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LARGE DIESEL ENGINE TESTING FOR OIL SHALE MINING

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by:
Southwest Research Institute
Department of Engine and Vehicle Research
6220 Culebra Road
San Antonio, Texas 78284

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FOREWORD

This report was prepared by Southwest Research Institute, San Antonio, Texas, under U.S. Bureau of Mines Contract Number J0265023. The contract was initiated under the Advancing Oil Shale Mining Technology program. It was administered under the technical direction of Twin Cities Mining Research Center, with Mr. Robert W. Waytulonis acting as the Technical Project Officer. Mr. John Arnold was the contract administrator for the Bureau of Mines.

This report is a summary of the work recently completed as part of this contract during the period June 1976 to January 1978.

ACKNOWLEDGEMENTS

The technical guidance and support of Mr. Robert Waytulonis, in the conduct of this project, is gratefully acknowledged. The cooperation and assistance of the following organizations and their respective personnel were essential to the program:

Caterpillar Tractor Company
Cummins Engine Company
Detroit Diesel Allison, Division of General Motors Corporation
Wagner Mining Equipment Company
MESA, Pittsburgh, Pennsylvania

SUBJECT INVENTIONS

Southwest Research Institute does hereby certify that no patents or inventions have resulted from this study.

NOTICE

The United States Government does not endorse products or manufacturers. Trade or manufacturers names appear herein solely because they are considered essential to the object of this report.

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I. INTRODUCTION

Future commercial underground oil shale mining operations will require mobile diesel powered equipment very similar to that presently used in large open pit mines. Studies of feasible underground mining methods for the economic extraction of oil shale deposits of the Green River Formation located in Colorado, Utah, and Wyoming, have concluded that in situ, room and pillar, and modified in situ (room and pillar combined with in situ) mining methods will achieve the lowest costs and highest production (1,2). By utilizing a room and pillar mining system, large diesel equipment can be used to excavate large stable openings for the mining of oil shale. Production requirements ranging from 5000 to 66,000 tons per day will demand the use of this large mobile equipment to provide the economics of scale well beyond that found in any underground mines. High performance equipment such as 50 to 200 ton capacity haulage trucks and 8 to 24 cu. yd. capacity front-end loaders may be necessary.

The Mining Enforcement and Safety Administration (MESA) regulations in Part 36, Title 30, Code of Federal Regulations set forth the requirements for mobile diesel-powered transportation equipment for gassy non-coal mines and tunnels (Schedule 31) (3). These regulations primarily require that intake and exhaust systems must be strong enough to completely contain internal explosions and have flame arresters to prevent flame propagation to a surrounding flammable mixture. An exhaust cooling system must be capable of reducing the exhaust gas temperature to 170°F maximum at the point of discharge, and the exhaust gas composition must be quantified, primarily NO_x, CO, and CO_2 to determine the mine ventilation necessary to dilute these pollutants to below the prescribed limits set forth in Part 36.45 (b). Also, the external surface temperature of this complete engine package cannot exceed 302°F in any operational condition.

Currently, engines used in gassy mines have a maximum output of approximately 150 horsepower. All are naturally-aspirated diesels and generally use the pre-chamber type of combustion system. This study was undertaken to evaluate the needs for large diesel engines for commercial oil shale operation, to select three engines that satisfy these needs and test them under MESA Schedule 31 criteria, to judge from the results of these tests the capability of the engines to meet selected criteria from Schedule 31, and to devise and recommend modifications to enable the engines to meet the Schedule 31 criteria.

II. EXECUTIVE SUMMARY & CONCLUSIONS

The use of the diesel engine underground is an important requirement from considerations of high power density, mobility, flexibility of torque-speed output, and safety. On the other hand, the rising concern over the health aspects of diesel engine exhaust will limit the application of the diesel engine to underground mines (including oil shale mines) until these problems are adequately quantified and solved. Environmental Protection Agency legislation concerning the exhaust emissions of automotive diesels up to 450 Bhp has led to extensive studies of the "regulated" exhaust emissions: unburned hydrocarbons carbon monoxide, oxides of nitrogen and exhaust smoke. Efforts to reduce these emittants have involved invesitgations of exhaust gas recirculation, catalytic converters, injection system modifications and combustion chamber modifications. However, only limited information is available on particulates, organic materials, or other unregulated chemical compounds which may be present in the diesel exhaust. Since even the measurement techniques for most of these compounds are currently in the development stages, the health affects of many of these compounds are unknown.

This program was an initial effort to identify and quantify the health and safety characteristics of large diesel engines of 600-1200 Bhp for use in underground oil shale mines. The testing program concentrated on the important aspects of currently published regulations specifically, surface temperatures, exhaust-gas composition, exhaust gas cooling system and explosion testing as specified in 30 CFR 36 (Schedule 31). Schedule 31 was chosen merely as a reference to judge the hazard potential of large diesel engines operated underground. Schedule 24 could as well have been chosen since it is not known whether oil shale mines will be classified as gassy or non-gassy.

One of each type of combustion system available in the 600-1200 Bhp range was selected for testing. Results indicated that none of the engines could meet Schedule 31 requirements without significant modifications. Other conclusions were as follows:

A. Gaseous Emissions

A survey of currently certified diesel engines shows that they are all naturally-aspirated and predominately use the pre-chamber combustion system. Typical ventilation rates are 100-140 cfm/Hp, which compares favorably with the Caterpillar D348 engine ventilation rate of 91 cfm/Hp. The Detroit Diesel 12V-149TI and Cummins VTA-1710 engines required about three times this flow of air which would make them less attractive from an economic standpoint due to the much larger fans required in an underground mine. The high NO_x emissions for the undiluted exhaust of the Cummins engine would require that the engine have a 12% lower power rating: 690 Hp at 2100 RPM versus the present 800 Hp at 2100 RPM. Engine manufacturers have not been required to meet Federal regulations for exhaust emissions with these large engines and in some cases this is the first time emissions data have been obtained. Rematching of the turbochargers, modification of injection timing, and altering the fuel rate could produce significant emissions reductions.

B. Exhaust Scrubbers

The initial attempt to design this scrubber based on technology used for smaller scrubbers has proven that an acceptable unit cannot be constructed merely by scaling up a smaller unit. With the baseline scrubber, the optimum water level for adequate exhaust cooling was 30 to 50 gallons below the minimum level to pass the explosion tests. Modifications to the scrubber accomplished in Phase II raised the optimum water level 40 gallons with a 7% increase in water consumption. Some redesign would be required to provide adequate explosion protection.

For these large diesel engines, however, the required size of the exhaust scrubber and its make-up water tank is critical with respect to vehicle design and available space. Current exhaust scrubber design appears to be based on a "cut and try" approach and there are no indications that the current designs are the optimum ones. It is therefore reasonable to expect that a careful approach to the design of diesel exhaust scrubbers could result in improvements in their performance, i.e., a reduction in scrubber size and/or a reduction in the required water consumption rate.

C. Intake Flame Arrestors

These units performed satisfactorily in the explosion and intake restriction tests.

D. Surface Temperatures

All engines had excessive surface temperatures on the turbine side of the turbocharger. Water jacketed enclosures around the turbocharger offer a possible solution to this problem, however, significant design and fabrication problems are encountered. Water jacketed exhaust manifolds and turbocharger enclosures would have to be designed taking into consideration the problems of thermal expansion and sealing. A turbocharger with flanges at the exhaust outlet would be required for sealing the turbocharger and the enclosure. These are not currently available from turbocharger manufacturers, but it should be possible to have them fabricated.

Water cooled turbochargers are a second possible solution and are manufactured by several companies for marine applications, but they are not available for any of these engines. A development effort could be required to match a new water cooled turbocharger to each engine and, in fact, there may still be local hot spots on the water cooled turbocharger that are above 302°F.

A third solution to the high surface temperature problem is the injection of water into the exhaust manifold before the turbocharger. Water injection into the exhaust system has been used on naturally-aspirated engines with some success, but has not been tried on a turbocharged engine. Explosion proofing of the exhaust system must be investigated using either a flame arrester or a small exhaust scrubber.

Water cooled exhaust manifolds between the turbochargers and the engine cylinder heads are capable of maintaining surface temperatures below 300°F if properly designed. As evidence of this, the Cummins engine had a completely water jacketed exhaust manifold and its surface temperature was below 275°F.

E. Engine Joints and Gaskets

Copper gaskets and bolted flanges were built for attaching the flame arresters to the turbochargers. No explosions were initiated due to leaks in these areas. Schedule 31 requires that all flanges be secured by through bolts or other suitable means. The Detroit Diesel and Caterpillar contained slip joints in the exhaust system at the connection with the turbocharger. The Cummins and Detroit Diesel contained flexible silicone sleeves connecting the turbocharger with the remainder of the intake system. These joints simplify installation during manufacture and overhaul, but would not be allowed on certified engines. Modifications to provide bolted joints could be easily made by either the engine or equipment manufacturer. New exhaust manifolds would have to be designed in one case, and flanges could be welded to the turbocharger and intake system in the other.

A final observation is that due to the small volume of equipment used in current underground gassy mines, very little has been published on currently available technology or state-of-the-art. With current interest in oil shale mines, additional research will be required to make possible the use of large diesel engine powered equipment for providing high load hauling capability.

III. EXPERIMENTAL PROCEDURES - PHASE I

A. Selection of Test Engines

A literature review was conducted to determine past and future oil shale mining methods and types of equipment used. Results of this survey indicated that, of the various mobile equipment, haul trucks and front-end loaders use the largest engines. Room-and-pillar mining techniques, as used in oil shale mines, allow the use of such large equipment. Typical room dimensions of future mines are 60 ft x 60 ft, with ceilings 80 ft high. Rubber-tired front-end loaders from 8-24 cubic yard capacities are projected, while haul truck capacities will be in the range of 50-200 tons.

Companies involved with oil shale mining were next interviewed to identify problem areas and the expected horsepower requirements of equipment. Shell Oil Company (Mr. R. E. Meeker) did not have detailed engine application data but did have general information on oil shale mining. Cameron Engineers has completed a project for the Bureau of Mines that produced a mining plan for oil shale. Several different mining methods were investigated including the use of front-end loaders, load-haul-dump equipment, hydraulic drill jumbos, rock bolt machines, and other types of equipment. The hydraulic drill jumbos had a horsepower requirement of about 135 Hp. The front-end loaders chosen were 12 yd^3 capacity with an installed horsepower of 612. The load-haul-dump equipment was also 12 yd^3 and required about 395 horsepower. Equipment manufacturers contacted by Cameron thought it would be possible to meet Schedule 31 conditions at some additional cost.

Colony Development Company indicated they have used up to 400 Hp in haul trucks in non-gassy oil shale mines. They feel that in the future very large horsepower engines will be required. They suggest up to 11,000 tons per day of oil shale production and extimate that diesel engines up to 2000 Hp might be required in such activities. Cleveland Cliffs Iron Ore Mining Company was next interviewed, and they have used large equipment in oil shale mining since 1963. The largest equipment used has been 75 ton haul trucks (800 Hp) and 20 yd^3 front-end loaders (1200 Hp). None of their work has been done in gassy mines, however, as none of the equipment was certified to Schedule 31. They have experience in other mining activities with smaller diesel engines in equipment modified to meet Schedule 31, and commented about the excessive size of the exhaust cooler and its required accessories.

Three basic types of diesel engines are available in the 600-1200 Hp range: (1) four-stroke, direct injection engines; (2) four-stroke, pre-combustion chamber engines; and (3) two-stroke, direct injection engines. All these engines are water-cooled but within each category there are numerous models with different numbers of cylinders, different bores and strokes, with and without turbocharging, and with many other minor modifications to satisfy special applications. It was felt that one specific engine design should not be emphasized to the point where other viable diesel engine designs were excluded, and it was thus decided to test one engine from each of the three major diesel engine manufacturers in the United States. Each engine represented a different combustion cycle-fuel injection system.

Engines in the 600-1200 Hp range tend to be turbocharged in order to achieve a high power density; this fact excluded naturally-aspirated engines from the test program. In actuality, only one manufacturer (Detroit Diesel) offers a naturally-aspirated engine in the horsepower range of interest. The marketing department of Cummins Engine Company felt that their 700-800 horsepower engines were the most likely candidates for large underground equipment. These engines are commonly used in haul trucks in the 70-80 ton range and front-end loaders in the 12 yd^3 range. A used, but completely rebuilt, VTA-1710 engine was obtained from Cummins on a consignment basis. Detroit Diesel provided a new 12V-149TI engine rated at 1200 Hp. This engine is commonly used in 105-130 ton haul trucks. The third engine tested was a new Caterpillar D348. It is rated at 920 Hp and is extensively used in open pit mines for 85-100 ton haul trucks with some derated versions being used in 10 yd^3 front-end loaders. Characteristics of these engines are shown in Table I.

B. Preliminary Set Up

Calculations were made of the intake and exhaust volume flow for each engine to provide design criteria for the exhaust cooler and intake flame arresters. Wagner Mining Equipment Company constructed the exhaust cooler based on their extensive experience in scrubber design for small diesel engines. The prototype scrubber is about four times larger than units used on current underground equipment and is 44 inches high, 48 inches wide, and 38-1/2 inches deep. Two flanged 8-inch diameter pipes are provided at the inlet while the outlet is a rectangular 48-inch by 10-inch opening.

Two 8-inch diameter intake flame arresters were purchased from G. W. Lisk Company. This company manufacturers smaller units which are approved for use in current underground equipment and the test units are of identical design. The honeycomb configuration consists of 1-inch wide plates spaced .018 inch apart.

A water-jacketed exhaust system was built for each engine to connect the scrubber with the engine turbochargers. Adapters were also fabricated to connect the intake flame arresters with the turbochargers.

Schedule 31 requires that tests be conducted to demonstrate that methane explosions in the engine intake and exhaust systems do not ignite a fuel-air mixture external to the engine. The explosion tests are made with the complete engine enclosed in a box containing a combustible natural gas-air mixture. Tests are made both with the engine at rest and with the engine being externally driven. A 10 ft x 12 ft concrete pad was laid adjacent to the Engine Laboratory approximately 50 ft from the building. A 10 ft x 9 ft x 7-1/2 ft high box was fabricated from angle iron and steel plate and provided with two blow out panels on the top. Joints in the box were sealed with silicone glue and foam rubber strips were attached along the bottom of the box. An explosion-proof blower was used to recirculate the natural gas-air mixture. Natural gas was metered to the suction side of the blower and manual

TABLE 1

Test Engines Selected

CUMMINS VTA-1710

- 800 Hp at 2100 RPM
- 4-stroke cycle, direct injection
- Turbocharged and after-cooled
- 12-cylinder V-type engine
- 5.5 inch bore x 6 inch stroke - 1710 c.i.d.
- Dual turbochargers, dual 5-inch exhaust pipes
- Approximate weight, 5800 lbs
- Maximum Exhaust Flow 7800 lb/hr

CATERPILLAR D-348

- 920 Hp at 2000 RPM
- 4-stroke cycle, pre-combustion chamber
- Turbocharged and after-cooled
- 12-cylinder V-type engine
- 5.4 inch bore x 6.5 inch stroke - 1787 c.i.d.
- Dual turbochargers, single 11-inch exhaust pipe
- Approximate weight, 7000 lbs
- Maximum Exhaust Flow 9600 lb/hr

DETROIT DIESEL 12V-149TI

- 1200 Hp at 1900 RPM
- 2-stroke cycle, direct injection
- Turbocharged and after-cooled
- 12-cylinder V-type engine
- 5.75 inch bore x 5.75 inch stroke - 1788 c.i.d.
- Four turbochargers, dual 6-inch exhaust pipes
- Approximate weight, 8600 lbs
- Maximum Exhaust Flow 18000 lb/hr

two-way valves were used to direct the mixture flow. Figure 1 is a schematic of the facility.

During emission testing, the exhaust gas was analyzed for carbon dioxide, oxygen, carbon monoxide, hydrogen, methane, nitrogen, nitric oxide, oxides of nitrogen, and aldehydes. Carbon dioxide and carbon monoxide were measured in accordance with SAE Recommended Practice J177a, using Beckman non-dispersive infrared analyzers. Unburned hydrocarbons were measured according to SAE Recommended Practice J215, using a Southwest Research Institute flame ionization detector with heated oven. Nitric oxide and oxides of nitrogen were analyzed using a chemiluminescent detector. Oxygen was measured with a Beckman meter. H_2 and N_2 measurements were made with a gas chromatograph equipped with a thermal conductivity detector. Methane and several other reactive and non-reactive hydrocarbons were measured using a multi-column, time-sequenced gas chromatograph equipped with a flame ionization detector. Non-reactive hydrocarbons evaluated included methane, ethane, acetylene, propane, and benzene. The measurement of these non-reactive hydrocarbons required the baseline separation of a number of reactive hydrocarbons, including ethylene, propylene, and toluene. Aldehydes were measured using the two, 4-dinitrophenylhydrazine (DNPH) method. This method is a wet collection technique, followed by chemical extraction and analysis using a dual column flame ionization gas chromatograph. This method measured a number of individual aldehydes, including formaldehyde, acetaldehyde, proponal, acrolein, butanol, crotonol, pentanol, hexanol, and benzaldehyde.

C. Cummins Engine Tests

The VTA-1710 engine was installed in Cell 5 of the Engine Laboratory and coupled to a 1000 Hp eddy-current dynamometer. The engine water cooling system required two heat exchangers for temperature control with a deaeration tank at the top to purge trapped air. The temperature of the cooling water leaving the engine was automatically controlled within 2°F. For the initial tests, two 5-inch exhaust pipes were built to connect the turbocharger exhausts with the laboratory exhaust ducts. Meriam Laminar Flowmeters were attached to each turbocharger air inlet for measuring intake air flow. A Flo-Tron linear mass flowmeter with a range of 0-500 lb/hr was used to measure fuel consumption rate. Engine torque, speed, and all relevant temperatures were digitally displayed at the operator's panel (see Figure 2). Engine speed and load were maintained at a desired set point by an automatic controller. Twelve pressures and twelve temperatures were monitored and recorded during all tests. After initial checkout, the engine was subjected to a 50-hour run-in at full load, 2100 RPM. A slight adjustment to the fuel rate was necessary to bring the power output to 800 Hp at 2100 RPM, with a maximum torque of 2300 lb-ft at 1500 RPM.

Two natural gas lines were installed in the intake air filters of the engine. A Beckman NDIR hydrocarbon analyzer was modified and calibrated to detect natural gas in the 1.5% range required for the

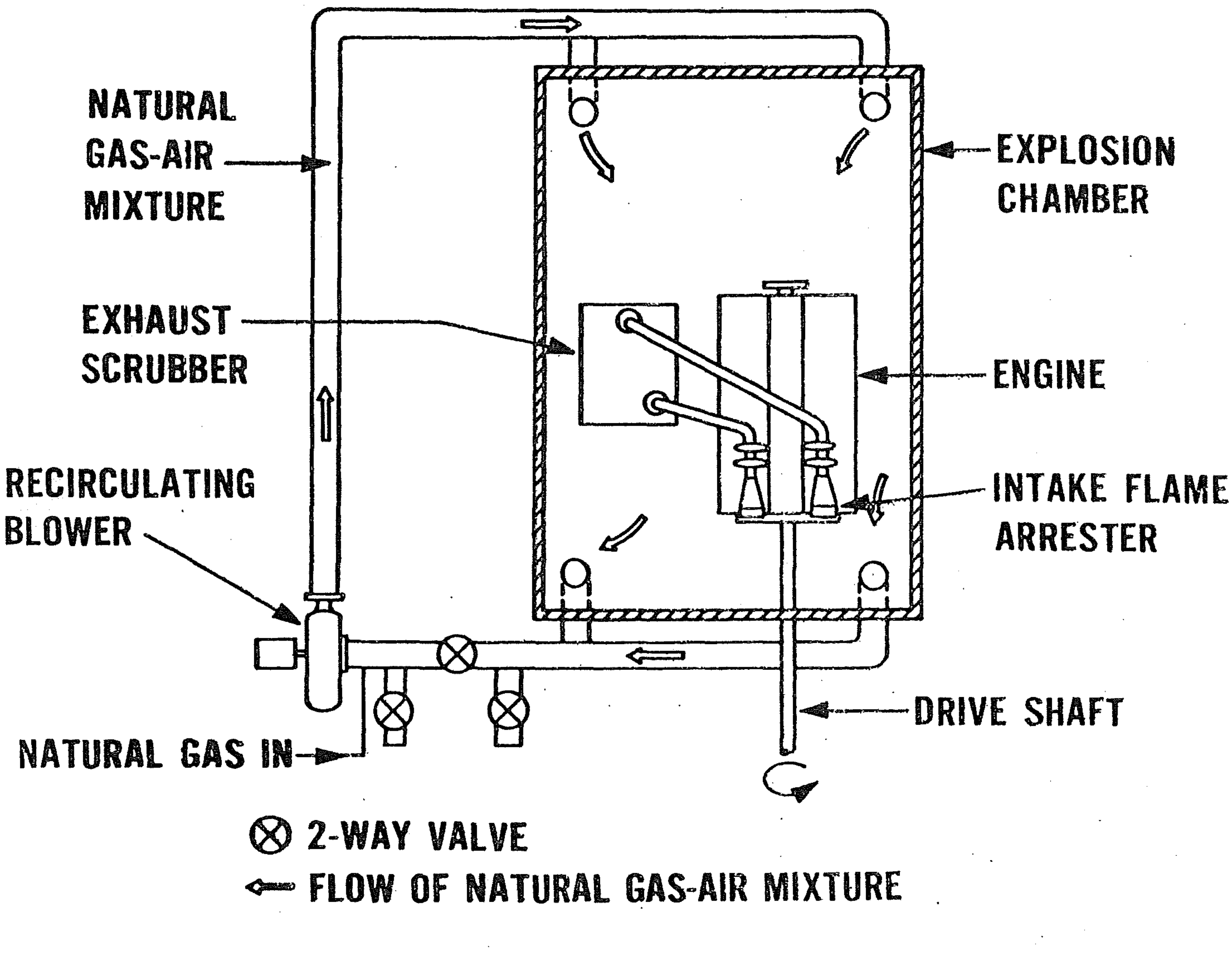

FIGURE 1.

SCHEMATIC OF EXPLOSION FACILITY

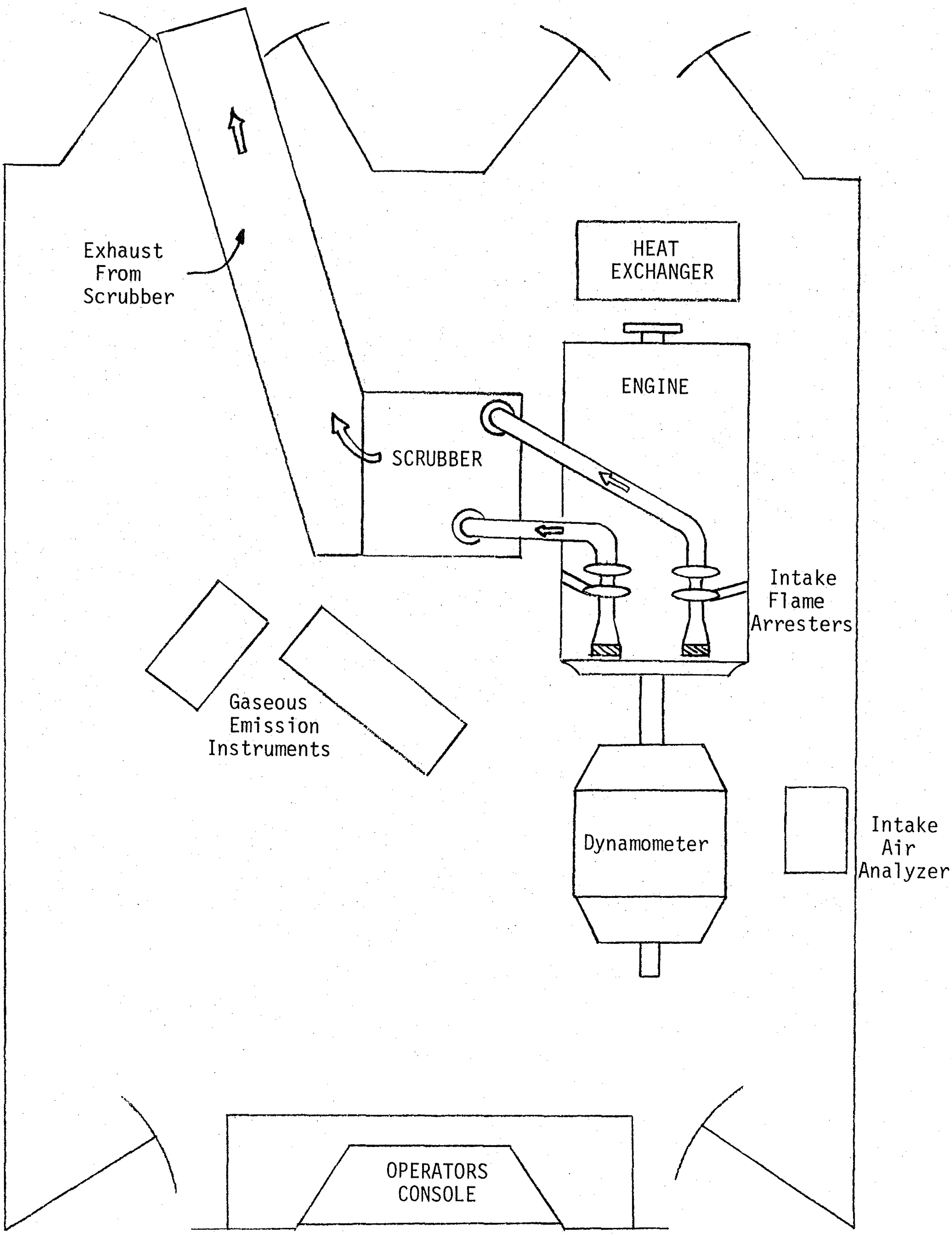

FIGURE 2 - SCHEMATIC DIAGRAM OF ENGINE TEST CELL

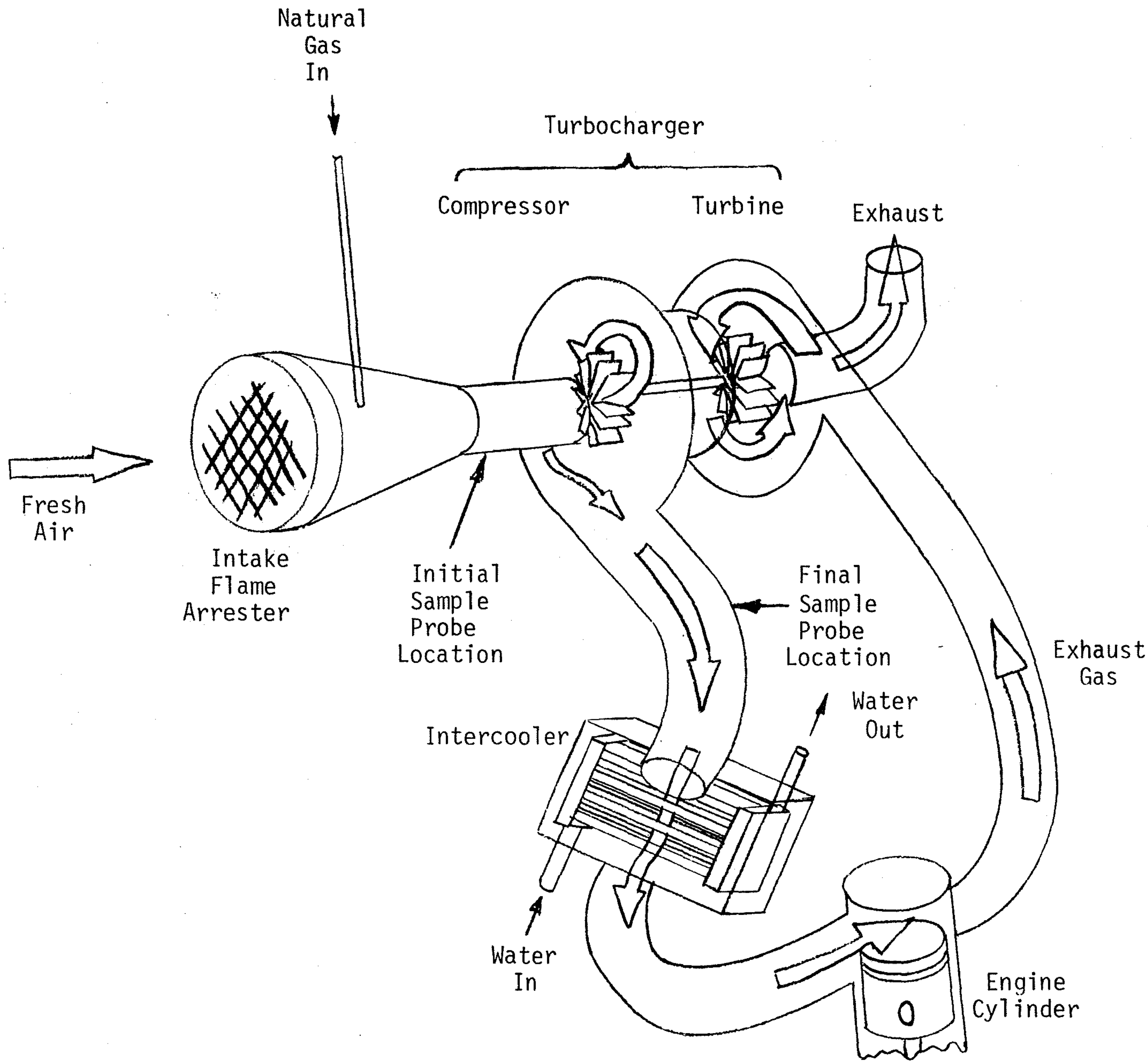

FIGURE 3 - SCHEMATIC DIAGRAM OF TURBOCHARGER AND INTERCOOLER

gaseous emissions tests. A single hole probe was placed in each intake air system about 18 inches upstream from the turbocharger compressor inlet with a two-way valve in the sample line to allow selection of the left or right bank. A series of runs was conducted at various speeds and loads to determine if the natural gas-air mixture was homogeneous across the diameter of the intake pipe. This information was vital in order to insure that the concentration of natural gas measured by the analyzer was in fact the actual concentration in the intake system. The single hole probe was traversed across the diameter of the intake pipe at one-half inch increments in both the left and right banks. Results showed that there was very severe stratification with approximately ten times higher gas concentration at the center of the pipe than at the outer edges. The probe was then located at a point midway between the turbocharger and aftercooler. The intake manifolds are oval shaped at this point and it was necessary to make the traverses in both the vertical and horizontal directions. Results from these data showed that the intake mixture was homogeneous at this point in both manifolds, since the turbochargers effectively mixed the air and natural gas. Multi-hole probes were then installed in each of the manifolds and were used to measure the concentration of natural gas in the intake air. Location of the inlet air probes is shown in Figure 3.

1. Gaseous Emissions Testing

A 1.5% (by volume) concentration of natural gas was introduced into the intake ducts to determine the effect on full load power output and to provide a final checkout of all systems. The 1.5% mixture is required by Schedule 31 and is designed to simulate a gassy mine atmosphere.

Preliminary gaseous emissions measurements were next made throughout the speed range of the engine at various loads. For these initial tests, measurements of the following contaminants were taken: CO, CO_2, NO_x, O_2, and unburned hydrocarbons. The engine was first operated at full load, 2100 RPM, and the restrictions were set to the maximum specified by the manufacturer: 25 in-H_2O intake depression and 3.0 in-Hg exhaust back pressure. Data were taken at 2100, 1900, 1700, and 1500 RPM and at 100%, 75%, 50%, 25%, and 0% load at each engine speed. At each test point the engine was first stabilized without introduction of natural gas and the readings of the emission instruments were recorded. Without changing the engine speed or throttle lever position, the 1.5% natural gas concentration was introduced and the instrument readings were again recorded.

After completion of the initial test runs, the remainder of the emissions instruments were installed and the engine was tested only with the natural gas-air intake mixture and diesel fuel. The same engine speeds and loads were used, but the no-load condition was eliminated and the idle speed was added.

2. Exhaust Gas Cooling Tests

The Wagner exhaust scrubber was installed on the engine in conjunction with the SwRI designed water-jacketed exhaust piping. A standpipe was attached to the outside of the scrubber and calibrated in five gallon increments. Water consumption of the scrubber was measured by a turbine flowmeter and read out on a frequency counter. The exhaust temperature into and out of the scrubber was continuously monitored by calibrated thermocouples. Natural gas was introduced into the intake air system at a concentration of 0.5%, by volume, as required by Schedule 31. The initial test was made with a scrubber water level of 110 gallons. The scrubber outlet was inside the test cell and upon engine start-up the cell filled with water-saturated exhaust gas. A large twenty ft-long duct was then built to carry the exhaust outside the laboratory. At full load and rated speed, the 110 gallon level was too high and entrained liquid water sprayed from the scrubber outlet. The scrubber water volume was next decreased in five gallon increments. The largest volume at which the scrubber could operate without entraining excess water was visually determined to be 105 gallons. The water level was further reduced to determine the low water level. At any water level less than 37 gallons, the exhaust temperature out of the scrubber rose dramatically. The water level difference between the high and low limit was ten inches. At all water levels between the high and low limits, the exhaust temperature out of the scrubber was 153-154°F.

3. Intake Flame Arresters

The honeycomb type flame arresters were next installed on the engine to determine if they were adequately sized for the engine air flow. The design pressure drop across the flame arrester was 10 in-H_2O, but at 800 Hp, 2100 RPM, the measured pressure drop was 14.1 in-H_2O.

4. Surface Temperature Tests

The Dynarad thermal imager was next used to determine which external surfaces of the engine exceeded the Schedule 31-specified maximum of 302°F. The Dynarad infrared camera collects and detects an object's thermal radiation, converts the infrared intensity into an electrical signal and displays the signal intensity of each point in the target image 60 times per second on an oscilloscope for a TV-like presentation.

The Dynarad system allowed a scan of the engine for high temperatures but it was impossible to take good photographs of the oscilloscope picture because the turbocharger was so hot that it caused a bright white spot on the print. Since the only excessive surface temperatures were the turbocharger and its adjacent hardware, no prints could be made which showed adequate details. In addition, the camera could not be tilted more than 10° from the horizontal direction and this

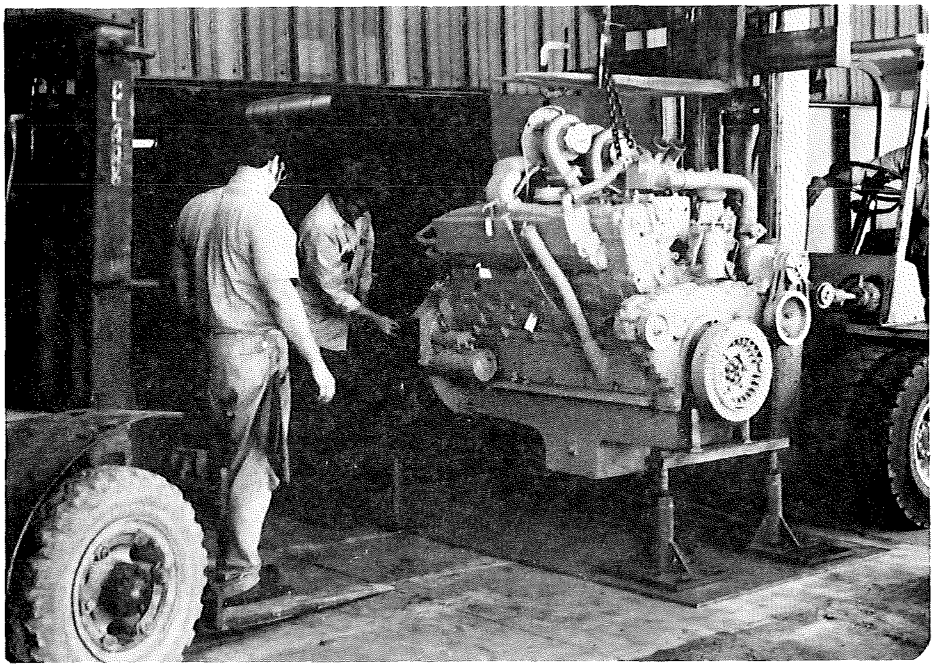

FIGURE 4 - PREPARATION FOR INSTALLATION
OF CUMMINS VTA-1710

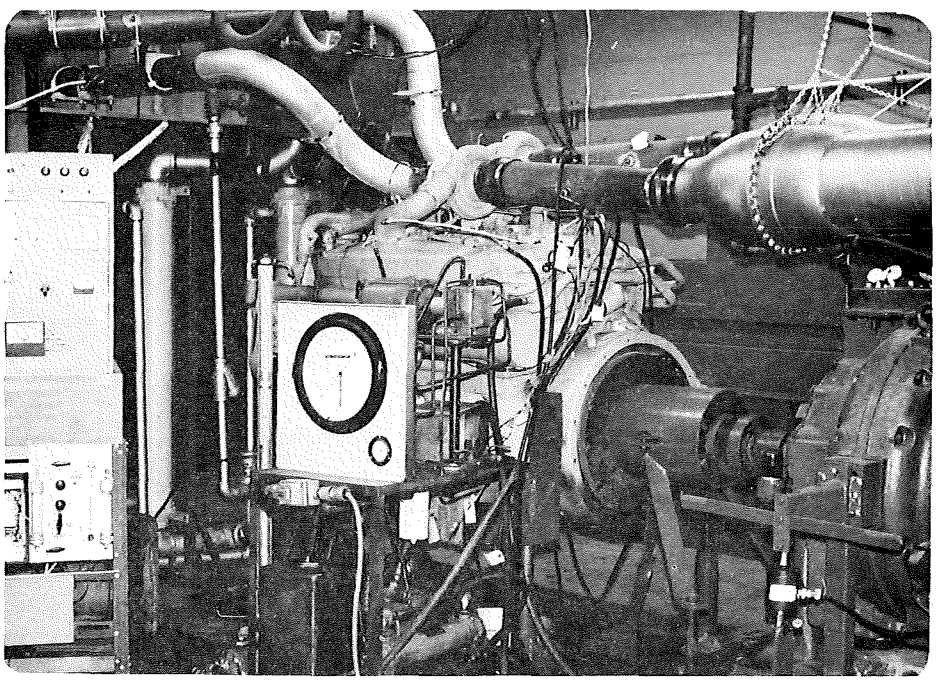

FIGURE 5 - LABORATORY INSTALLATION OF CUMMINS VTA-1710

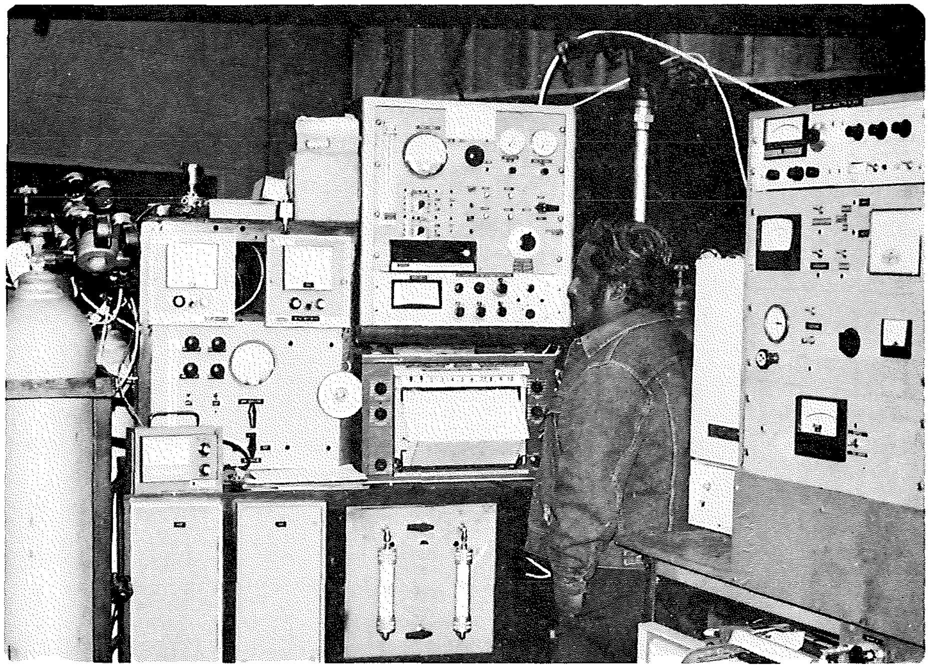

FIGURE 6 - GASEOUS EMISSIONS INSTRUMENTATION

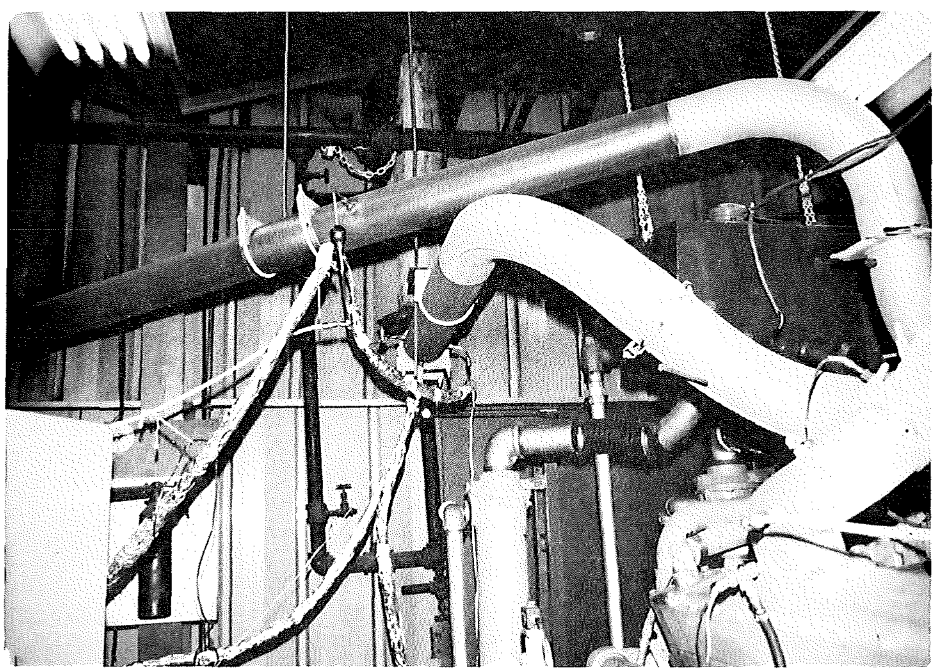

FIGURE 7 - TYPICAL SAMPLE PROBE LOCATIONS

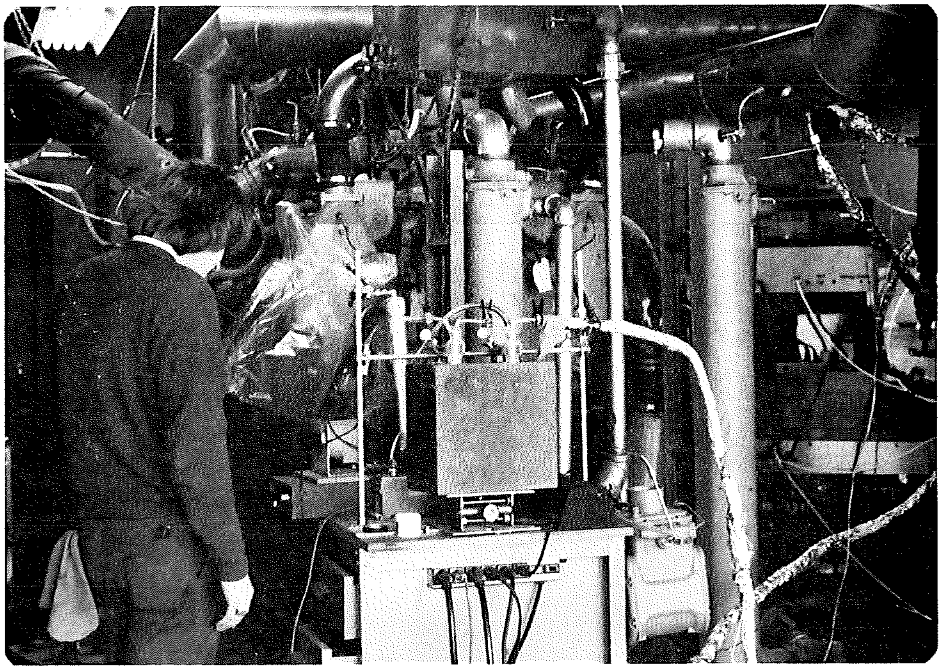

FIGURE 8 - TYPICAL ALDEHYDE INSTRUMENTATION

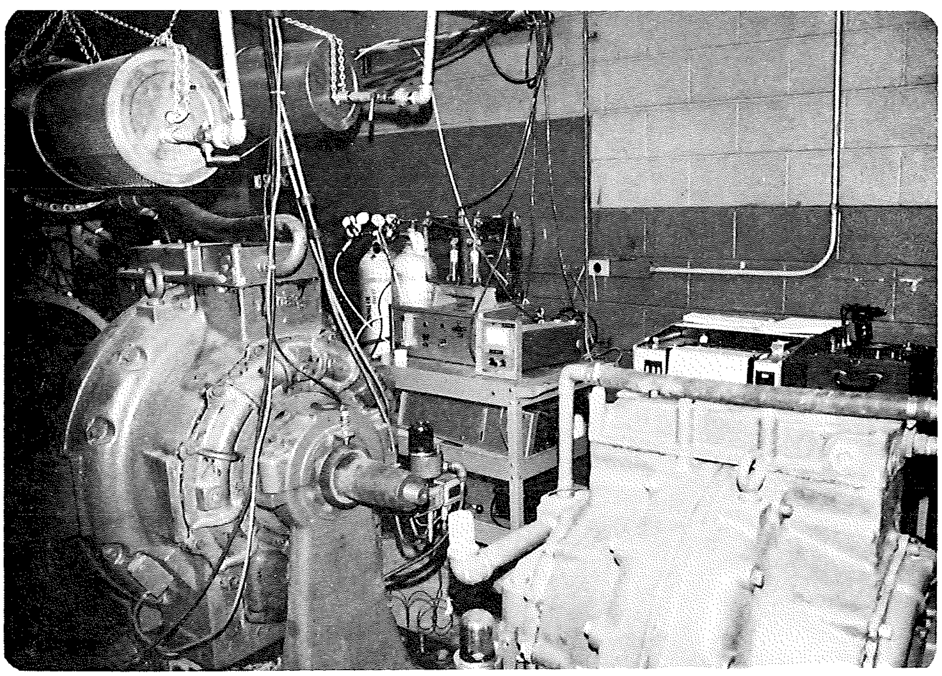

FIGURE 9 - INTAKE AIR SAMPLING INSTRUMENT

would not allow a thorough investigation of the temperatures within the V of the engine where the exhaust manifolds are located. Since the Dynarad system proved impractical, it was not further used.

The engine was next sprayed with a heat sensing paint which liquified when a temperature of 300°F was exceeded. The paint was applied in all areas expected to be near or above the temperature limit. This included the complete water-jacketed exhaust system, the inlet to the scrubber, the exhaust manifolds at their mating surfaces with the heads, the complete turbocharger, and adjacent manifolds. In addition, a set of temperature sensitive crayons were used to further identify the various surface temperatures. The crayons melted at 25°F intervals from 125°F to 800°F. The engine was then operated at full load, rated speed for a minimum of 20 minutes to insure temperature stabilization. Notes were made of all surfaces which exceeded 302°F and the crayons were used to find the extent to which that limit was exceeded.

D. Caterpillar Engine Tests

The D348 engine replaced the Cummins VTA-1710 on the dynamometer stand. Based on the Cummins engine results, it was expected that the power output of the D348 would be above the 1000 Hp capacity of the dynamometer. Accordingly, a 500 horsepower eddy-current dynamometer was added in tandem behind the 1000 horsepower dynamometer. A flexible coupling connected the two dynamometers and a second load cell readout system was added to the operator's panel. During testing, the load on the smaller dynamometer was set to a fixed value by one controller, while the load on the larger dynamometer was varied with the other controller to reach a desired setpoint (see Figures 11 and 12).

After initial startup and checkout, the engine was given a short break-in. Probes for determining the natural gas concentration in the intake air were again located in the connecting duct between the turbochargers and aftercoolers. A single-hole probe was traversed across the diameter of each connecting duct to verify that the intake air-natural gas mixture was homogeneous. Runs were made at numerous loads and speeds, and results showed that the mixture was quite uniform throughout the duct. Multihole probes were used for all subsequent measurements.

1. Gaseous Emission Testing

Maximum power measurements were made and then repeated with the introduction of a 1.5% (by volume) concentration of natural gas in the intake duct. Testing continued with the measurement of preliminary gaseous emissions with and without natural gas addition throughout the speed range of the engine. The test procedure was identical to that used on the Cummins engine. The maximum restrictions specified by Caterpillar were set while the engine was operating at full load, rated speed: 30 in-H_2O intake depression and 2.0 in-Hg exhaust back pressure.

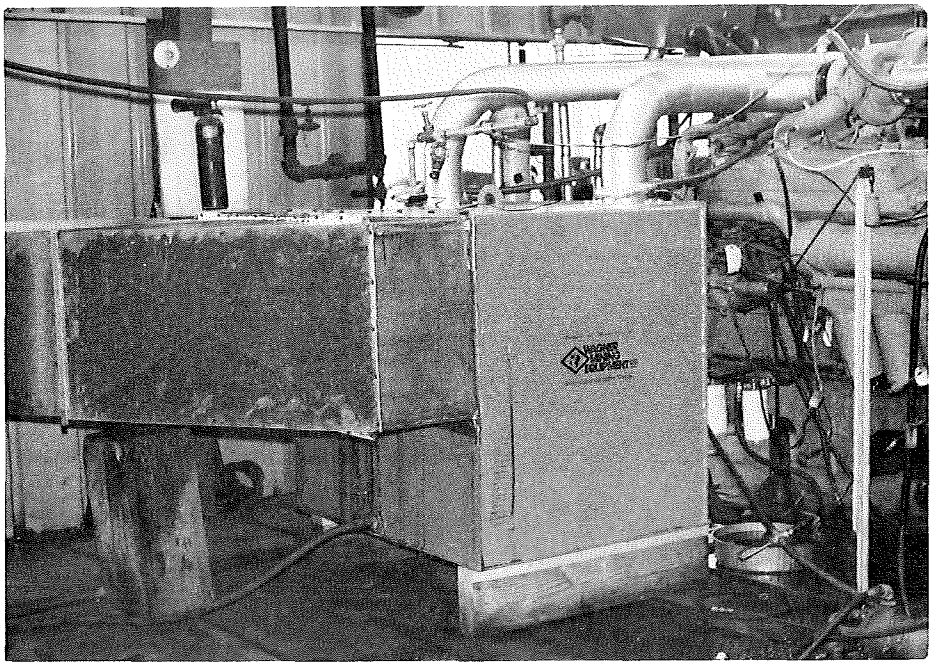

FIGURE 10 - EXHAUST SCRUBBER ATTACHED TO CUMMINS VTA-1710

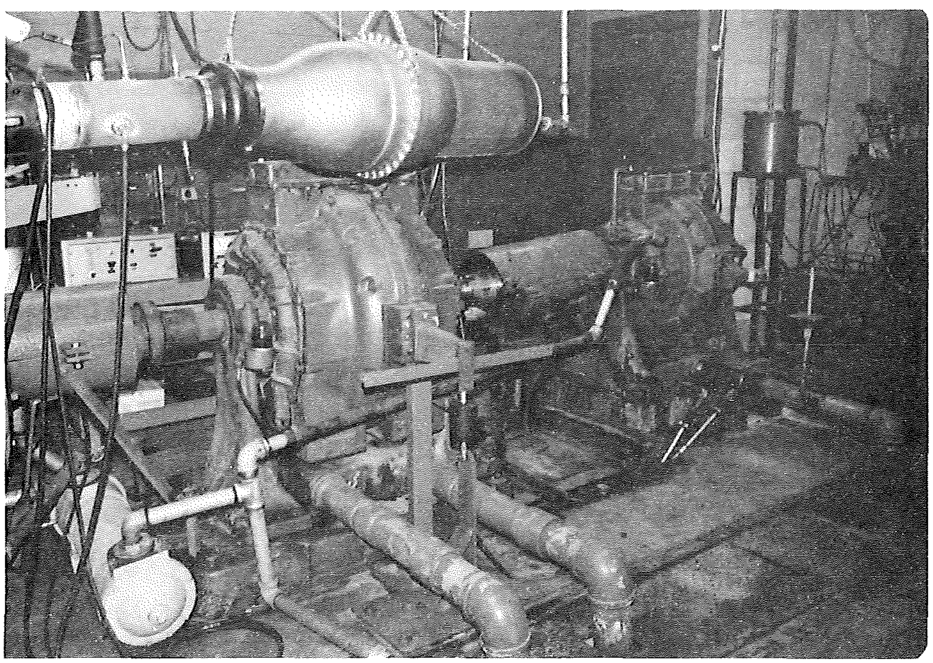

FIGURE 11 - TANDEM DYNAMOMETER SETUP

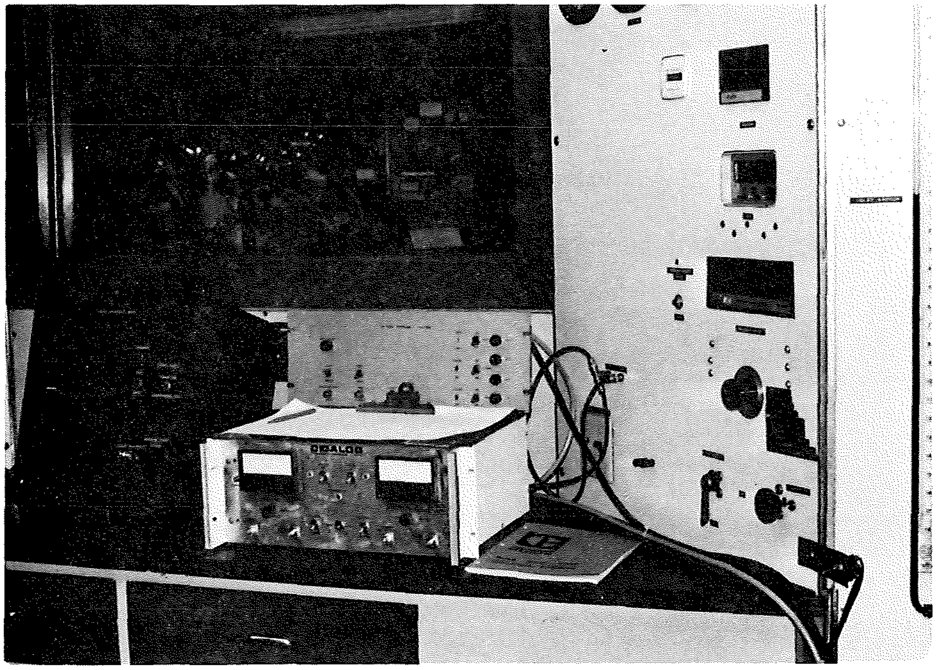

FIGURE 12 - DUAL CONTROLLERS USED FOR TANDEM DYNAMOMETERS

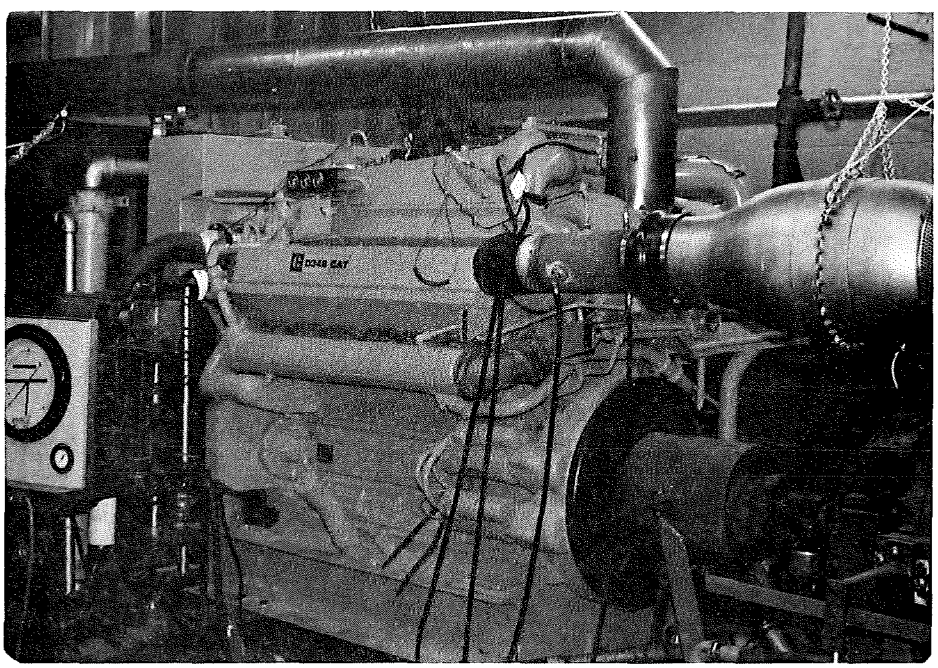

FIGURE 13 - SIDE VIEW OF CAT. D348 INSTALLATION

Data was taken at 2000, 1800, 1600, 1500, and 1400 RPM and at 100%, 75%, 50%, 25%, and 0% load for each engine speed.

The remaining emission instruments and sample lines were next installed and the engine was tested only with the natural gas-air mixture and diesel fuel. Procedures used were as specified in the Schedule 31 Regulations.

2. Exhaust Gas Cooling Test

The Wagner exhaust scrubber previously used on the Cummins engine was next installed on the Caterpillar engine in conjunction with a second set of SwRI-designed water-jacketed exhaust pipes. Instrumentation and test procedures were identical to previous tests. The low water level at which the scrubber failed to operate was 37 gallons. The high water level was again determined to be at the 105 gallon level. At all levels between these limits exhaust temperature out of the scrubber was 149-150°F, while exhaust temperature into the scrubber was 750-760°F.

3. Intake Flame Arresters

Pressure drop across the flame arresters was measured with the engine running at full load, rated speed to determine if the air flow restriction was excessive. The measured restriction was 14.1 in-H_2O.

4. Surface Temperature Tests

Heat sensing paint was sprayed on all exposed areas which could possibly be at, or above, the 302°F limit. Evaluation techniques and test procedures were identical to those used on the VTA-1710.

E. Detroit Diesel Engine Tests

The final dynamometer tests were made on the 12V-149TI engine. With a rated output of 1200 Hp at 1900 RPM, this was the largest engine tested and the tandem dynamometer setup used in the previous tests was required. The four turbochargers presented additional problems in constructing the intake and exhaust systems and in maintaining the correct natural gas concentration in the intake air. A pair of two-way valves was used to select the desired sample location when monitoring the natural gas concentration with the Beckman hydrocarbon analyzer. For all tests the flow restrictions were set to the maximum recommended by Detroit Diesel, with the engine operating at full load, rated speed: 20 in-H_2O intake restriction and 2.5 in-Hg exhaust back pressure.

1. Gaseous Emissions Tests

Comparisons were made of gaseous emissions with and without the 1.5% natural gas concentration in the intake air. Data was obtained at 1900, 1800, 1700, and 1600 RPM and at 100%, 75%, 50%, 25%, and 0% load at each speed. A final set of test runs was made with evaluations of all emissions required in the Schedule 31 Regulations.

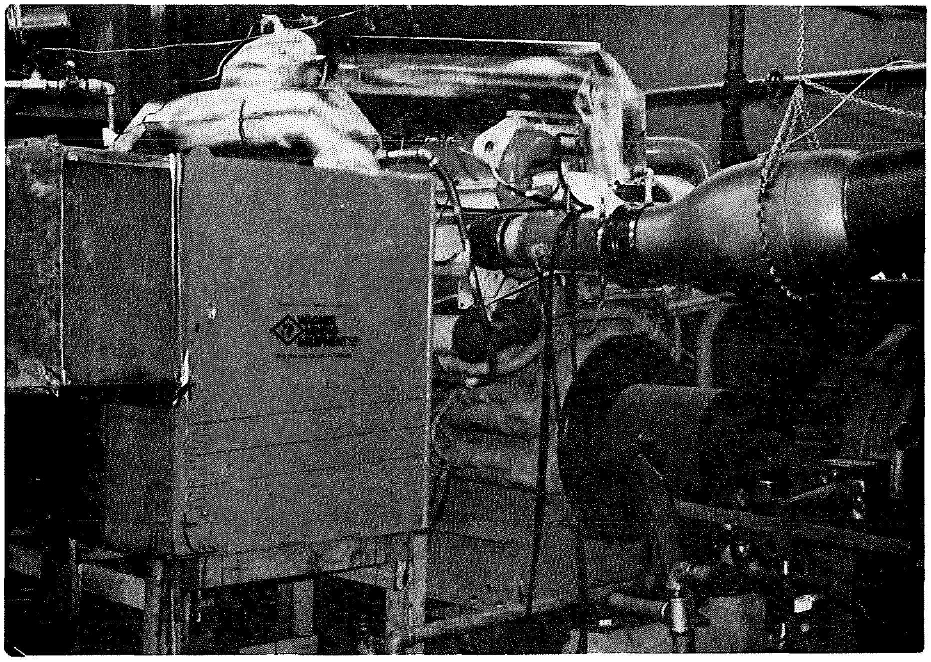

FIGURE 14 - SIDE VIEW OF CAT. D348 WITH SCRUBBER ATTACHED

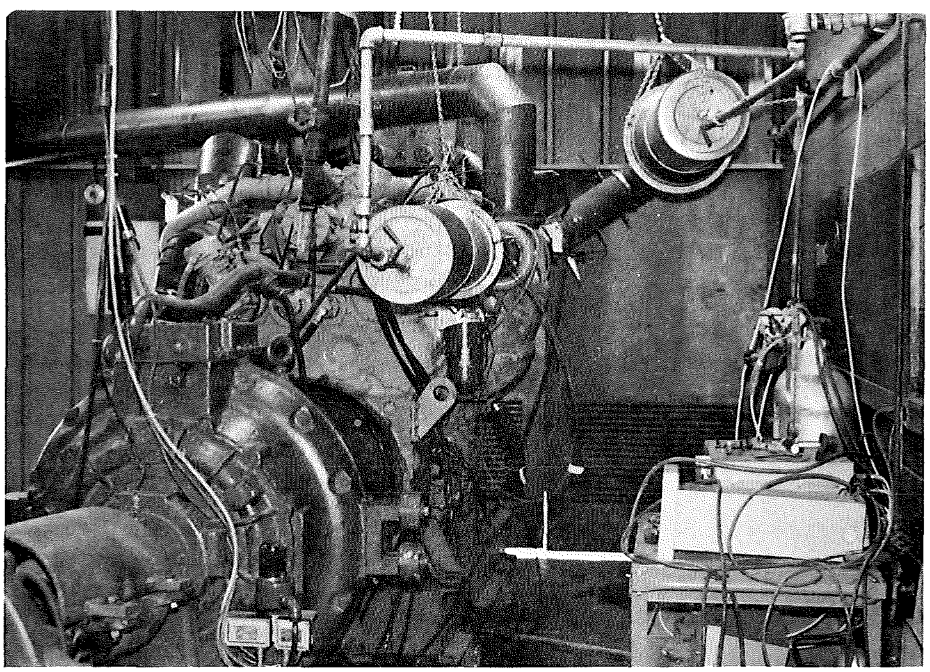

FIGURE 15 - REAR VIEW OF DETROIT DIESEL 12V-149TI INSTALLATION

2. Exhaust Gas Cooling Tests

The two-cycle diesel requires about 35% more air flow than a comparable four-cycle diesel. The total exhaust flow of the 12V-149TI was almost twice that of the Caterpillar D348, and this made it impractical to design a single scrubber for exhaust cooling. In a vehicle application two scrubbers would probably be necessary. Since both banks of the V-block engine are essentially identical, it was decided to install the scrubber only on the left bank of the engine so that only half the exhaust flow was ducted into it. The back pressure due to the scrubber was duplicated on the right side by adjusting a baffle in the exhaust pipe. Instrumentation and test procedures were the same as the previous two tests. The low water level at which the scrubber outlet temperature increased above the 170°F limit was 35 gallons. The high water level was determined to be 105 gallons for a difference in water level of 10.5 inches. Exhaust back pressure due to the scrubber was 23.4 in-H_2O with an 80 gallon water level. At all levels between the high and low water limits the exhaust temperature out of the scrubber was 147-149°F, while exhaust temperature into the scrubber was 770-790°F. Average horsepower during the runs was 1236 at 1900 RPM.

3. Intake Flame Arresters

Pressure drop across the flame arresters with the engine operating at full load, 1900 RPM, was 12.7 in-H_2O. A second measurement taken at the inlet to the converging duct confirmed that the majority of this restriction is caused by the converging duct transition piece between the flame arrester and turbocharger. The use of larger diameter intake piping would reduce this pressure drop to six in-H_2O.

4. Surface Temperature Tests

The same equipment and procedures used for the Caterpillar engine were duplicated on the Detroit Diesel engine to determine hot spots on external engine surfaces.

F. Explosion Tests

The Caterpillar engine was the first to be installed in the explosion test facility.

Concentration of the natural gas in the air was continuously monitored by a Beckman analyzer calibrated with bottles containing known methane concentrations. An initial explosion test in the entire enclosure was planned in order to determine the structural integrity of the box. The spark plug was suspended in the middle of the box and several attempts were made to ignite a measured 8% natural gas-air mixture. These attempts were unsuccessful. A "T" and petcock were placed in the analyzer sample line after the sample pump. When the analyzer indicated a combustible mixture was present in the box, the petcock was opened and an attempt was made to ignite the mixture from the sample line. The mixture strength indicated by the analyzer was approximately 30% in error. After shutting

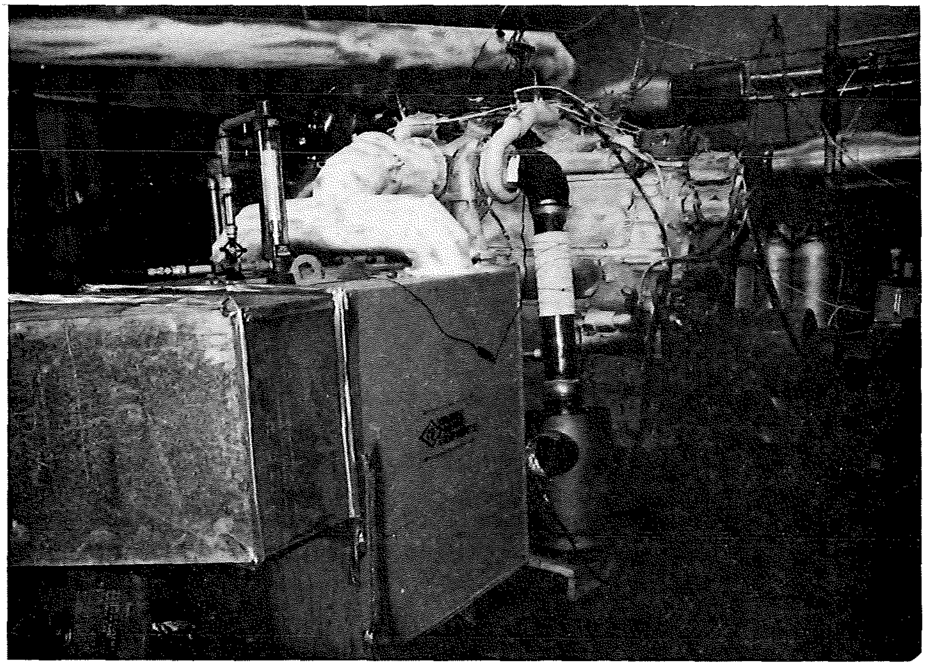

FIGURE 16 - SIDE VIEW OF DETROIT DIESEL 12V-149TI
WITH SCRUBBER ATTACHED

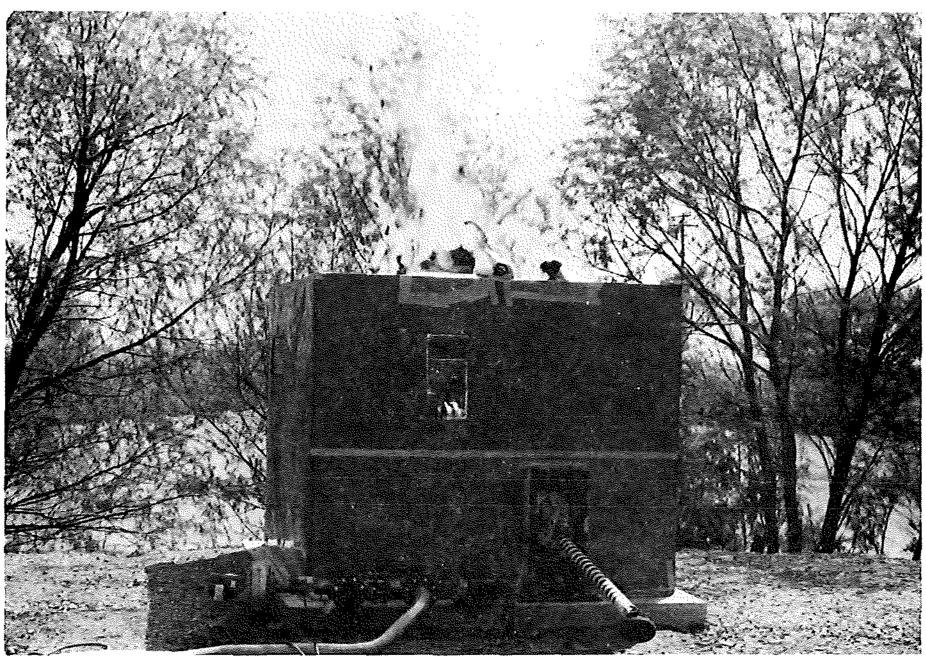

FIGURE 17 - TYPICAL EXPLOSION OF NATURAL GAS MIXTURE

off the recirculating fan and natural gas, the spark plug was triggered and the mixture exploded. Flames shot out of the top of the box about 10 ft, but the box suffered no damage. Figure 17 depicts this initial explosion. The next day the same procedure was followed and the mixture exploded again.

Numerous checks were made of the Beckman analyzer and sample lines to attempt to determine the reason for the large error in measuring methane concentrations in the explosion box. At the lower concentrations used for the emissions tests and cooling tests, the instrument read correctly. At the higher concentrations used for the explosion tests the instrument calibrated correctly, but when sampling from the box a consistently low reading was indicated. Bag samples were taken from the box at different methane concentrations and analyzed on a gas chromatograph to develop a new calibration curve for the Beckman analyzer.

Explosion tests were first conducted on the Caterpillar intake system. A high energy ignition system was used to fire a projected tip spark plug located between the turbocharger and the flame arrester, with a pressure transducer and thermocouple located in the same area. The output of the pressure transducer was recorded on a high-speed Honeywell oscillograph. Temperatures were monitored with a L&N dial readout. The explosion test facility is shown in Figures 18-20. The explosion box was charged to a ten percent methane/air mixture by adding methane in the duct of the explosion box air recirculating system. An 8-ft drive shaft was attached to the engine flywheel and to the power take-off unit of an agricultural tractor. The engine was motored over for about 15 seconds to draw methane into the intake system. The engine was then stopped, and the spark plug was triggered. An audible explosion could be heard. This procedure was repeated until the concentration of the mixture in the box was below eight percent. Eight to ten explosions could be made before it was necessary to recharge the box with a fresh air-methane mixture. Twenty shots were made with no ignition of the surrounding atmosphere in the box. Maximum pressure in the engine intake system observed during these static tests was 6 psi.

Dynamic tests of the intake system were next made by continuing to motor the engine while the spark plug was triggered. An audible explosion could be heard and the tractor shook from the force of the internal explosion. Maximum pressure in the intake system observed during these dynamic tests was only about 1 psi. Twenty successful shots were also made under these dynamic engine conditions. The flame arresters were removed for inspection and it was discovered that the compressor blades of one turbocharger were partially melted from excessive heat. This was undoubtedly caused by motoring the engine after a flame was initiated in the intake system during the dynamic tests, thus causing a continuous flame to be established in the intake system.

The spark plug and pressure transducer were next placed in the exhaust system to test the effectiveness of the scrubber as a flame arrester. The test procedure was essentially the same as used on the intake system. Ten static tests were made with 80 gallons of water in the exhaust scrubber with no explosion of the surrounding mixture. Maximum observed pressure in the exhaust system was 30 psi.

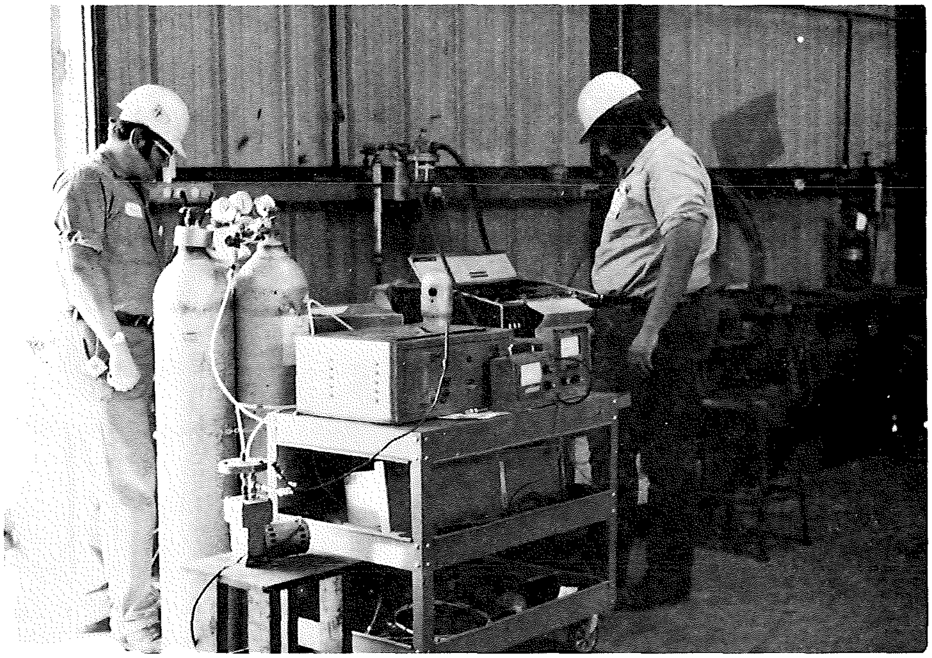

FIGURE 18 - INSTRUMENTATION FOR EXPLOSION TESTS

FIGURE 19 - TRACTOR ATTACHED FOR MOTORING ENGINES

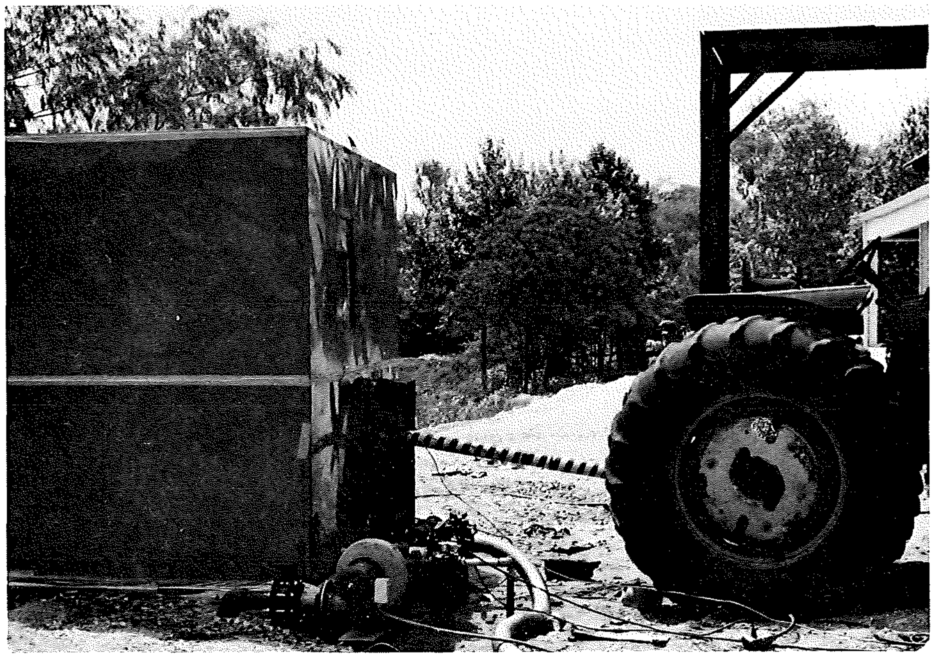

FIGURE 20 - EXPLOSION CHAMBER WITH ASSOCIATED PLUMBING

The water volume was next reduced to 50 gallons in an attempt to find the low water volume at which an explosion of the surrounding mixture would occur. On the first attempt, there was a violent explosion of the surrounding mixture and the box was badly damaged, although there was no apparent damage to the engine or scrubber. Four additional blow-out panels were added to the rear of the box on the side opposite the tractor, and bent side panels were straightened out and resealed. An initial run was made with 60 gallons of water in the scrubber after the box was rebuilt. This level also resulted in a strong explosion on the first attempt, but the box was not damaged due to the additional blow-out panels. Numerous other runs were made at 10 gallon increments up to 110 gallons. Each time the box exploded. At 110 gallons, 20 static tests and 13 dynamic tests were made before an explosion occurred.

The entire exhaust system was then pressurized and a close inspection was made for leaks. A significant leak was noted at a slip joint between the exhaust manifold and the turbocharger. Visual observations confirmed that the explosions initiated from this point, and the failure of the engine to pass the explosion test is attributed to this leak.

The Caterpillar engine was removed and replaced by the Cummins engine, with the spark plug initially located in the exhaust system. Tests to determine the low water level without an external explosion began with 70 gallons of water in the scrubber. The first static test initiated an explosion and the level was then raised to 80 gallons. At this level, twelve static internal explosions were made before the surrounding mixture was ignited on the last test. Ten tests were run at the 90 gallon level and nine tests at the 110 gallon volume before ignition of the mixture surrounding the engine. At 125 gallons, twenty-one static tests were made with no explosion, but the surrounding mixture did explode on the eighteenth dynamic test. The water volume was increased to 135 gallons and this proved to be an acceptable level. Twenty-five dynamic tests and thirty static tests produced no external explosion, with a maximum internal pressure of 40 psi for the dynamic tests and 58 psi for the static tests.

A similar set of twenty static and dynamic tests were conducted on the intake system with no external explosions. Maximum pressures observed were 4 psi and 2 psi, respectively. Both exhaust pipes were then removed from the scrubber and routed out the top of the box. The end of each pipe was left open but was sealed around its circumference at its exit from the box. This configuration was designed to investigate the effect on the exhaust system when an explosion was initiated in the intake system. Since the combustible mixture was being drawn into the engine and then expelled out of the exhaust pipe, only three or four explosions could be attempted before the concentration of the natural gas in the box fell below eight percent. After a static explosion in the intake system, white smoke trickled out of both exhaust pipes, and additional smoke was emitted during the motoring for the next shot. It is believed that the smoke from the explosion migrated into a cylinder with a partially open intake valve, around a partially open exhaust valve, and out the exhaust pipe. During motoring, the additional smoke may have been due to residual smoke from the explosion or partially-burnt methane which was being drawn into the engine.

The Detroit Diesel replaced the Cummins in the explosion chamber. The two available flame arresters were placed on the right front and right rear turbochargers and the inlets of the two turbochargers on the left side were sealed with reinforced tape to prevent air intake. During normal engine operation the two rear turbochargers direct air into one blower and the two front turbochargers direct air into the other blower. Thus, to motor the engine it is only necessary to have turbochargers on one side of the engine with open intakes. The scrubber was attached to the exhaust piping on the left side of the engine and the right exhaust was left open. A 135 gallon water volume was used in the scrubber with the spark plug in the left exhaust. On the third dynamic shot, the external mixture exploded. It was suspected that a flame had propagated back into the engine, into a cylinder with an open exhaust valve, through the air box, into a cylinder on the right side of the engine, and out the right exhaust.

To confirm this the right exhaust pipe was routed out the top of the box. This configuration also tended to quickly deplete the methane concentration in the box and allowed a maximum of four shots. Twenty static explosions were initiated in the left exhaust, and the exposed right exhaust pipe consistently emitted a flame with an accompanying noise similar to a rifle shot. Maximum pressure during these tests was 47 psi. Twenty dynamic explosions followed this with a maximum pressure of 40 psi and <u>no</u> external explosion in the box.

The spark plug was transferred to the right rear intake and the required series of twenty static and twenty dynamic tests were run with no resultant ignition of the surrounding mixture. A maximum pressure of 5 psi was observed during both tests. The water volume in the scrubber was next reduced to 125 gallons and the spark plug was relocated in the exhaust to determine if this engine required the same low water level as the Cummins. After nine dynamic shots, it became impossible to initiate an explosion in the exhaust system. The natural gas-air mixture and spark plug were checked and it was subsequently discovered that smoke was coming <u>out</u> of the intake system during motoring. From past experience it was expected that a blower drive shaft was broken. The blowers, intercoolers, and associated plumbing were removed and a broken blower drive shaft was discovered on the rear blower. An inspection of the shaft revealed a torsional overload with a subsequent fracture at the blower end. The cause was undoubtedly the severe shock loading caused by the repeated explosions inside the engine. Replacement parts were not available from any of the local distributors and had to be ordered from the factory.

IV. RESULTS AND DISCUSSION - PHASE I

A. Gaseous Emission Tests

Figures 21 through 23 depict the effect on power output when adding 1.5% natural gas to the intake air with the engines operating at full load. All engines experienced an 11-12% increase in power at rated speed, with the Cummins changing from 796 to 887 horsepower, the Caterpillar from 913 to 1026 horsepower, and the Detroit Diesel from 1198 to 1340 horsepower.

Results of the preliminary gaseous emission runs to determine the effect of adding 1.5% (by volume) natural gas to the intake air are shown in Figures 24 through 29. At identical power output, the addition of natural gas caused a slight decrease in oxides of nitrogen emissions of the Cummins engine, probably because of the decrease in available oxygen. Comparing identical liquid fuel rates, NO_x production is higher with natural gas added. CO emissions were consistently higher with the natural gas added than without it, particularly at the 1500 RPM peak torque speed under full load. This indicates insufficient oxygen present to complete combustion, which was verified by a significant increase in unburned hydrocarbons when the natural gas was present.

At identical power output, the Caterpillar D348 engine also displayed virtually no change in NO_x emissions when natural gas was added. At identical liquid fuel rates, NO_x production is 5-10 percent higher when natural gas is added. CO emissions <u>with</u> natural gas were higher at all loads and speeds, particularly at the lighter load points. NO_x emissions of the Detroit Diesel engine also were unaffected by the addition of natural gas at identical power outputs, but CO emissions were higher at all points. It should be noted that the highest CO emissions for the Caterpillar engine were at no-load conditions, while those of the Detroit Diesel and Cummins were at full-load conditions. CO_2 and O_2 were unchanged for all engines on an identical power basis, however, on an identical liquid fuel basis, the CO_2 increased with a corresponding decrease in O_2. A complete tabulation of the data are shown in Appendix A.

After completion of the initial runs, a second series of runs was made for each engine according to the requirements of Schedule 31. The exhaust gas was analyzed for CO_2, O_2, CO, H_2, CH_4, N_2, NO_x, and aldehydes. Figures 30 through 35 illustrate these results and the complete data are presented in Appendix B with graphical presentation in Appendix D. Results of these tests were used to calculate the air ventilation (cubic ft per minute) which would be required if these engines were used underground in mobile equipment. Schedule 31 specifies that the diluted mixture of exhaust gas and ventilation air shall not contain more than 0.25% CO_2, 0.005% CO, and 0.00125% NO_x (percent by volume). This requirement made the ventilation rates dependent on the NO_x concentrations and for all engines this occurred at full load and rated speed, where intake air flow is the highest (see Table 2).

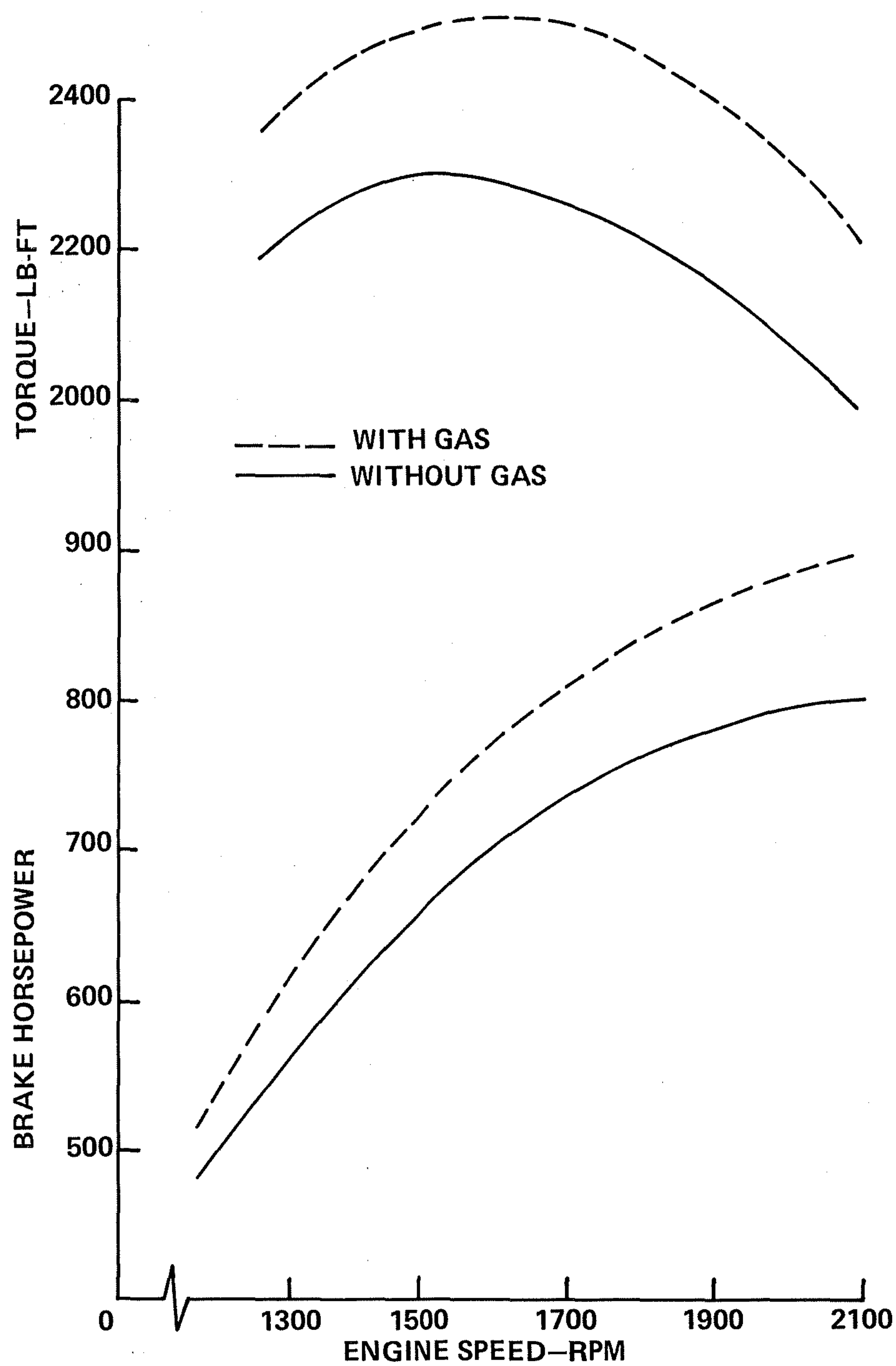

FIGURE 21 - EFFECT OF 1.5% NATURAL GAS ON FULL LOAD POWER OUTPUT OF CUMMINS VTA-1710 ENGINE

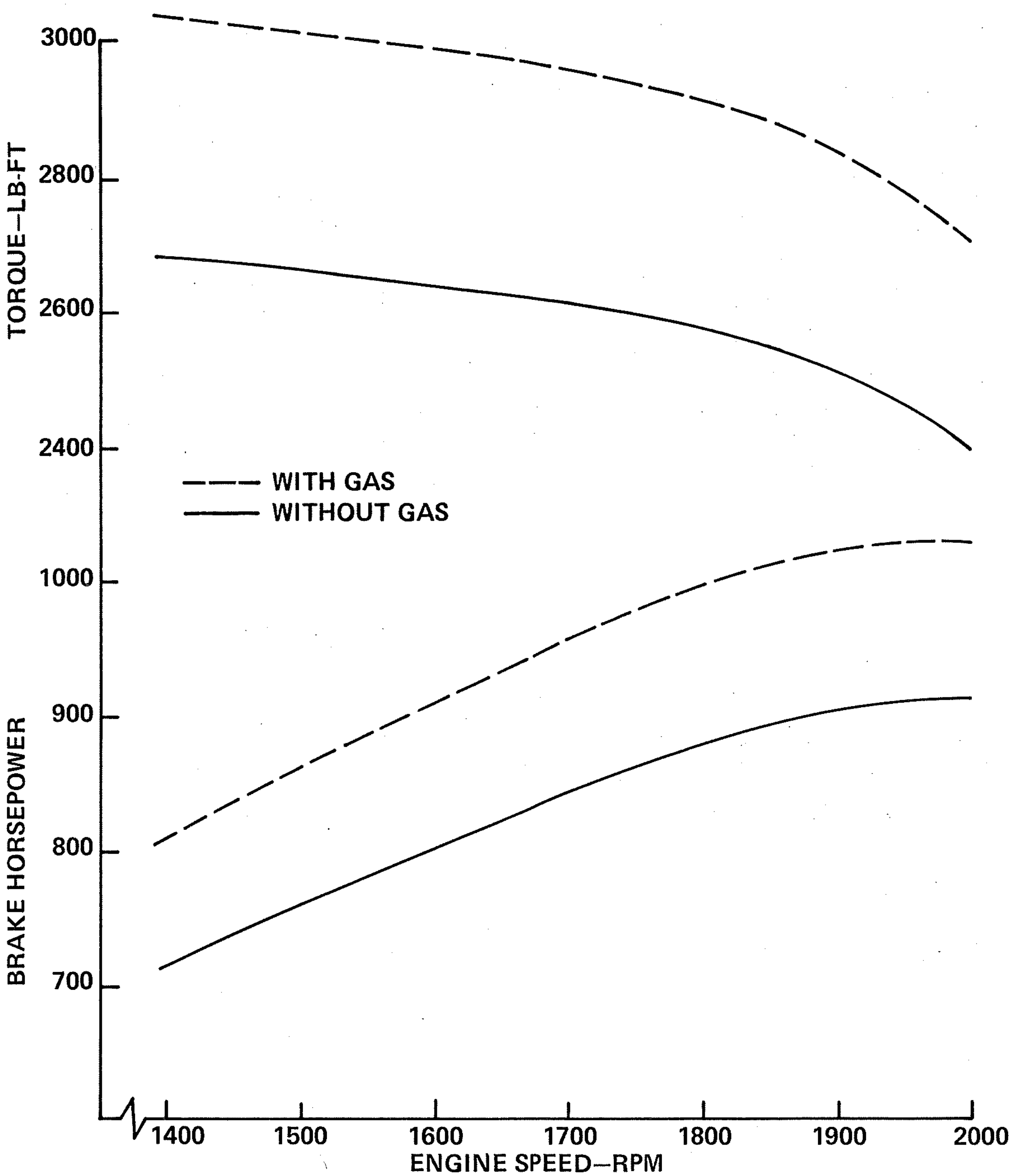

FIGURE 22 - EFFECT OF 1.5% NATURAL GAS ON FULL LOAD POWER OUTPUT OF CATERPILLAR D348 ENGINE

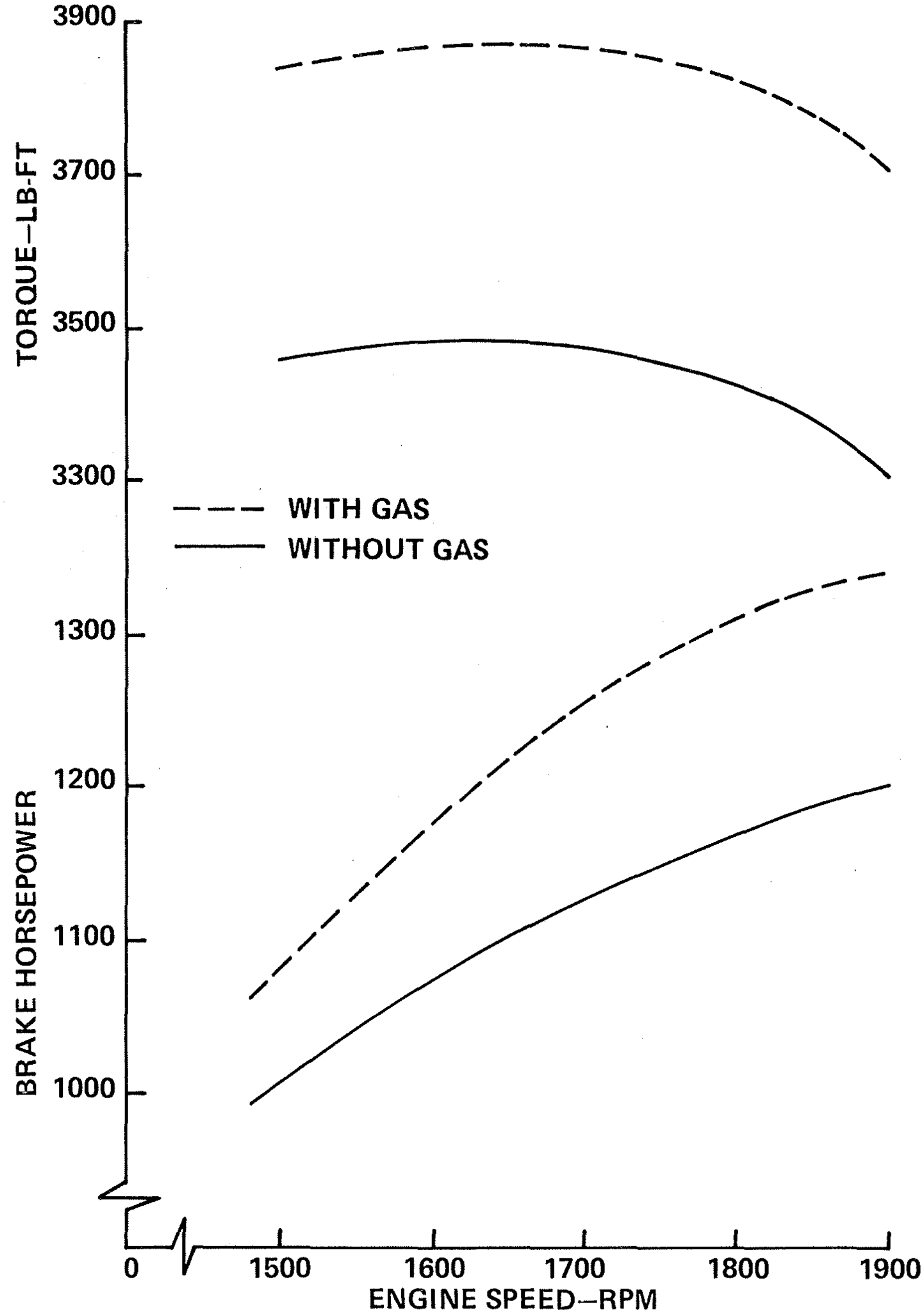

FIGURE 23 - EFFECT OF 1.5% NATURAL GAS ON FULL LOAD POWER OUTPUT OF DETROIT DIESEL 12V-149TI ENGINE

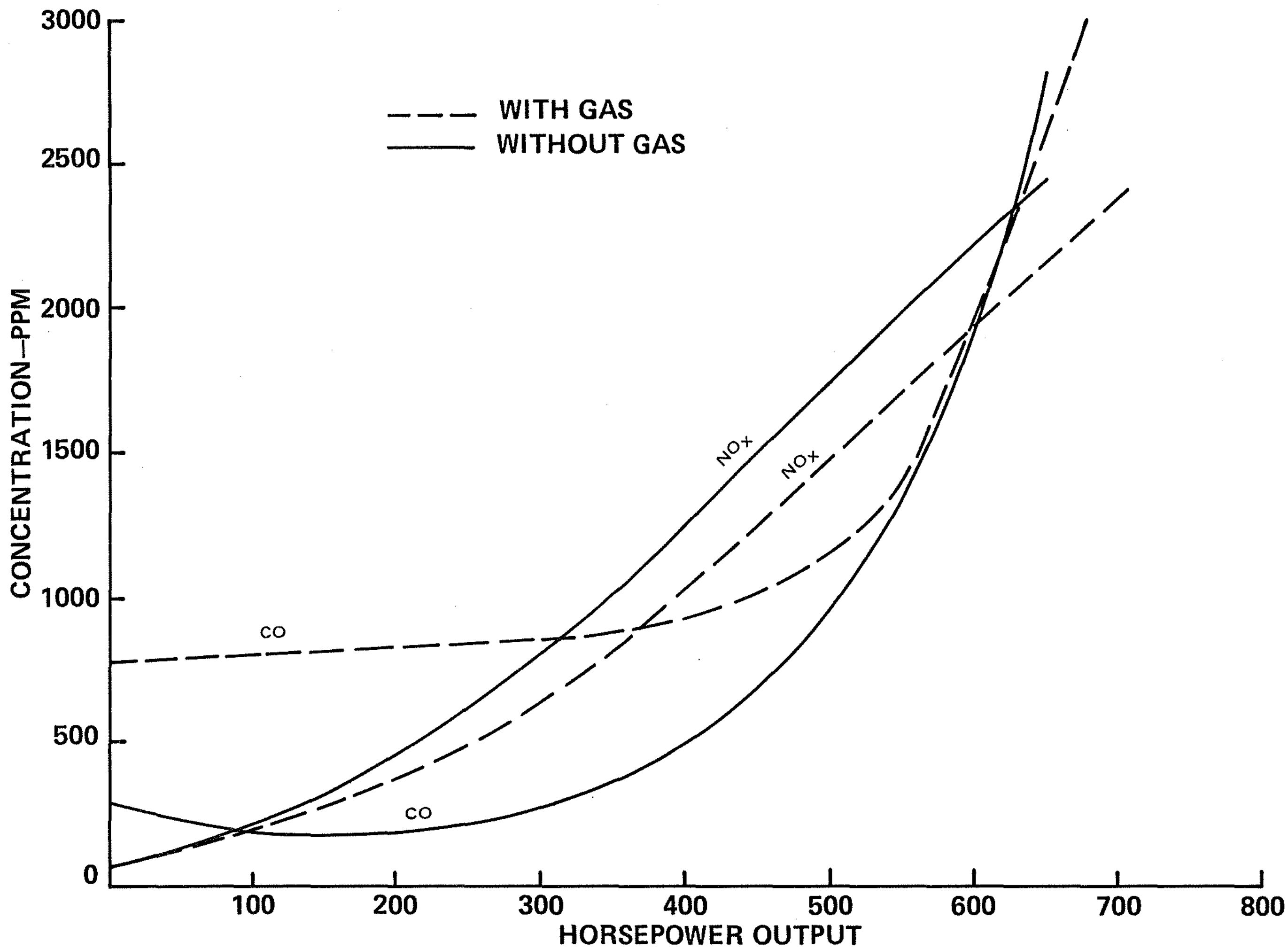

FIGURE 24 - EFFECT OF 1.5% NATURAL GAS ON CO AND NO_x EMISSIONS FOR CUMMINS VTA-1710 ENGINE AT 1500 RPM

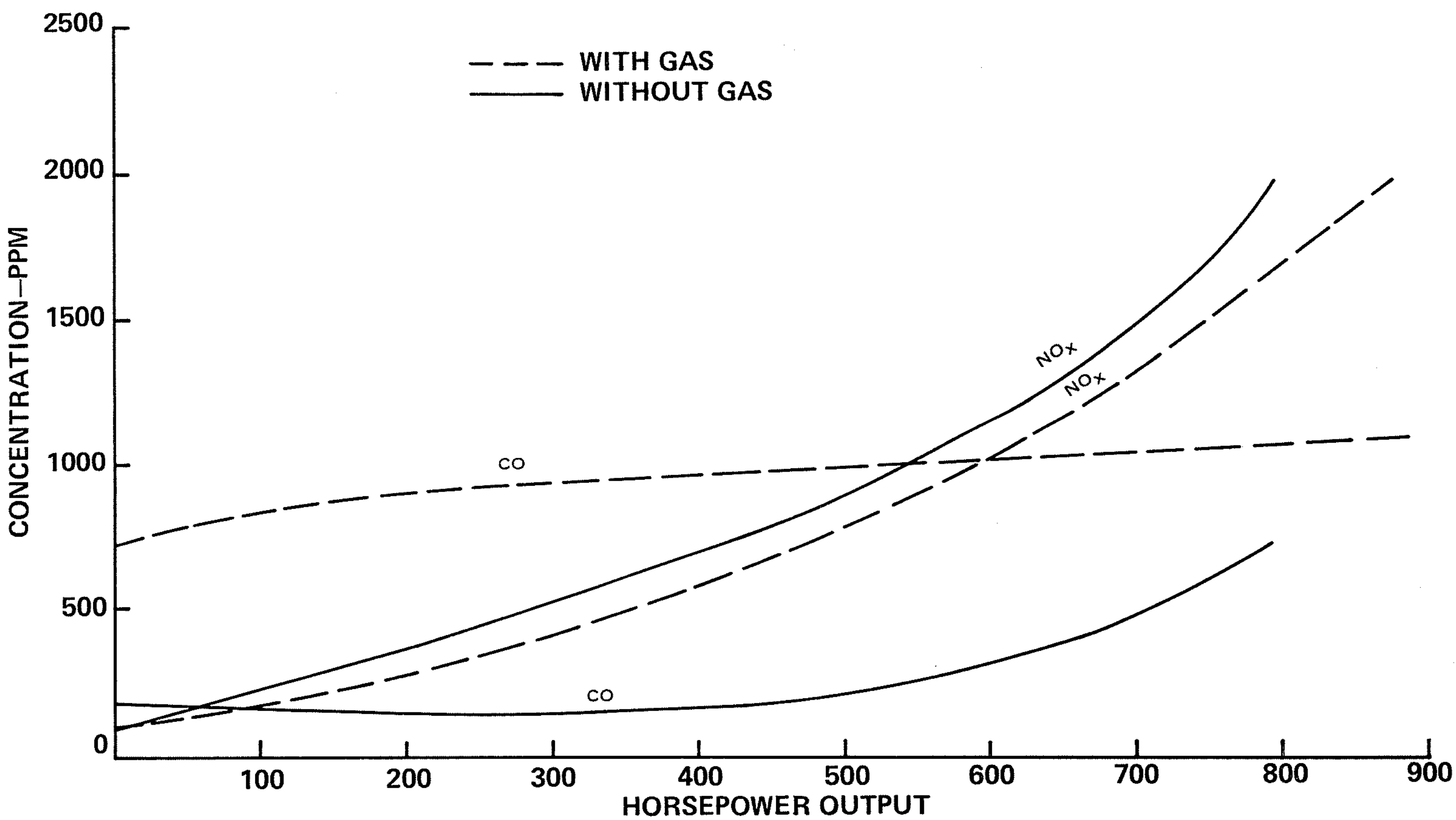

FIGURE 25 - EFFECT OF 1.5% NATURAL GAS ON CO AND NOx EMISSIONS FOR CUMMINS VTA-1710 ENGINE AT 2100 RPM

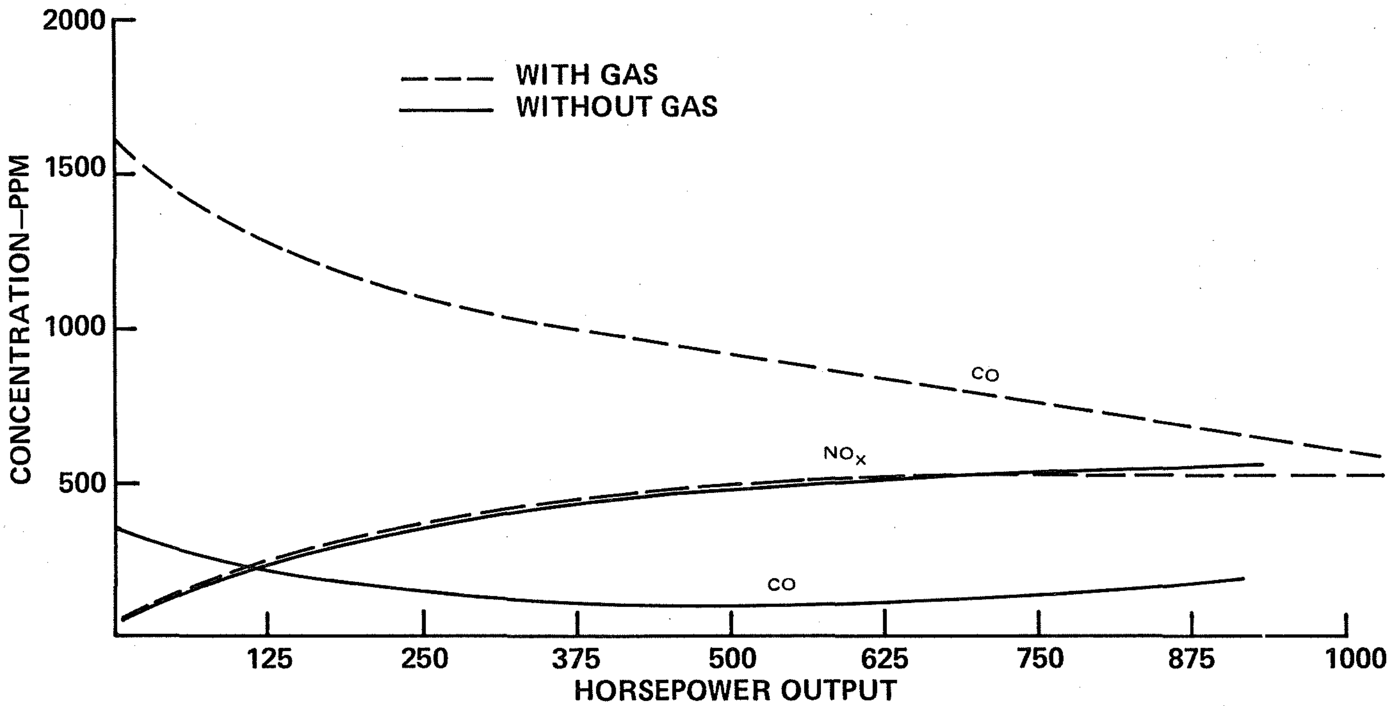

FIGURE 26 - EFFECT OF 1.5% NATURAL GAS ON CO AND NO_x EMISSIONS FOR CATERPILLAR D348 ENGINE AT 2000 RPM

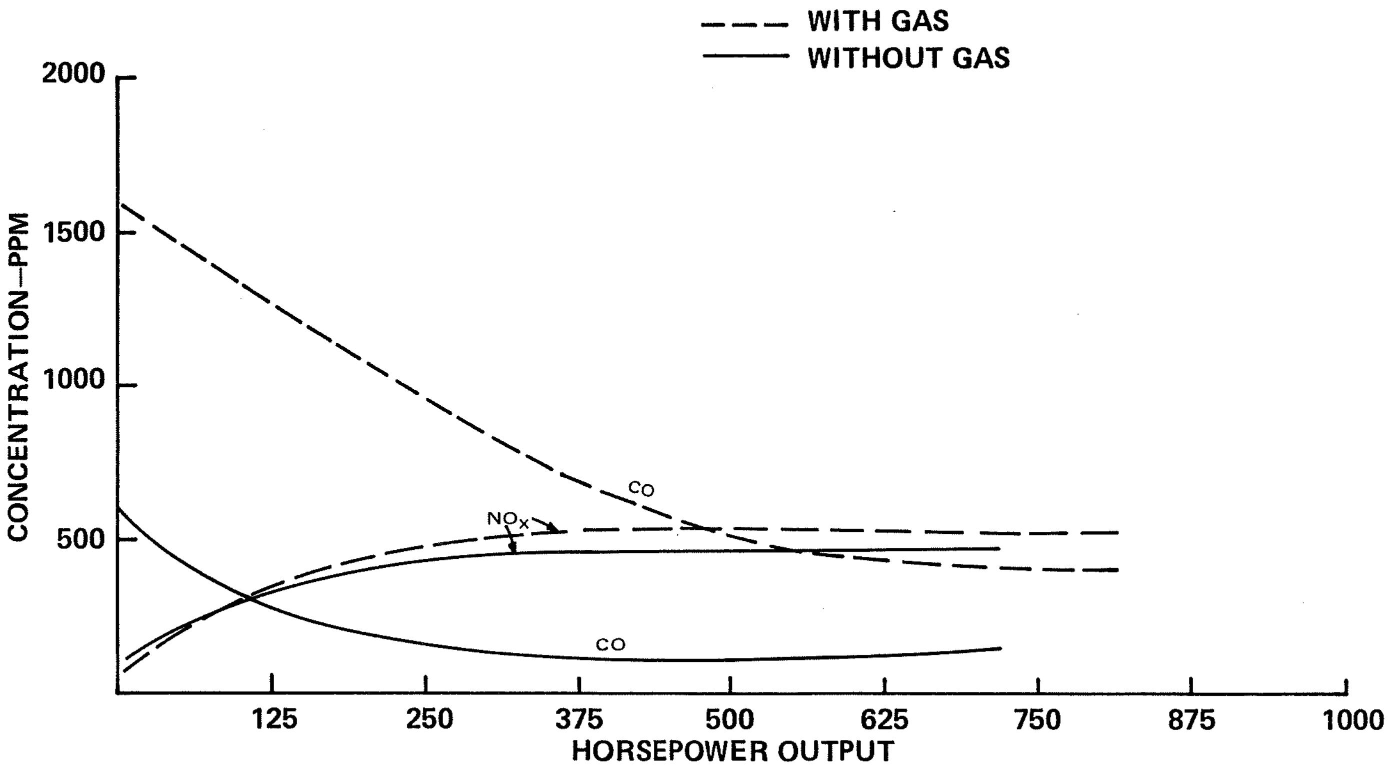

FIGURE 27 - EFFECT OF 1.5% NATURAL GAS ON CO AND NO_X EMISSIONS FOR CATERPILLAR D348 ENGINE AT 1400 RPM

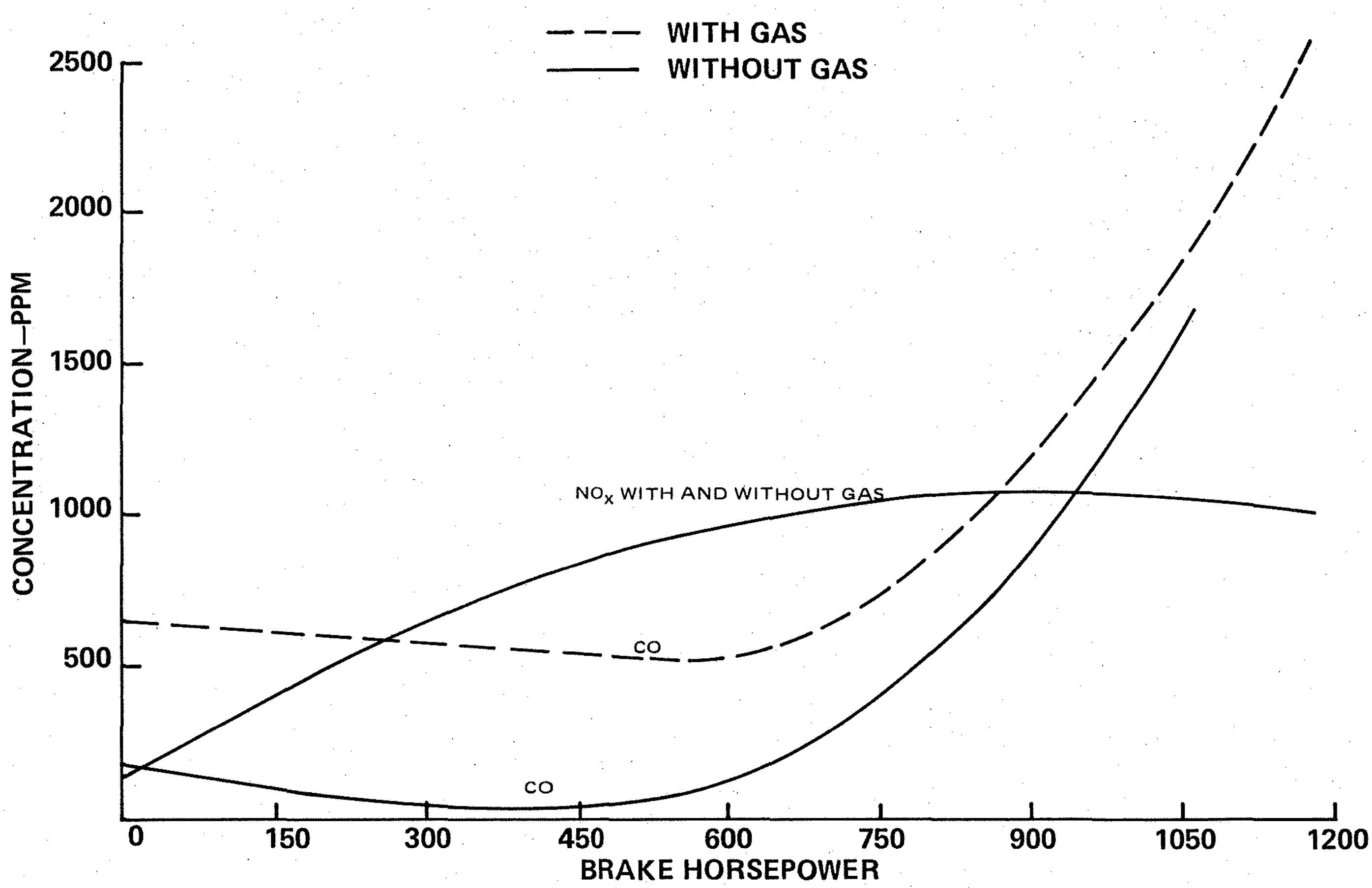

FIGURE 28 - EFFECT OF 1.5% NATURAL GAS ON CO AND NO_X EMISSIONS FOR DETROIT DIESEL 12V-149TI ENGINE AT 1600 RPM

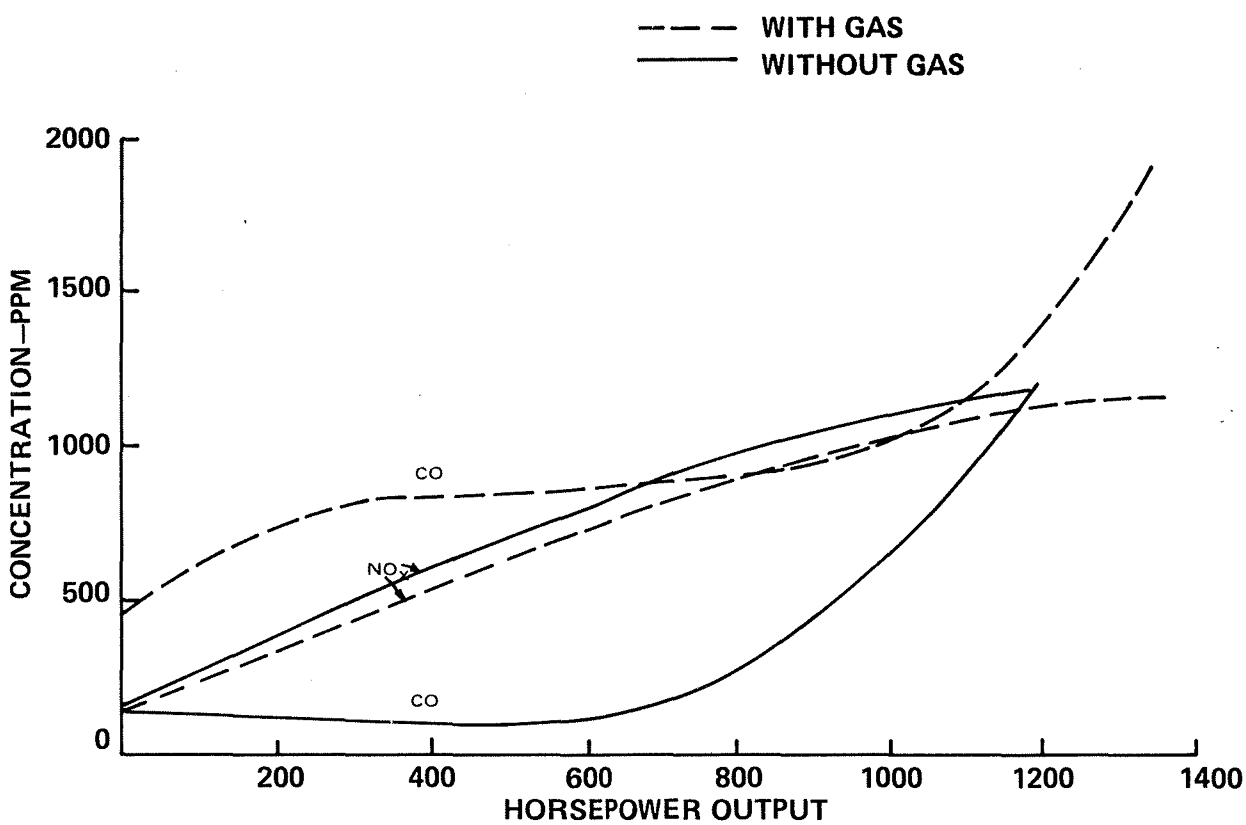

FIGURE 29 - EFFECT OF 1.5% NATURAL GAS ON CO AND NO_x EMISSIONS FOR DETROIT DIESEL 12V-149TI ENGINE AT 1900 RPM

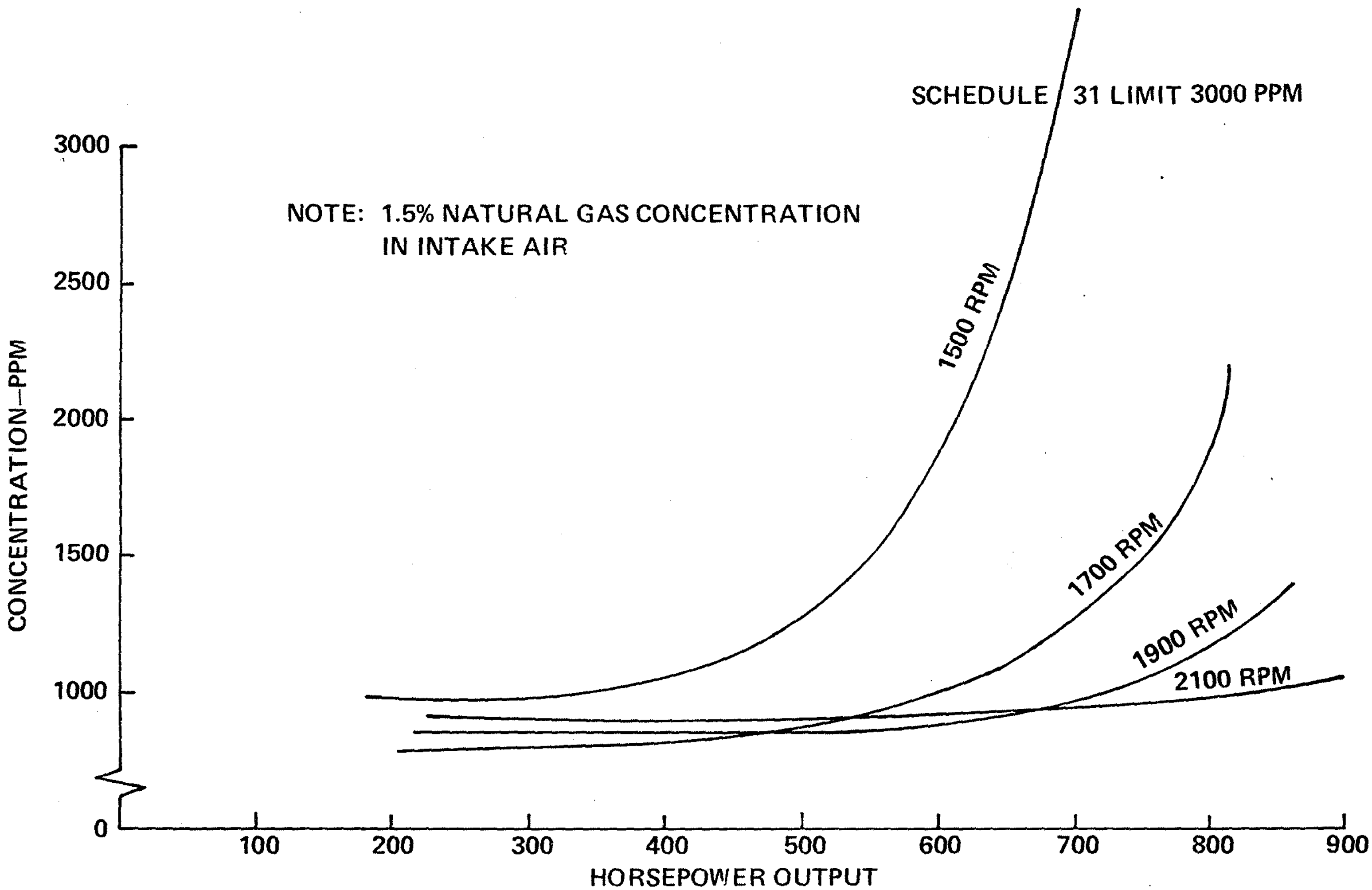

FIGURE 30 - EFFECT OF POWER OUTPUT ON CO EMISSIONS AT VARIOUS SPEEDS FOR CUMMINS VTA-1710 ENGINE

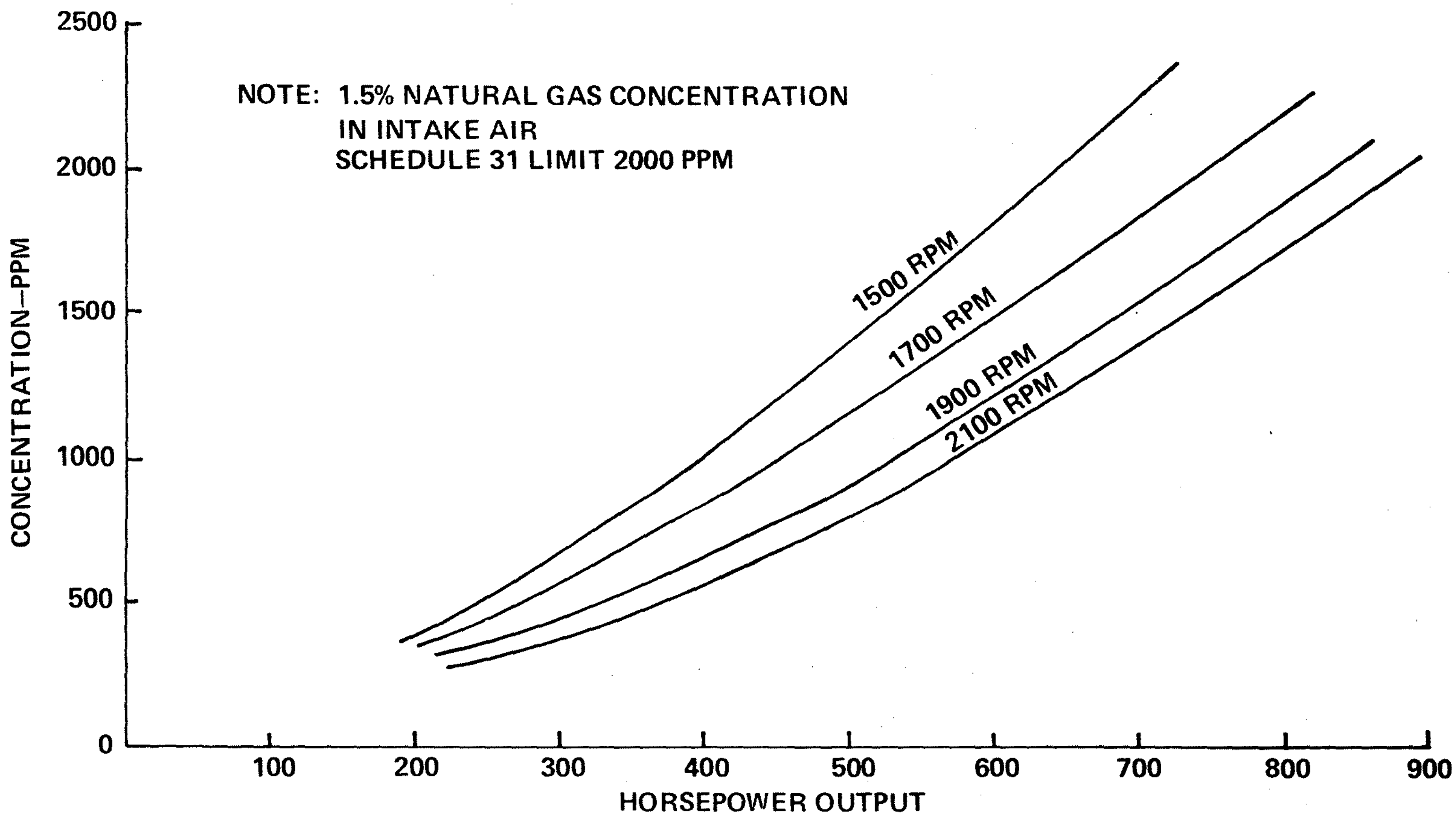

FIGURE 31. EFFECT OF POWER OUTPUT ON NO_x EMISSIONS AT VARIOUS SPEEDS FOR CUMMINS VTA-1710 ENGINE

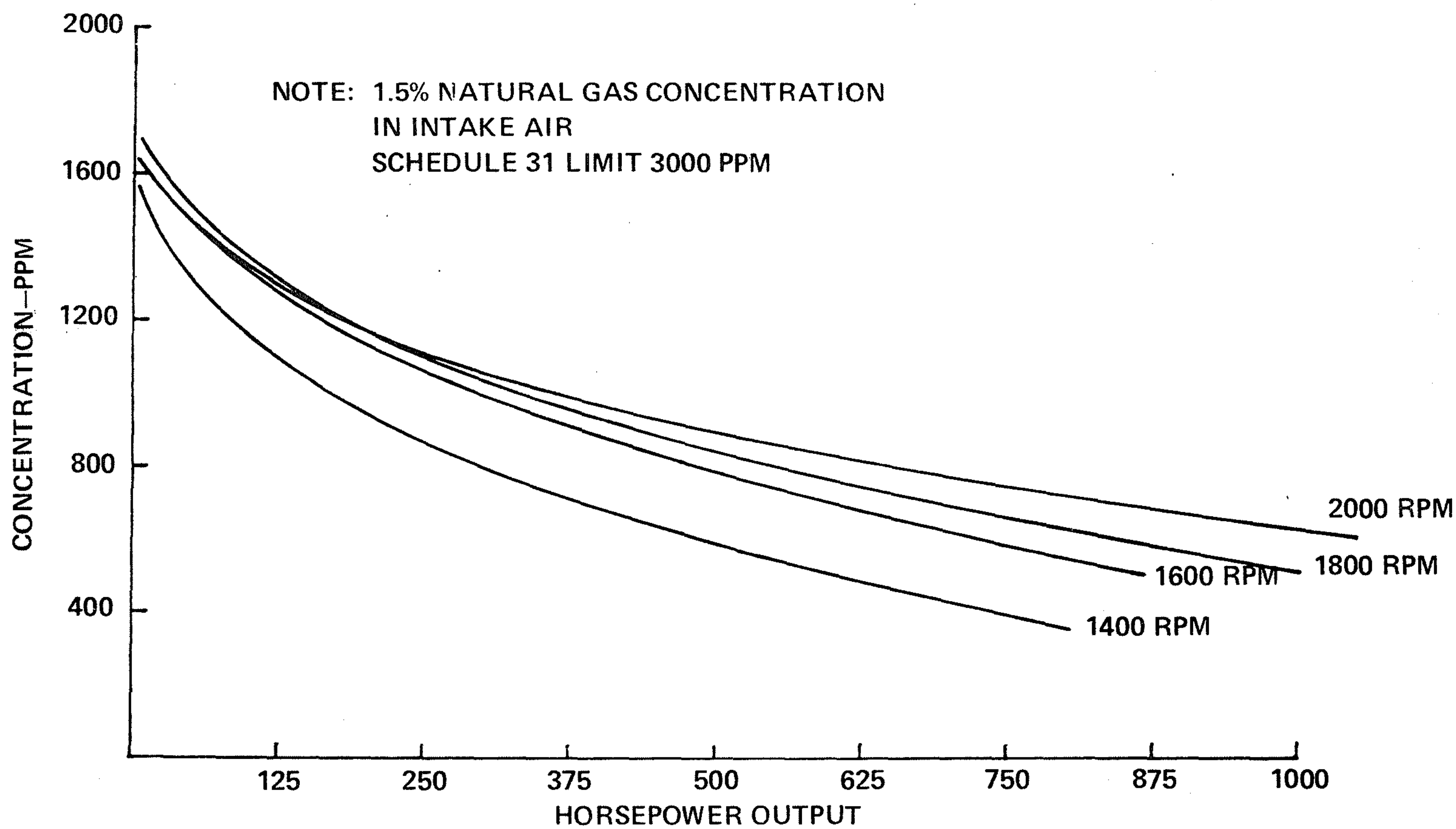

FIGURE 32. EFFECT OF POWER OUTPUT ON CO EMISSIONS AT VARIOUS SPEEDS FOR CATERPILLAR D348 ENGINE

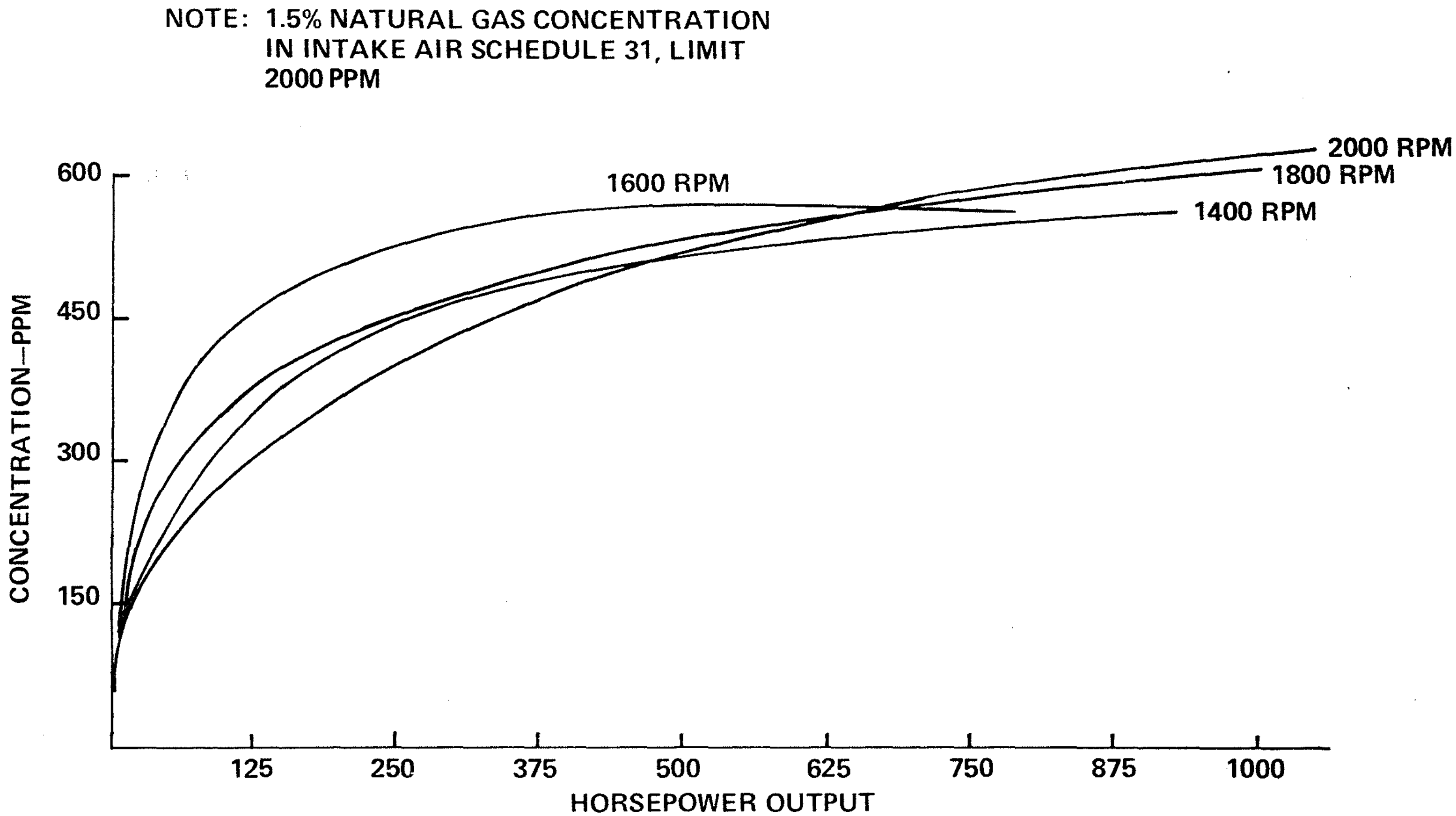

FIGURE 33 - EFFECT OF POWER OUTPUT ON NO_x EMISSIONS AT VARIOUS SPEEDS FOR CATERPILLAR D348 ENGINE

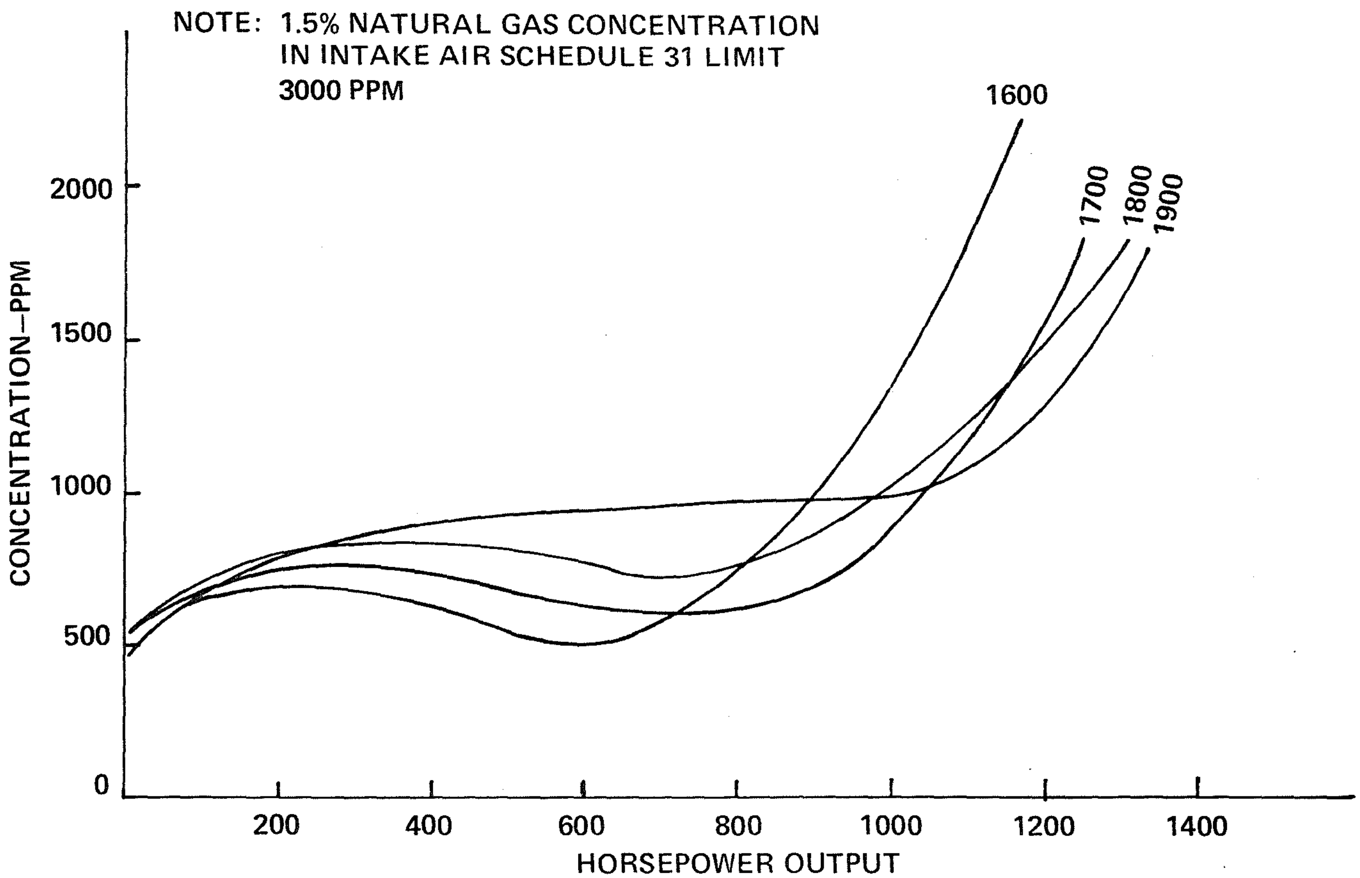

FIGURE 34. EFFECT OF POWER OUTPUT ON CO EMISSIONS AT VARIOUS SPEEDS FOR DETROIT DIESEL 12V-149TI ENGINE

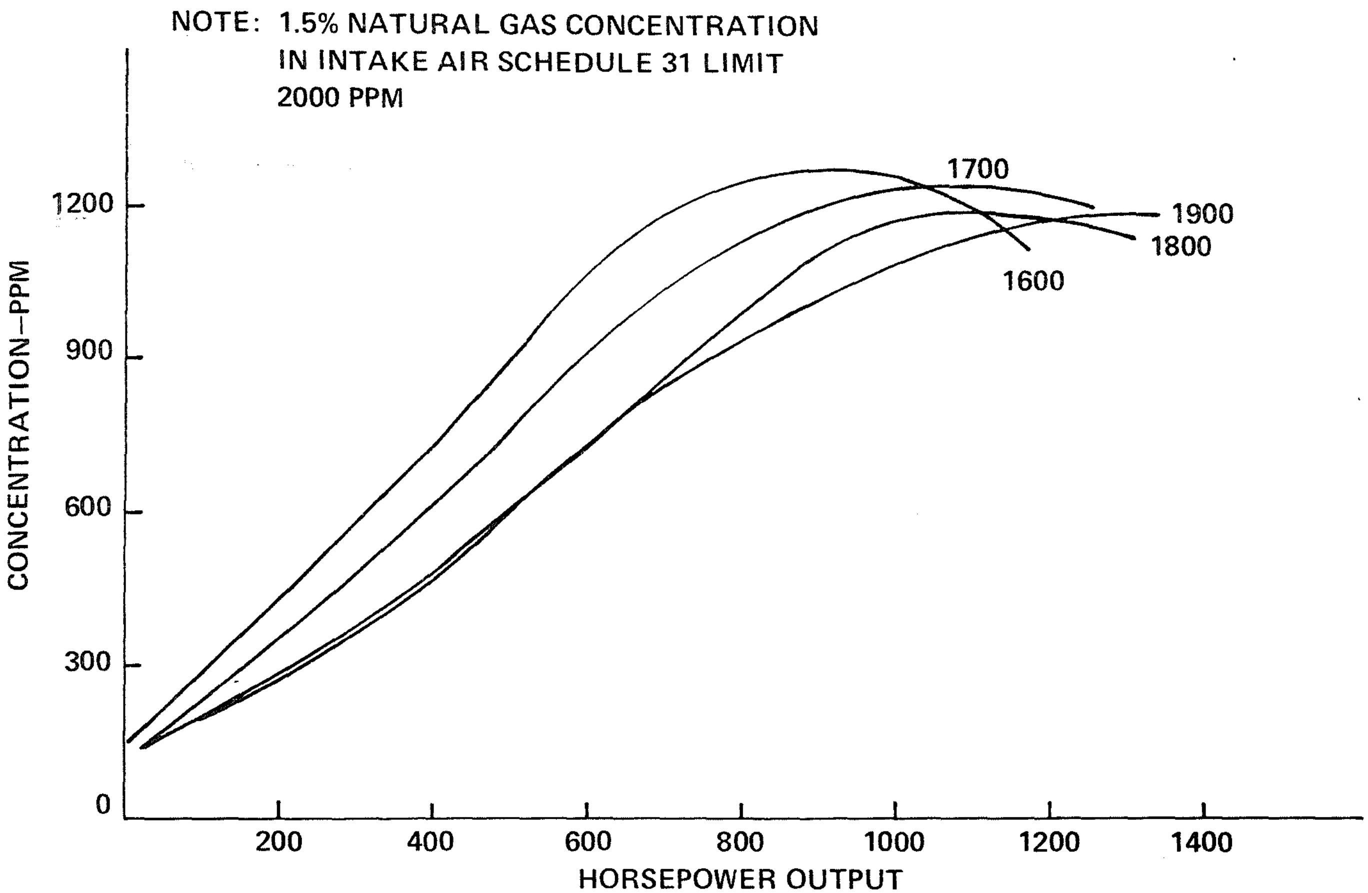

FIGURE 35 - EFFECT OF POWER OUTPUT ON NO_x EMISSIONS AT VARIOUS SPEEDS FOR DETROIT DIESEL 12V-149TI ENGINE

TABLE 2

Calculated Ventilation Rates

	Total cfm Required	cfm Required/Horsepower
Cummins VTA-1710	263,200	292
Caterpillar D348	95,100	91
Detroit Diesel 12V-149TI	394,800	262

The CO and NO_x emissions of the undiluted exhaust for the Cummins engine were above the allowable limit of 3000 ppm and 2000 ppm, respectively. The NO_x emission at 1500 RPM control the amount of power allowable to remain within the Schedule 31 limits. Referring to Figure 31, it can be seen that a maximum of 640 horsepower at 1500 RPM is possible in order to remain below 2000 ppm. This corresponds to 580 Hp @ 1500 ppm when the engine is operated without natural gas in the inlet air, and results in a 12% derate from the manufacturer's rating.

Both the CO and NO_x emissions of the Caterpillar D348 were well within the allowable Schedule 31 limits. NO_x concentrations were highest at full load, 2000 RPM but still did not exceed 600 ppm. CO concentrations were highest at idle but were only 1330 ppm. The undiluted CO and NO_x of the Detroit Diesel 12V-149TI were also within Schedule 31 requirements but the concentrations of both contaminants were significantly higher than the Caterpillar engine. Maximum NO_x concentrations always occurred when the engine was at full load but were virtually the same at all four engine speeds. Highest CO concentrations were at full load, 1600 RPM.

B. Exhaust Gas Cooling Tests

The exhaust scrubber was designed such that it could be used on all engines. The high water level at which large amounts of entrained water were carried out of the scrubber was visually determined to be 105 gallons, corresponding to a water depth in the scrubber of 15-3/4 inches. One inch of water is equivalent to 5.7 gallons. The low water level, at which the scrubber was unable to maintain exhaust temperatures below 170°F, was found to be 37 gallons (5.5 in. depth). Referring to Figures 36 through 38 and Table 3, it will be seen that the best operating range for minimum water consumption is between 7 and 13 in. Above the 13-in water level the water usage begins to increase significantly due to water entrainment. The ratio of scrubber outlet temperature, T_o to the scrubber inlet temperature, T_i changes less than two percent regardless of the water level. Although a maximum scrubber exhaust temperature of 170°F is allowed by Schedule 31, the Wagner system kept the outlet temperature below 155°F on all engines. The slight increase in scrubber inlet temperature with increasing water levels is to be expected, since the higher water levels increase the exhaust back pressure which in turn causes the exhaust temperature to rise.

TABLE 3

COOLING WATER CONSUMPTION AND FINAL EXHAUST GAS TEMPERATURE

	Water Level (gals)	Water Consumption (gal/minute)	Scrubber Inlet Temperature °F	Scrubber Outlet Temperature °F
Cummins VTA-1710	40	2.43	828	154
	50	2.59	823	153
	60	2.58	830	154
	70	2.78	834	153
	80	2.94	838	153
	90	2.92	842	153
Caterpillar D348	40	2.34	751	149
	50	2.34	751	149
	60	2.35	753	149
	70	2.34	756	150
	80	2.35	754	150
	90	2.33	755	149
	100	2.87	756	150
Detroit Diesel 12V-149TI (Half of Total Exhaust Flow)	40	2.27	774	147
	50	2.40	784	148
	60	2.40	784	149
	70	2.27	788	149
	80	2.40	792	149
	90	2.52	795	149

C. Intake Flame Arresters

Pressure drop due to the flame arresters was measured at the inlet to the turbochargers. The flame arresters were attached to the engine by means of a short converging duct and a short straight section of pipe the same diameter as the turbocharger. Measured pressure drop ranged between 12.7 and 16.7 in-H_2O for all three engines. It was later discovered that the converging duct and the straight pipe caused the majority of this intake restriction and the flame arrester units actually had a flow resistance of less than 6 in-H_2O.

D. Surface Temperature Tests

It was expected that all engines would exceed the allowable 302°F surface temperature since there were no provisions made on these engines to cool the turbocharger.

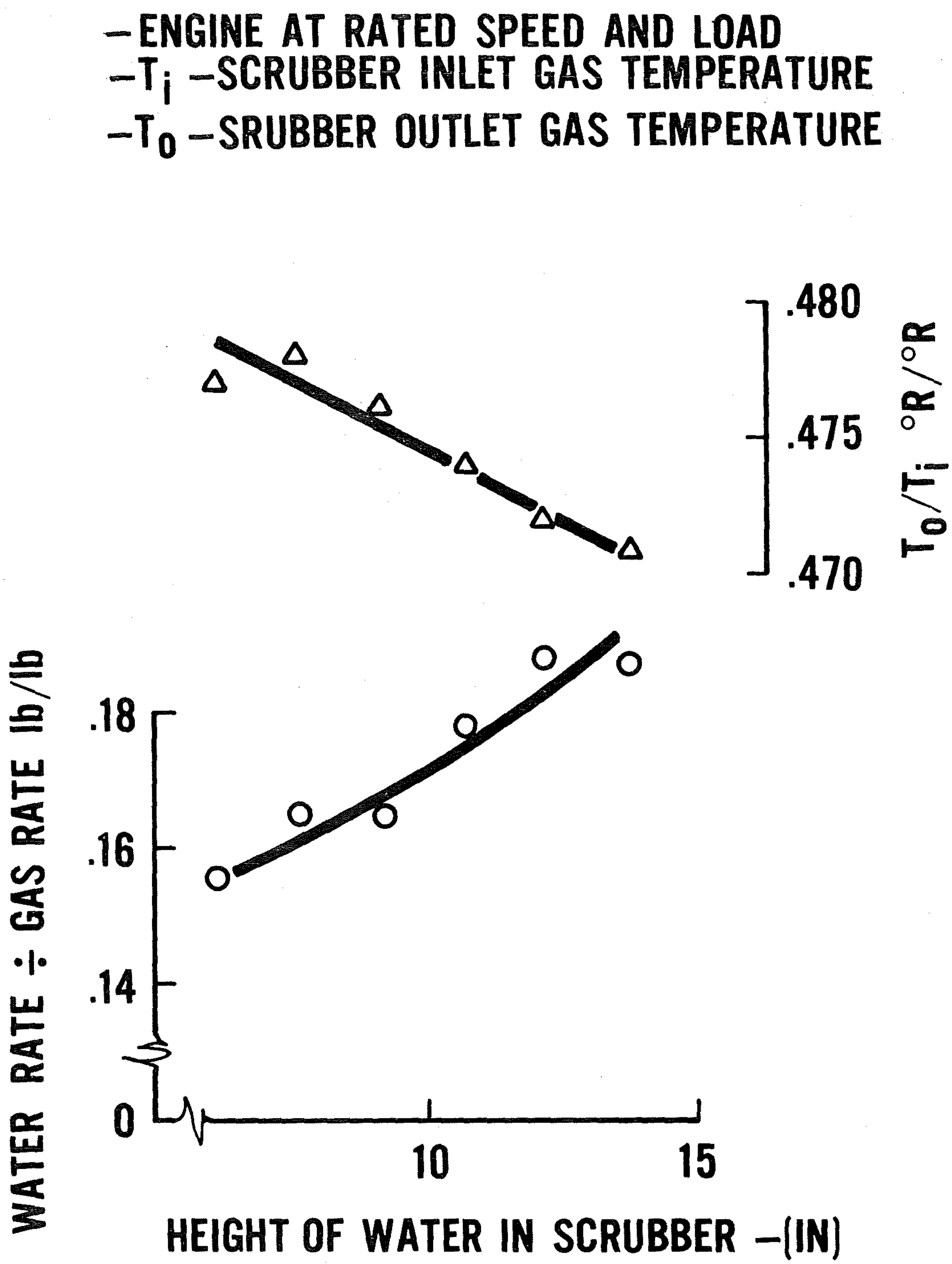

FIGURE 36
PERFORMANCE OF EXHAUST SCRUBBER ON CUMMINS VTA-1710 ENGINE

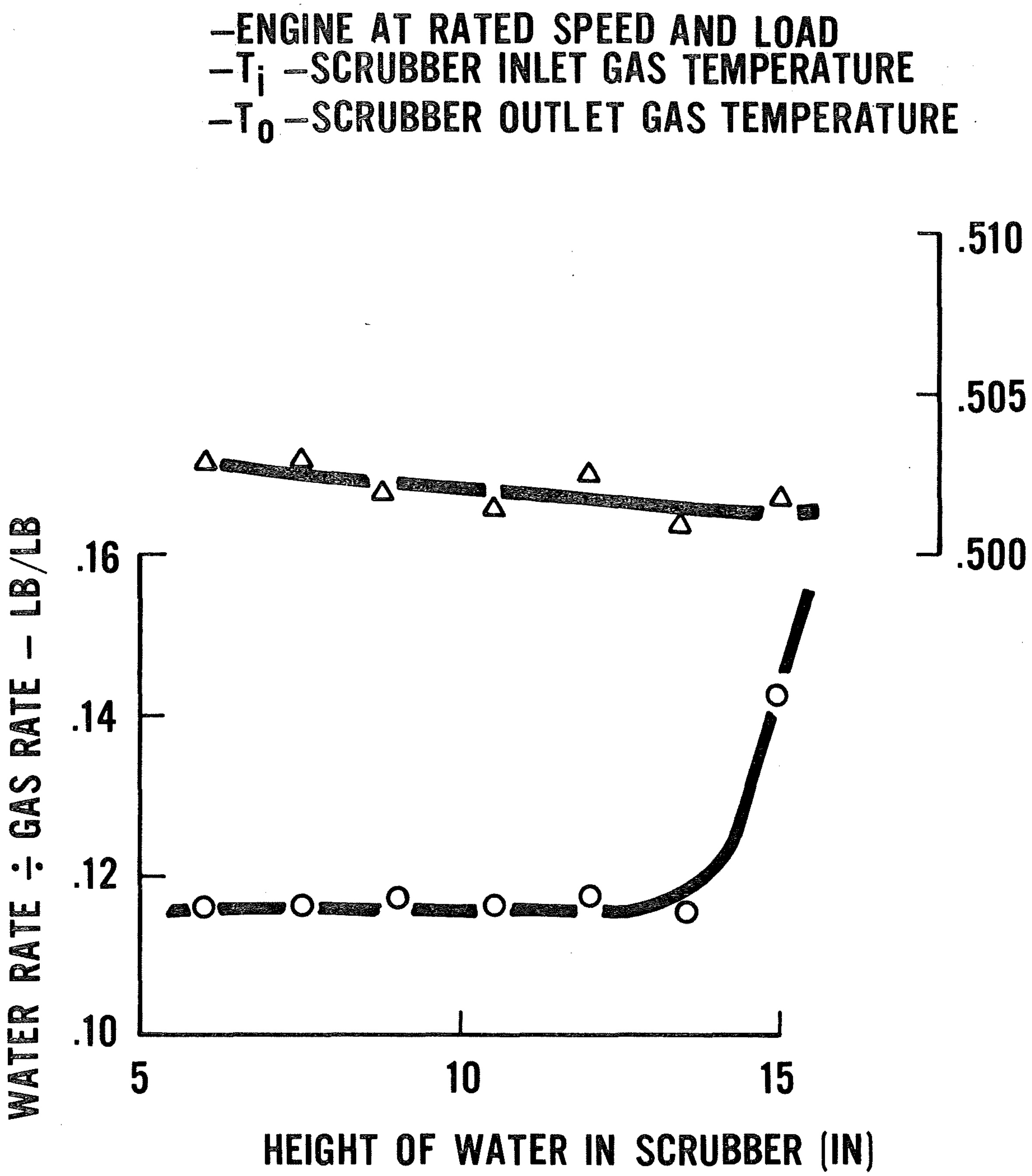

FIGURE 37

PERFORMANCE OF EXHAUST SCRUBBER ON CATERPILLAR D348 ENGINE

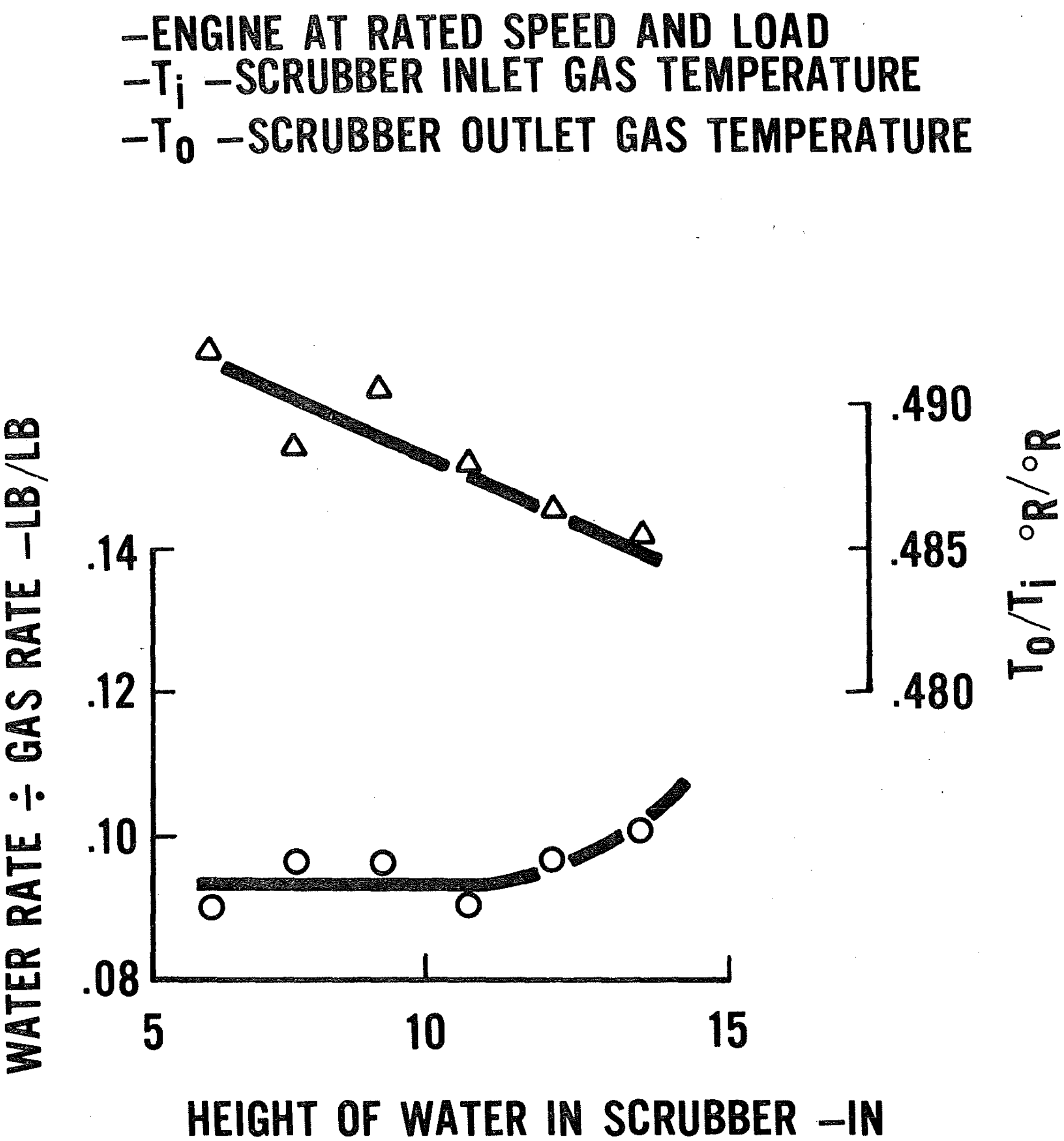

FIGURE 38
PERFORMANCE OF EXHAUST SCRUBBER ON DETROIT DIESEL 12V–149TI

1. Cummins Engine

This engine had a water cooled exhaust manifold but an uncooled turbocharger. As expected, the exhaust side of the turbocharger was the hottest point on the engine. Its surface temperature was about 900°F. The compressor side of the turbocharger was about 320°F from midway around the scroll to the intercooler manifold. The surface of the intercooler was about 310°F from air inlet to its midpoint. The surface temperature of the water-jacketed exhaust system was slightly below 300°F except at the mating flange between it and the turbocharger. The production exhaust manifolds between the turbocharger and the engine cylinder head were also under 300°F except at their attachment to the turbocharger and about one inch around it. Surface temperature of the water-jacketed exhaust pipes between the turbochargers and the scrubber were 250°F.

2. Caterpillar Engine

This engine employed a water-cooled exhaust manifold used in marine applications, but this water cooling did not extend to the flanges of the exhaust ports or for a distance of ten inches before the exhaust entered the turbochargers. The surface temperature of the turbine side of the turbocharger exceeded 800°F. The compressor side of the turbocharger was about 310°F over the last 1/3 of the scroll and on the connecting manifold to the intercooler. After the intercooler the surface temperature of the housing was 250°F. The flanges of the exhaust manifold at the exhaust ports and the exhaust manifold itself were more than 700°F. The partially water-jacketed exhaust manifold temperatures were under 300°F. All temperatures of the exhaust scrubber were below 200°F except that the flanges where the exhaust enters the scrubber were 225°F. The water-jacketed exhaust system maintained a surface temperature of 250°F.

3. Detroit Diesel Engine

Surface temperature tests of the Detroit Diesel revealed problems similar to those discovered in the previous engine tests. The turbocharger surface temperature on the exhaust side, as well as all exhaust manifolding between it and the engine block, were above 800°F. At the exhaust ports, the temperature decreased to under 300°F about 1/4 inch from the exhaust manifold mounting flange. The air box covers on the side of the engine were 235°F. On the compressor side of the turbocharger and on the associated ducting to the intercooler, the surface temperature was 305°F.

E. Explosion Tests

The intake flame arresters prevented propagation of internal explosions under all test conditions for all three engines. One of the flame arresters was subjected to a total of about 150 explosions with no evidence of deformation. Maximum explosion pressures in the intake system were quite low with only six psi observed during explosions with the engine stopped and five psi during explosions with the engine being motored.

1. Cummins Engine

A minimum water volume of 135 gallons (20 inches) was required to prevent ignition of the surrounding natural gas mixture when an explosion was initiated in the exhaust system. This is 30 gallons more than the acceptable high water volume determined in the cooling tests and would result in excessive water consumption without a redesign. Maximum explosion pressures observed in the exhaust system were 40 psi for the dynamic tests, and 58 psi for the static tests.

2. Caterpillar Engine

This engine was unable to pass the explosion tests on the exhaust system because it contained a slip joint between the exhaust manifold and the turbochargers which consistently allowed an internal explosion to propagate to the surrounding natural gas mixture. Consequently, it was not possible to determine the minimum scrubber water volume to prevent external explosions, although it is expected that it would be the same as the Cummins engine.

3. Detroit Diesel Engine

This engine passed the explosion tests at a scrubber water volume of 135 gallons. Maximum observed explosion pressures were 47 psi for the static tests and 40 psi for the dynamic tests. The water volume was reduced to 125 gallons to confirm that an external explosion could occur at this level. After nine dynamic shots, the drive shaft which turns the engine blowers broke and testing was discontinued. Even though the part was subsequently replaced, it was felt that further testing would have broken the new shaft. It is suspected that explosions were occurring within the air box.

V. EXPERIMENTAL PROCEDURE - PHASE II

Results from Phase I indicated that there were two major problem areas associated with all three engines; all engines had high surface temperatures on the turbine side of the turbocharger, and the optimum water level in the exhaust scrubber for adequate cooling was 30 to 50 gallons below the minimum level required to pass the explosion tests. The Cummins engine would require a derate to lower its emissions and the Detroit Diesel engine would require too large a volume of ventilation air. For this reason, the Caterpillar engine was chosen for retesting to demonstrate the feasibility of the modifications.

A. Surface Temperature Reduction

Water cooled enclosures around the turbocharger were investigated as a possible solution. Such enclosures in contact with the turbocharger surface only at the mounting flanges, exhaust system connection, and along the circumference of the compressor housing would not affect engine performance or emission characteristics, since only radiated heat would be transferred from the turbocharger. Measurements were made of all critical clearances around the two turbochargers of the Caterpillar engine. The left turbocharger had adequate clearances and did not present any unexpected problems. The right turbocharger, however, is adjacent to several water and air lines which did not allow adequate clearances for an enclosure. It was necessary to relocate the turbocharger four inches back from its original position. The final design was a double wall, cylindrical housing which attached to the turbocharger compressor along its inner circumference. The exhaust pipe was welded to a double walled plate which attached to the other end of the enclosure. Figures 39 and 40 show the enclosures mounted on the engine while Figure 41 shows sectional views of the installation. The surface temperature of the enclosures and adjacent areas were next determined with the engine operating at full load, rated speed. Test procedure was identical to that previously used in Phase I; however, two different cooling water conditions were investigated. For the first test, 180°F engine cooling water was circulated through the water cooled enclosures. A second evaluation was made using 75°F laboratory water as the cooling medium in the enclosures.

B. Exhaust Scrubber Investigations

Modifications were made to the exhaust scrubber in an attempt to reduce its water consumption and improve explosion protection. These modifications required virtually a complete rebuild of the scrubber (as shown in Figures 42 and 43) since the initial design had made no provisions for disassembly. The two horizontal swirl tubes at the bottom of the scrubber were rotated 45° such that the exhaust exited downward into the water rather than at an angle. It was expected that this change would allow a lowering of the required water depth for explosion integrity with some penalty due to increased exhaust back pressure. Figure 44 shows the location of the swirl tubes and the exhaust flow in the scrubber. A plexiglass window was added to one of the outer walls of the scrubber to allow an internal examination during engine operation. Exhaust gas cooling tests were next performed using the same procedures described on page 19.

A second modification was the addition of a water trap along the scrubber outlet. Two 24-inch long channels were fabricated with 2-inch flanges and a 1-inch web. The channels were attached just inside the scrubber outlet and sloped 2-inches from the center of the scrubber to its outer edge. It was expected that this water trap would catch any water droplets in the exhaust which impinge on the scrubber outlet, direct the water to the side of the scrubber, then back to the reservoir. A second set of exhaust gas cooling tests was then performed for comparison with previous results. The engine and scrubber were then installed in the explosion test facility to determine the effect of the scrubber modifications on explosion protection.

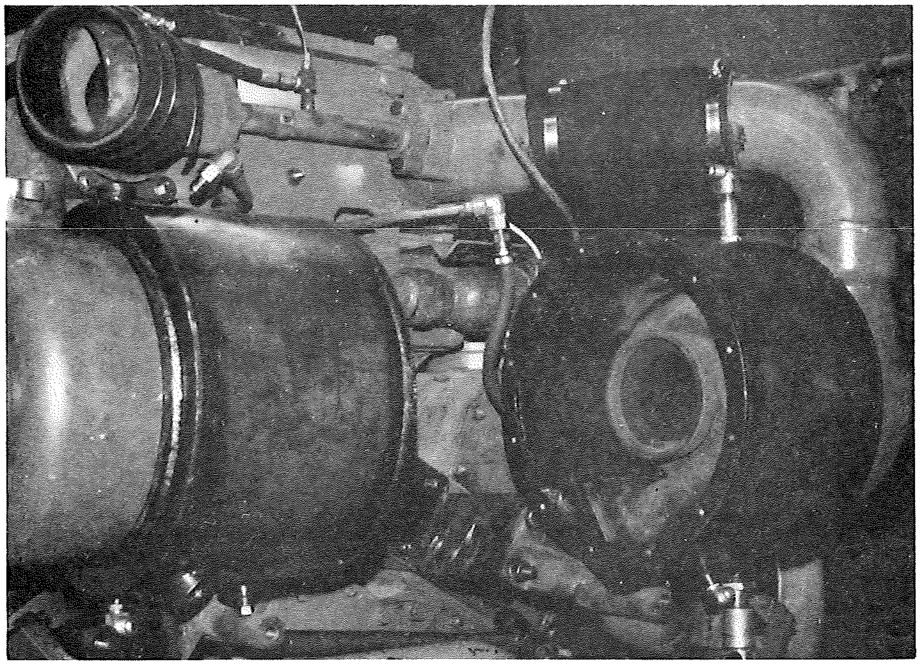

FIGURE 39
VIEW OF TURBOCHARGER ENCLOSURE ON
CATERPILLAR D348 ENGINE WITH EXHAUST OUTLETS REMOVED

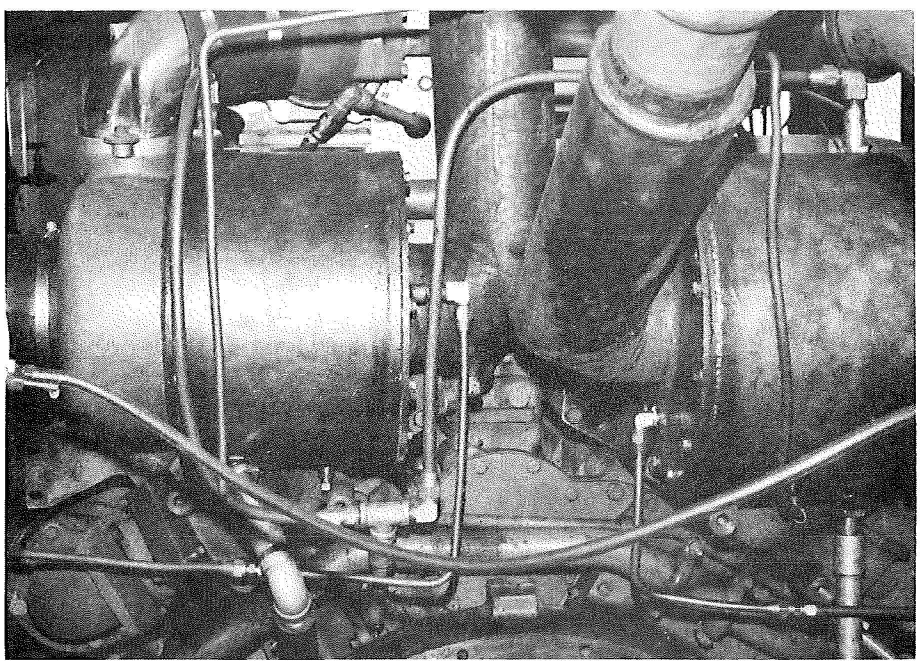

FIGURE 40
REAR VIEW OF TURBOCHARGER ENCLOSURES
ON CATERPILLAR D348 ENGINE

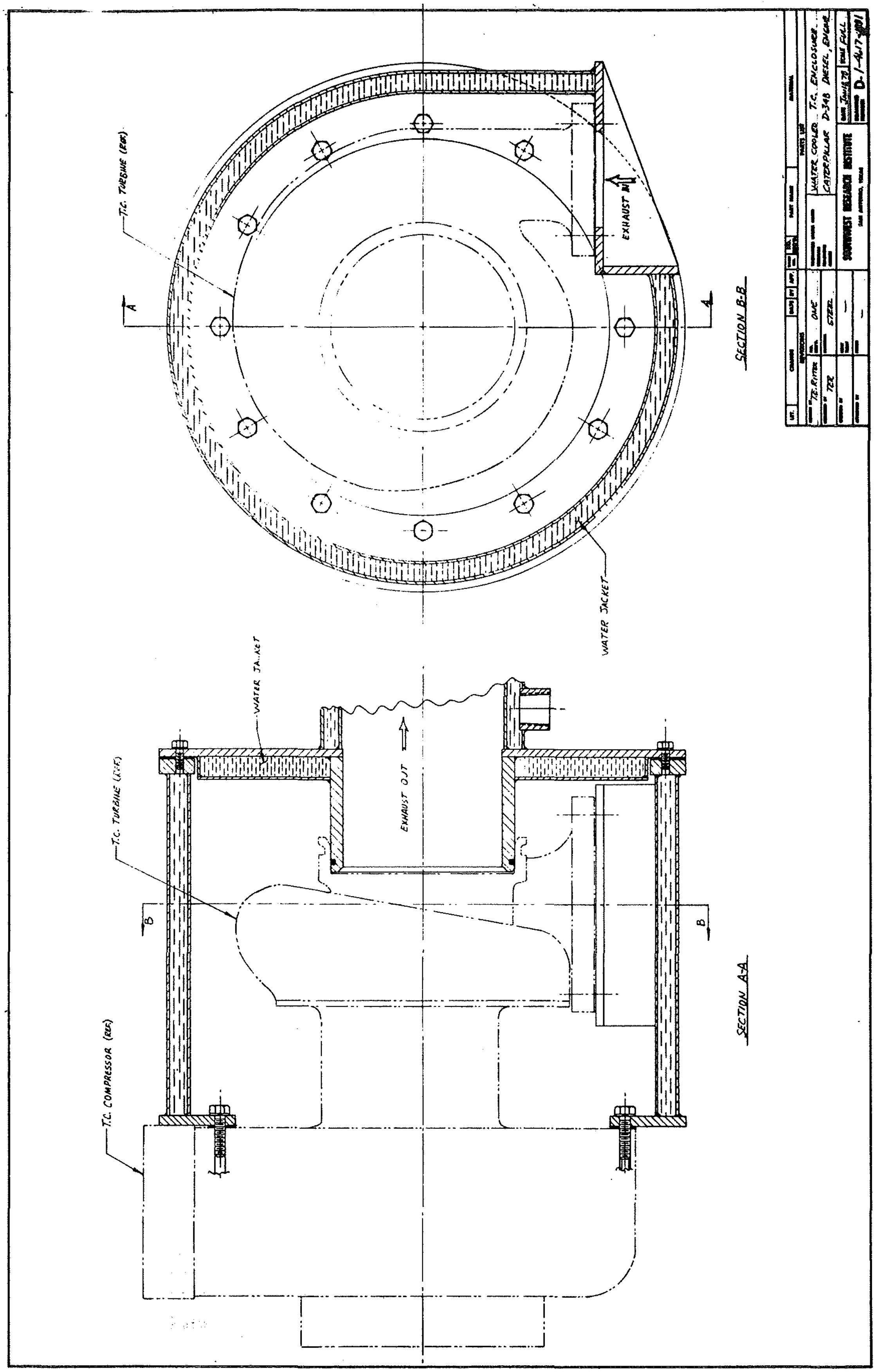

FIGURE 41--SECTIONAL VIEWS OF WATER JACKETED ENCLOSURES

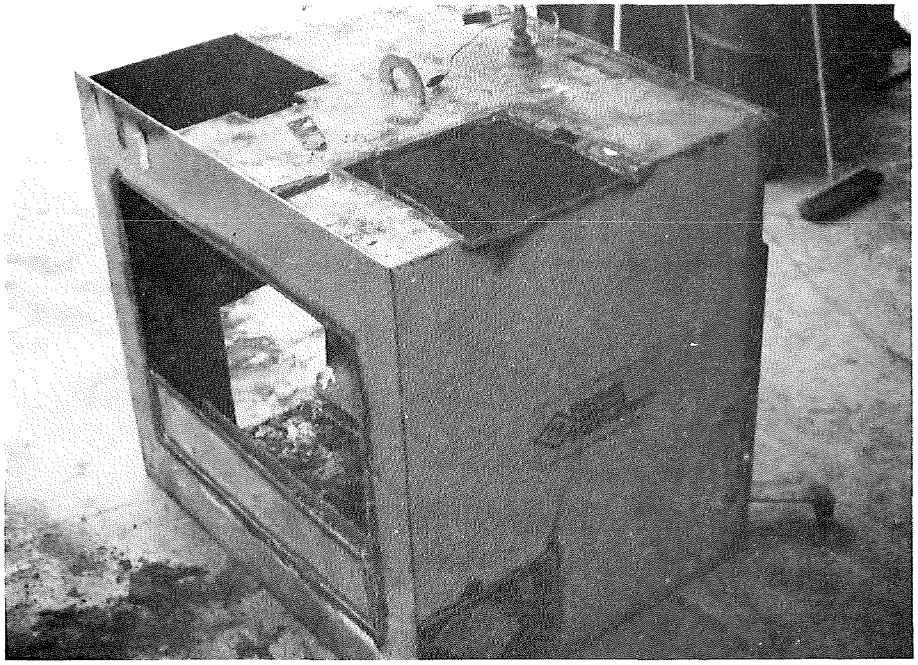

FIGURE 42
DISASSEMBLY OF SCRUBBER
FOR MODIFICATIONS

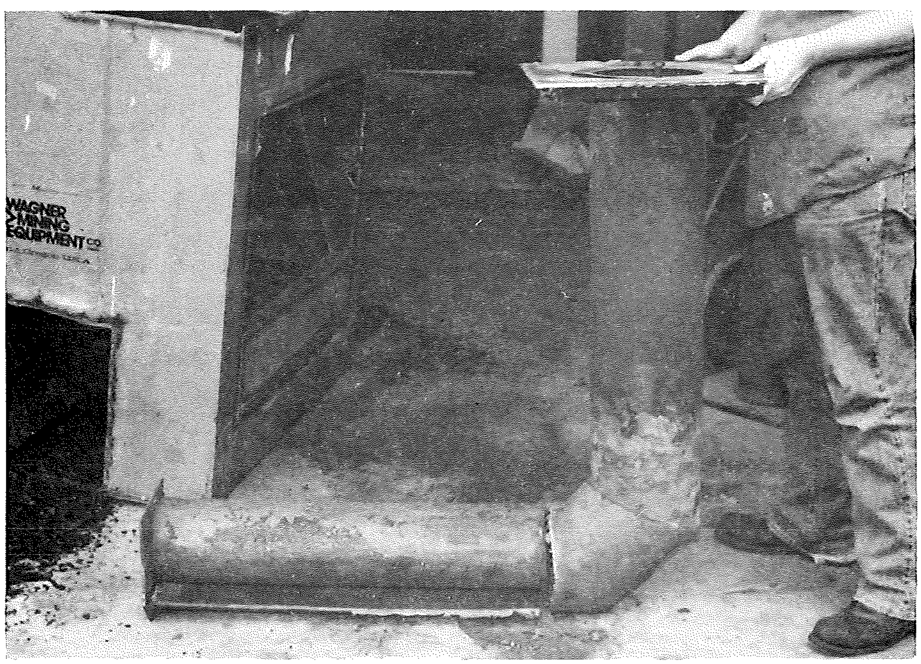

FIGURE 43
RELATIVE POSITION OF
EXHAUST PLUMBING INSIDE SCRUBBER

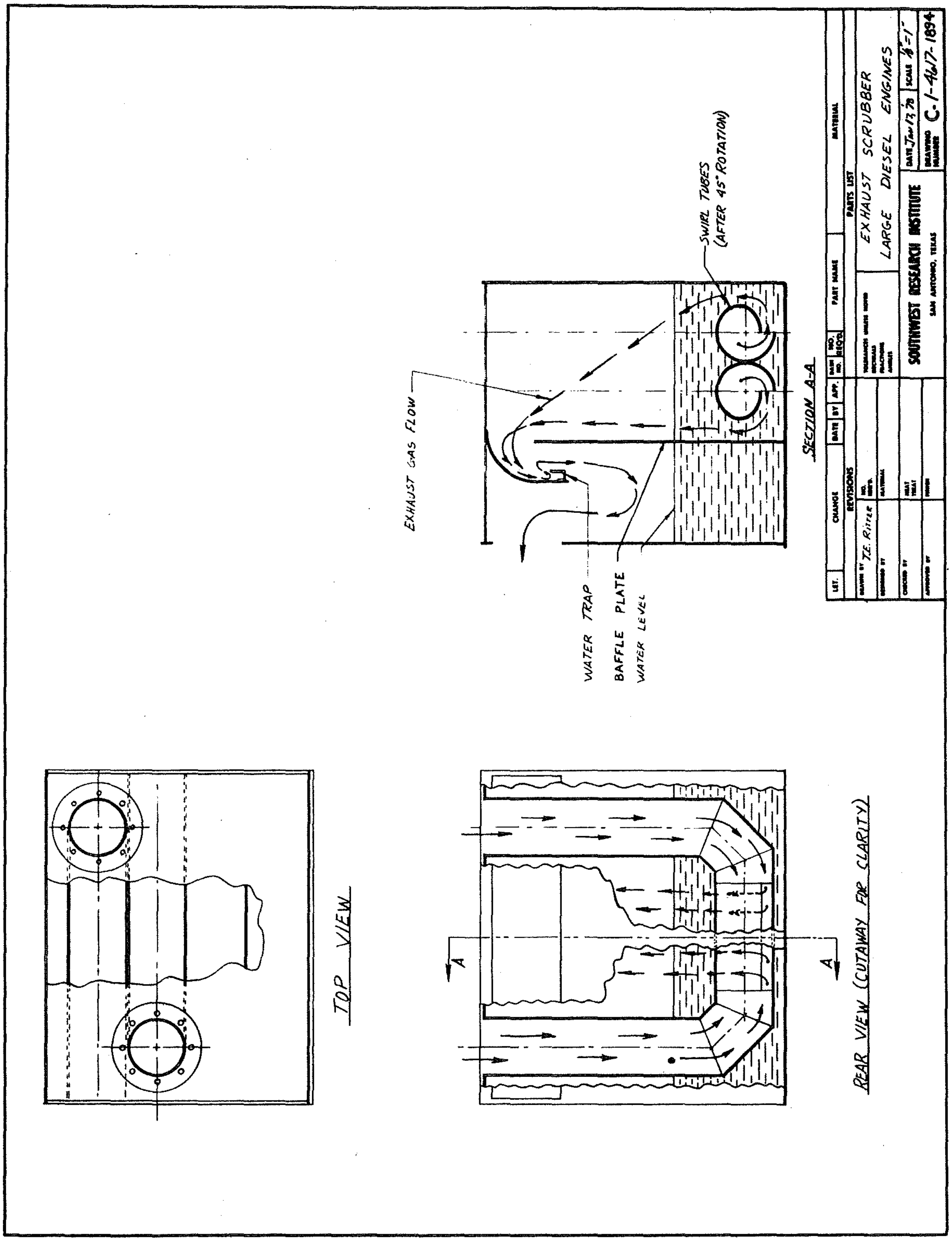

FIGURE 44--SCHEMATIC OF EXHAUST SCRUBBER

VI. RESULTS AND DISCUSSION - PHASE II

A. Surface Temperature Test

During the initial checkout it was determined that both turbochargers had significant exhaust leaks at the turbine mounting flanges and, in addition, the left turbocharger had a severe oil leak in the supply line that made it impossible to perform the surface temperature test. Subsequent inspection revealed that new gaskets were the only corrective action required. The oil leak in the supply line to the left turbocharger was caused by an improperly fabricated flange at the attachment to the turbocharger housing. A new flange was fabricated and both assemblies were carefully reassembled. Results of the initial tests with engine cooling water circulating through the water jacketed enclosures is shown in Figure 45. Surface temperatures were 300°F or less except at the flange where the enclosure attaches to the compressor of the turbocharger. This area could not be water jacketed due to insufficient clearances. A second run was made to determine the maximum power output which could be obtained with no surface temperature in excess of 300°F on the enclosures. At 775 horsepower, surface temperatures were below 300°F with the majority below 275°F. (See Figure 46)

A final evaluation was made using laboratory water as the cooling medium. Temperature of the water to the enclosures was 75°F and water out of the enclosures was 160°F. Under these conditions surface temperatures were below 175°F. During the subsequent exhaust gas cooling tests two separate turbocharger failures occurred on the right turbocharger. The first failure occurred after approximately ten hours of operation. The second failure occurred after the turbocharger had been rebuilt and operated for one hour. Both inspections found a severely worn turbocharger bearing that allowed the compressor blades to hit the case. Further investigation revealed the failure was caused by an external bending moment exerted on the turbocharger housing by the water jacketed enclosure. The problem stems from the necessity to bolt the enclosure to the vertical surface of the compressor housing and the horizontal base of the turbine flange. If either piece is not precisely fabricated, the turbocharger will be bent as the enclosure is bolted.

B. Exhaust Gas Cooling Tests

A comparison of old and modified exhaust scrubber performance on the Caterpillar engine is shown in Figure 47 and Table 4. The modifications to the scrubber resulted in an increase in water consumption of 7% over the baseline tests conducted in Phase I. This was probably due to increased contact between the exhaust gas and cooling water since the rotated swirl tubes allowed the exhaust gas to enter the water 1.35 inches lower. The low water level for the modified scrubber was found to be 25 gallons (3.75 inches depth). This was 1.75 inches less than the baseline scrubber. The high water level for the scrubber with rotated swirl tubes was 120 gallons, corresponding to a water depth in the scrubber of 18 inches. With the rotated swirl tubes and water trap the high water level was increased to 19-1/2 inches (130 gallon).

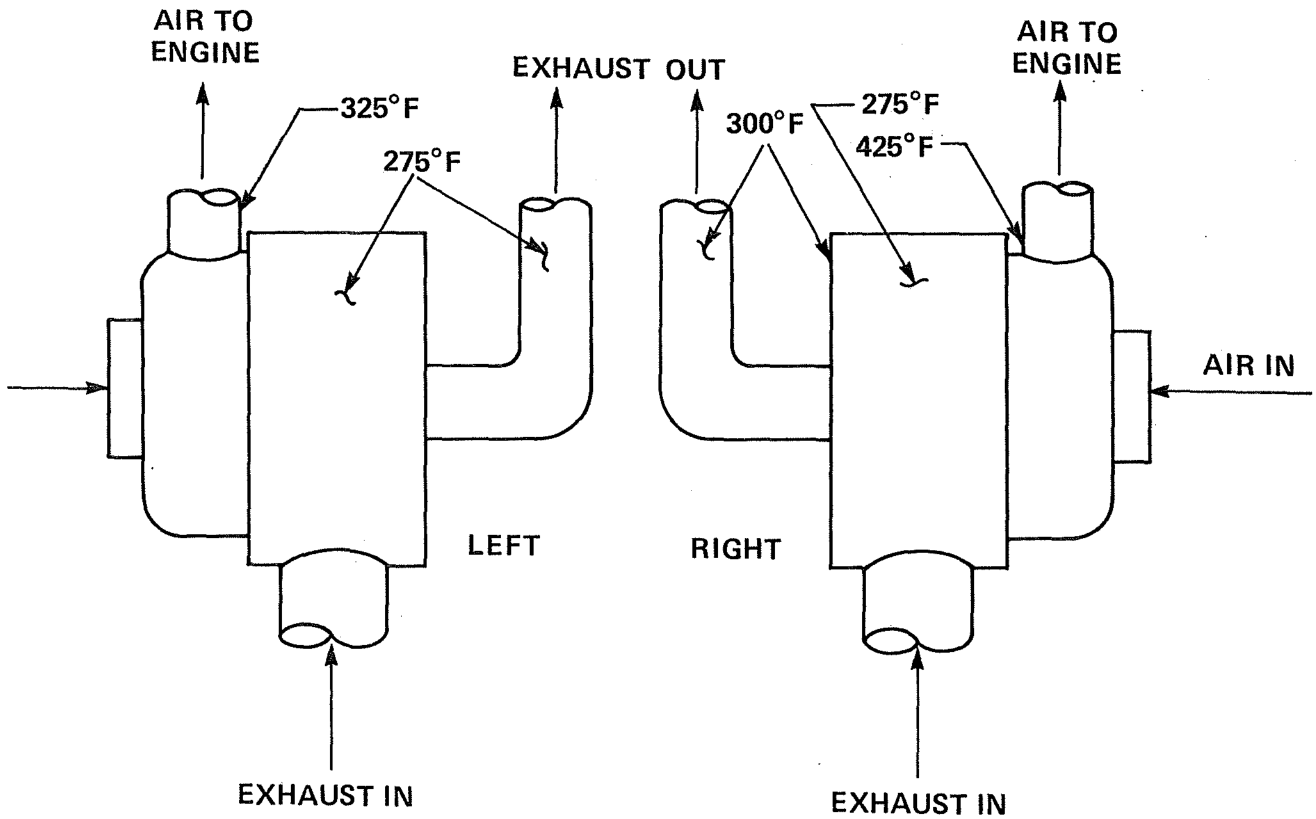

FIGURE 45 - SURFACE TEMPERATURES IN TURBOCHARGER AREA WITH ENGINE COOLING WATER

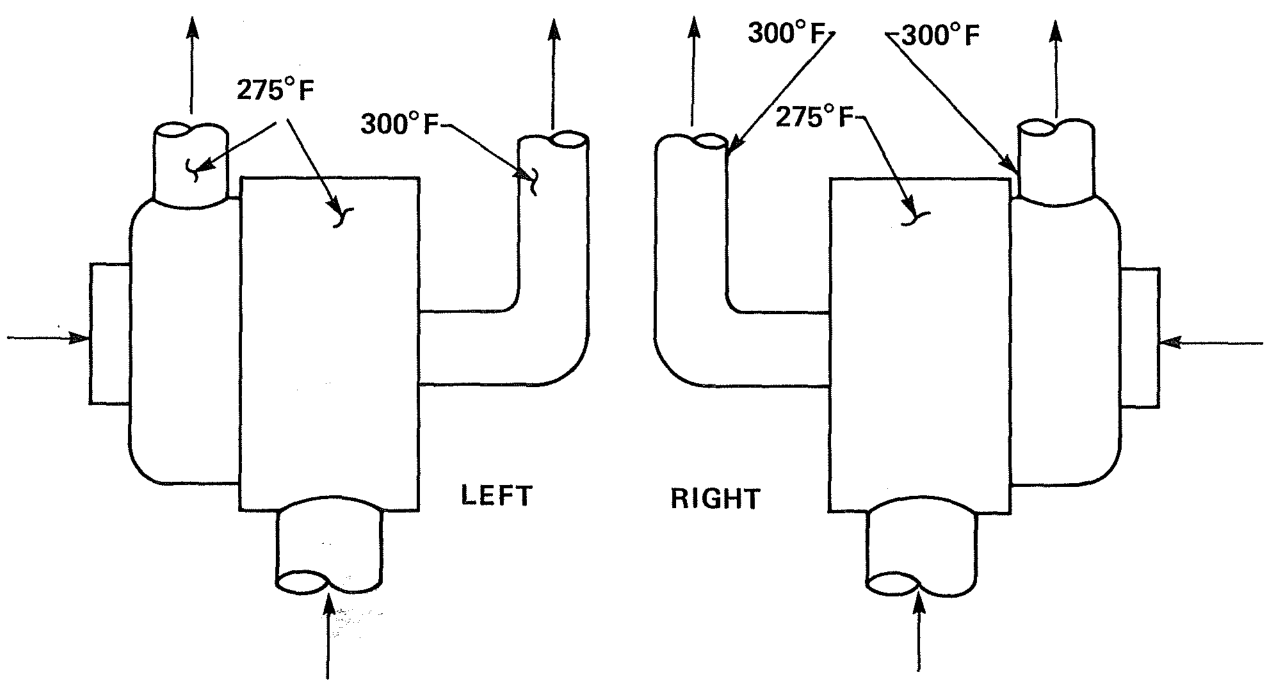

FIGURE 46 - SURFACE TEMPERATURES IN TURBOCHARGER AREA WITH POWER OUTPUT LIMITED TO 775 Bhp

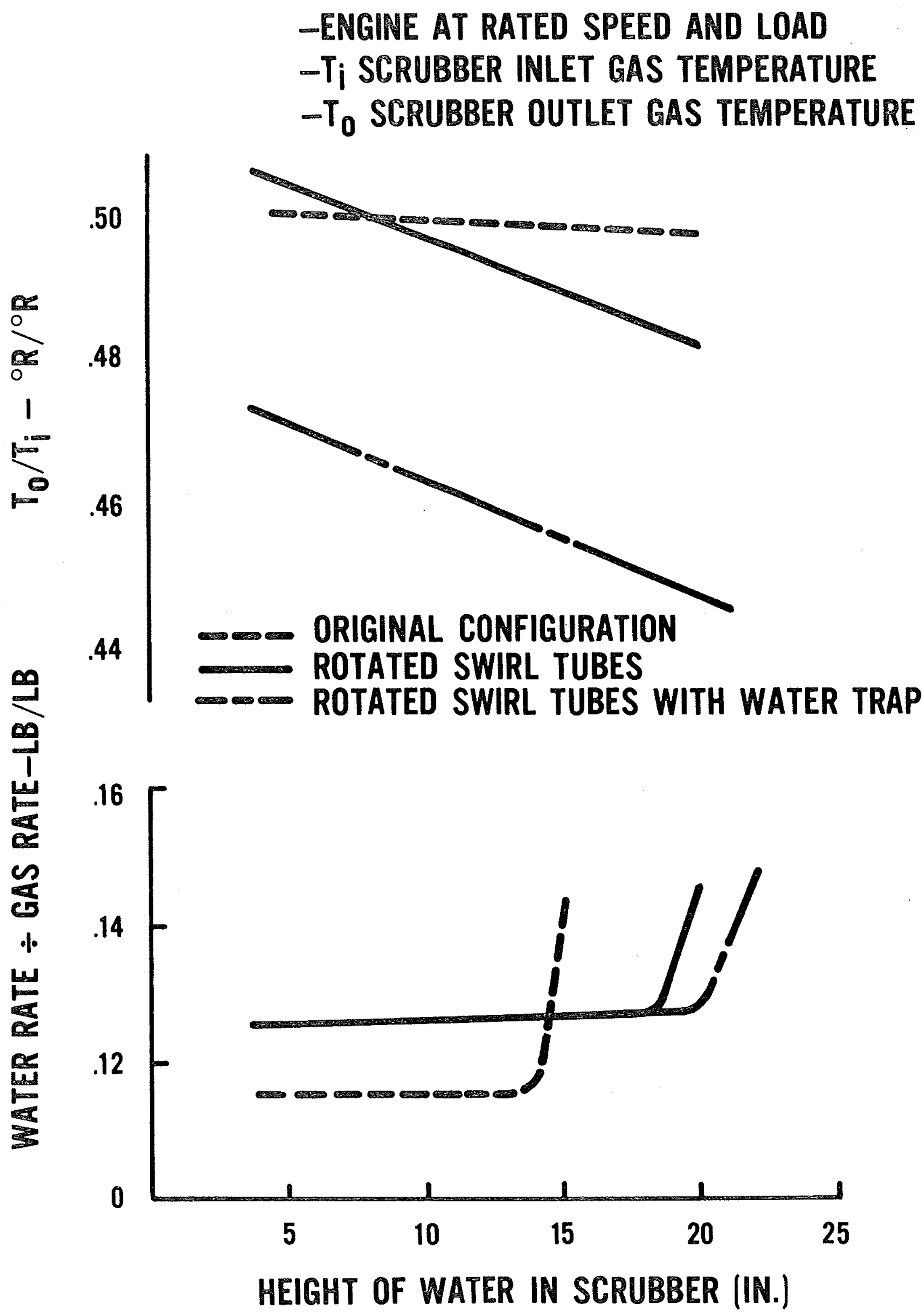

FIGURE 47

COMPARISON OF EXHAUST SCRUBBER PERFORMANCES ON CATERPILLAR D348 ENGINE

TABLE 4

Cooling Water Consumption and Final Exhaust Gas Temperature

A. Rotated Swirl Tubes

Water Level (gals)	Water Consumption (gpm)	Scrubber Inlet Temperature °F	Scrubber Outlet Temperature °F
40	2.51	744	147
60	2.53	755	147
80	2.55	760	147
100	2.55	778	148
120	2.55	796	150
130	2.85	804	150

B. Rotated Swirl Tubes and Water Trap

Water Level (gals)	Water Consumption (gpm)	Scrubber Inlet Temperature °F	Scrubber Outlet Temperature °F
40	2.55	837	150
60	2.55	852	150
80	2.55	864	152
100	2.63	880	152
120	2.59	899	152
130	2.59	924	154
140	2.81	914	153

This was an increase of 6 inches over the baseline scrubber. Comparing the temperature ratio across the scrubber for the three different configurations reveals an unexpected behavior. The decreased temperature ratio for the rotated swirl tube configuration indicates a lesser dependence on scrubber inlet temperature than with the original configuration except at water levels below 7.5 inches. The temperature ratio across the scrubber with rotated swirl tubes and a water trap was reduced approximately 0.035 at all water heights as compared to the scrubber with only the rotated swirl tubes.

C. Explosion Tests

With 120 gallons of water in the scrubber, 20 dynamic and 20 static tests produced no explosion of the gas-air mixture surrounding the engine. In order to determine the low water level at which an external explosion would occur, the water volume was lowered to 110 gallons. After 20 static tests, no external explosion had occurred but on the first dynamic test the surrounding gas-air mixture was ignited. The water volume was raised to 120 gallons once again but this time an explosion occurred on the second dynamic test. At 130 gallons, an explosion occurred on the first dynamic test. The exhaust system was then pressurized and when no leaks were discovered, it was suspected that an internal fialure of the scrubber had occurred. One wall of the scrubber was removed using a welding torch and the subsequent inspection revealed that the baffle plate adjacent to the swirl tubes had broken at the weld joining it to the bottom surface of the scrubber. This break allowed the flame to propagate out of the scrubber to the surrounding gas-air mixture. The failure apparently occurred during tests at the 110 gallon level and was caused by a combination of insufficient material thickness of the baffle plate and oxidation of the material.

D. Gaseous Emissions Test

Results of the gaseous emissions test with 1-1/2% natural gas in the intake are tabulated in Table 5 . Only HC, CO and NO were measured for this series of runs and these results compared favorably with those obtained in Phase I. This was expected since no significant changes were made to the engine system.

TABLE 5

Gaseous Emissions of Caterpillar D348 Engine with 1.5% Natural Gas Added to Intake Air

Engine Speed rpm	Power BHP	CO ppm	UBHC ppm	NO ppm	NO_x ppm
2000	1040	491	173	502	567
	825	658	332	423	559
	560	789	965	328	527
	300	1032	2153	240	442
1800	993	479	168	474	553
	778	592	375	418	550
	525	784	776	324	546
	277	917	2120	202	432
1600	920	421	196	405	488
	709	536	332	376	484
	480	760	703	280	475
	250	1051	1875	206	398
1400	805	362	168	456	502
	617	394	288	442	489
	425	649	544	333	499
	203	1010	1774	260	480

APPENDIX A

Data on Effect of 1.5% Natural Gas Addition

Table A-1. Effect of 1.5% Natural Gas Addition to Intake Air on Gaseous Emissions of Cummins VTA-1710 Engine

Engine Speed rpm		Power hp	CO ppm	CO_2 %	NO_x ppm	O_2 %	UBHC ppmC
2100	*	887	1000	8.49	1769	6.60	176
		796	675	8.46	1828	7.90	120
	*	666	827	7.35	1106	8.87	448
		588	233	7.16	1002	9.74	80
	*	444	944	6.19	609	10.83	1376
		387	150	6.10	644	11.50	80
	*	222	929	4.62	275	13.04	1776
		170	152	4.43	273	13.86	128
	*	–	726	2.56	100	16.88	6912
		–	185	2.68	113	16.75	112
1900	*	869	1299	8.68	1896	6.23	243
		779	909	8.47	1868	7.11	152
	*	652	827	7.62	1242	8.19	408
		574	291	7.48	1206	9.01	46
	*	435	811	6.37	681	10.67	1168
		380	173	6.27	702	10.95	26
	*	218	891	4.77	291	12.73	3088
		180	164	4.62	301	13.53	56
	*	–	726	2.34	100	17.25	7424
		–	220	2.39	108	17.31	160
1700	*	809	1725	8.86	1963	6.01	440
		734	1326	8.76	2070	6.81	352
	*	607	853	7.88	1269	7.99	392
		530	409	7.88	1366	8.74	32
	*	405	752	6.77	690	10.00	888
		365	210	6.69	713	10.60	24
	*	202	780	5.05	285	12.69	2272
		181	164	4.97	300	13.20	56
	*	–	662	2.17	85	17.50	5888
		–	276	2.23	95	17.62	296
1500	*	707	3157	9.42	2112	4.95	560
		656	2524	9.44	2214	5.63	480
	*	530	1147	8.51	1431	6.98	336
		488	771	8.54	1536	7.54	40
	*	354	803	7.11	764	9.33	728
		320	317	7.10	815	10.00	24
	*	177	833	5.07	298	12.32	2464
		155	189	4.97	303	13.20	52
Idle	*	–	765	1.91	75	17.75	8640
		–	328	2.01	85	18.00	432

NOTE: Runs with * are with gas; all others without gas

Table A-2. Effect of 1.5% Natural Gas Addition to Intake Air on Gaseous Emissions of Caterpillar D348 Engine

Engine Speed rpm		Power Bhp	CO ppm	CO_2 %	NO_x ppm	O_2 %	UBHC ppmC
2000	*	1026	497	8.32	501	7.63	192
		913	183	7.83	486	8.92	44
	*	809	644	7.19	474	8.97	372
		736	173	7.13	458	9.45	52
	*	551	796	5.96	446	12.09	936
		437	88	5.82	425	12.74	72
	*	293	954	4.62	362	13.16	2128
		223	44	4.38	318	14.06	128
	*	–	1638	2.17	129	17.25	5312
		–	354	2.39	156	17.05	184
1800	*	992	439	7.96	555	8.54	200
		876	136	7.77	534	9.62	88
	*	770	573	7.10	478	10.35	368
		648	112	6.71	434	10.85	64
	*	522	748	6.12	456	10.93	848
		433	113	5.82	420	11.68	80
	*	276	917	4.62	400	13.04	1936
		200	140	4.30	350	14.04	104
	*	–	1699	1.91	105	17.50	5632
		–	418	2.12	132	17.62	152
1600	*	915	322	7.11	546	10.01	184
		809	99	6.80	508	10.51	64
	*	708	439	7.27	491	9.10	272
		604	100	7.10	456	10.00	64
	*	479	699	6.35	484	10.46	752
		425	100	6.25	446	11.02	80
	*	250	929	4.83	442	12.22	1840
		207	152	4.56	373	13.02	136
	*	–	1653	1.66	82	17.25	6400
		–	508	1.86	105	17.37	192
1500	*	866	342	8.32	504	7.65	176
		763	99	8.34	466	8.30	32
	*	669	392	7.62	482	8.54	288
		598	86	7.66	463	8.80	88
	*	452	621	6.77	460	9.54	648
		398	100	6.58	422	10.20	104
	*	235	842	5.12	447	11.52	1552
		204	102	4.85	387	12.43	72
	*	–	1518	1.61	72	16.50	6528
		–	521	1.76	90	16.75	192
1400	*	810	376	8.60	477	7.54	176
		715	134	8.48	436	8.03	32
	*	624	404	8.05	482	8.19	272
		549	111	7.99	434	9.01	40
	*	421	573	7.35	488	9.43	496
		376	149	7.04	430	10.27	80
	*	219	941	5.20	442	11.99	1696
		189	177	4.84	382	12.88	112
	*	–	1576	1.51	58	17.25	7232
		–	585	1.71	112	17.50	240
Idle	*	–	1345	1.14	51	18.75	8832
		–	688	1.23	62	19.00	468

NOTE: Runs with * are with gas; all others without gas

Table A-3. Effect of 1.5% Natural Gas Addition to Intake Air on Gaseous Emissions of Detroit Diesel 12V-149TI Engine

Engine Speed rpm		Power Bhp	CO ppm	CO_2 %	NO_x ppm	O_2 %	UBHC ppmC
1900	*	1340	1700	5.47	1048	11.25	3200
		1198	1142	5.61	1115	11.38	204
	*	1001	924	4.75	969	13.20	3808
		909	424	4.90	1016	14.42	312
	*	668	800	3.86	727	14.26	4288
		623	115	4.11	818	15.15	200
	*	336	789	2.88	385	16.63	6080
		286	90	2.83	399	14.50	232
	*	–	469	1.47	131	20.00	8720
		–	133	1.51	155	20.50	372
1800	*	1310	1596	5.70	1095	10.68	3392
		1175	1098	5.82	1232	10.09	216
	*	984	875	5.40	1051	11.95	4096
		895	423	5.11	1114	12.38	288
	*	656	699	4.10	767	12.90	4288
		572	103	4.11	823	13.71	212
	*	332	776	2.94	410	14.73	5840
		260	104	2.83	419	15.70	276
	*	–	508	1.37	140	17.87	9120
		–	159	1.42	175	18.25	304
1700	*	1253	1952	5.86	1001	10.68	3648
		1125	1307	5.96	1032	11.07	188
	*	929	779	5.17	1010	11.03	4192
		815	312	5.19	1042	11.90	260
	*	619	591	4.23	748	12.32	4352
		534	77	4.17	797	13.20	204
	*	313	714	3.00	419	14.36	5840
		245	104	2.83	417	15.58	248
	*	–	508	1.32	122	18.13	9248
		–	159	1.42	160	18.37	296
1600	*	1176	2281	5.93	916	10.34	3456
		1060	1597	6.10	984	10.81	216
	*	887	1047	5.31	983	10.81	4192
		782	469	5.31	1052	11.61	261
	*	587	485	4.23	893	13.02	4448
		470	52	3.97	864	14.40	232
	*	295	641	2.88	499	15.31	5440
		239	78	2.78	507	16.19	248
	*	–	521	1.23	135	18.50	8800
		–	172	1.32	130	18.62	288
Idle	*	–	585	0.76	120	20.50	10,480

NOTE: Runs with * are with gas; all others without gas

APPENDIX B

Data on Gaseous Emissions

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3. U. S. Code of Federal Regulations, Title 30, Mineral Resources; Chapter 1 - MESA, Department of the Interior; Subchapter E - Mechanical Equipment for Mines; Tests for Permissibility and Suitability; Fees; Part 36 - Mobile Diesel-Powered Transportation Equipment for Gassy Noncoal Mines and Tunnels, Code of Federal Regulations, July 1, 1976.

APPENDIX A

Data on Effect of 1.5% Natural Gas Addition

Table A-1. Effect of 1.5% Natural Gas Addition to Intake Air on Gaseous Emissions of Cummins VTA-1710 Engine

Engine Speed rpm		Power hp	CO ppm	CO_2 %	NO_x ppm	O_2 %	UBHC ppmC
2100	*	887	1000	8.49	1769	6.60	176
		796	675	8.46	1828	7.90	120
	*	666	827	7.35	1106	8.87	448
		588	233	7.16	1002	9.74	80
	*	444	944	6.19	609	10.83	1376
		387	150	6.10	644	11.50	80
	*	222	929	4.62	275	13.04	1776
		170	152	4.43	273	13.86	128
	*	–	726	2.56	100	16.88	6912
		–	185	2.68	113	16.75	112
1900	*	869	1299	8.68	1896	6.23	243
		779	909	8.47	1868	7.11	152
	*	652	827	7.62	1242	8.19	408
		574	291	7.48	1206	9.01	46
	*	435	811	6.37	681	10.67	1168
		380	173	6.27	702	10.95	26
	*	218	891	4.77	291	12.73	3088
		180	164	4.62	301	13.53	56
	*	–	726	2.34	100	17.25	7424
		–	220	2.39	108	17.31	160
1700	*	809	1725	8.86	1963	6.01	440
		734	1326	8.76	2070	6.81	352
	*	607	853	7.88	1269	7.99	392
		530	409	7.88	1366	8.74	32
	*	405	752	6.77	690	10.00	888
		365	210	6.69	713	10.60	24
	*	202	780	5.05	285	12.69	2272
		181	164	4.97	300	13.20	56
	*	–	662	2.17	85	17.50	5888
		–	276	2.23	95	17.62	296
1500	*	707	3157	9.42	2112	4.95	560
		656	2524	9.44	2214	5.63	480
	*	530	1147	8.51	1431	6.98	336
		488	771	8.54	1536	7.54	40
	*	354	803	7.11	764	9.33	728
		320	317	7.10	815	10.00	24
	*	177	833	5.07	298	12.32	2464
		155	189	4.97	303	13.20	52
Idle	*	–	765	1.91	75	17.75	8640
		–	328	2.01	85	18.00	432

NOTE: Runs with * are with gas; all others without gas

Table A-2. Effect of 1.5% Natural Gas Addition to Intake Air on Gaseous Emissions of Caterpillar D348 Engine

Engine Speed rpm		Power Bhp	CO ppm	CO_2 %	NO_x ppm	O_2 %	UBHC ppmC
2000	*	1026	497	8.32	501	7.63	192
		913	183	7.83	486	8.92	44
	*	809	644	7.19	474	8.97	372
		736	173	7.13	458	9.45	52
	*	551	796	5.96	446	12.09	936
		437	88	5.82	425	12.74	72
	*	293	954	4.62	362	13.16	2128
		223	44	4.38	318	14.06	128
	*	–	1638	2.17	129	17.25	5312
		–	354	2.39	156	17.05	184
1800	*	992	439	7.96	555	8.54	200
		876	136	7.77	534	9.62	88
	*	770	573	7.10	478	10.35	368
		648	112	6.71	434	10.85	64
	*	522	748	6.12	456	10.93	848
		433	113	5.82	420	11.68	80
	*	276	917	4.62	400	13.04	1936
		200	140	4.30	350	14.04	104
	*	–	1699	1.91	105	17.50	5632
		–	418	2.12	132	17.62	152
1600	*	915	322	7.11	546	10.01	184
		809	99	6.80	508	10.51	64
	*	708	439	7.27	491	9.10	272
		604	100	7.10	456	10.00	64
	*	479	699	6.35	484	10.46	752
		425	100	6.25	446	11.02	80
	*	250	929	4.83	442	12.22	1840
		207	152	4.56	373	13.02	136
	*	–	1653	1.66	82	17.25	6400
		–	508	1.86	105	17.37	192
1500	*	866	342	8.32	504	7.65	176
		763	99	8.34	466	8.30	32
	*	669	392	7.62	482	8.54	288
		598	86	7.66	463	8.80	88
	*	452	621	6.77	460	9.54	648
		398	100	6.58	422	10.20	104
	*	235	842	5.12	447	11.52	1552
		204	102	4.85	387	12.43	72
	*	–	1518	1.61	72	16.50	6528
		–	521	1.76	90	16.75	192
1400	*	810	376	8.60	477	7.54	176
		715	134	8.48	436	8.03	32
	*	624	404	8.05	482	8.19	272
		549	111	7.99	434	9.01	40
	*	421	573	7.35	488	9.43	496
		376	149	7.04	430	10.27	80
	*	219	941	5.20	442	11.99	1696
		189	177	4.84	382	12.88	112
	*	–	1576	1.51	58	17.25	7232
		–	585	1.71	112	17.50	240
Idle	*	–	1345	1.14	51	18.75	8832
		–	688	1.23	62	19.00	468

NOTE: Runs with * are with gas; all others without gas

Table A-3. Effect of 1.5% Natural Gas Addition to Intake Air on Gaseous Emissions of Detroit Diesel 12V-149TI Engine

Engine Speed rpm		Power Bhp	CO ppm	CO_2 %	NO_x ppm	O_2 %	UBHC ppmC
1900	*	1340	1700	5.47	1048	11.25	3200
		1198	1142	5.61	1115	11.38	204
	*	1001	924	4.75	969	13.20	3808
		909	424	4.90	1016	14.42	312
	*	668	800	3.86	727	14.26	4288
		623	115	4.11	818	15.15	200
	*	336	789	2.88	385	16.63	6080
		286	90	2.83	399	14.50	232
	*	–	469	1.47	131	20.00	8720
		–	133	1.51	155	20.50	372
1800	*	1310	1596	5.70	1095	10.68	3392
		1175	1098	5.82	1232	10.09	216
	*	984	875	5.40	1051	11.95	4096
		895	423	5.11	1114	12.38	288
	*	656	699	4.10	767	12.90	4288
		572	103	4.11	823	13.71	212
	*	332	776	2.94	410	14.73	5840
		260	104	2.83	419	15.70	276
	*	–	508	1.37	140	17.87	9120
		–	159	1.42	175	18.25	304
1700	*	1253	1952	5.86	1001	10.68	3648
		1125	1307	5.96	1032	11.07	188
	*	929	779	5.17	1010	11.03	4192
		815	312	5.19	1042	11.90	260
	*	619	591	4.23	748	12.32	4352
		534	77	4.17	797	13.20	204
	*	313	714	3.00	419	14.36	5840
		245	104	2.83	417	15.58	248
	*	–	508	1.32	122	18.13	9248
		–	159	1.42	160	18.37	296
1600	*	1176	2281	5.93	916	10.34	3456
		1060	1597	6.10	984	10.81	216
	*	887	1047	5.31	983	10.81	4192
		782	469	5.31	1052	11.61	261
	*	587	485	4.23	893	13.02	4448
		470	52	3.97	864	14.40	232
	*	295	641	2.88	499	15.31	5440
		239	78	2.78	507	16.19	248
	*	–	521	1.23	135	18.50	8800
		–	172	1.32	130	18.62	288
Idle	*	–	585	0.76	120	20.50	10,480

NOTE: Runs with * are with gas; all others without gas

APPENDIX B

Data on Gaseous Emissions

Table B-1. Gaseous Emissions of Cummins VTA-1710 Engine with 1.5% Natural Gas Added to Intake Air

Engine Speed rpm	Power Bhp	CO ppm	CO_2 %	NO ppm	NO_x ppm	O_2 %	UBHC ppmC	M_{air} lb/min	M_{ch4} lb/min	M_{diesel} lb/min	F/A	BMEP psi	N_{Brake}	Q_d cfm
2100	901	925	8.50	1850	1877	7.20	232	128.6	1.12	5.30	.050	199	31.0	263,100
	676	829	7.28	1135	1198	9.81	384	114.3	.99	3.98	.044	149	29.9	148,000
	450	890	6.01	528	625	11.70	1344	100.5	.87	2.80	.037	99	26.8	66,800
	226	855	4.48	216	254	13.51	3392	85.4	.74	1.79	.030	50	19.4	22,100
1900	860	1243	8.65	1868	1882	6.94	240	115.8	1.01	5.01	.052	210	31.6	238,100
	645	825	7.51	1158	1234	8.85	365	99.6	.87	3.75	.046	157	30.8	133,100
	430	789	6.22	567	678	10.60	1168	85.1	.74	2.58	.039	105	28.4	61,500
	215	853	4.68	178	293	13.00	2936	71.7	.62	1.63	.031	52	20.7	21,700
1700	814	1952	9.15	1989	2003	4.79	344	100.5	.87	4.61	.055	222	32.8	220,600
	611	925	8.14	1391	1431	7.43	312	85.6	.75	3.52	.050	167	31.6	133,400
	407	759	6.89	680	790	9.83	744	71.4	.62	2.44	.043	111	29.2	60,600
	204	727	5.09	215	327	12.60	2192	58.6	.51	1.41	.033	56	23.1	19,900
1500	720	3489	9.64	2024	2037	4.85	544	84.1	.73	4.24	.059	222	32.1	188,700
	540	1311	8.67	1459	1476	6.63	432	71.1	.62	2.99	.051	167	33.1	114,400
	360	903	7.20	686	768	8.91	952	59.8	.52	2.08	.043	111	30.5	49,200
	180	913	5.10	197	295	11.46	2960	49.8	.43	1.16	.032	56	24.5	15,200
Idle	–	726	1.76	50	95	16.00	8832							

Table B-1 (Cont'd)

Engine Speed rpm	Power hp	HYDROCARBON CONCENTRATION, PPMC	Non-Reactive Hydrocarbons				Reactive Hydrocarbons			TC GAS Chromatograph Analysis	
		CH_4	C_2H_6	C_2H_2	C_3H_8	C_6H_6	C_2H_4	C_3H_6	C_7H_8	H_2 ppm	N_2 %
2100	901	85	2.1	3.1	tr	3.5	37.1	8.6	0.0	0	79.53
	676	271	9.6	tr	3.2	1.0	17.9	4.3	tr	0	79.37
	450	1141	43.6	tr	12.3	tr	41.1	6.5	0.0	0	78.70
	226	1540	143.9	1.3	49.1	tr	49.6	8.3	tr	0	77.69
1900	860	183	8.6	8.9	3.2	9.4	65.8	15.2	1.4	187	79.55
	645	286	10.7	0.6	4.1	1.0	18.7	4.6	tr	0	79.62
	430	900	37.1	0.7	13.0	0.9	35.4	7.4	tr	0	79.93
	215	1557	121.9	1.4	50.5	1.0	48.1	10.6	0.0	0	77.78
1700	814	164	4.2	11.3	1.1	9.7	79.9	17.4	1.7	160	79.86
	611	319	9.9	0.6	4.0	0.8	18.3	4.6	tr	113	79.12
	407	604	31.0	0.6	13.1	0.0	34.1	7.2	0.0	77	78.00
	204	1429	113.3	1.3	53.8	0.0	47.3	11.3	0.0	0	76.96
1500	720	139	9.6	24.0	3.1	4.7	146	33.1	0.0	540	78.82
	540	216	12.2	1.6	5.6	7.4	19.8	5.2	0.0	166	78.38
	360	739	31.4	0.7	6.9	2.7	23.7	3.6	tr	45	78.73
	180	1453	112.6	1.4	24.3	2.1	40.2	4.8	tr	0	77.61
Idle	–	1688	357.1	4.0	69.7	3.8	53.3	13.0	1.3	0	78.10

Engine Speed rpm	Power hp	INDIVIDUAL ALDEHYDES AS MEASURED BY DNPH-GC CONCENTRATIONS EXPRESSED IN PPM							
		Form	Acetaldehyde	Acetone[1]	ISO-Butraldehyde	Crotonaldehyde	Hexanol	Benzaklehyde	Total
2100	901	3.88	ND	ND	ND	0.06	ND	0.57	4.51
	676	4.50	ND	ND	ND	0.02	2.07	ND	6.59
	450	24.90	0.05	0.11	ND	ND	ND	ND	25.06
	226	56.69	1.62	0.48	ND	0.43	ND	ND	59.22
1900	860	2.92	ND	ND	ND	0.21	ND	1.09	4.22
	645	1.74	ND	ND	ND	ND	ND	ND	1.74
	430	9.17	0.14	ND	ND	ND	ND	ND	9.31
	215	75.10	1.68	0.60	1.01	ND	ND	ND	78.39
1700	814	14.50	ND	ND	ND	ND	ND	ND	14.50
	611	3.83	1.22	0.41	ND	ND	ND	2.69	8.15
	407	4.30	ND	ND	ND	ND	ND	ND	4.30
	204	45.00	1.19	0.32	ND	ND	ND	ND	46.51
1500	720	32.40	3.47	0.28	ND	5.43	ND	ND	41.58
	540	8.06	0.28	ND	ND	ND	ND	ND	8.34
	360	17.80	0.23	ND	ND	2.32	ND	ND	20.35
	180	83.60	1.70	0.46	ND	0.77	ND	ND	86.53
Idle	–	76.60	9.89	7.01	1.89	4.00	ND	ND	99.33

(1) Reported as acetone, includes acetone, propanol and acrolein

Table B-2. Gaseous Emissions of Caterpillar D348 Engine with 1.5% Natural Gas Added to Intake Air

Engine Speed rpm	Power Bhp	CO ppm	CO_2 %	NO ppm	NO_x ppm	O_2 %	UBHC ppmC	M_{air} lb/min	M_{ch4} lb/min	M_{diesel} lb/min	F/A	BMEP psi	N_{Brake}	Q_d cfm
2000	1048	473	8.20	495	572	7.84	152	155.4	1.35	5.74	.046	232	32.6	95,100
	823	655	7.09	404	537	9.87	344	135.7	1.18	4.34	.041	183	32.7	77,500
	561	845	6.04	330	526	11.27	912	109.6	.95	2.93	.035	124	31.5	60,900
	301	993	4.64	219	433	12.32	2064	86.1	.75	2.50	.038	67	20.2	39,200
1800	1001	473	8.30	481	554	8.29	160	143.8	1.25	5.36	.046	246	33.3	85,200
	782	583	7.32	403	527	9.16	392	122.0	1.06	4.02	.042	193	33.7	68,400
	531	755	6.24	319	513	10.04	776	92.3	.80	2.70	.038	131	33	50,100
	281	1032	4.78	231	443	11.54	2080	73.7	.64	1.50	.029	69	28.4	34,100
1600	926	437	8.28	376	476	9.63	192	128.7	1.12	4.94	.047	257	33.7	65,300
	718	559	7.58	380	481	9.74	352	107.8	.94	3.62	.042	199	34.7	55,100
	486	757	6.65	306	482	10.88	728	84.1	.73	2.38	.037	135	34.2	42,800
	255	989	4.83	222	421	12.50	1008	63.9	.56	1.30	.029	71	29.3	28,100
1400	804	331	8.89	492	532	7.11	144	109.5	.95	4.42	.049	255	33.0	62,500
	622	368	8.04	418	495	8.06	256	86.6	.75	3.24	.046	197	34.3	35,100
	421	608	7.17	376	519	8.84	544	65.8	.57	2.09	.041	133	34.6	36,300
	206	939	5.04	291	502	10.55	1824	49.4	.43	1.08	.031	65	29.3	26,100
Idle	–	1330	1.10	15	57	17.25	8768							

Table B-2 (Cont'd)

		HYDROCARBON CONCENTRATION, PPMC								TC GAS Chromatograph Analysis	
		Non-Reactive Hydrocarbons				Reactive Hydrocarbons					
Engine Speed rpm	Power hp	CH_4	C_2H_6	C_2H_2	C_3H_8	C_6H_6	C_2H_4	C_3H_6	C_7H_8	H_2 ppm	N_2 %
2000	1048	122	1.6	0.7	ND	1.1	19.3	4.4	ND	ND	80.69
	823	317	5.5	0.8	1.6	0.9	20.9	5.1	ND	ND	80.92
	561	830	17.8	0.6	5.9	0.8	29.0	6.5	ND	ND	79.83
	301	1737	44.0	0.9	15.1	1.1	40.4	9.2	ND	ND	80.46
1800	1001	131	2.0	0.4	ND	0.8	18.0	4.7	ND	ND	80.29
	782	320	6.0	0.5	1.9	0.7	16.1	3.8	ND	ND	80.94
	531	735	15.5	0.5	5.1	0.7	23.5	5.1	ND	ND	80.36
	281	1803	46.4	0.6	16.0	0.9	38.7	8.6	ND	ND	79.85
1600	926	157	2.4	0.3	0.6	0.8	14.0	3.7	ND	ND	81.57
	719	298	5.8	0.2	2.1	0.5	12.3	2.8	ND	ND	80.38
	486	654	13.3	0.3	4.5	0.7	21.0	4.3	ND	ND	80.84
	255	1740	43.4	0.4	15.4	0.8	36.2	7.9	ND	ND	79.61
1400	804	135	3.3	0.3	0.8	0.6	12.4	2.4	ND	ND	80.73
	622	228	8.0	0.2	2.0	0.4	12.6	2.9	ND	ND	81.20
	421	440	15.8	0.4	4.1	1.2	21.2	4.2	ND	ND	80.25
	206	1684	70.5	0.6	19.0	0.7	44.6	8.4	ND	ND	78.96
Idle		2441	382	2.0	115	3.5	110	25.3	2.7	ND	76 70

Engine Speed rpm	Power hp	INDIVIDUAL ALDEHYDES AS MEASURED BY DNPH-GC CONCENTRATIONS EXPRESSED IN PPM							
		Form	Acetaldehyde	Acetone[1]	ISO-Butraldehyde	Crotonaldehyde	Hexanol	Benzaldehyde	Total
2000	1048	2.91	ND	0.13	0.52	0.37	3.84	0.25	8.02
	823	1.62	ND	ND	0.20	0.19	4.65	ND	6.66
	561	2.81	ND	ND	0.46	16.39	ND	0.50	20.16
	301	21.74	0.85	ND	0.39	34.93	ND	ND	57.91
1800	1001	3.08	ND	ND	0.46	1.89	4.82	ND	10.25
	782	5.52	ND	ND	0.86	1.40	3.93	0.38	12.09
	531	4.23	ND	ND	0.34	10.78	3.85	ND	19.20
	281	6.02	0.40	ND	0.34	30.06	2.58	0.26	39.66
1600	926	2.50	ND	ND	0.67	20.91	3.02	ND	27.10
	719	3.98	ND	ND	0.60	3.24	3.12	1.01	11.95
	486	3.59	0.39	ND	0.53	19.89	2.54	0.63	27.57
	255	25.41	1.17	0.82	0.85	5.96	1.61	0.88	36.70
1400	804	2.35	0.23	ND	0.67	0.96	3.04	1.28	8.53
	622	2.90	ND	ND	ND	3.04	3.42	1.16	10.52
	421	1.50	ND	ND	0.54	4.77	2.93	1.67	11.41
	206	4.14	0.28	ND	0.61	25.91	1.53	0.64	33.11
Idle		92.77	12.13	11.34	8.64	7.93	3.87	3.09	139.77

(1) Reported as acetone, includes acetone, propanol and acrolein

Table B-3. Gaseous Emissions of Detroit Diesel 12V-149TI Engine with 1.5% Natural Gas Added to Intake Air

Engine Speed rpm	Power Bhp	CO ppm	CO_2 %	NO ppm	NO_x ppm	O_2 %	UBHC ppmC	M_{air} lb/min	M_{ch4} lb/min	M_{diesel} lb/min	F/A	BMEP psi	N_{Brake}	Q_d cfm
1900	1334	1668	5.73	1060	1102	10.89	3520	289.0	2.52	7.69	.035	156	33.1	349,800
	1001	917	4.97	973	1015	12.10	3552	256.8	2.24	5.57	.030	117	33.0	283,100
	668	899	4.05	668	768	13.51	3968	218.6	1.90	3.81	.026	78	30.8	180,200
	336	833	2.90	292	398	15.09	5647	188.0	1.64	2.37	.021	39	22.9	78,300
1800	1311	1679	6.02	1002	1058	10.44	3584	271.2	2.36	7.50	.036	161	33.5	315,500
	984	915	5.25	1041	1098	11.60	3904	230.8	2.00	5.26	.031	121	34.6	276,200
	656	687	4.24	681	762	13.01	4480	194.8	1.70	3.58	.027	81	32.5	159,500
	332	794	2.90	282	387	14.96	6080	169.8	1.48	2.20	.022	41	24.5	68,800
1700	1256	1682	6.04	1060	1116	10.70	3616	267.4	2.32	7.06	.035	164	33.9	327,500
	938	681	5.19	1098	1140	11.73	4032	224.4	1.96	5.08	.031	122	34.3	278,900
	628	566	4.17	814	913	13.24	4000	192.6	1.68	3.48	.027	82	31.9	189,400
	317	720	2.90	355	469	15.09	5344	164.0	1.42	2.03	.021	41	25.1	80,900
1600	1176	2031	6.17	945	1028	10.07	3424	247.8	2.16	6.89	.037	163	32.8	280,118
	887	885	5.45	1121	1191	11.21	4064	197.8	1.72	4.85	.033	123	34.3	257,800
	587	455	4.37	892	991	12.74	4160	168.2	1.46	3.20	.028	81	32.8	180,100
	295	647	3.03	407	541	14.61	5504	152.2	1.32	1.93	.021	41	24.7	87,000
Idle	–	495	0.72	65	160	18.75	9040							

Table B-3 (Cont'd)

Engine Speed rpm	Power hp	HYDROCARBON CONCENTRATION, PPMC Non-Reactive Hydrocarbons CH_4	C_2H_6	C_2H_2	C_3H_8	C_6H_6	Reactive Hydrocarbons C_2H_4	C_3H_6	C_7H_8	TC GAS Chromatograph Analysis H_2 ppm	N_2 %	INDIVIDUAL ALDEHYDES AS MEASURED BY DNPH-GC CONCENTRATIONS EXPRESSED IN PPM Form	Acetaldehyde	Acetone(1)	ISO-Butraldehyde	Crotonaldehyde	Hexanol	Benzaldehyde	Total
1900	1334	1081	130	1.4	26.6	1.4	36.2	8.6	0.0	155	80.01	1.92	0.0	0.89	0.70	3.56	0.19	0.31	7.57
	1001	1119	151	1.6	33.5	2.0	36.0	14.2	0.3	245	79.44	5.62	0.0	0.44	0.27	0.67	4.99	0.52	12.51
	668	1135	158	0.8	35.7	0.8	24.1	4.6	0.4	0	79.67	4.68	0.0	0.74	0.38	0.88	4.14	0.21	11.03
	336	1199	198	1.2	41.0	1.2	53.0	8.1	0.4	0	79.31	42.80	2.19	0.0	0.49	2.02	0.38	0.83	48.71
1800	1311	1099	143	1.8	28.5	1.9	45.4	9.9	0.1	164	80.14	10.14	0.37	0.0	0.38	1.47	3.06	0.31	15.73
	984	1122	161	2.0	33.9	2.3	42.1	15.5	1.0	78	79.44	4.50	0.0	0.94	0.44	1.52	5.83	0.73	13.96
	656	1149	174	1.0	36.9	2.7	21.2	4.4	0.3	217	80.22	4.00	0.49	0.0	0.16	0.0	4.78	0.31	9.74
	332	1210	214	1.6	42.8	3.2	50.1	7.1	0.3	0	78.99	29.98	1.06	0.56	0.33	1.10	0.96	0.84	34.83
1700	1256	1135	152	1.4	30.5	0.5	44.2	9.2	0.0	104	80.84	4.89	0.0	0.33	0.0	0.52	0.0	0.21	5.95
	938	1167	178	1.3	39.3	0.8	32.0	9.7	0.0	0	80.80	4.50	0.91	0.0	0.43	0.88	5.10	0.41	12.23
	628	1184	165	0.9	39.7	1.5	15.9	2.7	0.0	0	80.06	4.49	0.0	0.89	0.87	2.81	0.0	0.52	9.58
	317	1253	204	1.5	46.9	1.0	38.8	5.3	0.0	0	78.81	19.37	0.11	0.0	0.0	0.78	2.27	0.31	22.84
1600	1176	1083	139	2.1	27.2	2.7	56.8	8.1	0.0	0	79.68	2.66	0.0	0.71	0.11	3.84	0.0	0.52	7.84
	887	1174	180	1.3	37.4	2.8	31.7	6.7	0.0	67	79.98	2.91	0.0	0.86	0.33	0.79	3.15	0.54	8.58
	587	1189	187	0.8	39.1	1.8	15.3	2.2	0.7	0	79.13	3.63	0.0	0.98	0.38	0.63	4.87	0.10	10.59
	295	1247	185	1.1	56.9	2.0	32.7	4.4	1.4	0	78.16	7.28	1.30	0.0	0.49	0.84	0.10	1.05	11.06
Idle	–	1360	300	1.8	98.8	1.1	25.5	5.3	1.2	0	77.78	39.50	1.20	2.03	0.71	4.15	0.29	3.46	51.41

(1) Reported as Acetone; includes acetone, propanol, and acrolein

APPENDIX C

Equations Used in Calculations

CALCULATIONS

Schedule 31 requires that 1.5% by volume of methane be added to intake air ∴

$$\dot{M}_{air} = \rho_{air} \cdot V_{air}$$

Volume CH_4 Intake = .015 Volume Intake Air

$\dot{M}_{air}$ = Mass of Intake Airflow in lbs. per minute

ρ_{air} = Density of Air in lbs. per ft^3

V_{air} = Volumetric flow of air in ft^3 per minute

$$\dot{M}_{CH_4} = \rho_{CH_4} \cdot V_{CH_4} \text{ but } \rho_{CH_4} = 0.58\, \rho_{air}$$
$$= .58\, \rho_{air} \cdot .015\, V_{air} = .0087\, \dot{M}_{air}$$

$\dot{M}_{CH_4}$ = Flow of CH_4 in lbs. per minute

ρCH_4 = Density of CH_4 in lbs. per ft^3

V_{CH_4} = Volumetric Flow of CH_4 in ft^3 per minute

$$\dot{M}_{exhaust} = \dot{M}_{air} + \dot{M}_{CH_4} + \dot{M}_{diesel}$$
$$\text{Fuel/Air Ratio} = \frac{\dot{M}_{CH_4} + \dot{M}_{diesel}}{\dot{M}_{air}}$$

$\dot{M}_{exhaust}$ = Exhaust flow in lbs. per minute

$\dot{M}_{diesel}$ = Diesel Fuel Flow in lbs. per minute

BMEP = Brake Mean Effective Pressure in lbs. per in^2

$$BMEP = \frac{150.8 \cdot \text{Torque (lbs-ft)}}{\text{Engine Displacement } (in^3)} \quad \text{(4 cycle engine)}$$
$$= \frac{75.4 \cdot \text{Torque (lbs-ft)}}{\text{Engine Displacement } (in^3)} \quad \text{(2 cycle engine)}$$

CALCULATIONS (Cont'd)

$$\text{Brake Efficiency} = \frac{\text{Brake Horsepower} \cdot 2545 \cdot 100}{(\dot{M}_{diesel} \cdot \text{lower heating value}) + (\dot{M}_{CH_4} \cdot \text{lower heating value})}$$

Lower heating value of diesel = 18600 BTU per lb.

Lower heating value of CH_4 = 920 BTU per ft^3

$$Qd = \frac{\dot{M}_{exhaust}}{\rho_{air\text{-}standard}} \cdot \frac{NO_X}{\text{Schedule 31 Limit}} - 1$$

NO_X = NO_X Concentration in parts per million (ppm)

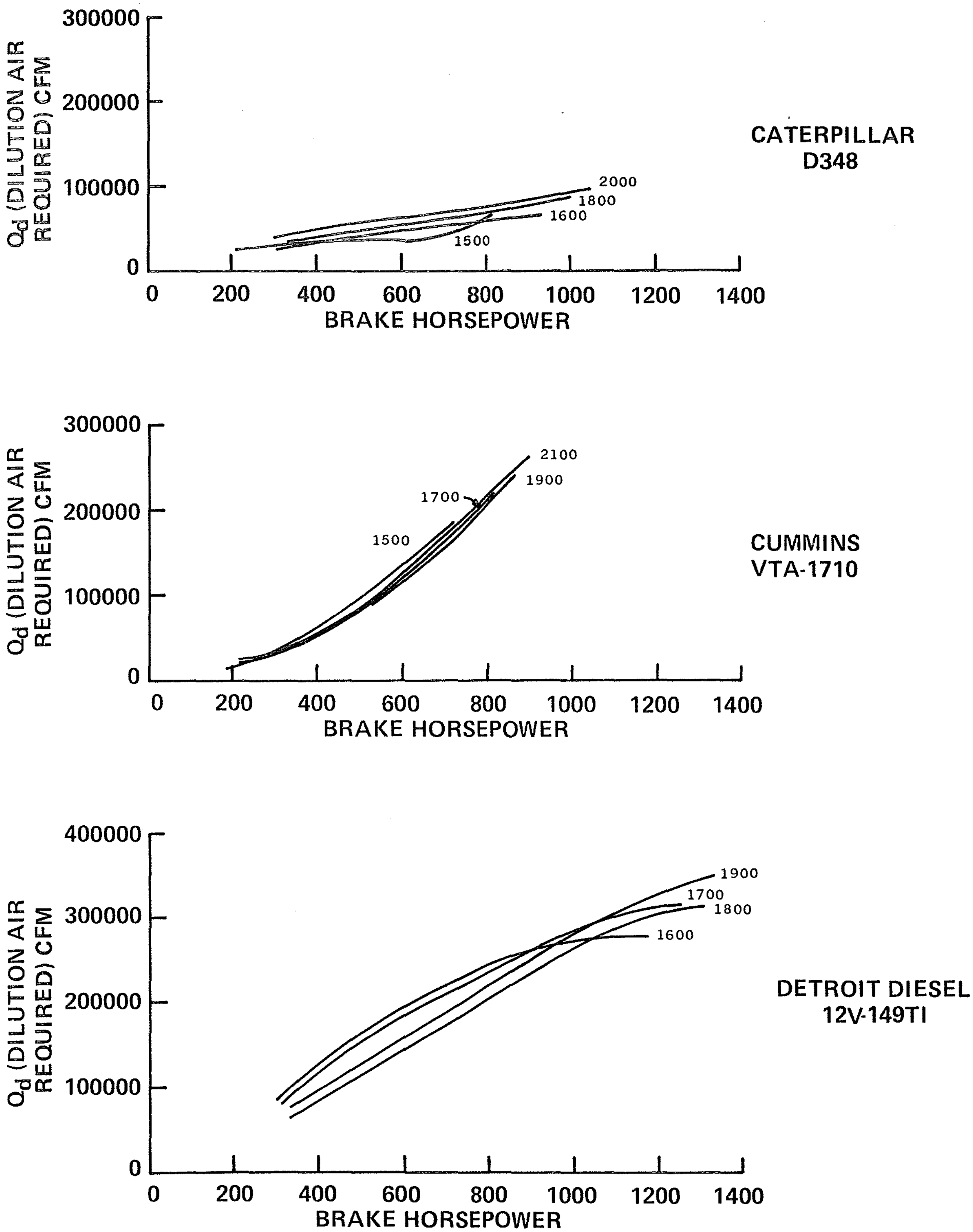
Qd = Dilution Air Required in cu. ft. per minute (CFM)

$\rho_{air\text{-}standard}$ = .0765 lbs. per ft^3 standard air density

Schedule 31 Limit = 12.5 parts per million oxides of nitrogen

APPENDIX D

Graphs of Emissions and Ventilation Requirements of Engines Tested with 1.5% Natural Gas Added to Intake Air



NOTE: ALL Q_d BASED ON NO_x EMISSIONS WITH 1-1/2% NATURAL GAS.

FIGURE D-1 - VENTILATION REQUIREMENTS OF TESTED ENGINES AT VARIOUS SPEEDS

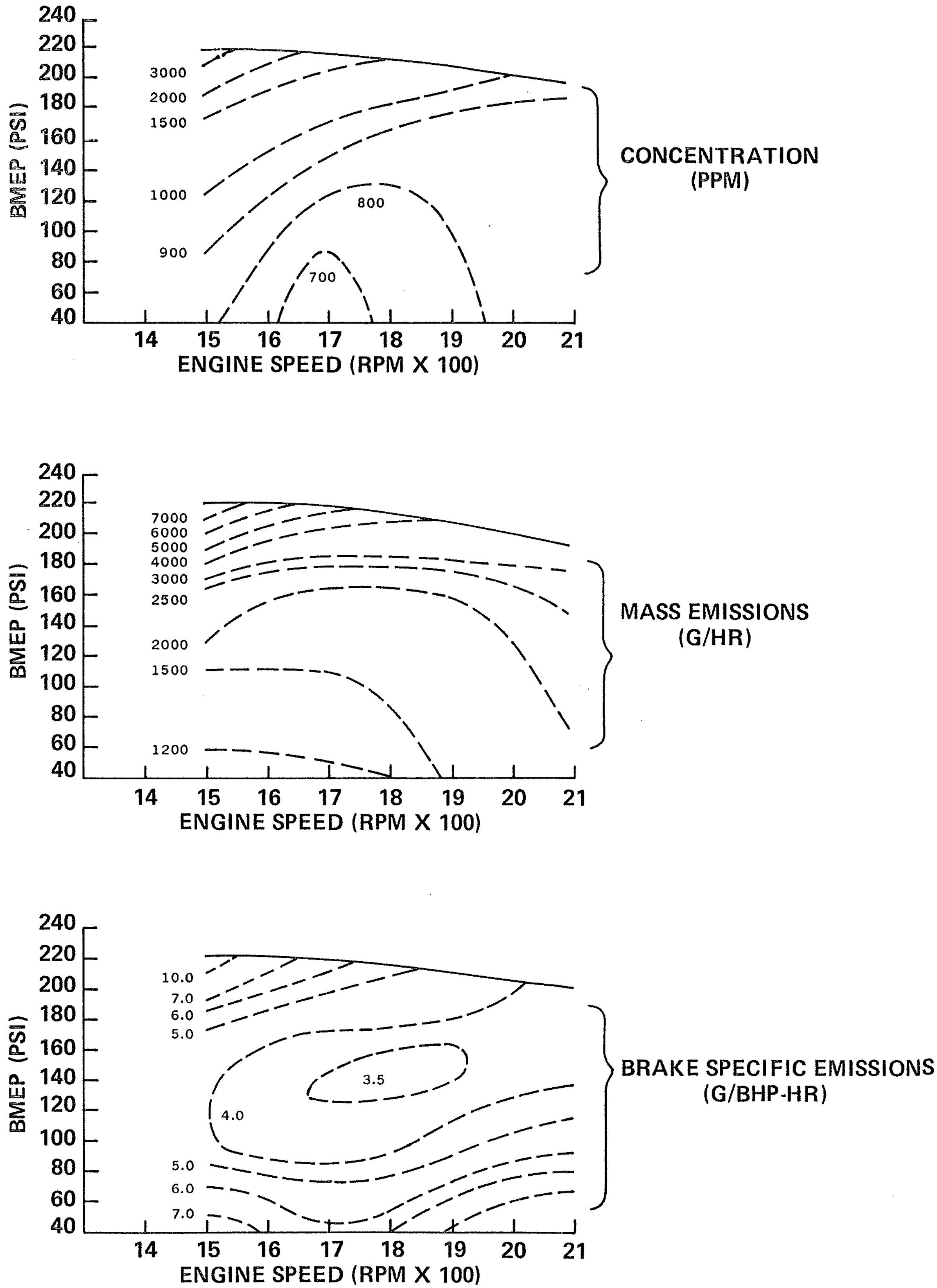


FIGURE D-2 - CO EMISSIONS OF CUMMINS VTA-1710 ENGINE WITH 1½% NATURAL GAS ADDED TO INTAKE AIR

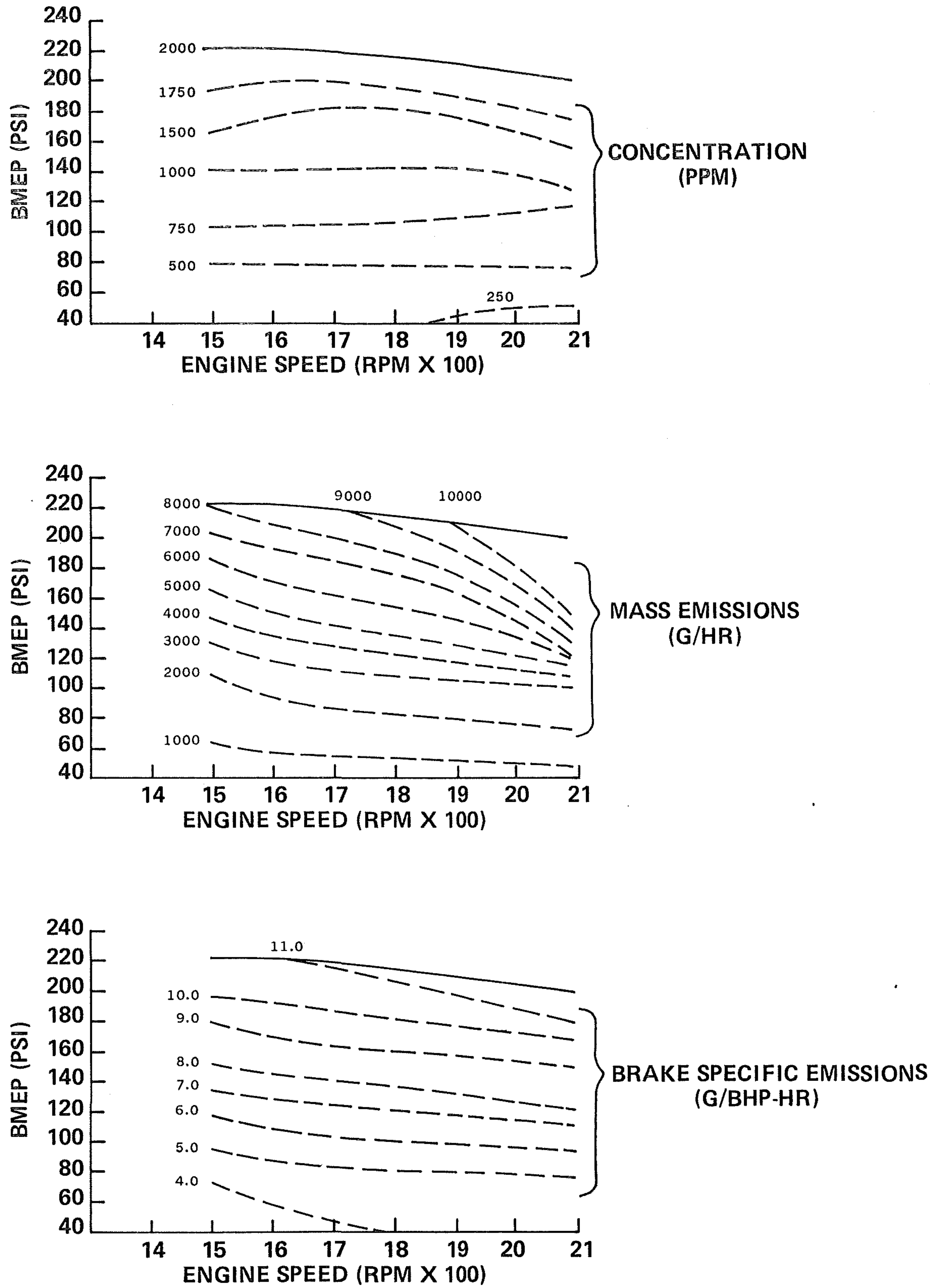


FIGURE D-3 - NO_X EMISSIONS OF CUMMINS VTA-1710 ENGINE WITH 1½% NATURAL GAS ADDED TO INTAKE AIR

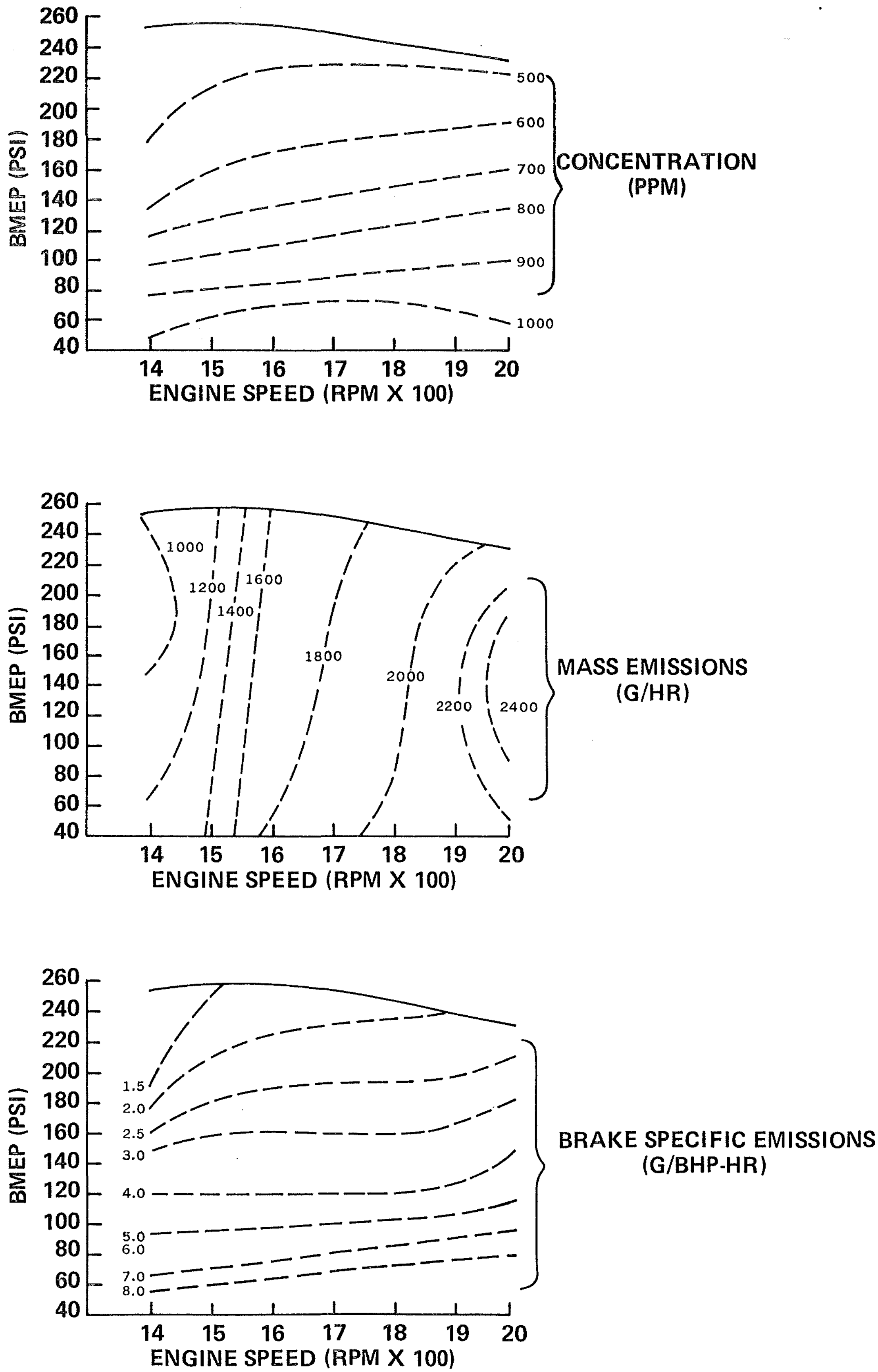


FIGURE D-4 - CO EMISSIONS OF CATERPILLAR D348 ENGINE WITH 1½% NATURAL GAS ADDED TO INTAKE AIR

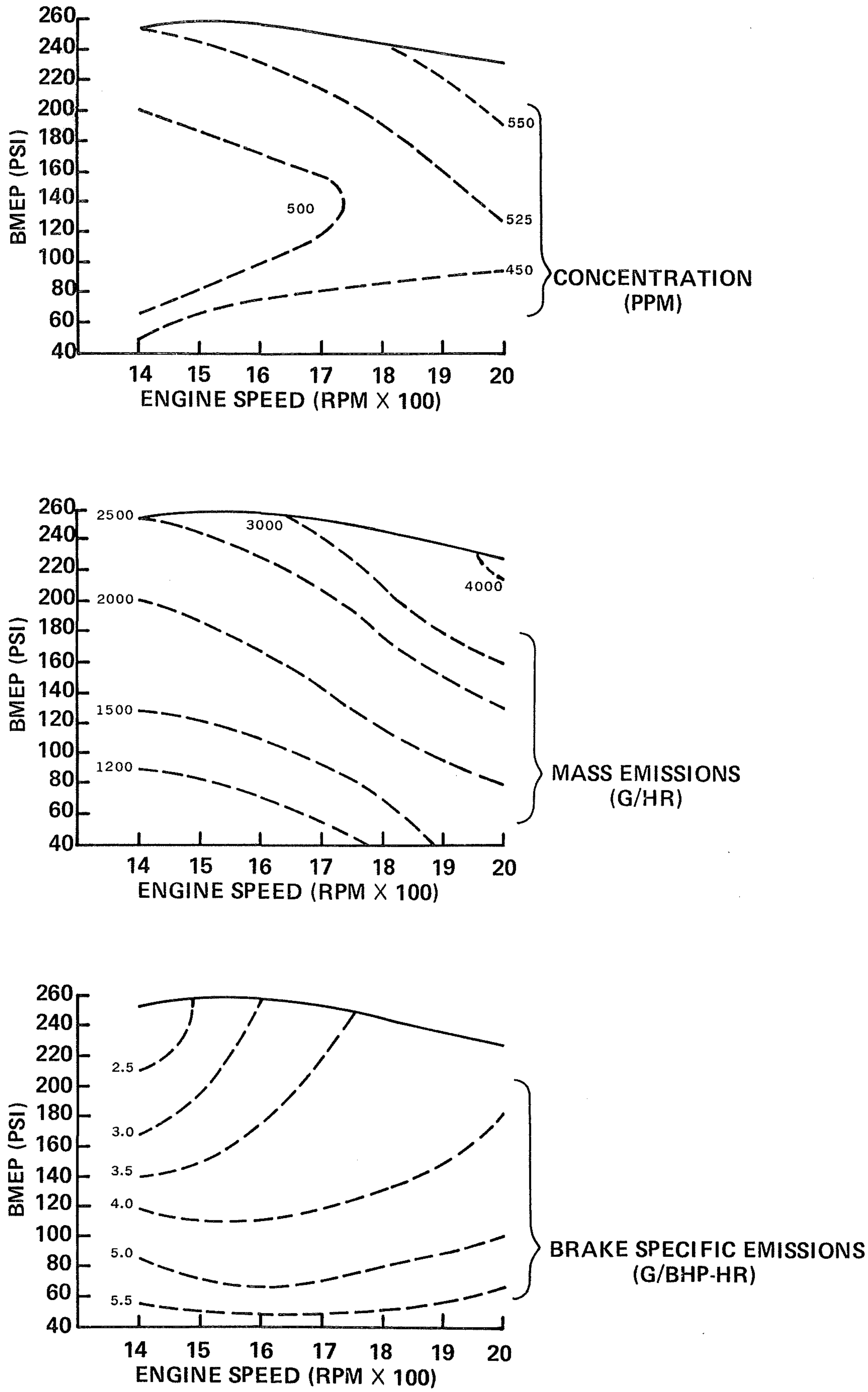


FIGURE D-5 - NO_x EMISSIONS OF CATERPILLAR D348 ENGINE WITH 1½% NATURAL GAS ADDED TO INTAKE AIR

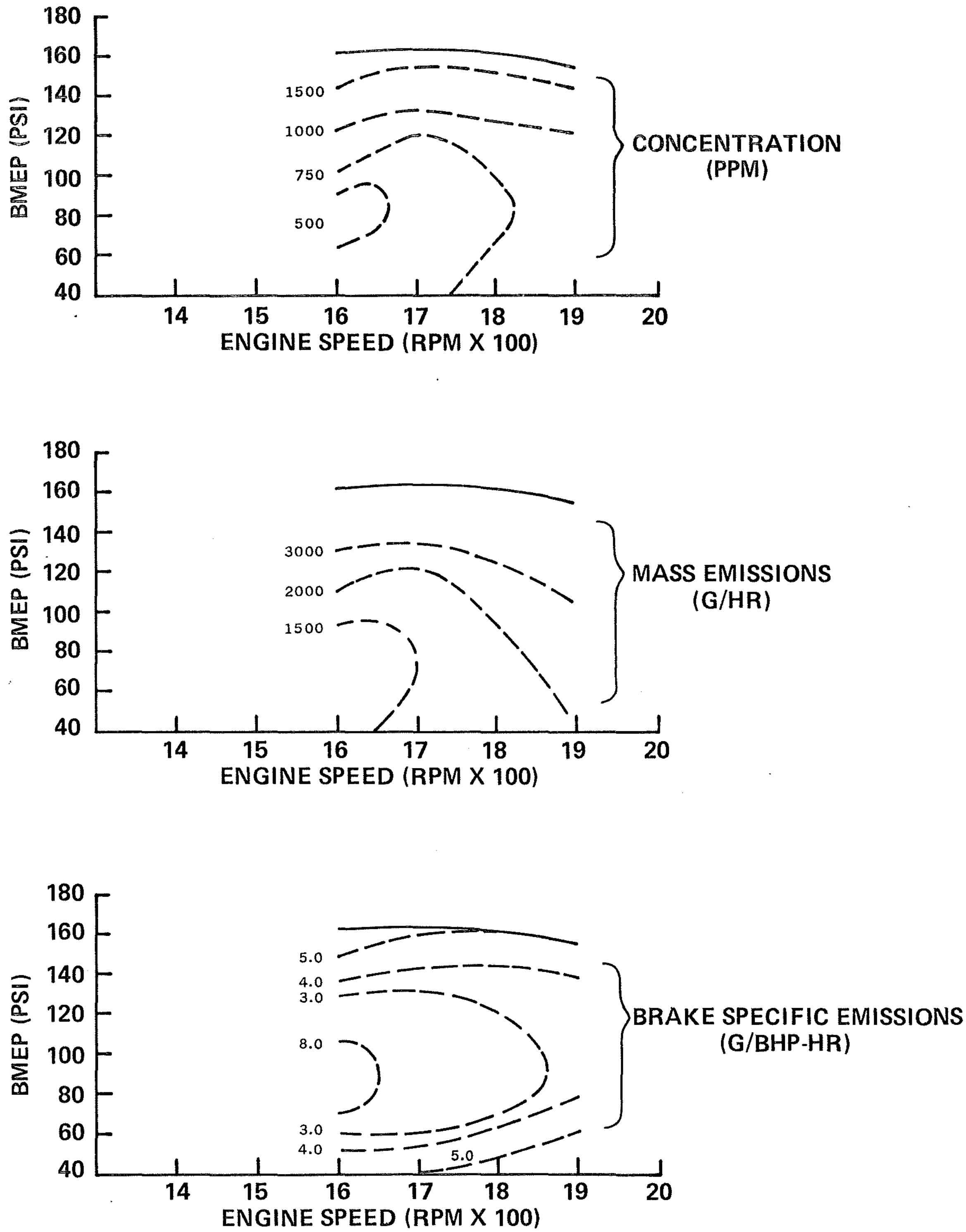


FIGURE D-6 - CO EMISSIONS OF DETROIT DIESEL 12V-149TI ENGINE WITH 1½% NATURAL GAS ADDED TO INTAKE AIR

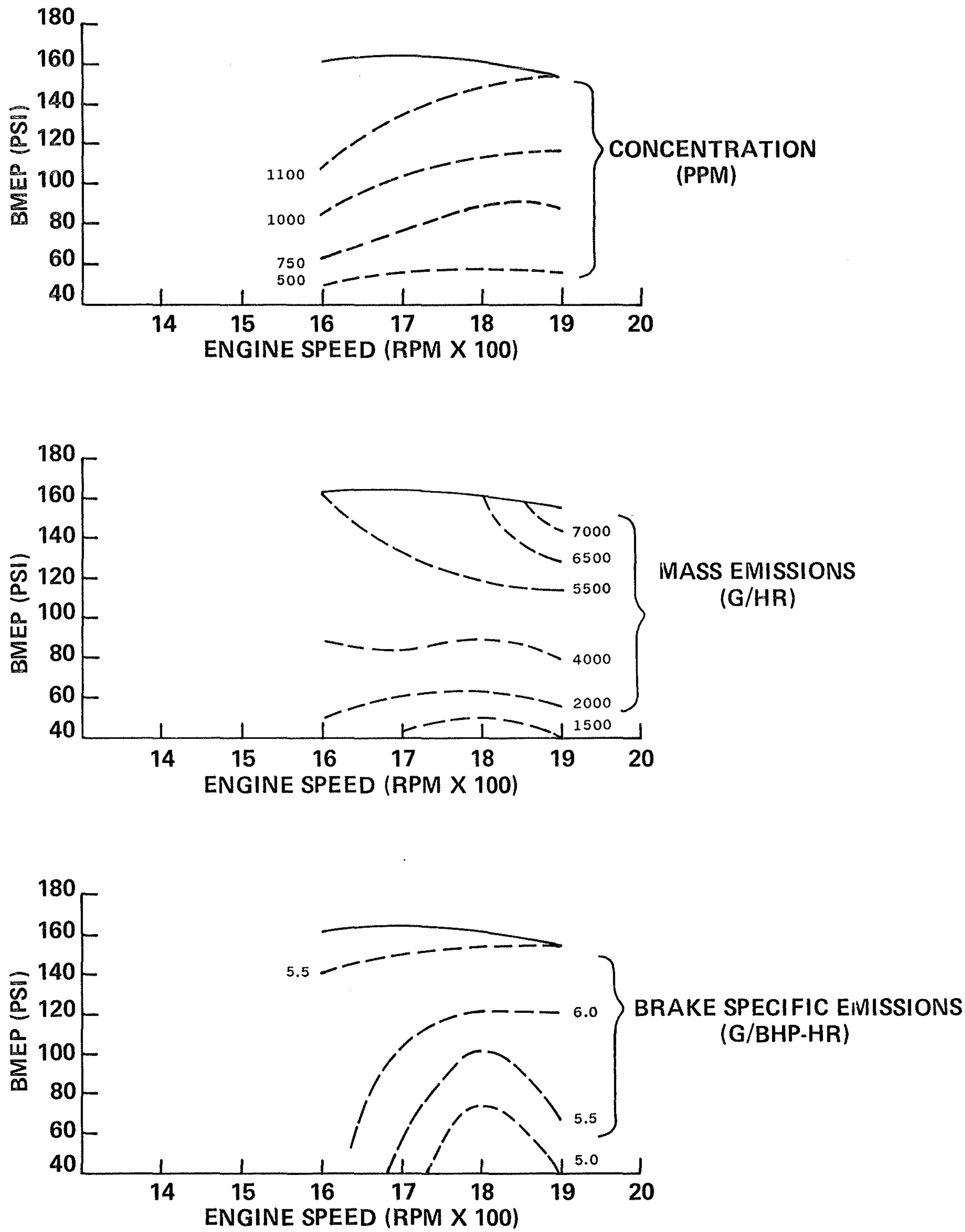


FIGURE D-7 - NO_X EMISSIONS OF DETROIT DIESEL 12V-149TI ENGINE WITH 1½% NATURAL GAS ADDED TO INTAKE AIR

APPENDIX E

Graphs of Emissions of Engines Tested with Normal Air

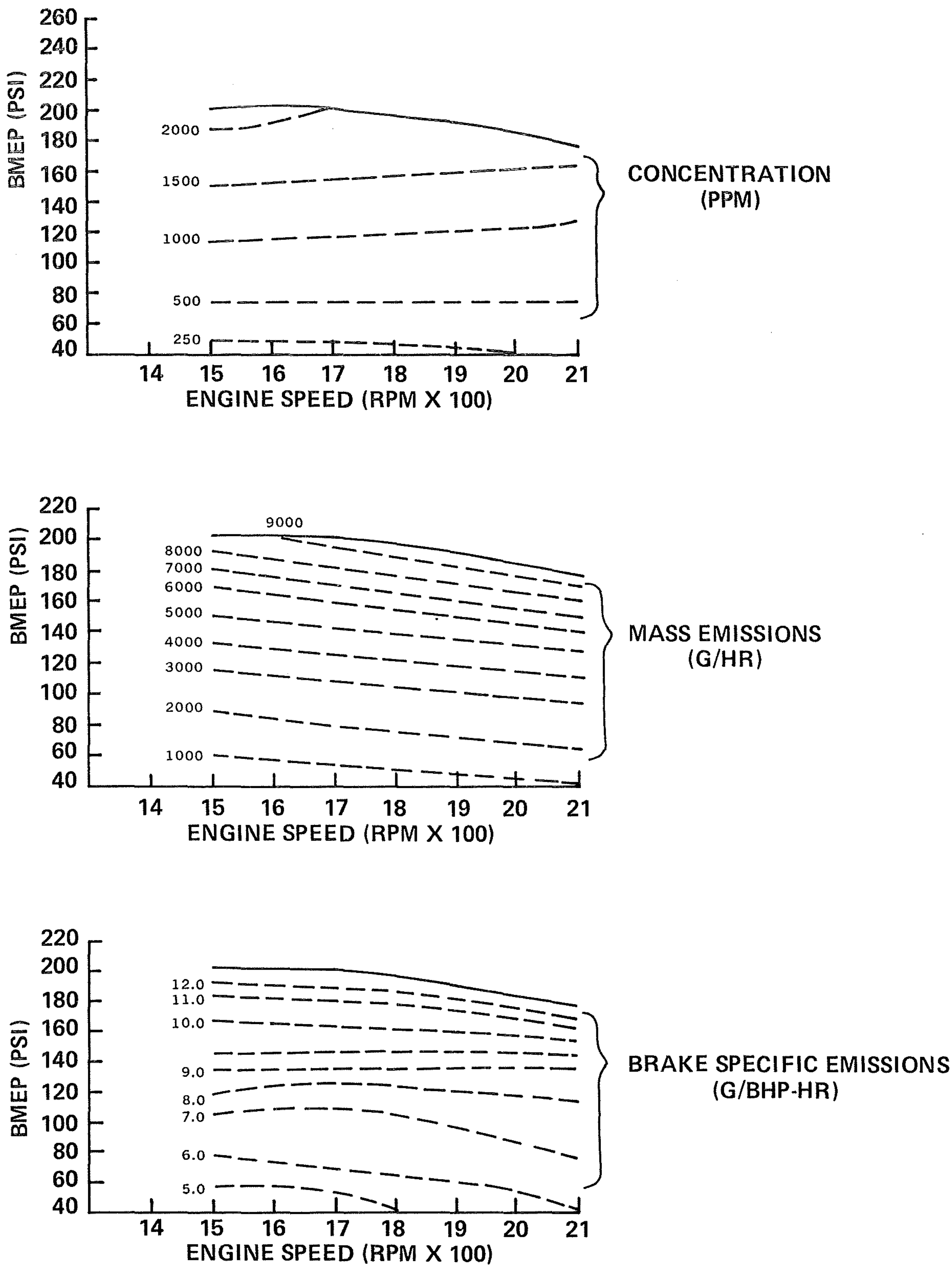


FIGURE E-1 - NO_X EMISSIONS OF CUMMINS VTA-1710 ENGINE WITH NORMAL AIR

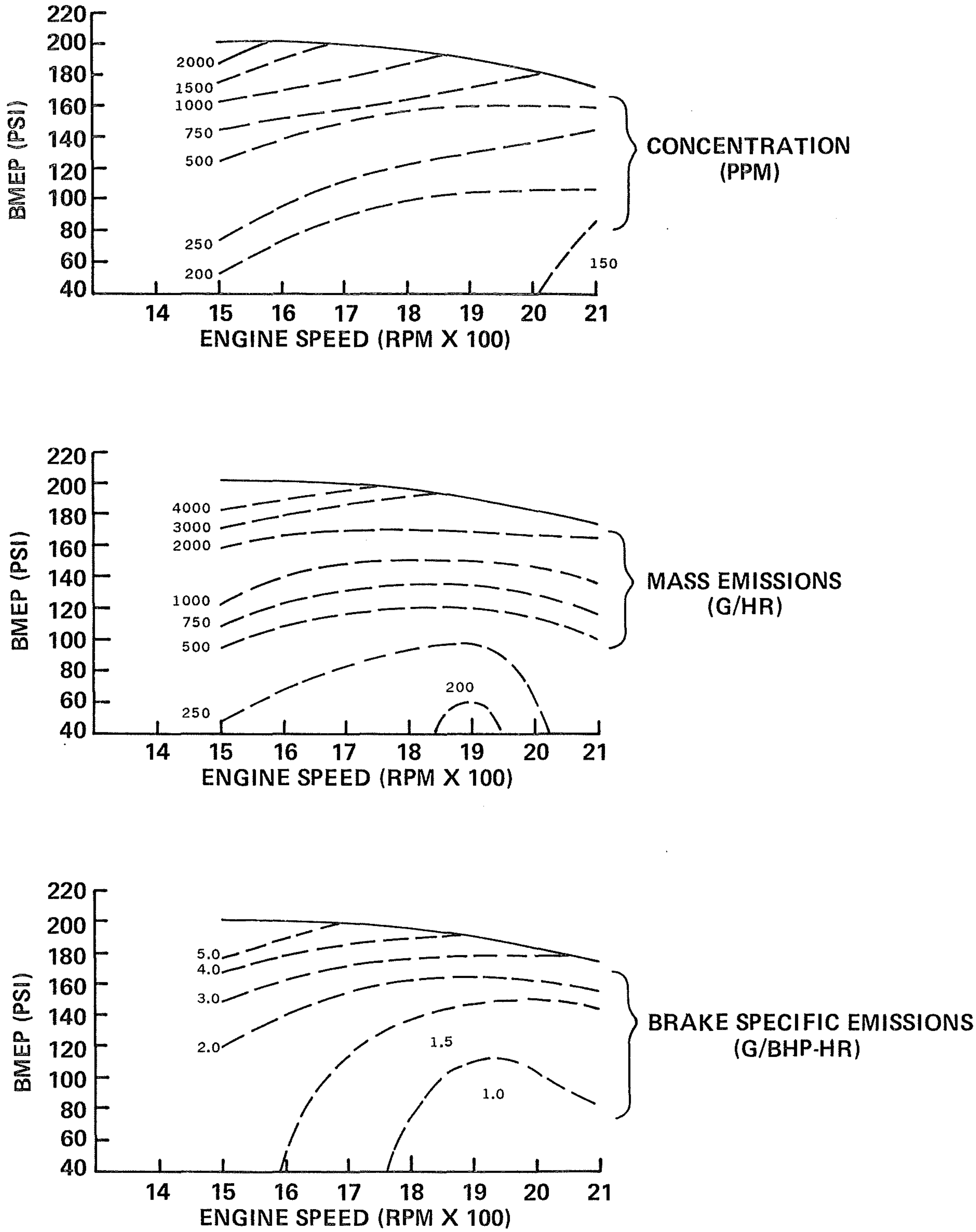


FIGURE E-2 - CO EMISSIONS OF CUMMINS VTA-1710 ENGINE WITH NORMAL AIR

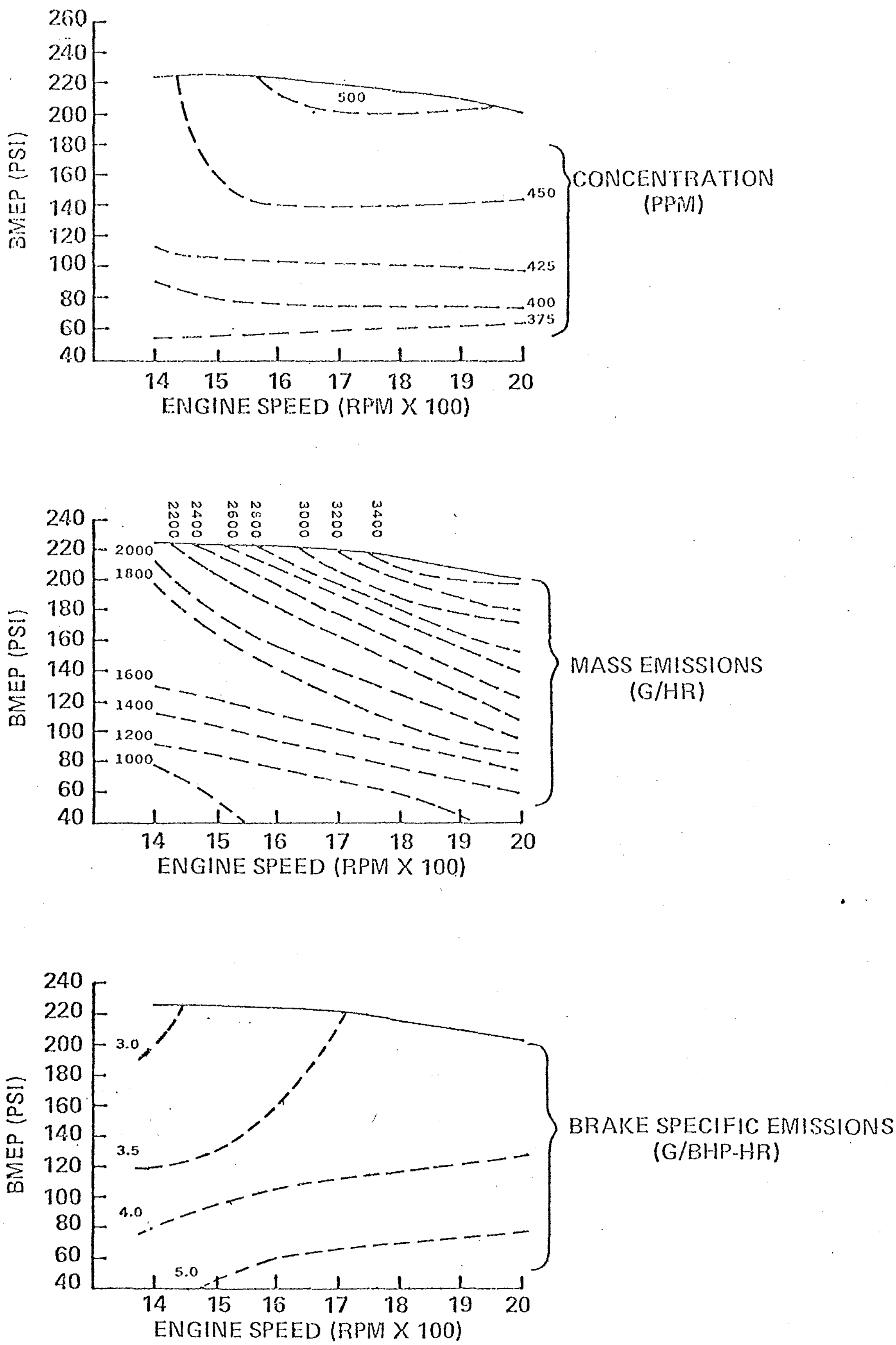


FIGURE E-3 - NO_X EMISSIONS OF CATERPILLAR D348 ENGINE WITH NORMAL AIR

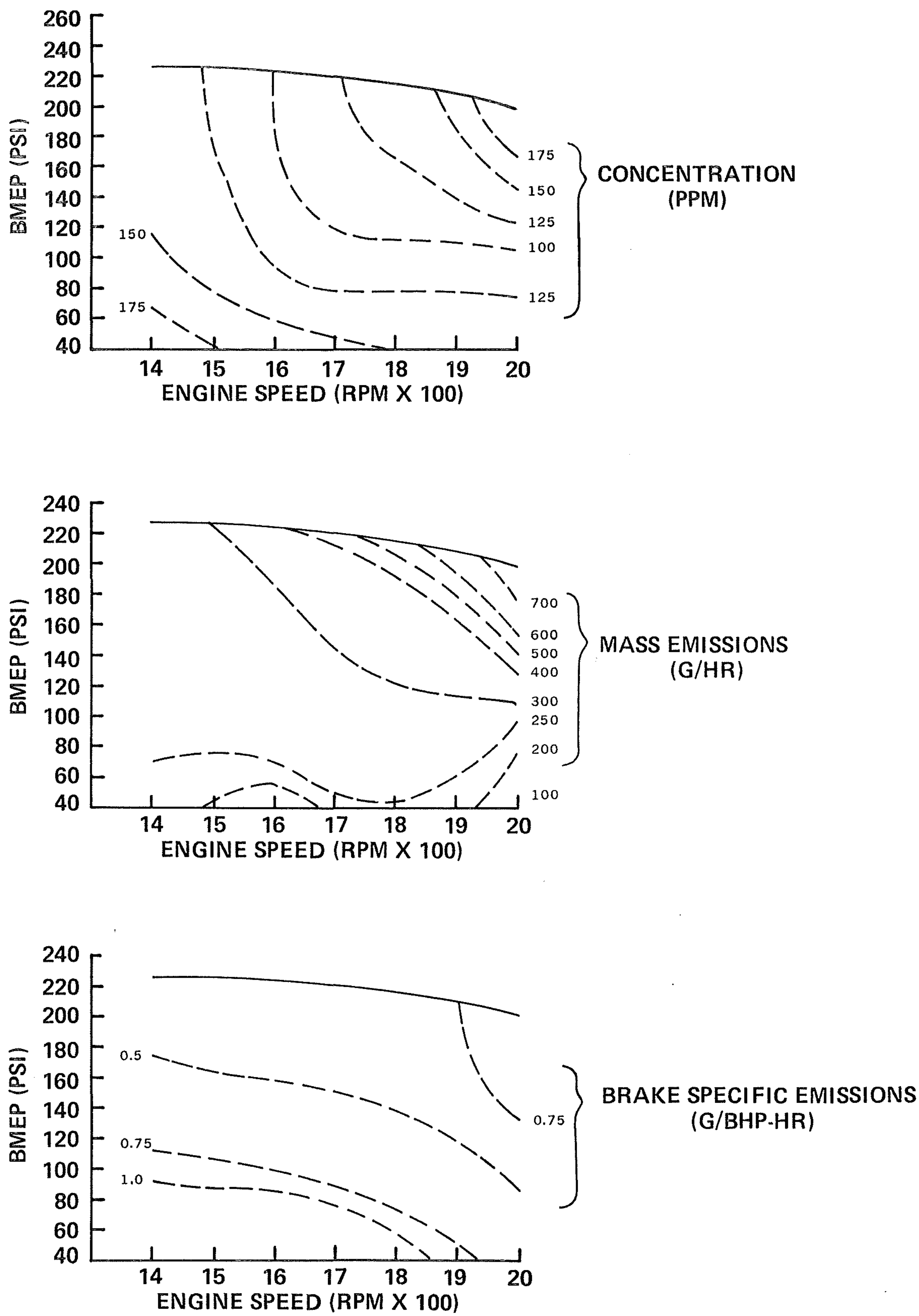


FIGURE E-4 - CO EMISSIONS OF CATERPILLAR D348 ENGINE WITH NORMAL AIR

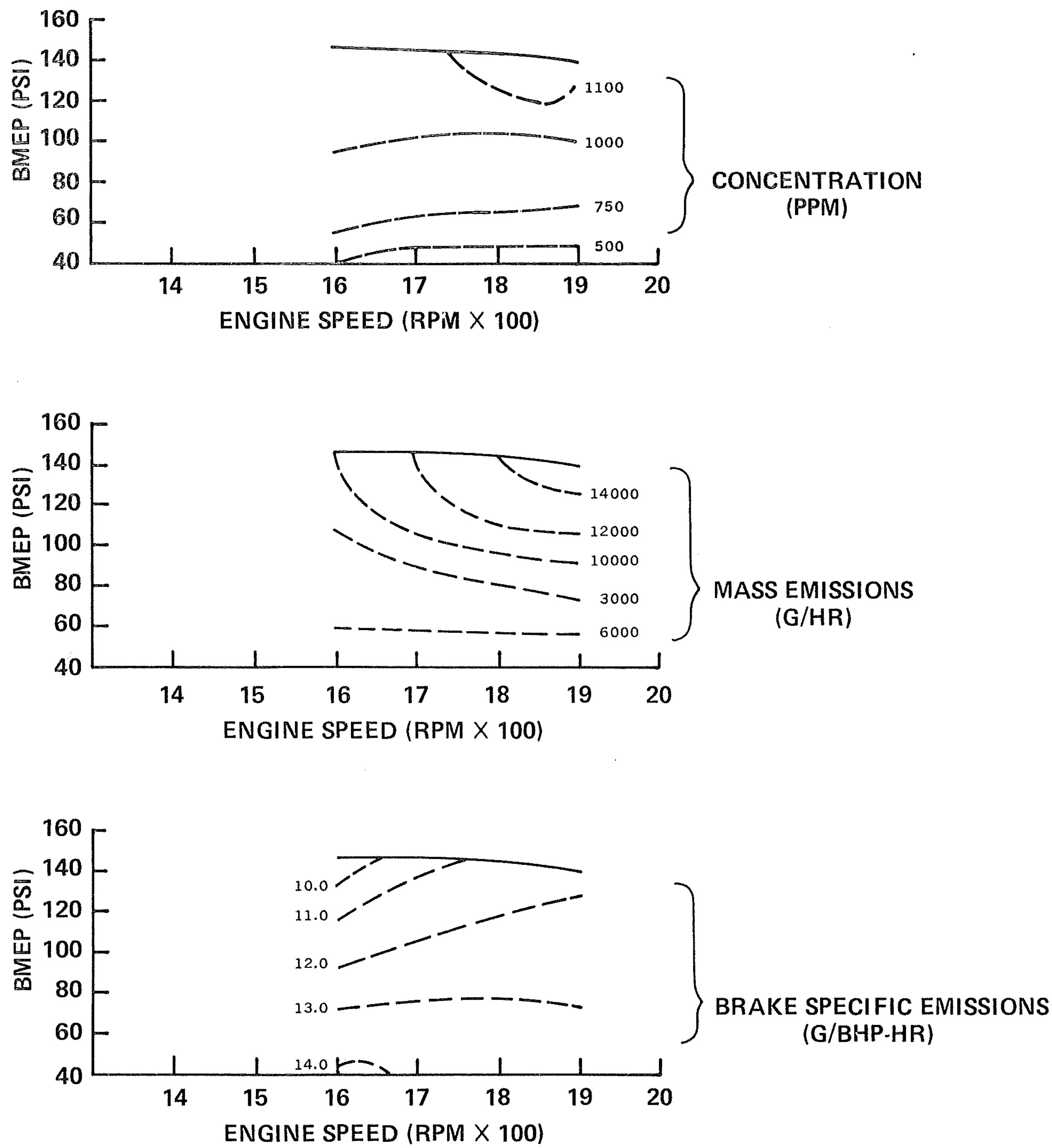


FIGURE E-5 - NO_X EMISSIONS OF DETROIT DIESEL 12V-149TI ENGINE WITH NORMAL AIR

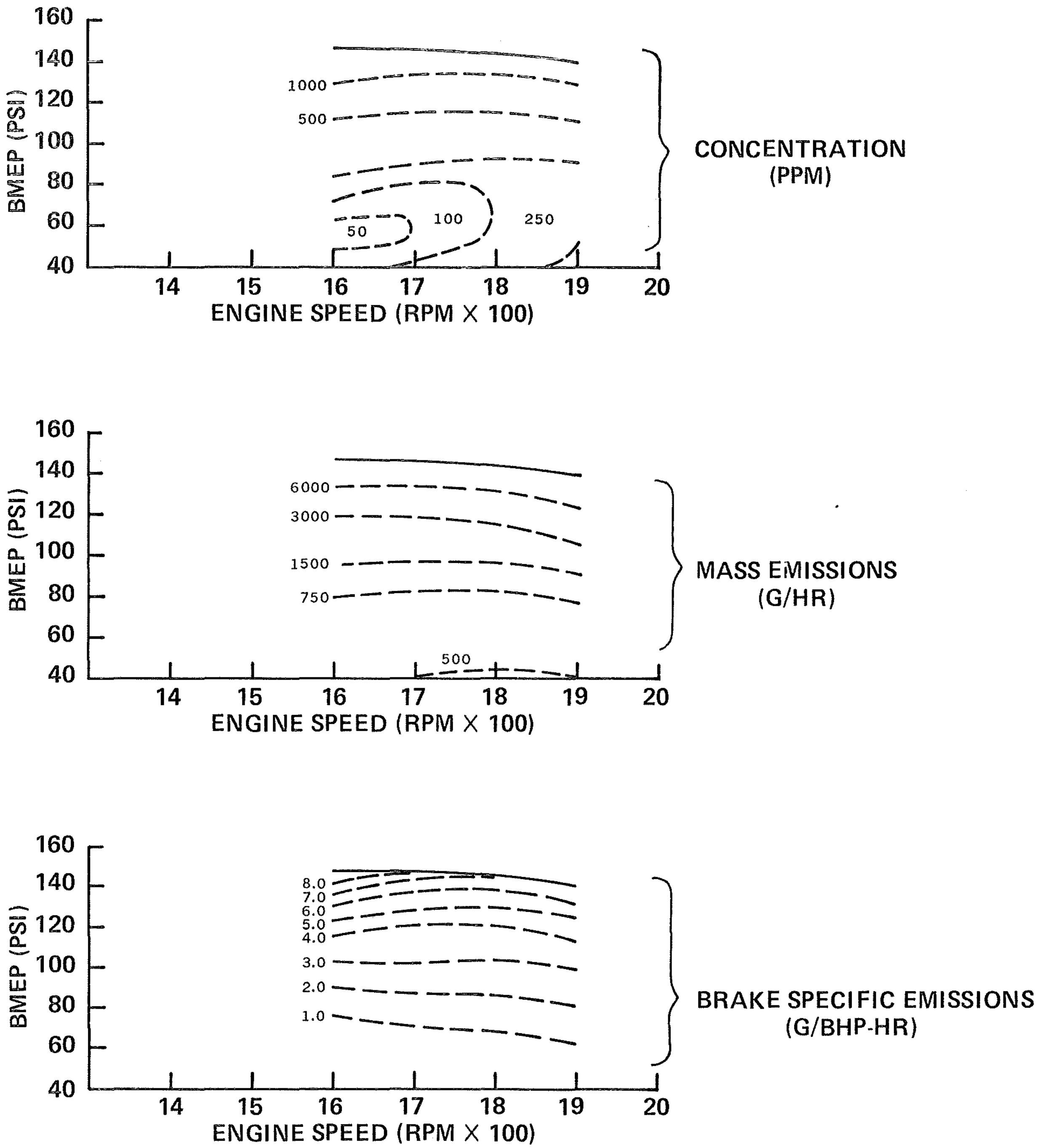


FIGURE E-6 - CO EMISSIONS OF DETROIT DIESEL 12V-149TI ENGINE WITH NORMAL AIR

APPENDIX F

Composition of San Antonio Natural Gas

APPENDIX F

Composition of San Antonio Natural Gas

COMPONENT	VOLUME PERCENT
Methane	92.55
Ethane	4.06
Carbon Dioxide	1.12
Nitrogen	.93
Iso-Butane	.12
N-Butane	.15
Iso-Pentane	.06
N-Pentane	.03
Propane	.91
Hexane	.07

Higher Heating Value - 1050 BTU/ft^3

COMPOSITION OF SAN ANTONIO NATURAL GAS