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DOE/NASA/0361-1
NASA CR-180833
CTR 0723-87001

Technical and Economic Study of Stirling and Rankine Cycle Bottoming Systems for Heavy Truck Diesel Engines

I. Kubo
Cummins Engine Co., Inc.
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September 1987

Prepared for
NATIONAL AERONAUTICS AND SPACE ADMINISTRATION
Lewis Research Center
Cleveland, Ohio
Under Contract DEN 3-361

for

U.S. DEPARTMENT OF ENERGY
Conservation and Renewable Energy
Office of Vehicle and Engine R&D
Under Interagency Agreement DE-A101-86CS50162

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Printed in the United States of America

Available from
National Technical Information Service
U.S. Department of Commerce
5285 Port Royal Road
Springfield, VA 22161

NTIS price codes[1]
Printed copy: A07
Microfiche copy: A01

[1]Codes are used for pricing all publications. The code is determined by the number of pages in the publication. Information pertaining to the pricing codes can be found in the current issues of the following publications, which are generally available in most libraries: *Energy Research Abstracts (ERA); Government Reports Announcements* and Index (*GRA* and I); *Scientific and Technical Abstract Reports (STAR)*; and publication, NTIS-PR-360 available from NTIS at the above address.

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FORWARD

The author would like to thank the guidance as well as helpful inputs by Murray Bailey of NASA Lewis during the execution of the entire program.

Following companies participated in this program and their effort as subcontractors are recognized.

- Mechanical Technology Inc., D. Najewicz
- Adiabatics Incorporated, R. Kamo
- CMC Aktiebolag, S. Carlqvist
- Argonne National Laboratory, R. Sekar
  R. Cole

In addition, the author would like to acknowledge the cooperation by Foster-Miller, Inc. and Thermo Electron Corporation which performed conceptual designs of Rankine bottoming systems under a previous program for NASA Lewis. Their cooperation were essential for us to complete the cost estimates of both Rankine systems.

In Cummins, many people participated in the program. Following are members of the bottoming cycle study team:

W. Brighton - Design
J. Wagner - Engine Application/Service
R. Linney - Manufacturing Cost Estimates
D. Reese - Thermal Analysis/Design

CONTENTS

SUMMARY

The main objective of the program is to evaluate the concept of bottoming cycle to heavy duty transport diesel engine applications. For that objective, following sub-objectives are set under this program.

- Develop conceptual design and cost data for a Stirling bottoming cycle system,

- Life-cycle cost evaluations of three bottoming systems: Organic Rankine, Steam Rankine, and Stirling cycles.

- Suggest future directions in waste heat utilization research.

For the Stirling bottoming cycle, MTI, Albany, N.Y. completed the study under the ground rule that the system only utilized "state of the art" technology. A conceptual design of a system which would fit in a "cab-over" type truck was developed and manufacturing cost estimates for the system were performed.

In addition to the above work on the Stirling system, Adiabatics Inc. and Stig Carlqvist of CMC Aktiebolag studied a new Stirling system called "high temperature combined cycle". Results of the "high temperature combined cycle" indicated that its combined thermal efficiency could reach 51%, similar to the diesel engine with a Rankine bottoming cycle. However, the system requires extremely high temperature materials and lubricant which are well beyond the current technology. Furthermore, a regenerator is required to recycle exhaust-gas heat into the intake air. Therefore, the improvement is considered too small for the risk factors involved in developing the system.

The above Stirling system study, particularly the work by MTI, completed the conceptual design phase of the bottoming cycle evaluation based on the life-cycle cost analysis. For steam- and organic-Rankine systems, results of the studies made by Foster-Miller and Thermo Electron respectively under a previous DOE/NASA program were used. Variables considered are initial capital investments, fuel savings, depreciation tax benefits, salvage values, and service/maintenance costs. Fuel savings are based on the truck mileage improvements calculated with the Cummins VMS (Vehicle Mission Simulation) computer code. All bottoming systems are to be used with advanced "adiabatic" engines. Comparisons were made against a turbocompound engine, seriously considered as a way to improve fuel economy for heavy duty

truck applications.

The turbocompound/aftercooled (TCPD) engine would provide a 18 to 19% IRR (Internal Rate of Return) investment opportunity for truck owners. However, currently none of the three bottoming systems studied are even marginally attractive. Manufacturing costs of the systems have to be reduced by at least 65% in order to become competitive against the TCPD engine. A new innovative approach is required for any bottoming system to be applied for heavy duty truck engines.

As such a system, an integrated Rankine/Diesel system was proposed. The system utilizes one of diesel cylinders as an expander. This would eliminate the need for the power transmission devices required for all conventional bottoming systems. Control requirements would be less. Another aspect of the proposed system is the capitalization of in-cylinder heat loss which is quite substantial for the "adiabatic" engine. The concept reduces the size of the exhaust evaporator.

Conceptual design of the system and a rough economic evaluation were made. Results indicate the system has a potential to become an attractive package for end-users, giving approximately a 20% IRR at the fuel cost of $1.25/gallon.

The study was intended for a rough evaluation of the concept and optimization of the system was not performed. Further optimization is possible by eliminating/combining some of the concepts built in the current design.

I. INTRODUCTION

The main objective of this waste heat utilization study is to evaluate the economic feasibility of the bottoming cycle concept to heavy duty truck engine applications. There are several bottoming cycle candidates which provide a good fuel economy improvement and should be evaluated under the study. Currently, Rankine, Brayton, and Stirling cycles fit the qualification.

In 1985, M. M. Bailey of NASA Lewis reported a comparative evaluation of three alternative bottoming power cycles (ref. 1). Alternatives studied were steam Rankine, organic Rankine, and an air Brayton cycles. The study was made under the following ground rules:

- Base engine is an "Adiabatic" turbocharged diesel.
- Engine output is 350 HP.
- Concepts for bottoming cycle systems only use "state of the art" technology (1985 - 1987 time period).

Results indicated that the Rankine cycles were substantially better than the Brayton cycle in terms of the payback to truck owners. The Stirling cycle was not included due to the lack of a conceptual design of the system at that time.

Under this program of which the major object is to complete the study initiated by Bailey, following tasks were set:

1. Development of conceptual design and cost data for diesel/stirling system,

2. Life-cycle cost evaluation of three bottoming systems; Organic Rankine, Steam Rankine, and Stirling cycle.

3. Preliminary evaluation of a new integrated Diesel/Rankine system.

Under the task 1, two concepts were evaluated. One is to follow the ground rules outlined above. This concept would complete the Bailey`s comparison study. The other concept is to use much more advanced technologies including new materials and lubricants. Therefore it can not be compared to other bottoming systems being evaluated here. However, this study was included because it was believed that a new approach to the bottoming cycle concept would be needed to make it economically competitive against the turbocompound engine.

The task 3 was an addition to the original program. Based on the evaluation study of task 2, a new concept to integrate a diesel engine with a Rankine bottoming system emerged. Here, a preliminary conceptual design/analysis was made on the system.

II. DIESEL STIRLING SYSTEM STUDY

1. "STATE OF THE ART" TECHNOLOGY

This portion of the study was performed by Mechanical Technology Incorporated, Latham, N.Y. Details of the study is described in a MTI report "MTI 87SESD33" (Ref. 2).

1.1 SCREENING OF COMPOUND ENGINE CONFIGURATIONS

There are many types of mechanical drive arrangements for a Stirling engine such as crank/connecting rod, rhombic drive, wobble plate, and free piston/hydraulic converter. After a brief review of the arrangements based on complexity, durability, performance, size, and weight; three basic configurations are selected for further examination. They are a connecting rod with crosshead concept, which is represented by the RESD V-4 (MTI automotive Stirling engine), a double-acting, four-cylinder "V" configuration; the SAV-4, a single-acting, pressurized crankcase concept; and a FPSE/hydraulic converter concept.

1.1.1. RESD V-4 Engine Concept

A cross section of the RESD V-4 is shown in Figure 2-1. Coolers and regenerators in this engine are arranged in an annular configuration about the piston, thus minimizing the number of pressurized parts in the engine, particularly the heater head castings. The working gas is sealed from ambient pressure at the piston rod seal. The rod seal utilizes a type of sliding seal known as a pumping Lenningrader (PL) seal. Main bearings are oil-lubricated rolling element bearings, and the connecting rod/crosshead assembly is of conventional design. The engine is based on the technology developed under the Automotive Stirling Engine (ASE) program and is supported by eight years of testing on Mod I and P-40 Stirling engines.

It is the most compact engine of the three concepts, and has advantages of smooth torque output and good manufacturability. Disadvantages of the engine are associated with the life of the piston rings and sliding rod seals. Currently, the design life for these items is around 3,500 - 5,000 hours.

A parametric study on the performance of the engine was performed to understand the effect of Stirling exhaust temperature and engine speed. As shown in Figure 2-2, the power recovery maximizes in the exhaust temperature range of 700-900°F (i.e., heater head temperature of 600-800°F) for 2,000 rpm Stirling engine speed. Considering the fact that the size and weight of the engine increase with reducing exhaust gas temperature, the optimum temperature is determined at around 800°F. It would give a Stirling power recovery of 30-31 hp with a turbocharged engine as a base.

1.1.2. Single-Acting V-4 (SAV-4) Engine Concept

The concept uses single-acting pistons as shown in Figure 2-3. Two cylinders are used as compressors and the other two act as expanders. Therefore, during one revolution of the crank, two power strokes take place, unlike four power strokes for the double acting four cylinder engines. In an attempt to eliminate the use of sliding seals, this concept utilizes a pressurized crankcase with rotary oil lubricated seals for sealing the crankshaft. Piston rings seal between the cycle pressure variation and the cycle mean pressure in the crank case. Connecting ducts between the top of pistons and the bottom of other pistons used for the double-acting engine are eliminated. This would minimize the dead volume and improve the engine performance.

A dry-lubricated design is required between piston/ cylinder liner. Sealed grease-lubricated, rolling element bearings are used at both ends of connecting rods. The oil lubricated face seals are located outside of the engine and isolated from the crankcase by "lip seals" which do not experience any pressure differential. The advantages of the SAV-4 engine include the overall simplicity of the design, a heater head configuration well suited for the bottoming cycle application, and a seal location easily accessible for maintenance.

A performance analysis was made for the SAV-4 engine as well. Results are similar to the RESD engine, except an improvement in power recovery (10% v. 11.5% in BSFC improvement at the rated condition). The parametric performance curves are similar to the RESD results. However, the SAV-4 is significantly heavier than the RESD V-4, mainly due to the inherent nature of the single-acting v. double-acting concept.

There are several technological unknowns associated with this concept. The life of the rotary seal is not well understood as is the problem of hydrogen diffusion through the lubricating oil. The dry-lubricated piston concept is also a major source of uncertainty.

1.1.3. FPSE/Hydraulic Transmission

The engine is similar to that shown in Figure 2-4, except that no combustor is required and that the linear alternator in the figure would be replaced with a hydraulic transmission. The engine would use a tubular heater head with an annular regenerator and an cooler. The hydraulic transmission utilize metal bellows or a metal diaphragm to pump a hydraulic fluid. The high-pressure fluid then drives a hydraulic motor to produce a shaft power. The advantage of this approach is that the Stirling engine is essentially decoupled from the diesel operation, i.e., Stirling engine speed is independent from the diesel speed and its location is not restricted by the location of the diesel crankshaft. The engine is hermetically sealed and usage of any sliding or rotating seals are minimized. However, the use of the hydraulic transmission/motor significantly increases the complexity of the system since a hydraulic circuit would require valves, accumulators, and many more components.

Overall, the FPSE had several potential advantages with regard to long life and flexible operation; however, the state of development of the engine is several years behind that of kinematic Stirling engines, and the size required for this application, 25-30 KW, is clearly larger than any existing FPSE under development today.

Performance of the FPSE was analyzed and the results are shown in Figure 2-5. It was optimized at the engine speed of 2,500 rpm to be compatible with its hydraulic system and the heater head temperature was set at 700°F based on the MTI's previous experience. Efficiencies for hydraulic pump and motor were assumed to be 97 and 95% respectively. BSFC improvement of 8.58% was achieved with the system.

1.1.4. Summary of Concept Evaluations

In addition to the performance discussed in the previous section, maintenance and manufacturing costs for each system were estimated. The results are shown in Tables 2-1 through 2-3. The overall comparison of the three systems is summarized in Table 2-4 in terms of several factors as performance, maintenance and manufacturing costs, and size. It appears that the RESD

V-4 is the best choice for bottoming cycle applications based on technical maturity, performance, and packagiability.

1.2 Detailed Analysis/Conceptual Design

After the basic configuration was selected, the integration of the system with the diesel was evaluated more in detail. The space available for packaging the Stirling bottoming system was one of the first issues addressed. Heater head design optimization was performed next. The optimization included the comparison of single-stage v. double-stage arrangement as well as redesigning of the heater head to improve performance. A brief summary of the work is described below.

1.2.1 Engine Size

The available packaging space around the diesel in a heavy-duty truck is very limited. Using the "cab-over" truck configuration in mind, the space was determined and is shown in Figure 2-6. The RESD engine used for the preliminary screening study was found to be significantly larger than the space available for the system. The MTI Mod II engine being developed for the automotive application, on the other hand, is smaller than the RESD V-4 engine (approx. 30%) and fits in the space. Therefore, it was decided to use the Mod II ASE engine for this program. A benefit of development a Stirling bottoming cycle based on the ASE is that the use of the existing technology would minimize the cost of development as well as any technical barriers for the introduction of the engine.

1.2.2 Single-Stage v. Double-Stage Power Recovery

Performance was analyzed using both a single-stage operation and a double-stage operation over the Stirling engine speed range. The analysis involved detailed heat transfer calculations between the diesel exhaust gas and the heater head metal as well as that between the heater head metal and the working fluid. Heat transfer coefficients used for the analysis are based on data developed by MTI during rig testing of heater head segments (Ref. 3). Results are shown in Figure 2-7.

As shown in the Figure, performance difference is not significant for this particular size engine and operating conditions. Therefore, a single-stage operation was considered for this study. The net performance improvement of the Mod II size engine was found to be 7.25%.

1.2.3 Heater Head Redesign

In order to improve the performance of this bottoming system, following engine modifications were considered: simplification of the heater head design by incorporating a individually tubed heater head instead of the four manifold arms used in the ASE engine, and increase in the heater tube length.

Elimination of the complex heater heat castings would provide: 1. cost reduction, and 2. the cycle performance improvement due to a reduction in the dead volume. The effect of the increase in heater tube length is significant as shown in Figure 2-8. However, if the length is increased more than a twice of the current design, flow losses in the heater tubes would negate the improvement and output would be reduced. The new heater head design is shown in Figure 2-9.

Each heater head will utilize 30 U-shaped heater tubes arrayed around each heater head. The tubes are finned only on the rear row and the fin spacing is a 0.5 mm (0.022 in.). The detailed heat transfer in the heater head area is studied further by using a two-dimensional computer program for a single-row tube heat exchanger with plate fins (MTI's FIN2D program). The analysis indicates the tube configuration described above will provide a heat transfer capacity of well above 80 KW required at the design point for this application.

1.2.4 Other Engine Modifications

The drive system will be essentially the same as the Mod II except that the hydrogen compressor will not be required for this application, since the engine will operate at a constant pressure. However, because of the leakage associated with the static and sliding seals, a hydrogen makeup tank will be required. A four-liter hydrogen storage bottle will be used and be recharged every six months. A cross section of the cold engine drive system is shown in Figure 2-10.

The overall mounting of the Stirling engine, including the power coupling is shown in Figures 2-11 and 2-12. As shown, the flywheel housing of the diesel is moved aft by approximately 1.5 inches to add an intermediate spacer/ chain cover. The Stirling engine itself is mounted on two brackets, one of which is supported by the flywheel housing and the other by the transmission casing. The power coupling incorporates a simple direct chain drive with torsional isolators and an electric clutch. The chain is a commercially available 1.5-in. "silent" chain. Since the rated speed of the diesel is 1,900 rpm, the drive ratio is nearly 1 to 1 (20:19). The electric clutch will allow the Stirling engine to be disconnected either for heater head cleaning or during starting or extended idling of the diesel. It also allows disconnection of the system in the event of a system malfunction. The clutch would be actuated by a thermocouple on the Stirling heater head. Thus at temperatures below the Stirling self-sustaining point (~350°F), the Stirling would not create a parasitic loss on the diesel.

Control of the Stirling bottoming system is simpler than the Stirling engine control required for the independent system. The engine pressure will remain at a fixed level set by a pressure regulator on the hydrogen makeup tank. During a "down-throttle" of the diesel, a "short- circuit" valve on the Stirling would connect the engine cycles such that the pressure-waves in cycles 180° out of phase cancel, reducing the output power instantaneously.

The Stirling engine will reject approximately 53 KW of heat at the design point. Thus it requires a water pump and a radiator similar to a conventional automotive one. The pressure drop of the diesel exhaust gas through the system is estimated to be on the order of 10 to 12 inches of water.

1.2.5 Summary of Conceptual Design/Analysis

The Mod II design was chosen as a final configuration for this study. The system will fit in the envelop available in a cab-over truck. Performance of the system under various engine operating conditions are shown in Table 2-5. At the diesel engine rated condition, it improves the BSFC of 9% for a turbocharged engine. Figure 2-13 shows BSFC improvement v. diesel engine load for 1300- and 1900-rpm operations.

1.3 MANUFACTURING/MAINTENANCE COST ESTIMATES

1.3.1 Manufacturing cost

MTI developed cost estimates for the bottoming cycle system based on a production rate of 10,000 units per year. It utilized the estimate prepared by the Pioneer Engineering and Manufacturing Co. for the Mod II ASE engine. The original MTI estimate was $1,789 which represents a "matured" manufacturing costs (ref. 2). Cummins estimated the cost to be at $2,281 which represents a 70% of the estimated manufacturing costs based on the Mod II drawings (See Appendix 1) and is supposed to be the matured costs.

The difference between the two estimates are due to; 1. labor rates used, 2. the manufacturing method assumed to be used for the system, and 3. quality of components used for the system. Cummins used a labor rate close to a $70/hour, while MTI assumed a rate of $30 to $35/hour. As for the manufacturing method, Pioneer's estimate is based on the use of capital intensive processes. This may be justifiable, if the Mod II ASE engine is a reality. The Cummins estimate, on the other hand, is based on a much less capital intensive method of manufacturing. Since Pioneer's estimate is for the automotive application, the components are all automotive quality parts. However, items such as the radiator required for our application are quite different from the automotive application. Therefore, Cummins' estimate was based on components of industrial quality.

Since all the bottoming cycle systems studied under this program are evaluated by the Cummins approach and also the ASE engine is still quite uncertain, the estimate made by Cummins will be used for the economic comparison of those bottoming systems.

1.3.2 Maintenance Costs

MTI estimated service/maintenance costs based on scheduled periodic overhauls, yearly general maintenance, and operator capital costs. The scheduled periodic maintenance is associated with major engine overhaul that will be required at 5,000-hour intervals for seal/ring replacement, as well as replacement of bearings and other renewable items. During the 7-year period (14,000 hours), two engine overhauls will be required and its yearly cost was estimated to be at $135.63 per year.

The yearly general maintenance of the engines cover such items as oil change, cooling fluid replacement, hydrogen recharging and other general maintenance. It was estimated to be 5% of the retail engine cost per year and, based on the MTI price, is $179/year. (Based on the Cummins cost ($4,562), it is $228/year) The last item, operator's capital, represents the capital investment by fleet operators to install support equipment and to train personnel. It is based on a 25-vehicle fleet and a 5-year recovery of the capital investment. The result was $116/year.

Total service/maintenance cost became $431/year by MTI cost and $480/year with Cummins cost. Both figures, however, are based on the MTI assumptions. Cummins also made the estimate and obtained different numbers as shown in the later chapter of this report.

1.4 CONCLUSIONS FOR THE STIRLING SYSTEM WORK

A conceptual design for a Stirling bottoming cycle system has been performed. The engine meets the packaging requirements of the heavy duty truck application and provides a 9% fuel savings over a baseline turbocharged adiabatic engine. No serious technical barriers can be foreseen for the system, though it still requires a limited amount of development in the areas such as heater tube fouling/cleaning, development of long-life seals/piston rings, and confirmation of the control approach.

2. HIGH TEMPERATURE COMBINED DIESEL/STIRLING CYCLE

This portion of the study was performed by the following subcontractors:

i) Preliminary Analysis ------ CMC Aktiebolag

ii) Conceptual Design ------ CMC Aktiebolag

iii) Computer Cycle Analysis ------ Adiabatics Inc.

Details of the study are described in a final report by Adiabatic Inc. (ref. 4)

A schematic of the concept is shown in Figure 2-14. Since the high exhaust gas temperature is critical for the performance improvement of any bottoming systems, the concept utilizes a heat recirculation from exhaust gas to intake air so as to obtain a extremely high exhaust gas temperature. A T-S diagram of the cycle is shown in Figure 2-15.

2.1 Preliminary Analysis

A preliminary thermodynamic calculation was made by using the T-S diagram shown in Figure 2-15. In the figure, combustion process is divided into three parts, C1, C2, and C3 which represent constant volume, isothermal and decreasing temperature expansions respectively. Expansion process is also separated into three parts as isothermal, decreasing temperature, and adiabatic expansion processes. STH is the part where the heat is extracted into the Stirling bottoming system.

Some of the assumptions made for the study are:

- In-cylinder conduction heat loss is assumed to be 5-8% of total fuel energy.

- Stirling engine efficiency is 40%.

- Mechanical efficiency of the diesel engine is 90%.
- Turbocharger overall efficiency is assumed to be in a range of 65-66.6%.

- Combustion gas dissociation effect is not taken into the consideration. (Variations in Cp/Cv with temperature and air/fuel ratio are assumed to be linear.)

- Peak cylinder pressure limit was set at 2,500 psi.

The results of the calculation by S. Carlqvist are shown in Figure 2-16 and table 2-6. It indicates that the total system efficiency obtainable with the concept is 60.5%. However, as shown in the table, there is a large error in the heat balance between the heat input to the engine and the total heat out from the engine including the diesel shaft output and the heat to the stirling system (Total output is 24% higher than the heat input, 602.7 vs 747.4.). This is primarily due to wrong values used for Cp during the cycle calculation. Therefore, the final result is not considered to be accurate.

If, instead of increasing the heat input as done by S. Carlqvist, the output from the diesel was decreased in order to balance the energy, resulting thermal efficiency becomes 53.6%, much smaller than the 60% shown in the figure and closer to the computer simulation result described in the next section.

2.2 Conceptual Design

Conceptual designs at several levels of integration were proposed. Unlike the MTI Stirling design, there was no space limitations imposed for this study. Figures 2-17 and 2-18 show the layouts of semi-integrated systems, while Figure 2-19 depicts a fully-integrated system.

2.3 Computer Simulation

Performance evaluation of the TSA-cycle was made with a use of a diesel cycle simulator by Adiabatic Inc. In order to calibrate/evaluate the simulation program, calculations were made on a Cummins L-10 turbocharged aftercooled engine case with the simulator and results were compared against actual engine data obtained by Cummins. After the accuracy of the computation was assured, the TSA-cycle calculation was made. Results of the calculation are shown in Table 2-7 and Figure 2-20.

The overall efficiency of the TSA-cycle is 50.7%. Since the base turbocharged "adiabatic" engine has an efficiency of 43%, the fuel economy improvement with the TSA-cycle is 17 to 18%, slightly higher than the steam Rankine cycle which shows the 16% BSFC improvement. Combination of the TSA-cycle with a turbocompound engine was also evaluated. However, the thermal efficiency did not improve.

2.4 Conclusions on the TSA-cycle Study

As mentioned before, the fuel economy improvement with the system over the baseline engine was only slightly better than the steam Rankine cycle. However, there are many technical obstacles, such as high temperature materials/lubricants, to be overcome before this system can be produced. The material temperature is extremely high, even way above some of the ceramic material capabilities. Therefore, it was concluded that the system was not worth pursuing any further, unless much higher pressure capabilities can be established such that the efficiency improvement can become significantly higher than the conventional bottoming systems.

III. COMPARATIVE EVALUATION OF THREE BOTTOMING CYCLES

With the information on the Stirling bottoming cycle available, the task of comparing bottoming cycles was initiated. The cycles evaluated and subcontractors who performed the conceptual design/performance analysis for each system are as follows:

- Stirling cycle ... MTI
- Organic Rankine cycle ... TECO (Ref. 5)
- Steam Rankine cycle ... Foster-Miller (Ref. 6)

The analysis was made based on a life cycle costs/ benefits to end-users. Variables considered for the analysis are:

- Initial Capital Investment

- Future Incomes/Expenses

  . Income: Fuel Savings
  Depreciation Tax Benefits
  Salvage Value of the engine

  . Expenses: Service/Maintenance Costs

Operational/economic assumptions made for this analysis are tabulated in Table 3-1. As shown in the table, sensitivity analyses were made for different oil prices and base truck fuel mileage.

3.1 FUEL ECONOMY EVALUATION

Instead of using the steady state rated condition to represent the fuel economy improvement, the Cummins Vehicle Mission Simulation (VMS) program was used to evaluate the improvement in truck mileage with bottoming cycles. Three routes are considered for the study. Those are: Reno-Sacramento, Indianapolis-Chicago, and Columbus-Louisville-Cincinnati-Columbus. The first one represents the most hilly case and the second is one of the flattest route in this country. The last one is more or less a mixture of hills and flat routes and is considered as a standard route which represents a typical truck route in this country.

For obtaining the truck mileage improvement, following procedure was used:

- First, the VMS simulations were run one time for each of the three cases. The results would give percentages of time spent for different engine operating conditions for the three routes.

- Engine performance maps for different engine configurations were obtained by using the diesel cycle simulation program and by predicting power recovery through various bottoming systems for different exhaust gas conditions.

- The maps were divided into several areas such that the truck mileage would be estimated by combining the maps and the VMS results.

As shown in Table 3-2, the organic Rankine system gives the best improvement, while the Stirling system is the worst. The performance improvement is better on the hilly route than on the flat one. For our comparison study, the results with the "mix" route were used. The reason for the poor performance of the Stirling system is illustrated in Table 3-3. The system relies on the high heater-head temperature to obtain a high power conversion efficiency. Because of its inherent characteristic, however, high heater-head temperature means high temperature of the Stirling exhaust gas which, in turn, reduces the efficiency of the energy extraction from the diesel exhaust gas.

Figure 3-1 shows the comparison in performance for various bottoming systems.

3.2 MANUFACTURING COST ESTIMATES

Since actual hardware for the organic Rankine system were available at TECO from the previous DOE program, the manufacturing cost estimates were made on that system first. Observation of actual hardware and drawings were utilized for the evaluation and detailed results are shown in Appendix 2. For the steam system, the estimate was made by comparing components with the organic system. Most of components are similar between the two systems, except the organic cycle uses a turbine type of expander with a rated speed of approximately 20,000 rpm. The steam system uses a two-cylinder reciprocating expander with a rated speed of 2,000 rpm. Detailed study for the steam system is seen in Appendix 3.

Table 3-4 compares manufacturing costs for the three systems. Estimates by subcontractors were used for vapor generators of Rankine systems. The vapor generator of the organic system is more expensive than that of steam system due to mainly a by-pass mechanism required for the organic cycle to prevent over-heating of the fluid. The reciprocator type steam expander costs more than the turbine expander used for the organic system. In summary, organic system's cost is the highest at $4,938 and steam Rankine costs second at $4,199. The Stirling cycle system cost is the lowest of all at $3,258. The figure is quite different from the one made by MTI as mentioned in a previous section. Figures for Rankine systems presented here, however, turned out to be close to those made by subcontractors in 1983.

All the cost figures are estimated from the current designs for each system. Considering the effect of the "learning curve", the costs were reduced by 30% to project "matured" costs of the systems. Figures for the matured costs are shown in the parentheses. The "matured" costs were used for the economic analysis in Section 3.4.

Output levels differ among engines with the three different bottoming systems due to the difference in power recovery. Therefore a correction was made on the manufacturing costs for a same output level (@350 HP) based on the "0.7-power law" correlation developed by Bailey for his analysis.

Actual prices which end-users have to pay for the bottoming cycles include mark-up by engine makers and OEMs. The mark-up changes depending on many factors including the entire economic situations and type of engines. Therefore it is difficult to pinpoint a certain number for the mark-up value. Based on inputs from Cummins marketing area and also from a OEM, the total mark-up was set to be at 100%. Thus the manufacturing costs were multiplied by 2 to obtain final prices for the end-users.

Final price figures are listed in Table 3-5. As seen in the table, difference in price among the bottoming systems becomes much smaller by the constant output comparison.

3.3 SERVICE/MAINTENANCE COSTS

Two different sources were used to determine service/ maintenance costs, namely a Cummins 7-year maintenance/ repair contract and service data accumulated for routine servicing/maintenance costs. Based on the data, following estimates are made:

- Regular Semi-Annual Inspection & Service
  3% of Equipment Original Price/100,000 miles

- Other Services
  7% of Equipment Price/100,000 miles

- Variable Element of Maintenance/Service

(Major overhaul and turbo replacement, etc.)
50% of Equipment Price/500,000 miles

Therefore, the total annual service/maintenance cost was assumed to be a 20% of original equipment price.

3.4 ECONOMIC EVALUATION

Based on the various economic data generated above, a comparative evaluation of the systems was made by using IRR (Internal Rate of Return) as a measuring criterion.

A summary of the manufacturing and maintenance costs to be used for this study is shown in Table 3-6, along with the estimates made by each subcontractor for its respective bottoming cycle system.

Results are shown in Figures 3-2 through 3-5. In Figure 3-2, IRRs for bottoming systems including a turbocompound/aftercooled engine are shown with fuel price as a parameter. A core-engine for this figure is a turbocharged/aftercooled engine. As seen in the figure, IRRs for bottoming cycles are all less than 10% while that for the turbocompound engine shows a above 20% return with a fuel price of $1.00/gallon. The best system is the Steam Rankine system. But it, too, won't be attractive unless fuel price goes over $1.50/gallon level. The result is based on the truck fuel economy of 8 miles/gallon. As shown in Figure 3-4, the net present value of the steam Rankine system becomes positive at a above 15% cost of capital level if the base truck fuel economy is below 6 miles/gallon. The better the base truck fuel economy is, the harder it is for any bottoming cycle systems to be used commercially. Due to future improvement on the base truck designs, the 8 miles/gallon assumption for the fuel economy is reasonable.

Figures 3-3 and 3-5 show similar comparisons as Figures 3-2 and 3-4, with a turbocompound engine rather than a turbocharged engine as a base. They will show us a comparison of systems when we have an option to buy either a turbocompound engine or another engine with a bottoming cycle system. From the figures, it is clear that the turbocompound engine is well superior to bottoming cycle systems unless fuel prices start to soar above $1.75/gallon levels. And even at a $2.00/gallon level, only the steam Rankine system becomes attractive and Stirling and Organic Rankine systems won't be feasible.

As a final analysis, calculations were made to obtain a system price of each bottoming system which would be competitive against the turbocompound engine. The results are shown in Table 3-7. For the study, fuel economy improvement with each bottoming system is assumed to be the same as the current system. As shown in the table, more than 35% cost reduction is required for the steam system to be competitive to the turbocompound system. Other systems require much higher cost reductions.

3.5 CONCLUSIONS OF COMPARATIVE EVALUATION

Based on the above analysis, following conclusions are made:

- Using a life-cycle costs/benefits concept model, an economic evaluation was made on three bottoming cycle systems and on turbocompound system.

- Assuming that fuel economy improvement with turbocompound engine (TCPD) is 6% over the baseline turbocharged/aftercooled engine and fuel cost is $1.00/gallon, TCPD system would provide a above 20% IRR investment opportunity. However, the system still required more than two years of payback period.

- None of the three bottoming cycle systems are even marginally attractive, unless diesel fuel price becomes close to $1.75/gallon.

- Manufacturing costs for bottoming systems have to be reduced, at least 35%, in order for them to become competitive against the TCPD engine in terms of return in investment.

Above results indicate that bottoming cycle systems as they are designed now will not be economically feasible for a foreseeable future, unless a totally different approach is introduced to reduce its cost substantially. As a new approach of making a bottoming system attractive, an integrated Rankine/diesel system concept has emerged. The concept will be analyzed and evaluated in the next chapter.

IV. INTEGRATED DIESEL/RANKINE SYSTEM

A schematic of the new integrated diesel/rankine cycle is shown in Figure 4-1. It utilizes one fluid for engine cooling as well as for a Rankine cycle. Thus the need for an additional radiator is eliminated. In addition, the size of the vapor generator required in the exhaust system can be reduced due to the fact that the working fluid picks up heat energy through the engine cooling before it reaches the vapor generator. This is am important factor from the cost point, since the vapor generator cost is one of the major items of the total manufacturing cost.

In this chapter, a thermal analysis of the diesel/Rankine system is described. Based on the available heat, a selection of a working fluid and a Rankine cycle optimization were made. Finally, a conceptual design of the system was proposed and a economical evaluation of the system was made.

4.1 THERMAL ANALYSIS OF THE ENGINE SYSTEM

Figure 4-2 shows passages for the working fluid. As shown later, steam was selected as a working fluid for this analysis. A total of 17.84 lbs/min of water would circulates through the condensor/radiator. The flow rate for the Rankine cycle is approximately 6.0 lbs/min. Thus at the rated condition, the flow of 12 lbs/min would just circulate through the oil cooler and the radiator only. For this analysis, the steam pressure was assumed to be 1,000 psi. As shown later, the pressure can be lowered to 500 psi with only a slight loss in the cycle efficiency. Amount of heat to be collected by the working fluid through various part of the engine is shown below.

Oil cooler:	2304 Btu/Min
To - Recirculating Water	1652 Btu/Min
- Rankine Working Fluid	652 Btu/Min
Cylinder Head:	1130 Btu/Min
Exhaust Manifold:	357 Btu/Min
Exhaust Stack Boiler:	5250 Btu/Min
Total Heat to Working Fluid:	7389 Btu/Min

Due to the heat energy collected through the engine cooling, the heat transfer at the vapor generator is reduced by approximately 30%. This reduction would be reflected in the size of the vapor generator. Here the oil temperature going into the oil cooler was assumed to be 374'F. If a lubricant which has a higher temperature capability can be developed , then the reduction can be much higher.

Physical conditions of the steam at different points of the process are shown below:

	Temperature (°F)	Pressure (PSIA)
Radiator/Condensor Outlet	236	30
Water Pump Outlet	236	95
Oil Cooler Outlet	362	95
High Press. Feed Pump Outlet	362	1000
Cylinder Head Outlet	544	1000
Boiler Inlet (11.5% Steam)	544	1000
Boiler Outlet (100% Steam)	950	1000

4.2 FLUID SELECTION AND OPTIMIZATION OF RANKINE CYCLE

The task of selecting the working fluid for the system was granted to Argonne National Laboratory based on their past experience on the subject. Dr. R. Cole selected water and toluene as two candidates from several possible working fluid candidates. Table 4-1 shows characteristics of various organic rankine cycle fluids. Reasons for selecting the above two fluids are described in the attached Argonne report (Appendix 4).

Cycle Analysis/Optimization

Thermodynamic analysis was performed for the following four cases:

1. Steam at 1000 PSI pressure as a working fluid
2. Steam at 500 PSI pressure as a working fluid
3. Toluene at 500 PSI pressure as a working fluid, without regeneration.

4. Toluene at 500 PSI pressure as a working fluid, with regeneration.

Following assumptions were made in all the analyses:

- Exhaust gas flow rate: 50 lbs/min
- Exhaust gas inlet temp. to evaporator: 1100°F
- Expander and booster pump efficiency: 70%
- Expander outlet pressure: 30 psia.

A summary of the analyses is shown in Table 4-2. It is clear that the cycle with toluene requires regeneration in order for its efficient to be comparable to the steam cycle. Toluene has considerable amount of energy after expansion in the power cylinder. Addition of the regenerator would increase complex to the system as well as the manufacturing cost. Therefore, it was decided to select water as a prime candidate for the system. Detailed analyses are depicted in the Appendix 4. The thermodynamic (T-S) diagram for the selected cycle is shown in Figure 4-3. Figure 4-4 shows component contributions to the heat input for the integral steam bottoming cycle.

4.3 CONCEPTUAL DESIGN OF THE SYSTEM

4.3.1 Heat Exchanger Sizing

Argonne National Lab. was subcontracted for this task and a detailed discussion on this subject is given in the Appendix 4. Here, a brief summary is discussed.

Figure 4-5 shows the size of the evaporator as a function of the working fluid inlet temperature to the evaporator for three levels of the fin spacing. Also shown is the size of the evaporator used by Thermo Electron for their DOE demonstration program. According to a heat exchanger manufacturer whose products include heat exchangers for diesel engine exhaust gas, it is recommended that the fin spacing should be limited to 6 fins/inch in order to avoid fouling on heat transfer surfaces (Ref. 7). For this analysis, the fin spacing of 6 fins/inch is selected. Details of the tube and fin dimensions are given in Figure 4-6. Compared to the demonstration unit, the size is reduced approximately by 40%. However, it is still too large to be attached to the base engine.

Another heat exchanger required for this system is the condensor/radiator. Slightly superheated steam from the expander outlet mixes with the engine coolant at or just before the top tank of the radiator. The mixing will cause the steam to condense into partially saturated steam since the mass flow rate of the working fluid is 1/2 of the engine coolant. As shown in the previous section, a total heat rejection of this system is 9041 BTU/Min at a rated 350 HP condition. Since the heat rejection of current diesel engines with a output power of 350HP is around 9000 Btu/Min, the size of the radiator for this system will be similar to current radiators being used in industry.

4.3.2 Power Expander

For this system, it is proposed that one of cylinders of the base diesel engine be used as the power recovery device. The steam reciprocating engine concept has been studied by a few (Ref. 6 and 8), including Foster Miller which performed the study for the NASA/DOE program described in this report earlier. However, when the reciprocator is integrated into the diesel engine, several design issues such as vibration and valve events must be addressed. Since the objective of this study is to perform a preliminary feasibility study of the concept, those issues were addressed from that objective in mind. More detailed analyses should be performed in the next phase, if the concept proved to be attractive, or at least worth pursuing further, based on this study.

Following are brief comments on those issues:

- Expander Displacement:

As described in Table 4-3, a displacement of approximately 100 cubic inches is required. This would fit nicely with a cylinder of Cummins L10 engine which was used as a base for this study.

- Intake Valve:

It is essential for the efficiency of the expander that the intake valve opens rapidly to allow the working fluid into the cylinder quickly and have adequate time for expansion work. Therefore, a sliding valve is considered as a intake valve as shown in Figure 4-7. The exhaust valve event can be more gradual and so can be the same as that of the base diesel engine.

- Vibration/Balance:

A detailed vibration analysis is beyond the scope of this work. There are several options as to how the configuration of the engine should be made. One of the option is to use the firing order of regular 5-cylinder-engines for the first five cylinders (1-2-4-5-3-1) and the expander cylinder would be positioned at a 180 crank angle degree from the NO. 1 cylinder. This would require a counter-balancer at the sixth cylinder to take the first order unbalance out. Another option would be to have the crank shaft same as the six-cylinder engine with the uneven firing order. Balance due to the inertia forces should be good for this configuration. However, torque on the crank shaft due to gas pressure should be evaluated.

- Lubrication

The oil sump of the steam cycle is separated from the rest of the engine as shown in Figure 4-8. Therefore, a special lubricating oil (Steam engine oil) can be used for the lubrication of components in the sixth cylinder. Those oils are commercially available through Exxon or Mobil (Ref. 5). As a future technology, dry lubricated steam reciprocators should be developed.

4.3.3 Engine Design/Layout

A cross-section of the base diesel engine is shown in Figure 4-9 along with a detailed explanation of each components. And a schematic, depicting a layout of the engine mounted on a truck, is shown in Figure 4-10.

4.4 PERFORMANCE PREDICTION

Based on the thermal analysis described in the sections 4.1 and 4.2, a total engine performance was made and the results are presented in Table 4-4. The best performance is obtained with the turbocharged/aftercooled turbocompound engine plus integrated bottoming cycle and its BSFC can be as low as 0.260 lbs/hp-hr. However, the performance with the non-aftercooled/turbocharged turbocompound engine is also close to the above figure and shows a good fuel economy.

4.5 MANUFACTURING COST ESTIMATE

Manufacturing cost estimate of the new system was made by comparing the engine against the base turbocharged engine. The detailed analysis of the study is shown in Appendix 5 of this report. Summary of the cost increase is shown in Table 4-5. The table compares the cost figures to those of the conventional steam Rankine system studied under this program, i.e., Foster Miller system. It shows the cost is almost a half of the conventional system. Main contribution of the difference comes from the expander, power train, and the condensor. However, due to its added complexity to the base engine for extracting the additional heat from the cylinder head and exhaust manifold, the cost of vapor generator and engine modification together would cost more for the new system than the conventional design. In addition, the new system only generates a 305 HP total output (Turbocharged version) as compared to the baseline engine which generates an output of 317HP. Therefore, when the comparison is made at the 350 hp rating (Here, the 0.7-power law was used to convert the cost at the 350 hp level.), the premium for the new system is increased to $2,400.

This disadvantage of the lower output can be reduced by increasing the BMEP of the engine. The design point of this particular engine was at the BMEP level of 195 psi. As the output level of the base engine increases, the total heat available for the bottoming system increases. Therefore, the output of the expander would increase and the reduction of output power due to the usage of one of the six cylinders as a expander would decrease as well. This concept should be studied as a next step if the system is to be pursued further.

4.6 ECONOMIC EVALUATION

Assumptions regarding to the maintenance cost, operations, etc, made for the evaluation of first three bottoming cycles are also used for this study. Results of the economic analysis is summarized in Figure 4-11. As seen from the figure, the new integrated system would give a much better return on investment for customers compared to any other bottoming systems. However, it still is not as good as the turbocompound engine. When the fuel price exceeds $1.25/gallon range, the new system seems to give a rather attractive investment opportunity for the end users.

V. CONCLUSION

Following conclusions can be made from the entire study described in this report.

1. Bottoming cycles offer good opportunities for large fuel economy gains. However, traditional bottoming cycles are not competitive against turbocompound engines due to its complexity and thus high costs (The initial investment as well as maintenance cost.)

2. A new integrated Rankine/Diesel system was proposed. Based on the preliminary study of the system, it offers the best return on investment among bottoming cycles studied.

3. The new system would give a more than 20% Internal Rate of Return (IRR) at the fuel price of $1.25/gallon.

4. Therefore, further studies should be made on the new system including:

- further optimization of the concept by studying such areas as,
  - . optimize the amount of heat recovery for each components.
  - . optimum BMEP of the base engine
- hardware demonstration of the 1-5 steam/diesel concept with an "ultra low" heat rejection engine.

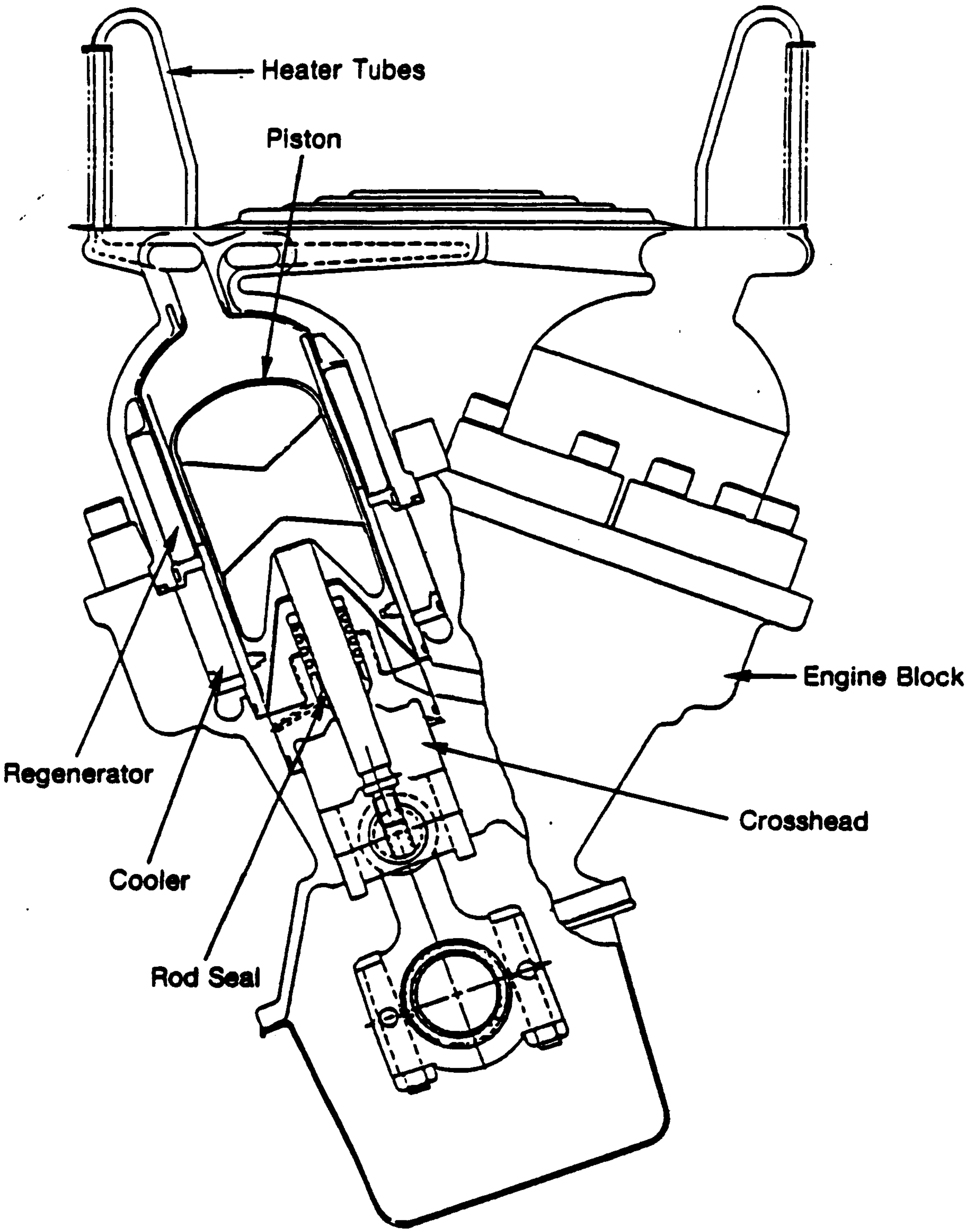


Figure 2-1. V-4 RESD (Front View)

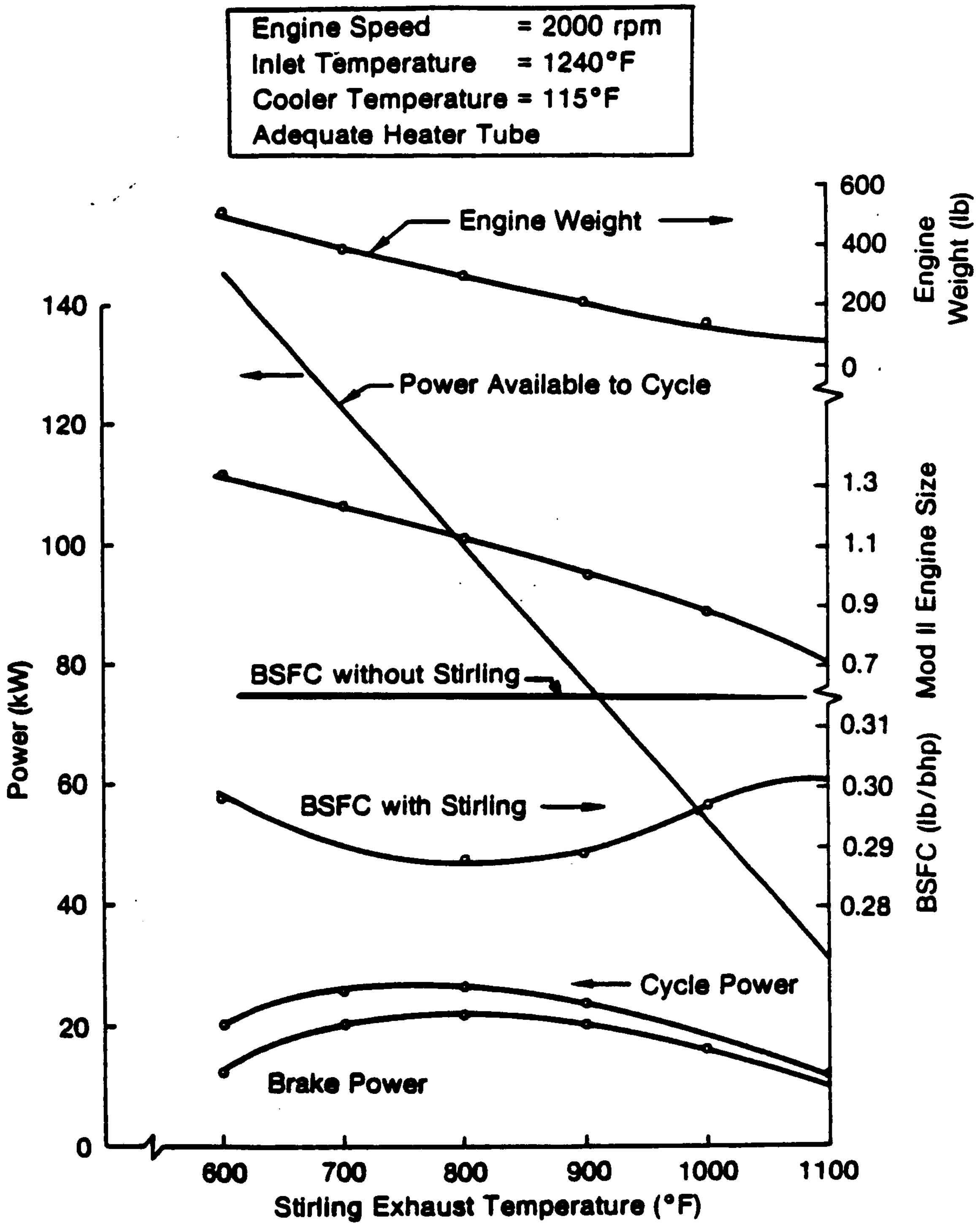


Figure 2-2. RESD V-4 Power Recovery as a Function of Heater Head Temperature (2000 rpm)

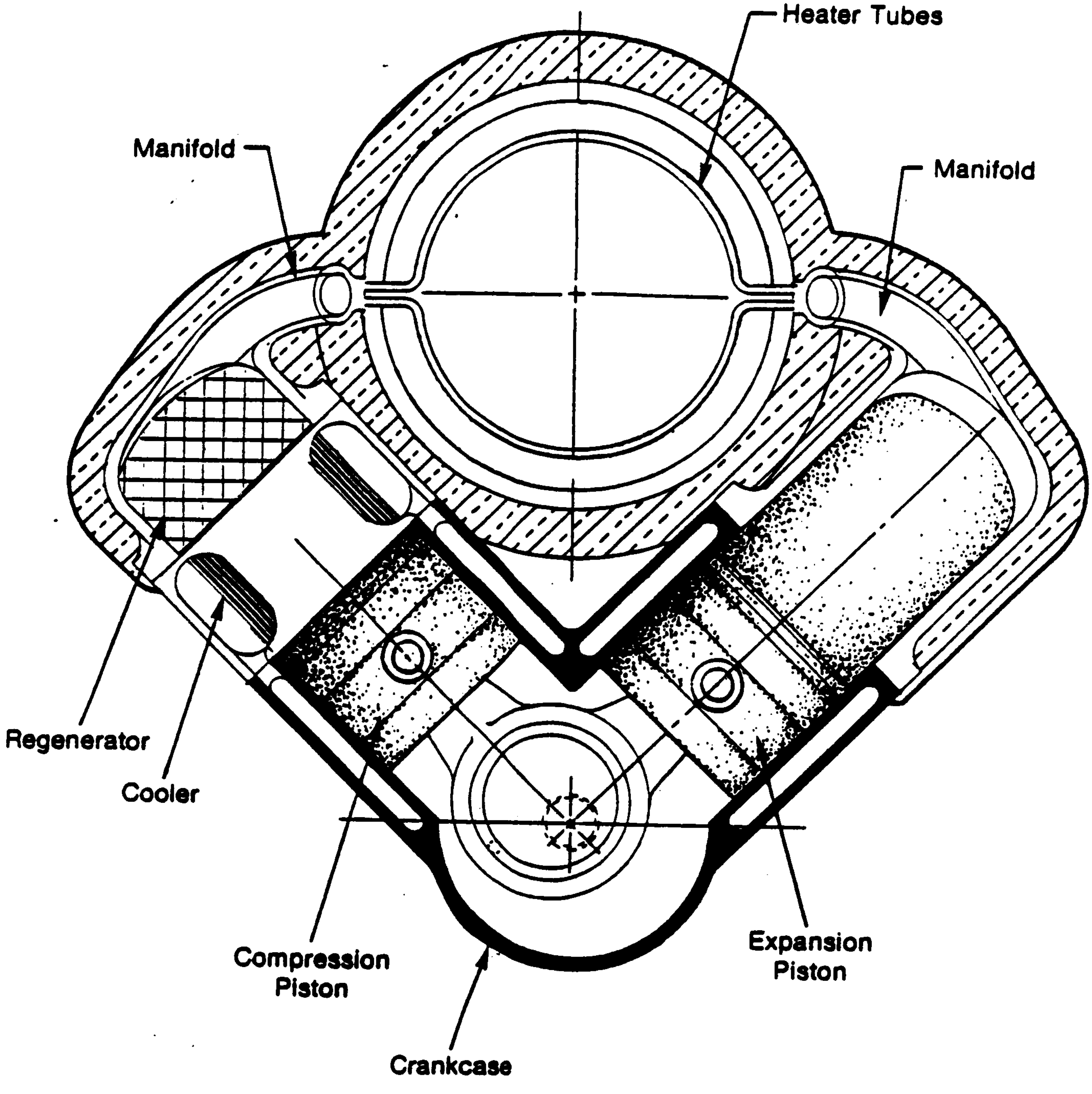


Figure 2-3. SAV-4 Concept

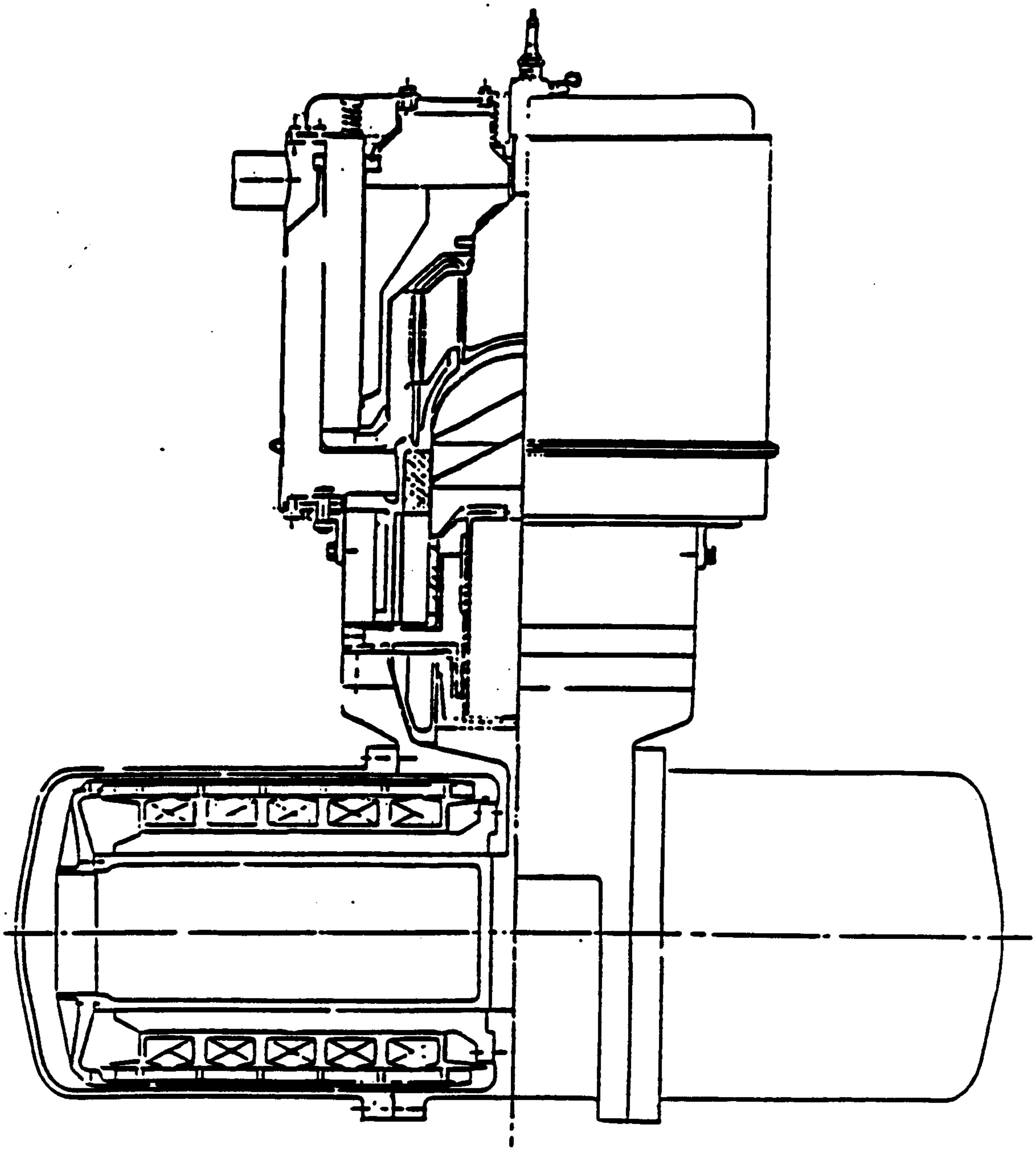


Figure 2-4. FPSE Concept

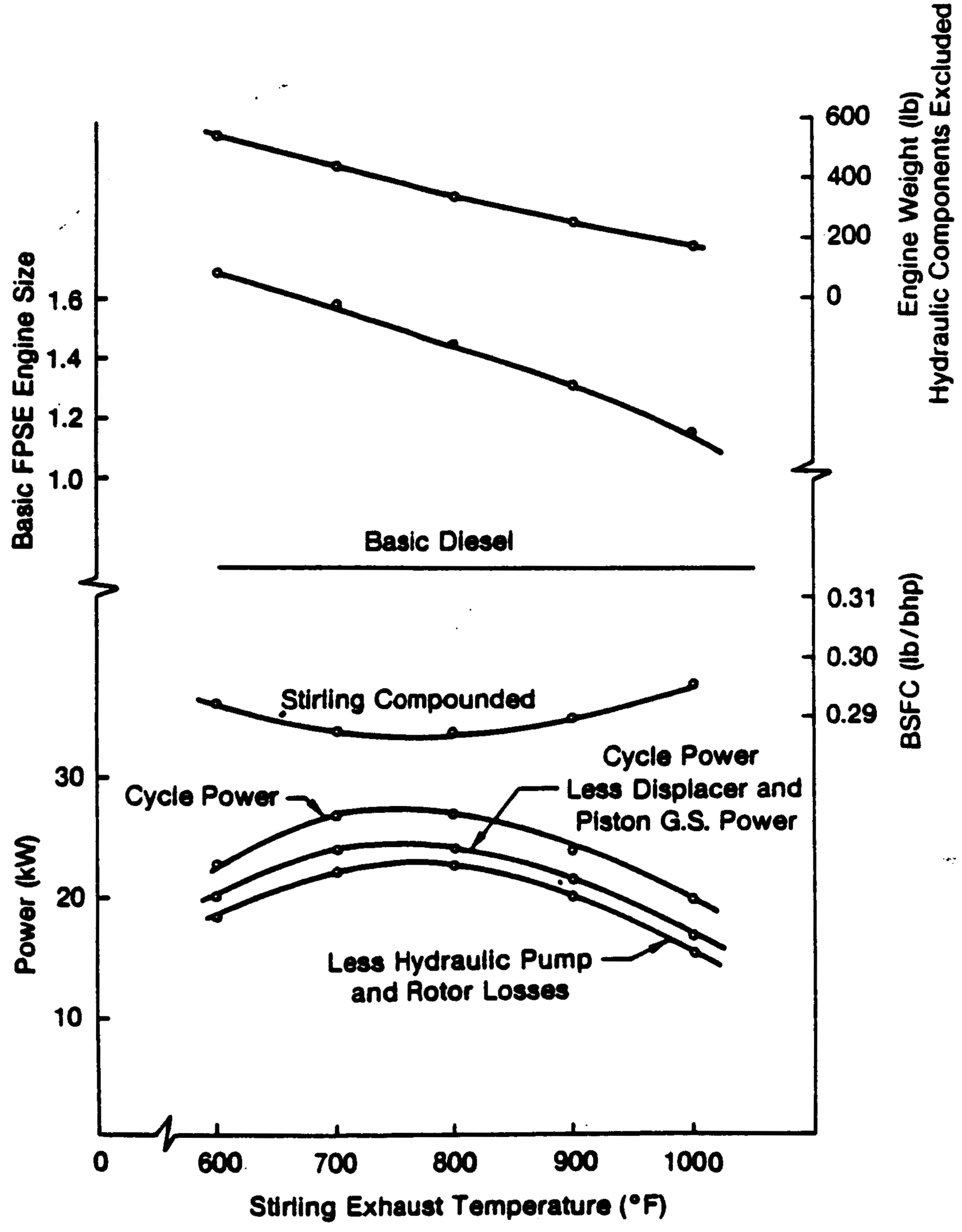


Figure 2-5. FPSE Optimized at 2500 rpm and 644 K Heater Tube Temperature

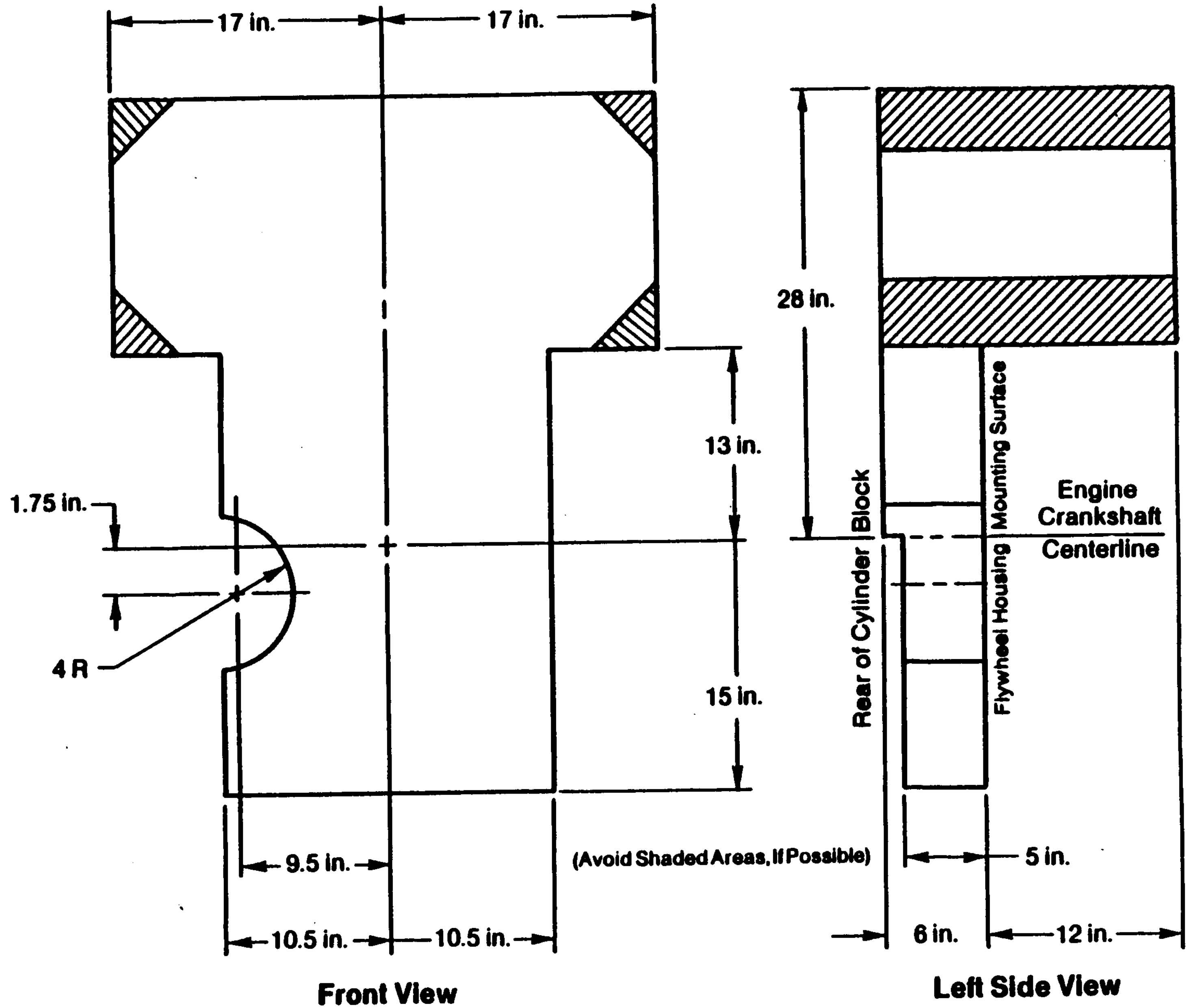


Figure 2-6. Bottoming Cycle Mounting and Drivetrain Envelope for COE Installation

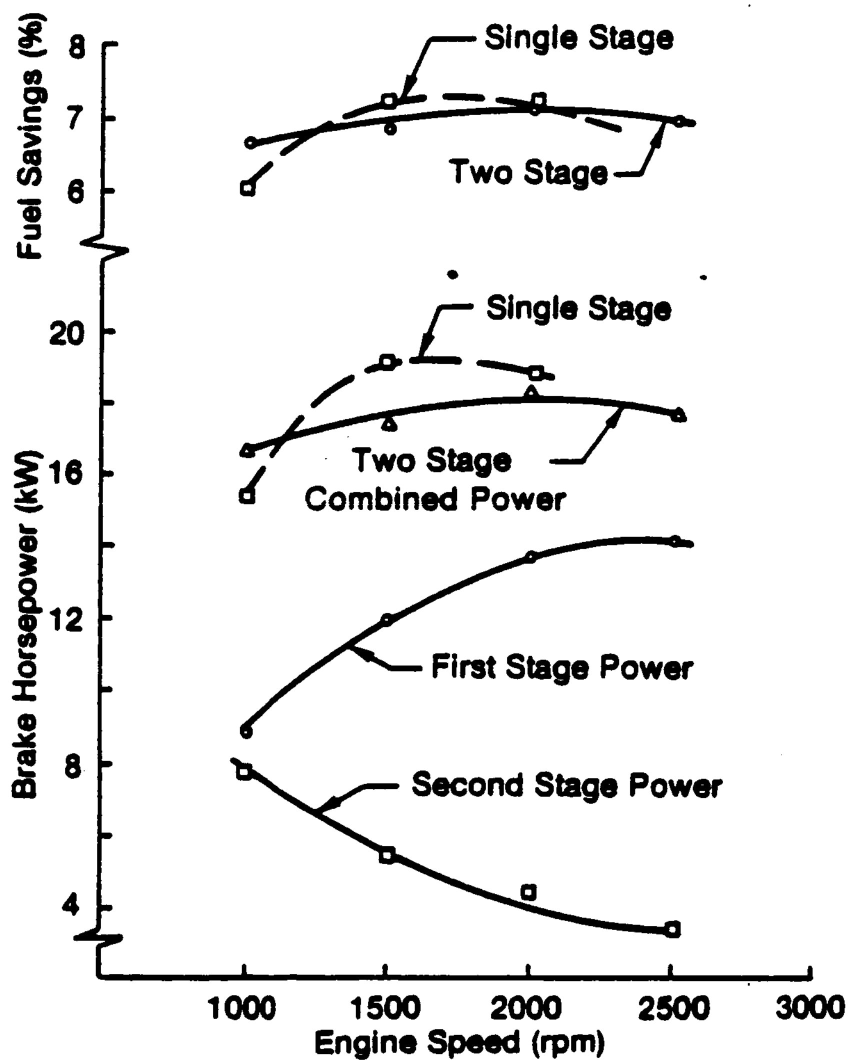


Figure 2-7. Comparison of Single- and Dual-Stage Power Recovery Based on Mod II Geometry

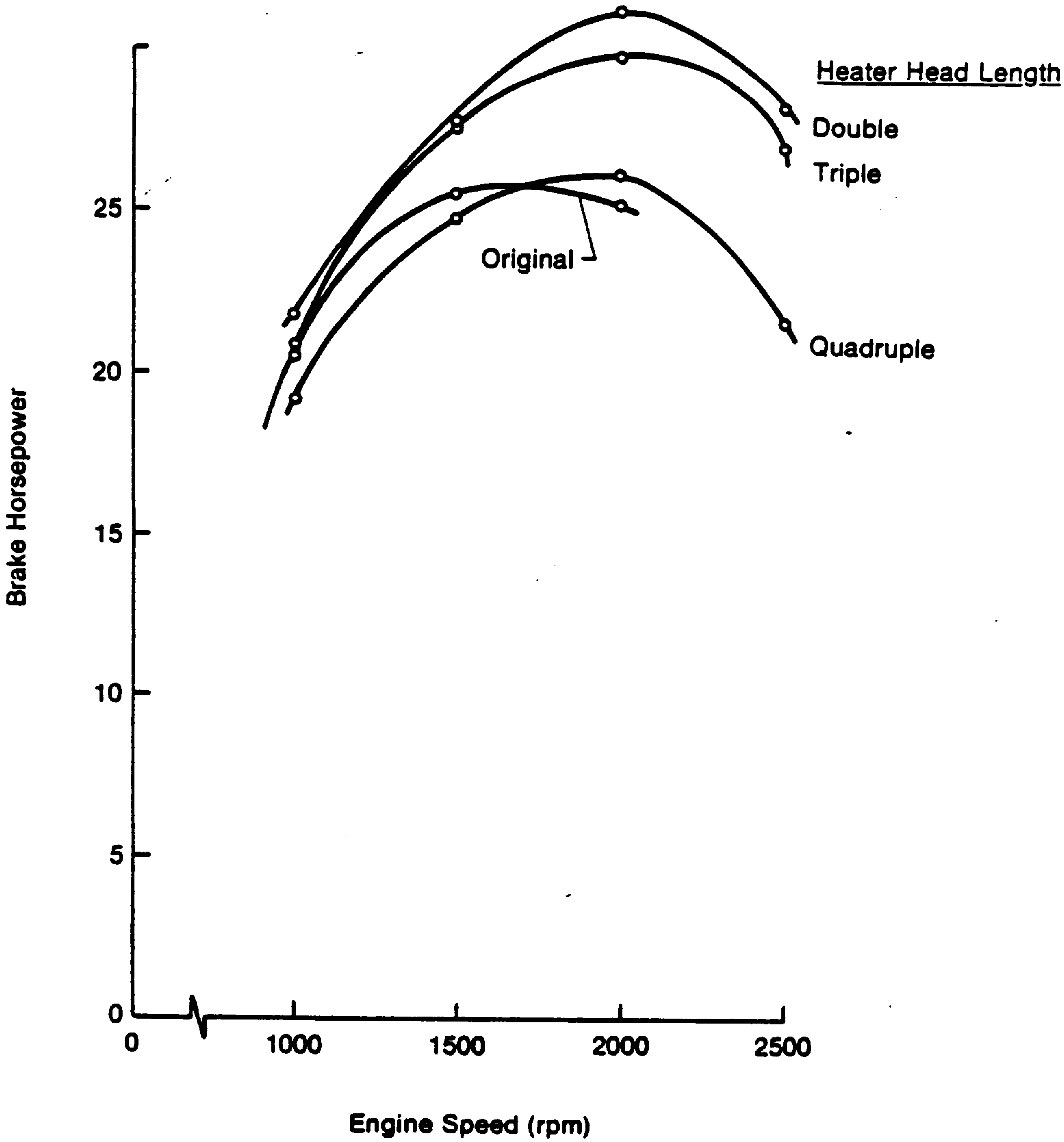


Figure 2-8. Mod II Design Power Recovery as a Function of Heater Head Tube Length

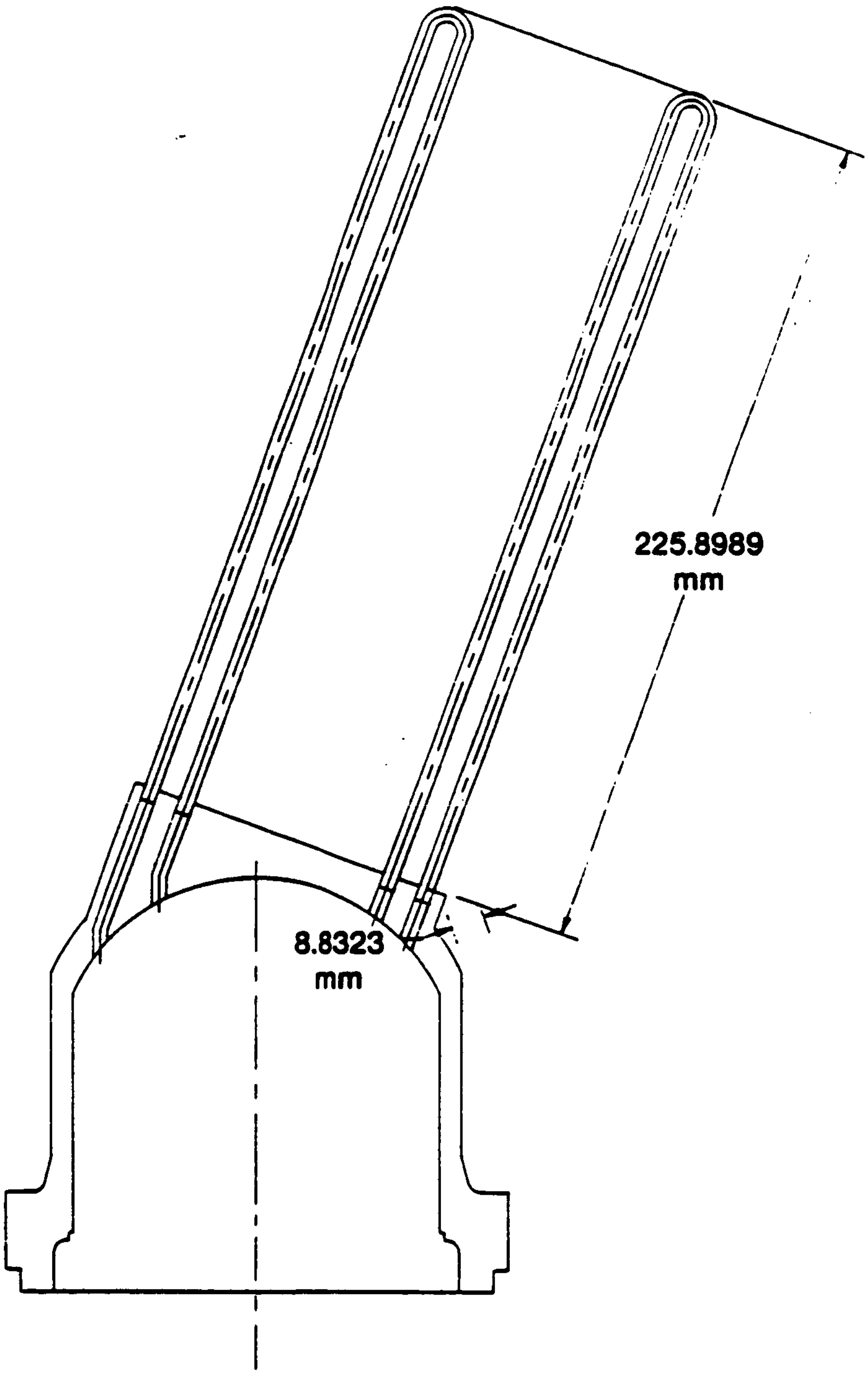


Figure 2-9. Individually Tubed Heater Head Concept

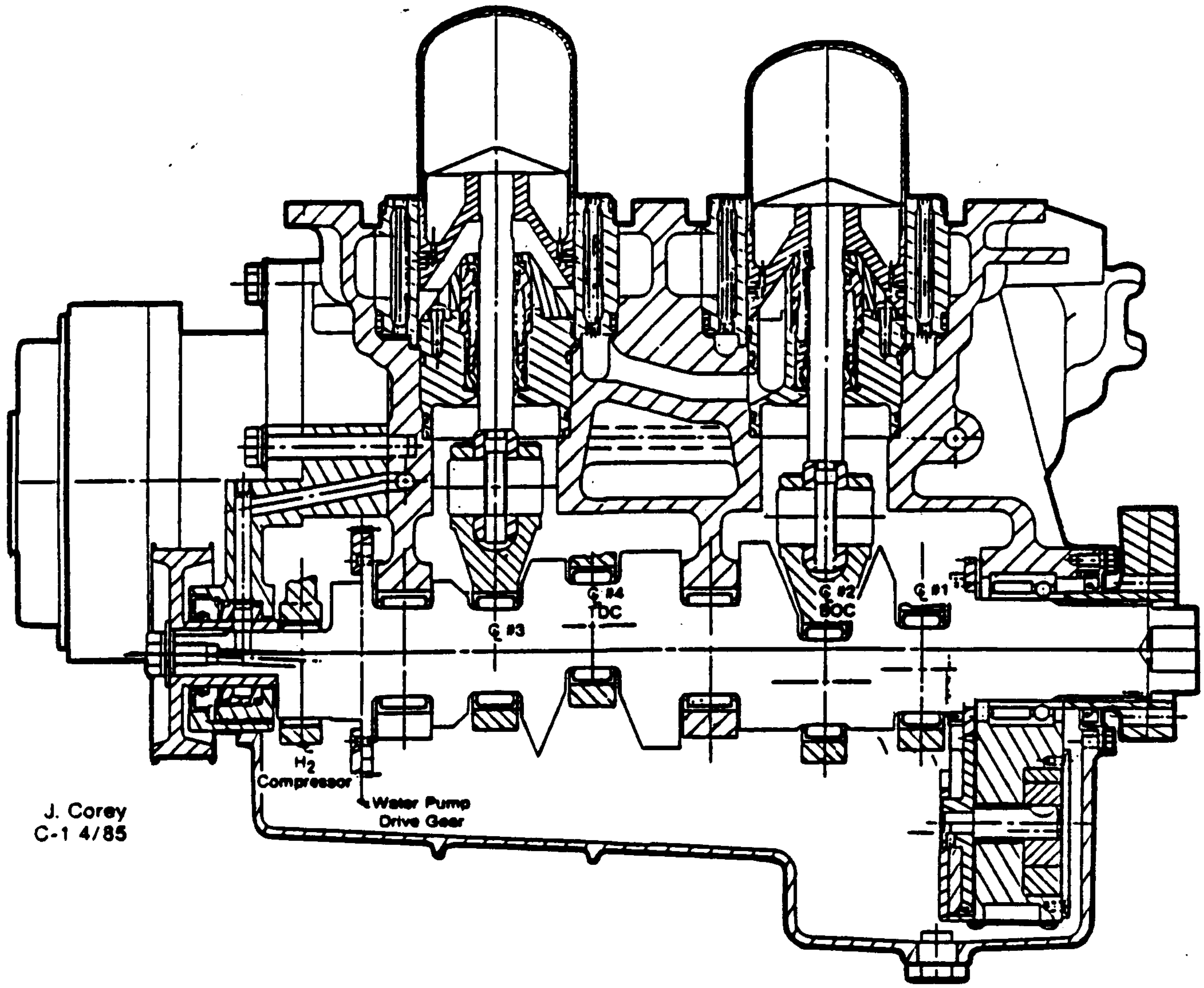


Figure 2-10. Cold Engine Drive System

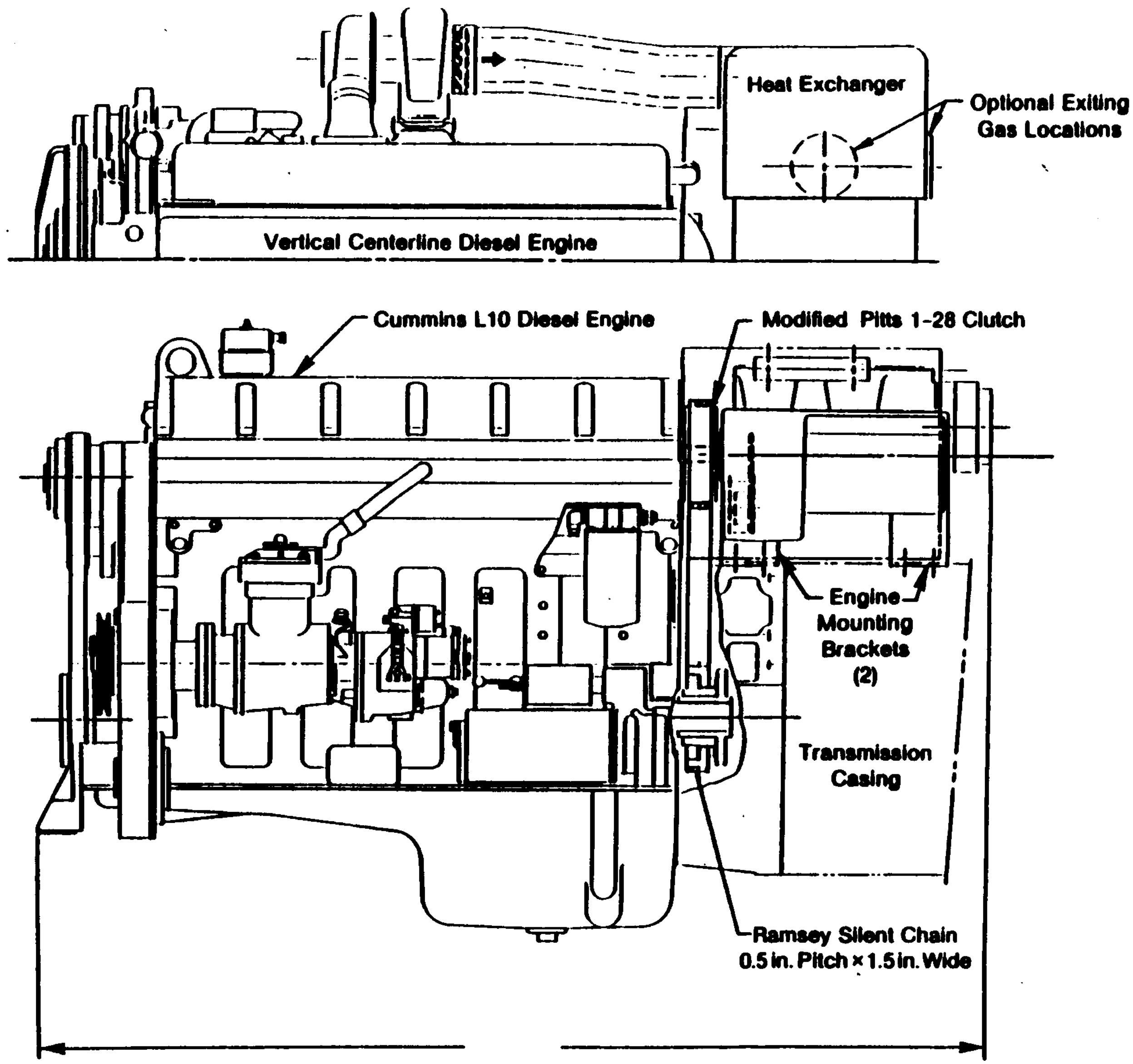


Figure 2-11. Diesel/Stirling Engine Integration (Side View)

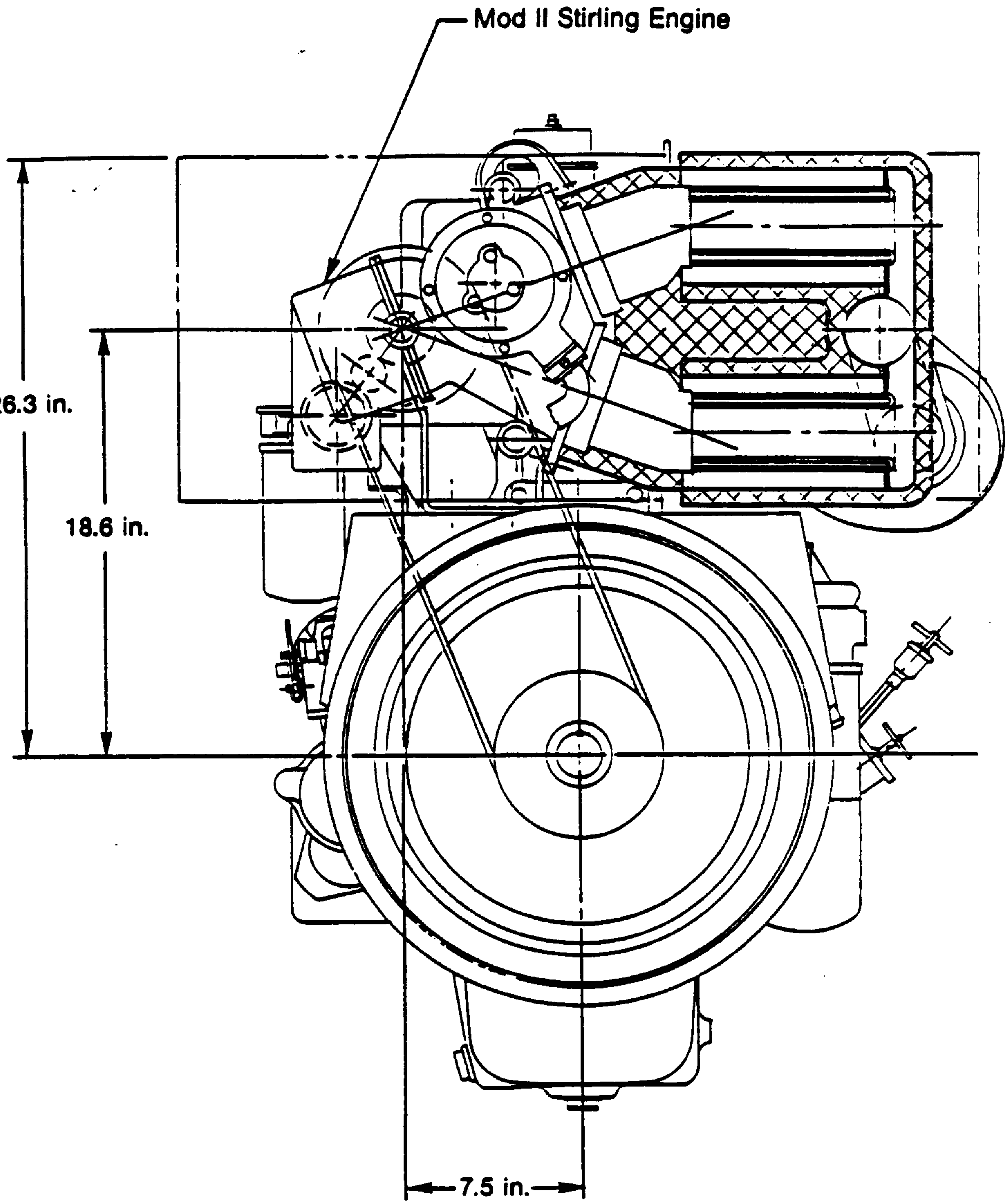


Figure 2-12. Diesel/Stirling Engine Integration (End View)

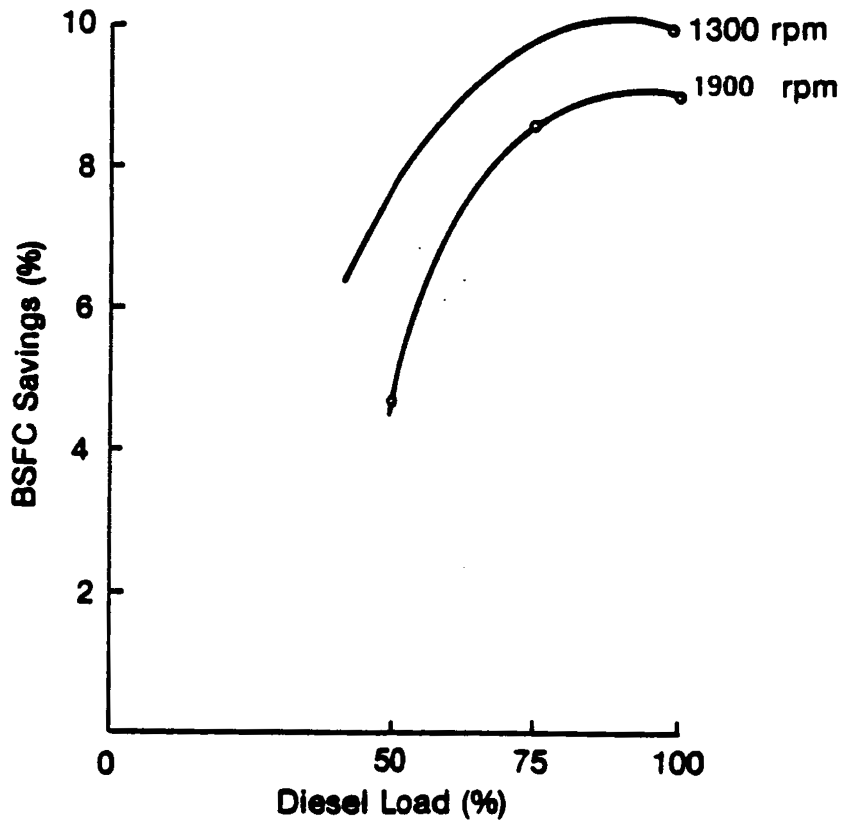


Figure 2-13. Stirling Bottoming Cycle Performance Map with Turbocharged Diesel

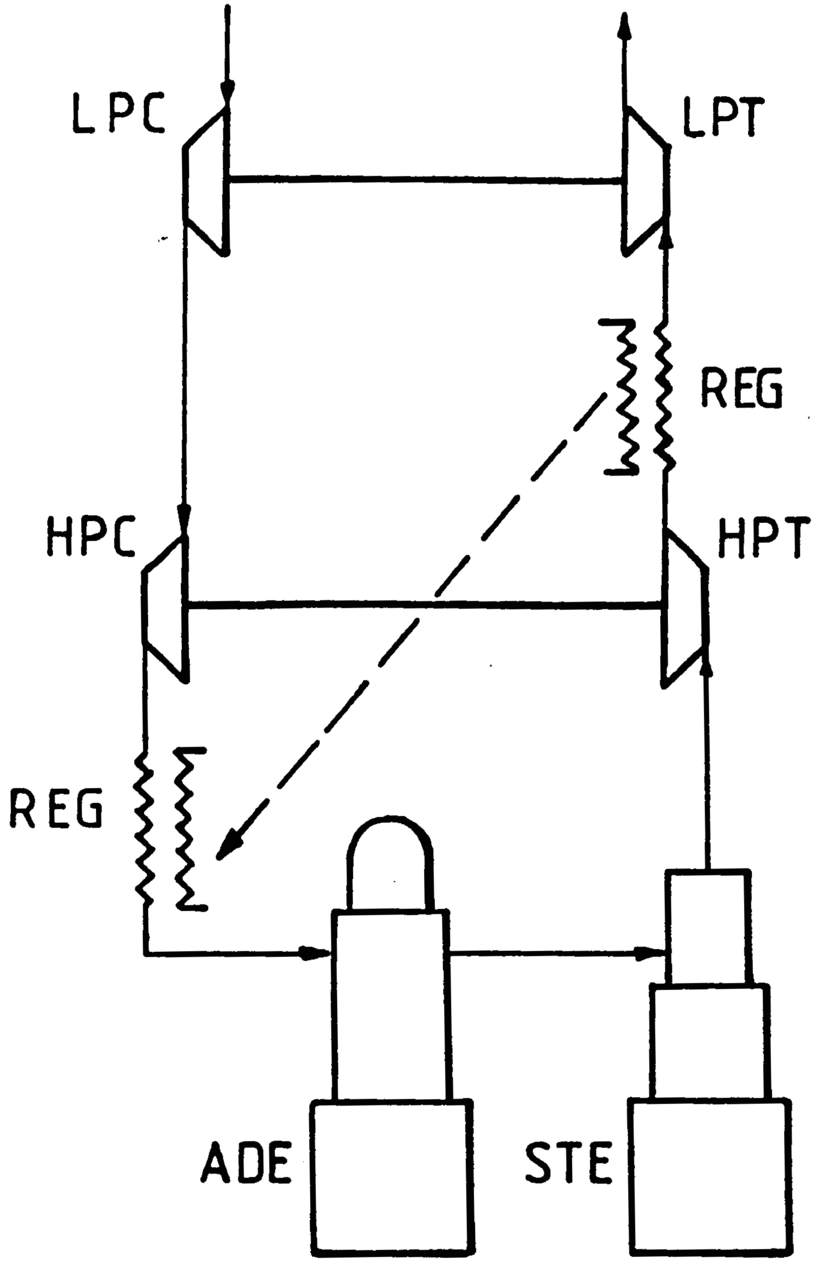


Figure 2-14 : Schematic of TSA Diesel Engine with the Following Notation:

LPC = Low Pressure Compressor
HPC = High Pressure Compressor
REG = Regenerator
ADE = Adiabatic Diesel Engine
STE = Stirling Engine
HPT = High Pressure Turbine
LPT = Low Pressure Turbine

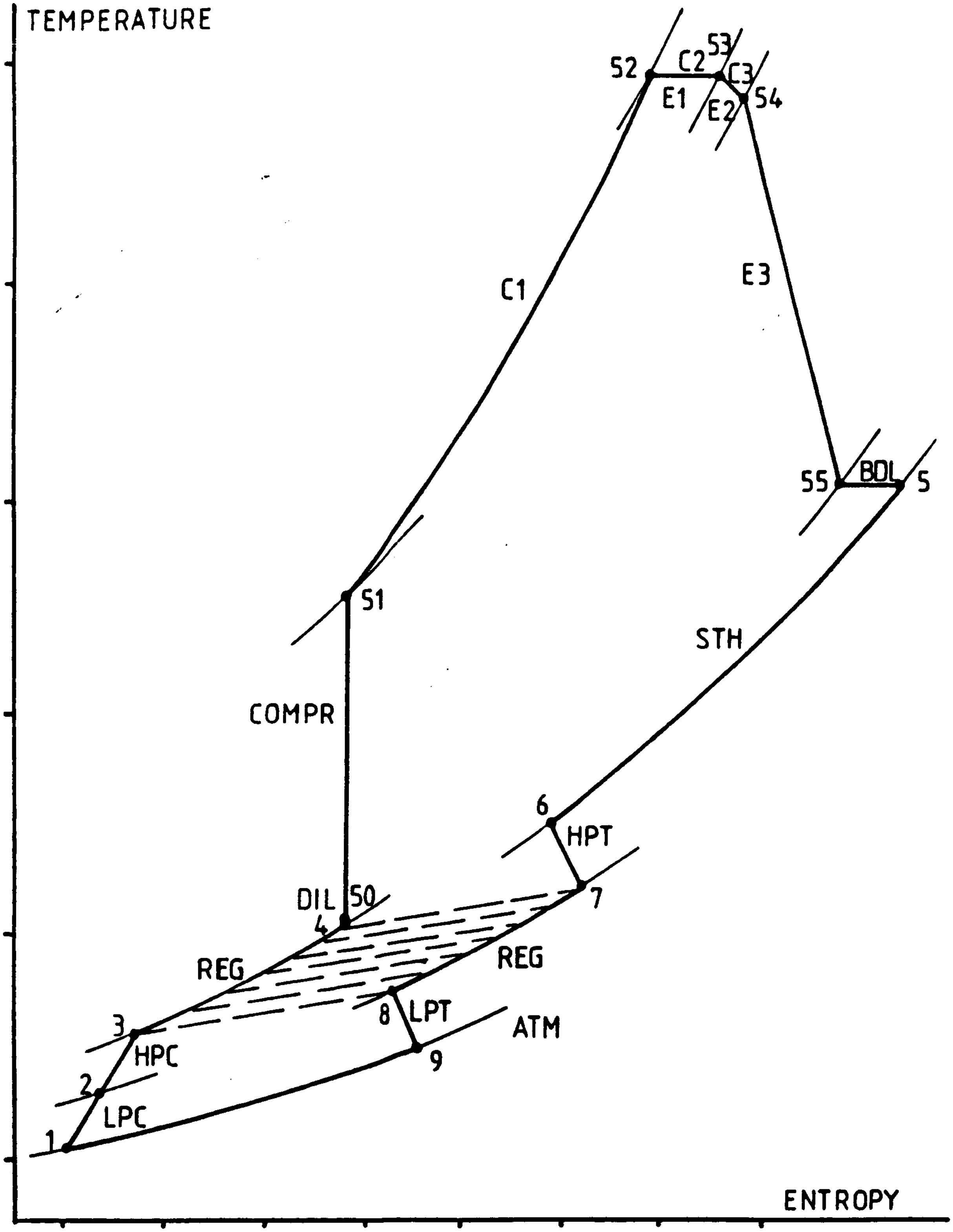


Figure 2-15 : T-s-Diagram for TSA Diesel Engine Showing Components and Part Processes.
For notation, see following page.

Attachment to Figure 2-15.
Notation to T-S Diagram

LPC	=	Low Pressure Compressor
HPC	=	High Pressure Compressor
REG	=	Heat Regenerator
DIL	=	Diesel Inlet Loss
COMPR	=	Compression in Diesel Engine
C1-C3	=	Combustion in Diesel Engine
E1-E3	=	Expansion in Diesel Engine
BDL	=	Blow-Down Loss
STH	=	Stirling Engine Heater
HPT	=	High Pressure Turbine
REG	=	Heat Regenerator
LPT	=	Low Pressure Turbine
ATM	=	Atmospheric Pressure Line

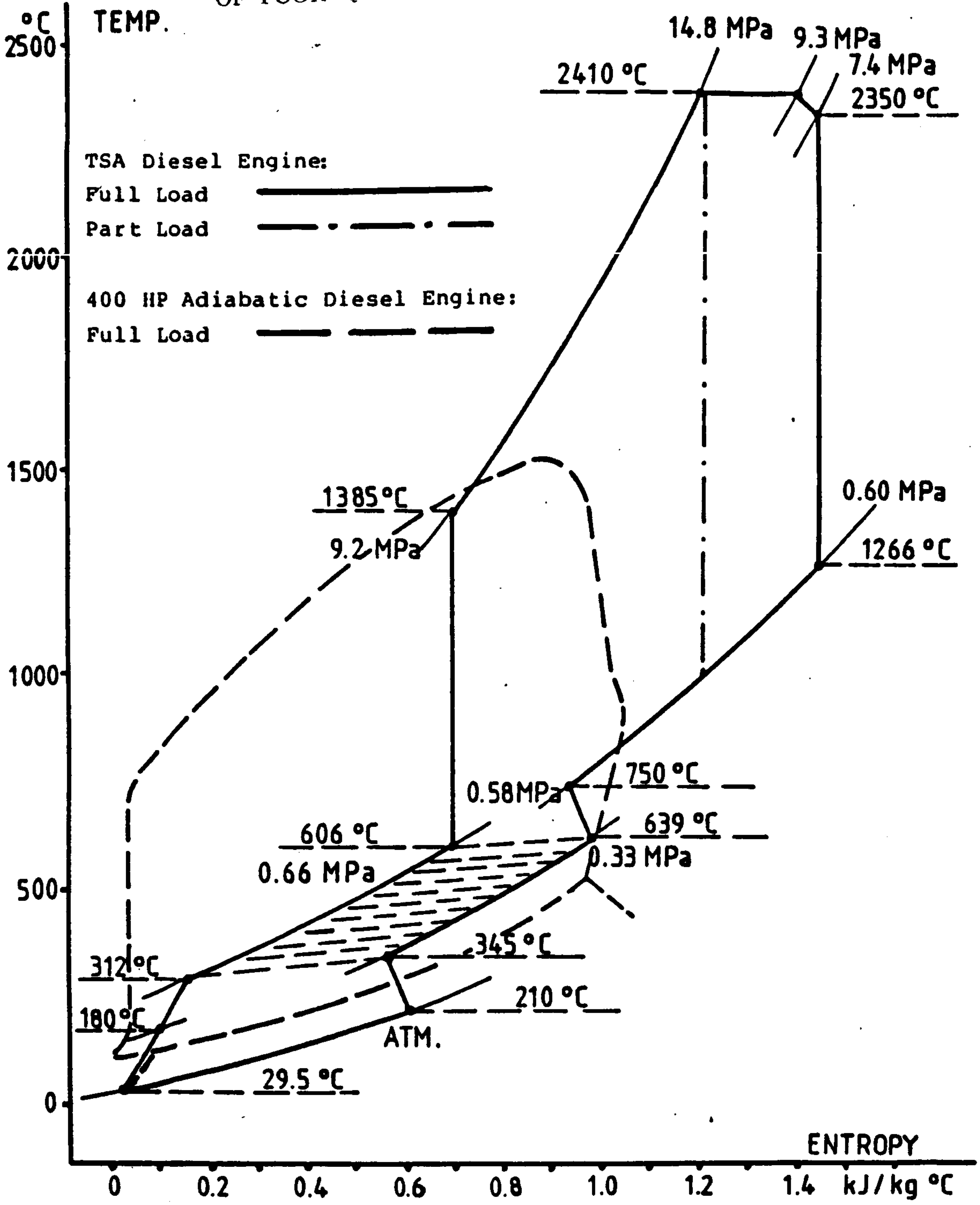


Figure 2-16 : T-s-Diagram for TSA Diesel Engine, Full Load and Part Load as well as, for Comparison, 400HP Adiabatic Diesel Engine. TSA Performance below:

Performance	ADE	STE	OVERALL
Power, kW	358	94	452
Efficiency, %	48	40	605
Fuel/Air Ratio	0.050	-	-
Air Flow, kg/s	0.35		

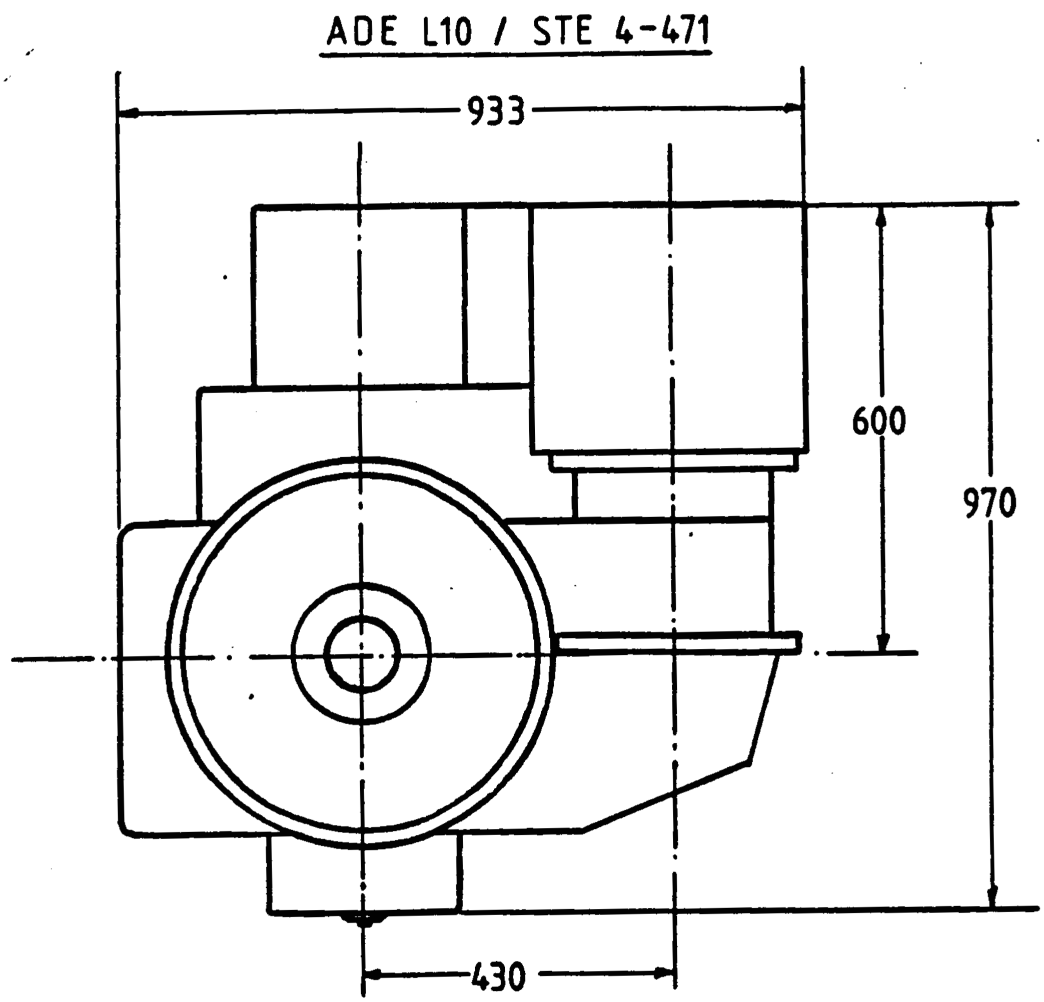


Figure 2-17. Semi-Integrated Design (In-Line Engines)

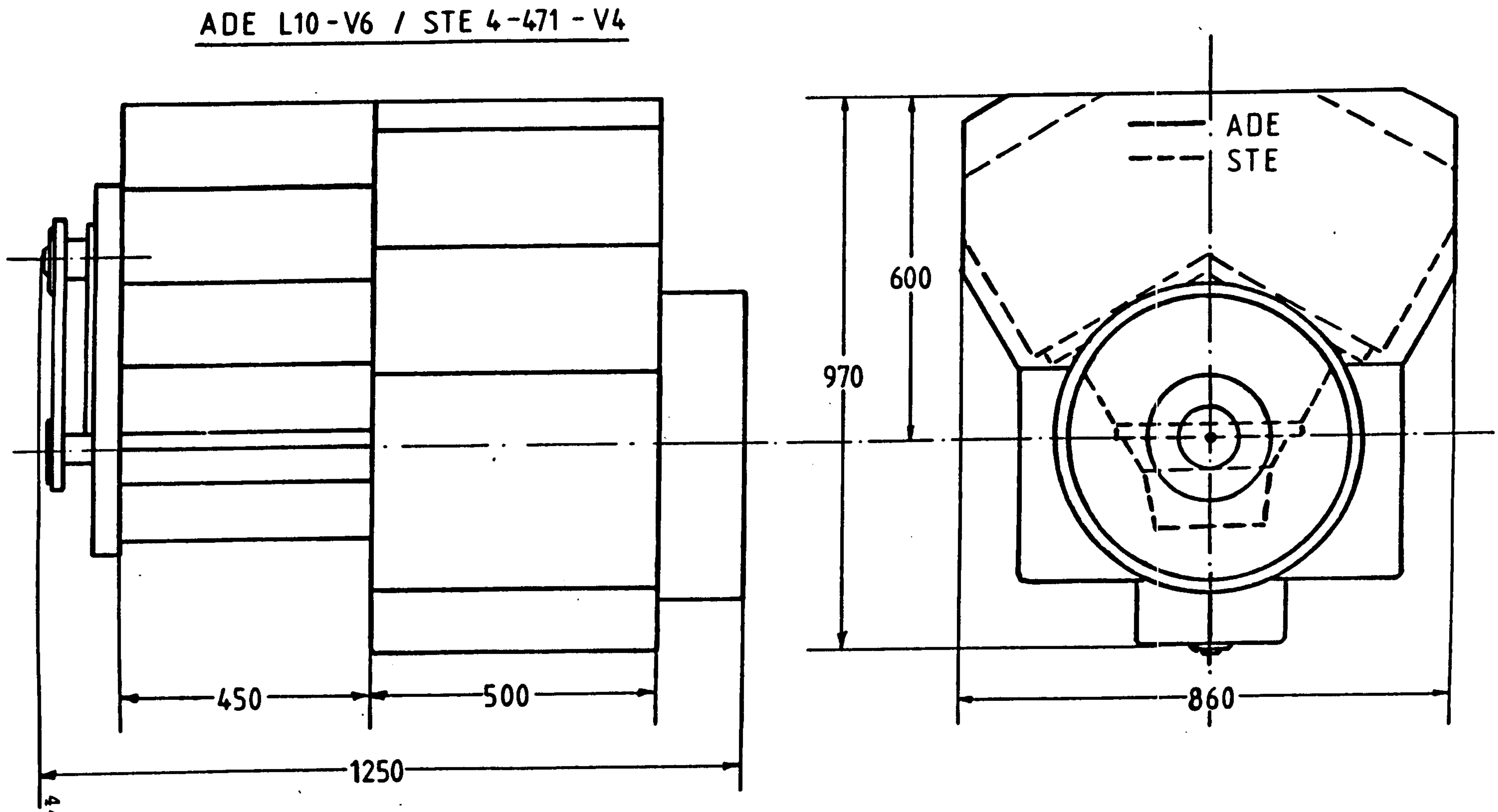


Figure 2-18. Semi-Integrated Design (V-Engines)

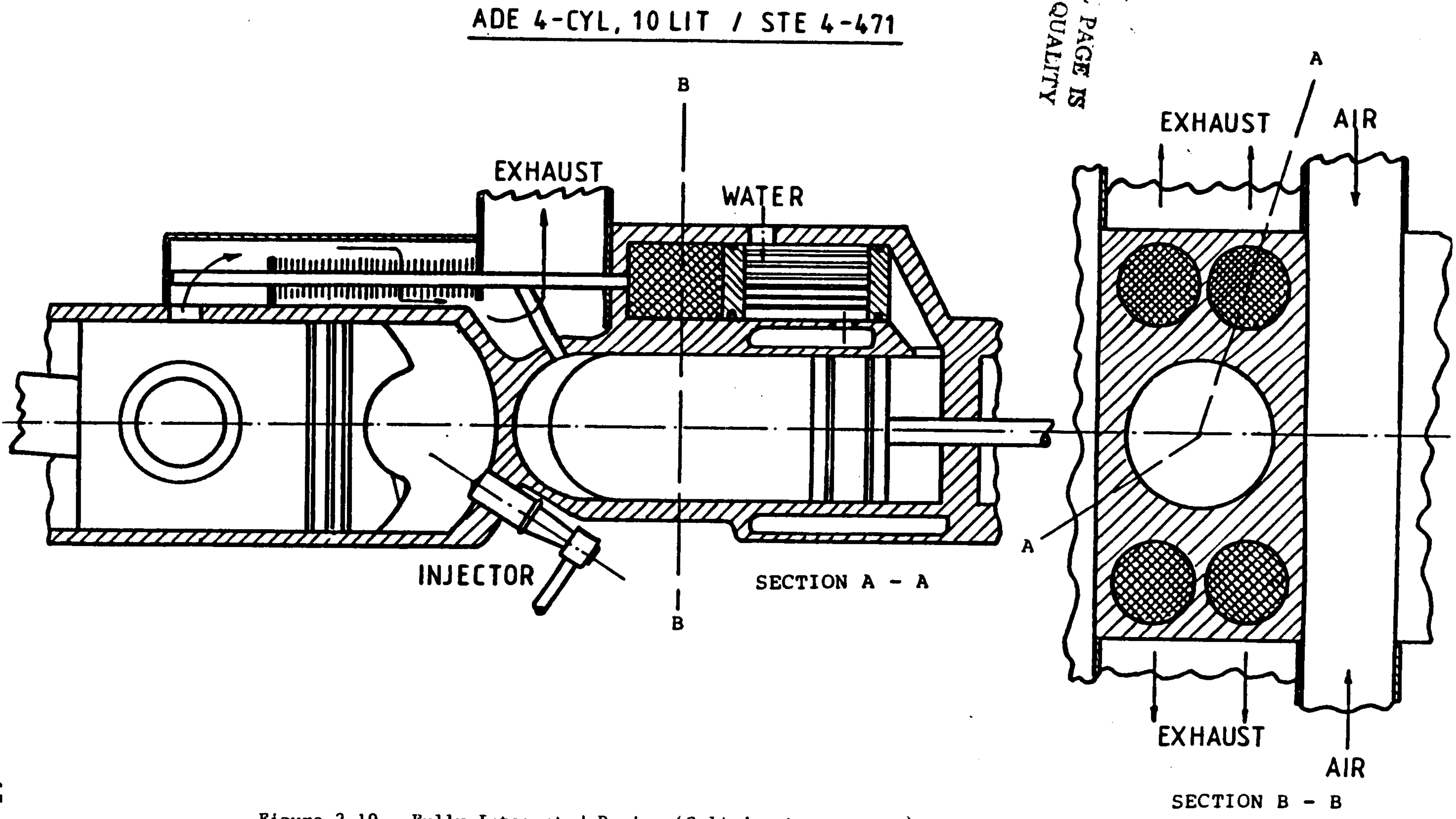


Figure 2-19. Fully Integrated Design (Cylinder Arrangement)

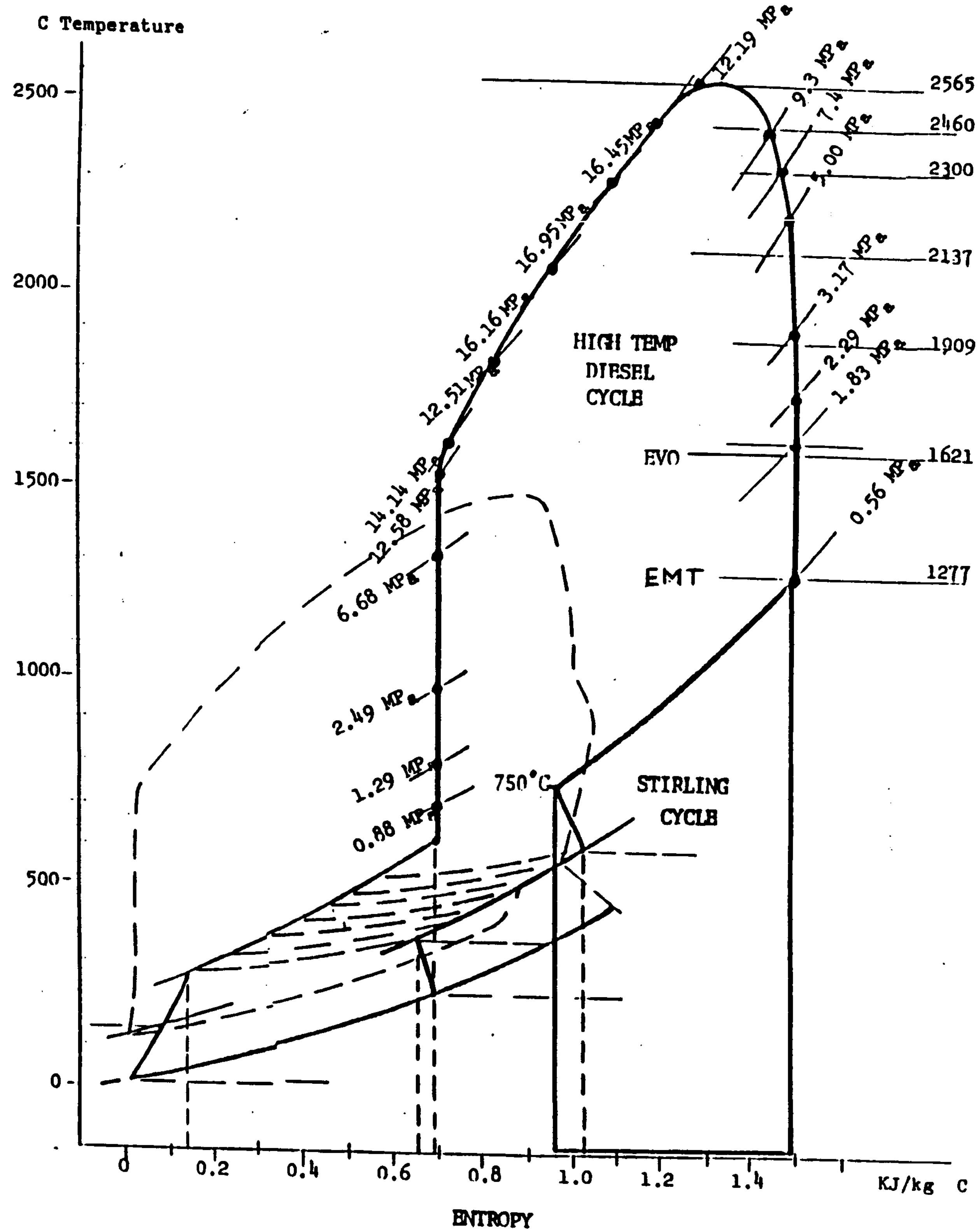


Figure 2-20. T-S Diagram for TSA Diesel Engine with High Temperature Diesel Cycle

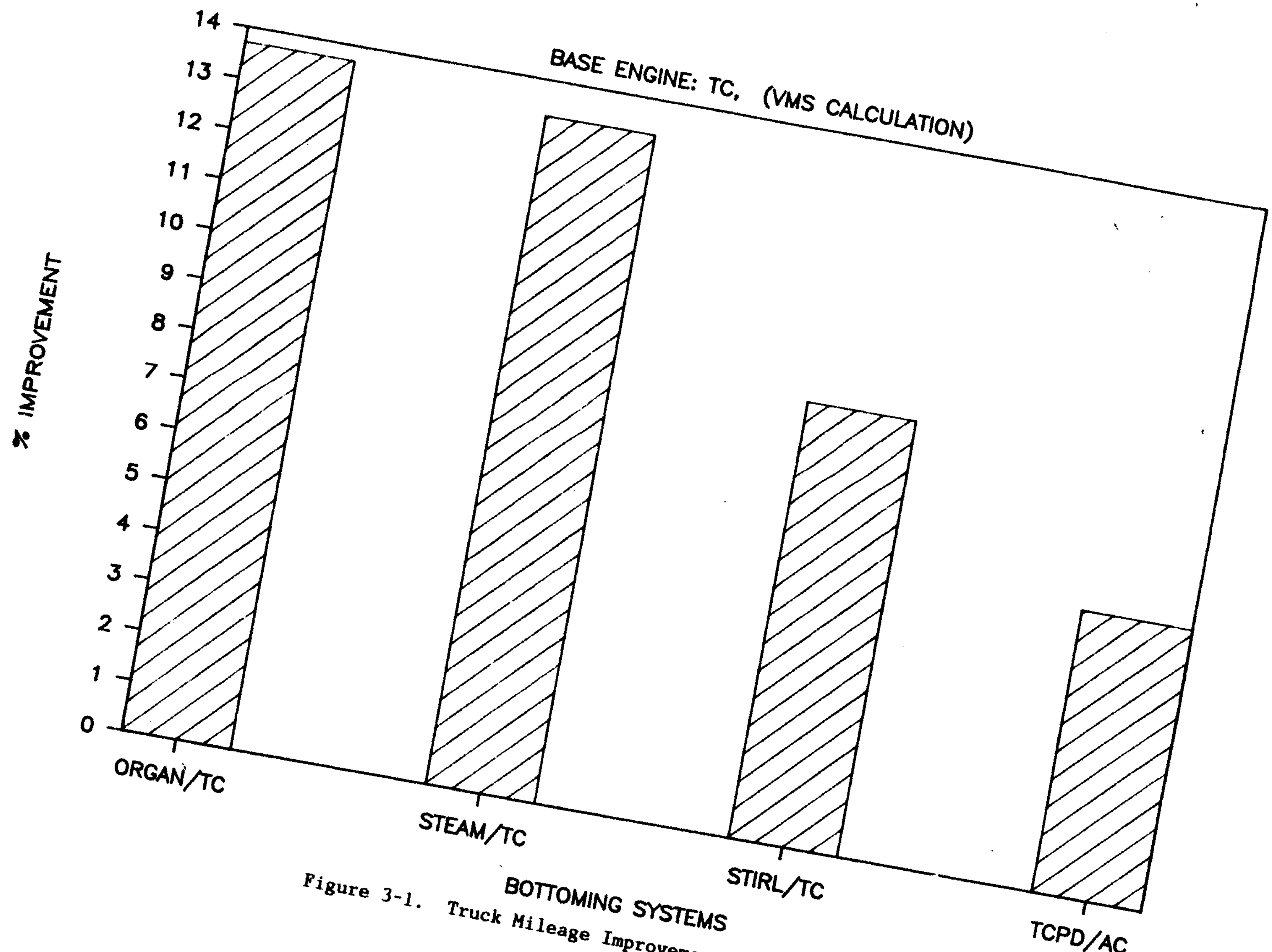


Figure 3-1. Truck Mileage Improvement (%)

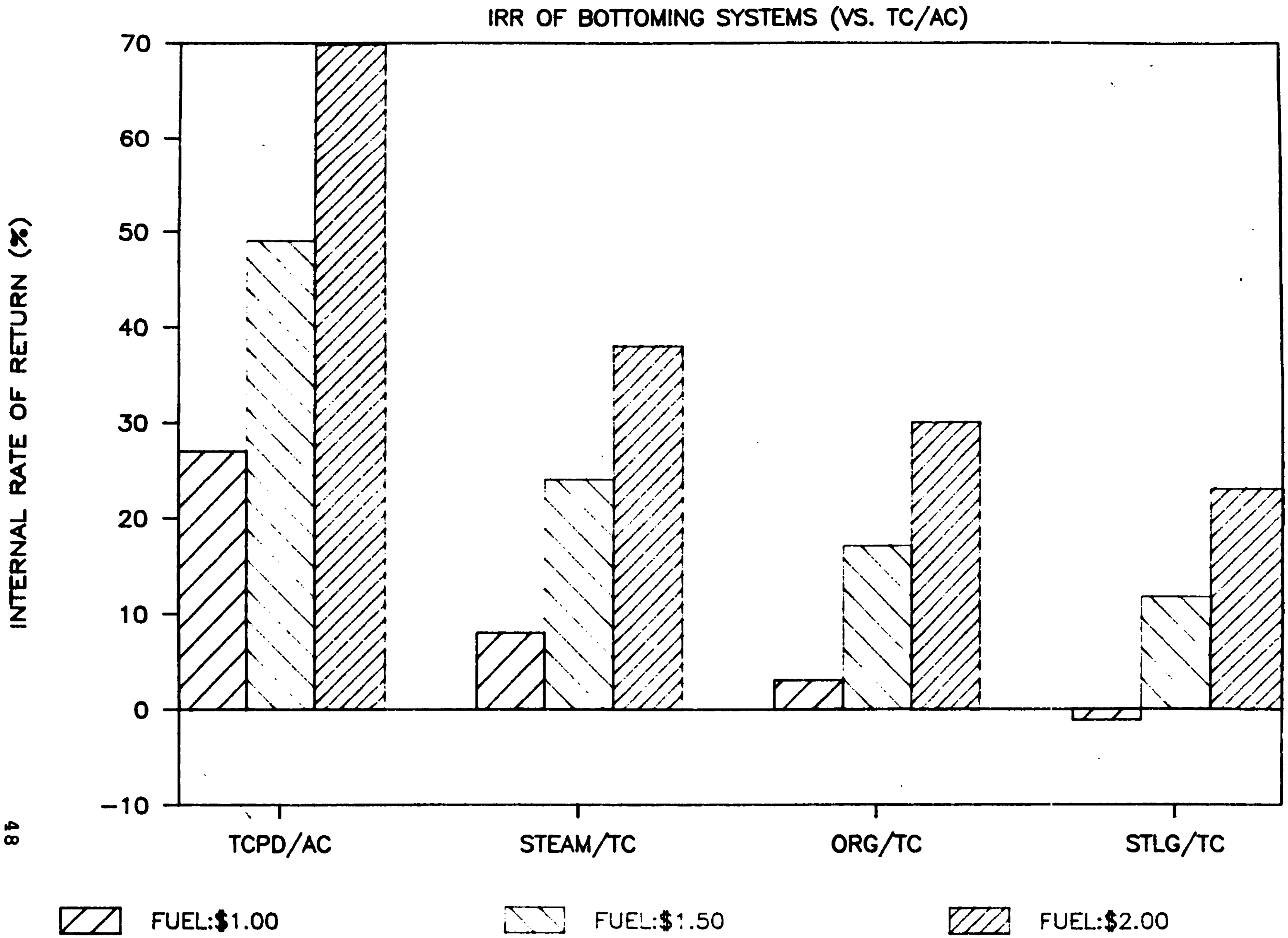


Figure 3-2. System Comparison

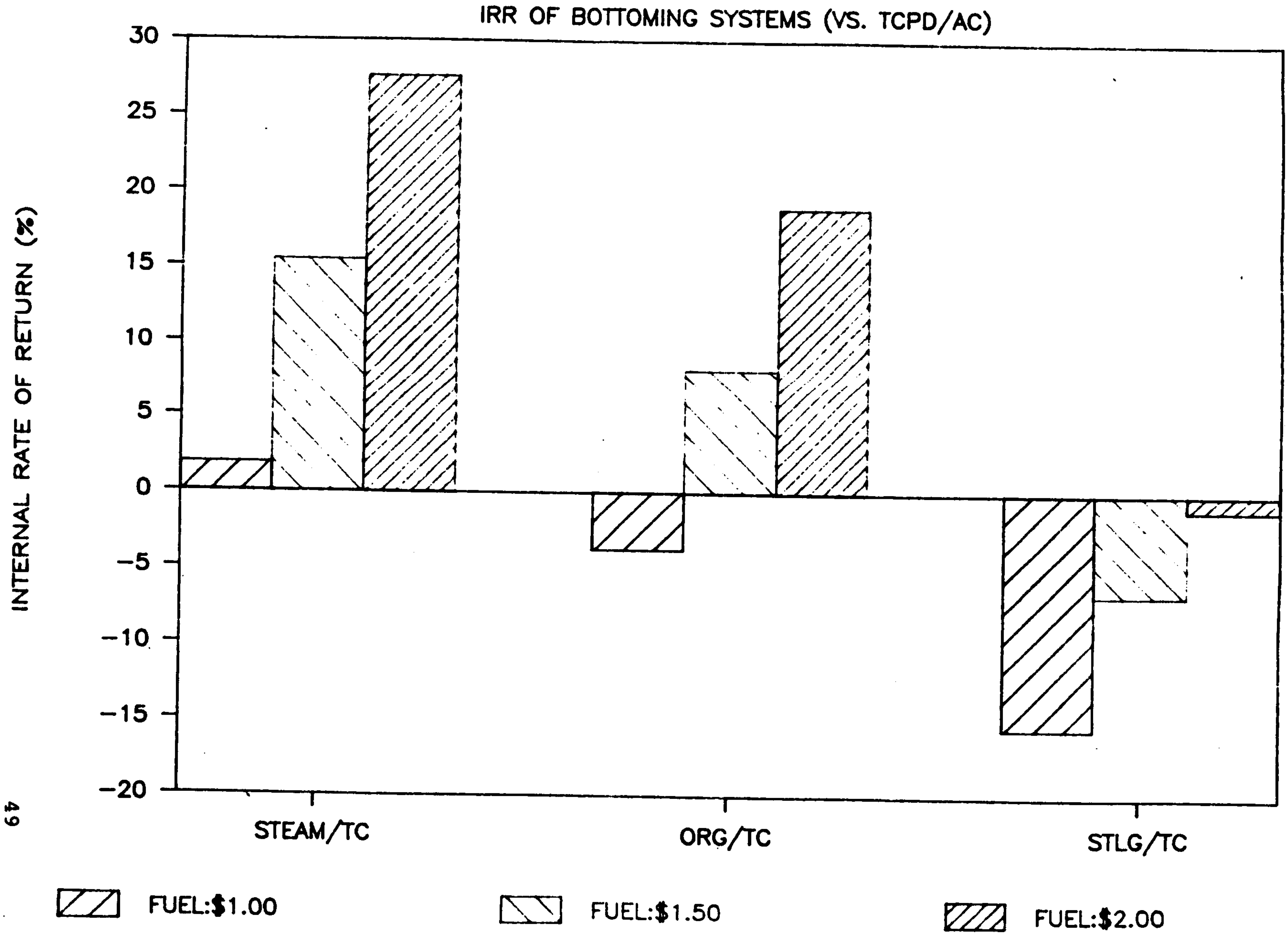


Figure 3-3. System Comparison

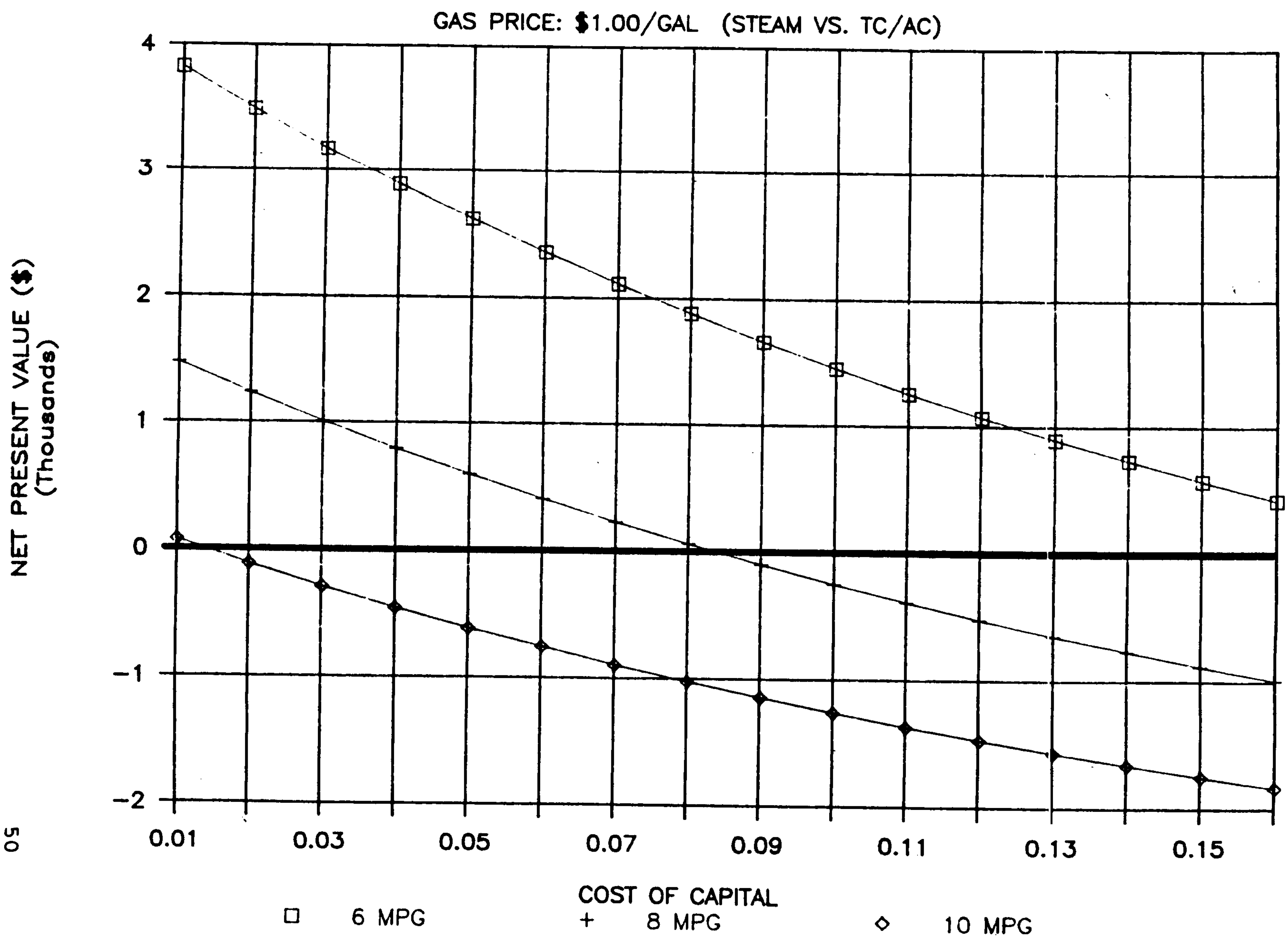


Figure 3-4. Truck Fuel Economy Effect

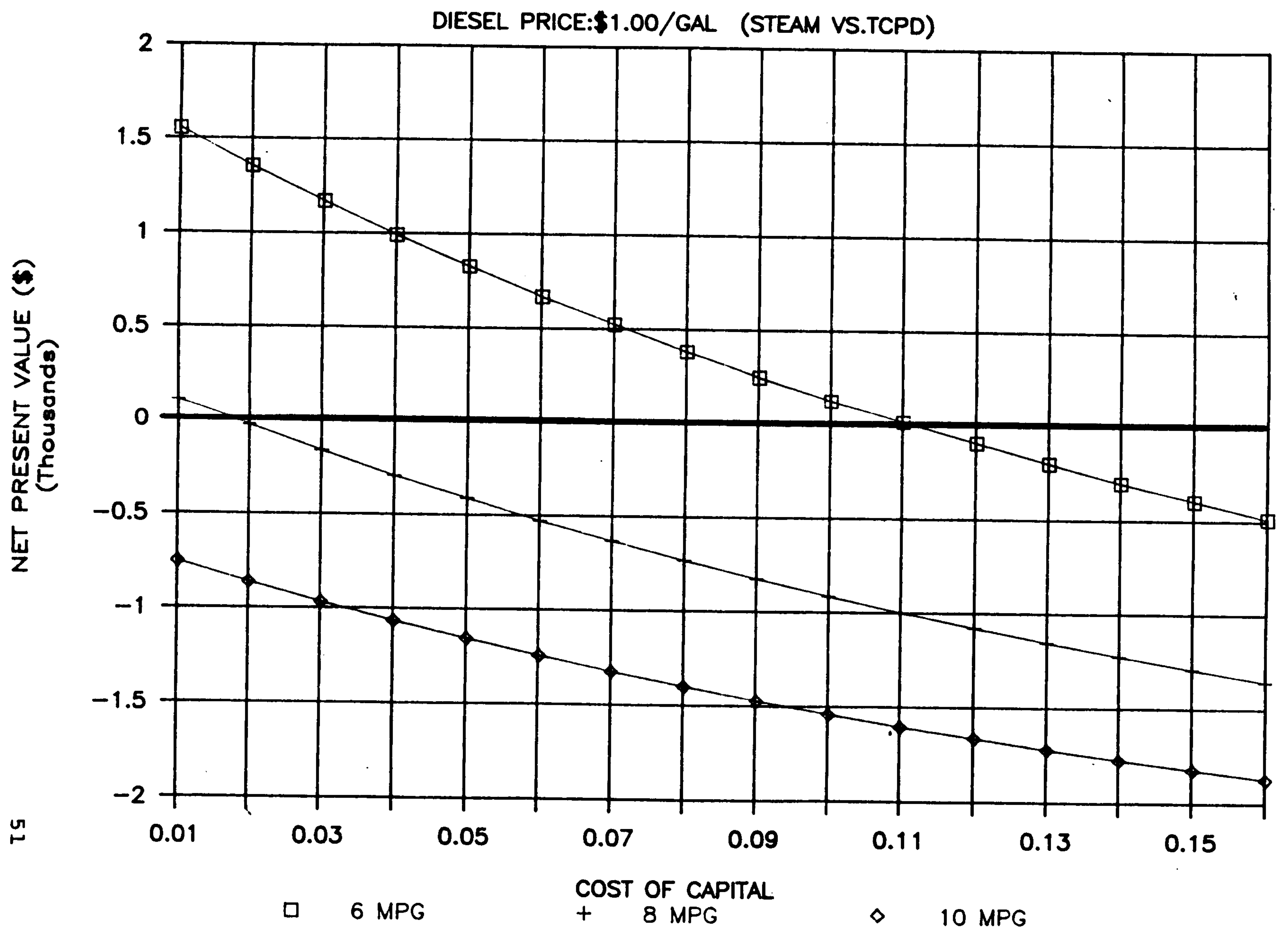
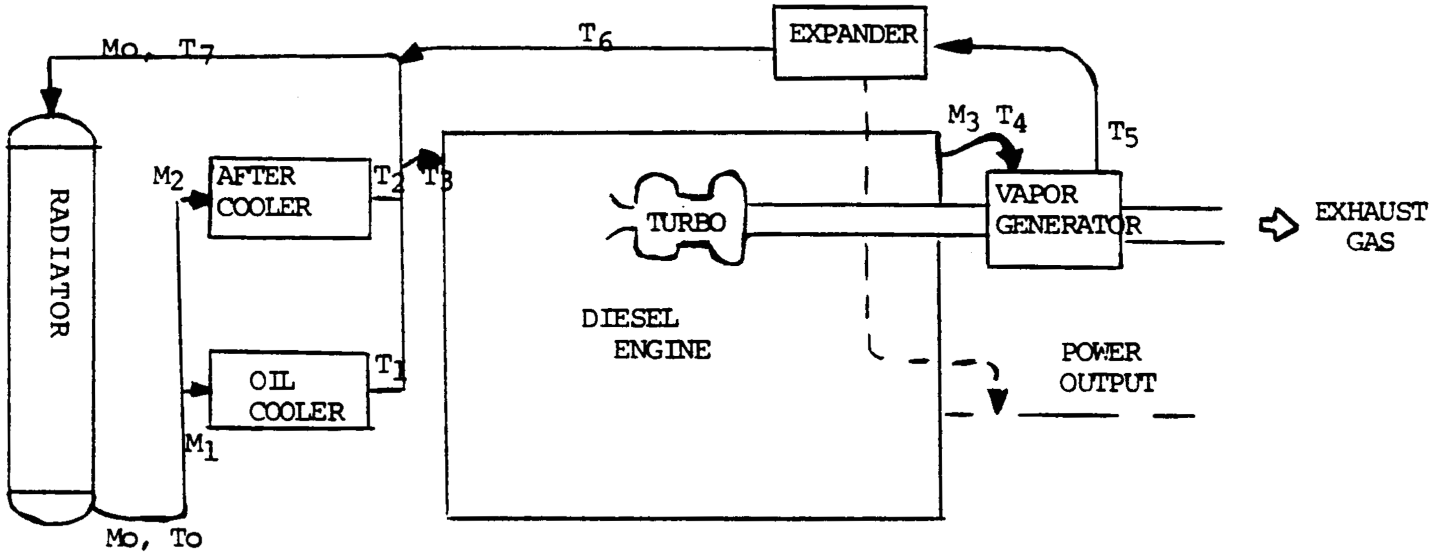


Figure 3-5. Truck Fuel Economy Effect

USE OF ONE FLUID FOR:

- DIESEL ENGINE COOLING
- RANKINE CYCLE WORKING FLUID



A SCHEMATIC OF A PROPOSED SYSTEM

Figure 4-1. Integrated Diesel/Rankine Cycle

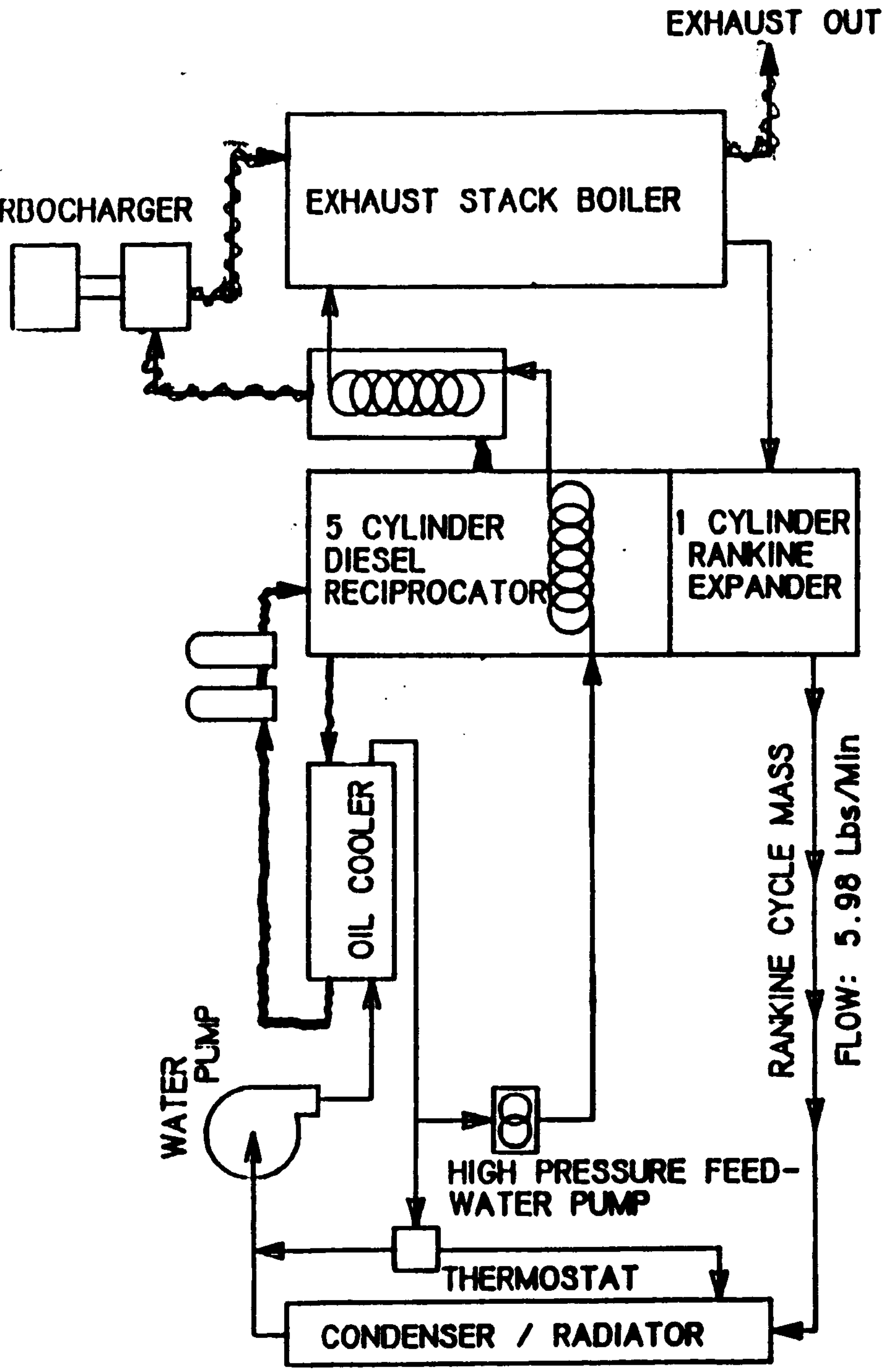


Figure 4-2. Integrated Steam Based Rankine Bottoming Cycle Schematic

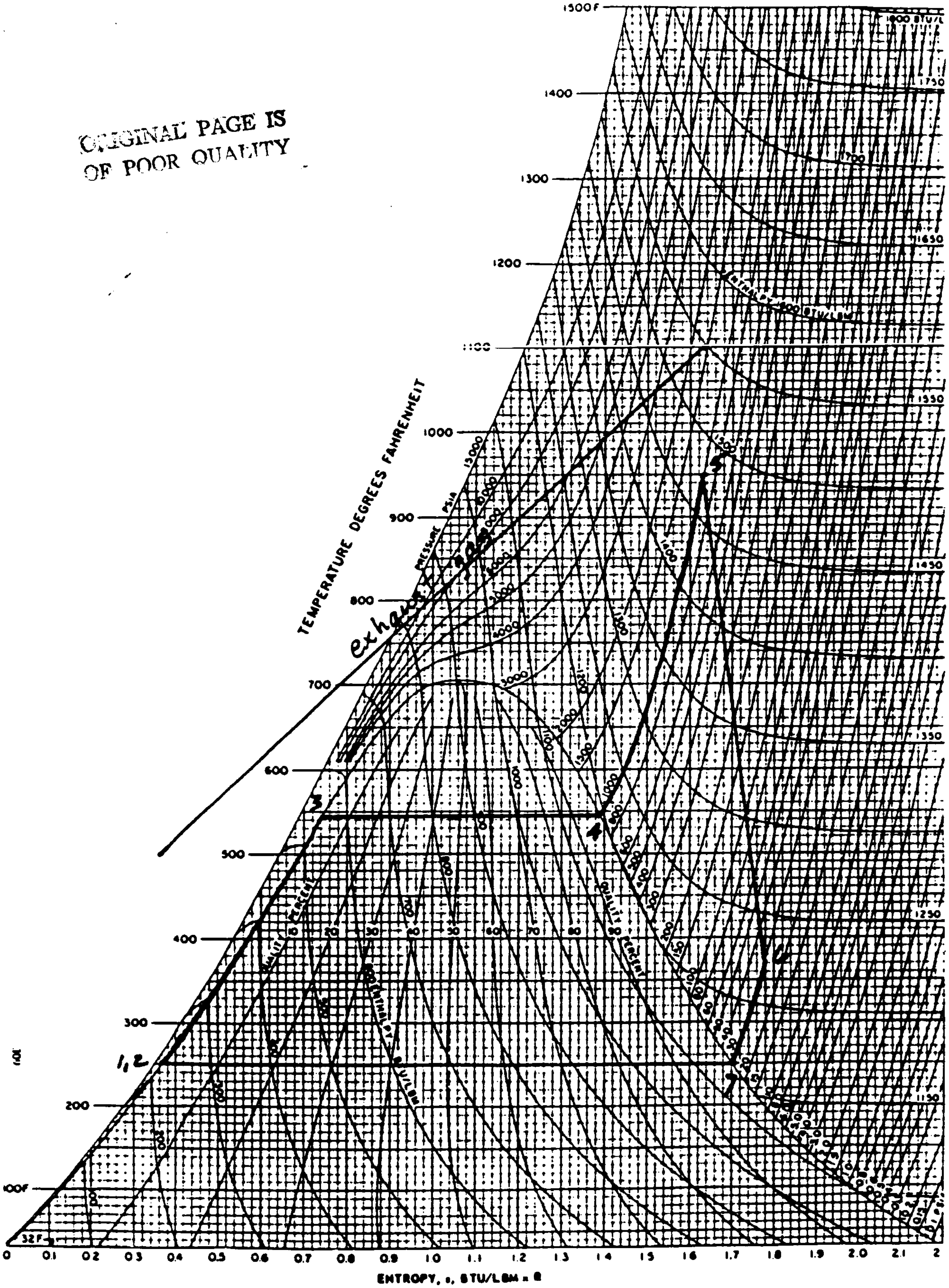


Figure 4-3. T-S Diagram for the Steam Rankine System

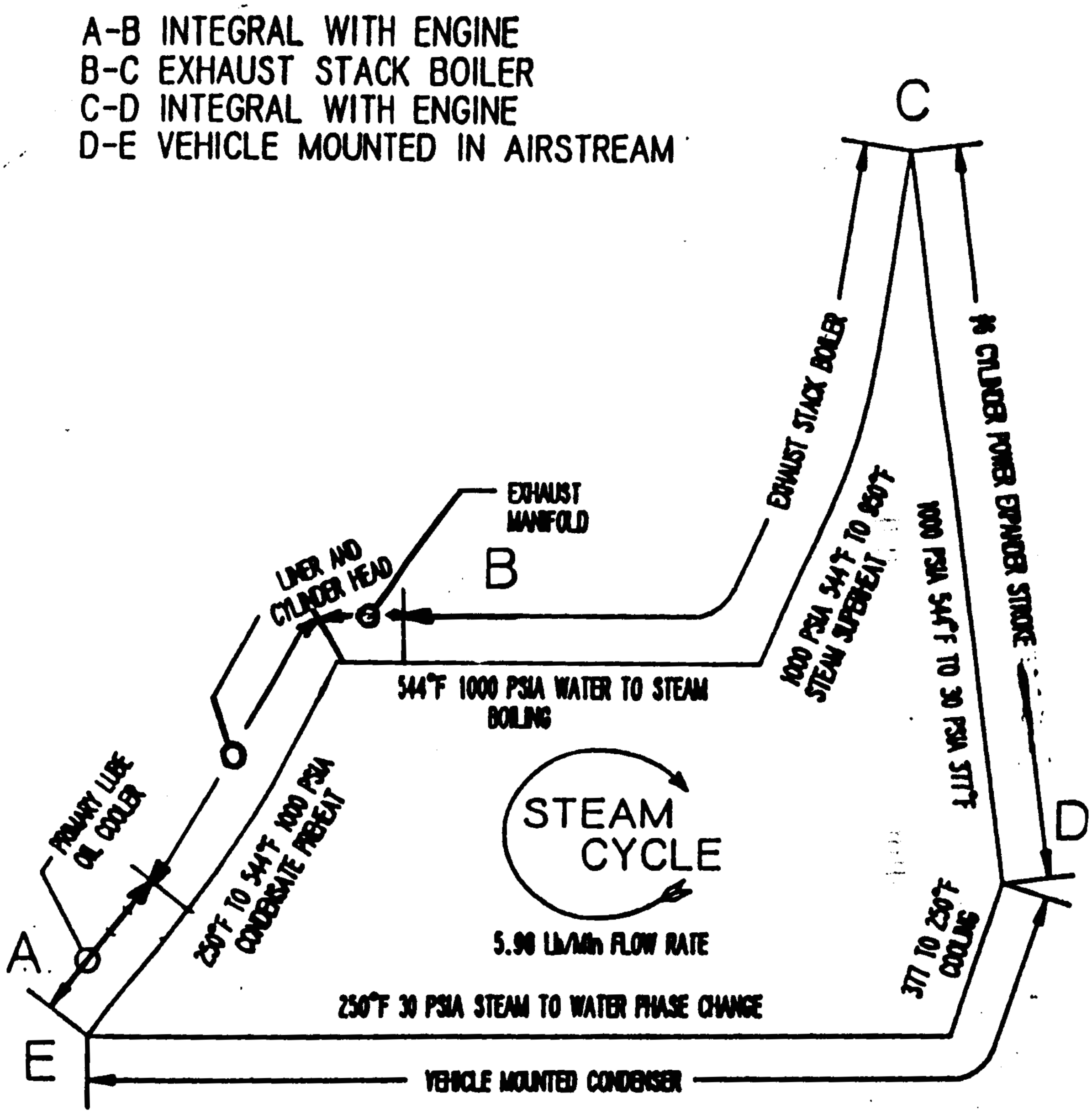
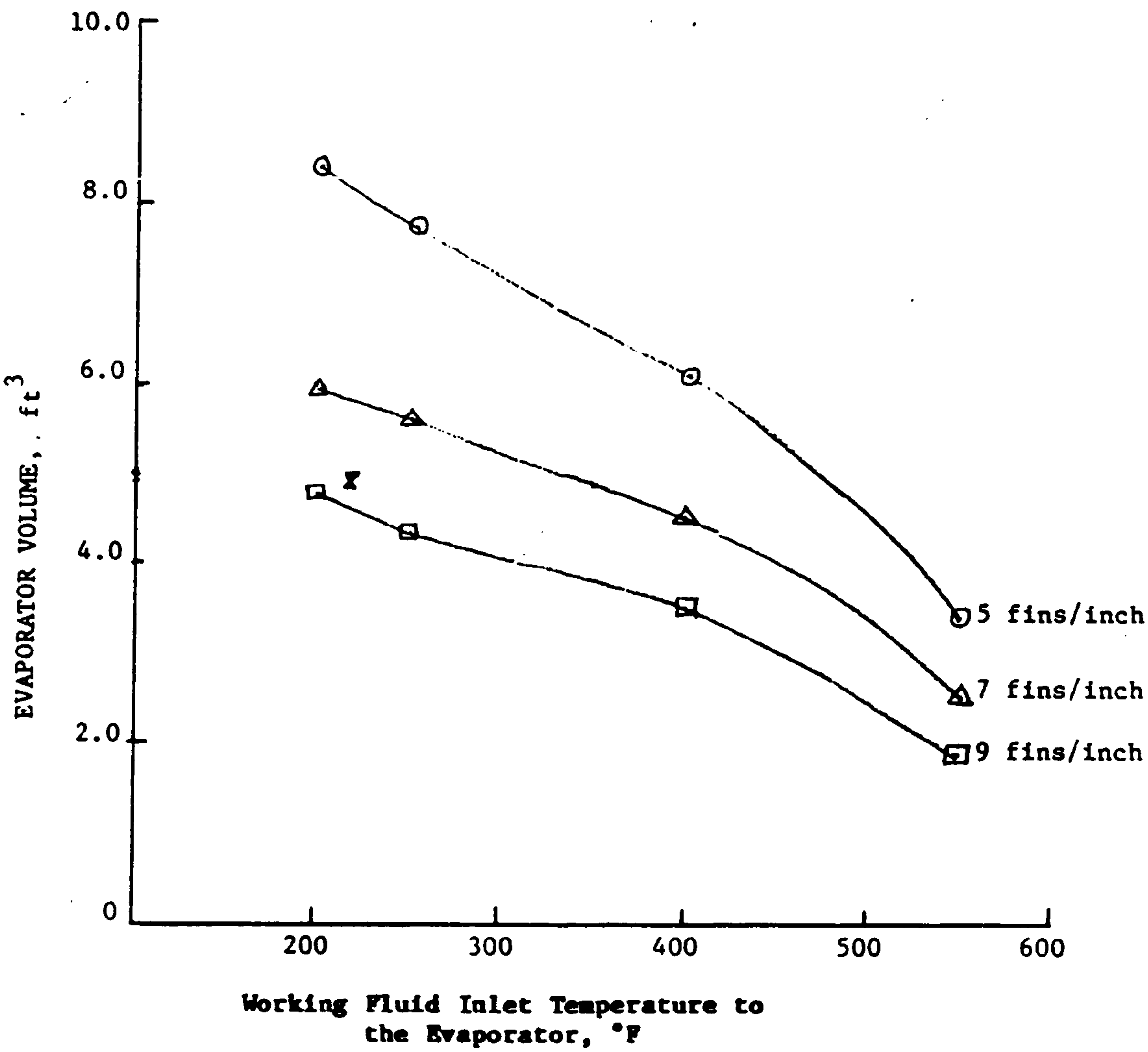
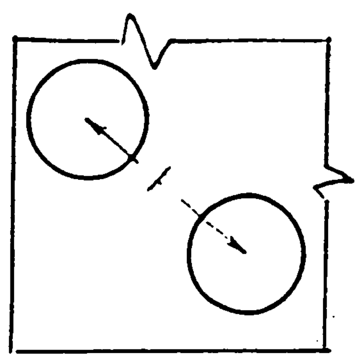


Figure 4-4. Steam Cycle Showing Integral Bottoming Cycle
Component Contributions



X - Department of Energy Demonstration Truck
Evaporator Designed & Built by
Thermoelectron

Figure 4-5. **Evaporator Size for the Integrated Rankine Bottoming Cycle for Truck Diesel Engine - Steam @ 1000 PSI**



Tube

Outside Diameter = 1/2" or 5/8"

Wall thickness = 0.04" to 0.05"

Material = carbon steel

pitch p = 3/4"

Fins

Thickness = 0.015" to 0.020"

Material = low carbon steel

No. of fins/inch = 6

- Multiple tube and fin compact Heat exchange design is recommended.
- Brazed tube to fin joint should be used
- Overall dimensions to suit packaging on the engine
- Total Fins surface area should match the figure chosen from the graphs.

Figure 4-6. Evaporator Tube and Fin Details

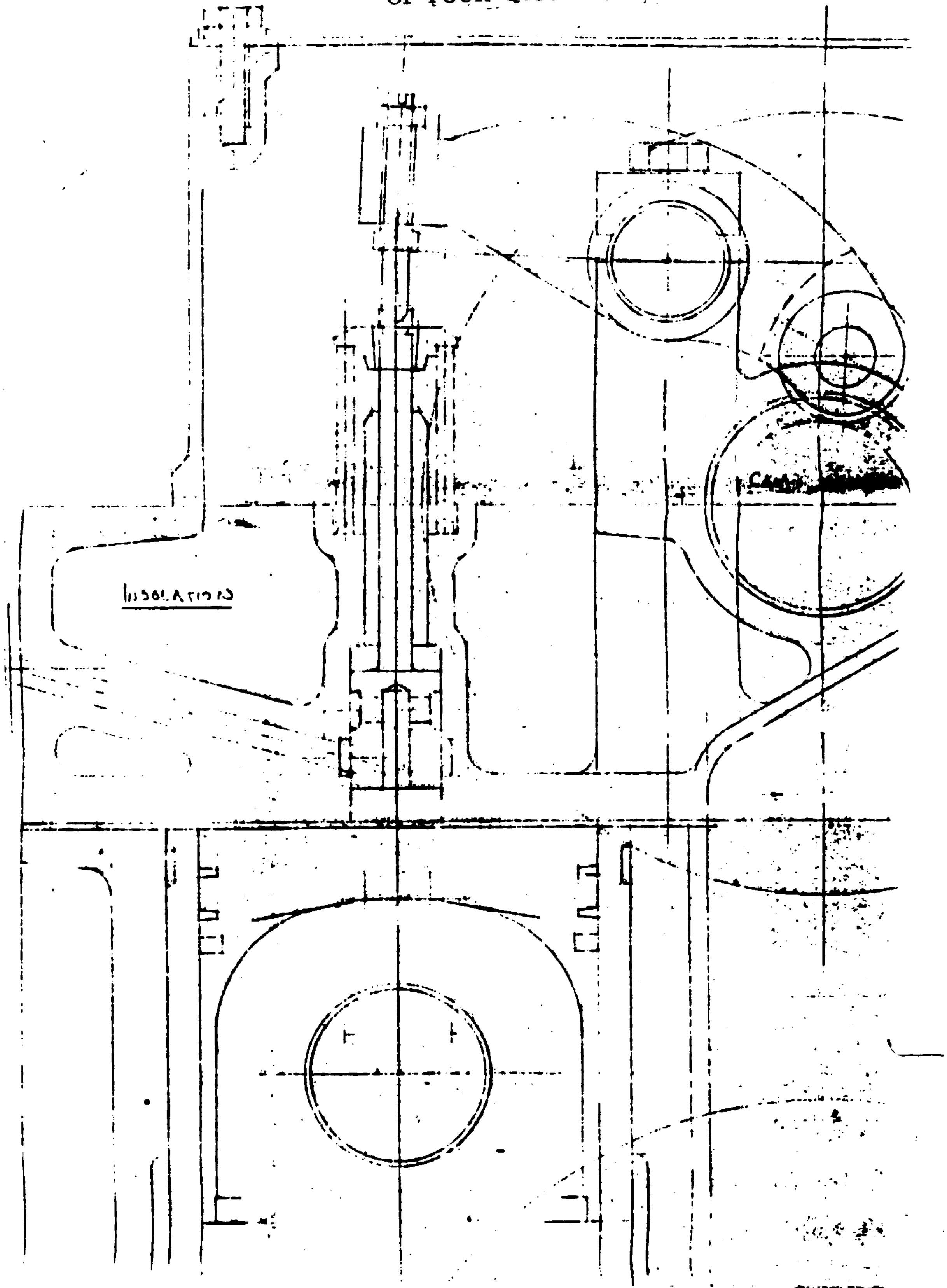


Figure 4-7. Intake Valve Arrangement for the Steam Expander

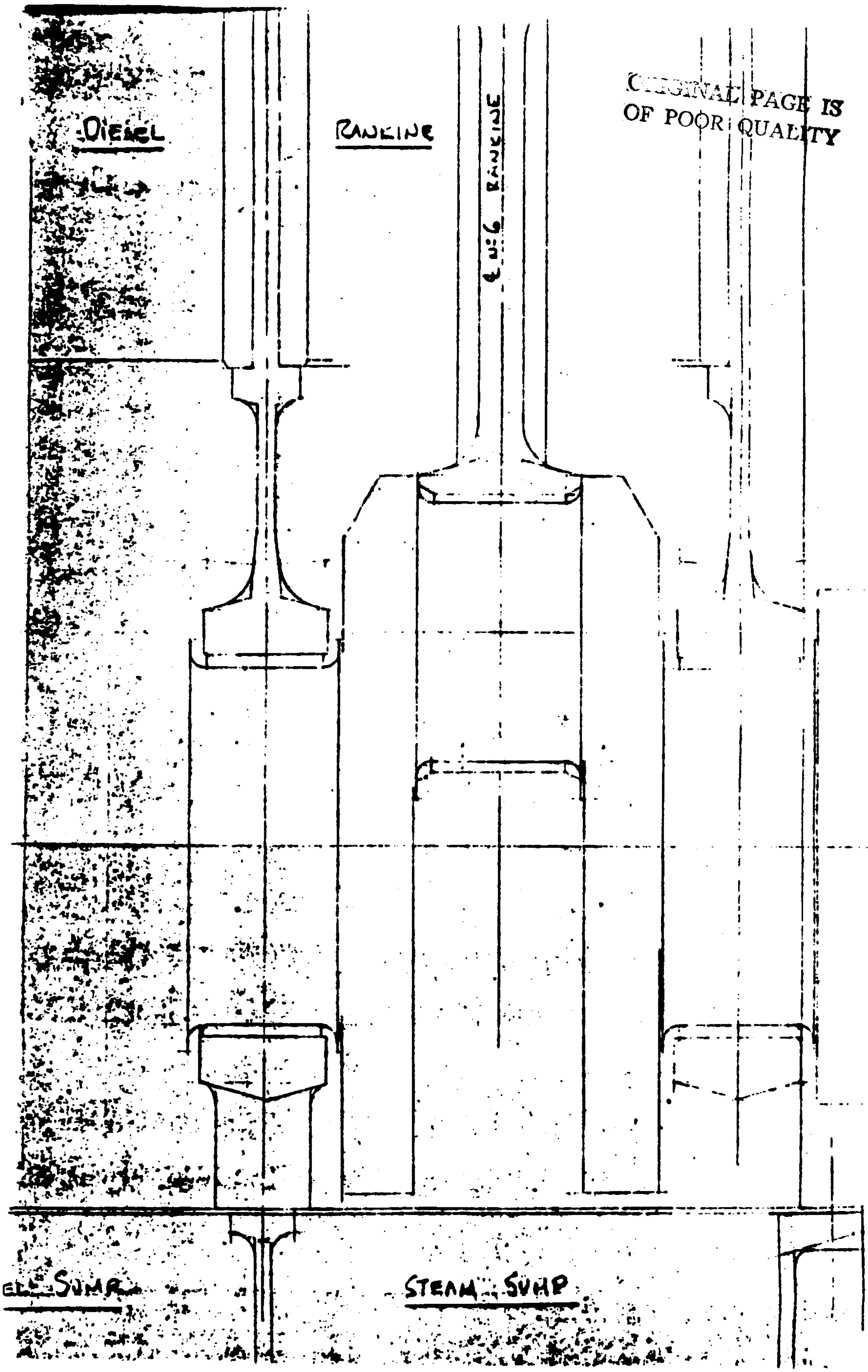


Figure 4-8. Crank/Sump Layout for the Steam Cylinder

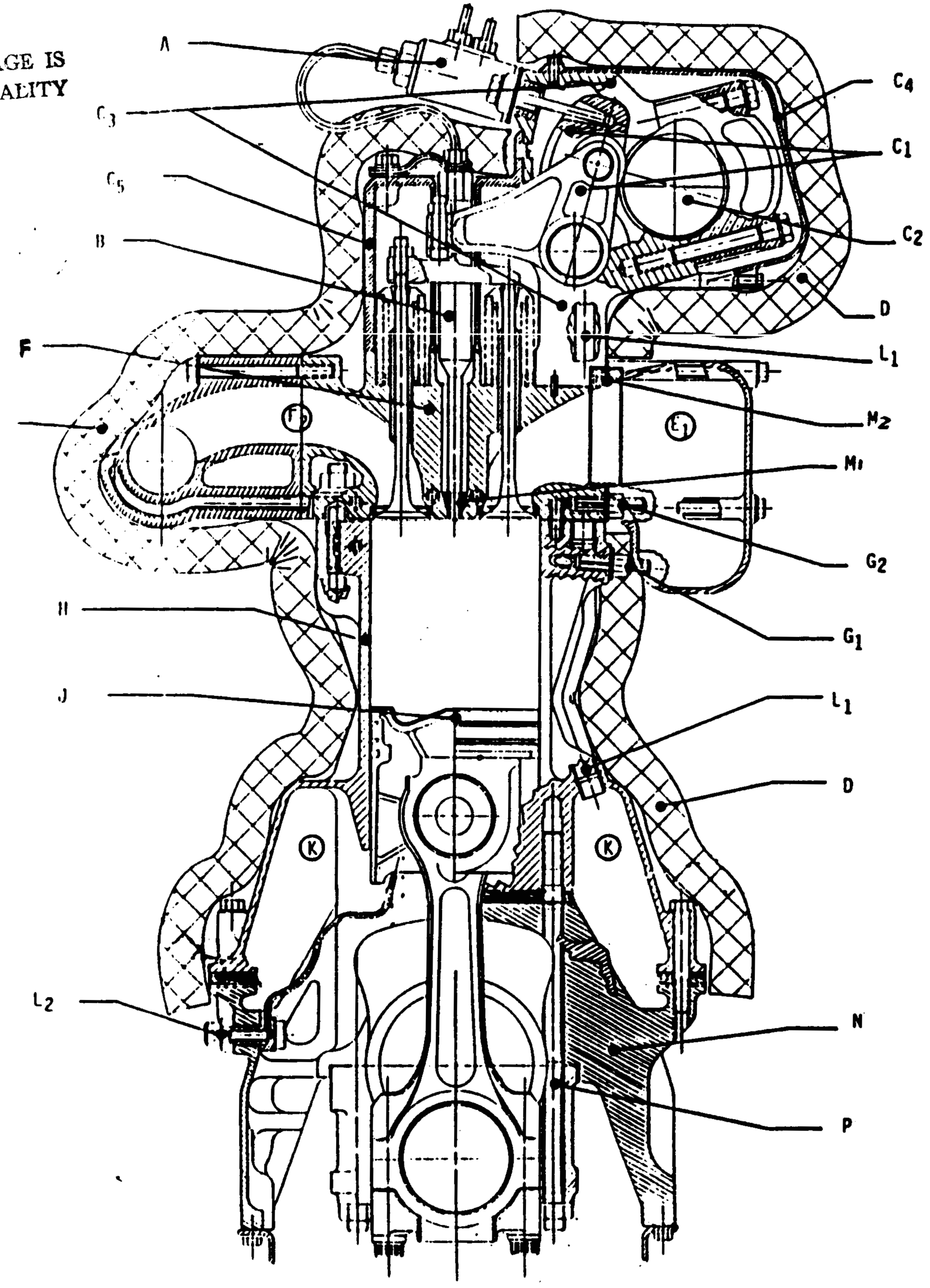


Figure 4-9. Cross Section of Diesel Reciprocator for Integral Bottoming Cycle Engine (See following page for corresponding design features)

Attachment to Figure 4-9.

DESCRIPTION OF DESIGN FEATURES FOR
INTEGRAL BOTTOMING CYLCLE ENGINE
(See sheet F for corresponding illustration)

A. Isolated Unit Pump Assembly. Isolated to reduce heat rejection from the head to fuel and for service. External to valvecover. 115 degree f fuel temp to injection nozzle. PT or ECI controled.

B. Fuel Nozzle. Isolated from head and valvetrain for reduced heat rejection. Cooled only by fuel used for combustion. Stanadyne type slim tip nozzle end features reduced tip area exposed to combustion chamber. Externally serviced. Surronded by an air gap, exposed to ambient.

C. Overhead Cam Valvetrain Assembly.
- C1. Pivoting valve and fuel injection rocker arms.
- C2. Overhead cam retained by split bearing inserts and bolt on caps.
- C3. Valve gear pedestal. Bolts to head as an assembly.
- C4. Cam cover
- C5. Valve cover, provides access to mechanism adjustments, locates fuel nozzle.

D. Thick insulativ blanket. Surrounds all areas used for heat recovery. Reduces heat loss by conduction, convection.

E. Engine Breathing System.
- E1. Isolated inlet charge plenum. Provides reduced heat rejection to charge air. Inlet head port area minimized.
- E2. Exhaust port/manifold retains pulse geometry and uses cast in high pressure steam/water tubes.

F. Iron Cylinder Head. Cooled only by boiling condensate in steel tube passages cast directly into head, and by incident valvetrain oil draining.

G. High Pressure Condensate Inlets. For increased strenght, the cast in steel tubes feature flared ends so that high pressure nipples may be threaded directly into the steel tubes.
- G1. Block Inlet. (1 per cylinder)
- G2. Head Inlet (1 per cylinder)

H. Engine Block. Simplified, incorporating linerless design, cooled by condensate boiling in cast in tubes in the top ring reversal area. Additional cooling provided by an underside piston/liner oil spray.

J. Zirconia spray coated ductile iron or CLAS cast steel piston. Cooling provided by forced oil mist on the underside. Convential 3 ring design.

K. Blow By and cylinder to cylinder breathing chambers. Also a colection point for high temperature oil to return to the scavenge pump.

L. High temperature lube oil system.
  L1. Valvetrain and head oil return tube.
  L2. Piston/Liner lube/cooling tube and connection.

M. Insulative Gasket System. Utilizes a steel and PSZ sandwich to isolate high and low temperature components.
  M1. Injector tip gasket. PSZ cone seals combustion pressure and isolates fuel cooled nozzle from cylinder head.
  M2. Inlet plenum gasket. PSZ or equivilent thermally isolates the cool inlet plenum and charge air (110 deg F) from the cylinder head.

N. Bed/Crankcase sump assembly. The bed assembly supports the crankshaft in the normal manor and forms the top half of the sump.

P. Main Cap/Bed/Block studs. For simplicity and improved alignment, a single set of studs is used to tie the main bearing cap, bed, and block together to form a single rigid unit.

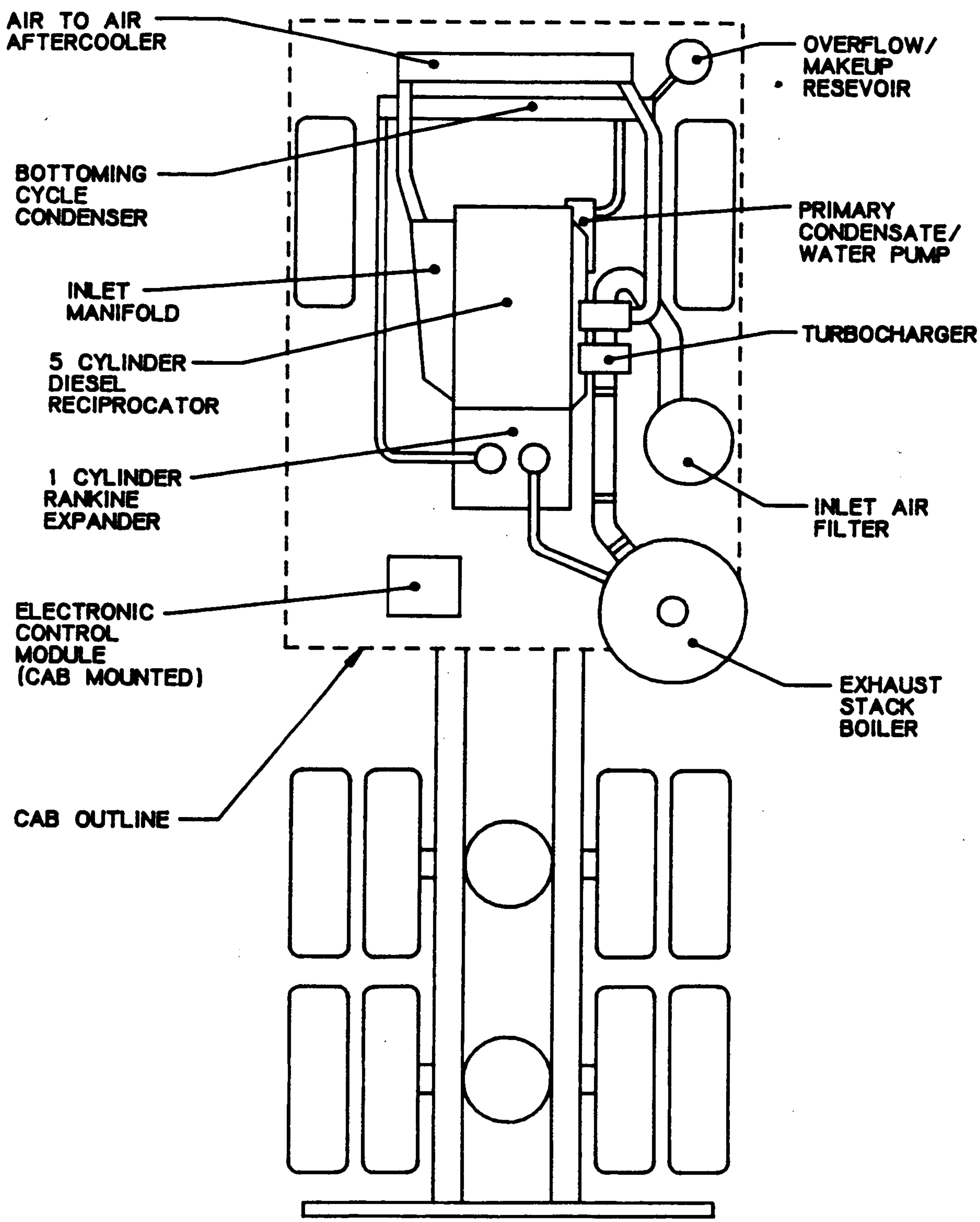


Figure 4-10. Schematic of Vehicle Mounted Equipment for Integrated Bottoming Cycle Engine (Overhead View)

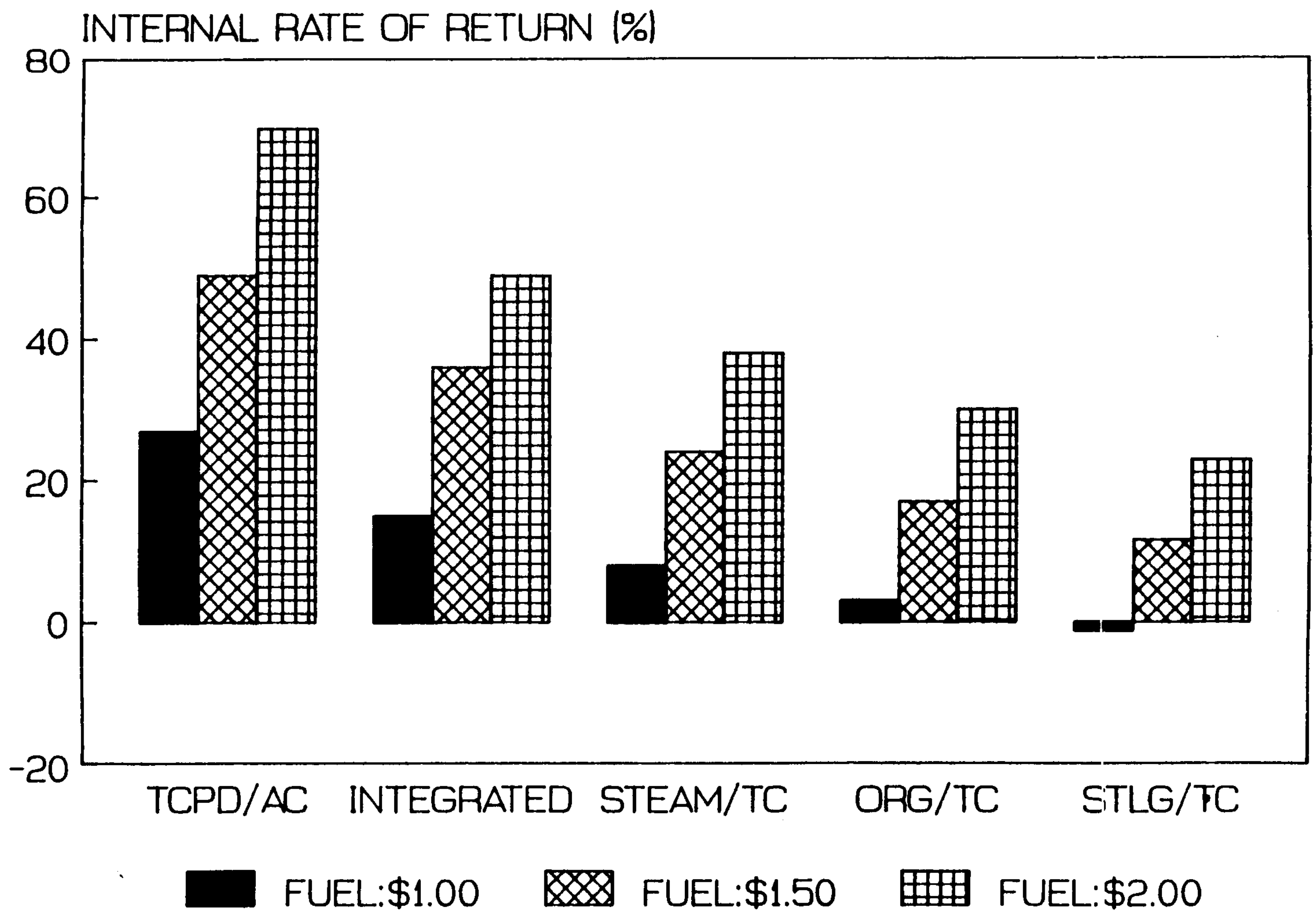


Figure 4-11. System Comparison - IRR of Bottoming Systems (Vs. TC/AC)

TABLE 2-1. MAINTENANCE SCHEDULE (HOURS)

	Configuration		
Description	Mod II	SAV-4	FPSE
Bearing Relubrication	-	5,000	-
Piston Seal	5,000	-	-
Piston Rings	5,000	5,000	-
Shaft Face Seals	5,000	2,000	-
Main Bearing Replacement	5,000	-	-
Change Oil	5,000	-	-
Flush Coolant	500	500	500
OK/Hydraulic Filter	10,000	-	10,000
Controls (External Check Valve)	10,000	10,000	-
H2 Supply	5,000	5,000	-

TABLE 2-2. MAINTENANCE SCHEDULE (COST)

	Mod II	SAV-4	FPSE
Bearing Relubrication			
Material		$3	
Labor*		Included	
Piston Seal			
Material	$20/Set of 4		
Labor*	Included		
Piston Rings			
Material	$28/Set of 4	$20/Set of 4	
Labor	Included	Included	
Shaft Face Seal			
Material	$10/Set of 2	$111/Set of 2	
Labor*	Included	$105/Set	
Main Bearing Replacement			
Material	$168/Set of 7		
Labor*	Included		
Oil Change			
Material	$5/4 Quarts		
Labor	$26		
Flush Coolant			
Material	$6/Gallon	$6/Gallon	$6/Gallon
Labor	$52	$52	$52
Oil/Hydrogen Filter			
Material	$6		$12
Labor	$18		$18
Controls			
Material	$8	$8	
Labor	$9	$9	
H2 Supply			
Material	$5	$5	
Labor	$17	$17	
Tear Down Engine			
Labor	$560 (16 hr)	$420 (11 hr)	

*Main Labor Cost is Included in Engine Tear-Down Cost
Production Rate of 10,000 Units/Year
Based on $35/hr Labor Rate

TABLE 2-3. MAINTENANCE SUMMARY

• Evaluated Over the Life (1400 Hours) of the Engine

MOD II	SAV-4	FPSE
This includes two complete changes of • Piston Seals • Piston Rings • Shaft Seals • Main Brgs • Oil • Hydrogen • Engine tear down plus one change of: • Oil filter • Control valve plus (3) coolant flushes	This includes two complete changes of: • BRG Lubrication • Piston Rings • Engine tear down plus one change of control valves Plus (7) changes of: • Shaft Seals • Hydrogen	Includes (3) changes of coolant and one change of Hydraulic Filter
MAT'L - $ 504 LABOR - $1,389 TOTAL - $1,893	MAT'L - $ 884 LABOR - $1,859 TOTAL - $2,743	MAT'L - $ 30 LABOR - $ 174 TOTAL - $ 204

TABLE 2-4. COMPARISON OF THE THREE ENGINE CONCEPTS

(BY MTI)

	MOD-II	SAV-4	FPSE
FUEL SAVINGS (%) *1	10	11	8
MANUFACTURING COST *2	$2,580	$1,940	$7,000
MAINTENANCE COST *3	$1,893	$2,743	$204
WEIGHT (LBS)	539	781	620
TECHNICAL RISK	LOW	HIGH	VERY HIGH
PACKAGABILITY	BEST	DIFFICULT	VERY DIFF
OTHER FACTORS	SMOOTH TORQ	ROUGH TORQ	HIGHLY COMPLEX
OVERALL	BEST CHOICE		

*1: BASED ON TURBOCHARGED CORE ENGINE AND OTHER PARAMETERS ARE
STIRLING EXHAUST GAS TEMPERATURE: 700'F
WORKING FLUID: HYDROGEN.
STIRLING ENGINE SPEED: 1,000 RPM
COOLER TEMPERATURE: 115'F

*2: ROUGH ESTIMATES BY MTI (10,000 UNITS PRODUCTION/YEAR)

*3: FOR A 7-YEAR LIFE TIME

TABLE 2-5. STIRLING BOTTOMING CYCLE PERFORMANCE MAP
TURBOCHARGED ADIABATIC DIESEL

Turbocharged Aftercooled

Diesel Engine Condition		Exhaust Gas				Without Bottoming Cycle		With Bottoming Cycle		
Speed	Load	Temperature (-F)	Mass Flow (lb/sec)	Heater Head Temperature	Stirling Net Power (hp)	hp	BSFC	hp	BSFC	% Saved
1900	Full	1240	48.1	832	31.1	317	0.315	348.1	0.286	9.0
	75%	1134	39.0	677	21.7	229	0.327	250.7	0.299	8.6
	50%	983	31.0	478	7.0	141	0.353	148.0	0.336	4.7
	0%	620	18.8	-	0	-	-	-	-	-
1300	Full	1375	32.5	943	27.3	247	0.317	247.3	0.285	9.9
	75%	1198	27.5	743	19.7	182	0.324	201.7	0.292	9.8
	50%	1000	22.8	518	9.5	114	0.343	123.5	0.316	7.7
	0%	620	13.7	-	0	-	-	-	-	-
700	0%	500	7.4	7	12.1*	-	-	-	-	-
OTHER ENGINE CONFIGURATIONS										
Turbocompound Aftercooled										
1900	Full	1060	48.4	683	22.1	380	0.293	362.1	0.275	6.1
Turbocharged Aftercooled										
1900	Full	1120	47.6	725	24.6	320	0.310	344.6	0.288	7.1
Turbocompound										
1900	Full	1140	47.8	742	23.02	335	0.297	25.7	0.276	7.1

TABLE 2-6 : HEAT BALANCE FOR TSA DIESEL ENGINE:

HEAT AND SHAFT POWER OUTPUT OF THE DIESEL ENGINE:

	kW
Conduction Losses (10% of Fuel Heat Input)	74.7
Diesel Engine Shaft Power (Exclusive of Friction)	357.5
Diesel Engine Friction Losses	11.1
Diesel Exhaust Heat to Stirling Engine	235.3
System Exhaust Loss (From 210 to 29.5 Degrees C)	67.2
Total Heat Out of Diesel Engine	747.4

HEAT INTO DIESEL ENGINE:

Calculated Input of Fuel Heat	602.7
Added Heat Input to Compensate for the Use of Extrapolated c_p- Values	144.7
Total Heat Flow Into Diesel Engine	747.4

HEAT CONVERSION IN STIRLING ENGINE:

Total Heat Flow Into Stirling Engine	235.3
Stirling Engine Shaft Power (Efficiency = 40%)	94.1

TABLE 2-7 **TSA DIESEL ENGINE (4 CYLINDER DIESEL & STIRLING)**
Simulation Results - Full Load Condition

DIESEL SHAFT POWER(kW)	DIESEL EFF. (%)	HEAT LOSS TO WALLS (%)	HEAT TO STIRLING (Kw)	STIRLING EFF. (%)	STIRLING POWER (Kw)	TOTAL POWER (Kw)	OVERALL EFF.(%)
235.9	37.8	11.6	201.1	40	80.4	316.3	50.7

TABLE 3-1. ECONOMIC/OPERATIONAL ASSUMPTIONS

- Tax
  . Corporate Tax Rate 34%
  . Investment Tax Credit 0%

- Equipment Price/Salvage Value/Life *1
  . Annual Production Rate .... 10,000 units
  . Selling Price/Mfg Cost 2.0
  . Future Cost Reduction (Learning Curve Effect) 30%
  . Salvage Value (% of Original) 20%
  . Hardware Useful Life .... 7 years

- Fuel Economy
  . Anuual Truck Mileage .... 100,000 miles
  . Diesel Oil Price $1.00/Gal *2
  . Base Engine (TCPD/AC) Truck Fuel Mileage .... 8.0 MPG *2

*1: Assumed a 10% penetration on a market of 100,000 class-8 trucks

*2: Sensitivity Analysis was made around this base case

TABLE 3-2. TRUCK MILEAGE IMPROVEMENT (%)

(VMS STUDY)

	RATED	HILLY[1]	FLAT[2]	MIX[3]
ORGANIC RANKINE	(15.0%)	13.9%	13.5%	13.7%
STEAM RANKINE	(14.1%)	13.6%	13.2%	13.3%
STIRLING[4]	(10.0%)	9.4%	8.9%	9.1%
BRAYTON	(11.1%)	10.6%	9.5%	10.3%

1 RENO----SACRAMENTO

2 INDIANAPOLIS----CHICAGO

3 COLUMBUS----LOUISVILLE----CINCINNATI----COLUMBUS

4 AUTOMOTIVE STIRLING ENGINE IS USED (FOR SIZE & COST), LARGER ONE WOULD GIVE 12.9% IMPROVEMENT AT RATED CONDITION.

TABLE 3-3. EXHAUST HEAT UTILIZATION

	ENERGY EXTRACTION η	POWER CONVERSION η	OVERALL UTILIZATION η
TURBOCOMPOUND	0.12	0.72	9%
BRAYTON CYCLE	0.85	0.15	13%
STIRLING CYCLE	0.37	0.30	11%
RANKINE CYCLE	0.88	0.21	18%

ENERGY EXTRACTION $\eta = \dfrac{T_1 - T_2}{T_1 - 300}$ T_1: GAS SOURCE, T_2: STACK

POWER CONVERSION $\eta = \dfrac{\text{SHAFT POWER}}{\text{HEAT INPUT}} = \dfrac{\text{SHAFT POWER}}{\dot{m}_g \cdot CP \cdot (T_1 - T_2)}$

TABLE 3-4.

MANUFACTURING COST OF BOTTOMING CYCLE SYSTEMS

MAJOR COMPONENTS	RANKINE		STIRLING
	ORGANIC	STEAM	
1. VAPOR GENERATOR OR HEATER HEAD	$1,200	$800	$534
2. EXPANDER/HOUSING	$863	$1,336	$939
3. CONDENSOR/REGENERATOR (INCL. FAN & OIL COOLER)	$690	$650	$734
4. POWER TRAIN/CLUTCH	$538	$470	$470
5. WORKING FLUID SYSTEM (PLUMBING, PUMPS, ETC.)	$1,030	$472	$146
6. CONTROL SYSTEM	$305	$305	$254
7. ASSEMBLY/ PRE-SHIPMENT TEST	$312	$166	$203
TOTAL	$4,938	$4,199	$3,258
OPTIMISTIC ESTIMATE	($3,457)	($2,939)	($2,281)

TABLE 3-5. ENGINE SYSTEM PRICE TO CUSTOMERS

(@ 350 HP level)

	Base	Rankine Steam	Rankine Organic	Stirling
Turbocharge (TC)	$15,005	$23,192 ($20,434)*	$24,670 ($21,469)	$21,210 ($19,047)
TC/AfterCool (TC/A)	$15,439	$23,692 ($20,934)	$25,170 ($21,969)	$21,710 ($19,547)
Turbocompound (TCPD)	$16,121	$24,826 ($22,068)	$26,304 ($23,103)	$22,844 ($20,681)
TCPD/Aft.Cool (TCPD/A)	$16,134	$25,326 ($22,568)	$26,804 ($23,603)	$23,344 ($21,181)

* Numbers in parenthesis indicate "optimistic" value.

- Price is based on the 0.7 power law (1)

$$\text{Price (350 HP)} = \text{Base Engine Price} * (350.0/\text{Combined System Output})^{0.7}$$

- The price includes the installation costs at OEM. Installation cost data are provided by Navistar

TABLE 3-6. A SUMMARY OF MANUFACTURING/MAINTENANCE COSTS

	STIRLING	RANKINE	
		ORGANIC	STEAM
MANUFACTURING COST	$3,258	$4,938	$4,199
	($1,789)*1	($4,189)*2	($3,035)*2
MAINTENANCE COST	$737	$1,034	$850
	($431)	(1,100)*2	($580)*2

*1: NUMBERS IN PARENTHESIS ARE ESTIMATES BY SUBCONTRACTORS FOR EACH BOTTOMING SYSTEM, I.E., MTI, TECO, AND FOSTER-MILLER

*2: FIGURES ARE BASED ON A 1983 $ VALUE.

TABLE 3-7. COMPETITIVE PRICE OF BOTTOMING SYSTEMS

	PRESENT ANALYSIS PRICE	COMPETITIVE W/ TCPD PRICE	REDUCTION (%)
RANKINE SYSTEM			
ORGANIC	$7,469	$4,368	41.5
STEAM	$6,434	$4,138	35.7
STIRLING SYSTEM	$5,047	$2,470	51.1

TABLE 4-1. FLUID CHARACTERISTICS

Identification	Average Molecular Weight	Max. Use Temp. °F	Flow or Freezing Point °F	I Factor	Atmospheric Boiling Point °F	Pressure at 220°F psia	Toxicity: inhalation	Toxicity: oral	Toxicity: skin	Fire Hazard	Explosion Hazard	Toxic Decomposition Products	Toxic Partial-Oxidation Products
Fluorinol 85 (85 mole % TFE/15% water)	87.74	550	-82	1.26	168-169	42	high	high				F^-,HF	CO,COF
toluene	92.13	650-750	-139	0.66	231	12.3	mod	low	low	slight	mod		CO
2-methylpyridine/ water (25 mole % 2MP/ 65 mole % water)	44.3	575-670	-40	1.38	200	21	mod	mod	mod	mod-high		HCN	NO_x,CO
RC-1 (60 mole % penta-fluoro-benzene/40 mole % hexafluorobenzene)	175.3	750?	-44	0.72	172	30	low					F^-,HG	COF
50 vol. % ethylene glycol/50% water	40.05	unknown	-30	>1	225	13		mod	mod	slight	mod		CO
50 wt % methanol/50% water	20.04	unknown	-30	2.16 for 100% methanol			mod	high	mod	dangerous	mod		CO
water	18.02	1050	32	2.81	212	17.2	none	none	none	none	none	none	none
benzene	78.1	800?	42	0.89	176	29.4	—[high, carcinogen]—			dangerous	mod		CO
Flutec PP3, $(CF_3)_2C_6F_{10}$	400.08	700?	-67									F^-	COF
monochlorobenzene	112.5	630?	-50	0.74	269		low	mod		dangerous	mod	Cl^-,HCl	CO,COCl
thiophene, C_4H_4S	84.14	550?	-37	1.35 (depends on temp.)	183		mod			dangerous			CO,SO_2
isobutane	58.12	450?	-255		11	313	low			very dangerous	severe		CO
R-113, CCl_2F-$CClF_2$	187.39	300	-31	0.69	118	70						Cl^-,F^-	COCl,COF

TABLE 4-2. RESULTS OF THE CYCLE ANALYSIS FOR THE DIFFERENT CASES

	Working Fluid	Operating Pressure	Cycle Efficiency	Flow Rate lb_m/min	Expander Power H.P.
1.	Superheated Steam	1000	19.8	5.98	34.9
2.	Superheated Steam	500	19.5	6.10	34.4
3.	Superheated Toluene with Regeneration	500	18.0	32.40	32.1
4.	Superheated Toluene without Regeneration	500	13.6	24.30	24.1

TABLE 4-3. CALCULATION OF EXPANDER DISPLACEMENT

Specific volume of steam at exapnder outlet = 16.44 $\frac{\text{c. ft}}{\text{lb}}$

Steam flow rate = 6 lb/min.

= 6 x 16.44 $\frac{\text{c.ft}}{\text{min}}$ 98.64 $\frac{\text{c.ft}}{\text{min}}$

If the engine is reated at 1900 rpm,
Steam flow rate = $\frac{\text{98.64 c.ft}}{\text{1900 rev}}$ 0.0519 $\frac{\text{c.ft}}{\text{rev.}}$

One exh. stroke/rev. displacement = 0.0519 $\frac{\text{c.ft}}{\text{stroke}}$

= 89.68 c.in 90 c.in
Vol. 0.9, Displacement = 100 c.in

TABLE 4-4. FUEL CONSUMPTION COMPARISON WITH AND WITHOUT INTEGRATED BOTTOMING CYCLE SYSTEM FOR VARIOUS ENGINE CONFIGURATIONS

ENGINE CONFIGURATION	BHP	BSFC (LB/BHP-HR)	EXHAUST (°F)	EXHAUST (LB/MIN)
TURBOCHARGED-NONAFTERCOOLED (TC)	317	0.315	1240	48.1
SAME WITH INTEGRATED BOTTOMING CYCLE (TC+BC)	305	0.273	700	40.1
TURBOCHARGED-AFTERCOOLED (TC/A)	320	0.310	1120	47.6
SAME WITH INTEGRATED BOTTOMING CYCLE (TC/A+BC)	304	0.272	700	39.7
TURBOCOMPOUND-NONAFTERCOOLED (TCPD)	335	0.297	1140	47.8
SAME WITH INTEGRATED BOTTOMING CYCLE (TCPD+BC)	317	0.262	700	39.8
TURBOCOMPOUND-AFTERCOOLED (TCPD/A)	340	0.293	1060	48.4
SAME WITH INTEGRATED BOTTOMING CYCLE (TCPD/A+BC)	320	0.260	700	40.3

INTEGRATED BOTTOMING CYCLE ENGINES USE
5 DIESEL CYLINDERS AND 1 RANKINE CYLINDER

TABLE 4.5. MANUFACTURING COST OF STEAM RANKINE BOTTOMING CYCLE SYSTEMS

MAJOR COMPONENT	CONVENTIONAL	INTEGRATED	(@350HP)
1. VAPOR GENERATOR	$800	$700	$772
ENGINE MODIFICATION		$353	$389
2. EXPANDER/HOUSING	$1,336	$326	$360
3. CONDENSOR (INCL. FAN $ OIL COOLER)	$650	$307	$339
4. POWER TRAIN/CLUTCH	$470	$0	$0
5. WORKING FLUID SYSTEM (PLUMBING, PUMPS, ETC.)	$472	$315	$347
6. CONTROL SYSTEM	$305	$125	$138
7. ASSEMBLY/ PRESHIPMENT TEST	$166	$50	$55
	$4,199	$2,176	$2,400

APPENDIX 1

M.T.I.

COST STUDY OF STIRLING

BOTTOMING CYCLE SYSTEM

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PART NO.	DESCRIPTION	MIW BOF	QTY. REQD	LABOR COST @ $73/HOUR	MATERIAL COST	TOTAL MATERIAL & LABOR COST	PATTERN COST	MATERIAL TYPE	REMARKS
HEATER HEAD	HEATER HEAD	MIW	4	$109.08	$44.60	$153.68	$20,000.00		
"	TUBES HEATER HEAD	MIW	120	$16.00	$33.31	$49.31			
"	FINS HEATER HEAD	MIW	10150	$37.27	$81.20	$118.47			
"	ASSEMBLY HEATER HEAD	MIW	4	$45.09	$0.90	$45.99			
"	SUB-TOTAL					$367.45			
DUCTING HTR. HD. COVER ASSEMBLY	DUCTING/HEATER HEAD COVER ASSEMBLY								
"	TOP COVER-OUTER	MIW	1	$1.82	$1.90	$3.72			
"	TOP COVER-INNER	MIW	1	$1.82	$1.66	$3.48			
"	ASSEMBLY TOP COVER	MIW	1	$21.45	$2.54	$23.99			
	SUB-TOTAL					$31.19			

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M.T.I. STIRLING

DATE: 05-01-1986

PART NO.	DESCRIPTION	MIW BOF	QTY. REQD	LABOR COST @ $73 HOUR	MATERIAL COST	TOTAL MATERIAL & LABOR COST	PATTERN COST	MATERIAL TYPE	REMARKS
ASSEMBLY ENCLOSURE WITH OUTLET DUCT	ASSEMBLY - ENCLOSURE WITH OUTLET DUCT								
	ENCLOSURE OUTER	MIW	2	$10.54	$3.36	$13.90			
"	ENCLOSURE INNER	MIW	2	$10.54	$2.92	$13.46			
"	DUCT OUTLET	MIW	2	$2.36	$0.58	$2.94			
"	ASSEMBLY ENCLOSURE WITH OUTLET DUCTS	MIW	2	$61.45	$5.94	$67.39			
	SUB-TOTAL					$97.69			
ENCLOSURE COVER	ENCLOSURE COVER	MIW	2	$1.82	$1.12	$2.94			
ASSEMBLY PLENUM WITH INLET DUCT	ASSEMBLY - PLENUM WITH INLET DUCT								
	PLENUM - OUTER	MIW	1	$1.82	$1.91	$3.73			
"	PLENUM - INLET	MIW	1	$1.82	$1.91	$3.73			
"	PLENUM - COVER	MIW	1	$0.91	$0.62	$1.53			
"	DUCT - INLET INSULATED	BOF	1		$8.00	$8.00			
"	ASSEMBLY - PLENUM WITH DUCT		1	$15.72	$1.73	$17.45			
	SUB-TOTAL					$34.44			
CLAMPS OTHER	CLAMPS / OTHER				$3.00	$3.00			

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DATE: 05-01-1986

PART NO.	DESCRIPTION	MIW BOF	QTY. REQD	LABOR COST @ $73 HOUR	MATERIAL COST	TOTAL MATERIAL & LABOR COST	PATTERN COST	MATERIAL TYPE	REMARKS:
DRIVE SYSTEM	DRIVE SYSTEM								
"	TORSION ISOLATOR ASSEMBLY	BOF	1		$77.60	$77.60			
"	STOCK SPROCKET 1/2 PITCH	BOF	1		$32.40	$32.40			
"	STANDARD CHAIN 1/2 PITCH	BOF	1		$35.00	$35.00			
	CLUTCH (PITTS MODEL I-28 P/N 10536)	BOF	1		$200.00	$200.00			
	SUB-TOTAL					$345.00			
RADIATOR SYSTEM	RADIATOR SYSTEM				$250.00	$250.00			
"	RADIATOR CORE	BOF	1						
"	SHROUD	BOF	1						
"	HOSES & CLAMPS	BOF							
"	FAN	BOF	1						
"	ELECTRIC MOTOR BRACKETS & FAN SHROUD								
"	**ASSEMBLY RADIATOR SYSTEM**		1						
	SUB-TOTAL					$250.00			
MOUNTING PLATE	MOUNTING PLATE	MIW	1	$64.27		$64.27			
ENGINE MOUNTING	ENGINE MOUNTING	MIW		$54.55		$54.55			

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M.T.I. STIRLING

DATE: 05-01-1986

PART NO.	DESCRIPTION	MIW BOF	QTY. REQD	LABOR COST @ $73 HOUR	MATERIAL COST	TOTAL MATERIAL & LABOR COST	PATTERN COST	MATERIAL TYPE	REMARKS:
CONTROL BLOCK ASSEMBLY MODIFIED	CONTROL BLOCK ASSEMBLY MODIFIED	BOF	1		$87.80	$87.80			
ELECT PACKAGE	ELECTRONIC PACKAGE MODIFIED	BOF	1		$76.37	$76.37			
ROTARY CONTROL VALVE	ROTARY CONTROL VALVE - MODIFIED	BOF	1		$39.91	$39.91			
TRANS. PRESSURE	TRANSDUCER - PRESSURE (ONE PIECE)	BOF	1		$49.67	$49.67			
HYDROGEN STORAGE	HYDROGEN STORAGE SYSTEM	BOF	1		$59.61	$59.61			
ENGINE BLOCK	ENGINE BLOCK MACHINING	MIW	1	$73.00	$95.00	$168.00	$[illegible]5,000.00		
ASSEMBLY	HOUSING REAR MAIN BEARING CAP & OIL PUMP	MIW	1	$4.00	$2.50	$6.50	$2,500.00		
"	BEARING CAP MAIN	MIW	2	$17.52	$6.00	$23.52	$3,500.00		
"	PIN, DOWEL	BOF	6		$1.00	$1.00			
"	SCREW, SOCKET HEAD M12 X 1.75 X 45 GR 8.8	BOF	5		$3.50	$3.50			
"	SCREW, SOCKET HEAD M12 X 1.75 X 75 GR 8.8	BOF	1		$0.70	$0.70			
"	SCREW, SOCKET HEAD SET FLAT POINT M12 X 1.75 X 12 GR. 8.8	BOF	3		$1.20	$1.20			

M.T.I. STIRLING DATE: 05-01-1986

PART NO.	DESCRIPTION	MIW BOF	QTY. REQD	LABOR COST @ $73 HOUR	MATERIAL COST	TOTAL MATERIAL & LABOR COST	PATTERN COST	MATERIAL TYPE	REMARKS:
ENGINE	BEARING - BOSTON GEAR	BOF	1		$6.00	$6.00			
BLOCK									
ASSEMBLY	SCREW, SOCKET HEAD SET	BOF	1		$0.50	$0.50			
CONT'D	FLAT POINT M10 X 1.5 X 10								
"	GR. 8.8								
"									
"	WASHER, LOCKING M12	BOF	6		$0.12	$0.12			
	SUB-TOTAL					$211.04			
FRONT	FRONT COVER	MIW	1	$21.90	$15.00	$36.90	$4,500.00		
COVER									
ASSEMBLY	LEE PLUS (SHORT) - ALUMINUM	BOF	1		$0.30	$0.30			
"									
"	HELICOIL INSERT	BOF	4		$2.80	$2.80			
"	M 6 X 1 X 9								
"									
"	HELICOIL INSERT	BOF	4		$3.75	$3.75			
"	M10 X 1.5 X 20								
"									
"	HELICOIL INSERT	BOF	1		$0.70	$0.70			
"	M10 X 1.5 X 15								
"									
"	BEARING, NEEDLE ROLLER	BOF	1		$4.00	$4.00			
"	TORRINGTON #FJ-1512								
	SUB-TOTAL					$48.45			
CRANK-	CRANKSHAFT	MIW	1	$36.50	$30.00	$66.50	$10,000.00		
SHAFT									
BALANCING	WATER PUMP DRIVE GEAR	BOF	1		$2.00	$2.00			
ASSEMBLY									
"	OIL PUMP DRIVE GEAR	BOF	1		$6.00	$6.00			
"									
"	DRIVE HUB	MIW	1	$7.30	$5.00	$12.30	$2,500.00		

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M.T.I. STIRLING

DATE: 05-01-1986

PART NO.	DESCRIPTION	MIW BOF	QTY. REQD	LABOR COST @ $73 HOUR	MATERIAL COST	TOTAL MATERIAL & LABOR COST	PATTERN COST	MATERIAL TYPE	REMARKS:
CRANK-SHAFT	NUT, RETAINING	BOF	1		$0.50	$0.50			
BALANCING ASSEMBLY	BALANCE WEIGHT - GEAR	MIW	1	$7.30	$2.50	$9.80	$2,500.00		
CONT'D	BALANCE WEIGHT - POWER PISTON CRANK	MIW	4	$21.90	$8.00	$29.90			
"	SPACER	MIW	1	$2.19	$2.50	$4.69			
"	SCREW, FLAT SOC HD MACHINE M 4 X .7 X 20 GR.10.9	BOF	3		$1.20	$1.20			
"	SCREW, FLAT SOC HD MACHINE M 4 X .7 X 10 GR.10.9	BOF	11		$5.50	$5.50			
"	CRANKSHAFT ASSEMBLY	MIW	1	$12.17		$12.17			

	SUB-TOTAL					$150.56			
WATER PUMP ASSEMBLY	WATER PUMP ASSEMBLY	BOF	1		$60.00	$60.00			
"	BACK WATER PUMP HOUSING	BOF	1						
"	FRONT WATER PUMP HOUSING	BOF	1						
"	WATER PUMP SHAFT	BOF	1						
"	WATER PUMP DRIVE GEAR	BOF	1						
"	WATER PUMP INBOARD THRUST WASHER	BOF	1						
"	WATER PUMP OUTBOARD THRUST WASHER	BOF	1						
"	ASSY. - GEROTOR UNIT ULTEM 400,410SS 1.5"THICK	BOF	1						
"	VALVE,RELIEF - 1/4"MALE NPT	BOF	1						
"	BEARING - DRAWN CUP ROLLER CLUTCH	BOF	1						

M.T.I. STIRLING

DATE: 05-01-1986

PART NO.	DESCRIPTION	MIW BOF	QTY. REQD	LABOR COST @ $73 HOUR	MATERIAL COST	TOTAL MATERIAL & LABOR COST	PATTERN COST	MATERIAL TYPE	REMARKS:
WATER PUMP ASSEMBLY CONT'D	BEARING, NEEDLE ROLLER	BOF	1						
"	BEARING, DEEP GROOVE BALL M12 X 28 X 8 FAG#6001.2RSR	BOF	1						
"	SEAL, FACE SHAFT TYPE 792 3/4"ID SERIES 300 CUPS&HOUSINGS	BOF	2						
"	SEAL, SHAFT LIP TYPE DRW INDUSTRIES #6420-CRW1-R	BOF	1						
"	O-RING PARKER #2-124N674-70 'BUNA-N'	BOF	2						
"	RING, SNAP - BEVELED TRUARC #N5002-106	BOF	1						
"	RING, SNAP - BOWED TRUARC #5101-46	BOF	1						
"	KEY, WOODRUFF 3/16 X 1.0" NO.608	BOF	1						
"	ASSEMBLY - ELECTRIC MOTOR - PMI MOTORS TYPE #12FP ITEMS 23-31	BOF	1						
"	SCREW, SOC HD CAP M 8 X 1.25 X 55 GR.8.8	BOF	2						
"	WASHER, FLAT M 9	BOF	4						
"	NUT, HEX M 8 X 1.25 GR.8.8 #934	BOF	4						
"	WATER PUMP GEAR HUB	MIW	1						
"	WATER PUMP BALANCE WEIGHT	MIW	1						
"	SCREW, SOC HD CAP M 5 X .8 X 10 GR.10.9	BOF	6						

M.T.I. STIRLING

DATE: 05-01-1986

PART NO.	DESCRIPTION	MIN BOF	QTY. REQD	LABOR COST @ $73 HOUR	MATERIAL COST	TOTAL MATERIAL & LABOR COST	PATTERN COST	MATERIAL TYPE	REMARKS:
WATER PUMP ASSEMBLY CONT'D	SCREW, HEX HD CAP M 5 X .8 X 25 GR.12.9	BOF	3						
"	SCREW, HEX HD CAP M 4 X .7 X 12 GR.8.8	BOF	1						
"	SCREW, HEX HD CAP M 5 X .8 X 6 GR.8.8	BOF	1						
"	SCREW, SLOTTED FLAT HD M 5 X .8 X 10 GR.8.8	BOF	3						
"	SCREW, PAN HD MACHINE M 5 X .8 X 10 #85BR	BOF	4						
"	SCREW,SHOULDERED SOC HD CAP M 8 X 30 GR.12.9	BOF	2						
"	SCREW,SHOULDERED SOC HD CAP M 8 X 40 GR.12.9	BOF	2						
	SUB-TOTAL					$60.00			
ASSEMBLY-MAIN SEAL HOUSING-CYL.2&4	MAIN SEAL HOUSING SEAT	BOF	1						
"	ASSEMBLY - MAIN PL SEAL	BOF	1						
"	MAIN SEAL HOUSING SPRING	BOF	1						
"	MAIN SEAL HOUSING CARRIER	BOF	1						
"	MAIN SEAL HOUSING FOLLOWER	BOF	1						
"	MAIN SEAL HOUSING CAPSEAL	BOF	1						
"	MAIN SEAL HOUSING INJECTION BUSHING	BOF	1						
"	MAIN SEAL HOUSING BACKUP WASHER	BOF	1						

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M.T.I. STIRLING

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PART NO.	DESCRIPTION	MIW BOF	QTY. REQD	LABOR COST @ $73 HOUR	MATERIAL COST	TOTAL MATERIAL & LABOR COST	PATTERN COST	MATERIAL TYPE	REMARKS:
ASSEMBLY-	COVER - MAIN SEAL HOUSING	BOF	2						
MAIN	ALL CYLINDERS								
SEAL									
HOUSING-	ASSY - MAIN SEAL HOUSING	BOF	2	$0.00	$25.00	$25.00			
CYL.2&4	CYL. 2 & 4								
CONT'D									
"	SCREW, BUTTON SOC HD CAP	BOF	6						
"	M 5 X .8 X 14								
"									
"	O-RING PARKER #2-198V646-75	BOF	2						
"									
"	O-RING PARKER #2-115V747-75	BOF	2						
"									
"	O-RING PARKER #2-122V747-75	BOF	4						
"									
"	O-RING	BOF	4						
"	APPLE #6.73 X 1.52 - 70								
"									
"	OIL JET - MAIN PL SEAL	BOF	4						
"									
"	BACK UP RING - PARKER	BOF	2						
"	PARBAK #8-122N300-90								
	SUB-TOTAL					$25.00			

PART NO.	DESCRIPTION	MIW BOF	QTY. REQD	LABOR COST @ $73 HOUR	MATERIAL COST	TOTAL MATERIAL & LABOR COST	PATTERN COST	MATERIAL TYPE	REMARKS:
ASSEMBLY-	MAIN SEAL HOUSING SEAT	BOF	2				$2,500.00		
MAIN									
SEAL	ASSY. - MAIN PL SEAL	BOF	2						
HOUSING-									
CYL.1&3	MAIN SEAL HOUSING SPRING	BOF	2						
"									
"	MAIN SEAL HOUSING CARRIER	BOF	2				$3,500.00		
"									
"	MAIN SEAL HOUSING FOLLOWER	BOF	2				$3,000.00		
"									
"	MAIN SEAL HOUSING CAPSEAL	BOF	2						
"									
"	MAIN SEAL HOUSING INJECTION	BOF	2						
"	BUSHING								

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M.T.I. STIRLING

DATE: 05-01-1986

PART NO.	DESCRIPTION	MIW BOF	QTY. REQD	LABOR COST @ $73 HOUR	MATERIAL COST	TOTAL MATERIAL & LABOR COST	PATTERN COST	MATERIAL TYPE	REMARKS:
ASSEMBLY-MAIN SEAL HOUSING-CYL.1&3 CONT'D	MAIN SEAL HOUSING BACKUP WASHER	BOF	2						
	COVER - MAIN SEAL HOUSING - ALL CYLINDERS	BOF	2						
"	MAIN SEAL HOUSING CYL. 1&3 ASSEMBLY	BOF	2	$0.00	$25.00	$25.00			
"	SCREW, BUTTON SOC HD CAP M 5 X .8 X 14	BOF	6						
"	O-RING PARKER #2-109V747-75	BOF	2						
"	O-RING PARKER #2-115V747-75	BOF	2						
"	O-RING PARKER #2-122V747-75	BOF	4						
"	O-RING APPLE #6.73 X 1.52 - 70	BOF	2						
"	OIL JET - MAIN PL SEAL	BOF	4						
"	BACK UP RING - PARKER PARBAK #8-122N300-90	BOF	2						
	SUB-TOTAL					$25.00			
OIL PUMP ASSEMBLY	HOUSING, REAR MAIN BEARING CAP & OIL PUMP	BOF	1						
"	COUNTERWEIGHT	BOF	1						
"	GEROTOR UNIT NO 4113-0.6875 CLASS 111	BOF	1						
"	DRIVE GEAR	BOF	1						
"	DRIVE SHAFT	BOF	1						
"	KEY, WOODRUFF MFG. #3	BOF	1						

M.T.I. STIRLING

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PART NO.	DESCRIPTION	MIW BOF	QTY. REQD	LABOR COST @ $73 HOUR	MATERIAL COST	TOTAL MATERIAL & LABOR COST	PATTERN COST	MATERIAL TYPE	REMARKS:
OIL PUMP	COVER, PLATE	BOF	1						
ASSEMBLY	GEAR COVER PLATE	BOF	1						
"	SCREEN	BOF	1						
"	SCREW, FLAT SOCKET HD M 4 X .7 X 10 GR. 8.8	BOF	8						
"	SCREW, BUTTON SOC HD CAP M 4 X .7 X 6 GR 8.8	BOF	8						
"	PIN, ROLL M 4 X 16 NO 1481	BOF	1						
"	OIL PUMP ASSEMBLY	BOF	1	$0.00	$40.00	$40.00			
	SUB-TOTAL					$40.00			
ASSEMBLY CHECK VALVE BODY CYL 2 & 3	CHECK VALVE BODY CYL 2 & 3	BOF	1		$12.00	$12.00			
	ASSY. CHECK VALVE STRAIGHT FLOW	BOF	2						
"	ASSY. CHECK VALVE NINETY DEGREE FLOW	BOF	4						
"	PLUG	BOF	4						
"	PLUG M 10 X 1 - TEMETO AB	BOF	8						
"	ORIFICE	BOF	2						
"	O-RING M 17.10 X 1.60 VITON 70 DURO	BOF	4						
"	O-RING M 4.00 X 1.80 VITON 70 DURO	BOF	2						
	SUB-TOTAL					$12.00			

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DATE: 05-01-1986

PART NO.	DESCRIPTION	MIN BOF	QTY. REQD	LABOR COST @ $73 HOUR	MATERIAL COST	TOTAL MATERIAL & LABOR COST	PATTERN COST	MATERIAL TYPE	REMARKS:
ASSEMBLY CHECK VALVE BODY CYL 1 & 4	CHECK VALVE BODY	BOF	1		$12.00	$12.00			
	ASSY CHECK VALVE STRAIGHT FLOW	BOF	2						
"	ASSY CHECK VALVE NINETY DEGREE FLOW	BOF	4						
"	PLUG	BOF	4						
"	PLUG M 10 X 1 TEMETO AB	BOF	8						
"	ORIFICE	BOF	2						
"	O-RING M 17.10 X 1.60 VITON 70 DURO	BOF	4						
"	O-RING M 4.00 X 1.8 VITON 70 DURO	BOF	2						
	SUB-TOTAL					$12.00			

PART NO.	DESCRIPTION	MIN BOF	QTY. REQD	LABOR COST @ $73 HOUR	MATERIAL COST	TOTAL MATERIAL & LABOR COST	PATTERN COST	MATERIAL TYPE	REMARKS:
ASSEMBLY PORT SEAL	TUBE PORT SEAL	BOF	1		$4.00	$4.00			
	ELASTOMER - PORT SEAL	BOF	2						
"	SPACER - PORT SEAL	BOF	1						
"	WASHER - PORT SEAL	BOF	1						
"	NUT - PORT SEAL	BOF	1						
"	ASSEMBLY PORT SEAL	BOF	1						
	SUB-TOTAL					$4.00			

M.T.I STIRLING

DATE: 05-01-1986

PART NO.	DESCRIPTION	MIW BOF	QTY. REQD	LABOR COST @ $73 HOUR	MATERIAL COST	TOTAL MATERIAL & LABOR COST	PATTERN COST	MATERIAL TYPE	REMARKS:
ASSEMBLY PORT SEAL	TUBE - PORT SEAL	BOF	1		$4.00	$4.00			
SEAL LEAKAGE	ELASTOMER - PORT SEAL	BOF	1						
"	WASHER - PORT SEAL	BOF	1						
"	NUT - PORT SEAL	BOF	1						
"	ASSEMBLY PORT SEAL	BOF	1						

	SUB-TOTAL					$4.00			
COOLER ASSEMBLY	MACHINING - COOLER FINAL	MIW	4	$46.55	$80.72	$127.27			
"	FLOW BLOCKER	MIW	8						
"	COOLER CYLINDER	MIW	4						
"	COOLER SHELL SET	MIW	4						
"	TUBE - COOLER	MIW	1392						

	SUB-TOTAL					$127.27			
REGENER-ATOR ASSEMBLY	REGENERATOR PARTITION WALL	MIW	4	$110.80	$181.48	$292.28			
"	REGENERATOR STUFFER	MIW	4						
"	REGENERATOR SEAL RING	MIW	4						
"	REGENERATOR MATRIX	MIW	4						

	SUB-TOTAL					$292.28			

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M.T.I STIRLING

DATE: 05-01-1986

PART NO.	DESCRIPTION	MIW BOF	QTY. REQD	LABOR COST @ $73 HOUR	MATERIAL COST	TOTAL MATERIAL & LABOR COST	PATTERN COST	MATERIAL TYPE	REMARKS:
ASSEMBLY-POWER PISTON CONN. ROD	POWER PISTON CONNECTING ROD	MIW	4	$48.68	$24.00	$72.68	$7,500.00		
	POWER PISTON CONNECTING ROD BOLT	BOF	8		$8.00	$8.00			
"	NUT, HEX - POWER PISTON CONNECTING ROD	MIW	8		$2.00	$2.00			
"	POWER PISTON CONNECTING ROD FINAL MACHINING	MIW	4	$14.60		$14.60			
"	POWER PISTON CROSSHEAD	MIW	4	$7.30	$4.00	$11.30	$2,500.00		
"	WRIST PIN	MIW	4		$8.00	$8.00			

	SUB-TOTAL					$116.58			
ASSEMBLY-PISTON MACHINING & RIDER RING	WELDMENT - PISTON - VENTED RINGS	MIW	4	$14.60		$14.60			
	RULON LD 7.84-7.85 WIDE X 0.8 THK X 225.6 LG	BOF	8		$8.00	$8.00			
"	ASSEMBLY - PISTON MACHINING AND RIDER RING	MIW	4	$43.80	[illegible]	$83.80	$5,000.00		
"	PISTON RING - SOLID	BOF	8		$8.00	$8.00			
"	PISTON RING - SPLIT	BOF	8		$12.00	$12.00			
"	EXPANDER RING	BOF	8		$4.00	$4.00			
"	ASSEMBLY - PISTON ROD/BASE VENTED RINGS	MIW	4	$9.75		$9.75			
"	ASSEMBLY - RADIATION SHIELD	MIW	4						
"	PISTON DOME	MIW	4		$40.00	$40.00			

	SUB-TOTAL					$180.15			

M.T.I STIRLING

DATE: 05-01-1986

PART NO.	DESCRIPTION	MIW BOF	QTY. REQD	LABOR COST @ $73 HOUR	MATERIAL COST	TOTAL MATERIAL & LABOR COST	PATTERN COST	MATERIAL TYPE	REMARKS:
	ASSEMBLE UNIT COMPLETE	MIW	1	$182.50		$182.50			
	TEST UNIT	MIW	1	$109.50		$109.50			
OIL PAN ASSEMBLY	OIL PAN	MIW	1	$14.60	$10.00	$24.60	$10,000.00		
"	OIL PAN GASKET	BOF	1		$2.50	$2.50			
"	CAPSCREW	BOF	14		$1.40	$1.40			
"	WASHER, FLAT	BOF	14		$0.42	$0.42			
"	WASHER, LOCK	BOF	14		$0.28	$0.28			
"	PLUG, OIL PAN	BOF	1		$1.50	$1.50			
"	GASKET, PLUG	BOF	1		$0.20	$0.20			
	SUB-TOTAL					$30.90			
	GRAND TOTAL			$1,264.79	$1,930.33	$3,195.12	$134,500.00		
	PREPARED BY: ROBERT L. LINNEY								

THERMO ELECTRON CORP.
COST STUDY OF ORGANIC
RANKINE BOTTOM CYCLE
SYSTEM

ORIGINAL PAGE IS
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DATE: SEPT. 25, 1985

MIW = MAKE IN WORKS
BOF = BOUGHT OUT FINISHED

PART NO.	PAGE 1 OF 11 DESCRIPTION	MIW BOF	QTY. REQD	LABOR COST - HOUR	MATERIAL COST	TOTAL MATERIAL & LABOR COST	PATTERN COST	MATERIAL TYPE	REMARKS
GEAR HOUSING ASSEMBLY	GEAR HOUSING	MIW	1	$127.75	$75.00	$202.75	$6,000.00	DUCT IRON	REFERENCE LAYOUT 100 GBL 551 SHEET 1 & 5
	PIPE PLUG 1/8	BOF	7		$0.14	$0.14			
	SEAL, GEAR HSG. TO BLOCK	BOF	1		$0.50	$0.50			
	GASKET, GEAR HSG. TO BLOCK	BOF	1		$3.00	$3.00		SILASTIC RTV #732	
	DOWEL PIN, GR. HSG. TO BLK.	BOF	5		$5.00	$5.00			
	GASKET, BRACKET TO BLOCK	BOF			$1.00	$1.00			
	BRACKET, GEAR HSG. ANGLE	MIW		$10.04	$3.00	$13.04			
	BRACKET,GEAR HSG. SUPPORT	MIW		$14.96	$5.00	$19.96			
	BOLT, LIFTING	BOF	2						
	NUT 5/8	BOF	2		$1.50	$1.50			
	CAPSCREW, BRKT. TO GR. HSG.	BOF	2		$0.50	$0.50			
	WASHER	BOF	2		$0.10	$0.10			
	CAPSCREW BRKT. TO BRKT. 3/8	BOF	2		$0.40	$0.40			
	WASHER	BOF	2		$0.10	$0.10			
	CAPSCREW BRKT. TO BLOCK 1/2	BOF	4		$1.40	$1.40			
	WASHER	BOF	4		$0.40	$0.40			
	SUB-TOTAL					$249.79			
FLYWHEEL HOUSING ASSEMBLY	HOUSING, FLYWHEEL	MIW	1		$125.00	$125.00	$4,500.00	DUCT IRON	
	ASSEMBLY FLYWHEEL HOUSING	MIW	1	$219.00		$219.00			MACHINING ASSEMBLY
	SHAFT, IDLER	BOF	1		$10.00	$10.00			
	PIN, IDLER SHFT TO FLY HSG	BOF	1		$2.00	$2.00			
	CAPSCREW, SHAFT TO FLY HSG	BOF	2		$1.00	$1.00			
	WASHER, SHAFT TO FLY HSG	BOF	2		$0.50	$0.50			
	O'RING, SHAFT TO GEAR HSG	BOF	1		$1.25	$1.25			
	GASKET, FLY HSG TO GR HSG	BOF	1		$3.50	$3.50		SILASTIC RTV #732	

APPENDIX 2

THERMO ELECTON CORP.

COST STUDY OF ORGANIC RANKINE

BOTTOM CYCLE SYSTEM

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THERMO ELECTRON CORP.
COST STUDY OF ORGANIC
RANKINE BOTTOM CYCLE
SYSTEM

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DATE: SEPT. 25, 1985

MIW = MADE IN WORKS
BOF = BOUGHT OUT FINISHED

PART NO.	DESCRIPTION	MIW BOF	QTY. REQD	LABOR COST @ $73/HOUR	MATERIAL COST	TOTAL MATERIAL & LABOR COST	PATTERN COST	MATERIAL TYPE	REMARKS
GEAR HOUSING ASSEMBLY	GEAR HOUSING	MIW	1	$127.75	$75.00	$202.75	$6,000.00	DUCT IRON	REFERENCE LAYOUT 100 GBL 551 SHEET 1 & 5
	PIPE PLUG 1/8	BOF	7		$0.14	$0.14			
	SEAL, GEAR HSG. TO BLOCK	BOF	1		$0.50	$0.50			
	GASKET, GEAR HSG. TO BLOCK	BOF	1		$3.00	$3.00		SILASTIC RTV 8732	
	DOWEL PIN, GR. HSG. TO BLK.	BOF	5		$5.00	$5.00			
	GASKET, BRACKET TO BLOCK	BOF			$1.00	$1.00			
	BRACKET, GEAR HSG. ANGLE	MIW		$10.04	$3.00	$13.04			
	BRACKET,GEAR HSG. SUPPORT	MIW		$14.76	$5.00	$19.76			
	BOLT, LIFTING	BOF	2						
	NUT 5/8	BOF	2		$1.50	$1.50			
	CAPSCREW, BRKT. TO GR. HSG.	BOF	2		$0.50	$0.50			
	WASHER	BOF	2		$0.10	$0.10			
	CAPSCREW BRKT. TO BRKT. 3/8	BOF	2		$0.40	$0.40			
	WASHER	BOF	2		$0.10	$0.10			
	CAPSCREW BRKT. TO BLOCK 1/2	BOF	4		$1.40	$1.40			
	WASHER	BOF	4		$0.40	$0.40			
FLYWHEEL HOUSING ASSEMBLY	HOUSING, FLYWHEEL	MIW	1		$125.00	$125.00	$4,500.00	DUCT IRON	
	ASSEMBLY FLYWHEEL HOUSING	MIW	1	$219.00		$219.00			MACHINING ASSEMBLY
	SHAFT, IDLER	BOF	1		$10.00	$10.00			
	PIN, IDLER SHFT TO FLY HSG	BOF	1		$2.00	$2.00			
	CAPSCREW, SHAFT TO FLY HSG	BOF	2		$1.00	$1.00			
	WASHER, SHAFT TO FLY HSG	BOF	2		$0.50	$0.50			
	O RING, SHAFT TO GEAR HSG	BOF	1		$1.25	$1.25			
	GASKET, FLY HSG TO GR HSG	BOF	1		$3.50	$3.50		SILASTIC RTV 8732	
	CAPSCREW, FLY HSG TO GR HSG	BOF			$0.30	$0.30			
	WASHER, FLY HSG TO GR HSG	BOF			$0.05	$0.05			

PART NO.	PAGE 2 OF 9 DESCRIPTION	MIN BOF	QTY REQD	LABOR COST @ $73 HOUR	MATERIAL COST	TOTAL MATL & LABOR COST	PATTERN COST	MATERIAL TYPE	REMARKS
FLYWHEEL									
HOUSING	CAPSCREW, FLY HSG TO BLOCK	BOF	7		$1.40	$1.40			
ASSEMBLY	WASHER, FLY HSG TO BLOCK	BOF	7		$0.28	$0.28			
CONTINUED									
	OIL SEAL REAR CRANK	BOF	1		$3.00	$3.00			
IDLER	IDLER GEAR	BOF							
GEAR	ASSEMBLY, IDLER GEAR	BOF							
ASSEMBLY	GEAR, IDLER	BOF	1		$44.00	$44.00			
	BUSHING, IDLER GEAR	BOF	1		$5.00	$5.00			
	THRUSTWASHER, IDLER GEAR	BOF	2		$10.00	$10.00			
CRANKSHAFT	CRANKSHAFT								
ASSEMBLY									
	CRANKSHAFT	BOF	1						
	GEAR, CRANK REAR GEAR TRAIN	BOF	1						
	CAPSCREW 3/8 GR TO CRANK	BOF	3						
	WASHER GEAR TO CRANK	BOF	3						
FLYWHEEL	FLYWHEEL	MIN							
ASSEMBLY									
	FLYWHEEL (PER APPLICATION)	MIN	1						
	FLYWHEEL RING GEAR	BOF	1						
	CAPSCREW FLYWHEEL TO CRANK	BOF	6						
	WASHER FLYWHEEL TO CRANK	BOF	6						
VAPOR	VAPOR GENERATOR ASSEMBLY	BOF			$1,200.00	$1,200.00			
GENERATOR									
ASSEMBLY	VAPOR GENERATOR ASSEMBLY	BOF							
	SHELL, OUTER	BOF							
	SHELL, INNER	BOF							
	COIL ASSY VAPOR GENERATOR	BOF							
	DIVERTER VALVE ACT. ASSY.	BOF							
	DIVERTER VALVE ASSEMBLY	BOF							
	INSULATION, SHELL	BOF							
	INSIDE CAN BAFFLE	BOF							
	VAPOR GENERATOR	BOF							

PART NO.	PAGE 3 OF 9 DESCRIPTION	MIN PUR	QTY. REVD	LABOR COST @ $73 HOUR	MATERIAL COST	TOTAL MATL & LABOR COST	PATTERN COST	MATERIAL TYPE	REMARKS
TURBINE	TURBINE GEAR BOX								
GEAR BOX									
ASSEMBLY	ASSY GEAR BOX FINAL MACH	MIN	1	$10.95		$10.95			
	GEAR BOX HSG CVR MFG	MIN	1					CAST IRON	
	COVER GEAR BOX	MIN	1	$14.97	$6.00	$20.97	$2,000.00	CAST IRON	
	PILLOW BLOCK	POF	1		$5.00	$5.00		CAST IRON	
	CAPSCREW PILLOW BLOCK	POF	4		$2.00	$2.00			
	WASHER	POF	4		$1.00	$1.00			
	CAPSCREW COVER TO GEAR BOX	POF	9		$2.25	$2.25			
	DOWEL COVER TO GR BOX HSG	POF	2		$0.50	$0.50			
	DOWEL PILLOW BLP TO GR BOX	POF	2		$0.50	$0.50			
	O RING, CVR TO GR BOX HSG	POF	1		$1.50	$1.50			
	O RING GR BX HSG TO G.B HSG	POF	1		$1.50	$1.50			
	CAPSCREW GEAR BOX HSG TO	POF	8		$2.00	$2.00			
	GEAR BOX HSG.	MIN	1	$18.25	$8.00	$26.25	$2,500.00		
	DOWEL GR BX HSG TO G.B HSG	POF	8		$4.00	$4.00			
	HOUSING,GR BOX OUTPUT	MIN	1	$27.56	$10.00	$37.56	$3,500.00	DUCT IRON	
	BEARING,BALL GEAR	POF	2		$5.00	$3.00			MRC 819075
	BEARING BALL REAR	POF	1		$3.00	$3.00			MRC# 3085
	BEARING BALL FRONT	POF	1		$3.00	$3.00			MRC# 2065
	SNAP RING	POF	1		$0.05	$0.05			TRUARC 5100-118
	GEAR DRIVER (TO GEAR TRAIN)	POF	1		$15.00	$15.00			
	SPACER, BRG REAR	POF	1		$2.50	$2.50			
	SPACER, BRG FRONT	POF	1		$2.50	$2.50			
	STUD OUTPUT HSG TO COVER-	POF	3		$3.00	$3.00			
	MOUNTING HOUSING					$0.00			
	NUT	POF	3		$0.15	$0.15			
	WASHER	POF	3		$0.15	$0.15			

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OF POOR QUALITY

PART NO.	PAGE 4 OF 9 DESCRIPTION	MIN BOF	QTY. REQD	LABOR COST @ $7[illegible] HOUR	MATERIAL COST	TOTAL MATL & LABOR COST	PATTERN COST	MATERIAL TYPE	REMARKS
TURBINE GEAR BOX ASSEMBLY	TURBINE GEAR BOX	MIN	1						
	GEAR - SHAFT	BOF	1		$25.00	$25.00			ON SHAFT WITH GEAR DRIVING INTO GEAR TRAIN
	BEARING, BALL	BOF	1		$3.00	$3.00			
	BEARING, BALL	BOF	1		$3.00	$3.00			
	OIL SEAL	BOF	1		$3.50	$3.50			
	SEAL	BOF	1		$1.50	$1.50			
	SPACER, BEARING	BOF	1		$2.50	$2.50			
	O RING	BOF	1		$1.00	$1.00			
	SNAP RING	BOF	1		$0.05	$0.05			
	GEAR	BOF	1		$75.00	$75.00			ON SHFT WITH BULL GR
	GEAR - SHAFT, BULL	BOF	1		$5.00	$5.00			
	BEARING, BALL	BOF	1		$3.00	$3.00			
	BEARING,BALL DOUBLE ROW	BOF	1		$10.00	$10.00			
	SNAP RING	BOF	1		$0.05	$0.05			
	CAPSCREW GEAR MOUNTING	BOF			$0.50	$0.50			
	DOWEL GEAR MOUNTING	BOF			$1.00	$1.00			
	COVER	MIN	1	$14.97	$10.00	$74.97	$2,500.00		
	O RING	BOF	1		$1.50	$1.50			
	CAPSCREW	BOF			$0.25	$0.25			
	TURBINE GEAR BOX	MIN		$27.56	$12.00	$37.56	$3,000.00		
	OVERRUNNING CLUTCH	BOF	1		$250.00	$250.00			
VIBRATING ISOLATING ASSEMBLY	VIBRATING ISOLATING - COUPLING ASSEMBLY	BOF	1		$150.00	$150.00			
		BOF							
	SHAFT ISOLATING	BOF	1						
	QUILL SHAFT	BOF	1						
	HOUSING, TUBE		1						PART OF ISOLATING ASSEMBLY

PART NO.	PAGE 5 OF 9 DESCRIPTION	MFG/BOF	QTY. REQD	LABOR COST @ $73/HOUR	MATERIAL COST	TOTAL MAT'L & LABOR COST	PATTERN COST	MATERIAL TYPE	REMARKS
TURBINE	TURBINE ASSY SINGLE STAGE								
ASSEMBLY									
SINGLE	ASSY SINGLE STAGE TURBINE	BOF	1	$14.97		$14.97			
STAGE	GEAR, PINION TURBINE	BOF	1		$30.00	$30.00			
	SHAFT, TURBINE	BOF	1	$18.25	$10.00	$28.25			
	BEARING, BALL	BOF	1		$3.00	$3.00			
	BEARING, BALL	BOF	1		$3.00	$3.00			
	SPACER, END	BOF	1		$1.50	$1.50			
	LOCKNUT	BOF	1		$0.50	$0.50			
	SPACER, BEARING INNER	BOF	1		$3.00	$3.00			
	SPACER, BEARING OUTER	BOF	1		$4.00	$4.00			
	SEAL, TURBINE	BOF	1		$6.00	$6.00			
	WHEEL, TURBINE - SHAFT	MIW	1	$36.50	$300.00	$336.50	$6,000.00		
	PLUG, SHAFT ORIFICE	BOF	1		$1.50	$1.50			
	SLINGER	BOF	2		$5.00	$5.00			
	NOZZLE BLOCK TURBINE	MIW	1	$73.00	$50.00	$173.00			
	INLET HOUSING	MIW	1	$54.75	$40.00	$94.75	$4,000.00		
HOUSING	EXHAUST, HOUSING	MIW	1	$18.25	$20.00	$38.25	$4,500.00		
ASSEMBLY	O RING METALIC	BOF	1		$3.50	$3.50			
	O RING METALIC	BOF	1		$3.50	$3.50			
	CAPSCREW	BOF			$0.50	$0.50			
HOUSING	HSG., MARK 3 SINGLE STAGE	BOF	1	$54.75	$35.00	$87.75	$2,500.00		
ASSEMBLY	CAPSCREW	BOF	8		$4.00	$4.00			
	WASHER	BOF	8		$0.80	$0.80			
	PLUG	BOF	1		$0.25	$0.25			
TURBINE	TURBINE ASSY SINGLE STAGE	BOF		$36.50		$36.50			
END COVER									
ASSEMBLY	COVER, TURBINE SHAFT END	BOF	1	$18.25	$75.00	$93.25	$2,500.00		
	O'RING, COVER	BOF	1		$2.50	$2.50			
	CAPSCREW	BOF	4		$2.50	$2.50			
	O RING	BOF	1		$1.50	$1.50			
BELLOWS	CONNECTION, TURBINE OUTLET-	BOF	1		$35.00	$35.00			
ASSEMBLY	BELLOWS ASSY								
	O RING	BOF	1		$1.50	$1.50			
	CAPSCREW	BOF			$0.25	$0.25			
	WASHER	BOF			$0.05	$0.05			

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PART NO.	PAGE 6 OF 9 DESCRIPTION	MIN- ROF	QTY REQD	LABOR COST @ $73/ HOUR	MATERIAL COST	TOTAL MATL & LABOR COST	PATTERN COST	MATERIAL TYPE	REMARKS
	REGENERATOR -CONDENSER -	ROF			$625.00	$625.00			
	BOOST PUMP	ROF			$250.00	$250.00			
	TURBINE OIL PUMP - PLUMBING				$15.00	$15.00			
FILTER	FILTER ASSEMBLY	ROF	1		$45.00	$45.00			FLOURINOL CIRCUIT
ASSEMBLY	CAN FILTER	ROF	1						
	FLANGE	ROF	1						
	FLANGE	ROF	1						
	FLANGE	ROF	1						
	O RING	ROF	1						
	NIPPLE	ROF							
	CAPSCREW	ROF							
BRACKET	BRACKET,SUPPORT REGENERATOR	MIN	1	$18.25		$18.25			
MOUNTING	CONDENSER ASSY								
	CAPSCREW	ROF	4		$0.60	$0.60			
	WASHER	ROF	4		$0.20	$0.20			
BRACKET	BRACKET,FILTER ASSEMBLY	MIN	1	$18.25		$18.25			
MOUNTING	CAPSCREW	ROF	4		$0.60	$0.60			
	WASHER	ROF	4		$0.20	$0.20			
	GEROTOR PUMP - TURBINE OIL	ROF	1		$20.00	$20.00			
OIL COLLER	OIL COOLER - TURBINE OIL	ROF	2						
MOUNTING	BRACKET,OIL COOLER MTG.	MIN	2		$15.00	$15.00			
	CAPSCREW	ROF	4		$0.60	$0.60			
	WASHER	ROF	4		$0.20	$0.20			
	REGENERATOR - CONDENSER BOOST PUMP - TURBINE OIL PUMP - PLUMBING		1		$30.00	$30.00			
	PLUMBING,FLOURINOL,CONDENSER TO BOOST PUMP		1		$20.00	$20.00			
	PLUMBING,FLOURINOL,BOOST PUMP TO FILTER.		1		$20.00	$20.00			
	PLUMBING,FLOURINOL,FILTER TO FEED PUMP.		1		$20.00	$20.00			

PART NO.	PAGE 7 OF 9 DESCRIPTION	MIN ROE	QTY. REQD	LABOR COST @ $73 HOUR	MATERIAL COST	TOTAL MAT'L & LABOR COST	PATTERN COST	MATERIAL TYPE	REMARKS
	PLUMBING,OIL,TURBINE TO OIL COOLER.		1		$20.00	$20.00			
	PLUMBING,OIL,OIL COOLER TO GEROTOR.		1		$15.00	$15.00			
	PLUMBING,FLOURINOL,TURBINE TO CONDENSER.		1		$20.00	$20.00			
CONNECTION ASSEMBLY	CONNECTION,TURBINE TO REGENRATOR (FLOURINOL)	MIN	1		$25.00	$25.00			
	GASKET	BOF	1		$1.50	$1.50			
	CAPSCREWS	BOF	4		$0.60	$0.60			
	WASHERS	BOF	4		$0.20	$0.20			
	CLAMP	BOF	2		$7.00	$7.00			
	FLUID RESEVIOR ASSEMBLY	BOF	1		$50.00	$50.00			
BRACKET MOUNTING	BRACKET, FLUID RESEVIOR MTG.	MIN	1		$15.00	$15.00			
	CAPSCREW	BOF			$0.75	$0.75			
	WASHER	BOF			$0.20	$0.20			
	CAPSCREW	BOF			$0.50	$0.50			
	WASHER	BOF			$0.20	$0.20			
	PUMP, VACUUM	BOF	1		$25.00	$25.00			
	PLUMBING,FLOURINOL,TURBINE TO FILTER.		1		$20.00	$20.00			
	PLUMBING,FLOURINOL,FILTER TO VACUUM PUMP		1		$20.00	$20.00			
	PLUMBING,FLOURINOL,VACUUM PUMP TO RESEVIOR.		1		$20.00	$20.00			

PART NO.	PAGE 8 OF 9 DESCRIPTION	MIN POF	QTY. REQD	LABOR COST @ $73 HOUR	MATERIAL COST	TOTAL MATL & LABOR COST	PATTERN COST	MATERIAL TYPE	REMARKS
	FLUMPING FLOURINOL,RESEVIOR TO CONDENSER		1		$20.00	$20.00			
FEED FUMP & DRIVE ASSEMBLY	FEED FUMP & DRIVE								
	ASSEMBLY FEED FUMP - FLOURINOL		1	$36.50		$36.50			
	HOUSING FEED PUMP	MIN	1	$36.50	$20.00	$56.50	$3,500.00	MEEHANITE	TYPE GA 50
	FEED FUMP	POF	1						
	CAPSCREW FEED FUMP MOUNTING	POF	4		$0.60	$0.60			
	WASHER	POF	4		$0.70	$0.20			
	HOUSING,FEED FUMP DRIVE	MIN	1	$36.50	$10.00	$46.50	$4,000.00	DUCT IRON	
	BEARING,BALL REAR	POF	1		$4.50	$4.50		MRC #2075	
	SNAP,RING	POF	1		$0.05	$0.05			TRUARC #5000-291
	SNAP,RING	POF	1		$0.05	$0.05			TRUARC #5100-137
	OIL SEAL	POF	1		$2.50	$2.50			
	BEARING,BALL FRONT	POF	1		$3.00	$3.00		MRC #2065	
	SNAP RING	POF	1		$0.05	$0.05			TRUARC #5100-118
	SHAFT,DRIVE FEED FUMP	POF	1		$12.00	$12.00			
	GEAR,DRIVE FEED FUMP	POF	1		$9.00	$9.00			
	KEY,WOODRUFF 3/16 X 1	POF	1		$1.50	$1.50			
	O'RING HSG TO FLY HSG	POF	1		$1.50	$1.50			
	COVER,FEED FUMP DRIVE	MIN	1	$14.60	$5.00	$19.60	$2,500.00		
	O'RING COVER TO GR HSG	POF	1		$1.50	$1.50			
	CAPSCREW F.P HSG TO F.P COVER	POF	5		$0.75	$0.75			
	WASHER	POF	5		$0.25	$0.25			
	FLUMPING,FLOURINOL,FEED PUMP TO PEGENERATOR.		1		$25.00	$25.00			

PART NO.	PAGE 9 OF 9 DESCRIPTION	MIN POF	QTY. REQD	LABOR COST @ $73 HOUR	MATERIAL COST	TOTAL MATL & LABOR COST	PATTERN COST	MATERIAL TYPE	REMARKS
	OIL PAN REAR DRIVE TRAIN	MIN	1	$36.04	$50.00	$86.04	$3,500.00		
	ASSEMBLY OIL PAN	MIN	1	$14.60		$14.60			
OIL PAN ASSEMBLY	OIL PAN	MIN	1						
	PLATE	MIN	1						
	RIB	MIN	2						
	GASKET,OIL PAN	POF	1						
	GASKET,OIL PAN REAR	POF	1		$1.50	$1.50			
	CAPSCREW	POF			$5.00	$5.00			
	WASHER	POF			$2.00	$2.00			
	CONTROL SYSTEM	POF	1		$305.00	$305.00			
	ASSEMBLE UNIT	MIN	1	$292.00		$292.00			
	TOTALS			$1,314.47	$4,478.42	$5,792.89	$57,000.00		

ALL COSTS ARE IN 1985 DOLLARS.

LABOR COST @ $73.00 HOUR IS COMPLETE MANUFACTURING COST INCLUDING
ALL OVERHEAD EXPENSES.

R.L. LINNEY 9/25/85

APPENDIX 3

FOSTER - MILLER INC.

COST STUDY OF STEAM BOTTOMING

CYCLE SYSTEM

FOSTER - MILLER INC.
COST STUDY OF STEAM
BOTTOMING CYCLE SYSTEM

MIW = MAKE IN WORKS
BOF = BOUGHT OUT FINISHED

DATE: NOVEMBER 01, 85

PART NO.	PAGE 1 OF 11 DESCRIPTION	MIW BOF	QTY. REQD	LABOR COST @ $73 HOUR	MATERIAL COST	TOTAL MATERIAL & LABOR COST	PATTERN COST	MATERIAL TYPE	REMARKS
HOUSING, GR.	HOUSING, GEAR REAR	MIW	1	$127.75	$90.00	$217.75	$7,000.00	DUCT IRON	
	SEAL GEAR HOUSING TO BLOCK	BOF	1		$7.00	$7.00			
	DOWEL PIN GR. HOUSING TO BLOCK	BOF	2		$0.70	$0.70			
	SUB-TOTAL					$225.45			
HSG. , FLYW.	HOUSING, FLYWHEEL	MIW	1	$131.40	$79.95	$211.35	$5,500.00		
	DOWEL, FLYWHEEL HOUSING	BOF	2		$0.50	$0.50			
	CAPSCREW FLYWHEEL HSG TO BLK.	BOF	8		$1.04	$1.04			
	WASHER	BOF	8		$0.32	$0.32			
	CAPSCREW FLYW. HSG. TO GR. HSG	BOF	4		$0.48	$0.48			
	WASHER	BOF	4		$0.16	$0.16			
	SUB-TOTAL					$213.85			
CRANKSHAFT	CRANKSHAFT (REAR GEAR TRAIN)	BOF	1						
	SPROCKET, CHAIN	BOF	1		$14.50	$14.50			
	CHAIN, EXPANDER TO ENG. CRANK	BOF	1		$24.00	$24.00			
	DOWEL, FLYWHEEL	BOF	1		$0.15	$0.15			
	SUB-TOTAL					$38.65			

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PART NO.	PAGE 2 OF 11 DESCRIPTION	MIW BOF	QTY REQD	LABOR COST @ $73 HOUR	MATERIAL COST	TOTAL MATERIAL & LABOR COST	PATTERN COST	MATERIAL TYPE	REMARKS
FLYWHEEL	FLYWHEEL ASSEMBLY	MIW	1						
	FLYWHEEL	MIW	1						
	GEAR, RING	BOF	1						
	CAPSCREW FLYWHEEL TO CRANK	BOF	6						
	DOWEL PIN	BOF	2						
HOUSING REAR OIL SEAL	HOUSING REAR OIL SEAL	BOF	1	$48.93	$29.00	$77.93	$3,500.00		
	GASKET	BOF	1		$0.50	$0.50			
	CAPSCREWS	BOF	8		$1.50	$1.50			
	WASHERS	BOF	8		$0.30	$0.30			
	SEAL, OIL	BOF	1		$3.00	$3.00			
	SUB-TOTAL					$83.23			
PAN, OIL	PAN, OIL	MIW	1		$71.00	$71.00			
	GASKET, OIL PAN	BOF	1		$3.50	$3.50			
	CAPSCREWS BLOCK & GEAR COVER	BOF							
	WASHERS	BOF							
	CAPSCREWS	BOF	4		$0.60	$0.60			
	WASHER	BOF	4		$0.16	$0.16			
	SUB-TOTAL					$75.26			

PART NO.	PAGE 3 OF 11 DESCRIPTION	MIW BOF	QTY. REQD	LABOR COST @ $73 HOUR	MATERIAL COST	TOTAL MATERIAL & LABOR COST	PATTERN COST	MATERIAL TYPE	REMARKS
BOILER ASSY	BOILER ASSEMBLY	BOF	1		$880.00	$880.00			
	SHELL OUTER	BOF	1						
	SHELL INNER (WIRE MESH)	BOF	1						
	INSULATION	BOF	1						
	GARTER SPRING	BOF	2						
	COIL ASSEMBLY	BOF	1						
	CONNECTION WATER IN	BOF	1						
	CONNECTION STEAM OUT	BOF	1						
	FLANGE EXHAUST IN	BOF	1						
	FLANGE EXHAUST OUT	BOF	1						
	FLANGE BOTTOM HOUSING	BOF	1						
	INNER CORE	BOF	1						
	BOTTOM HOUSING	BOF	1						
	GASKET	BOF	1						
	CAPSCREW	BOF	12						
	WASHER	BOF	12						
	FLANGE BOTTOM HOUSING	BOF	1						
	SUB-TOTAL					$880.00			
	STEAM EXPANDER ASSEMBLY		1	$91.25		$91.25			
	BLOCK	MIW	1	$114.49	$44.00	$158.49	$20,000.00		
LINER	LINER (WITH EXH. MANIFOLD)	MIW	2	$129.94	$22.00	$151.94	$5,500.00		
	GASKET LINER TO BLOCK	BOF	1		$1.25	$1.25			
	CAPSCREWS	BOF	4		$3.20	$3.20			
	WASHERS	BOF	4		$0.80	$0.80			
	SUB-TOTAL					$157.19			
CONN. ROD	CONNECTING ROD	MIW	2	$39.42	$16.50	$55.92			
	SHELL CONNECTING ROD	BOF	4		$8.00	$8.00			
	BOLT, CONNECTING ROD	BOF	4		$3.00	$3.00			
	SUB-TOTAL					$66.92			
	PIN, PISTON	BOF	2		$11.00	$11.00			
	BOLT, ROD TO PIN	BOF	4		$2.80	$2.80			
	SUB-TOTAL					$13.80			

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PART NO.	PAGE 4 OF 11 DESCRIPTION	MIN BOF	QTY. REQD	LABOR COST @ $73 HOUR	MATERIAL COST	TOTAL MATERIAL & LABOR COST	PATTERN COST	MATERIAL TYPE	REMARKS
PISTON	PISTON, SKIRT	MIN	2		$7.70	$7.70			ALUMINUM
	PISTON, CROWN	MIN	2		$6.50	$6.50			STEEL
	CAPSCREW, CROWN TO SKIRT	BOF	16		$14.08	$14.08			
	RING, SEALING CROWN TO SKIRT	BOF	2		$5.50	$5.50			
	RING, PISTON TOP	BOF	2		$5.50	$5.50			
	RING, PISTON INTERMEDIATE	BOF	2		$5.50	$5.50			
	RING, PISTON BOTTOM	BOF	2		$5.50	$5.50			
	SUB-TOTAL					$50.28			
CAMSHAFT	CAMSHAFT	BOF	1		$19.80	$19.80			
	GEAR, CAMSHAFT	BOF	1		$4.95	$4.95			
	NUT	BOF	1		$0.45	$0.45			
	KEY	BOF	1		$1.25	$1.25			
	BEARING, BALL CAM	BOF	1		$4.25	$4.25			
	BEARING,BALL CAM	BOF	1		$4.25	$4.25			
	SUB-TOTAL					$34.95			
CRANKSHAFT	CRANKSHAFT	BOF	1		$88.00	$88.00			
	GEAR, CRANKSHAFT	BOF	1		$4.95	$4.95			
	BEARING CRANKSHAFT	BOF	2		$16.50	$16.50			
	SUB-TOTAL					$109.45			
CLUTCH	CLUTCH ASSEMBLY (EXPANDER	1	BOF		$75.00	$75.00			
	NUT, CLUTCH	1	BOF		$1.00	$1.00			
	SUB-TOTAL					$76.00			
ISOLATOR ASSEMBLY	TORSIONAL ISOLATOR ASSEMBLY (EXPANDER TO ENGINE)	1	MIN	$18.25		$18.25			
	HUB, CLUTCH ISOLATOR	1	BOF	$18.00		$18.00			
	BUSHING, HUB	2	BOF	$4.00		$4.00			
	LEAF SPRING	20	BOF	$6.00		$6.00			
	PLATE, RETAINER	2	BOF	$0.20		$0.20			
	RETAINER, SPRING	1	BOF	$10.45		$10.45			
	COVER, SPRING RETAINER	1	BOF	$10.36		$10.36			
	PIN, SPRING RETAINER	4	BOF	$2.00		$2.00			
	RIVET, SPRING RETAINER	8	BOF	$1.20		$1.20			
	SUB-TOTAL					$70.46			

PART NO.	PAGE 5 OF 11 DESCRIPTION	MIW BOF	QTY. REQD	LABOR COST@ $73 HOUR	MATERIAL COST	TOTAL MATERIAL & LABOR COST	PATTERN COST	MATERIAL TYPE	REMARKS
SPROCKET	SPROCKET, CHAIN DRIVE	1	BOF		$14.50	$14.50			
	BUSHING, SPROCKET	1	BOF		$1.50	$1.50			
	CAPSCREW	6	BOF		$3.00	$3.00			
	WASHER	6	BOF		$0.90	$0.90			
	SUB-TOTAL					$19.90			
HSG., REAR	HOUSING, REAR BRG. CARRIER	MIW	1	$27.65	$8.00	$35.65			
	BEARING,BALL CAM	BOF	1		$4.25	$4.25			
	GASKET, HSG TO BLOCK	BOF	1		$1.75	$1.75			
	CAPSCREWS	BOF	8		$4.00	$4.00			
	WASHER	BOF	8		$0.40	$0.40			
	SUB-TOTAL					$46.05			
HSG., O SEAL	HOUSING, OIL SEAL EXPANDER	MIW	1	$54.75	$15.00	$69.75	$3,000.00		
	OIL SEAL	BOF	1		$3.75	$3.75			
	GASKETS	BOF	1		$1.50	$1.50			
	CAPSCREWS	BOF	4		$1.20	$1.20			
	WASHERS	BOF	4		$0.20	$0.20			
	SUB-TOTAL					$76.40			
FLYWHEEL	FLYWHEEL	MIW	1		$33.00	$33.00	$3,500.00		
	GUARD, FLYWHEEL	BOF	1		$4.75	$4.75			
	CAPSCREWS	BOF	8		$1.60	$1.60			
	WASHERS	BOF	8		$0.24	$0.24			
	SUB-TOTAL					$39.59			
HEAD EXP.	HEAD, EXPANDER	MIW	2	$73.00	$6.60	$79.60	$5,500.00		
	GASKET, HEAD	BOF	2		$6.00	$6.00			
	CAPSCREW, HEAD TO BLOCK	BOF	8		$6.16	$6.16			
	WASHERS	BOF	8		$2.00	$2.00			
	SUB-TOTAL					$93.76			

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PART NO.	PAGE 6 OF 11 DESCRIPTION	MIW BOF	QTY. REQD	LABOR COST @ $73 HOUR	MATERIAL COST	TOTAL MATERIAL & LABOR COST	PATTERN COST	MATERIAL TYPE	REMARKS
STEAM RELIEF VALVE ASSY.	STEAM RELIEF VALVE ASSEMBLY	BOF	1		$60.00	$60.00			
	INSERT, RELIEF VALVE	BOF	2						
	BODY, RELIEF VALVE	BOF	2						
	VALVE, RELIEF	BOF	2						
	WASHER, BODY	BOF	2						
	TAPPET, INTAKE VALVE	BOF	2						
	SOCKET, TAPPET	BOF	2						
	SUB-TOTAL					$60.00			
HSG. INT VLV	HOUSING INTAKE VALVE & MFLD.	MIW	2	$109.50	$30.00	$139.50	$4,500.00		
	VALVE, INTAKE	BOF	2		$6.00	$6.00			
	INSERT, INTAKE VALVE	BOF	2		$0.76	$0.76			
	GUIDE, VALVE	BOF	2		$1.20	$1.20			
	VALVE LASH ADJUSTER	BOF	2		$3.00	$3.00			
	PUSH ROD	BOF	2		$4.00	$4.00			
	SPRING, VALVE	BOF	2		$0.52	$0.52			
	RETAINER, VALVE	BOF	2		$1.50	$1.50			
	COLLET	BOF	4		$0.04	$0.04			
	TUBE, PUSH ROD ENCLOUSER	BOF	2		$3.75	$3.75			
	RETAINER SPRING TUBE	BOF	4		$3.00	$3.00			
	O,RING INT HSG MTG TO BLK.	BOF	2		$1.50	$1.50			
	CAPSCREW HOUSING MOUNTING	BOF	4		$1.00	$1.00			
	GASKET, INT. HSG. TO HEAD	BOF	1		$1.25	$1.25			
	CAPSCREW , INT. HSG. TO HEAD	BOF	2		$0.50	$0.50			
	WASHER, INT. HSG. TO HEAD	BOF	2		$0.10	$0.10			
	BODY	BOF	2		$3.00	$3.00			
	WASHER	BOF	2		$0.75	$0.75			
	SUB-TOTAL					$171.37			
TUBE	INLET TUBE, BOILER TO EXPANDER	BOF	2		$3.00	$3.00			
	FLANGE, INLET TUBE	BOF	2		$1.50	$1.50			
	CAPSCREW	BOF	4		$0.80	$0.80			
	SUB-TOTAL					$5.30			

PART NO.	PAGE 7 OF 11 DESCRIPTION	MIN BOF	QTY. REQD	LABOR COST @ $73 HOUR	MATERIAL COST	TOTAL MATERIAL & LABOR COST	PATTERN COST	MATERIAL TYPE	REMARKS
COVER, PAN	COVER, OIL PAN EXPANDER	BOF	1		$11.00	$11.00			
	GASKET	BOF	1		$2.25	$2.25			
	CAPSCREWS	BOF	6		$0.48	$0.48			
	WASHERS	BOF	6		$0.06	$0.06			
	PLUG, DRAIN	BOF	1		$0.30	$0.30			
	SUB-TOTAL					$14.09			
	FILTER, LUBE OIL EXPANDER	BOF	1		$8.00	$8.00			
	PUMP, LUBE OIL EXPANDER	BOF	1		$50.00	$50.00			
EXPANDER	GASKET, EXPANDER TO GR HSG MTG	BOF	1		$3.00	$3.00			
MOUNTING	CAPSCREWS, EXPANDER MOUNTING	BOF	8		$2.00	$2.00			
	WASHERS, EXPANDER MOUNTING	BOF	8		$0.80	$0.80			
	SUB-TOTAL					$5.80			
PUMP ASSY	FEEDWATER PUMP ASSEMBLY	BOF	1						
	FEEDWATER PUMP ASSEMBLY (ELECTRONIC FLOW CONTROL SOLENOID OF INTAKE VALVE UNLOADER INCLUDED)	BOF	1		$93.50	$93.50			
	GASKET FEED PUMP MOUNTING	BOF	1			$1.75			
	CAPSCREWS	BOF	4			$1.00			
	WASHERS	BOF	4			$0.40			
	SUB-TOTAL					$96.65			

PART NO.	PAGE 8 OF 11 DESCRIPTION	MIW BOF	QTY. REQD	LABOR COST @ $73 HOUR	MATERIAL COST	TOTAL MATERIAL & LABOR COST	PATTERN COST	MATERIAL TYPE	REMARKS
PUMP DRIVE	DRIVE FEEDWATER PUMP								
	SHAFT, IDLER GEAR	BOF	1		$20.00	$20.00			
	GEAR, IDLER	BOF	1		$30.00	$30.00			
	BEARING, DOUBLE ROW BALL	BOF	1		$7.00	$7.00			
	SHAFT, ACCESSORY DRIVE	BOF	1		$12.50	$12.50			
	GEAR, ACCESSORY DRIVE	BOF	1		$15.00	$15.00			
	HOUSING, ACCESSORY DRIVE	MIW	1	$54.75	$12.50	$67.25	$3,500.00		
	BEARING, BALL	BOF	1		$2.50	$2.50			
	BEARING, BALL	BOF	1		$2.50	$2.50			
	NUT, SHAFT GEAR END	BOF	1		$0.50	$0.50			
	GASKET, HOUSING	BOF	1		$1.00	$1.00			
	CAPSCREWS	BOF	4		$1.00	$1.00			
	WASHERS	BOF	4		$0.40	$0.40			
	SUB-TOTAL					$159.65			
	RADIATOR ASSEMBLY	BOF	1			$0.00			
	RADIATOR, CONDENSOR	BOF	1		$407.00	$407.00			
	RADIATOR, SUBCOOLER CONDENSATE	BOF	1		$38.50	$38.50			
	RADIATOR, OIL COOLER ENGINE	BOF	1		$33.00	$33.00			
	SHUTTER	BOF	1		$66.00	$66.00			
	CONTROLS, SHUTTER	BOF	1			$0.00			

PART NO.	PAGE 9 OF 11 DESCRIPTION	MIW BOF	QTY. REQD	LABOR COST @ $73 HOUR	MATERIAL COST	TOTAL MATERIAL & LABOR COST	PATTERN COST	MATERIAL TYPE	REMARKS
FAN DR. ASSY	FAN DRIVE ASSEMBLY	MIW	1	$18.25		$18.25			
	HOUSING, FAN DRIVE	MIW	1	$36.50	$10.00	$46.50	$2,500.00		
	BEARING	BOF	1		$2.50	$2.50			
	BEARING	BOF	1		$2.50	$2.50			
	SEAL, OIL	BOF	2		$1.50	$1.50			
	SHAFT	BOF	1		$19.00	$19.00			
	PULLEY, FAN DRIVE	BOF	1		$20.00	$20.00			
	KEY	BOF	1		$1.00	$1.00			
	BRACKET, FAN HOUSING	MIW	1	$26.28	$18.00	$44.28	$3,500.00		
	CAPSCREWS BRACKET MOUNTING	BOF	6		$1.80	$1.80			
	WASHERS, BRACKET MOUNTING	BOF	6		$0.60	$0.60			
	SUB-TOTAL					$157.93			
CLUTCH ASSY	THERMOSTATIC CLUTCH	BOF	1		$50.00	$50.00			
	FAN	BOF	1		$38.50	$38.50			
	FAN SPACER	BOF	1		$2.00	$2.00			
	CAPSCREWS, FAN MOUNTING	BOF	6		$1.50	$1.50			
	WASHERS, FAN MOUNTING	BOF	6		$0.60	$0.60			
	NUT, FAN SHAFT	BOF	1		$1.00	$1.00			
	SUB-TOTAL					$93.60			
PULLEY	PULLEY, CRANKSHAFT	MIW	1	$27.81	$10.53	$38.34			
	CAPSCREWS, PULLEY MOUNTING	BOF	6		$1.20	$1.20			
	WASHERS, PULLEY MOUNTING	BOF	6		$0.12	$0.12			
	BELTS	BOF	2		$4.00	$4.00			
	SUB-TOTAL					$43.66			
	SCREW, FAN ADJUSTING	BOF	1		$2.65	$2.65			
	NUT, FAN ADJUSTING	BOF	1		$0.30	$0.30			
	SUB-TOTAL					$2.95			

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PART NO.	PAGE 10 OF 11 DESCRIPTION	MIW BOF	QTY. REQD	LABOR COST @ $73/HOUR	MATERIAL COST	TOTAL MATERIAL & LABOR COST	PATTERN COST	MATERIAL COST	REMARKS
	INSTALLATION BOILER ASSEMBLY		1			$0.00			
	BRACKET, MOUNTING BOILER ASSY	MIW	1			$0.00			
	CLAMPS, MOUNTING BOILER ASSY	BOF	2			$0.00			
	CAPSCREWS BRACKET	BOF	8			$0.00			
	SUB-TOTAL					$0.00			
	EXHAUST PIPE (FROM EXH MFLD)	BOF	1			$0.00			
	FLANGE	BOF	1						
	V BAND CLAMP	BOF	2			$0.00			
	BELLOWS, FLEXIBLE	BOF	2			$0.00			
	SUB-TOTAL					$0.00			
	THERMAL INSULATION (ENGINE EXHAUST SYSTEM)								
	EXHAUST PIPE (STACK BOILER OUTLET)	BOF	1			$0.00			
	V BAND CLAMP	BOF	1			$0.00			
	SUB-TOTAL					$0.00			
	SHIELDS (HIGH PRESSURE & HIGH TEMPERATURE PIPING, BYSTANDER PROTECTION)	BOF							
	BRACKET, RADIATOR	MIW	2			$0.00			
	BRACKET, RADIATOR	MIW	1			$0.00			
	CAPSCREWS	BOF	10			$0.00			
	WASHERS	BOF	10			$0.00			
	SUB-TOTAL					$0.00			

PART NO.	PAGE 11 OF 11 DESCRIPTION	MIM BOF	QTY. REQD	LABOR COST @ $73 HOUR	MATERIAL COST	TOTAL MATERIAL & LABOR COST	PATTERN COST	MATERIAL TYPE	REMARKS
	BRACKET, SHUTTER CONTROLS	BOF				$0.00			
	CAPSCREWS	BOF				$0.00			
	WASHERS	BOF				$0.00			

	SUB-TOTAL					$0.00			
	CONTROL SYSTEM	BOF			$440.00	$440.00			
	ASSEMBLY UNIT	MIM		$146.00		$146.00			
	TEST UNIT	MIM		$73.00		$73.00			
	TOTALS			$1,401.13	$3,289.15	$4,690.28	$67,500.00		

APPENDIX 4

INTERGRATED BOTTOMING CYCLE FOR TRUCK DIESEL ENGINES

ARGONNE NATIONAL LABORATORY
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INTEGRATED BOTTOMING CYCLE FOR TRUCK DIESEL ENGINES

by

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Energy and Environmental Systems Division

February 1987

Work sponsored by

Cummins Engine Company
Columbus, Indiana
under Work for Others Contract No. 31-109-ENG38-85402

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Integrated Bottoming Cycle for Truck Diesel Engines

by

Raj Sekar and Roger L. Cole

ABSTRACT

This study was undertaken to assess the feasibility of incorporating a Rankine Bottoming Cycle as part of a truck type diesel engine. The Organic Rankine Bottoming Cycle (ORBC) that was previously demonstrated by the Department of Energy (DOE) in a heavy duty long haul truck showed about 12% improvement in fuel consumption. However, that system was considered to be too complex and costly to be commercialized. The integrated system described here is an attempt to simplify and reduce the cost of the ORBC system. The main features of the integrated system are: 1. One cylinder of a six cylinder truck diesel engine will be used for power recovery, rather than the turbine and reduction gears employed in the previous ORBC system. 2. Same fluid will be used for engine cooling and as working fluid in the bottoming cycle. 3. The radiator used to cool the engine coolant will serve as the condenser for the bottoming cycle as well. Toluene and steam were considered in this assessment and it was concluded that steam will be more practical working fluid. Steam at 1000 psi, partially vaporized to about 33% saturation in the cylinder head and superheated in the evaporator, is the recommended working fluid. The heat exchanger sizes are smaller than the previously demonstrated ORBC system but still may pose a challenge in under the hood installation of a truck. Design and layout drawings and cost comparisons are beyond the scope of this effort by ANL and are being done separately by the sponsors. Overall the concept appears to be feasible.

1. INTRODUCTION

Rankine cycle has been the mainstay of industrial and utility power generation for over a century. In these applications fuel is directly burned in boilers to generate steam, which is then used for driving a prime mover to generate power. Simultaneously diesel engines were developed to be a highly reliable prime mover for transportation and stationary applications. It is generally well known that both Otto cycle and Diesel engines have thermal efficiencies in the 25%-40% range. This means the remaining fuel input energy of about 60% or more is lost to the ambient through the coolant and the exhaust gases. Reacting to the petroleum price escalations and shortages of the 1970's, governement and industry started research work on utilizing the wasted exhaust energy from engines to generate useful power. The U.S. Department of Energy funded the development and demonstration of the turbocompound system (Ref. 1,2,3) and Organic Rankine Bottoming Cycle (ORBC, Ref. 4,5,6) for long haul heavy duty diesel truck applications. The efficient recovery of waste heat becomes even more critical for adiabatic diesel engine concept, which has been getting considerable attention in recent years. The complexity and cost of implementing the ORBC system in a diesel truck have been the main reason for the industry's reluctance to commercialize the concept. A comparative evaluation of the waste heat recovery systems is reported in Ref. 7. It is clear that unless significant cost reduction and simplifications are demonstrated, ORBC will not be attractive to the industry. Cummins Engine Company and Argonne national Laboratory (ANL) undertook this feasibility study to make the Rankine Bottoming Cycle (RBC) more practical and attractive for commercialization.

2. DESCRIPTION OF THE CONCEPT

The basic ORBC system is shown schematically in figure 1. The working fluid forms a separate loop with its own evaporator and condenser. The engine exhaust gas is the source of heat and a power turbine extracts work from the working fluid and, through reduction gears, feeds the power to the crankshaft. The features included in the integrated system are:

a) The power recovery turbine and reduction gears are eliminated and one of the existing power cylinders is used for the bottoming cycle.

b) The working fluid for the bottoming cycle is the same as the engine cooling fluid.

These two features have system simplification and cost reduction potentials. The engine coolant is pre-heated in the head and therefore has the potential to reduce the size of the evaporator. The schematic diagram of the integrated rankine bottoming cycle is shown in figure 2. Since a large percentage of truck diesels have aftercoolers, air-to-air heat exchangers would be used for that purpose. This approach will allow the coolant to boil in the engine without adversely affecting intake air temperatures. Oilcooling might be done with engine coolant without adverse effects. A portion of the coolant is diverted to the inlet of the booster pump where the coolant is pressurized to 1000 psi before it enters the engine cylinder head. While cooling the head, the coolant is vaporized and steam at an estimated quality of 33% would come out of the head. This working fluid is then routed through a waste heat recovery heat exchanger where superheated steam at 1000 psi is produced from the energy in the engine exhaust gases. This superheated steam is then expanded in one cylinder of the engine. The power generated in this cylinder is designed to be 1/6 th of the rated engine power of a six cylinder truck engine. The exhaust steam is then routed to the truck radiator, where it mixes with the rest of the coolant from the engine and condensed. In order to improve the thermal efficiency, sometimes a regenerator is also included in the system. However, in this application installation space is limited. Hence use of a regenerative heat exchanger in the cycle is not advisable.

3. SELECTION OF WORKING FLUID

Several organic fluids have been considered for use in the bottoming cycle. Most common among them are Toluene, Fluorinol 85 and RC-1 (60% penta-fluoro-benzene, 40% hexa-fluoro-benzene). The DOE demonstration truck used Fluorinol 85. Obviously water has to be considered as a candidate. In addition to the shape of the temperature - entropy diagram, toxicity, degradation temperature, products of decomposition, fire hazard and other physical properties of the fluid must be considered in the selection of the working fluid. Since availability and cost are always important commercial concerns, these characteristics should also be kept in mind. Since the working fluid is the same as engine coolant, the selected fluid should also have good specific heat and other properties required of a coolant. From literature search and previous work done at ANL, the following tables (Tables 1-5) were prepared for a few candidate fluids.

Table 1 contains characteristics of various organic rankine cycle fluids. The significance of the various columns is as follows:

Average molecular weight: A high molecular weight implies a dense vapor in the condenser and a smaller, less costly condenser. A high pressure at 220°F also implies a dense vapor in the condenser. If the condensing vapor were a perfect gas, its specific volume would be:

$$v = \frac{1}{m.w.}\left(\frac{RT}{p}\right)$$

where m.w. is the molecular weight, R is the universal gas constant, T is the absolute temperature, and p is the absolute pressure. Although the condensing vapor is not a perfect gas, the perfect gas assumption gives a rough estimate of specific volume that is adequate for ranking the various fluids.

Maximum use temperature: A high maximum use temperature implies a high theoretical cycle efficiency and maximum extraction of usable energy from the waste heat. Only the values for Fluorinol 85, toluene and water are well known; values for the other fluids listed should be considered optimistic because they are based on static capsule tests. Table 2 gives some rules of thumb for estimating the decomposition temperatures. Maximum use temperatures will be 100-300°F less than the temperatures given in Table 2.

Given an 1100°F exhaust temperature and a 220-250°F condensing temperature, the best fluids can be expected to have about a 20% theoretical cycle efficiency. Therefore, fluid characteristics other than theoretical cycle efficiency are likely to have a major influence on the choice of fluid.

Flow or Freezing Point places a lower limit on system operation although an automatic drain-down system could be designed for a water system.

I-factor is formally defined as:

$$I = 1 - \frac{T/Cp}{\left(\frac{dT}{ds}\right)_D}$$

where D refers to the dewpoint line. Fluids with I greater than about 1.0-1.3 may not operate satisfactorily with turbine expanders, but may be acceptable with positive-displacement expanders (i.e. piston-, vane-, or screw-types). A regenerative heat exchanger will be required for small I-factor fluids, but for I-factors greater than 1.0, the regenerative heat exchanger may not always be required.

Pressure at 220°F: In addition to giving information on the condenser volume, (see also average molecular weight), the pressure at 220°F gives an indication of whether air could leak into the system and oxidize the fluid or whether the fluid could leak out through shaft seals, piston rings, valve packings, and so forth.

Toxicity gives an indication of hazard to personnel. Table 3 (from I. Sax, Dangerous Properties of Industrial Materials) defines the various levels of toxicity as used in Table 1. For comparison purposes, gasoline is rated a moderate-to-high hazard via inhalation.

Fire and Explosion Hazards are further indications of hazards to personnel. Table 5 (from I. Sax, Dangerous Properties of Industrial Materials) defines these hazards. For comparison, gasoline is a dangerous fire hazard and a moderate explosion hazard. No. 2 diesel fuel is a dangerous fire hazard. Additional fire hazard data is given in Table 5.

Toxic Decomposition Products & Toxic Partial Oxidation Products: These products are listed where they are known even if they are produced in very low concentration.

From an analysis of known properties discussed above, water and Toluene were chosen as the two practical fluids for this application. Water is a benign liquid that is easily available and has been widely used both as coolant and Rankine Cycle working fluid. Hence its acceptance will be easy. Besides steam reciprocators are well understood in practice. However there is one important concern with water: it freezes at 32 F. Trucks commonly use 50% ethylene Glycol - Water Mixture. This mixture as bottoming cycle working fluid would create problems with high temperature oxidation and decomposition products. This issue has to be recognized and solved during experimental phase of the project. For this feasibility study, steam and pure water properties were used. Among the organic compounds considered toluene appears to be the most suitable. Availability and cost of toluene are reasonable, primarily due to the industrial research on this fluid over many years. Hence water is recommended as the primary fluid and toluene is the second choice for this application.

4. CYCLE ANALYSIS

Thermodynamic cycle analysis was performed for the following cases:

1. Steam at 1000 psi pressure as working fluid. Four levels of feed water preheat were considered.

2. Steam at 500 psi pressure as working fluid. Four levels of feed water preheat were considered.

3. Toluene at 500 psi as working fluid, without regeneration.

4. Toluene at 500 psi as working fluid, with regeneration.

The following assumptions were made in all the analyses:

Exhaust gas flow rate----------	50 lbs/min
Exhaust gas inlet temp to evaporator	1100 F
Expander efficiency	70%
Booster pump efficiency	70%
Expander outlet pressure	30 psi

The various cycles analyzed are presented in figures 3 - 6. The cycle calculations are described in full detail for the case of steam at 1000 psi.

All other cases followed similar logic.

State point 1 refers to water at the inlet to the booster pump, 30 psi pressure, liquid state

State point 2 refers to water at the outlet of the booster pump, 1000 psi pressure, liquid state

State point 3 corresponds to the pure liquid state of 1000 psi steam. The process represented by 2 - 3 is the feed water preheating phase, which is accomplished to varying degrees in cooling the cylinder head. Cummins estimated that the maximum preheating that could be accomplished within the engine is to generate 33% quality steam, which then can be superheated in an evaporator. The main impact of preheating the working fluid in the engine is to reduce the size of the evaporator.

State point 4 corresponds to the saturated steam state of the working fluid. The working fluid is brought to this state by a combination of preheating in the engine and evaporation in the boiler.

State point 5 is the superheated state of the working fluid. Process 4-5 is the superheating process accomplished in the evaporator.

Process 5-6 is the expansion of the working fluid in one of the power cylinders of the engine. It is in this process that exhaust energy is recovered as useful power. State point 6 is selected to be on the 30 psi pressure line and to conform to our assumption of 70% isentropic efficiency.

Process 6-7 is the condensation of the superheated steam from the power cylinder exhaust to the state of saturated steam. Process 7-1 is the continued condensation to pure liquid state. The entire condensation process is to be accomplished in the radiator.

The diesel engine exhaust gas enters the evaporator at 1100 F and transfers the energy to the working fluid as shown by the line marked "exhaust gas". The slope of this line is chosen to provide reasonable temperature differential in the evaporator, especially at the "pinch point", which is the state point 3.

The cycle analysis for the case of steam at 1000 psi is shown in detail in Table 6. Results of similar analysis for the other cases are shown in Table 7. It is important to note the difference in the shape of the saturation curves for steam and toluene (figures 1 and 3). It is due to the shape of the curve that toluene requires a regenerator to avoid efficiency loss. It can be clearly seen from figures 3 that the working fluid (toluene) still has considerable amount of energy after expansion in the power cylinder, and that a large condenser is required to bring toluene back to the liquid state. The calculated cycle efficiency is in the 20% range for the bottoming cycle and this agrees with previous estimates.

5. HEAT EXCHANGER CONSIDERATIONS

Design, construction and installation of the evaporator poses the biggest challenge in a practical application of the bottoming cycle in a truck. Even though the DOE demonstration truck provided a solution, a production version should be much more compact and cost effective. The integrated concept studied here provides an opportunity to make the bottoming cycle part of the engine and, therefore, should be under the hood of the truck. The critical component to make such an installation possible is the evaporator. This section describes a first cut at the design of the evaporator.

For the purpose of analysis, the evaporator is divided into three distinct sections, namely the economizer, the evaporator and the superheater. Since the source of heat for this heat exchanger is exhaust gas, tube and fin type heat exchanger should be used with the exhaust gas passing over the fins. The tube side heat transfer coefficient can be calculated from equation (1).

$$Nu = Re^{.8} \; Pr^{.4} \qquad \text{Eq. (1)}$$

where:

the Reynold's number, $Re = VD/\nu$,

the Prantl's number, $Pr = cp/k$ and

the Nusselt number, $Nu = hi\ D/k$

The heat transfer coefficient on the exhaust gas side is of the orderof 10 Btu/(hr-ft2- deg. F). Since the tube side heat transfer coefficients are generally two orders of magnitude greater than the fin side heat transfer coefficients, the overall heat transfer coefficient is essentially limited by the gas side. Reference 8 recommends a value of 5-6 Btu/(hr-ft2- Deg. F) for the overall heat transfer coefficient for a heat exchanger with 1/8" thick steel tube. A value of 5.5 was used in this study. Table 8 gives the calculated heat transfer surface areas and box volumes for the evaporator for the various cases. Since the box volume is of great importance to the engine designer, these results are also shown as graphs in figures 7-9 compared to the demonstrated system in the DOE truck. Figure 10 shows one practical design of the evaporator. Since the exhaust side fins are subject to fouling, the recommendation of a heat exchanger manufacturer (Ref 9) should be followed and the fin spacing should be limited to 6 fins/inch.

An examination of figures 7-9 indicates that the evaporator size would be smaller in this integrated bottoming cycle compared to the unit in the DOE demonstration truck. However, the actual size is still too large for "under the hood" installation unless some clever packaging is designed.

The condenser part of the bottoming cycle is the same as the truck radiator. Slightly superheated steam from the expander outlet mixes with the engine coolant at or just before the top tank of the radiator. Since the pressures of the two streams are designed to be the same, it is expected that the mixing process alone will condense the superheated steam into atleast partilly saturated steam. This assumption is reasonable due to the fact that the mass flow rate of the working fluid is much smaller than the mass flow rate of the engine coolant. In order to handle the extra heat load, it is estimated that the radiator will have to be enlarged by 25-30%. More detailed description of the methodology and economic analysis of the bottoming cycles with various working fluids can be found in ref. 10.

6. OTHER DESIGN CONSIDERATIONS

The major difference between this system and the DOE demonstration truck system is the power expander. It is proposed that one of the six cyliders of the engine be used as the power recovery device. Such a reciprocating expander has been studied before (ref. 6,11). Besides, steam reciprocating engine concept is quite old and well known. However, when the reciprocator is integrated into the diesel engine, several design issues must be addressed. The speed and power output of the expander must always be the same as the rest of the cylinders. In a transportation application this might pose a problem at light loads, idle and low speeds when the bottoming cycle is not very efficient. The expander cylider will produce power every revolution whereas the rest of the cylinders of a Cummins engine operate on a four stroke cycle. This means the cam and the valve systems for the expander must be different than for the other five cylinders. The intake valve should be open very rapidly to allow all the working fluid into the cylinder quickly and still have adequate time for expansion work. The exhaust event can be more gradual. The displacement of the expander is determined by the "swallowing capacity" needed to accommodate the required mass of the working fluid at the lowest pressure in the cycle. This calculation is illustrated in Table 9. The same cam that operates the valves and the injectors in the five diesel cylinders can be used for the expander, if the required valve events could be accomplished by simple modifications such as double lobed, steep ramp cam profile. Some innovative valves such as sliding valves should be considered for the expander cylinder. While these changes appear to be difficult, this approach has the advantage of eliminating the turbine and the gear train. A detailed analysis of the reciprocating expander should be done in the next phase of the concept evaluation.

7. CONCLUSIONS

1. The integrated rankine bottoming cycle is a feasible concept for truck applications. This concept presents significant cost reduction and system simplification potentials for the ORBC system which has been previously demonstrated.
2. Evaporator size could be significantly reduced compared to the ORBC system. But the size is still quite large and requires innovative ideas for under the hood installation.
3. Steam at 1000 psi pressure is the first choice for the working fluid and toluene at 500 psi is the second choice. Evaluation of the suitability of 50- 50 ethylene glycol / water mixture as working fluid should be more thoroughly investigated.

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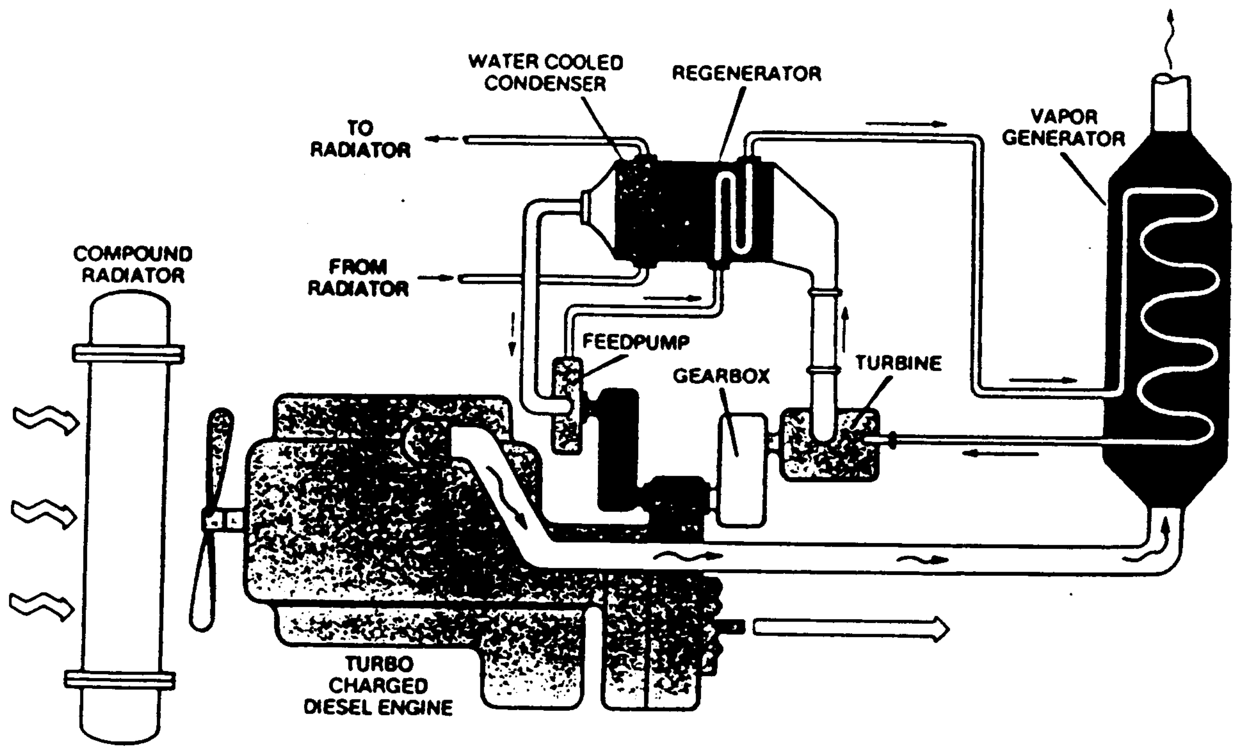
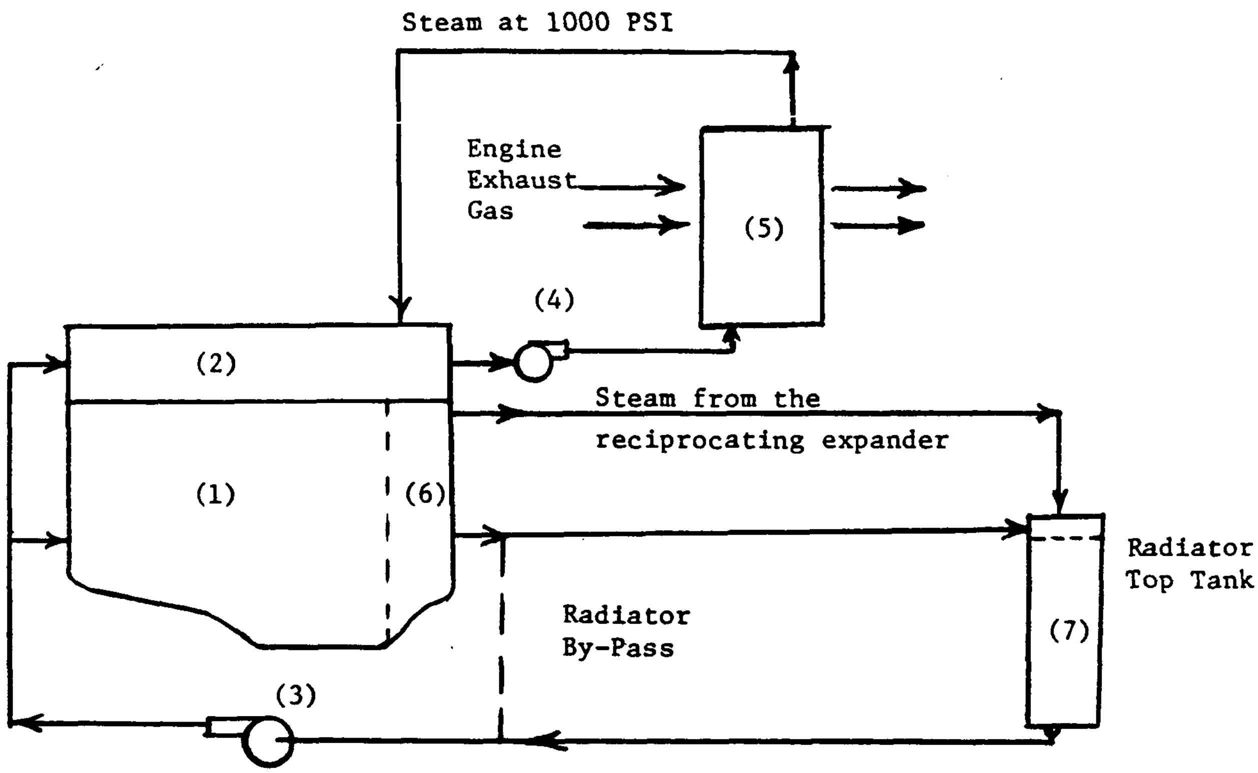


FIGURE 1 Schematic Diagram of the Organic Rankine Bottoming Cycle Demonstrated in a DOE Test Vehicle

(Figure adopted from Reference 4)



(1) Six cylinder truck diesel engine
(2) Engine head
(3) Main engine coolant pump
(4) Bottoming cycle booster pump
(5) Evaporator
(6) One cylinder of the engine used as bottoming cycle expander
(7) Truck radiator, also condenser for the IRBC

FIGURE 2 Integrated Rankine Bottoming Cycle (IRBC) Concept

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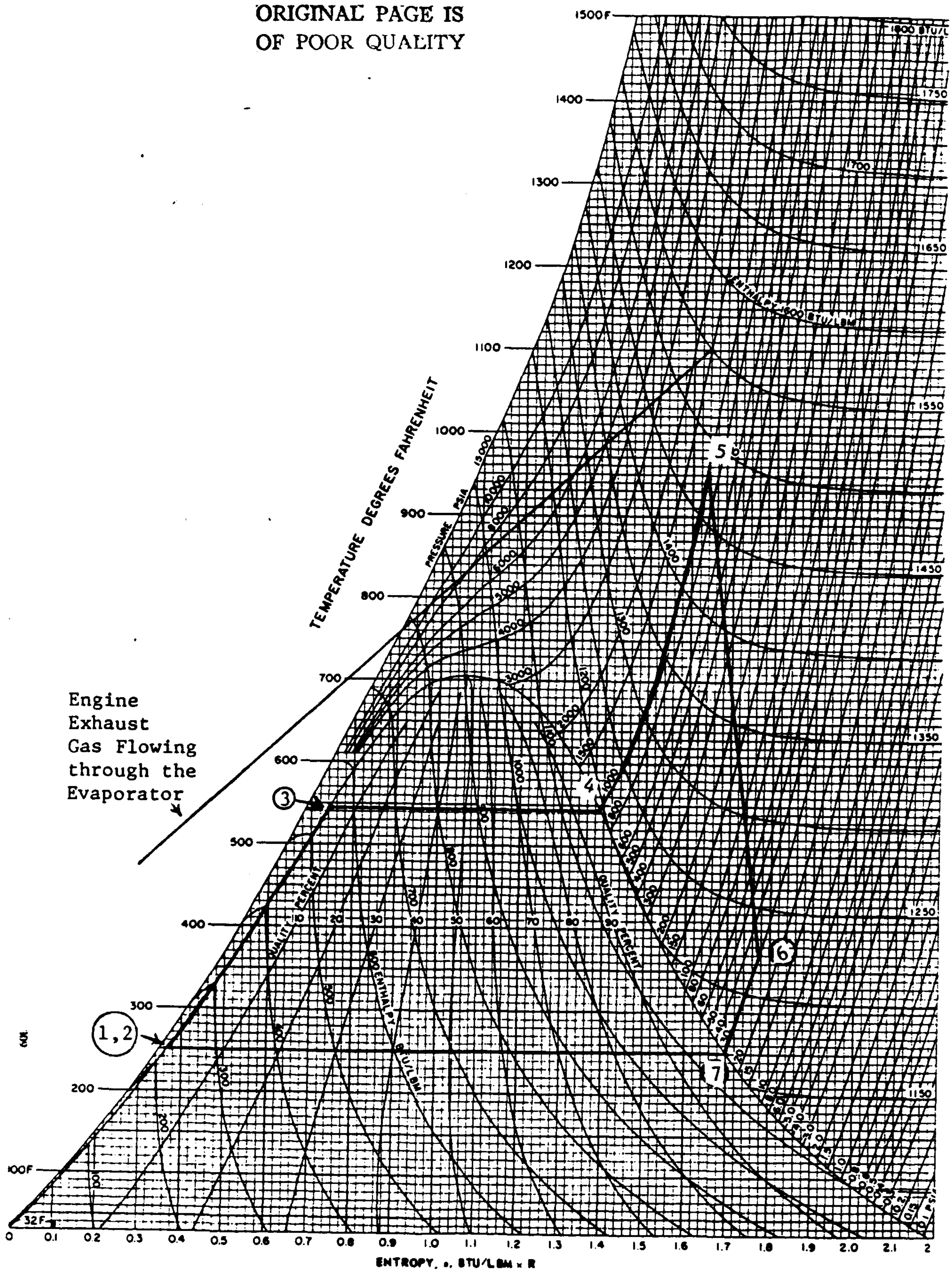

Engine
Exhaust
Gas Flowing
through the
Evaporator

FIGURE 3 T-S Diagram of the IRBC: Steam at 1000 PSI as Working Fluid

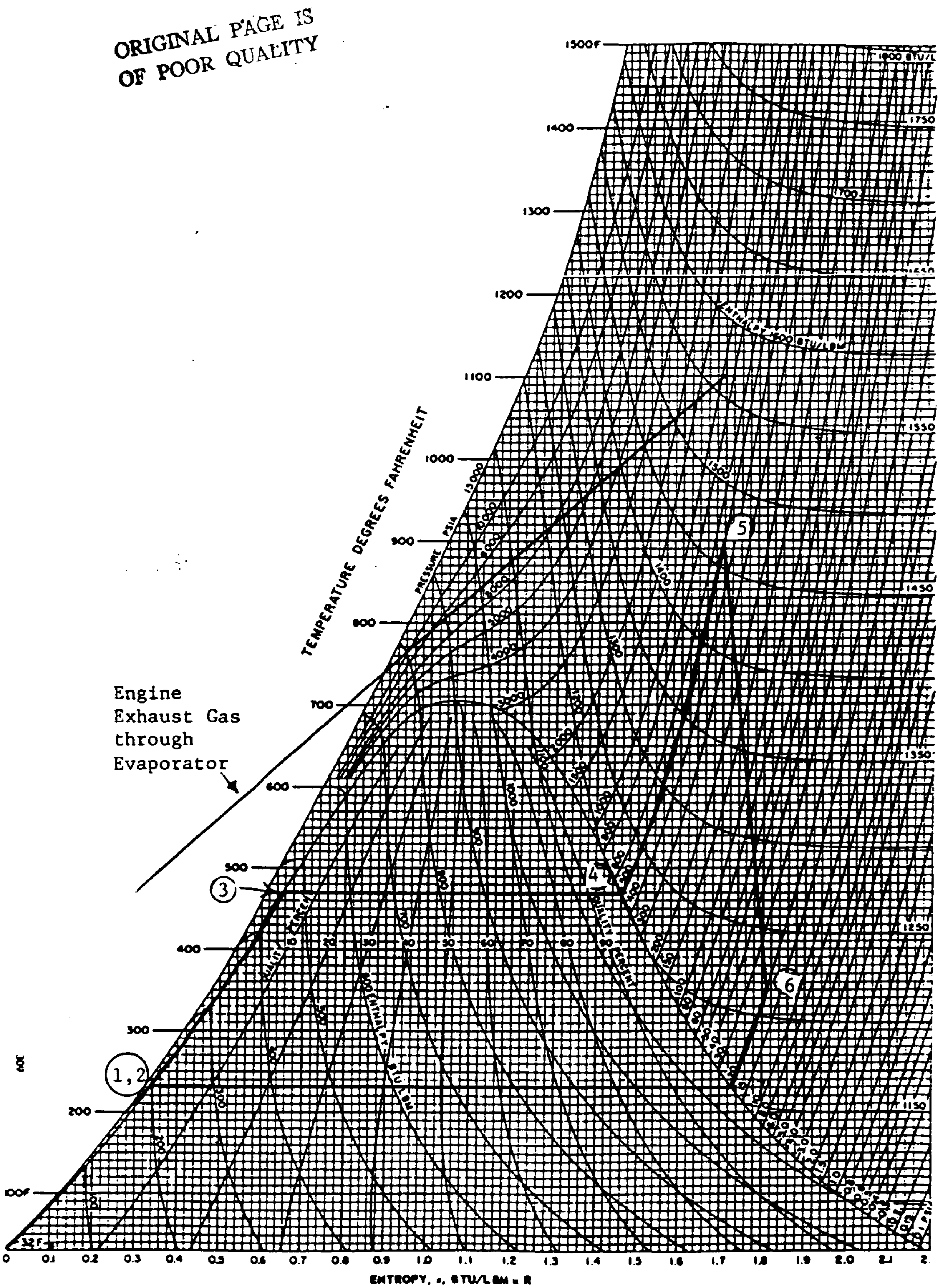

FIGURE 4 T-S Diagram of IRBC: Steam at 500 PSI as Working Fluid

FIGURE 5 T-S Diagram of IRBC: Toluene at 500 PSI with Regeneration

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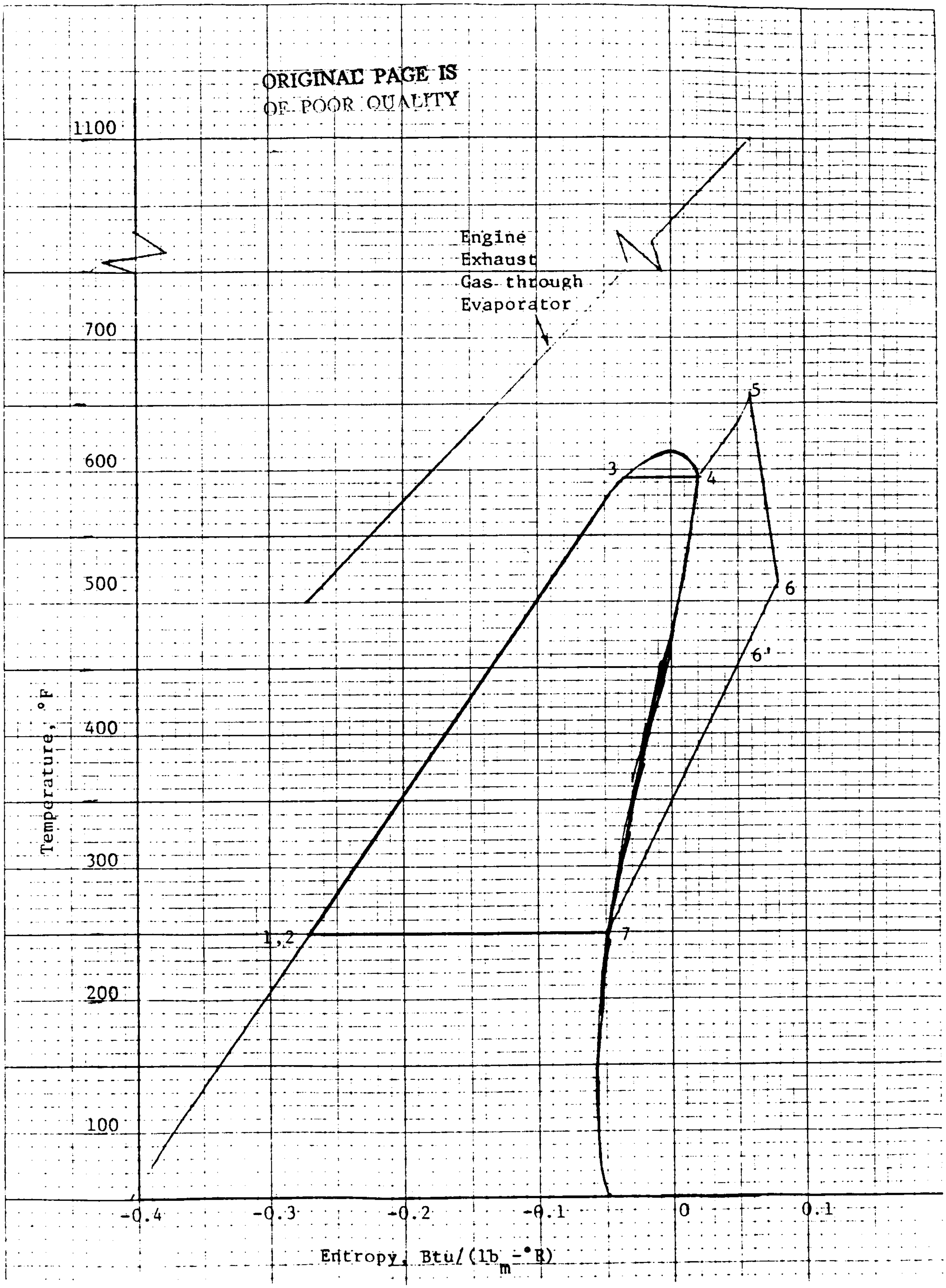
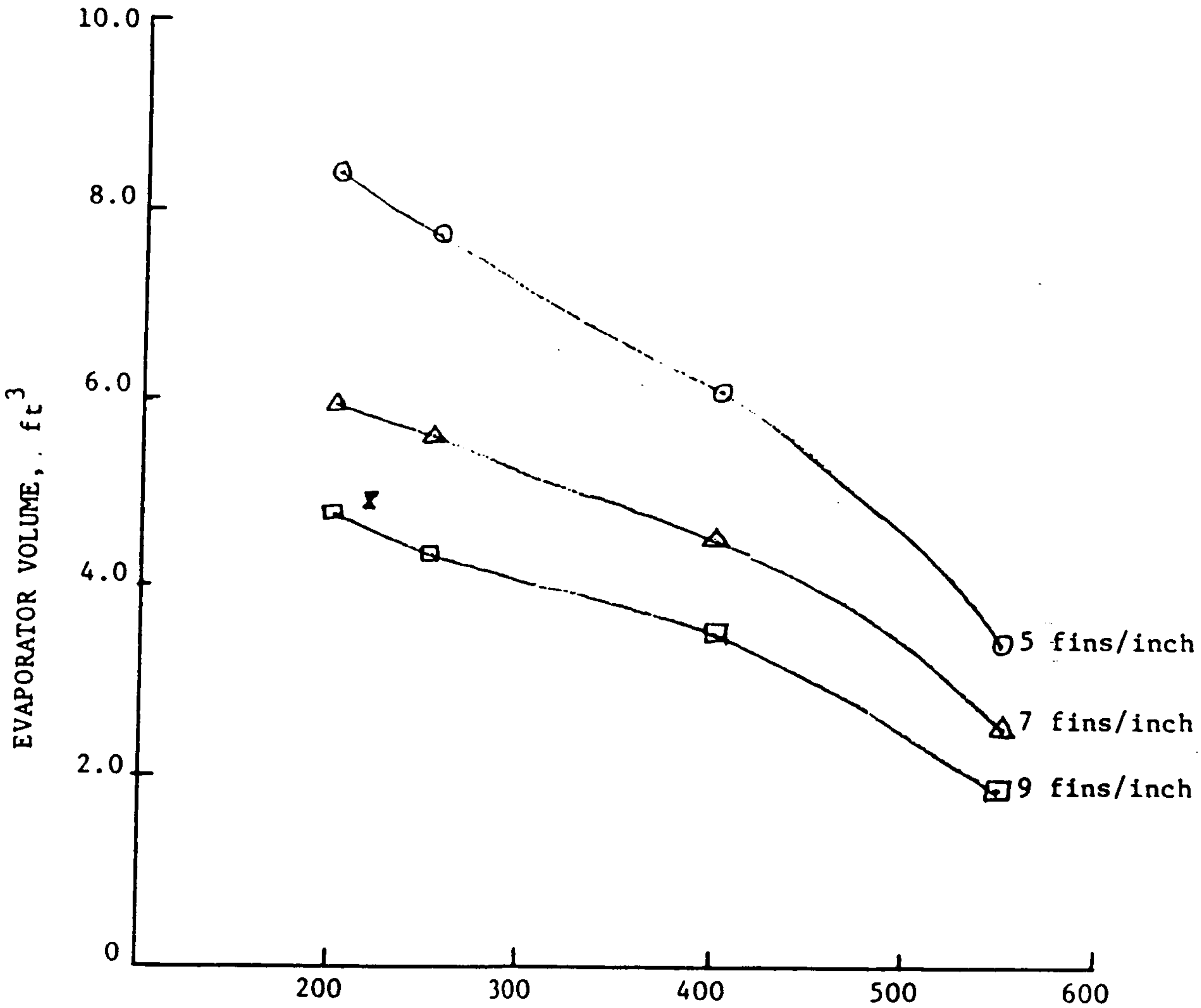


FIGURE 6 T-S Diagram of IRBC: Toluene at500 PSI, No Regenerator

EVAPORATOR SIZE FOR THE INTEGRATED RANKINE BOTTOMING CYCLE FOR TRUCK DIESEL ENGINE

Sream @]000 PSI



Working Fluid Inlet Temperature to the Evaporator, °F

X - Department of Energy Demonstration Truck Evaporator Designed & Built by Thermoelectron

FIGURE 7 **Evaporator Size for the Integrated Rankine Bottoming Cycle for Truck Diesel Engine - Steam @ 1000 PSI**

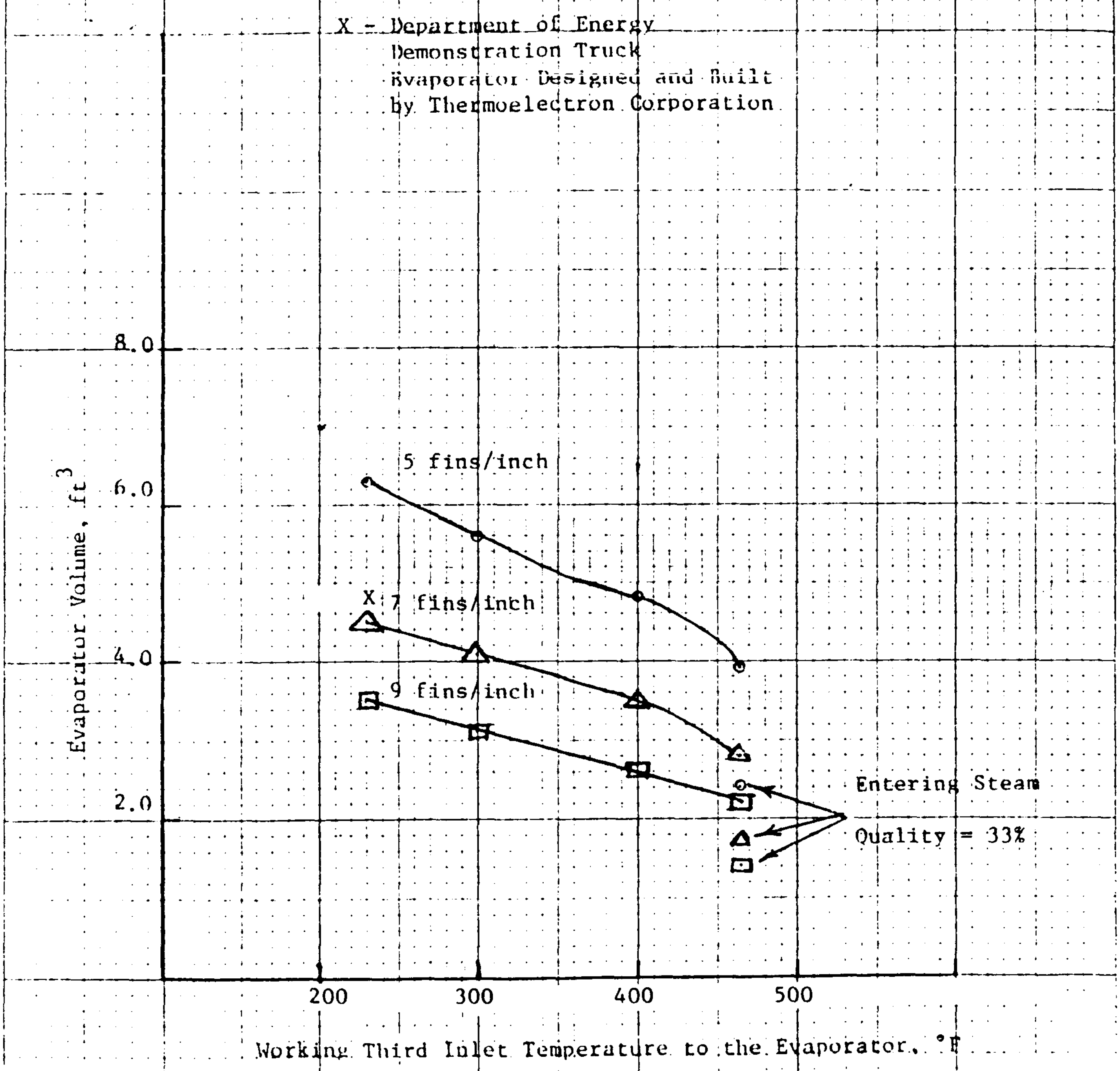
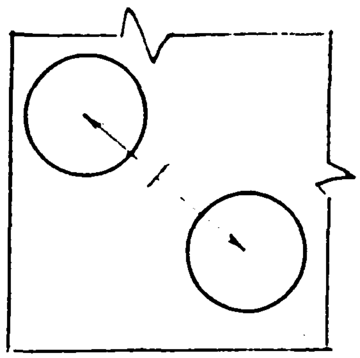


FIGURE 8 Evaporator Size for IRBC: Steam at 500 PSI



Tube

Outside Diameter = 1/2" or 5/8"

Wall thickness = 0.04" to 0.05"

Material = carbon steel

pitch p = 3/4"

Fins

Thickness = 0.015" to 0.020"

Material = low carbon steel

No. of fins/inch = 6

- Multiple tube and fin compact
  Heat exchange design is recommended.
- Brazed tube to fin joint should be used
- Overall dimensions to suit packaging on the engine
- Total Fin surface area should match the figure
  chosen from the graphs.

FIGURE 10 Evaporator Tube and Fin Details

Table 1 Fluid Characteristics

Identification	Average Molecular Weight	Max. Use Temp. °F	Flow or Freezing Point °F	I Factor	Atmospheric Boiling Point °F	Pressure at 220°F psia	Toxicity: inhalation	Toxicity: oral	Toxicity: skin	Fire Hazard	Explosion Hazard	Toxic Decomposition Products	Toxic Partial-Oxidation Products
Fluorinol 85 (85 mole % TFE/15% water)	87.74	550	-82	1.26	168-169	42	high	high				F^-,HF	CO,COF
toluene	92.13	650-750	-139	0.66	231	12.3	mod	low	low	slight	mod		CO
2-methylpyridine/ water (25 mole % 2MP/ 65 mole % water)	44.3	575-670	-40	1.38	200	21	mod	mod	mod	mod-high		HCN	NO_x,CO
RC-1 (60 mole % penta-fluoro-benzene/40 mole % hexafluorobenzene)	175.3	750?	-44	0.72	172	30	low					F^-,HG	COF
50 vol. % ethylene glycol/50% water	40.05	unknown	-30	>1	225	13		mod	mod	slight	mod		CO
50 wt % methanol/50% water	20.04	unknown	-30	2.16 for 100% methanol			mod	high	mod	dangerous	mod		CO
water	18.02	1050	32	2.81	212	17.2	none	none	none	none	none	none	none
benzene	78.1	800?	42	0.89	176	29.4	—[high, carcinogen]—			dangerous	mod		CO
Flutec PP3, $(CF_3)_2C_6F_{10}$	400.08	700?	-67									F^-	COF
monochlorobenzene	112.5	630?	-50	0.74	269		low	mod		dangerous	mod	Cl^-,HCl	CO,COCl
thiophene, C_4H_4S	84.14	550?	-37	1.35 (depends on temp.)	183		mod			dangerous			CO,SO_2
isobutane	58.12	450?	-255		11	313	low			very dangerous	severe		CO
R-113, CCl_2F-$CClF_2$	187.39	300	-31	0.69	118	70						Cl^-,F^-	COCl,COF

Table 2 Thermal Stability Rules of Thumb

Aliphatic Compounds	Generalized Structure	Approximate Decomposition Temperature, °F
alcohols	RCH_2OH	250 - 650
amines	RCH_2NH_2	250 - 650
ketones	$RCH_2\overset{O}{\overset{\|}{C}}-CH_2R$	250 - 450
ethers	RCH_2OCH_2R	500 - 600
acids	$R\overset{O}{\overset{\|}{C}}-OH$	200 - 600
esters	$R\overset{O}{\overset{\|}{C}}-OR'$	350 - 615
hydrocarbons	$RCH_2CH_2CH_2R'$	630 - 680
chlorides & fluorides	RCH_2X	200 - 500
silanes & silicates	R_4Si or $(RO)_4Si$	580 - 680
perfluorocarbons	C_nF_{2n+2}	800 - 900
perfluoroethers	$(RF)_2O$	750 - 850
perfluoroamines	$(RF)_3N$	750 - 850
Aromatic Compounds		
hydrocarbons	ArH, $ArCH_3$, Ar_2CH_2	800 - 1000
ethers, amines, sulfides	$ArOr$, Ar_3N, Ar_2S	850 - 900
chlorides	$ArCl$	700 - 800
fluorides	ArF	800 - 850

Table 3 Definitions of Toxicity

Toxicity	LD_{50}		Approximate lethal oral dose for a 70 kg man
none	>15	g/kg	>1 quart
slight	5-15	g/kg	1 quart
moderate	0.5-5	g/kg	1 pint
high	50-500	mg/kg	1 ounce
serious	1-50	mg/kg	1 teaspoonful
dangerous	<1	mg/kg	a taste

Table 4 Definitions of Fire and Explosion Hazard

Flash point	Fire and explosion hazard
<100°F	dangerous
100–200°F	moderate
>200°F	low

Table 5 Additional Fire Hazard Data

Fluid	Flash Point °F	Fire Point °F	Autoignition Temperature °F
RC-1	none	none	none
Fluorinol 85	105	160	
2-methylpyridine/water	130	145	1060
toluene	40		900
benzene	12		928–1044
100% methanol	54–55		727–878
100% ethylene glycol	232		748–752
kerosene		100–160	
gasoline	-50		536–853
diesel #2	100		494

TABLE 6

Steam Rankine Cycle Expander Cycle Calculations (An Example)

State Point	T	P	v	h
1	250	29.82	.017006	218.59
2	-	1000	-	222.95
3	545	1000	.02159	542.60
4	545	1000	.44596	1192.90
5	950	1000	.7953	1477.10
6'	250	29.82	13.128	1116.67
6	373	29.82	16.440	1224.80
7	250	29.82	13.819	1164.00

<u>Notes</u> a) T = Temperature, ^{o}F

p = Absolute pressure, psia

v = specific volume, c. ft/lb

h = enthalpy, Btu/lb

b) $$h_2 = h_1 + \frac{v_1 \ (p_2 - p_1) \times 144}{\eta_{pump} \quad 778.2}$$

$$h_6 = h_5 - \eta_{exp} \ (h_5 - h_{6'})$$

c) pump efficiency, $\eta_{pump} = 0.7$

expander efficiency $\eta_{exp} = 0.7$

TABLE 6 (cont.)

Sample Calculation

Steam Rankine Cycle - Steam at 1000 psi

Cycle eficiency $= \dfrac{(h_5-h_6) - (h_2-h_1)}{h_5-h_2}$

$$\frac{(1477-1255) - (223 - 219)}{1477}$$

$= 0.198$

Energy input $= \dot{m}\ c_p\ \Delta T$ of exhaust gas

$= 50 \times 0.25\ (110-500)$

$= 7500$ Btu/min

Power output $= \dfrac{7500 \times 0.198}{42.5}$

$= 34.94$ hp $\simeq 35$ H.P.

Theoretical steam flow rate $= \dfrac{7500}{h_2-h_2}$

$= \dfrac{7500}{1477-223}$

$= 5.98$ lb/min

$\simeq 6$ lb/min

TABLE 7 Results of Cycle Analysis for the Different Cases

	Working Fluid	Operating Pressure	Cycle Efficiency	Flow Rate lb_m/min	Expander Power H.P.
1.	Superheated Steam	1000	19.8	5.98	34.9
2.	Superheated Steam	500	19.5	6.10	34.4
3.	Superheated Toluene with Regeneration	500	18.0	32.40	32.1
4.	Superheated Toluene without Regeneration	500	13.6	24.30	24.1

TABLE 8 Heat Exchanger Size Calculation (Example)

1.	Liquid inlet temp. °F	200	250	400
2.	Steam ouitlet temp. °F	950	950	950
3.	Exh. gas inlet temp. °F	1100	1100	1100
4.	Inlet temp diff. °F	470	500	600
5.	Inlet temp diff. °F	270	250	200
6.	Outlet temp diff. °F	150	150	150
7.	*LMTD °F	204	196	174
8.	Heat transfer rate Btu/min	7876	7411	6211
9.	Overall heat transfer coefficient. Btu/hr-ft^2			
10.	Heat transfer area, ft^2	386	378	357
11.	$\frac{\text{Surface area}}{\text{Core Volume}}$ $\frac{ft^2}{ft^3}$	55	55	55
12.	Heat exchanger core volume, ft^3	7.02	6.87	6.49

*Long-MEAN temperature difference

TABLE 9 Calculation of Expander Displacement

Specific volume of steam at exapnder outlet = 16.44 $\frac{\text{c. ft}}{\text{lb}}$

Steam flow rate = 6 lb/min.

= 6 x 16.44 $\frac{\text{c.ft}}{\text{min}}$ 98.64 $\frac{\text{c.ft}}{\text{min}}$

If the engine is reated at 1900 rpm,
Steam flow rate = $\frac{\text{98.64 c.ft}}{\text{1900 rev}}$ 0.0519 $\frac{\text{c.ft}}{\text{rev.}}$

One exh. stroke/rev. displacement = 0.0519 $\frac{\text{c.ft}}{\text{stroke}}$

= 89.68 c.in 90 c.in
Vol. 0.9, Displacement = 100 c.in

APPENDIX 5

ADDITIONAL COST FOR INTEGRAL

BOTTOMING CYCLE ENGINE

VS.

L10 TURBOCHARGED AFTERCOOLED ENGINE

DATE:DECEMBER 17, 1986

ADDITIONAL COST FOR INTEGRAL BOTTOMING CYCLE ENGINE
VS L 10 TURBOCHARGED AFTERCOOLED ENGINE

PRESENT L 10 TURBOCHARGED AFTERCOOLED ENG.					PROPOSED BOTTOMING CYCLE ENGINE				
DESCRIPTION	QTY. REQD ENG.	MATERIAL COST ENGINE	LABOR & M.E. DOLLARS ENGINE	TOTAL MATERIAL & LABOR ENGINE	DESCRIPTION	QTY. REQD ENG.	MATERIAL COST ENGINE	LABOR & M.E. DOLLARS ENGINE	TOTAL MATERIAL & LABOR ENGINE
					SOLENOID VALVE	5	$100.000		$100.000
INJECTOR	6	$260.754	$0.000	$260.754	FUEL NOZZLE	5	$30.000		$30.000
FUEL PUMP	1	$55.944	$126.343	$182.287	FUEL SUPPLY PUMP	1	$60.000		$60.000
SLEEVE, INJECTOR	6	$3.000		$3.000	INJECTOR	5	$100.000		$100.000
					CAMSHAFT BEARING CAPS	7	$42.000		$42.000
					PEDESTAL VALVE GEAR	2	$120.000		$120.000
HOUSING, ROCKER LEVER	1	$116.670		$116.670	CAM COVER	1	$105.000		$105.000
COVER, ROCKER HOUSING	1	$25.580		$25.580	VALVE COVER	1	$130.000		$130.000
					PLENUM INLET CHARGE	1	$115.000		$115.000
MANIFOLD, EXHAUST CENTER	1	$24.320		$24.320	MANIFOLD, EXHAUST -1- PC.	1	$55.000		$55.000
MANIFOLD, EXHAUST ENDS	2	$16.524		$16.524					
HEAD, CYLINDER	1	$191.170		$191.170	HEAD, CYLINDER (5 CYL'S)	1	$160.000		$160.000
GASKET, HEAD	1	$22.570		$22.570	GASKET,HEAD (5 CYL'S)	1	$20.000		$20.000
					HEAD, CYLINDER (1 CYL)	1	$85.000		$85.000
					GASKET, HEAD (1 CYL)	1	$6.000		$6.000
					NIPPLES HIGH PRESSURE BLOCK	6	$21.000		$21.000
					NIPPLES HIGH PRESSURE HEAD	6	$18.000		$18.000
BLOCK, CYLINDER	1	$234.765		$234.765	BLOCK, CYLINDER	1	$305.000		$305,000
					BED CRANKCASE SUMP	1	$195.000		$195,000
PISTON	6	$210.000		$210.000	PISTON (DUCTILE IRON)	6	$210.000		$210,000

PRESENT L 10 TURBOCHARGED AFTERCOOLED ENG.					PROPOSED BOTTOMING CYCLE ENGINE				
DESCRIPTION	QTY. REQD ENG.	MATERIAL COST ENGINE	LABOR & M.E. DOLLARS ENGINE	TOTAL MATERIAL & LABOR ENGINE	DESCRIPTION	QTY. REQD ENG.	MATERIAL COST ENGINE	LABOR & M.E. DOLLARS ENGINE	TOTAL MATERIAL & LABOR ENGINE
					TUBE, OIL RETURN	1	$5.000		$5.000
PISTON COOLING NOZZLE	6	$9.300		$9.300	PISTON COOLING NOZZLE	5	$9.300		$9.300
					OIL COOLER CONNECTION	1	$5.500		$5.500
					GASKET, INJECTOR TIP	6	$2.400		$2.400
					GASKET, INLET PLENUM	1	$15.000		$15.000
ROD, PISTON	6	$71.388	$132.492	$203.880	ROD, PISTON	6	$212.880		$212.880
STUDS, MAIN BEARING CAP	14	$12.432		$12.432	STUDS, MAIN BEARING CAP	14	$70.000		$70.000
					CONDENSER	1	$407.000		$407.000
					VALVE RELIEF	1	$25.000		$25.000
					OVERFLOW & MAKEUP TANK	1	$5.000		$5.000
					PRIMARY CONDENSATE BOOST PUMP	1	$225.000		$225.000
OIL COOLER (TO AIR)	1	$250.000		$250.000	PRIMARY OIL COOLER	1	$150.000		$150.000
THERMOSTAT	1	$0.000		$0.000	THERMOSTAT	1	$5.200		$5.200
					SECONDARY CONDENSATE BOOST PUMP	1	$80.000		$80.000
					OIL SCAVENGE PUMP	1	$95.000		$95.000
					EXHAUST STACK EXCHANGER	1	$700.000		$700.000
ASSEMBLY/TESTS							$50.000		$50.000
TOTAL		$1,504.417	$258.835	$1,763.252	TOTAL		$3,939.280	$0.000	$3,939.280
					TOTAL COST DELETED PARTS				$1,763.252
					TOTAL ADDITIONAL COST FOR BOTTOMING CYCLE ENGINE				$2,176.028

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Report Documentation Page

1. Report No.	2. Government Accession No.	3. Recipient's Catalog No.
NASA CR-180833	-----	------

4. Title and Subtitle	5. Report Date
Technical and Economic Study of Stirling and Rankine Cycle Bottoming Systems for Heavy Truck Diesel Engines	September 1987
	6. Performing Organization Code ------
7. Author(s) I. Kubo	8. Performing Organization Report No. CTR 0723-87001
	10. Work Unit No. -------
9. Performing Organization Name and Address Cummins Engine Company, Inc. Box 3005 Columbus, IN 47202-3005	11. Contract or Grant No. DEN 3-361
	13. Type of Report and Period Covered Contractor Report
12. Sponsoring Agency Name and Address U.S. Department of Energy Office of Vehicle and Engine R & D Washington, DC 20585	14. Sponsoring Agency Code DOE/NASA/0361-1

15. Supplementary Notes

Final Report. Prepared under Interagency Agreement DE-AI01-86CE50162
Project Manager, M. Bailey, Propulsion Systems Division, NASA Lewis Research Center, Cleveland, OH 44135

16. Abstract

For the evaluation of bottoming cycle concepts on heavy duty transport engine applications, following tasks were performed:

1. Develop conceptual design and cost data for Stirling systems.
2. Life-cycle cost evaluation of three bottoming systems: Organic Rankine, Steam Rankine, and Stirling cycles.
3. Suggest future directions in waste heat utilization research.

Variables considered for the second task were initial capital investments, fuel savings, depreciation tax benefits, salvage values, and service/maintenance costs. The study shows that none of the three bottoming systems studied are even marginally attractive. Manufacturing costs have to be reduced by at least 65%.

As a new approach, an integrated Rankine/Diesel system was proposed. It utilizes one of diesel cylinders as an expander and capitalizes the heat energy to the engine coolant. The concept eliminates the need for the power transmission device and a sophisticated control system, and reduces the size of the exhaust evaporator. The system would offer an attractive package for end-users, giving roughly a 20% IRR at a $1.25/gallon fuel price. Further optimization of the system is possible by eliminating/combining some of concepts built in the current design.

17. Key Words (Suggested by Author(s))	18. Distribution Statement
Waste heat recovery Stirling cycle Rankine cycle	Unclassified-unlimited Star Category 85 DOE Category UC-96

19. Security Classif. (of this report)	20. Security Classif. (of this page)	21. No of pages	22. Price*
Unclassified	Unclassified		----

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