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# POTENTIAL OF DIESEL ENGINE, DIESEL ENGINE DESIGN CONCEPTS, CONTROL STRATEGY AND IMPLEMENTATION



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U.S. Department of Transportation
Research and Special Programs Administration
Transportation Systems Center
Cambridge MA 02142

MARCH 1980

FINAL REPORT



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16. Abstract

Diesel engine design concepts and control system strategies are surveyed with application to passenger cars and light trucks. The objective of the study is to indicate the fuel economy potential of the technologies investigated. The engine design parameters discussed are related to the engine configuration, combustion process, valving, friction, compression ratio, and heat transfer. Various engine control strategies and control implementation are considered.



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# PREFACE

This report, DOT-TSC-NHTSA-79-41, is one of a series of four companion reports to DOT-TSC-NHTSA-79-38 "Potential of Diesel Engine, 1979 Summary Source Document."* It assesses the fuel economy potential of design improvements to diesel engines. The authors wish to acknowledge the assistance of Giorgio Cornetti, Fiat Central Research, Torino, Italy.

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*"Potential of Diesel Engine, 1979 Summary Source Document," by T. Trella, Report No. DOT-TSC-NHTSA-79-38, March, 1980.

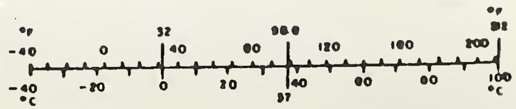
## METRIC CONVERSION FACTORS

### Approximate Conversions to Metric Measures

| Symbol | When You Know | Multiply by | To Find | Symbol |
|---|---|---|---|---|
| | | **LENGTH** | | |
| in | inches | 2.5 | centimeters | cm |
| ft | feet | 30 | centimeters | cm |
| yd | yards | 0.9 | meters | m |
| mi | miles | 1.6 | kilometers | km |
| | | **AREA** | | |
| $in^2$ | square inches | 6.5 | square centimeters | $cm^2$ |
| $ft^2$ | square feet | 0.09 | square meters | $m^2$ |
| $yd^2$ | square yards | 0.8 | square meters | $m^2$ |
| $mi^2$ | square miles | 2.6 | square kilometers | $km^2$ |
| | acres | 0.4 | hectares | ha |
| | | **MASS (weight)** | | |
| oz | ounces | 28 | grams | g |
| lb | pounds | 0.45 | kilograms | kg |
| | short tons (2000 lb) | 0.9 | tonnes | t |
| | | **VOLUME** | | |
| tsp | teaspoons | 5 | milliliters | ml |
| Tbsp | tablespoons | 15 | milliliters | ml |
| fl oz | fluid ounces | 30 | milliliters | ml |
| c | cups | 0.24 | liters | l |
| pt | pints | 0.47 | liters | l |
| qt | quarts | 0.95 | liters | l |
| gal | gallons | 3.8 | liters | l |
| $ft^3$ | cubic feet | 0.03 | cubic meters | $m^3$ |
| $yd^3$ | cubic yards | 0.76 | cubic meters | $m^3$ |
| | | **TEMPERATURE (exact)** | | |
| °F | Fahrenheit temperature | 5/9 (after subtracting 32) | Celsius temperature | °C |


### Approximate Conversions from Metric Measures

| Symbol | When You Know | Multiply by | To Find | Symbol |
|---|---|---|---|---|
| | | **LENGTH** | | |
| mm | millimeters | 0.04 | inches | in |
| cm | centimeters | 0.4 | inches | in |
| m | meters | 3.3 | feet | ft |
| m | meters | 1.1 | yards | yd |
| km | kilometers | 0.6 | miles | mi |
| | | **AREA** | | |
| $cm^2$ | square centimeters | 0.16 | square inches | $in^2$ |
| $m^2$ | square meters | 1.2 | square yards | $yd^2$ |
| $km^2$ | square kilometers | 0.4 | square miles | $mi^2$ |
| ha | hectares (10,000 $m^2$) | 2.5 | acres | |
| | | **MASS (weight)** | | |
| g | grams | 0.035 | ounces | oz |
| kg | kilograms | 2.2 | pounds | lb |
| t | tonnes (1000 kg) | 1.1 | short tons | |
| | | **VOLUME** | | |
| ml | milliliters | 0.03 | fluid ounces | fl oz |
| l | liters | 2.1 | pints | pt |
| l | liters | 1.06 | quarts | qt |
| l | liters | 0.26 | gallons | gal |
| $m^3$ | cubic meters | 35 | cubic feet | $ft^3$ |
| $m^3$ | cubic meters | 1.3 | cubic yards | $yd^3$ |
| | | **TEMPERATURE (exact)** | | |
| °C | Celsius temperature | 9/5 (then add 32) | Fahrenheit temperature | °F |



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# 1. INTRODUCTION

With increasing demand for energy conservation in transportation, diesel powered vehicles have become more and more popular because of their superior fuel economy in comparison with the spark ignition engines. Good engine performance is based upon a more comprehensive understanding of the engine's basic design concepts as well as an understanding of their control during the vehicle's operation. This document presents some basic aspects of engine design concepts and control strategies.

The traditional disadvantages of the diesel engine have always caused it to be considered unacceptable for wide use in vehicles. Drawbacks such as a lower power-to-weight ratio, higher idle noise and vibration, a higher initial cost coupled with a lower maximum speed are problems which have always seemed insurmountable. For this reason the use of diesel engines has been confined mostly to heavy duty operations.

Today's diesel engine has the following advantages:

- greater fuel economy (about 25 percent);
- comparable cost to the spark ignition engine;
- substantially lower regulated emissions (as low as 0.41/3.4/2.0 grams/mile of HC/CO/$NO_x$).

These advantages make the Light Duty diesel engine preferable to the spark ignition engine. Thus, a diesel would seem to be the prefered power plant for passenger car and light truck vehicles.

Future $NO_x$ emission levels of 1.5 grams/mile in 1981 and 1 gram per mile at a subsequent date, and the proposed particulate levels of 0.6 gram/mile for 1981 model year vehicles and the later 0.20 gram/mile particulate standard beginning with 1983 model year vehicles, have recently put the possibility of diesel engine implementation into discussion.

In this document, various injection methods, preparation of the air-fuel mixture and size are described and the influence of engine configuration and size on overall performance is discussed. The effects of several other engine design variables on performance are also examined, which include valving, compression ratio, heat loss and friction. Information on various strategies for control of emissions and fuel economy is also integrated in this report.

# 2. ENGINE DESIGN PARAMETERS

Engine design parameters in this section include the design of parts of the engine which are related to the engine cyclinder and its direct auxiliaries. Generally speaking, the engine cylinder is the heart of the engine which produces power as well as emissions. Thus, a good design is extremely critical. The combustion processes which occur in the cylinder provide large influence on engine performance and is directly influenced in many cylinder design parameters. Some of these cylinder design parameters include valving operation, compression ratio, degree of heat transfer through the wall, and frictional power loss. This section provides some introductory remarks on these parameters.

## 2.1 PARAMETERS OR OPERATIONS RELATED TO COMBUSTION PROCESSES

### 2.1.1 Injection Operation

2.1.1.1 - High Speed Direct Injection - Direct injection diesels are currently used almost exclusively in all types of automotive applications apart from the light duty automobile and truck. In general, direct injection offers better starting, lower heat losses, lower thermal loadings and better fuel economy with an advantage of 8-10 percent compared to indirect injection.[1] Table 2-1 shows a performance comparison of Mercedes Benz - 220D, with both indirect and direct fuel injection systems. The comparison shows that the performance of the direct injection engine is much better than the comparable indirect injection engine.

The schematic diagram of a typical unit injection system is shown in Figure 2-1. It basically contains a plunger and nozzle combination and the camshaft driving mechanism together with the fuel supply and connecting pipings. The unit injection system results in maximum power saving for delivering fuel into the cylinder. Furthermore, because of the simplicity of the mechanism, it can control more precisely the injection quantity and the injection

TABLE 2-1. COMPARISON OF IDI PRODUCTION ENGINE WITH DI SUBSTITUTE

| Vehicle | Injection Type | Inertia Weight (lbs) | CID ($in^3$) | HC (gms/mile) | CO (gms/mile) | $NO_x$ (gms/mile) | Particulates (gms/mile) | Composite FTP-Cycle (mpg) | Percent Difference DI/IDI |
|---|---|---|---|---|---|---|---|---|---|
| Mercedes Benz* 220 D | IDI | 3500 | 134 | 0.27 | 1.25 | 1.40 | 0.46 | 27.6 | +20% |
| | DI | 3500 | 134 | 0.22 | 1.0 | 1.40 | 0.34 | 33.2 | |

*Acceleration Performance 0 to 50 mph, 18 sec.

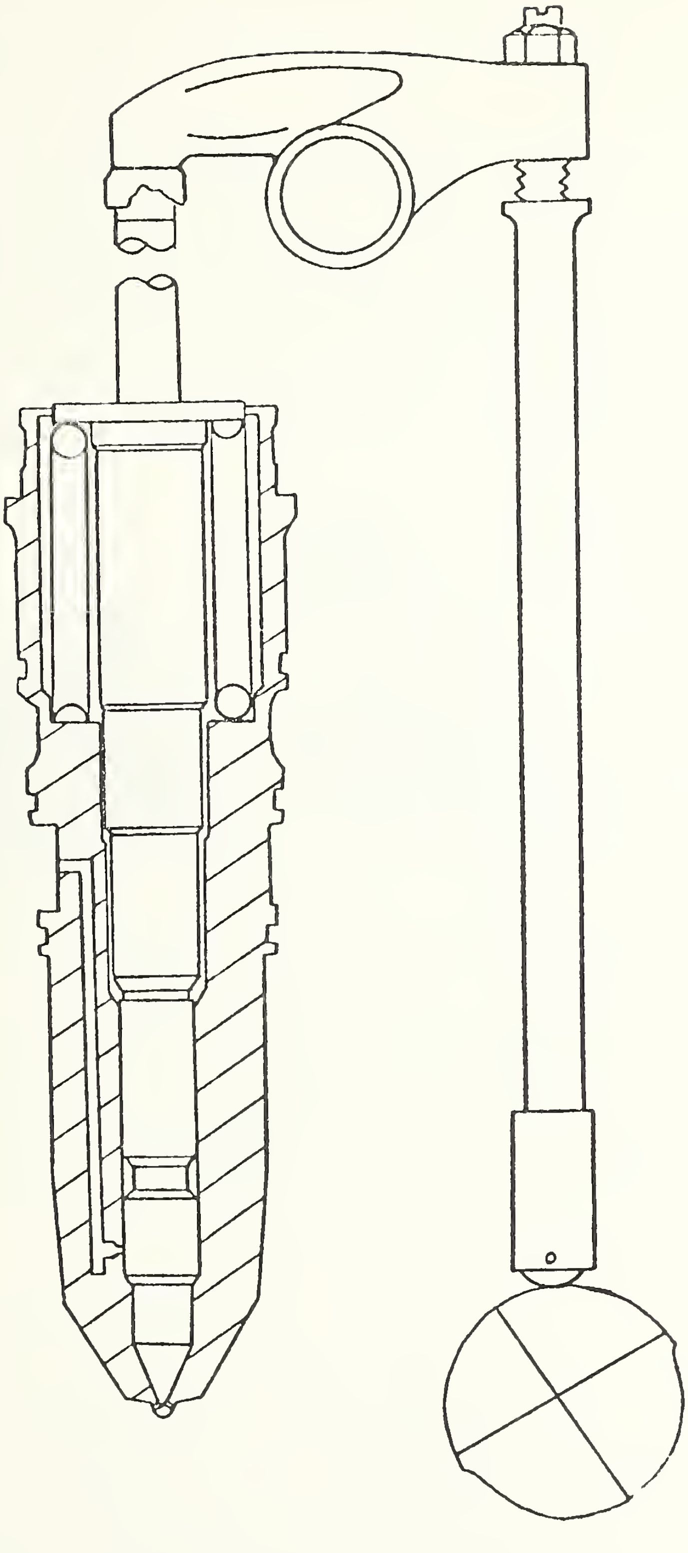


FIGURE 2-1. SCHEMATIC OF INJECTION SYSTEM

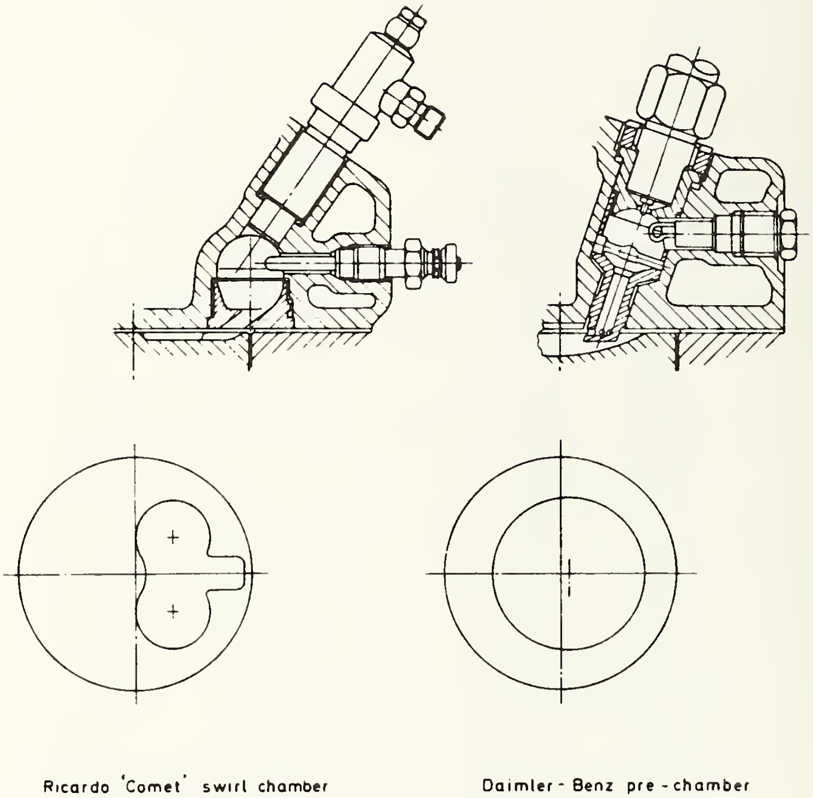



FIGURE 2-2. SCHEMATIC DIAGRAMS OF RICARDO 'COMET' SWIRL CHAMBER AND DAIMLER-BENZ PRE-CHAMBER

timing; less mechanism also means less injection equipment noise. By increasing the injection pressure, high atomization conditions can be obtained, which result in quick mixing suitable for high speed diesel operation.

AVL (Austalt für Verbrennungsmotoren, Prof. Dr. h.c. Hans List)[2] installed a direct injection system in a converted 2.2 liter Daimler Benz diesel engine. See Table 2-1. It obtained 20 percent better fuel economy than current light duty diesel engines at equal output and equal gaseous emissions, almost instant unaided cold starting, low particulate emissions, practically no visible emissions, and less exhaust odor and irritancy than current I.D.I. diesel engines.

Currently, the D.I. system is not yet fully developed for the high speed diesel engines which are needed for light vehicles.

2.1.1.2 Indirect Injection - This is an earlier version of a fuel injection system for the diesel engine. It contains a prechamber (or swirl chamber) in addition to the main combustion chamber. Fuel is first injected into the prechamber to allow more time for evaporation and mixing. The swirl, introduced by specially designed chamber geometry, enhances the mixing mechanism. This version of fuel supply is still most commonly used in all the light duty diesel vehicles. Figure 2-2 shows a schematic diagram of two typical indirect injection systems - Ricardo's "Comet" and Daimler-Benz. The former is an example of the swirl chamber type and the latter is the prechamber one. IDI requires an injection pump with lower pressure.

### 2.1.2 Turbulent Evaporation

This is a process that precedes combustion in the cylinder. In diesel, the requirement is different from that in spark ignition engines. In the diesel case, an appropriate amount of fuel must be evaporated and mixed before ignition occurs. As to the gasoline engine it appears that the closer to the completion of evaporation and mixing, the better would be the performance.

In open chamber diesel engines, the main chamber swirl is mainly used to promote the fuel evaporation and fuel-air mixing. For the divided chamber diesel, the initial preparation of the mixture is mainly done in the pre- or swirling-chamber.

Swirl and turbulence are frequently induced from the specially designed cylinder configuration. Specifically, for induced primary swirl, i.e., the swirl before combustion process, any of the following methods[3] can be used:

(i) Orient the angle of the inlet ports such that they deviate from the radial direction.

(ii) Use a masked valve. Here, a part of the flow area is blocked by a circular arc on the inlet valve.

(iii) Use a masked port. By masking a part of the flow area with a shelf or projection in the passageway, the air flow is diverted away from the shelf side of the port.

(iv) Use a directed inlet port. Here, the passageway is laid out to direct the inflowing air in the desired tangential direction.

(v) Use a vortex port. Here, the incoming air is made to rotate around the valve stem before entering the cylinder. The induced swirl motion, together with the air motion induced because of the internal configuration of the piston and cylinder, such as squish, toroidal, or jet (from prechamber), will generate strong turbulence in the cylinder.

### 2.1.3 Hot Wall Evaporation - MAN System

Hot wall evaporation is one kind of mechanism which dispenses and evaporates the fuel mainly through heat and no other mechanism during the pre-combustion period. It was first designed and tested by Maschinenfabrik Augsburg Nürnberg company; it is called MAN system or sometimes, M-system. Figure 2-3 shows a schematic diagram of the M-engine. Basically, it contains an injection system with injection pressure at about 2500 psia and a piston with bowl shaped top. During the injection period, most of the

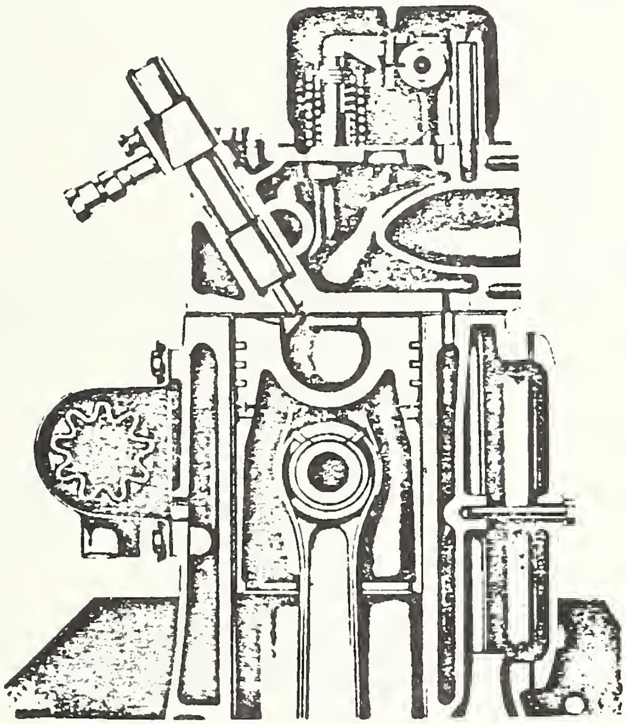


FIGURE 2-3. CROSS-SECTION OF 6-CYL. 4.4 x 5.5-IN PRODUCTION ENGINE, 200 HP. MAX. OUTPUT (supercharged) AT 2300 RPM

injected fuel (almost 95 percent according to Meurer[4]) will impinge on the hot combustion chamber wall to form a thin fuel film. Meanwhile, because of the thermal effect of the wall, fuel films are evaporated, decomposed and mixed with high-speed air swirl. With the appropriate time table, a reduced exhaust smoke level and low specific fuel consumption can be obtained over a wide speed range. The MAN group has conducted extensive test programs. The results are amazing. In particular, the typical diesel type auto-ignition knocking has disappeared over the entire speed range, even during idling and starting of the cold engine; this has been obtained without sacrificing specific output or fuel economy. Figure 2-4 shows some BSFC results of a 6-cylinder naturally aspirated M-engine. The curves of iso-BSFC extend over wide speed and map ranges. The turbocharged version of the same engine has a peak output of 200 hp and its BSFC is less than 0.37 lb per BHP-hr over wide speed and map ranges. Exhaust smoke level is influenced by the injection direction. Typical results are shown in Figure 2-5. It can be seen that as the injection is directed towards the wall, the smoke level remains low for the whole speed range. This behavior is related to the minimization of the initial auto-ignition fuel quantity.

Finally, this engine concept is used in heavy trucks and has not yet received attention fof automobile and light truck applications.

## 2.2 ENGINE CONFIGURATION RELATED PARAMETERS

### 2.2.1 Rotary Engine Configuration

Rotary diesel engines currently have the following development problems:

1) the difficulty of obtaining a high enough compression ratio,
2) a high surface/volume ratio at TDC,
3) a shallow and elongated combustion space,
4) a special gas sealing system, which contains single element with line contact.

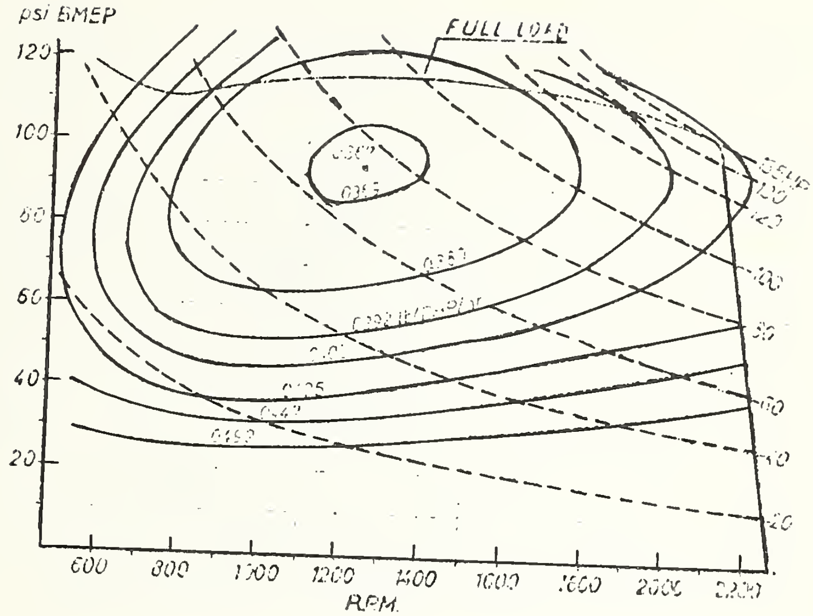



FIGURE 2-4. SFC CHARACTERISTICS IN RELATION TO ENGINE SPEED AND MEN AND FULL-LOAD POWER CURVE FOR 6-CYL., D 1246 M ENGINE WITH NATURAL ASPIRATION

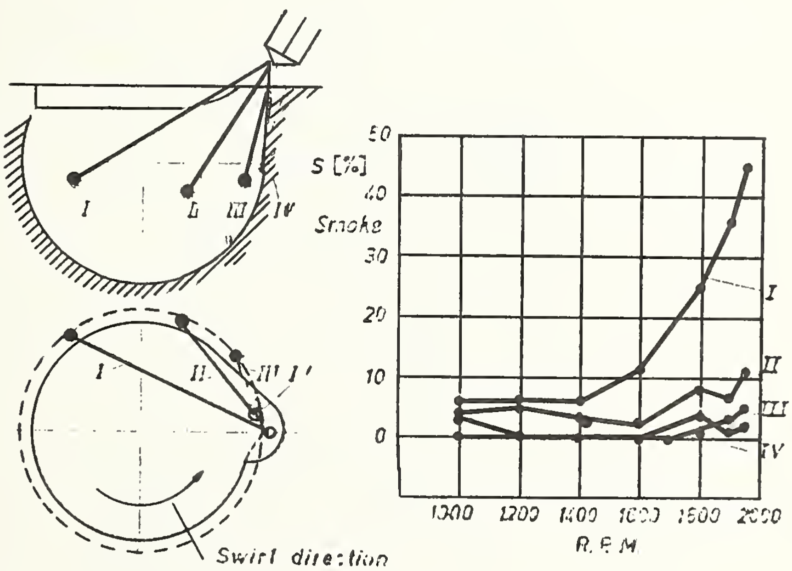



FIGURE 2-5. INFLUENCE OF DIRECTION OF FUEL SPRAY UPON EXHAUST SMOKE. INJECTION WELL BELOW PLANE OF CHAMBER FORMAT

To overcome these difficulties, one of the methods is to use a two stage rotary engine. To demonstrate its practical feasibility, Research Rotary Engines, Rolls-Royce, Ltd., has conducted research programs since 1964. The major achievement of this development program was the development of an engine with tremendous compactness. Figure 2-6 shows a schematic diagram of the typical two-stage rotary engine. The basic principle is to compress and expunge the gas medium in two successive rotating cylinders connected by passages. Gas intake and exhaust occur at low pressure cylinder and burn occurs in high pressure cylinder. Different fuel injector positions have been tested. Optimized injector position is found as shown in Figure 2-7. The combustion chamber is constructed in such a way that air swirl can be induced. A set of stepped Apex seals, Figure 2-8, having the advantage of reduced mass, was developed and the desired operation was achieved. Figure 2-9 shows some typical results. No emission data are available.

### 2.2.2 'Squish Lip' Piston Design

It is well known that the interior cylinder configuration design influences the engine performance as well as emissions a great deal. Here is a specific example, developed by the Perkins Engine Group Ltd in England.[6] In this program, a series of direct injected diesel engines with bowled pistons were tested. The purpose was to find a low emission combustion system without sacrificing fuel economy. The bowl configuration was systematically varied for different tests. The major test parameters were throat diameter, bowl volume, flank angle, lip shape, and central pip. Totally, fourteen different piston bowl configurations were used as shown in Figure 2-10. Figure 2-11 shows some typical performance and emission data of an optimized piston configuration with a comparison to the results of a standard open chamber engine. It is clear that both smoke level and BSFC are improved, yet the $NO_x$ level gets worse because of the increased combustion rate and peak temperature. For the various specific test series, smaller throat diameter gives lower smoke level and higher $NO_x$ level due to the improved combustion rate. Yet, the BSFC was optimized at moderate

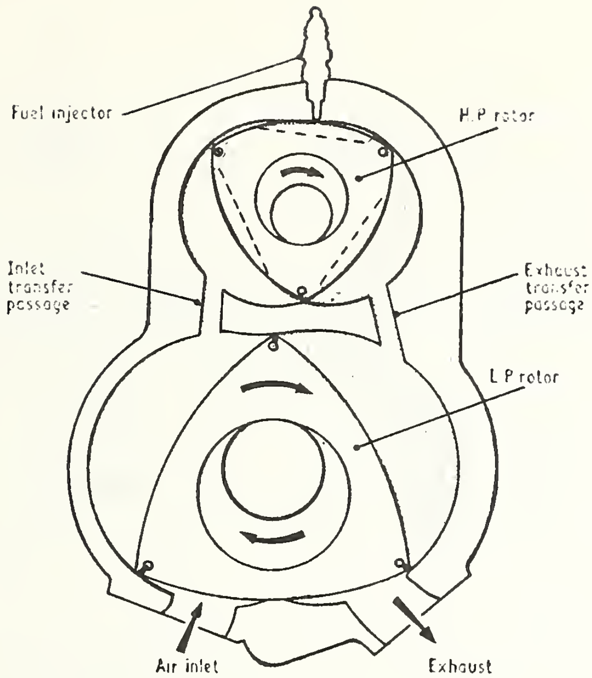



FIGURE 2-6. 2-STAGE DESIGN WITH TWO ROTORS

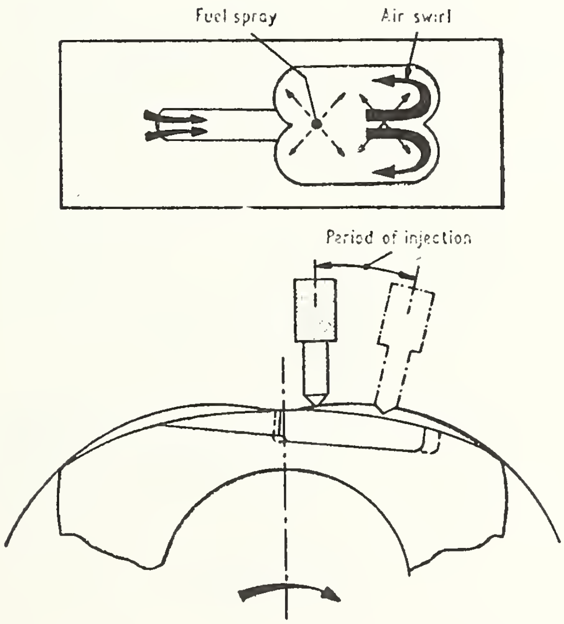



FIGURE 2-7. PREFERRED COMBUSTION SYSTEM

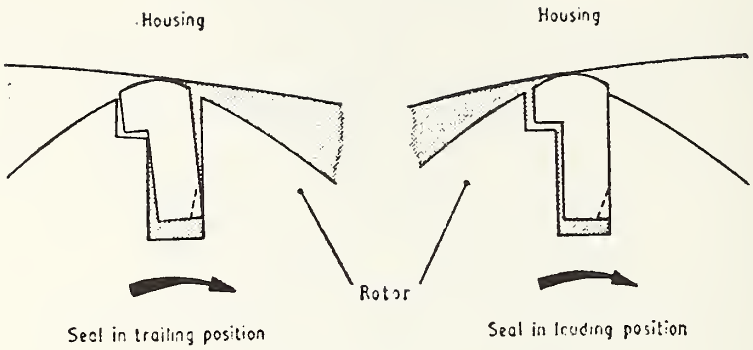



FIGURE 2-8. STEPPED APEX SEAL

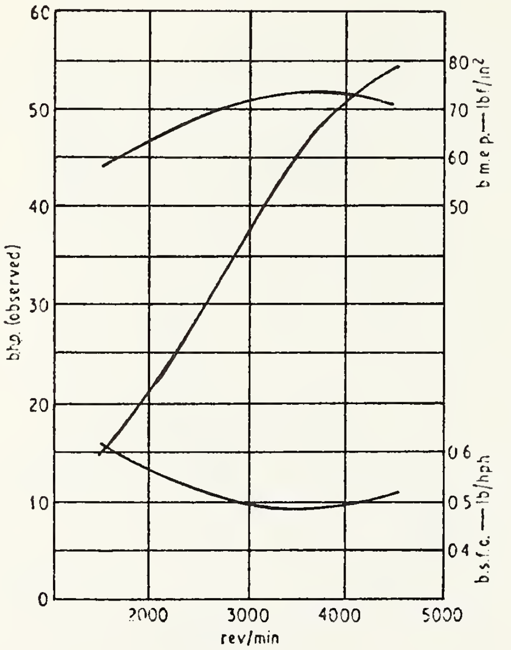



FIGURE 2-9. PERFORMANCE OF A ROTARY ENGINE WITH AN AIR/FUEL RATIO OF 30:1

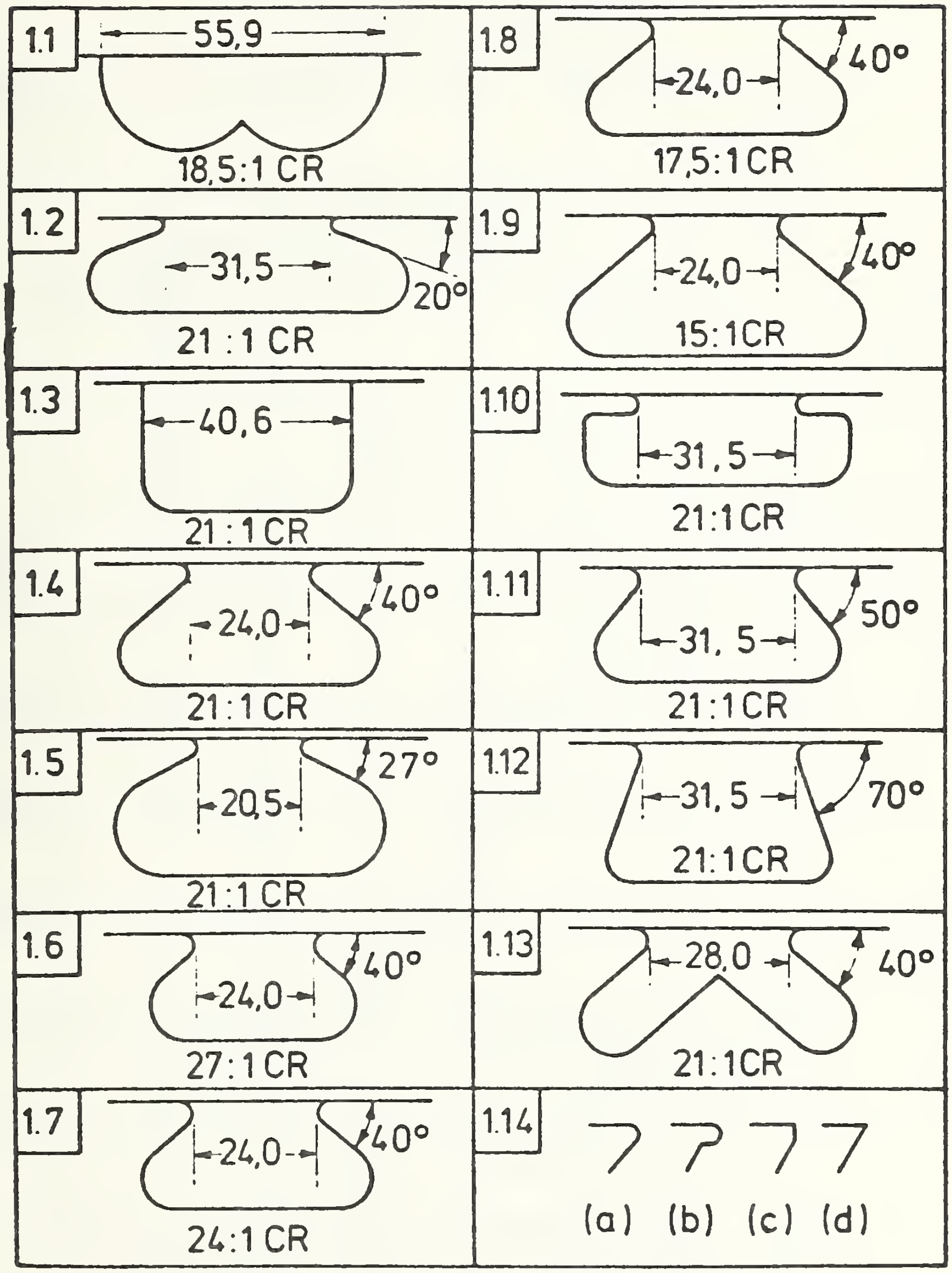



FIGURE 2-10. STANDARD COMBUSTION CHAMBER AND VARIOUS RE-ENTRANT CONFIGURATIONS

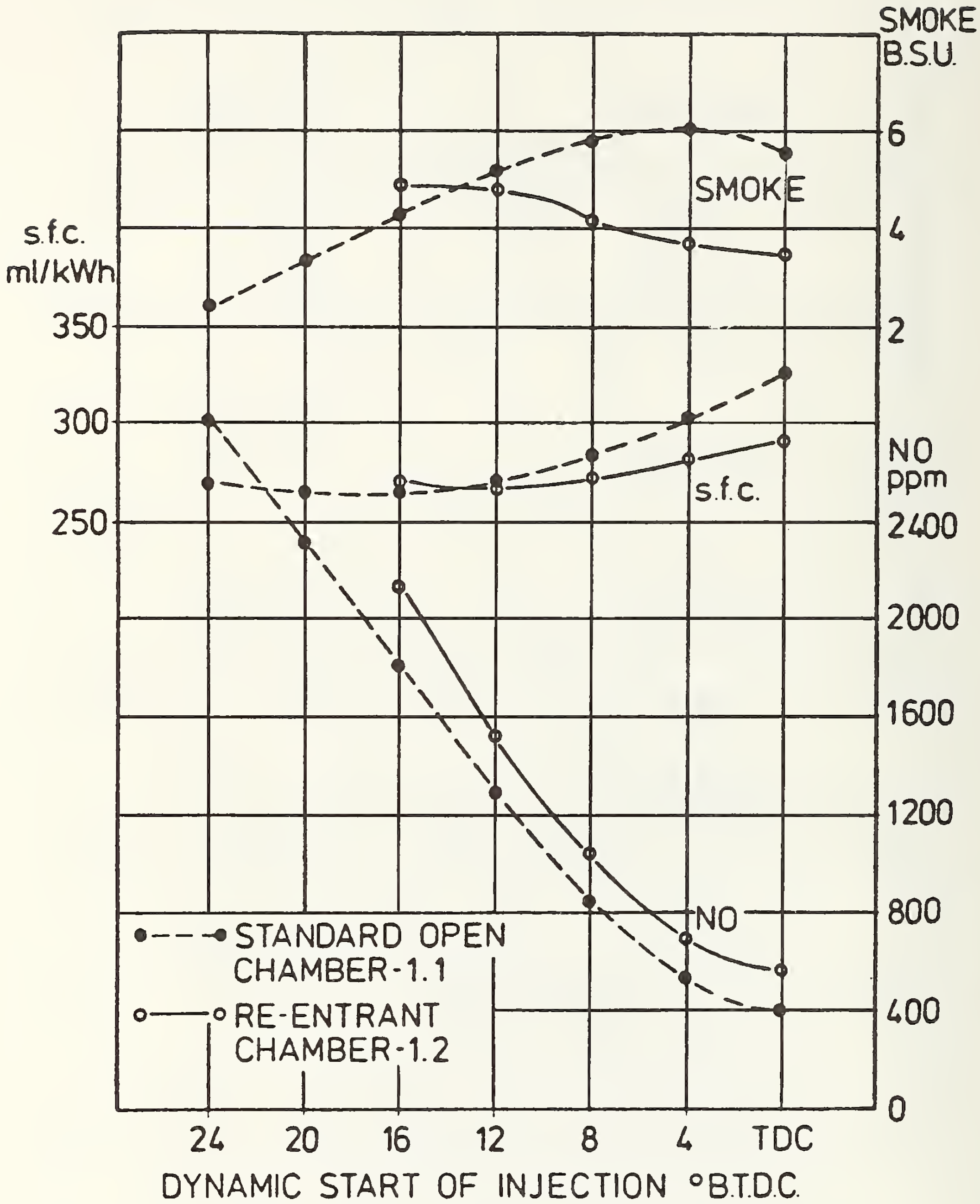



FIGURE 2-11. EFFECT OF INJECTION TIMING, STANDARD OPEN CHAMBER, AND RE-ENTRANT CHAMBER, ENGINE SPEED 1400 REV/MIN, EQUIVALENCE RATIO

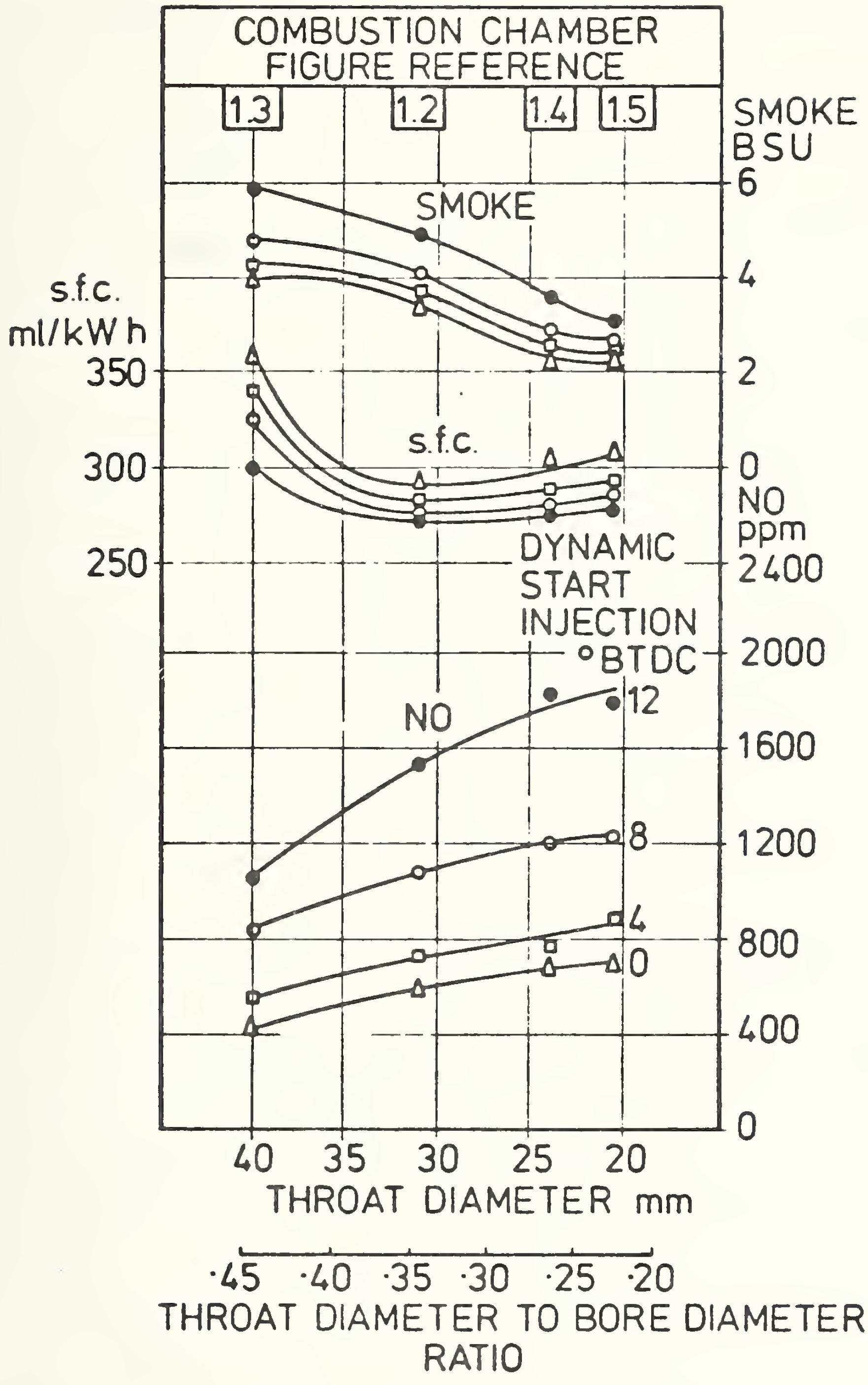



FIGURE 2-12. EFFECT OF THROAT DIAMETER ON SMOKE, SFC, AND NO EMISSIONS, ENGINE SPEED 1400 REV/MIN, EQUIVALENCE RATIO

throat diameter as shown in Figure 2-12. It is possible that with a very small throat diameter, turbulence velocity is too large so as to cause large friction loss and heat loss, which results in high BSFC. The various bowl volume tests indicate that as bowl volume decreases, both ignition delay time and turbulence levels were reduced, which results in higher smoke level, lower BSFC, and lower $NO_x$ level. Flank angle tests indicate that the optimized angle range is around 20 to 40 degrees. Bowls with central pip yield slightly better BSFC's than those without.

### 2.2.3 Engine Sizing

Diesel engines are built with piston diameters of 2 to 37 inches, and with speeds ranging from 100 to 4,400 rpm while delivering from 1.5 to 33,400 bhp on one crankshaft[3]. With such a wide range in engine size, a basic understanding of the effect of cylinder size on engine performance is important. In a group of cylinders of similar design and the same materials of construction, Taylor[5] summarized the effects of differing cylinder size as follows:

1) Stresses due to gas pressure and inertia of the cylinder assembly are the same at the same crank angle, provided (a) mean piston speed is the same, (b) indicator diagrams are the same and (c) there is not serious vibration on the engine structure.

2) When inlet and exhaust conditions and fuel-air ratio are the same, similar cylinders will have the same indicator diagrams at the same piston speed and the same friction mean effective pressure. Under these conditions brake power is proportional to bore squared or to piston area.

3) Since the weight of a cylinder is proportional to the bore cubed or to the total piston displacement, when the mean pressure and piston speed are the same, the weight per horsepower increases directly with the bore.

4) The temperature of the parts exposed to hot gases will increase as cylinder size increases.

5) In diesel engines, as the cylinder bore increases speed of revolution is reduced; it becomes easier to control maximum cylinder pressures and maximum rates of pressure rise. Consequently, fuels of lower ignition quality can be used.

6) As the cylinder bore increases, wear damage in a given period of time decreases; that is, the engine lasts longer between overhauls or parts replacement.

7) With the same fuel, fuel-air ratio, and compression ratio, efficiency tends to increase with increasing cylinder size because of reduced direct heat loss.

In general, engine size can correlate with many parameters. In practice, bmep, piston linear speed, specific output (bhp/in$^2$ of piston area) and ignition delay time appear to tend to fall slowly as cylinder bore increases. Relative to engine performance, (except for very small engines) as the bore increases, the engine indicated thermal efficiency increases slightly, as does the specific fuel consumption. Tests on cylinders of less than 2-in bore usually show very poor brake thermal efficiency because of the relatively large heat loss and friction loss.

## 2.3 VALVING

Valves control the inlet and exhaust of the engine. Valve geometry, timing, duration and lift will all influence engine performance. Basically, the valve geometry, together with intake duct and cylinder design, relates to the intake air swirl and turbulence, their pattern and levels, and the valve timing, duration, and lift directly control the cylinder and volumetric efficiency. Certain operations of valves, such as the valve overlap, will also influence the intake air composition and physical conditions.

Currently, a universally accepted valve used in 4-cycle engines is the poppet valve. None of the other types, such as sleeve, piston and rotary valves, etc. can compete with it. The poppet valve has the following features:

1) It can give larger values of valve-flow area to piston area than most other types,
2) excellent flow coefficient,
3) low manufacturing cost,
4) very little friction so that it requires less lubrication, and
5) needs cooling on exhaust valves.

Poppet-valve design must achieve satisfactory results in respect to the amount of gas flow, cooling and heating flow, structural strength, lubrication and wear and provision for repair and replacement.

An extensive parametric study of the influence of valve operations on volumetric efficiency has been made.[5] Figure 2-13 shows the correlation of the volumetric efficiency and the $(\frac{b}{D})\ \frac{s}{a}$ where: b is the cylinder bore; D is the valve outside diameter; s is the mean piston speed; and a is sound velocity at inlet temperature. This conclusion indicates that the volumetric efficiency decreases as D increases, which we would expect. Figure 2-14 presents some of Ricardo's test data on the valve closing effect on volumetric efficiency. It can be seen that there is an optimized engine speed for maximum volumetric efficiency, and valve closing time has a significant effect on volumetric efficiency at either high or low engine speeds.

As mentioned above, exhaust valve cooling is a very important problem. Ordinarily, valves are made of austenitic steels, EV3 to EV11. In regard to the cooling problem, a relatively large stem diameter, plenty of material in the valve head, minimum exposure, of the stem to hot gas, coolant passages all around the seat and stem, and a minimum length of heat paths to coolant are considered. For valves of more than about 2-inches, internal cooling becomes necessary. Sodium is used frequently for internal cooling material.

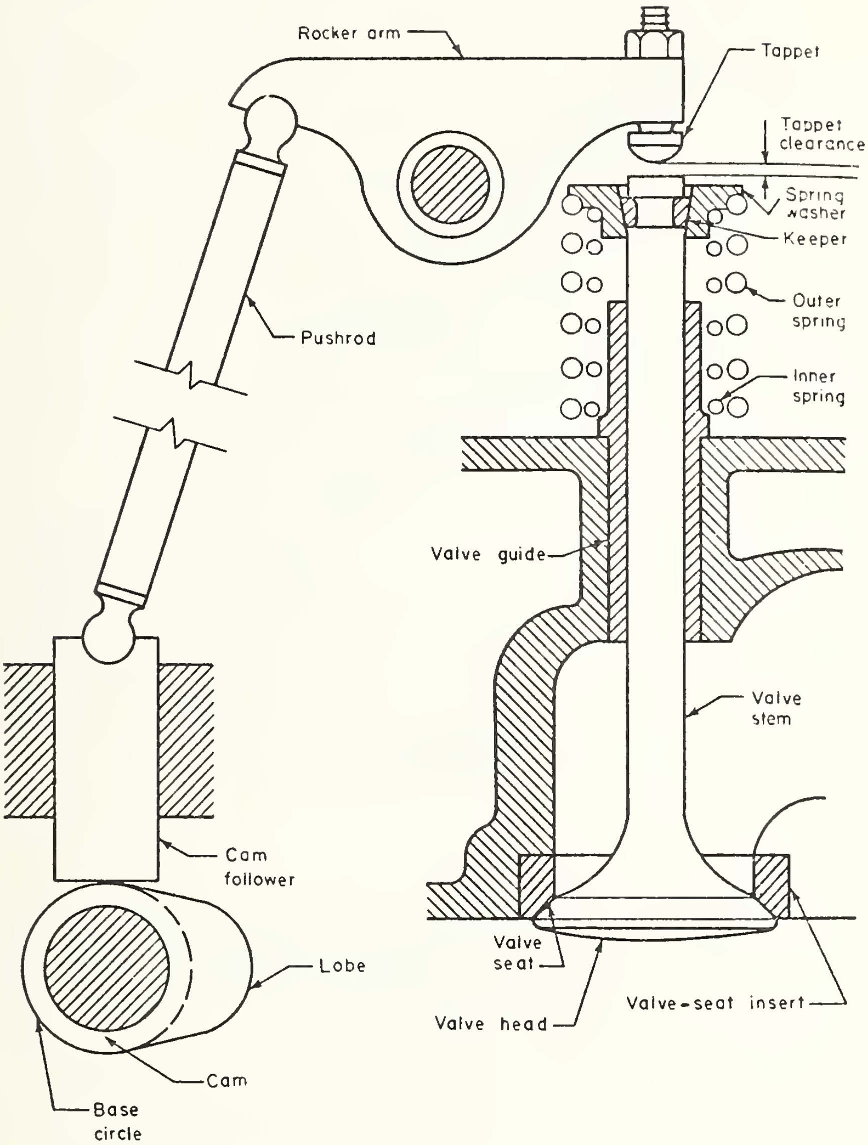
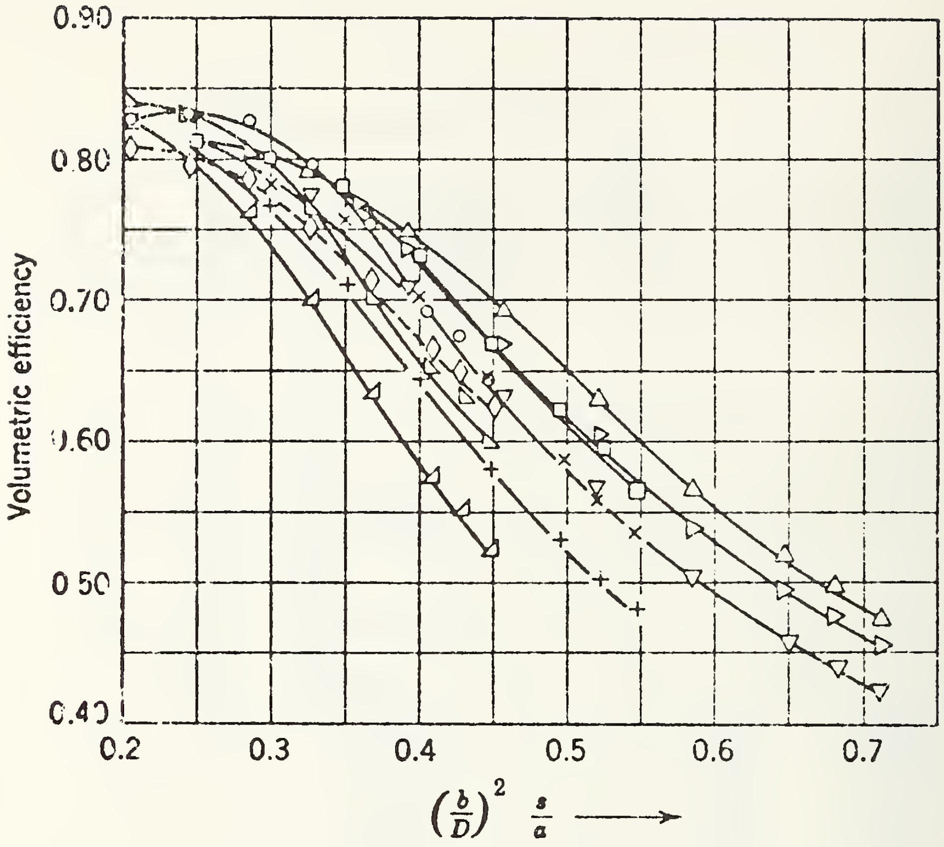



FIGURE 2-13. POPPET-VALVE SCHEMATIC DIAGRAM




| *D* (in) \ Lift (in) ⟶ | 0.208 | 0.238 | 0.262 | Design |
|---|---|---|---|---|
| 1.050 | ◁ | ◺ | ○ | *A* |
| 0.950 | + | × | □ | *A* |
| 0.830 | ▽ | ▷ | △ | *A* |
| 1.050 | | | ◇ | *B* |

FIGURE 2-13. VOLUMETRIC EFFICIENCY WITH SEVERAL INLET-VALVE SIZES, LIFTS, AND SHAPES: b = cylinder bore; D = : VALVE OUTSIDE DIAMETER: s = MEAN PISTON SPEED; a = VELOCITY OF SOUND AT INLET TEMPREATURE; CFR 3.25 x 4.5 in. CYLINDER: 4 - 4.92.

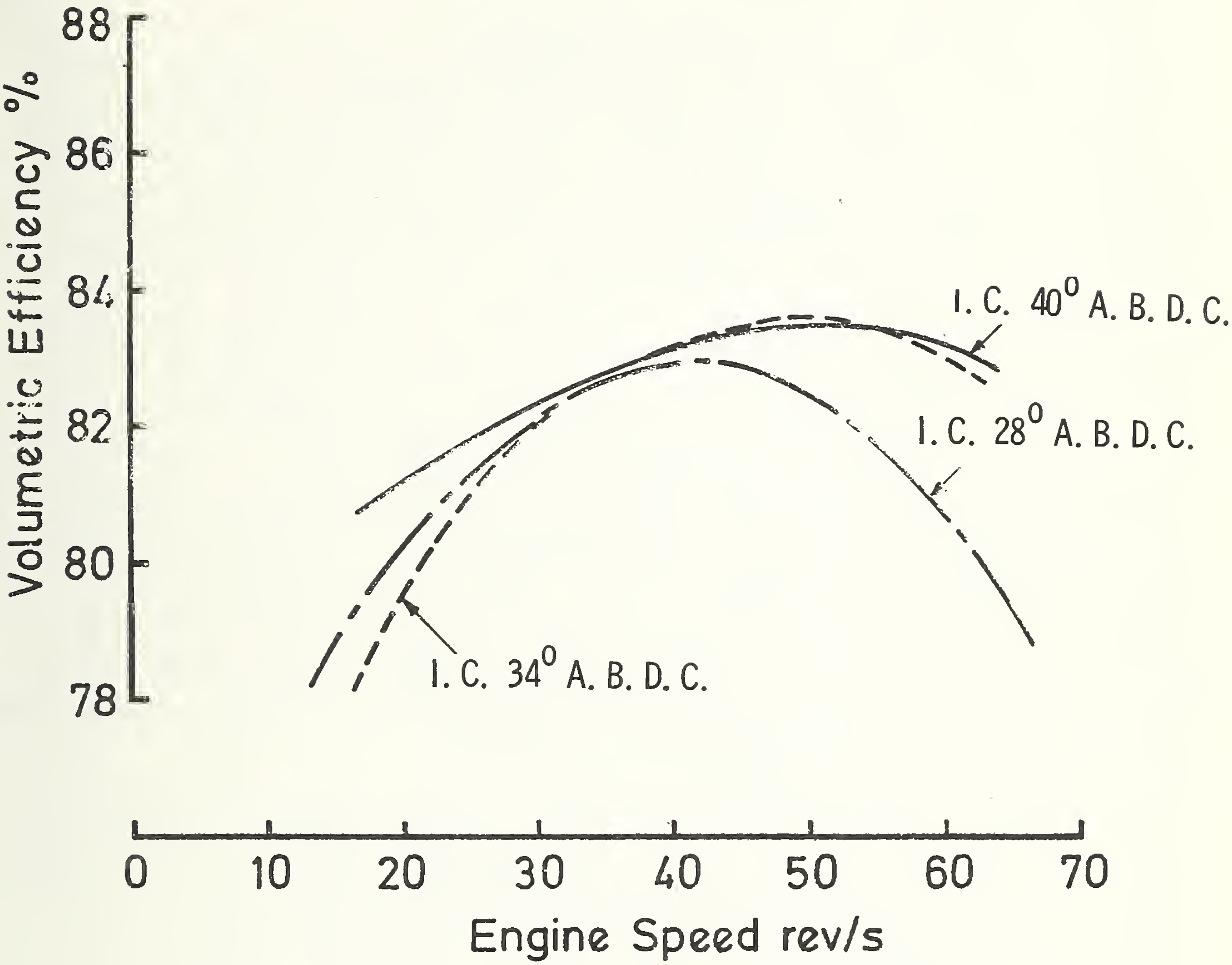



FIGURE 2-14. THE EFFECT ON VOLUMETRIC EFFICIENCY OF VARYING INLET VALVE CLOSING ON A 3.6 LITER 'COMET'ENGINE

## 2.4 COMPRESSION RATIO

From a thermodynamic point of view, higher compression ratios in general result in higher thermal efficiencies. Diesel engines usually run at much higher compression ratios (12-24) than gasoline engines (7-10). This is one of the reasons that diesel engines have better fuel economy than gasoline engines. However, as we attempt to apply high compression ratios to an engine, we have to be careful about the interaction among the compression ratio and other important parameters which influence the combustion process. Figure 2-15[6] gives a vivid example. As the compression ratio increases, we might expect, according to Hardenberg and Frankle[7] that the engine thermal efficiency increases (relfected by the BSFC behavior), smoke level decreases, and $NO_x$ level increases accompanying the decrease of ignition delay time. Yet, the actual test data, as shown in Figure 2-15, indicate that all the behaviors are the opposite of what would be expected. Because of the increase of compression ratio, the turbulence level in the cylinder is also greatly reduced; in consequence, the mixing is poor even after ignition, combustion is incomplete, and peak temperature is low. Therefore, a lower thermal efficiency (higher BSFC), a higher smoke level, and a low $NO_x$ level are obtained.

The minimum compression ratio of diesel engines depends on the cetane number of the fuel used as well as on the cetane number requirement of the engine. This limitation basically relates to the auto-ignitability of the compressed fuel-air medium in a diesel cylinder. Cetane number is the index which characterizes the auto-ignitability of the fuel. The maximum compression ratio of a diesel engine depends on the engine stress condition, the manufacturing limitation (for high compression ratio, the cylinder clearance becomes small so the precision requirement becomes relatively high), and the power used for compression. It is not always true that the higher the compression ratio, the better (as illustrated above) the performance of the engine. Hence, some investigators study the variable compression ratio (VCR) concept[8] to optimize engine operation in terms of compression ratio effect. This particular paper describes programs to reduce the peak trans-

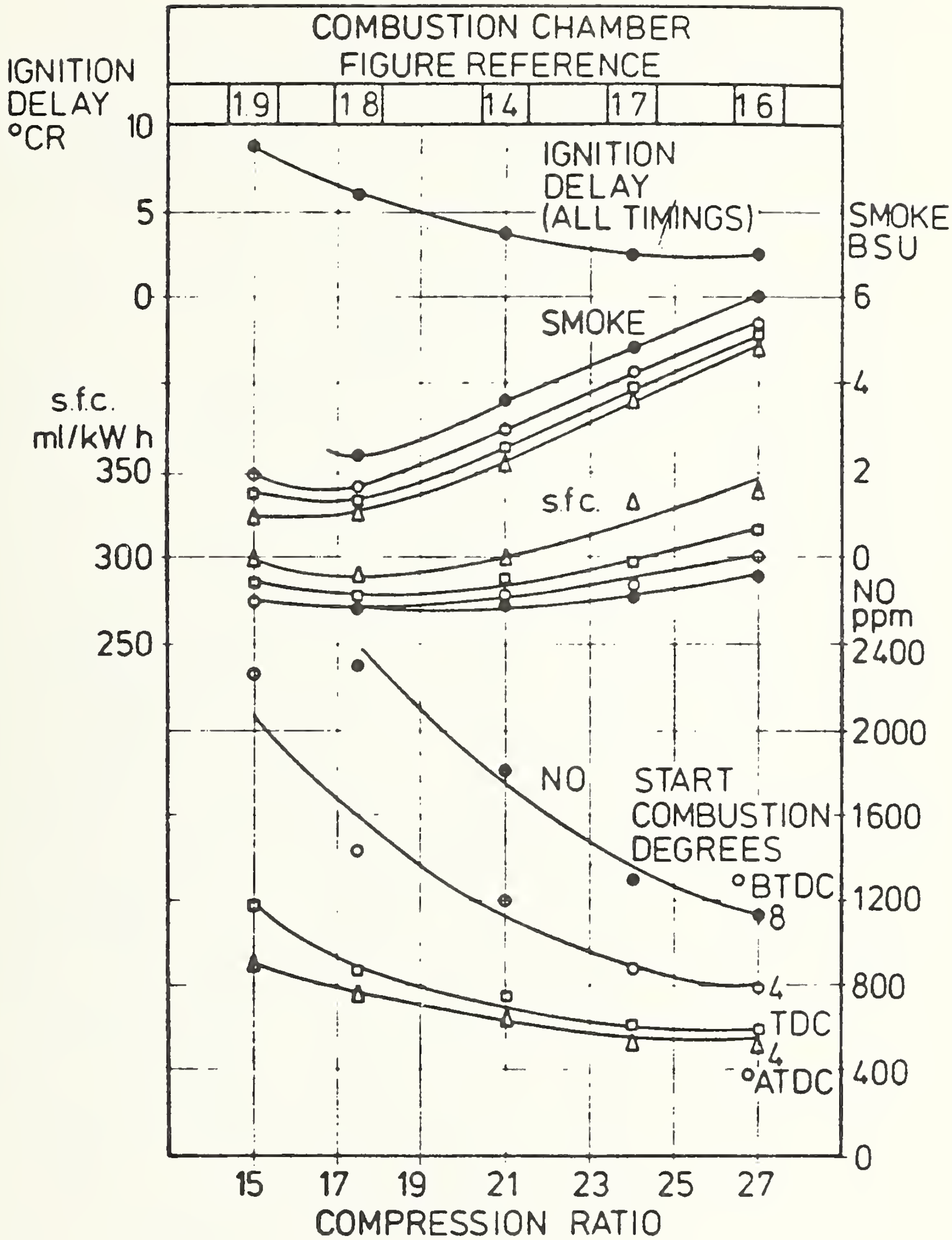



FIGURE 2-15. EFFECT OF COMPRESSION RATIO ON SMOKE, SFC AND NO EMISSIONS, ENGINE SPEED 1400 REV/MIN, EQUIVALENCE RATIO

ient smoke burst to acceptable levels. They used a 1360 $m^3$ displacement volume, 12-cylinder engine as their test apparatus. The cylinder clearance is varied by adjusting the piston head volume. The compression ratio is varied from 16 to 9, from low to high engine power range. The actual thermal efficiency of the AVCR 1360-2 is higher than what would initially be assumed for an engine employing a 9.1 compression ratio. With optimized components, the test engine smoke level (Figure 2-16) is greatly reduced in comparison with that of a standard engine.

## 2.5 ADIABATIC WALL AND TURBOCOMPOUND CONCEPT

By looking at a typical energy balance diagram of the diesel engine, as shown in Figure 2-17, it is clear that almost 2/3 of the energy input is wasted either through the coolant or by the outgoing exhaust gas. To recover that amount of energy it is natural to think of using an adiabatic wall surrounding the combustion chamber and some power producing device downstream of the engine exhaust. The U.S. Army Tank-Automotive Research and Development Command (TARADCOM) and Cummins Engine Company have jointly explored the adiabatic wall concept.[9] Figure 2-18 shows a schematic diagram of an adiabatic turbocompound engine. As shown, both the engine and the exhaust turbine produce power for the engine flywheel. Figure 2-19 shows another assembly diagram which contains a Rankine bottoming cycle on an adiabatic diesel engine. The adiabatic engine is supplied with turbocharged air through an aftercooler. The exhaust gas, flowing through an exhaust turbine, is used to generate vapor. The vapor generated in the boiler is used to turn an expander to produce power. The performance of an insulated naturally aspirated diesel engine has also been predicted. The comparison with a standard diesel is shown in Table 2-2. From the listed value, it appears that little is gained by insulating the engine alone since more energy is exhausted. This situation leads to testing the insulation, together with a turbocompound (or Rankine cycle or both) mechanism. Table 2-3 shows some predicted results of several combinations and their comparison. It is clear that by using adiabatic, turbocompound and

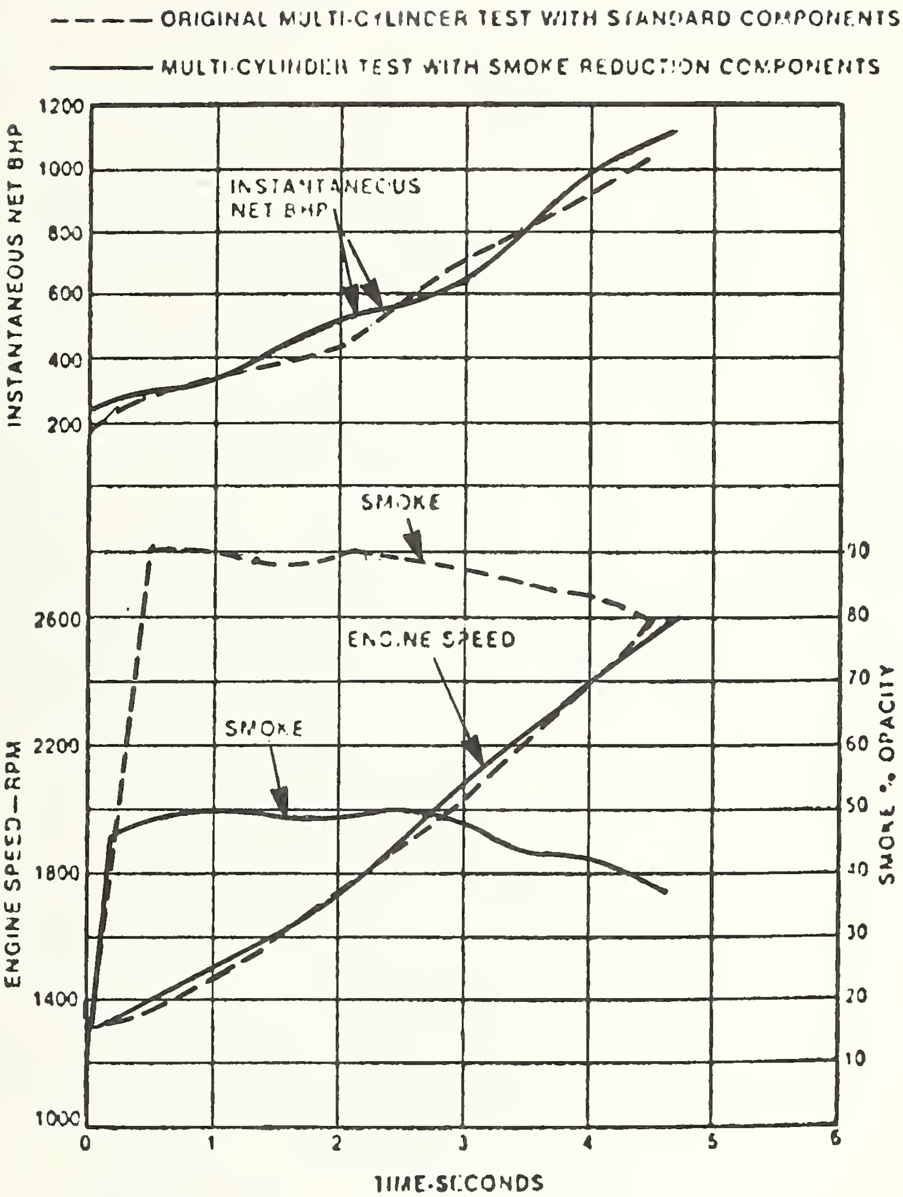



Figure 2-16. MULTI-CYLINDER ENGINE TRANSIENT ACCELERATION SMOKE TEST

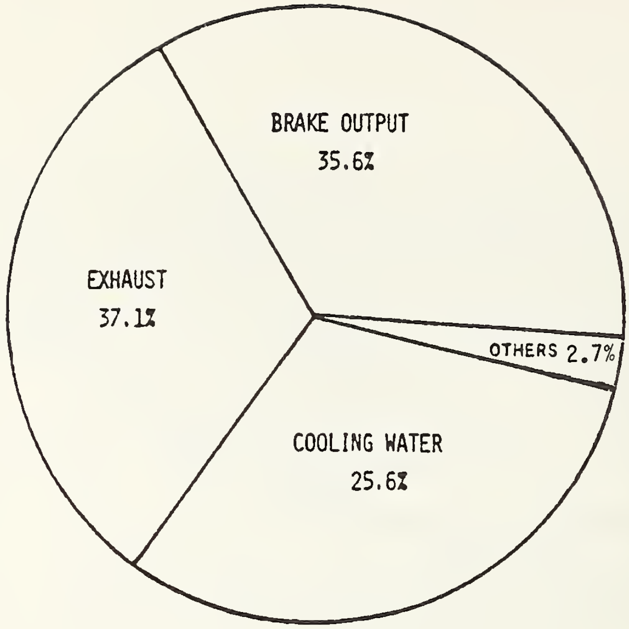



FIGURE 2-17. TYPICAL ENERGY BALANCE OF DIESEL ENGINE

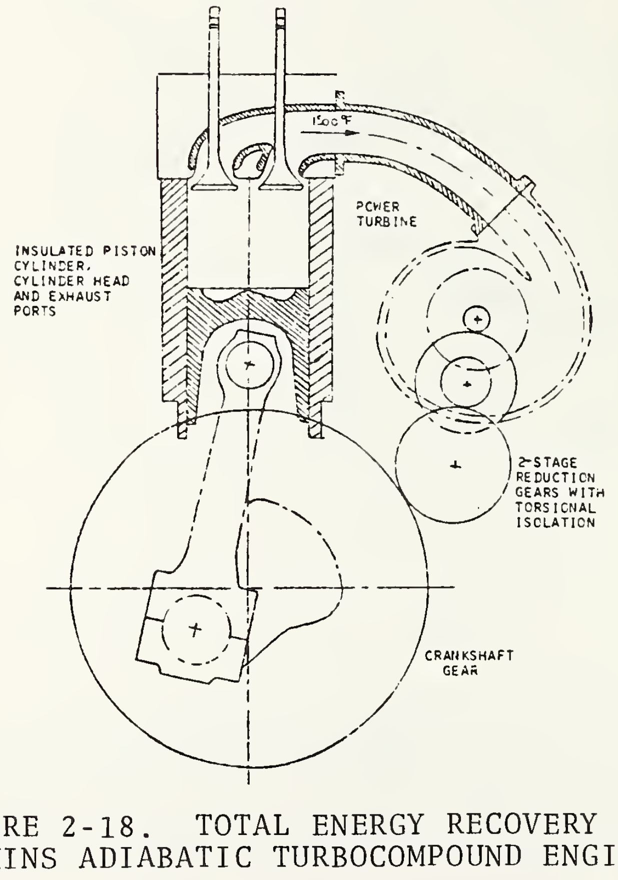



FIGURE 2-18. TOTAL ENERGY RECOVERY VIA
CUMMINS ADIABATIC TURBOCOMPOUND ENGINE

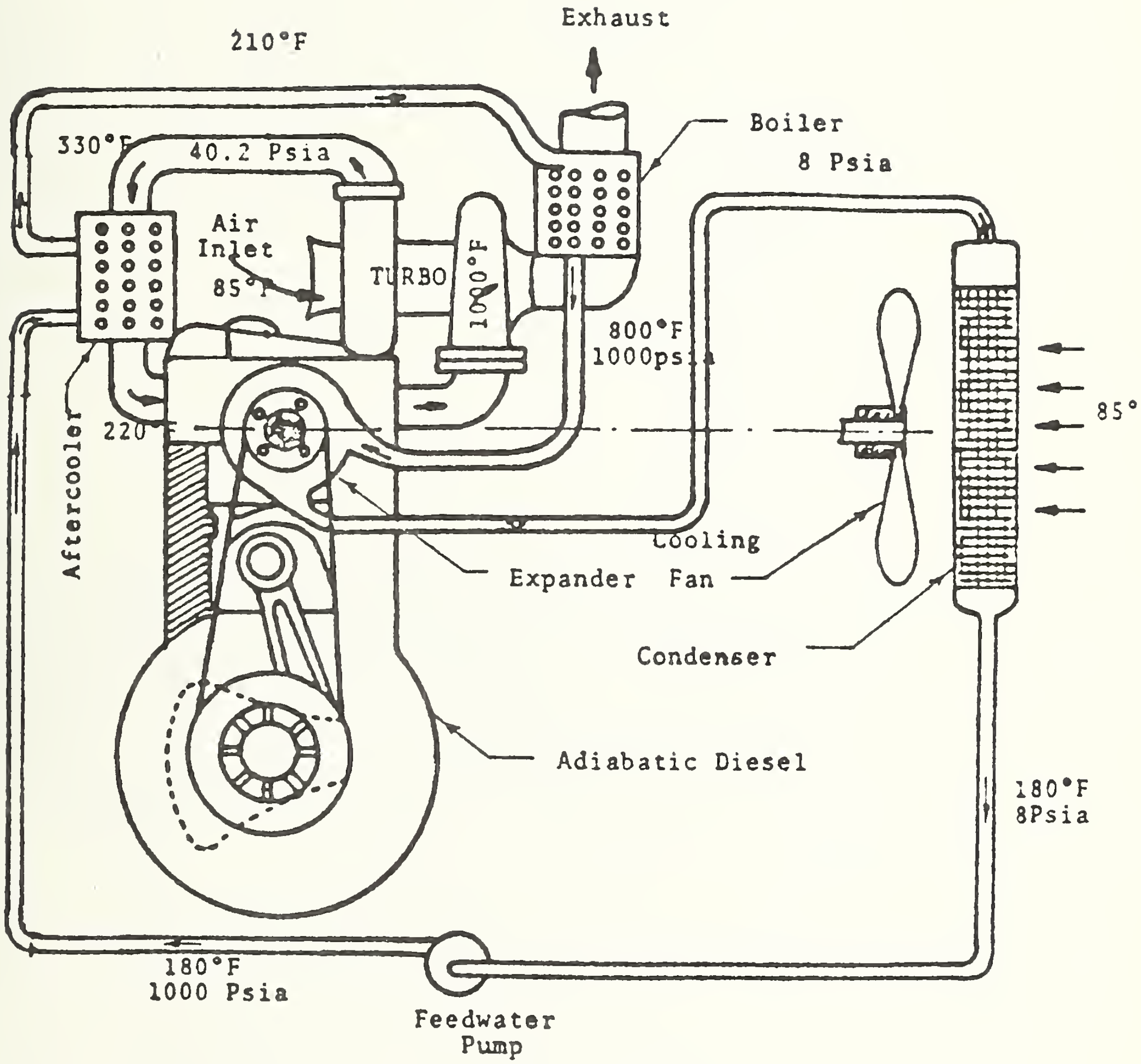



FIGURE 2-19. SCHEMATIC OF A RANKINE BOTTOMING CYCLE ON AN ADIABATIC DIESEL ENGINE SHOW-THERMODYNAMIC STATE POINTS AND NUMBERS

TABLE 2-2. PREDICTED PERFORMANCE FOR 250 BHP STANDARD AND INSULATED NATURALLY ASPIRATED DIESEL ENGINE

| | Standard | Insulated |
|---|---|---|
| Engine Speed, rpm | 2100 | 2100 |
| Intake Manifold Pressure, psia | 14.1 | 14.1 |
| Exhaust Manifold Pressure, psia | 15.0 | 15.0 |
| Exhaust Temperature, $^{o}F$ | 1280 | 1860 |
| Volumetric Efficiency, $^{o}F$ | 0.843 | 0.680 |
| Heat Projection Rate, Btu/min | 5842 | 556 |
| Start of Heat Release, $^{o}CA$ | 340 | 340 |
| BSFC, lb/BHP-HR | 0.400 | 0.398 |
| Installed BSFC, lb/BHP-HR | 0.415 | 0.398 |
| $BSNO_2$, Grams/BHP-HR | 5.0 | 5.0 |

TABLE 2-3. DIESEL CYCLE SIMULATION RESULTS OF VARIOUS ADVANCED DIESEL BASED POWER PLANT

| | Turbo-compound (cooled) | Adiabatic Turbo-compound (Insulated) | Adiabatic Rankine $\eta_R$ = 0.158 | Adiabatic Turbocompound Rankine $\eta_R$ = 0.158 |
|---|---|---|---|---|
| SPECIFICATIONS | | | | |
| Engine RPM/Air Fuel | 2100/28 | 2100/28 | 2100/28 | 2100/28 |
| Peciprocator, BMFP | 177.8 | 177 | 183 | 117 |
| PERFORMANCE | | | | |
| Peciprocator, BHP | 403.2 | 401.8 | 415.1 | 401.8 |
| Turbine, BHP | 40.8 | 109.6 | - | 109.6 |
| Rankine, BHP | - | - | 77.6 | 64.4 |
| Total, BHP | 444.0 | 511.4 | 492.7 | 575.8 |
| EFFICIENCIES | | | | |
| Peciprocator, BSEC | 0.341 | 0.363 | 0.323 | 0.366 |
| Overall | 0.300 | 0.285 | 0.272 | 0.255 |
| EMISSION | | | | |
| $BSNO_2$, Grams/BHP-HP | 5.0 | 5.0 | 5.5 | 4.4 |

Rankine cycle together, significant gains in fuel economy and emission level may be achieved.

## 2.6 FRICTION

Frictional loss stands for the unavoidable part of power loss from diesel engine. By definition, it represents the difference between the indicated and brake power output. Frictional loss mainly consists of mechanical friction loss, pumping power loss, compressor power loss, auxiliary power loss, etc. Important types of mechanical friction may be divided into four classes, i.e., fluid-film friction, partial-film friction, rolling friction, and dry friction. Under normal operating conditions, all the rubbing parts of an engine are supposedly operated under fluid-film friction. So, the partial-film friction, like dry friction, is of little importance as a contributor to engine friction. Fluid-film friction depends heavily on fluid thickness, and surface shape.[10, 11]

The above parameters can be regrouped into dimensionless form so that the dimensionless frictional coefficient can be expressed as Reynolds number as well as those dimensionless parameters characterizing the surface geometry. Rolling friction associates with ball and roller bearings and with cam-follower and tappet rollers. These bearings have a coefficient of friction which is nearly independent of load and speed. The frictional force is due partly to the fact that the roller is continuously "climbing" the face of a small depression in the track created by the contact surfaces as they deflect under the load. Hence, the elastic property, such as Young's modules of the surface material, will influence the magnitude of the rolling friction. Frictional loss is ordinarily expressed as frictional mean effective pressure (FMEP - $lb_f/in^2$). In practice, one of the methods for evaluating FMEP is to use so-called Willans Line, i.e., a curve recording the relation between fuel-energy input and BMEP. FMEP can be estimated by extrapolating the Willans Line towards the negative MEP-axis direction until it intercepts the MEP axis. The magnitude between the intercept point and the origin on the MEP axis is presumably to represent the FMEP, as shown in Figure 2-20. This certainly provides a convenient method for estimation of overall friction loss of the engine. From statistical records, the mechanical friction losses can be broken down into a distribution as shown in Figure 2-21.

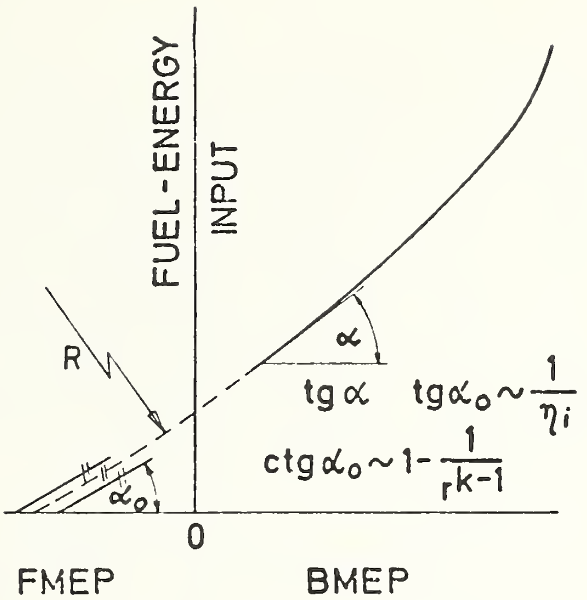



FIGURE 2-20. WILLANS-LINE

| | % |
|---|---|
| FRICTION PISTON | 40-45 |
| FRICTION BEARINGS | 25-30 |
| VALVE TRAIN & GEARS | 14-17 |
| WATER + OIL PUMPS | 9-10 |
| INJECTION PUMP | 6- 8 |
| | ~100% |

FIGURE 2-21. BREAK-DOWN OF 100% OF MECHANICAL LOSSES

# 3. ENGINE CONTROL STRATEGIES

Until recently, the task of calibrating a given engine/power train configuration was simply an iterative procedure of testing, adjusting the parameters of the injection system (mainly fuel delivery and injection timing) and then re-testing. The objective was to obtain good performance, low fuel consumption, low smoke, and low noise. As the number of control variables grew, and as the complexity of the ways of controlling emissions increased due to more stringent emission limits evaluated under transient running conditions, the task became overwhelming.

Mathematical models are used to compute the 1975 FTP cycle emissions and fuel economy on steady state bench data. A good correlation exists with chassis dynamometer tests. In fact, for diesel engines, any transient condition can be assumed as quasi-steady because the fuel is directly injected into the combustion chamber. Therefore, if the emission (HC/CO/$NO_x$) concentration and fuel consumption maps (in grams per hour) are given, modal emission and fuel economy characteristics can be calculated as follows. Main data requested for performing program are: vehicle weight; wheel, transmission and engine inertia; gear ratios; tire sizes; vehicle frontal area; air and rolling drag; transmission efficiency. The cycle is divided into one second intervals. The program calculates, for each interval, average speed and acceleration of the vehicle and then the power needed to perform the cycle. The program supposes that the engine runs at constant speed and load for every one second interval. According to the tire and total transmission ratio, the program calculates a correspondence between the engine speed and the actual vehicle speed. From these data (engine speed and power) the program reads, on the steady state maps, fuel consumption and HC/CO/$NO_x$ emission at each second of the cycle. By summing all the calculated values, total fuel consumption and emissions (in grams) emitted during the cycle are obtained. The final value of the pollutant and fuel consumption (grams/mile) is computed dividing the previous value by total mileage.

The overall prediction of the behavior of an engine/vehicle system must, however, lead to underlining the main engine operational points, which affect emissions and fuel consumption. Fiat[12] developed an optimization program identifying the regions giving the maximum contribution to regulated emissions. The model is based on the computation of the emission level that can be obtained on an FTP urban driving cycle from steady-state emission maps. To each point of the steady-state emission map an increment of 50 percent has been applied. Then the corresponding variation in emission level has been mapped in order to identify the engine running conditions, giving the maximum contribution to each pollutant. Figures 3-1, 3-2, and 3-3 show the pattern of HC, CO and $NO_x$, respectively.

Consider for example Figure 3-1 related to HC emission. Consider the percentage variation of two different points on the HC steady map. The effect on the FTP-urban cycle of one point in the zone 1, 1 bar - 2100 rpm is 100 percent while the effect of another point placed on line 20 is five times lower. In other words, the same percentage of improvement (or decline) of the steady-state emission causes the same effect on the FTR-Urban cycle if the selected points are on the same line. The relative variation when passing from one line to another is expressed by the numbers reported on the Figure. Figure 3-3 related to $NO_x$ has the zone to maximum influence on the FTP-Urban cycle resulting from the steady-state emission changes at about 3 bar of BMEP and 2000 rpm of engine speed. Because of the difference in the maximum zone between HC and $NO_x$ emissions, it is possible to perform different adjustments of injections (or setting or configurations) system parameters, EGR and other emission control devices in order to achieve minimum HC and $NO_x$ emission.

Fiat under contract to DOT/TSC summarized the different emission targets and estimated fuel economy improvements that can be achieved by introducing different engine and vehicle modifications for a 3000 lb vehicle. See Table 3-1. The first level to reduce emissions is to optimize the injection system once the best combustion chamber configuration is found. Furthermore, postinjections must be drastically reduced by a proper arrangement of the injection

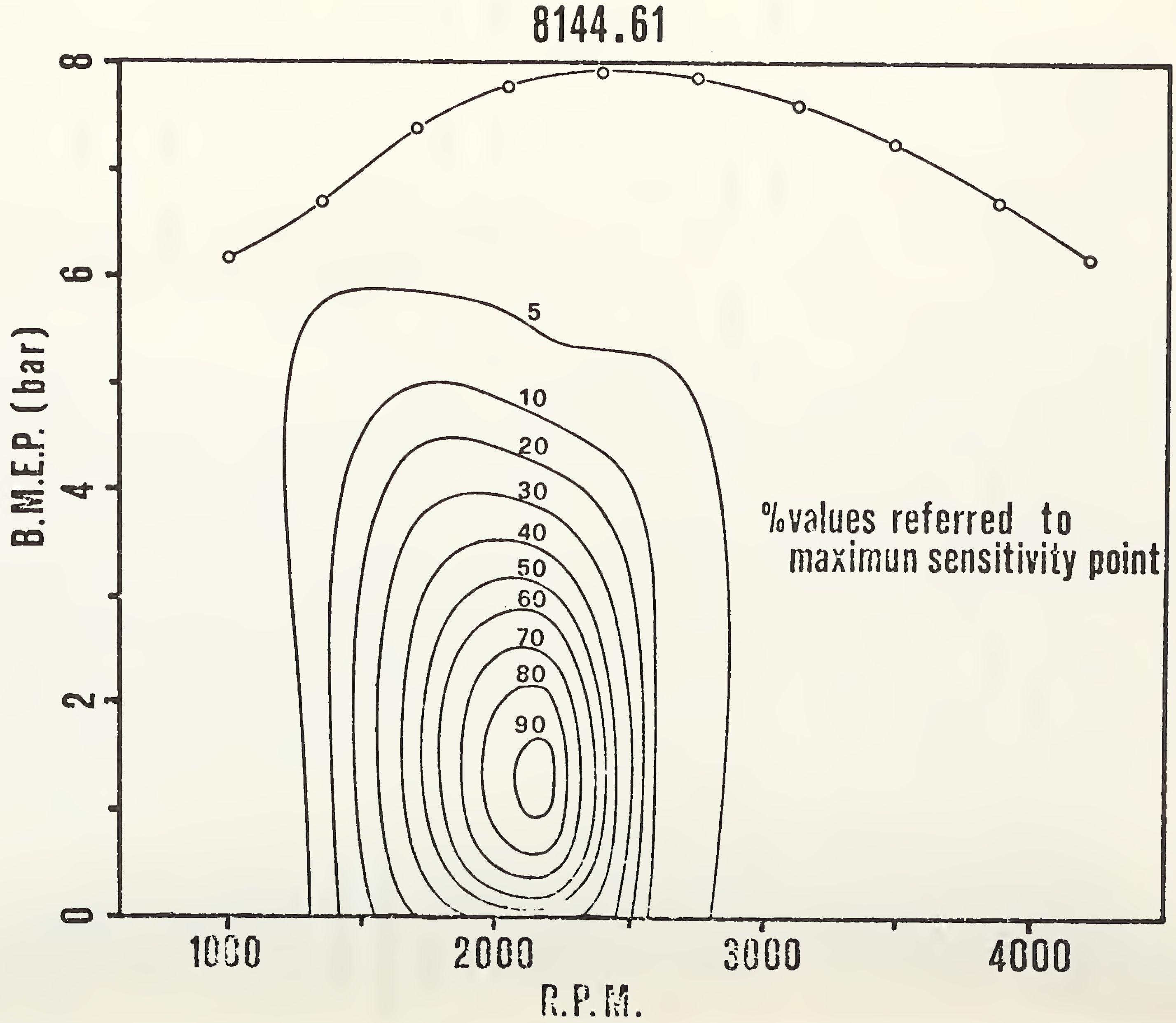



FIGURE 3-1. 3000 lb I.W. 3.2 REAR AXLE RATIO SENSITIVITY MAP OF HC EMISSION.

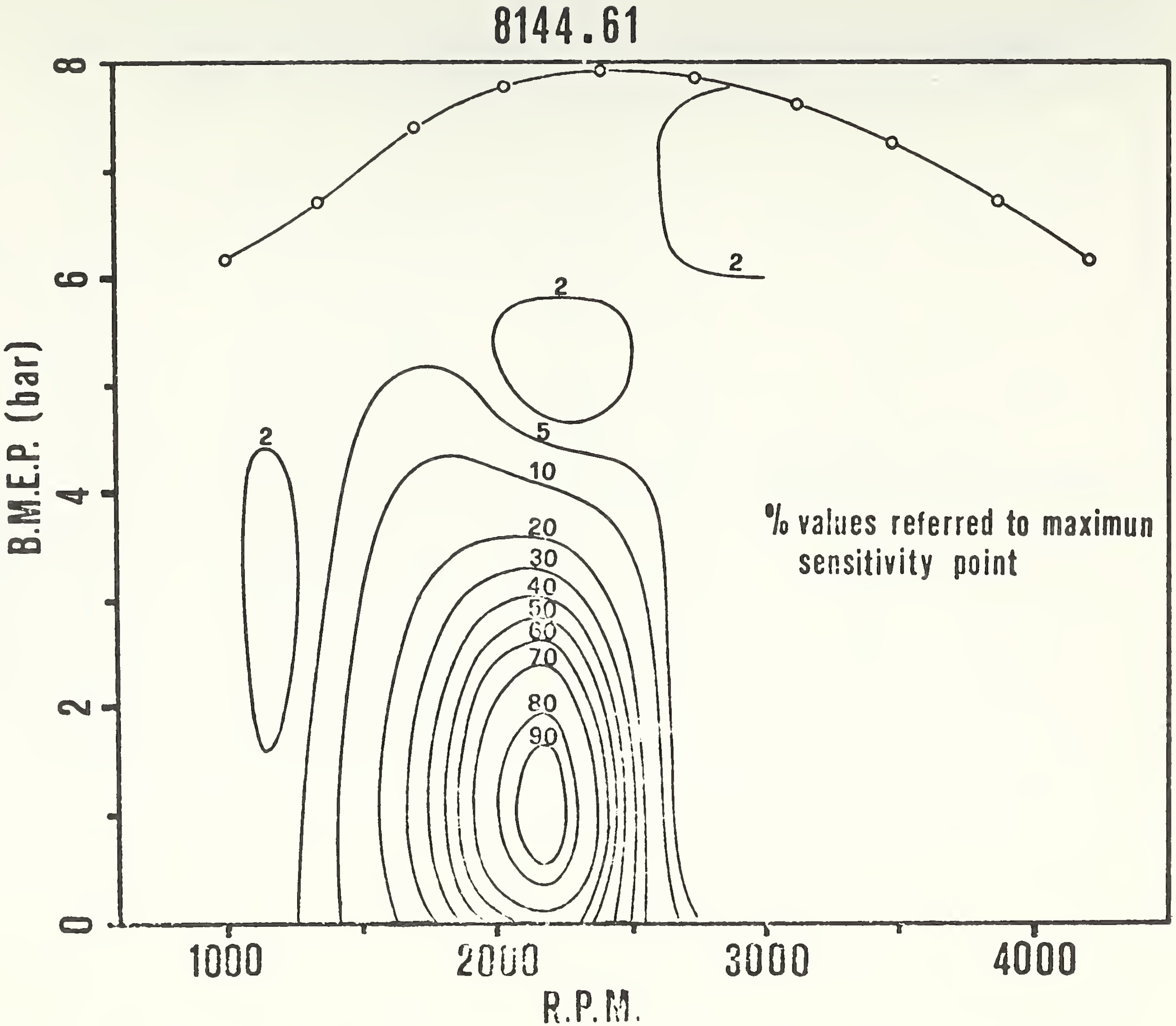



FIGURE 3-2. 3000 lb I.W. 3.2 REAR AXLE RATIO SENSITIVITY OF CO EMISSION

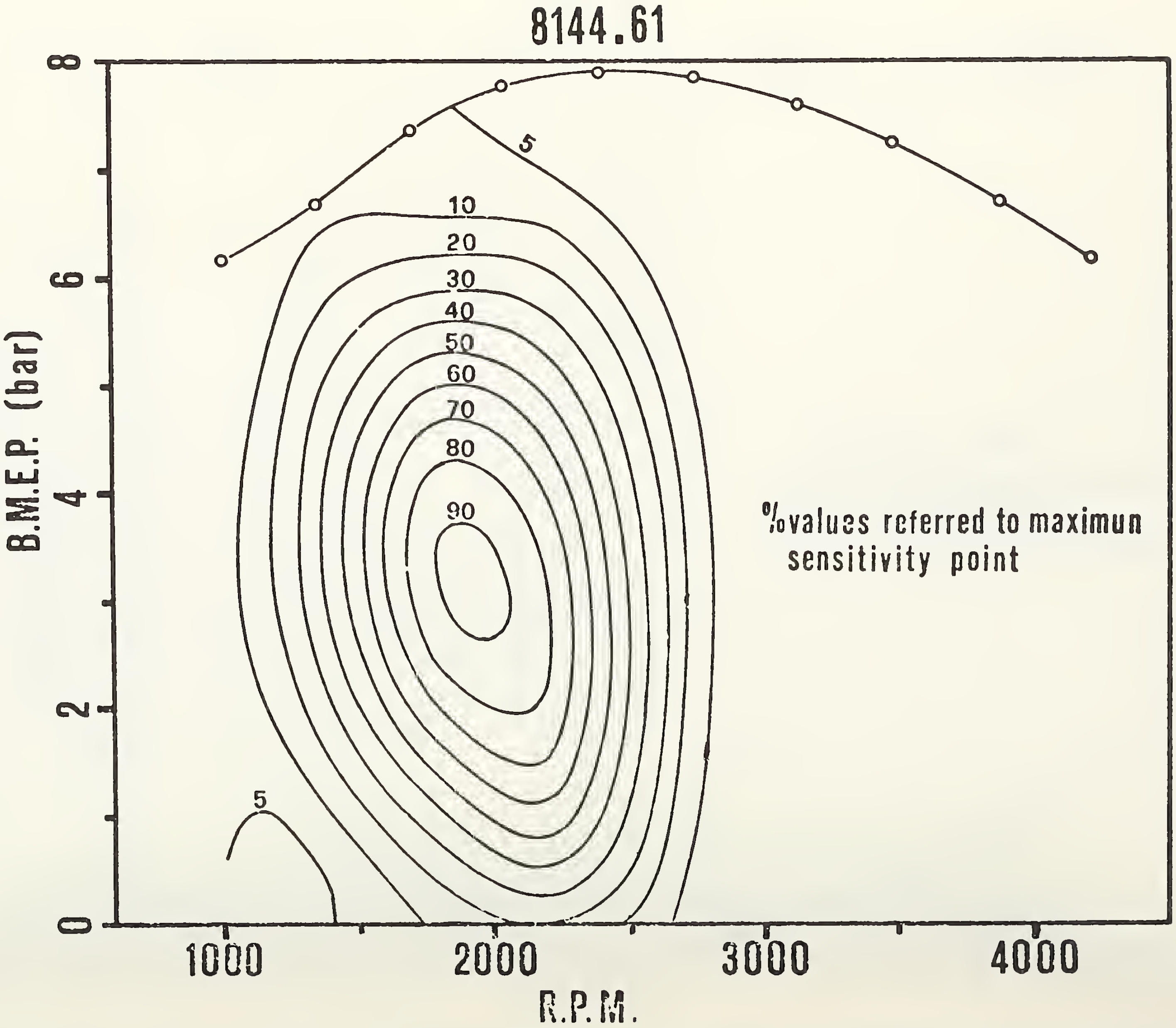



FIGURE 3-3. 3000 lb I.W. 3.2 REAR AXLE RATIO SENSITIVITY MAP OF $NO_x$ EMISSION

TABLE 3-1. IMPACT OF VEHICLE MODIFICATIONS ON EXHAUST EMISSION LEVELS

| Engine | HC | CO | $NO_x$ | Partic-ulate | Average Relative Fuel Economy Changes |
|---|---|---|---|---|---|
| N.A. Present Solution | >0.41 | <3.4 | >2.0 | <0.6 | - |
| Fuel Injection Optimization (F.I.O.) | <0.41 | <3.4 | <2.0 | <0.6 | 0 |
| N.A. + F.I.O. + Vehicle Modification (V.M.) | " | " | " | " | +10% |
| N.A. + F.I.O. + V.M. + E.G.R. | >0.41 | <3.4 | <1.0*<br><1.5** | >0.6 | + 6% |
| Turbocharged (T.C.) + F.I.O + V.M. | <0.41 | <3.4 | <2.0 | <0.2 | Between +15 and 20% |
| T.C. + F.I.O. + E.G.R. | <0.41 | <3.4 | <1.0*<br><1.5** | >0.2 | Between +10 and 15% |

*For vehicle inertia weight less than or equal to 2,500 lbs (Reference 14).
**For vehicle inertai weight ranging from 2,750 to 3,500 lbs (Reference 12).

system and nozzle "sac" volume of the injectors reduced. In this way low HC levels are obtained and the injection timing can be retarded for achieving low $NO_x$ in those cycle zones that give maximum contribution to the grams per miles of $NO_x$ emission. Thus, injection timing has to be controlled against engine speed and load.

Vehicle modifications (i.e., rear axle ratio) are important in order to achieve better fuel economy without unacceptable loss in performance.[12] At the same time slight reduction in $NO_x$ is obtained. Lower inertia weight is another important factor to attain better fuel economy and lower $NO_x$.[12] EGR prototypes and oxidation catalysts have been tested to achieve lower $NO_x$ and lower HC levels, respectively. A system to control the engine particulate emission is turbocharging. Fiat has shown that particulate levels can be as much as 35 percent of that obtained with the naturally aspirated version[12, 13] and fuel economy can be improved too.[14] Unfortunately, EGR has to be used to obtain lower levels of $NO_x$ (1gm/mi). EGR studies till now[14] cause a strong increase in particulate (up to two times). Thus, the present technology emission level of 0.41/3.4/1.5/0.6 gms/mi of HC/CO/$NO_x$/particulate can be achieved with very good fuel economy while the lower emission level of 0.41/3.4/1.0/0.20 will be the subject of future research.

# 4. IMPLEMENTATION OF ENGINE CONTROL

## 4.1 GENERAL

The use of indirect injection and direct injection light duty diesels in passenger cars and light trucks is a subject under discussion.[15] The basic issue is particulates (or better carbon formation) and their associated compounds. Some examples of particulate reductions have been cited in the previous section. Particulates may be controlled through aftertreatment devices, for example, particulate traps and oxidation catalysts. However, these devices are in the early stages of development. Electronic control of EGR, turbocharging and injection timing offer added potentials to reduce particulate emissions. Finally, fuels of suitable characteristics for passenger cars and light truck diesels require explorations.

## 4.2 AFTERTREATMENT DEVICES

Diesel engine oxidation catalysts must satisfy more demanding requirements than present day catalysts used for spark ignition engines. The minimum, average and maximum gas temperatures in the exhaust manifold of a Naturally Aspirated Light-Duty LDT Diesel engine are about 120, 200 and 550°C respectively during the 1975 FTP. These values are much lower than that of a gasoline powered vehicle. Therefore, an oxidation catalyst with good efficiency at lower temperature has to be used, for example, the platinum oxidation catalyst.

The variability in air/fuel ratio (a catalyst works well in the range of 30 and 70 percent maximum load) also requires attention. HC and CO reduction of 50 and 65 percent respectively were found when using a platinum oxidation catalyst.[12, 16] Other tests at DOE-Bartlesville showed a PNA reduction. In particular B(a)P decreased to 40 percent under steady state low temperature conditions.

Exhaust particulate trap systems which use the available lead trap technology are being studied. Tests have been made on a 5 cylinder Mercedes 300-D with automatic transmission at the 4000 pound weight. A particulate reduction of something greater than 50 percent was found when a alumina-coated steel wool trap was used. The efficiency was negligible after 3000 miles. A substantial reduction in odor was also found. Not only was there less odor intensity, but there was a different quality. Hydrocarbons were substantially less (40 percent) while there was no change in the other regulated emissions nor in economy. Sulfates were sensibly low (90 percent when trap was new and 60 percent at the end of the test) and Benz (a) Pyrene was half of the base value. It appears that the trap acts as a catalyst while HC and PNA first and later particulate are aborted effectively, usually through an upstream catalyst system and a downstream trap (DOE-Bartlesville: tests on stationary diesel power plants). Since these data represent initial findings, however, further research still remains in order to find effective aftertreatment systems (catalyst and/or traps) for diesel engines.

## 4.3 FUEL INJECTION SYSTEMS

Historically, injection systems are designed to satisfy power, consumption and exhaust emission requirements.

A diesel engine injection system consists of an injection pump, a flow adjustment system and the same number of injectors as the engine cylinders. The injection pump is either of the "in-line" or "rotary piston and distributor" type. An in-line pump consists essentially of the same number of pumping elements as the engine cylinders. Each element is actuated by a cam mechanism, the eccentric motion of which gives high pressure of fuel inside the pumping element. The fuel, under high pressure, actuates the injector needle valve and is atomized inside the combustion chamber through the injectors.

Unlike the Bosch gr. "P" MW in-line pumps, the Bosch "VE" and CAV "DPA" distributor and rotating piston pumps are provided with a distributor that opens as many openings as the engine cylinders. These pumps are commonly mounted on prechamber engines, mainly because of their small size but also because injection pressure and duration are not so important as for the direct injection engine.

Nozzle type holes are used for direct injection engines due to the configuration of the engine head and the need to send fuel directly to the combustion chamber. Pre-and swirl chamber engines, can more easily use pintle-nozzle.

The need to provide lower emission engines has recently prompted studies into the injection system features in order to identify the main major controlling parameters. (Typical examples are elimination of the after-injection of fuel which does not affect performance and consumption but which drastically alter the emission of HC, as shall be seen later.)

The parameters for emission by engine type can thus be shown.

D.I. Engines

1) Injection pressure: Increased injection pressure does not alter the values of $NO_x$ and HC substantially, while particulate can be reduced without increasing fuel consumption.[17]

2) Timing: Delaying injection reduces $NO_x$ and HC by about (10/12 percent)/degree but increases consumption by 1 percent per degree, particulate by (10/15 percent)/degree and exhaust gas temperature by about (10°C)/degree. Decrease of HC is voided and changes trend when timing is close to T.D.C. (-4 and 5° B.T.D.C.).[17] Speed and load changes at the above rate show marked variations of emission levels. Trend inversion occurs from 2 to 8 degrees of timing before top dead center.[18]

3) Post-injection: The parameters chosen for a certain injection system (speed, load, timing and type of nozzle) can create pressure waves in the piping so as to form post-injections which tend to increase HC emission and

even consumption. This can be eliminated by reducing the internal diameter of pressure piping, increasing the delivery valve retraction volume or by adding a reserve flow throttle valve on the piping from the pump to the injector.[16]

4) Injection duration: Faster injection together with timing adjusted to load and speed reduce $NO_x$ and HC. Interesting results were obtained by using the FIAT DRF experimental pump with high injection speed that can be coupled with an electronic timing adjuster.[19]

5) Nozzles: HC emissions are reduced by reducing nozzle sac volume. Interestingly, HC emission per cycle is about 20 percent of sac content. If the nozzle is inclined on the engine head with respect to the cylinder center line, sac reduction is greatly limited. Reduction of nozzle holes, together with higher pressure adjustment and injection pressure, reduce particulates.[20]

Prechamber and Swirl Chamber Engines

1) Injection Pressure: The effect of injection pressure on emissions is relatively negligible, though it varies according to the type of swirl chamber or prechamber.

2) Timing: Particulates are reduced by delaying timing, which is not true with D.I. engines. Reduction of $NO_x$ emission is to the order of about (5 percent)/degree, HC decreases by the same amount down to 6° - 8° of timing in respect to T.D.C. The trend then changes and values increase markedly. Optimum timing at minimum HC must thus be established. Timing adjustment must be made as a function of load besides speed.

HC increases by 100 percent from 50 to 100 percent of load[17] on a 2.9771. swirl chamber engine at 2000 r.p.m.

3) Post-injection: Research on an engine of the type mentioned above has led to the conclusion that shortening

the nozzle throttle length by 50 percent (to decrease the residual pressure and the peak pressure in the injection pipe) and adopting a reverse flow damping valve for damping the reflected pressure in the injection pipe decreases post-injection from 90 to 6 percent, reduces HC emission by 80 percent and CO by 40 percent.

4) Injection duration: The effects of injection duration on HC and $NO_x$ emissions are relatively scant. It can be said, however, that with a shorter injection time, injection delay can be chosen so as to obtain a minimum level of $NO_x$ before HC increases.

5) Nozzle: No real sac exists on pintle nozzles but noxious volumes can be reduced in this case too by properly designing the pintle shape. Volume reductions of 0.8 to 0.3 $mm^3$ have shown as much as 30 percent and 40 percent HC and CO emission reduction respectively.[17]

Great importance should be attributed to the position of the nozzle inside the swirl chamber. HC emission can in fact be reduced 50 percent by moving the jet towards the swirl chamber center. The jet should be towards the wall to obtain lower $NO_x$ values.[16]

## 4.4 ELECTRONIC

Small and medium displacement engine injection adjustment systems are currently based on centrifugal mass mechanical adjusters. Injection timers are either mechanical (with in-line pumps) or hydraulic (with rotating pumps).

The need to adjust timing more precisely to further reduce emissions requires more sophisticated adjustment. Injection pump manufacturers are producing adjustment systems based on electrical or hydraulic actuators slaved to a logically present electronic center. Such a type of adjustment system was designed by Fiat on the DRF pump.[18]

## 4.5 EXHAUST GAS RECIRCULATION

Exhaust gas recirculation has so far been studied as a means to reduce $NO_x$ emissions, at the expense of a slight HC increase. EGR actually increases particulates, and leads to corrosion of the first piston ring found in durability tests. It is thus important to have an EGR system with extremely precise control.

An EGR system capable of precisely controlling air flow as well as partial control of load, by the air/fuel ratio, has been developed by Bosch.[21] The main components of the system are:

- mixture-control unit, consisting of air-flow sensor and fuel-flow sensor.
- throttle valve with actuator for the control of the recirculated exhaust gas.
- The air-flow sensor consists of an air funnel and an air-flow sensor plate which can move within it and which is mounted on a lever. A hydraulic plunger acts on this lever so that the air-flow sensor plate is loaded with an essentially constant force against the direction of the flow of the air. The position of the air-flow sensor plate is a measure of the rate of air flow. The fuel which is fed to the engine passes through a slit throttle the section of which can be altered by the displacement of a plunger. A different pressure valve regulates the pressure drop at this throttle to a constant value. With a constant pressure drop - i.e., with a constant stroke of the diaphragm of the differential pressure valve - the position of the plunger is a measure of the rate of fuel flow. The throttle valve is fitted at the end of the exhaust gas recirculation pipe in the intake manifold of the engine. The position of the throttle valve determines the quantity of exhaust gas recirculated. The valve can both close the exhaust gas pipe and reduce the section of the intake manifold. It is hydraulically operated by an actuator. The actuator is acted upon on the one side by a variable pressure, and on the other side by a spring.

The individual components work together as follows: the position of the throttle valve determines the air-exhaust gas ratio of the cylinder charge. The throttle valve is regulated by the mixture-control unit so that a set air-fuel ratio is maintained. In the mixture-control unit the air-flow sensor and the fuel-flow sensor are connected by means of a lever, so that when the travel of the air-flow sensor plate in the air-flow sensor increases, the cross-section of the variable slit throttle of the fuel-flow sensor is increased. The air-flow sensor is now tuned with the fuel-flow sensor so that, when the desired air-fuel ratio is reached, there is a constant pressure drop at the slit throttle for each chosen quantity of fuel. This tuning is achieved by shaping the air funnel of the air-flow sensor. In this condition the diaphragm in the differential pressure valve of the fuel-flow sensor opens a certain cross-section, through which a pump forces fuel which flows back to the fuel tank via a restriction in the actuator. In this way a pressure $p_2$ is set up between the differential pressure valve and the restriction, which balances the spring force at the actuator. The resulting position of the throttle valve ensures that the quantity of fuel and the quantity of air are in the desired ratio.

If, for example, the rate of fuel flow of the engine is increased, then there is a greater pressure drop at the slit throttle of the fuel-flow sensor. The diaphragm of the fuel-flow sensor opens a larger cross-section. In this way the pressure at the actuator is increased. The throttle valve is now opened until the rate of air flow of the engine has again reached the level assigned to the new quantity of fuel.

Another EGR system has been tested on a 3 liter 4 cylinder Toyota engine. Relatively low emissions have been obtained, but not within limits fixed. The following values have been obtained by adding a catalytic exhaust muffler:

| HC gm/mile | CO gm/mile | $NO_x$ gm/mile | Inertia weight lb |
|---|---|---|---|
| 0.30 | 0.75 | 0.50 | 2250 |
| 0.30 | 0.95 | 0.75 | 3500 |

Toyota's EGR system consists of varying the percentage of EGR versus the vacuum created by an engine driven pump. A valve adjusts passage of exhaust gases and maximum value is set by a calibrated aperture on the manifold.[16]

Volkswagen has carried out in-depth research work on emissions from diesel powered light vehicles.[22] Their work is summarized below.

| Inertia Weight | Engine | HC | CO | $NO_x$ |
|---|---|---|---|---|
| 1750 | N.A. + EGR | 0.45 | 2.5 | 0.4 |
| 2000 | " | 0.45 | 2.5 | 0.46 |
| 2250 | " | 0.40 | 2.5 | 0.47 |

## 4.6 ELECTRONIC CONTROL SYSTEM

Emission limits also require more precise and sophisticated controls than currently available. There is an urgent need to know all parameters involving emissions by means of electronic control which monitors injection pump adjustment, turbocharging, EGR, etc. It is unknown whether prototype systems exist which can satisfy the above requirements.

## 4.7 TURBOCHARGING

Turbocharging has been applied extensively in heavy duty diesels to increase rated power. Turbocharging has recently been considered for light duty vehicles to increase engine delivery and vehicle fuel economy because of more favorable working conditions.

Turbocharging does, however, pose some problems:

1) Duration, because of increased heat
2) High combustion chamber pressure

3) High levels of smoke at low r.p.m.

Several solutions have been studied, the main being:

1) Piston cooling: with oil under pressure to a cavity in the inside face of the piston top.

2) Inter cooling: with a heat exchanger.

3) Waste gate: by reducing gas flow to the turbine at high r.p.m. where excess air is produced, to ensure air flow at low r.p.m. This requires exact adjustment of fuel injection.

4) LDA: pump pressure adjustment versus the inlet manifold air turbocharging pressure has been adopted on Bosch VE pumps.
Air pressure acts on a membrane attached to a cam which varies pump delivery.
The results obtained on HC/CO/$NO_x$ emissions from a VW engine can be summarized as follows:[14]

| HP | Engine type | HC gm/mi | CO gm/mi | $NO_x$ gm/mi |
|---|---|---|---|---|
| 50 | NA | 0.16 | 1.0 | 1.2 |
| 70 | TC | 0.11 | 0.8 | 0.9 |

Particular attention should be given to fuel economy. A gain of some 10 percent is in fact obtained with turbocharging.[14] The main advantage, however, is reduced particulates, the mechanism of which is not yet well know.

Particulate reductions have been obtained by VW as follows:

| HP | Engine type | Partic. Emissions gm/mi |
|---|---|---|
| 50 | NA | 0.35 |
| 70 | TC | 0.2 |

A 35 percent reduction[12] has also been obtained on a 2.5 liter Fiat engine. Particulate levels increase again from 0.35 to 0.45 g/mile when E612 (turbocharged versus naturally aspirated) is used with turbocharging.

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