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A REVIEW OF ADVANCED VEHICULAR DIESEL RESEARCH AND DEVELOPMENT PROGRAMS WHICH HAVE POTENTIAL APPLICATION TO STATIONARY DIESEL POWER PLANTS

Energy Conversion Branch
Aerospace Power Division (POOC)

March 1980

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AERO PROPULSION LABORATORY
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This report, prepared for the Aerospace Power Division, Aero Propulsion Laboratory, Wright-Patterson AFB, reviews, assesses, and summarizes the research and development status of advanced diesel engine/vehicular component technologies, and identifies those systems which may have application to diesel power plants utilized as stationary engine power sources.

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The reviewed R&D projects include: variable area turbocharging, variable compression ratio piston, high pressure fuel injection systems, turbocompounding, the adiabatic engine, and the Rankine bottoming cycle.

Several of the development programs reviewed, when combined into one power plant, offer the best potential for diesel stationary power plants. The turbocompound, Rankine bottoming cycle, adiabatic engine potentially can achieve thermal efficiency of 50-60% thereby achieving a dramatic increase over today's diesel engines.

FOREWORD

This report, prepared for the Aerospace Power Division, Aero Propulsion Laboratory, Wright-Patterson AFB, reviews, assesses, and summarizes the research and development status of advanced diesel engine/vehicular component technologies, and identifies those systems which may have application to diesel power plants used as stationary engine power sources.

The approach adopted to complete this report was to draw on information published in the open literature. In particular, material presented at recent Department of Energy (DOE) contractors coordination meetings, joint Department of Transportation/Transportation System Center (DOT/TSC) reports, and DOT reports provided substantial inputs to this report. This information was aided by discussions with recognized experts in the field, and by a review of many Society of Automotive Engineers (SAE) technical papers and other available technical reports.

This report was submitted by the Aero Propulsion Laboratory, under project/task/work unit 3145/24/02, with Richard G. Honeywell, 2Lt, USAF, as project engineer. Andrew W. Kaupert, Capt, USAF, of the Detroit Detachment, ASD Reserves (XOOR), Wright-Patterson AFB, Ohio, is technically responsible for the work.

TABLE OF CONTENTS

TABLE OF CONTENTS (Cont'd)

LIST OF ILLUSTRATIONS

LIST OF TABLES

EXECUTIVE SUMMARY

Because of the ever increasing scarcity of fossil fuels, the USAF Energy Conversion Branch of the Aero Propulsion Laboratory has intensified its efforts to review existing and future potential energy conversion power plants for application to stationary power units. As part of this analysis, this report reviews the advanced diesel engine research and development activities conducted by both government agencies and industry. This report identifies several advanced diesel power components and total engine systems which merit continued monitoring by the USAF.

The reviewed R&D projects include: variable area turbocharging, variable compression ratio piston, high pressure fuel injection systems, turbocompounding, the adiabatic engine, and the Rankine bottoming cycle. With the exception of the adiabatic engine, the other projects are component development programs designed to improve the power output with a corresponding reduction in the brake specific fuel consumption (bsfc) of the basic engine. The component development programs are of such a nature that when these components are taken to maturity during the mid to late 1980's, significant thermal energy improvements ranging from 5 to 15% can be achieved. These thermal energy improvements can be directly translated into fuel savings, smaller power-plant physical sizes, and concomitant savings of other types.

Several of the development programs reviewed, when combined into one power plant, offer an excellent potential for diesel stationary power plants. The component development programs are described in detail in the report. In summary, the turbocompound, Rankine bottoming cycle, adiabatic engine looms as the most thermal efficient diesel engine proposed to date. This system has been projected to achieve brake specific fuel consumption which translates into thermal efficiency of 50-60%. This is a dramatic improvement over today's diesel thermal efficiency, which ranges from 30 to 40%. Additionally, the increased thermal efficiency of the adiabatic engine may allow the diesel engine to achieve its potential as a multifuel engine. Remember, Rudolph Diesel's first experiments leading to the

development of the compression ignition cycle were with an air-fuel mixture designed to use powdered coal. Although the adiabatic engine may not be able to go to these extremes, it could use a broader range of fossil fuels than is now mandated for diesel engines.

SECTION I

INTRODUCTION

During late 1977, the Aerospace Power Division of the Aero Propulsion Laboratory conducted a study assessing the United States Air Force terrestrial energy requirements for the post-1985 time frame. The results of this study culminated in a report entitled "USAF Terrestrial Energy Study (Reference 1).

The Aerospace Power Division plans to expand the terrestrial energy study by determining the market penetration of advanced power systems in the post-1985 period and ascertaining the types of power systems which will be needed to meet future USAF auxiliary, emergency, and remote site requirements.

It has long been recognized that diesel engines have the capability of achieving high engine output and low exhaust pollutants with excellent fuel consumption, reliability, and durability characteristics. While stationary diesel power plant development has been steadily progressing, interest in the diesel as an alternative power plant for combat vehicles has also steadily increased over the past few years. Because of the basic combustion/configuration similarity of diesel engines for both stationary and vehicular applications, this study will concentrate on those vehicular advanced diesel power plant/component developments which may have potential for stationary power sources. Comments will also be made on the applicability or desirability of a technology transfer from vehicular application to a stationary electrical generation power plant.

The principal objective of this study was to review, assess, and summarize the current status of advanced diesel engine-related research and development efforts conducted or planned by industry and government agencies. The second objective was to identify those areas in which government support is needed to develop the information required for a comprehensive assessment of the future potential of diesel engines for electrical generation.

The approach adopted to accomplish these study objectives was to draw on information published in the open literature. In particular, the presentations reported by industry and government representatives at several DOE contractor coordination meetings, DOT/TSC presentations and DOT contractor reports. The information was supplemented by (1) telecommunications with individuals from several government agencies, (2) an examination of reports made by industry representatives to the Environmental Protection Agency (EPA), and (3) a review of many pertinent SAE papers and other reports. It should be noted, however, that technology-related programs conducted by industry are frequently considered to be proprietary, and therefore were not included in this study.

SECTION II

ADVANCED DIESEL RESEARCH AND DEVELOPMENT

Several government agencies are currently involved in a variety of in-house and contracted programs in the areas of diesel engine technology research and development. These include the Department of Transportation, Environmental Protection Agency, U.S. Army, U.S. Bureau of Mines, and the Department of Energy. Also, a number of automobile manufacturers, both domestic and foreign, have active light duty diesel engine technology development programs. Domestically, Chrysler, Ford, and General Motors have active light duty diesel engine and diesel engine technology development programs. Foreign manufacturers include British Leyland, Citreon, Daimler-Benz, Fiat, Peugeot, and Volkswagon.

The U.S. Army Tank/Automotive Research and Development Command (USATARADCOM) has been involved in diesel R&D programs for many years. The principal objective of these programs is to improve power train efficiency in military propulsion systems. The work sponsored by TARADCOM has maintained and supported, sometime solely, many of the advanced diesel programs now vital to progress for reduction of fuel consumption and exhaust emissions. Although it might be argued that those systems necessary for a high power, quick, mobile combat vehicle may be inappropriate for stationary power sources, the technology developed has the potential for transfer to stationary power plants with concomitant results.

1. VARIABLE AREA TURBOCHARGING

Turbocharging is a technological process that could be used extensively in the 1980's and offers considerable advantages for weight reduction, and improved fuel economy and acceleration performance. The primary benefits of variable area turbochargers are higher power outputs with the same engine displacement and improvements in efficiency relative to fixed-area turbocharging (Reference 2). Besides providing better fuel economy, turbocharging also has the potential for lowering the exhaust emission levels of diesel engines. It should be noted, however, that many electrical power generator diesel engine sets are now turbocharged and, therefore, the impact of turbocharging may not be as substantial for stationary units.

But, turbocharging is not without problems. The response time of turbocharging during low speed acceleration is still a major problem which limits the use of the engine full throttle torque capability. Various schemes are under development to reduce this disadvantage. These schemes include reducing compressor and turbine rotor inertia, using a pelton wheel or burners, electronic feedback systems, and variable area turbocharging. Other turbocharging disadvantages include higher thermal loading on pistons and the correct sensing of inlet manifold pressure to reduce over-fueling problems.

a. System Description

The horsepower available from a diesel engine is a direct function of the amount of air and fuel available for combustion in the cylinders at that given instant of time. For a turbocharged engine, the airflow is primarily a function of turbocharger speed. By maintaining high turbocharger speeds at all engine speeds, it is possible to substantially increase the airflow and the instantaneously available horsepower.

However, if turbocharger speed is maximized at a low engine speed, it will overboost the engine at high engine speeds. This results in either excessive structural loads on the engine or failure of the turbocharger by overspeeding. The approaches proposed to overcome these limitations include "wastegating" (bypassing the exhaust energy around the turbine) and using exhaust augmenters or combustors (wasteful of fuel, costly, and complex), and the variable area turbocharger (VAT).

An approach using the VAT involves adjusting the turbine stator (nozzle) flow area as a function of engine speed. In this way it is possible to maintain high turbine speed at low engine speeds by reducing the nozzle area. As the engine airflow rate rises with engine speed, the nozzle area is opened to prevent over-boosting. By this technique it should be possible to maintain a constant turbocharger speed (and thus intake manifold pressure) regardless of engine speed.

Additionally, the compressor side of the turbocharger is also constrained by some stringent limitations. High pressure ratio centrifugal compressors are limited in flow range. The flow range is restricted between surge and choke, which can occur in either the diffusor or inducer. However, choking flow limits can be increased if the area which is choked can be increased, and the surge flow limits can be reduced if the area where surge occurs is reduced. These are precisely the goals of the VAT (Reference 3).

b. Factors Considered

The factors the author considered concerning the potential for this system are listed in Table 1. These factors are the basis for the projection concerning the application of this system to the stationary diesel power plant. Additionally, an attempt has been made to quantify the impact of the described components in terms similar to the USAF Terrestrial Energy Study (Reference 1). Because of the nature of this report, it is not possible to exactly duplicate the methodology of the Terrestrial Energy Study. This stems from the fact that this discussion centers on components rather than engine systems, the advanced development nature of these components, and the necessity for brevity. However, an attempt to follow the Terrestrial Energy Study methodology has been constructed and can be seen in Tables 2, 3, and 4.

c. Potential for Stationary Powerplants

As denoted in Table 1, VAT has the potential for improved brake specific fuel consumption, lower emissions and signature, and improved engine response with little or no weight, size, or cost penalty. These potentials may be laudable goals for a mobile diesel engine, but are these goals essential for stationary diesel engines primarily designed for electrical generation?

The answer must be yes! Stationary power units will be affected by the world shortages of liquid fossil fuels as well as the mobile sources. VAT offers the potential, as noted in Tables 1 through 4, to minimize or lessen the impact of these shortages. Additionally, VAT provides quicker

TABLE 1

VAT FACTORS CONSIDERED

FACTOR	COMMENTS
Materials	Uses common turbocharger materials with some exotic materials needed for high speed rotational parts (Reference 2)
Costs	More costly than presently available turbochargers, but less costly than advanced aircraft turbo machinery (Reference 4)
Development	Bench tested hardware, needs further engine testing and durability testing.
Weight	Minimal additional weight.
Production Capability	Needs to be determined.
When Producible	Mid to late 1980's (Reference 5)
Probability of Success	Good to excellent (Reference 4)
Potential Benefits	Increased horsepower with no increase of size or weight, faster transient response, and reduced engine signature and emissions.

TABLE 2

PERFORMANCE PARAMETER DEFINITIONS

A. Engine Efficiency - The described components's impact on total engine efficiency is defined as the rate of energy output divided by the rate of energy input.

B. Reliability - A qualitative and quantitative estimate of the engine components' "on-line" availability for operational use over the designated lifetime and duty cycle.

C. Lifetime - An estimated number of years the engine/component system is expected to produce its designed power output.

D. Operation and Maintenance - A qualitative estimate of the power system's ease of operation and maintenance, including technical expertise necessary for operation and maintenance.

E. Component Growth Potential - Indicates the capability of the component described to be scaled-up or scaled-down to impact the rated power output of applicable engines.

F. Start-Up/Shut-down Time - Estimates the impact of the described component on the engine's ability to go from a ready to start condition to a full power steady condition. Shutdown time refers to the time required to bring a system from a full power operation to an off or standby/start mode.

G. Thermal Energy Available - An approximation of the amount of currently wasted thermal energy which may be recoverable from the total engine/component system.

H. Fuel Consumption - Impact of described component on fuel consumption.

I. Volume/Size - An estimate of the described component's impact on the land area occupied for a fixed engine system and the volume of the system's envelope which indicates the amount of space required for transportation of the engine system.

J. Weight - An estimate of the described component's impact on system weight.

TABLE 2 (Cont'd)

K. Environmental Contraints - The six types of environmental impacts considered are thermal discharge, air pollution, noise, solid waste, chemical discharge, and radioactive wastes.

L. Acquisition Cost - An estimate of the described component's impact on total system costs or an estimated cost for the component itself.

M. Life Cycle Cost - An estimate of life cycle costs including the total cost incurred during the lifetime of the engine/component system.

TABLE 3

ADVANCED DIESEL ENGINE SYSTEMS (VAT)

Performance Parameter	Estimated System Impact vs. Present Diesel Systems
A. Engine Efficiency	Improved engine efficiency by 5-20%.
B. Reliability	Over 700 hrs shown to date; should represent very little difference to present systems (Reference 3)
C. Lifetime	No difference; 5 yrs between overhauls.
D. Operation and Maintenance	No different than today's TCs, no special expertise needed.
E. Component Growth Potential	Can be scaled up and down with corresponding increase or decrease in power and torque.
F. Start-Up/Shutdown Time	Improved start-up time, no difference in shutdown time (Reference 3).
G. Thermal Energy Available	Reduced thermal energy available due to improved TC* energy utilization (Reference 3)
H. Fuel Consumption	Reduced 5-10% (Reference 3)
I. Volume/Size	No impact on size or volume of TC units.
J. Weight	Slight increase in weight, approximately 30-50 lbs depending on VAT size (Reference 4).
K. Environmental Constraints	Lower smoke levels (<3% opacity); assumed lower emission levels; lower thermal discharge; increase in noise levels; no solid, chemical or radioactive wastes.
L. Acquisition Cost	Present costs are 1500-3000 dollars per KW due to low production volumes - reduced to approximately 300 dollars per KW with volume production.
N. Life Cycle Cost	Life cycle cost should be reduced due to improved fuel consumption, no special maintenance necessary.

*TC - Turbocharger

TABLE 4

ADVANCED DIESEL SYSTEM (VAT)

Engine Size	Estimated Potential Introduction Date and Application				
	1980	1985	1990	1995	2000
A. 50 MW Cont	If diesel engines of this size were to be developed VAT would be an integral part of development				
B. " " 1 hr	NA	X	X	X	X
C. 10 MW Cont	NA	X	X	X	X
D. " " 8 hr	NA	X	X	X	X
E. " " 1 hr	NA	X	X	X	X
F. 750 KW Cont	NA	X	X	X	X
G. 250 KW Cont	NA	X	X	X	X
H. 50 KW Cont	X*	X	X	X	X
I. " " 8 hr	X	X	X	X	X
J. " " 1 hr	X	X	X	X	X
K. 10 KW Cont	X	X	X	X	X
L. " " 8 hr	X	X	X	X	X
M. " " 1 hr	X	X	X	X	X

X *Indicates availability*
* Approximately 1982 to 1983

start-up times for diesel stationary units, increased horsepower while maintaining size/weight configuration, and should reach maturity just as the United States Air Force will be seeking new stationary power sources. This system will help make the diesel engine a worthy competitor to other advanced power systems.

2. VARIABLE COMPRESSION RATIO PISTON

Nearly all of the research and development effort expended on the variable compression ratio (VCR) piston has been funded by the United States Army. The principal objective of the VCR piston development program was to provide a fully developed and tested engine system for a vehicular application (Reference 6). The Army application required a diesel engine rated at 1500 gross brake horsepower at 2600 rpm with a high power-to-weight ratio, exceptionally high instantaneous horsepower characteristic, and low transient smoke.

The aforementioned Army application was the XM-1 main battle tank. The initial design qualification and test run-off resulted in the VCR piston engine (AVCR 1360) losing to the AGT-1500 gas turbine powered version of the tank. The Army has a modestly funded program to continue component improvements on the VCR piston engine.

Until the 1975 time frame, DOE considered the VCR piston as a viable candidate in their heat engine program and they provided modest funding to further evaluate the VCR piston potential (Reference 7).

Teledyne Continental Motors conducted an extensive study of the VCR piston during 1976 (Reference 8). Teledyne Continental concluded that the VCR diesel engine could maintain the fuel economy and durability of the diesel while providing excellent performance, light weight, reduced bulk, low cost, excellent cold starting, reduced noise level, and very low emission levels. Additionally, this engine could have the capabilities of being converted from an existing gasoline engine using the same basic engine structure and burning a broad range of fuels.

Although this Teledyne Continental Motors study was essentially a paper study and mathematical predictive model, Teledyne concluded that the VCR concept coupled with a turbocharger, exhaust gas recirculation and pre-chamber diesel engine could achieve the lowest emission levels proposed by the EPA for vehicular mobile sources. Verification of these predictions needs to be completed. Work on the VCR piston concept has also been conducted by the General Motors Corporation. The International Harvester Company has conducted basic research under funding by the U.S. Army.

a. System Description

The VCR piston principle is essentially an automatic, hydraulically actuated piston assembly. The assembly consists of two main parts: the piston ring carrier and the piston pin carrier. See Figure 1.

The piston pin carrier, attached to the connecting rod, travels a definite path, but, the piston ring carrier is free to move within predetermined limits relative to the pin carrier. This movement provides a variable height from the center of the piston pin to the top of the piston crown. The variation in compression ratio is achieved through a change in combustion chamber clearance volume. Movement of the piston ring carrier is restrained hydraulically by engine lubricating oil in the upper and lower chambers.

Oil from the lubricating system is fed through the connecting rod to the non-return piston oil supply valve by the piston oil collector. As the upper chamber fills with oil, the piston ring carrier moves to its extreme position or maximum compression ratio. Oil also enters the lower chamber through the orifice connecting the two chambers.

The piston remains in the high compression ratio mode until power demand increases and the force of combustion pressure causes the pre-set oil discharge value to release oil from the upper chamber. The VCR piston regulates combustion pressure until the low compression ratio is reached.

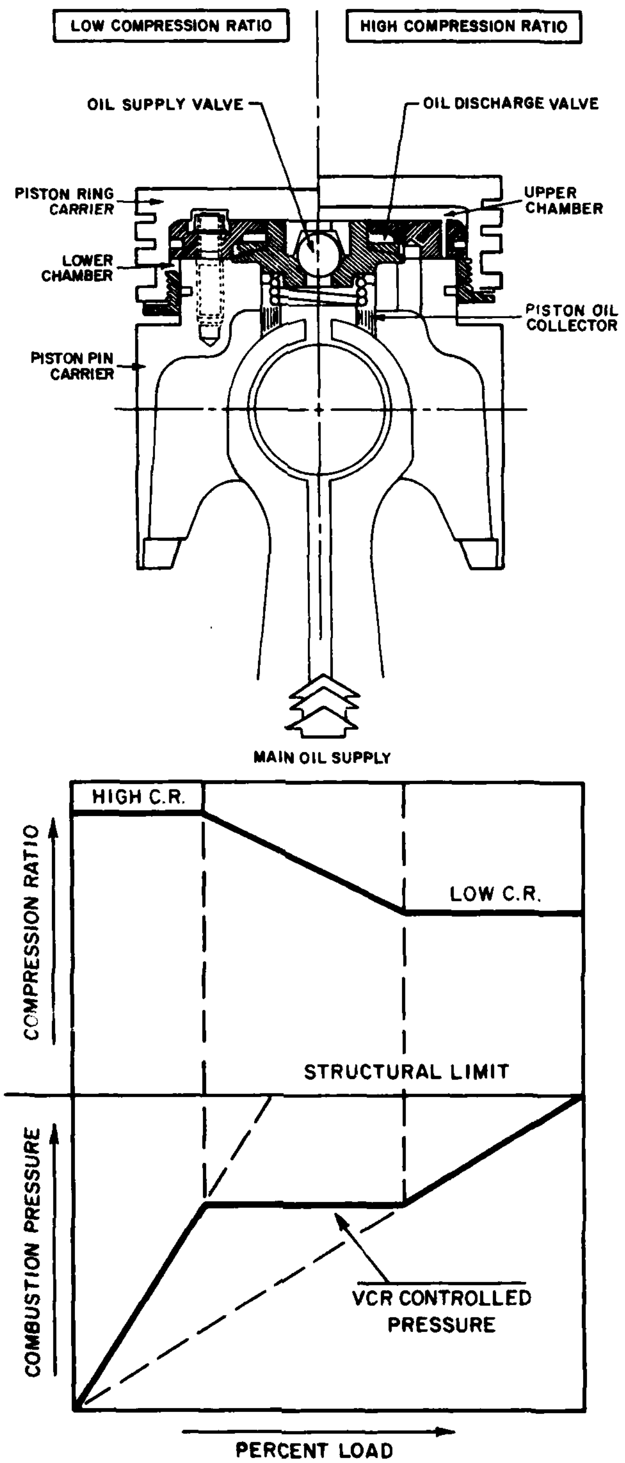


Figure 1. VCR Piston Principle (from Reference 8)

The high compression ratio mode aids in starting and in low power operation where a high compression ratio is essential. The low compression ratio is essential at high load for low fuel consumption, lower frictional load, and low exhaust emissions.

Inertia forces tend to force the ring carrier to move upwards during the end of the exhaust and the beginning of the intake strokes.

Oil in the lower chamber, which has a fixed orifice to the upper chamber, controls the amount of movement between the piston pin carrier and ring carrier during the exhaust and intake strokes. As combustion pressures are reduced, the ring carrier returns to the higher ratio until equilibrium is achieved.

b. Factors Considered

The factors the author considered in evaluating the future potential of the VCR piston are listed in Table 5. The potential for application to stationary diesel power plants are detailed in the following section. The Terrestrial Energy Study type of comparisons are shown in Tables 6 and 7.

c. Potential for Stationary Power Plants

Although the VCR piston does not have the immediate potential for application to stationary power sources, the potential improvements indicated in the aforementioned figures dictate further monitoring of the VCR piston concept. Because of the low development effort being expended currently, due partly to gas turbine interest by the U.S. Army and the need for further definition of the United States energy policy, the VCR piston engine is projected to be a viable stationary source candidate during the mid 1980's. There should be no difficulty in translating or transferring the technology to large bore, slower speed stationary diesel engines. The VCR piston development program should be continually monitored to determine when the VCR piston is applicable to stationary power sources.

TABLE 5

VCR PISTON FACTORS CONSIDERED

FACTOR	COMMENTS
Materials	Utilizes cast iron piston. Some need for stainless steel for oil filtration (Reference 9).
Development	Completed military qualification tests. Need to be productionized and further compression ratio cycling tests need to be conducted. Could be produced on gasoline engine production lines thereby saving cost (Reference 8).
Costs	Modest cost increase above conventional cast iron piston but has potential to reduce total materials costs due to reduced structural requirements.
Size and Weight	Reduces both.
Production Capability	Good.
When Producible	Early 80's for smaller size engines; mid 80's for larger engines.
Probability of Success	Good to excellent (Reference 8).
Potential Benefits	Lower exhaust emissions and signature, lower fuel consumption, reduced size and weight, improved startability, multifuel capability, lower noise emissions.

TABLE 6

ADVANCED DIESEL ENGINE SYSTEM (VCR PISTON)

Performance Parameter	Estimated System Impact vs. Present Diesel Systems
A. Engine Efficiency	Improve total efficiency 5-10%.
B. Reliability	As reliable as today's systems but needs further definition (Reference 10).
C. Lifetime	No change - 15-20 years.
D. Operation and Maintenance	No difference from present systems, needs no additional expertise.
E. Component Growth Potential	Will have no growth difficulties in mid-range engines; may have difficulty in large bore engines.
F. Start-Up/Shutdown Time	Improved start-up and no difference on shutdown; transient operation needs definition (Reference 10).
G. Thermal Energy Available	Reduced thermal energy due to improved combustion efficiency over load range.
H. Fuel Consumption	Lower (10-15%) than present systems.
I. Volume/Size	No effects, replace present system one-for-one.
J. Weight	Approximately 7.0 lbs more than piston replaced.
K. Environmental Constraints	Reduced thermal discharge, air pollution, and noise. No solid waste, chemical discharge or radioactive wastes (Reference 10).
L. Acquisition Cost	Present cost for VCR pistons is approximately $500/piston. Cost can be reduced with increased production volume.
M. Life Cycle Cost	Marginal gains - increased fuel economy partially offset by decreased oil change intervals.

TABLE 7

ADVANCED DIESEL SYSTEM (VCR PISTON)

Engine Size	Estimated Potential Introduction Date & Application				
	1980	1985	1990	1995	2000
A. 50 MW Cont	NA	NA	NA	NA	NA
B. " 1 hr	NA	NA	NA	NA	NA
C. 10 MW Cont	NA	?	X	X	X
D. " 8 hr	NA	?	X	X	X
E. " 1 hr	NA	?	X	X	X
F. 750 KW Cont	NA	X	X	X	X
G. 250 KW Cont	NA	X	X	X	X
H. 50 KW Cont	NA	X	X	X	X
I. " 8 hr	NA	X	X	X	X
J. " 1 hr	NA	X	X	X	X
K. 10 KW Cont	X*	X	X	X	X
L. " 8 hr	X	X	X	X	X
M. " 1 hr	X	X	X	X	X

X Indicates Availability
* Approximately 1982 to 1983

3. HIGH PRESSURE FUEL INJECTION SYSTEM

Historically, fuel preparation, distribution, and shutoff have been the weak point of diesel engine combustion. Yet, in addition to air utilization, fuel injection system parameters are the most critical and the most poorly understood area of diesel combustion. However, the criticality of fuel injection has become apparent to researchers in the field of diesel combustion; and in the past several years, research into fuel systems and fuel system impacts on exhaust emissions and consumption have increased dramatically.

The fuel injection system (fuel pump and injectors) for high speed diesel engines is important in the design, operation, and performance of the engine. The fuel injection equipment must break up the fuel into small droplets or "atomize" the fuel by the injection process and spray the fuel within the combustion chamber in a penetration pattern which is dependent on the size of the injector opening (orifice or holes) and the opening orientation.

Presently there are four basic hydraulic/mechanical types of fuel injection systems, and modifications thereof, in use on high RPM diesels. These systems include (a) pressure-time fuel system, (b) multiple pump system, (c) unit injector system, and (d) distributor system.

The pressure-time fuel systems use injectors that meter and inject the fuel into the combustion chamber. The pressure at the injector is supplied by a low pressure fuel feed pump. Fuel metering is determined by how long the injector metering orifice remains open, an interval determined by the engine rotational speed via the camshaft-driven injector plunger which forces fuel into the cylinder at a pressure higher than compression pressure.

The multiple pump system uses individual plungers for metering and injecting fuel into each engine cylinder. These pumps are remotely mounted and are connected to the injectors, which are located in the cylinder head, by long high pressure fuel lines. The advantage of the multiple pump system lies in its compactness, since all functions are combined in a separate pump for each cylinder. However, the cost is high because each pump on a multicylinder unit must precisely match its companions at all operating speeds and loads.

The unit injector system locates both the metering and injection functions in the injector. The plunger pumps are similar to those of the multiple pump system but are positioned in the individual injectors, thus eliminating the use of high pressure fuel lines. Rotation of the plunger by the fuel rack controls the metering.

The distributor system uses a single-plunger high pressure metering pump to meter the fuel for all cylinders. The governor and throttle control the pump stroke length and thus the amount of fuel delivered on each stroke. Distribution to the cylinders is controlled by a rotating disc or shaft with holes located to synchronize the metering pump delivery stroke with the various cylinders.

The foregoing mechanical design arrangements and control mechanisms in use today represent a compromise between the ideal of instant and precise response to load demand variation for each cylinder under all operating conditions of atmospheric and fuel quality, and the practical considerations of injection equipment cost, serviceability, and maintainability of precision equipment. Wear of mechanical fuel injection equipment, including cams, roller followers, plungers and barrels, governor linkages, and control rods and bearings, all contribute to the need for regular repair and maintenance by skilled personnel to avoid damage, performance loss, and excessive exhaust emissions (Reference 10).

Requirements on the injection equipment of the diesel engine are mainly dependent on combustion system type. This is illustrated in Figure 2 where typical fuel injection pump injection rates are shown as a function of peak injection pressure for various diesel engines in the power range between 50 and 500 horsepower (Reference 11).

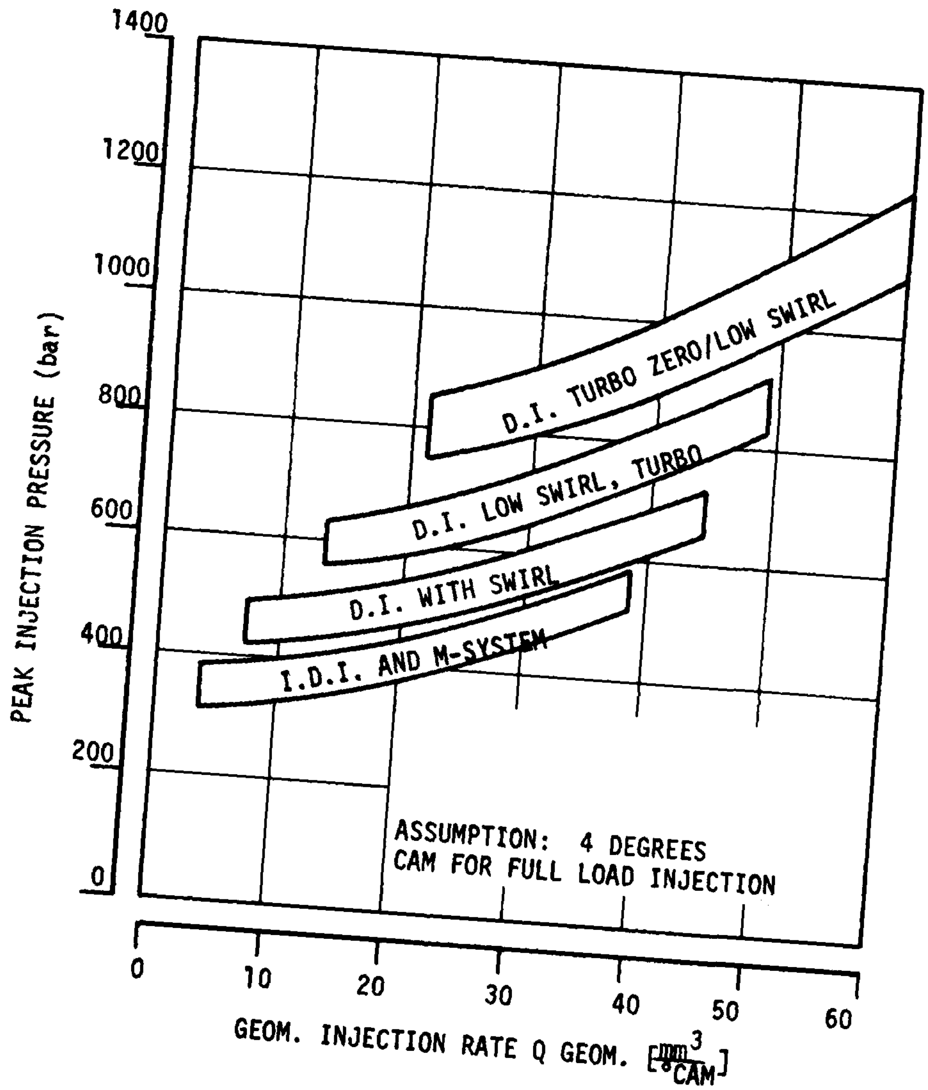


Figure 2. Relationship of Combustion System, Injection Pressure, and Injection Rate (from Reference 11)

The acronym UFIS means Universal Fuel Injection System. As the name implies, the UFIS was targeted to be applicable to any and all types of combustion chamber designs or configurations. The development of the system originated in 1968 with the U.S. Army Tank Automotive Command (USATARADCOM's predecessor) and American Bosch Division of AMBAC Industries with the goal of duplicating the injection characteristics required of any of the known diesel combustion configurations.

For a four cycle engine, the specific requirements for a UFIS injection unit were 500 mm^3/stroke, 25° crank shaft duration at 3000 RPM (1.38 milliseconds), and the ability to vary the rate of injection as a function of injection duration. It should be noted that at that time (1968) there was very little emphasis on gaseous emissions or noise. American Bosch developed a rail type unit injector to meet the above requirements. The system has (1) hydraulic pressure amplification, (2) positive displacement low pressure metering, and (3) electronic control of injection timing. With such a system it is possible to perform research work on an engine by adjusting timing, injection quantity, and injection pressure while the engine is running. Once the requirements of the engine are determined, the control can be programmed for automatic operation.

Engine testing with this new fuel injection concept started in 1970. With the added emphasis on gaseous emissions, several new injection characteristics had to be considered above and beyond the original concept. The system was redesigned to include these new characteristics which included higher injection pressure and various combinations of nozzle valve control. Since the initial U.S. Army funding, several industrial firms have shown an interest in the UFIS because of its inherent potential for reduced fuel consumption and exhaust emissions. These firms included a medium sized manufacturer of diesel engines and a farm machinery manufacturer. Several of the research versions of the UFIS are located in government research facilities and universities. To date no production units have been developed, although production quantities are contemplated.

a. System Description

Figure 3 is a block diagram of the UFIS. To the left of the vertical dotted line are the components housed in an engine-driven control unit. To the right of the dotted line is the injector. There is one injector for each cylinder.

The control unit consists of a low pressure pump, an intermediate pressure pump, a timer control, a fuel motor, and a governor. The electronic control circuits are mounted off the engine.

Each injector consists of a magnetic servo, a hydraulic servo, a hydraulic amplifier, and a nozzle.

When the UFIS is used as a research tool to study combustion, the components of the control unit are off-engine mounted with the exception of the meter and timer which are engine driven.

The low pressure pump output (typically 200 psi) is fed to a positive displacement, shuttle-type meter. The metered output from the shuttle is directed to the appropriate injector by a distributor shaft running at half engine speed. The distributor shaft is timed to the engine such that the charge of metered fuel is introduced into the injector anytime during the 690° (345° in the case of 2 cycles) rotation of the crankshaft between injections.

The intermediate pressure pump (adjustable from 1500 to 3000 psi) supplies the driving (rail) fuel for the injectors. The fuel is prevented from entering the injector by the hydraulic servo valve until the time for injection. The timing control, which is engine driven, provides an electrical signal to the magnetic servo valve to start injection. Opening the magnetic valve opens the hydraulic servo valve, thus letting the intermediate pressure drive fuel into the hydraulic amplifier. The metered fuel which has been stored in the injector is pressurized by the hydraulic amplifier and fed to the nozzle under high pressure (9000 to 18,000 psi typical). Injection ends when the metered charge has been displaced from

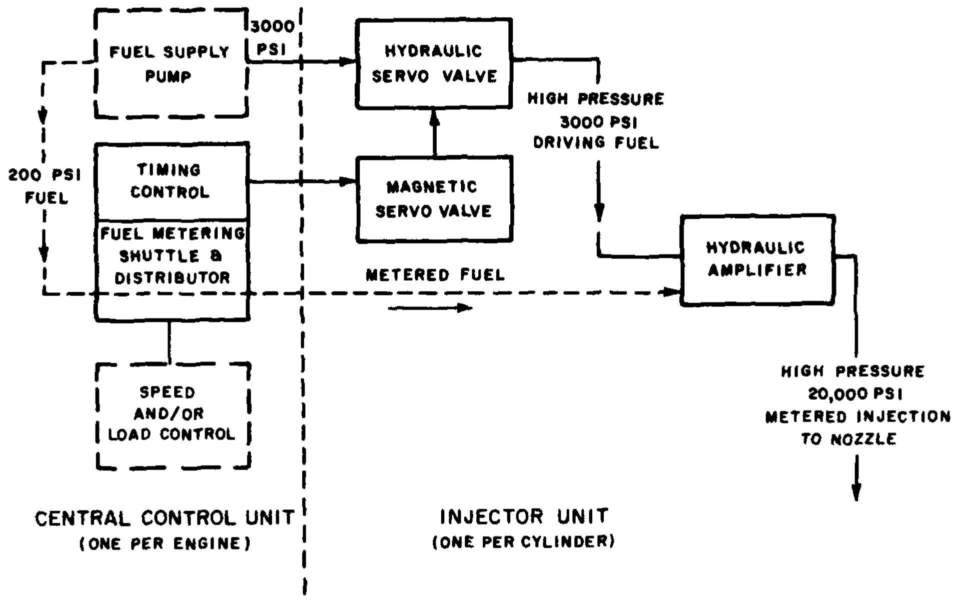


Figure 3. Block Diagram of UFIS System

under the injection piston. At no time is there any interconnection between the driving fuel and the metered fuel.

Three injector sizes have been developed for a wide range of applications. They range in size from 300 mm^3/stroke to 2400 mm^3/stroke. They are being used to study engines rated from 50 horsepower to 4000 horsepower and with 2 to 16 cylinders per engine.

b. Factors Considered

Evaluation of the factors considered are shown in Tables 8 through 10. Because of the proprietary nature of the UFIS, some of the factors considered reflect the state-of-the-art as known from the U.S. Army program. Estimates of the system's potential are given based on production quantities.

TABLE 8

UFIS FACTORS CONSIDERED

FACTOR	COMMENTS
Materials	Research units are high strength, exotic materials. Extensive work is needed to develop production units.
Development	Research, pre-production units are available. Production units are underway but at what levels are not known.
Costs	Present costs are prohibitive. Production costs will be higher than conventional units but may be recoverable due to lower fuel consumption.
Size and Weight	Both are increased with research units - production attempts directed at reducing both (Reference 4).
Production Capability	Research units - poor to good. Production units remain to be determined.
When Producible	Research units available now, production is targeted for mid 1980's.
Probability of Success	Moderate - depends on manufacturers desire to meet governmental regulations and energy usage reductions.
Potential Benefits	Lower exhaust emissions, lower fuel consumption, increased power (assuming good air utilization), improved engine performance.

TABLE 9

ADVANCED DIESEL ENGINE SYSTEMS (UFIS)

Performance Parameter	Estimated System Impact
A. Engine Efficiency	Improved efficiency up to 20% depending on combustion system and cycle.
B. Reliability	Needs to be determined, will be designed to be comparable to or exceed present system.
C. Lifetime	Units will be designed to meet or exceed current engine lifetimes - 21 years.
D. Operation & Maintenance	The system will be as easy to operate as present systems - repairs will necessitate a higher skill level than presently available.
E. Component Growth Potential	Good growth potential ranges from 300 mm^3/stroke with 10:1 turn down ratio (Reference 12).
F. Start-Up/Shutdown Time	Improve start-up time - may improve shutdown due to rapid shutoff of fuel.
G. Thermal Energy Available	Improved cycle efficiency hence 5-15% less thermal energy available.
H. Fuel Consumption	Improved - conservatively 5-20% depending on combustion system.
I. Volume/Size	Increased engine height 12-18 inches - to improve with production; 40-60 lbs more than present systems.
J. Environmental Constraints	Reduction in exhaust emissions, noise and thermal discharge. No other effects.
K. Acquisition Cost	Present unit cost $5000 per cylinder - target for production $100/cylinder.
L. Life Cycle Cost	Slightly higher if production volume doesn't increase (initial costs will not be offset by lower fuel consumption). Assumed 10-20% LCC increase if production volume doesn't increase.

TABLE 10

ADVANCED DIESEL SYSTEM (UFIS)

Engine Size	Estimated Potential Introduction Date & Application				
	1980	1985	1990	1995	2000
A. 50 MW Cont	NA	NA	NA	NA	NA
B. " " 1 Hr	NA	NA	NA	NA	NA
C. 10 MW Cont	NA	X	X	X	X
D. " " 8 Hr	NA	X	X	X	X
E. " " 1 Hr	NA	X	X	X	X
F. 750 KW Cont	X*	X	X	X	X
G. 250 KW Cont	X*	X	X	X	X
H. 50 KW Cont	X*	X	X	X	X
I. " " 8 Hr	X*	X	X	X	X
J. " " 1 Hr	X*	X	X	X	X
K. 10 KW Cont	X*	X	X	X	X
L. " " 8 Hr	X*	X	X	X	X
M. " " 1 Hr	X*	X	X	X	X

X Indicates Availability
* Available 1982 - 1983

c. Potential for Stationary Power Plants

It is apparent that a UFIS could be applied directly to a stationary engine now. The work with the 4000 horsepower engine could be translated directly to a stationary power plant engine. It is the author's opinion that the first marketable application of the UFIS will be a stationary power plant engine rather than a vehicular application no matter how attractive the low exhaust emissions and fuel economy benefits. Presently, low diesel vehicular engine volume precludes the mass introduction of the UFIS due to cost. However, if market penetration of the vehicular diesel engine continues to increase and government regulations requiring increased fuel enconomy and lower air pollution levels remain, the high production volume and reduced cost UFIS may result. The stationary source engines could take advantage of the economy of mass production in that event.

The UFIS system offers a potential utilization to stationary source engines unlike any other of the components discussed in this report. Presently, the U.S. Army is funding the UFIS on a low priority basis. These commendable efforts should be enhanced, and a demonstration program of a large size UFIS for stationary application should be considered.

4. TURBOCOMPOUNDING

The concept of turbocompounding is neither revolutionary nor new. Turbocompounding, reduced to its simplest terms, is mechanically adding the excess exhaust energy from a turbocharger (excess energy over that energy necessary to compress intake charge air) to the driveshaft of the engine. For example, the Napier Nomad aircraft engine manufactured in the 1950's was a turbocompounded engine (Reference 13). This application is described as a low-pressure, free-power turbine used to extract additional power from the high temperature exhaust gas by transferring the additional power to the crankshaft via reductive gears and a torsional isolator (Reference 10).

Because turbocompounding is usually associated with the adiabatic engine concept discussed later, no attempt will be made to completely disassociate the two concepts. Some data exists which give estimates of the effects of turbocompounding, but the value of turbocompounding becomes evident only when the adiabatic or insulated engine concepts are used to produce high exhaust gas temperatures.

a. System Description

The typical energy balance of a diesel engine shows approximately 35.6% of the input energy goes to brake output, 37.1% is lost through the exhaust, 25.6% is lost to the cooling water, and the remaining 2.7% goes to other losses. The energy losses to the exhaust and cooling systems represent approximately two thirds of the energy input (Reference 14). Consequently, the U.S. Army and the Cummins Engine Company have engaged in the development of a high temperature diesel, which will greatly reduce these exhaust and cooling system losses. Part of this development program is to turbocompound the engine. Cummins Engine Company is currently evaluating a turbocompound engine for application to a basic diesel engine (non-adiabatic), and will be making marketing decisions for the early 1980's based on these test results.

A turbocompound engine is quite similar to the turbocharged after-cooled engine except for a low pressure free power turbine which extracts additional power from the high temperature exhaust gas and transfers it to the crankshaft by means of reduction gears and a torsional isolator, as shown in Figure 4.

It should be noted that the performance of an insulated adiabatic turbo-compound engine has been estimated to make a worthwhile improvement in engine bsfc (9% at maximum rated speed and load at a constant NO_x emission level) by utilizing the exhaust gas energy. The insulated engine also offers an additional 10% reduction in bsfc over the uninsulated adiabatic turbocompound engine.

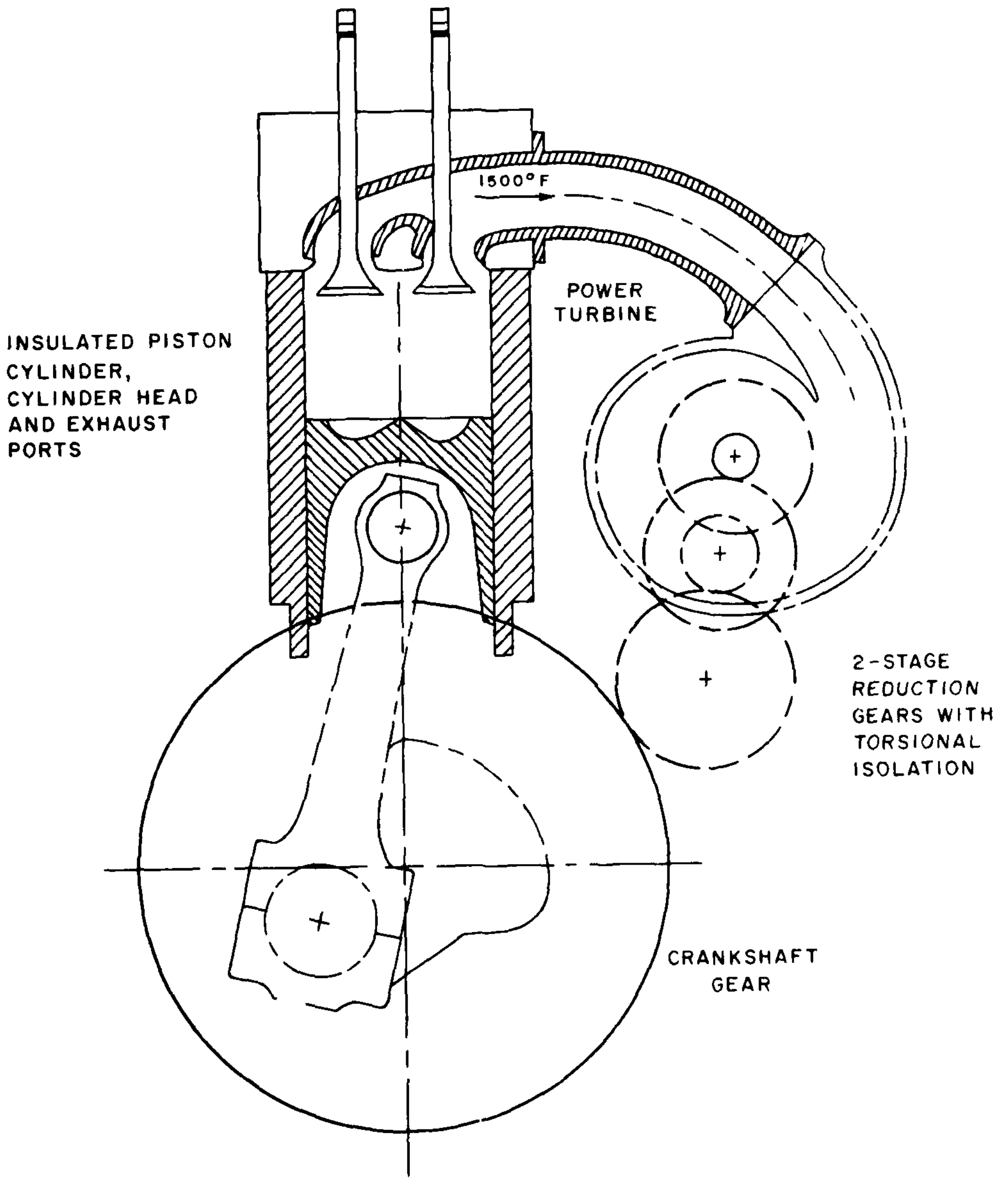


Figure 4. Turbocompound Engine Concept

Kamo and Bryzik (Peference 14) also indicated a horsepower boost of approximately 17% at no additional cost, with a corresponding fuel consumption decrease and a modest volume increase. Recent test results indicate that turbocompounding a non-adiabatic diesel engine will result in 50 additional horsepower capacity with an 8-12% fuel economy increase. Other advantages claimed for turbocompounding are improved mechanical and thermal loading and improved engine acceleration (Reference 15).

b. Factors Considered

Because turbocompounding is so intimately tied to the adiabatic engine concept, the factors considered are not as extensive as they were for previously discussed concepts. The factors the author considered are shown in Table 11 through 13.

c. Potential for Stationary Power Plants

The potential for stationary sources application is not significant when viewed in light of a non-adiabatic engine. Since many stationary engines are presently turbocharged and several engines are also after-cooled, the turbocompound engine would simply be a substitution for the turbocharger/after-cooler package. Weight and volume savings would be possible, but no other significant improvements are readily apparent.

These unapparent improvements are not the case for an insulated adiabatic turbocompound engine because of its dramatic increase in thermal efficiency. Consequently, the turbocompounded engine concept needs to be explored in conjunction with the insulated engine concept. Also the work started by the U.S. Army needs to be followed closely for the potential technology transfer to stationary engine applications.

5. ADIABATIC ENGINE

One of the most exciting concepts for reduced fuel consumption, lower exhaust emissions, and higher engine output is the adiabatic insulated turbocompound diesel engine. As previously mentioned, this concept is in the R&D stages at the Cummins Engine Company and is being sponsored by the U.S. Army. This program is to improve power train efficiency in military

TABLE 11

TURBOCOMPOUNDING FACTORS CONSIDERED

FACTORS	COMMENTS
Size and Weight	Reduced size and weight compared to turbocharged/aftercooled diesel engine. In other applications, slight increase in weight and size (220 lbs).
Costs	Reduced costs over turbocharged/after-cooled engine - no cost benefits over naturally aspirated engine.
Materials	No change.
Development	Development hardware presently not available - still in research stage.
Production Capability	Should not pose any production difficulties.
When Producible	Will follow adiabatic engine development program - targeted for mid to late 80's time frame (Reference 9). Non-adiabatic engine turbocompounding could be as soon as early 1980's.
Probability of Success	Good to excellent (Reference 14).
Potential Benefits	Reduced size and weight, improved fuel economy over naturally aspirated engine (Reference 15).

TABLE 12

ADVANCED DIESEL ENGINE SYSTEMS (TURBOCOMPOUND ENGINE)

PERFORMANCE PARAMETER	ESTIMATED SYSTEM IMPACT VS. PRESENT DIESEL SYSTEMS
A. Engine Efficiency	Improved by 5-15% over standard engines, less over T/C aftercooled engine (Reference 14).
B. Reliability	Expected to be as good as present naturally aspirated (NA) systems.
C. Lifetime	No difference 15-20 years, 5 years between overhaul.
D. Operation & Maintenance	No different from present system, with no impact on operator's skill levels.
E. Component Growth Potential	Some growth potential, has greater impact on power output than the 19% increase offered by T/C engines.
F. Start-Up/Shutdown Time	Improved start-up, 2 to 3 secs over T/C engines, no impact on shutdown (Reference 10).
G. Thermal Energy Available	Reduced due to improved thermodynamic cycle improvement.
H. Fuel Consumption	Lower by 9% (Reference 10), including an 18% power increase (Reference 18).
I. Volume/Size	A 1.1 ft^3 increase over present NA engine but no different over T/C after-cooled engine (Reference 5).
J. Weight	Increased 150 lbs over NA engine but 900 lbs less on adiabatic engine.
K. Environmental Constraints	Improved exhaust emissions (especially particulates), 2.6 db reduction in noise (Reference 14).
L. Acquisition Cost	No difference in costs at same power output of NA engine. Approximately 0.7 times NA costs at 2.0 times NA power output.
M. Life Cycle Cost	Should be lower due to lower fuel consumption; needs to be quantified after development proceeds.

TABLE 13

ADVANCED DIESEL SYSTEMS (TURBOCOMPOUND ENGINES)

ENGINE SIZE			POTENTIAL INTRODUCTION DATE & APPLICATION				
			1980	1985	1990	1995	2000
A.	50 MW Cont		NA	NA	NA	NA	NA
B.	"	1 hr	NA	NA	NA	NA	NA
C.	10 MW Cont		NA	NA	NA	NA	NA
D.	"	8 hr	NA	NA	NA	NA	NA
E.	"	1 hr	NA	NA	NA	NA	NA
F.	750 KW Cont		NA	X	X	X	X
G.	250 KW Cont		NA	X	X	X	X
H.	50 KW Cont		NA	X	X	X	X
I.	"	8 hr	NA	X	X	X	X
J.	"	1 hr	NA	X	X	X	X
K.	10 KW Cont		NA	X	X	X	X
L.	"	8 hr	NA	X	X	X	X
M.	"	1 hr	NA	X	X	X	X

X indicates availability

propulsion systems. Except for smoke (signature), none of the Army programs are concerned with exhaust emissions at this time. In terms of its applicability to light duty and heavy duty highway vehicles, the adiabatic engine is the most interesting program (Reference 2).

Insulated combustion chamber components are used in this engine to reduce heat losses into the engine block. The insulated components, currently under development by Cummins Engine Company, include the piston, cylinder head, valves, cylinder liner, and exhaust ports (References 14 and 16). Since the cooling system has been eliminated, the combustion chamber temperature is increased, providing a significant efficiency improvement potential. Maximum benefits will be achieved by combining the adiabatic engine with turbocompounding and/or a Rankine bottoming cycle. The projected specific fuel consumption of these combined systems falls into the 0.25 to 0.28 lb/bhp-hr range, compared to about 0.35 lb/bhp-hr for a large conventional turbocharged diesel engine (Reference 2).

The adiabatic engine is currently in an early stage of development, with single cylinder engine and component work in progress at Cummins. While the use of ceramic materials in experimental diesels is not new, large concentrated efforts remain to be undertaken before acceptable reliability and cost-effectiveness can be demonstrated. These efforts include the development of ceramics and insulated components which are capable of tolerating the high temperatures, high pressures, and pressure rise rates typical of adiabatic diesel engine operation.

a. System Description

The problem of vehicular cooling systems in military applications has prompted the U.S. Army to intensify its efforts to eliminate the engine cooling system, including cooling fans, pumps, radiators, hoses, and shrouds. The increased combat survivability of a military vehicle without the need for a cooling system is evident.

The impact of adiabatic diesel engine operation includes fuel economy improvements, specific weight and volume reductions, smoother, quieter combustion, and improved multifuel characteristics because of high temperature operation.

The elimination of the cooling system in a diesel engine can eliminate much of the reliability, maintenance, and parasitic losses and failure problems. Equally important is the matter of utilizing the conserved heat which is normally lost in the cooling system. These strong two-fold advantages can be realized in an adiabatic diesel engine.

There are several methods of recovering the heat normally rejected to the cooling water. The adiabatic engine contains the heat in the combustion chamber at high temperature; and, upon opening the exhaust valves, the high temperature exhaust gases are available for work. This work can be recovered with turbocompounding (see Section II.4) and by Rankine bottoming cycle (see Section II.6). However, the combination of the turbocompound system and the Rankine bottoming cycle offers the maximum benefit (Reference 14).

The insulated adiabatic engine (Figure 5) will obviously result in higher cylinder temperatures. Cylinder wall temperatures will exceed 2000°F. The insulative effects of the adiabatic engine are reflected in an increase in component temperature, and, as shown in Table 14, a decrease in overall engine heat rejection rate of various engine components fabricated with glass ceramics (zirconia and lithium aluminum silicate). Note that a completely insulated engine with components made of glass ceramics can reduce the heat transfer rate by 79%. In terms of overall engine heat rejection rate, the reduction is 41.5% (including intercooler and oil cooler). This translates into a fuel consumption improvement of 0.021 lb/bhp-hr. Noise is also reduced (Reference 14).

Although a completely insulated engine is highly desirable, a partially insulated engine could still be useful, since the greatest improvement to engine efficiency is made by the insulated pistons, cylinder liners, and valves. Kamo and Bryzik predict that 0.28 bsfc could be realized in an adiabatic turbocompound diesel engine (Reference 14).

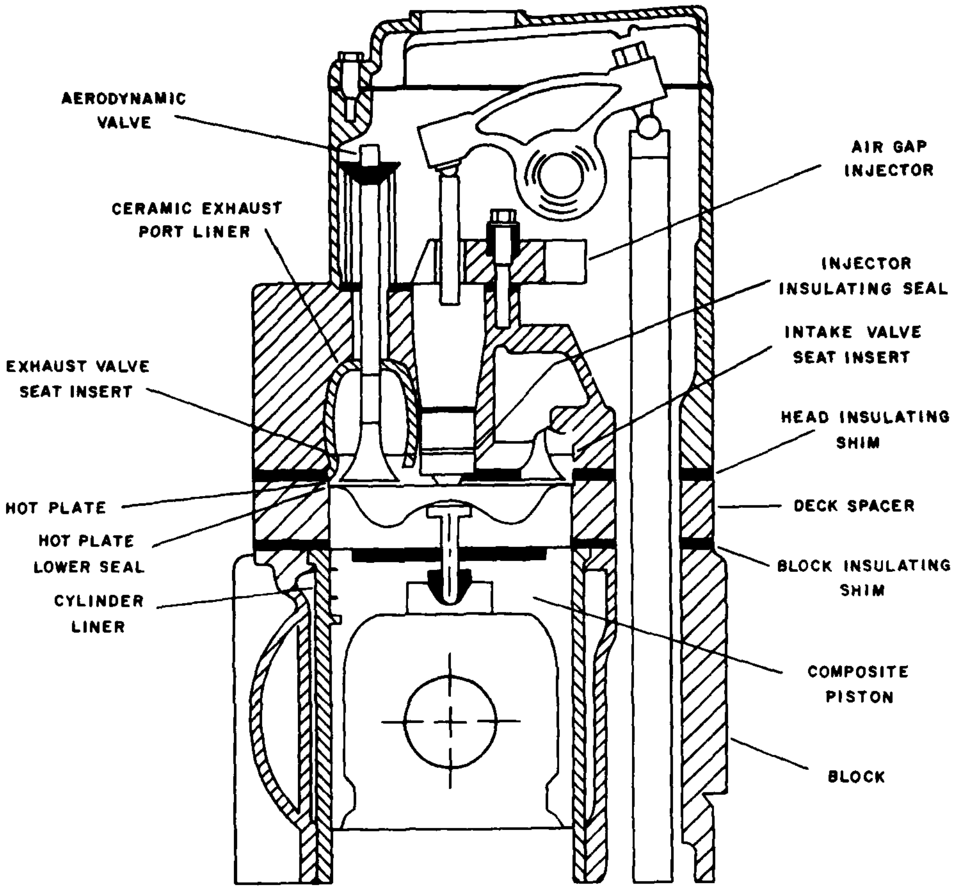


Figure 5. Cross Section of Cummins Basic Adiabatic (Insulated) Diesel Engine (from Reference 17)

TABLE 14

REDUCTION IN ENGINE HEAT REJECTION
RATE - ALL GLASS CERAMIC COMPONENTS (FROM REFERENCE 17)

INSULATION CONFIGURATION	PERCENT REDUCTION HEAT TRANSFER RATE
Standard	0
Piston	19.7
Piston and Liner	40.8
Piston + head + valves	37.4
Piston + liner + head + valves	61.2
Piston + liner + head + valves + ports	79.0

- Reduction in overall engine heat rejection rate
    - Include intercooler and oil cooler 41.5%
- Corresponding in bsfc via cooling fan/waterpump
    - All glass ceramic parts 0.021 lb/bhp-hr
- Fan noise reduction 10 dbA
- Overall engine noise reduction @ 3 ft. 2.6 dbA

Based on the developmented progress to date, the following conclusions were projected by Kamo and Bryzik (Reference 14).

• The turbocompound systems of the Rankine bottoming cycle are essential in taking full advantage of the inherent cycle efficiency of the insulated adiabatic engine.

• The lack of cooling systems in a reciprocating engine does much toward improving its maintenance and reliability as well as improving the system fuel economy.

• Turbocharging the adiabatic engine is essential in restoring the power output. Air-to-air aftercooling of the turbocharger discharge before it enters the engine must be considered.

- Other improvements which can be expected are:

  Power plant system weight
  Power plant system size
  Power plant system cost
  Noise reduction
  Multifuel capability
  Emissions improvements

b. Factors Considered

Reference 14 provides a good summary of the factors considered and the data from Reference 14 are reproduced in Tables 15 and 16. Additionally, an analysis similar to that of the Terrestrial Energy Study is shown in Table 17. This analysis departs from previous analyses in that the adiabatic engine is more of a complete system while the other systems discussed in this report are components attached to a complete system.

The materials under investigation include the glass ceramics, hot press silicon nitride, and metallic components (Reference 17). The results of this extensive materials review and development program indicate the original projections may be achievable. However, it must be cautioned that ceramics developed for the adiabatic engine rival the ceramics development problems encountered in vehicular gas turbine programs. The reliability of the ceramic piston is claimed to require no more sophisticated cooling than metallic pistons. Producibility of the ceramic components needs to be defined. Consequently, the success rate of future developments rests on the successful completion of the ceramics development program.

By way of future consideration, the U.S. Army has asked Cummins to consider friction reduction of the adiabatic turbocompound engine (Reference 17). A 50% reduction in the overall adiabatic engine friction could reduce brake specific fuel consumption from 0.28 lb/bhp-hr in the adiabatic turbocompound case to 0.25 lb/bph-hr in the minimum friction engine case. To meet these reductions in overall engine friction and

TABLE 15

DIESEL CYCLE SIMULATION RESULTS OF TESTS OF VARIOUS ADVANCED DIESEL BASED POWER PLANTS FROM (REFERENCE 14)

	Turbocompound (cooled)	Adiabatic Turbocompound (insulated)	Adiabatic Rankine η_R=0.158	Adiabatic Turbocompound Rankine η_R=0.158
SPECIFICATIONS				
Engine RPM/Air:Fuel	2100/26	2100/28	2100/28	2100/28
Reciprocator, BMEP	177.8	177	183	177
PERFORMANCE				
Reciprocator, 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				
Reciprocator, BSFC	0.341	0.363	0.323	0.366
Overall, BSFC	0.309	0.285	0.272	0.255
EMISSION				
$BSNO_2$, Grams/BHP-HR	5.0	5.0	5.5	4.4

TABLE 16

ESTIMATED INSTALLED ADVANTAGES OF ADIABATIC TURBOCOMPOUND ENGINE
(FROM REFERENCE 14)

	WEIGHT	COST	SIZE		BHP
Diesel (Advanced)	3260		71.0	0.367	465
Turbocompound	3410		1.1	0.327	545
Piston		-	-	(0.010)	
Block-Liner Water	(410)		-	(0.006)	
Water Pump/Drives	(80)	(105.90)	(9.7)	(0.004)	
Radiator/Water	(345)	(300)	-		
Holes, Pulley, etc.	(25)	(128.87)	-		
Fan	(15)	(10.00)	-	(0.013)	
Cylinder Head	(120)	-	-	(0.014)	
Oil System		-	-	0.001	
TOTAL	2550	UNCHANGED	52.4	0.281	639

(LOSS)

TABLE 17

ANALYSIS OF THE ADIABATIC INSULATED DIESEL ENGINE

System Description: Adiabatic insulated diesel engine with turbocompounding and/or Rankine bottoming cycle.

Fuel: No. 2 distillate fuel oil.

Working Fluid: Air

System: Assume diesel engine/generator with no improvements in generator efficiency.

System Definition: See Section 5a - assume no 50 MW units available or contemplated to be produced, 10 MW units are 10 1 MW units in series. (Note: because of compact size of adiabatic engine it may be possible to use fewer units for 10 MW).

Physical Descriptions: Similar to present diesel engine, however, needs further definition.

List of Requirements and Time Frames:

Requirements	1985	1990	1995	2000
50 MW Cont	NA	NA	NA	NA
" 1 hr	NA	NA	NA	NA
10 MW Cont	NA	X	X	X
" 8 hr	NA	X	X	X
750 KW Cont	X	X	X	X
250 KW Cont	X	X	X	X
50 KW Cont	NA	X	X	X
" 8 hr	NA	X	X	X
" 1 hr	NA	X	X	X

TABLE 17 (Cont'd)

Requirements	1985	1990	1995	2000
10 KW Cont	NA	X	X	X
" "	NA	X	X	X
" "	NA	X	X	X
" "	NA	X	X	X
" 8 Hr	NA	X	X	X
" "	NA	X	X	X
" 1 Hr	NA	X	X	X

X Indicates availability

Acquisition Cost (1977 Dollars)

Requirements	1985*	1990
10 MW Cont	NA	2,655,000
" 8 Hr	NA	2,457,800
" 1 Hr	NA	2,457,800
750 KW Cont	266,300	379,275
250 KW Cont	172,250	261,700
50 KW Cont	NA	218,625
" 8 Hr	NA	213,900
" 1 Hr	NA	213,900
10 KW Cont	NA	202,000
8 Hr	NA	201,240
1 Hr	NA	201,240

*Costs were determined by multiplying the 1977 cost data presented in Reference 1 by an assumed inflation rate and modest developmental cost inclusion (see Equation 1). Costs may vary +25 to 30% to -10% depending on system developmental success. Costs could vary greatly because of the highly developmental nature of this system.

TABLE 17 (Cont'd)

$$C = C_{77}(1 + I_r)^y + C_D;\ \text{Assumed } I_r = 5\%/\text{yr} \qquad (1)$$

C_D = \$15,000/yr average

Y = Number of years

Life Cycle Cost (1977 Dollars)

The life cycle cost is defined by the following equation:

LCC = AC + FC + OMC

where AC = Acquisition cost

FC = Total fuel cost over system lifetime

OMC = Operation and maintenance cost over system lifetime

Requirements	LCC (10^3) 1977\$	LCC/yr (10^3) 1977\$
10 MW Cont	51,389	2,335
" 8 Hr	19,992	883
" 1 Hr	9,301	258
750 Cont	5,887	327
250 Cont	2,258	125
50 KW Cont	2,953	223
" 8 Hr	1,660	116
" 1 Hr	1,862	51
10 KW Cost	587	65
" 8 Hr	414	41
" 1 Hr	323	9

TABLE 17 (Cont'd)

Lifetime*

Requirement	Lifetime 1985	Lifetime 1990	Years Between Overhauls 1985	Years Between Overhauls 1990	No. of Overhauls 1985	No. of Overhauls 1990
10 MW Cont	NA	22	NA	4.9	NA	4
" 8 Hr	NA	24	NA	5.5	NA	4
" 1 Hr	NA	36	NA	15	NA	2
750 KW Cont	18	22	4.4	4.9	4	4
250 KW Cont	18	22	4.4	4.9	4	4
50 KW Cont	NA	13	NA	2.8	NA	4
" 8 Hr	NA	14	NA	2.9	NA	4
" 1 Hr	NA	36	NA	20.5	NA	1
10 KW Cont	NA	9	NA	1.6	NA	7
" 8 Hr	NA	10	NA	1.6	NA	8
" 1 Hr	NA	35	NA	27.5	NA	1

*Assume lifetime in 1985 will be 90-95% of lifetime shown in Reference 1, page 105, due to uncertainty of development program. Goal for 1990 time frame is thought to be 110% of current lifetime requirements.

TABLE 17 (Cont'd)

Volumes*

Requirements	Volume ft^3	Volume m^3
10 MW Cont 8 hr, 1 hr	8850*	250
750 KW Cont	528	15
250 KW Cont	176	5
50 KW Cont 8 hr, 1 hr	70	2
10 KW Cont	21	0.59
" 8 Hr, 1 hr	21	0.59

*Volume assumed to be 0.88 of volume shown in Reference 8, page 106 and Reference 14.

**10 MW units consists of 1000 units in parallel.

Weight*

Requirement	System Weight lb	System Weight kg
10 MW Cont, 8 hr, 1 hr	257,400**	117,000
750 KW Cont	14,625	6,630
250 KW Cont	5,538	2,496
50 KW Cont, 8 hr, 1 hr	1,895	858
10 KW Cont	483	218
" 8 Hr, 1 Hr	483	218

*Weight assumed to be 0.78 of weight shown in Reference 1, page 107, and in Reference 14.

**10 MW units consists of 1000 units in parallel.

TABLE 17 (Cont'd)

Fuel*

Requirements	Amount Per Year 10^3 gal	10^3 kg	Average Cost Per Year 10^3 Dollars (1977)
10 MW Cont	2,891	9,452	1,971
10 MW 8 Hr	1,376	4,500	628
10 MW 1 Hr	172	563	182
750 KW Cont	226	738	294
250 KW Cont	78	257	102
50 KW Cont	16.5	53.7	202
" 8 Hr	7.8	25.6	97.6
" 1 Hr	0.98	3.2	45.0
10 KW Cont	3.4	11.0	39.4
" 8 Hr	1.6	5.2	18.9
" 1 Hr	0.2	0.6	3.2

*Costs calculated as in Reference 1, page 108.

TABLE 17 (Cont'd)

Environmental Constraints

The environmental constraints of the adiabatic engine are indicated in the following table.

Emission	X	Y	Z
Thermal Discharge	0	0	-
Air Pollution			
HC	0	0	-
CO	-	-	-
NO_x	◐	◐	0
SO_x	0	-	-
Particulates	0	0	-
Noise	◐	-	-
Solid Waste	-	-	-
Chemical Waste	-	-	-
Radioactive Waste	-	-	-

Where: X = amount of uncontrolled emission

Y = amount of pollution which would be emitted with no controls

Z = degree of difficulty in meeting more stringent regulations

Key: blank - none

0 - minor

◐ - moderate

● - major

TABLE 17 (Cont'd)

System Efficiency

Requirements	Efficiency %*
10 MW Cont, 8 Hr, 1 Hr	0.50
750 KW Cont	0.48
250 KW Cont	0.46
50 KW Cont; 8 Hr, 1 Hr	0.44
10 KW Cont;	0.43
" 8 Hr, 1 Hr	0.43

*From Reference 14, assume 90% electrical conversion efficiency, and the achieved thermal cycle efficiency of the adiabatic engine ranges from 48-56%.

Start-Up/Shutdown Time

Requirements	Start-Up	Shutdown
10 MW Cont; 8 Hr, 1 Hr	6 - 10 sec	8 - 10 sec
750 KW Cont	6 - 10 sec	8 - 10 sec
250 KW Cont	6 - 10 sec	8 - 10 sec
50 KW Cont; 8 Hr, 1 Hr	6 - 10 sec	8 - 10 sec
10 KW Cont, 8 Hr, 1 Hr	6 - 10 sec	8 - 10 sec

Growth Potential

Some modest growth potential could be achieved if scaling-up of the technology does not present major production difficulties.

Reliability

The conditions existing in the adiabatic diesel engine which seriously impact reliability are the numerous moving parts and thermal cycling. Those conditions having moderate impact on the system reliability are high temperature operation, high stress levels, and non-modular design.

TABLE 17 (Cont'd)

Maintenance and Operation

The annual maintenance and operating costs are estimated to range from 110 to 120% above those estimates presented in Reference 1 on page 118. This uncertainty is caused by the developmental nature of the adiabatic engine.

Requirement	Operation (1977 $/Yr)	Maintenance (1977 $/yr)
10 MW Cont	29,400	214,800
" 8 Hr	14,040	88,560
" 1 Hr	1,752	6,360
750 KW Cont	2,208	16,080
250 KW Cont	1,470	6,840
50 KW Cont	882	4,320
" 8 Hr	420	3,120
" 1 Hr	54	3,540
10 KW Cont	528	2,880
" 8 Hr	252	2,160
" 1 Hr	30	276

Other Energy Production

Requirement	10^6 Btu/Hr	10^3 KW Thermal
10 MW Cont; 8 Hr, 1 Hr	20.4	6
750 KW Cont	1.6	0.48
250 KW Cont	0.6	0.17
50 KW Cont; 8 Hr, 1 Hr	0.4	0.04
10 KW Cont; 8 Hr, 1 Hr	0.03	0.008

brake specific fuel consumption, the following engine components require careful study and scrutiny: (1) Gas lubricated piston/ring/liner combinations; (2) Unlubricated roller bearings for wrist pin, crank pin, and main bearings; and, (3) Solid lubricant cooling for gears, valve guides, rocker arm, and push rod assemblies.

All of the above component designs involve the use of advanced high performance ceramics technology. Not only would the engine be uncooled, but i successful, a good possibility exists for the development of an unlubricated diesel engine.

The minimum friction engine approach is still in the very early stages of development. If these research and development programs are successful, the Army expects:

(1) Uncooled diesel engine design of very high thermal efficiency (48%) in conjunction with turbocompounding. Presently, diesel engines have a thermal efficiency range of 30-40%.

(2) Unlubricated engine design of higher thermal efficiency (56%) with significant exhaust energy still available for a bottoming cycle.

(3) The success of the minimum friction adiabatic turbocompound engine will be highly dependent on the contribution of the high performance ceramics.

(4) New dimensions in technology bases in engine design, combustion and application engineering will have to be reckoned with (Reference 17).

c. Potential for Stationary Power Plants

The adiabatic insulated diesel engine, either turbocompounded or with the Rankine bottoming cycle, represents the ultimate in advanced diesel engine research and development. All of the other diesel component research and development programs discussed in this report represent either modest improvements in engine cycle efficiency or address a specific problem, i.e., air pollution. Therefore, the adiabatic diesel engine has the potential of advancing the diesel engine state-of-the-art dramatically.

The adiabatic diesel engine can be translated directly to the stationary power plant. In the author's opinion, the adiabatic diesel engine, because of its compact size, improved cycle efficiency, lower fuel consumption, and potential for lower exhaust emissions, may be amenable to increased power sizes greater than 10 MW electrical output.

The major obstacle to the adiabatic diesel engine is the development and a reliability demonstration of the ceramics used in these engines. If the engine can be expected to demonstrate approximately 0.25 lb/bhr-hr fuel consumption, the break through in ceramic components will have to be demonstrated first. The producibility of ceramic components must be demonstrated before a stationary power plant can be demonstrated.

Because of these high risk areas, it is estimated that the adiabatic diesel engine will not be available in stationary power plants until the late 1980's. However, if the ceramic problems are solved, the adiabatic diesel engine will be a worthy competitor to all other power-producing units which are dependent on fossil or synthetic fuels.

6. RANKINE BOTTOMING CYCLE

Considerable interest has been shown in recent years by industry and government in the organic working fluid Rankine cycle bottoming system, or diesel bottoming cycle, as a means of capturing and utilizing waste heat to improve overall diesel engine cycle efficiency. A bottoming cycle engine is an engine powered by heat that is rejected, and usually discarded, from another heat power cycle. The amount of recovered power that could be produced is dependent on the working fluid characteristics, cycle efficiency, amount and temperature of the waste heat, and space available for the sizing of the bottoming system components. Industrial Rankine cycle units powered by diesel engine exhaust heat (or other sources such as flue gases) have been built and operated. (Systems for vehicular application have also been constructed and are undergoing testing; see References 19, 21, and 22). These units are primarily engine-driven electrical generation plants. The author of Reference 20 states that experience with these stationary units and other experimental units shows that the organic

Rankine engine could produce output power in the range from a few percent to more than 20% of the basic diesel engine output power. Fuel consumption of the basic diesel engine could be reduced by a comparative amount as an alternative to producing additional shaft horsepower (Reference 20).

a. System Description

The schematic diagram shown in Figure 6a represents a closed cycle organic Rankine bottoming system engine, with a turbine expander and power supplied by diesel engine exhaust gases (approximately 700°F). Figure 6b is a schematic of the Rankine bottoming cycle as adapted to the adiabatic insulated engine (see Section II.5).

Figure 6a shows that a heated, high pressure vapor (state 1) is expanded through the turbine engine and produces shaft work. The expanded vapor (state 2) then passes through a regenerator (heat exchanger) where it is cooled (state 3) by the transfer of heat to the liquid state of the working fluid. The slightly superheated working fluid exits the regenerator, enters the condensor, where it is cooled and condensed to liquid (state 4), and enters a feed pump which raises the liquid pressure to the expander inlet pressure (state 5). The working fluid is heated as it passes through the liquid side of the regenerator (state 6). The liquid enters the vapor generator where it is heated by the heat source to produce high temperature vapor (state 1) and the cycle is complete.

Figure 7 shows the economy of adding a bottoming cycle system to a diesel engine. Shaft output power (P) can be produced at a lower output power level (B) of the basic diesel engine if the difference in output power (between A and B) is generated by the bottoming cycle system. The difference in the fuel consumption of the basic diesel engine when operated at state points A and B results in fuel savings, or alternatively, in increased engine plant efficiency. Also, additional power (represented by the difference between A and C) can be produced and used at the same total fuel consumption rate as the basic engine, but at point A power output.

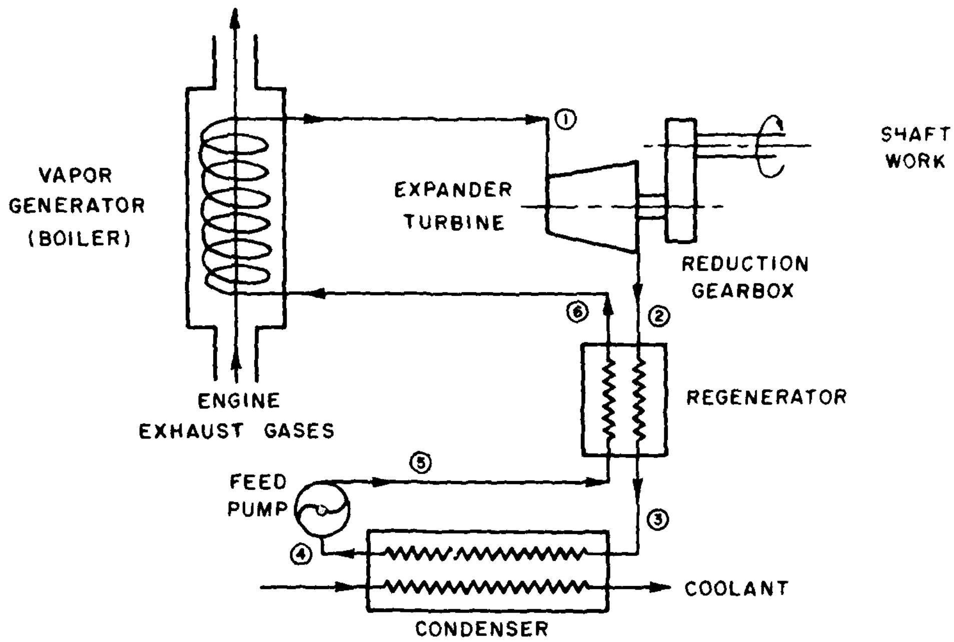


Figure 6a. Schematic of an Organic Rankine Cycle Bottoming System (from Reference 20)

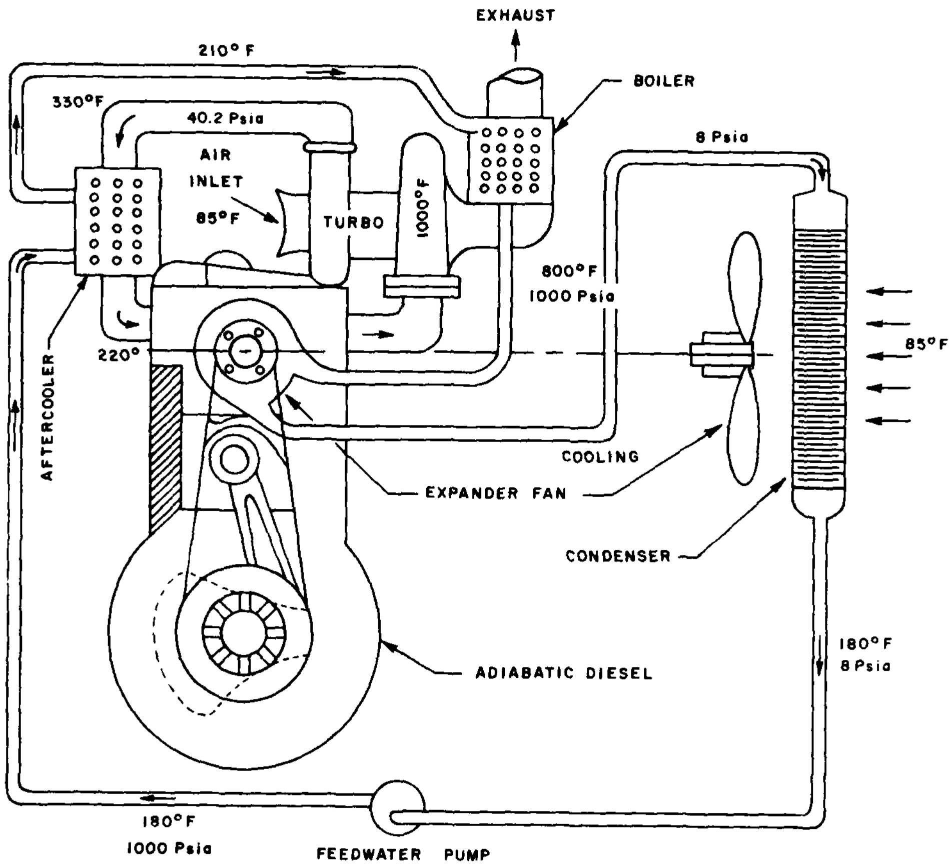


Figure 6b. Rankine Bottoming Cycle on an Adiabatic Diesel Engine (from Reference 14)

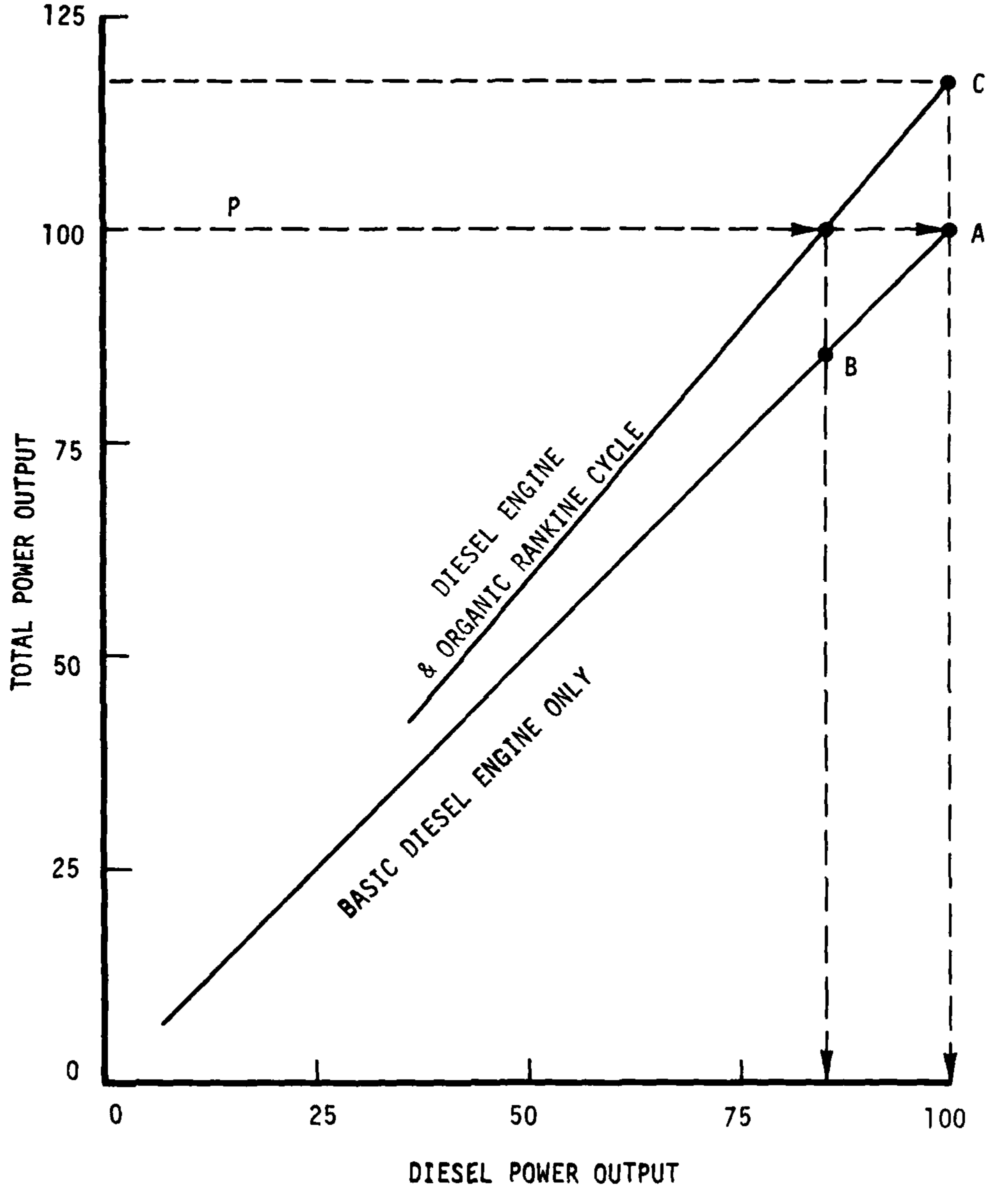


Figure 7. The Relationship Between Diesel/ORC Power Output and Basic Diesel Engine Power Output (from Reference 20)

b. Factors Considered

Table 18 details those factors the author considered important in the evaluation of the Rankine bottoming cycle. An attempt to put the analysis in the order of Reference 1 can be seen in Tables 19 and 20.

c. Cost of Organic Rankine Cycle Bottoming Systems

The following discussion is extracted from Reference 20 and presents the best analysis of Rankine bottoming cycle costs to date.

The cost of organic Rankine cycle bottoming systems was developed from data on existing units, published data, data supplied by the three developers, and engineering analysis. Costing of Rankine cycle bottoming systems was difficult because of the great variance in existing system designs, heat sources, organic fluid types, system recovery efficiencies, type of heat rejection systems, single and dual heat source system designs, size of systems, and application of recovered power. In addition, current cost data available represent single unit systems under development or small quantities. Cost data on past systems are several years old and reflect construction of a single or limited number of units. Cost data on many units do not necessarily include all associated costs such as development costs. Installation, maintenance, and operating costs also vary greatly from one installation to another. Cost estimates of units in 1973/1974 ranged from approximately \$350 to \$750/KW (\$260 to \$560/bhp). Recently manufactured units are priced from approximately \$1,000 to \$1,300/KW (\$746 to \$970/bhp), also on a very limited number of units.

The production costs for marine diesel engines is comprised of all costs leading to multiple unit production of organic Rankine cycle bottoming systems. The production costs of diesel bottoming systems, shown in Figure 8, are installed costs and include initial system start-up, operational checkout, and warranty costs. The costs are fully burdened and include profit, but do not include the amortization of development costs and any tooling involved, which could increase the cost by about 5%. The three specific bottoming cycle systems identified in this figure are for specific designs, cycle efficiencies, and basic diesel engines, and may

TABLE 18

RANKINE BOTTOMING CYCLE FACTORS CONSIDERED

Factor	Comments
Materials	Uses materials more exotic than those commonly used with diesel engines. Working fluids are expensive, pose hazards, and require complex handling systems.
Costs	Increases costs, but cost can be recovered by fuel savings, thereby achieving a net cost reduction (Reference 21).
Development	Vehicular applications are undergoing testing and pre-production validation; stationary power plants are apparently ready for sale (Reference 20).
Weight	Increases vehicle weight, suited for large trucks and stationary power plants where weight may not be a limiting factor.
Production Capability	Presently produced on a small volume basis, mass production needs to be determined.
When Producible	Early 1980's, some units now ready for application; adiabatic engine units will not be ready until mid 1980's.
Probability of Success	Excellent, major drawback is working fluid, cost, and hazards.
Potential Benefits	Lower fuel consumption and/or increased power (Reference 20); slight potential for reduced exhaust emission (Reference 14).

TABLE 19

ADVANCED DIESEL ENGINE SYSTEMS (RANKING BOTTOMING CYCLE)

Performance Parameter	Estimated System Impact
A. Engine Efficiency	Cycle efficiency has been reported to be improved from a few percent to 20% (Reference 20).
B. Reliability	Needs to be determined; a 3000 mile road test is projected (Reference 19).
C. Lifetime	Needs to be determined, expected to be as long as diesel engine life.
D. Operation & Maintenance	No additional labor or experience levels will be required over basic diesel engine (Reference 20).
E. Component Growth Potential	Good growth potential, especially scaling-up (Reference 20).
F. Start-Up/Shutdown Time	No effect on shutdown time; may have 20-30 sec thermal lag on start-up.
G. Thermal Energy Available	Lower because of basic operation of system; up to 20% less than basic engine.
H. Fuel Consumption	Improved 5 to 20% (Reference 20).
I. Volume/Size	Negatively impacted, to be determined, estimated at 20-35% greater.
J. Weight	Increased weight, estimated at 20-30% more weight.
K. Environmental Constraints	Lower thermal discharge, possibly lower specific exhaust emissions, lower noise, potential (hazard) chemical discharge, no solid or radioactive discharge.
L. Acquisition Cost M. Life Cycle Cost	Reference 20 gives the cost estimates as discussed in Paragraph II 6c.

TABLE 20

ADVANCED DIESEL SYSTEM (RANKINE BOTTOMING CYCLE)

Engine Size		Potential Introduction Date and Application				
		1980	1985	1990	1995	2000
A. 50 MW	Cont	NA	NA	NA	NA	NA
B. "	1 Hr	NA	NA	NA	NA	NA
C. 10 MW	Cont	NA	X	X	X	X
D. "	8 Hr	NA	X	X	X	X
E. "	1 Hr	NA	X	X	X	X
F. 750 KW	Cont	X	X	X	X	X
G. 250 KW	Cont	X	X	X	X	X
H. 50 KW	Cont	X	X	X	X	X
I. "	8 Hr	X	X	X	X	X
J. "	1 Hr*	X	X	X	X	X
K. 10 KW	Cont	X	X	X	X	X
L. "	8 Hr*	X	X	X	X	X
M. "	1 Hr*	X	X	X	X	X

X Indicates availability

* May not be practicable in these applications

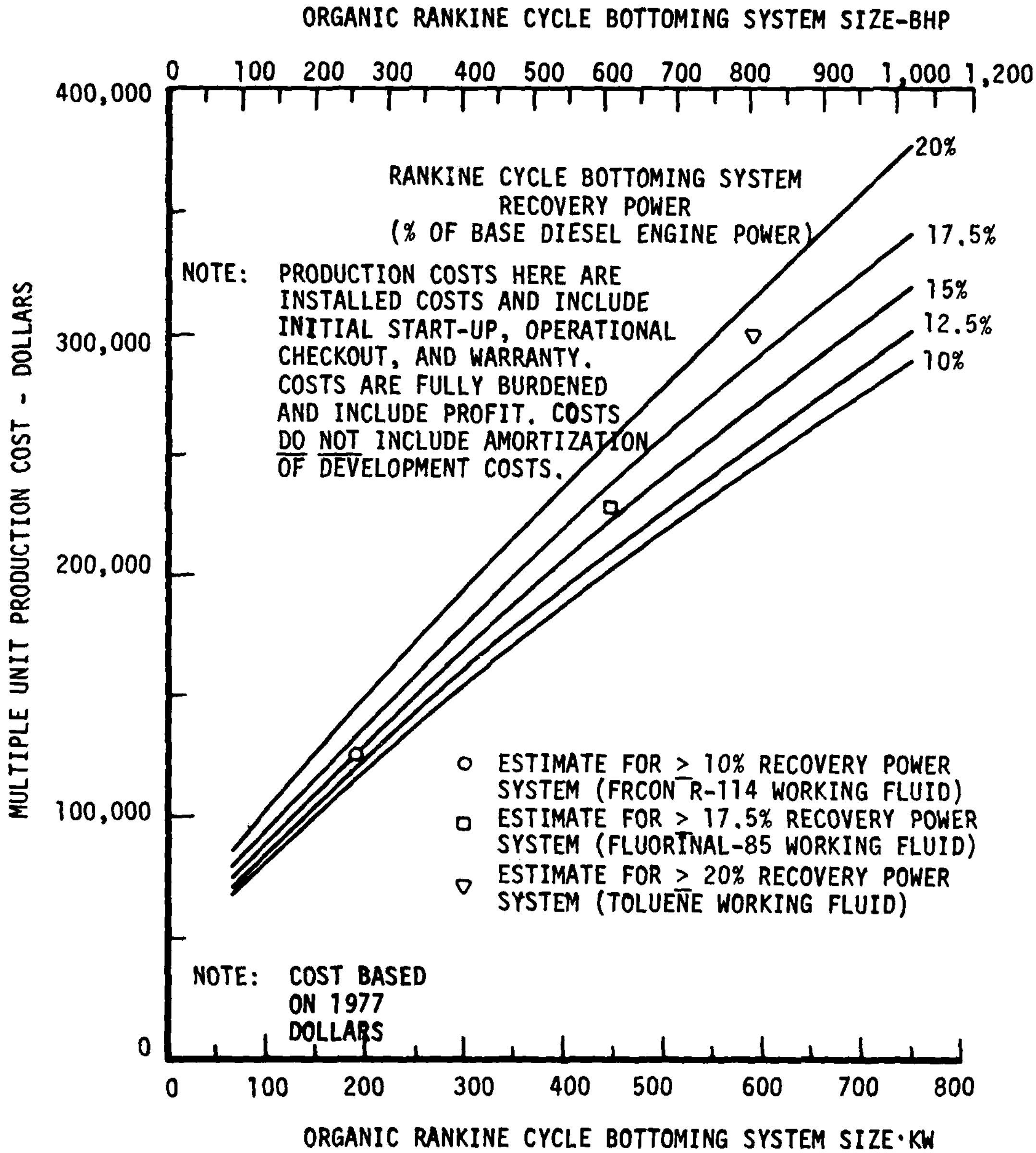


Figure 8. Organic Rankine Cycle Diesel Bottoming System Production Coat Estimates (from Reference 20)

not be representative of the maximum amount of power recovery capability for each particular design (Reference 20).

Figure 8 and Table 21 show a comparison of typical operating and maintenance costs for electr.cal plants and marine diesel engines. Based on these data, it is estimated that maintenance and repair costs of organic Rankine cycle bottoming systems will be approximately $0.0015 per kilowatt-hour for approximately 6000 hours per year (in 1976 dollars).

d. Potential for Stationary Power Plants

Several high-technology industrial firms are actively engaged in the development of organic Rankine cycle bottoming systems. These firms have all proposed, are developing, and are demonstrating bottoming systems for various vehicular and stationary electrical generation applications.

Mechanical Technology, Inc. (MTI) has been engaged in the development of both binary and single loop organic Rankine cycle bottoming systems for industrial applications. The binary systems use Freon R-11 and steam as the working medium, and the single-loop systems use Freon R-114 in connection with diesel and gas turbine engine exhaust heat sources. MTI currently is engaged in the development of a binary bottoming cycle system utilizing the exhaust gas reject heat from a diesel engine of approximately 7000 bhp as the heat source to drive a generator set. Net recovered energy from this system is approximately 599 bhp or 447 KW.

The Sundstrand Energy Systems Company has been engaged in the development of organic Rankine cycle bottoming systems utilizing toluene as the working fluid. They have developed several varied types of organic Rankine cycle systems below the 6 KW size. The company's recent experience includes an 88 bhp organic Rankine cycle bus engine and 40 KW and 100 KW total energy systems. They currently are developing a 600 KW (approximately 800 bhp or approximately 900 bhp when shaft power is used) organic Rankine cycle system to drive a generator. Three of the six units will be powered by the exhaust gas reject heat from a diesel engine rated at about 4,200 bhp. Recovered horsepower is approximately 20%.

TABLE 21

COMPARISON OF TYPICAL OPERATING AND MAINTENANCE COSTS FOR ELECTRIC PLANTS AND MARINE DIESEL ENGINES (REFERENCE 20)

TYPE OF UNIT	O&M COST $/kW/H	O&M Cost $/BHP-HR
Average Electric Plant (100% Load Factor)	≅0.0030	≅0.00220
Well Managed New Electric Plant (100% Load Factor)	≅0.0015	≅0.00110
Small Electric Plants	≅0.0013	≅0.00095*
Marine Diesel Engines, 4-Cycle	-	≅0.00082*
Marine Diesel Engines, 2-Cycle	-	≅0.00068*
Large Diesel Marine Engines	-	≅0.00090*

*Based on 6,000-hours/year operation.

Costs factored to December 1976 dollars.

Thermo Electron Corporation has been engaged in the development of organic Rankine cycle bottoming systems using Fluorinol-50 and Fluorinol-85 as the working fluids. This system is a large over-the-road diesel tractor (Reference 21 and 22). This bottoming system is powered by the exhaust reject heat of a diesel engine rated at approximately 275 bhp. The recovered horsepower, amounting to approximately 19% for a total engine/bottoming system output of approximately 325 bhp, will be used to drive the vehicle. Fuel savings of approximately 15% can be accomplished by throttling back. This corporation is developing a 620 bhp bottoming system which will drive a 450 KW generator. The system is powered by the exhaust reject heat from a 3,600 bhp diesel engine and operates at a power recovery effectiveness of 17 1/2%.

Other concerns engaged in the development of organic Rankine cycle energy recovery systems include Barber/Nichols, Allied Chemical/Foster-Wheeler, and B&C Associates.

As can readily be seen, the application of the Rankine bottoming cycle to a stationary power plant is at hand. What remains is to accomplish increased production volume to reduce the system cost.

7. OTHER SYSTEMS

Although the systems discussed above can be considered to be the more mature and complex systems under development for diesel engine cycle efficiency improvements, it should be noted there are a myriad of development programs which in themselves are targeted for other goals (exhaust emissions reduction, etc.). These programs should be reviewed for diesel cycle efficiency improvement potentials.

It was considered to be beyond the scope of this report to expend effort on developing the narrative of these systems beyond brief remarks concerning their potential. Therefore, complete analyses of these systems, although technically possible, would not be as beneficial as the aforementioned systems. Additionally, many of the following discussed developments are very preliminary and not quantifiable.

a. Systems to Reduce Exhaust Emissions

Diesel engines, while inherently low emitters of hydrocarbons and carbon monoxide, have the potential to be high emitters of oxides of nitrogen, particulate matter, sulfur compounds, and various currently unregulated exhaust emissions. Consequently, both industry and government are intensifying their efforts, collectively and individually, to resolve these problems. By far the hardest pollutants for vehicular diesel engines to overcome will be oxides of nitrogen and particulate matter. These may not be as difficult, or as necessary, for stationary diesel engines to overcome.

The research categories, investigators, and comments shown in Table 22 are a summary of those areas of research which will impact the diesel engine's ability to meet exhaust emission standards.

TABLE 22

PROGRAMS TO REDUCE DIESEL EXHAUST EMISSIONS

Program	Conducted by	Comments
Water Injection	U.S. Government, Auto Industry	Total cooling of a diesel engine by direct injection of water into the cylinder is feasible for stationary and marine engines. Fuel economy and power are improved. HC and CO emissions tend to increase while NO_x is reduced. Sulfur oxides are scrubbed. Engine durability is unchanged.
Reduction of Blow-by	U.S. Government	Reduction of blow-by increases engine efficiency and reduces oil contamination and emissions, favorably impacts several unregulated exhaust emissions.
Exhaust Gas Recirculation (EGR)	U.S. Government, Auto Industry, Component Suppliers	EGR is used for NO_x control which impacts fuel economy, particulate and HC emissions. Engine durability problems have been noted, 40-60% reduction of NO_x has been demonstrated. Durability and lube oil contamination problems need investigation.
Particulate Traps and Scrubbers	U.S. Government, Auto Industry, Component Suppliers	Traps have achieved 30-40% particulate reduction, 90% efficiency for sulfates but deteriorates rapidly at low mileage; back pressure buildup is a problem, may be able to be regenerated; stationary units may be amenable to disposable traps or scrubbers more so than vehicular units.
Fuel Composition	U.S. Government, Auto Industry, Component Suppliers	Following fuel parameters being studied: density, aromatics, sulfur content, smoke suppressant additives, cetane number, boiling point range, H_2O-fuel ratio. Study efforts concentrate on emissions and fuel economy improvements; some results indicate effect may be function of combustion chamber rather than fuel.

TABLE 22 (Cont'd)

Program	Conducted by	Comments
Fuel Preheating	Auto Industry	Under investigation for effects on fuel economy and emissions.
Basic Engine Parameter Optimization	U.S. Government, Auto Industry Component Suppliers	The following are the parameters under study: injection timing, injection rate, valve timing, compression ratio, fuel spray, patterns, inlet air temp., pilot injection, chamber wall temp., exhaust back pressure, fumigation, basic combustion chamber modifications, and fuel injection system modifications.

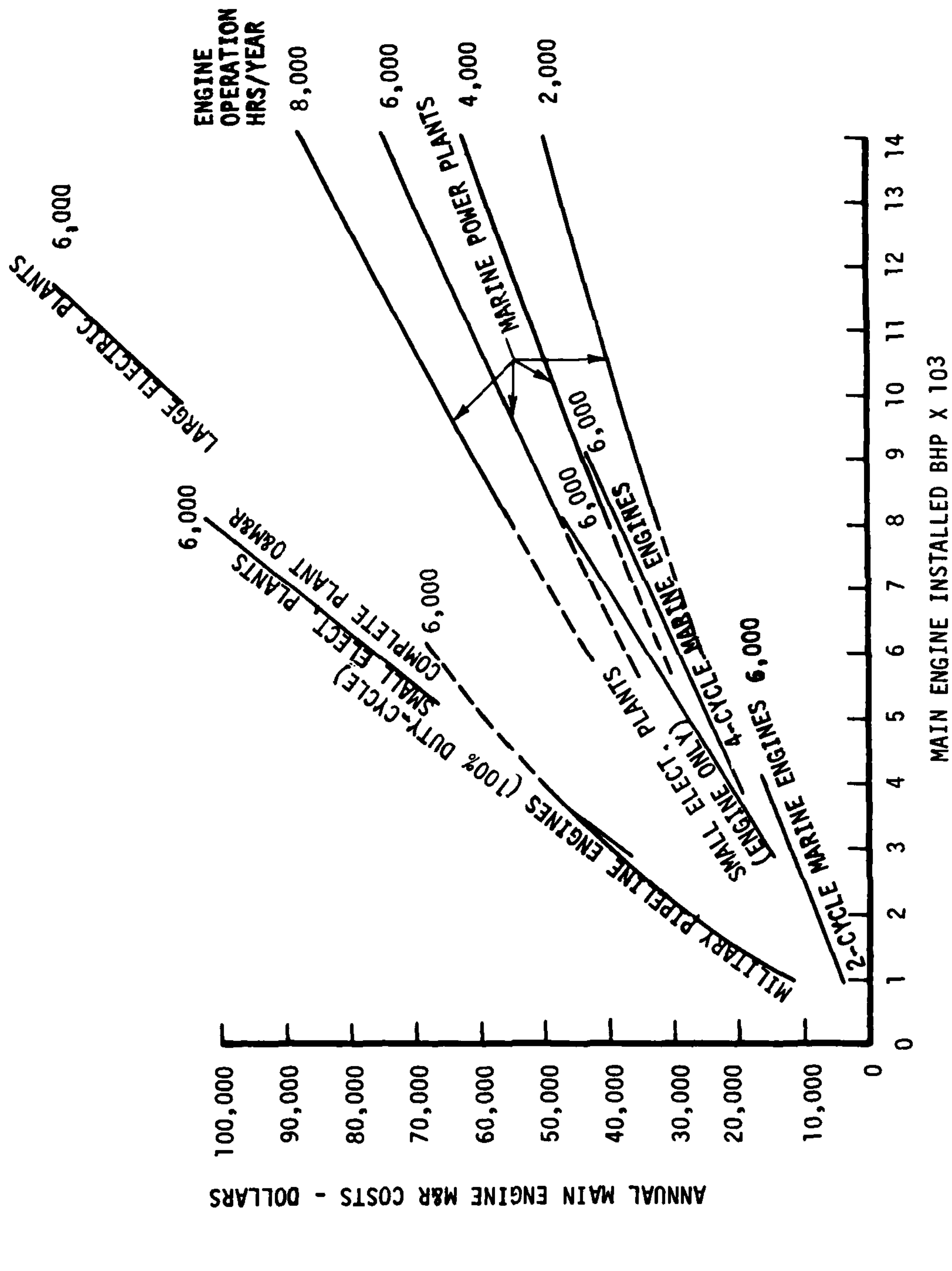
b. Systems to Reduce Noise and Systems to Improve Vehicular Fuel Economy

There are more programs designed to improve diesel-powered vehicles' fuel economy center on downsizing (reduction of weight), improved transmissions, optimization of engine operating parameters in relation to vehicle test cycle, and subtle vehicle/power train improvements. Discussion of these programs are beyond the intent of this report but are well documented in Reference 10.

Programs to reduce diesel engine noise appear to be targeted to a 10 db reduction in noise level from today's engines. Figure 10 portrays the noise levels from present vehicular large diesel engines. The reference from which this figure was extracted presents a broad overview of the problem associated with the reduction of diesel engine noise (Reference 23). Reference 24 details the potential systems (enclosures, structural changes, etc.) that will reduce diesel engine noise. The reader is encouraged to review these references.

c. High BMEP Engines

The history of the U.S. Army diesel engine development work has centered around not only advanced components but also high brake mean effective pressure (BMEP) engine for high horsepower in small power-plant



NOTE: ALL COSTS FACTORED TO 1976 DOLLARS (U.S. STAT.AB. 1976 & S.C.A. 4TH QTR., 1976) SHP CONVERTED TO INSTALLED BHP.

Figure 9. Comparison of Annual Maintenance and Repair Costs for Industrial Plants and Marine Engines (from Reference 20)

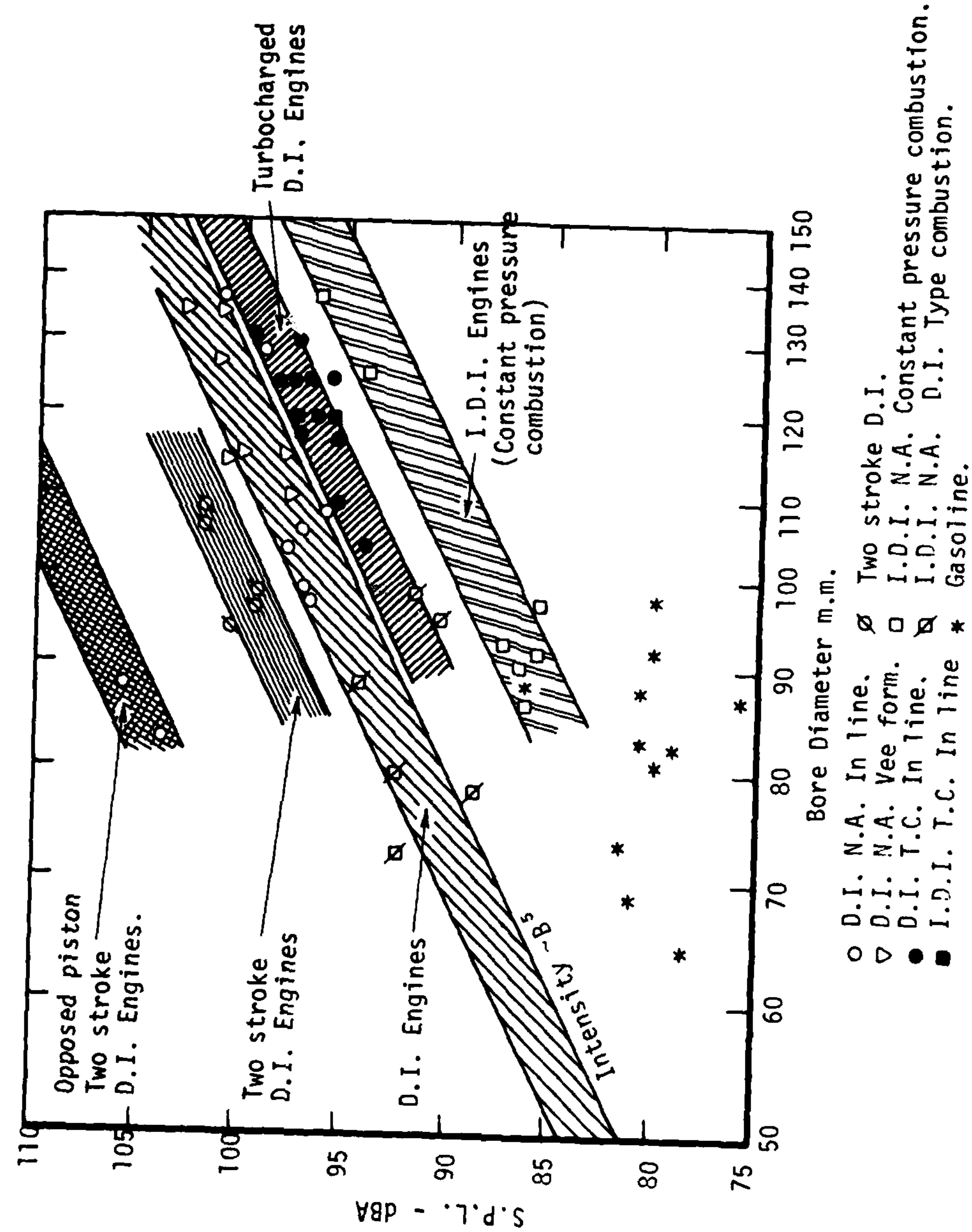


Figure 10. Relation Between Overall Noise and Bore Size at 2000 Rev/Min Full Load (from Reference 11)

packages. This work should be reviewed again in light of the fossil fuel shortages and new advances in materials, technology, and improved fuel and air management systems. Development programs such as the VCR, very high output (VHO) diesel and Hyperbar (Bypass Turbo Assist) diesels are discussed in Reference 15. The U.S. Army is also investigating constant pressure combustion (Reference 4).

Table 23, extracted from Mr. Kamo's paper (Reference 15), summarizes the potential for these systems. In Table 23a, Class I refers to conventional midly turbocharged engines, Class II engines are advanced turbocharged aftercooled engines pressing today's technology, and Class III engines are engines of high technology advanced design.

Table 23b shows, for Class III engines, the various possible engine systems which will overcome the cold start, mechanical load, transient response, emissions, and fuel economy problems of present-day engines.

Table 23c shows Mr. Kamo's overall comparison of a Class III high BMEP engine, and Figure 11 illustrates the cost per unit power of this proposed engine versus the other types of engines (Reference 15).

These engines are proposed for the mid to late 1980's, but a successfully completed development program will ensure the diesel engine a competitive position for those applications which will use liquid fossil fuels. These developments should be monitored and encouraged if at all possible.

TABLE 23

BMEP - PSI

100 ←→ 140 ←→ 220 ←→ 300

CLASS I — CLASS II — CLASS III

	TURBO	VARIABLE TIMING	AFTERCOOLING	VARIABLE GEOMETRY	2 STAGE TURBO	COMPREX	BURNER (COMBUSTION AID)	LOW CR	ADV. TURBO	TURBO ASSIST	VCR	P_{CONST} COMBUSTION	TURBOCOMPOUND
FUEL ECONOMY	✓	✓	✓						✓				✓
LOADING													
MECHANICAL			✓					✓			✓	✓	✓
THERMAL			✓					✓			✓	✓	✓
HIGH BOOST													
RANGE				✓	✓				✓				
PRESSURE RATIO	✓		✓		✓				✓				
EMISSIONS													
WHITE SMOKE		✓					✓			✓	✓	✓	
BLACK SMOKE	✓									✓			
HYDROCARBONS	✓	✓					✓			✓			
NO_x	✓	✓	✓					✓			✓	✓	
ACCEL SMOKE						✓				✓			
DRIVEABILITY													
STARTING TORQUE				✓		✓				✓			
RESPONSE				✓		✓				✓			✓
LOW SPEED TORQUE				✓		✓				✓			
COLD START							✓			✓	✓	✓	

a. Matrix of three classes of turbocharged engines, their design variables and how they affect engine performance

	COLD START	PEAK PRESSURE LIMIT	TRANSIENT RESPONSE	EMISSION CONTROL	FUEL ECONOMY
VCR	HIGH C.R.	LCR	VG TURBO	HIGH C.R.	NONE
LCR*	COMBUSTION AID	LCR	2-STAGE	COMB. AID	NONE
HYPERBAR	TURBO ASSIST ELECTRIC MOTOR	LCR	TURBO ASSIST BY PASS	EXH. BURNER	NONE
LCR-2	TURBO ASSIST (JET AIR)	LCR	TURBO ASSIST BY PASS	EXH. BURNER	NONE
LCR-3	COMBUSTION AID	LCR	TURBO ASSIST BY PASS	EXH. BURNER	NONE
P_{CONST} COMBUSTION	HIGH C.R.	INJ. CONTROL	VG TURBO	HIGH C.R. INJ. CONTROL	HIGH C.R.
TURBO-COMPOUND	COMBUSTION AID	LCR/P_{CONST} COMB.	2-STAGE	COMBUSTION CONTROL	EXHAUST RECOVERY
VHO	HIGH C.R.	NONE	VG TURBO	HIGH C.R.	HIGH C.R.
OTHER FORMS OF HIGH OUTPUT:	DESIGN, COMPACT HIGH SPEED TWO-STROKE				

*LCR = LOW COMPRESSION RATIO

b. Possible high BMEP engine systems for Class III vehicular diesel engine

	NA	CLASS I	CLASS II	CLASS III
HP	265	320	450	600
BMEP	102	125	175	233
HP/IN^3 Displacement	0·337	0·407	0·572	0·76
Box Volume - FT^3	42·8	49·1	52·0	53·0
Weight - Pound	2080	2125	2286	2336
Cost - $	1·0	1·04	1·176	1·23
HP/FT^3	6·2	6·6	8·65	11·32
LB/HP	7·85	6·64	5·1	3·89
$/HP	1·0	0·85	0·68	0·52
BSFC, LB/BHP-HR	0·385	0·370	0·375	0·375

c. Specific output comparison of four classes of diesel engine

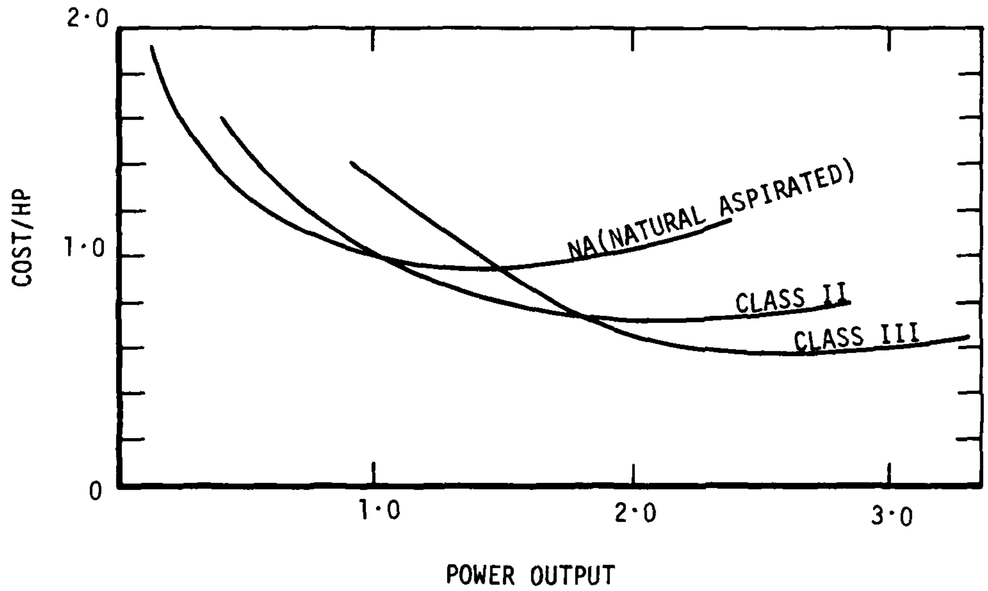


Figure 11. Expected Relative Cost of Different Classes of Diesel Engines on Equal Volume Production (from Reference 15)

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