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Utilization Of Waste Heat In Trucks For Increased Fuel Economy

Final Report

June 1978

Prepared For
U.S. Department Of Energy
Assistant Secretary For
Conservation And Solar Applications
Division Of Transportation Energy Conservation

Under Interagency Agreement No. EX-76-A-31-1011

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Utilization Of Waste Heat In Trucks For Increased Fuel Economy

Final Report

June 1978

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For
U.S. Department Of Energy
Assistant Secretary For
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DEFINITION OF SYMBOLS AND ABBREVIATIONS

$\overline{C}_p$	average specific heat of exhaust gases at constant pressure
$\overline{C}_v$	average specific heat of exhaust gases at constant volume
$\dot{m}_{ex}$	mass flow rate of exhaust gases
P_{atm}	atmospheric pressure
$P_{exhaust}$	exhaust pressure of base engine
P_{inlet}	inlet pressure of base engine
P_{max}	maximum cylinder pressure of Diesel engine
$Q_{regeneration}$	energy available from regeneration
Q_{total}	total fuel energy supplied to base engine
T_{ex}	temperature of base engine exhaust downstream of turbocharger or power turbine outlet
$T_{reg\ out}$	temperature of base engine exhaust downstream of vapor generator outlet
Δ	difference between values
ϵ_{vg}	effectiveness of vapor generator
η_{act}	actual thermal efficiency of Rankine engine
η_b	brake efficiency of base engine
η_{C_1}	overall efficiency of compressor
η_{fan}	actual fan efficiency

η_{id}	ideal thermal efficiency of Rankine engine using Fluorinol-50 at maximum temperature of 315°C
η_m	mechanical efficiency of base engine
η_{mech}	mechanical efficiency of drive train connecting Rankine engine to base engine
η_{th}	indicated efficiency of base engine
η_{T_1}	overall efficiency of turbocharger turbine
η_{T_2}	overall efficiency of power turbine
BHP	gross brake horsepower of base engine
bmep	brake mean effective pressure of base engine
fmep	frictional mean effective pressure of base engine
$HP_{fan,id}$	ideal fan horsepower requirement for condenser of Rankine engine compounding
imep	indicated mean effective pressure of base engine

ABSTRACT

Trucks currently reject up to 40% of the total fuel energy in the exhaust. Since petroleum costs are continuing to increase, there is growing interest in techniques that can utilize this waste heat to improve overall system efficiency. This report evaluates and compares improvement in fuel economy for a broad spectrum of truck engines and waste heat utilization concepts.

The engines considered are the Diesel, spark ignition, gas turbine, and Stirling. Principal emphasis is placed on the four-stroke Diesel. Because there will be a significant increase in the amount of exhaust energy, the still-to-be-developed "adiabatic" Diesel is also examined.

The waste heat utilization concepts include preheating, regeneration, turbocharging, turbocompounding, and Rankine engine compounding. Predictions are based on fuel-air cycle analyses, computer simulation, and engine test data. All options are evaluated in terms of maximum theoretical improvement, but the Diesel and adiabatic Diesel are also compared on the basis of maximum expected improvement and expected improvement over a driving cycle.

The study indicates that Diesels should be turbocharged and after-cooled to the maximum possible level. At higher boost pressures, the engine power and the fuel economy can be increased, and leaning out the fuel-air mixture or aftercooling the compressor outlet air will reduce the NOx. Turbocharging also increases the potential for turbocompounding if compressor and turbine efficiencies can be maintained. The results reveal that Diesel driving cycle performance can be increased by 20% through increased turbocharging, turbocompounding, and Rankine engine compounding. The Rankine engine compounding provides about three times as much improvement as turbocompounding but also costs about three times as much. Performance for either can be approximately doubled if applied to an adiabatic Diesel.

Additional results indicate that gas turbine performance can be improved substantially through Rankine engine compounding, but because of a lack of energy in the exhaust, only minimal improvement is possible for the Stirling. Except for regeneration, approximately the same improvement is possible for the spark ignition engine as for the Diesel. Because of higher exhaust temperatures, it would be more efficient to regenerate a spark ignition engine.

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SECTION I

INTRODUCTION

Increased fuel costs and diminishing petroleum supplies are forcing both government and industry to reexamine a number of conservation measures. The Office of Highway Systems within the Department of Energy is responsible for managing programs that conserve fuel by improving overall vehicle system efficiency. One such program is currently examining the utilization of waste heat in trucks. Although trucks lose up to 40% of the total fuel energy in the exhaust, much of this can be recovered using techniques such as turbocharging, turbocompounding, or Rankine engine compounding. Industry has evaluated these options, but the results have been difficult to compare since they are based on different engines and analytical approaches. There has also been interest in the potential improvement that could be obtained by combining techniques. The Vehicle Systems Project at the Jet Propulsion Laboratory has conducted this study to better characterize and compare a broad range of waste heat recovery concepts.

SECTION II

GENERAL DESCRIPTION OF ENGINES AND OPTIONS

A. ENGINES

The truck engines included in this study are the Diesel, spark ignition, gas turbine, and Stirling. Diesels are typically used in long-haul trucks because of their relatively low operating costs and long life. Spark ignition engines are more common in smaller, short-haul trucks because of their lower initial cost and weight. While the gas turbine and Stirling engines are expected eventually to offer higher efficiency and lower emissions, they are not yet in commercial service. Due to the limited scope of this study, most of the emphasis was placed on the four-stroke Diesel engine. Because of the moderately high loads and relatively constant driving cycle of a long-haul truck, it is probably the most amenable to waste heat recovery.

To conduct the analyses on a rational basis, representative baseline engines were selected. These are described in Table 1. As can be seen from the table, both a non-aftercooled baseline and an aftercooled baseline were selected for the Diesel. Except for the aftercooling and turbocharging pressure, the engines are similar. The aftercooled engine is turbocharged to a slightly higher level.

Most Diesels are turbocharged and about half are aftercooled in order to increase power and reduce NOx emissions. Turbocharger pressures are currently limited to below 2.7 atm because of turbocharger costs and performance tradeoffs. Peak engine pressures are limited to 1.38×10^7 N/m^2 (2000 psi) because of mechanical and thermal considerations. In order to minimize exhaust smoke, the air-fuel ratio is maintained above 21:1. This represents an equivalence ratio of 0.7.

Figure 1 illustrates the flow of energy in a typical heat engine. The distribution between shaft work, coolant, and the exhaust for the

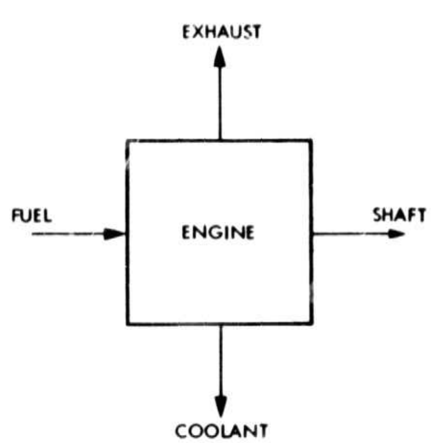

Figure 1. Energy Flow in a Typical Heat Engine

Table 1. Baseline Engines

Engine	Specifications		
		Non-aftercooled	Aftercooled
Diesel Engine	Model	Mack ENDT 675	Mack ENDT 676
	Stroke cycle	4	4
	Displacement	1.1×10^{-2} m^3 (672 $in.^3$)	1.1×10^{-2} m^3 (672 $in.^3$)
	Compression ratio	14.9:1	14.9:1
	Turbocharging pressure[a]	1.8 - 2.1 atm	2.3 - 2.7 atm
	Peak cylinder pressure	1.38×10^7 N/m^2 (2000 psi)	1.38×10^7 N/m^2 (2000 psi)
	Air-to-fuel ratio[a]	21:1 - 29.5:1	21:1 - 29.5:1
	Maximum gross brake power at 1200 RPM	145.5 kW (195 HP)	187.9 kW (252 HP)
	1800 RPM	167.8 kW (225 HP)	214.8 kW (288 HP)
	2100 RPM	166 kW (223 HP)	212.5 kW (285 HP)
	Brake specific fuel consumption[b]	0.225 kg/kW h (0.37 lb/HP h)	0.216 kg/kW h (0.356 lb/HP h)
Spark Ignition Engine	Chevrolet, 350, V-8, naturally aspirated engine		
	Stroke cycle	4	
	Displacement	5.74×10^{-3} m^3 (350 $in.^3$)	
	Compression ratio	8.5:1	
	Air-to-fuel ratio	14.7:1	
	Maximum gross brake power	126.5 kW (170 HP)	
	Brake specific fuel consumption	0.322 kg/kW h (0.53 lb/HP h)	
Gas Turbine	Detroit Diesel Allison, IGT 404, regenerated engine Maximum gross brake power, 224 kW (300 HP)		
Stirling Engine	Philips Ford, Model 4-125 preheated engine Maximum gross brake power: 126.5 kW (170 HP)		

[a] Maximum torque and maximum horsepower points.
[b] Maximum efficiency point.

baseline engines is presented in Fig. 2. Although up to 50% of the energy is lost to the coolant, it is too low in temperature to be useful. However, if this energy were redirected to the exhaust manifold, it could significantly improve the performance of waste heat utilization devices. There could also be some savings in fan power. Cummins Engine Company, Inc., and the U. S. Army Tank Command (Ref. 1) are attempting to develop an "adiabatic" Diesel, which could accomplish this through the use of ceramic liners and components.

Even though an adiabatic Diesel engine is not currently available, an attempt was made to assess the potential improvement in fuel economy that could be obtained through waste heat recovery. It was assumed that the heat loss to the coolant would be eliminated and that the exhaust temperatures would increase in proportion to the exhaust heat. Other characteristics were assumed to be identical to those of the baseline Diesel.

B. DRIVING CYCLE FOR DIESEL TRUCKS

For the analyses of the Diesel, a long-haul truck driving cycle was defined. It should be noted that it is almost impossible to define a truly "typical" driving cycle for Diesel trucks. They operate on different routes and under different engine conditions. Manufacturers compile typical driving cycle data by route or geographic region. The driving cycle used for certifying exhaust emissions compliance in heavy-duty vehicles is a 13-mode cycle, with 3 min at each condition, as shown in Table 2. This cycle makes no attempt to assign weighting values to various speed-load points (other than idle) to represent a realistic driving cycle. It does, however, cover the entire useful operating range of the engine. It was well beyond the scope of this study to assess the performance of engine systems and various subsystems at so large a number of operating conditions. After review of the NAPCA driving cycle used by Thermo Electron Corporation (TECO) in their Rankine engine bottoming cycle work (Ref. 2) and discussions with industry, a three-point "mini-cycle" was selected. This driving cycle is presented in Table 3. The engine would operate between points 1 and 2 while ascending grades or accelerating and around point 3 during cruise.

The mini-driving cycle was designed specifically for high-torque-rise 150- to 225-kW (200- to 300-HP) engines, such as the baseline Diesels, and does not apply directly to engines outside of this general category. Calculations carried out by graphical integration of the NAPCA driving cycle show that the mini-cycle is fairly representative, and additional analysis indicates that load variations of up to 25% will not significantly affect overall results.

C. OPTIONS

Five waste heat utilization options were studied: preheating, regeneration, turbocharging (or increased turbocharging if the baseline is already turbocharged), turbocompounding, and Rankine engine compounding. Preheating the air, fuel, or fuel-air mixture (depending on the engine) occurs prior to compression. Regeneration takes place after compression.

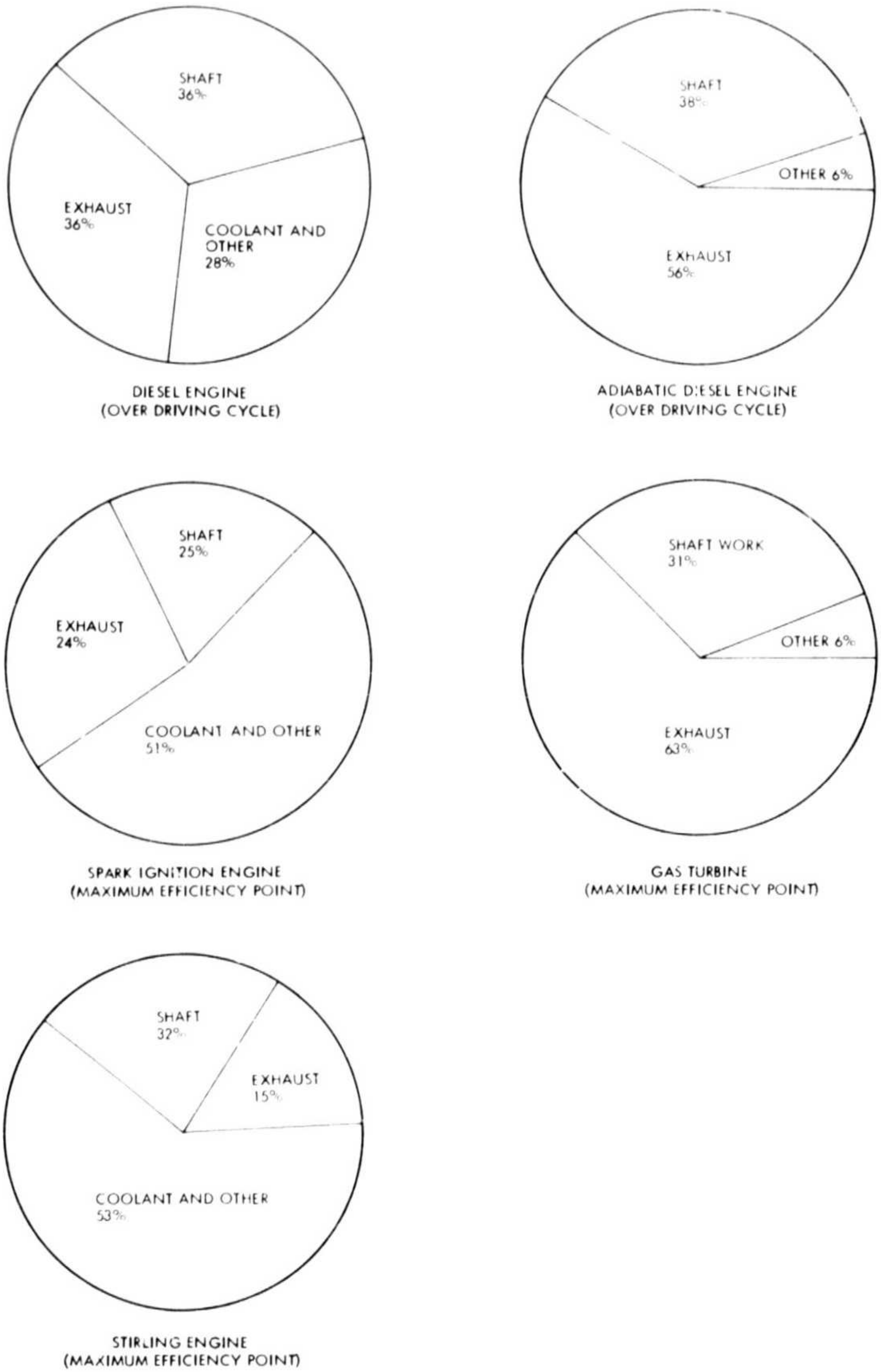

Figure 2. Energy Distribution in Baseline Engines

Table 2. Driving Cycle for Certifying Exhaust Emissions in Heavy-Duty Vehicles

Mode	bmep, %	RPM
1	0	Idle
2	100	Rated
3	75	Rated
4	50	Rated
5	25	Rated
6	0	Rated
7	0	Idle
8	100	Higher of maximum torque and 60% of rated
9	75	Higher of maximum torque and 60% of rated
10	50	Higher of maximum torque and 60% of rated
11	25	Higher of maximum torque and 60% of rated
12	0	Higher of maximum torque and 60% of rated
13	0	Idle

Table 3. Three-Point Mini-Driving Cycle for Diesel Engines

Point	Mode	bmep, %	RPM	Time Weighting
1	Maximum horsepower, maximum speed	100	2100	1/3
2	Maximum torque	100	1200	1/3
3	Cruise	75	1800	1/3

Regeneration differs from recuperation in that, with the former, hot and cold fluids flow alternately over a fixed bed, whereas with the latter, hot and colds fluids flow through separate paths and exchange heat continuously. The terms regeneration and recuperation are often used interchangeably.

An engine is turbocharged to increase engine inlet pressure, flow rate, and density. A typical configuration is presented in Fig. 3. The classical reason for turbocharging has been to increase the power for a given engine size and weight. The inlet density and power can be further increased by aftercooling the compressor outlet air. This will also reduce NOx emissions. Aftercooling can be done through air-to-water or air-to-air heat exchange. An air-to-air aftercooler requires a fan to circulate the cooling air but can generally cool to lower temperatures. One method of driving the fan is to couple it to a tip turbine operated by compressor bleed air (Ref. 3).

In a turbocompounded engine the exhaust gases are expanded in a turbine and the power is transmitted back to the engine crankshaft. The turbocompounding (or power) turbine would generally be placed downstream of a turbocharger turbine if the engine is turbocharged. A typical configuration is shown in Fig. 4. Turbocompounding can also be employed in naturally aspirated engines although to a more limited degree.

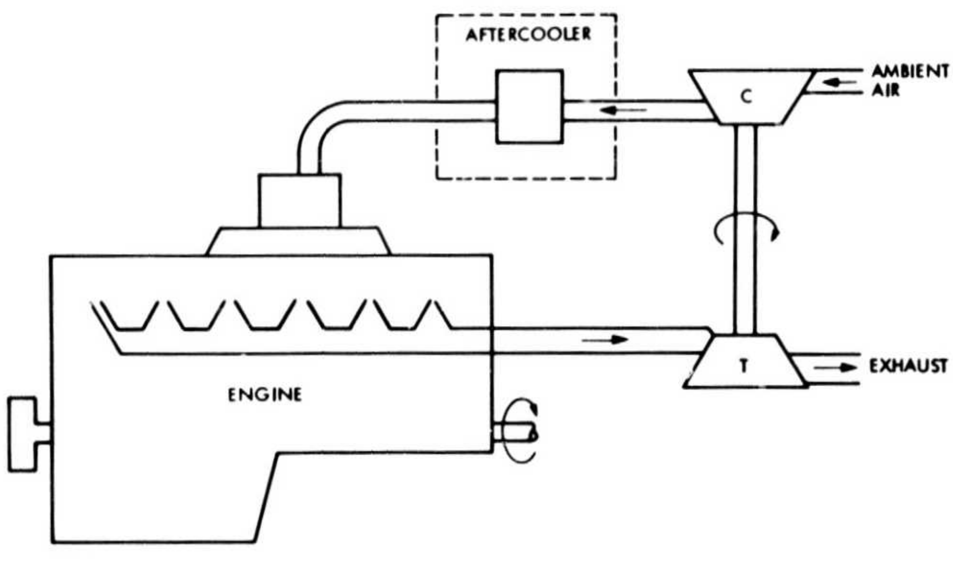

Figure 3. Schematic of Turbocharging

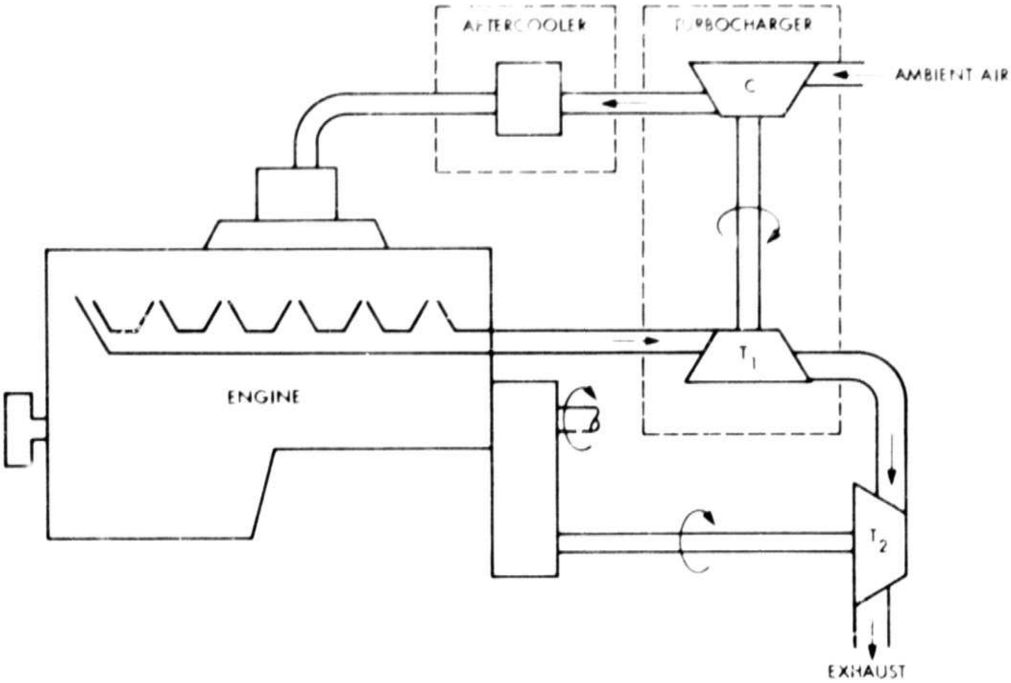

Figure 4. Schematic of Turbocompounding

In Rankine engine compounding, the exhaust energy is used to vaporize a low-boiling-point liquid (usually an organic liquid). Power is transmitted back to the base engine crankshaft through a speed-matching gear box and power takeoff. An overriding clutch may be used to prevent transfer of negative power from the bottoming engine. Figure 5 shows a simplified schematic of one implementation. The exhaust gases of the base engine pass through a vapor generator, vaporizing the working fluid. The vaporized fluid is then expanded in an expander to deliver power. The expander may be either aerodynamic or positive displacement machinery. The expanded fluid passes through a recuperator, releasing most of its heat, and then through a condenser for further cooling. In this configuration, the primary purpose of the recuperator is to reduce the size of the condenser. After flowing through the condenser, the liquid is pumped through the recuperator, where it is heated, and then into the vapor generator to complete the circuit. The flow diagram is shown in Fig. 6.

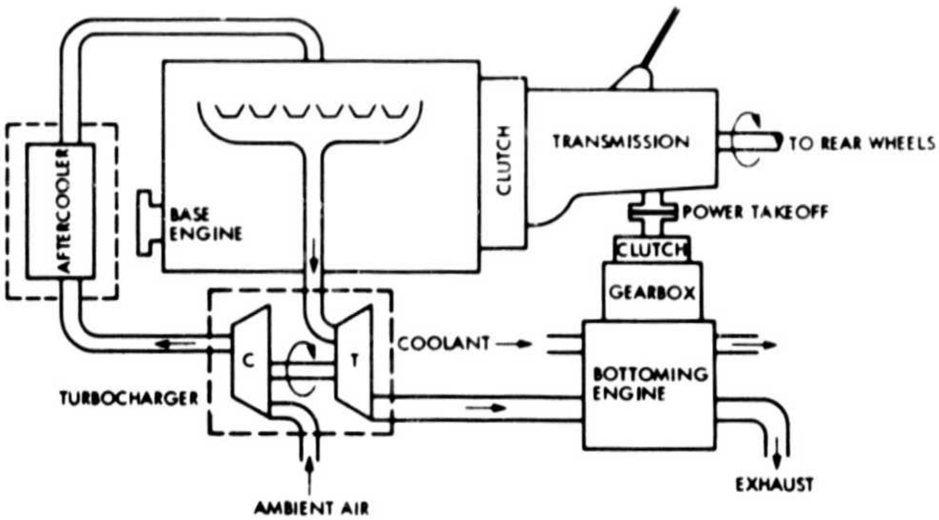

Figure 5. Schematic of Rankine Engine Compounding

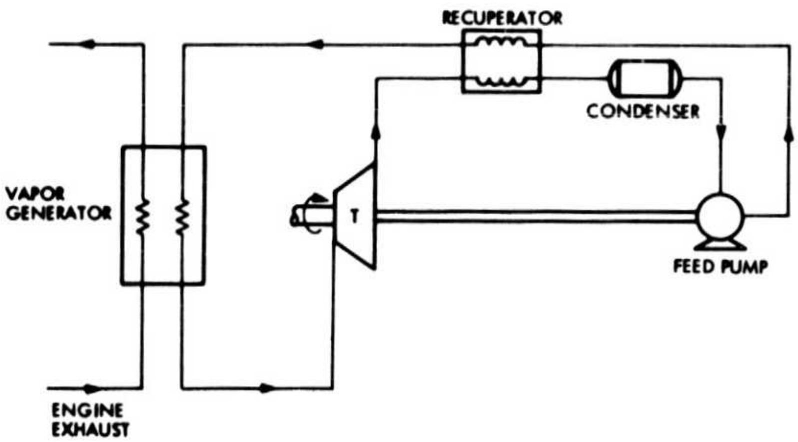

Figure 6. Flow Diagram of Rankine Engine Compounding

SECTION III

APPROACH

At the beginning of the study, an attempt was made to identify the most promising options. The quantity and quality (temperature) of the energy in the coolant and exhaust were estimated. Based on the results, emphasis was placed on the exhaust energy since the low-temperature heat in the coolant is too difficult to recover. The exhaust energy and temperatures of the baseline engines under representative driving conditions are presented in Table 4. As may be noted, a substantial amount of waste heat is available in the exhaust of the adiabatic Diesel and the gas turbine, a moderate amount in the exhaust of the Diesel and the spark ignition, and very little in the exhaust of the Stirling. Exhaust temperatures are highest for the spark ignition and adiabatic Diesel and lowest for the gas turbine and the Stirling.

Each of the options was evaluated in greater depth to identify areas of potential improvement and to assess the impact on engine design and operation. For example, a Rankine engine bottoming cycle is expected to convert waste heat into useful work with essentially no impact on the base engine, whereas turbocharging and turbocompounding can involve complicated interaction with the engine. Based on the assessment, the waste heat utilization matrix presented in Table 5 was established. Only Rankine engine compounding appeared appropriate or feasible for the gas turbine and Stirling engines. The gas turbine is already regenerated, and the Stirling is preheated. As indicated in Table 4, there is not enough energy in the exhaust of the Stirling for regeneration or turbocompounding, and preheating the gas turbine did not appear advantageous since it would reduce flow rate and power.

The study was conducted using engine test data, standard air cycle analysis, and fuel-air cycle analysis employing real gas properties and variable specific heats. Fuel-air cycle results were modified to account for frictional losses and heat loss to the coolant. Actual engine test data were utilized for this purpose. A computer program was written to calculate the improvement to be gained from turbocompounding. The effects of engine back pressure on volumetric efficiency, pumping loss, and exhaust gas temperature were included in the engine model. Visits were made to Mack Trucks, Inc., Cummins Engine Co., Inc., and Garrett AiResearch to discuss assumptions, critical tradeoffs, and overall approach.

Results were calculated for the baseline engines. These are summarized in terms of (1) <u>maximum theoretical improvement</u> in fuel economy, (2) <u>maximum expected improvement</u> in fuel economy, and (3) <u>expected improvement</u> in fuel economy <u>over the driving cycle</u>. Maximum theoretical improvement was calculated assuming realistic engine performance, realistic exhaust gas temperatures, and maximum theoretical waste heat recovery performance, e.g., 100% regenerator effectiveness, 100% thermodynamic and 100% mechanical power turbine efficiencies, and maximum theoretical Rankine cycle efficiency. All of the waste heat recovery options were compared on this basis.

Table 4. Exhaust Temperatures and Waste Heat of Baseline Engines Over Typical Operating Range

Engine	Downstream Exhaust Temperature, °C	Waste Heat, % of Total Fuel Energy
Diesel	400 - 600	30 - 40
Adiabatic Diesel	700 - 950	60 - 65
Spark Ignition	600 - 900	20 - 35
Gas Turbine	150 - 300	55 - 70
Stirling	150 - 300	15

Table 5. Waste Heat Utilization Configuration Matrix

Engine	Option			
	Pre-heating	Regeneration/ Recuperation	Increased Turbo-charging	Rankine Engine Compounding
Turbocharged Diesel	X	X	X	X
Turbocharged Adiabatic Diesel	X	X	X	X
Spark Ignition	-	X	X	X
Regenerated Gas Turbine	-	Baseline	-	X
Preheated Stirling	Baseline	-	-	X

Maximum expected improvement and expected improvement over the driving cycle were calculated only for the Diesel and the adiabatic Diesel. In contrast to maximum theoretical improvement, they were based on realistic performance of the waste heat recovery devices. The maximum expected improvement represents the most that can be obtained at any point in the driving cycle, whereas expected improvement represents an average over the driving cycle.

Fuel economy is defined in terms of kilometers per liter of fuel. Improvement in fuel economy can be calculated by dividing the increase in power by the base engine power per Eq. (1):

$$\% \text{ improvement} = \frac{\Delta kW}{kW} \times 100 \qquad (1)$$

Power generated by waste heat recovery techniques would not have the same transient response as the base engine power and therefore cannot be considered as "true" replacement power. However, under quasi-steady-state conditions, it would be available to reduce fuel consumption.

SECTION IV

DISCUSSION OF RESULTS

The improvement in fuel economy afforded by each of the waste heat utilization options is discussed below. Major emphasis is placed on the Diesel.

A. PREHEATING

Some preheating occurs in Diesel engines as the air flows through the warm intake manifold and inlet ports. If the preheating were increased, the inlet air density would decrease, and this in turn would reduce the mass flow rate and power. In addition, the higher temperatures resulting from preheating would tend to increase NOx emissions. Preheating the fuel, on the other hand, would complicate fuel metering without significantly improving the combustion efficiency (Ref. 4). Because of these deficiencies, preheating was not considered in further detail.

B. REGENERATION/RECUPERATION

To regeneratively heat an internal combustion engine, heat must be transferred from the exhaust gas to the compressed charge prior to combustion. If it were possible to implement this, fuel requirements could be reduced. This is illustrated on temperature-entropy diagrams in Fig. 7. Figure 7a represents a conventional Diesel engine. In Fig. 7b, the compressed charge at point 2 is regeneratively heated to point 2'. The percentage improvement in fuel economy can be calculated by Eq. (2):

$$\% \text{ improvement} = \frac{Q_{regeneration}}{Q_{total} - Q_{regeneration}} \times 100 \qquad (2)$$

where $Q_{regeneration} = \bar{C}_v(T_{2'} - T_2)$.

Although no successful technique for regenerating a Diesel engine has yet been demonstrated, a theoretical analysis was performed to evaluate the potential. The results indicate a maximum theoretical fuel economy improvement of 8% and a maximum expected improvement of 5%. For the latter, a regenerator effectiveness of 70% was assumed.

Regeneration could be equally well applied to spark ignition engines. Because of higher exhaust temperatures, theoretical improvements of up to 25% are possible. If properly designed and implemented, this could be an attractive technique for improving the fuel economy of future spark ignition engines.

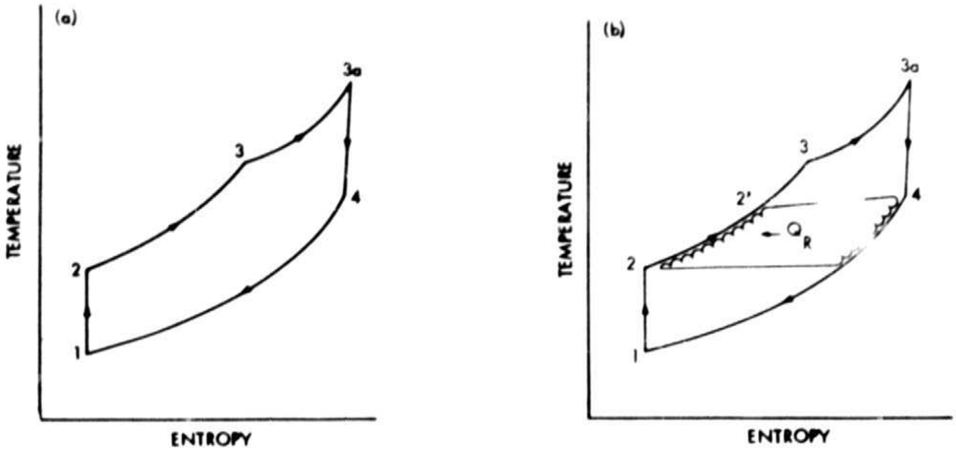

Figure 7. Temperature-Entropy Diagram of Conventional and Regenerated Diesel Engines

C. TURBOCHARGING

Under certain conditions, turbocharging can improve fuel economy without adversely affecting engine performance. This is especially true for Diesel engines. The Diesel turbocharge alternatives considered in this study are:

(1) Allow the maximum cylinder pressure to increase to improve indicated efficiency.

(2) All the mean effective pressure to increase to improve mechanical efficiency.

(3) Reduce the piston speed to increase mechanical efficiency.

(4) Lean out the fuel-air mixture to increase indicated efficiency.

To conduct the analysis a maximum turbocharger pressure had to be established. Based on economic considerations the pressures are currently limited to 2.7 atm. If sufficient incentive is provided, this could be increased to 3 to 3.5 atm. Above this range, the compression pressure would exceed the maximum engine pressure of 1.38×10^7 N/m^2 (2000 psi). If the engine pressure limit were also relaxed, the turbocharger ratio could be increased significantly, provided the fuel-air mixture is not leaned-out. If it is leaned-out, an energy limit is reached at about 4 atm. Based on all of the above considerations, a maximum turbocharger pressure of 3 atm was assumed. To maintain this limit at high engine speeds, some waste-gating may be necessary. The analysis also assumed that turbocharger efficiencies could be maintained at higher boost pressures. If this cannot be accomplished, the benefits from turbocharging will decrease. From this point of view, the turbocharging results may be optimistic. The turbocharging options are discussed below.

1. Increasing the Maximum Cylinder Pressure

Although fuel economy can be improved by raising the cylinder pressures, Diesels are currently limited to 1.38×10^7 N/m^2 (2000 psi). Increasing the maximum pressure will either adversely affect durability or require redesign of the engine. Durability is a primary selling point for Diesels, and retooling for stronger components may not be economically feasible. Nevertheless, the theoretical improvement was calculated.

Raising the maximum pressure tends to make the Diesel cycle approach the Otto cycle (spark ignition) thermodynamically. Since heat addition at constant volume is more efficient than at constant pressure, the indicated efficiency is increased. The engine indicated mean effective pressure will also increase. Because there will be only a relatively small increase in friction at higher pressure (Ref. 5), the mechanical efficiency will probably also improve. This is illustrated by Eq. (3):

$$\eta_m = \frac{\text{imep} - \text{fmep}}{\text{imep}} \tag{3}$$

The increase in indicated and mechanical efficiencies results in an increase in brake efficiency. Analysis indicates that if the maximum cylinder pressure could be raised to 2.06×10^7 N/m^2 (3000 psi), fuel economy would be increased by 10 to 13%. Unfortunately, the combustion temperatures would increase by 125°C, probably increasing the NOx. Because of engine design and NOx constraints, this option may not be feasible.

All the remaining options are based on maximum cylinder pressures of 1.38×10^7 N/m^2 (2000 psi). It is assumed that this is accomplished through retardation of the ignition timing. Fuel-air cycle analysis indicates that over the limited turbocharge range considered retarding is slightly more efficient than reducing compression ratio. Little or no net improvement in brake efficiency can be obtained if a Diesel is turbocharged to a higher pressure and reduced in compression ratio. However, some modest improvement can be gained by turbocharging and retarding. Some increase in NOx emissions may occur unless the fuel-air mixture is leaned out. Aftercooling will alleviate this potential problem but may not eliminate it. If the compressor outlet air on the non-aftercooled engine were cooled from 140 to 60°C, it would result in a 95 to 120°C decrease in peak temperature. This amount of aftercooling will also increase the power by 16% and the brake efficiency by 2%.

2. Increasing Mean Effective Pressure to Increase Mechanical Efficiency

By increasing the turbocharge pressure of the baseline non-aftercooled engine to 3 atm and retarding to maintain constant maximum pressure, the indicated mean effective pressure and power can be increased by 37%. This will increase the mechanical efficiency by 7%. Since the indicated efficiency is decreased by 3% due to retardation, the improvement in brake efficiency is only 4%. Average combustion temperature would increase by less than 100°C.

3. Reducing Piston Speed to Increase Mechanical Efficiency

The brake efficiency can be further increased if the piston speed is decreased. One approach is to reduce the rear axle ratio. Since the piston friction is proportional to the mean piston speed (Ref. 5), there could be a measurable improvement in the mechanical efficiency. There could also be a slight penalty in the indicated efficiency due to poorer mixing of the charge. Turbocharging to 3 atm, retarding, and reducing piston speed to maintain constant power yielded about 9% improvement in mechanical efficiency. Indicated efficiency is expected to be slightly lower than in alternative 2. The net improvement in brake efficiency is predicted to be between 4 and 6%. About the same improvement is expected over the driving cycle.

4. Leaning-out the Fuel-Air Mixture to Increase Indicated Efficiency

Another alternative for increasing fuel economy is to turbocharge and lean-out the fuel-air mixture, maintaining constant power. The leaned-out mixture reduces the combustion temperatures and thus the specific heats. Because of the lower specific heats, there is a higher temperature rise per unit mass of fuel. In addition, the residuals tend to have a lower molecular weight. Both of these effects increase indicated efficiency. Since the mechanical efficiency remains essentially unaffected, the net result is an increase in brake efficiency. If the turbocharging is increased from 1.8 to 3.0 atm, the equivalence ratio must be reduced from 0.7 to 0.5 to maintain constant power. Fuel-air cycle analysis indicates about 3% improvement in fuel economy. Since the peak combustion temperature decreases by 315°C, there will also be a reduction in NOx.

Further improvement in performance is possible for engines operating at lower turbocharge ratios. To a limited degree, manufacturers have already taken advantage of this to reduce NOx without paying large penalties in efficiency or power. This potential is plotted in Fig. 8. Turbocharging results are summarized in Table 6.

5. Turbocharging a Spark Ignition Engine .

As discussed above, turbocharging can improve Diesel fuel economy without sacrificing power. This may not be true for short-haul trucks with spark ignition engines. These engines cannot be turbocharged much

Table 5. Effects of Various Turbocharging Alternatives on Diesel Performance (ENDT 676)

Alternative	Point on Cruising Cycle	Effect on Base Engine: Power %	Indicated Efficiency	Mechanical Efficiency	Brake Efficiency	Maximum Expected Fuel Economy Improvement %	Comments
Increase maximum engine pressure from [illegible] to [illegible] N m⁻² ([illegible] to 3000 psi)	Maximum torque	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	Peak combustion temperatures would increase by [illegible] and exhaust temperatures would decrease by [illegible]. This might result in increased NO emissions. The higher pressures and temperatures would adversely affect the durability of the engine.
Increase turbocharging pressure from [illegible] to [illegible] atm and retard to maintain maximum pressure at [illegible] N m⁻² (2000 psi)	Maximum torque	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	Peak temperatures would decrease slightly, but the maximum compression temperature would increase by [illegible] and the exhaust temperature by [illegible]. This might tend to increase the NO emissions. The fuel economy improvement over the cruising cycle is estimated to be slightly less than 4%. Approximately half of this improvement is possible for the aftercooled baseline engine since it is already turbocharged to a higher level.
Increase turbocharging pressure from [illegible] to [illegible] atm and decrease piston speed by 20% to maintain constant power	Maximum torque	Constant	[illegible]	[illegible]	[illegible]	[illegible]	Lower piston speeds would adversely affect sizing. About the same fuel economy improvement ([illegible]%) is expected over the cruising cycle.
Increase turbocharging pressure from [illegible] to [illegible] atm and reduce equivalence ratio from [illegible] to [illegible] to maintain constant power	Maximum torque	Constant	[illegible]	Constant	[illegible]	[illegible]	Peak combustion temperatures would decrease by [illegible] and exhaust temperatures by [illegible]. This might help in reducing the NO emissions. About [illegible]% fuel economy improvement is expected over the cruising cycle.
Aftercool compressor outlet air from [illegible] to [illegible] and retard to maintain maximum pressure of [illegible] N m⁻² (2000 psi)	Maximum torque	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	Peak temperatures would decrease [illegible] and exhaust temperatures would decrease slightly. This might reduce the NO emissions. About the same fuel economy improvement ([illegible]%) is expected over the cruising cycle.
Aftercool and lean out to maintain constant power	Maximum torque	Constant	[illegible]	Constant	[illegible]	[illegible]	This might help in reducing the NO emissions significantly because of reduced peak combustion temperatures and exhaust temperatures. A fuel economy improvement of about [illegible]% is estimated over the cruising cycle.

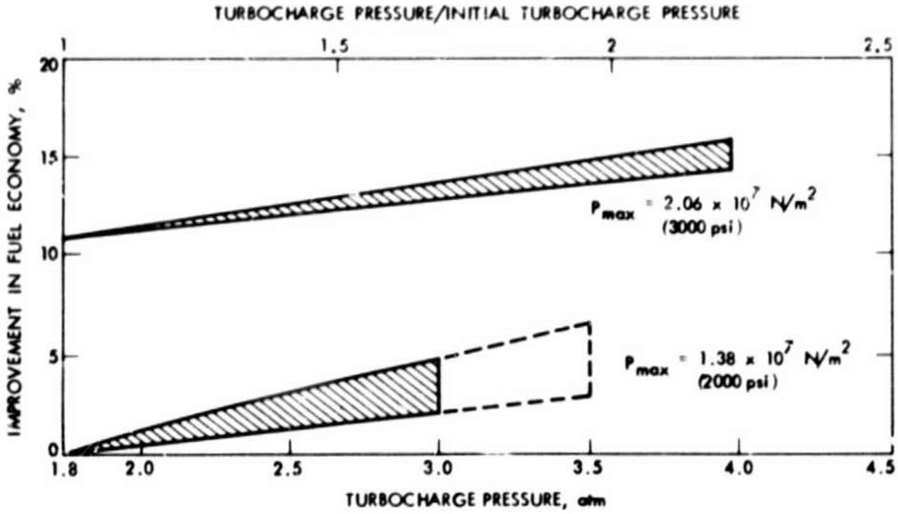

Figure 8. Effect of Turbocharging on Fuel Economy of Diesel Engines

without reducing the compression ratio or there will be detonation and NOx problems. Since the efficiency of a spark ignition engine decreases rapidly with decreasing compression ratio, this is not a favorable tradeoff. Even so, by aftercooling and using fuel with better anti-knock characteristics, some turbocharging may be possible. The results of fuel-air cycle analysis indicate that a maximum theoretical improvement of 5% is possible if the naturally aspirated baseline engine is turbocharged to 2 atm.

D. TURBOCOMPOUNDING

Because of relatively low upstream manifold pressures, only 15 to 30% of the exhaust energy is available to a power turbine. Fortunately, turbines operate at fairly high efficiencies of 60 to 80%. Another major limitation of turbocompounding is that it increases engine back pressure.* Higher back pressure increases the engine pumping losses, reduces volumetric efficiency and power, and can cause reverse flow. Although an upper limit could not be firmly established, it was assumed that the back pressure could not exceed the inlet pressure by more than 6.9 x 10^4 N/m^2 (10 psi).

It is important to understand how exhaust energy is converted to useful power. Figure 9 represents the exhaust energy distribution on a P-V diagram for a turbocharged four-stroke engine. The solid line in the diagram (4-5-5'-6) represents the exhaust process. The

* The back pressure is defined as the average exhaust manifold pressure immediately downstream of the exhaust valves.

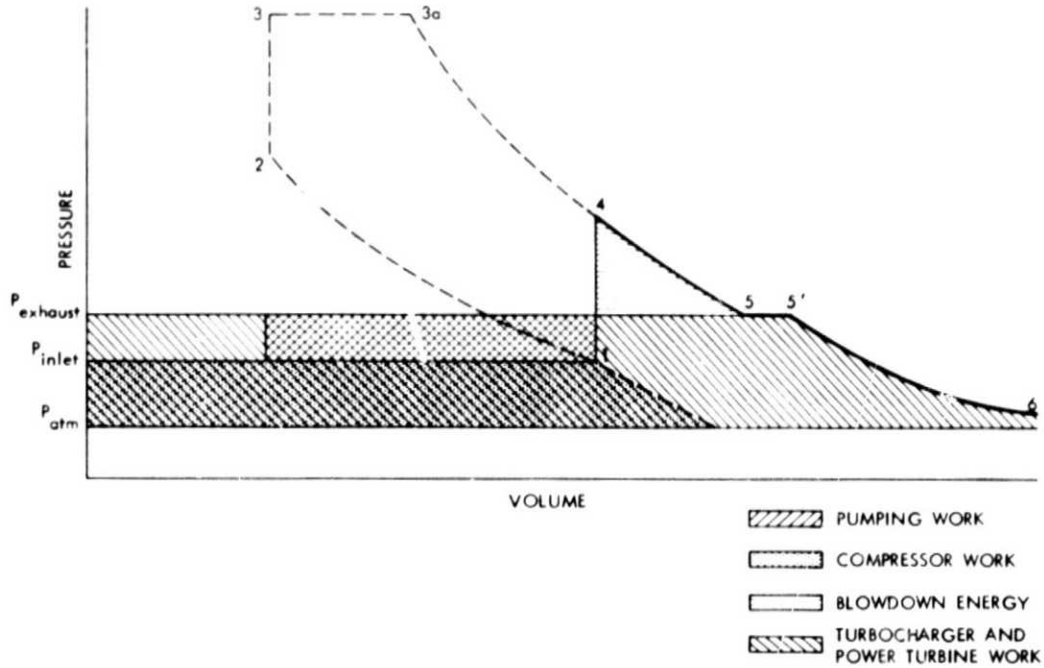

Figure 9. Energy Available from Ideal Exhaust Process

hot gases at the end of the expansion stroke (point 4) suddenly expand to a low exhaust manifold pressure (point 5). This expansion process from 4 to 5 is called the blowdown process. Because of the very short time interval, it is difficult to capture the kinetic energy acquired during this process. When the expanding gases stagnate, the kinetic energy is converted to heat (process 5-5'). If the process is adiabatic, no energy will be lost but the thermodynamic availability will be decreased. The hot gases at point 5' are then expanded to point 6 in two steady-flow turbines. The first turbine drives the turbocharger compressor and the second generates power that is transmitted to the crankshaft.

Figure 9 also identifies the engine pumping work, the energy required to drive the compressor, and the turbine output. The pumping work represents the work done by the engine during the inlet and exhaust strokes. If the exhaust pressure is higher than the inlet pressure, the pumping work is negative. These pumping losses can detract from the engine output. Even though the figure illustrates theoretical pumping work, actual work is assumed to be 50% higher if negative or 50% lower if positive because of flow restrictions in the manifolds and valves. Inlet pressures tend to be lower than theoretical and exhaust pressures higher.

Turbine output and pumping losses both increase with increasing back pressure, but because of the slope of the adiabatic line, turbine work does so at a decreasing rate. The system is theoretically optimized when the power turbine output minus the pumping loss is maximized.

Additional improvement can be obtained if the engine is turbocharged to higher inlet pressures and reasonable turbocharger efficiencies can be maintained.

Some of the turbocompounding alternatives examined in this study are discussed below. They include the following:

(1) Allow the engine back pressure to increase.

(2) Maintain a constant pressure differential across the engine by increasing the turbocharge ratio.

(3) Increase the utilization of blowdown energy.

(4) Increase the amount of energy in the exhaust by retarding the injection timing.

1. Increasing Engine Back Pressure

In this case, inlet pressure was maintained constant as back pressure was increased. The effects on pumping work, volumetric efficiency, engine power, and turbine power were simulated with a computer program. Fuel flow rate was decreased as volumetric efficiency decreased in order to maintain a constant equivalence ratio. Turbocharger and power turbine component efficiencies were obtained by evaluation of test data and varied parametrically. Calculations were made at all three points in the driving cycle. Results for the non-aftercooled baseline Diesel are plotted in Figs. 10, 11, and 12. The results for the baseline aftercooled engine are plotted in Fig. 13. The expected improvement in fuel economy illustrated in Figs. 10 and 13 is less than that in Fig. 11 because of lower component efficiencies at the higher boost pressures. The sensitivity of turbocompounding performance to exhaust temperatures is shown in Fig. 12.

It can be seen from the Figs. 10 to 13 that the peak improvement in fuel economy occurs with pressure differentials of -3.5×10^4 to -6.9×10^4 N/m^2 (-5 to -10 psia). It can also be noted that the results are a strong function of the turbocharger and power turbine efficiencies. For the baseline non-aftercooled Diesel, a maximum theoretical improvement of 11% is predicted. The maximum expected improvement is 6%, and the expected improvement over the driving cycle is 3 to 4%. Only half as much improvement is possible for the aftercooled baseline because of the lower effective turbocharger component efficiencies.

2. Increasing Turbocharge Ratio to Maintain Constant Pressure Differential Across the Engine

There are several reasons for increasing the turbocharge ratio. Higher boost pressures can alleviate problems associated with high back pressures, improve base engine performance, and increase the amount of energy available to the power turbine. For the analysis, it was assumed that the turbocharge pressure was increased to 3 atm without

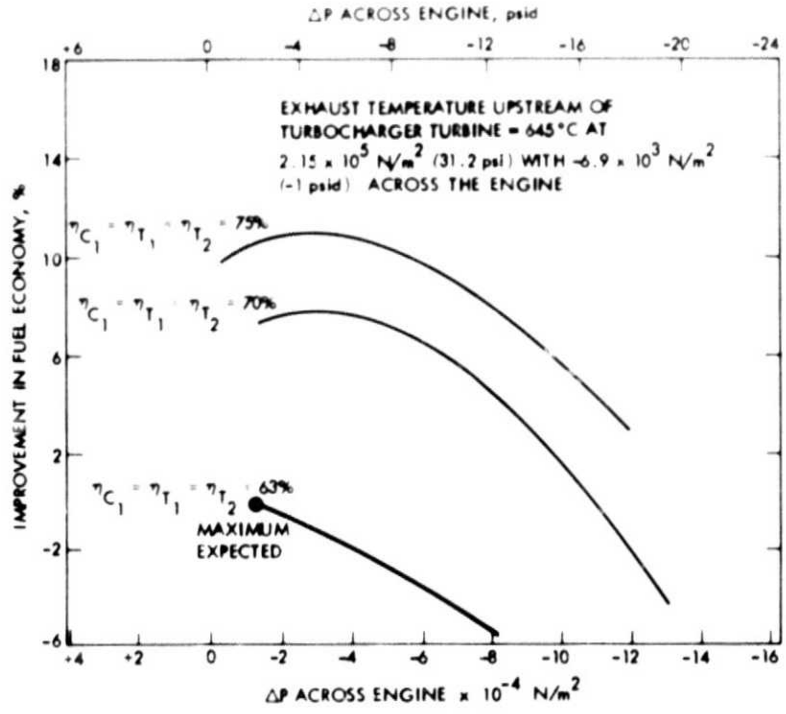

Figure 10. Effect of Back Pressure and Component Efficiencies on Turbocompounding Performance of Baseline Non-aftercooled Engine at Maximum Horsepower

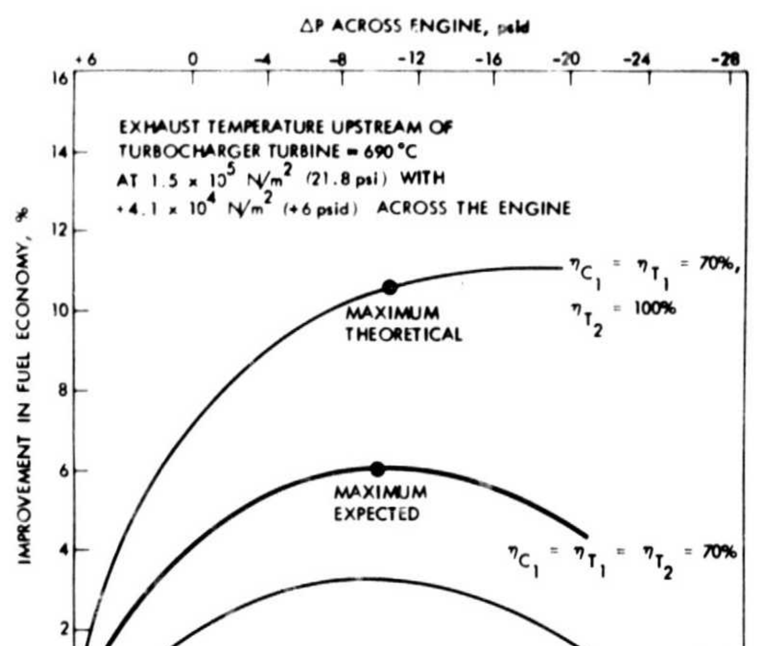

Figure 11. Effect of Back Pressure and Component Efficiencies on Turbocompounding Performance of Baseline Non-aftercooled Engine at Maximum Torque

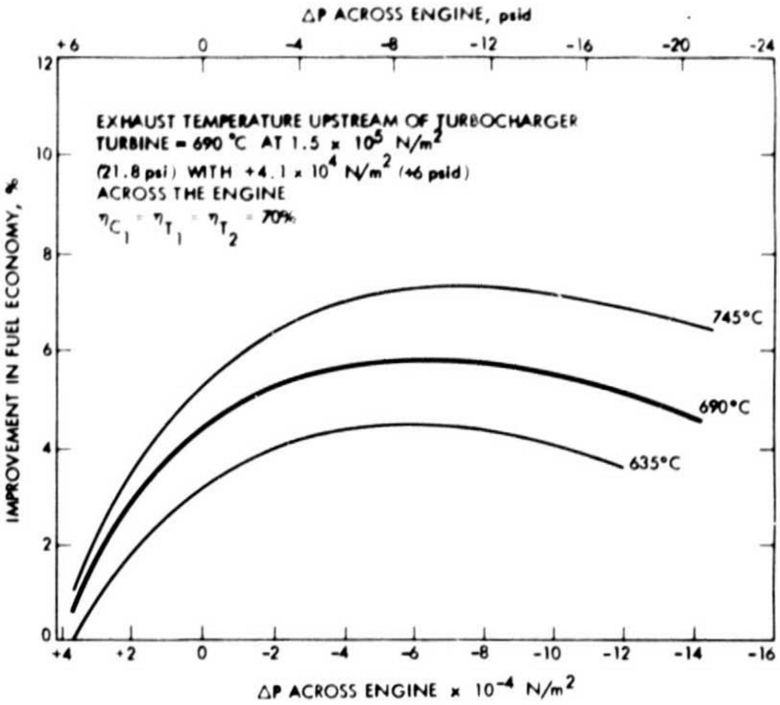

Figure 12. Effect of Exhaust Temperature on Turbocompounding Performance of Baseline Non-aftercooled Engine at Maximum Torque

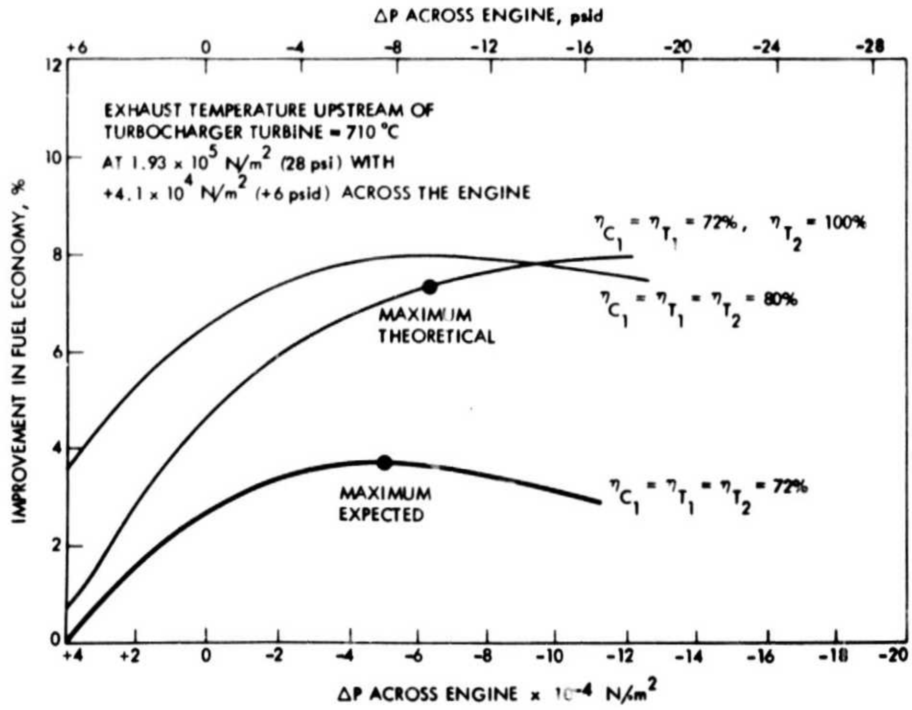

Figure 13. Effect of Back Pressure and Component Efficiencies on Turbocompounding Performance of Baseline Aftercooled Engine at Maximum Torque

any decrease in compressor or turbine efficiency. The results indicate that the improvement that can be obtained from turbocompounding about doubles. For the baseline aftercooled engine, only about half as much improvement would be possible since it is already turbocharged to a higher level.

3. Increasing the Percentage of Usable Blowdown Energy

During a short interval following the exhaust valve opening, hot gases in the cylinder expand to a low exhaust manifold pressure. The kinetic energy achieved during this blowdown process is difficult to capture because of the unsteady flow, short time period, and irreversible flow losses across the exhaust valve. Because of these difficulties, a significant amount of available energy is lost. Refinement of the exhaust valve design and coupling impulse turbines to every two or three cylinders could possibly recover a portion of this energy, but implementation is difficult. Even so, an analysis was performed to determine the potential improvement. The results indicate that if 100% of the blowdown energy could be utilized, turbocompounding performance could be increased by a factor of 2 to 2.5.

4. Increasing the Energy Available in the Exhaust by Retarding the Injection Timing

The possible benefits of retarding engine timing in order to increase exhaust energy were investigated using fuel-air cycle analysis. The effect was simulated by reducing the maximum pressure from 1.38×10^7 to 1.24×10^7 N/m^2 (2000 to 1800 psi). This resulted in a slight increase in exhaust temperature and a 1% improvement in turbocompounding performance. However, the retardation reduced base engine efficiency and power by more than 3%. Thus the tradeoff does not appear favorable. Diesel turbocompounding results are presented in Table 7.

5. Turbocompounding Spark Ignition Engines

Turbocompounding can also be employed in spark ignition engines. As discussed earlier, it is difficult to turbocharge spark ignition engines without reducing the compression ratio and severely penalizing fuel economy. In this study, the naturally aspirated baseline was turbocompounded to a back pressure differential of 6.9×10^4 N/m^2 (10 psi). The results indicate a maximum theoretical improvement of about 13%. This is comparable to the improvement for the Diesel. Expected improvement over the driving cycle would be much less.

6. Turbocompounding Costs

Components needed to implement turbocompounding include a turbine, turbine housing, speed reduction gear box, and a coupling device. Although some design problems might be encountered, the basic concept is relatively simple. Based on limited data, the manufacturing cost is estimated to be between $300 and $800.

Table 7. Results of Diesel Turbocompounding (ENDT 675)

Alternative	Improvement in Fuel Economy, %		
	Maximum Theoretical	Maximum Expected	Expected Over Driving Cycle
Increase Engine Back Pressure	11	6	3 - 4
Increase Turbo-charging Pressure to 3 atm	23	15	7
Utilize 100% Blow-down Energy	25	17	8
Retard Injection	--	-3	--

E. RANKINE ENGINE COMPOUNDING

Theoretically, any external combustion engine may be employed as a bottoming engine. However, only two cycles have received much attention as of today. Mechanical Technology, Inc., studied the use of a free-piston Stirling engine/linear alternator for this application. Their analytical predictions were encouraging, but there has been no further development. TECO has studied, built, and tested Rankine bottoming engines. Because of the limited scope of this effort, only the Rankine engine bottoming cycle was quantitatively evaluated. Consistent with TECO's study, Fluorinol-50 was selected as the working fluid, even though this imposes an unnecessary penalty on the Rankine engine performance when exhaust temperatures exceed 540°C.

In contrast to the turbocompounding options, most of the exhaust energy is usable but the conversion efficiencies are low. The most significant advantages of Rankine engine compounding are: (1) the ability to use low-quality heat and (2) minimum interference with the base engine. There is a minimum effect on engine back pressure. During the study, total exhaust energy, usable exhaust energy, ideal Rankine cycle efficiency, expected Rankine engine efficiency, and fan horsepower requirements were evaluated. The percent increase in fuel economy was calculated using Equation (4). The parameters are listed in Table 8.

Table 8. Parameters Used in Evaluating Rankine Engine Bottoming Cycle (ENDT 676)

	Maximum Theoretical Performance (Maximum Power Point)	Maximum Expected Performance (Maximum Power Point)	Expected Performance (Maximum Torque Point)
$\dot{m}_{ex}$	1561 kg/h (3442 lb/h)	1561 kg/h (3442 lb/h)	850.5 kg/h (1875 lb/h)
$\overline{C}_p$	1.09 kJ/kg K (0.26 Btu/lb R)	1.09 kJ/kg K (0.26 Btu/lb R)	1.09 kJ/kg K (0.26 Btu/lb R)
T_{ex}	537°C	537°C	613°C
$T_{reg\ out}$	107°C	107°C	107°C
ϵ_{vg}	100%	87%	87%
η_{id}	26.9%	26.9%	26.9%
$\frac{\eta_{act}}{\eta_{id}}$	1.0	0.71	0.71
η_{mech}	100%	96%	96%
$\frac{HP_{fan,id}}{\eta_{fan}}$	1.56 kW (2.1 HP)	2.7 kW (3.6 HP)	1.56 kW (2.1 HP)
Net Engine Power	195.4 kW (262 HP)	195.4 kW (262 HP)	175.2 kW (235 HP)
% Improvement	26%	15%	11%

$$\% \text{ improvement} = \frac{\dot{m}_{ex}\bar{C}_p(T_{ex} - T_{reg\ out})\varepsilon_{vg}\eta_{id}\dfrac{\eta_{act}}{\eta_{id}}\eta_{mech} - \dfrac{HP_{fan,id}}{\eta_{fan}}}{BHP} \times 100 \quad (4)$$

The largest improvement is possible at the maximum horsepower point, but in contrast to turbocompounding, there is only a slight decrease over the driving cycle. Percentage-wise there is very little difference between the aftercooled and non-aftercooled baselines. The results indicate a maximum theoretical improvement of 26%, a maximum expected improvement of 15%, and an expected improvement over the driving cycle of 12%. These numbers are in agreement with TECO's projections. In their work, they combined the base engine radiator and fan with the condenser radiator and fan. By using a low-pressure-drop radiator, high-efficiency fan, and variable-speed fan drive, they were able to improve performance by another 3 percentage points.

Basic analysis was also performed for the spark ignition, gas turbine, and Stirling engines. The maximum theoretical improvement in fuel economy was calculated to be 22% for the spark ignition engine and 37% for the gas turbine. Only 7% improvement is possible for the Stirling engine because of the small amount of energy in the exhaust (see Table 4).

The preceding analysis neglects transient response time. The waste heat recovery devices do not respond instantly to increased power demands. This is especially true for the Rankine engine bottoming cycle. It would therefore be misleading to include the power from the bottoming cycle in the base engine power rating, even though this power would be available under quasi-steady-state conditions to reduce fuel consumption.

The system integration and packaging challenges of compounding with a Rankine engine are significant, although they appear to have been solved, at least conceptually, by TECO and Mack Trucks, Inc. Based on limited data, the manufacturing cost of a complete Rankine engine bottoming system is estimated to be between $1000 and $2000.

SECTION V

CONCLUSIONS

The results were quite sensitive to several second-order effects such as the varation of specific heat with temperature and equivalence ratio. It was also difficult to determine the effects of slower piston speed on mixing and of higher peak pressure on piston friction. In general, the results are expected to be accurate to within ±3 percentage points.

The most promising Diesel alternatives are summarized in Table 9. Maximum theoretical improvement, maximum expected improvement, and expected improvement over the driving cycle are included. It is concluded that preheating would not improve engine efficiency and, in fact, could reduce power and increase NOx. Regeneration could provide moderate improvement but would pose complex implementation problems. Diesels should be turbocharged and aftercooled to the maximum possible

Table 9. Fuel Economy Improvement for Promising Diesel Alternatives

Alternative	Improvement in Fuel Economy, %		
	Maximum Theoretical	Maximum Expected	Expected Over Driving Cycle
Aftercool	3	2	1 - 2
Increase Turbocharging to 3 atm	-	4 - 6	2 - 5
Regeneration	8	5	3 - 4
Turbocompound	11	6	3
Increase Turbocharging to 3 atm and Turbocompound	23	15	7
Compound With Rankine Engine	26	15	12
Turbocompound and Rankine Engine Compound	28	15	15
Increase Turbocharging to 3 atm. Turbocompound and Rankine Engine Compound	40	26	20 - 26

level. The baseline Diesel driving cycle performance can be increased by 6% through aftercooling and increased turbocharging. Reduction in NOx emissions can be obtained by leaning out the fuel-air mixture or aftercooling. Increasing the maximum allowable cylinder pressure appears to offer additional improvement, but retooling costs and NOx problems may preclude this option.

Turbocompounding can improve performance over the driving cycle by 3 to 4%. If the turbocharge pressure is raised to 3 atm, the improvement can be doubled. Turbocompounding performance can also be improved if more blowdown energy is utilized.

Rankine engine compounding offers three times more fuel economy improvement than turbocompounding but may also cost three times as much. The Rankine engine predictions appear to be consistent with TECO's results.

The effects of turbocompounding and Rankine engine compounding over the driving cycle are approximately additive. If the baseline Diesel is turbocharged to 3 atm, turbocompounded, and Rankine engine compounded, its performance could be improved by 20%. Even this is not an upper limit. Performance of an adiabatic Diesel could be improved by nearly 40%.

The results for all five engines are summarized in Table 10. They are specified in terms of maximum theoretical improvement. The spark ignition projections are similar to those of the Diesel except for regeneration. As a result of higher exhaust temperatures, there would be about three times as much potential improvement for regenerating a spark ignition engine as for a Diesel. It also appears that gas turbines should be Rankine engine compounded. Due to the large amount of energy available in the exhaust, a significant improvement can be obtained. Conversely, because of the low amount of energy in the exhaust of a Stirling engine, only a very limited improvement is possible.

Table 10. Maximum Theoretical Fuel Economy Improvement in Truck Engines

Engine	Regeneration	Turbocharging	Turbocompounding (100% Efficient Power Turbine)	Rankine Engine Compounding
Diesel	8	6	11	26
Adiabatic Diesel	25	6	29 - 34	40
Spark Ignition	25	5	13	22
Gas Turbine	-	-	-	37
Stirling	-	-	-	7

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