.

2. Base Line Emissions Converted to a Mass Base

Although the concentration emissions stated previously are meaningful when considering the specific performance of a given engine it is more significant to consider the total mass rate of emissions when judging the engines contribution to air and water pollution. The base line mass hydrocarbon emissions for the various speeds and loads are shown in Figs. 18 a-d from the four exhaust ports and the exhaust pipe. These results are determined according to the technique outlined in Appendix D, which utilizes both the concentration hydrocarbon emissions and the total air and fuel flow rate delivered to the engine. Note that the mass emissions increase very rapidly with speed and load.

Figure 19 shows the mean average (of the four cylinder ports) and mean exhaust pipe mass rate of hydrocarbon emissions as a function of the boat load test conditions. As a basis for comparison mass hydrocarbon emission levels from a typical automotive engine at the same operating conditions are also plotted in Fig. 19 for the same engine speeds and loads. This plot focuses attention on the seriousness of the problem facing the 2-stroke gasoline engine. Note particularly the extremely high emission levels at the higher speeds and loads.

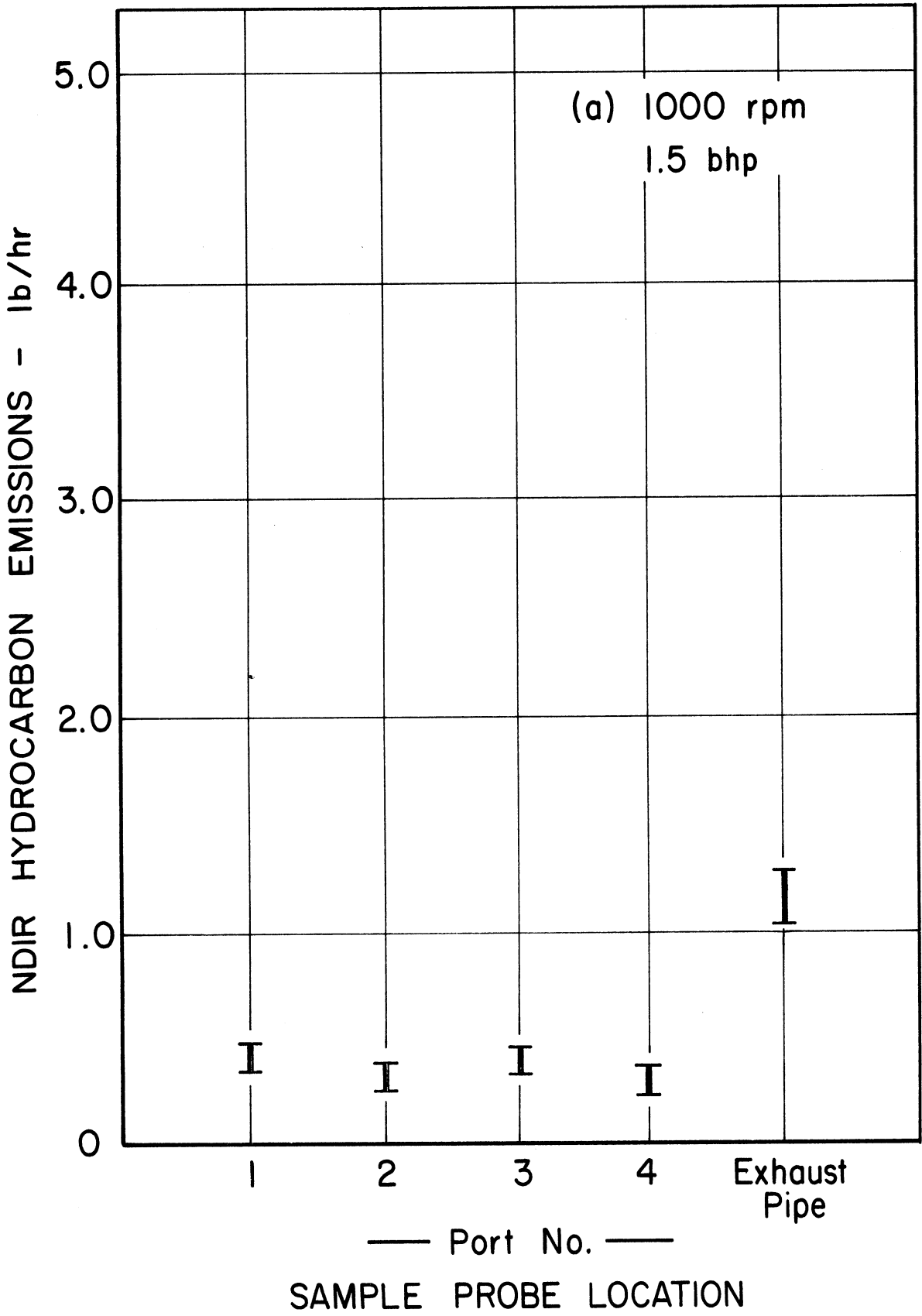


Fig. 18. Base line hydrocarbon mass emissions, lb/hr, boat load.

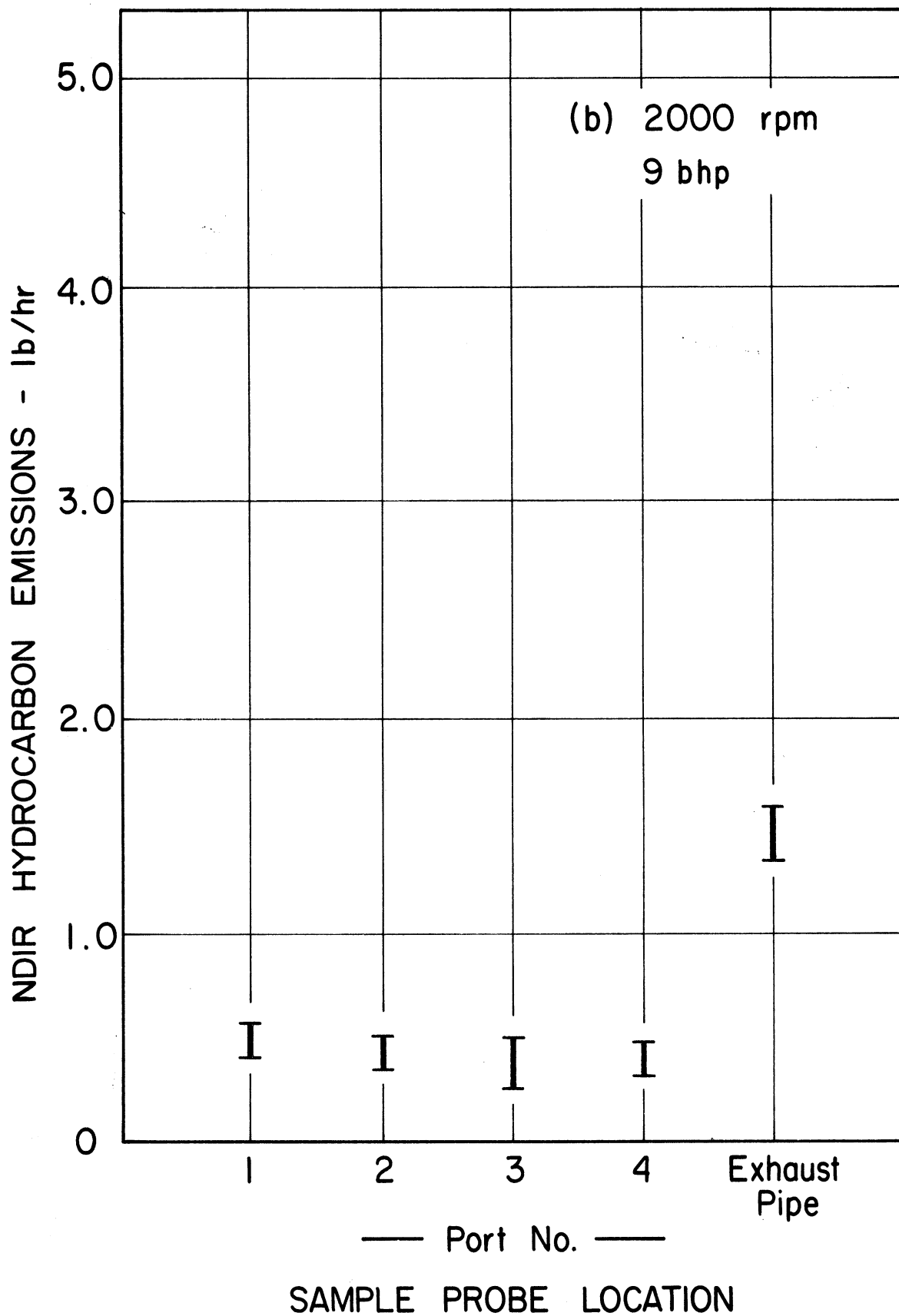


Fig. 18. (Continued).

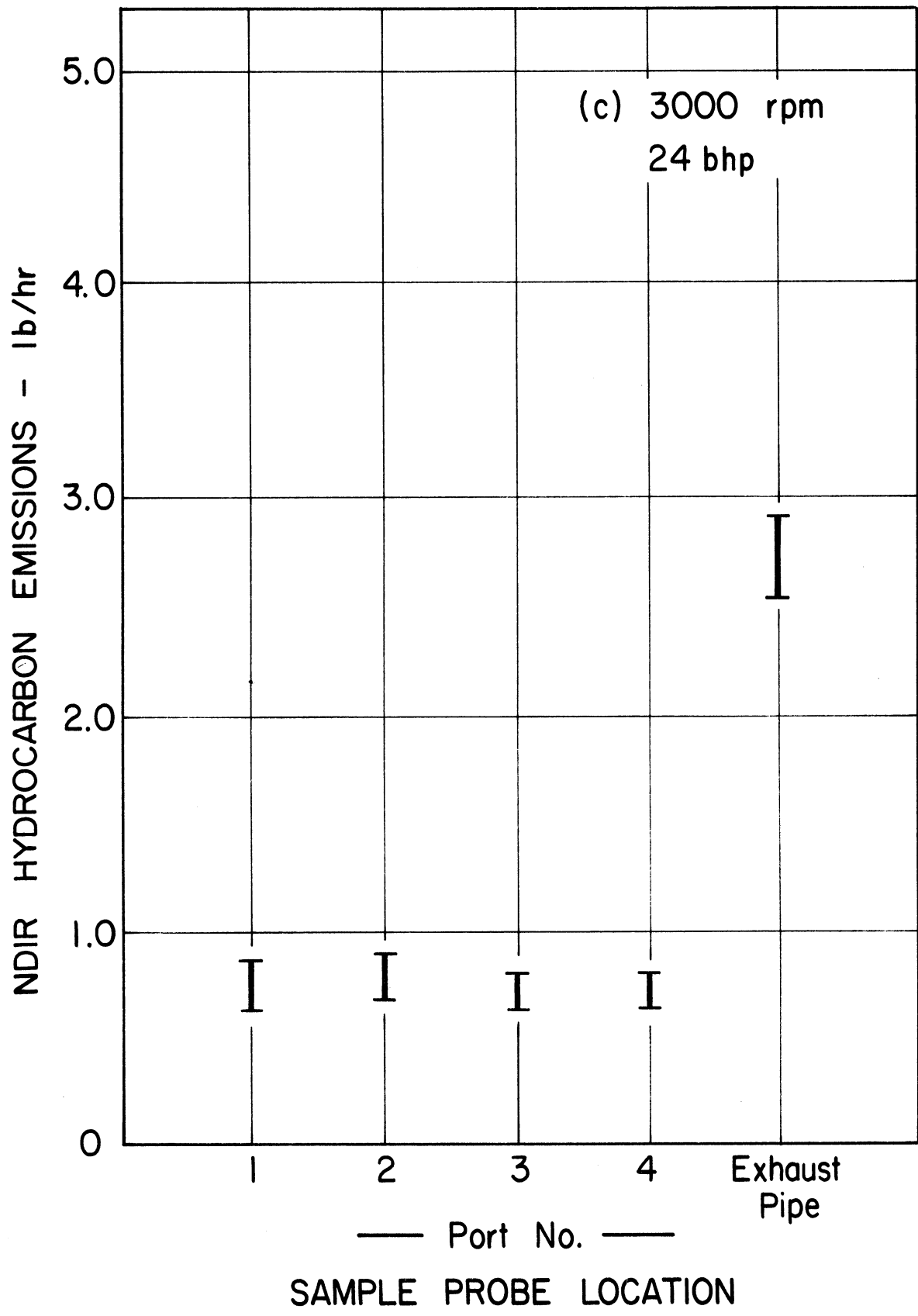


Fig. 18. (Continued).

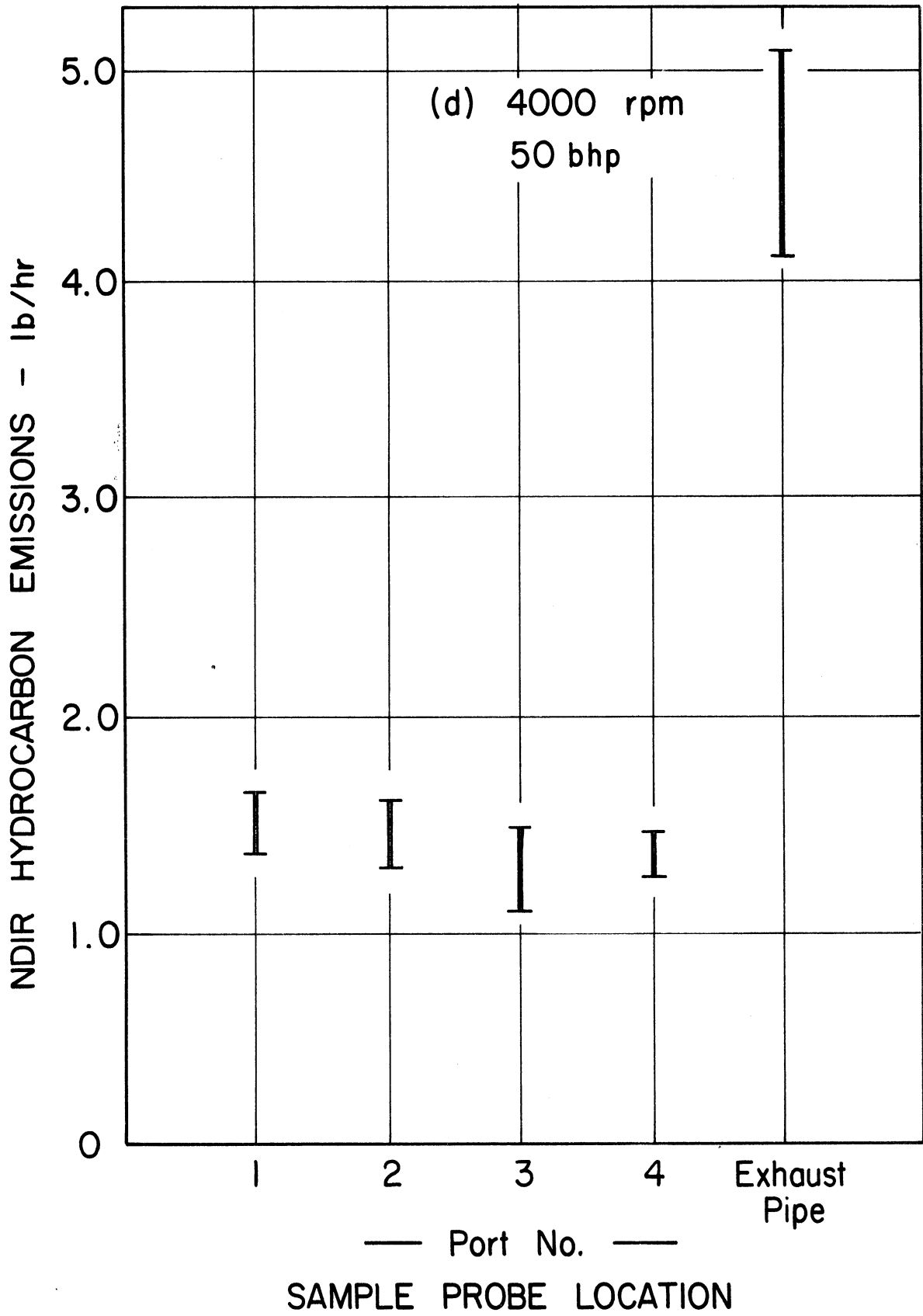


Fig. 18. (Concluded).

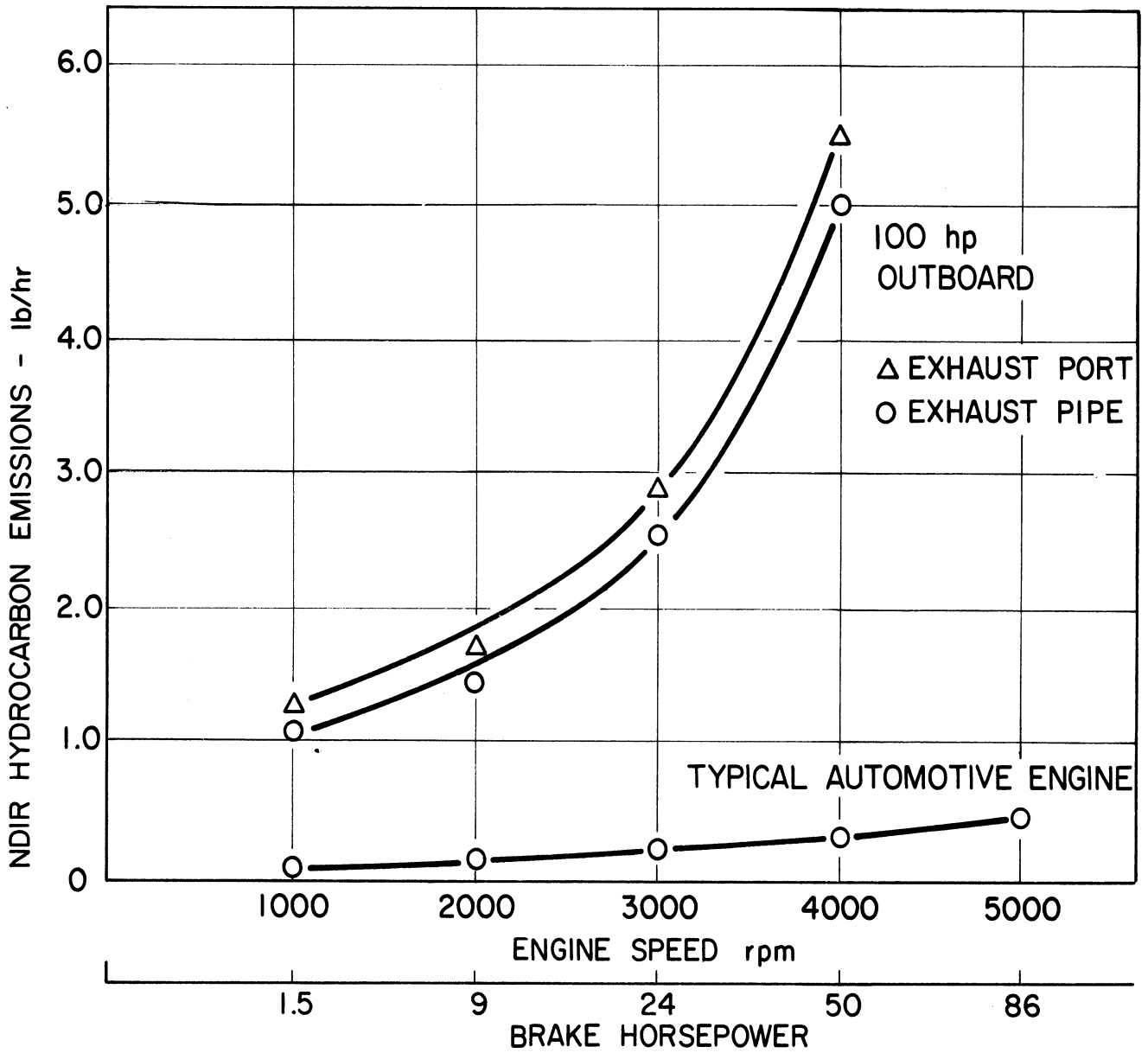


Fig. 19. Average mass hydrocarbon emission level from the exhaust ports and exhaust pipe.

B. COMPARISON OF EMISSIONS MEASURED WITH THE DIFFERENT SAMPLE COLLECTION AND CONDITIONING SYSTEMS

It is interesting to note that the hydrocarbon emissions measured above the water in the underwater exhaust simulator and at the outlet of the condenser system were essentially the same at the same test condition. This says, in effect, that the high boiling point compounds (including most of the water, some of the unburned gasoline and lubricating oil) which are trapped in the condenser are likely the same constituents "washed out" of the bubbled exhaust gas. Since the combustion products, with the exception of water, consist of primarily low boiling point constituents, the condensate from the condenser system or the material trapped by the water in the simulator is related directly to "through scavenging" of the unburned charge. Thus the contribution of the unburned hydrocarbons to water pollution can conceivably be evaluated by measuring the difference between the mass of fuel and oil which is scavenged and the mass of unburned hydrocarbons which pass through the two sample conditioning systems into the analyzer. It should also be mentioned that the "quench" exhaust port probe used in part of our work acted as condenser and precipitated the high boiling point compounds out of the sample. Within the range of accuracy obtainable with the analytical system it does not appear to make much difference (with respect to air pollution measurements) which sample system is used. It was also observed in the underwater exhaust simulator that as the ratio of water flow rate to exhaust flow rate was increased there was only a slight decrease in the measured emissions. In fact, the change was much smaller than the variance observed from test to test at any given condition. A detailed analysis of the underwater exhaust simulation device is presented in Appendix B.

C. EFFECT OF AIR/FUEL RATIO ON HYDROCARBON EMISSIONS

Experience with 4-stroke gasoline engines suggested that air/fuel ratio would be one of the most important engine parameters effecting the hydrocarbon emissions. The early testing of the outboard engine certainly proved the validity of this assumption. In fact, it may be the most critical variable for several reasons:

- . The unburned hydrocarbon content of the mixture and therefore of the "through scavenged" charge is a direct function of air/fuel ratio.
- . Misfiring frequency is strongly dependent on air/fuel ratio.
- . The concentration of unburned hydrocarbons in the combustion products is related to mixture ratio.

Limited total engine air/fuel data has been obtained in the test program to date because of the base line reproducibility problems. However, the emissions from cylinder no. 1 were measured and plotted as a function

of air/fuel ratio. Unfortunately the overall engine air/fuel ratio was measured instead of the more meaningful individual cylinder mixture ratio. This deficiency was rectified with the recent development of an individual cylinder fuel metering device. The air/fuel ratio error could result in a slight shift of the curves to the right or left. Another error was associated with the poor test-to-test reproducibility which could result in a significant variance of the measured emissions. By taking data over a relatively brief period of time it was hoped that this error could be minimized and that the functional relationship of air/fuel ratio to the exhaust emissions could be determined.

Sample results from exhaust port no. 1 are shown in Fig. 20 and demonstrate the significant effect of air/fuel ratio on the hydrocarbon emissions. In this test the engine load was continuously regulated by throttle adjustment to the boat load for a given speed as the air/fuel ratio was changed. Thus, for example, at mixture ratios leaner than best power as the air/fuel ratio was increased, the brake load would normally decrease unless the throttle were opened to increase the mass flow of charge to the engine. This emphasizes the importance of basing final judgment on the mass rate of emissions rather than purely a concentration base, exactly the type of standard the federal government will be applying to the automotive industry in 1970. At 3000 and 4000 rpm the minimum emissions occurred at a slightly rich mixture ratio ($\approx 14/1$). With mixture enrichment the emissions increased primarily due to an increase in the hydrocarbon content of the scavenged, unburned charge. As the mixture was leaned, unburned hydrocarbons increased due to poorer combustion (partial misfiring). It was interesting to note that as in the base line test the 3000 rpm results were generally lower than those at 4000 rpm suggesting that the scavenging is most efficient near the 3000 rpm test point in cylinder no. 1.

At 1000 and 2000 rpm the data initially made little sense. The emissions continuously decreased with a decrease in air/fuel ratio. An explanation was not obvious until the oxygen content of the cylinder no. 1 exhaust gas was measured and suggested that either the trapping efficiency of the engine was changing with air/fuel ratio or that the misfiring frequency was changing. It was concluded that the latter was the most probable cause because the scavenging dynamics should change only slightly. As the engine air/fuel ratio was decreased, the air/fuel ratio supplied to cylinder no. 1 also decreased with a consequent decrease in misfiring frequency. It was learned later that poor adjustment of the carburetor idle system caused the air/fuel ratio in cylinder no. 1 to be substantially leaner than the average of the four cylinders at light loads.

This data also focused attention on the necessity of measuring the misfiring frequency as well as the trapping efficiency.

Figure 21 is plot of 4 times the mass hydrocarbon emission rate from port no. 1 as a function of air/fuel ratio. This provides a comparison with the total engine emission level assuming each cylinder has the same hydrocarbon concentration as cylinder no. 1. In this plot the significance of the emission levels at the high speed boat-load conditions is again emphasized.

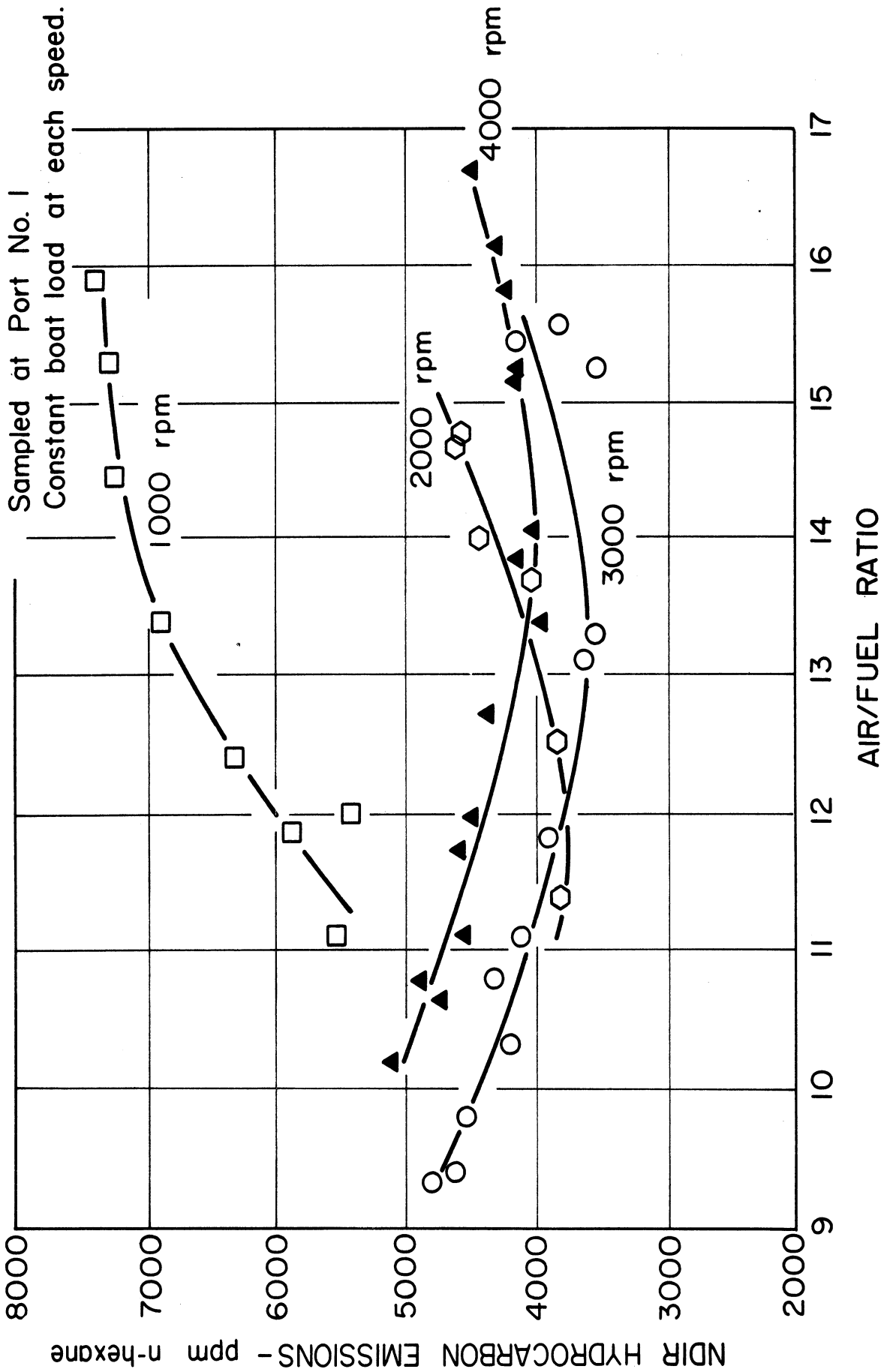


Fig. 20. Effect of air/fuel ratio on exhaust hydrocarbon concentration.

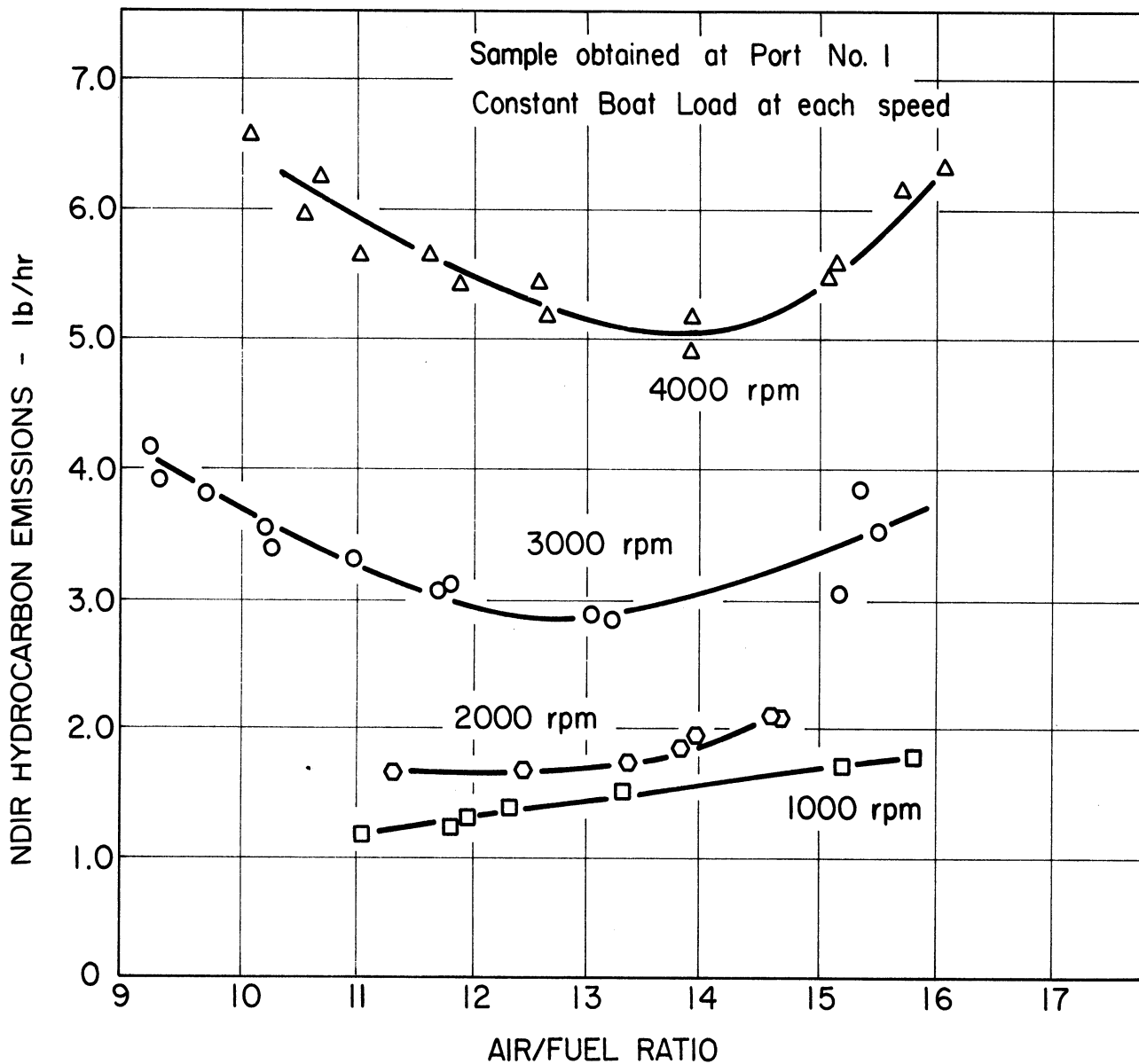


Fig. 21. Effect of air/fuel ratio on the exhaust hydrocarbon mass emissions, lb/hr.

D. DETERMINATION OF TRAPPING EFFICIENCY

It is generally conceded that the primary source of unburned hydrocarbon emissions from the 2-stroke engine is the air/fuel mixture that is short circuited through the engine during the scavenging process. Since previous techniques^{9,10} devised to quantify the effectiveness of the scavenging process have proved to be relatively inadequate, a new method was developed which used a tracer element in the intake charge. Conveniently, it was found that atmospheric oxygen in the inducted mixture appeared to meet the necessary requirements of a tracer.

- . Oxygen can easily and accurately be measured with new analyses techniques. In our study a Beckman Model 715 process oxygen analyzer was used.
- . The oxygen which reacts with the fuel during the combustion process loses its identity and is not measured in the oxygen analysis. Of course rich mixture ratios must be used to assure a minimum oxygen content in the combustion products. The products of combustion with an air/fuel ratio of 13/1 show only about (.3%) oxygen.
- . The oxygen which is scavenged through the engine is not measurably affected or altered by conditions within the engine. Generally the gasoline type hydrocarbons are quite nonreactive at the temperatures observed during scavenging. To prevent partial oxygenation in the higher temperature exhaust plenum a "quench" probe was used to collect the sample.

A complete discussion of the oxygen tracer technique is presented in Appendix E.

This technique for measuring the trapping efficiency is only valid if the engine is not misfiring. With misfiring neglected the engine trapping efficiency, Γ , which is defined as the ratio of mixture supplied to the engine to the mixture that is actually trapped in the cylinder, is given as follows:

$$\Gamma = (1 - \% O_2 \text{ in exhaust})$$

However, if the engine is misfiring the contribution of both the scavenged mixture and misfired charge to the total emission level is measured, i.e., the oxygen tracer technique really measures the percentage of the supplied mixture lost to the exhaust. Whether the source is related to scavenging or misfiring is immaterial with respect to environmental pollution. Conversely, the source distribution is vitally important to the designer. If the misfiring frequency were known, it would be possible to approximately determine the proportion of the total unburned hydrocarbons caused by scavenging and by misfiring and therefore to proportion the engineering design effort appropriately.

Sample oxygen data from the exhaust port of cylinder no. 1 is plotted in Fig. 22 as a function of air/fuel ratio. At the low speeds (1000 and 2000 rpm) the unusual trend of decreasing oxygen concentration with decreasing air/fuel ratio was observed. This result was totally unexpected since one would normally expect only small changes in trapping efficiency with changing air/fuel ratio. The only reasonable cause appeared to be engine misfiring. As the air/fuel ratio was richened, cylinder no. 1 fired more regularly with a resulting decrease in the charge entering the exhaust plenum. As mentioned previously, the data should be used with caution because the mixture ratio measured was the overall mixture ratio furnished to the engine which was richer than the mixture furnished to cylinder no. 1.

At the 3000 and 4000 rpm test points the results were much more understandable. The curves became flat below air/fuel ratios of about 13 to 1. The up-turn at leaner mixture ratios was caused by a combination of increasing oxygen content in the combustion products, increased misfiring and possibly small changes in the trapping efficiency.

The inverse of the oxygen concentration in the exhaust is a measure of the quantity of fuel which burns in the engine, trapping efficiency, and is also plotted in Fig. 22. Certainly the higher this number the less the hydrocarbon emissions should be which is exactly what is found when comparing Figs. 20 to 22.

It is interesting to note that the oxygen content is close to a minimum at 3000 rpm. Assuming that misfiring is not a problem at the higher speeds it appears that the engine trapping efficiency is a maximum near 3000 rpm.

Our oxygen results as with most other data is still incomplete. In the test program the major effort to date has been devoted to the development of meaningful measurement techniques.

E. TEMPERATURE EFFECT ON HYDROCARBON EMISSIONS

Limited data was taken with engine coolant or cylinder heat temperatures as controlled variables. However, it is interesting to look at the possible functional relationship of the unburned hydrocarbons with the cylinder head temperatures that were observed in the base line testing. Figure 23 shows this relationship at the various test points and relates the hydrocarbon emissions from a given cylinder to the head temperature in that cylinder. It is readily observed that the higher the head temperature the lower the emissions at a given speed. Of course it must be recognized that both the temperature and the emission level may be and probably are, directly related to variation in some other engine parameter such as mixture ratio or misfiring frequency, i.e., there may be only a limited cause and effect relationship between head temperature and unburned hydrocarbons.

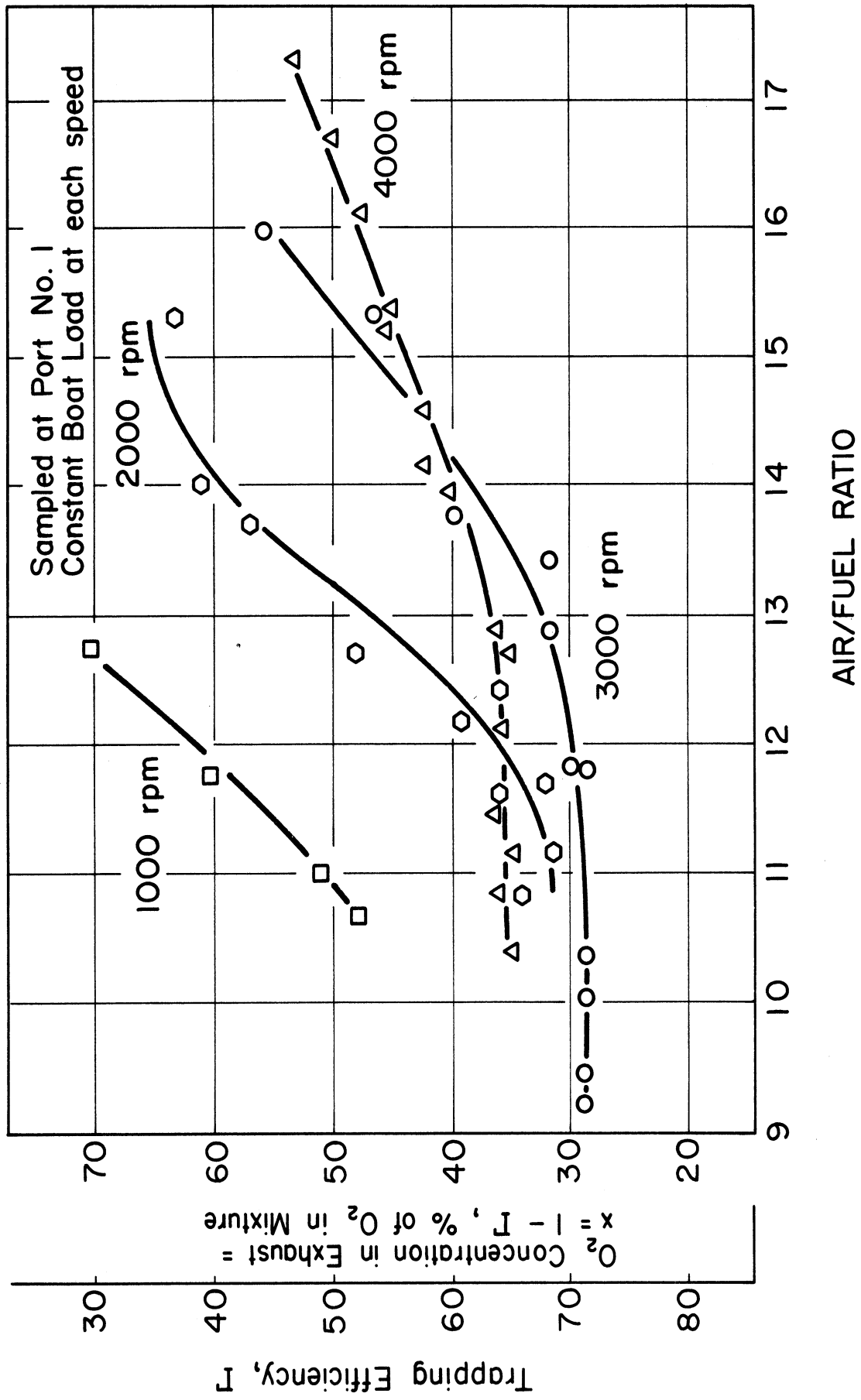


Fig. 22. Effect of air/fuel ratio on exhaust gas oxygen concentration and trapping efficiency.

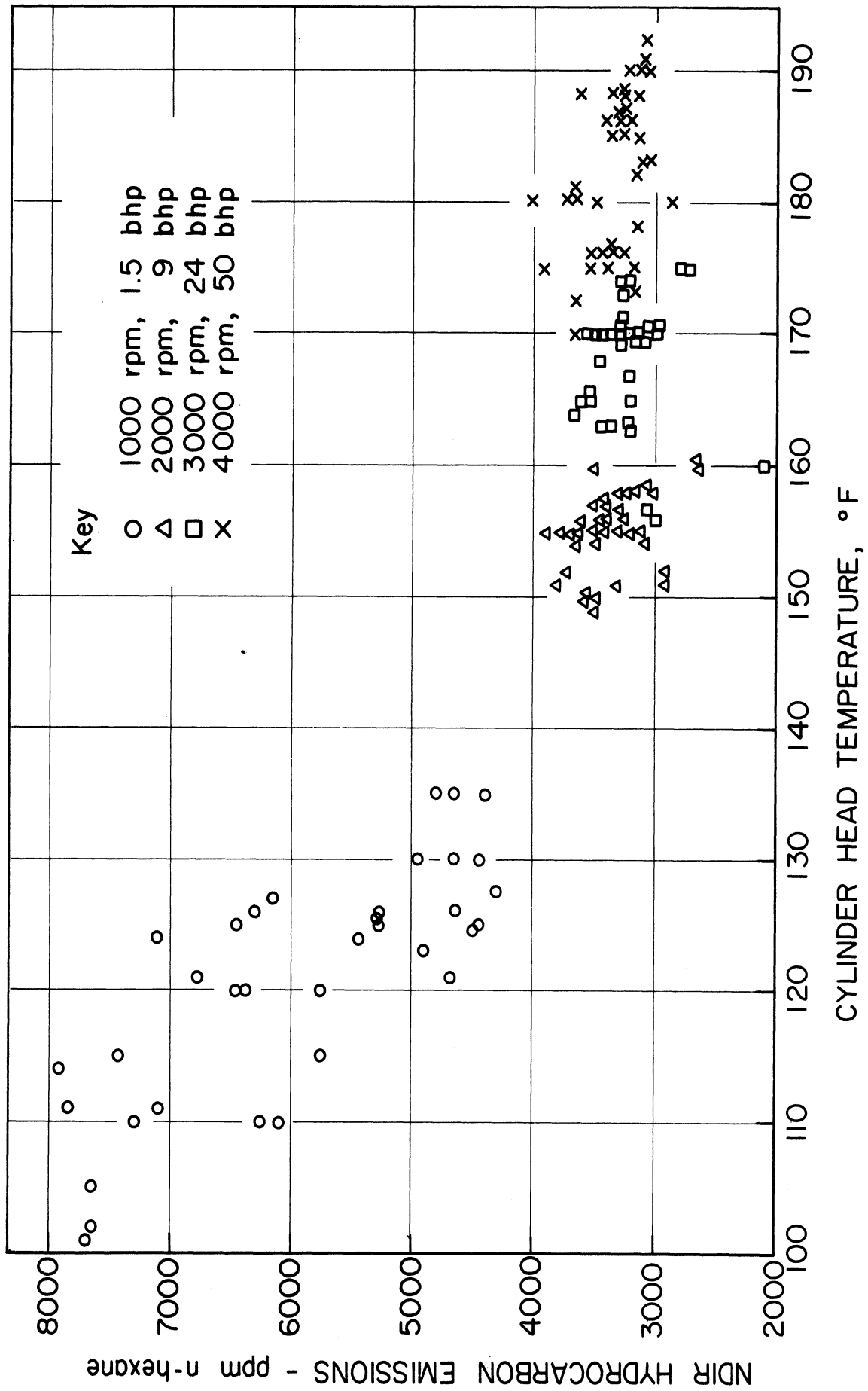


Fig. 23. Effect of cylinder head temperature on exhaust gas hydrocarbon concentration.

This same observation can be made about the apparent relationship between exhaust port temperature and hydrocarbon emissions measured at the individual ports, which is plotted from the base line data in Fig. 24. However, it is believed that there is a relationship between the exhaust pipe emissions and the exhaust temperature because of the possible "after engine" reaction in the exhaust plenum. This point will be discussed in a following section.

F. SOURCES OF THE UNBURNED HYDROCARBON EMISSIONS

There are three principal causes or sources of the unburned hydrocarbons in the outboard engine's exhaust gas. Perhaps these sources can be best illustrated with the aid of the schematic diagram shown in Fig. 25.

In the well tuned engine the scavenged air/fuel mixture is undoubtedly the most significant source and may account for 70-90% of the total hydrocarbon emissions. The unburned mixture has an extremely high concentration of unburned hydrocarbons, about 20,000 ppm when considered theoretically and is strongly a function of air/fuel ratio. The relationship between hydrocarbon concentration and air/fuel ratio is discussed in Appendix F. Note that the measured concentration is also a function of the type of instrumentation used. The contribution of the lubricating oil in the charge is relatively unimportant in comparison to the fuel because it constitutes a very small part of the air-fuel mixture.

In our experimental work the engine was run with one cylinder totally misfiring. An exhaust gas sample was taken from the misfiring cylinder and then conditioned in the condenser. Analysis of the sample with the NDIR technique indicated hydrocarbon emissions levels of about 8,000-10,000 ppm. Comparing this number with the theoretically expected NDIR analysis it is found that approximately 80% of the scavenged charge enters the atmosphere.

It is interesting to look at an example from the test data of the scavenged mixture contribution to the measured hydrocarbon emissions. At 3000 rpm boat load the hydrocarbon emissions measured at exhaust port no. 1 are approximately 3500 ppm. The oxygen data shows that approximately 30% of the mixture is scavenged through the engine. Since the NDIR analyzer indicates about 9000 ppm in the raw unburned mixture (measured from cylinder no. 1 with a shorted spark plug) the unburned hydrocarbon concentration due to scavenging is approximately $.30 \times 9000 = 2700$ ppm or $2700/3500 \times 100 = 78\%$ of the total. These calculations assume that the misfiring frequency at the stated test condition is zero.

In the products of combustion a much lower concentration of unburned hydrocarbons is present, on the order of magnitude of 300-1000 ppm. During the combustion process the flame does not propagate completely to the relatively cool walls of the combustion chamber. Instead the flame is quenched a finite distance from the wall. In this "quench zone" the 200 or so species of hydrocarbons are formed which constitute the combustion products unburned hydrocarbons. This source is the principal cause of 4-stroke engine emissions.

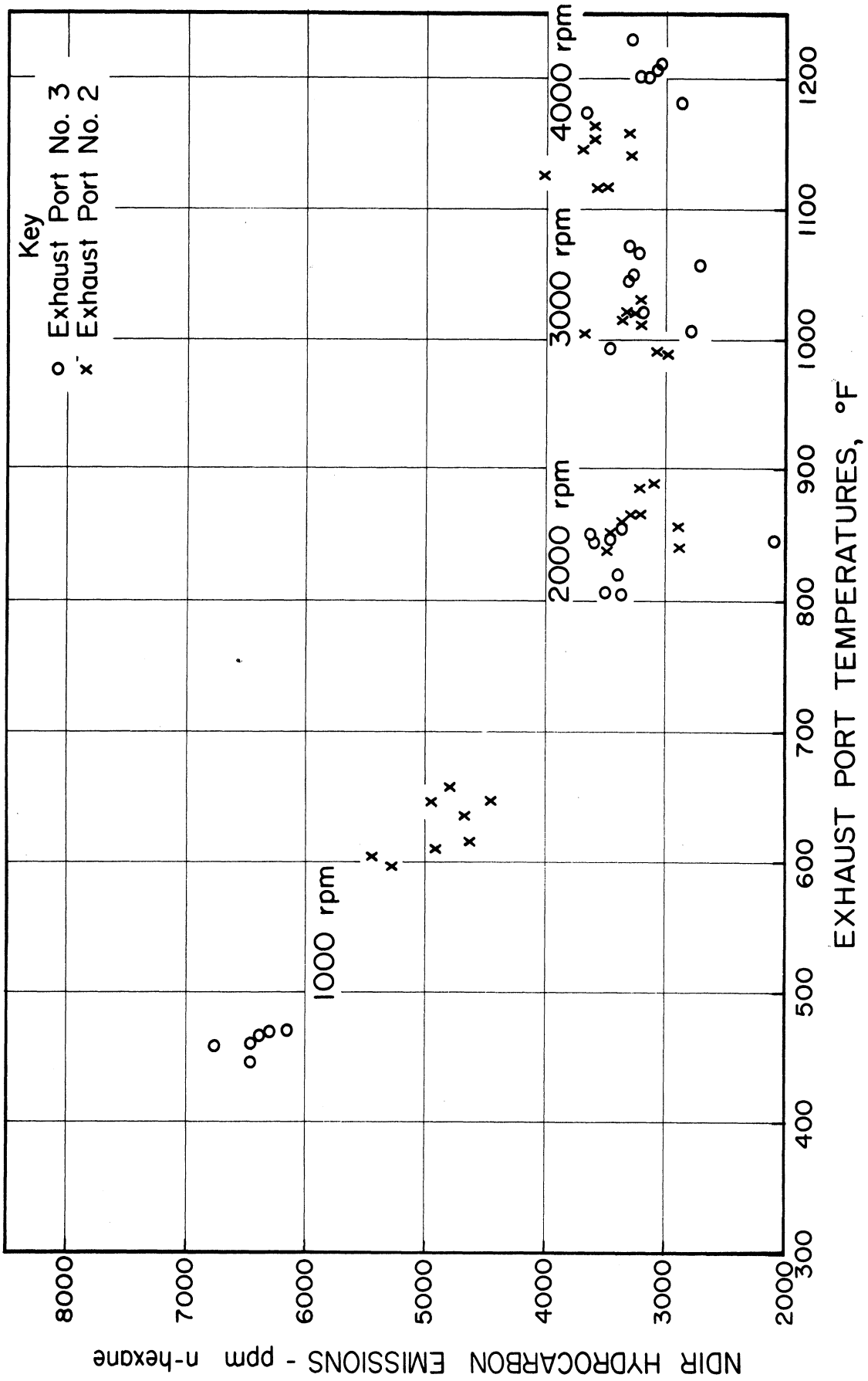


Fig. 24. Effect of exhaust port temperature on exhaust hydrocarbon concentration.

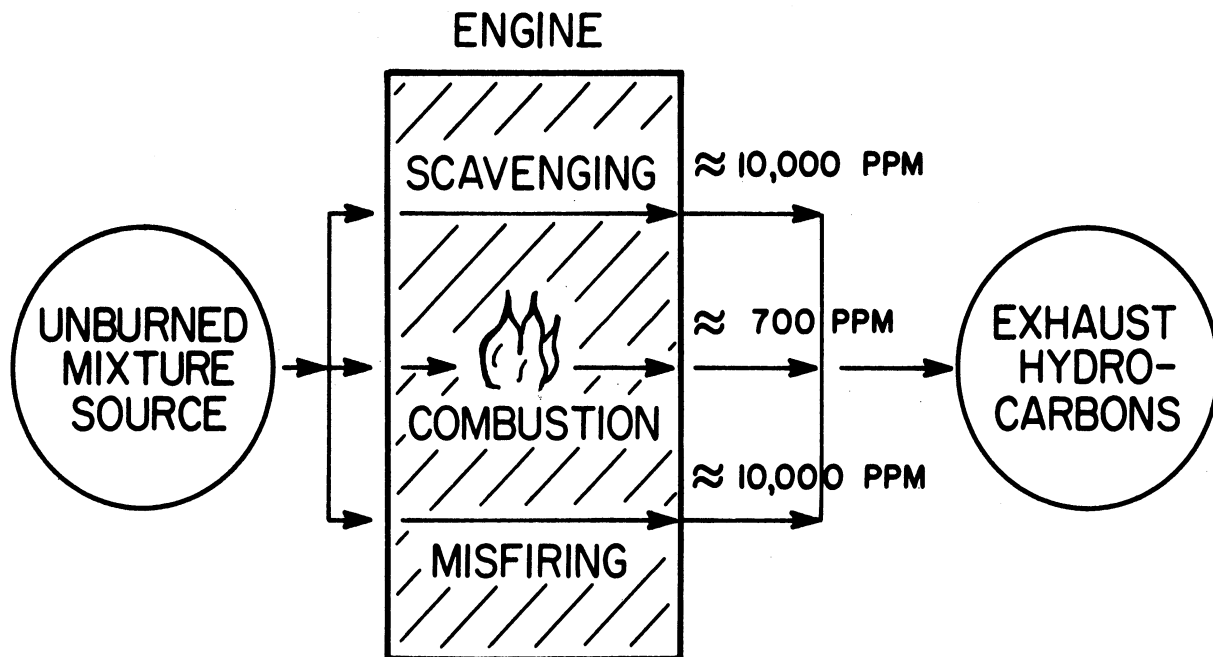


Fig. 25. Schematic diagram showing the sources of the hydrocarbon emissions in the 2-stroke engine.

Because of the relatively low hydrocarbon concentration in the combustion products, even though a major portion of the charge is burned, its contribution to the total emission problem is quite small. It is believed that design and development to minimize this source of emissions would be attacking the total problem at its least significant point and would be largely ineffective.

The final source is again related to the hydrocarbons in the unburned charge and is due to misfiring of the engine. When a cylinder misfires a charge of unburned mixture is released without burning. The higher this frequency the higher the emission level. When the cylinder is misfiring 100% of the time the exhaust consists of 100% fresh mixture. It is certainly recommended that work be done to minimize misfiring at all normal operating conditions because misfiring can have such a great effect on the overall hydrocarbon emission level as demonstrated by the 1000 rpm base line data. The quantitative analysis of misfiring is discussed in the next section.

G. MISFIRING FREQUENCY DETERMINATION AND ALLOTMENT OF THE EXHAUST HYDROCARBONS TO THE ENGINE PROCESSES

As mentioned in the previous section, it is desirable to quantify the misfiring frequency. With this data it should be possible to allot the unburned hydrocarbons to the several major internal sources as follows:

Let

x = percentage of atmospheric O_2 in the exhaust (% of 100%),
 q = misfiring frequency,
 Γ = engine trapping efficiency,
ppm NC = hydrocarbon concentration measured in the supplied mixture,
100% misfiring, no combustion,
ppm Com = hydrocarbon concentration in only the combustion products,
ppm = measured exhaust hydrocarbon concentration.

The trapping efficiency, Γ , is given by

$$\Gamma = \frac{100 - x}{100 (1 - q)}$$

$1 - \Gamma$ = proportion of mixture lost during the scavenging process.

With this information the following can be determined:

Percent of emissions due to the unburned mixture = $x \cdot \text{ppm NC/ppm}$
Percent of emissions due to the scavenging losses = $(1 - \Gamma) \cdot \text{ppm NC/ppm}$
Percent of emissions due to the misfiring = $x - (1 - \Gamma) \cdot \text{ppm NC/ppm}$
Percent of emissions due to combustion = $(1 - x) \cdot \text{ppm Com/ppm}$

It is hoped that ultimately the hydrocarbon emissions may be distributed or allotted to the proper internal engine processes through a series of curves, a qualitative example of which is shown in Fig. 26.

A number of techniques are currently being explored for measuring the misfiring frequency and include the following:

1. Thermocouple in the exhaust port. This technique uses a very small diameter, high frequency response thermocouple. It was hoped that the high temperature blowdown from a firing cycle would cause a measurable voltage response at the thermocouple junction. The capacitor-discharge ignition system, however, has created problems with electrical noise. The signal-noise ratio is too low for adequate resolution.
2. Cylinder pressure record. Direct evaluation of the pressure time record from the cylinders does not provide the required data resolution. At light loads where misfiring is most significant there is very little difference in peak cycle pressure between a misfiring and firing cycle. This is a result of the significant spark retardation at these conditions. Consequently, we have integrated the pressure signal to gain a measure of indicated cylinder work. Integration also alleviates high frequency noise problems. By sampling the cumulative cylinder work near the end of the expansion stroke, it is much easier to detect a misfiring cycle. Unfortunately, the resolution at very light loads is still a problem.

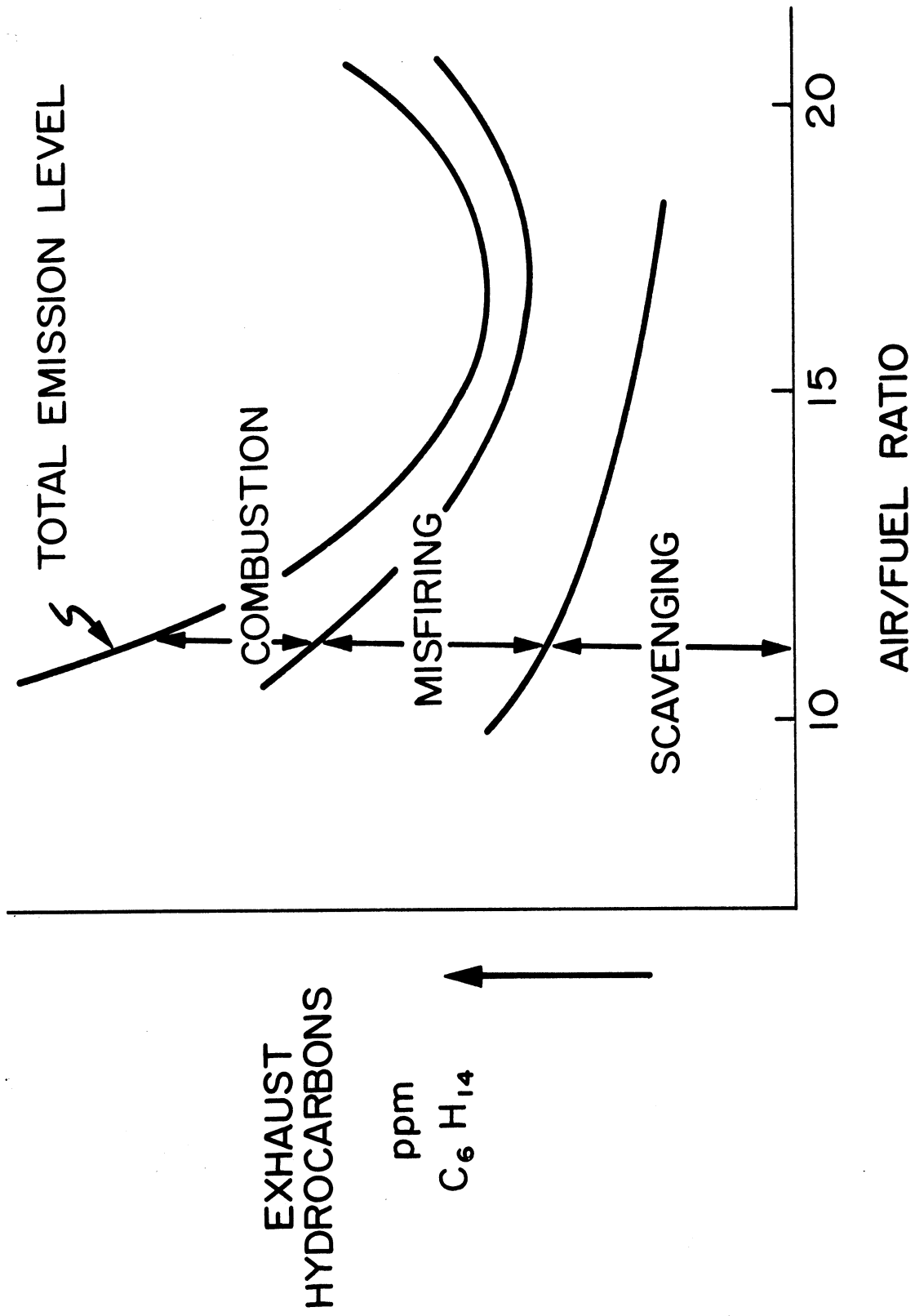


Fig. 26. Hypothetical distribution of the hydrocarbon emissions from the various internal engine sources as a function of air/fuel ratio.

3. Ion gap in the combustion chamber. This technique should permit detection of a flame front in the combustion chamber. The high temperature flame causes ionization of an air gap which, if placed in an electrical circuit with a voltage potential, will result in a significant decrease in circuit resistance and therefore increase in current flow. We are presently installing a fitting in the head to hold the ion-gap probe. High output voltages are attainable which minimizes the ignition noise problem.

4. Photoelectric measurement of combustion. In this technique a quartz window is fitted to the head and either a solar cell or phototube used to record the incidence of combustion generated light. We have substantial experience with photoelectric devices. The combustion chamber fitting used for the ion probe will be used to hold the quartz window.

5. Ion gap in the exhaust port. With an ion gap placed in the exhaust port it should be possible to determine if combustion occurred by observing ionization caused by the high temperature blowdown. This, of course, assumes that combustion in the other cylinders will not ionize the gap during their respective blowdown processes.

It is our intent to develop at least two of the before mentioned techniques to the point where their data corroborates one another.

H. POSSIBILITY OF EXHAUST CLEAN-UP BY OXIDATION IN THE EXHAUST PLENUM

It has been shown by a number of investigators¹¹⁻¹³ that significant oxidation of the unburned hydrocarbons and carbon monoxide can occur in the engine exhaust system if the conditions are proper. The most important engine variables effecting this reaction are:

- . Temperature - temperature is perhaps the single most important variable affecting the clean-up reaction. In fact above 1200°F for every 40°F temperature increase there is a corresponding doubling of the overall reaction rate.
- . Volume - increasing the volume at the high exhaust port temperatures increases the clean-up.
- . Pressure - raising the back pressure also increases the reaction rate.
- . Time - clean-up is improved by prolonging the time the exhaust is at the elevated temperatures.
- . Oxygen - free oxygen must be present for oxidation to occur, added either from an external source (air pump) or carried along with the charge (lean mixtures or scavenged, unburned mixture). Increasing the

concentration of oxygen causes improved oxidation if the other conditions are appropriate.

- . Mass Flow Rate - increasing the flow rate causes decreased clean-up by decreasing the time at the elevated temperatures. However, the increase in flow rate causes increased turbulence which may cause an increase in the reaction rate.
- . Reactivity of the Unburned Species - clean-up is obviously improved if the exhaust gas compounds have a high reactivity.

Eltinge, Marsee, and Warren¹¹ have attempted to quantify the after engine oxidation clean-up, assuming a homogeneous reaction, with the following relationship:

$$C_o = C_i e^{-\left(\frac{K_r P^2 O_2 V}{K_3 T^2 W}\right)}$$

where

- C_o = concentration of hydrocarbons in the exhaust gases leaving the reaction volume, ppm (vol)
- C_i = concentration of hydrocarbons in the exhaust gases entering the reaction volume, ppm (vol)
- e = base of natural logarithms
- P = exhaust pressure, psia
- O_2 = oxygen concentration in the exhaust gases, % (vol)
- V = volume available for exhaust gas reaction, ft³
- T = absolute temperature, °R
- W = mass flow rate of air, lb sec⁻¹
- K_3 = numerical constant
- K_r = specific reaction rate, ft³ lb-mol⁻¹ sec⁻¹

Clearly the relative importance of the various parameters is shown. A measurable exhaust clean-up reaction occurs in the 2-stroke outboard as demonstrated by the decrease of unburned hydrocarbon concentration from the exhaust port to the exhaust pipe. This also suggests that with improved design of the exhaust system an even more efficient clean-up might be obtained.

A number of factors make the 2-stroke engine attractive for the application of this technique for decreasing certain undesirable emissions.

- . The current design of the exhaust collector on the V-4 is quite efficient in that the ports exhaust into a volume with a relatively low surface/volume ratio. This conserves energy by minimizing heat transfer to the surroundings. Automotive engines are at a disadvantage in this respect because the exhaust ports are not in close proximity.

- . The 2-stroke engine exhaust is twice the frequency of the 4-stroke exhaust which results in a high average exhaust temperature. However, cooling caused by scavenged mixture and significant exhaust dilution of the reactants in the combustion process decrease the apparent temperature advantage.
- . Oxygen is present in the exhaust due to inefficient scavenging.

Unfortunately there are some difficulties in attempting significant clean-up in the exhaust system such as metallurgical problems at the high temperatures and the possibility of cylinder charge ignition during scavenging.

I. THE TWO-STROKE ENGINE AND WATER POLLUTION

The measurement of the effect of the 2-stroke engine exhaust on water pollution was of secondary importance in this research program. However, a technique was developed to obtain a sample of the compounds which remain in or on the water after the engine exhaust process. The condensate trap of the condenser sample conditioning system served this purpose and permitted collection of most of the low boiling point compounds which are generally the same compounds that would contaminate the lake or river water. Special care was taken to minimize "hang up" of any of these compounds in sample system locations other than the condensate trap.

We collected a number of condensate samples and initiated a quantitative analyses study with the Sanitary Engineering Department at The University of Michigan. Due to the base line experimental difficulties discussed earlier and a time limitation we have not obtained any meaningful results. The primary constituent was, of course, water but it was found that the percentage of unburned hydrocarbons in the condensate was significant.

With the quantitative analysis of the hydrocarbons in the condensate, the ratio of sample flow rate to the engine air and fuel rate, and mass of condensate obtained in a given length of time it should be possible to determine the mass of water pollution emissions per unit time.

Qualitative observation of the condensate shows a very dank, murky, light brown fluid. The odor is particularly foul with a noticeable aldehyde flavor. Some of the constituents are water soluble while others are not and form a greasy film on the water surface. This separation between the soluble and insoluble compounds presents a formidable but hopefully not insurmountable analysis problem.

It is quite possible that the water pollution aspects of the 2-stroke outboard may be even more important than the air-pollution problem. At any rate, it is likely to be a problem of more immediate concern to the government.

ACKNOWLEDGMENT

The authors would wish to acknowledge the services of a number of Research Assistants: Peter Brown, Peter Van deWalker, David Lebūt, Terry Cadmus, and Steven Sawyer who collected the data and designed and built much of the experimental apparatus. Without their substantial efforts the project would not have been successful.

Also the work of the several groups of the Office of Research Administration who assisted in the preparation of this report is gratefully acknowledged.

APPENDIX A

OUTBOARD MARINE CORPORATION HYDROCARBON EMISSIONS AND
ENGINE OPERATING VARIABLE DATA REDUCTION PROGRAM

The purpose of this program is to reduce and print out certain engine performance data including:

1. Engine speed—rpm
2. Brake horsepower—BHP
3. Brake torque—TORQUE
4. Air/fuel ratio—AF
5. Brake mean effective pressure—BMEP
6. Brake thermal efficiency—THEFF
7. Brake specific fuel consumption—BSFC
8. Hydrocarbon emission

Concentration—PPM

Mass per unit time—LB/HR

COMPUTER PROGRAM

FORTRAN IV G COMPILER	MAIN	08-14-68	19:27.31
0001		IMPLICIT REAL(L)	OMC 0
0002		INTEGER PPM(50,5), REV	OMC 1
0003		DIMENSION BHP(50),TORQUE(50),AF(50),BSFC(50),THEFF(50),BMEP(50),	OMC 2
		1HEXANE(50,5),SPEED(50)	OMC 3
0004		NAMELIST/NAM1/BARR,VPRESS,TEMP,TC,PC	OMC 4
0005		CALL FCVTHB(REV)	OMC 4.5
0006	1014	READ (5,NAM1)	OMC 5
0007		IF(BARR.EQ.0.0) GO TO 1015	OMC 5.5
0008		DBARR=BARR-VPRESS	OMC 6
0009		PWPPCA=(VPRESS*18.0)/(DBARR*28.85)	OMC 7
0010		PWPPMA=PWPPDA/(1.0+PWPPDA)	OMC 8
0011		CF=(1.0-PWPPMA)*0.075*TC*PC*130.0*60.0	OMC 10
0012		DC=4000.0*28.0/16.0	OMC 11
0013		DO 103 I=1,50	OMC 12
0014	101	FORMAT (F6.1,I6,F6.3,F6.3,F6.3,I6,I6,I6,I6)	OMC 13
0015		READ (5,101)LOAD,REV,TIME,WT,H2O,PPM(I,1),PPM(I,2),PPM(I,3),PPM(I,	OMC 14
		14),PPM(I,5)	OMC 15
0016		IF(LOAD.EQ.0.0) GO TO 1012	OMC 16
0017		SPEED(I)=REV/TIME	OMC 17
0018		BHP(I)=LCAD*SPEED(I)/DC	OMC 18
0019		TORQUE(I)=5252.0*BHP(I)/SPEED(I)	OMC 19
0020		BMEP(I)=75.4*TORQUE(I)/89.5	OMC 20
0021		FRATE=(WI/TIME)*60.0	OMC 21
0022		ARATE=I20*CF	OMC 22
0023		AF(I)=ARATE/FRATE	OMC 23
0024		BSFC(I)=FRATE/BHP(I)	OMC 24
0025		THEFF(I)=33000.0*60.0*100.0/(BSFC(I)*19000.0*778.0)	OMC 25
0026		DO 103 J=1,5	OMC 26
0027		IF(PPM(I,J).EQ.0) GO TO 102	OMC 27
0028		HEXANE(I,J)=(ARATE+FRATE)*86.0*PPM(I,J)/(86.0*PPM(I,J)+30.3*((1.0EOMC 28	
		16)-PPM(I,J)))	OMC 29
0029		GO TO 103	OMC 30
0030	102	HEXANE(I,J)=0.0	OMC 31
0031	103	CONTINUE	OMC 32
0032	1012	M=I-1	OMC 32.5
0033		DO 1011 K=1,M	OMC 33
0034		IF(K.NE.1) GO TO 1013	OMC 34
0035	104	FORMAT ('1',41X,50H OUTBCARD MARINE CORPORATION - JOHNSON 100 HP VOMC 35	
		1-4)	OMC 35.5
0036	105	FORMAT (48X,36H SPONSORED RESEARCH PROJ. NO. 34856)	OMC 36
0037	106	FORMAT (48X,34H TWO-CYCLE EXHAUST EMISSION STUDY)	OMC 37
0038	107	FORMAT ('0',30HENGINE BRAKE BRAKE AIR,13X,12HTHERM BSFC,OMC 38	
		124X,33HHYDROCARBONS MEASURED AS N-HEXANE)	OMC 39
0039	108	FORMAT (11H-SPEED HP,6X,22HTORQUE FUEL BMEP,4X,13HEFF OMC 40	
		1(LBM/,6X,6HPORT 1,9X,6HP CRT 2,9X,6HP CRT 3,9X,6HPORT 4,6X,12HEXHAUSOMC 41	
		2T PIPE)	OMC 42
0040	109	FORMAT (132H(RPM) (BHP) (LBF-FI) RATIO (PSI) (%) HP-OMC 43	
		1HR) (PPM) (LB/HR) (PPM) (LB/HR) (PPM) (LB/HR) (PPM) (LB/HR) (OMC 44	
		2PPM) (LB/HR))	OMC 45
0041		WRITE (6,104)	OMC 46
0042		WRITE (6,105)	OMC 47
0043		WRITE (6,106)	OMC 48
0044		WRITE (6,107)	OMC 49
0045		WRITE (6,108)	OMC 50
0046		WRITE (6,109)	OMC 51
0047	1010	FORMAT ('0',F6.1, 3X,F4.1,4X,F5.1,4X,F5.2,4X,F5.2,4X,F4.1,4X,F4.2,OMC 52	
		14X,I4,2X,F5.3,4X,I4,2X,F5.3,4X,I4,2X,F5.3,4X,I4,2X,F5.3,4X,I4,2X,FOMC 53	
		25.3)	OMC 54
0048	1013	WRITE (6,1010)SPEED(K),BHP(K),TORQUE(K),AF(K),BMEP(K),THEFF(K),BSFCOMC 55	
		1C(K),PPM(K,1),HEXANE(K,1),PPM(K,2),HEXANE(K,2),PPM(K,3),HEXANE(K,3)OMC 56	
		2),PPM(K,4),HEXANE(K,4),PPM(K,5),HEXANE(K,5)	OMC 57
0049	1011	CONTINUE	OMC 58
0050		GO TO 1014	OMC 58.2
0051	1015	CALL SYSTEM	OMC 58.5
0052		END	OMC 59

SAMPLE CALCULATIONS

CUIBOARD MARINE CORPORATION - JOHNSON 100 HP V-4
 SPONSORED RESEARCH PROJ. NO. 34856
 TWO-CYCLE EXHAUST EMISSION STUDY

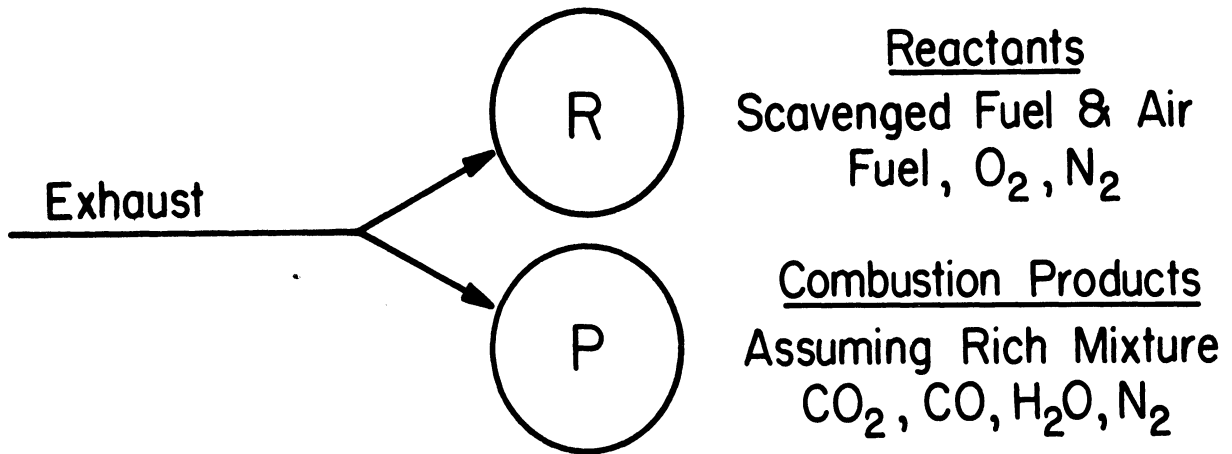
July 12, 1968

ENGINE SPEED (RPM)	BRAKE HP (BHP)	BRAKE TORQUE (LBF-FT)	AIR FUEL RATIO	BMEP (PSI)	I/TERM EFF (%)	BSEC (LBM/HP-HR)	HYDROCARBONS MEASURED AS N-HEXANE							
							PORT 1 (PPM)	PORT 2 (PPM)	PORT 3 (PPM)	PORT 4 (PPM)				
1004.7	1.6	8.4	12.78	7.08	3.7	3.62	7650	1.718	0	0.0	0	0.0	0	0.0
1100.4	1.7	8.0	12.96	6.70	3.9	3.45	7700	1.728	0	0.0	0	0.0	0	0.0
1021.9	1.5	7.8	11.76	6.57	3.5	3.87	0	0.0	4450	0.939	0	0.0	0	0.0
995.2	1.6	8.3	11.71	7.02	3.6	3.74	0	0.0	4675	0.987	0	0.0	0	0.0
996.2	1.5	7.8	11.83	6.57	3.4	3.95	0	0.0	0	0.0	6375	1.340	0	0.0
1007.2	1.5	7.7	12.47	6.51	3.6	3.74	0	0.0	0	0.0	6450	1.351	0	0.0
1005.5	1.5	7.7	12.22	6.51	3.5	3.82	0	0.0	0	0.0	0	0.0	4400	0.93
1001.8	1.6	8.3	12.47	6.95	3.8	3.52	0	0.0	0	0.0	0	0.0	4450	0.94
1000.9	1.5	7.8	11.98	6.57	3.5	3.88	0	0.0	0	0.0	0	0.0	0	0.0
1098.7	1.7	8.0	12.01	6.70	3.9	3.46	0	0.0	0	0.0	0	0.0	0	0.0
1994.9	9.2	24.3	12.94	20.48	11.8	1.13	3800	1.564	0	0.0	0	0.0	0	0.0
1998.3	9.0	23.7	13.10	19.97	11.7	1.15	3550	1.461	0	0.0	0	0.0	0	0.0
2012.7	9.0	23.6	12.30	19.85	10.8	1.24	0	0.0	2900	1.224	0	0.0	0	0.0
2011.7	8.9	23.1	13.38	19.47	11.5	1.17	0	0.0	2900	1.216	0	0.0	0	0.0
2017.3	8.6	22.5	12.41	18.96	10.8	1.24	0	0.0	0	0.0	3625	1.468	0	0.0
2005.0	9.1	23.9	13.53	20.10	11.9	1.12	0	0.0	0	0.0	3600	1.507	0	0.0
2008.5	8.9	23.3	13.15	19.59	11.3	1.18	0	0.0	0	0.0	0	0.0	3000	1.26
2010.3	9.1	23.9	13.02	20.10	11.5	1.16	0	0.0	0	0.0	0	0.0	3000	1.26
2003.4	9.1	23.9	12.90	20.10	11.6	1.15	0	0.0	0	0.0	0	0.0	0	0.0
2016.6	9.1	23.7	13.36	19.97	11.9	1.13	0	0.0	0	0.0	0	0.0	0	0.0
3059.2	23.9	41.1	12.66	34.64	15.9	0.84	3300	2.563	0	0.0	0	0.0	0	0.0
3020.2	24.1	41.9	13.16	35.27	16.5	0.81	3225	2.525	0	0.0	0	0.0	0	0.0
3012.5	23.9	41.7	13.46	35.14	16.2	0.83	0	0.0	3075	2.478	0	0.0	0	0.0
3012.3	23.7	41.3	13.85	34.83	16.4	0.82	0	0.0	3000	2.438	0	0.0	0	0.0
3014.4	23.9	41.7	12.92	35.14	15.9	0.84	0	0.0	0	0.0	3300	2.613	0	0.0
3007.7	24.0	41.9	13.42	35.27	16.5	0.81	0	0.0	0	0.0	3275	2.586	0	0.0

APPENDIX B

DESIGN OF UNDERWATER EXHAUST SIMULATION

Two types of compounds are identified in the exhaust, those with the properties of the products of combustion, and those with the properties of the reactants (fuel and air).



At each engine condition of speed and load the relationship between the reactions (R) and products (P) can be found if the trapping efficiency (Γ) is known; the trapping efficiency being defined as the ratio of the mixture trapped in the cylinder to the mixture supplied,

$$\Gamma = \frac{\text{trapped}}{\text{supplied}}$$

In the exhaust gas let N_T = total number of moles. If it is assumed that the reactants and combustion products have approximately the same molecular weights

$$N_T = \Gamma(a \text{ CO}_2 + b \text{ CO} + c \text{ H}_2\text{O} + d_P \text{ N}_2) + (1 - \Gamma)(e \text{ O}_2 + d_R \text{ N}_2 + f \text{ C}_8\text{H}_{18})$$

Note: C₈H₁₈ (octane) is assumed to represent the average composition of the fuel.

It is reasonable to assume that all of the water in the sample is trapped in the NDIR system condenser and therefore is not included in the volume flow measurement through the system,

$$(\dot{Q}_{\text{dry sample}} = \dot{Q}_{\text{ds}}) ,$$

and also reasonable to assume that the C_8H_{18} is not trapped and is included in the dry sample. Actually some H_2O is not condensed and some fuel is.

y = mass fraction of H_2O in the sample

$$\approx \frac{\Gamma c(18)}{\Gamma(44a + 28b + 18c + 28d_p) + (1 - \Gamma)(e + d_r) 29 + f114)}$$

$\approx .08$ for typical conditions

Let \dot{M}_{ds} = mass rate of sampling (dry) and

\dot{M}_{ts} = mass rate of sampling (total including H_2O).

These quantities may be computed with the use of the continuity equation

$$\dot{M}_{\text{ds}} = \dot{Q}_{\text{ds}} (\rho_{\text{ds}}) \quad \text{and} \quad \dot{M}_{\text{ts}} = \dot{Q}_{\text{ds}} [\rho_{\text{ds}} (1 + y)]$$

$\rho_{\text{ds}} \approx .079 \text{ lb/ft}^3$ = density at the point where dry sample flow rate is measured.

For an accurate simulation of the normal exhaust flow into a lake or river

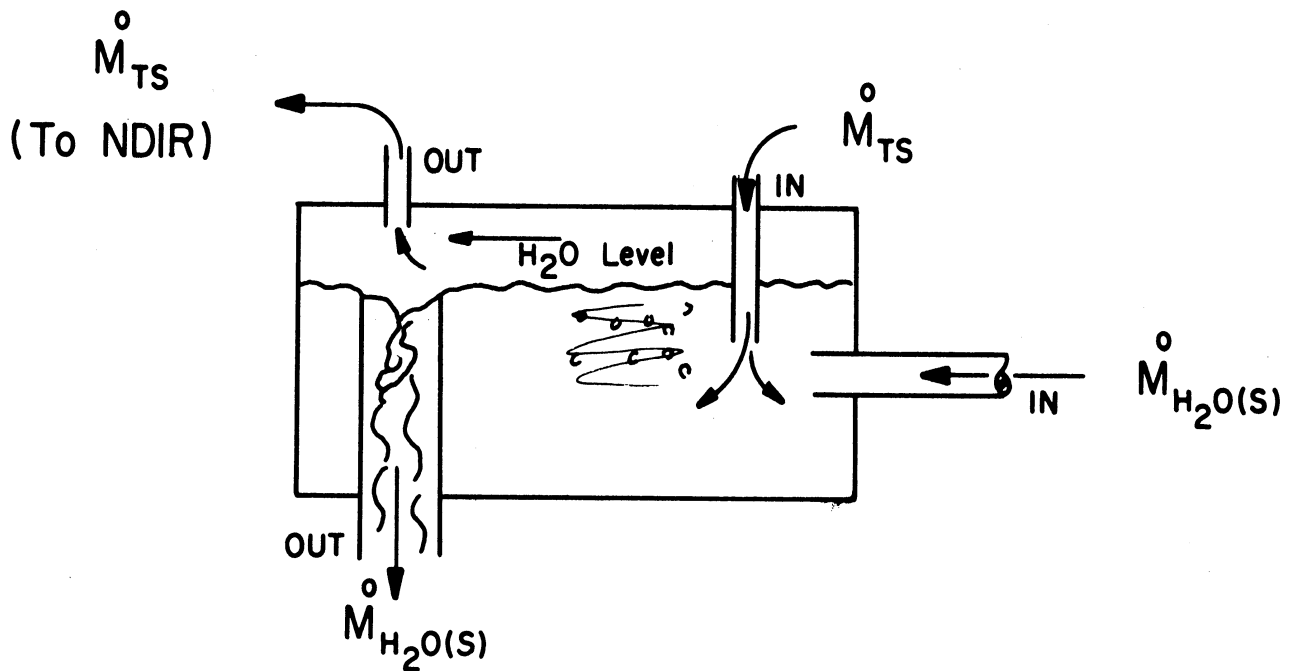
$$\frac{\dot{M}_{\text{ts}}}{\dot{M}_{\text{H}_2\text{O}}(s)} \text{ should equal } \frac{\dot{M}_{\text{E}}}{\dot{M}_{\text{H}_2\text{O}}}$$

where $\dot{M}_{\text{H}_2\text{O}}(s)$ = water flow rate in the simulated system

$\dot{M}_{\text{H}_2\text{O}}$ = water flow which the total engine exhaust sees when propelling a boat

\dot{M}_{E} = total engine exhaust mass flow rate

The following schematic diagram shows the basic simulation device in which the exhaust sample is bubbled through the highly turbulent water flow.



In the next figure the lower unit is shown passing through the water at a given velocity (V).

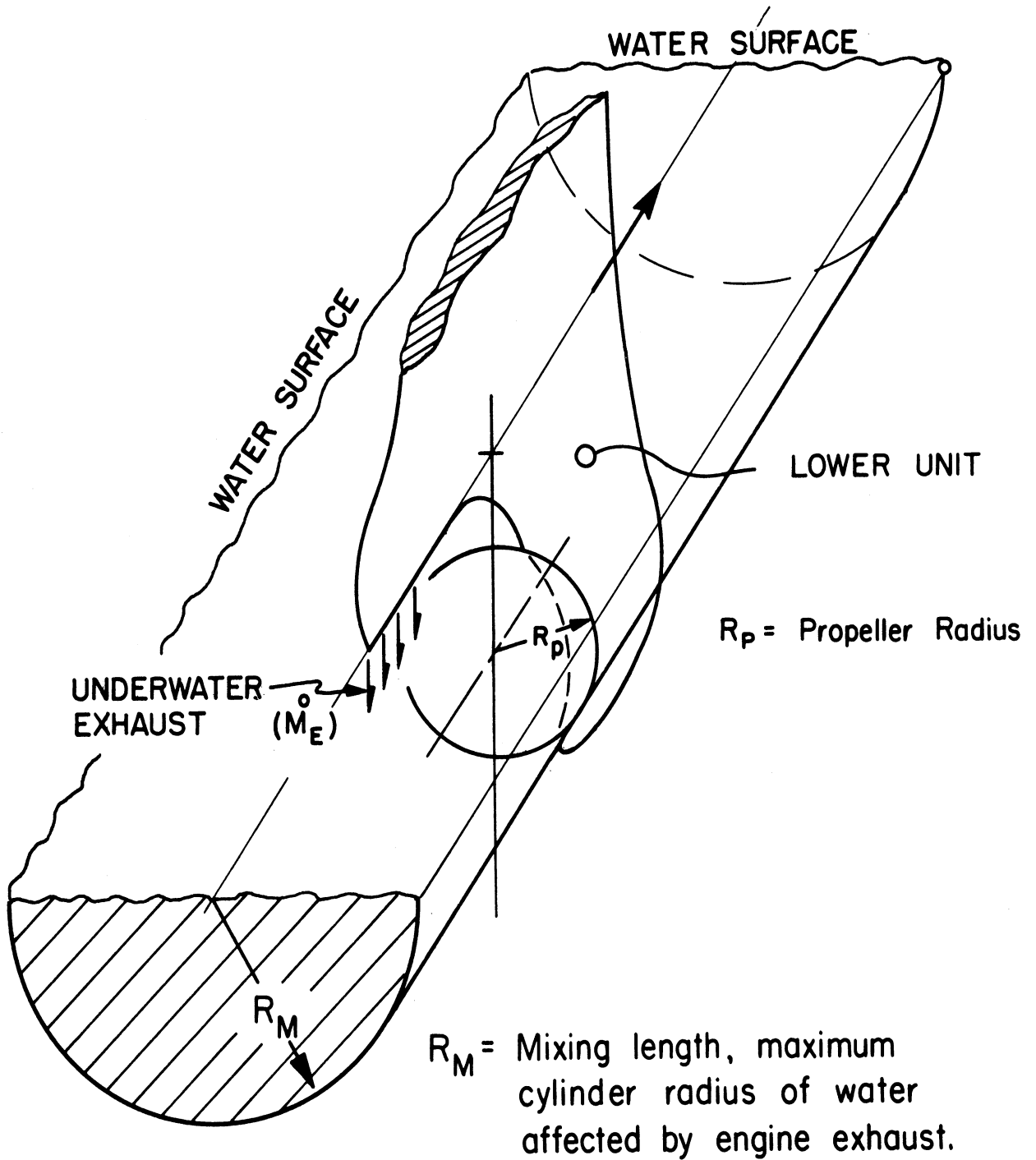
The exhaust gas influences a semi-cylinder of water which is, in effect, moving relative to the lower unit at a volume flow rate of \dot{Q}_{H_2O} .

Let \dot{Q}_{H_2O} = volume flow of H_2O past exhaust outlet which is effected by the exhaust, and $A = \frac{\pi R_m^2}{2}$ = area of section through the semi-cylinder, then

$$\dot{M}_{H_2O} = \frac{\pi R_m^2}{2} \times \dot{Q}_{H_2O} = \frac{\pi R_m^2}{2} \times \rho_{H_2O} \times V$$

The ratio of exhaust gas flow to the water flow is,

$$\begin{aligned} \frac{\dot{M}_E}{\dot{M}_{H_2O}} &= \frac{\dot{M}_A (1 + F/A)}{\frac{\pi R_m^2}{2} \cdot V \cdot \rho_{H_2O}} \\ &\approx \frac{\dot{M}_A}{V} (1 + F/A) \times .26 \times 10^{-5} \\ &= \frac{\dot{M}_{ts}}{\dot{M}_{H_2O(s)}} \end{aligned}$$



where \dot{M}_a = engine air flow rate (lb/hr)
 F/A = fuel/air ratio
 V = boat forward velocity (ft/sec)

and with the assumption that the mixing length $R_m = 2 \times R_p = 1.04$ ft.

A summary of the computations is presented in Table III for the boat-load test conditions. The required simulator water flow rate is specified in gallons per minute (gpm) for an exhaust sample flow rate of 5 cubic feet per hour (CFH).

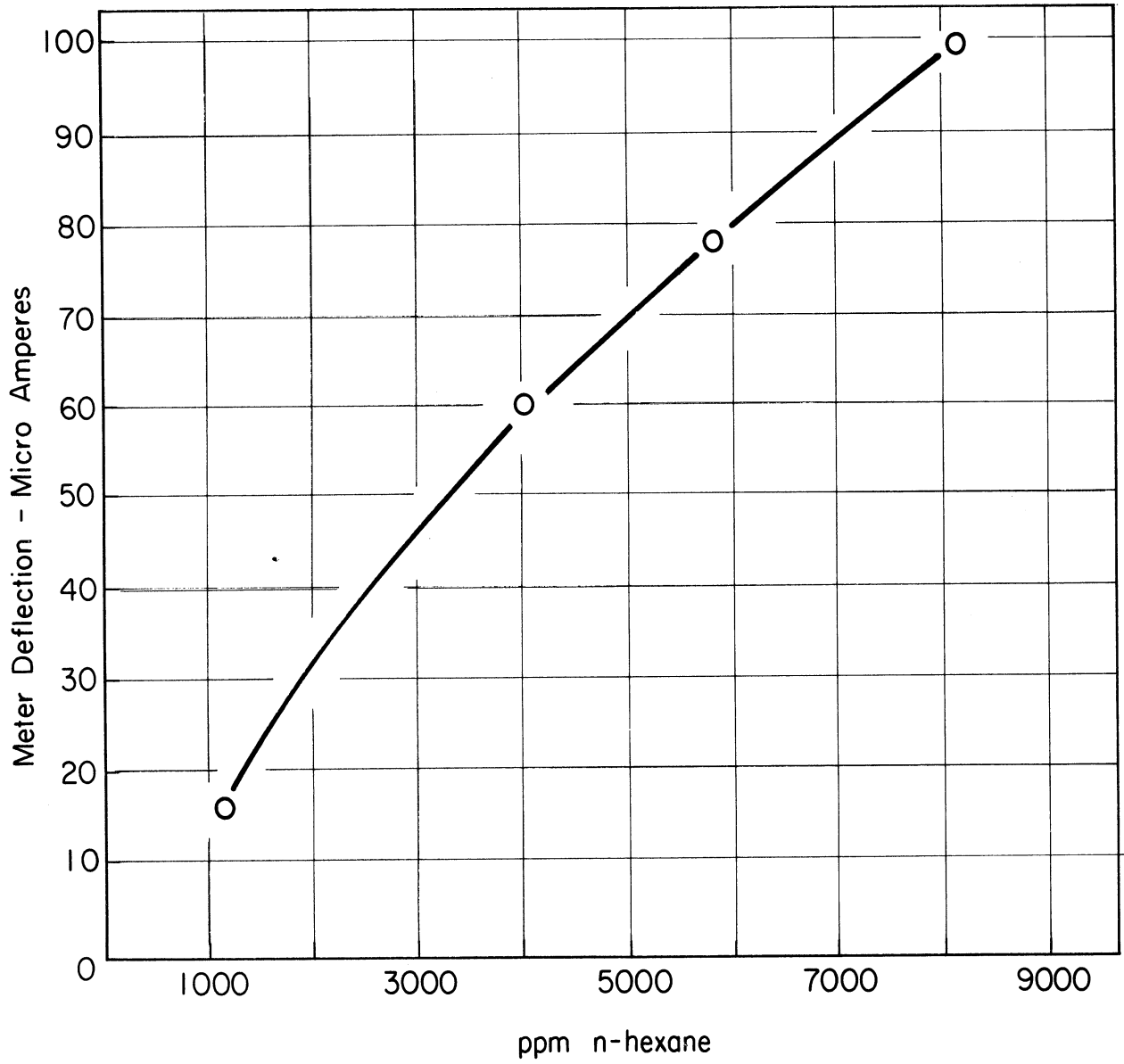
TABLE III

UNDERWATER EXHAUST SIMULATION DATA FOR A DRY EXHAUST GAS
SAMPLE FLOW RATE, \dot{Q}_{ds} , OF 5 CFH

Engine, rpm	Engine, BHP	Assumed Boat Speed, V - ft/sec	$\dot{M}_E = \dot{M}_a(1+F)$	$\dot{M}_E/\dot{M}_{H_2O} \times 10^5$	$\dot{M}_{H_2O}(s)$ lb/hr x 10^{-4}	$Q_{H_2O}(s)$ gal/min
1000	1.5	5.0	80	4.2	1.01	20
2000	9	8.6	175	5.3	.81	16.2
3000	24	17.5	315	4.7	.91	18.2
4000	50	32	500	4.1	1.04	20.8
5000	86	54	750	3.6	1.18	23.6

APPENDIX C

BECKMAN NDIR CALIBRATION CURVE



APPENDIX D

CONVERSION OF CONCENTRATION HYDROCARBON EMISSIONS TO MASS RATE OF HYDROCARBON EMISSIONS

It is reasonable to assume that the dry exhaust gas* is composed primarily of the following components together with their respective molecular weights: CO₂ - 44, N₂ - 28, and C₆H₁₄ - 86.

For each mole of CO₂ there are approximately 6 moles of N₂, therefore the

$$\text{Molecular weight of the CO}_2 \text{ and N}_2 \text{ mixture} \approx \frac{(1 \cdot 44) + (6 \cdot 28)}{7} \approx 30.3$$

If $x = \text{ppm C}_6\text{H}_{14}$ (n-hexane), then $86x$ is proportional to the mass of C₆H₁₄, and $30.3(10^6 - x)$ is proportional to the mass of CO₂ + N₂ and

$$\frac{\text{Mass C}_6\text{H}_{14}}{\text{Mass of Dry Exhaust}} \approx \frac{86x}{30.3(10^6 - x) + 86x} ,$$

which can be simplified to $86x/30.3(10^6)$. This simplification introduces less than 1% error for $x < 10,000$ ppm.

Since Total Mass of Exhaust/Mass of Dry Exhaust ≈ 1.05 for the average hydrocarbon fuel, the

$$\frac{\text{Mass C}_6\text{H}_{14}}{\text{Mass of Total Exhaust}} \approx \frac{86x}{30.3 \cdot 10^6 (1.05)} \approx \frac{90x}{30.3 \cdot 10^6}$$

The total mass flow rate from the engine is:

$$\dot{M}_T = \dot{M}_A + (F/A)\dot{M}_A = (1 + F/A)\dot{M}_A$$

*The water vapor can be assumed to be totally condensed in the analysis system condenser and the unburned hydrocarbons to have negligible mass compared to the other constituents.

where

\dot{M}_T = total mass flow rate

\dot{M}_A = air flow rate

F/A = fuel/air ratio

Thus the mass rate of unburned hydrocarbon emission is:

$$\dot{M}_{C_6H_{14}} \approx \dot{M}_A (1 + F/A) \left(\frac{90x}{30.3 \cdot 10^6} \right)$$

$$\dot{M}_{C_6H_{14}} \approx \text{lb/hr n-hexane}$$

APPENDIX E

DETERMINATION OF 2-CYCLE ENGINE TRAPPING EFFICIENCY (Γ)
USING OXYGEN (O_2) AS A TRACER

There are two primary classes of constituents in the exhaust from a spark-ignition 2-stroke cycle engine.

1. Those with the properties of the combustion products:

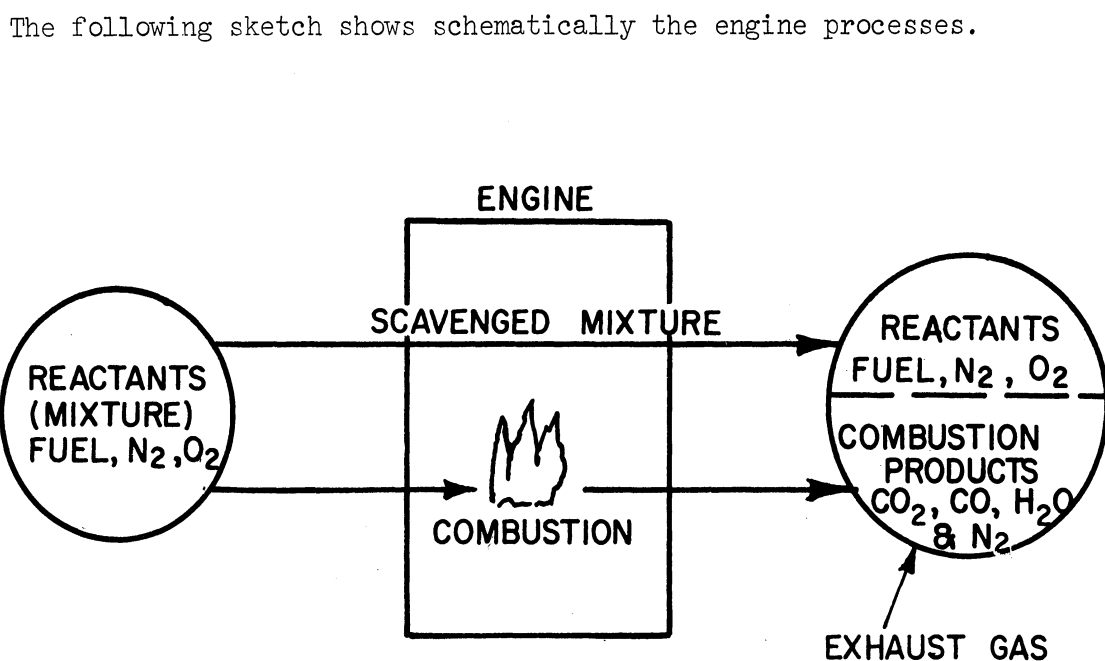
CO_2 , CO , H_2O , N_2 , and O_2

For rich mixtures $O_2 \approx 0\%$

NO , NO_2 , H_2 , etc., are neglected because their concentrations are very low.

2. Those with the properties of the reactants: fuel and air.

Note: This analysis will assume that a rich mixture is used, the engine is not misfiring, and that the sample is quenched in the sample probe preventing any oxidation in the sample system.



Since the exhaust sample is drawn through a condensor at 32°-34°F, it is a reasonable assumption that the fuel and water are condensed. Thus, the dry sample consists primarily of:

1. O₂ and N₂ from the unburned mixtures.
2. CO₂, CO, and N₂ from the products of combustion

If x = the mole fraction of O₂ in the dry sample, expressed as percent of 100, the engine trapping efficiency, Γ, on a mass base is approximated (within ±2% accuracy), by the following relation:

$$\Gamma \approx 1 - x$$

The previous equation is a simplification of the exact relationship,

$$\Gamma = \frac{(57.2 - 57.2x)\frac{1}{\Phi}}{x(a + b - \frac{12}{\Phi}) + \frac{57.2}{\Phi}}$$

which assumes the use of octene, C₈H₁₆, as the fuel and where

Φ = equivalence ratio,
a = moles of CO₂ formed in the reaction of 1 mole of C₈H₁₆ with air,
and b = moles of CO formed in the reaction of 1 mole of C₈H₁₆ with air.

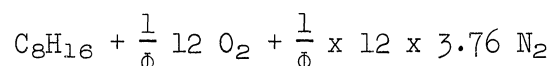
APPENDIX F

HYDROCARBON CONCENTRATION IN THE UNBURNED AIR-FUEL MIXTURE

It is reasonable to make several simplifying assumptions in the analysis of unburned hydrocarbon concentration in the air-fuel mixture without introducing appreciable error.

- . Since the lubricating oil is such a small part of the gasoline-oil mixture (1 in 50 by volume) it will be neglected.
- . The gasoline will be assumed to have the composition of octene C_8H_{16} .

The left side of the air-fuel combustion equation can be written as:



where Φ = equivalence ratio or relative air/fuel ratio. The volume percentage of C_8H_{16} in the mixture is:

$$\frac{\text{Moles } C_8H_{16}}{\text{Total No. of Moles of Mixture}} = \frac{1 \times 100}{\frac{12}{\Phi} + \frac{12}{\Phi} \times 3.76 + 1} = \frac{100}{\frac{57}{\Phi} + 1}$$

The concentration of unburned hydrocarbons in parts per million (ppm) C_8H_{16} can be found by multiplying the volume percent by 10^4 .

$$\text{ppm } C_8H_{16} = \frac{100}{\frac{57}{\Phi} + 1} \times 10^4 = \frac{10^6}{\frac{57}{\Phi} + 1}$$

Since n-hexane (C_6H_{14}) is currently used as the reference gas for much of the exhaust emission work the ppm C_8H_{16} can be changed to the hexane equivalent by multiplying by $8/6 = 1.33$.

However, this implies that a flame ionization detector (FID), which measures essentially all of the nonoxygenated hydrocarbons, is used as the analytical device. In our work we have used, to date, a nondispersive infrared (NDIR) technique which does not respond linearly to all hydrocarbons. For example the olefins have a very small (low) response in a NDIR detector sensitized to a paraffin such as n-hexane. Generally the NDIR indicates approximately 50% of the FID concentration in the average engine exhaust.

For a given sample

$$\text{ppm}_{\text{FID}}(\text{C}_6\text{H}_{14}) \cong 2 \times \text{ppm}_{\text{NDIR}}(\text{C}_6\text{H}_{14})$$

or

$$\text{ppm}_{\text{NDIR}}(\text{C}_6\text{H}_{14}) \cong .5 \times \text{ppm}_{\text{FID}}(\text{C}_6\text{H}_{14})$$

The computations for a series of mixture ratios from $\phi = .8$ to $\phi = 1.5$ are presented in Table IV.

TABLE IV

HYDROCARBON CONCENTRATION IN THE UNBURNED AIR-FUEL MIXTURE

ϕ	F/A Ratio	% C_8H_{16} by Vol	$\text{ppm}_{\text{C}_8\text{H}_{16}}$	$\text{ppm}_{\text{C}_6\text{H}_{14}}$ FID	$\text{ppm}_{\text{C}_6\text{H}_{14}}$ NDIR
.8	.0542	1.38	13,800	18,400	9,200
.9	.0610	1.56	15,600	20,800	10,400
1.0	.0678	1.72	17,200	22,300	11,100
1.1	.0746	1.88	18,800	24,400	12,200
1.2	.0813	2.06	20,600	26,800	13,400
1.3	.0882	2.22	22,200	28,900	14,500
1.4	.095	2.41	24,100	31,200	15,600
1.5	.1016	2.56	25,600	33,300	16,700

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