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# Waste Heat Recovery from Adiabatic Diesel Engines by Exhaust-Driven Brayton Cycles

United Technologies Research Center

December 1983

Prepared for  
NATIONAL AERONAUTICS AND SPACE ADMINISTRATION  
Lewis Research Center  
Under Contract DEN 3-304

for  
**U.S. DEPARTMENT OF ENERGY**  
**Conservation and Renewable Energy**  
**Office of Vehicle and Engine R&D**

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East Hartford, Connecticut 06108

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Office of Vehicle and Engine R&D  
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## FOREWORD

The work described in this report was performed by the Advanced Systems Technology group at the United Technologies Research Center (UTRC) with subcontract assistance from the John Deere Product Engineering Center and the Hamilton Standard Division (HSD) of United Technologies. Mr. G. Melikian was the UTRC Project Manager and Dr. H. E. Khalifa was the Principal Investigator. Contributing to the UTRC effort were Mr. A. I. Chalfant, Mr. G. T. Peters, Dr. T. N. Obee, Dr. L. E. Greenwald, Mr. E. D. Kostic, all of UTRC, and Mr. G. Wilmot of HSD. The John Deere effort was conducted by Dr. M. Goyal.



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## SUMMARY

The temperature and energy content of the exhaust gases discharged by adiabatic diesel engines are expected to be much higher than in the case of conventional turbocharged diesel engines. Considerable improvements in the efficiency of adiabatic engines can be effected through the use of efficient bottoming systems that can utilize the energy contained in the exhaust gases for the generation of additional shaft work. Exhaust-driven Brayton bottoming systems (BBS) are particularly suitable for this function because they represent a logical extension of the familiar turbocompounding concept and can be added to an adiabatic engine either instead of turbocompounding or in tandem with it.

The objective of this project was to provide an analytical assessment of the technical and economic feasibility of the BBS for adiabatic diesel engine applications in heavy-duty, long-haul trucks. The project was structured around three major work packages: (1) characterize the performance and design features of adiabatic diesel engines over a wide range of operating conditions; (2) perform performance, design and cost analyses of the BBS; and (3) carry out simulation of vehicle performance and a comparative evaluation of the economics of trucks equipped with adiabatic engines with or without BBS.

Parametric analyses were performed to evaluate the effect on BBS performance of external design parameters such as exhaust temperature and flow rate and internal design parameters such as turbomachinery pressure-ratio and efficiency and heat exchanger temperature difference and pressure drop. The results of these parametric studies, combined with qualitative assessments of the advantages and disadvantages of various BBS configurations, led to the selection of an intercooled pressurized BBS for further analysis. Conceptual design and trade-off studies were undertaken to estimate the optimum values of the key internal design parameters. The critical sizing parameters for the key hardware components were estimated along with the projected mass-production cost for these components and for the entire system. The addition of a pressurized BBS to a turbocharged adiabatic diesel engine was shown to be capable of a 12-percent improvement in fuel economy. Produced in 10,000 units annually, such a system would have an installed cost of about \$170/Bhp (1983\$).

Off-design performance analyses were performed to assess the sensitivity of the BBS performance to ambient temperature variations and external leakage as well as to evaluate the effect of heat exchanger fouling and cleaning frequency on the average performance of the BBS. The benefits of BBS speed regulation through a variable ratio transmission were investigated and complete part-load performance maps were generated for the BBS-equipped engines.

The power-plant maps were incorporated into a drive-train and vehicle/driving-cycle simulation routine to calculate the truck average fuel consumption and drivability characteristics. The fuel savings that can be attributed to the BBS were computed for numerous driving cycles. It was found that, with the addition of a BBS to a turbocharged non-aftercooled engine, as much as 2000 gallons of fuel (12%) can be saved over 100,000 miles of annual driving in a 73,000 lb, 300 Bhp long-haul truck. However, when compared with the aftercooled turbocompound engine,

the BBS-equipped engine would offer only 4.4 percent of fuel savings. The installation of the BBS in tandem with the aftercooled turbocompound engine would result in about 7.2 percent of fuel savings.

Discounted cash-flow and payback calculations were performed to assess the economic merit of the BBS. It was found that, although the BBS would be an economically justifiable addition to turbocharged or turbocompounded engines, it would, at best, be marginally economical as a substitute for turbocompounding. Lower capital costs, longer equipment life, higher fuel prices and the elimination of other technical and economic uncertainties could significantly improve the economic attractiveness of the BBS, particularly when used in tandem with an aftercooled turbocompound engine.

# 1. INTRODUCTION

## 1.1 Background

Conventional diesel engines convert approximately one-third of their fuel energy input into mechanical work. The remaining two-thirds are rejected into the atmosphere either directly with the exhaust gases or indirectly through the cooling of the engine surfaces, lubricating oil, and charge air. In a typical diesel engine, the indirect heat rejection may account for as much as one third of the fuel energy, usually through a water jacket maintained at 180-200 F. The remaining fraction of fuel energy is carried with the exhaust gases at temperatures in the range of 800-1000 F.

Significant improvements in diesel engine performance can be effected by converting part of the rejected energy into additional mechanical work through exhaust-driven devices such as turbocompounding systems (Ref. 1.1) and bottoming Rankine cycle systems (Refs. 1.2, 3). None of these devices, however, is currently in large-scale commercial use because the additional complexity and cost associated with them appear to outweigh their performance benefits, at least under present and projected near-term economic conditions. The conversion of the low-temperature energy contained in the jacket cooling water into work is much less economically justifiable, particularly in mobile engine applications.

Recent advances in material development have made it possible to fabricate some of the internal parts of diesel engines from insulating (low-thermal conductivity) ceramics, thus reducing energy losses to the coolant and lubricating oil and enabling the engine to approach adiabatic operation more closely (Refs. 1.4-7). The most obvious advantage of this is the elimination of much of the cooling system in the engine along with the associated bulk, complexity, lower reliability and cost. This advantage constitutes the major impetus for the development of adiabatic (insulated) diesel engines for tank propulsion.

Because insulated diesel engines (IDE) retain a larger fraction of the fuel energy in the working medium (combustion products), their thermal efficiency has been shown to be significantly higher than that of their conventional counterparts. Furthermore, insulated diesel engines discharge their exhaust gases at much higher temperatures approaching, and possibly exceeding 1400 F for a well-insulated engine. The energy balance of an insulated engine is considerably different from that of a conventional engine, with most of the rejected energy in insulated engines discharged with the exhaust gases, leaving only a small fraction to be rejected through the lubricating oil or residual cooling.

The higher exhaust gas temperatures and the larger fraction of energy contained in the exhaust gases of an insulated engine present an attractive opportunity for exhaust power recovery and overall thermal efficiency improvement through the use of advanced cost-effective and reliable bottoming systems in conjunction with these engines. Such an improvement will have the strongest economic impact in the heavy-duty transport sector, where space limitations are not as critical as they are in tank applications, and where large quantities of valuable petroleum fuel are being consumed.

Other advantages that have been claimed for the insulated diesel engine include wider fuel quality tolerance, lower particulate emissions and lower thermal pollution. These advantages strengthen the case for developing insulated diesel engines with integrated exhaust energy recovery systems for heavy-duty transport applications.

Previous R&D efforts on power recovery from high temperature waste energy streams in general, and diesel engine exhaust in particular, have focused almost exclusively on Rankine cycles (e.g., Refs. 1.2, 1.3, 1.9, 1.10) using either organic working fluids or steam. Organic Rankine cycles are usually limited by the lack of safe, inexpensive working fluids that will remain chemically stable at the high exhaust temperatures of insulated diesel engines. In fact, acceptable organic fluids for Rankine cycle applications appear at present to be limited to maximum temperatures below 700 F. This condition limits the ability of organic Rankine cycles efficiently to utilize exhaust gases at temperatures in excess of 1000 F such as those discharged by insulated diesel engines. Such is not the case with steam Rankine cycles which, nonetheless, suffer from the performance shortcomings of small high-expansion-ratio steam expanders.

The preceding arguments provide the motivation for the DOE/NASA Advanced Waste Heat Utilization Program, whose goal it is to explore new concepts for higher temperature waste heat recovery systems applicable to the heavy-duty transport sector. The present report describes the work performed by the United Technologies Research Center (UTRC) to assess the feasibility of one class of such systems, viz.; exhaust-driven Brayton Bottoming Systems (BBS).

## 1.2 Program Objectives

The specific objective of the UTRC program is to provide an analytical assessment of the technical and economic feasibility of exhaust-driven Brayton bottoming systems for insulated diesel engines in long-haul heavy-duty truck applications. A further objective of this program is to identify the critical technologies that must be developed before these systems can be reduced to commercial practice.

## 1.3 Program Scope

The UTRC program was structured around three main work packages. The purpose of the first package was to characterize the performance and design features of insulated diesel engines over a range of operating conditions. John Deere Product Engineering Center was subcontracted by UTRC to provide this information. The second package comprised all the performance and design analyses pertaining to the Brayton cycle bottoming system itself and was performed by UTRC with assistance from the Hamilton Standard Division (HSD) of United Technologies Corporation (UTC). The third package, which engines with or without BBS. The union of these three work packages enabled UTRC to assess BBS both inasmuch as they are affected by the engine, and inasmuch as they affect truck performance over typical driving cycles.

#### 1.4 Report Organization

This report comprises eight (8) chapters of which this introduction is the first. Chapter 2 describes the general characteristics of BBS and presents a qualitative comparison among several BBS configurations. Chapter 3 covers the unique characteristics of insulated diesel engines particularly those that could affect the design and operation of BBS. A parametric analysis of the performance of BBS over a wide range of design conditions is given in Chapter 4. The bulk of the BBS design analysis and trade-off studies is given in Chapter 5 which also provides the specifications and layout of the selected BBS and its major components. Chapter 6 contains a description of the vehicle performance analysis including fuel economy comparisons among various powertrain configurations and duty cycles. Chapter 7 contains the results of the life-cycle cost analysis and comparisons, and Chapter 8 presents a discussion of the results, an assessment of the technical barriers, conclusions and recommendations for future work.

## 2. EXHAUST-DRIVEN BRAYTON CYCLES

### 2.1 Preliminary Considerations

Power recovery from hot exhaust gas streams has been under investigation and development by UTRC and others for many years. Most of these investigations have focused on Rankine cycles (Refs. 1.2, 1.3, 1.9), although studies of Brayton cycles (Refs. 1.10, 2.1) and Stirling cycles (Ref. 1.9) have also been conducted.

Rankine cycles using suitable organic or inorganic working fluids can be adapted to a wide variety of waste heat recovery applications in the low- to moderate temperature range. However, no satisfactory organic fluid yet exists that will provide high performance with a minimum of fire and health hazard while remaining chemically stable at temperatures in excess of about 700 F. With such fluids, elaborate seals must be used to minimize leakage. Furthermore, when space is limited, as it is in a truck, the complexity and bulk of the Rankine cycle heat exchangers (preheater, boiler, regenerator, and condenser) and their associated plumbing, constitute a serious drawback that may prevent the commercial acceptability of Rankine cycle bottoming of truck diesel engines, despite the performance advantage this cycle may offer.

The use of steam may eliminate the thermal stability and sealing problems encountered with organic fluids but will not alleviate the drawback associated with the complexity and bulk of the heat exchangers and their plumbing. Furthermore, the design of an efficient, reliable and cost-effective steam expander in the 20-50 HP size range still constitutes a formidable challenge (Refs. 1.9, 1.10) owing to the very high volume expansion ratio such expanders must cope with in all but the most inefficient low-pressure steam Rankine cycles.

Stirling cycles, which are capable of high thermal efficiency and not limited by working fluid decomposition, are still in an early stage of development. Little reliable information is available on their commercial potential since no version of Stirling engine is currently in commercial production and none is expected to become so in the near future.

Brayton cycles, on the other hand, use air or combustion products as the working medium and are, therefore, free from the major shortcomings of Rankine cycles. Brayton bottoming systems constitute a natural extension of the turbocompounding concept which has been recognized for a long time as a relatively simple method to improve engine performance within acceptable levels of hardware complexity (Ref. 1.1). In fact, the Brayton bottoming concept can be thought of as enhanced turbocompounding as can be seen in Fig. 2.1 which contrasts a turbocompound engine with a BBS-equipped engine.

Because Exhaust-driven Brayton-Bottoming Systems (BBS) are not temperature-limited by working fluid decomposition, they are better suited

**COMPARISON OF TURBOCOMPOUNDING AND BRAYTON BOTTOMING SYSTEM CONFIGURATIONS**

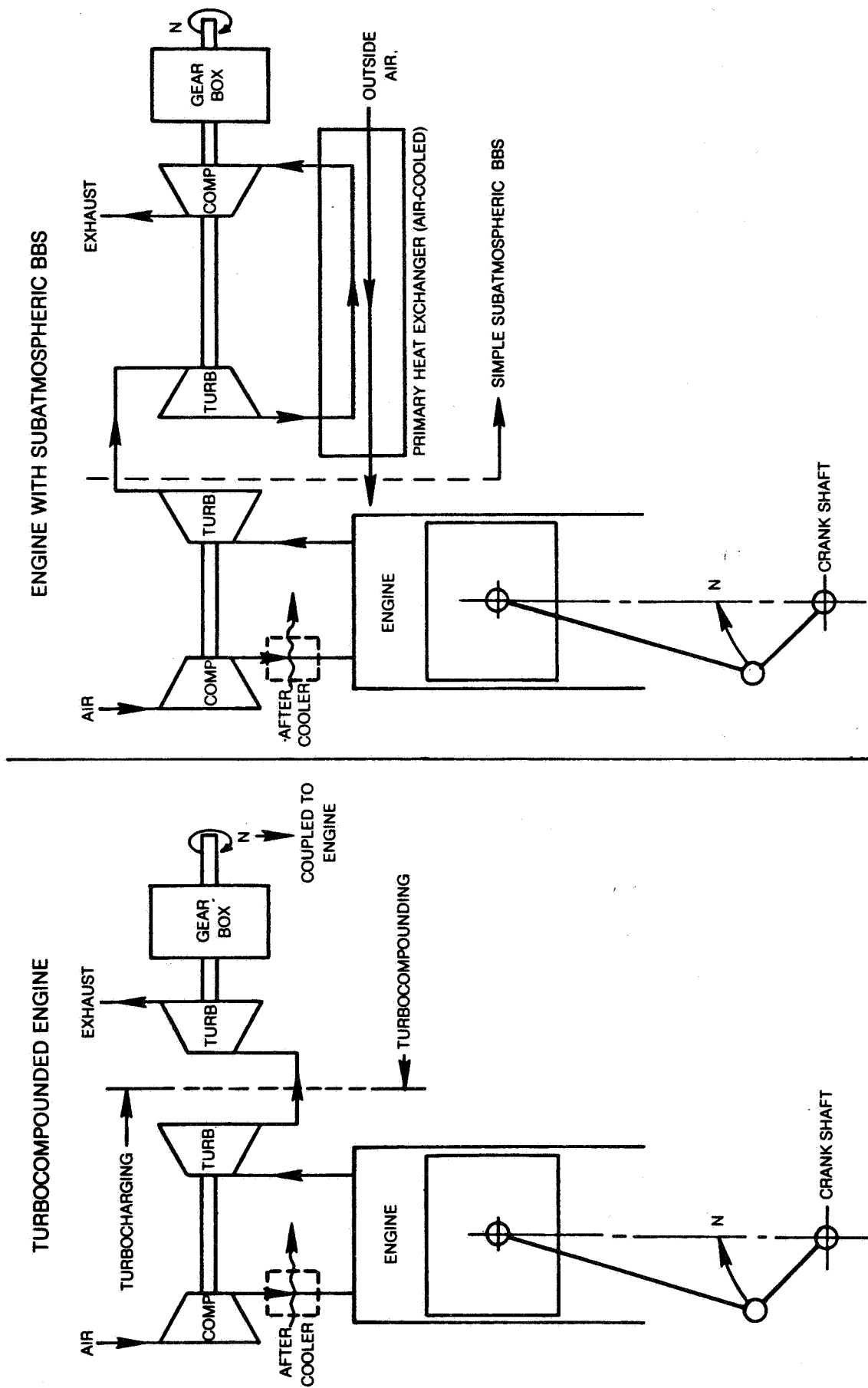


FIG. 2.1

than Rankine systems to take advantage of the much higher temperature of insulated engine exhaust.\* Exhaust-driven Brayton bottoming systems comprise hardware components with which the heavy-duty truck industry is quite familiar, namely:

- . small high-speed turbomachines like those widely used as superchargers and exhaust turbines (for turbocharging or turbocompounding)
- . gas-to-gas compact heat exchangers like those used as supercharger aftercoolers in many diesel engines.

Both items can be easily integrated with diesel engines. In particular, it may be possible to integrate the engine turbocharger with the turbomachinery of the BBS to yield a simpler package. Another advantage of Brayton systems is that their response to engine modulation will be relatively fast. This is attributed mainly to the lower thermal inertia of the Brayton cycle primary heat exchanger in comparison with the fluid-filled heat exchangers in Rankine cycle systems. This advantage is particularly important in automotive power plant applications.

## 2.2 System Configurations

### 2.2.1 Brayton Bottoming Systems (BBS)

Brayton cycle bottoming systems can be configured in many ways to match the specific characteristics of the engine. Two basic types of Brayton bottoming systems (BBS) can be identified.

- a. Direct subatmospheric BBS (e.g., Fig. 2.2)
- b. Indirect pressurized BBS (e.g., Fig. 2.3)

The main configurational difference between the two is the position of the primary heat exchanger. In the first case (subatmospheric), the hot exhaust gas from the engine is expanded directly in the turbine to a subatmospheric pressure then cooled in an air-cooled surface heat exchanger before pumping it up to slightly more than one-atmosphere for disposal. In the second case, the hot engine exhaust is used to heat air that has been drawn from the atmosphere and compressed before entering the primary heat exchanger. The hot compressed air is then expanded in the turbine down to atmospheric pressure. Both recuperative-type heat exchangers and regenerative\*\*--type heat exchangers can be used.

---

\*Contrariwise, Rankine cycles constitute a better match to lower exhaust temperatures (<1000 F) than Brayton cycles.

\*\*NASA LeRc is analyzing a system of this kind in which an automotive gas turbine ceramic matrix regenerative heat exchanger is to be used.



# EXHAUST-DRIVEN BRAYTON SYSTEM

SUBATMOSPHERIC/INTERCOOLED

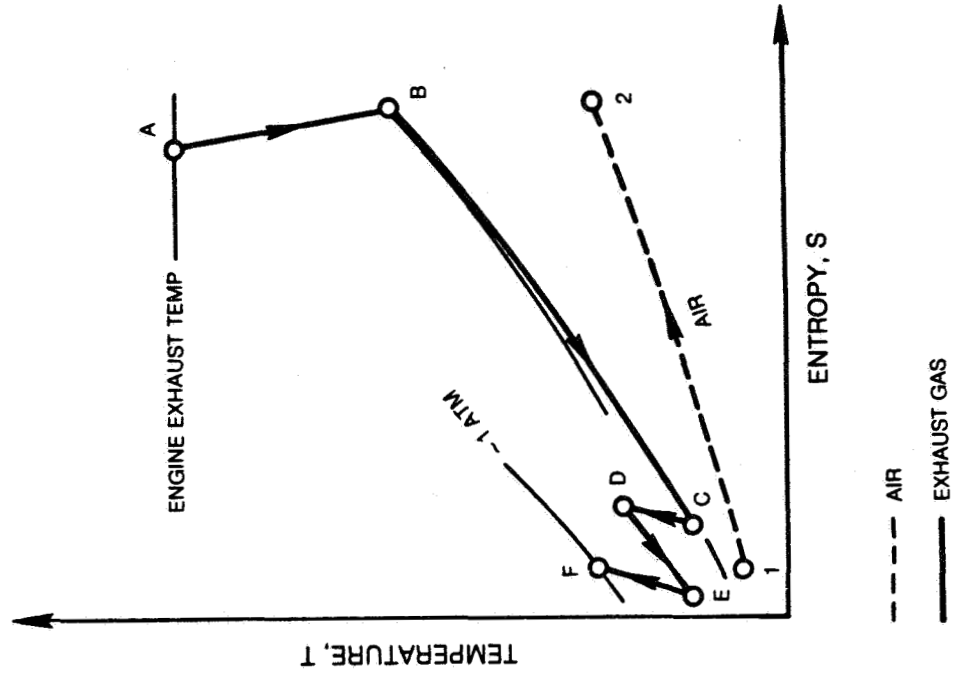
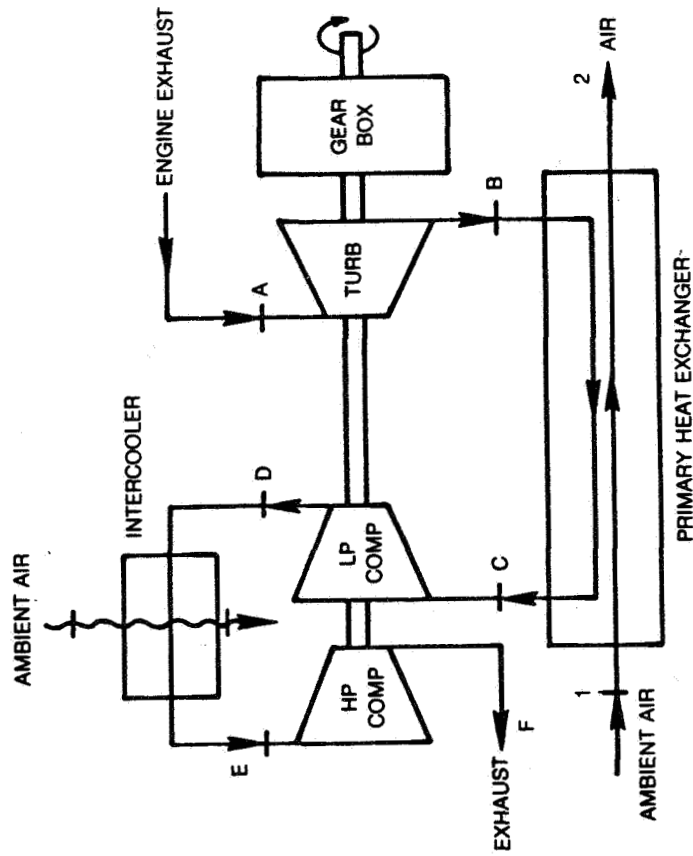


FIG. 2.2

**EXHAUST-DRIVEN BRAYTON SYSTEM**  
(PRESSURIZED/INTERCOOLED)

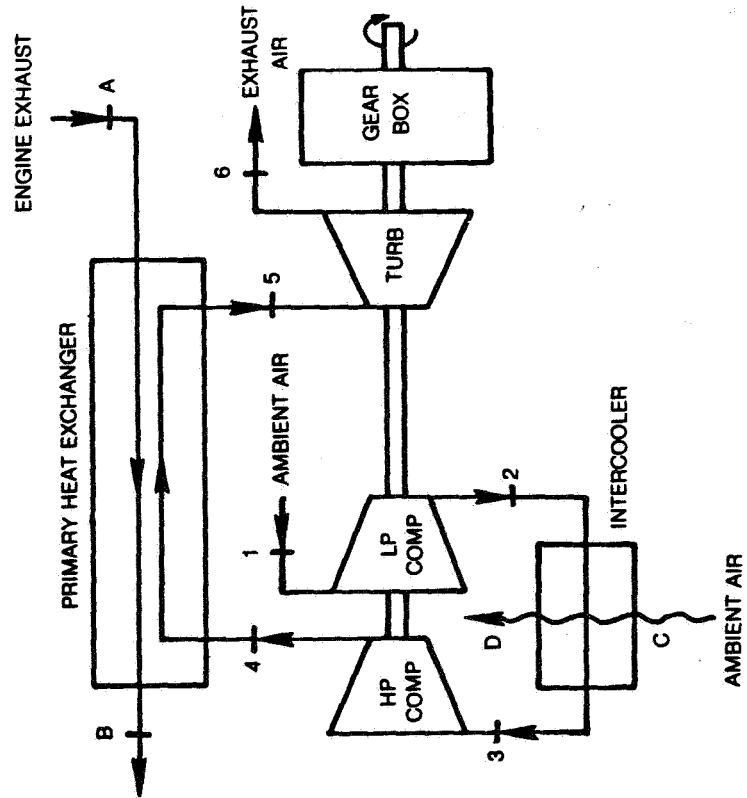
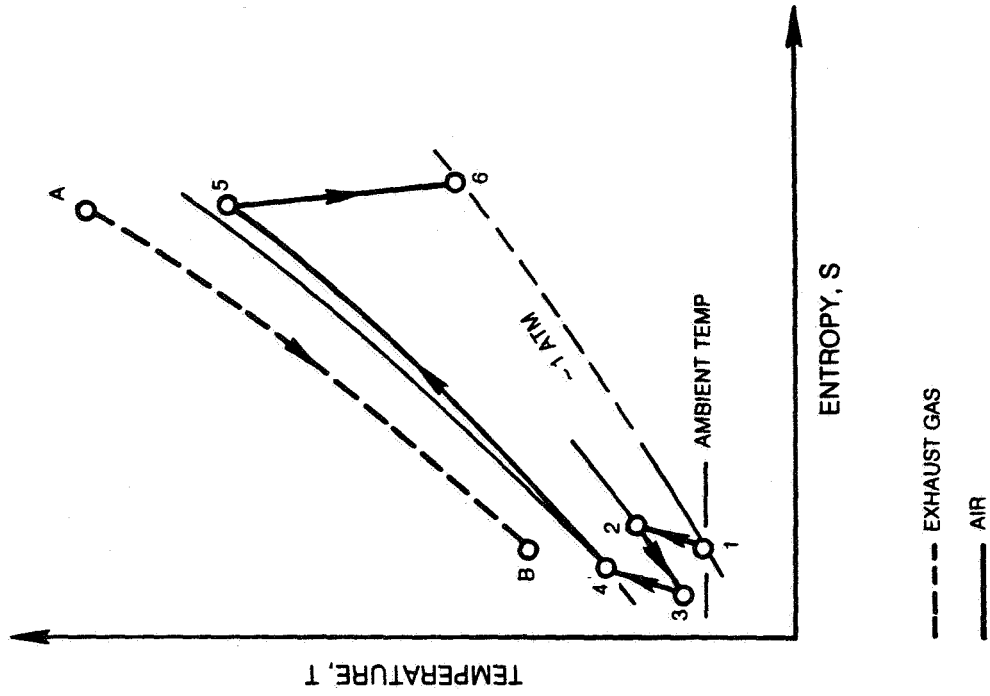


FIG. 2.3

### 2.2.3 Turbocomachinery Arrangement

The simplest arrangement of a subatmospheric or a pressurized BBS is one which employs a single-shaft turbine/compressor unit with a single-stage turbine and a single-stage compressor. The subatmospheric version of this arrangement is shown in Fig. 2.1. Of intermediate complexity are the two arrangements shown in Figs. 2.2 and 2.3 which comprise a single-shaft turbine/compressor unit with a single-stage turbine and an intercooled two-stage compressor. A more complex arrangement which may offer additional design flexibility could employ a two-shaft turbine/compressor that comprises a compressor and its drive turbine on one shaft and a free-power turbine on another shaft.

### 2.2.4 Integration with Engine

The above-mentioned variations constitute Brayton bottoming systems that are independent of the engine turbocharging unit. Additional variations can be configured in which the engine turbocharger (or supercharger drive) is integrated with the BBS. However, because the design of the turbocharger unit is intimately connected with the design of the engine itself, it was deemed more appropriate in this program to consider the turbocharger as part of the engine and to focus the effort on Brayton systems that receive the hot gases from the engine after these gases have passed through the turbocharger turbine.

Brayton bottoming systems must be operated at very high speeds in order to achieve high performance. A speed reducing device must be interposed between the BBS and engine shaft. A constant ratio reduction gearbox is by far the simplest device. The main disadvantage of such a device is that it will force the BBS to run at a fixed multiple of the engine speed which may not be the optimum BBS speed under the corresponding engine exhaust temperature and flow rate. A variable speed reducer with a suitable controller will alleviate this drawback, albeit with additional complexity and cost. A variable speed reducer for BBS applications could comprise a constant ratio reduction gearbox, followed by a simple, low-ratio variable speed reducer, e.g., a belt and cone drive. Regardless of the type of speed reducing device used, a freewheeling clutch must be interposed between the output shaft of the speed-reducer and the engine in order to ensure that the engine will not drive the bottoming system when the latter is not generating power.

### 2.3 Qualitative Comparison of BBS Configurations

The two basic versions of BBS differ in a number of details that can significantly affect their suitability for use with diesel engines in general and insulated diesel engines in particular. The most significant difference is that the engine exhaust gas passes directly through the BBS turbomachines in the subatmospheric version (the direct BBS), whereas only air passes through the turbomachines of the pressurized version (the indirect BBS). The implications of this are discussed briefly in the following paragraphs.

### 2.3.1 Response Time

Although both versions of the BBS are expected to have much faster response to engine modulation than Rankine cycle systems, the direct (subatmospheric) BBS is expected to have the fastest response. This is because no metallic parts with large thermal inertia intervene between the hot engine exhaust and the active components of the BBS in this case. In fact, because the heat exchanger surface will start from a colder state, the start up transient power output of the subatmospheric BBS may exceed its steady-state output for a brief interval of time.\* The primary heat exchanger surface material intervenes between the hot engine exhaust and the working fluid (air) in the pressurized BBS, which will result in a slower response compared with the subatmospheric BBS.

The fast response time of the subatmospheric BBS places it nearly on an equal footing with turbocompounding in this regard. It is an advantage that is not shared by any of the indirect bottoming systems, particularly Rankine cycle systems, which employ dense working fluids with a large thermal inertia.

### 2.3.2. Exposure to Exhaust Gases

Because diesel engine exhaust gases contain particulates, sulfur oxides and water vapor, fouling and corrosion may affect those parts of a diesel bottoming system that come in contact with these gases. Oxidation and high temperature corrosion may attack the exposed hot parts of the system, whereas acid corrosion may attack the low-temperature parts, unless the materials are carefully selected or unless special provisions are made to protect the vulnerable surfaces.

The two basic BBS configurations differ considerably in their vulnerability to corrosion and fouling. In the pressurized BBS, only the hot-side of the primary heat exchanger will be exposed to the hot engine exhaust. Fouling and corrosion, if they occur, will be confined to this part of the system. Being exposed directly to the hot gases discharged by the engine, the hot section of the primary heat exchanger must be fabricated from a suitable material. Low-temperature acid corrosion is not likely to be a serious problem in this case because the gas temperature remains relatively high, as indicated in the representative T-S diagram of Fig. 2.3.

The situation is quite different in the subatmospheric BBS, where the turbine, the hot side of primary heat exchanger, compressor, and, if one is used, the hot-side of the intercooler will all be exposed to the engine exhaust. Problems of high-temperature corrosion, if present, will be confined to the turbine. As shown in the T-S diagram of Figure 2.2, the hot end of the primary heat exchanger will be exposed to gases whose temperature has been reduced in the turbine. This will diminish the

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\* The heat exchanger will appear to have a higher effectiveness during start-up or sudden loading of the BBS.

probability of high-temperature corrosion in the heat-exchanger. However, the exhaust gases must be cooled in the heat exchanger to as low a temperature as possible in order to lower the compressor power consumption and improve system efficiency (see Fig. 2.2). Consequently, the cold section of the system, viz., primary heat exchanger cold end, compressor and intercooler (if any), will be more vulnerable to acid corrosion and associated fouling.

### 2.3.3 Relative Size

The operating pressure of the subatmospheric cycle will be much lower than that of the pressurized cycle. This will lead to larger turbomachinery and heat exchangers in the subatmospheric cycle. Although turbomachinery cost generally increases with size, larger turbomachines tend to be more efficient than small ones. The net effect of these two opposite trends can be determined only through a detailed design and life-cycle cost analysis.

### 2.3.4 Compatibility with Engine

As mentioned earlier, the subatmospheric BBS represents a logical extension of the turbocompounding concept. Because of the lack of a physical separation between the gas flow paths in the engine and the BBS in this case, the latter cannot be treated as a simple add-on. Unless the subatmospheric BBS is carefully matched to the engine over their common operating range, the BBS may impose adverse back pressures on the engine that could lower the overall performance. This strongly suggests that, like the exhaust turbines in turbocompounded engines, the subatmospheric BBS must be designed simultaneously with the engine and the combination should be optimized as one composite system. This may offer opportunities for a combined design in which the engine and the BBS complement each other synergistically.

Integrating a pressurized cycle with the engine is much less restrictive owing to the physical separation between the gas flow paths. In fact, at least insofar as the fluid path design is concerned, the pressurized BBS can be designed as an add-on package to be connected to the engine exhaust pipe and linked mechanically with the engine shaft.

### 2.3.5 Concluding Remarks

On the basis of the preceding arguments, it may be concluded that the three main advantages of the subatmospheric BBS relative to the pressurized BBS are:

- 1) Faster response to engine modulation
- 2) Better high temperature capability
- 3) Higher potential for synergism in integration with engine.

The most serious weakness of the subatmospheric system resides in its higher vulnerability to acid corrosion and fouling and the potential adverse impact of these problems on engine back pressure.

The main advantages of the pressurized BBS relative to the subatmospheric BBS are:

- 1) Lower vulnerability to acid corrosion and fouling
- 2) More compact size
- 3) Lower potential for affecting the engine adversely
- 4) Wider design flexibility.

The last two advantages stem from the add-on nature of the pressurized BBS.

This qualitative comparison suggests that the design of a pressurized BBS will be less problematic and would, therefore, have a better chance for success than the design of a subatmospheric BBS, despite the attractive advantages of the latter.

### 3. INSULATED DIESEL ENGINE CHARACTERISTICS

Insulated diesel engines achieve their high thermal efficiency and increased exhaust temperature and energy content by eliminating a large fraction of the coolant heat loss. This can be accomplished by substituting low-thermal-conductivity ceramics for the conventional materials used in the fabrication of hot engine parts such as the combustion chamber walls, piston crown, cylinder liners, valves and valve guides and exhaust ports. Major developments in insulated engine technology by Cummins and others have focused on eliminating the engine primary cooling system, providing only for the cooling of the lubricating oil.

#### 3.1 Design Features

True adiabatic operation of an internal combustion engine can only be achieved through perfect insulation, including the total elimination of oil cooling. The development of such an engine is a formidable task that is contingent upon the availability of ceramics and lubricants that can withstand the high pressures and temperatures inside fully insulated engines. Much of the current adiabatic engine development effort is focused on semi-insulated engines that can operate without a cooling jacket but still use lubricating oil and possibly an oil cooler. Insofar as power generation through exhaust energy utilization is concerned, the degree of insulation, DI, of an engine can be measured by

$$DI = (W + Q_e)/Q_f \quad (3.1)$$

in which  $W$  is the fraction of fuel energy,  $Q_f$ , converted into shaft work and  $Q_e$  is the fraction remaining in the exhaust gases. The degree of insulation is unity for fully insulated (adiabatic) engines. The degree of insulation for a conventional diesel engine is about 0.7; semi-insulated engines fall between these two limits.

If no exhaust energy utilization is contemplated, the thermal efficiency of the engine ( $W/Q_f$ ) becomes the most significant performance factor. However, when exhaust-driven bottoming systems are added to the engine, the efficiency of the combination is measured by the overall conversion efficiency

$$\eta_o = (W + \eta_b Q_e)/Q_f ; \quad (3.2)$$

$$\eta_o = \eta_e + \phi \eta_b , \quad (3.3)$$

where  $\eta_e$  is the engine thermal efficiency,  $\eta_b$  is the bottoming system thermal efficiency and  $\phi$  is the fraction of energy discharged from the engine with the exhaust gases. In such an arrangement, engine modifications that may increase  $\eta_e$  may not lead to higher  $\eta_o$  if they adversely affect the performance of the bottoming system at either full-load or part-load

conditions. An example of this circumstance is the use of a supercharger aftercooler. It is known that the use of such a device in a conventional engine both increases volumetric efficiency and improves thermal efficiency. Aftercooling, however, results in a lower exhaust temperature, which could mean lower bottoming system output, and possibly lower overall output for aftercooled engines equipped with bottoming systems.

Recognizing the importance of engine/bottoming system inter-relation, UTRC decided not to base the design of the bottoming system on a single set of insulated engine data and decided, instead, to consider several variations of insulated diesel engine design, with and without supercharger aftercooling and covering a range of degrees of insulation. Pertinent data on insulated diesel engines were obtained from two sources:

- (a) NASA, who provided UTRC with the full-load performance characteristics of turbocharged (TC), turbocharged and aftercooled (TC/A), turbocompounded (TCPD) and turbocompounded and aftercooled (TCPD/A) engines.
- (b) John Deere Product Engineering Center, who, under subcontract to UTRC, provided typical full- and part-load performance characteristics of turbo-charged semi-insulated engines with and without aftercoolers as well as for a hypothetical near adiabatic without aftercoolers as well as for a hypothetical near adiabatic "fully insulated" engine.

Given below are the pertinent performance data for these engines.

### 3.2 Engine Performance Characteristics

An engine performance map provides contours of constant brake specific fuel consumption (bsfc) in a torque-speed plane. Combined with the vehicle and drive train characteristics, this map can be used to calculate the fuel consumption of the vehicle over a specified driving cycle. However, in order to design an exhaust-driven bottoming system and match it with the engine over its likely operating domain, the engine exhaust flow rate, temperature and thermal and chemical characteristics must also be provided over the operating domain. In addition to this information, the influence of engine back-pressure on engine performance must also be given because the addition of a bottoming system may result in a higher engine back pressure.

#### 3.2.1 Engine Data Provided by NASA

The engine data provided by NASA-LeRc reflect the performance of turbocharged semi-insulated engines in which about 60 percent of the energy that would normally be lost to the coolant has been retained in the combustion products owing to the elimination of the water jacket. All these engines are in the 300-350 hp class suitable for powering heavy duty (class-8) long-haul trucks. The data provide the full-rack performance characteristics of four versions of one such engine for both aftercooled and nonaftercooled configurations with or without turbocompounding. These data cover only a limited range of engine operation extending from the maximum torque point (about 1300 rpm) to the maximum power point (about 1900 rpm) at full-load (full rack position). The performance characteristics of these engines at their rated output are summarized in



Table 3.1. The torque rise for these engines is about 15 percent which is typical for heavy-duty truck applications. The degree of insulation of these engines was estimated to range from 0.83 for the TC/A engine to 0.88 for the TC engine.

### 3.2.2 Engine Data Provided by Deere

Much more detailed information was provided to UTRC by the John Deere Product Engineering Center on conventional and insulated engines. The data were generated from a computer simulation of a turbocharged six-cylinder direct-injection diesel engine. The approximate physical characteristics of the baseline engine block are given in Table 3.2. The rated output of the engine is about 300 bhp at a speed of 1800 rpm; the torque rise is about 20 percent at 1300 rpm (maximum torque point).

Several versions of this baseline engine were modeled. In all cases the engine and turbocharger characteristics were chosen so that the engine will develop 300 bhp at 1800 rpm with a 20 percent full-rack torque rise at 1300 rpm. Because a high exhaust temperature after the turbocharger turbine is desirable for the operation of a Brayton bottoming cycle, the air to fuel ratio was decreased in some cases to as low as 21:1, which still provides sufficient excess air to ensure efficient, smokeless combustion.

Several engine design iterations were performed during the course of this study. These iterations were guided by parallel calculations of the Brayton bottoming system performance with the goal of increasing the combined output of the engine and bottoming system. Starting with the conventional baseline engine, the degree of insulation was increased in three steps toward a near-adiabatic design. The degree of insulation of the conventional baseline turbocharged, after-cooled engine was estimated to be about 0.67; the degree of insulation of the insulated engines are as follows:

Engine	Degree of Insulation
IDE-1: Semi-insulated-I (TC and TC/A)	0.79
IDE-2: Semi-insulated-II (TC)	0.87
IDE-3: Near-adiabatic (TC)	0.92

It can be seen that the three insulated engines cover a range of insulation both lower and higher than that of the NASA engines; the second engine (IDE-2) has about the same degree of insulation as the corresponding NASA engine. The characteristics of all these engines at the full rack position between maximum speed (1800 rpm) and peak-torque speed (1300 rpm) were obtained, together with the part-load performance data for the first insulated engine (IDE-1) without the aftercooler. The part-load characteristics of the second semi-insulated engine (IDE-2), which is a highly turbocharged, better insulated version of the first engine, were obtained from a combination of direct computer simulation and judicious scaling of the IDE-1 data. Table 3.3 presents a comparison of the important characteristics of the TC and TC/A insulated engines provided by both NASA and Deere. It is evident that the TC NASA engine and the IDE-2 TC engine are fairly similar in characteristics, at least with full power operation, especially when the flow is adjusted for the difference in Bhp.

TABLE 3.1

## CHARACTERISTICS OF NASA ENGINES AT RATED OUTPUT

Parameter Engine	TC	TC/A <sup>+</sup>	TCPD	TCPD/A <sup>+</sup>
Max. Bhp	317	320	335	340
Speed (rpm)	1900	1900	1900	1900
Bsfc (lb/Bhph)	0.315	0.310	0.297	0.293
Exhaust Flow (lb/min)	48.0	47.5	47.8	48.4
A/F Ratio	27.8	27.7	27.8	28.1
Exhaust Temp. * (°F)	1240	1120	1140	1060
Degree of Insulation	0.88	0.83	0.86	0.84

<sup>+</sup> Charge air cooled down to 140 F after compressor.

\* Outlet of turbocharger or turbocompounding turbine.

TABLE 3.2

## CHARACTERISTICS OF INSULATED DIESEL ENGINES

Type	Turbocharged
No. of Cylinders	6
Bore (inch)	5.12
Stroke (inch)	5.51
Compression Ratio	16:1
Intake Valve Diam. (in.)	1.72 x 2
Exhaust Valve Diam. (in.)	1.72 x 2
Torque Rise (from 1800 to 1300 rpm)	20%
Maximum Power (hp)	300
Speed @ Max. Power (rpm)	1800

TABLE 3.3

COMPARISON OF FULL-POWER PERFORMANCE  
OF INSULATED DIESEL ENGINES

Parameter Engine	NASA-TC	NASA-TC/A	IDE-1/TC	IDE-1/TC/A	IDE-2/TC	IDE-3/TC
Max. Bhp	317	320	300	300	300	300
Speed (rpm)	1900	1900	1800	1800	1800	1800
Bsfc (lb/Bhph)	0.315	0.310	0.315	0.304	0.315	0.289
Exhaust Flow (lb/min)	48.0	47.5	36.5	33.5	43.8	33.1
A/F Ratio	27.8	27.7	22.1	21.0	26.7	21.9
Exhaust Temp. *	1240	1120	1265	1240	1260	1510
Degree of Insulation	0.88	0.83	0.79	0.78	0.87	0.92

---

\* Outlet of turbocharger turbine

The NASA TC/A and the IDE-1/TC/A engines provide an interesting contrast; the former discharges 47.5 lb/min of exhaust at 1120 F, whereas the latter discharges only 33.5 lb/min of exhaust at a higher temperature of 1240 F. The exhaust energy amounts nearly to 2440 Btu/bhph in the first case and to 2060 Btu/bhph in the second case. This may appear to be an advantage of the NASA engine over the Deere engine when used in conjunction with bottoming systems that are not very sensitive to variations in exhaust temperature above a certain level (e.g., Rankine cycles). However, bottoming systems whose performance depends sensitively on the exhaust gas temperature (e.g., Brayton cycles) may show greater merit when used with an engine like the IDE-1/TC/A engine, despite its lower exhaust energy content.

Figure 3.1 depicts the bsfc contours for the IDE-1/TC engine. The variations of the exhaust temperature flow rate, Bhp and bsfc with speed for this engine at full-load is shown in Fig. 3.2. Within the uncertainty limits of the engine simulation model, Figs. 3.1 and 3.2 can be taken to provide adequate representations of the performance data for the highly supercharged, better insulated version of the same engine, i.e., IDE-2/TC engine, except for the exhaust flow rate which would be about one fifth higher in the second engine.

### 3.2.3 Particulate Emissions from Insulated Engines

Excessive emission of particulate matter (soot) presents a major problem in the design of exhaust-driven bottoming systems for diesel engines. The adhesion of these particulates to heat exchanger surfaces results in heat exchanger fouling with an accompanying decrease in heat exchanger effectiveness and increase in pressure drop. Higher heat exchanger pressure drop, in turn, leads to higher engine back-pressure and lower engine performance. Although the exact mechanism of particulate formation in diesel engines is not reliably known, there is sufficient evidence to suggest that particulate emissions from insulated diesel engines would be considerably lower than those from conventional diesel engines owing to the former's much higher operating temperatures (Ref. 1.7).

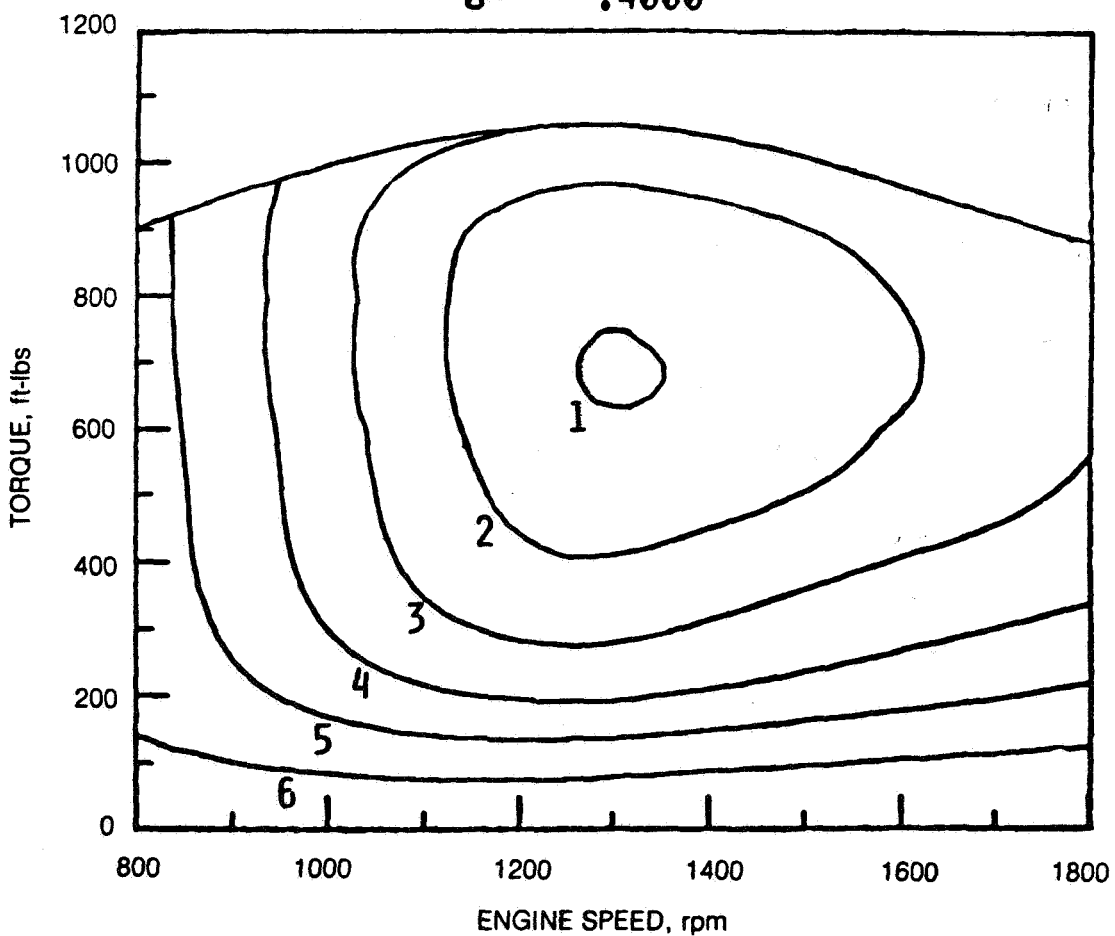
Other efforts are underway to reduce the particulate emissions from diesel engine through the use of fuel additives and catalysts and through modifications in the engine fuel injection system and combustion chamber design. These efforts are primarily motivated by the need to reduce the deleterious environmental effects of particulate emissions but will also prove very beneficial to the development of cost effective diesel bottoming system.

SFC CONTOUR MAP

CONTOUR LEVELS

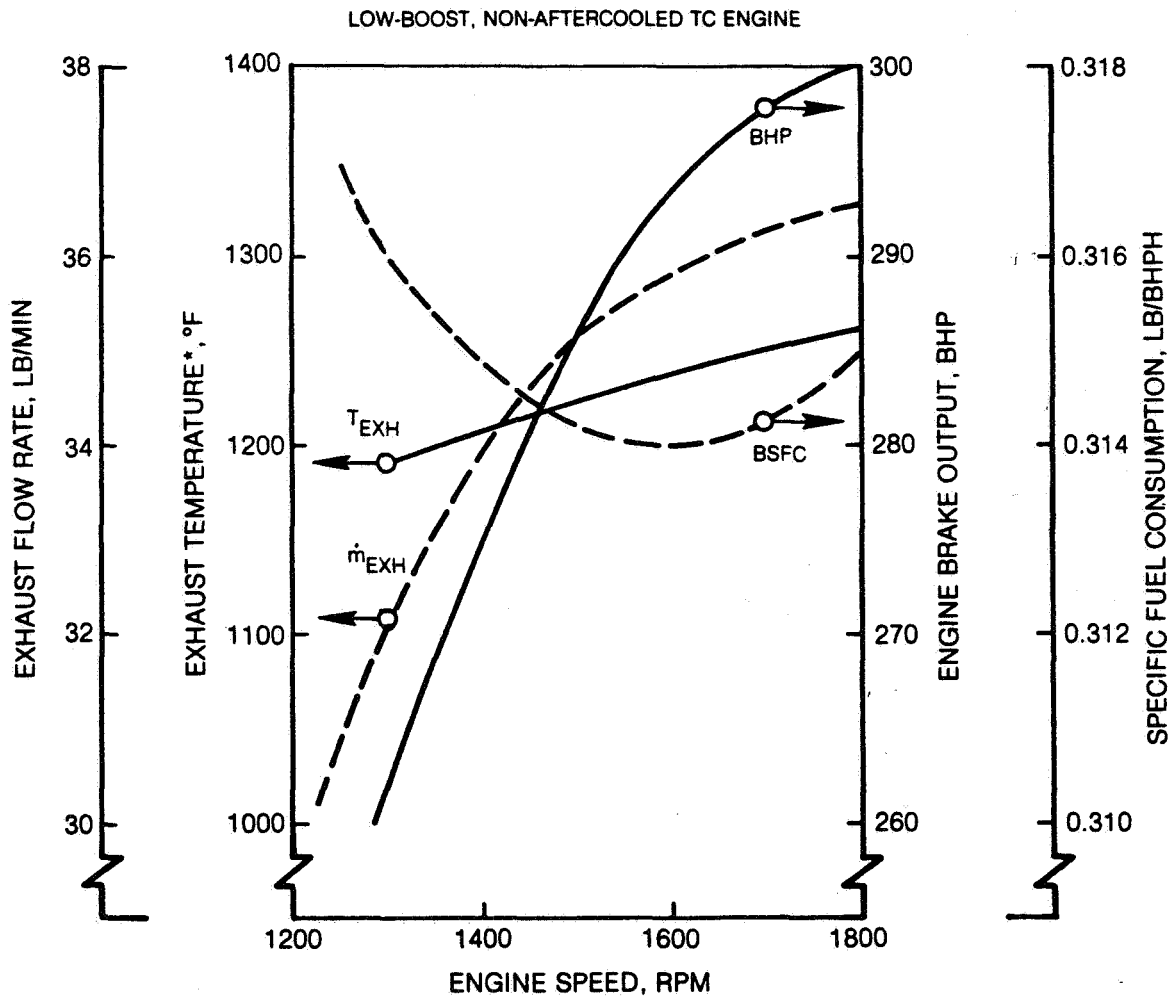
NO. SFC  
(LB/HP-HR)

- 1- .2970
- 2- .3030
- 3- .3120
- 4- .3250
- 5- .3450
- 6- .4000



**FULL-RACK PERFORMANCE OF DEERE SEMI-INSULATED ENGINE**

(IDE-1/TC ENGINE)



\*OUTLET OF TURBOCHARGER TURBINE

#### 4. PARAMETRIC ANALYSIS OF BOTTOMING SYSTEM PERFORMANCE

The performance and cost of BBS depend on a large number of internal and external design factors each of which may vary over a wide range. The most important external factors include exhaust gas flow rate and temperature, and ambient air temperature; the most important internal factors include cycle pressure ratio, heat exchanger minimum temperature difference, heat exchanger pressure drop and turbomachinery efficiency. In a design study, the influence of these factors on system performance and equipment cost must be assessed so that the optimum design parameters can be determined.

A detailed optimization study in which all the internal design parameters are allowed to change in search of the economically optimum values is beyond the scope of this project. Nevertheless, much can be learned about the sensitivity of the design to changes in the design parameters, and about the approximate values of the optimum parameters by performing an analysis of the system performance in which the values of the key design variables are parametrically varied over suitable ranges.

##### 4.1 Assumptions and Groundrules

The parametric analysis was performed for both the subatmospheric and the pressurized BBS with and without compressor intercoolers. The calculations were performed first for the baseline case characterized in Table 4.1 and subsequently for cases in which the key parameters were varied one at a time. Because of the higher gas density in the pressurized cycle, and hence the smaller turbomachinery size, the efficiency of the turbomachines in this case were chosen to be somewhat lower than the values chosen for the subatmospheric cycle. Figure 4.1 depicts the effect of size on the relative efficiency of turbomachines.

##### 4.2 Analysis Methodology

In order to allow maximum flexibility in the use of the parametric data, the performance of the bottoming system was expressed as an efficiency increment,  $\Delta\eta$ , such that

$$\Delta\eta = \Delta W/Q_f \quad (4.1)$$

in which  $\Delta W$  is the incremental power output of the BBS and  $Q_f$  is the rate of fuel energy input to the engine. In terms of the BBS specific output,  $\Delta w$  (per unit exhaust flow rate), the air/fuel ratio,  $A/F$  and the fuel heating value (HV), the efficiency increment can be written as

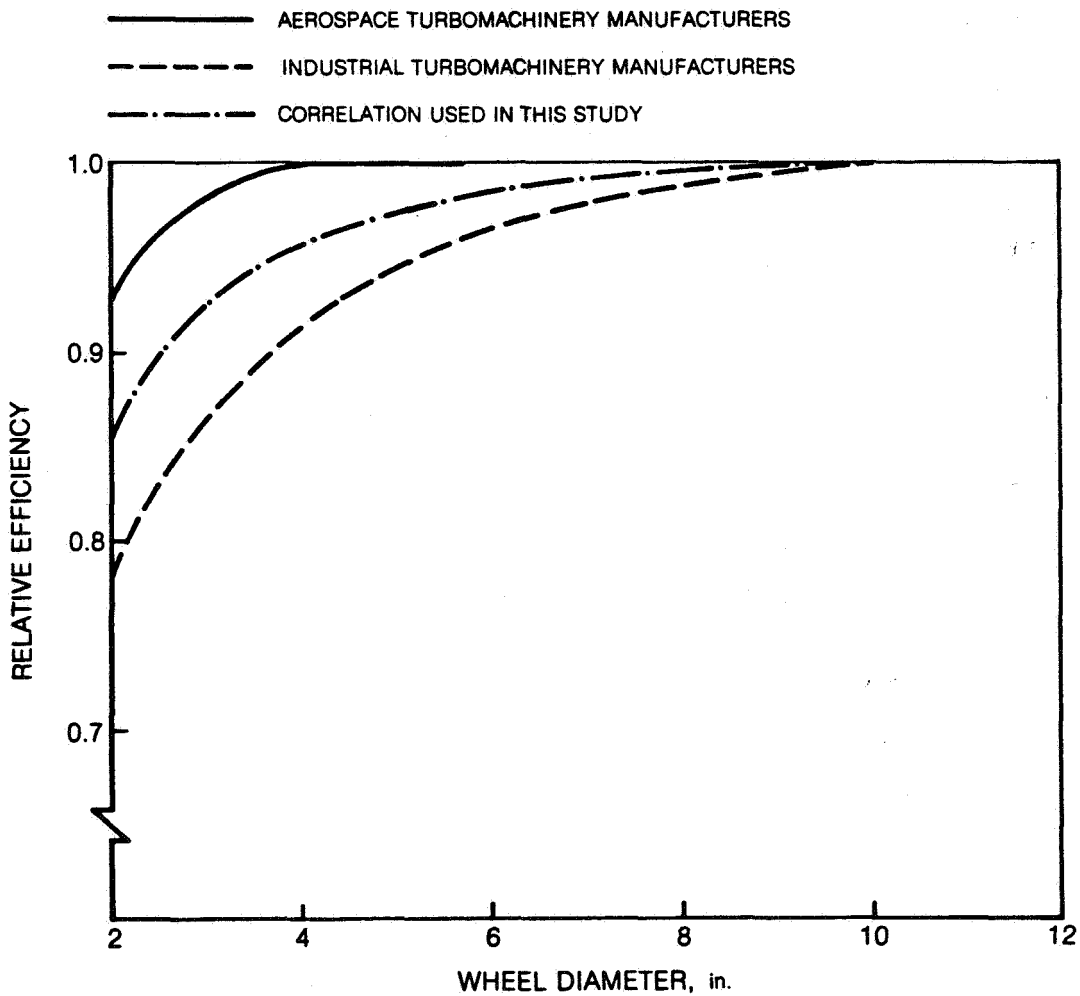
$$\Delta\eta = \Delta w \cdot [(A/F) + 1]/(HV), \quad (4.2)$$

which indicates that for given values of  $(A/F)$  and  $HV$ ,  $\Delta\eta$  will be directly proportional to the specific output of the BBS.





### EFFECT OF SIZE ON RADIAL TURBOMACHINERY EFFICIENCY



For each parameter, the efficiency increment was computed as a function of the cycle pressure ratio,  $P$ , for both simple and intercooled subatmospheric and pressurized cycles. Figure 4.2 presents the results of these computations for a subatmospheric cycle with the exhaust temperature as the parameter. Similar computations were performed to quantify the effects of the heat exchanger minimum temperature difference, turbomachinery product efficiency, cycle pressure losses, and the engine air/fuel ratio. For the purpose of this parametric analysis, it was assumed that the temperature difference is uniform in the heat exchanger of the pressurized cycles.

#### 4.3 Parametric Results

Figures 4.3 through 4.7 display the loci of the maxima in the  $\eta$  plots for typical ranges of system design parameters. Figure 4.3 and Table 4.2 present the effect of exhaust temperature on BBS output. The results shown in Fig. 4.4 (effect of heat exchanger minimum temperature difference,  $\Delta T_{\min}$ ) are based on equal  $\Delta T_{\min}$  in both the primary heat exchanger and intercooler of a subatmospheric BBS and a fixed  $\Delta T_{\min}$  of 50 F in the intercooler of a pressurized BBS.

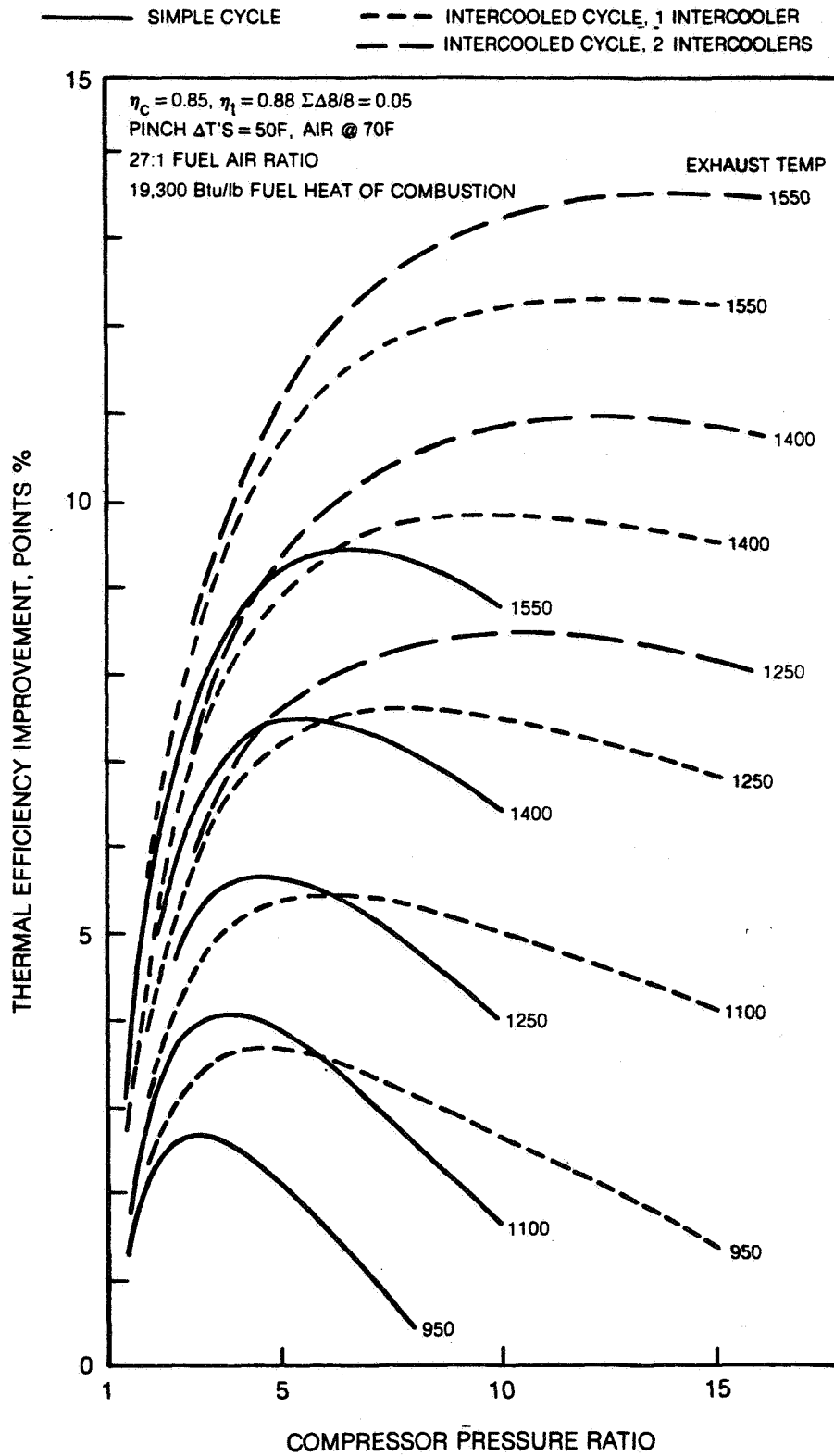
Figure 4.5 indicates that for the same  $\Delta T_{\min}$  and machinery efficiency, the pressurized BBS is inherently more efficient than the subatmospheric BBS. This is because of the occurrence of  $\Delta T_{\min}$  at the high temperature end of the cycle in pressurized BBS and at the low temperature end, where it constitutes a severe thermodynamic penalty, in the subatmospheric BBS. (See Figs. 2.2 and 2.3). However, because of its larger and more efficient turbomachines, the subatmospheric BBS appears to offer a slight performance advantage over the pressurized BBS at low values of  $\Delta T_{\min}$  (< 100 F).

Figures 4.3 through 4.7 also indicate that the intercooler offers a significant performance improvement, particularly in the case of the subatmospheric cycles, whose performance is much more sensitive to changes in the compressor power consumption. On balance, however, there is no decisive performance advantage of one type of BBS over the other, except insofar as the effect of heat exchanger minimum temperature difference is concerned; the pressurized cycle shows a definite advantage over the subatmospheric cycle at values of  $\Delta T_{\min}$  exceeding 150 F.

#### 4.4 Performance Limitations

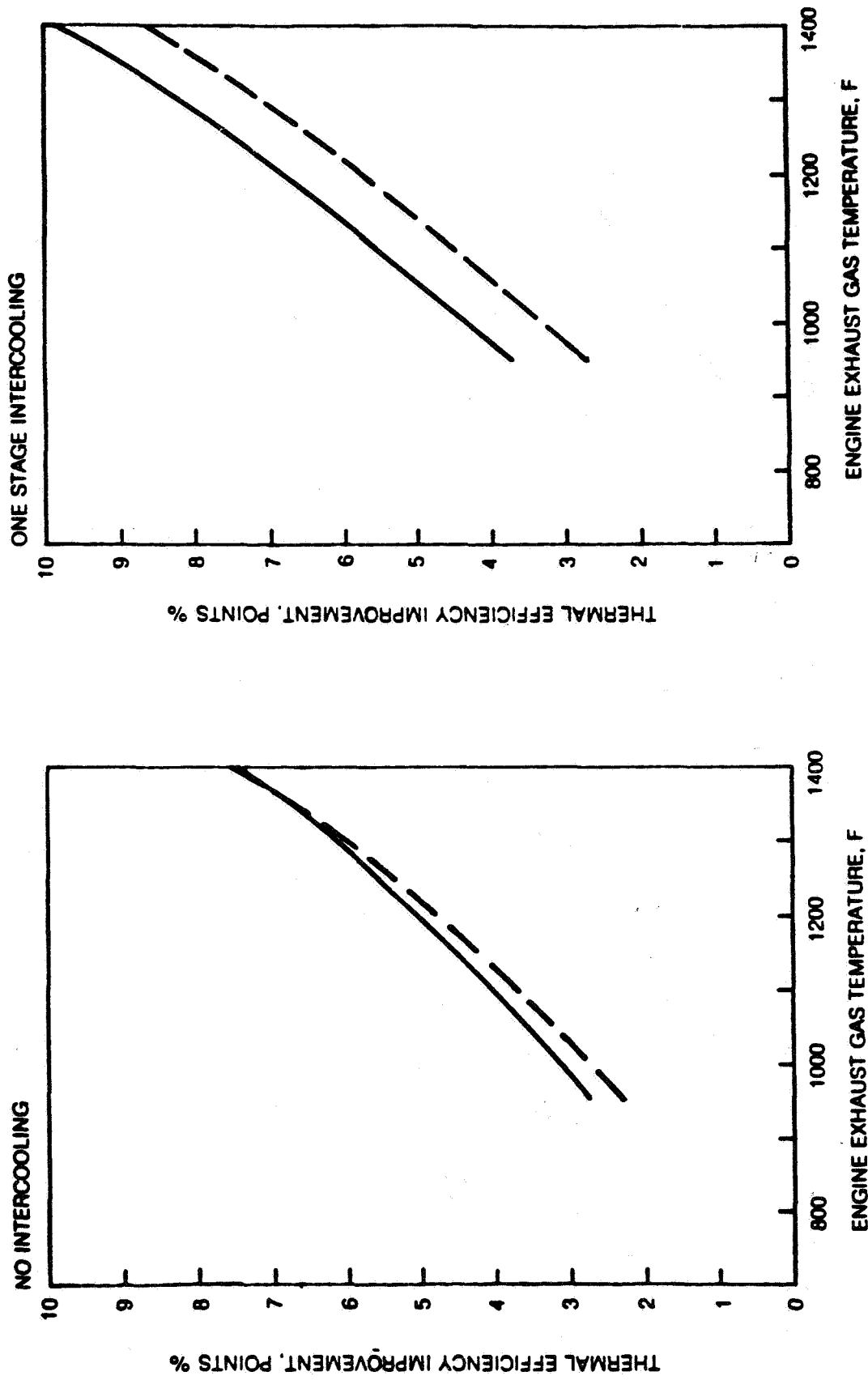
The results reported in the preceding paragraphs were obtained parametrically, paying only cursory attention to the limitations of the key components of the system. In many cases, such limitations can restrict the range of acceptable values of the design parameters and will, consequently, influence the design process. Section 5.1 presents a discussion of the most important limitations and assesses their influence on BBS selection and design. Among the most important of these limitations are those pertaining to heat exchanger fouling and corrosion, which could restrict the minimum temperature difference in the subatmospheric BBS heat exchangers to values higher than 200 F to ensure that the surface temperature exceeds the exhaust gas acid dew point. The effect of this restriction is to lower the performance of the subatmospheric BBS quite significantly, as can be seen in Fig. 4.4.

**EFFECT OF EXHAUST TEMPERATURE AND PRESSURE RATIO ON THE EFFICIENCY OF SUBATMOSPHERIC BOTTOMING CYCLES**



### EFFECT OF EXHAUST TEMPERATURE ON BBS PERFORMANCE

SYSTEM PRESSURE DROP = 5%  
 AIR/FUEL RATIO = 27:1 MINIMUM HX TEMPERATURE DIFFERENCE = 50F  
 — SUBATMOSPHERIC CYCLE,  $\eta_c/\eta_t = 85\%/88\%$   
 - - - PRESSURIZED CYCLE,  $\eta_c/\eta_t = 83\%/84\%$

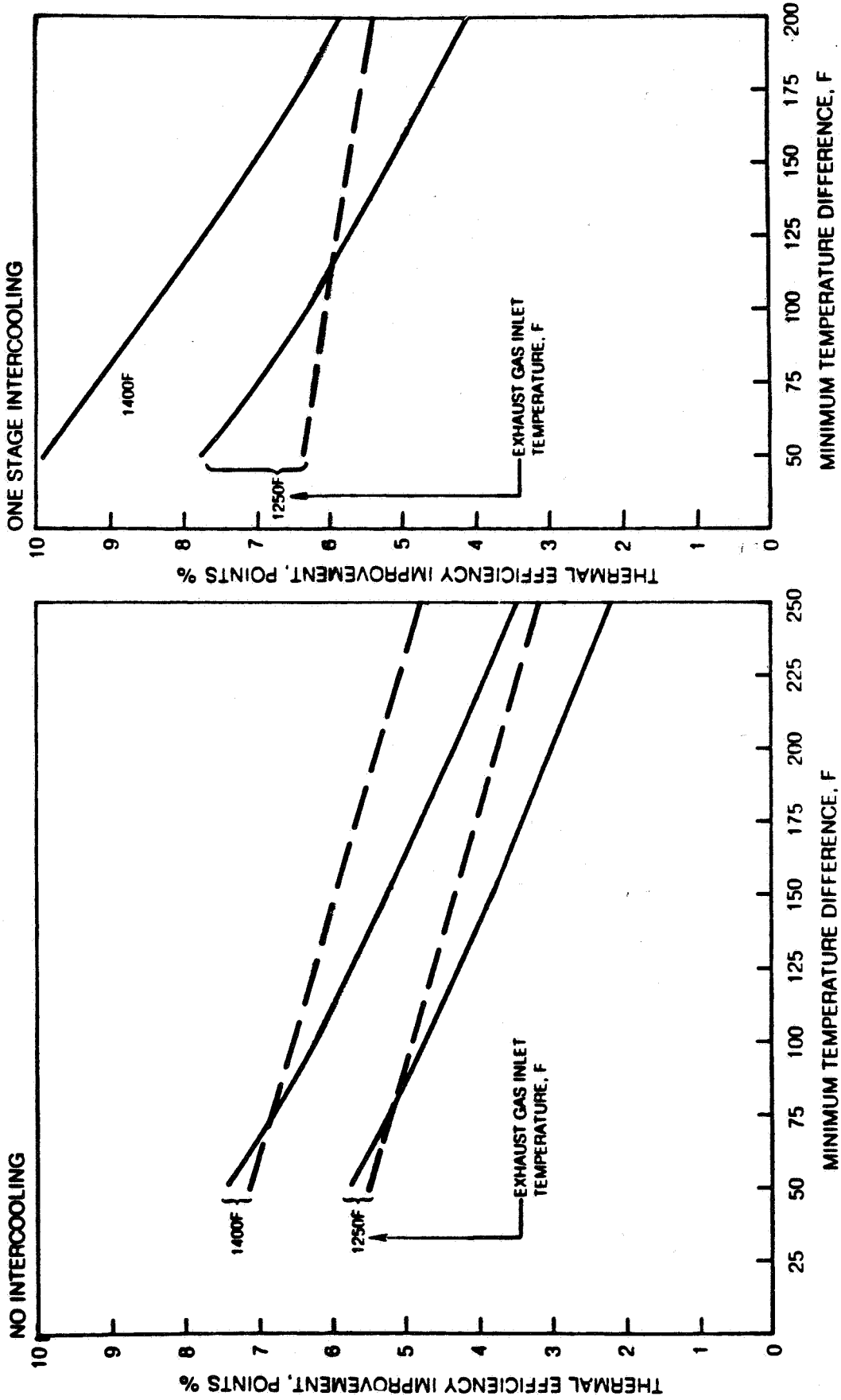


1-9-1-28

**EFFECT OF HX MINIMUM TEMPERATURE DIFFERENCE ON BBS PERFORMANCE**

SYSTEM PRESSURE DROP = 5%  
AIR/FUEL RATIO = 27.1

- SURATMOSPHERIC CYCLE,  $\eta_c/\eta_1 = 85\%/88\%$
- - - PRESSURIZED CYCLE,  $\eta_c/\eta_1 = 83\%/84\%$



83-3-48-2

**EFFECT OF TURBOMACHINERY EFFICIENCY ON BBS PERFORMANCE**

SYSTEM PRESSURE DROP = 5%

AIR/FUEL RATIO = 27:1

MINIMUM HX TEMPERATURE DIFFERENCE = 50F

— SUBATMOSPHERIC CYCLE

- - - PRESSURIZED CYCLE

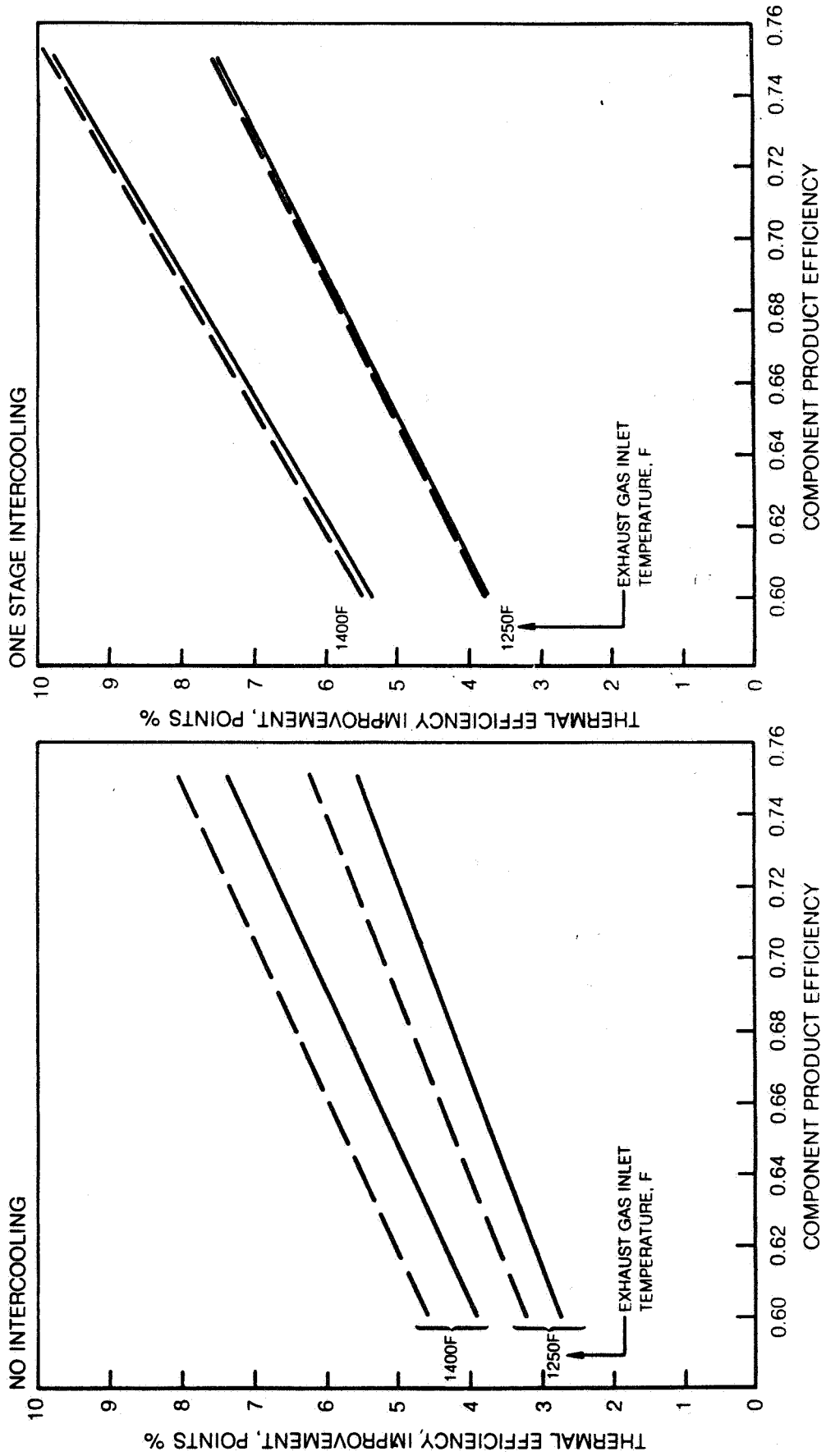


FIG. 4-6

**EFFECT OF PRESSURE DROP ON BBS PERFORMANCE**

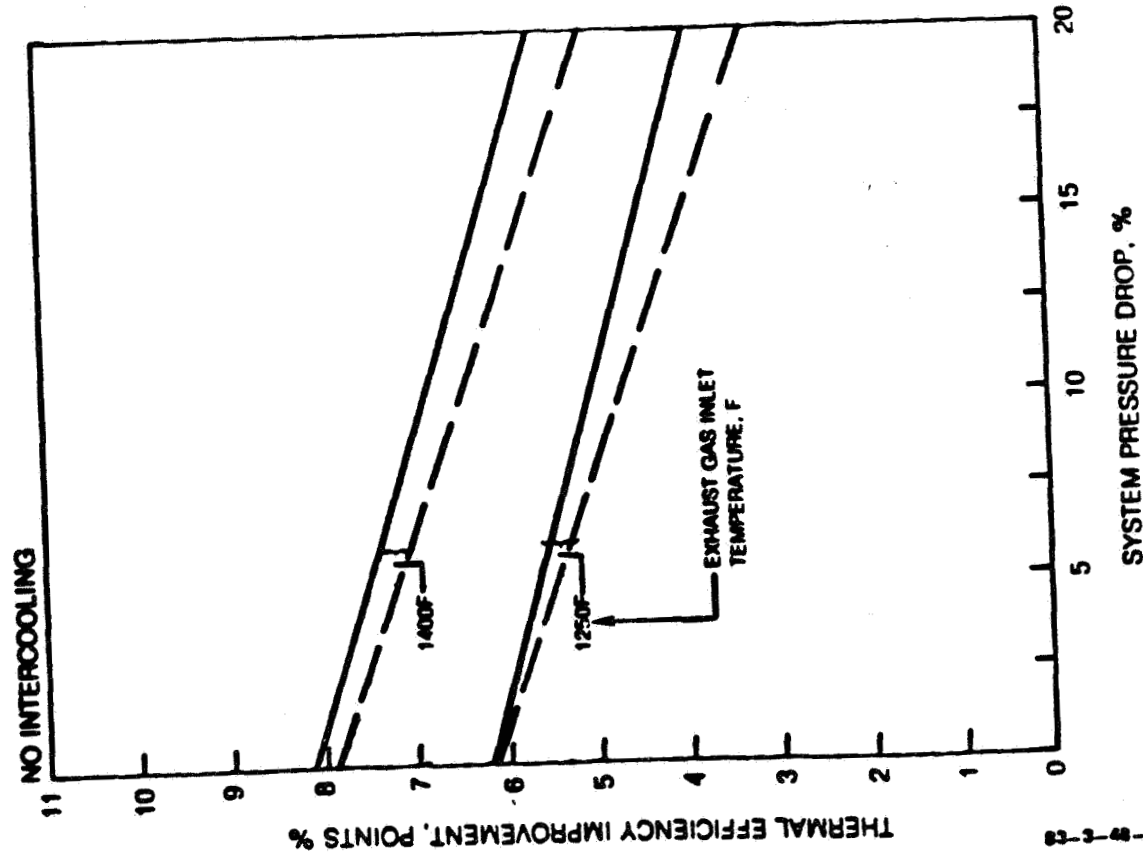
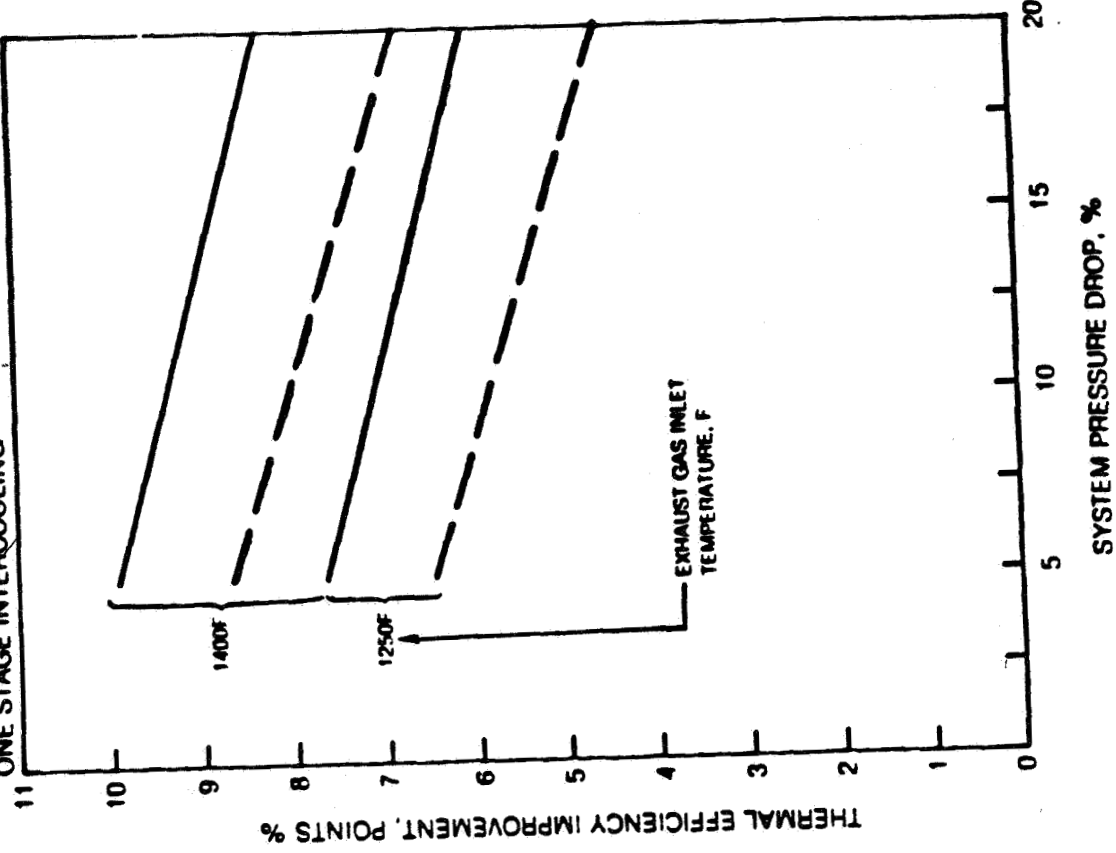
MINIMUM HX EXHAUST TEMPERATURE = 50F

SUBATMOSPHERIC CYCLE,  $\eta_c/\eta_i = 85\%/88\%$

PRESSURIZED CYCLE,  $\eta_c/\eta_i = 83\%/84\%$

— — — — —  
 — — — — —

**ONE STAGE INTERCOOLING**



83-3-48-4



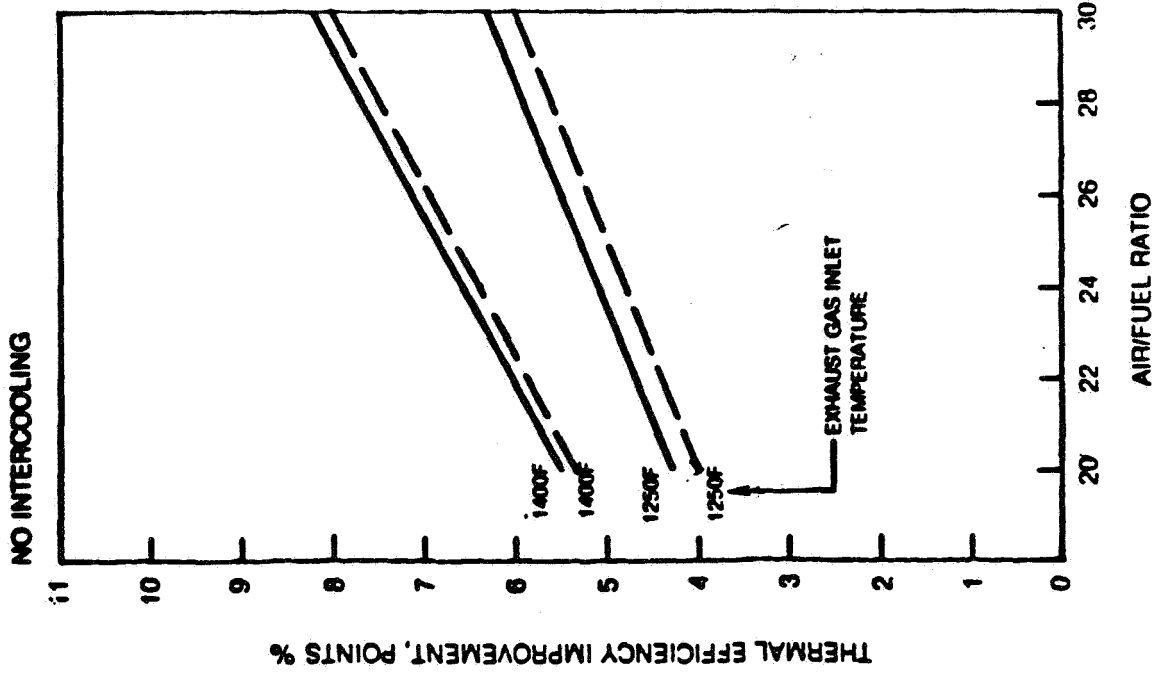
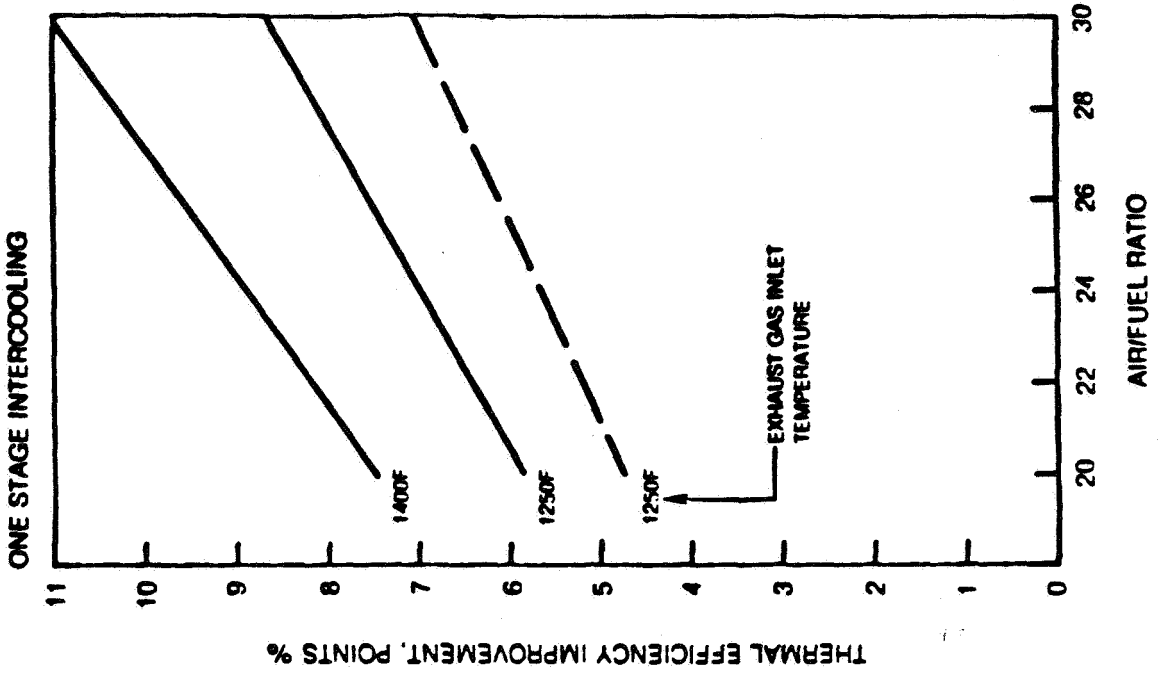
**EFFECT OF ENGINE A/F RATIO ON BBS PERFORMANCE**

SYSTEM PRESSURE DROP = 5%

MINIMUM HX TEMPERATURE DIFFERENCE = 50F

— SUBATMOSPHERIC CYCLE,  $\eta_c/\eta_1 = 85\%/88\%$

- - - PRESSURIZED CYCLE,  $\eta_c/\eta_1 = 83\%/84\%$



1-5-1-28

TABLE 4.2

## EFFECT OF EXHAUST TEMPERATURE ON BBS OUTPUT

(150 F HX Minimum Temp. Diff.)

Exh. Temp. (F)	Normalized Output		
	Pressurized*	Subatm. (a)*	Subatm. (b)**
1000	0.45	0.26	0.32
1100	0.66	0.47	0.57
1200	0.88	0.70	0.85
1300	1.12	0.94	1.15
1400	1.38	1.22	1.48
1500	1.65	1.51	1.83
1600	1.92	1.82	2.21

---

\* Relative to output of pressurized BBS with 1250 F exhaust.

\*\* Relative to output of subatmospheric BBS with 1250 F exhaust.

## 5. SYSTEM HARDWARE DESIGN ANALYSIS

The parametric results given in the preceding chapter established the sensitivity of BBS performance to the important internal and external design parameters. Unlike the external design parameters, which are imposed on the system and cannot be arbitrarily changed, the internal parameters can be varied over a fairly wide range, subject only to the physical, operational or economic limitations of the equipment. Among the numerous internal design parameters of BBS design, the heat exchanger pinch temperature difference, the heat exchanger pressure drop, and the turbomachinery efficiencies are the most important. This chapter provides a discussion of the considerations influencing the BBS hardware design and the design and optimization procedures and presents the results of applying these considerations and procedures to the design, sizing and costing of the BBS major components.

### 5.1 System Design Considerations

The considerations governing the design of a BBS for diesel exhaust energy recovery can be classified into three major categories: (1) those pertaining to heat exchanger design, (2) those pertaining to turbomachinery design, and (3) those pertaining to overall system integration.

#### 5.1.1 Heat Exchangers Design Constraints

The primary heat exchanger in both the subatmospheric and the pressurized BBS, as well as the intercooler in the subatmospheric system, will be subject to the hot exhaust gases discharged by the insulated diesel engine. Because diesel exhaust gases contain both soot and sulfur oxides, fouling and corrosion of heat transfer surfaces may occur, resulting in a precipitous deterioration in equipment performance and life.

Numerous investigations of heat exchanger performance in exhaust gas service (e.g., Refs. 5.1, 5.2, 5.3, 5.4) concluded that hydrocarbon and acid condensation on heat exchanger surfaces is often a precursor to the formation of a fouling film which adheres to the wet surface and grows in thickness. This film reduces the heat transfer efficiency and raises the temperature of the fouled surface at its interface with the gas. When the interfacial temperature exceeds the dew point, condensation ceases and the fouling film eventually reaches an asymptotic thickness. Although the presence of a fouling film reduces the heat exchanger effectiveness (and hence, the BBS performance) it has been argued that it may also protect the surface from severe corrosion by the condensing sulfuric acid. From an economic viewpoint, however, it may not be desirable to lower the heat exchanger surface temperature below the acid dew point in order to avoid the use of expensive corrosion-resistant materials or coatings. Maintaining the surface temperature above the dew point to reduce the adhesion of the fouling film is even more desirable when the exhaust gases contain a large concentration of particulates, as in the case of present truck diesel engines.

The problems of fouling and corrosion impose several limitations on the design and performance of BBS, particularly the subatmospheric system. If the surface temperature of the primary heat exchanger and the intercooler

in the subatmospheric BBS is to be maintained above the acid dew point, which is typically 200-250 F for diesel exhaust gases, the minimum temperature difference in these heat exchangers may have to be restricted to values in excess of 200 F. It can be seen from Fig. 4.4 that the subatmospheric system will be inferior to the pressurized system under these conditions, i.e., corrosion limits the performance of the subatmospheric BBS.

Even if the heat exchanger surfaces can be protected from the corrosive effect of condensed acid, lowering the surface temperature below the dew point can aggravate the fouling problem. Aside from the expected deterioration of heat exchanger performance as a result of fouling, the build-up of a fouling layer on the heat exchanger surfaces also increases the pressure drop in the heat exchanger, which, in turn, leads to a higher engine back pressure and a lower engine performance. For this reason, the minimum temperature difference in the heat exchangers of the subatmospheric BBS may still have to be high enough to ensure that the surface temperature will be higher than the dew point value.

If components of the BBS are to operate below the dew point temperature, measures must be taken to counteract the harmful effects of corrosion and fouling. Special corrosion resistant materials, such as Hastelloy and several coating such as TEFLON, have been proposed for service in corrosive environments but do not appear at present to be economically feasible, especially in the case of complex geometries such as those to be found in plate-fin heat exchangers and three-dimensional turbomachinery passages.

The effect of fouling on the performance of the system can be reduced through frequent cleaning and the proper selection of the heat exchanger flow passages to avoid plugging. Frequent cleaning represents an additional maintenance expense which, if excessive, may render the system uneconomical. Several cleaning methods have been tried and range from soot blowing by a gas blast (e.g., so-called burp system in trucks), to baking the heat exchanger at high temperature ( 1000 F) to burn-off the fouling film. So far, washing the heat exchanger with water appears to be the most reasonable approach (Ref. 5.5).

Another important constraint on the design of BBS heat exchanger arises from the high temperature limitations of the heat exchanger material. Because the heat exchanger in the pressurized BBS is directly exposed to the hot exhaust discharged by the insulated engine, high temperature corrosion is likely to be more of a limiting factor in pressurized BBS than in subatmospheric BBS. The use of carbon steel, which is inexpensive and easy to fabricate, is limited to metal temperatures that are not much higher than 1000 F. High-temperature alloys and ceramics, which are considerably more expensive or more difficult to fabricate than carbon steel, can be used at higher exhaust temperature but may prove to be uneconomical in the bottoming system applications.

#### 5.1.2 Turbomachinery Design Constraints

The turbomachinery in the subatmospheric BBS, particularly the compressors, are subjected to the corrosive, soot-laden engine exhaust gases and must be designed to meet the corrosion and fouling constraints discussed in

conjunction with the heat exchanger design. The material selection for these turbomachines is further complicated by the high stress conditions under which their rotors have to operate. The material selection process for the pressurized BBS turbomachines is considerably less complicated than in the case of the subatmospheric BBS because only air flows through the passages of the BBS turbomachines. In fact, many of the stationary components in the compressors of the pressurized BBS can be fabricated from aluminum and aluminum-base alloys.

Aside from the effects of corrosion and fouling, the effect of pressure ratio on the design and performance of the turbomachines must also be assessed. An examination of data such as those shown in Fig. 4.2 reveals that the optimum pressure ratio for BBS could be as high as 8:1, particularly in the case of high-temperature, intercooled cycles. Although single-stage radial inflow turbines can achieve high efficiency (80%) at these high pressure ratios, such is not the case with single-stage centrifugal compressors owing to Mach number limitations (Ref. 5.6). Even for turbines, a single-stage design for such a high pressure ratio may necessitate a tip speed that is too high to satisfy the strength limitations of most materials. BBS for near-adiabatic engines with very hot exhaust (1400 F) may require a two-stage turbine to expand the gas efficiently over a pressure ratio approaching 8:1.

For insulated diesel engines such as the NASA TC engine, the optimum pressure ratio is close to 5:1 and is, therefore, within the capability of high-efficiency single-stage radial inflow turbines. A two-stage centrifugal compressor with intercooling should have no difficulty handling this pressure ratio very efficiently.

### 5.1.3 System Integration Constraints

The components of the BBS must be matched together to ensure high system performance and reliability. Furthermore, the BBS must be integrated with the engine to form the compound engine/BBS power plant which is constrained by the following considerations:

- a) If a constant ratio gearbox is employed, the BBS speed will be constrained to a fixed multiple of the engine speed. The BBS speed is not likely to remain within its optimum range over the entire operating range of the engine. A variable-ratio speed reducer offers better matching of the BBS and engine speeds over a much wider range, albeit at a considerable increase in cost and complexity.
- b) With a constant ratio gearbox, no control system is needed to regulate the operation of the BBS other than a simple over-running clutch to prevent the engine from driving the BBS at low part loads. With a variable ratio transmissin, a control system is needed so that optimum or near-optimum BBS speed can be chosen at various engine operating conditions.
- c) In a subatmospheric BBS the engine exhaust gases flow directly through the BBS turbomachines. These machines must be very carefully designed as part of the engine exhaust system in order to ensure that the BBS will not impose an excessive back pressure on the engine over its operating range. In fact, the BBS in this case must be treated as part of the engine, very much like a turbocompounding system, and should not be simply added-on to

the engine. Being separated from the engine by the primary heat exchanger, the pressurized BBS is essentially independent of the engine and can be treated very much like an add-on device.

Other system integration constraints arise from the limited space available in the truck for placing such a system. The BBS components must be packaged in such a way as to allow the placement of the system behind the cab where it will offer minimum interference with the engine or drive train. This arrangement also provides a relatively simple coupling of the BBS gearbox with the engine flywheel, power take-out unit or gearbox.

## 5.2 Preferred BBS Configuration

The performance data presented in Chapter 4 and the design constraints provided in the preceding section strongly suggest that the pressurized BBS, while offering about the same performance advantages as the subatmospheric BBS, will be much less problematic to design, particularly for insulated diesel engines with exhaust temperatures below 1300 F (e.g., all the NASA engines as well as IDE-1 and IDE-2 engines). The subatmospheric BBS may prove a better match to future adiabatic diesel engines with higher exhaust temperature and much lower particulate emissions (e.g., IDE-3). For these reasons, the pressurized BBS system was chosen for further design, performance and economic analyses.

## 5.3 Pressurized BBS Design

The design of the pressurized BBS system involved an iterative process in which the design and sizing parameters of the major components of the system were varied in search of a near-optimum design that will achieve a reasonable compromise between performance and cost.

### 5.3.1 Design Groundrules and Assumptions

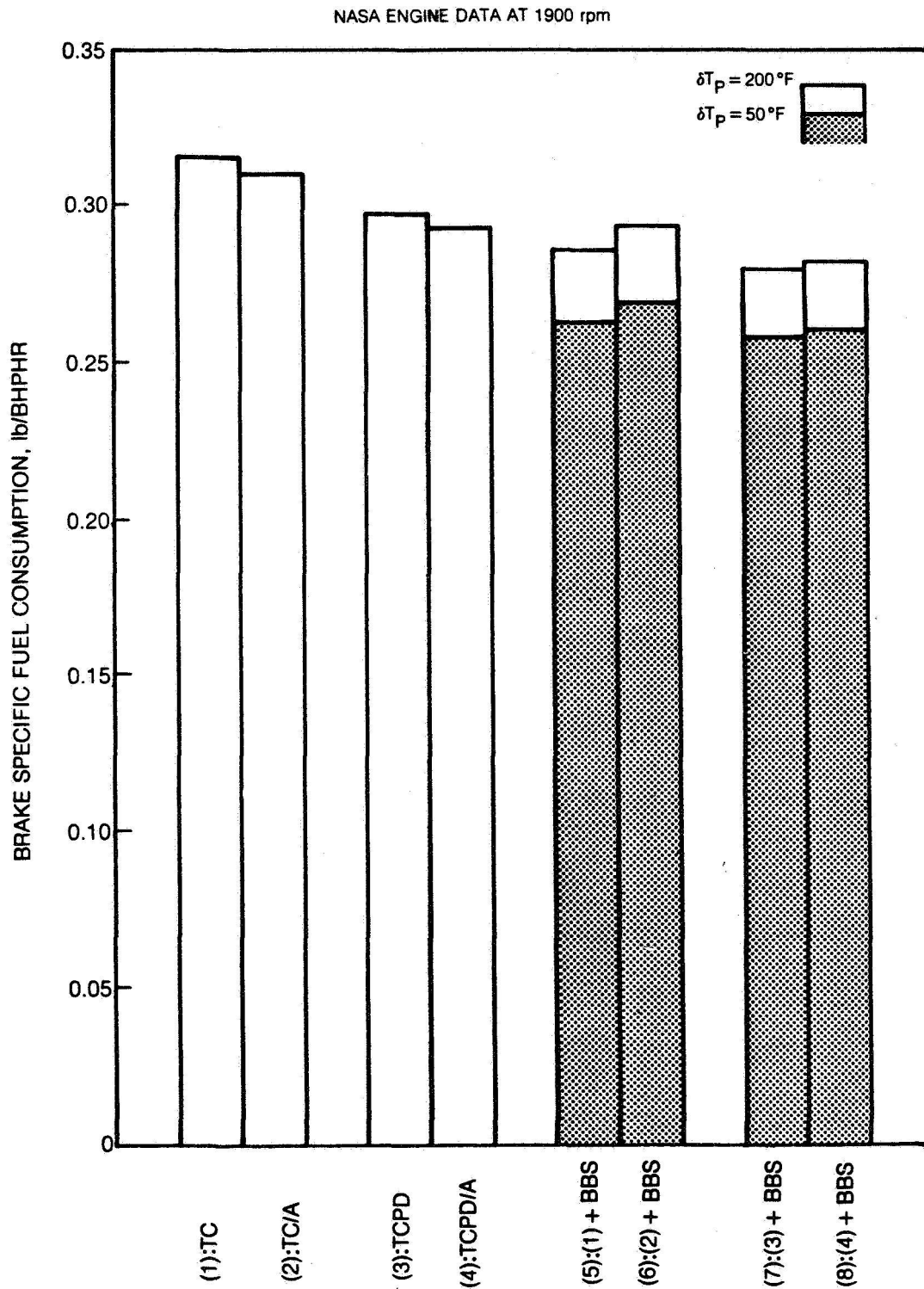
Engine: The BBS was designed to match the exhaust characteristics of non-aftercooled turbocharged engines, specifically, the IDE-2/TC engine. Though less efficient than the aftercooled and turbocompound versions, these engines discharge about the same quantity of exhaust at considerably higher temperatures, which is beneficial to the BBS. Figure 5.1 shows that, for the NASA-engines, the potential overall performance of the TC engine/BBS power plant will be better than or equal to that of the BBS combined with the other more efficient engines.

Design Point: All design calculations were performed at the full-power point, i.e., 300 Bhp 1800 rpm for the IDE-2/TC engine.

Primary Heat Exchanger: The heat exchanger design and sizing was based on clean-surface performance. A fouling penalty was subsequently introduced as a correction to the clean surface results.

Engine Back Pressure: Since all engine data were based on a 5 percent pressure loss through the exhaust muffler, the exhaust side of the primary heat exchanger was designed for a pressure loss of 5 percent or less in order not to present an additional burden on the engine.

COMPARISON OF SPECIFIC FUEL CONSUMPTION



Ambient Conditions: Equipment design and sizing were based on an average ambient temperature of 70 F at sea level rather than the traditional 85 F/500 ft altitude conditions. The 70 F ambient temperature is thought to represent a more typical condition.

Economic Assumptions: For the purpose of performance-cost tradeoffs, it was assumed that the annual fuel consumption of the track will scale in direct proportionality to the specific fuel consumption of the engine/BBS power plant. Annual fuel savings obtained in this way were translated into allowable capital cost increments following the procedure and detailed assumptions described later in Chapter 7. Other assumptions and ground rules pertaining to specific components will be discussed when appropriate.

### 5.3.2 Turbomachinery

The design values of many of the external parameters of the BBS turbomachines can be inferred from previous parametric analyses of system performance and from experience with similar systems. For example, an overall system pressure ratio in the vicinity of 5:1 was found to yield the maximum BBS output for 1250 F exhaust gases. Small departures from this value are not likely to be consequential. However, too high a value will both lower the system performance and increase the cost of the turbomachines by necessitating additional turbine and compressor stages. Too low a value, on the other hand, eliminates the need for a two-stage compressor but, inevitably, results in an unacceptable loss of system performance.

The optimum compressor pressure ratio split is that at which the compressor work is a minimum. Figure 5.2 shows the optimum first-stage pressure ratio for an intercooled two-stage compressor. For a total pressure ratio close to 5:1, and intercooling to within 50 F of first-stage inlet temperature, the optimum first stage design pressure ratio is close to 2.7:1.

Small, high-speed radial-inflow turbines and centrifugal compressors can achieve efficiencies in excess of 80 percent if their specific speed ( $N_s$ ), specific diameter ( $D_s$ ) and Mach and Reynolds numbers are properly selected (Ref. 5.6). Since a single-shaft turbine/compressor module represents the simplest, most rugged design, the turbine and compressor specific speeds are not independent and must be selected together to ensure high product efficiency ( $\eta_t \cdot \eta_c$ ). Furthermore, the design of the turbomachines cannot be divorced from the design of the primary heat exchanger whose air inlet temperature is the compressor discharge temperature and whose air exit temperature is the turbine admission temperature.

Table 5.1 presents the estimated values of the design and sizing parameters of the turbine, low-pressure (LP) compressor and high-pressure (HP) compressor of the pressurized BBS for the IDE-2/TC engine exhaust conditions. These values correspond to the optimum value of the primary heat exchanger temperature difference determined in accordance with the procedure described in Section 5.3.4. The listed efficiencies represent the total-to-static values for the turbine and the total-to-total values for the compressor. Efficiency levels as high as these have been achieved in numerous small turbomachines built for aircraft environmental control and auxiliary power applications by the Hamilton Standard Division of United Technologies.



FIG. 5.2

OPTIMUM PRESSURE RATIO FOR TWO-STAGE INTERCOOLED COMPRESSORS

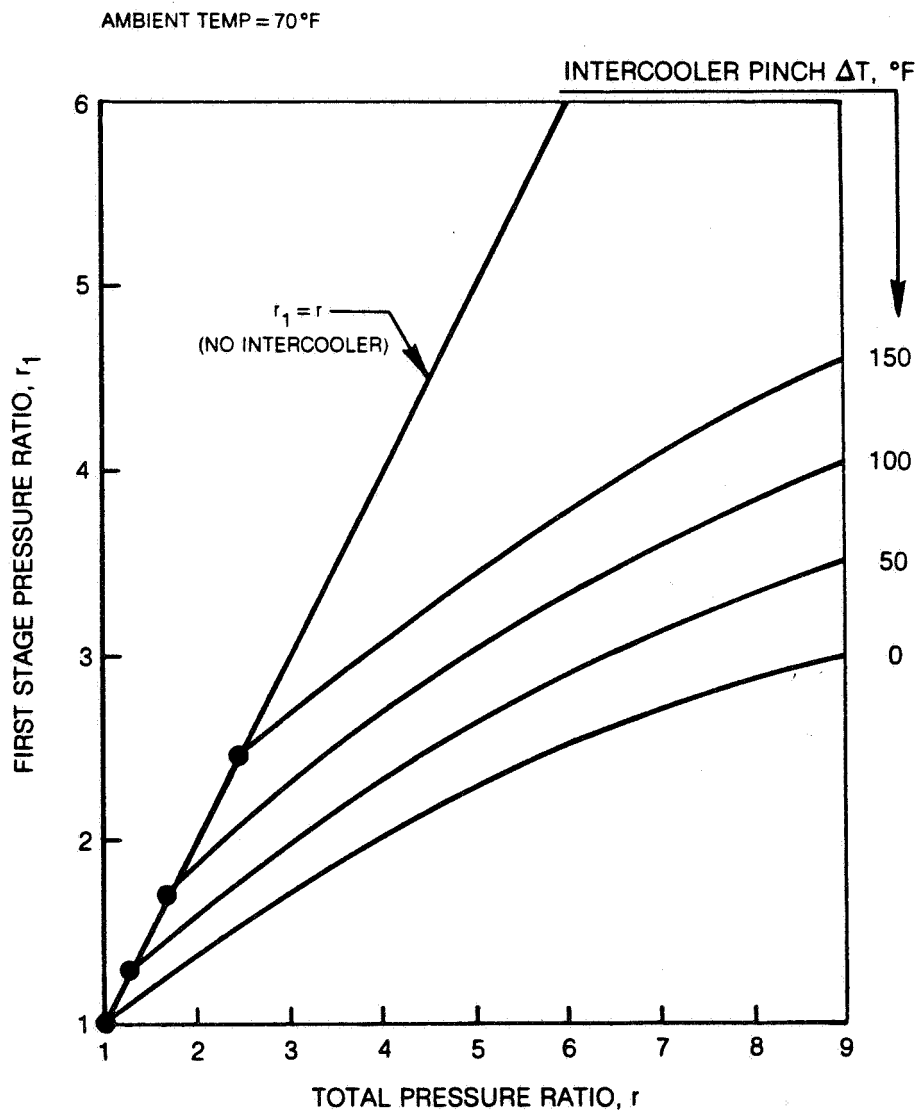


TABLE 5.1

## PRESSURIZED BBS ROTATING EQUIPMENT SPECIFICATIONS

(IDE-2/TC Engine)

Component	Turbine	LP	HP
		Compressor	Compressor
Type	Radial	Radial*	Radial*
Pressure Ratio	5.0:1	2.75:1	1.9:1
Specific Speed, $N_s^{**}$	55	96	88
Specific Diameter, $D_s^{**}$	1.5	1.8	1.85
Speed (rpm)		70-80,000	
Rotor Tip Diam. (in.)	5.5	5.3	4.1
Peak Efficiency, %	0.84	0.835	0.835
Stator Material	347 SS	XA140AR	
Stator Fabrication Method	Invest, Cast	Sand Cast + Machined	
Rotor Material	UDIMET 500	356A	
Rotor Fabrication Method	Invest, Cast	Invest Cast + PPT Hardened	
Shaft Material		AMS6415	
Bearings		3 Ball Bearings	
Gearbox: Type	2-Stage Planetary w/Overrunning Clutch		
Reduction Ratio	40:1		

\* Swept back vanes.

\*\* In engineering units, see Ref. (5.6).

The turbine rotor material must be capable of withstanding the centrifugal stresses at high operating temperatures (~1000 F); UDIMET 500 was chosen for this application. The compressor rotors are not subjected to high-temperature or corrosive gases; an aluminum-base alloy (356A) was chosen for this purpose. Stainless steel 347 was chosen for the turbine stator to meet the high temperature requirements and another aluminum-base alloy (XA140 AR) was chosen for the compressor stators. Precision investment casting is the method of fabrication chosen for the critical parts of the turbine and the compressor. Table 5.1 also contains a summary of the materials and fabrication methods selected.

The turbine/compressor shaft bearings can be either air-lubricated foil-type or oil-lubricated ball bearings. It is expected that three such bearings will be needed, one of which must be capable of taking the unbalanced axial thrust. Three ball bearings were chosen for this application because of their high efficiency and low cost.

Figure 5.3 shows a conceptual cross-sectional view of the turbine/compressor modules envisioned for the pressurized BBS application.

### 5.3.3 Intercooler

The intercooler of a pressurized BBS is not subjected to any corrosive or soot-laden gases (being an air/air heat exchanger operating at low pressures and temperatures). The most suitable design was judged to be a compact plate-fin heat exchanger with an aluminum core (Ref. 5.7). A cross-flow geometry with large frontal area and small depth (like a conventional automobile radiator) was selected for the present application. This arrangement was found to offer packaging convenience, reduce cooling air pressure drop (fan power) and allow the utilization of the ram cooling effect while the truck is traveling at highway speed. The BBS compressor intercooler will be very much similar to the turbocharger air/air aftercooler used in the TC/A and TCPD/A engines.

Compact plate-fin aluminum heat exchangers are relatively inexpensive. The optimum design for this type of heat exchanger tends towards lower terminal temperature differences because the cost penalty associated with a low temperature difference (large surface area) design are not steep enough to offset the performance gain. For this reason, the intercooler in the pressurized BBS was designed for a minimum temperature difference of only 50 F. Table 5.2 summarizes the pertinent design and sizing data for the intercooler.

### 5.3.4 Primary Heat Exchanger

The temperature difference in the primary heat exchanger exerts a strong influence on both the size (and hence the cost of the heat exchanger) and on the performance of the entire system. The design and sizing of the turbomachines are also influenced by the temperature difference in the heat exchanger as mentioned in Section 5.3.1. Because of its strong influence on the system cost and performance, the heat exchanger design must be carefully evaluated and optimized.

A counter-flow plate fin compact design was chosen for the BBS application to achieve high effectiveness. A stainless steel construction was deemed

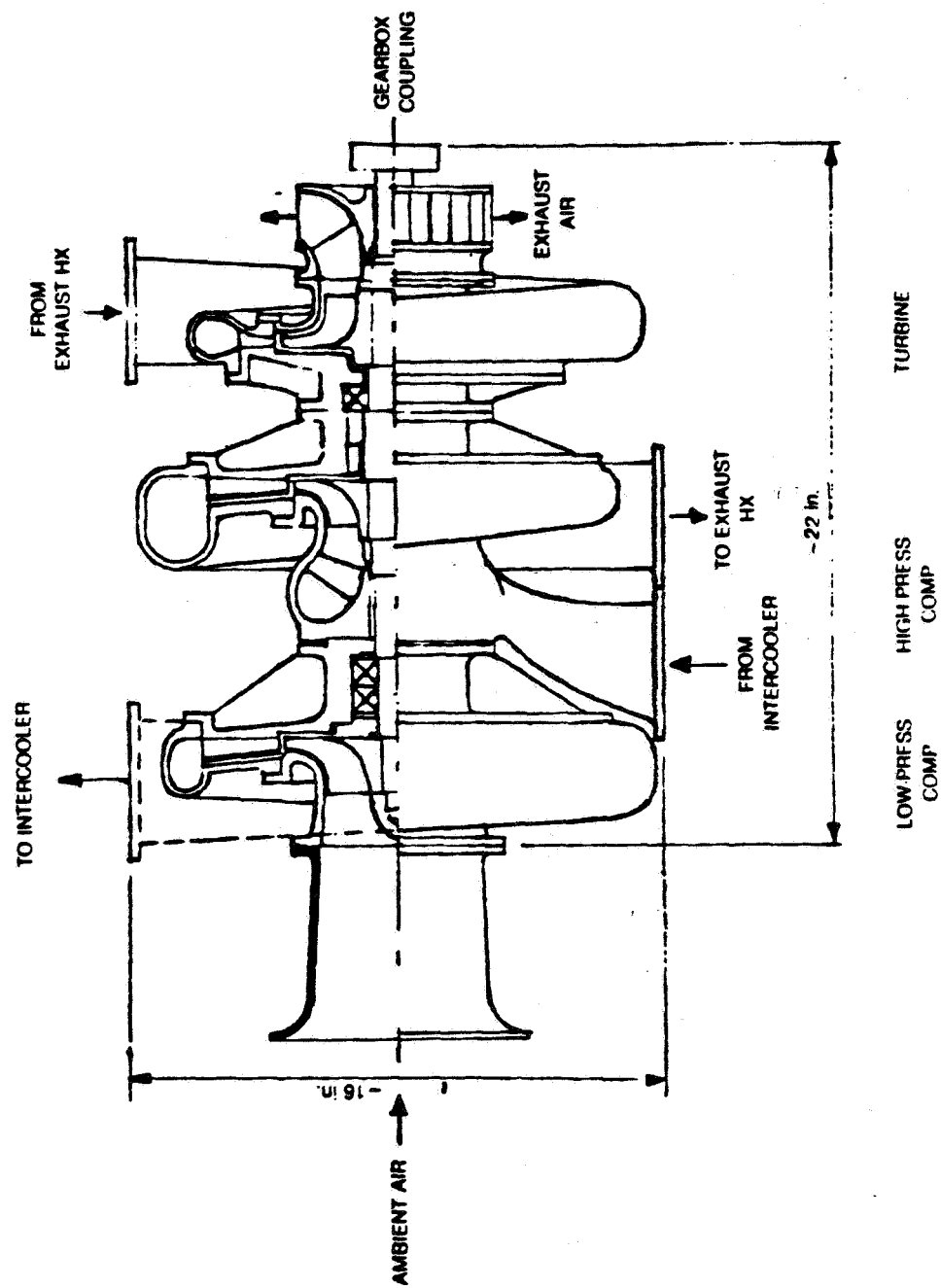
TABLE 5.2

PRESSURIZED BBS INTERCOOLER SPECIFICATIONS  
(IDE-2/TC Engine)

Type	Plate-Fin/Cross Flow
Cold Air Flow Length, in.	8.1
Hot Air Flow Length, in.	17.5
No-Flow Length, in.	19.5
Total Weight, lb	48
Total Volume, ft <sup>3</sup>	1.6
Heat Transfer Area, ft <sup>2</sup>	835 (both sides)
<u>Fin Characteristics:</u>	(Both Sides)
. Geometry	Shallow Wavy Fin (HSD Data)
. Height, in.	0.25
. Spacing, in.	0.05
. Thickness, in.	0.006
. Material	Aluminum
Design Effectiveness	0.76
Minimum Temp. Diff., F	50
Design Pressure Drop (psi)	
. Hot Side	1.05
. Cold Side	0.35

FIG. 5.3

**TURBINE/COMPRESSOR UNIT FOR PRESSURIZED BOTTOMING BCWHRS**



desirable to limit the corrosive effect of acids that may condense during cold start transients, although carbon steel would be adequate under normal operating conditions. Rather than selecting the air flow rate on the cold side of the heat exchanger to achieve a near-uniform temperature difference, a flow rate was chosen that provides a divergent temperature difference toward the heat exchanger hot end. This is compatible with good heat exchanger design practices which suggest that a reasonable compromise between cost and performance can be achieved if  $\Delta T/T$  is about the same at both ends of the heat exchanger.

A computerized proprietary compact heat exchanger design procedure was employed to calculate the dimensions of the heat exchanger given the inlet and outlet temperatures of the hot and cold streams, the flow rates, pressures and specific heats of the two streams, and the admissible pressure drop on each side. The computer program has access to the geometrical, heat transfer and fluid friction characteristics of a wide variety of finned surfaces of which several were tried for the present application. The heat exchanger dimensions were obtained for a range of minimum (cold end) temperatures; Figure 5.4 depicts the variation of the heat exchanger volume, weight and heat transfer area (both sides) with the cold-end temperature difference for IDE-1/TC engine exhaust flow rate. The corresponding values for the IDE-2/TC engine are about 20 percent larger.

The effect of the minimum temperature difference on the cost of BBS system, including the heat exchanger and the turbomachines, is depicted in Fig. 5.5. Shown also in Fig. 5.5 is the allowable installed capital cost of the BBS that can be justified by the annual fuel savings attributed to it. The allowable cost was based on the fuel savings computed in accordance with the procedure described earlier in Section 5.2 and the economic assumptions and methodology given in Chapter 7. The difference between the allowable cost and the actual cost is a measure of the BBS excess benefit. The BBS meets the minimum acceptable rate of return (MARR) specified in the calculation of the allowable cost (12 percent real after tax value) when the allowable cost equals the actual cost. The optimum value of  $T$ , however, is that at which the "excess benefit" reaches a maximum, i.e., the value at which the incremental value of the rate of return equals the MARR. Figure 5.5 shows that this value is close to 150 F rather than the 75 F value which yields zero excess benefit.

Table 5.3 summarizes the characteristics of the primary heat exchanger for the system of interest (IDE-2/TC engine with pressurized BBS).

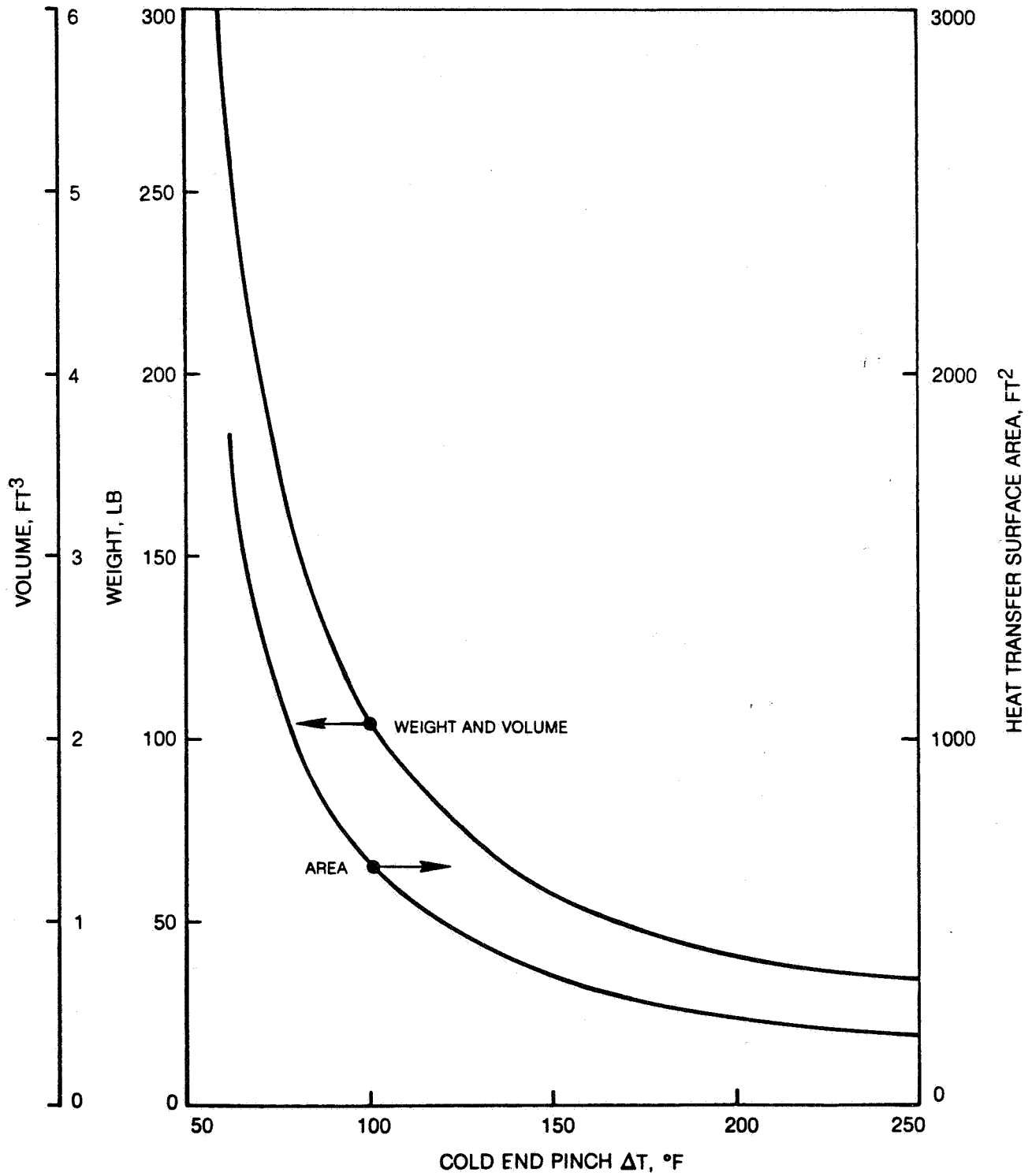
#### 5.3.4 System Integration

Figure 5.6 presents the system layout, excluding the intercooler and the gearbox/overrunning clutch unit. The system will be placed behind the truck cab with the turbine/compressor shaft parallel to the engine crank shaft. The intercooler will be placed either under the hood in front of the engine (where the radiator of a conventional engine is usually located) or on top of the cab; both locations allow the intercooler to receive ram air for cooling. The gearbox, which will be a two stage design, and the overrunning clutch will be placed at the rear of the engine close to the main engine gearbox. The preferred orientation of the system is with the length of the primary heat exchanger in the vertical direction although horizontal placement is also possible.

FIG. 5.4

### BBS PRIMARY HEAT EXCHANGER SIZE AND WEIGHT

PRESSURIZED-INTERCOOLED BBS  
IDE-1/TC ENGINE



83-9-60-9

**OPTIMUM PINCH TEMPERATURE DIFFERENCE IN  
BBS PRIMARY HEAT EXCHANGER**

- PRESSURIZED-INTERCOOLED BBS
- IDE-2/TC ENGINE

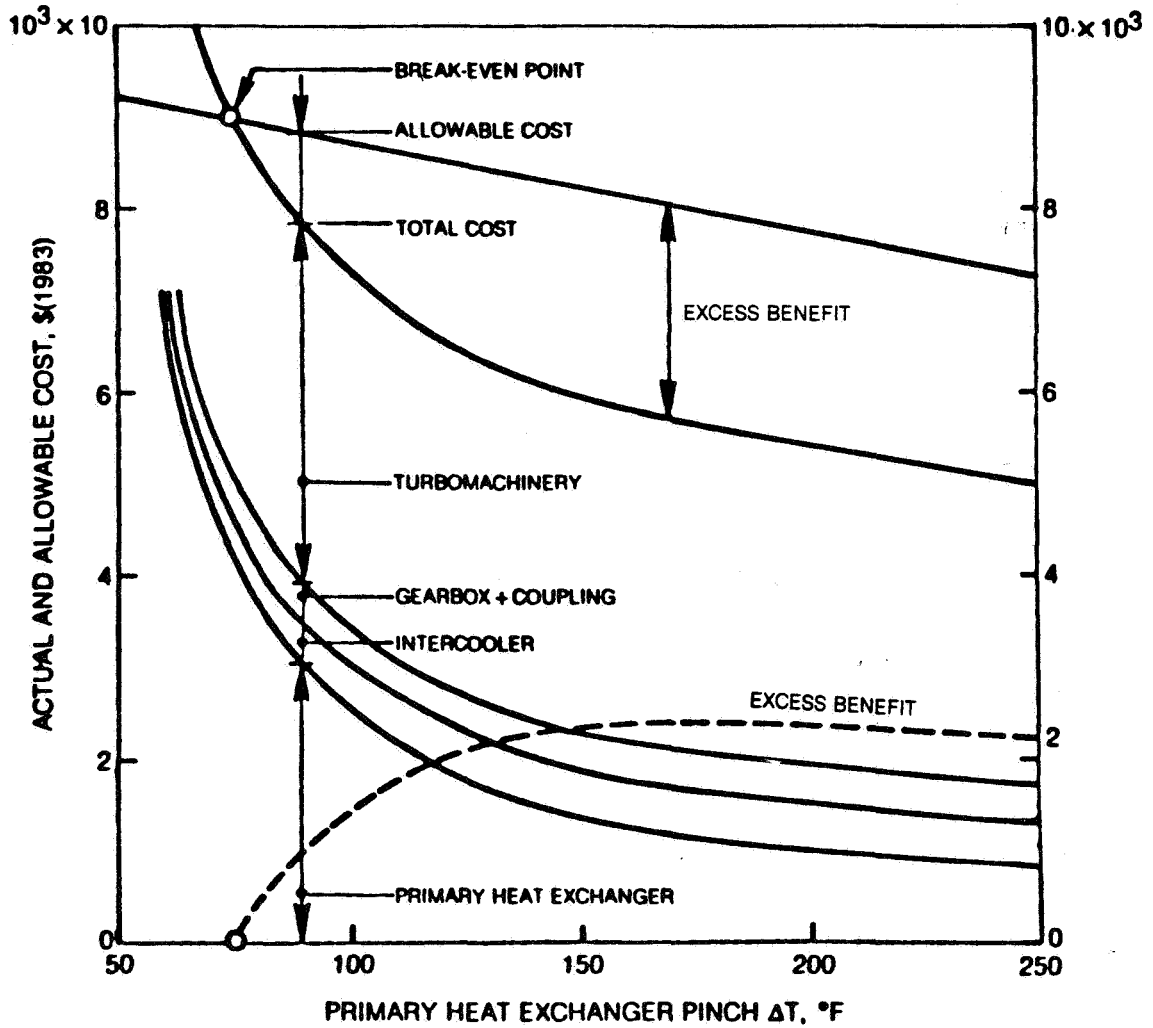




TABLE 5.3

## PRESSURIZED BBS PRIMARY HEAT EXCHANGER SPECIFICATIONS

(IDE-2/TC Engine)

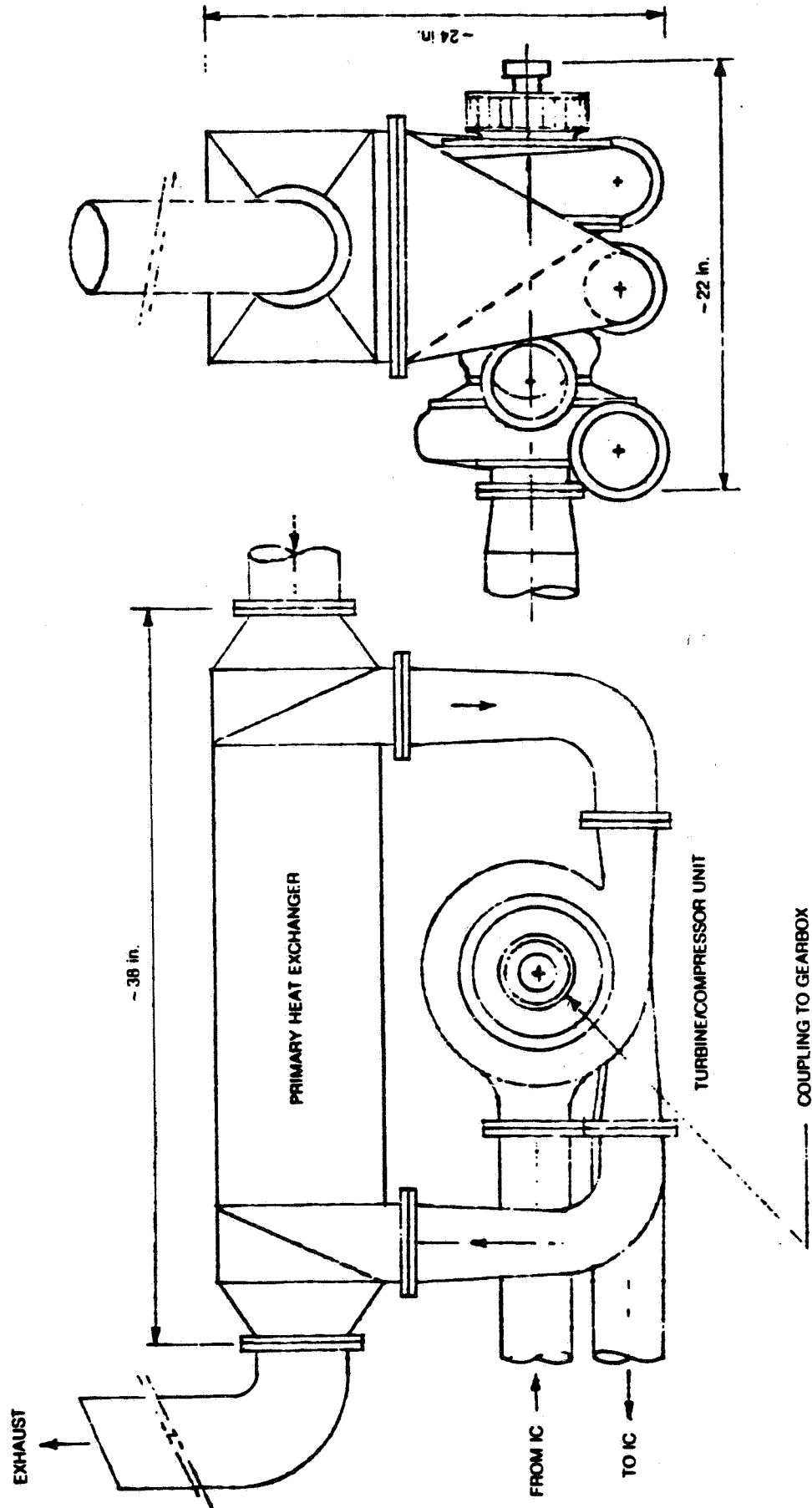
Primary Heat Exchanger

Type	Plate-Fin/Counterflow	
Flow Length, in.	23	
No-Flow Length, in.	12	
Height, in.	9.5	
Tent Length, in.	2 x 5	
Total Weight, lb	85	
Total Volume, ft <sup>3</sup>	1.7	
H.T. Area, ft <sup>2</sup>	840 (Both Sides)	
Fin Characteristics:	HOT SIDE	COLD SIDE
. Geometry	Wavy Fin K&L* Fig. 10.66	Ruffled Fin HSD Data
. Height, in.	.375	.250
. Spacing, in.	.087	.077
. Thickness, in.	.005	.005
. Material	Stainless Steel	Stainless Steel
Design Effectiveness	0.85	
Minimum Exhaust Gas Temp., F	425	
Minimum Temp., Diff., F	130	
Design Pressure Drop, psi		
. Hot Side (Exhaust Gas)	0.8	
. Cold Side (Air)	3.9	

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\* Kays and London, Ref. (5.7).

PRESSURIZED BBS LAYOUT



#### 5.4 System Performance and Cost Summary

Table 5.4 presents the thermodynamic conditions at the key points of the pressurized BBS (see Fig. 2.3) designed to match the IDE-2/TC engine, i.e., based on the component characteristics given in Section 5.3. Table 5.5 summarizes the system performance data when all heat exchanger surfaces are clean.

A cost breakdown of the system is given in Table 5.6. The total installed cost of the system was assumed to be double the major equipment manufacturing cost which is typical for this type of equipment. All the unit cost data were obtained from correlations that were prepared from a compilation of manufacturers cost estimates for similar equipment, e.g., automotive gas turbines, aircraft air-cycle environmental control systems and automotive turbochargers and aftercoolers. These correlations also reflect the effect of production numbers on the unit cost. The cost data given in Table 5.6 are based on annual production runs of about 10,000 BBS units which corresponds to less than 15 percent of the projected heavy duty transport market (Ref. 5.8). At this production rate, it is estimated that the installed cost of pressurized BBS in the 30 to 40 hp range will be about \$170/Bhp (1980\$). This is considerably higher than the cost of the engine itself (\$40/Bhp) and reflects to a large extent the lower production volume of the BBS. The BBS system cost would be much lower if production volumes were comparable to those of automotive engines. The annual O&M allowance for lubricants and filters was estimated to be about \$100. The annual expenses for heat exchanger periodic cleaning was estimated separately as presented in Section 5.5.3 below.

#### 5.5 OFF-Design Performance

The external and internal design parameters of the BBS are not expected to remain at their design values during most of the truck operating time. Exhaust flow rate and temperature change with engine torque and speeds; ambient conditions change with time and location; and equipment performance deteriorates as a result of wear and fouling. A realistic assessment of the economics of diesel engine bottoming systems must be based on their performance not only under ideal steady-state design conditions, but also under varying off-design conditions. The effect of these off-design conditions on BBS performance are discussed in the following paragraphs.

##### 5.5.1 Effect of Ambient Temperature and Leakage

The design calculations were performed for a constant ambient temperature of 70 F and no external leakage.\* The sensitivity of the BBS output to variations in the ambient temperature and the percentage of compressor flow that leaks out of the system is shown in Fig. 5.7. It should be remembered, however, that the BBS studied here should be able to operate

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\*Leakage inside the turbine or the compressor is accounted for in the overall efficiency of these machines.

TABLE 5.4

PRESSURIZED BBS THERMODYNAMIC DATA  
(IDE-2 TC Engine)

Station <sup>+</sup>	Fluid	Pressure (psia)	Temp. (F)	Flow Rate (lb/min)	Comments
A	Exh. Gas	15.5	1260	43.8	Engine Exhaust
B	Exh. Gas	14.7	425	43.8	BBS Exhaust
C	Air	15.05 <sup>*</sup>	70	58.8	Upstream of Intercooler
D	Air	14.7	222	58.8	Downstream of Intercooler
1	Air	14.7	70	53.1	LP Compressor Inlet
2	Air	42.5	293	53.1	LP Compressor Exit
3	Air	41.5	125	53.1	Intercooler Exit
4	Air	82.1	277	52.8	HX Inlet
5	Air	81.6	1030	52.8	Turbine Inlet
6	Air	14.7	545	52.8	Turbine Exhaust

<sup>+</sup> See Fig. 2.3

<sup>\*</sup> Includes half the dynamic head at 55 mph.

TABLE 5.5

PRESSURIZED BBS DESIGN POINT  
PERFORMANCE SUMMARY

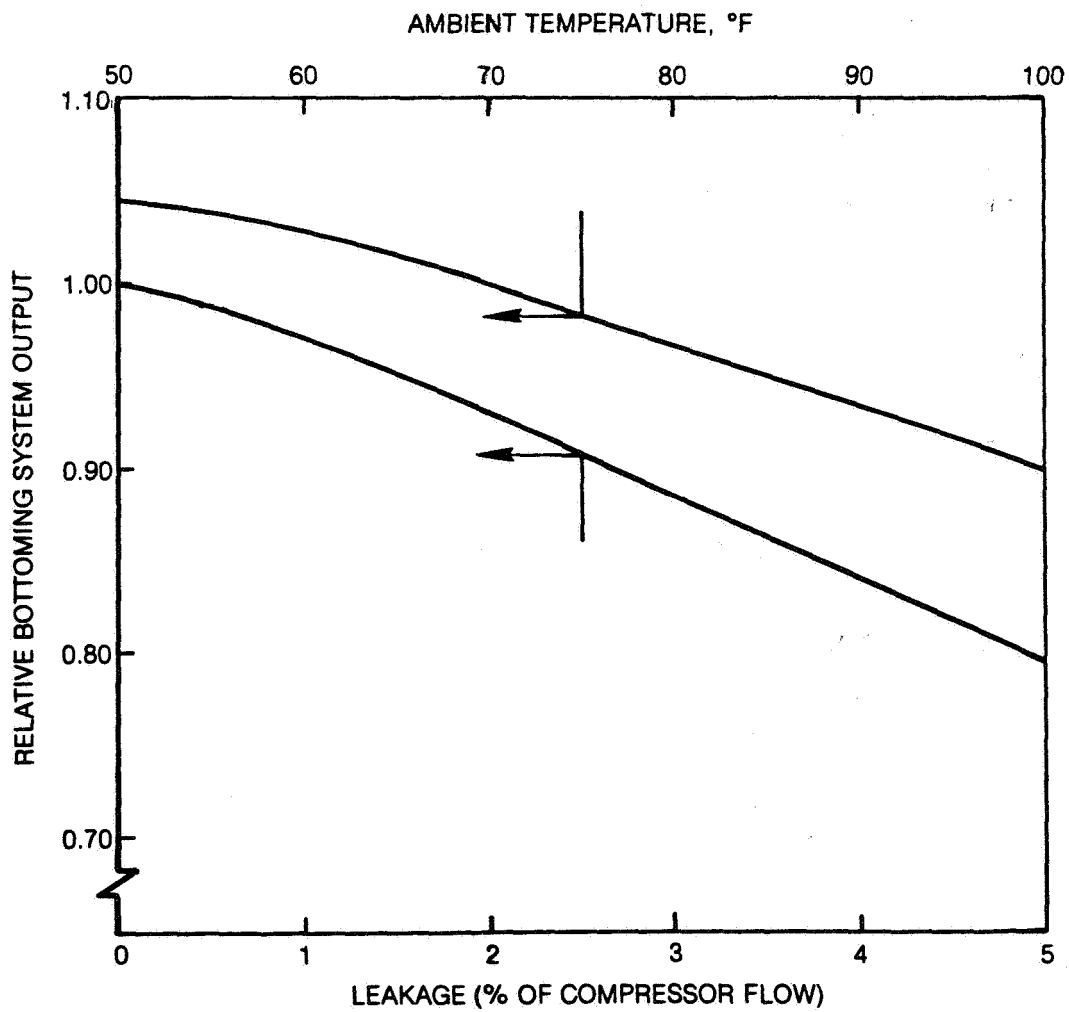
<u>Engine</u> (Non-Aftercooled)	IDE-1/TC	IDE-2/TC	NASA TC
Rated hp	300	300	317
Rated rpm	1800	1800	1900
Exhaust Flow (lb/min)	36.5	43.8	48.0
Exhaust Temp. (F)	1265	1260	1240
<u>Bottoming Cycle</u>	Pressurized Intercooled Brayton		
Turbine, hp	129.3	154.9	162.8
LP Comp. hp	-56.3	-67.6	-70.7
HP Comp. hp	-38.0	-45.6	-47.6
Bearing & Leaking Loss (hp)	-1.9	-2.3	-2.4
Gear Box Loss (hp)	-1.0	-1.2	-1.3
Intercooler Fan (hp)	-1.2	-1.4	-1.6
Net HP (Clean HX)	30.0	36.8	39.2
Primary HX Effectiveness (Clean)	0.85	0.85	0.84
Hot Side P (psi)	0.85	0.85	0.85
Min. Exhaust Temp. (F)	429	425	427
Intercooler Effectiveness	0.75	0.75	0.76

TABLE 5.6

COST BREAKDOWN OF A 36-BHP BBS  
(For IDE-2/TC)

	<u>Material</u>	Cost, \$(1983) <u>Fabrication</u>	<u>Total</u>
<u>Turbine</u>			
Rotor	60	160	220
Nozzles	110	70	180
Scroll	215	75	290
Diffuser	15	35	50
<u>Compressors</u>			
1st Stage Rotor	10	115	125
1st Stage Diffuser	10	35	45
2nd Stage Rotor	10	115	135
2nd Stage Diffuser	10	35	45
Housing	35	125	160
<u>Bearings &amp; Shaft</u>			
Bearings (3x)			135
Shaft			50
(A) <u>Turbomachinery Total Manuf. Cost</u>			1435
(B) <u>Gearbox &amp; Coupling</u>			185
(C) <u>Rotating Equipment Assembly</u>			400
(D) <u>Intercooler</u>			270
(E) <u>Primary Heat Exchanger</u>			750
<u>Total Installed Cost = 2x(A+B+C+D+E)</u>			6080
<u>Installed Cost/Bhp</u>			170

**EFFECT OF AMBIENT TEMPERATURE AND LEAKAGE ON THE PERFORMANCE OF PRESSURIZED BRAYTON SYSTEMS**



without external leakage or, at worst, with a negligible amount of leakage which is attributed mainly to failure of seals and gaskets. An external leakage of 0.5 percent of compressor flow was assumed in all subsequent calculations; this corresponds to a 1 percent output loss. However, the 70 F ambient temperature was retained as a representative seasonal average value. The BBS net output would be about 4 percent lower if the system were designed for an 85 F ambient temperature.

#### 5.5.2 Part Load Operation

The part load performance of the BBS was evaluated using a proprietary computer program that combines the performance maps of the major components of the system and establishes the correct steady-state heat and mass balances for each component and for the entire system, corresponding to a new set of external parameters. The performance maps for the components are presented to the computer in the form of tabulated data or computational subroutines. For the pressurized BBS these include:

- . Low-pressure compressor map
- . Intercooler subroutine
- . High-pressure compressor map
- . Primary heat exchanger subroutine
- . Turbine map.

The turbomachinery maps were obtained by scaling from maps of similar existing machines designed and built by Hamilton Standard. The heat exchanger performance subroutine is a variant of the heat exchanger design subroutine.

The exhaust flow rate and temperature corresponding to an engine operating point (torque and speed) constitute the external input parameters that determine the BBS part-load performance when a constant ratio gearbox is used between the BBS and the engine. The BBS part-load performance can be improved by using a variable ratio transmission with a suitable controller. This is shown in Fig. 5.8 which presents the BBS gross\* output as a function of engine power and speed over a range of BBS speeds. A 43:1 constant-ratio gearbox designed to match the engine exhaust conditions at maximum power (300 Bhp at 1800 rpm) will force the BBS to operate at suboptimum speeds at lower engine speeds, viz., 67,000 rpm at an engine speed of 1550 rpm and 56,000 rpm at an engine speed of 1,300 rpm.

Allowances were made for the mechanical losses between the turbine and compressors (bearing losses) and between the turbine/compressor module and the gearbox. The bearing losses were taken as 1.5 percent of the compressor shaft power. The gearbox loss was assumed to be proportional to the cube of the speed, amounting to about 3 percent of the net design output at maximum speed (see Table 5.5). Other minor losses were assumed to be a constant fraction of the output.

Two BBS performance maps were generated and combined with the engine maps to generate combined power plant maps: one corresponding to a BBS with a constant-ratio gearbox and the other corresponding to a BBS with a variable ratio transmission that is controlled to operate the BBS at the

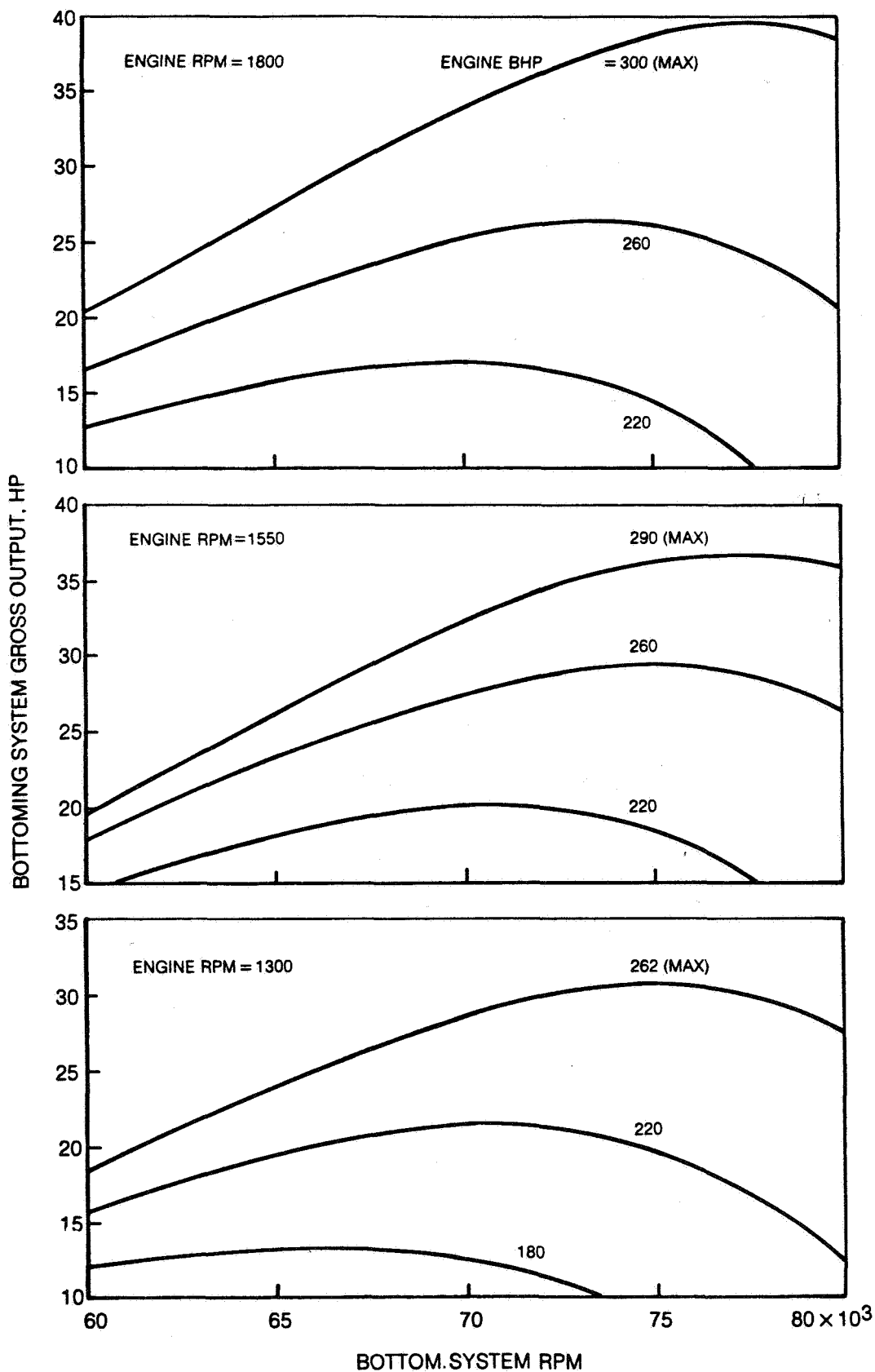
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\*No penalty for mechanical and leakage losses.



**EFFECT OF SPEED ON BRAYTON BOTTOMING SYSTEM GROSS OUTPUT**

(IDE-2/TC ENGINE/PRESSURIZED BBS)



speeds corresponding to the maxima in Fig. 5.8. The second case, which may require a complex or impracticable controller, corresponds to the best that can be achieved with a variable ratio transmission; its economic feasibility can only be determined through an analysis of the truck performance over a realistic drive cycle as will be shown in Chapter 6. Complete performance maps for these two compound power plant versions are given in Chapter 6.

### 5.5.3 Effect of Heat Exchanger Fouling

The effect of a uniform film of fouling on the performance and pressure drop characteristics of the primary heat exchanger was evaluated using a modified version of the abovementioned computer code. The calculations were performed for two values of the fouling film thermal conductivity; 0.02 and 0.05 Btu/h ft F, which are typical for carbon deposits. Figure 5.9 depicts the relationship between uniform fouling thickness and the heat exchanger effectiveness and pressure drop. The assumption of uniform thickness over the entire exhaust side surface provides only a first-order approximation to the actual case in which fouling will tend to build up much faster at the colder end of the heat exchanger than at the hotter end.

The combined effect of heat exchanger performance and pressure losses was translated into a power plant output loss and specific fuel consumption increase, as shown in Fig. 5.10. The power plant output loss, normalized in Fig. 5.10 with respect to the BBS output when the heat exchanger is clean, is the sum of two losses:

- . BBS performance loss caused by the lower heat exchanger effectiveness.
- . Engine output loss caused by the higher back-pressure.

The calculation of the latter loss was based on a small proportional adjustment of the engine brake mean effective pressure (194 psig at rated power). Although this loss is not a direct BBS output loss, it is caused entirely by the BBS and would not have arisen in the absence of the BBS. It can be seen that the pressure loss effect can be as significant, if not more significant than that attributed to the heat exchanger loss of effectiveness.

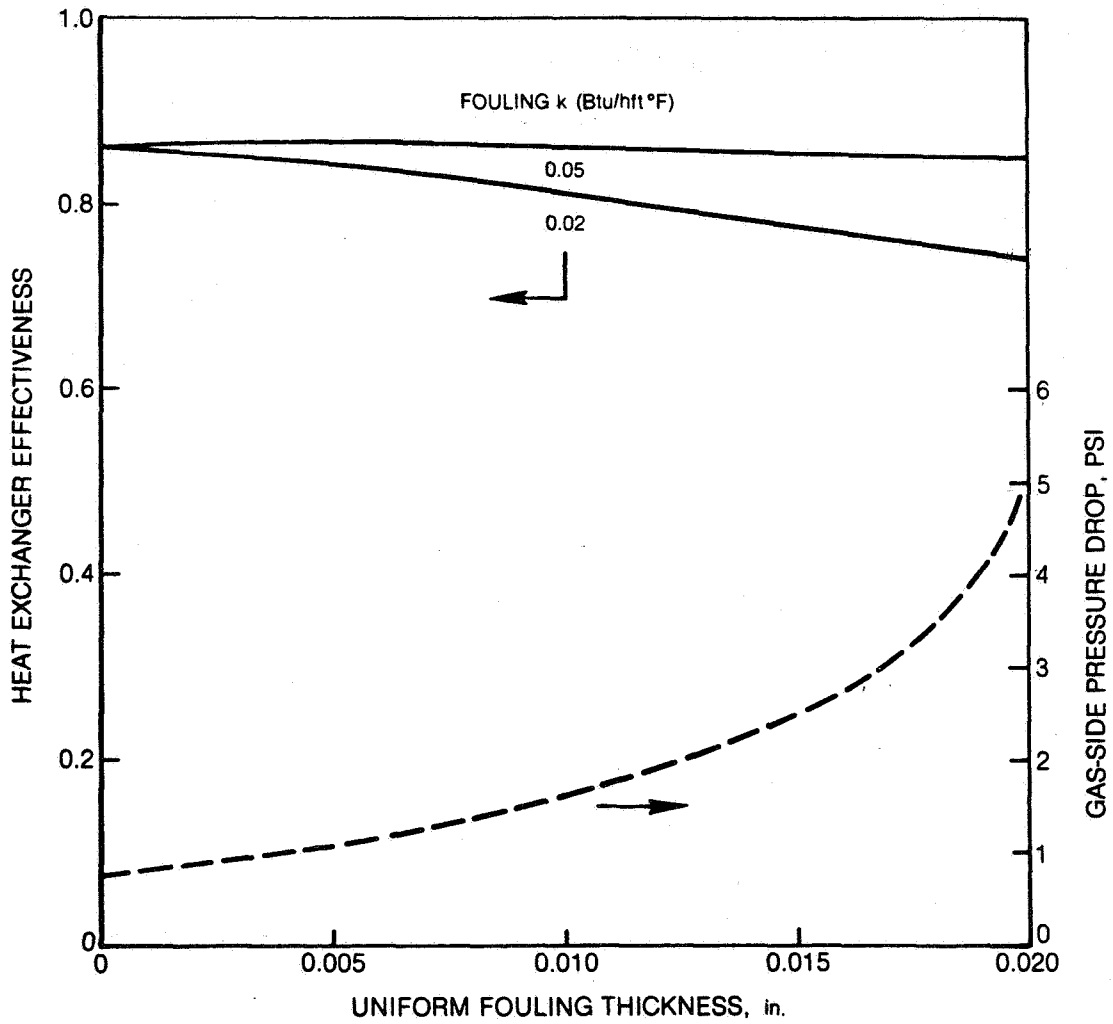
Since fouling films grow asymptotically to a steady-state value, the average BBS output loss will depend both on the fouling thickness growth rate and on the heat exchanger cleaning frequency. The fouling film growth was modeled by the simple relation

$$\delta/\delta_{\infty} = 1 - \exp(-t/\tau), \quad (5.1)$$

in which  $\delta_{\infty}$  is the asymptotic film thickness and  $\tau$  is fouling time scale. Because the heat exchanger of a pressurized BBS will be maintained comfortably above the acid dew point under normal operating conditions, the asymptotic film thickness was taken as 0.01 in, which is commensurate with the findings of Ref. (5.2) for uncooled heat exchangers. The time scale in Eq. (5.1) was taken as 240 hours. Figure 5.11 depicts the BBS average

**EFFECT OF FOULING ON HX PERFORMANCE**

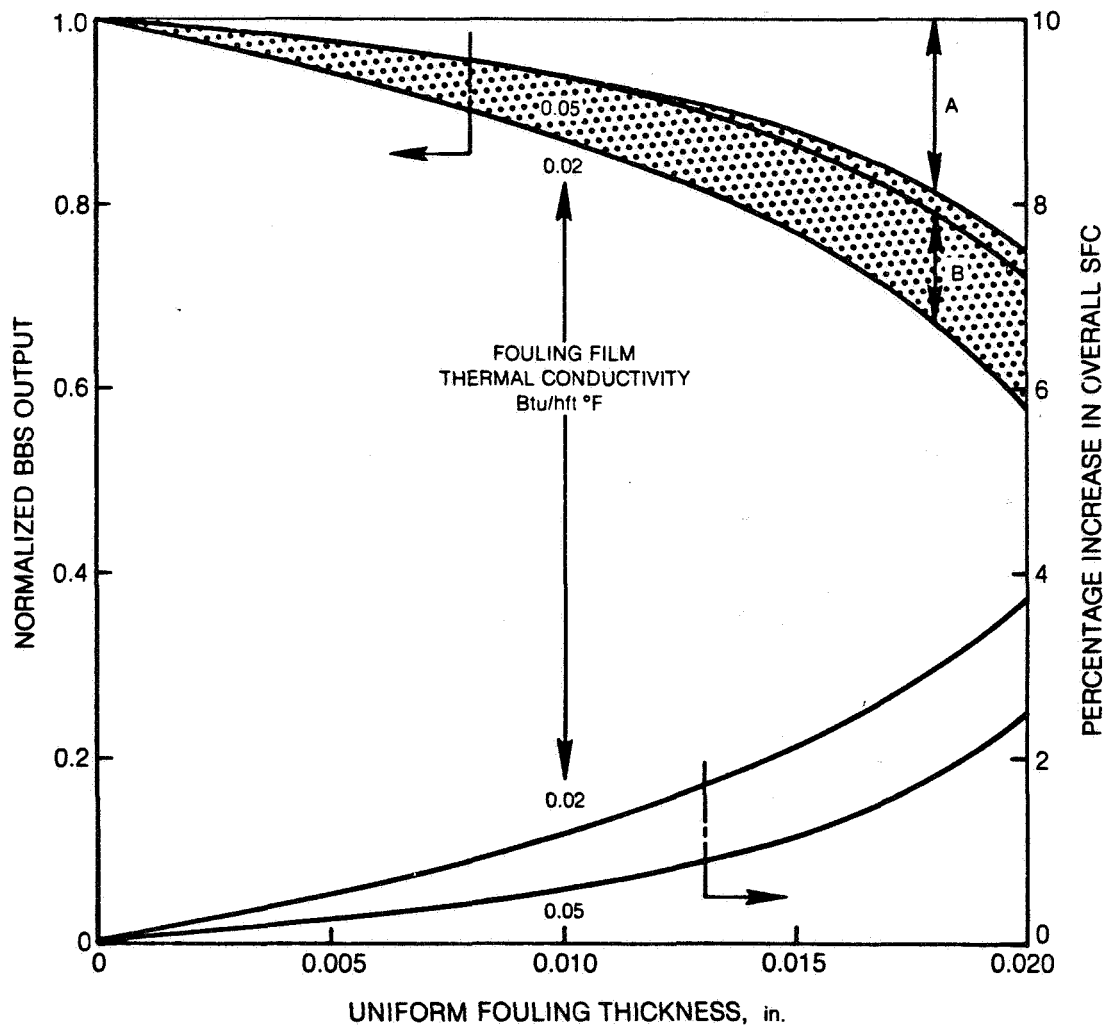
PRESSURIZED BBS WITH  $\Delta T_{COLD} = 130^\circ\text{F}$  AND  
 IDE-1/TC ENGINE  
 AIR-SIDE  $\Delta P/P < 5\%$



ENGINE BMEP @ 1800 RPM AND FULL LOAD = 194 PSIG

**EFFECT OF UNIFORM FOULING ON BBS OUTPUT AND COMBINED SYSTEM PERFORMANCE**

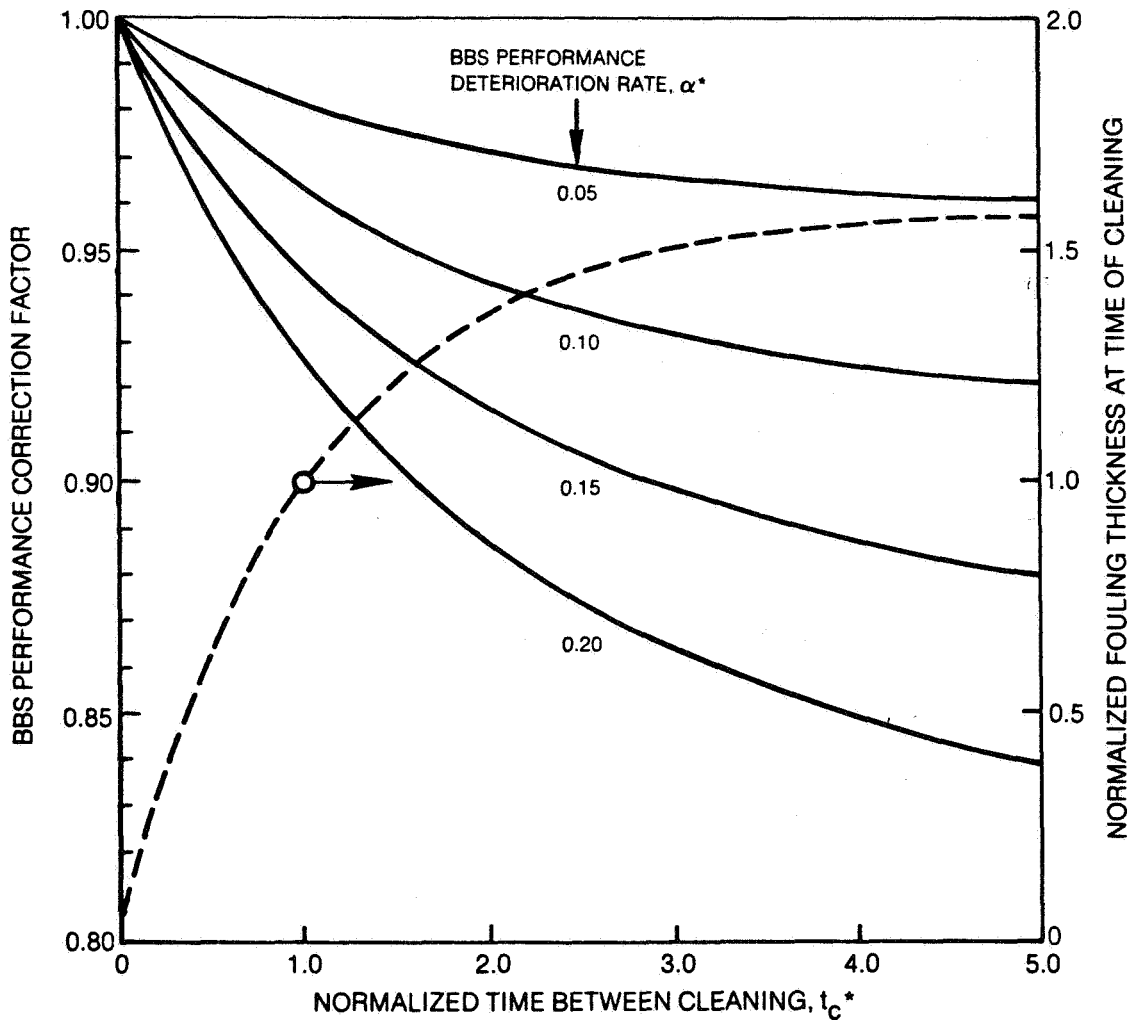
PRESSURIZED INTERCOOLED BBS  
IDE-1/TC ENGINE



A: LOSS DUE TO HIGHER ENGINE BACK PRESSURE  
B: LOSS DUE TO LOWER HEAT EXCHANGER EFFECTIVENESS

**EFFECT OF HEAT EXCHANGER FOULING  
AND CLEANING FREQUENCY ON BBS OUTPUT**

PRESSURIZED BBS  
IDE-1/TC ENGINE



$t^* = \frac{\text{TIME BETWEEN CLEANING}}{\text{FOULING BUILD-UP CHARACT. TIME}}$   
 $\alpha^* = \text{RATE OF OUTPUT LOSS}$

performance correction factor\* as a function of the normalized time between cleaning and the BBS performance deterioration rate  $\alpha$ , defined here as the BBS relative output loss at  $\delta = \delta_{\infty}$ . Figure 5.10 indicates that typical values for  $\alpha$  in the 10-15 percent range would be reasonable.

The average time between cleanings was chosen to coincide with engine regular maintenance (oil and filter change every 5000 - 7000 miles, say). This corresponds to a cleaning frequency of once every 100 to 150 hours of normal truck operation. Under these conditions the annual average BBS would be 2-3 percent lower than that corresponding to no fouling at all; i.e., the BBS average output would be 97-98 percent of its no-fouling output. The BBS outputs used in all subsequent calculations of fuel economy and life-cycle cost were adjusted by a factor of 0.97 to account for the fouling penalty.

It is estimated that for a 100,000 miles per year service the heat exchanger will have to be cleaned between 13 and 20 times annually. Washing with water would probably be the most effective cleaning method. The annual O&M expenses for this procedure was estimated to be less than \$200.

#### 5.6 System Specifications

Table 5.7 lists the most important specifications of the pressurized BBS matched to the IDE-2/TC engine. Within the level of accuracy acceptable for this study, the unit installed cost (\$170/Bhp) and the relative annual O&M cost (4.9 percent of installed capital cost) can be taken to apply also to pressurized Brayton bottoming systems designed for other engines in the same output class, e.g., TC/A, TCPD and TCPD/A engines.

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\*To be multiplied by the clean surface value.

TABLE 5.7

PRESSURIZED BBS CHARACTERISTICS

(NASA TC Engine)

<u>Performance</u>	
Net Output (Clean HX)	39.2 Bhp
Fouling Penalty	1.3 Bhp
Bsfc Improvement	~12%
<u>Weight (Without Gearbox)</u>	330 lb
<u>Installed Cost (1983\$)</u>	\$170/Bhp
<u>Total Annual O&amp;M Expenses (1983\$)</u>	\$300

## 6. VEHICLE PERFORMANCE ANALYSIS

### 6.1 Introduction

This chapter documents the results of the vehicle performance analysis conducted for the various powerplants considered in this study. Vehicle performance is broken into two categories: fuel economy and drivability. Drivability refers to the vehicle's grade capability, starting acceleration, and maximum speed for a given gross vehicle weight (GVW). These two figures of merit are presented for a wide range of driving conditions and GVWs. A proprietary UTRC truck performance simulation was used to calculate vehicle performance for the study. The computer code is described in Section 6.2.

### 6.2 Powertrain Simulation Description

The major tool of this analysis is a computer program which evaluates the performance and operational capabilities of large diesel trucks. This program was developed under UTRC Corporate funding and is documented in Ref.(6.1).

The main function of the program is to predict fuel economy for truck operation over arbitrary driving cycles. The driving cycles are specified in a novel way wherein the cycle is represented by a matrix whose entries define the distance traveled at selected combinations of speed and grade. Fuel economy is computed in terms of both mpg and ton-mpg.

The program can accept the engine and drivetrain characteristics of a given vehicle or will select the proper axle ratio, number of gears and gear ratios. The gear split, or minimum rpm change is determined by the ratio of the maximum rpm to the peak-torque rpm while number of gears (and the minimum first gear ratio) is determined by the starting torque requirements of the vehicle. Drive axle ratio depends on the desired maximum speed of the vehicle and the maximum rpm of the engine.

Vehicle performance in trucks is often specified by comparing the power available from the engine in each gear to the power required for different grades, both as functions of vehicle speed. The program prepares computer-graphic plots of these functions as well as plots of maximum grade capability in each gear as a function of vehicle speed. Engine speed versus vehicle speed is also plotted. The computer graphics package produces contour plots of engine fuel flow and specific fuel consumption (sfc). The figures in this chapter are illustrative of the graphic output available from this program.

### 6.3 Vehicle and Chassis Data

The vehicle and chassis data used in this study is shown in Table 6.1. This data describes a diesel truck considered to be representative of modern long-haul vehicles. Several sources were used in the compilation of this data; the GVW was defined by NASA and the remainder of the data was obtained from References (6.2) and (6.3).



TABLE 6.1

BASELINE VEHICLE SPECIFICATIONS

Gross Vehicle Wt.	73,000 lb
Geared Top Speed	60 mph
Number of Gears	8
Axle Ratio	3.4101
Transmission Eff.	0.95
Rear Axle Eff.	0.95
Maximum Power	300 hp
Drag Coeff. x Frontal Area	65.8 sq ft
Rolling Resis- tance Coefficients	0.0068 lb/lb 0.00074 lb/lb/mph
Wheel Radius (520 rev/mile)	1.592 ft
Fuel Density	7.0 lb/gal

#### 6.4 Engine Data

Details of the engine/BBS performance and its derivation are given in Chapters 3 and 5 of this report. Figure 3.1 summarizes the performance of the baseline IDE-TC engine. The performance of the three engine/BBS power plants, IDE-1/TC/BBS, IDE-2/TC/BBS, and IDE-2 TC/BBS with variable reduction transmission, are summarized in Figures 6.1 through 6.3, respectively. The first figure for each engine displays fuel flow as a function of engine speed and torque. This, along with the full-rack torque curve, is the format in which the engine characteristics are input to the computer program. The second figure shows lines of constant fuel flow in the torque-rpm plane bounded by the full-rack torque curve. The third figure for each engine shows the familiar sfc contour map with lines of constant sfc. The sfc plots are a convenient way to display engine data because, in addition to showing hp-normalized fuel consumption, sfc is inversely proportional to thermal efficiency.

Two trends can be identified when the baseline (IDE-1/TC) engine is compared with the three engines equipped with a Brayton Bottoming System (BBS). First, the values of the sfc contours are significantly lower for the three BBS-equipped engines than for the baseline engine. This indicates that the BBS engines will have improved fuel economy over the baseline at all engine operating points. It can also be noted that the position of the center of the "islands"--the point of greatest fuel economy--shifts from engine to engine. In general, the island shifts to points of greater engine speed and torque when the bottoming cycle is added since more waste heat is available at higher power levels. This implies that while the BBS engine will always out-perform the baseline engine in terms of fuel economy, the actual change will vary with the particular driving cycle over which the vehicle is operating. Hence, it is important that the driving cycles used for comparing fuel economy be considered carefully in terms of their relevance to actual vehicle operation conditions.

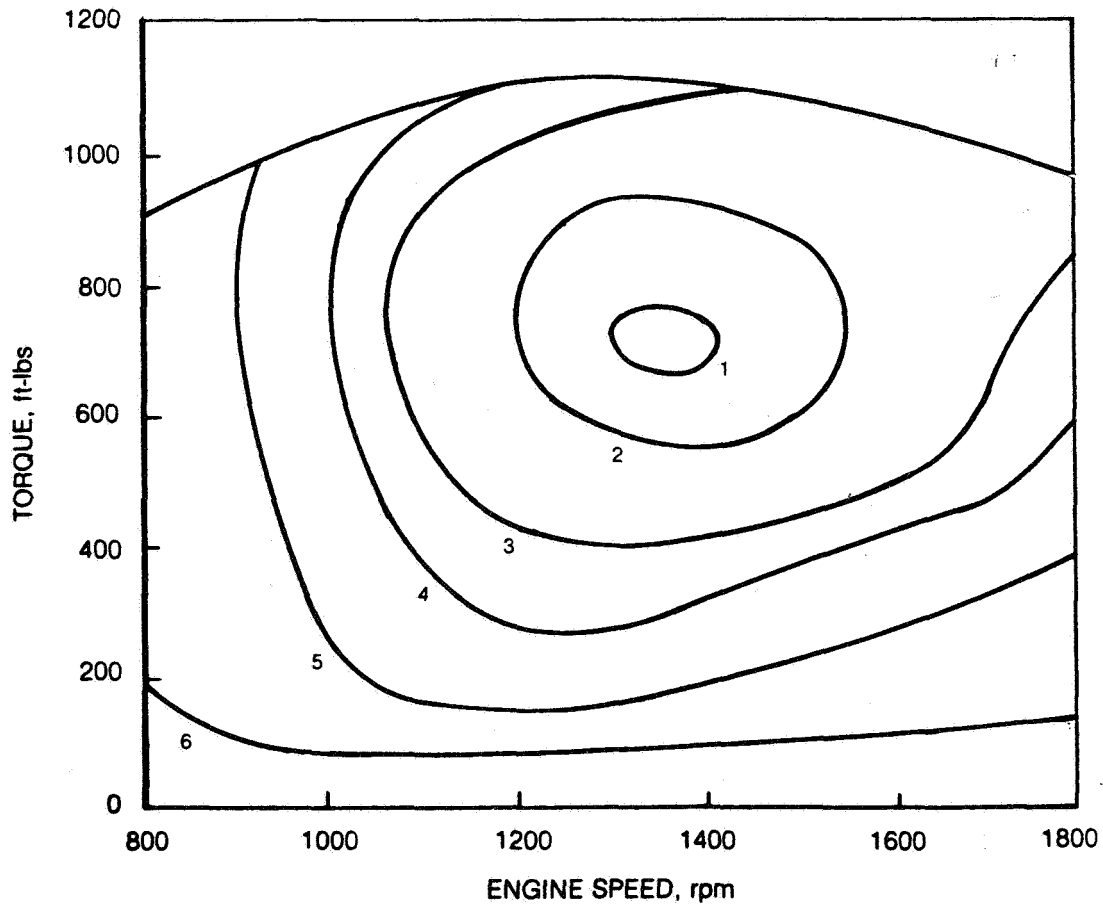
#### 6.5 Engine Scaling

The BBS-equipped engines offer improved performance relative to the baseline engine because each BBS engine has at least 30 hp more power than the baseline engine. This means there will be improvement in top speed, grade capability, etc. when the bottoming cycle is added. Since vehicle performance includes both fuel economy and gradability, comparing only the fuel economy results for the BBS-equipped engines (with increased performance) to those of the baseline engine may not be somewhat misleading; the BBS engines will always out-perform (out-accelerate) the baseline engine. A BBS-equipped engine scaled (reduced in size or displacement) will yield higher fuel economy than an unscaled engine. Thus, another way to evaluate the engine would be to compare fuel economy of power plants of equal power ratings, in effect giving a "normalized" comparison between engines. The computer program has a facility to allow this comparison to be made. An engine scaling feature has been included so that the effects of variable engine ratings can be investigated using only

**SFC CONTOUR MAP**

IDE-1/TC/BBS

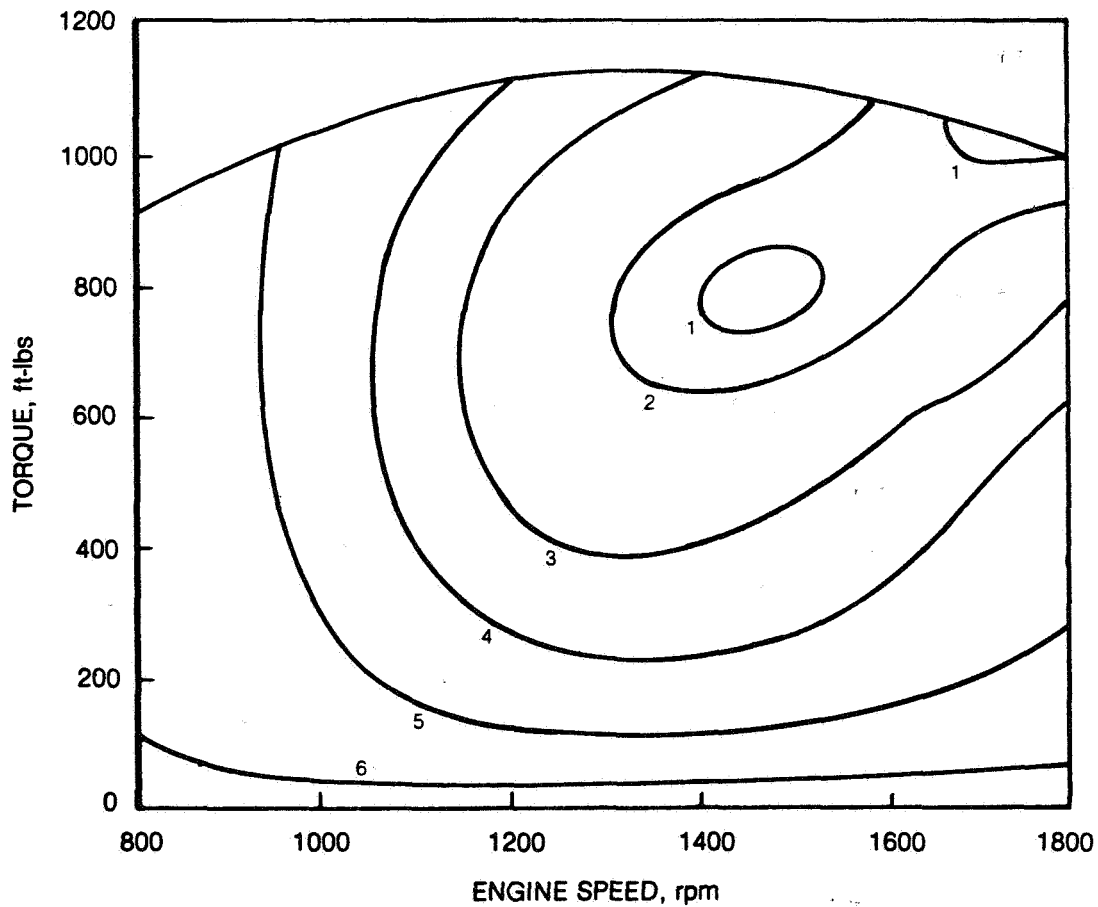
CONTOUR LEVELS	
No.	SFC (lb/hp-hr)
1	0.280
2	0.285
3	0.295
4	0.305
5	0.325
6	0.400



**SFC CONTOUR MAP**

IDE-2/TC/BBS

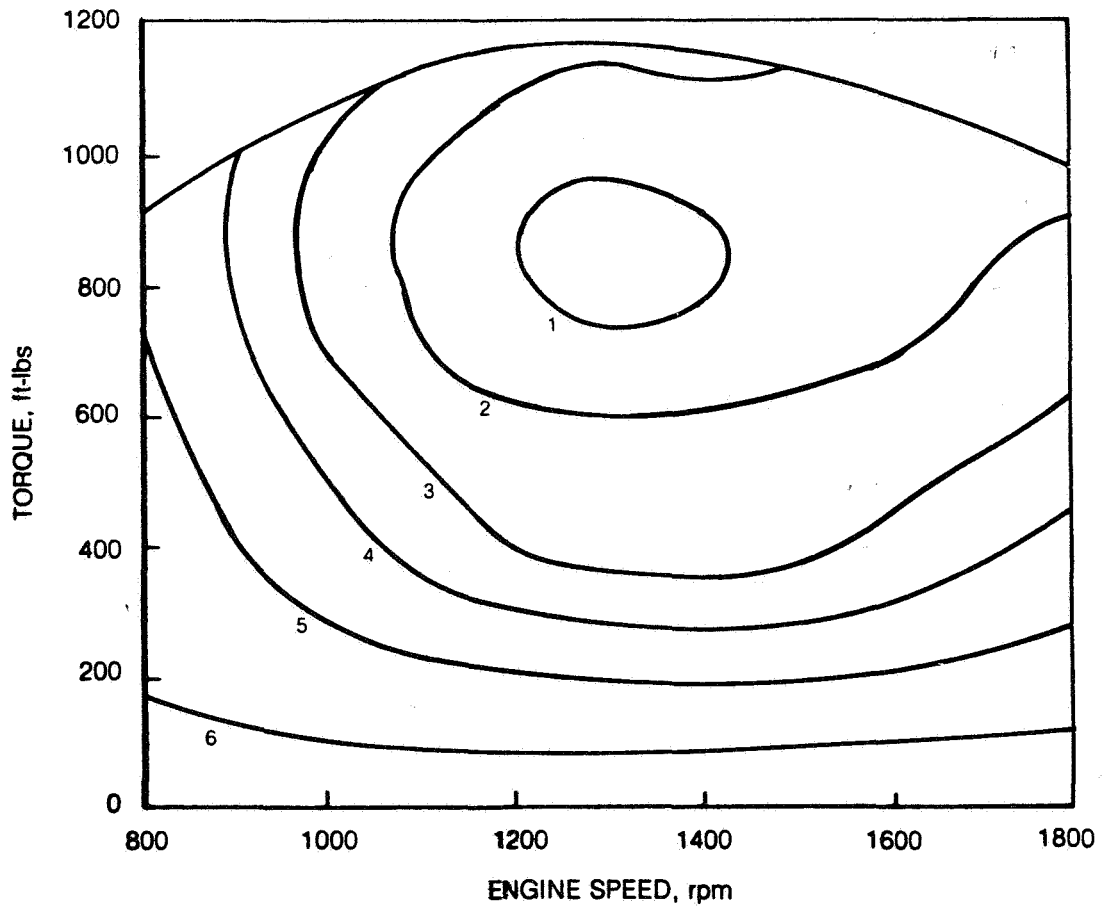
CONTOUR LEVELS	
NO.	SFC (LB/HP-HR)
1	0.282
2	0.285
3	0.295
4	0.305
5	0.325
6	0.400



**SFC CONTOUR MAP**

IDE-2/TC/BBS WITH VARIABLE REDUCTION TRANSMISSION

CONTOUR LEVELS	
NO.	SFC (lb/hp-hr)
1	0.278
2	0.285
3	0.295
4	0.305
5	0.325
6	0.400



one set of engine data. This "rubber engine" can be changed in size (torque or displacement) by changing only one scaling parameter. This feature was utilized on the three BBS-equipped, resulting in a total of seven different engines.

To illustrate the effect of scaling, the change in fuel economy was examined at three different engine speeds for each of the BBS engines. The results are shown in Figures 6.4 through 6.6. There are two important points to note. First, there may be either an increase or a reduction in fuel economy depending on the particular operating point of the engine. In Figure 6.4, for example, the scaled IDE-1 TC/BBS power plant shows an increase in fuel economy over the unscaled one for an engine speed of 1300 rpm and up to 70 percent torque. When torque is increased above 70 percent, the scaled engine suffers a reduction in fuel economy compared to the unscaled engine. It should be noted, however, that the magnitude of these changes is small--on the order of a few percent. This scaling effect is lessened even further in the course of operation as these differences will tend to cancel one another as torque and speed vary. Note that points of average engine speed and average engine torque have been marked on the figure for various driving cycles which are described later. The effect of engine scaling on overall driving cycle fuel economy is seen to be small. Since engine scaling was shown to have only a slight effect on fuel economy, the fuel economy data given in the following sections apply both to the scaled and unscaled power plants. The performance benefits resulting from the addition of the bottoming systems to the unscaled engine are presented as improvements in both acceleration (or gradability) and fuel economy.

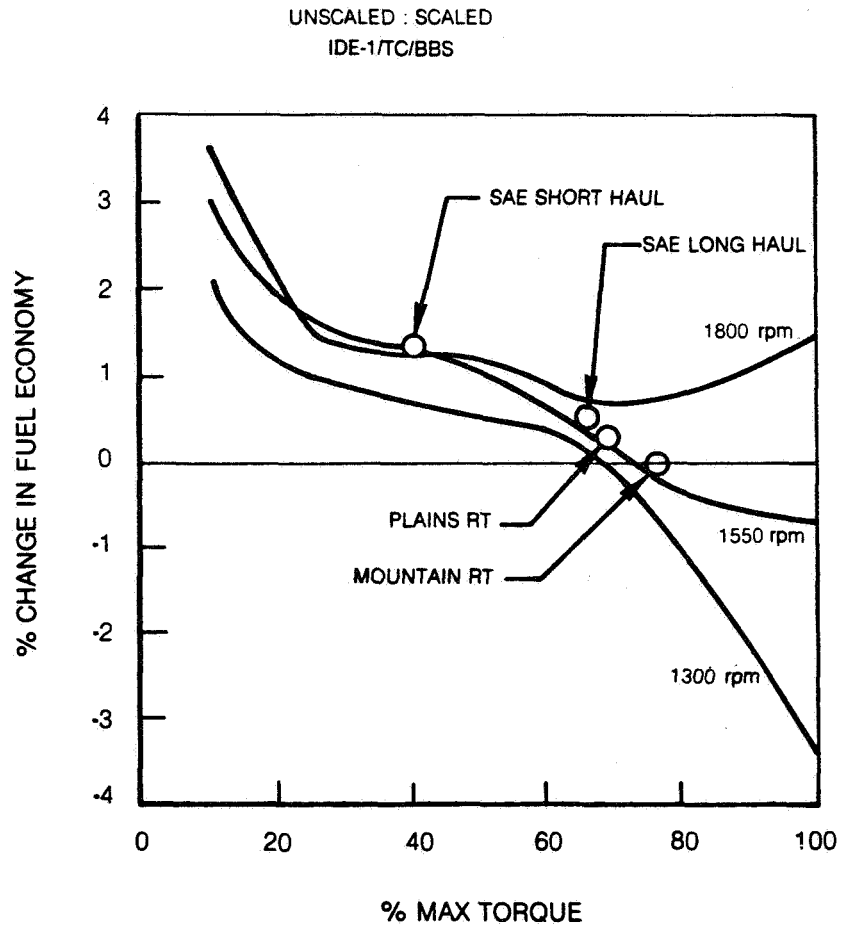
#### 6.6 Measures of Fuel Economy

The importance of choosing the correct driving cycle over which to measure fuel economy was discussed in Section 6.4; different driving cycles will yield varying improvements in fuel economy for the BBS-equipped engines when compared to the baseline engine. A driving cycle which realistically simulates actual operating conditions must be chosen.

Several measures of fuel economy are examined in this study. The first measure is fuel economy at the design point, which is the point of maximum rpm and full-rack torque. This is the traditional measure used in engine design. Clearly, the design point method is inadequate in terms of realism; all trucks do not operate at only one point on the engine map. It will be seen that the fuel economy results based on the design point do not agree with those derived from a typical driving cycle. Nevertheless, this measure has been recommended by NASA for this study to facilitate comparison with other types of bottoming systems.

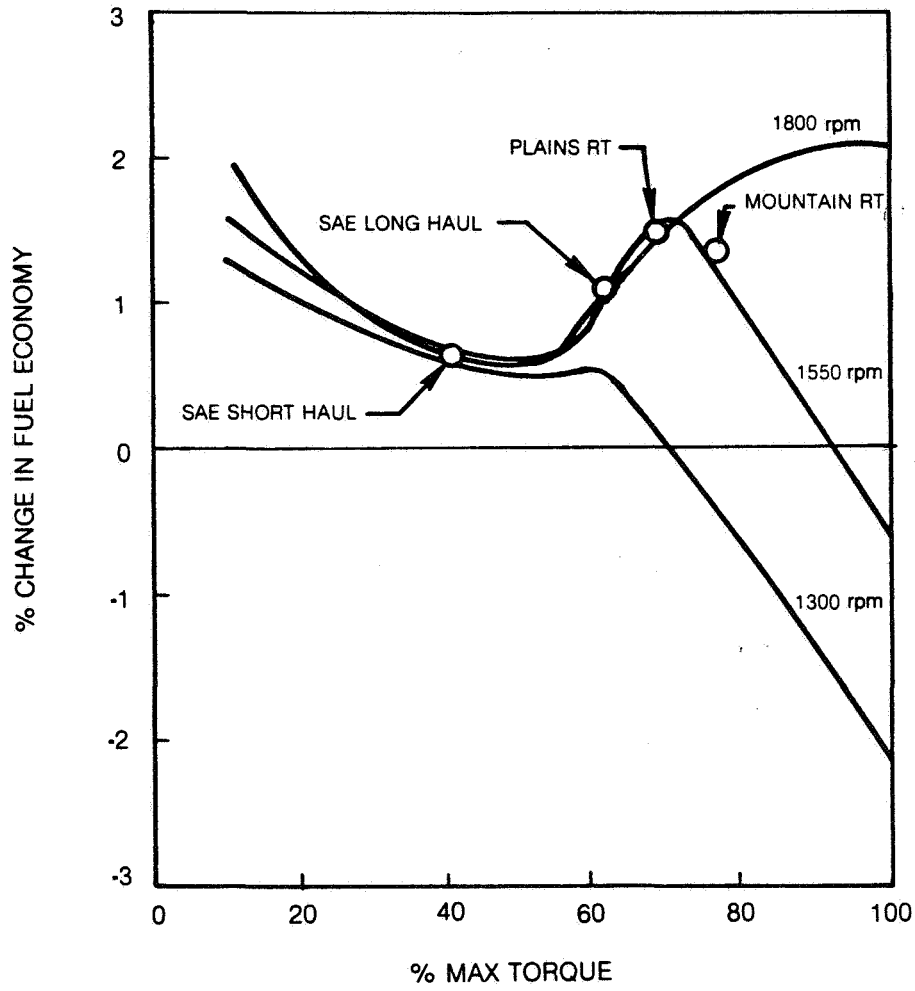
There is no industry-wide standard driving cycle over which to measure fuel economy for trucks like that used for passenger cars. Two cycles promulgated for the trucking industry are the SAE Short Haul and Long Haul cycles (Ref. 6.4) shown in Figure 6.7. There is one major shortcoming of the SAE cycles, namely, that they have no grade requirements. The fuel economy predictions based on these cycles will tend to yield optimistic results because most trucks spend a significant portion of their operational life traveling up and down hills with at least 0.5 percent grade. Even grades this shallow have a definite affect on truck

### EFFECT OF SCALING ON FUEL ECONOMY



### EFFECT OF SCALING ON FUEL ECONOMY

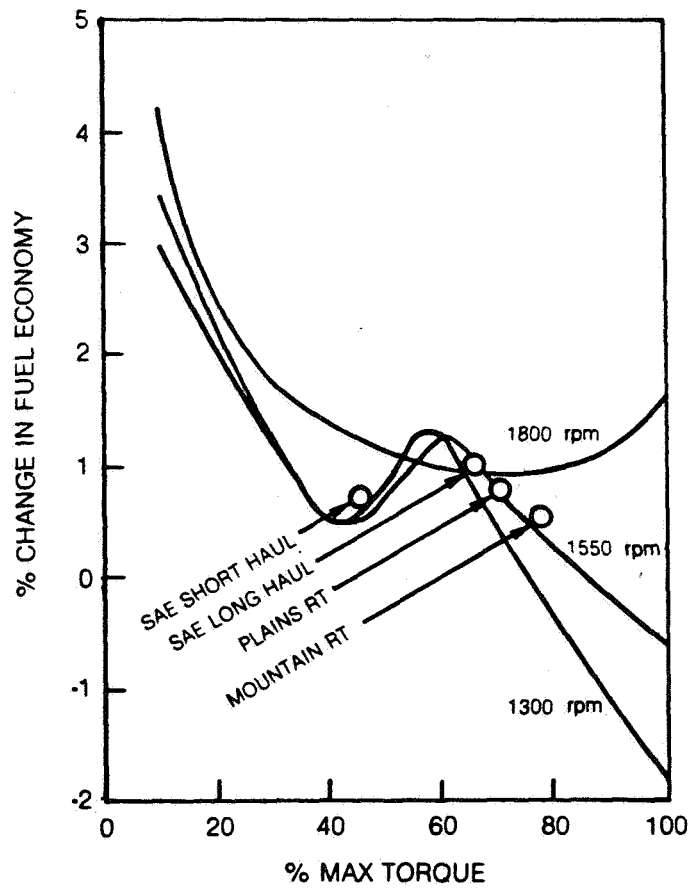
UNSCALED:SCALED  
IDE-2/TC/BBS



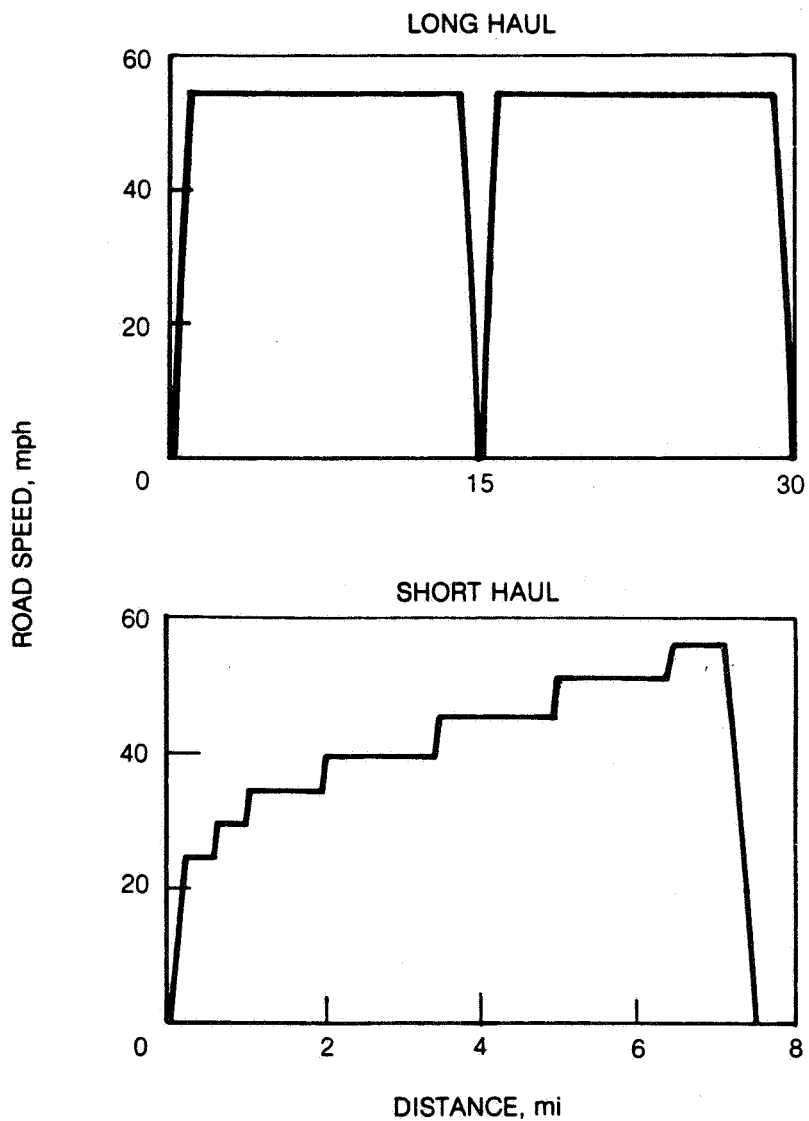


### EFFECT OF SCALING ON FUEL ECONOMY

UNSCALED SCALED  
IDE-2/TC/BBS WITH  
VARIABLE REDUCTION TRANSMISSION



### SAE DRIVING CYCLE SPECIFICATIONS



performance, owing to the great weight of the vehicle. Hence, fuel economy results derived from the SAE cycles will not accurately reflect actual road fuel economy. The SAE figures are included herein for the purpose of comparison.

A driving route which more accurately reflects the typical operation of trucks is the Cummins Plains Route (Ref. 6.5). The Cummins Mountain Route described in the same report is also considered. The specifications for these routes are given in Fig. 6.8 in terms of grade and distance. Speed is not a specified parameter, hence the trucks may travel as fast as possible, being constrained by only engine power and the applicable speed limit. Thus, there is an additional parameter, vehicle speed, in the Cummins fuel economy results. The Mountain Route probably contains grade requirements which are too severe to represent average conditions.

Fuel economy was also examined parametrically. In these studies, either gross vehicle weight or speed was varied to obtain sensitivities about a baseline condition.

### 6.7 Fuel Economy Results

Five measures of fuel economy were considered in this study: the design point, SAE long and short haul routes, and the Cummins Plains and Mountain routes. The design point fuel economy figures were calculated by hand using a straightforward formula for calculating the mpg as a function of vehicle speed and power. The GVW was assumed to be 73,000 lbs and the maximum speed was assumed to be 60 mph, this being the geared top speed of the vehicle. The equation for computing fuel economy at a steady state speed can be expressed as

$$\text{mpg} = \rho V / (\text{HP} \cdot \text{sfc}) \quad (6.1)$$

where

HP = power, hp  
sfc = specific fuel consumption lb/hp-hr  
V = speed, mph  
 $\rho$  = fuel density, lb/gal

For design point fuel economy, the formula assumes that the engine is operating at full power at its rated speed. Operation of the vehicle at this condition tends to yield optimistic values of fuel economy because of the high speed; on the average, the vehicle will not be in high gear at top speed even though average power is high.

The fuel economy results comparing the three power plants for the selected driving cycles are presented in bar graph form in Figure 6.9. The SAE short haul cycle was not included because the low speed and zero-grade requirements are not sufficiently representative of heavy duty, long-haul truck operation. For operation over this low-power cycle, average fuel economy was about 7.5 mpg. Average fuel economy figures of 4 to 5 mpg were obtained with the exception of the SAE long haul driving cycle.

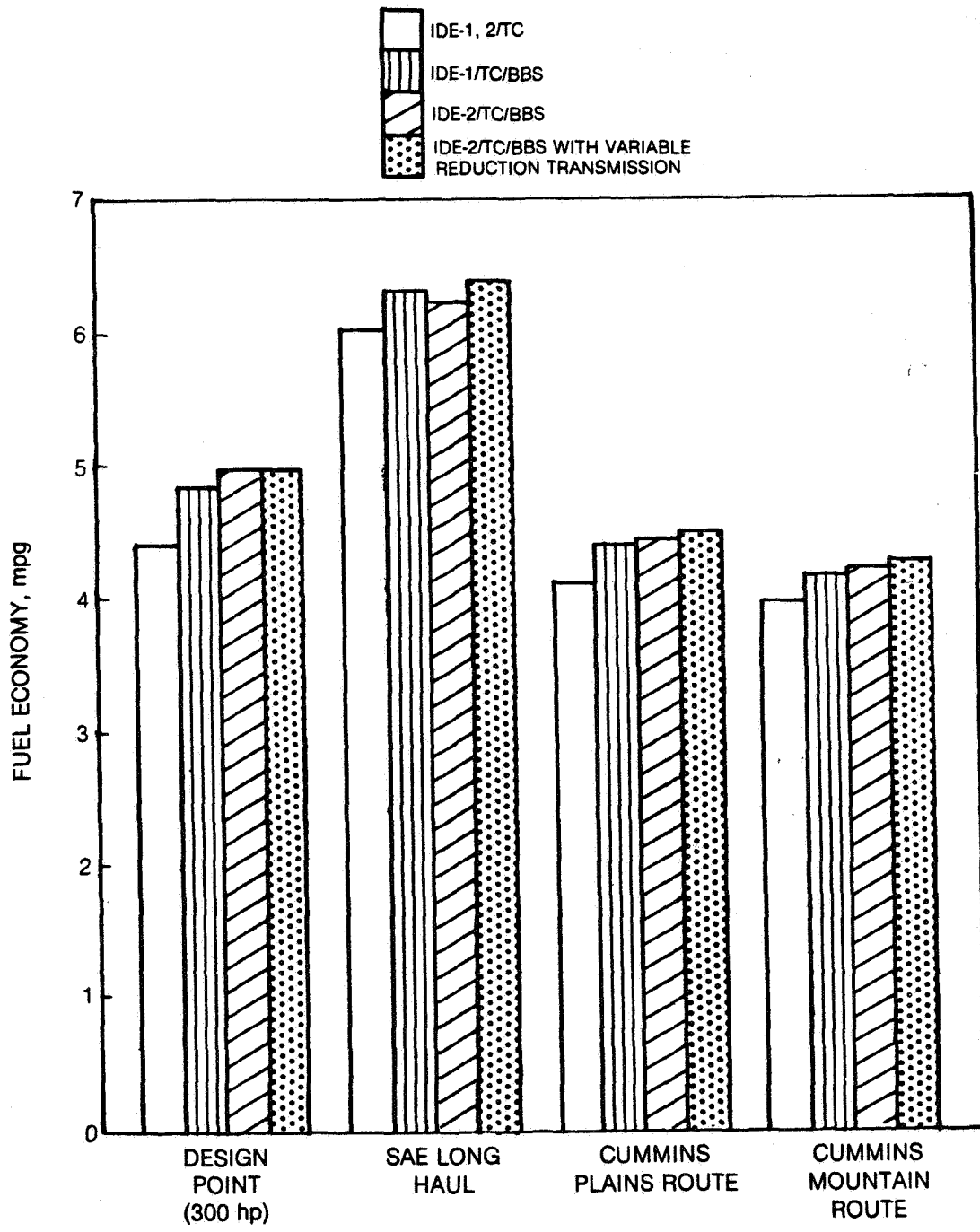
Substantially better fuel economy was obtained for the SAE long haul driving cycle than for the other measures presented in Figure 6.9. Fuel economies in the range of approximately 6.25 mpg were observed. This value

**CUMMINS DRIVING CYCLE SPECIFICATIONS**

ROUTE	GRADE RANGE, %	UPGRADE DISTANCE, mi	DOWNGRADE DISTANCE, mi
PLAINS	0 to 1	17.8*	8.3
	1 to 2	10.0	10.1
	2 to 3	1.7	2.1
	3 to 5	0.0	0.0
	5 to 7	0.0	0.0
	Over 7	0.0	0.0
MOUNTAIN PASS	0 to 1	8.8*	5.8
	1 to 2	10.6	10.5
	2 to 3	13.0	13.8
	3 to 5	18.4	15.8
	5 to 7	0.8	2.5
	Over 7	0.0	0.0

\*LEVEL ROAD IS INCLUDED IN THESE FIGURES

**FUEL ECONOMY RESULTS — MPG**



is greater than for the Cummins cycles because the zero-grade, SAE long-haul cycle places lower loads on the vehicle than the latter cycles. As noted in Section 6.6, these loads are lower than can be expected in normal heavy-duty truck operation. Hence, the SAE long haul figures should not be used in assessing the economics of the BBS in this study. It is interesting to note how engine scaling affects the five measures of fuel economy. Figures 6.4 through 6.6 show the effect engine scaling will have for the three BBS engines. The circled points in the graphs correspond to the average torque and rpm for the respective cycles, and indicate improvements of only one percent or so when engine scaling reduces the design-point power to 300 hp. For example, when the IDE-2/TC/BBS engine is scaled, there is a 1.5% increase in fuel economy on the Cummins Plains Route.

A more informative display of the fuel economy results is shown in bar graph form in Figure 6.10. Here the annual amount of fuel saved by each of the three BBS engines, as compared to the baseline engine, is displayed. Savings are seen to range from 800 to 2300 gallons a year. These numbers are easily derived from the mpg figures: The amount of fuel consumed on an annual basis for each engine can be found by dividing the distance traveled in one year by the mpg. Here it is assumed the vehicle will travel 100,000 miles annually. The numbers shown in Figure 6.10 are then the differences between amount of fuel consumed by the BBS engines and the baseline engine.

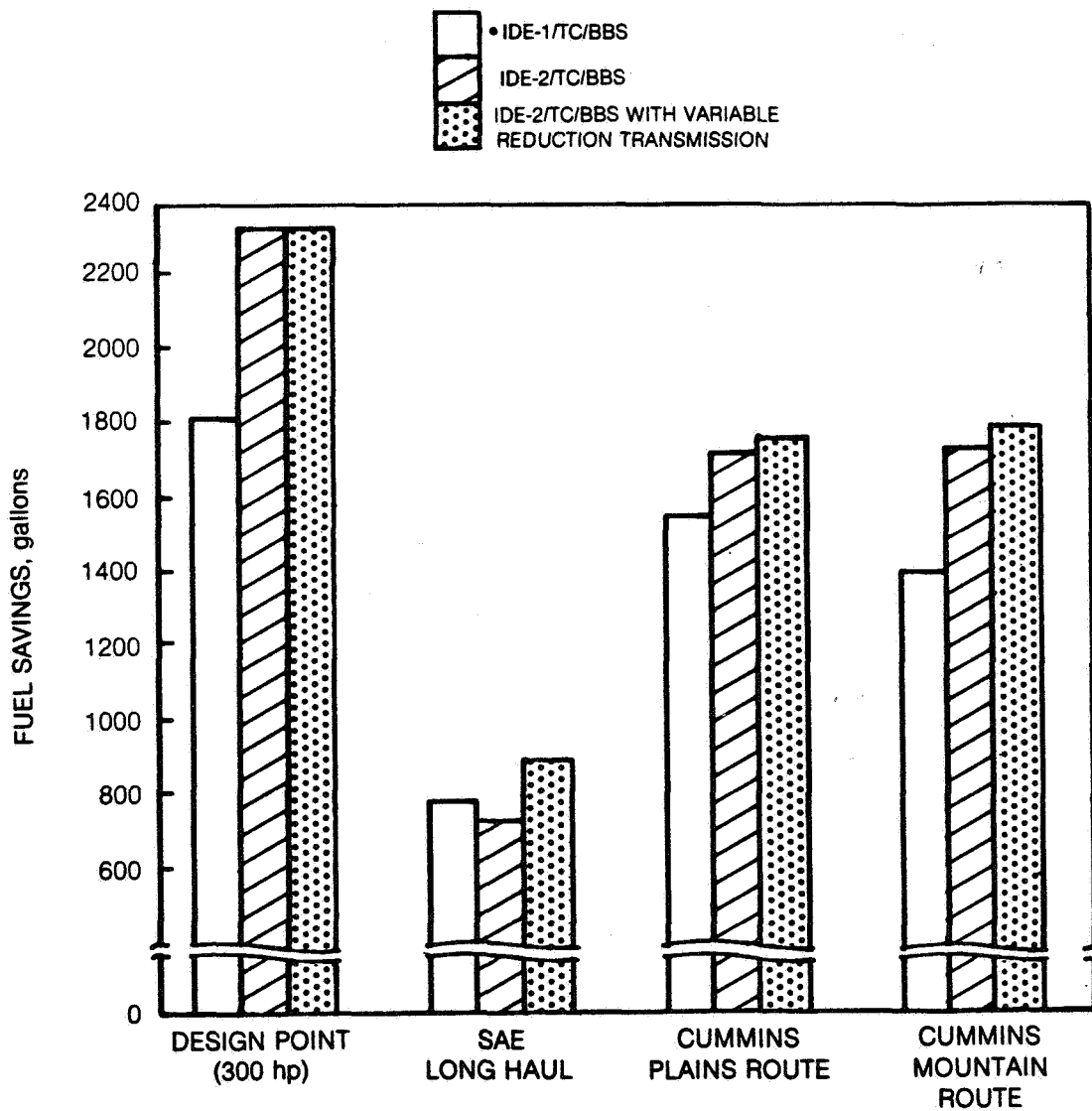
The design point figure is quite optimistic in its prediction, with about 2300 gallons of fuel being saved per year as compared to 800-1600 gallons being saved by the other fuel economy measures. If the criterion that a measure of fuel economy should accurately portray the vehicle's anticipated driving conditions is applied, then actual fuel savings would most likely be reflected by the Cummins plains route. In general, annual fuel savings increase as the average power expended by the cycle increases. This is because more waste heat becomes available to the BBS at the higher power conditions.

In addition to the one-point estimates, fuel economy as a function of weight for each of the four driving cycles was obtained. These results are shown in Figures 6.11 through 6.13. The sensitivity of fuel economy to weight can be judged with the aid of these curves. It should be noted that for the Cummins routes, as weight increases both fuel economy and vehicle speed decrease. The Cummins plains route results (Figure 6.12) provide a typical example. For this example, a 10,000 lb increase in GVW results in a drop in fuel economy of about 0.5 mpg and a decrease in average speed of over 5 mph.

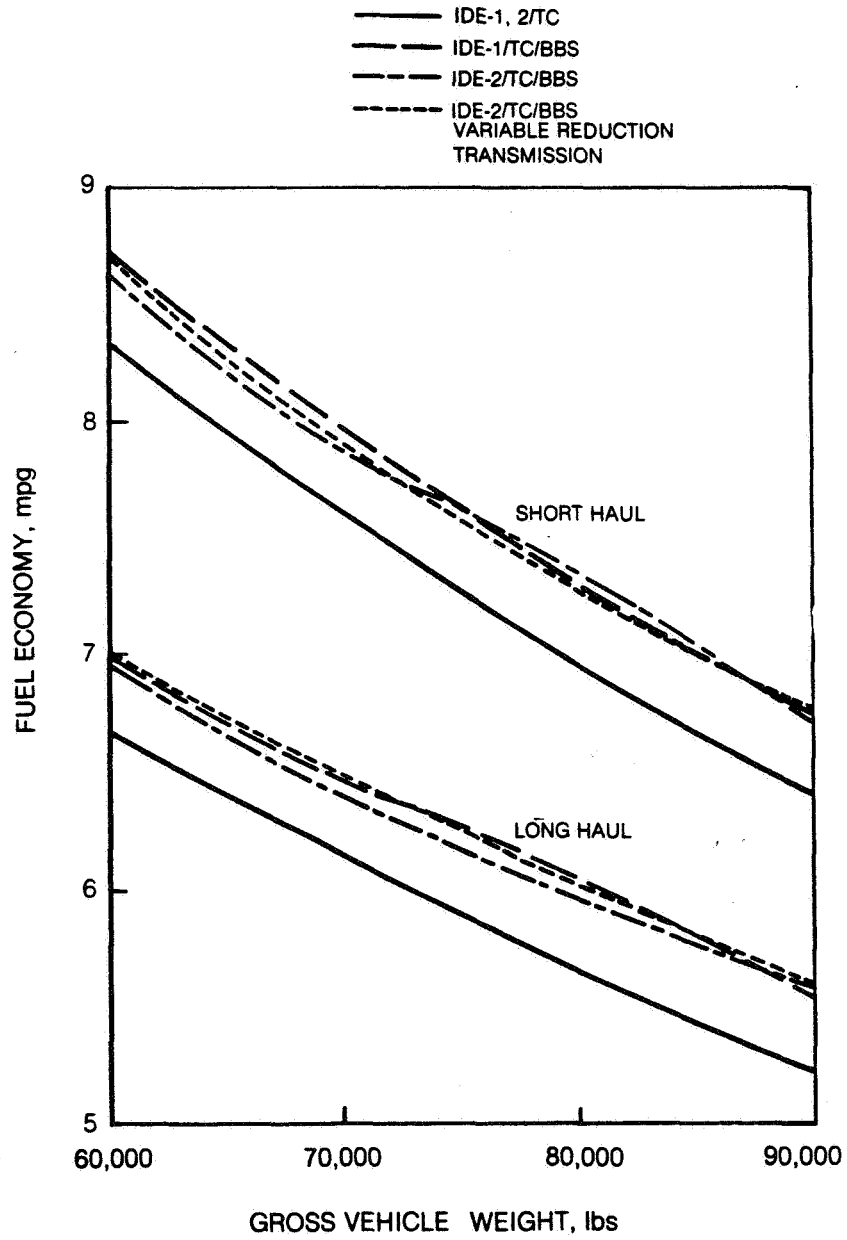
Other parametric studies were undertaken as well. In Figure 6.14, the dependence of fuel economy on GVW and percent grade is shown for a steady speed of 55 mph. It can be seen that for a given grade and constant fuel economy, the BBS-equipped engines offer improved load-carrying ability compared to the baseline engine; up to 7500 lb for the IDE-2 TC/BBS power plant. The profound impact of even one half percent grade can also be seen in this figure; to maintain constant fuel economy on a 0.5 percent grade, GVW must decrease by about 25,000 lbs. Alternately, an increase in grade of 0.5 percent, at a constant GVW reduces fuel economy by about 1.4 mpg (over 20 percent).

**FUEL ECONOMY RESULTS — ANNUAL FUEL SAVINGS**

100,000 MILES DRIVEN PER YEAR



SAE DRIVING CYCLE — FUEL ECONOMY

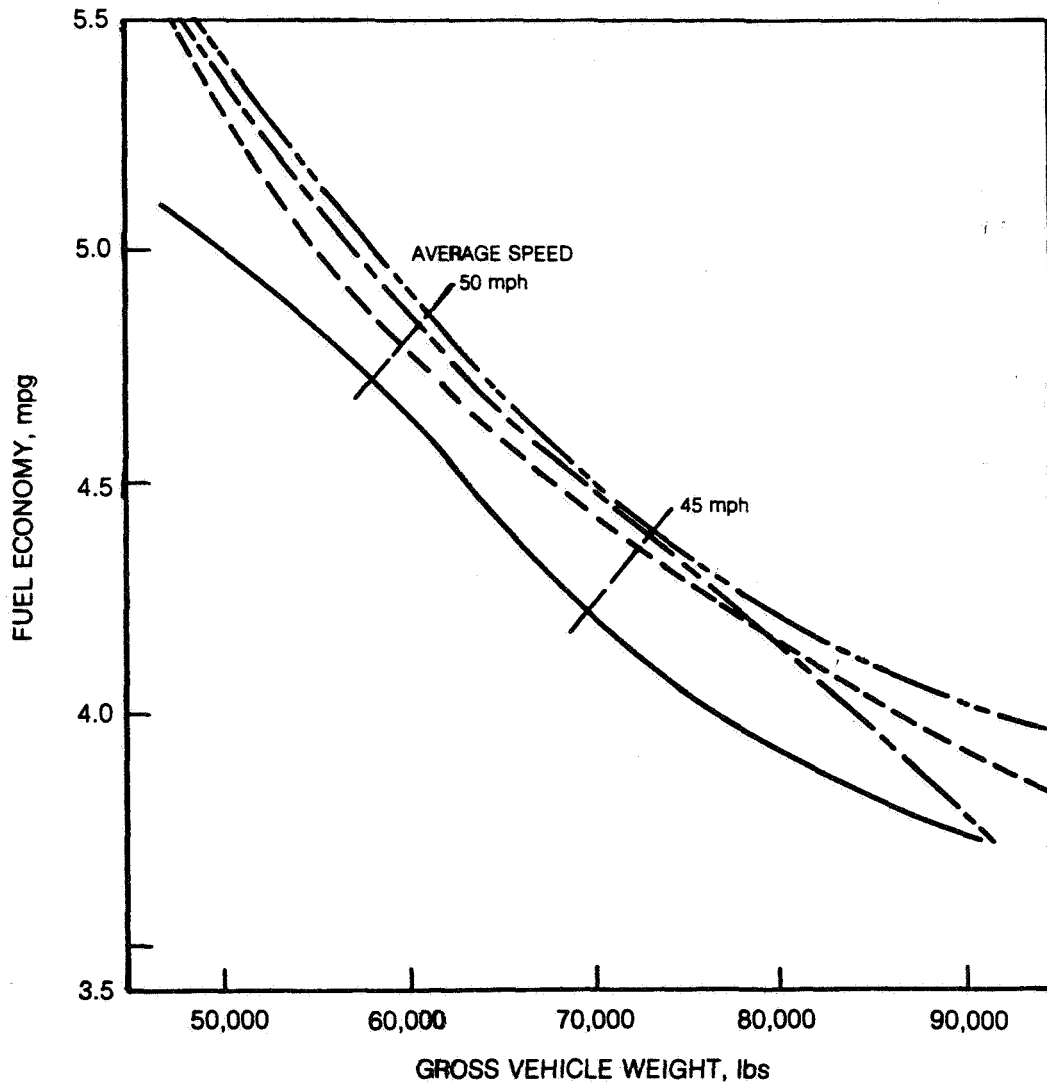




### CUMMINS PLAINS ROUTE FUEL ECONOMY

60 mph GEARED TOP SPEED

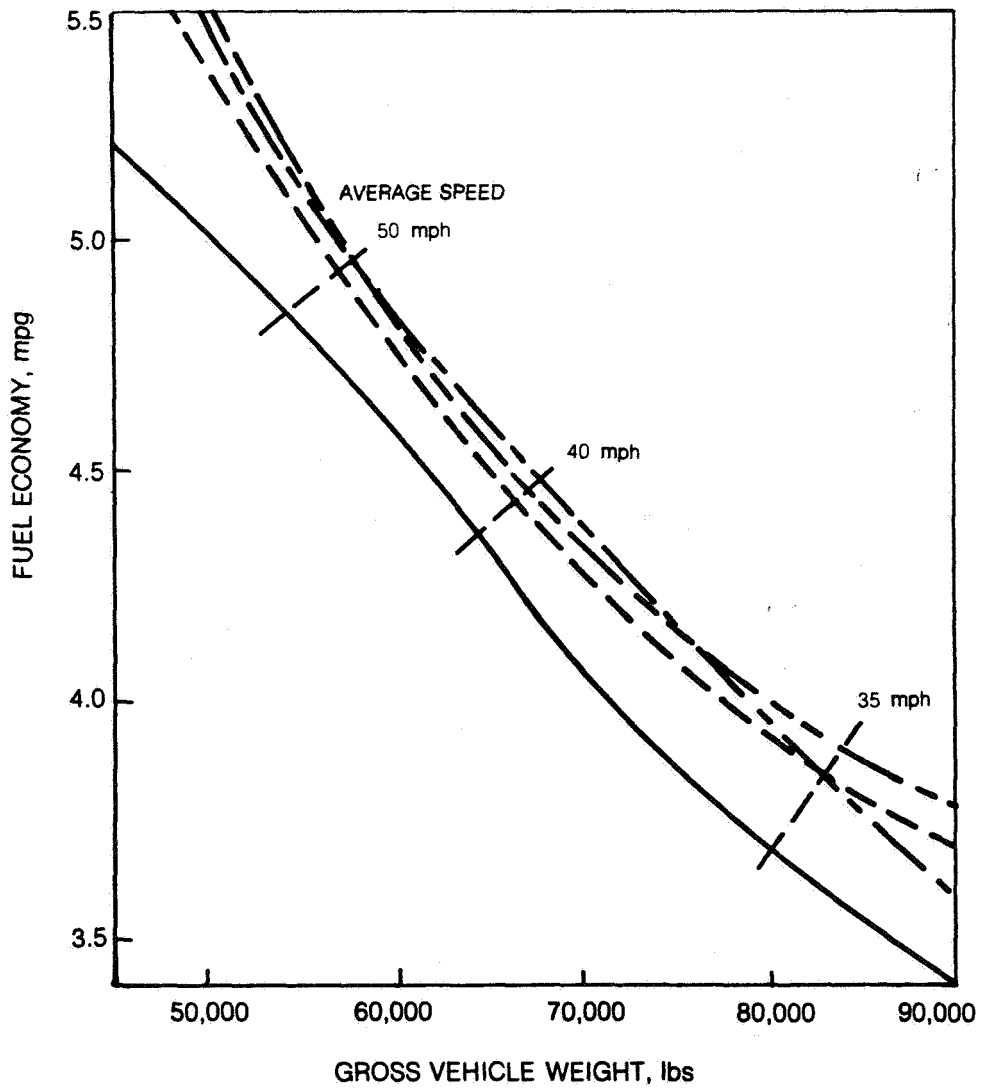
- IDE-1, 2/TC
- - - IDE-1/TC/BBS
- · - IDE-2/TC/BBS
- · - IDE-2/TC/BBS WITH VARIABLE REDUCTION TRANSMISSION



### CUMMINS MOUNTAIN ROUTE FUEL ECONOMY

60 mph GEARED TOP SPEED

- IDE-1, 2/TC
- - - IDE-1/TC/BBS
- · - · IDE-2/TC/BBS
- · - · IDE-2/TC/BBS WITH VARIABLE REDUCTION TRANSMISSION



### STEADY STATE FUEL ECONOMY

VARIATION OF GVW  
 60 mph GEARED TOP SPEED  
 CONSTANT 55 mph

- IDE-1, 2/TC
- - - IDE-1/TC/BBS
- - - IDE-2/TC/BBS
- - - IDE-2/TC/BBS WITH  
 VARIABLE REDUCTION  
 TRANSMISSION

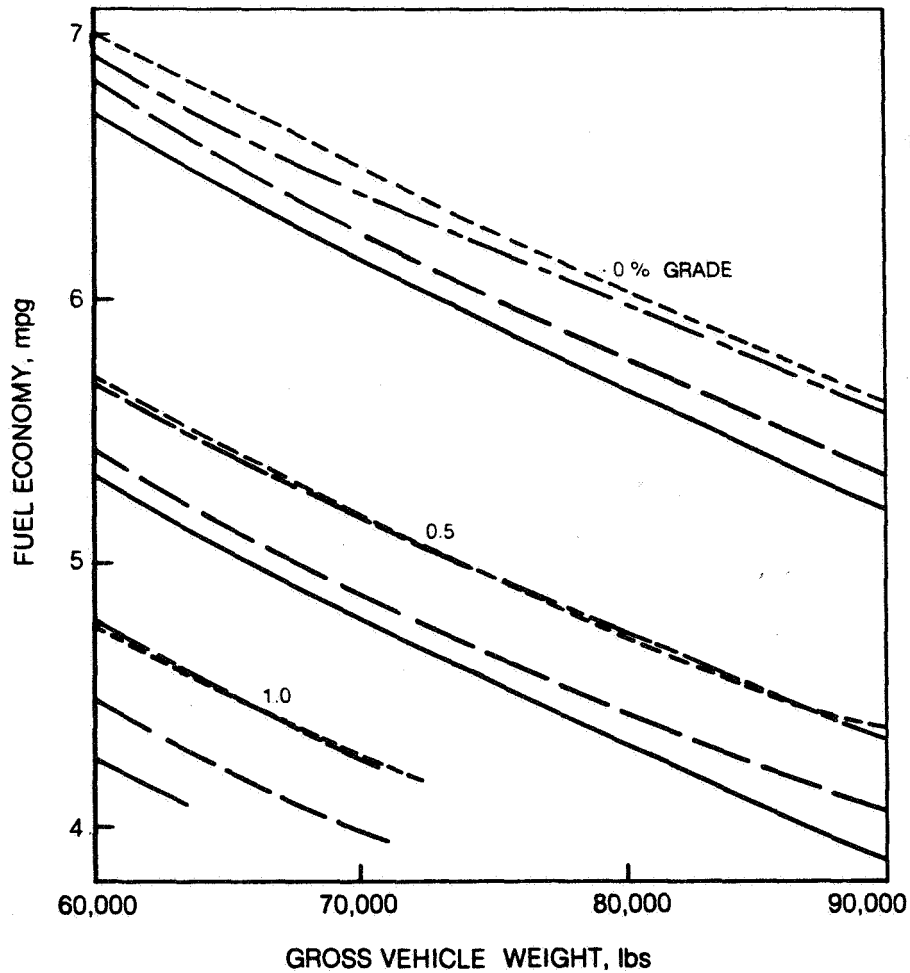


Figure 6.15 shows fuel economy as a function of vehicle speed for the standard 73,000 lb GVW truck traveling on various grades. The sudden jump in the curves is caused by a gear shift which occurs at about 43 mph. After the shift, the vehicle is operating at a higher torque and a lower sfc than before the shift, although the power is constant. The large values for fuel economy for low-speed, level-road operation compared to realistic driving cycle can be assessed by comparing these results to those shown in Figure 6.9.

## 6.8 Drivability

It is not possible to arrive at a single number which characterizes truck drivability in the same fashion that mpg characterizes fuel economy. Instead, this question can be addressed by parametric studies which compare the performance of trucks equipped with each of the four power plants. Figure 6.16 shows the relation between GVW and grade capability for a constant speed of 55 mph. It can be seen that for a 73,000 lb truck, the baseline engine will allow the truck to maintain speed on a maximum grade of 0.7 percent. However, the BBS-equipped engines will carry the same truck up a 1.0 percent grade at the same 55 mph. Another way to interpret this graph is as follows: if the truck is required to climb a 1.0 percent grade at 55 mph, the baseline engine will be able to haul a maximum of 63,000 lbs. The BBS-equipped engines will, however, haul approximately 73,000 lbs.

Figure 6.17 shows the grade capability of the baseline 73,000 lb GVW truck as a function of speed. The curves show that for most speeds, the BBS-equipped trucks are capable of ascending grades about 0.4 percent greater than the baseline vehicle. The discontinuity in the curves is caused by the reduction in power at the upshift which occurs at 43 mph.

The final parametric drivability evaluation examines the maximum vehicle speed on a one percent upgrade as a function of weight (see Fig. 6.18). The trucks have a geared top speed of 60 mph, hence the curves are truncated at that value. For example, a 73,000 lb vehicle powered by the baseline engine has a maximum speed of 51 mph on the upgrade, while the same truck fitted with one of the BBS-equipped engines will be able to reach a speed of about 55 mph. For a fixed weight, the BBS-equipped vehicles can travel about 3 to 5 mph faster on the upgrade.

The scaled engines speed, acceleration and grade capability results would be almost identical to those of the baseline engine.

## 6.9 Conclusions

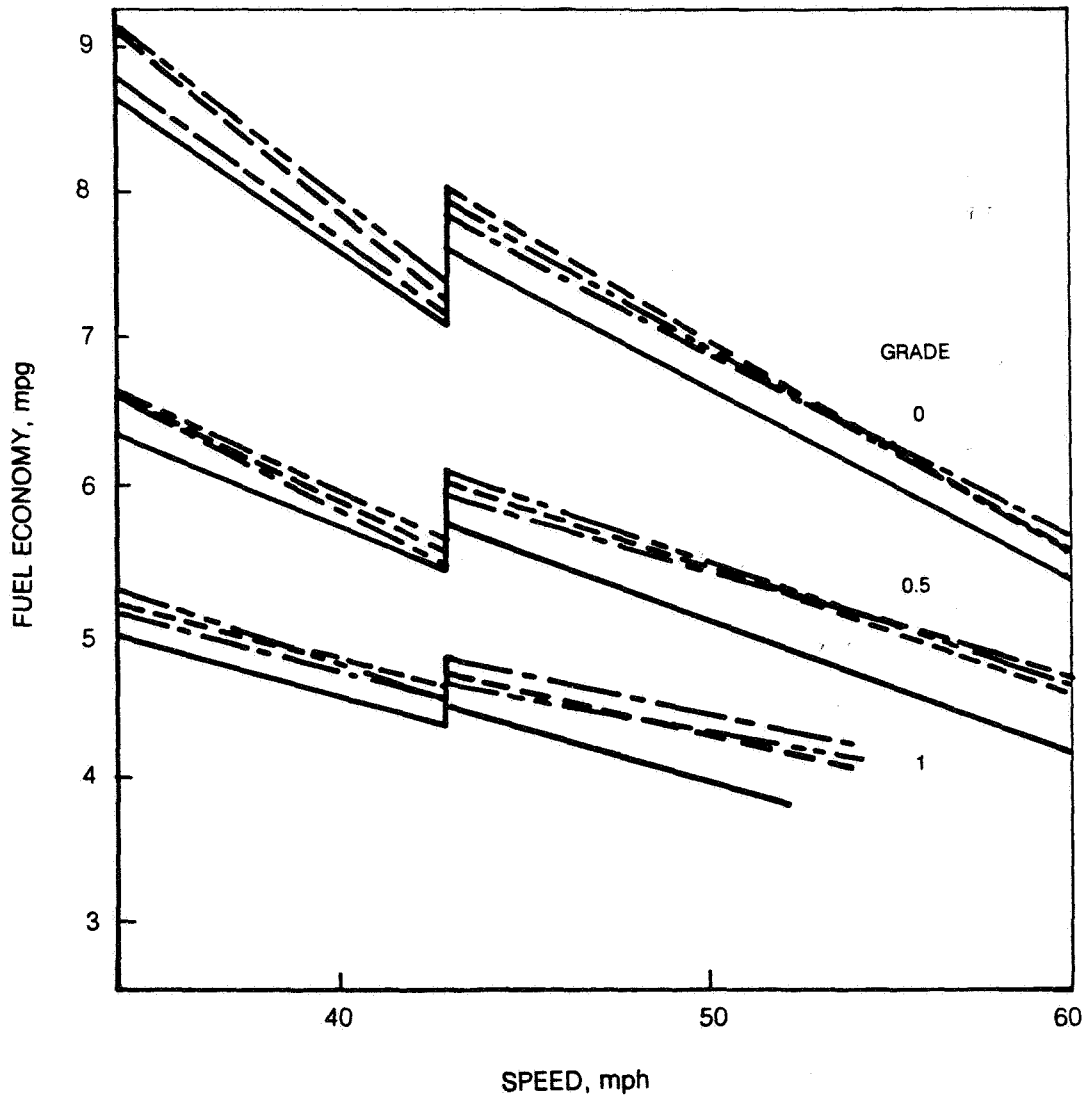
With the aid of the UTRC truck performance simulation program, a thorough analysis of the performance of the engines of interest has been conducted. Driving cycle and steady speed fuel economy has been computed including variations with respect to vehicle speed and weight. Also, truck drivability has been characterized and questions of engine scaling explored. Several conclusions can be drawn.

First, it has been shown that the measure of fuel economy must be carefully chosen. For any two of the engines considered, the difference in engine economy varies as the operating point is varied. Hence, should an

**STEADY STATE FUEL ECONOMY**

VARIATION OF CRUISE SPEED  
 60 mph GEARED TOP SPEED  
 73,000 lb GVW

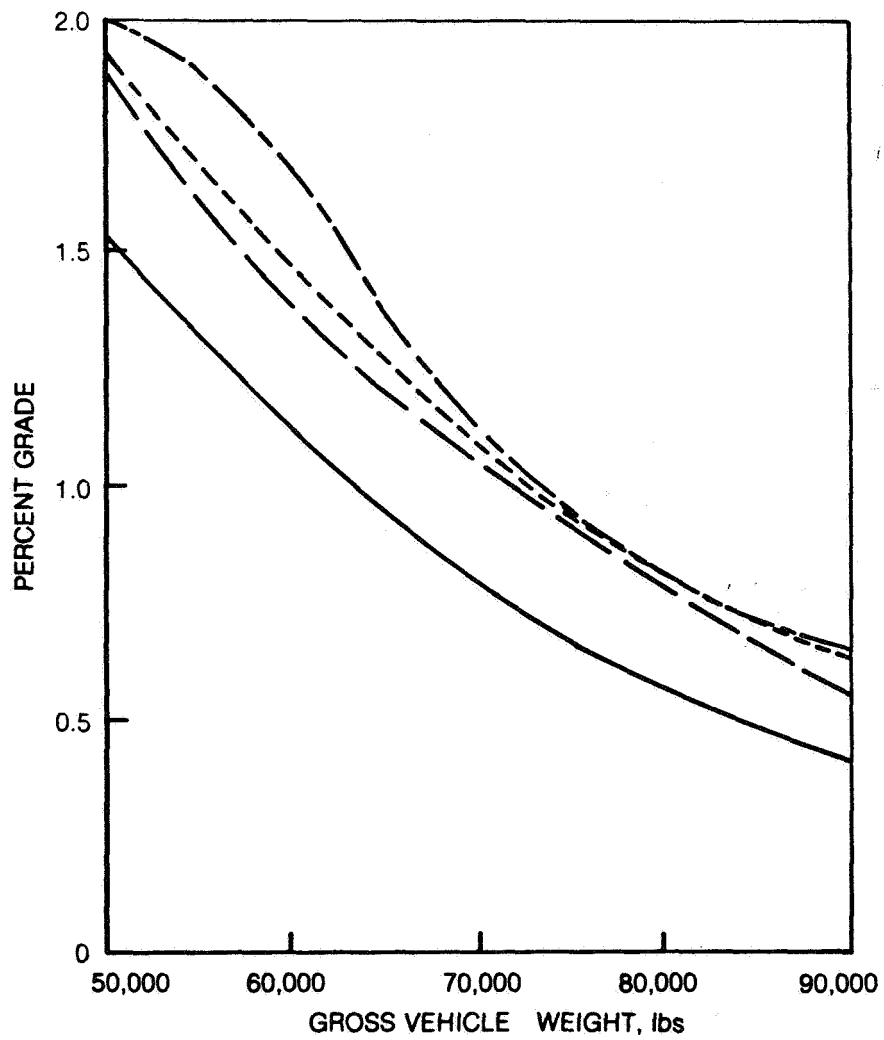
- IDE-1, 2TC
- - - IDE-1/TC/BBS
- · - IDE-2/TC/BBS
- · · - IDE-2/TC/BBS WITH VARIABLE REDUCTION TRANSMISSION



### GRADE CAPABILITY AT CONSTANT SPEED

55 mph CRUISING SPEED  
DEERE-TC ENGINE  
60 mph GEARED, TOP SPEED

- IDE-1, 2/TC
- IDE-1/TC/BBS
- - - IDE-2/TC/BBS
- - - IDE-2/TC/BBS WITH  
VARIABLE REDUCTION TRANSMISSION

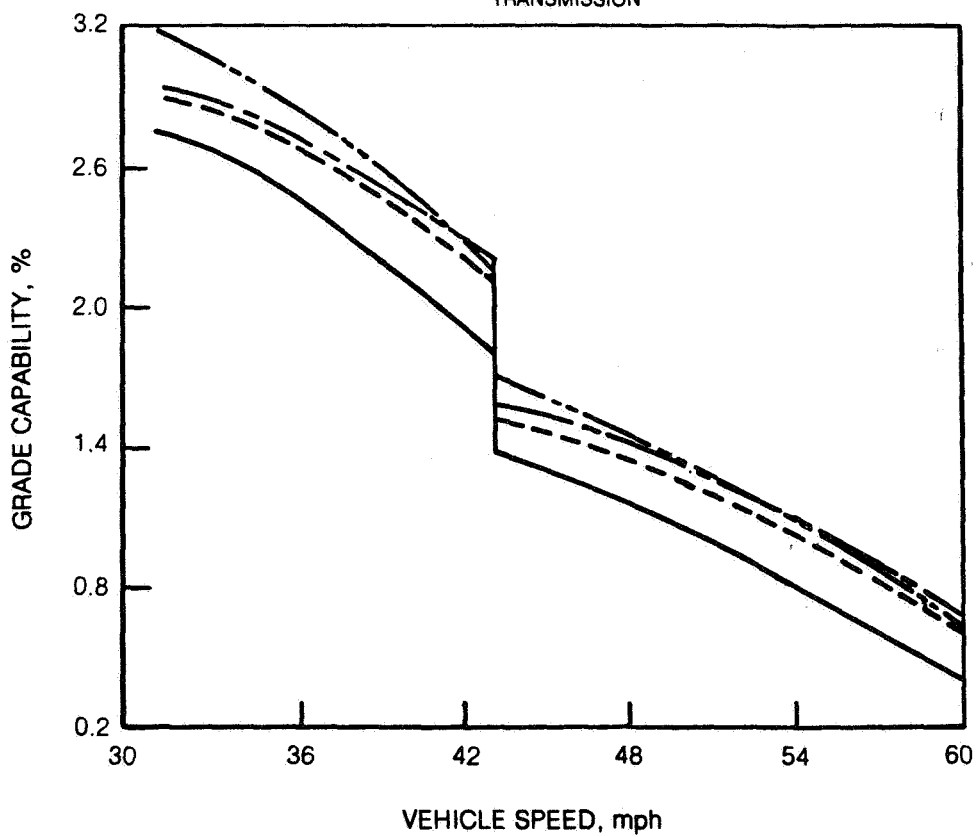


### GRADE CAPABILITY AT CONSTANT GWW

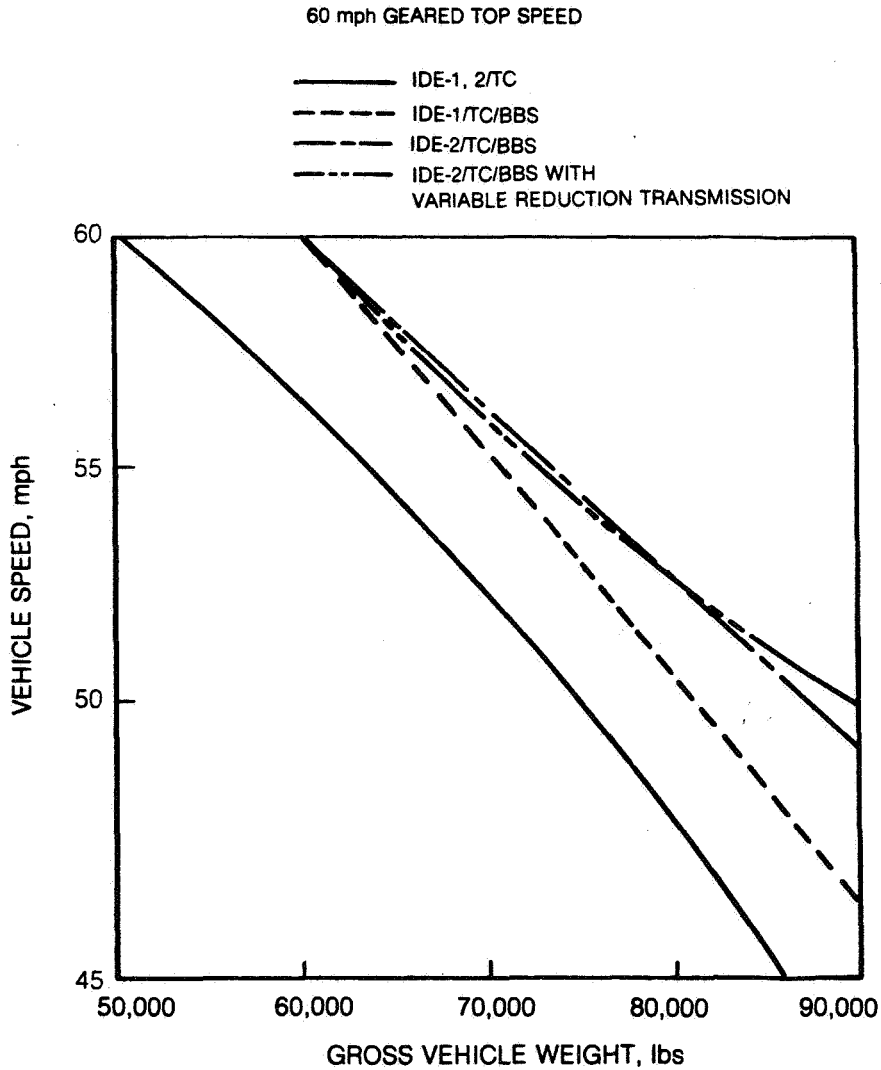
73,000 lb GWW

60 mph GEARED TOP SPEED

- IDE-1, 2/TC
- - - IDE-1/TC/BBS
- · - IDE-2/TC/BBS
- · - IDE-2/TC/BBS WITH VARIABLE REDUCTION TRANSMISSION



### MAXIMUM SPEED ON ONE PERCENT GRADE





unrealistic driving cycle be chosen, not only will the fuel economy figure be incorrect, but also any comparison made between engines will be misleading.

In general, there is an increase in fuel economy over the baseline engine for each of the BBS-equipped engines. The magnitude of this difference is on the order of 10 percent. This difference is also reflected in annual fuel savings for the BBS-equipped engines compared to the baseline engine. Typical annual savings in fuel are about 1600 gallons.

The effect of engine scaling was examined and shown to have a very slight effect on overall fuel economy because of the driving cycles employed and nature of the BBS engine performance maps. The value of the BBS variable-reduction transmission was also shown to have a negligible effect on fuel economy. The use of this more costly transmission cannot be justified on the basis of fuel savings under typical driving conditions.

## 7. LIFE-CYCLE COST COMPARISON

The purpose of this chapter is to provide a comparative assessment of the life-cycle costs and economics of BBS for insulated diesel engine applications in long-haul trucks. The data presented in this chapter are based on the performance and cost data given in previous chapters and on performance and incremental cost data for baseline insulated engines provided by NASA-LeRC.

### 7.1 Assumptions and Methodology

The assumptions and methodology employed in the economic analysis are based on a set of guidelines provided by NASA-LeRC. The real escalation rate applicable to the price of diesel fuel is based on recent projections of fuel prices by Data Resources, Inc. (Ref. 7.1), which predict a 2-3% fuel price real escalation rate for the 1990's. Table 7.1 contains a summary of the basic assumptions employed for the economic evaluation of BBS-equipped engines.

A discounted cash-flow methodology was used for evaluating the economic justifiability of the BBS as an addition to adiabatic diesel engines. In this methodology, an alternate system is deemed economically justifiable with respect to a baseline less costly (but less efficient) reference system if the present value of the life-cycle costs of the former is lower than that of the latter. The life cycle costs for truck power plants include capital costs, fuel costs, operation and maintenance expenses, taxes and other miscellaneous expenses. Other economic figures of merit that were used include the simple payback period, which must be less than a maximum acceptable value, and the annual fuel savings per dollar of additional capital cost, which must exceed a minimum acceptable value. The assumptions given in Table 7.1 imply a maximum acceptable payback period of 3.62 years and a minimum acceptable annual fuel savings of 0.23 gal per dollar of incremental capital cost of installed equipment.

The fuel savings used in the economic analysis were based on the design-point performance of BBS-equipped NASA engines to enable NASA to compare the BBS with other bottoming systems on a competitive footing. As recommended by NASA, annual fuel consumption scales in direct proportion to the design-point sfc, assuming that the average fuel mileage of the truck equipped with the TC engine, is 5.66 mpg. It should be emphasized, however, that fuel savings computed from simulations of more realistic drive cycles are likely to be about one-third lower than those computed on the basis of design point performance as was shown in Chapter 6. The economic figures of merit of the bottoming system must be within a comfortable margin of the limiting acceptable values to account for this discrepancy. For this reason, the scaled power plants, which would be as fuel efficient but less expensive than their unscaled counterparts, are much more likely to prove economically justifiable.

Two sets of economic comparisons were performed: (1) BBS-equipped TC engine versus the most efficient of NASA engines, namely, the TCPD/A engine; and (2) BBS-equipped engines versus the same engines without the BBS. In addition, a study of the sensitivity of the results of the first comparison to fuel price and equipment life was also performed.

TABLE 7.1

ECONOMIC COMPARISON ASSUMPTIONS

Real Cost of Money <sup>*</sup>	12%
Tax Rate	46%
Investment Tax Credit	7%
Diesel Fuel Price (1983) \$/US Gal.	1.2
Fuel Price Real Escalation (Per Annum)	2%
Annual Truck Mileage	100,000
Hardware Useful Life (Years)	7
Annual O & M (% of Capital Cost)	5%
Depreciation Method	Straight Line-5 Years

---

\* To be used also as the real after-tax minimum acceptable rate of return.

## 7.2 Comparison of TC/BBS and TCPD/A Power Plants

The annual fuel savings of the BBS-equipped TC engine relative to the TCPD/A engine were estimated and combined with the estimated incremental installed cost of the BBS (\$170/Bhp) and its annual O&M expenses (~5% of installed cost) to yield the present value of the life-cycle cost savings and the other economic figures of merit. These calculations were carried out both for the unscaled power plants and for scaled versions of these power plants. The reciprocating engine capital cost was scaled down on the basis of an incremental cost of \$30/Bhp. Table 7.2 presents a comparison of these two power plants in their scaled and unscaled versions. The incremental cost of turbocompounding (\$2000) and aftercooling (\$500) used in generating the data in this table were provided by NASA.

Table 7.2 shows that the unscaled TC/BBS power plant, which has a simple payback period exceeding 3.6 years, cannot be economically justified as an alternative to turbocompounding of aftercooled engines (unscaled TCPD/A engine). The scaled TC/BBS power plant shows a marginal disadvantage relative to the scaled TCPD/A engine, i.e., the fuel savings is not sufficient to justify the additional investment for the bottoming system, particularly since fuel savings under realistic driving conditions would be about one-third less than those reported in Table 7.2 (see Chapter 6). A BBS cost reduction amounting to 35-40 percent would be necessary to make the BBS economically justifiable as a substitute for turbocompounding.

## 7.3 Comparison of BBS-Equipped Engines with Baseline Engines

The preceding comparison of the TC/BBS and TCPD/A power plants stipulates implicitly that the bottoming and turbocompounding systems are mutually exclusive. It is evident, however, that a bottoming system such as the pressurized BBS can be added to any of the four NASA engines (TC, TC/A, TCPD and TCPD/A), forming the four compound power plants TC/BBS, TC/A/BBS, TCPD/BBS and TCPD/A/BBS. In fact, it is reasonable to regard the BBS as an extension of the turbocompounding principle (see Chapter 2 and Fig. 2.1 for a discussion of the similarities of BBS and turbocompounding).

Table 7.3 presents a summary of the performance improvements that can be obtained by adding a pressurized BBS to each of the four NASA engines and the corresponding incremental cost. The BBS outputs shown in Table 7.3 are approximate unscaled values obtained from the parametric data of Chapter 4, and incorporating some of the data generated in Chapter 5 for the optimum design of a BBS system matched to the IDE-2/TC engine. The fuel savings were computed on the basis of design-point B<sub>sf</sub>c improvements and the incremental installed capital costs were estimated on the basis of a constant specific cost of \$170/Bhp for all pressurized BBS.\*

Figure 7.1 contains a graphical comparison of the economic figure of merit [(gal/yr)/\$] for various power plant improvements in both scaled and unscaled versions. The first three bars show that the addition of

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\*For power ratings spanning a  $\pm 25\%$  range, the assumption of constant specific cost yields cost increments that are within only  $\mp 7\%$  of the values predicted by a 0.7 power law.

TABLE 7.2

## COMPARISON OF TC/BBS AND TCPD/A POWER PLANTS

(TC/BBS-TCPD/A)	Scaled	Unscaled
Incremental Output, Bhp	0	15
Bsfc Improvement, %	4.4	4.4
Annual Fuel Savings, gal/yr	729	729
Incremental Cost (1983\$)	3350	3960
Simple Payback Period, <sup>*</sup> yr	3.8	4.5

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\* Maximum acceptable value = 3.6 yr.

TABLE 7.3

COMPARISON OF BBS FUEL SAVINGS AND INCREMENTAL  
COST RELATIVE TO NASA ENGINES  
(Unscaled Power Plants)

Power Plant	TC/BBS	TC/A/BBS	TCPD/BBS	TCPD/A/BBS
Base Engine	TC	TC/A	TCPD	TCPD/A
Base Engine BSFC, lb/hp,hr	0.315	0.310	0.297	0.293
Compound Power Plant BSFC, lb/hp,hr	0.280	0.284	0.273	0.272
Improvement, %	11.1	8.3	8.1	7.2
BBS Output, Bhp	38	29	30	26
Annual Fuel Savings, <sup>*</sup> gal/yr	1963	1458	1346	1178
BBS Installed Cost <sup>**</sup>	6460	4930	5100	4420

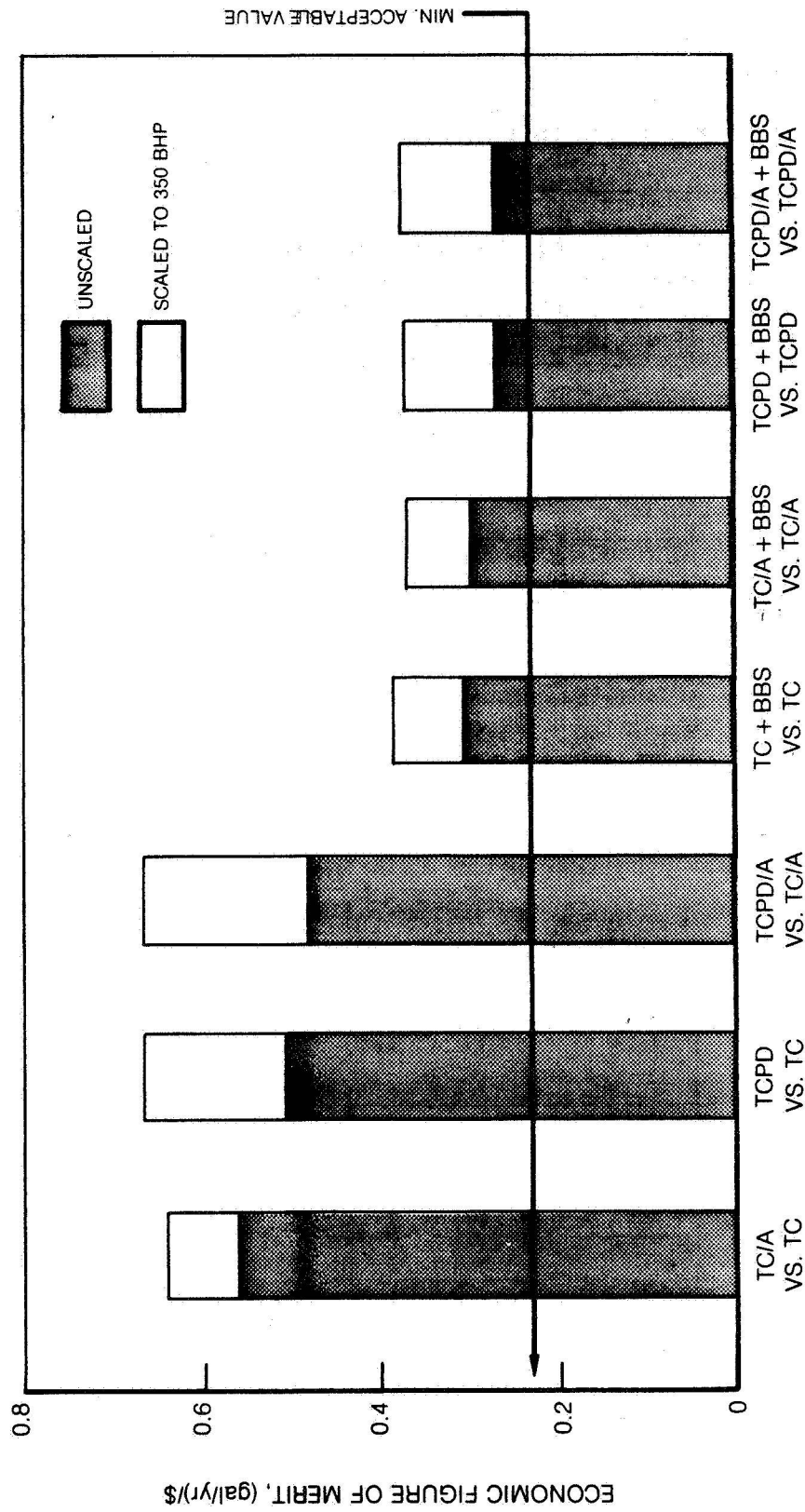
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\* Computed in accordance with NASA's recommendations.

\*\* Based on \$170/Bhp

FIG. 7.1

COMPARISON OF POWER PLANT ECONOMICS



aftercooling and turbocompounding to a baseline TC engine are highly desirable in both scaled and unscaled versions. The last four bars show that the addition of a BBS to any of the four NASA engines is economically justifiable in the scaled version but only marginally so in the unscaled version. In general, all alternatives appear to be economically justifiable relative to their respective baseline engines because the economic figure of merit exceeds the minimum acceptable value of 0.23 (gal/yr)/\$ in all cases.

Care must be exercised, however, in comparing the heights of the various bars; for example, it is erroneous to conclude that because the first bar is higher than the fourth, aftercooling is economically preferable to the addition of a BBS. In fact, the fifth bar indicates that the BBS is a desirable addition to an aftercooled engine (TC/A/BBS vs. TC/A).

In order to avoid this confusion, the standard method of economic comparison must be used. In this method, the alternatives are arranged in an ascending order of capital cost, and compared two at a time, starting with the least expensive option (baseline TC engine) and the next more expensive option. The superior of the two then becomes the baseline for the next pair to be compared, and so on. Here, superiority is measured by an incremental figure of merit exceeding the minimum acceptable value, a simple payback period shorter than the maximum acceptable value or, equivalently, a real after-tax rate of return exceeding the minimum hurdle rate.

The application of this method to the various power plant options considered here provides the appropriate economic ranking as shown in Table 7.4. The ranking in this table proceeds from less costly to more costly options; the payback periods and the economic figure of merits apply to the incremental investment between one option and the preceding one. Options that are not economically justifiable with respect to less expensive ones were omitted.

It is evident from Table 7.4 that the most successful application of the BBS concept will be to add it to a scaled TCPD/A engine. Such a combination may even be more economical than indicated in Table 7.4 if the gearboxes of the turbocompounding and bottoming systems are combined into one unit and if the design of the BBS were optimized for the exhaust characteristics of the TCPD/A engine.

#### 7.4 Sensitivity Analysis

All the preceding economic analyses were based on the assumptions given in Table 7.1. Because future prices of petroleum fuels are quite uncertain, the sensitivity of the economic comparison results to fuel prices was estimated for the case of the unscaled TC/BBS and TCPD/A power plants with 7 and 10 years equipment life. Figure 7.2 shows the present value of the life-cycle cost savings of these two power plants relative to the baseline TC engine as a function of the fuel price.

It can be seen that for a 7-year equipment life, the unscaled BBS-equipped TC engine would become economically superior to the TCPD/A engine only if fuel prices exceed \$1.4/gal (1983\$). With a 10-year equipment life, however, the unscaled BBS-equipped TC engine would compete with the TCPD/A engine at fuel prices in the vicinity of \$1.2/gal (1983\$) which are very close to diesel fuel prices at present. The corresponding results for the



TABLE 7.4

## ECONOMIC RANKING OF SYSTEMS

<u>Rank</u>	<u>Power Plant</u>	<u>Payback*</u> <u>Period</u> <u>[Years]</u>	<u>Economic Figure of Merit**</u> <u>[(gal/yr)/\$]</u>
a) <u>Unscaled</u>			
1	TC (Baseline)	--	--
2	TC/A	1.48	0.56
3	TCPD	1.71	0.49
4	TCPD/A	1.86	0.45
5	TCPD/A/BBS	3.13	0.27
b) <u>Scaled to 300 Bhp</u>			
1	TC (Baseline)	--	--
2	TC/A	1.13	0.74
3	TCPD	1.06	0.78
4	TCPD/A	1.12	0.75
5	TCPD/A/BBS	2.01	0.41
c) <u>Scaled to 350 Bhp</u>			
1	TC (Baseline)	--	--
2	TC/A	1.32	0.63
3	TCPD	1.24	0.67
4	TCPD/A	1.30	0.64
5	TCPD/A/BBS	2.35	0.35

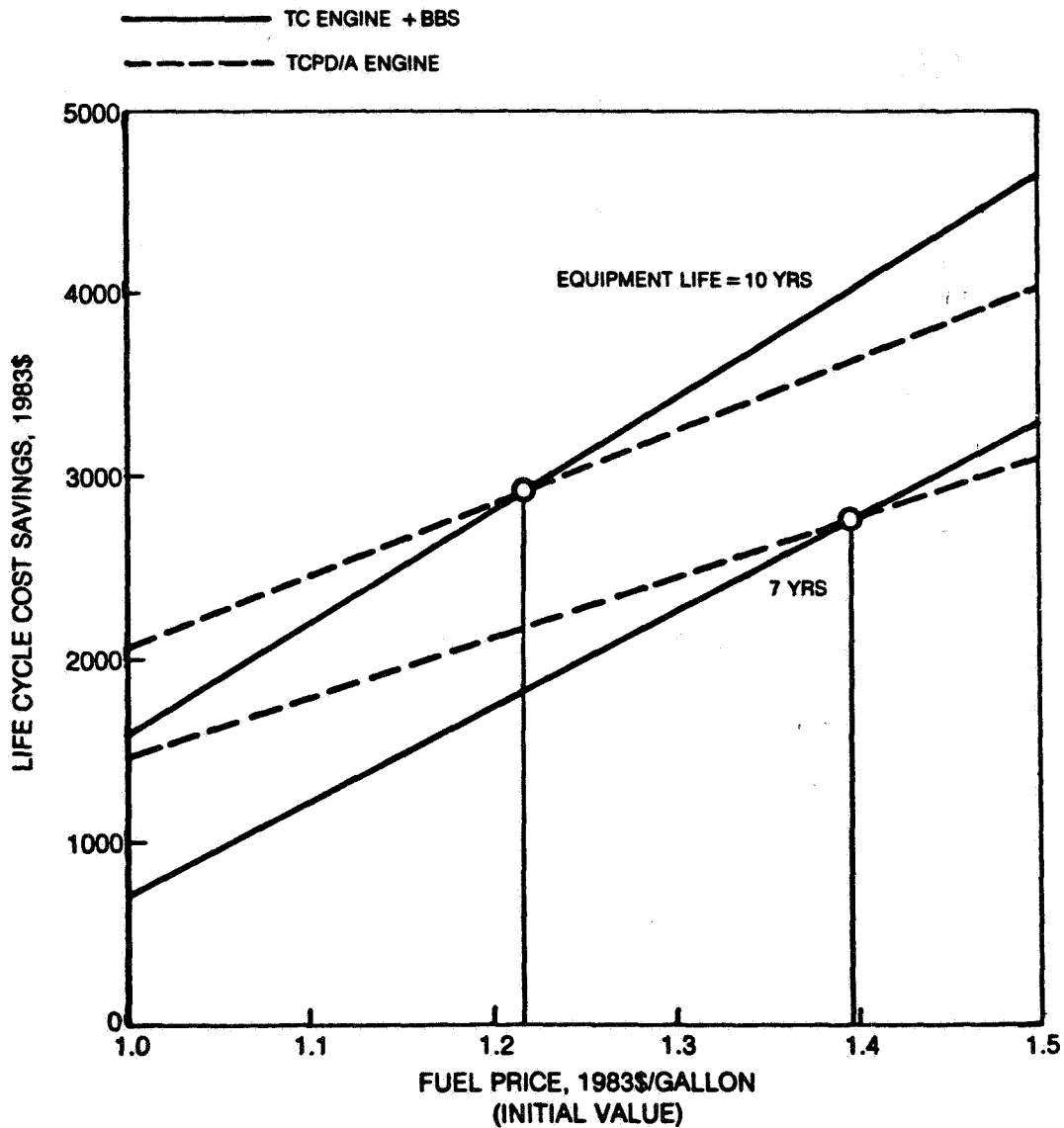
\* Maximum acceptable value = 3.62 years

\*\* Minimum acceptable value = 0.23 (gal/yr)/\$

scaled power plants would place the intersection point for the 7-year life slightly below \$1.2/gal, as could be inferred from the data given in Table 7.2. The corresponding value for the 10-year life would be lower than \$1.0/gal.

**EFFECT OF FUEL PRICE AND EQUIPMENT LIFE ON LIFE CYCLE COST SAVINGS FOR TURBOCOMPOUND AND BRAYTON BOTTOMING SYSTEMS**

(BASELINE: TC ENGINE)  
 REAL DISCOUNT RATE = 12% ; REAL FUEL PRICE ESCAL. = 2%



## 8. DISCUSSION AND CONCLUSIONS

### 8.1 General Discussion

It has been shown that Brayton bottoming systems, particularly the pressurized BBS, can be easily integrated with adiabatic diesel engines, effecting significant fuel savings over typical long-haul truck drive cycles. Brayton bottoming systems represent a logical extension of the well-known turbocompounding concept and can be used either instead of, or in tandem with it. The BBS is relatively simple in construction and is free from the complexity associated with the use of organic or high pressure working fluids. Therefore it is quite likely that the unit costs (\$/Bhp) will be considerably lower than that of Rankine cycle systems. This has been shown to be true in a wide variety of commercial and developmental Brayton and Rankine cycle power plants (e.g., gas turbine vs. steam turbine power plants and automotive gas turbines vs. automotive steam engines).

The fuel savings attributed to the BBS would be sufficient to justify the additional cost of the BBS relative to the baseline engine if the technical and economic uncertainties associated with the adiabatic engine/BBS power plant could be reduced to an acceptable level. The BBS approach appears, at best, to be marginally justifiable as an alternative to the less risky turbocompounding approach. However, the BBS could prove justifiable as an addition to a turbocompounded adiabatic diesel engine, provided also that technical and economic uncertainties are reduced to an acceptable level.

### 8.2 Technical and Economic Barriers

Technical and economic uncertainties constitute the major barriers that may hinder the development of BBS or other bottoming systems for adiabatic diesel engine applications. The major technical and economic uncertainties are those associated with:

- . Adiabatic diesel engine performance, exhaust conditions, and particulate emissions
- . Primary heat exchanger performance, corrosion and fouling in diesel exhaust service
- . Turbomachinery efficiency, reliability and cost
- . Future fuel prices
- . Market size

Furthermore, the economic worth of power plant performance improvements through the use of bottoming systems may diminish significantly owing to the commercial introduction of simpler methods for improving the fuel economy of heavy duty trucks. These methods include drag reduction, more efficient engines, better drive-trains, etc. The advent of these methods would increase the relative economic uncertainty associated with the use of bottoming systems and could increase the traditional reluctance of the trucking industry to the introduction of devices that are perceived to be untraditional.

### 8.3 Recommendations

The results and discussions presented in this report suggest that the development of a BBS is likely to be much less problematic than the development of a commercially mature adiabatic diesel engine that would be acceptable to the trucking industry. Therefore, a bottoming system major development effort, particularly prototype fabrication and testing, should be paced in accordance with the development schedule of the engine itself. Several aspects of bottoming system developments could be investigated in parallel with adiabatic engine development. These include both component- and system-related efforts, viz.:

#### (1) Turbomachines

- . Development of small, high performance turbomachines (gas turbines and compressors) that can be easily integrated with adiabatic engines as BBS turbomachines, turbocompounding turbines or turbochargers.
- . Investigation of novel methods for integrating these turbomachines with the engine to achieve synergism, e.g., configurations that allow the integration of turbocompounding turbines with BBS turbines; use of mechanically driven supercharger in turbocompound or BBS-equipped engines to achieve more favorable torque-speed characteristics, etc.

#### (2) Heat Exchangers

- . Detailed investigation of fouling and corrosion abatement in adiabatic diesel exhaust heat exchangers.\* This effort would be particularly beneficial to subatmospheric Brayton and Rankine cycle development.
- . A study of the transient thermal behavior of heat exchangers used in bottoming systems.

#### (3) Systems

- . Preliminary assessment of the feasibility of using a BBS in tandem with a highly insulated turbocompounded engine. This effort could also include the investigation of integrating the turbomachines and gearboxes for the turbocompounding system and the BBS.

Finally, it should be emphasized that an assessment of the relative economic merit of various bottoming system options should be based on a careful evaluation of the fuel savings potential of the various options under realistic truck operating conditions (typical drive cycles) and a consistent estimation of the capital and O&M costs of each option. This is necessary to ensure that the cumulative uncertainties of over- and under-estimation of the performance and cost of the various options do not render the results of the comparison meaningless.

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\*This is a problem that has been and is still under investigation.

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16. Abstract  This report presents an evaluation of Brayton Bottoming Systems (BBS) as waste heat recovery devices for future adiabatic diesel engines in heavy duty trucks. Parametric studies were performed to evaluate the influence of external and internal design parameters on BBS performance. Conceptual design and trade-off studies were undertaken to estimate the optimum configuration, size, and cost of major hardware components. The potential annual fuel savings of long-haul trucks equipped with BBS were estimated. The addition of a BBS to a turbocharged, nonaftercooled adiabatic engine would improve fuel economy by as much as 12%. In comparison with an aftercooled, turbocompound engine, the BBS-equipped turbocharged engine would offer a 4.4% fuel economy advantage. It is also shown that, if installed in tandem with an aftercooled turbocompound engine, the BBS could effect a 7.2% fuel economy improvement. The cost of a mass-produced 38 Bhp BBS is estimated at about \$6460 or \$170/Bhp. Technical and economic barriers that would hinder the commercial introduction of bottoming systems were identified. Related studies in the area of waste heat recovery from adiabatic diesel engines are NASA CR-168255 (Steam Rankine) and CR-168256 (Organic Rankine).					
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