THE STUDY OF LOW EMISSION VEHICLE POWER PLANTS USING GASEOUS WORKING FLUIDS

FINAL REPORT

REPORT NO. SR-20

AUGUST 1972

THERMO MECHANICAL SYSTEMS COMPANY
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COPY NO.:

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ABSTRACT

This report presents a preliminary technical study of low emission vehicle powerplants which use gas as the powerplant working fluid. The scope of the program concerns evaluation of the thermodynamics and preliminary design of several cycles which include but are not necessarily limited to:

1. The External Combustion Piston Engine
2. The Closed and Open Brayton Cycle Engines
3. The Ackeret-Keller Cycle Powerplant
4. The Stirling Cycle Engines
5. Evaluation of Rankine Cycle and Other Closed Cycle Working Fluid Hazards

This report includes consideration of those factors which affect the general suitability of the powerplant to the automotive type vehicle, and a comparison of the overall relative merits of the several power systems.

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1.0 INTRODUCTION

Both the gas turbine and the Rankine cycle engines are included in the EPA Advanced Automotive Power Systems Program because of their potential for achieving low exhaust emissions that meet Federal 1976 standards. The prime reason that these engines merit serious consideration is that they use a continuous combustion system in which the combustion process can be controlled to produce relatively low values of exhaust emissions. These emissions include carbon monoxide, unburned hydrocarbons, and oxides of nitrogen. The achievement of the Federal 1976 oxide of nitrogen standards for the internal combustion gas turbine has not been demonstrated, but encouraging results have been reported. The Rankine cycle engine, which utilizes an external combustion system, has demonstrated in combustion rig tests that this engine can achieve Federal 1976 emission standards.

The gas turbine or open Brayton cycle engine has the potential of achieving low emissions. The Rankine cycle engine is an external combustion engine but there are other engine thermodynamic cycles based on gaseous working fluids that utilize potentially low emission external combustion which also merit investigation. This program includes a study of both the open Brayton cycle and the following closed cycle heat engines:

1. Closed Brayton cycle
2. Stirline engine
3. External Combustion Piston Engine
4. Ackeret-Keller cycle.

Factors considered during the evaluation of the above cycles included efficiency (at full and part load), power plant weight and volume, operating characteristics, cost hazards, etc.

An additional objective of the present program was to evaluate the potential hazards associated with organic Rankine cycle and candidate gaseous cycle working fluids. The combustive, explosive, and toxic characteristics of these fluids were compared to similar hazards now common to existing motor vehicle power plants.

2.0 SUMMARY

This final report presents the results of an analysis and design study of closed and open cycle gaseous working fluid powerplants and an evaluation of their suitability for use as low emission vehicle powerplants. The primary areas of investigation and the conclusions drawn therein are as follows:

1. An in-depth study of all Brayton cycle concepts which hold potential as automotive powerplants was performed. The first step in the Brayton cycle investigation was to determine the relative merits of open versus closed cycles. For the closed cycle, working fluids (helium, argon, freon, etc.) otherthan air were evaluated and, in addition, the complex Ackeret-Keller cycle was also investigated with varying degrees of intercooling and reheat. In general, the closed cycle gas turbines did not appear as feasible as the open cycles due to their lower efficiencies, increased complexity, and added cost.

   A thorough investigation into state-of-the-art turbine technology was then undertaken to determine the turbomachinery components and associated characteristics which were best suited for an automotive application. It was found that any automotive gas turbine powerplant will be limited to certain maximum temperatures and pressures as determined by the availability and cost of refractory alloys. The number and sizes of compressor, turbine, and heat exchanger components must be kept to a minimum to reduce the very important initial cost. The best powerplant package was determined to be a single stage radial compressor and single stage radial turbine with a maximum compressor pressure ratio of about 7:1 (depending upon heat exchanger effectiveness) and maximum turbine inlet temperature of 1800°F. The issue of single-shaft versus free-turbine was found to allow considerable trade-offs in such areas as weight, size, cost, and operational characteristics. For this reason, either form is feasible, although the single-shaft seems to have the advantage with respect to initial cost and amount of high temperature material required.

Having determined the best combination of turbomachinery for the automotive powerplant, detailed cycle analyses were performed. It became immediately clear that the simple cycle with a single stage radial compressor and maximum turbine inlet temperature of 1800°F had unacceptable part load fuel consumption. This led to the analyses of gas turbine cycles which use various types of heat recovery systems. Both regenerative and recuperative heat exchangers were examined, with the plate-fin type recuperator appearing to be the superior type of unit with respect to size, cost, and complexity, although the reliability and expected life of this type of heat exchanger have not been proven. The tubular recuperator, though reliable, was shown to be somewhat bulky and expensive, while the ceramic or metallic regenerator, though comparable in size and performance to the plate-fin recuperator, was determined to be more complex because of possible seal and bearing problems. Further cycle analysis with various plate-fin recuperators revealed that the optimum recuperator size depends on the available powerplant space and the associated heat exchanger manufacturing cost.

An exhaust gas bypass arrangement was evaluated in combination with the various types of heat exchangers. Bypassing the heat exchanger at higher engine power levels gives a more compact less costly unit, since the heat exchanger is sized for low power operation only. Such an arrangement can provide the power of a medium pressure ratio non-recuperative cycle for acceleration and high speed operation, while retaining the efficiency of a low pressure ratio recuperative cycle at low road load power levels. The study showed that the bypass concept offers advantages with respect to size, weight, and cost while giving somewhat poorer fuel consumption than could be obtained with a larger heat exchanger designed for full power operation without the bypass feature.

In addition, a preliminary investigation was also directed towards determining the potential of the "Comprex" as the gas

generator unit for a small free-turbine powerplant. A literature search and preliminary cycle calculation shows that this concept holds potential as a successful automotive powerplant.

2. The External Combustion Piston Engine (ECPE) is a new and unusual powerplant concept which is intended to improve the emissions of conventional reciprocating piston engines in the power class which is needed for automobiles. The concept is centered on the use of an external combustor which transfers the energy of combustion to a closed cycle reciprocating engine through an external heat exchanger. Open cycle operation was investigated originally and shown to require excessive displacements to produce the required horsepower (160 BHP). Various closed cycle working fluids (helium, argon, freon, etc.) other than air were also investigated; however, the performance, availability, and cost factors were such as to make air the most attractive. Optimization studies were conducted to determine the best combination of cycle parameters with close attention being paid to such factors as material cost, material availability, stress limits and design sophistication. Preliminary cycle analysis showed that a 267 CID, 4 stroke engine with maximum closed cycle temperature and pressure of 1300°F and 1500 psia with a pressure ratio of 10 would produce the required 160 BHP at a design speed of 3600 rpm. A conventional six cylinder 250 CID Chevrolet engine was used for prototype layouts, although a somewhat heavier expander will be required for the development program to withstand the maximum 1500 psia cylinder pressure. This base engine was modified with a slight increase in cylinder bore and a new head modified for the four valves which are required to establish the hot and cold air flow paths. The pressure ratio and maximum cycle temperature were such as to make regeneration of the closed cycle air unattractive. Maximum combustion products temperature was limited to 2540°F to reduce both the burner NO_x emissions and the amount of critical metals in the combustion heat exchanger.

Control system operation, heat exchanger designs, and engine characteristics, as a function of speed and power, were determined and used to evaluate engine performance under all power demands. Acceptable efficiency and fuel economy was obtained for the ECPE powered vehicle throughout the normal operating range. The ECPE powerplant engine with all auxiliaries and accessories was found to fit in the engine compartment of a medium size vehicle (1970 Chevrolet Caprice).

The ECPE powerplant was evaluated with respect to cost, hazards, and various other economic and social considerations and shown to be a potential solution to the air pollution problem.

3. A very thorough literature survey of past and present effort on the Stirling engine was the first step in determining the feasibility of this engine as an automotive powerplant. The information obtained was analyzed and coalesced into a form from which the most feasible design was constructed. Various working fluids were investigated with hydrogen proving to be the most efficient and helium the least hazardous.

   Weight and size considerations indicate that the Stirling engine may not be a very practical automotive powerplant even with its capability for high efficiency and low emissions. Major design problems must be solved to produce a suitable automobile engine.

4. A study was conducted to evaluate the potential hazards of a number of candidate closed cycle working fluids for automotive powerplants. The study considered such characteristics as toxicity, combustibility, and susceptibility to explosion, and compared the proposed closed cycle working fluids with fuels which are now in common automotive use.

   In general, it was determined that any closed cycle system requiring a toxic or flammable working fluid should not be considered unless the total quantity of fluid is sufficiently small so as to minimize the hazard.

5. The final effort in this study was a comparison of the various powerplant concepts investigated, in addition to a comparison with other existing more conventional powerplants (Otto, diesel, Wankel, etc.). The various powerplants were compared with respect to weight, cost, complexity, full and part power efficiency, driveability and, of prime importance, emissions. The comparisons have indicated that: (1) the gas turbine, Rankine, ECPE, and other concepts being considered will require further development work to prove them acceptable automotive powerplants; (2) the Stirling engines appear to be less attractive than other contending powerplants; and (3) based on demonstrated emission levels and suitability for automotive application, various versions of the internal combustion engine have significant leads over other proposed powerplants.

3.0 RECOMMENDATIONS

Based on the results of the present program, the following recommendations are presented:

1. Further effort should be directed toward developing the medium pressure ratio (6 or 7:1) plate-fin type recuperative gas turbine powerplant.
2. Additional effort is recommended to determine the feasibility of using a "Comprex" type gas generator unit for an automotive free-turbine powerplant.
3. Due to similarities in size, cost, complexity, etc., the ECPE and Rankine cycle engines may be considered as equally feasible low emission automotive powerplants. Sufficient design and development work should be carried forward on the ECPE to bring it to a stage where a more authoritative comparison can be made with the Rankine and other powerplant concepts.
4. The cost and complexity of the present Stirling engine make it impractical for an automotive powerplant. The use of hydrogen as the working fluid, higher engine speeds, and a barrel engine configuration can improve the performance of this engine. Re-evaluation of the Stirling engine for the automotive powerplant application should be made after recent development details are made available from Phillips.
5. Additional effort is recommended on promising variations of the Otto and diesel engines such as the stratified charge concept, low compression ratio diesels, and variable displacement engines.
6. Any system requiring a toxic or flammable working fluid should not be considered unless the total quantity of fluid is sufficiently small so as to minimize any hazard.

4.0 EXTERNAL COMBUSTION PISTON ENGINE (ECPE)

4.1 INTRODUCTION

The External Combustion Piston Engine is an unusual powerplant concept which is intended to improve the emissions of conventional reciprocating piston engines in the power class which is needed for automobiles. While there has been much progress in reducing emissions from conventional Otto and diesel cycle engines, and more progress can be anticipated in the future, progress to date has been achieved at the expense of additional complexity and it is anticipated that further reductions, which will be essential to meet future goals, will be achieved at the expense of further increases in complexity, fuel consumption, maintenance, etc. At the present time, the intermittent combustion process of the internal combustion engine produces higher than desireable levels of emissions (CO, CH_x, NO). While it would be unwise to predict that the emissions of the internal combustion engine cannot reach the low levels required by current and future regulations, available data indicates that some form of power system which involves steady flow, external combustion would be more favorable.

Such external combustors are most effectively used with closed cycles. The total emissions of an engine of a given power, however, is the product of the total emissions of the combustor and the overall efficiency of the engine. Thus, it is important that the energy converter has a high efficiency over a broad range. The Stirling engine is one form of closed gas cycle which has demonstrated a relatively flat, high efficiency level (on the order of 30-40%) and should, therefore, exhibit very favorable overall emission characteristics.

While many millions of dollars and many years have been spent on the Stirling engine, the promising performance characteristics have not resulted in a production type engine. The Stirling engine uses intermittent flow heat exchangers and hydrogen or helium working fluid. It is necessary to achieve a fine balance between heat transfer coefficients, fluid friction, and void volume of the heat exchanger to achieve the desired performance of the engine.

Possibly the Stirling type of engine would be much more favorably received if it were possible to retain the high efficiency reciprocating

compression and expansion processes and replace the intermittent flow heat exchangers with steady flow heat exchangers. This would remove the requirement for special working fluids, and ordinary ambient air could then be used. This is the basis for the External Combustion Piston Engine (ECPE).

It was the objective of the ECPE cycle to eliminate as many of the undesirable factors of the Stirling engine as possible while retaining its favorable features. High on the list of changes considered was the development of a concept which would be capable of attaining Stirling engine efficiency, but which would use ambient air as the working fluid. This means that the heat exchanger design had to be changed radically to eliminate the engine's dependence on heat exchangers which must be so compact that a hydrogen working fluid is required to maintain acceptable pressure losses. To achieve this objective, it appeared most feasible to design a cycle which used steady flow, or essentially steady flow, through the heat exchangers so that the required heat can be transferred with minimum pressure loss. With a reciprocating system, steady flow can be approximated by using multiple cylinders so that the fluctuations are minimized.

In addition, in small sizes, such as engines in the 160 HP class, the reciprocating compressor (and expander) is usually more efficient than the corresponding turbomachinery units. It is not unusual for small reciprocating compressors to attain over 90% efficiency, but it is unusual for a small turbine or compressor to achieve over 80%, and more often, the value is on the order of 70-75%. A turbocharger compressor for a 160 HP diesel engine, for example, attains an efficiency of about 72%. The reciprocating compressor and expander thus has a distinct advantage from the overall cycle efficiency standpoint for the ECPE cycle. In its basic form, the cycle is similar to the classical diesel cycle, and the total efficiency is of the order of 20-25% at maximum cycle temperature of about 1300°F using air as the working fluid. Both regenerative and non-regenerative versions have been studied; however, as will be shown, the results indicate that regeneration is of marginal or no value for the range of pressure ratios and temperatures under consideration. Several working fluids have been investigated and while a monotomic gas such a Argon appears to be favorable from the

efficiency standpoint, the overall availability and performance of the cycle using air was found to be more favorable in the practical sense.

The literature shows that there have been attempts to design reciprocating open Brayton cycle engines (or modifications thereof). They have always failed, however, because such concepts have always used one cylinder (or set of cylinders) for compression and an equal number of cylinders for expansion. In all cases, extreme design sophistication is essential to cool the expansion pistons and cylinders due to the exposure to the hot gas. No successful engine has been developed, although there are many patents and many designs which have been tried. However, in the case of the ECPE design, both compression and expansion take place in the same cylinder as is the case for the conventional internal combustion engine. The cylinder wall is heated during the expansion stroke and cooled during the compression stroke. To control the process events, four valves are used per cylinder and the cycle operates on the 4-stroke principle. Modifications to the valve events can be made to convert the ECPE classical diesel cycle to a reciprocating Brayton cycle. The intake valve could be maintained in the open position during part of the compression stroke so that the compression ratio would equal the expansion ratio thereby increasing the cycle efficiency. The work per cycle would decrease, of course, but the power can be increased by increasing the average gas pressure.

In addition, a hybrid version of the basic cycle can use the classical diesel cycle at high power and convert to the Brayton cycle for higher efficiency at lower power requirements, or the intake valve timing can be modified to provide any cycle between the two. This can be used as an alternate control method compared to that conventionally used in closed gas cycles (control of base cycle pressure). Since broad power and torque range is important for applications such as automobile powerplants, the intake valve method of controlling the cycle output may have some advantages which merit investigation in a later program.

4.2 PRINCIPLE OF THE ECPE

The External Combustion Piston Engine is unique in that the compression and expansion processes take place in the same cylinder while the

heat addition and removal processes are conducted external to the cylinder. Thus, the engine retains the well established reciprocating engine characteristic of using the cooler gas of compression to cool the cylinder structure which was heated by the expansion process. The ECPE cycle can be designed for either open or closed cycle, and with or without regeneration.

The concept of the ECPE is illustrated in Figure 4.1 for the closed cycle version. In both the open and closed cycles, the engine machinery is similar, and Figure 4.1 can be used for descriptive purposes. Cold gas is inducted into the cylinder through the intake valve as the piston moves downward. This valve is closed at or near bottom center and the gas is compressed to some value which is determined by the overall cycle thermodynamics and is discharged from the cylinder through valve no. 2. The gas then flows through the regenerator (if there is one) and through the combustor heat exchanger or burner, adding heat to the working fluid. The expansion valve (no. 3) then opens and the piston moves downward. This valve remains open for part of the piston downstroke, and the point at which it closes is determined by the overall cycle thermodynamics. Once this valve is closed, the gas in the cylinder expands to the end of the stroke. The exhaust valve (no. 4) then opens and the cylinder gas "blows down" to the exhaust pressure. This valve remains open as the working fluid remaining in the cylinder is discharged while the piston moves to top center. If there is a regenerator, the exhaust gas flows through it before being discharged to the atmosphere (open cycle) or the waste heat exchanger. Figure 4.2 shows the basic P-V diagram for the ECPE cycle. The working fluid is drawn into the cylinder from 0' to 1, compressed from 1 to 2, forced out of the cylinder from 2 to 2', heated at almost constant pressure and returned to the cylinder from 3 to 4, expanded within the cylinder from 4 to 5, blown down to base pressure from 5 to 6, and forced out of the cylinder from 6 to 0. The difference in pressure levels between 0 and 0', and 2' and 3 is due to valve losses and heat exchanger pressure drops.

At this point it was necessary to determine whether the open or the closed ECPE cycle was the most feasible as an automotive powerplant, keeping in mind the fact that the closed cycle has the potential of using various gaseous working fluids other than air. Various monatomic and

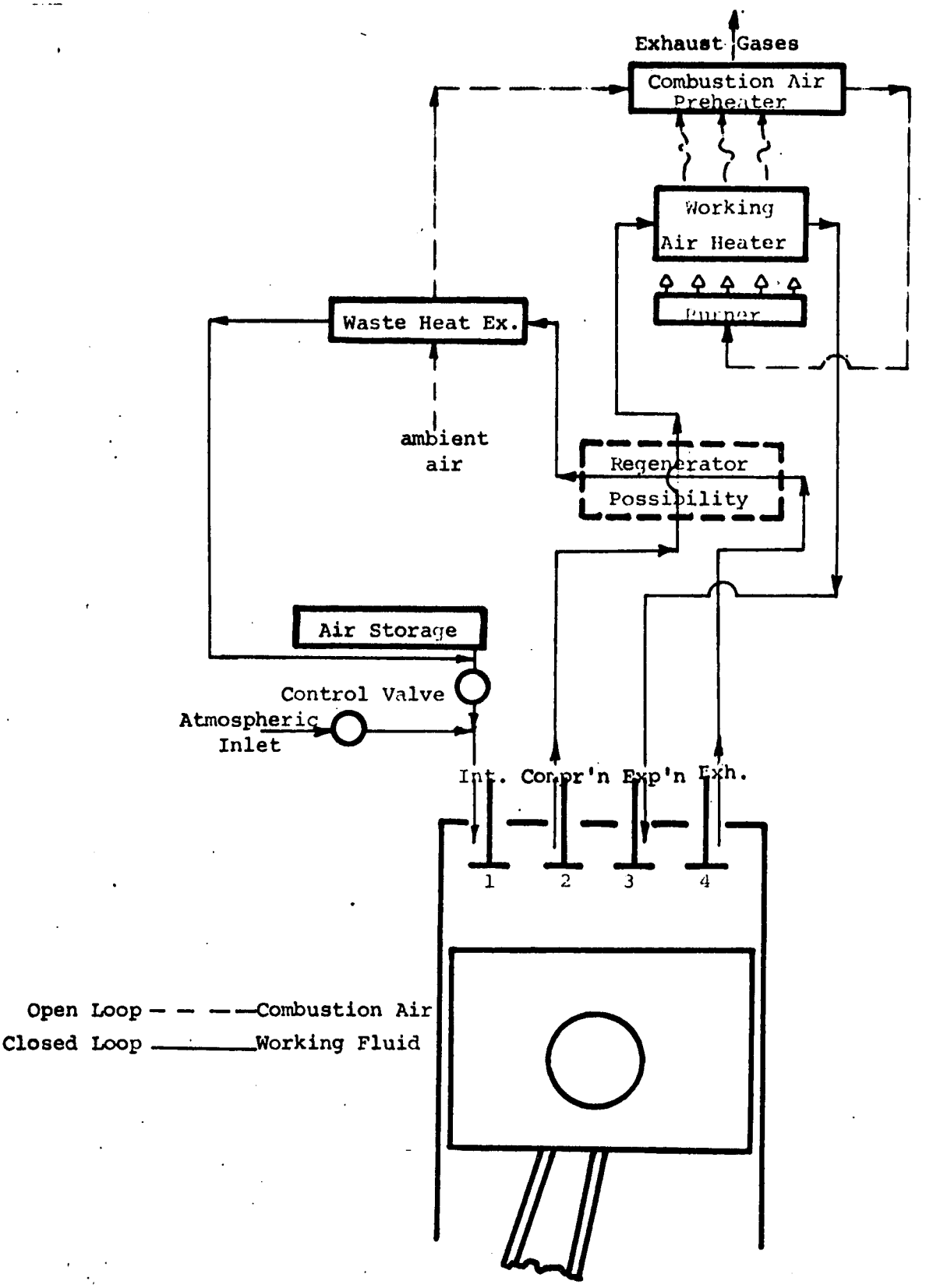


FIGURE 4.1 Diagram of External Combustion Piston Engine (ECPE).

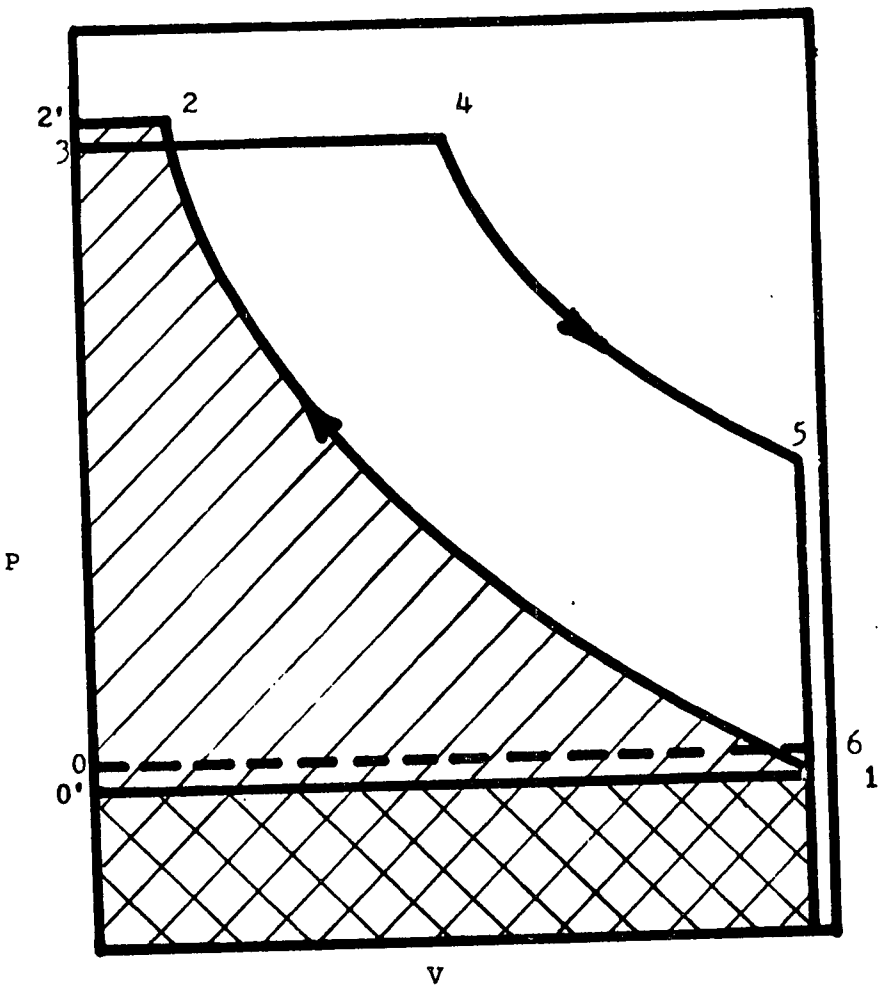


FIGURE 4.2. Basic P/V diagram for the External Combustion Piston Engine (ECPE).

diatomic gases were reviewed as possible ECPE working fluid candidates, and the following are chosen as having the greatest potential:

1. Air (open and closed cycle)
2. Argon (closed cycle)
3. Helium (closed cycle)
4. Hydrogen (closed cycle)
5. Freon (closed cycle)

Freon was used for comparative purposes inasmuch as it is not capable of withstanding the peak cycle temperature used in the analysis. For this preliminary investigation, assumptions of the closed cycle parameters included: peak cycle temperature of 1540°F, peak pressure of 1000 psia, pressure ratio of 10, and minimum cycle temperature and pressure of 140°F and 100 psia respectively. For the open cycle the peak pressure was 147 psia and cycle inlet conditions of 85°F and 14.7 psia were considered (all other parameters comparable). At this point in the analysis cycle parameters (T_{max}, P_{max}, etc.) were chosen which would give relatively high efficiencies in a reasonably sized powerplant while taking into consideration material limitations of reciprocating engine components. It will be shown later that heat exchanger limitations will change the cycle parameters as presented here; however, such changes do not affect the general results of the working fluid analysis given here.

Figures 4.3 to 4.6 present ideal P-V diagrams for the closed air, argon, helium and hydrogen ECPE cycles, plotted using logarithmic scales. The cycle calculations were performed assuming isentropic expansion and compression with no valve, heat exchanger of pumping losses. In short, the work output and efficiencies were determined on an ideal basis to be used for comparison purposes only, not to be used as engine performance capabilities. For computing working fluid mass flow rate and engine size (CID), an indicated horsepower requirement of 250 HP was assumed. The absolute value of this horsepower requirement is unimportant as the results here are to be used for relative comparisons only.

The results of the cycle calculations for the various gases are tabulated in Table 4-I. The open air cycle requires an engine with a displacement of 1471 cubic inches which eliminates it as a possible contender for an automotive type powerplant. Of course, decreasing the

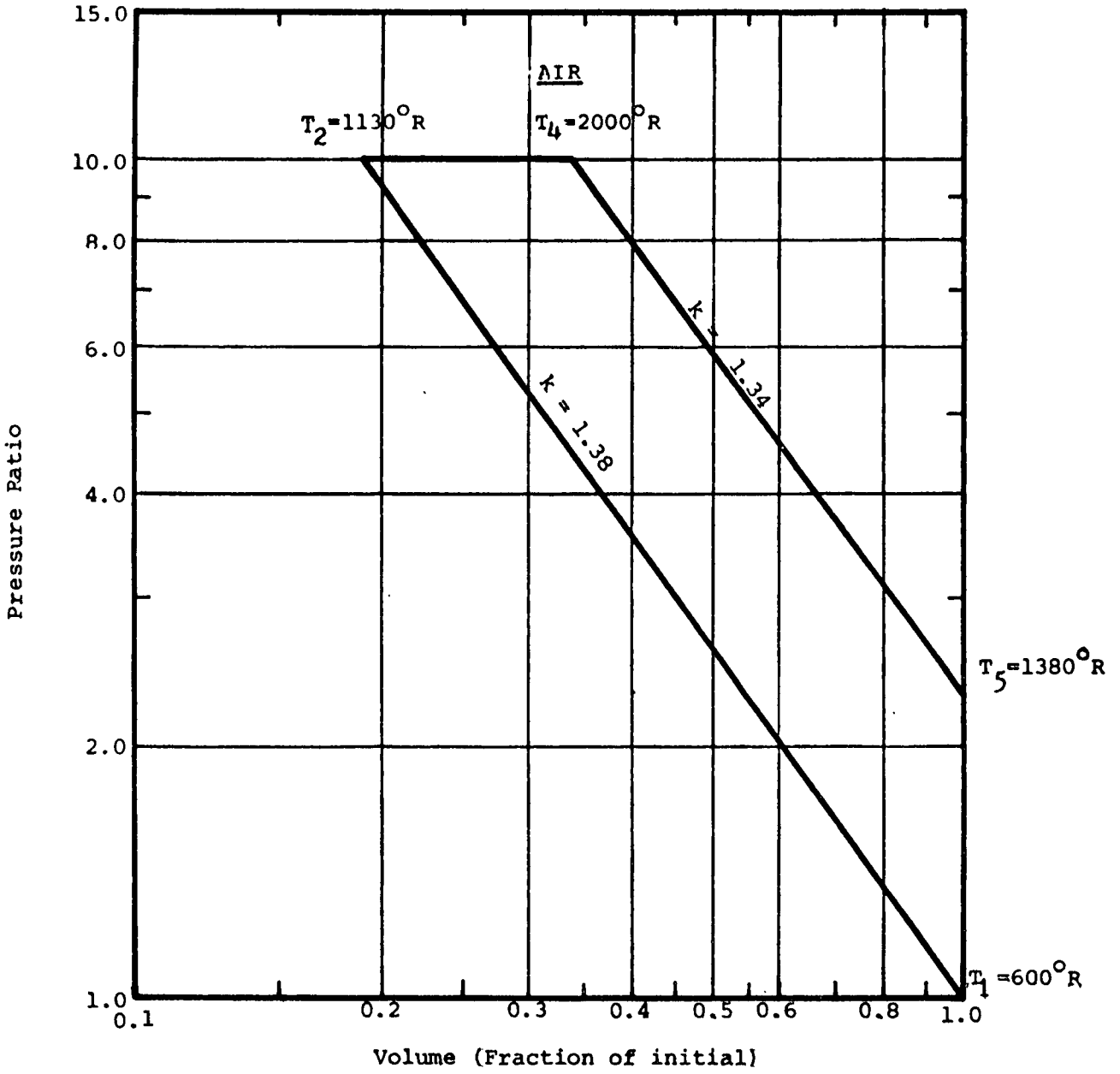


FIGURE 4.3 External Combustion Piston Engine
Closed cycle, ideal pressure-volume
diagram for air.

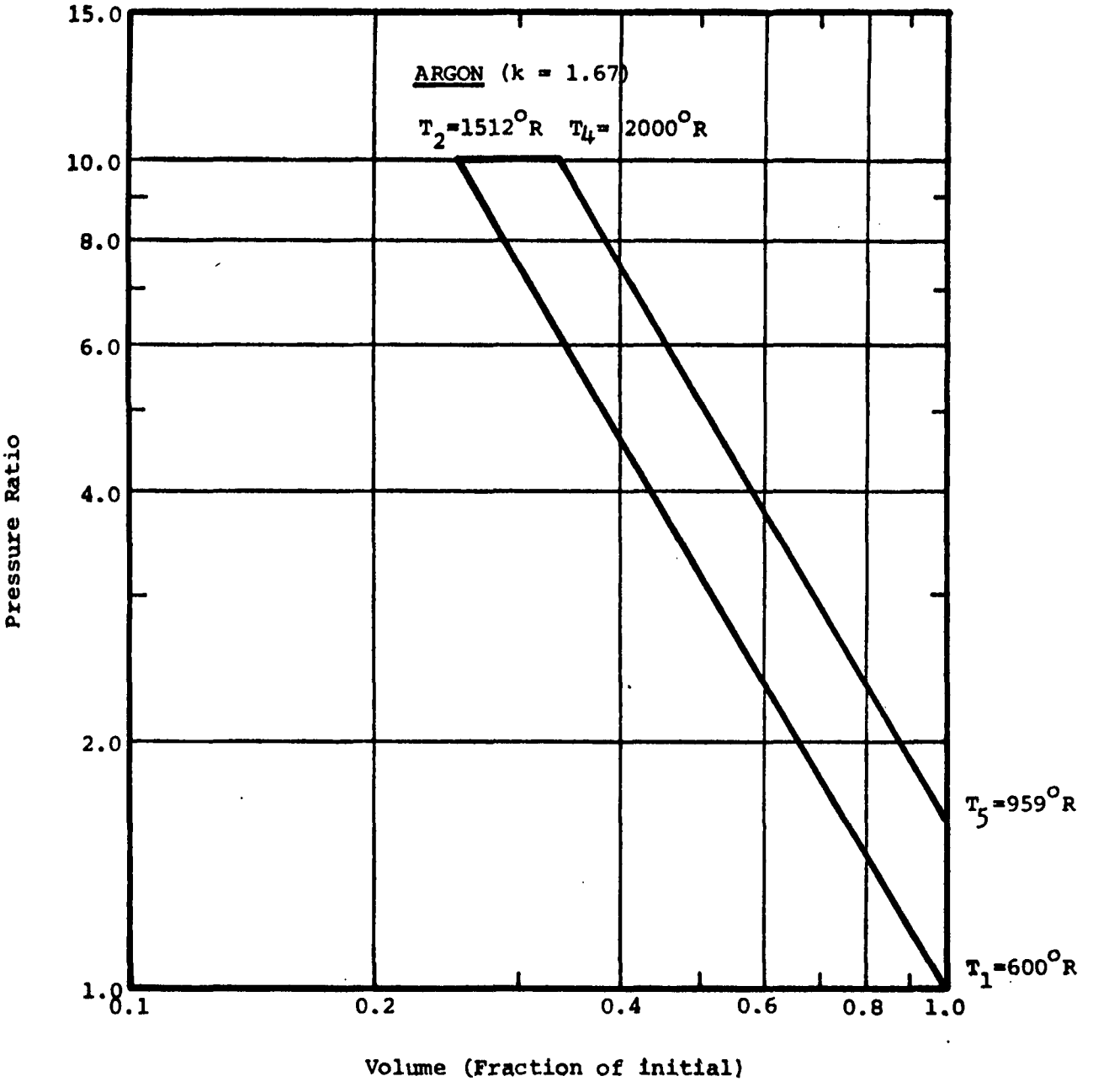


FIGURE 4.4 External Combustion Piston Engine
Closed cycle, ideal pressure-volume diagram for argon.

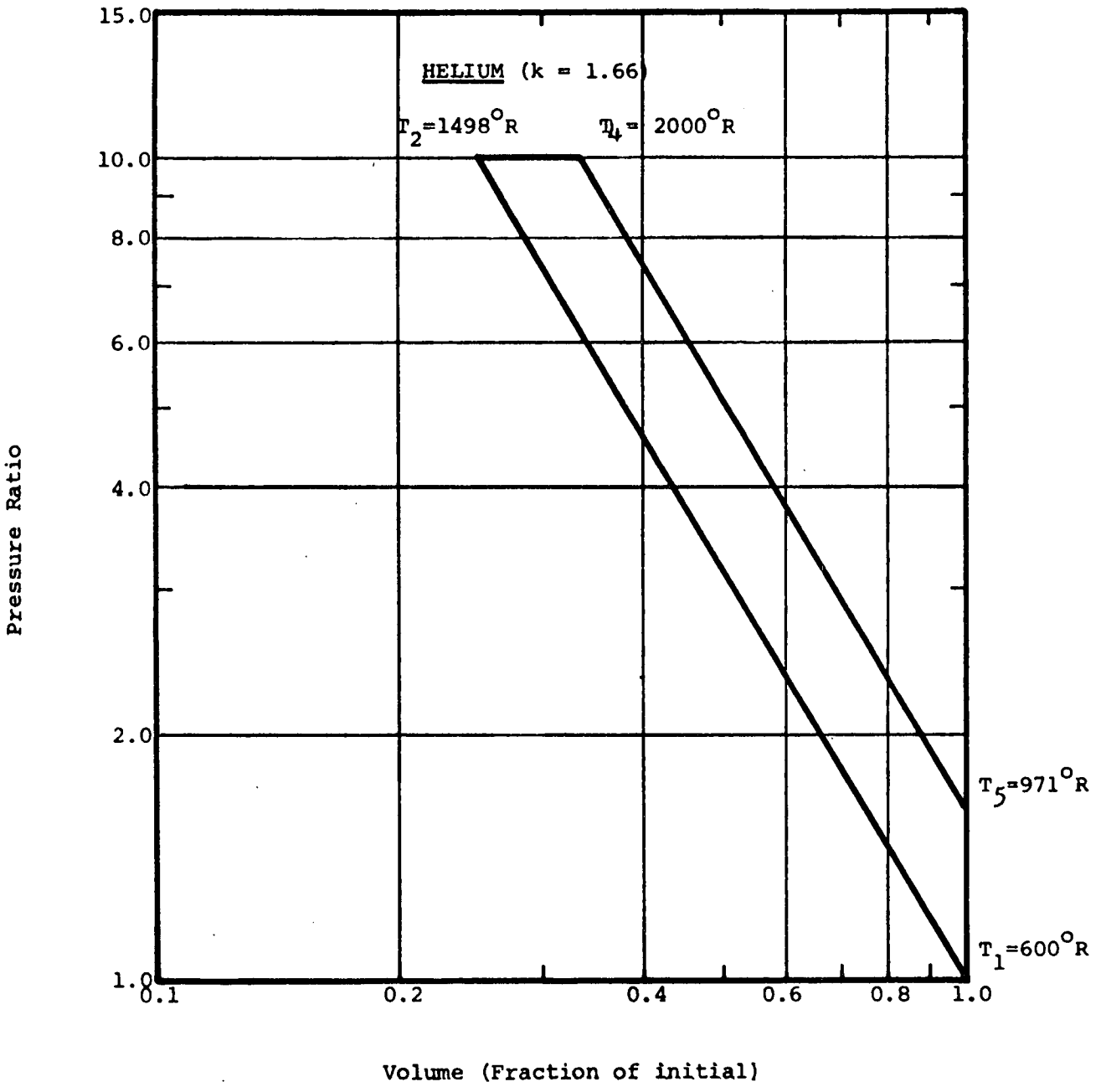


FIGURE 4.5 External Combustion Piston Engine
Closed cycle, ideal pressure-volume diagram
for helium.

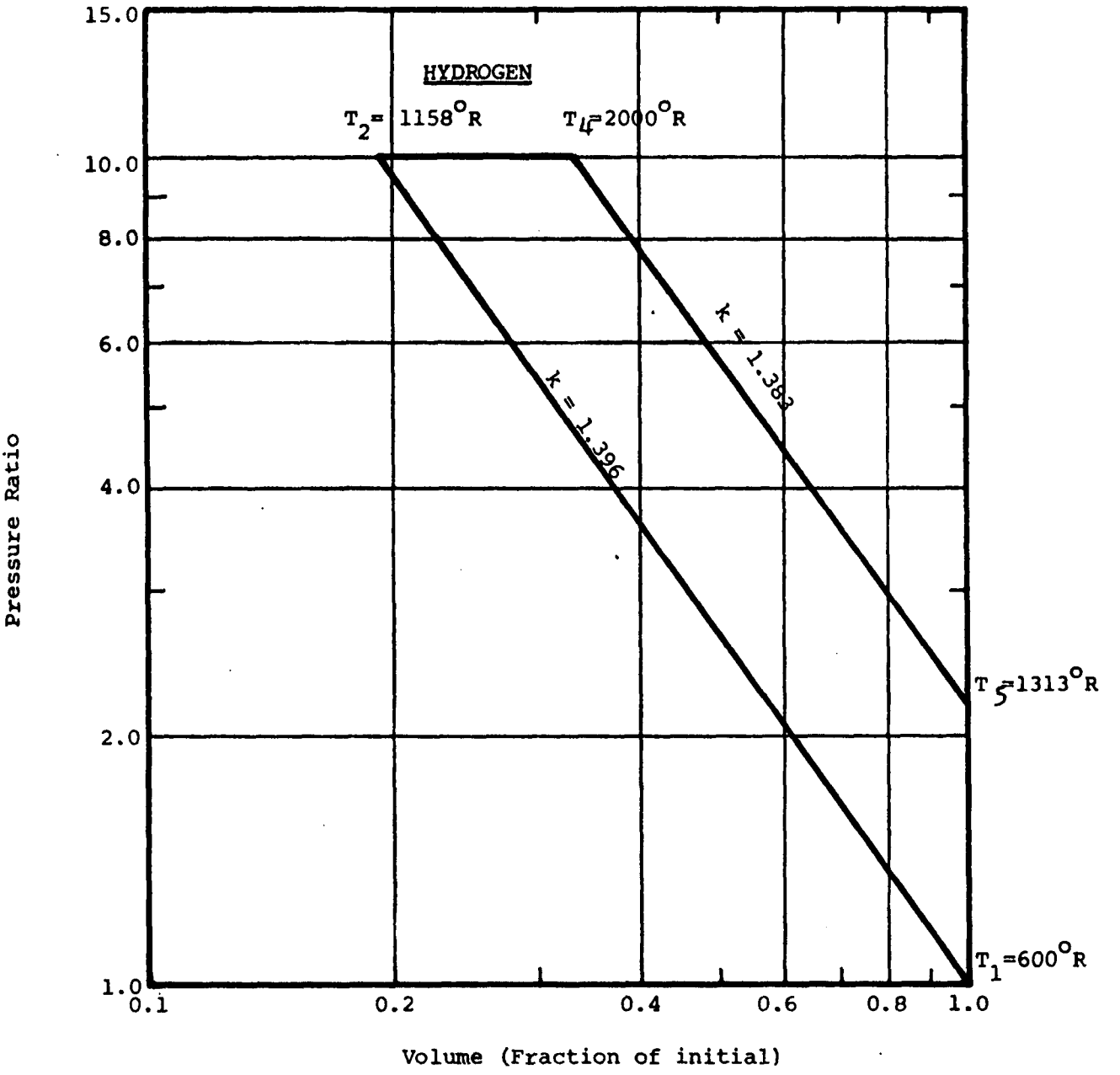


FIGURE 4.6 External Combustion Piston Engine
Closed cycle, ideal pressure-volume diagram
for hydrogen.

TABLE 4-I

EXTERNAL COMBUSTION PISTON ENGINE WORKING FLUID CYCLE CHARACTERISTICS

CYCLE	T_1 P_1 V_1	T_2 P_2 V_2	T_4 P_4 V_4	T_5 P_5 V_5	Ideal Cycle Work BTU/lb	CID for 250 IHP at 3600 RPM in^3	Mass Flow Rate lb/min.	Ideal Cycle Efficiency
AIR (Closed Cycle)	$600^{\circ}R$ 100 psia 2.22 ft^3/lb	$1130^{\circ}R$ 1000 psia .420 ft^3/lb	$2000^{\circ}R$ 1000 psia .744 ft^3/lb	$1380^{\circ}R$ 231 psia 2.22 ft^3/lb	90	253	119	.396
ARGON (Closed Cycle)	$600^{\circ}R$ 100 psia 1.613 ft^3/lb	$1512^{\circ}R$ 1000 psia .406 ft^3/lb	$2000^{\circ}R$ 1000 psia .537 ft^3/lb	$959^{\circ}R$ 159.5 psia 1.613 ft^3/lb	34	484	313	.560
HELIUM (Closed Cycle)	$600^{\circ}R$ 100 psia 16.10 ft^3/lb	$1498^{\circ}R$ 1000 psia 4.025 ft^3/lb	$2000^{\circ}R$ 1000 psia 5.373 ft^3/lb	$971^{\circ}R$ 162 psia 16.10 ft^3/lb	350	472	31	.554
HYDROGEN (Closed Cycle)	$600^{\circ}R$ 100 psia 31.96 ft^3/lb	$1158^{\circ}R$ 1000 psia 6.17 ft^3/lb	$2000^{\circ}R$ 1000 psia 10.64 ft^3/lb	$1313^{\circ}R$ 218 psia 31.96 ft^3/lb	1220	267	9	.407
FREON (Closed cycle)	$600^{\circ}R$ 100 psia 0.525 ft^3/lb	$782^{\circ}R$ 1000 psia 0.0685 ft^3/lb	$2000^{\circ}R$ 1000 psia 0.175 ft^3/lb	$1698^{\circ}R$ 240 psia 0.525 ft^3/lb	35	154	305	.202
AIR (Open Cycle)	$545^{\circ}R$ 14.7 psia 13.73 ft^3/lb	$1041^{\circ}R$ 147 psia 2.62 ft^3/lb	$2000^{\circ}R$ 147 psia 5.04 ft^3/lb	$1420^{\circ}R$ 38.2 psia 13.73 ft^3/lb	95	1471	112	.377

pressure ratio would increase the work output and reduce the required displacement; however, decreasing it enough to give a reasonable size powerplant would produce an unfavorable cycle efficiency. Hydrogen requires a displacement of 267 cubic inches and provides an ideal thermal efficiency of 41%, values which are essentially the same as calculated for the closed air cycle, Inasmuch as there is a fire hazard with hydrogen, this gas it not considered a likely working fluid for the ECPE cycle. The two monatomic gases, argon and helium, provide a significantly improved ideal efficiency of about 56%. However, the engine size increases to 472 cubic inches with helium and to 484 cubic inches with argon. Of course, just as for the open air cycle, the displacement can be reduced by reducing the pressure ratio; however, this will lower the efficiency to the point where there is little advantage over the closed air cycle, even with suitable regeneration. Furthermore, since there is a very limited supply of helium, it does not appear to be a strong candidate for the ECPE. In addition, the use of Argon will require a large resupply source for the working fluid (and the associated distribution problems) thereby eliminating the advantage of using readily available ambient air. Therefore, a closed cycle using air as the working fluid was chosen for further ECPE analysis and design studies.

Parametric studies to determine the best combination of inlet pressure, pressure ratio, maximum system pressure, and maximum system temperature on engine size, efficiency, and power output were then initiated. The maximum system temperature was limited to 1300°F by heat exchanger metal temperature limitations and the need to avoid the use of high alloy stainless steels (Hastelloy, etc.). The maximum system pressure was limited to 1500 psia, a level intermediate between conventional Otto and diesel cycle maximum cylinder pressures. The minimum closed cycle temperature was 165°F and the waste heat exchanger is designed to control this temperature under all operating conditions on an 85°F day. In addition, it was decided to design the engine for a maximum rotational speed of only 3600 rpm to reduce friction losses and excessive engine stresses. Finally, the peak brake horsepower requirement at the design speed was calculated to be 160 BHP (see Appendix A).

Figures 4.7 and 4.8 show the results of these parametric studies for the closed ECPE air cycle. Since the maximum pressure (1500 psia)

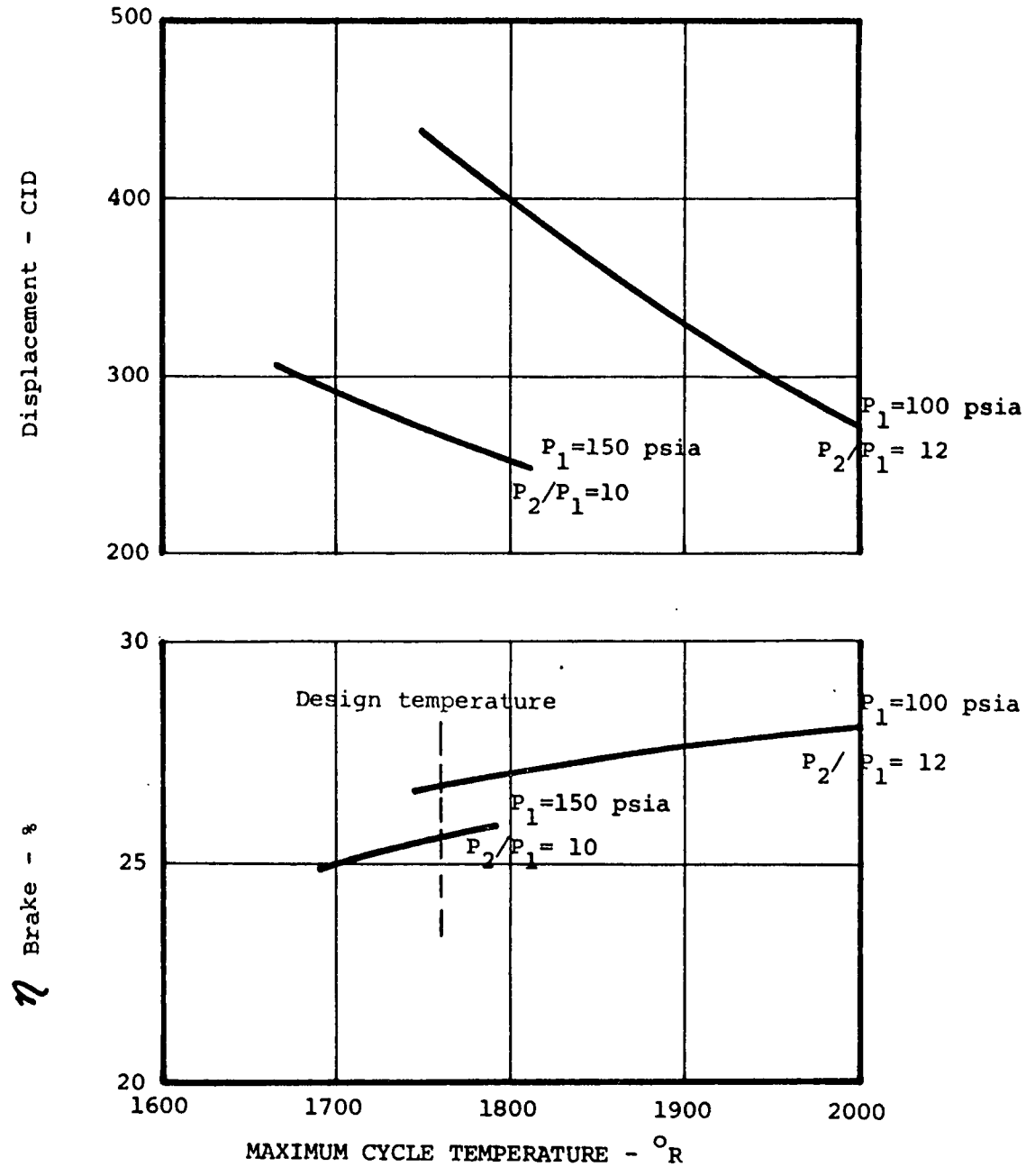


FIGURE 4.7 Effect of Maximum Cycle Temperature on Engine Displacement and Brake Cycle Efficiency.

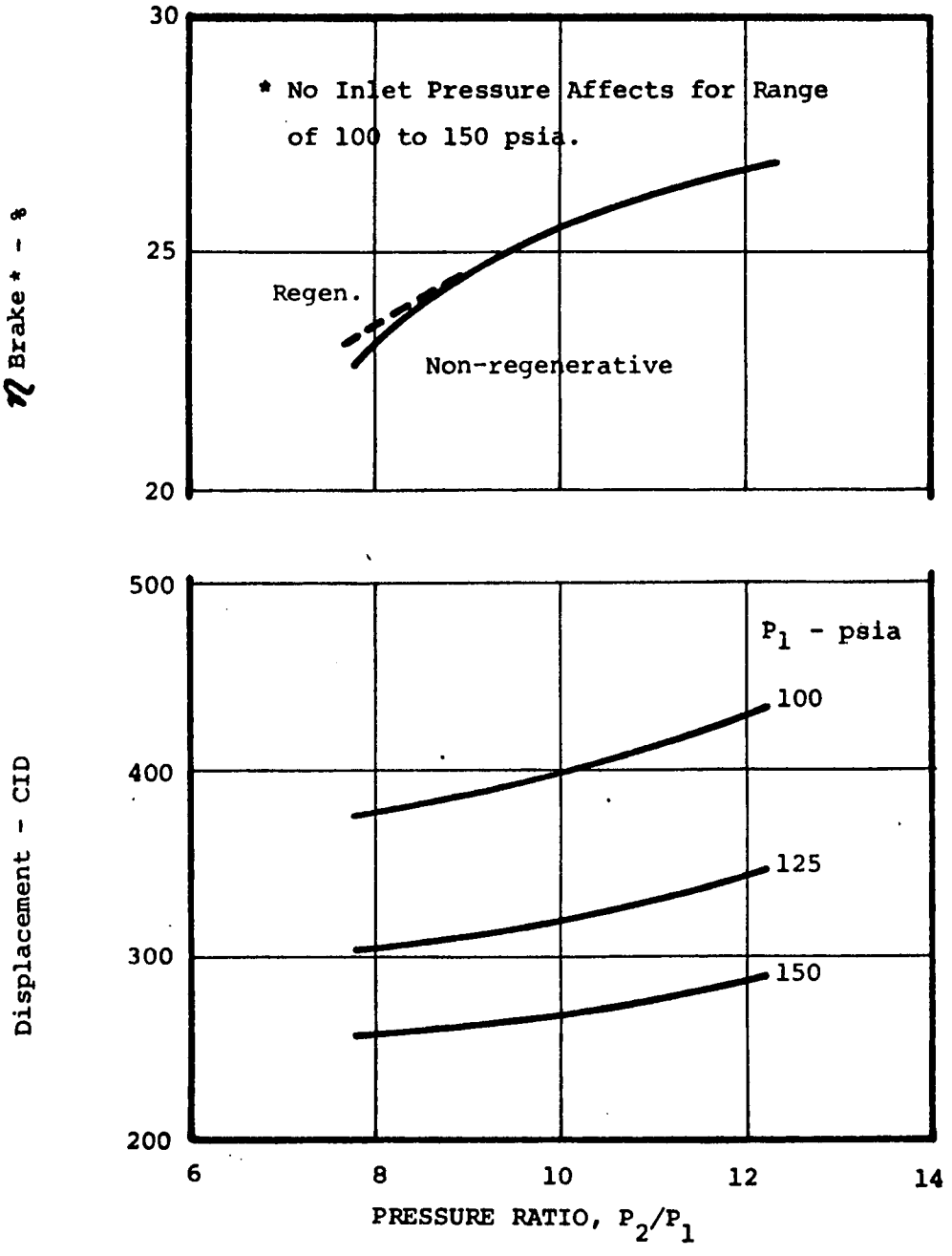


FIGURE 4.8 Effect of Pressure Ratio and Inlet Pressure on Brake Cycle Efficiency and Engine Displacement. (T_4 = 1760°R.)

and temperature (1300°F) limits were defined, it was merely necessary to determine the best combination of base pressure level and pressure ratio which gives the best combination of engine size and efficiency. It can be seen that a base pressure of 150 psia with a pressure ratio of 10 gives a medium size engine (267 CID) with a relatively high brake cycle efficiency (25.5%). It should be pointed out that this efficiency does not take into account the combustion loop heat losses and that it applies only at the design point. As expected, Figure 4.7 shows that the highest efficiency and smallest powerplant is obtained at the highest temperature, and therefore there is no question here but to use the maximum allowable, or 1300°F. Figure 4.8 shows that to use a higher pressure ratio than 10 has the effect of increasing the efficiency slightly and increasing the required displacement (for 160 BHP) quite significantly. Therefore, the more detailed total efficiency calculations will now be presented for the ECPE closed air cycle using base pressure level of 150 psia and a pressure ratio of 10 as the design point parameters.

4.2.1 Cycle Analysis

Figures 4.9 and 4.10 are the P-V and T-S diagrams for the closed air cycle ECPE concept at the design point. Point 0 represents the air as it is just about to be inducted into the cylinder through the intake valve (no. 1). As the piston moves from top center (TC) to bottom center (BC) on the intake stroke, the air is drawn into the cylinder (0-0'-1) with a resulting pressure drop due to losses across this valve. This pressure drop was estimated to be about 15% from valve pressure drop and volumetric efficiency information, and will be discussed in detail in the next section. This pressure drop is handled by merely letting the pressure at 0 be 15% higher than the desired induction pressure at 1, (in this case P_o = 176 psia, $P_{o',1}$ = 150 psia). To simplify the cycle calculations, 1 lb. of air was chosen as the amount inducted during the intake stroke. Therefore, for induction conditions of 150 psia and 625°R we have:

$$v_1 = \frac{R\,T_1}{P_1} = \frac{(53.3)\ (625)}{(150)\ (144)} = 1{,}548\ \text{ft}^3/\text{lbm}$$

and

ECPE P- . T-3 CYCLE DIAGRAMS

160 HP DESIGN POINT CHARACTERISTICS

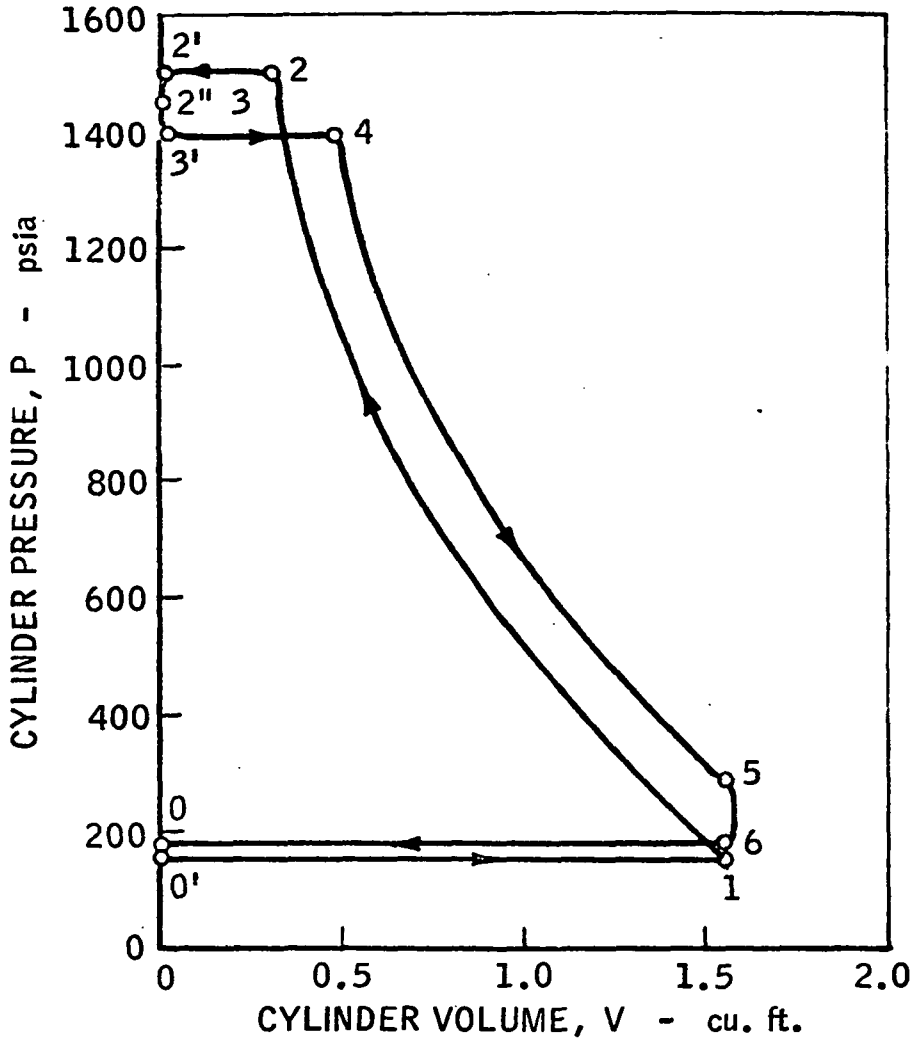


FIGURE 4.9

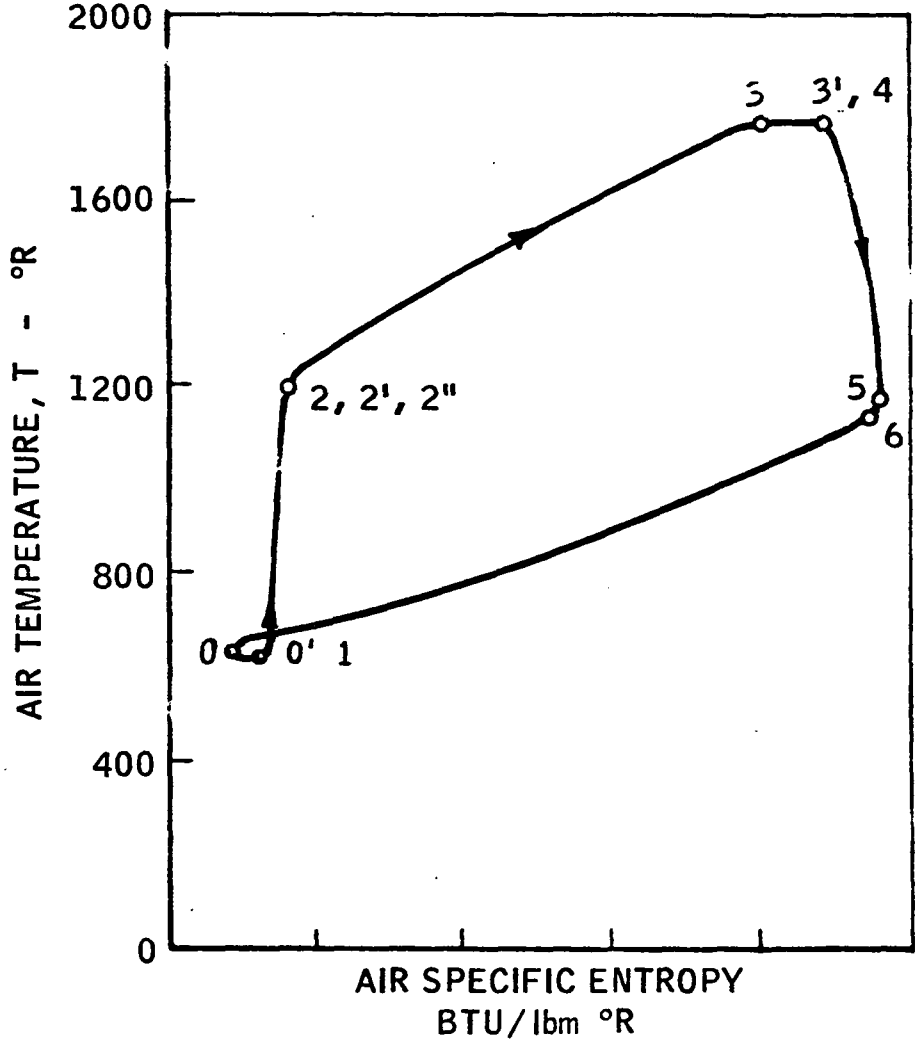


FIGURE 4.10

$$V_1 = 1.548 \text{ ft}^3$$

Where: R = air gas constant $(\frac{\text{ft-lbf}}{\text{lbm } ^\circ\text{R}})$

v = air specific volume (ft^3/lbm)

V = cylinder volume (ft^3)

The work done by the gas on the piston during the induction stroke is:

$$W_{0'-1} = \frac{P_1 v_1}{J} = \frac{(144)\ (150)\ (1.548)}{778} = 43.0 \text{ BTU/lbm}$$

Where: W = work (BTU/lbm)

J = mechanical conversion factor (778 ft-lbf/BTU)

The air is then compressed to the maximum system pressure (1500 psia) from state 1 to 2, and this compression process is assumed to occur at about 98% isentropic efficiency (η_s). This value was estimated from information available on reciprocating compressor design technology. Therefore, for a 10 to 1 pressure ratio we calculate*:

$$T_{2s} = T_1 \left(\frac{P_2}{P_1}\right)^{\frac{k-1}{k}} = 1186^\circ\text{R}$$

k = ratio of specific heats for air = 1.386

$$W_{1-2} = \frac{u_{2s} - u_1}{\eta_s} = (u_2 - u_1) = 101.90 \text{ BTU/lbm}$$

u = air specific energy (BTU/lbm)

where

$$u_2 = W_{1-2} + u_1$$

and $T_2 = 1198^\circ\text{R}$

$$v_2 = \frac{RT_2}{P_2} = .295 \text{ ft}^3\text{/lbm}$$

$$V_2 = .295 \text{ ft}^3$$

When the pressure of the gas reaches 1500 psia (or the volume reduced to .295 ft^3), the compression valve (no. 2) opens, allowing the high pressure air to flow through the combustion heat exchanger. The characteristics of this valve will be discussed in the next section, and the reason that no regeneration is employed will be brought out at the end of

*Values of u, h, Cp, and k used in the cycle calculations are taken from Keenan and Kayes Air Tables.

this analysis. The piston continues doing work to TC pushing the gas out of the cylinder (from 2 to 2'). This work is equal to:

$$W_{2-2'} = \frac{P_2 V_2}{J} = \frac{(144)\ (155)\ (.295)}{778} = 82.04 \text{ BTU/lbm}$$

When the air has been pushed out of the cylinder and through the combustion heat exchanger, it will be forced back into the cylinder through the expansion valve (no. 3) as the piston moves from TC to BC. The temperature of the gas will be at the maximum cycle temperature ($1760^{\circ}R$); however, the pressure will not be at the maximum pressure (1500 psia) since the air has experienced pressure drops through valve nos. 2 and 3, and through the heat exchanger. The pressure drop across the heat exchanger (2"-3) was found to be about 2% (see ECPE heat exchanger section), and the pressure drops across valve no. 2 (2'-2") and valve no. 3 (3-3') were calculated to be 3% and 4% respectively. The pressure drops across these valves will also be discussed in more detail in the next section. The pressure at position 3' and 4 is, therefore, 1365 psia, and the amount of heat added through the combustion is:

$$Q_{in} = h_{1760} - h_{1198} = 148.21 \text{ BTU/lbm}$$

h = air specific enthalpy (BTU/lbm)

Valve no. 3 will remain open until the 1 lb of air at 1365 psia and $1760^{\circ}R$ has been inducted into the cylinder or until the cylinder volume is:

$$V_4 = \frac{RT_4}{P_4} = \frac{(53.3)\ (1760)}{(1365)\ (144)} = .477 \text{ ft}^3/\text{lbm}$$

$$V_4 = .477 \text{ ft}^3$$

Before the valve closes, the gas will have done work on the piston equal to:

$$W_{3'-4} = \frac{P_4 V_4}{J} = \frac{(144)\ (1365)\ (.477)}{778} = 120.57 \text{ BTU/lbm}$$

After the valve closes the gas expands, again with 98% isentropic efficiency (η_e), to position 5. The thermodynamic conditions at this point are:

$$v_5 = v_1 = 1.548 \text{ ft}^3/\text{lbm}$$

$$V_5 = V_1 = 1.548 \text{ ft}^3$$

$$T_{5s} = T_4 \left(\frac{V_4}{V_5} \right)^{k-1} = 1154^{\circ}R$$

The work done by the gas on the piston is:

$$W_{4-5} = (u_4 - u_{5s}) \times \eta_e = u_4 - u_5 = 115.5 \text{ BTU/lbm}$$

and $$u_5 = u_4 - W_{4-5}$$

$$T_5 = 1165^{\circ}R$$

Where: η_e is the isentropic expansion efficiency = .98 (chosen as an average value for the given expansion pressure ratio, 5)

The pressure at position 5 is:

$$P_5 = \frac{RT_5}{V_5} = \frac{(53.3)\ (1165)}{(1.548)\ (144)} = 278 \text{ psia}$$

At this point in the cycle the piston is near BC and valve no. 4, the exhaust valve, begins to open. The high pressure gas in the cylinder immediately "blows down" to the discharge pressure (state 5 to 6). This discharge pressure is 15% higher than the initial induction pressure (+.55% due to waste heat exchanger pressure loss), or:

$$P_{discharge} = P_6 = 176 \text{ psia}$$

The complete blowdown process is best described by referring to the T-S diagram of Figure 4.11. The air in the cylinder, before valve no. 4 opens, is at state i, in the sketch. The "first particle" of air to leave the cylinder expands from state i to state 2 and accelerates through the exhaust port (this is shown as an isentropic process in the sketch). After entering the discharge line, the air dissipates its kinetic energy at the constant discharge pressure (P_d) along path 2-3-4. Note that during the short time required for the particle to execute this process, the flow is essentially a steady throttling process so that the initial and final enthalpies are equal ($h_i = h_4$).

A subsequent particle will expand within the cylinder from state i to state 1, using its energy as change of volume work in forcing the previous

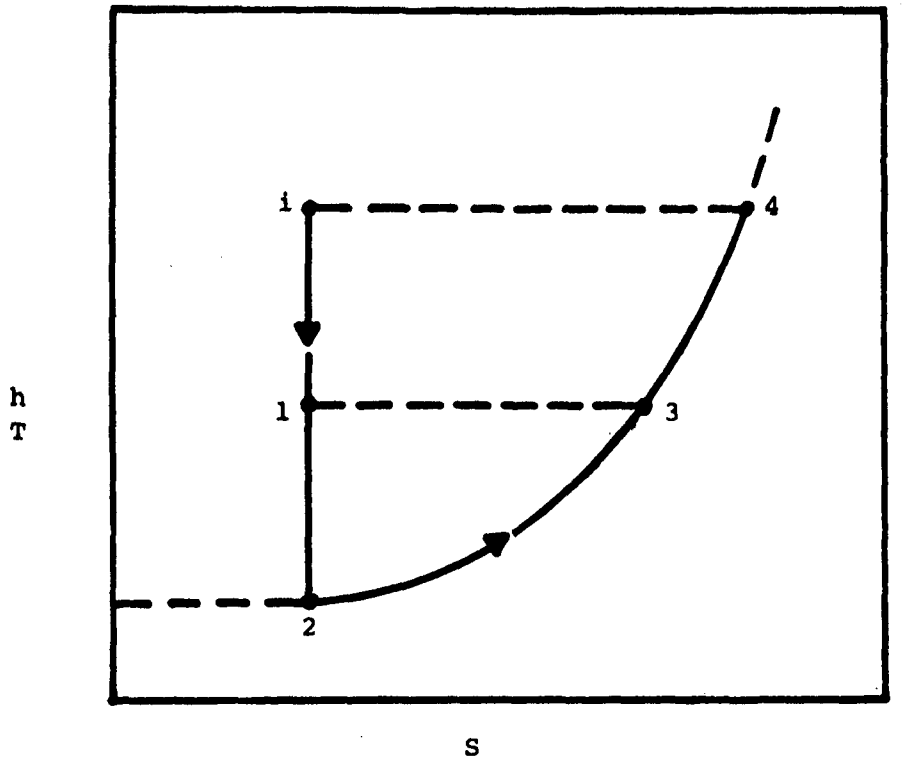


FIGURE 4.11 T-S Diagram of Blowdown Process

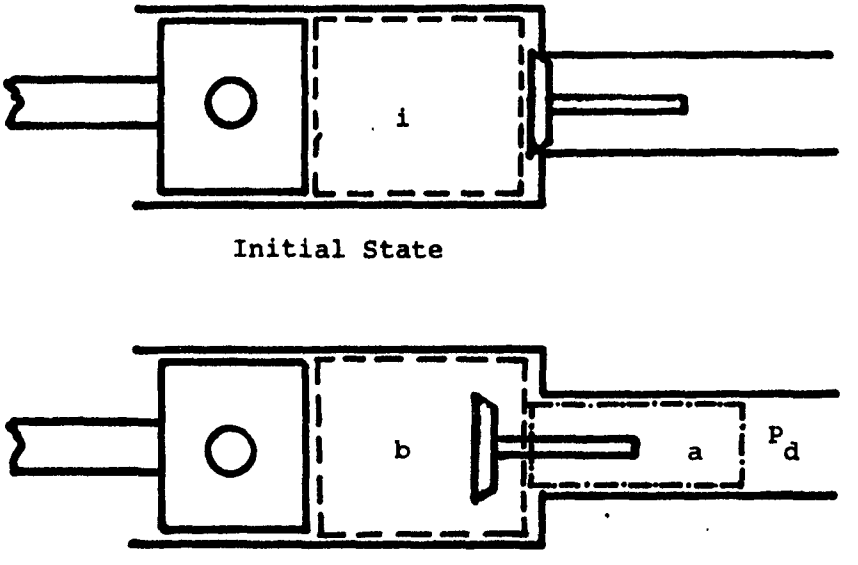


FIGURE 4.12 Sketch of Initial and Final Blowdown Conditions

particles from the chamber. It may leave the cylinder at state 1, for example, and accelerate through the discharge port to state 2. Its velocity at this point will be lower than that of previous particles since its energy potential for accelerations is only $(h_1 - h_2)$. This will dissipate its kinetic energy in the discharge line to a final state of enthalpy $h_3 - h_1$, a lower value than previous particles, due to its lower kinetic energy. All the air which has not left the cylinder when the process is complete (when the cylinder and discharge line pressures are equal) will have followed the process i-2 and have a final enthalpy h_2, having given up their energy as change of volume work, and not having been accelerated. Thus, all air particles having left the cylinder will have a mean energy level lying between h_4 and h_2, whereas all air particles remaining in the cylinder will have the same final state, h_2. Figure 4.12 shows the state-of-the-air before and after the blowdown process. The temperature of the fluid remaining in the cylinder after blowdown (T_b) is:

$$T_b = T_i \left(\frac{P_b}{P_i} \right)^{\frac{k-1}{k}}$$

$$= 1029^{\circ}R$$

Where: $P_i = P_5 = 278$ psia

$T_i = T_5 = 1165^{\circ}R$

$P_a = P_b = P_6 = 176$ psia

$V_b = 1.548$ ft^3

The mass of the fluid remaining in the cylinder after blowdown is:

$$M_b = \frac{P_b V_b}{R T_b} = \frac{(144)\ (176)\ (1.548)}{(1029)\ (53.3)} = .715 \text{ lbm}$$

Now, the energy equation for the blowdown process is:

Irreversible Work = Initial Energy - Final Energy

$$P_a v_a M_a = M u_i = (M_a u_a + M_b u_b)$$

Where: $M = M_a + M_b = 1$ lbm

$u_b = 82.29$ BTU/lbm

$u_i = 107.21$ BTU/lbm

or

$$M_a h_a = M_b u_b = M u_i$$

$$.285\ (h_a) + .715\ (82.29) = (1)\ (107.21)$$

$h_a = 169.6$ BTU/lbm

$T_a = 1096^{\circ}R$

Therefore, the temperature of the air pushed out of the cylinder is 1096°R and the temperature of the air remaining in the cylinder is 1029°R.

Having completed blowdown, the piston begins its upstroke in the cylinder, pushing the remaining air at state b out of the cylinder (6-0).

The work done by the piston on the gas is:

$$W_{6-0} = \frac{P_6 v_6}{J} = \frac{(144)\ (176)\ (1.548)}{778} = 50.7 \text{ BTU/lbm}$$

The final mixed state of the gases after the blowdown and exhaust process is:

$$M_a\ u_a + M_b\ u_b = M_6\ u_6$$

$$(2.85)(94.50) + (.715)(82.29) = u_6$$

$$u_6 = 85.76 \text{ BTU/lbm}$$

$$T_6 = 1048^\circ R$$

T_6 is the mixed final exhaust temperature which would be available for regeneration. The temperature at state 2, however, is higher than this temperature, therefore making regeneration impossible. It should be pointed out here that regeneration may have been feasible if a lower pressure ratio or a higher maximum cycle temperature had been chosen. The maximum cycle temperature was limited to 1300°F, however, due to heat exchanger considerations. Due to this somewhat low maximum temperature, regeneration would not be feasible unless the pressure ratio was dropped below 8 (see Figure 4.8) and even then the cycle efficiency is several points lower than that which can be obtained with the present design parameters.

Thus, the air flow through the waste heat exchanger transferring to the coolant air heat equal to:

$$Q_{6-0} = h_6 - h_0 = 103.6 \text{ BTU/lbm}$$

The air is now at its original state, point 0, and is again ready to repeat the cycle process. The net ideal cycle work is:

$$W)_{\text{net ideal}} = (+43.0) + (-101.9) + (-82.04) + (+120.57) + (+115.52) + (-50.7)$$
$$= 44.5 \text{ BTU/lbm} \times 1 \text{ lbm} = 44.5 \text{ BTUs}$$

therefore, for 1 lbm of air, the cycle gives 44.5 BTUs of mechanical work (ideal). The average frictional loss for internal combustion reciprocating engines (at 3600 rpm) is about 15% of full load power (this will be expanded in the performance section). Therefore,

$$W)_{brake} = 44.5 \times 85\% = 37.8 \text{ BTU/lbm}$$

and

$$\eta_{brake} = \frac{W)_{brake}}{Q \text{ in}} = \frac{37.8}{148.21} = 25.5\%$$

The required air flow rate and engine displacement to produce 160 BHP at full load (3600 rpm) are:

$$\dot{m} = \frac{(160 \text{ BHP}) \ (42.44 \text{ BTU/min})}{(37.8 \text{ BTU/lbm}) \ (1 \text{ BHP})} = 180 \text{ lbm/min.}$$

$$CID = \frac{(180 \text{ lbm/min}) \ (1.548 \text{ ft}^3\text{/lbm}) \ (1728 \text{ in}^3)}{(1800 \text{ cycles/min.}) \ (\text{ft}^3)} = 267 \text{ in}^3$$

In the above closed cycle analysis and in the following open cycle analysis, the only heat losses included are those from the waste heat exchanger and the exhaust gases from the recuperator. The heat losses from these areas will account for most of the heat lost from the engine since these areas act at the only significant heat sinks from both the open and closed loops. Direct heat losses to both the coolant and the surrounding air have been neglected. However, due to the low average operating temperature of the ECPE and to the insulation which will be required on the hot air path, these direct heat losses should be kept to a minimum and should not significantly affect the cycle performance calculations.

Having analyzed the closed loop air system, it is now necessary to analyze the open loop combustion system to determine the overall powerplant performance characteristics. The characteristics of the control system and the heat exchanger designs presented in Sections 4.3 and 4.4 will be used here and their derivation saved for those sections. It will be shown in the performance section (4.5) that it is possible to retain high levels of efficiency over a broad range of vehicle operating conditions with the chosen power control system.

The basic components of the ECPE cycle are illustrated in Figure 4.13. The basic function of the waste heat exchanger is to remove waste heat from the closed cycle; however, it also assists in warming the ambient air before it passes to the recuperator for additional heating. In effect, this preheating of open cycle air in the waste heat exchanger reduces the heating requirements of the burner. Both the recuperator and the combustion heat exchanger are designed for an effectiveness (ϵ) of .75 at 75% of design power.*

*Design power is defined to be 160 BHP at full throttle (150 psi base pressure and full speed, 3600 rpm).

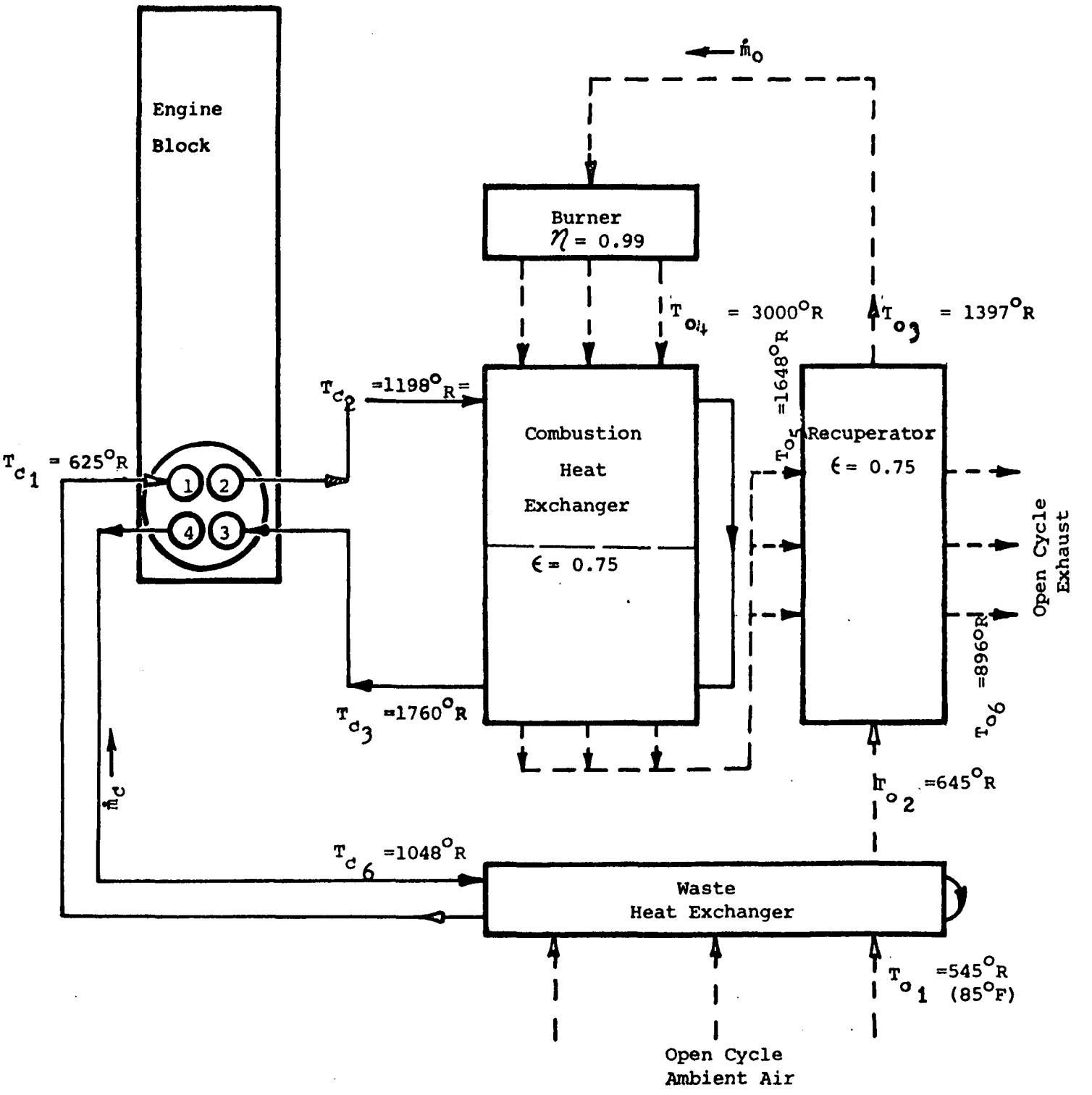


FIGURE 4.13 Diagram of ECPE Basic Cycle and Temperature History at 75% Design Power.

For the ECPE performance analysis it was assumed that the heat transfer effectiveness, combustion efficiency, and the burner outlet temperature ($3000^{\circ}R$) will be constant for the entire engine operating range. In reality, however, heat exchanger effectiveness values will increase as the flow rate through them is reduced, and for all power conditions corresponding to less than 75%, the performance analysis is somewhat conservative. Accordingly, heat exchanger effectiveness will be somewhat less than design for higher power conditions, but automatic transmission shift characteristics (the full throttle shift point occurs approximately at 75% of design speed) essentially limit the operation of the engine to less than 75% power except for the very high speed regime (over 100 mph). This effect will be developed more completely in Section 4.4.

The above heat transfer characteristics were used to calculate the engine efficiency and fuel consumption at full power. Referring to Figure 4.13, the subscripts "C" and "O" refer to the closed cycle and open cycle respectively. Initially, the open cycle (ambient) air enters the waste heat exchanger (WHX) at a temperature (T_{o_1}) of $545^{\circ}R$. This air is heated as it passes through the WHX to approximately $645^{\circ}R$, and is directed through an air cleaner to the recuperator.

Before the heating effect of the recuperator on the air can be determined, the temperature of the hot combustion gases into it (or similarly, the temperature of the gases out of the CHX) must be determined. Using the 75% effectiveness value for the CHX as determined in Section 4.4, we have,

$$\epsilon_{CHX} = .75 = \frac{T_{o_4} - T_{o_5}}{T_{o_4} - T_{c_2}}$$

or

$$T_{o_5} = 3000 = .75\ (3000-1198)$$
$$= 1648^{\circ}R$$

Now the temperature of the air having been heated in the recuperator with 75% effectiveness is given by,

$$\epsilon_{recup} = .75 = \frac{T_{o_3} - T_{o_2}}{T_{o_5} - T_{o_2}}$$

or

$$T_{o_3} = 645 + .75\ (1648-645)$$
$$= 1387^{o}R$$

and similarly the temperature of the hot combustion products leaving the recuperator is:

$$T_{o_6} = 1648-.75\ (1648-645)$$
$$= 896^{o}R$$

Now since the combustion temperature is equal to $3000^{o}R$ (constant), the temperature rise across the combustor is $3000^{o}-1397^{o} = 1603^{o}R$. The ideal fuel-air ratio required to provide this temperature rise (assuming constant pressure combustion and a fuel lower heating value of 18,700 BTU/lb, from Ref. 4). is .0265. Assuming 99% combustion efficiency, this ratio becomes .0268.

It has been shown previously that 148.21 BTU/lbm of heat is transferred in the CHX per cycle, and that at full speed, full power, the flow rate is 180 lb/min. Therefore, the heat transfer rate is:

$$Q_{CHX} = 148.21 \text{ BTU/lbm} \times 180 \text{ lbm/min.}$$
$$= 26,677 \text{ BTU/min}$$

Knowing this and the temperature of the combustion gases into and out of the CHX, the mass flow rate of the combustion products ($\dot{m}_{cp}$) can be determined at full power as:

$$\dot{m}_{cp} = \frac{Q_{transf}}{C_p (T_{o_4} - T_{o_5})}$$

$$= \frac{26,677}{(.2975)(3000-1648)} \qquad C_p = \text{specific heat of the combustion products}$$

$$= 66.31 \text{ lbm/min}$$

and knowing the fuel air ratio = .0268, the mass flow rate of the air ($\dot{m}_a$) before the burner is 64.58 lbm/min. and the flow rate of fuel ($\dot{m}_f$) is 1.73 lb/min.

Therefore, the fuel heat input (Q_f) at full power is:

$$Q_f = \dot{m}_f \ (LHV)$$
$$= (1.73)(18,700)$$
$$= 32,360 \text{ BTU/min}$$

and the overall efficiency of the combustion loop (η_{CL}) is:

$$\eta_{CL} = \frac{Q_{transf}}{Q_f} \times 100$$

$$= \frac{26,677}{32,360} \times 100$$

$$= 82.4\%$$

Finally, the total system efficiency is the product of the cycle efficiency and the combustion loop efficiency or:

$$\eta_{total} = (.255 \times .824) \times 100$$

$$= 21\%$$

and the brake specific fuel consumption at design is:

$$BSFC = \frac{\dot{m}_f \times 60}{BHP}$$

$$= \frac{(1.73)(60)}{160}$$

$$= .649 \text{ lb/BHP-hr}$$

The pressure and temperature history as a function of crank angle for the reference cycle is plotted in Figure 4.14. The analysis to determine the crank angle at which valves no. 2 and no. 3 must open and close will be presented in the following section.

4.2.2 Valve Analysis

A detailed valve analysis is necessary so that the correct valve pressure drops can be incorporated into the ECPE cycle analysis. Since there are four valves in the ECPE cycle, a simple estimating procedure for the valve pressure drops could result in large errors. A preliminary timing sequence for the valve events has been estimated and is shown in Figure 4.15. To understand the mechanics behind this diagram, one must first understand the role played by each of the valves in determining cycle performance. The following discussion will examine the important characteristics of each of the valves.

The operation of the intake valve (no. 1) is relatively straightforward. On the intake stroke the piston moves from top center (TC) to bottom center (BC) drawing a new charge of air into the cylinder. To obtain the maximum amount of work from the cycle, it is necessary to take in as much air as possible during this stroke. Therefore, the valve is opened slightly

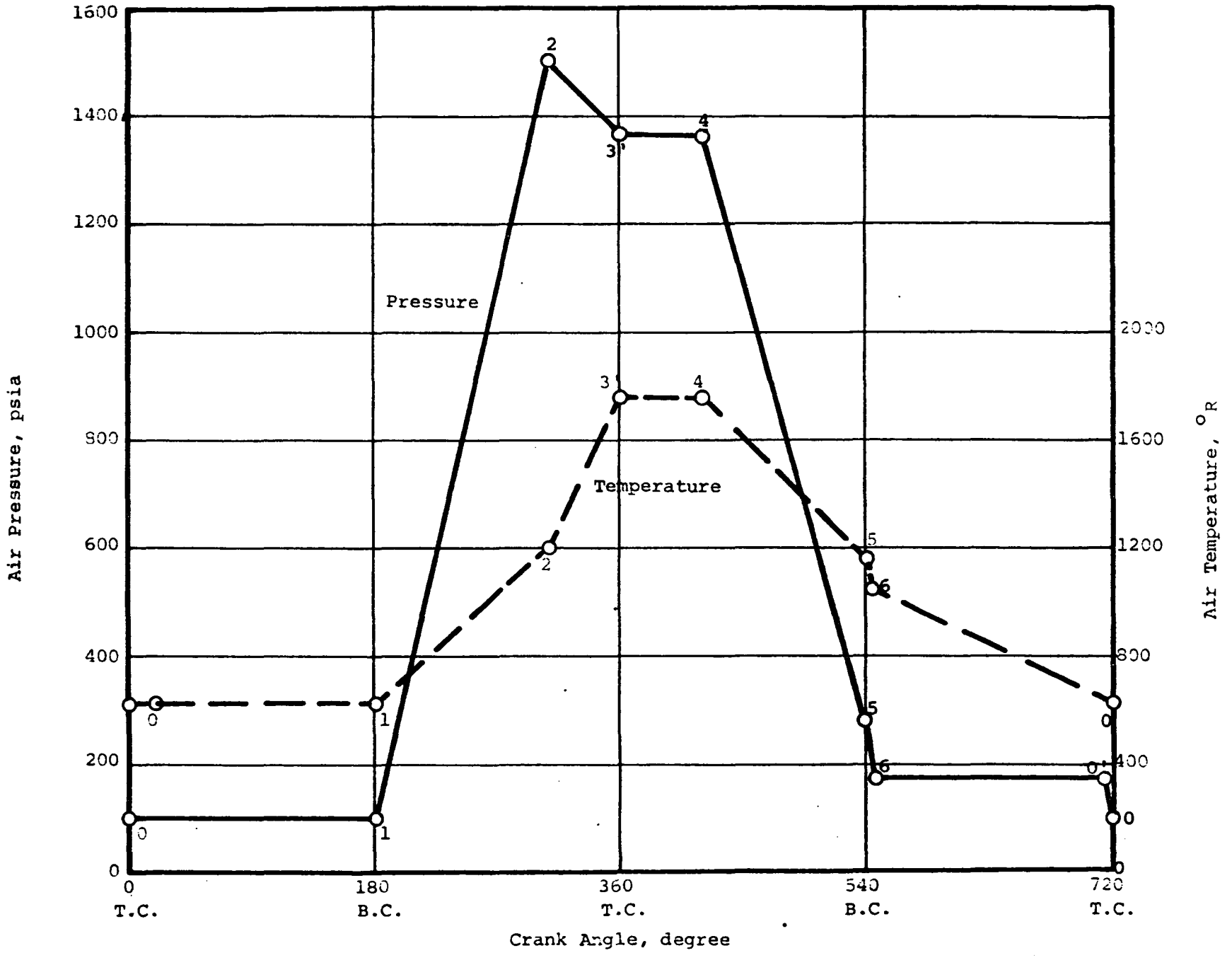


FIGURE 4.14 **Working Fluid (Air) Temperature-Pressure History for the ECPE Cycle.**

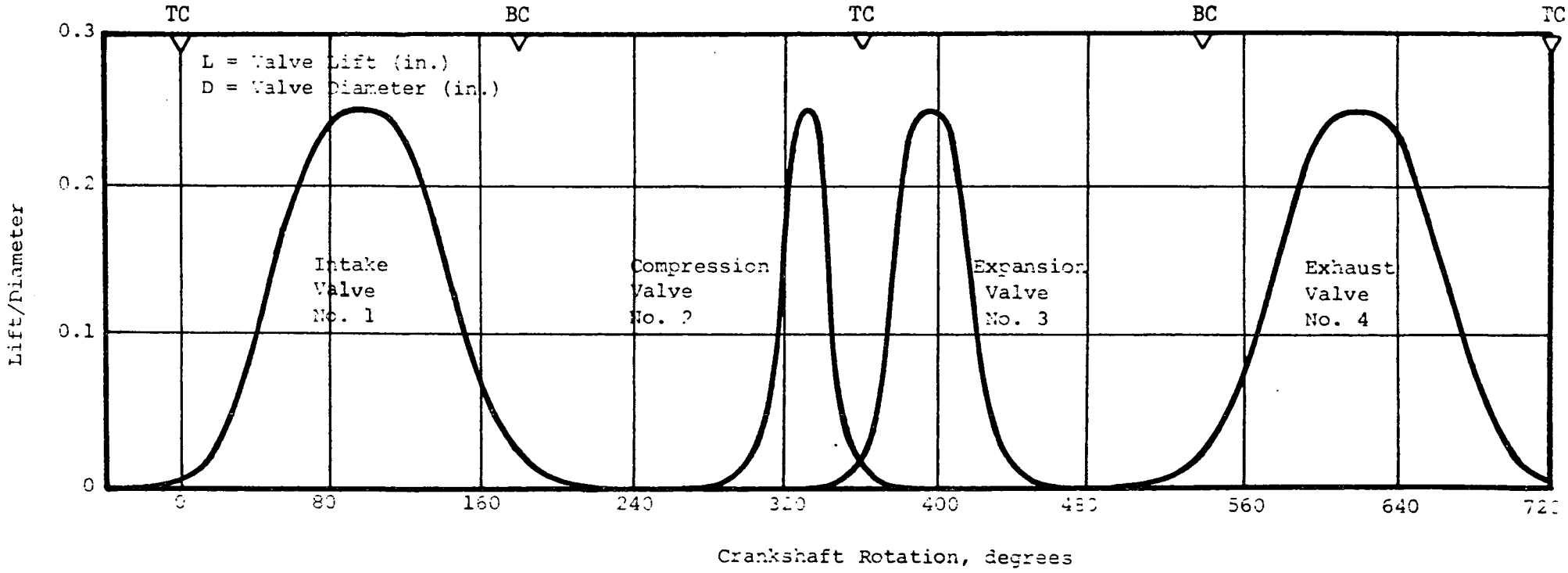


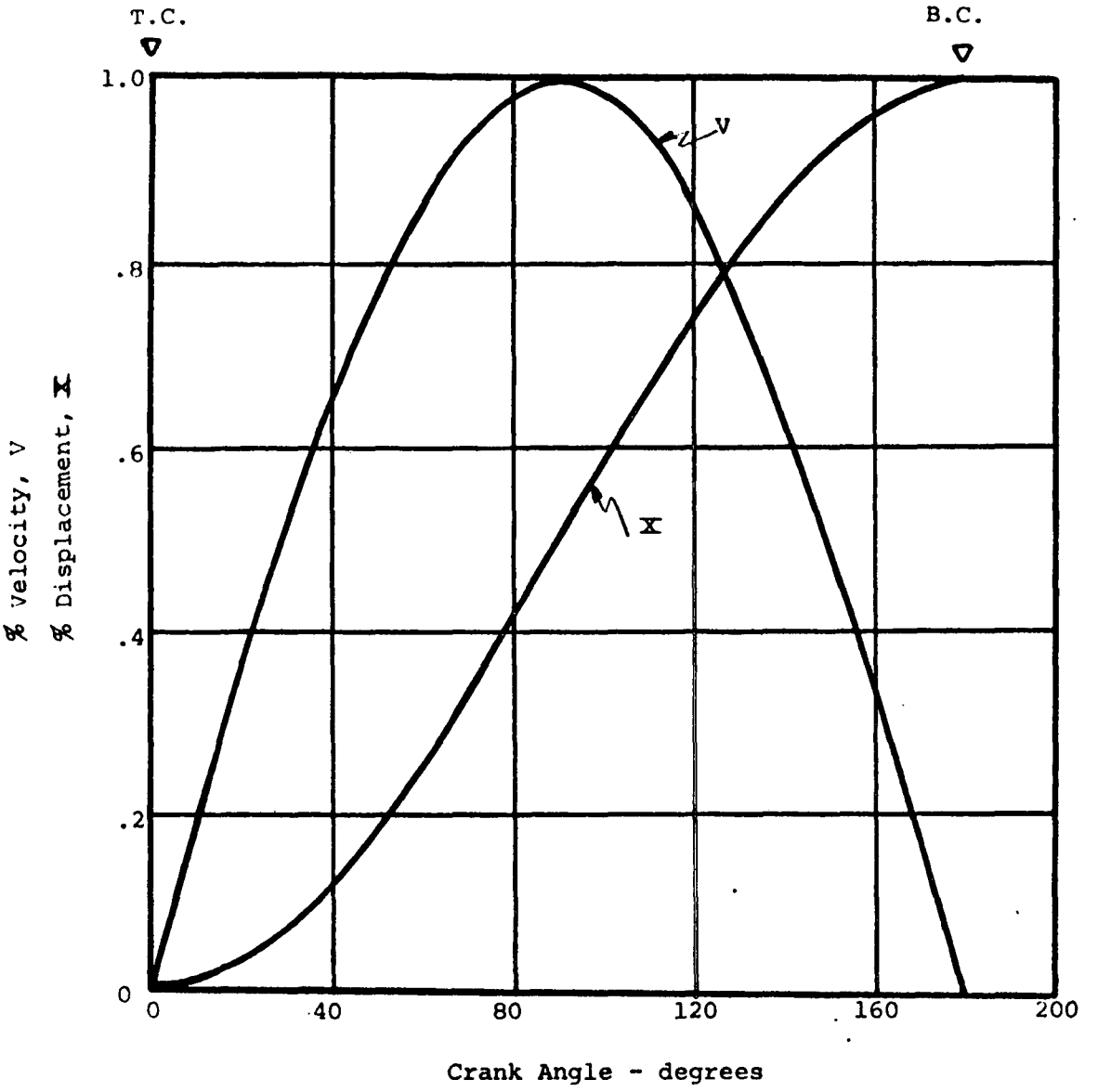
FIGURE **4.15** Preliminary Valve Timing Sequence Diagram for One Complete ECPE Cycle.

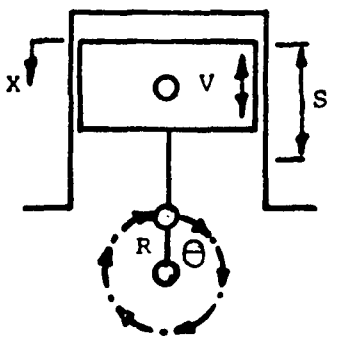
before TC (10°) and closed slightly after BC (30°), giving a total open angle of about 220°. (The detailed analysis of the valve overlap angles will not be attempted here. The angles used are merely typical internal combustion engine values.) Next, as the piston travels from BC to TC, the fresh charge of air is compressed to a pressure and temperature as determined by the cycle thermodynamics. Referring to the reference cycle calculations of the previous section, the pressure, temperature, and specific volume of the air at the maximum compression point are 1500 psia, 1198°R, and .295 ft^3/lbm. To hold these conditions at state 2, valve no. 2, the compression valve must open when the piston is .20 ($= V_2/V_1$) of its stroke from TC.

To determine the crank angle at which valve no. 2 opens requires knowing the piston position as a function of crank angle. This relationship and its derivation are shown in Figure 4.16 where the piston was assumed to move with simple harmonic motion. The crank angle corresponding to X = .20 is about 58° BTC. Valve no. 2 will remain open until just past TC (about 5°) at which time valve no. 3 begins to allow the high temperature air back into the cylinder. It has been assumed that about 10% on each side of the lift curve is a zero flow region. Therefore, the total opening of valve no. 2 including the zero flow region is about 75°.

Now valve no. 3, the expansion valve, will have started to open slightly before TC (5°) and continues to open as the piston moves downward, allowing the high temperature air to fill the cylinder. When the original mass of air has again filled the cylinder, valve no. 3 will close, allowing the air to expand for the remaining portion of the downstroke. For the reference cycle, the properties of the air at the required closing position are $T_4 = 1760^{\circ}$R, $P_4 = 1365$ psia, and $V_4 = .477$ ft^3/lbm. From Figure 4.16 this gives a crank angle at X = .31 ($= V_4/v_5$) of 70° ATC. Therefore, including the 10% zero flow regime on each side of the valve event, the total open angle for valve no. 3 is about 90°.

Now that valve no. 3 has closed, the gases expand in the cylinder until BC at which time valve no. 4, the exhaust valve, opens. Actually, this valve opens slightly before BC so that the blowdown process is completed before the piston starts its upward stroke. Valve no. 4 then remains open during the exhaust stroke to allow the spent air to be pushed





$$X = \left(\frac{1-\cos\theta}{2}\right) = \frac{x}{S}$$

$$V = \sin\theta = \left(\frac{v}{0.262\ nS}\right)$$

x = distance from TC (in.)
v = piston velocity (fps)
s = stroke (in.)
n = rpm

FIGURE 4.16 Diagram Showing Piston Position and Velocity as a Function of Crank Angle.

out of the cylinder and through the waste heat exchanger. It closes slightly after TC (20°) to allow complete removal of the spent gases from the cylinder, giving a total open angle of about 220°. Valve no. 1, the induction valve, is again opened about 10° BTC to allow the new charge of air to purge the spent air from the cylinder and to provide the fresh charge for the next cycle.

Now that the operation of the four valves has been discussed, it remains to determine the appropriate sizes and the resulting pressure drops. Taylor (Ref. 1) uses a term called the Mach Index to determine the volumetric efficiency through valves of various lifts and diameters. This Mach Index (Z) is defined as:

$$Z = (b/D)^2 \times (S/C_i a)$$

Where: b = piston bore, inch
D = valve diameter, inch
S = average piston speed, fpm
C_i = average flow coefficient
a = speed of sound = $2940\sqrt{T_s}$. fpm
T_s = static temperature, $^{\circ}R$

The average flow coefficient used in computing Z was obtained by integrating the flow coefficients at various L/D ratios over the schedule of valve lift versus crank angle. Taylor shows that lifting the valve more than one quarter of its diameter gives diminishing returns in flow capability. Therefore, a maximum L/D ratio of .25 was used for all valves (as shown in Figure 4.15) and the average flow coefficient for this lift for typical internal combustion engine valve design was found to be .33 (see Taylor Volume I, page 176).

Taylor also shows that for the best volumetric efficiency (η_v), and therefore, the best inlet valve design, the Mach Index should remain below about 0.5 or 0.6. Figure 4.17 shows that as Z exceeds 0.6, the volumetric efficiency falls rapidly. Therefore, Z = 0.6 was chosen as the Mach Index for the inlet valve design of the ECPE engine. The resulting inlet valve diameter and pressure drop for a proposed 4.0" bore, 3.530" stroke engine having full load at 3600 rpm is:

$$Z = (b/D)^2 \times (S/C_i a)$$

$$D = b\ (S/C_i a\ Z)^{1/2}$$

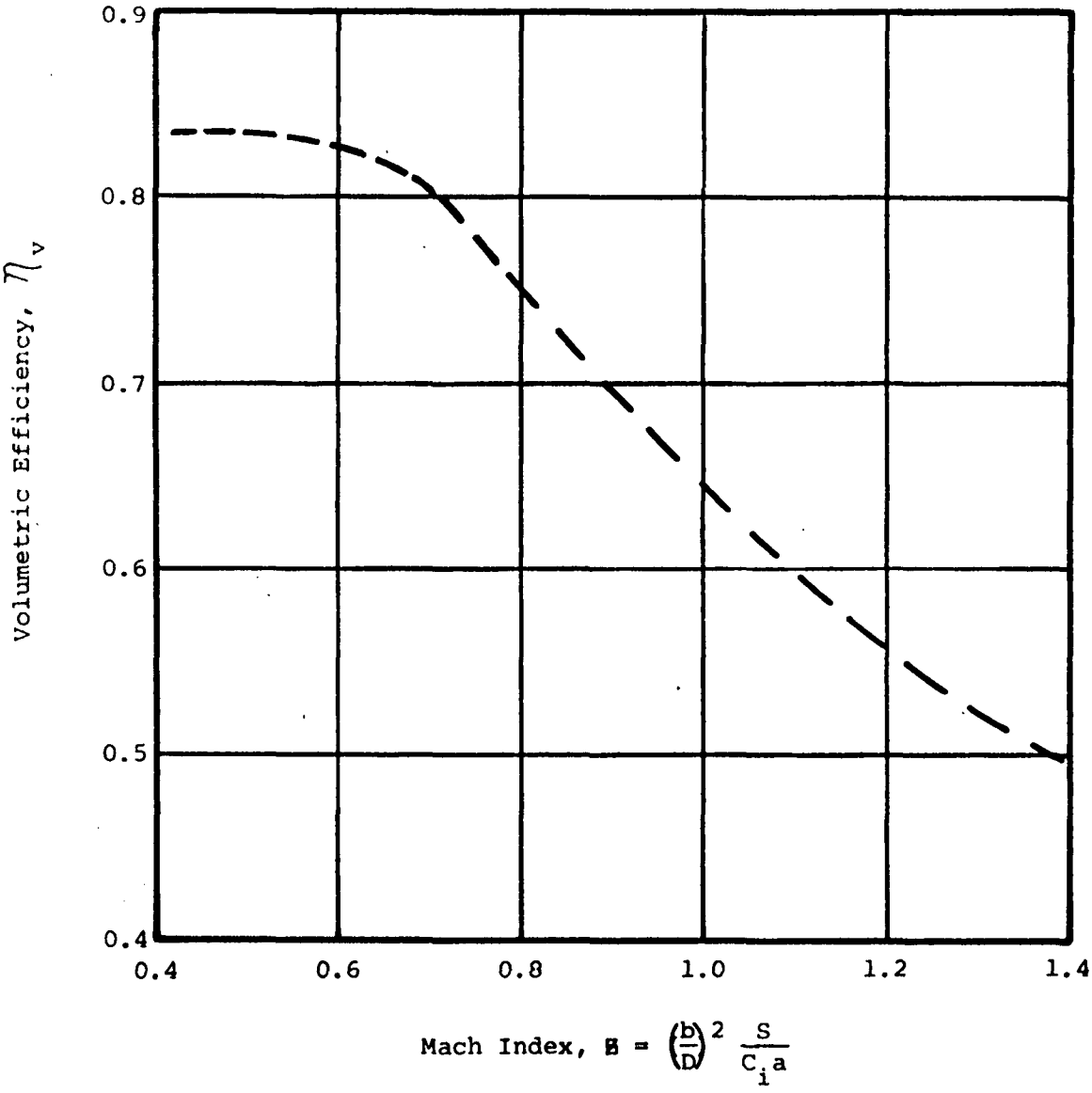


FIGURE 4.17 Volumetric Efficiency Vs Inlet-Valve Mach Index.

where: $b = 4.0$ inches

$S = (3600 \frac{rev}{min}) (\frac{3.53 \text{ in}}{stroke}) (\frac{2 \text{ strokes}}{rev}) (\frac{ft}{12 \text{ in}}) = 2100$ ft/min

$Z = 0.6$

$C_i = .33$

$a = 2940 \times \sqrt{625} = 73{,}500$ ft/min

gives

$D = 1.63$ inch and from Figure 4.17, $\eta_v = .82$ at $Z = .60$.

If all the volumetric losses are assumed to occur across the inlet valve, then the inlet valve pressure loss is 18%. The assumption that the entire volumetric loss is due to pressure drop across the inlet valve is a very good one for the ECPE engine since very little heating will occur between the cylinder walls and the fresh air since there is not a large temperature difference between them. The volumetric efficiency that Taylor gives takes into account the higher air heating that occurs in the spark ignition engine. Therefore, the ECPE volumetric efficiency and the pressure drop are probably less than 18%, probably closer to about 12%. Thus, due to lack of better data, and to be conservative, a value of 15% was chosen.

For the compression valve (no. 2), the Mach Index can again be calculated; however, the volumetric efficiency curve of Figure 4.17 cannot be used to determine the resulting pressure drop. The pressure drop for this valve is determined by finding the pressure ratio corresponding to the calculated Z value from the Mach number tables. A valve diameter of 1.25 inches was chosen as a reasonable size and the resulting Mach Index calculation was:

$Z = (b/D)^2 \times (S/C_i a)$ Where: $b = 4.0$ inch

$D = 1.25$ inch

$C_i = .33$

$a = 2940 \times \sqrt{1200} = 102{,}000$ fpm

The value of S, the average piston speed, is the average speed from the time at which valve no. 2 opens to TC. This average speed can be determined from Figure 4.16 where the piston velocity is plotted as a function of crank angle. Valve no. 2 opens at about 58° BTC for the reference cycle at which time the piston velocity is about .80 V_{max}.

Therefore, $\bar{V}_p = S = .4\ V_{max}$

$$S = (.4)(.262)(3600)(3.530)$$

$$S = 1340 \text{ ft/min.}$$

Consequently,

$$Z = \frac{4.0}{1.25}^2 \quad \frac{1340}{(.33)(102{,}000)}$$

$$Z = .41$$

From the Mach number tables of Ref. 59, the pressure ratio is 0.0875 or a 12.5% drop across the valve. This analysis assumed that the entrance to the valve seat was shart-cornered and that there was no pressure recovery past the valve. Reference 3 gives the correction factors (K_c,C_c) to account for actual valve design as:

$$\Delta P_{actual} = K_c\ C_c \Delta P$$

where:

K_c = contraction loss coefficient = .43

C_c = entrance correction factor = .50 (Ref. 3, page 87)

therefore:

$$\frac{\Delta P_2}{P_{in}} = (.43)(.50)(12.5) = 3\%$$

For the expansion valve (no. 3), the Mach Index is again used in conjunction with the Mach number tables. Several sized valves were studied in an attempt to keep the pressure losses through this valve to a minimum. The final design calculation used a 1.63 inch valve, the same size as the intake valve, and gave the following Mach Index calculations:

$$Z_3 = (b/D)^2 \times (S/C_i a)$$

where: $b = 4.0$ inch

$D = 1.63$ inch

$C_i = .33$

$a = 2940 \times \sqrt{1760} = 123{,}480$ fpm

The value of S was determined in the same manner as for valve no. 2. Valve no. 3 closes at about 70° ATC for the reference cycle giving:

$$\bar{V}_p = S = 2000 \text{ ft/min.}$$

Therefore,

$$Z = (4.0/1.63)^2 \times \frac{2000}{(.33)(123{,}480)} = .29$$

From the Mach number tables of Ref. 2, the resulting pressure ratio is 0.944 or a 5.6% drop across the valve. The correction factor (K_e) to account for the sudden expansion pressure drop is given in Ref. 3 as:

$$\Delta P_{actual} = K_e \Delta P$$

$$K_e = .74$$

therefore:

$$\frac{\Delta P_3}{P_{in}} = (.74) \times (5.6\%)$$

$$= 4.2\%$$

The pressure drop across the exhaust valve (no. 4) was not rigorously calculated due to the fact that the blowdown process includes this pressure drop, and therefore, it does not directly affect the cycle characteristics. The only limitation on valve no. 4 is that it be large enough to allow essentially complete blowdown to discharge pressure before the piston starts its exhaust stroke. With the proper valve opening, it was felt that a 1.25 inch diameter valve would be sufficient for complete blowdown.

The proposed valve arrangement is shown in Figure 4.18 for a 4.0 inch bore cylinder having 4 in-head valves. The design is symmetrical with two 1.63 inch D inlet valves, and two 1.25 inch D outlet valves. (These are standard production Chevrolet valves.) The valve design criteria given by Taylor (Ref. 1) was used to determine the minimum spacing between valves and between the valves and the cylinder wall. In addition, if the calculated clearances between the valves is not sufficient then additional area can be obtained by merely doming the cylinder heads.

4.2.3 Clearance Volume Analysis

In estimating the clearance volume required for the ECPE head design, it must first be determined whether the piston need be recessed to prevent interference between the piston and valves. The sizes and operating characteristics of the four valves has been reviewed in the previous section and it was determined that lifting any of the valves more than one quarter of its of its diameter gives diminishing returns in flow capability. Therefore, the maximum lift for valves no. 1 and no. 3 (D = 1.63 inch) is .407 inch, and the maximum lift for valves no. 2 and no. 4 (D = 1.25 inch) is .312 inch. The valve and piston displacements are plotted as a function of crank angle

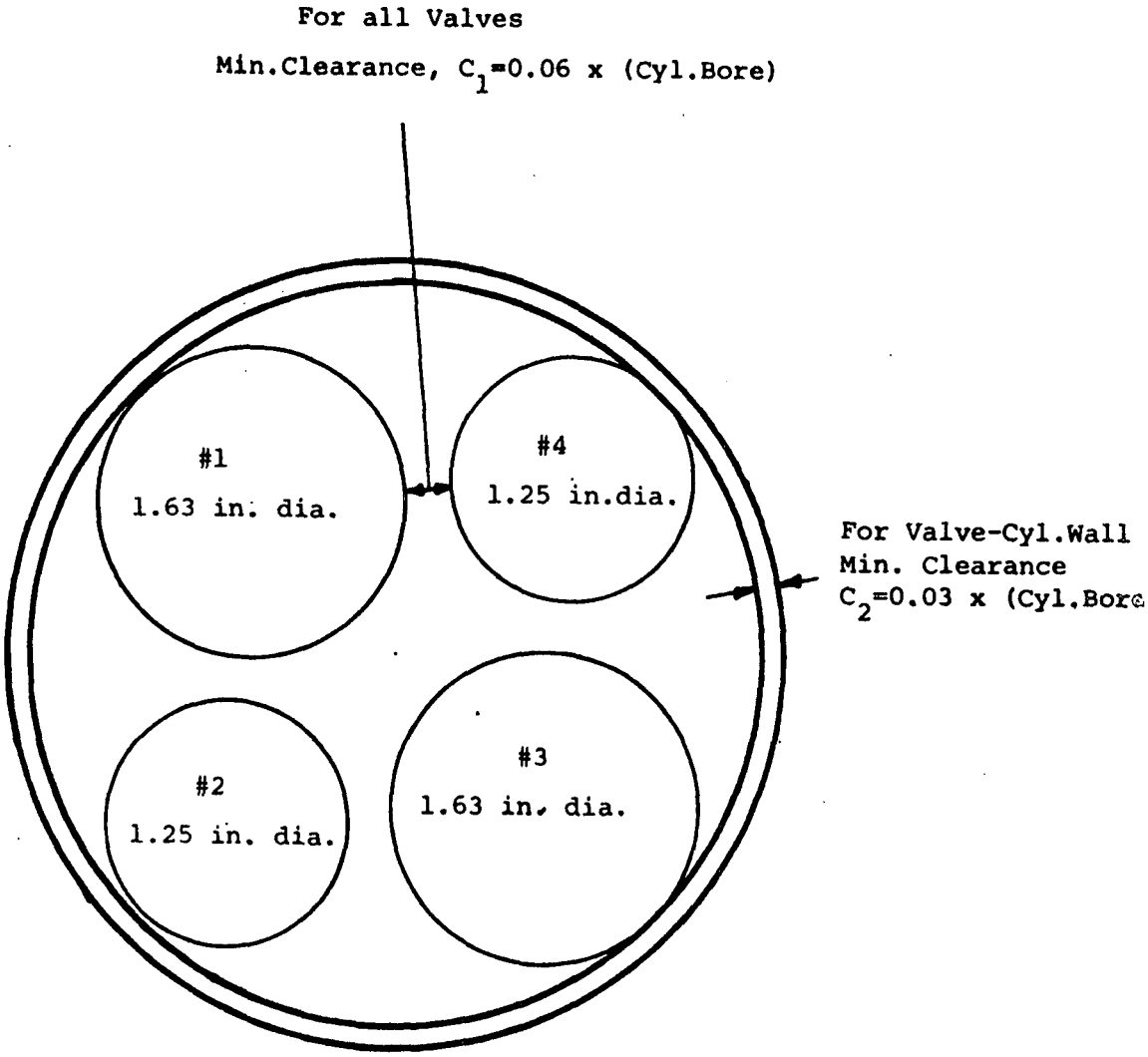


FIGURE 4.18 Proposed ECPE Engine Valve Sizes and Arrangement in Cylinder Head.

in Figure 4.19. The opening and closing angles for the valves are as determined in the previous section and both the valves and piston are assumed to move with simple harmonic motion.

The ECPE powerplant is being developed around a conventional 250 CID L-6 OHV Chevrolet engine. The standard bore and stroke is 3.88 inches x 3.53 inches; however, to provide the required power output (160 BHP), the ECPE displacement needs to be increased to 267 CID. To accomplish this increase in displacement, the bore has been increased to 4.00 inches (stroke remaining constant). This increased displacement and bore should be within the tolerable bore limitations of the stock engine.

Figure 4.19 reveals that there will be no interference between valves no. 1 and no. 4 and the piston for a design having .050 inch mechanical clearance between piston and head. However, to keep the minimum .050 inch mechanical clearance between the valves and pistons (Taylor Ref. 1) requires recessing the piston about .025 inch beneath these valves. Valve numbers 2 and 4, however, will interact with the piston and it appears that the minimum recess required for these valves is about .2118 inch plus the .050 inch mechanical clearance, or .2618 inch. Therefore, allowing .050 inch mechanical clearance on the valve diameters, the total clearance volume is:

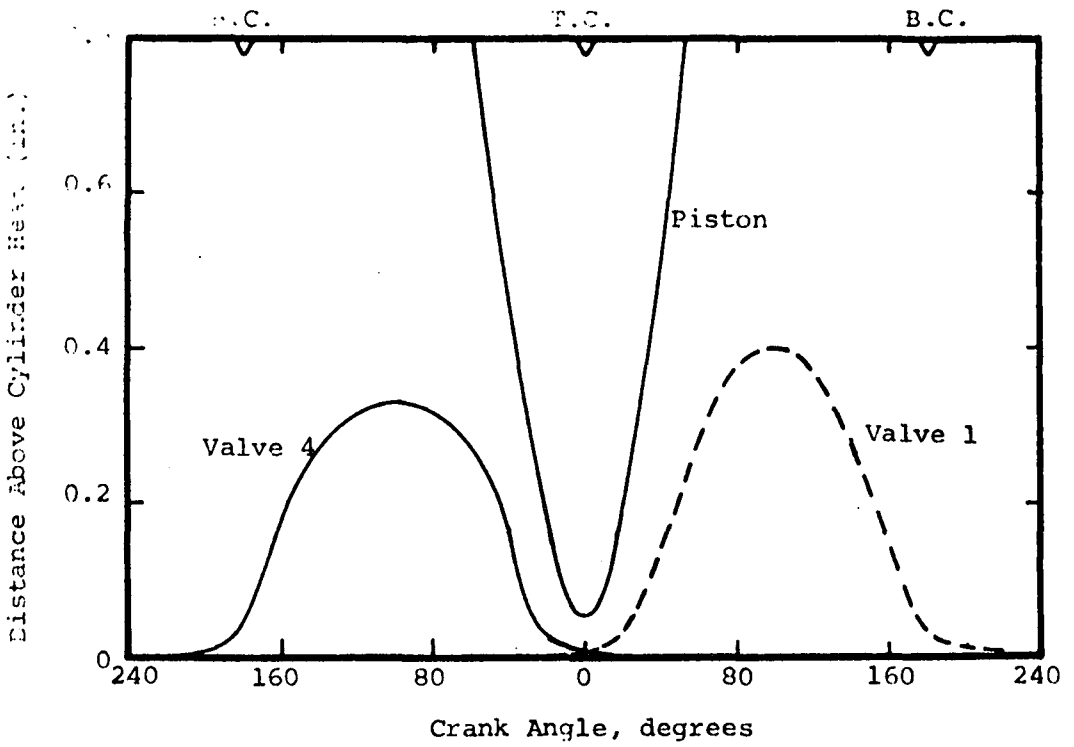
$= 1.60 \text{ in}^3/\text{cyl.} \times 6 \text{ cyl.} = 9.6 \text{ in}^3$

% clearance volume = 9.6/276.6 = 3.4%.

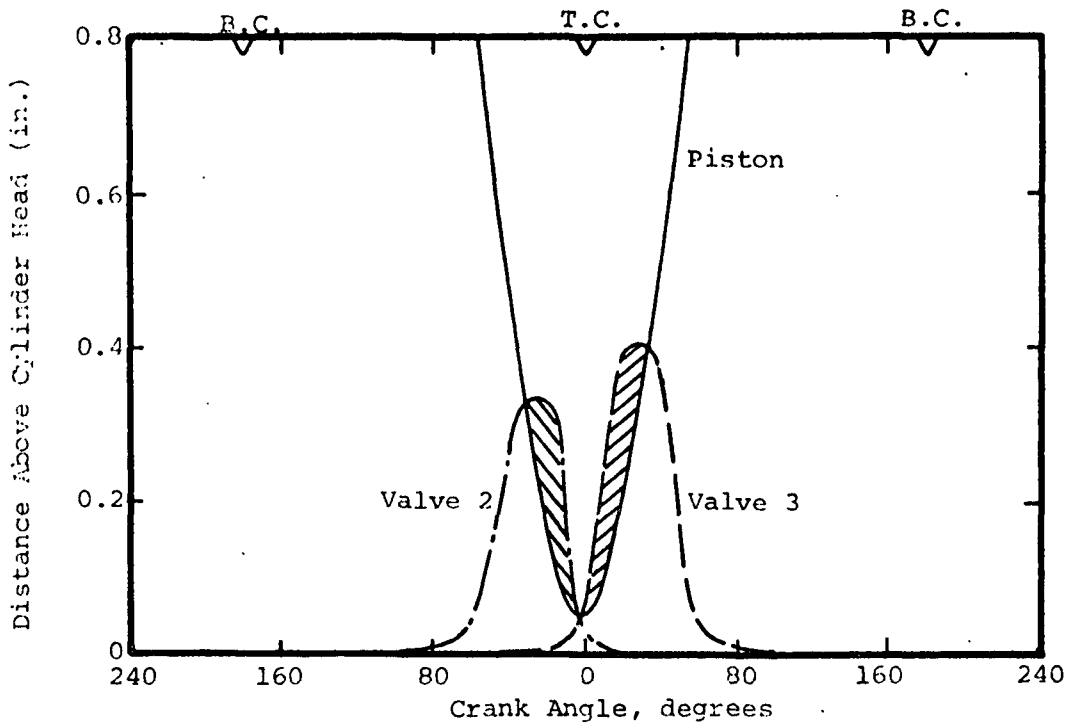
Therefore, the total clearance volume in the proposed engine is only about 3%. The effects of this clearance volume on the previously calculated cycle performance data would be quite small and can be considered to be taken into account in the somewhat conservative value of volumetric efficiency (85%) used in that analysis.

4.3 CONTROL OPERATION

Three power control methods were initially proposed for the ECPE powerplant. These were: (1) control of inlet pressure and pressure ratio, (2) control of maximum cycle temperature, and (3) control of inlet density. It was shown early in the study that the cycle efficiency as a function of power level for each of the three methods was comparable, and since the first two methods require a complex variable valve timing system, the third



a) Valves #1 and #4



b) Valves #2 and #3

FIGURE 4.19 Valve and Piston Motion as a Function of Crank Angle.

method was chosen as the power control system for the ECPE engine. This power control system is illustrated in Figure 4.20. The power controller is envisioned as being similar to the conventional gas pedal in that a mechanical actuation is used to regulate:

1. The base pressure in the closed cycle loop through power valve A, and
2. The fuel flow and air flow to the combustor.

Additional control auxiliaries to the ECPE powerplant include an ignition device to start the burner (some type of battery powered glow plug), a starter motor, a burner air and fuel control, an air storage tank to hold the closed loop air, an air compressor to provide system make-up air, and a conventional automatic transmission.

The air storage tank is a cylindrical steel bottle designed to hold 1.46 ft^3 of air at a maximum pressure of 375 psia. The maximum tank pressure was defined to be equal to the maximum cycle pressure at the proposed cycle idle condition, and the tank was designed to provide almost twice the mass of cycle air required at the full power condition. The air storage bottle is pressurized during deceleration by closed loop bleed air passing through the power valve (A), which is actuated by a deceleration signal from the throttle pedal. Upon completion of the deceleration to idle, the pressure in the high pressure bleed line and the tank should have equalized at 375 psia minus the small drop due to the mass of air which has been lost to the atmosphere by means of blow-by past the rings and valves. The mass of air which remains in the closed loop at idle power is assumed to be lost during extended engine shutdown periods, while the air in the storage tank is retained for the next powerplant start-up.

The mass of air lost each time an engine shutdown occurs can be estimated from the component average volumes, temperatures, and pressures. Bu summation of the engine, heat exchangers, and manifold fluid volumes, it was found that the total ECPE fluid volume was 3030 in^3. The "low pressure" side of the engine, including the waste heat exchanger, low pressure intake and exhaust manifolds, and one-half of the engine displacement, comprises 2170 in^3 of the total volume. The approximate mass of air contained in this volume at idle can be calculated from volume-averaged pressures and temperatures (37.5 psia and 800°R) as:

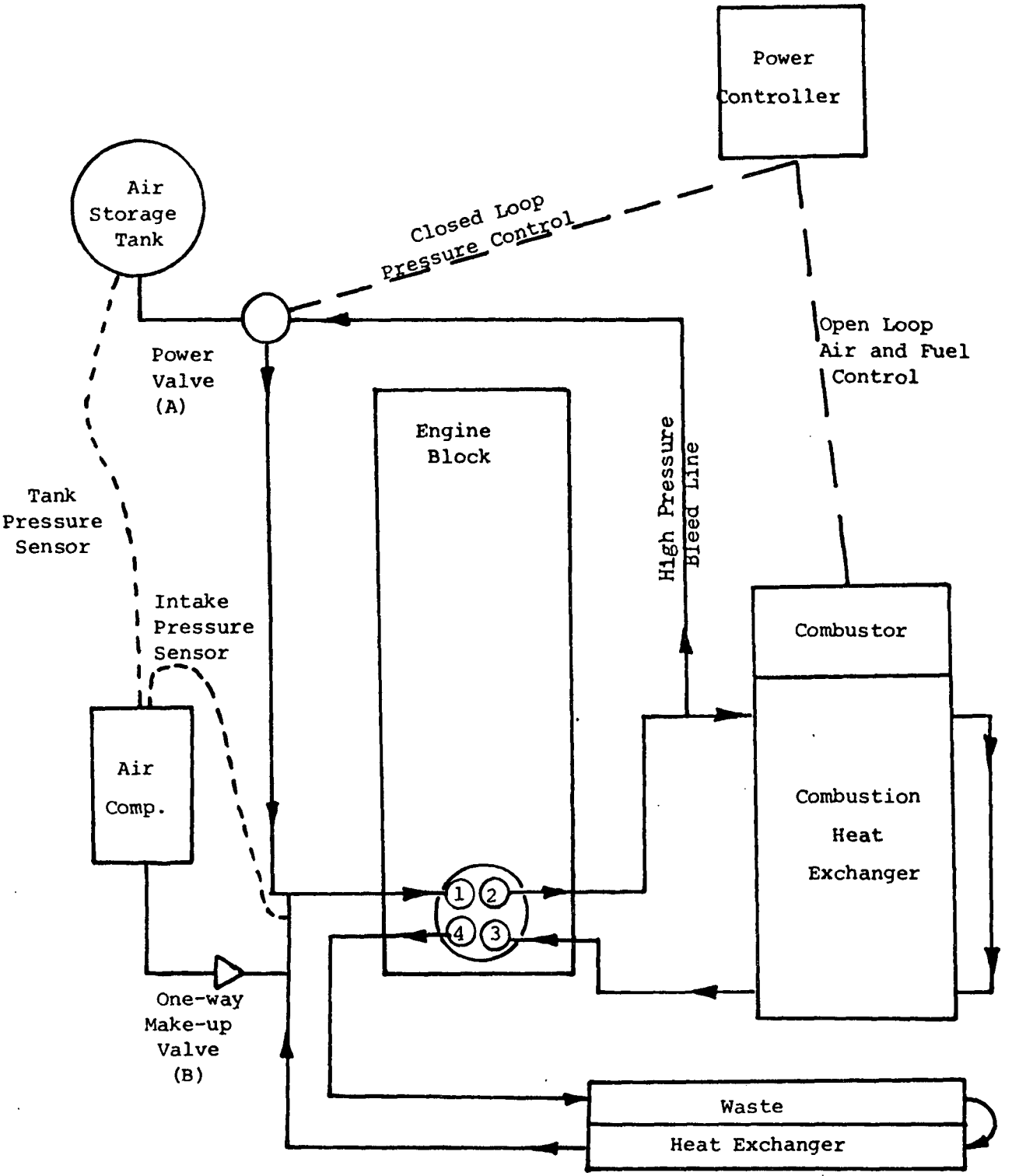


FIGURE 4.20 Schematic of ECPE Power Control System.

$$M_{\text{low side}} = \frac{\bar{P}\,V}{R\,T}_{\text{low}} = \frac{(37.5)(144)(2170)}{(53.3)(800)(1728)} = .16 \text{ lbm.}$$

A similar procedure applied to the "high pressure" side, including the combustion heat exchanger, high pressure intake and exhaust manifolds, and one-half of the engine displacement, results in a mass (at idle) of:

$$M_{\text{high side}} = \frac{(350)(144)(860)}{(53.3)(1400)(1728)} = .34 \text{ lbm}$$

where: $\bar{T} = 1400^{\circ}R$

$\bar{P} = 350$ psia

therefore, the total mass of air lost during an extended engine shutdown period is:

= .50 lbm - (air mass at ambient conditions)

= .50 - .128

= .372 lbm

The storage tank has been pressurized to contain:

$$M_{\text{tank}} = \frac{P\,V}{R\,T} = \frac{(375)(144)(1.46)}{(53.3)(545)} = 2.72 \text{ lbm}$$

The shutdown loss is therefore seen to represent almost 12% of the total system mass. A loss of this amount of air plus an additional amount of blow-by past the rings and valves during normal operation must be replaced by the air compressor during the next start-up. An air compressor installed for this purpose will be actuated automatically at idle power conditions when the tank pressure falls below 735 psia. Once actuated, the compressor will supply make-up air to the system through a one-way valve (B) located on the low pressure intake side of the cycle. The compressor will provide approximately 3:1 pressure ratio air from the atmosphere at such a rate that operation at idle power for 10 second upon starting will resupply the system with the mass loss and pressurize the tank to 375 psia. The compressor requires about 2.0 HP at a maximum volumetric flow rate of 19.3 ft^3/min. to supply this required make-up air.

A description of the operation of the system from "ignition on" to "ignition off" will be useful in describing the function of the individual components. It begins with the transmission in neutral, the ignition key turned to start, and the accelerator pedal depressed. Current from the battery heats the igniter (glow) plug and engages the starter motor, which "turns over" the engine. A preliminary investigation has shown that

the power required for the starter should be only slightly larger than that required for an internal combustion engine starter. This is due primarily to the slightly higher than ambient base pressure, which results in a higher work of compression while turning over the engine.

The mechanical motion of the power controller pedal then opens the power control valve (A) to increase the working pressure level. The position of the pedal is also sensed by the burner control, which precisely regulates the air and fuel quantities through the open burner cycle. The closed cycle air should be brought up to idle conditions within a few revolutions of the engine, and with the glow plug operating, the engine should sustain idle power. The air compressor will immediately begin to supply system make-up air if needed at the defined idle inlet conditions. When a dashboard light (for instance, a red light indicates that the air compressor is operating) goes off, the driver can put the car in gear and begin to accelerate, assured that he has a fully pressurized storage tank for maximum acceleration requirements.

As the pedal is depressed, the air from the storage tank flows through the power valve (A) to bring the cycle up to the desired power level (full pedal will be 150 psia inlet pressure). It should be mentioned here that the time lag to go from, say, idle to full power, has not been fully determined. The time lag due to increasing the combustion heat exchanger internal pressure from that at idle power (300 psia) to that at full power (1500 psia) has been roughly determined to be under 1 second (at the idle speed of 600 rpm). However, it is felt that the heat transfer lag is of secondary inportance. Since burner operation (including transients) is being studied by other OAP contractors, it will be necessary in any follow-on study to combine the ECPE transients with the burner transients to determine the overall engine performance with respect to such considerations. At the full power condition, almost 50% of the total system mass remains in the air storage tank, which would be sufficient to supply cycle air lost due to blow-by over extended driving periods or to supply air for a sudden burst of power above the 100% design power level.

Deceleration is accomplished by release of the pedal, which allows high pressure air to flow back into the air storage tank through the power valve (A). Engine power and vehicle speed are reduced as the closed

cycle air and the open cycle fuel and air flow are simultaneously reduced to the desired level. In addition, during deceleration, the engine acts as a compressor, storing energy in the high pressure storage tank and increasing engine braking ability. Finally, turning off the ignition switch will extinguish the igniter spark and stop the burner fuel flow, thereby stopping the engine.

4.4 HEAT TRANSFER ANALYSIS

The three heat exchangers employed in the ECPE cycle (disregarding the coolant radiator, for the moment) are the combustion heat exchanger (CHX), the waste heat exchanger (WHX), and the recuperator. These three components exert a significant influence on engine performance and operating characteristics, cycle efficiency, engine cost, etc. Because of the importance of these heat exchanger elements, considerable effort has been devoted to establishing the interrelationship between cycle parameters and heat exchanger designs in order to reduce the required sizes (reduce weights and costs), improve the shapes (for better packaging in the engine compartment), and reduce the fan power requirements. The interrelationship between the open and closed cycle loops and between the various heat exchanger elements is illustrated in Figure 4.21. The closed loop air flow path has been reviewed in a previous section and the open loop flow path is as follows:

Ambient air ($85^{o}F$) is pulled through the WHX by a fan (not shown in Figure 4.21) which sits directly behind it and which is belt driven from the engine. This fan may be the same fan or a separate fan from that which is needed for the coolant radiator. The air temperature after the WHX is approximately $185^{o}F$.

A portion of this heated air is then ducted to the recuperator inlet, where another fan will provide the required pressure rise to force the air through the entire open loop combustion system (including air cleaner, recuperator, burner, CHX, and exhaust ducting). This fan will be an integral part of the fuel control system and it is expected that it will be electrically driven to provide a controlled amount of air-flow for proper air-fuel ratios and clean emissions.

ECPE COMBUSTION LOOP SCHEMATIC

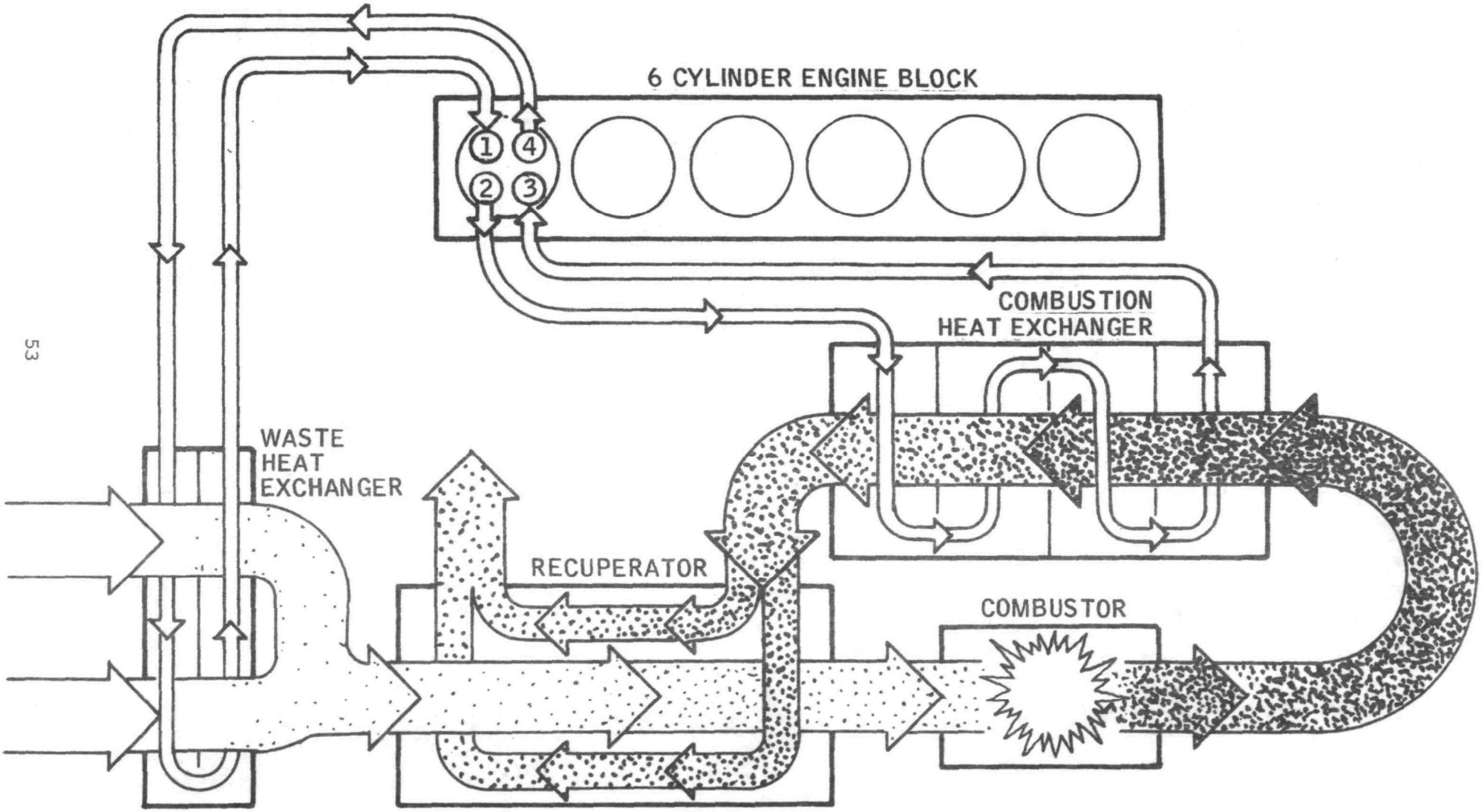


FIGURE 4.21

Having passed through the air cleaner, the heated air passes directly through the counterflow, plate-fin recuperator. The air is heated to approximately 1000°F and ducted to the combustor where it is mixed with fuel and burned to its maximum temperature of 2540°F. The high-temperature combustion gases then pass straight through the cross-flow combustion heat exchanger where heat is transferred to the closed cycle working fluid, these gases leaving the CHX at about 1200°F. Then the combustion gases are turned into the recuperator where heat is transferred to the incoming open cycle air, and finally the gases leave the recuperator and enter the exhaust pipe at about 400°F.

It should be evident from Figure 4.21 and the above flow description that the ECPE heat exchanger designs play an important part in the feasibility of the entire ECPE concept. The efficiency of the powerplant is very dependent on the ability of the heat exchangers to transfer the heat efficiently from the open loop combustion gases to the closed loop working fluid (air). In addition, the heat exchanger designs must be such that they fit comfortably in the available under-the-hood space of the prototype vehicle. Therefore, high efficiency and compact size were the two prime requisites for the ECPE heat exchanger designs.

To facilitate high heat exchanger efficiency in a compact package, it was decided to use 75% of maximum power for heat exchanger design purposes. This 75% of maximum power was arbitrarily selected as a compromise between the high speed cruise (40 to 50% power) and maximum speed/acceleration vehicle operating conditions, and results in less drastic heat exchanger off-design operation. By designing for 75% of full power, the heat exchanger material requirements were made less severe, and overall sizes and costs were reduced. This was directly due to the fact that lower mass flows, pressures and temperatures are required at 75% power than 100%. Full power will still be available, however, and this will be possible by pushing the heat exchanger maximum pressures and temperatures slightly above design for these short high power periods (less than 2% of time spent near full power (Ref. 15)). The decision to use 75% of maximum power as the design point was also motivated by the fact that with an automatic transmission the engine rpm will rarely exceed 75% of design speed due to

design shifting characteristics (except at high road speeds in high gear). Additional assumptions and characteristics used in the heat exchanger and combustion system analyses are:

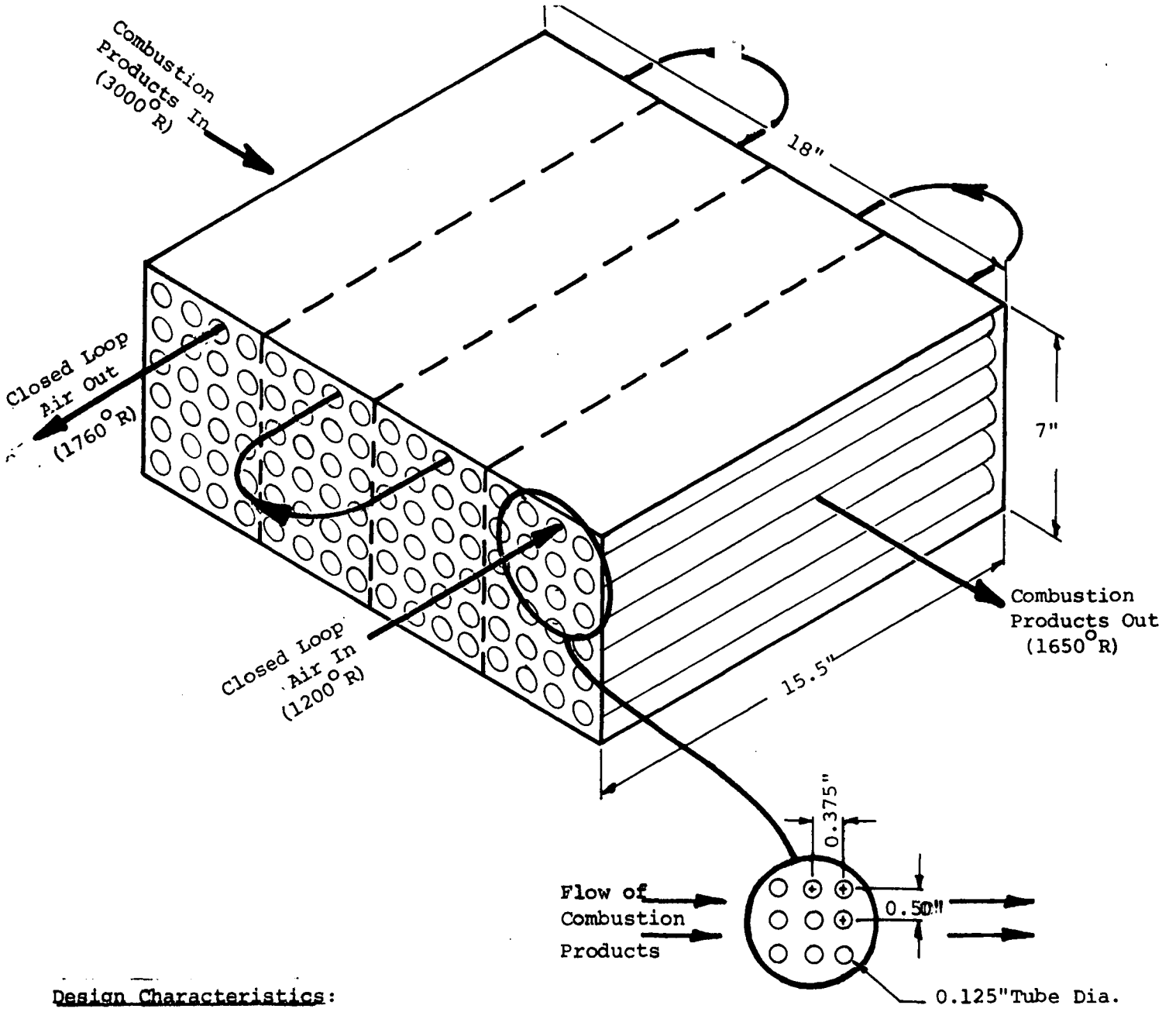
1. Waste heat exchanger and recuperator fan air is 185°F and 14.7 psia on an 85°F day.
2. Normal combustion products temperature is 2540°F.
3. The open and closed loop temperatures remain approximately constant for all normal operating conditions. This will be true except for slight heat exchanger effectiveness (ϵ) changes as a function of speed and ambient temperature variations.
4. The burner pressure drop is 3% of inlet. (The low-emission external burner is being developed under other OAP contracts.)
5. The air cleaner and exhaust pipe pressure drop was calculated to be about 4%.

The combustion loop thermal efficiency (defined as heat transferred in CHX divided by heat released from fuel) was calculated to be about 82% at design point. The detailed open loop calculations for this efficiency have been presented in section 4.2. The individual heat exchanger designs will now be presented.

4.4.1 Combustion Heat Exchanger (CHX)

The representative CHX and pertinent design data are presented in Figure 4.22. It is a bare tube, four pass, cross-counterflow design based on a maximum combustion products temperature of 3000°R and exchanger design effectiveness of 75%. The overall dimensions are 7 x 15.5 x 18 inches (H x W x D) with an estimated total weight of about 35 lbs, based upon the use of type 347 stainless steel tubes with .0125 inch wall thickness. Among the more important of the CHX design characteristics are:

1. 1/8 inch OD tubes with vertical in-line spacing to increase the air-side heat transfer coefficient (h), thereby lowering the metal temperature (1400°F limit). Harrison Radiator states that 1/8 inch diameter tubes are realistic, however, they tend to be expensive. Future optimization studies should investigate increasing tube diameter and



Design Characteristics:

Tube Outside Diameter = .125 inch
Tube Arrangement - Vertical, In-line (see sketch)
Tube Material - ~~18-8~~ stainless steel, .0125 inch thickness
Maximum Tube Wall Temperature = 1400°F
Header Diameters, Inlet and Outlet = 2 inch
Estimated Total Weight (frames, shells, headers, etc.) = 35 lbs.
Estimated Cost About $160.00

Operating Characteristics:

Combustion Products Flowrate = 50 lb/min.
Combustion Products Pressure Drop = .6%
Ideal Air Fan Horsepower = 0.3 HP
Closed Loop Air Flowrate = 135 lb/min.
Closed Loop Air Pressure Drop = 2.2%
Effectiveness (ϵ) = 75%

FIGURE 4.22 Representative ECPE Combustion Heater Design.

using internal turbulators or ring dimpled tubes to increase the internal h coefficient.

2. Type 347 stainless steel, which is representative of a stabilized 18-8 grade, is used in the CHX primarily because of its known favorable high temperature, corrosion-resistant brazing characteristics. It has adequate short-time tensile strength and ductility at 1200°F, with stress rupture properties high enough for uprating up to possibly 1400°F. It has adequate corrosion resistance and excellent resistance to oxidation at temperatures up to 1700°F. It is also possible that with the great strides being made in the use of ceramics for high temperature applications, much of the CHX casing, tube supports and manifolds can be made from this material, thereby greatly reducing the overall cost. It is unlikely, however, that the tubes can be made of ceramics due to the large internal pressures (1500 psia).

The CHX design is such as to allow easy manifolding between the burner and the recuperator, and to and from the no. 2 and no. 3 valves of each of the six cylinders.

4.4.2 Recuperator

The recuperator and pertinent design data are presented in Figure 4.23. It is a counterflow, strip-fin, plate-fin heat exchanger designed for an effectiveness of 75%. The overall dimensions are 9 x 9 x 18 inches (H x W x D) with an estimated total weight of about 30 lbs based on the use of a low-alloy steel with .004 inch fin thickness. AiResearch Manufacturing Co. has stated that the counterflow design as presented here can be manufactured as easily as a crossflow design and with a significant increase in effectiveness. Future optimization studies should investigate reducing the plate thickness (perhaps from .01 inch to .006 inch) and also reducing the plate spacing. The recuperator manifolding allows for the exhaust gas to be ducted toward the ground for easy exhaust pipe pickup and back towards the rear of the car (the exhaust piping will further lower the exhaust temperature to a safe level).

ECPE RECUPERATOR DESIGN

COUNTERFLOW STRIP-FIN, PLATE-FIN HEAT EXCHANGER
SURFACE 1/4(a) - 11.1 FOR AIR SIDE
SURFACE 3/32 - 12.22 FOR COMBUSTION PRODUCTS SIDE

DESIGN CHARACTERISTICS:

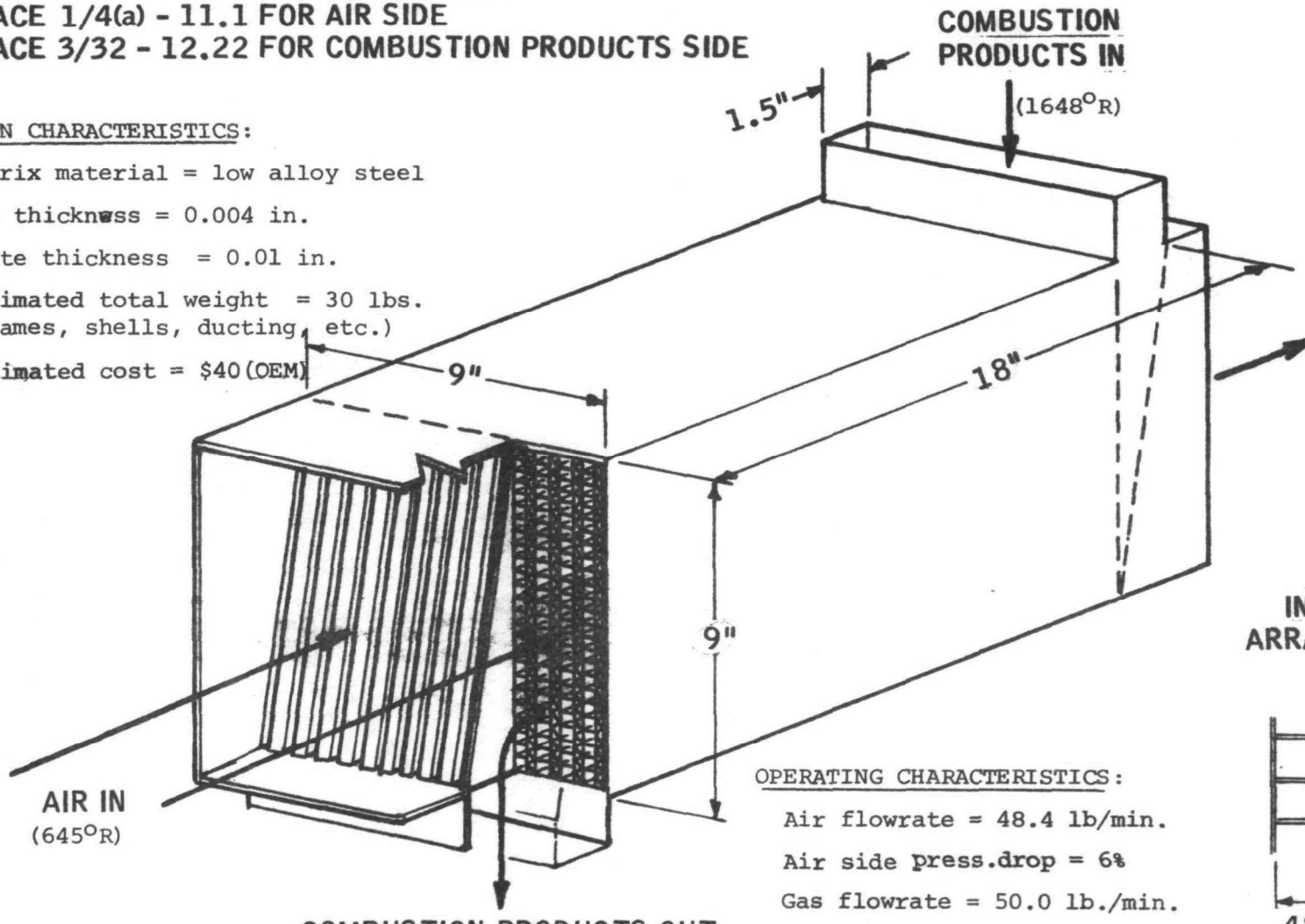
Matrix material = low alloy steel

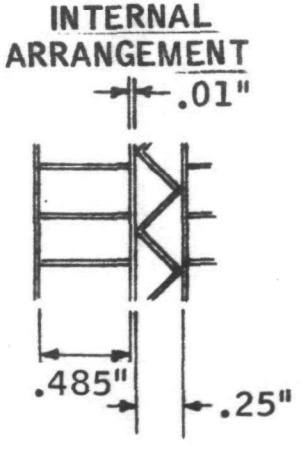
Fin thickness = 0.004 in.

Plate thickness = 0.01 in.

Estimated total weight = 30 lbs.
(frames, shells, ducting, etc.)

Estimated cost = $40 (OEM)





OPERATING CHARACTERISTICS:

Air flowrate = 48.4 lb/min.

Air side press.drop = 6%

Gas flowrate = 50.0 lb./min.

Gas side press. drop = 5%

Ideal fan hp = 7 hp

Effectiveness (€) = 0.75

FIGURE 4.23

4.4.3 Waste Heat Exchanger (WHX)

The waste heat exchanger and pertinent design data are presented in Figure 4.24. It is a two-pass, cross-counterflow with plain, plate-fin surface on both sizes (19.86 fins/inch). The overall dimensions are 22 x 18 x 4 inches (H x W x D) with an estimated total weight of about 25 lbs based on the use of aluminum with .006 inch fin thickness and .01 inch plate thickness on both sides. This design was found to be much more effective and also much less expensive than a finned-tube type arrangement (as used in automotive radiators) of the same size. The two pass arrangement on the working fluid side permits easy manifolding to and from the no. 1 and no. 4 valves of each of the cylinders. In addition, the manifolding arrangement to and from the WHX has been designed to allow as much cooling as possible from the headers and piping. With finned manifolds it is felt that at least 10% of the waste heat can be removed in this manner. As mentioned previously, the WHX serves the double purpose of cooling the working fluid and also preheating the ambient air into the recuperator. One additional point to note is the fact that the ideal fan horsepower required does not take into account the fact that, under cruising conditions, the WHX will take advantage of the dynamic head due to vehicle motion. For instance, for cruising (no acceleration) at 75% of maximum power, the vehicle will be travelling approximately 95 mph and will supply almost 80% of the required ideal HP, or the ideal HP required will only be about 2 HP. In addition, it has been determined that with the proper fan arrangement (including a slipping clutch or possibly even an electrically driven fan) and ducting, fan efficiencies on the order of 65-75% should be obtainable.

4.4.4 Off-Design Heat Exchanger Performance

As mentioned previously, the heat exchangers were designed at 75% of full power and the performance characteristics (ϵ, ΔP) determined at that level were used for the entire power range. This was done to reduce the number of laborious heat exchanger calculations and, as will be shown, represents a very conservative approach for engine performance over the normal operating regime. At power levels below the design point, which represents most driving conditions (about 95%) the effectiveness will increase

ECPE WASTE HEAT EXCHANGER (WHX)

CROSS-COUNTERFLOW, PLAIN, PLATE-FIN HEAT EXCHANGER

SURFACE 19.86 FOR BOTH SIDES

DESIGN CHARACTERISTICS:

- Matrix material = aluminum
- Fin thickness = 0.006 in.
- Plate thickness = 0.01 in.
- Estimated total Weight = 25 lb.
- Estimated cost = $35.00 (OEM)

OPERATING CHARACTERISTICS:

- Cooling air flowrate = 524 lb/min.
- Cooling air side ΔP = 1.3%
- Working Fluid flowrate=135 lb/min.
- Working fluid ΔP = 0.55%
- Ideal fan HP = 9 hp
- Effectiveness = 0.82

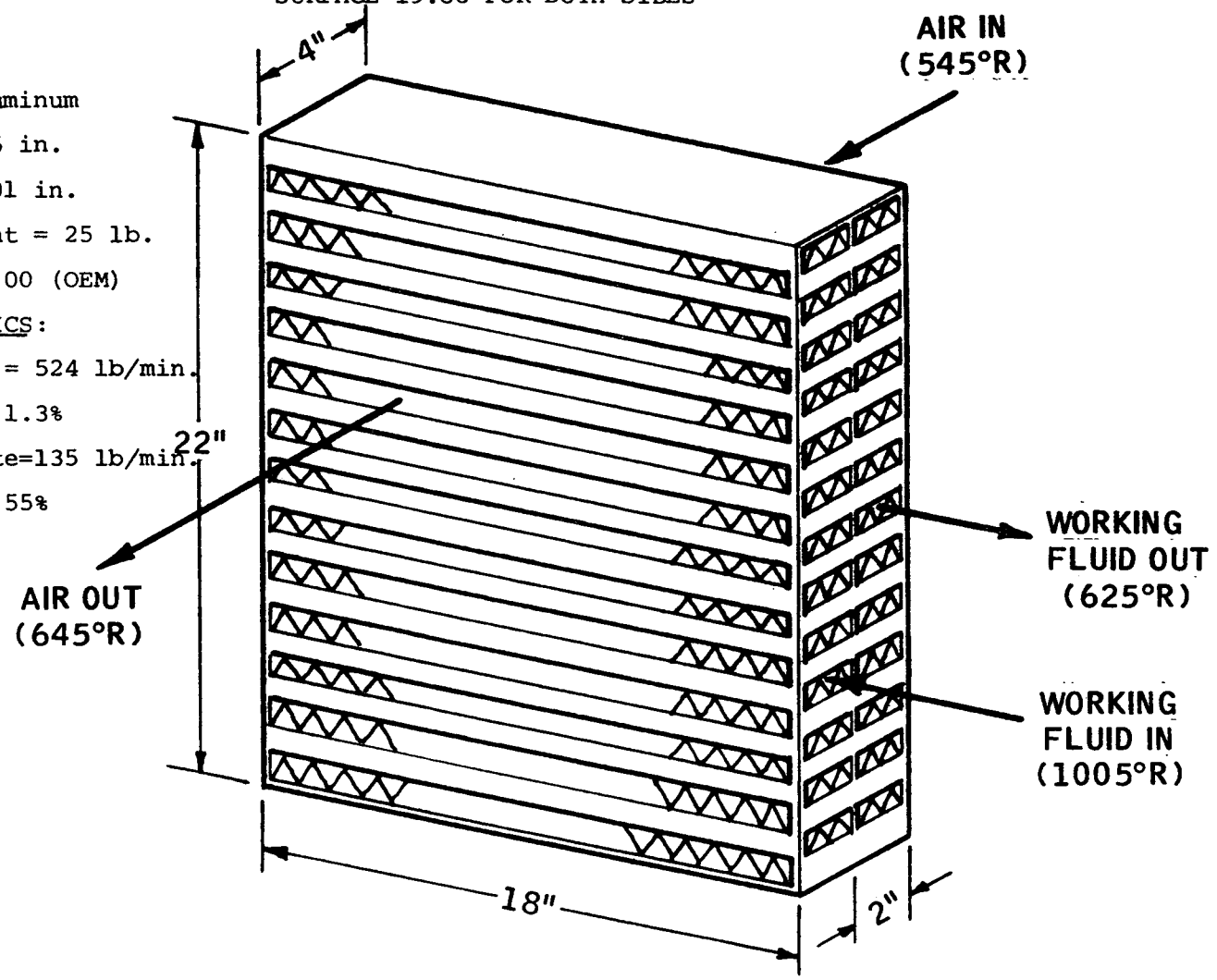


FIGURE 4.24

and the pressure drops decrease, thereby increasing cycle efficiency and reducing fan horsepower requirements. In fact, at idle conditions, the flow rate in both the open and closed loops is only about 5% of that at design and the resulting fan horsepower requirement is almost negligible. Also of importance, however, is the performance at 100% which requires about a 33% increase in both open and closed system flow rates from the design point (75% power). The 100% power level calculations have indicated that:

1. The CHX will suffer very little in effectiveness (about 2 points). This is due to the fact that the working fluid flowing through it is very highly turbulent and an increase in flow shows an almost corresponding increase in hot side h coefficient. The combustion products side is in a transition flow region and shows a similar increase in the cold side h coefficient (although not as large). These h increases tend to keep effectiveness constant as flow rate increases. Of course, the pressure drops on both sides will increase (about as the 1.8 power of flow rate), however, the design pressure drops were low enough so that this does not significantly effect engine performance.
2. The recuperator will suffer more of a drop in effectiveness (about 4 points) than the CHX due to the fact that the flow through both hot and cold sides is in the laminar flow regime (characteristic of most plate-fin heat exchangers) and an increase in flow does not change the h coefficient much. Therefore, an increase in flow rate decreases the NTU value and correspondingly decreases ϵ. This decrease in ϵ will require a higher fuel-to-air ratio in the burner to hold the same maximum combustion temperature, which tends to reduce cycle efficiency (increase BSFC). In addition, the pressure drop will go up almost linearly with the flow rate, thereby increasing the recuperator fan power requirements. Thus, at high cycle flow rates, the cycle efficiency and the power output are somewhat lower than predicted based on the present recuperator design. Full power will be available, however, by increasing either the maximum cycle pressure or temperature above design conditions, since full power will only be required for short intervals.

3. The WHX will suffer in a manner similar to the recuperator since it is also a plate-fin type design, although the flow through it on both sides is more in the transition flow regime than the laminar. It will experience a slight drop in effectiveness; however, it is felt that with proper manifolding design (finned manifolds) that the WHX will be able to provide the required cooling (closed cycle minimum temperature of 625°R) even at maximum flow. The pressure drops will go up as the mass flow rates on both sides go up and this will result in higher losses, especially on the cooling air side. However, with a properly designed fan and shroud arrangement, it is felt that these losses can be kept to an acceptable level.

   Ambient temperature effects were also investigated as they directly affect the WHX and indirectly affect the powerplant performance. An increase in ambient temperature of 20° (85 to 105°F) showed up as an increase in minimum cycle temperature of about 15°F (from 625°R to 640°R). This increase in minimum cycle temperature had only a slight effect on cycle efficiency and power output calculations and it is felt that any further definitive studies should be saved for actual engine dynamometer testing.

In summary, it is felt that the heat exchanger designs presented here should compliment the ECPE powerplant package during the initial prototype development stage. We are fully aware that they may not be the optimum designs; however, it is felt that further studies must be guided by actual hardware testing as the entire field of heat exchanger preliminary design is quite uncertain until proven in the actual operating environment. Such factors as cycle efficiency, power output, fan horsepower, fan noise, etc. should be considered in such a heat exchanger development stage to optimize the powerplant for the combination of on and off-design operating conditions.

4.5 AUTOMOTIVE ENGINE PERFORMANCE

The cycle analysis presented in Section 4.2 indicated that a 267 CID powerplant would be capable of producing the required 160 BHP at the design speed of 3600 rpm. The maximum closed cycle temperature and pressure were

defined to be 1760°R and 1500 psia at the design point. The total powerplant efficiency at the design point was found to be 21% with a resultant BSFC of about .65 lb/Bhp-hr. It now remains to determine the powerplant performance characteristics for the entire range of vehicle operation.

Evaluation of the engine power output at part speed and at part load required an estimation of the valve pressure losses and the frictional horsepower requirements at these part load conditions. Pressure losses for valves no. 2 and no. 3 as a function of engine speed are shown in Figure 4.25 and were determined by the same method as shown in Section 4.2 for the design point pressure loss calculations. Since the volumetric losses are assumed to occur across the inlet valve (no. 1), and since these losses do not necessarily decrease with decreased engine speed, it was decided to retain the design point pressure drop of 15% for valve no. 1 at all speeds. In addition, the pressure drop across the exhaust valve (no. 4) is included in the blowdown process and therefore, does not directly affect cycle calculations (as long as the valve is large enough to allow proper blowdown). Frictional horsepower data from a larger Ford engine was scaled down to be compatible with the requirements for the ECPE powerplant and is presented in Figure 4.26. As can be seen, the frictional horsepower increases with rpm, reaching a maximum of 28.2 BHP or 15% of the design indicated horsepower at 3600 rpm.

The brake horsepower, brake efficiency, torque, and brake mean effective pressure of the powerplant were developed from cycle calculations (with the above losses) as a function of engine speed at power levels of 100, 80, 50, 33, and 20%. Power level, as defined for the ECPE, refers to the percentage of maximum closed cycle base pressure (150 psia), which is not necessarily the percentage of maximum power as a result of various cycle losses (valve and friction). The brake horsepower as a function of engine speed, presented in Figure 4.27, shows the effect of the increased frictional losses at high speeds, especially at the low power levels where the brake horsepower may even decrease as engine speed is increased. This is due to the fact that the frictional horsepower requirements increase at a greater rate than the indicated horsepower.

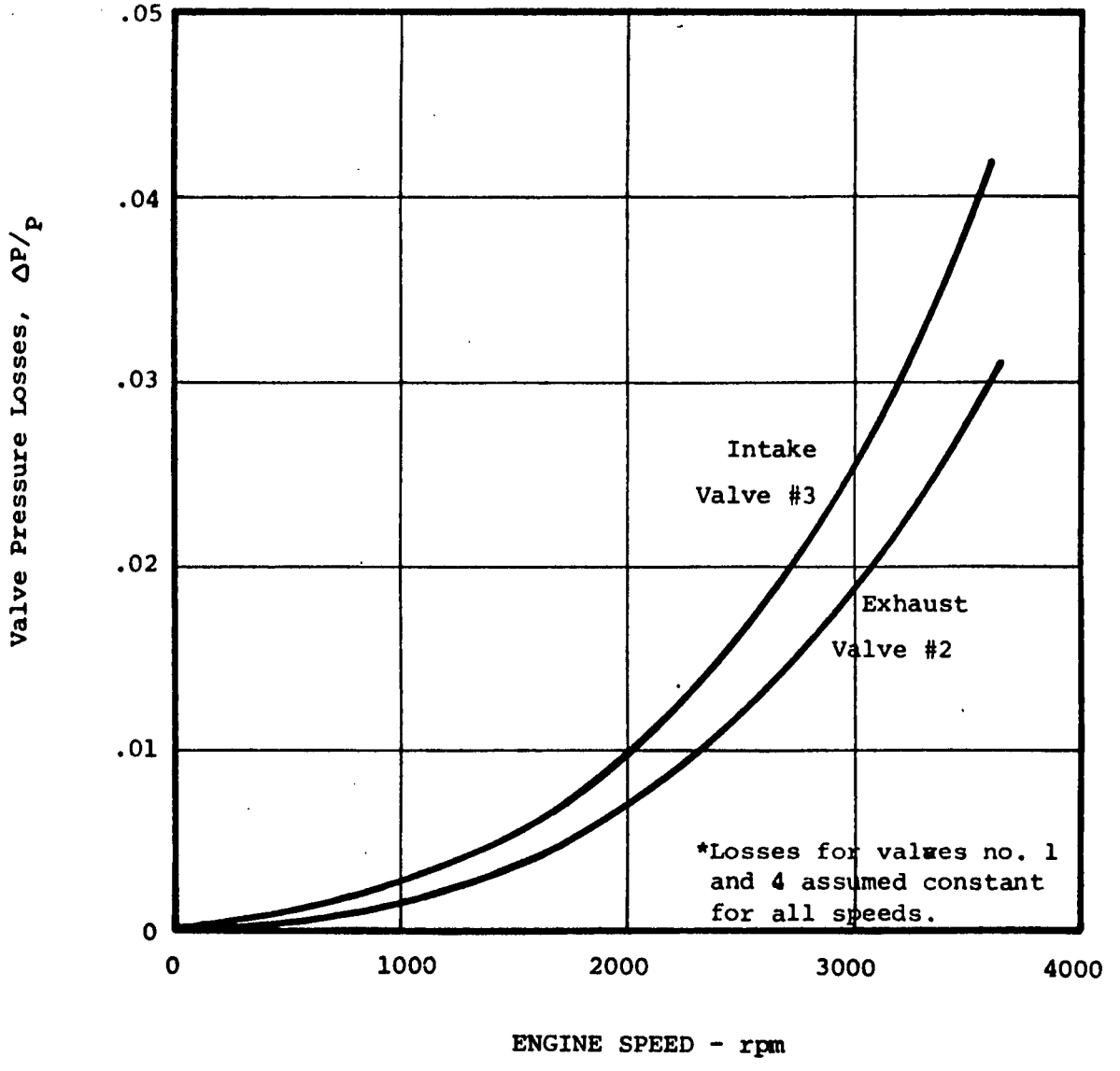


FIGURE 4.25 Valve Pressure Drops vs. Engine Speed.

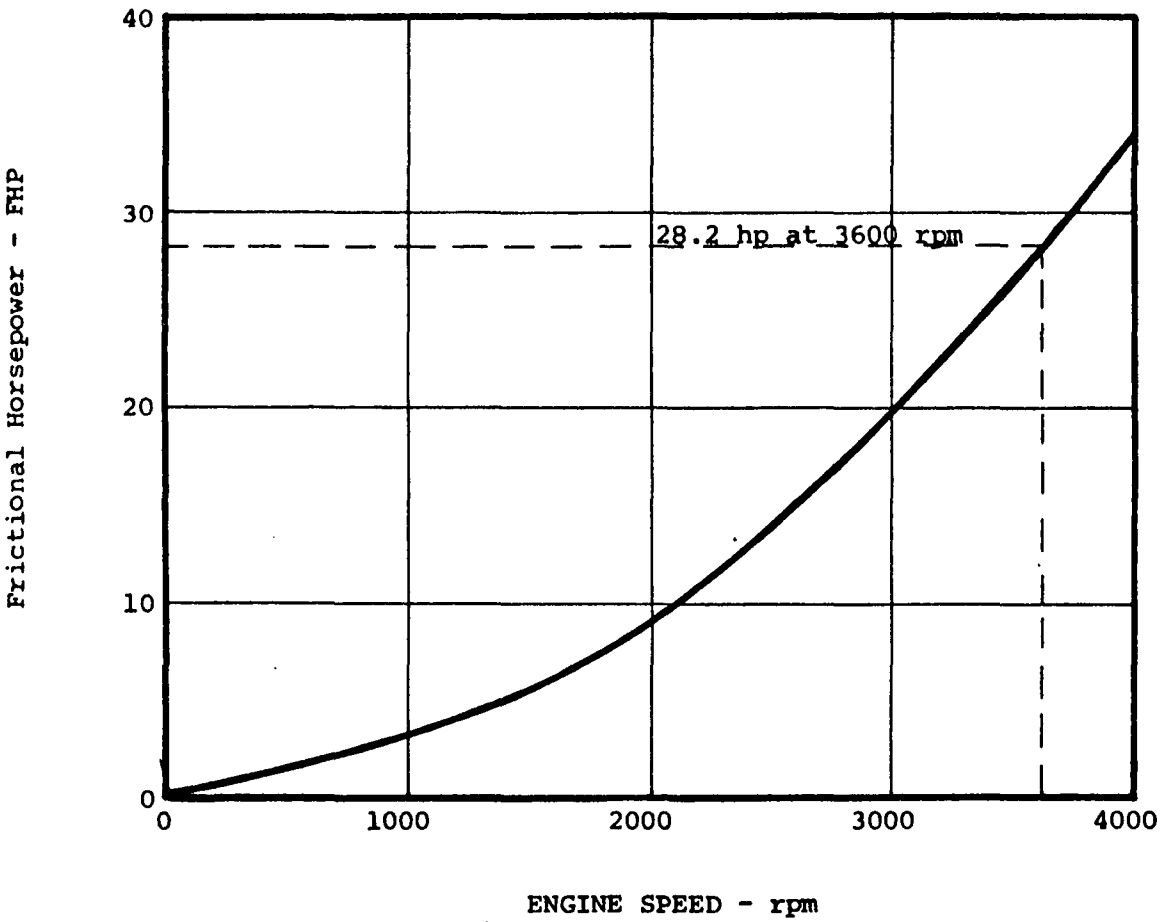


FIGURE 4.26 Frictional Horsepower Requirement vs. Engine Speed for the ECPE.

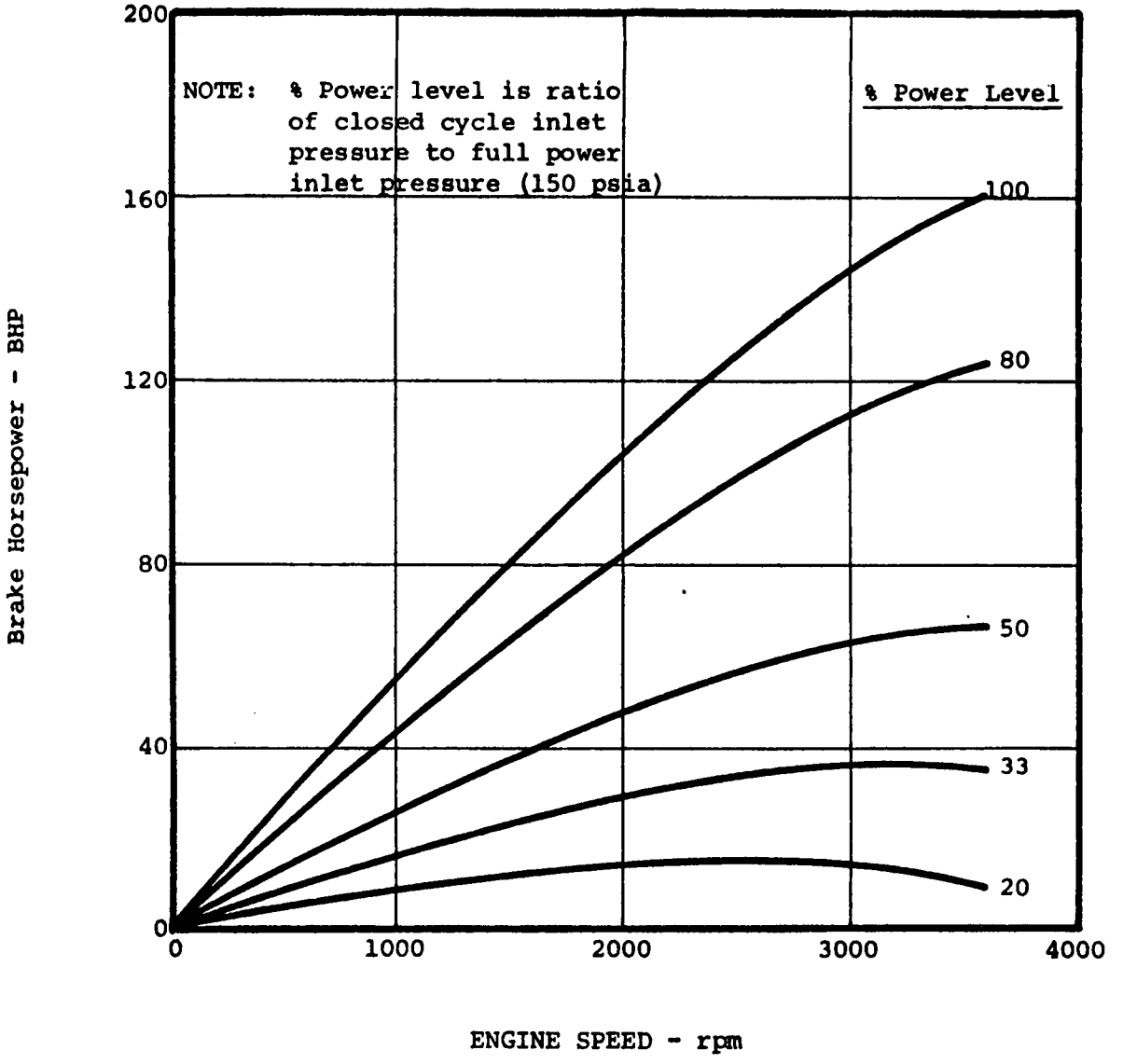


FIGURE 4.27 Brake Horsepower vs. Engine Speed for ECPE Powerplant.

The engine torque and BMEP as a function of engine speed at full throttle are shown in Figure 4.28 and illustrate the characteristic of the ECPE to develop maximum torque for acceleration at very low engine speeds. This fact makes it possible for the ECPE 267 CID engine to match the performance of larger, more powerful spark ignition engines in acceleration runs, and yet demonstrate the economy that a smaller displacement engine can provide both in initial cost and fuel savings. The published maximum torque @ rpm ratings for several standard 1970 domestic spark ignition engines are shown to illustrate the favorable torque characteristics of the ECPE powerplant. It can be seen that the ECPE provides more torque than comparable size six-cylinder engines, and provides almost the same maximum torque as the somewhat larger V-8 engines, although at much lower engine speeds (where it is most needed).

The ECPE total thermal efficiency and brake specific fuel consumption, presented as a function of engine speed for the various power levels, is shown in Figure 4.29 and 4.30. The reduction in efficiency at higher speeds is caused primarily by the increased frictional horsepower requirements at these speeds. As mentioned in Section 4.4, the effectiveness of the various heat exchangers will increase at lower than design flow rates (75% of maximum) and decrease at higher than design flow rates. The effectiveness (and pressure drop) values used in the analysis here were assumed constant and equal to the design point values for all engine speeds. This is somewhat of a simplification. However, due to the unknown degree of error present in other areas of the analysis caused by the rather vague information available on such things as valve losses, isentropic compression and expansion efficiencies, blowdown losses, etc., it was felt that using the design point values would be sufficiently accurate at this point in the study. In addition, since about 95% of all automotive vehicle operation is below 50% power, the efficiency and horsepower curves presented here are actually somewhat lower than realizable for the area of operation where we are most concerned. As shown in Figure 4.30, the ECPE, unlike most automotive powerplants (both conventional and unconventional), operates most efficiently at low speeds. This is due to the fact that the ECPE cycle losses do not increase at low speeds (such as the Otto cycle with its pumping losses); in fact, the losses decrease due to the lower valve and friction losses. The increased BSFC at lower power levels for the same engine speed

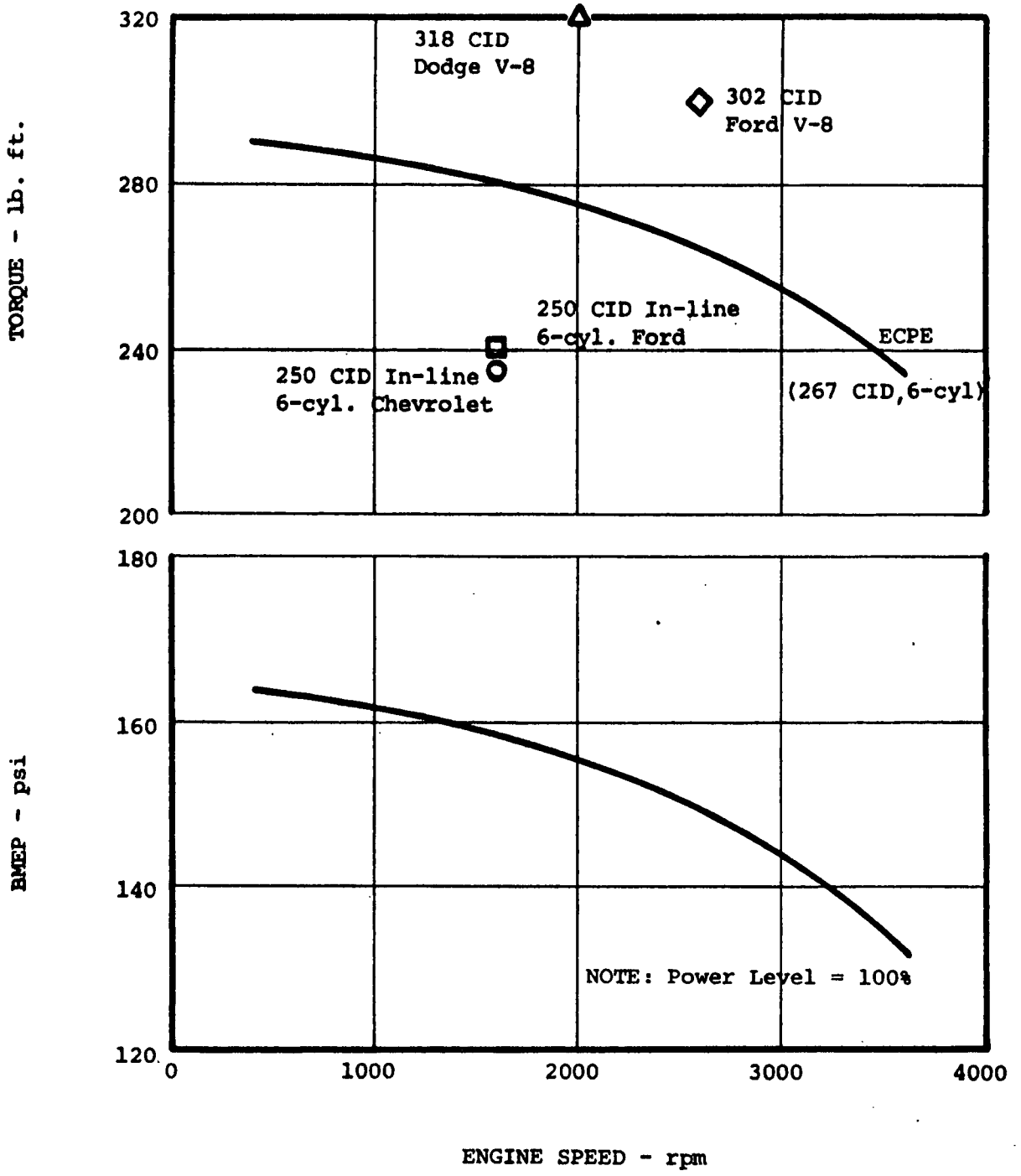


FIGURE 4.28 ECPE Full Power Torque and BMEP vs. Engine Speed (Compared to Conventional Spark Ignition Engines Published Maximum Torque @ RPM).

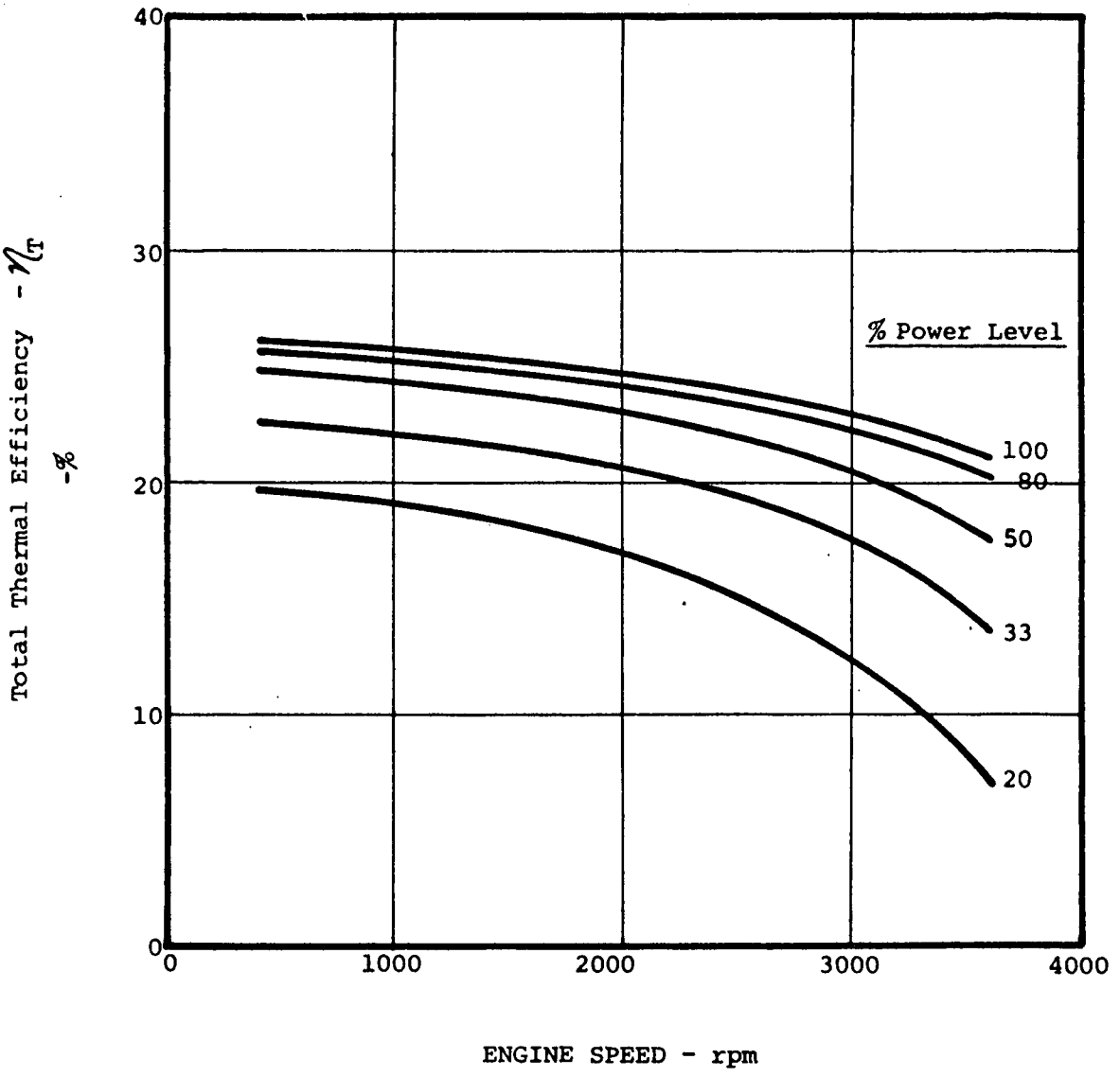


FIGURE 4.29 ECPE Total Thermal Efficiency vs. Engine Speed.

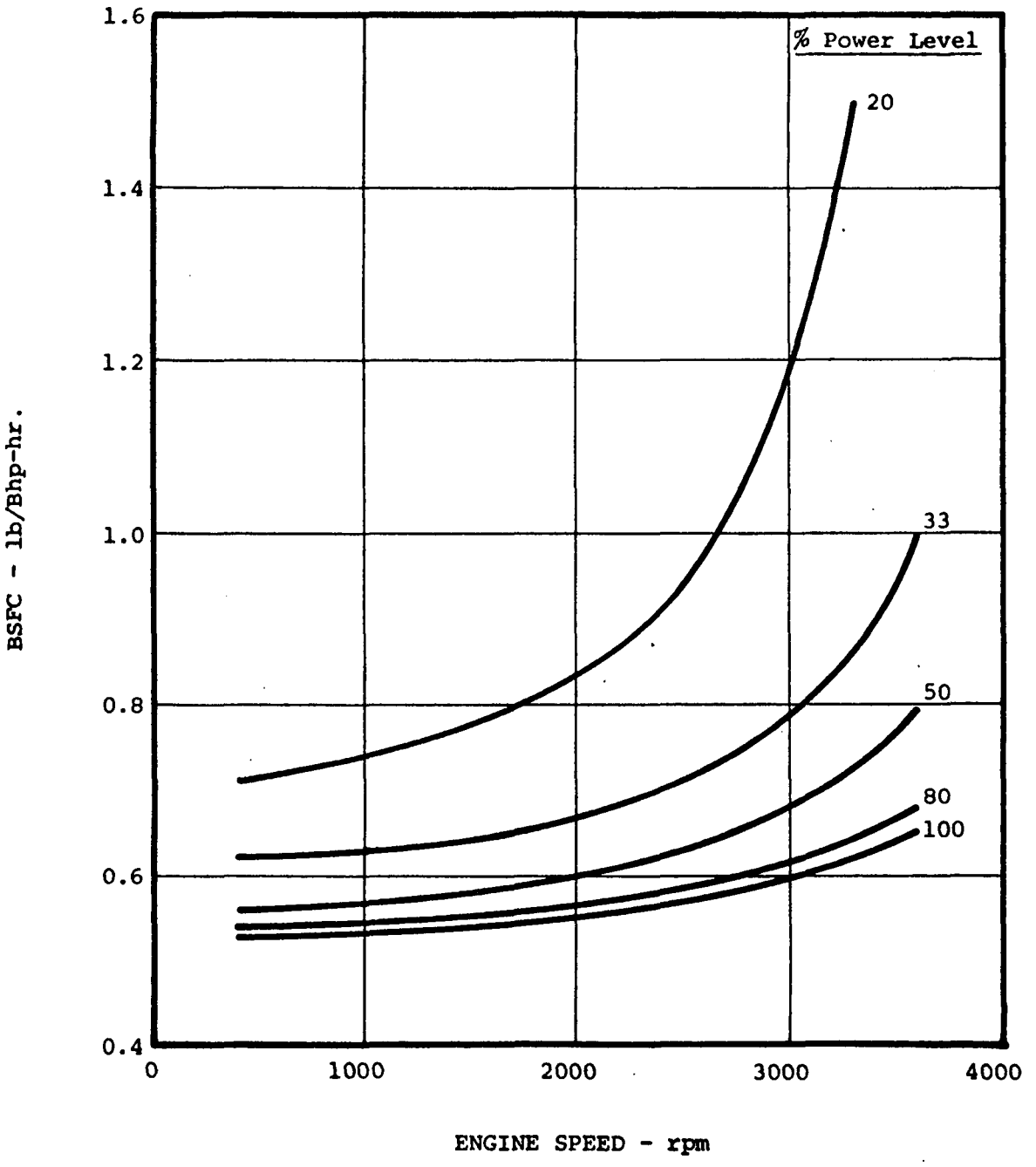


FIGURE 4.30 Brake Specific Fuel Consumption vs. Engine Speed for the ECPE Powerplant.

is due to the fact that the friction losses remain approximately the same as power level decreases (at constant speed), therefore, taking a greater percentage of the indicated power at the lower power levels. The individual BSFC curves level out at idle speed (600 rpm) due to the fact that the engine is considered to be putting out about 5 BHP at idle to handle the accessory load (see Appendix I).

Increased horsepower can be obtained from the ECPE powerplant (keeping maximum temperature the same) by several methods:

1. Overloading the present 267 CID engine by increasing cycle base pressure (up to 180 psia);
2. Increasing the engine displacement;
3. Increasing the maximum speed (3600 rpm).

The first of these, overloading, may be admissible only for short bursts of extra power, since the maximum cylinder pressure will reach 1800 psia (at a compression pressure ratio of 10), and this pressure is considered too high for sustained operation. In addition, the air storage tank must be enlarged to provide additional cycle air (an additional 32 BHP is available with a tank volume increase of 1.2 ft^3) for this overload condition. The second means of power gain, increased engine displacement, requires a simple scaling of all powerplant components, which, of course, adds to both cost and weight. Similarly, the third method, increasing maximum engine speed, would require enlarging many of the cycle components to handle the increased flow rate and would require a greater degree of design sophistication for the valves, cams, bearings, etc.

After having evaluated the engine performance as a function of engine speed, it now becomes pertinent to select the engine drive train components and evaluate the overall vehicle performance. A conventional 3-speed automatic transmission with gear ratios of 2.52, 1.52, and 1.00:1 and the corresponding rear axle gear ratio of 2.73:1 were chosen as being compatible with the ECPE powerplant. The shift characteristics of the transmission both as a function of power level and engine speed were combined with the previously shown curves of engine performance to give the vehicle performance as a function of vehicle speed.

Brake horsepower of the ECPE powerplant as a function of vehicle speed is shown in Figure 4.31. Typical shift patterns are shown which demonstrate

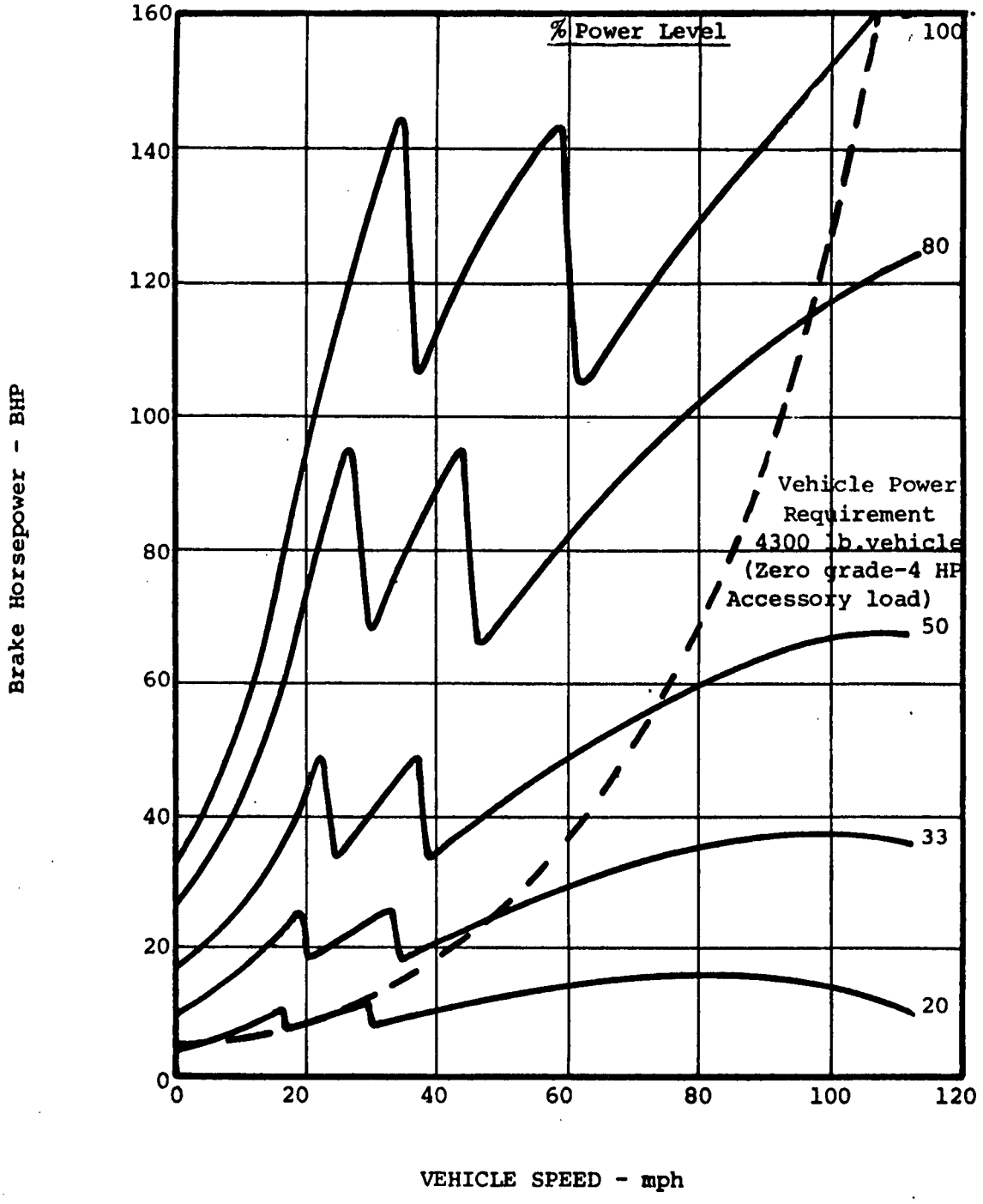


FIGURE 4.31 ECPE Brake Horsepower vs. Vehicle Speed.

the tendency of transmission shift points to be a function of both throttle position and engine speed. Power available for acceleration is represented by the difference between full throttle BHP and the vehicle power requirement, which has been superimposed on the figure. (The derivation of the vehicle power curve is presented in Appendix I.) The vehicle power curve shows that at a near zero vehicle speed the power requirement is about 5 HP. The 20% power level will conveniently handle this power at about 600 rpm, and therefore, this power level and engine speed have been defined as the engine idle condition. As shown in Appendix I, and again here, the maximum vehicle speed is approximately 107 mph since at that speed the vehicle power requirement (including all auxiliaries and accessory requirements) is equal to the maximum engine output.

The total efficiency of the ECPE powerplant as a function of vehicle speed is presented in Figure 4.32 where the non-accelerating vehicle operation curve has been determined from the vehicle power curve of Figure 4.31. Total efficiency for non-accelerating vehicle operation is shown to range from about 18 to 22%. This represents a significant advantage over gas turbine and conventional spark ignition engines, which are generally known to display poor efficiency at low vehicle speeds. Several sources have been identified, including the AiResearch Manufacturing Co.'s mid-term report to EPA, "Automobile Gas Turbine Optimum Configuration Selection Study", and they indicate that load range requirements of 0-15 BHP, corresponding to vehicle speeds of approximately 0-30 mph, comprise over 60% of most average vehicle driving time. This fact points out the importance of good efficiency at low load for good fuel economy and low emissions.

Next, the brake specific fuel consumption of the ECPE was evaluated as a function of vehicle speed and is shown in Figure 4.33. The same general conclusions made concerning thermal efficiency apply to these curves, and BSFC at non-accelerating vehicle operation is shown to range from .63 to .76 lb/Bhp-hr. As in Figure 4.30, these BSFC values are based on 5 BHP output at idle or zero vehicle speed to run the accessories.

Finally, the steady-speed fuel economy (in high gear) was calculated for the ECPE powered vehicle and the results are presented in Figure 4.34. The calculated fuel economy, which was developed from a generally conservative performance analysis, peaks at about 24 mpg at 20 mph, and there-

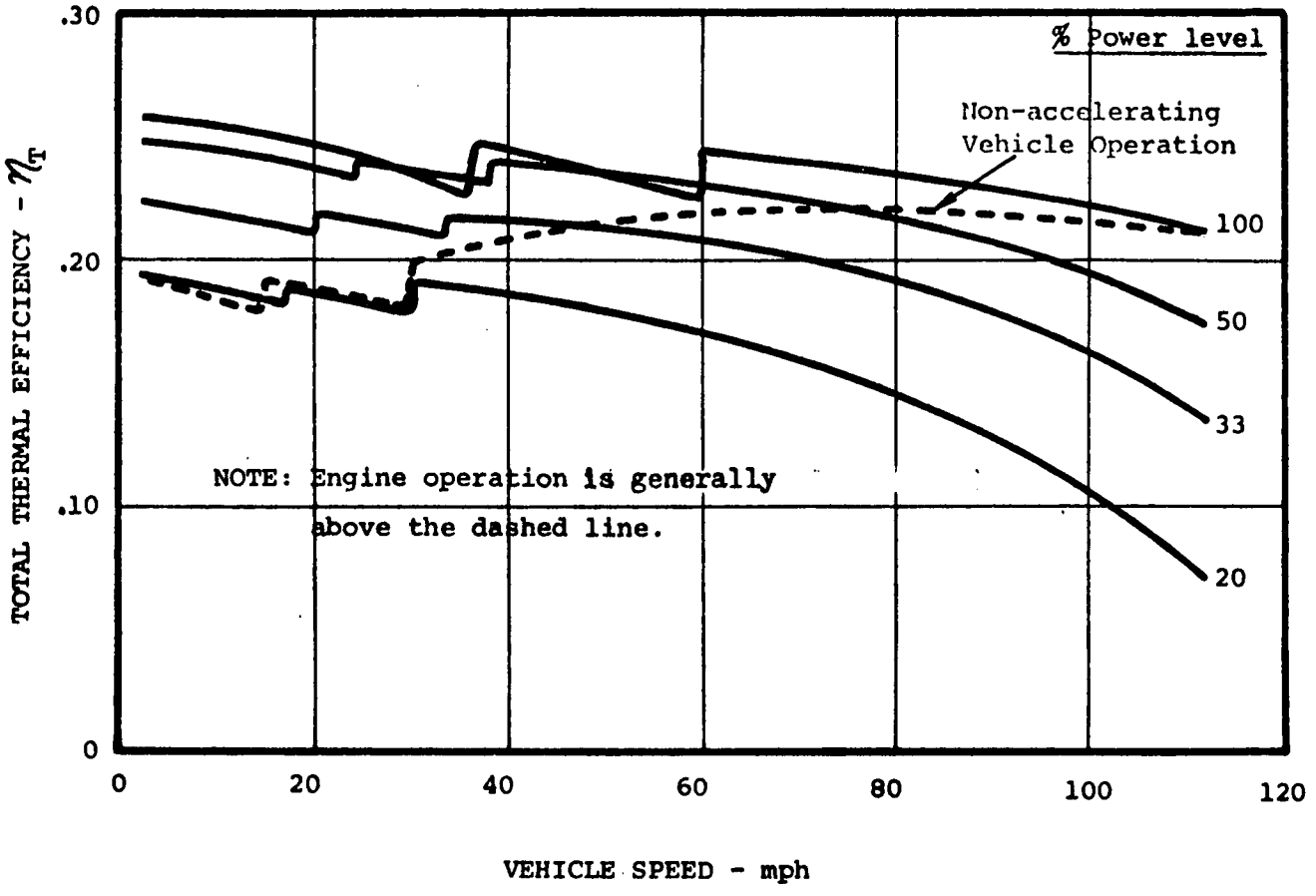


FIGURE 4,32 Total Thermal Efficiency vs. Vehicle Speed for ECPE Powerplant.

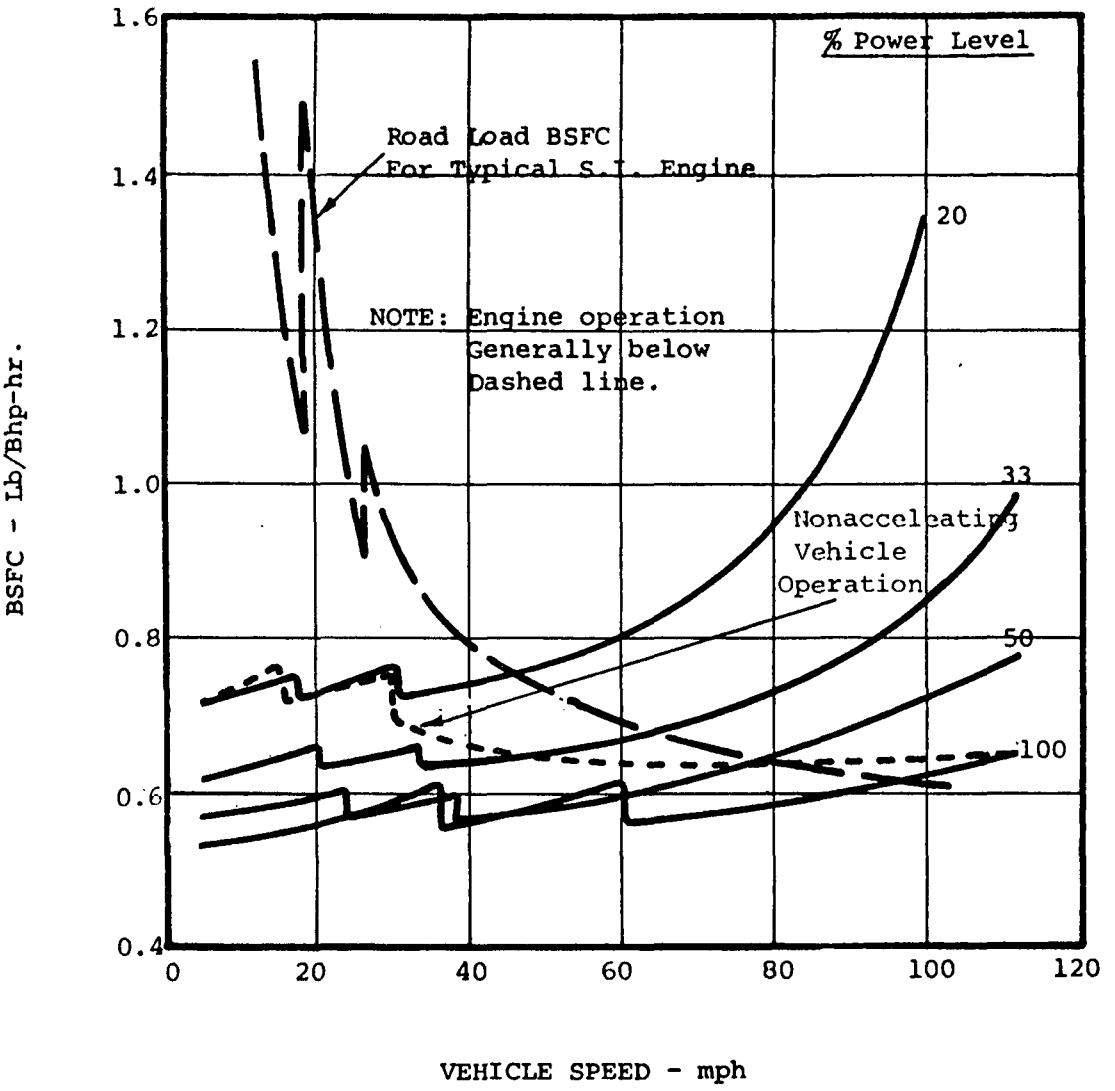


FIGURE 4.33 Brake Specific Fuel Consumption vs. Vehicle Speed.

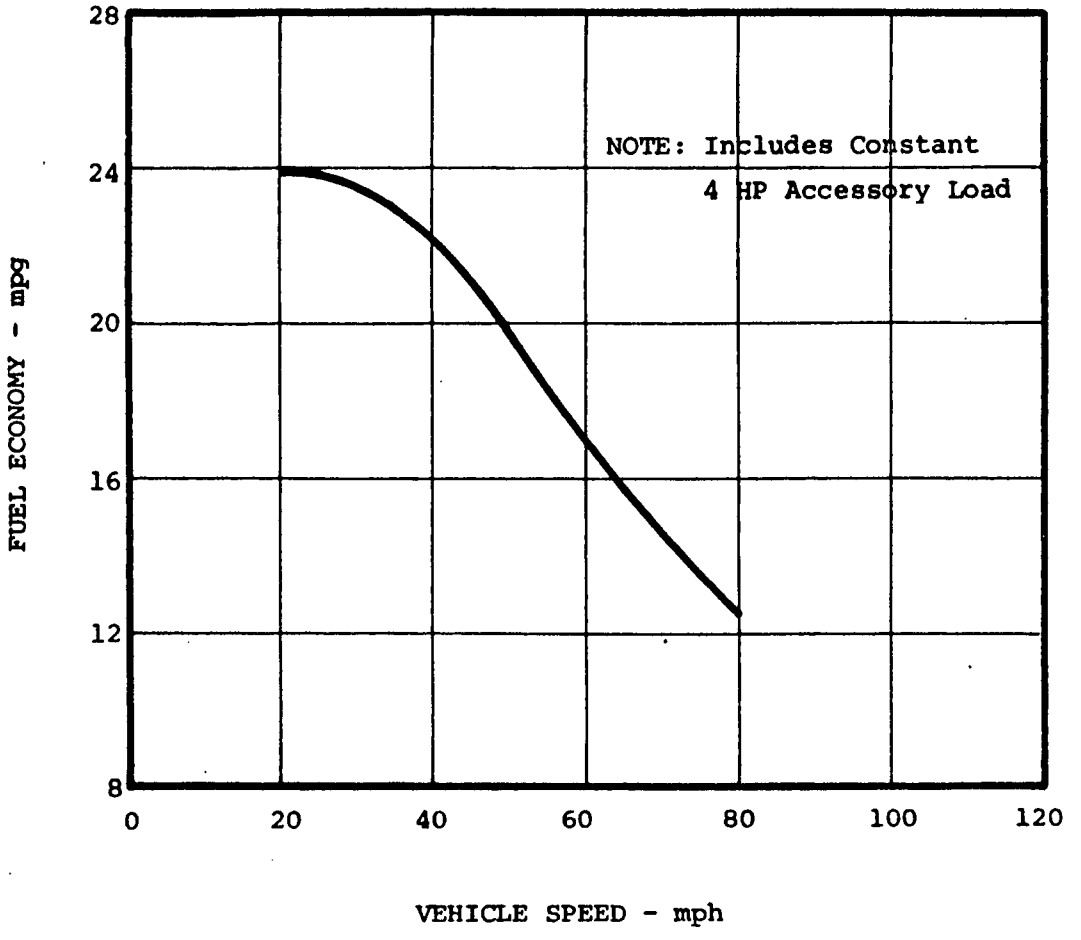


FIGURE 4.34 Steady Speed Road Load Fuel Economy of ECPE Powerplant.

after decreases to 12.5 mpg at 80 mph. Fuel economy of this level is considered quite good, and will be shown comparable or superior to other types of automobile powerplants in the powerplant comparison section. It is also noteworthy that the ECPE, like other external combustion cycles, requires less highly refined fuels such as kerosene, which promises to be an additional benefit in vehicle operational cost.

It should be noted here that the vehicle operation curves in the previous figures take into account all parasitic, auxiliary, and accessory losses in addition to the rolling and aerodynamic power requirements (see Appendix I). In short, what has been done is that all power losses other than engine friction losses have been lumped together in the vehicle power curve, so that the BHP, as calculated by the cycle analysis of Section 4.2, represents power available to satisfy all power draws (auxiliaries, accessories, wind resistance, etc.). This appeared to be the best method of presenting the results since the ECPE engine can now be evaluated in combination with any vehicle by retaining all the engine performance curves and merely using the vehicle power curve which represents the particular vehicle. For instance, additional auxiliaries and accessories can be added to the ECPE engine and the only effect on the results presented here will be to shift the position of one curve, the vehicle power curve. In addition, the vehicle power curve used here may be slightly lower than that required at the high speed end since average values of transmission, fan, accessory, etc. losses were used. Conversely, the curve may be slightly higher than that required at the low speed end. This will tend to improve performance characteristics for the low speed region and degrade them for the high speed region; however, as mentioned previously, the low speed region is of highest concern since the vehicle will spend most of its operating life in this region.

Finally, the ECPE vehicle performance with the three speed automatic transmission and associated rear end was checked against the OAP vehicle design goals as presented in Appendix I and found to satisfy all requirements. This was done by actually determining vehicle acceleration as a function of vehicle speed for full power operation. The proper vehicle shift points for full pedal operation were adhered to and the 4300 lb vehicle was found to

have the necessary acceleration to meet all OAP goals. This was done without taking into account any torque multiplication which would result from the standard torque converter.

4.6 AUTOMOTIVE DESIGN AND INSTALLATION

Early in the study it was decided that a six-cylinder in-line engine arrangement would be best suited for the ECPE powerplant due to the fact that the heat exchangers and associated manifolding could be installed alongside the block in the available under-the-hood space of a medium sized vehicle. A search of the technical literature revealed that the 250 CID IL-6 OHV Chevrolet engine would provide a convenient base powerplant for the ECPE concept. The required 267 CID could easily be obtained by increasing the bore from the stock 3.87 inches to 4.00 inches and leaving the stroke (3.53 inches) the same. In addition, preliminary calculations have shown that there is ample bore area in the head to accomodate the four valves (see valve analysis Section 4.2) and therefore engine design changes would be basically limited to: the cylinder head, valves, valve train, manifolding, and fan and accessory pulleys. As will be show, these changes do not produce any significant changes in engine volume.

Generally, the effort was made to keep changes to the basic stock engine to the very minimum to reduce prototype costs while not infringing on the optimum cycle performance. Figure 4.35 shows the front view section of the proposed ECPE powerplant. Due to the number of cam operated valves (24) and also because of the necessary porting arrangement, the double overhead cam design was chosen. The stock camshaft has been left on this figure due to the fact that it may be required to operate various powerplant auxiliaries (fuel pump, oil pump, etc.). The proposed arrangement of the cylinder head, roughly indicating valve and port locations, is indicated in Figure 4.36. As shown, the valves are so arranged that the same function (i.e., exhaust, intake, high pressure release, and high pressure return) parts of two adjacent cylinders may be combined into a single manifolding port. The two sets of high pressure valve ports are both located on the left side of the engine, in order to make them accessible to the combustion heat exchanger manifolding, while the intake and exhaust valve ports are located on the right. This valve arrangement and

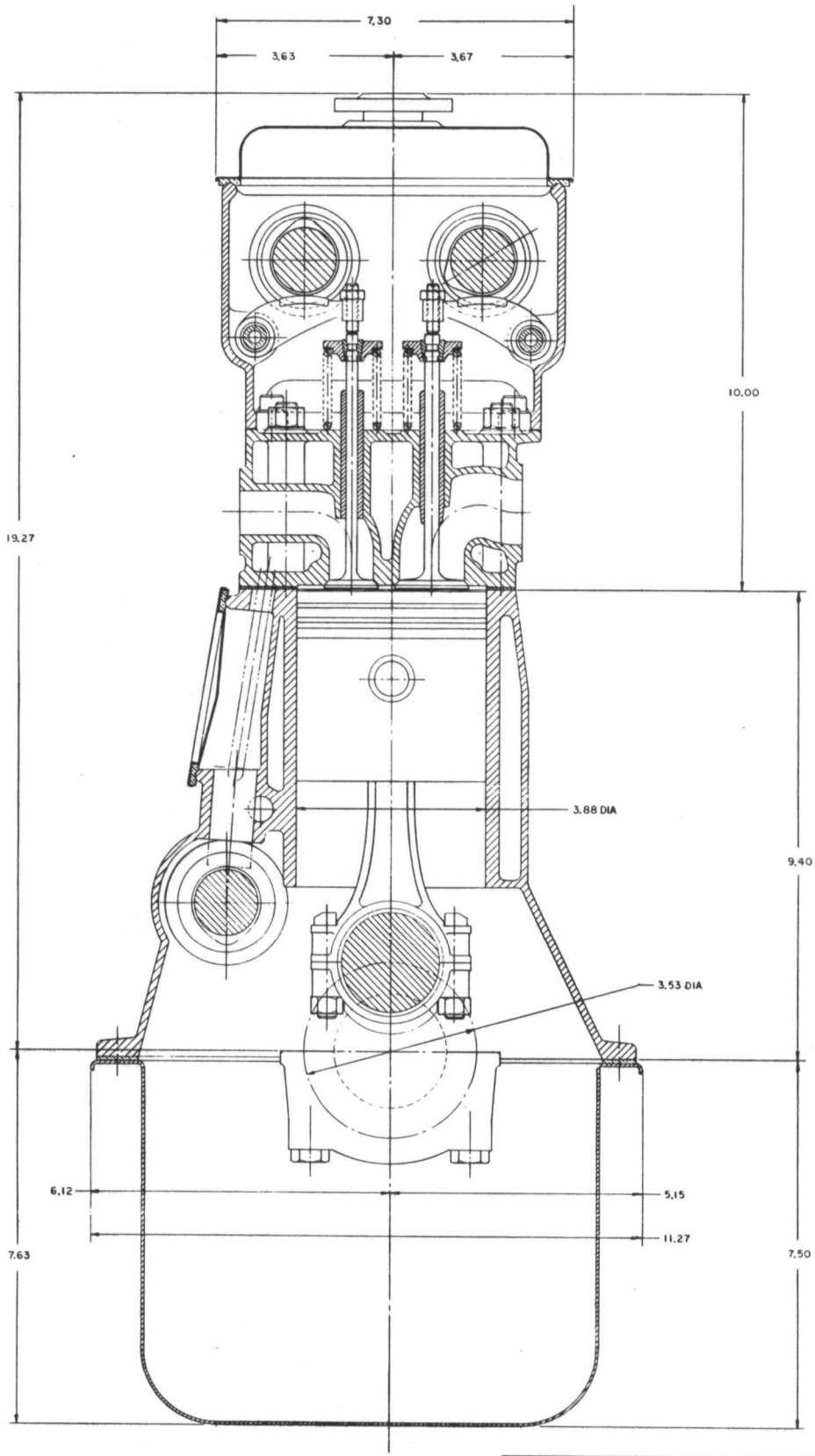


FIGURE 4.35

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DATE: 6-16-71		REVISED:
STUDY-ECPE ENGINE SECTION, FRONT VIEW		
CHEVROLET 250 CU.IN.-6 CYL. IN LINE		DRAWING NO. LO-31R

VIEW ON THE TOP

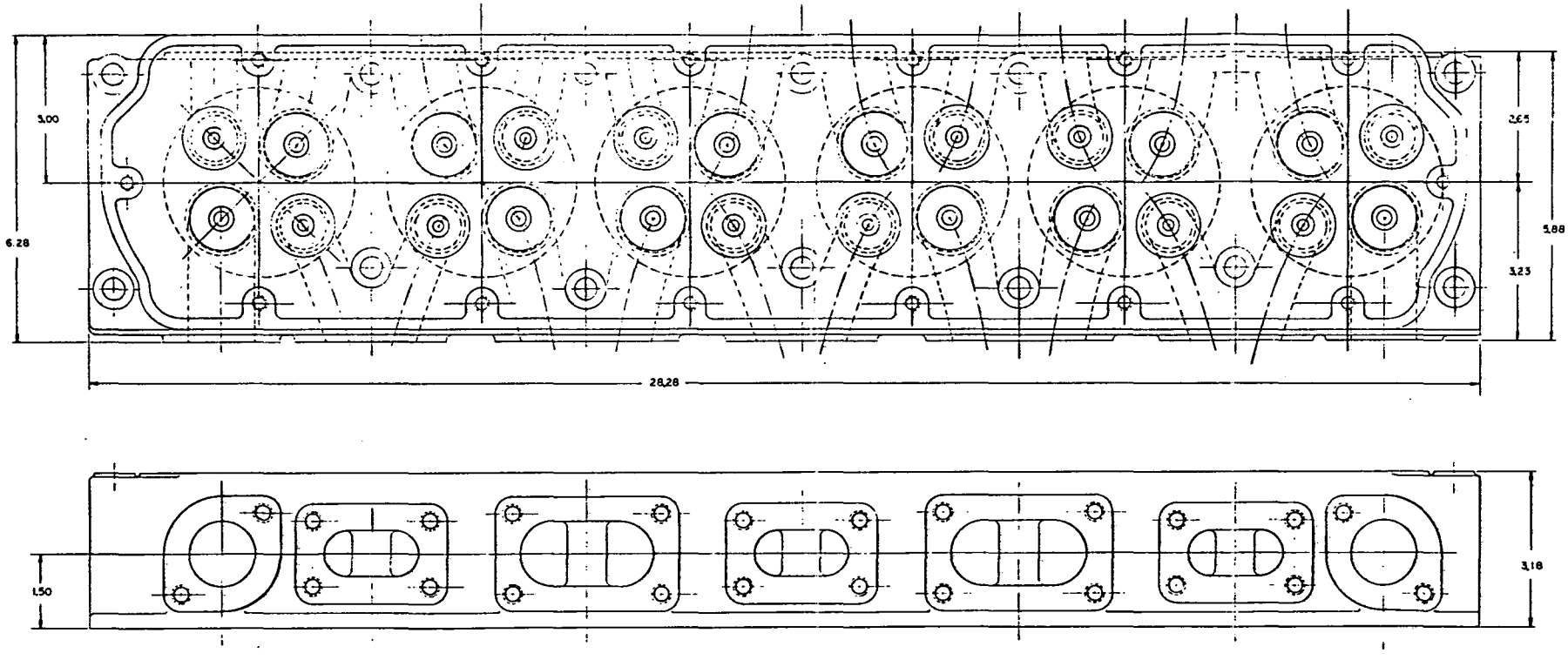

FIGURE 4.36

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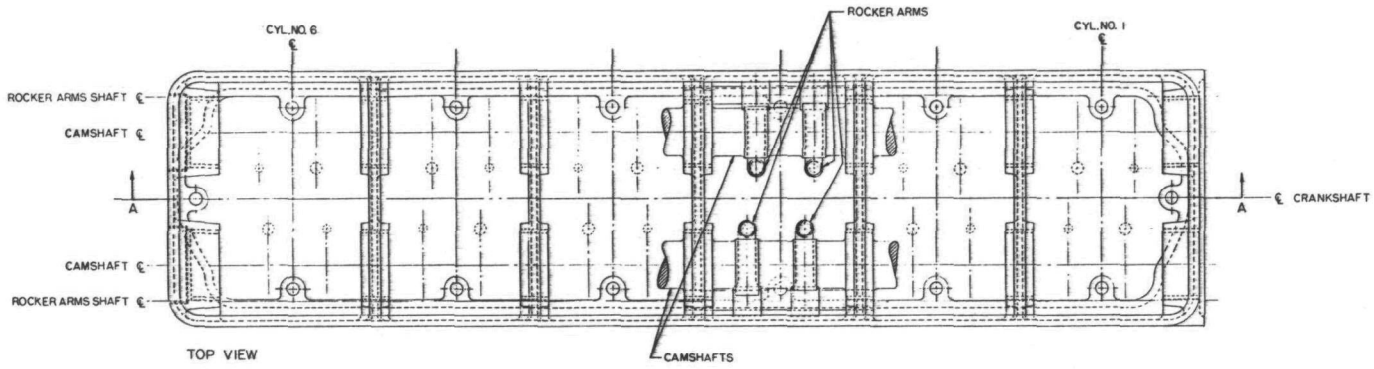
STUDY – ECPE ENGINE, CYLINDER HEAD		
CHEVROLET 250 CU.IN. – 6 CYL. IN LINE		LO-32R

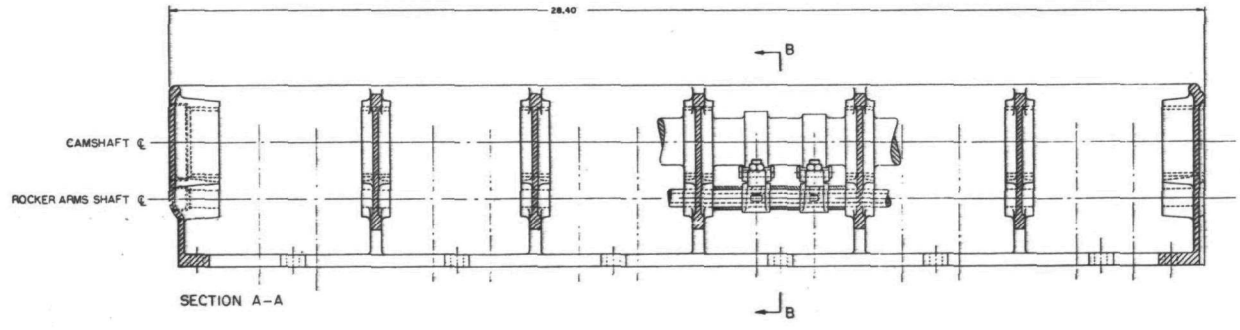
the tolerance between valves, and valves and cylinder wall, are shown in more detail in the valve analysis section (4.2). The head has been designed to utilize the standard mounting bolts and tentatively to use the same water cooling passages between the head and block.

The rocker arm and cam housing design, shown in Figure 4.37, indicates the general approach to bearing support for the cams and valve train. Cam lobes have been indicated but are only of a preliminary conceptual nature. The finalized details of optimum cam shape, shaft and rocker stress, and bearing loads are critical but were not calculated at this point in the ECPE development program. A very preliminary analysis into such area as valve seals and piston ring designs was done and it is felt that the technology is available, when the time comes for prototype design, to justify the preliminary configurations presented here. The importance of acceptable mechanical designs to the feasibility of the ECPE powerplant is realized, however, and detailed designs will be of first concern in any follow-on program.

Next, the belt drive system for the cams and various auxiliaries and accessories was determined and is shown in Figure 4.38. This belt drive system has a separate belt for the two cam pulleys and for each of the fan and accessory pulleys which are to the right and left sides of the engine. The cam drive will use a cog-toothed belt similar to that employed on the Ford Pinto four-cylinder engine and the accessories and fans will be driven by conventional V-belts and pulleys. As indicated in the figure, the cam drive incorporates a driving cog-toothed pulley shaft driven by the original cam drive gear, an idler pulley mounted on the front of the new cylinder head, and driven cog-toothed pulleys at the front of each of the camshafts. The only change to the basic engine block involves replacement of the cam drive gear access plate with a mounting plate for the cam cog-toothed pulley drive. All of the drive system are typical of current automotive production techniques.

After having completed the basic engine design phase, attention was directed towards designing the heat exchangers to fit comfortably around the engine in the available under-the-hood space of a medium size automobile. Actual under-the-hood dimensions were used, taking into account such restrictions as wheel wells, steering column, master cylinder, etc. The





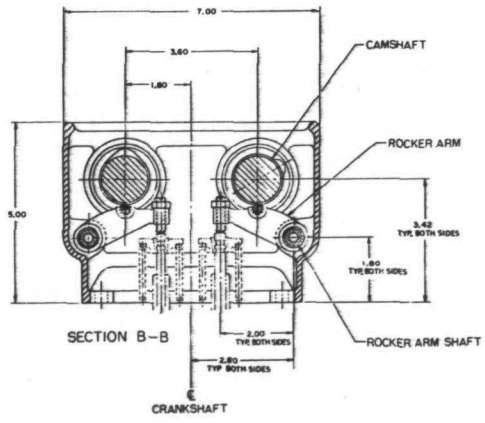


FIGURE 4.32

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STUDY-ECPE ENGINE, ROCKER ARMS BOX	
CHEVROLET 250 CU. IN.-6 CYL IN LINE	DRAWING NO. LO-38R

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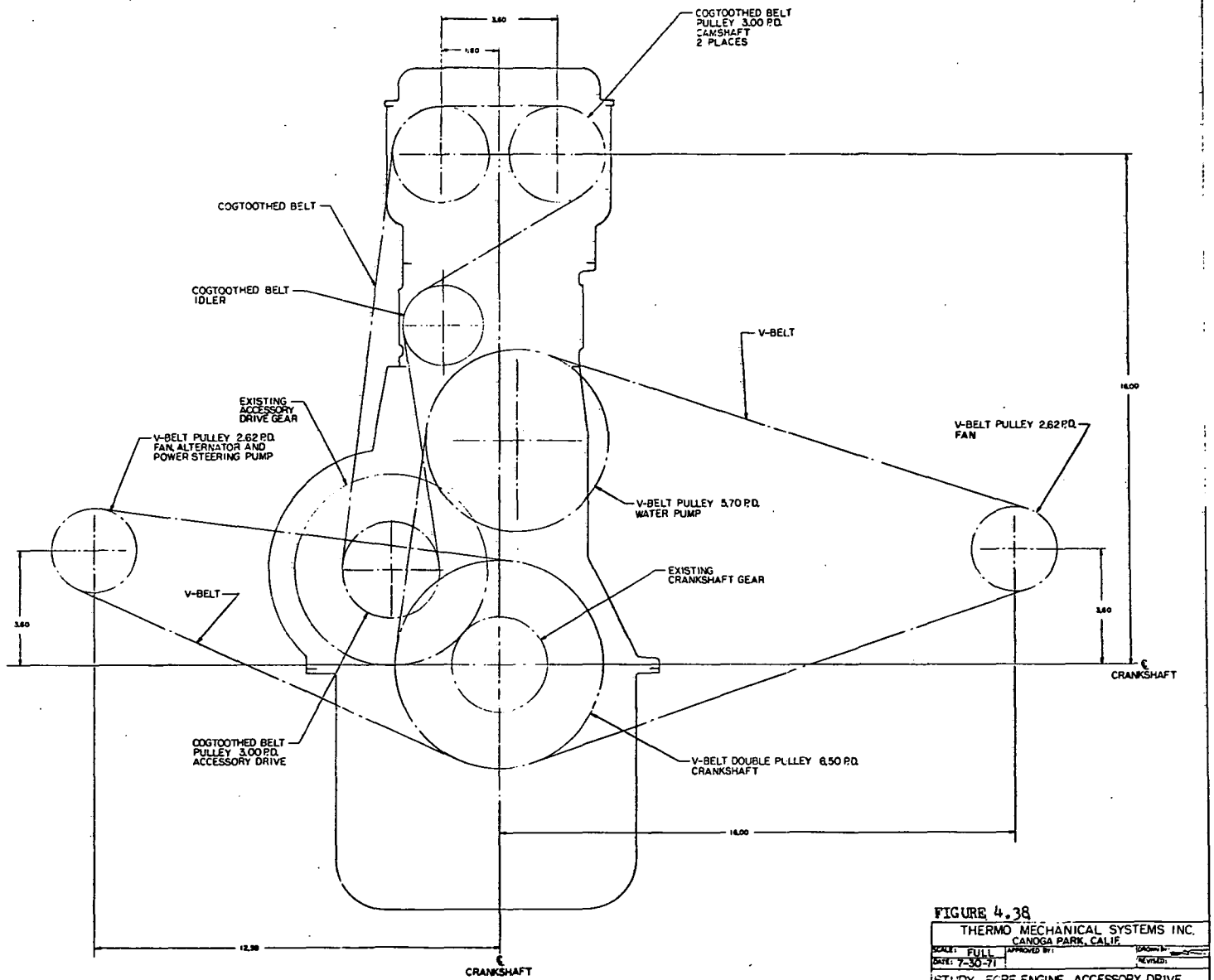

FIGURE 4.38

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DATE: 7-30-71	REVISED:
STUDY-ECPE ENGINE, ACCESSORY DRIVE	
CHEVROLET 250 CU.IN-6 CYL.IN LINE	LO-39R

heat exchanger designs are presented in Section 4.4 and the recuperator-burner-CHX combination is shown in Figure 4.39. As will be shown in Figure 4.40, the heat exchanger arrangement was such as to make maximum use of the available space. Further heat exchanger optimization studies will be a part of future ECPE development work; however, it is felt that such studies will not change the general arrangement as shown in Figure 4.39.

Finally, the complete three-dimensional representation of the ECPE powerplant installed in a medium size automobile is shown in Figure 4.40. This representation includes all the required accessories, among which are: the complete air conditioning system, the make-up air compressor, the alternator, and the power steering pump. It can be noted that the system make-up air compressor has been mounted on a common shaft with the alternator and that the reserve air cylinder has been located at the back, right hand side of the engine compartment. In addition, the power controller and associated valves, transfer lines, linkage, etc. referred to in Section 4.3 should present no design or packaging problems and can be installed where most convenient on the right or left side of the engine. To facilitate manifolding, the combustion heat exchanger and the combustor are mounted along the left side of the engine opposite the cylinder head, with the recuperator located directly beneath. The close packaging arrangement plus a certain amount of insulation should keep convective heat losses from the air heater, recuperator, and manifold to a minimum. The waste heat exchanger and radiator are indicated to be located side by side in front of the engine with a separate fan for each; however, it is felt that perhaps one fan is sufficient to provide the required air flow for both. In fact, analysis of the engine cooling system has indicated that the ECPE radiator should be much smaller than that required for the stock 250 CID engine due to the fact that the ECPE average operating temperature is well below that of the typical internal combustion engine. Future development work will include a detailed cycle heat loss analysis and will investigate the possibility of removing the radiator altogether through proper design of the head to air cool and properly isolate the high temperature no. 3 valve and valve seat.

A preliminary weight estimation for the ECPE powerplant and a comparison with that of the conventional 250 CID powerplant is given in Table 4-II.

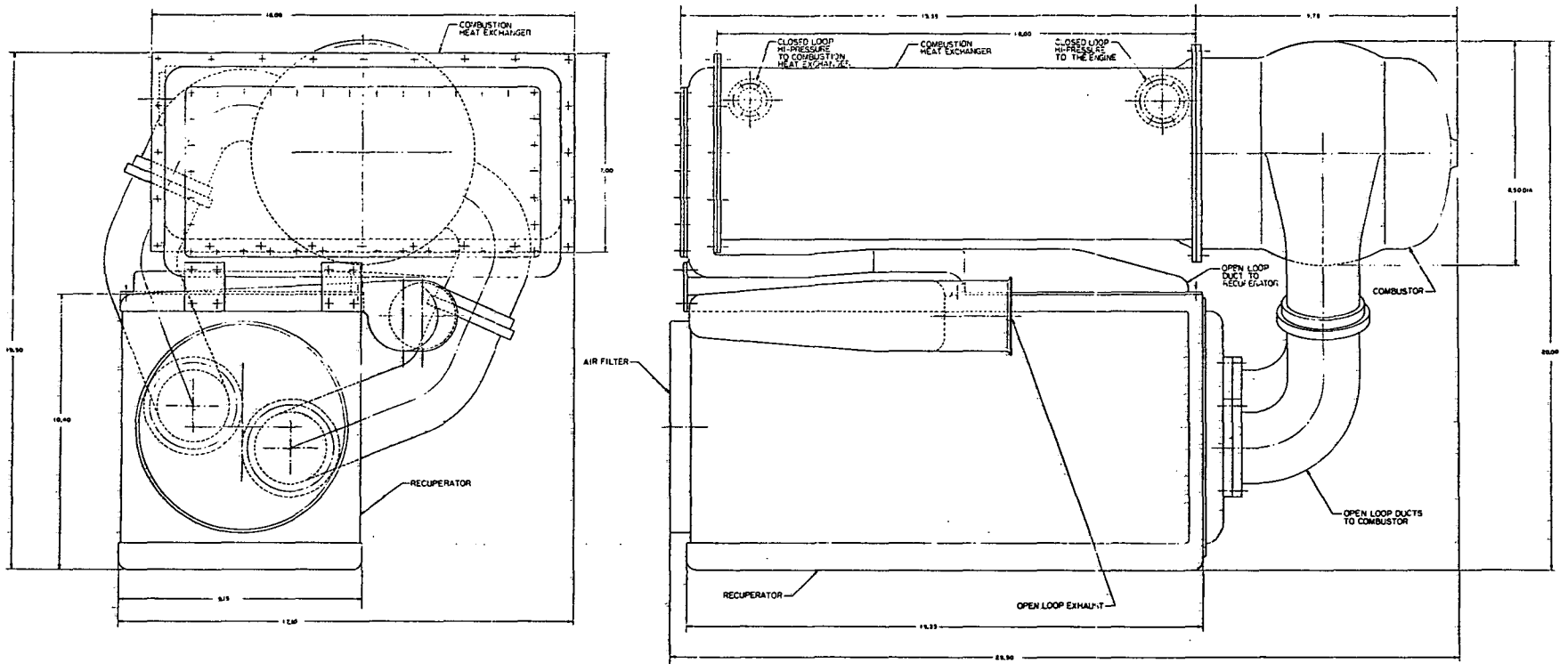

FIGURE 4.39

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STUDY-SCALE ENGINE-COMBUSTION HEAT EXCHANGER, RECUPERATOR AND COMBUSTOR	
CHEVROLET 250 CU IN.-6 CYL. IN LINE	LO-40R

1250

EXTERNAL COMBUST I N PISTC N EN G N E

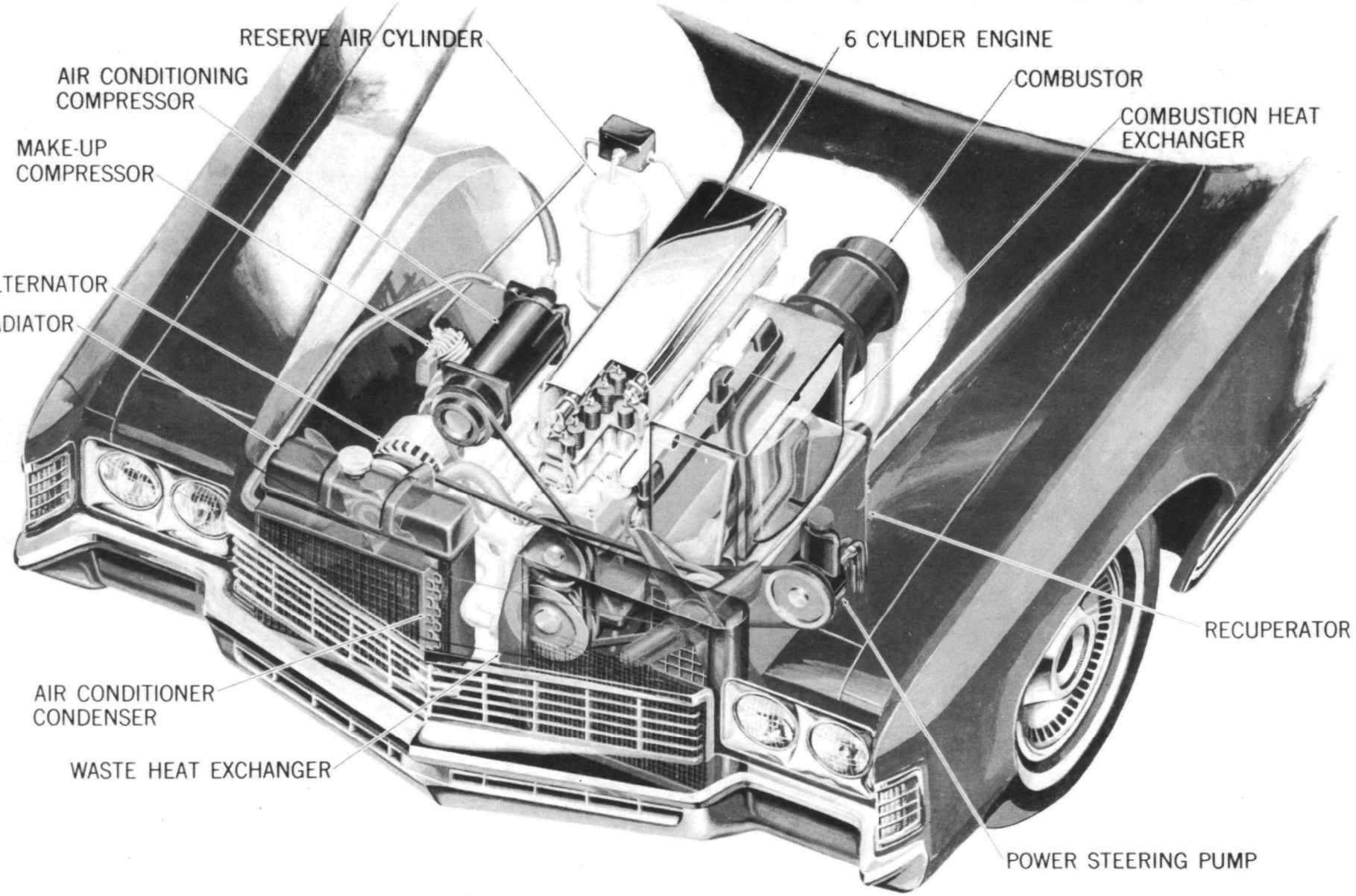

FIGURE 4.40

TABLE 4-II

PRELIMINARY POWERPLANT WEIGHT ESTIMATION

	ECPE 267 CID 6-cyl engine	Conventional S.I. 250 CID 6-cyl engine
Engine (including manifolds)	600	400
Radiator	25	35
Starter	20	20
Generator	20	20
Combustor (plus ducting)	20	
Combustion Heat Exchanger (plus ducting)	35	
Recuperator (plus ducting)	30	
Waste Heat Exchanger (plus ducting)	40	
Air Compressor (plus pulleys, belts)	20	
Air Storage Tank (plus lines and controls)	25	
Total	835 lbs	475 lbs
Transmission	170	170
Rear Axle and Drive Train	170	170
Battery	40	40
Fuel and Tank (200 mile range)	145	145
Exhaust	40	40
Miscellaneous	50	50
Total	615 lbs	615 lbs
Total Propulsion System Weight	1450	1090
Vehicle without Propulsion System Weight	2700	2700
Vehicle Curb Weight	4150	3790

The ECPE propulsion system is estimated to weigh a total of about 1450 lbs, or about 350 lbs more than the conventional six-cylinder propulsion system from which it was developed. Since a somewhat heavier expander than that of the 250 CID engine will be required to withstand the full 1500 psi pressure, the ECPE base engine weight (including manifolds) has been increased over that of the standard engine. It must be remembered, however, that the predicted ECPE performance (horsepower, torque) is superior to most conventional six-cylinder engines and therefore, when compared to a V-8 of similar performance, the ECPE total powerplant weight should be more comparable.

Unlike the gas turbine powerplant, the total amount of critical materials used in the ECPE (nickel, chromium, and cobalt) is not considered to be a major problem. In the ECPE essentially all of the critical metal usage is in the open combustion loop components, including the combustor, combustion heat exchanger, recuperator (or pre-heater), and their associated hot manifolding and ducting. The estimated total weight of these metals is for nickel, about 4.5 lbs, for chromium, about 8 lbs, and for cobalt, about .02 lbs. These estimations are, unfortunately, very rough at the present time and may vary considerably depending on further optimization studies and development work on the heat exchangers and burner. The critical metal content of the burner, for instance, was determined using available information from the OAP contractors engaged in burner development work (Solar, Ref. 60). Since burner development work is still progressing, it becomes a very difficult task to approximate the content of critical materials that will be required. The primary purpose for estimating the critical material content of the ECPE powerplant was to show that critical material availability is not a major problem.

4.7 GENERAL CHARACTERISTICS

The previous sections have reviewed the principle and design of the ECPE concept and have shown that it holds good promise as an automotive powerplant. The performance and fuel economy characteristics were shown to be quite good in Section 4.5, and the design of the various system components in Section 4.6 was shown to be well within the reach of present technology. This section will attempt to identify some of the major social

and economic considerations applicable to the ECPE which will necessarily play a major role in its acceptance or non-acceptance as an automotive vehicular powerplant. These major social and economic considerations are:

1. cost
2. hazards
3. emissions
4. development time, costs, and risks.

4.7.1 Cost

The first of these considerations, cost, can only have meaning when evaluated in comparison to the present internal combustion engine. For the ECPE powerplant, the primary cost difference as compared to the I.C. engine is in the heat exchangers and the external burner. Preliminary cost estimates made for these components (see Heat Exchanger Analysis, Section 4.4) and also for the additional engine equipment (valves, cams, belts, fans, air compressor, storage tank, etc.) place the initial cost at about 1.5 to 2.0 times the cost of a conventional I.C. engine of comparable performance. This value is estimated to represent the cost as compared to an I.C. engine without emission controls. With controls, to provide comparable emissions, the ECPE cost may be favorable to the I.C. engine. These cost estimates account for the additional equipment required by the ECPE and not required by the I.C. engine and, in turn, also account for the equipment needed for the I.C. engine and not needed by the ECPE.

In addition to initial cost, the maintenance and operating cost of the ECPE should be comparable to the I.C. engine, especially if emission control equipment is taken into account. As mentioned, the troublesome electrical system of the I.C. engine is essentially eliminated with the ECPE (1 glow plug required), therefore eliminating the frequent, expensive tune-ups characteristic of the I.C. engine. Also, because of the clean (combustion-free) air circulating in the engine, carbon deposits should be eliminated and overall engine life extended. Finally, fuel costs should be less due to the fact that the external burner can operate with less highly refined fuels than the I.C. engine.

4.7.2 Hazards

The hazards of the ECPE engine have been investigated and the general conclusion is that the ECPE poses no additional hazards over those of the conventional I.C. engine. It is felt that with proper design and the necessary safety controls any danger of a fuel or lubricant fire or explosion can be eliminated to the same degree as presently realizable in the I.C. engine. The proper design will include: (1) high tolerance, high performance piston rings with the very minimum of lubricant leakage, (2) the possible use of a lubricant having a higher maximum operating temperature than conventional automotive lubricants, (3) an oil trap (or scrubber) within the air storage tank to continuously clean the closed cycle working fluid (air), and (4) valve seals with a high degree of sealing capability (possibly a labyrinth type seal). It is felt that the danger of a lubricant explosion will be further reduced due to the fact that the crankcase is not pressurized (in fact, it will be continuously vented to the burner) and also because after each shutdown the closed cycle air not stored and filtered in the air storage tank will escape from the system taking entrained lubricant with it. In addition, the danger of a fuel explosion is of about the same magnitude as the danger present in conventional I.C. engines, which is quite small.

Additional hazards, such as excessive exhaust temperatures, high rotative speeds, etc., appear to be no more or no less a danger than that present by conventional I.C. powerplants.

4.7.3 Emissions

Recent developments have shown that the type of external burner used in the ECPE powerplant is capable of meeting the 1976 Federal emission standards. The burner development studies are being handled by other OAP contractors and their most recent test results are tabulated below:

EMISSION STANDARDS (GM/MILE)

	1971 Federal Standards	1976 Federal Standards	TECO Ext. Burner (test results)	AEROJET Ext. Burner (projected
CO	47	4.7	.25	.30
UHC	4.6	.14	.2	.01
NO_x		.4	.27	.10

4.7.4 Development

The development of the ECPE into a road-tested, low-emission vehicle powerplant appears to be one of minimum lead time, few risks, and relatively low cost. Its close resemblance to the present I.C. engine should require very few retooling changes for the automotive industry. The production of the heat exchangers should be relatively straightforward, the primary effort here being to keep material and manufacturing costs to a minimum. This, however, should be a simple task for the automotive industry, which has developed the techniques to produce I.C. engines at little more than the cost of the material used to make them.

4.8 CONCLUSIONS

The preceding ECPE concept study has shown that the ECPE powerplant could provide a solution to the air pollution problem. Preliminary design work has been completed and the next phase will involve further detailed design and analysis, hardware development, and powerplant installation in a test vehicle. Hardware development includes head design and heat exchanger fabrication (or acquisition), while powerplant installation primarily includes the acquisition and coordination of the burner, control valves, air storage tank, auxiliary air compressor, and the various drive belts, manifolds and supporting equipment.

5.0 BRAYTON CYCLE ENGINES

5.1 INTRODUCTION

The outstanding success of the gas turbine engine in aircraft applications has generated significant interest in utilization of small turbomachines for automobile powerplants. In aircraft applications very large gains were established quite readily in terms of reliability and maintainability characteristics over reciprocating internal combustion engines. In addition, the gas turbine engine has demonstrated a potential for low weight and size, for quiet, smooth operation, good torque at low speeds (for free-turbine type), and finally, low exhaust emissions. But despite intensified efforts in the past decade to demonstrate the advantages of turbine power for passenger cars, disappointingly slow progress has been made toward the introduction of a mass produced turbine powered vehicle.

Failure of previous efforts to capitalize on the good features of gas turbines have been largely economic. During this period the greatest challenge has been to show sufficiently superior economics and operational characteristics over the reciprocating engine to persuade the automobile industry to replace the massive investment in tooling and capability to produce the conventional reciprocating engine developed over the past 50 years. However, with the adoption of stringent federal emission standards for years 1975 and 1976, the position of all new automotive powerplant concepts has been strengthened and renewed effort has been generated to overcome remaining difficulties with the automotive gas turbine powerplant.

The most commonly noted detriment to extended application of gas turbine power is deficiency in fuel economy when it is necessary to operate a gas turbine at low power settings for extended periods of time (as in the case of the automobile). Other areas of concern include acceleration lag from idle to full power, an increased requirement for air contamination protection due to its high specific air consumption, and sensitivity to ambient temperature conditions. Additionally, gas turbine production requires large quantities of costly materials of limited availability, close manufacturing tolerances, and necessitates an extensive training program to provide competent mechanics for servicing of precision turbomachinery components.

The objective of this study was to examine the various gas turbine concepts under consideration for passenger car application and define the characteristics of these systems so that an overall comparison could be made with conventional and Rankine cycle systems. Past and present work in automotive gas turbine development has been reviewed to determine progress to date, including, among others, the fifty Chrysler produced turbine cars of the mid-sixties, the Williams Research turbine powered Hornet, and gas turbine programs sponsored by the Office of Air Programs to study feasibility, optimize designs, and further the development of gas turbine systems. Such efforts in the field render valuable assistance in evaluation of the vehicular gas turbine powerplant and have provided the direction needed to channel efforts in the areas requiring improvement. Both open and closed Brayton cycle systems were considered. Various gaseous working fluids were compared to determine their relative merit in a closed Brayton system.

5.2 GENERAL

5.2.1 Combustors and Gas Turbine Emissions

Intensified interest in the development of low cost, small gas turbines for automobile and industrial usage have resulted in significant investments in component improvement research. In addition, the impact of emission standards has required that a great deal of attention be given to gas turbine combustors. Attention focused upon combustor development includes both independent research and research funded by government agencies (such as the EPA funding of low NO_x burner development). It is not the intent of this study to contribute to combustor or other component technology, but rather to correlate known characteristics in synthesizing a realistic evaluation of gas turbine systems.

A variety of gas turbine combustor designs are possible, including the more conventional can type (single or multi) and annular combustors. A cross section of a typical annular combustor is illustrated in Figure 5.1, a design which uses several discrete fuel nozzles to inject fuel into the primary zone as sheets of droplets. Fuel-rich combustion results and the primary mixture flows downstream where secondary air is introduced and mixed with the primary combustion products to reduce the overall mixture strength

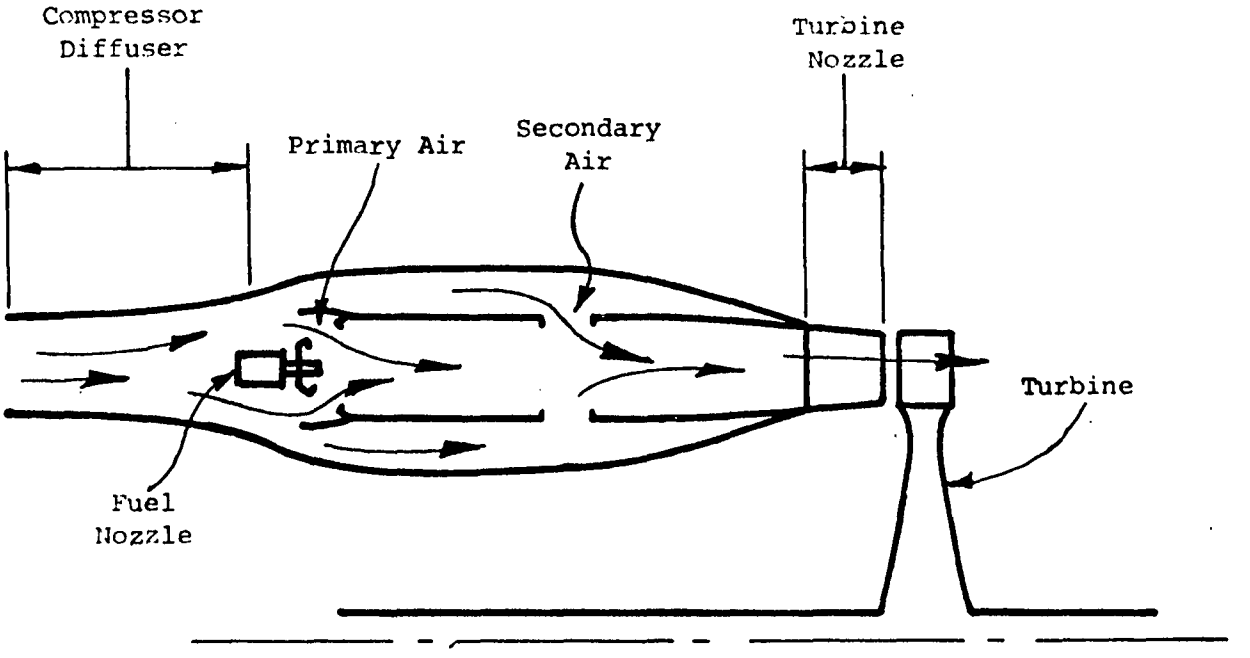

FIGURE 5.1 Representative Gas Turbine Combustor.

(fuel-air ratio) to a value of the order of 0.02 to 0.025 at the design turbine temperature level. Providing the mixture is more or less uniform, there is very little free hydrocarbon or carbon monoxide left in the combustion products and, if the design is proper, there are very few particulates. However, the oxides of nitrogen with this type of combustor can be a problem because there is a variable mixture field around each fuel nozzle, part of which may burn in a high temperature region which for the maximum flame temperature is slightly leaner than stoichiometric, thereby forming excessive NO_x. The through-flow Mach number of these types of combustors is on the order of .2 to .3, so there is inadequate time for the NO_x to dissociate and the combustion process remains essentially frozen as the gas flows through the turbine. Thus, while the present gas turbine can have low emissions of hydrocarbons, carbon monoxide, and particulates, it does have relatively high emissions of the oxides of nitrogen.

Adding to the emissions problem in open cycle gas turbine engines is the high specific air consumption (about 3 to 4 times that of the conventional Otto cycle engine) and the generally unfavorable part load efficiency characteristics of gas turbine engines. Even with high specific air consumption, however, the open Brayton cycle can provide low emissions and, in overall terms, can provide the basis of a very favorable low emission engine. Williams Research, for example, has recently emission tested (in a federal test facility) its 80 horsepower free-turbine engine in a Volkswagen squareback (53), and the results indicate its ability to meet 1975 emission goals:

	Proposed 1975 Standard g/mi	Williams Gas Turbine g/mi	Proposed 1976 Standards g/mi
Hydrocarbons	0.46	0.34	0.14
Carbon Monoxide	4.7	4.5	4.7
Oxides of Nitrogen	3.0	2.15	0.4

These results, although promising, require extensive additional improvement.

Three important characteristics of the open cycle gas turbine combustor which affect gas turbine performance are combustion efficiency, total pressure drop, and average outlet gas temperature. It has been convenient in preliminary cycle analyses done for this and other gas turbine studies to make the simplifying assumptions of constant efficiency and

pressure drop over the entire spectrum of combustor operation. Throughout this study these values have been fixed at 99% and 3% respectively.

Combustion efficiency should always be close to 100% if fuel and air are well mixed in proper proportions, ignited, and given time to burn. In typical aircraft gas turbines the combustor size is critical and combustion may be somewhat less than complete, resulting in off-design efficiency degradation. The automotive gas turbine combustor, however, will be required to burn cleanly and evenly over the entire load range in order to meet stringent emissions standards. Such an advanced burner would have a burner efficiency near 100% over the entire operating range.

Combustion system pressure drop is made up of two components (assuming the diffusion of high velocity compressor air is accounted for in compressor efficiency); the friction loss through the combustion chamber, and the momentum loss associated with acceleration of the gases to a high exit velocity. These losses are related to reference velocity, defined as the theoretical velocity for flow of combustor inlet air through the maximum burner cross-section flow area. Mass flow varies with load, and static pressure varies in a similar manner, so volumetric air flowrate or velocity is nearly constant with load. This characteristic permits the use of a constant combustor pressure drop for all operating conditions.

Fuel types also influence combustor performance primarily because of the effect of differences in viscosity and evaporation rate on atomization and volatility (22). Performance calculations have assumed aircraft-type fuels (JP4, JP5), with lower heating values of approximately 18,400 BTU/lb, although a variety of fuel types are possible.

The materials chosen for manufacture of the combustion chamber are subjected to the highest temperatures prevailing in the gas turbine. A major requirement of the alloys is good oxidation resistance, ease of fabrication in the desired shape, and satisfactory vibration resistance. Refractory alloys such as Hastelloy X have been commonly used for high temperature cycles, but designs utilizing ferritic alloys containing reduced quantities of cobalt and nickel should be investigated in the interest of limiting the use of these critical materials to the absolute minimum. Again, cost is the overwhelming criteria upon which the practicality of the gas turbine as an automotive powerplant will be judged.

5.2.2 Compressors

Advancements in the state of technology of small compressor machinery have resulted largely from attempts to develop compressors for small industrial and aircraft engines, including new high bypass ratio turbofan engines in small thrust sizes which necessarily have very small gas generator components. Efforts in these fields have led to significant improvements in the maximum stage loading capability (pressure ratio) and efficiency of small compressors, particularly centrifugal compressor machinery, which has long been the choice for small turbine designs. Table 5-I lists some important characteristics of radial and axial compressor types in regard to vehicular gas turbine applications.

Aside from these general characteristics, some significant improvements have resulted from advanced centrifugal compressor research. Earlier passenger car turbine efforts, including free-turbine regenerative designs announced by General Motors, Chrysler, Ford, and Williams Research, have utilized a single stage centrifugal compressor of approximately 4:1 pressure ratio. Recent developments, however, have demonstrated the ability to generate pressure ratios on the order of 6 to 7 per stage with compressor efficiencies of the order of .80 and no significant decrease in the surge free range.

Utilization of high stage pressure ratios has several beneficial effects on the cycle, most important of which is better optimization of cycle conditions at lower load conditions for improved part load BSFC. This is illustrated in Figure 5.2, where a comparison of the SFC characteristics of comparable regenerative engines with design compression pressure ratios of 4 and 6 is presented. At the design point there is little to choose between the fuel consumption of these two cycles. However, at part load as the compressor pressure ratio falls and turbine inlet temperature is reduced, the operating line of the higher pressure ratio cycle intersects lines of constant T_4 at nearer the minimum BSFC for that turbine temperature. The equilibrium operating line of a perfectly optimized cycle is defined by the locus of the minimum BSFC points at any operating temperature. Figure 5.2 shows that a regenerative cycle (ϵ =.75) with a design compressor pressure ratio of about 6 tends to follow this locus of minimum BSFC for reduced turbine inlet temperatures. Because of the tendency to much higher

TABLE 5-I

COMPARISON OF GENERAL CHARACTERISTICS OF

RADIAL AND AXIAL FLOW COMPRESSOR TYPES

RADIAL COMPRESSORS:	AXIAL COMPRESSORS:
(1) Have high structural integrity, allowing high tip speeds and higher pressure ratio per stage than axials.	(1) Are limited aerodynamically and structurally to lower tip speeds and stage pressure ratios so more stages are required for a given pressure ratio.
(2) Are less costly due to the ability to investment cast the parts and also require little or no finish machining.	(2) Are costly due to the requirement for close attention to manufacturing details such as leading and trailing edge sharpness and additional stages required to achieve a given pressure ratio.
(3) Are relatively insensitive to losses associated with small airflow sizes and are competitive with axials in efficiency for the same cost.	(3) Are more sensitive to small flowpath heights and clearance effects.
(4) Are inherently more vibration resistant than axials.	(4) Require sophisticated vibration damping methods.
(5) Adapt easily to flowpath requirements for ducting to a heat exchanger or burner which are typically located outside of the axial flowpath.	(5) Require tortured ducting flowpaths to burners and heat exchangers outside the axial flowpath.
(6) Are shorter in axial length for the same pressure ratio but usually exibit larger frontal area.	(6) Provide a low frontal area profile due to smaller O.D. but a longer axial length (due to larger number of stages).
(7) Are generally heavier and have higher polar moment of inertia and therefore are slower to accelerate to high speeds from idle.	(7) Are usually lighter and have low polar moment of inertia if acceleration of the gasifier is an important consideration.
(8) Are less sensitive to erosion caused by air contamination.	(8) Are especially sensitive to sand and dust erosion.

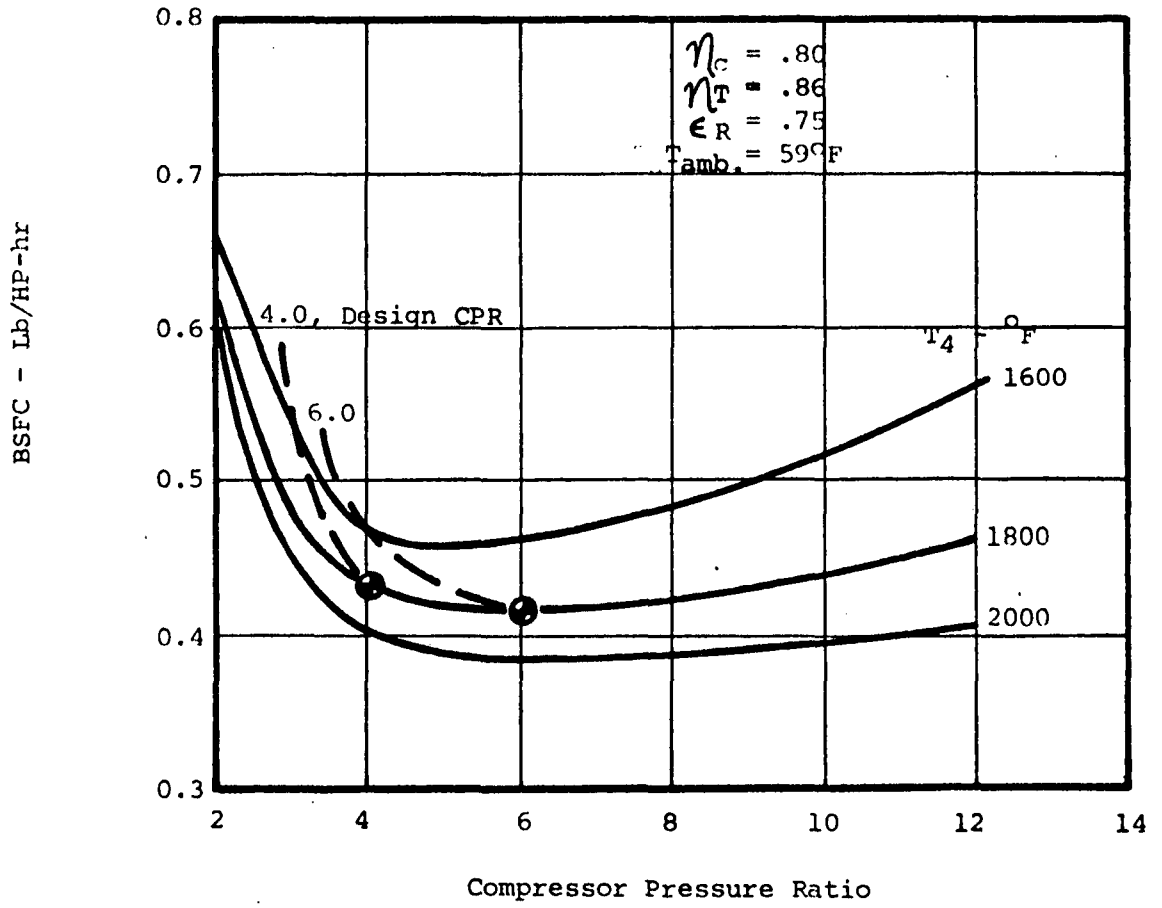

FIGURE 5.2 Effect of Design Compressor Pressure Ratio on Part-Load Specific Fuel Consumption.

fuel consumption at lower cycle operating temperatures and the large amount of time spent in part load conditions in normal passenger car operation, it is absolutely necessary to optimize the design compressor pressure ratio for any cycle, whether simple or recuperative, in this manner.

Optimization of the cycle to higher compression pressure ratios has benefits not only in reducing specific fuel consumption but also in increasing specific air consumption, thereby reducing size, weight, and cost of the entire engine package. A 15% increase in air consumption is possible with an increase in compressor pressure ratio from 4 to 6.

Because of the cost, size, and difficulty in maintaining the heat exchanger component, a somewhat less efficient simple cycle may be desirable. Non-recuperated cycles require higher compressor pressure ratios (typically of the order of 10-12) for optimum performance. Recent advancements in radial compressor technology have demonstrated that such pressure ratios can be achieved in a single stage, but the present drawbacks of such machinery include:

1. vertical speed lines with little surge free range
2. dependence upon variable compressor inlet guide vane geometry for adequate surge margin during off-design operation
3. potential efficiency loss due to very small airflows and associated low Reynolds numbers
4. requirement for titanium parts, precise tolerances, and small clearances for realizing efficiency potential
5. bearing problems resulting from very high rotational speeds (over 100,000 rpm)
6. inertia effects on response time.

Since tortured flowpaths and added costs result if more than one stage of radial compression is used, it is important that the above problems be solved before the elimination of the regenerative type powerplant can be considered.

Exploding the myth that centrifugal compressors are inherently low in efficiency is another objective of extensive research. Approximate 1968 state-of-the-art of small centrifugal compressors (23) is presented in Figure 5.3.

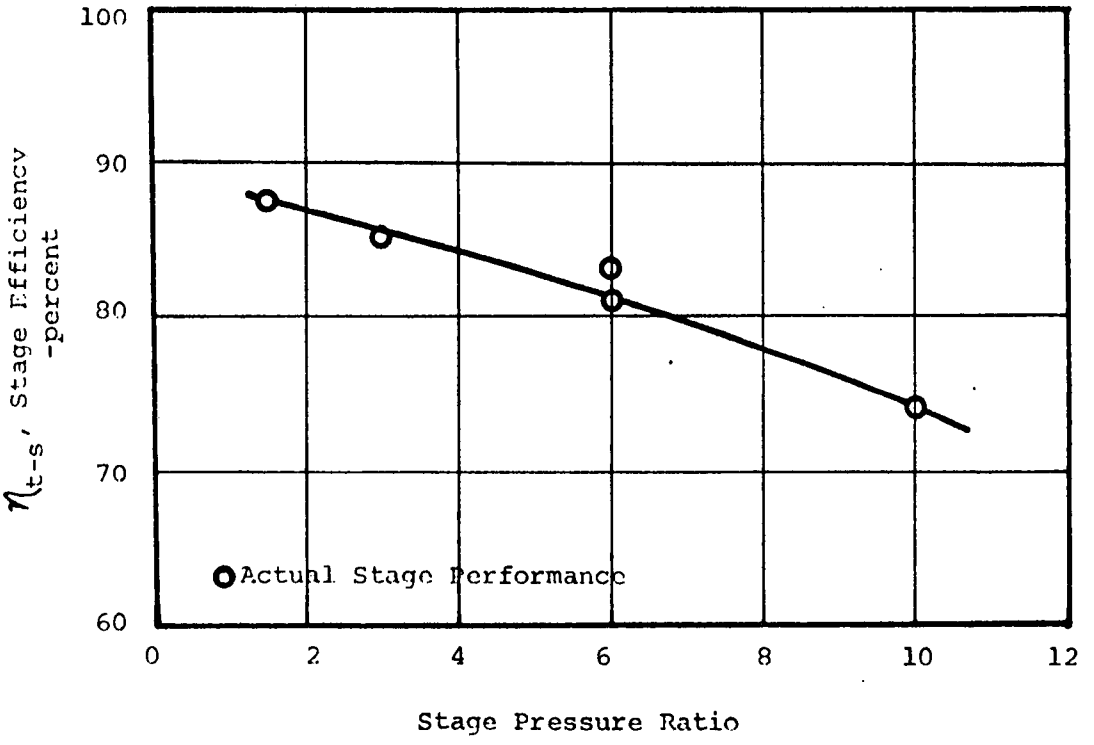

FIGURE 5.3 Present Performance of Small Centrifugal Compressors[23].

Comparison of various gas turbine concepts requires that careful consideration be given to component efficiency values. Compressor efficiency characteristics corresponding to present state-of-the-art technology were used in this study. The incremental difference in BSFC for various levels of compressor design efficiency for a representative range of compressor pressure ratio (CPR) is presented in Figure 5.4 for a simple cycle with 86% turbine efficiency, and 2000°F turbine inlet temperature. An error in estimating compressor efficiency level of plus or minus 2% from the base level of 80% results in an incremental change in BSFC of .010 - .015 lbm/hp-hr, depending upon the design CPR. For a turbine inlet temperature of 1800°F, which is representative of current uncooled turbine blade temperature limits, the incremental change in BSFC is between .017 - .015 lbm/hp-hr. In Figure 5.5 it is shown that this inability to predict compressor efficiency level by a 2% margin affects the design cycle BSFC by approximately the same amount as a change in turbine inlet temperature of 100°F. These characteristics point out the care that should be taken in estimating turbomachinery component performance.

It has been mentioned that variable compressor geometry or preswirl is necessary in the case of high pressure cycles to provide adequate surge margin and optimum engine performance at part power. The effect of variable inlet guide vane geometry on compressor map performance characteristics is illustrated in Figure 5.6, where a typical high CPR machine with variable pre-swirl characteristics is shown. The surge line has been effectively shifted to higher pressure ratios and to lower flows, allowing equilibrium operation at higher cycle temperatures in the normally unstable operating region. Operation at higher T_4, which may be limited by turbine outlet (hib) over-temperature considerations, results in a significant part-load efficiency improvement. The variable geometry, however, affects the mechanical and control system simplicity, cost, and reliability of the simple cycle engines. This type of variable geometry actuation system, however, is considerably less costly, more reliable, imposes less severe component efficiency degradation, and represents less of a development risk than the variable power turbine geometry located in hot sections of a free turbine engine. The variable power turbine geometry will be discussed in following sections.

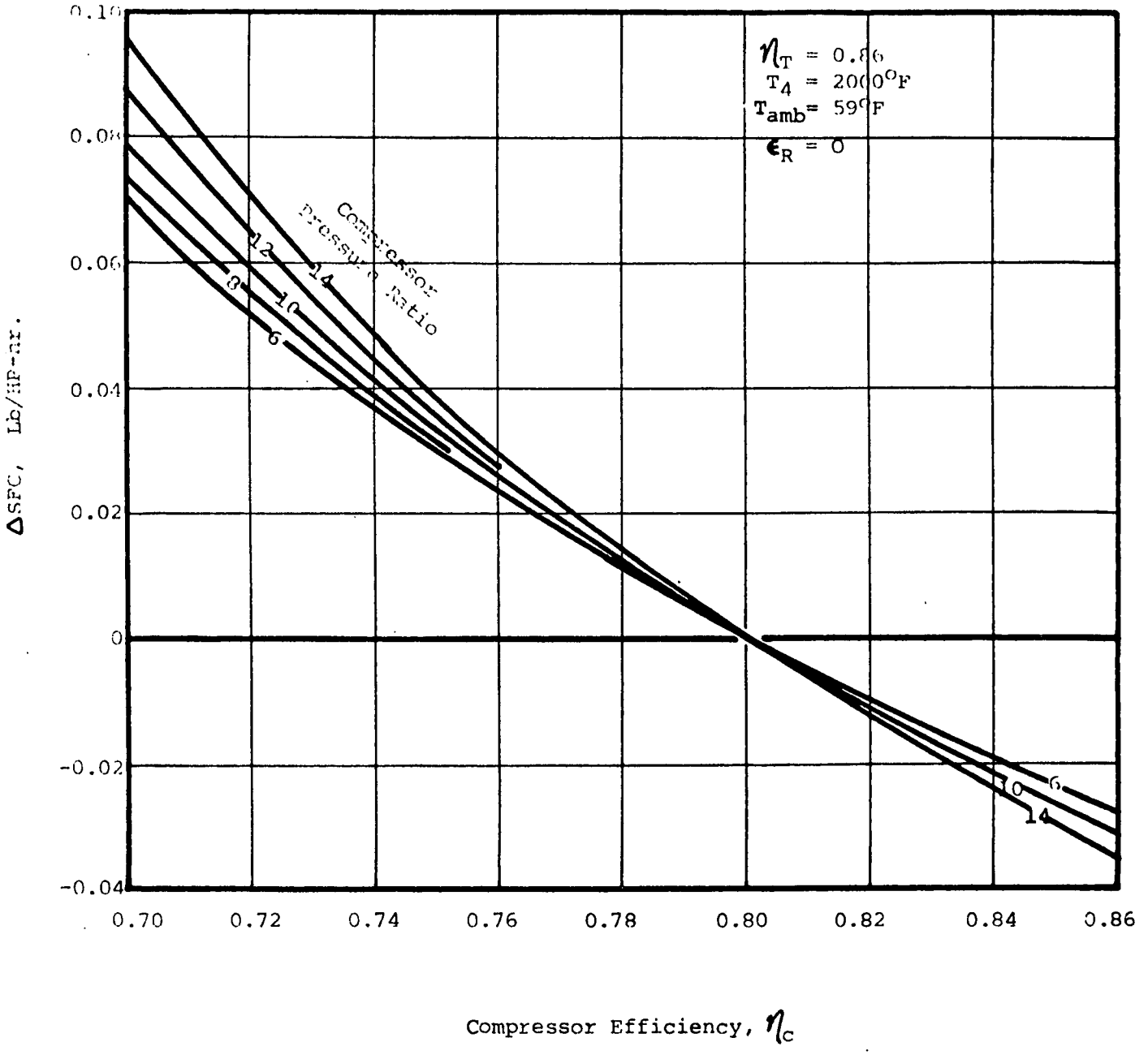

FIGURE 5.4 Effect of Change in Compressor Component Efficiency on Incremental Change in Specific Fuel Consumption (ΔSFC).

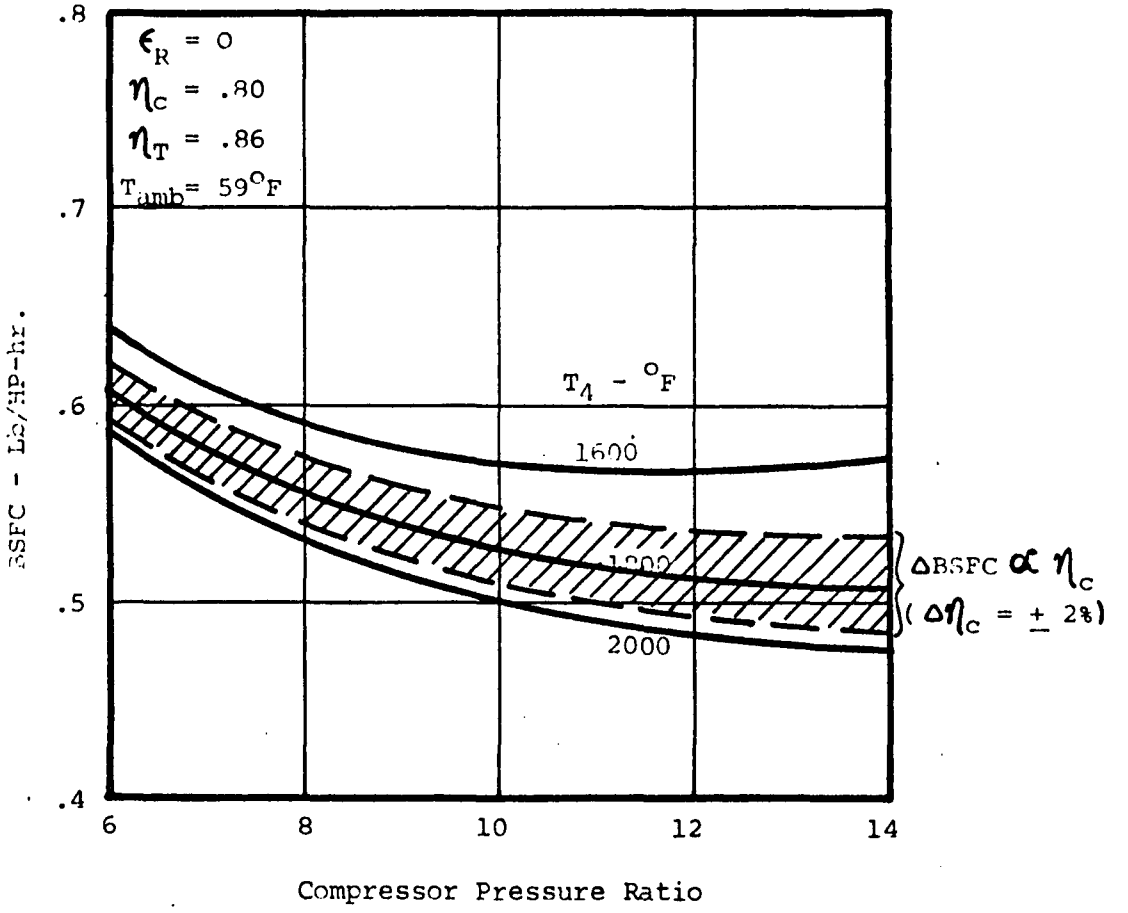

FIGURE 5.5 Degree of Uncertainty of Specific Fuel Consumption Resulting from Plus or Minus Two Percentage Points in Compressor Efficiency.

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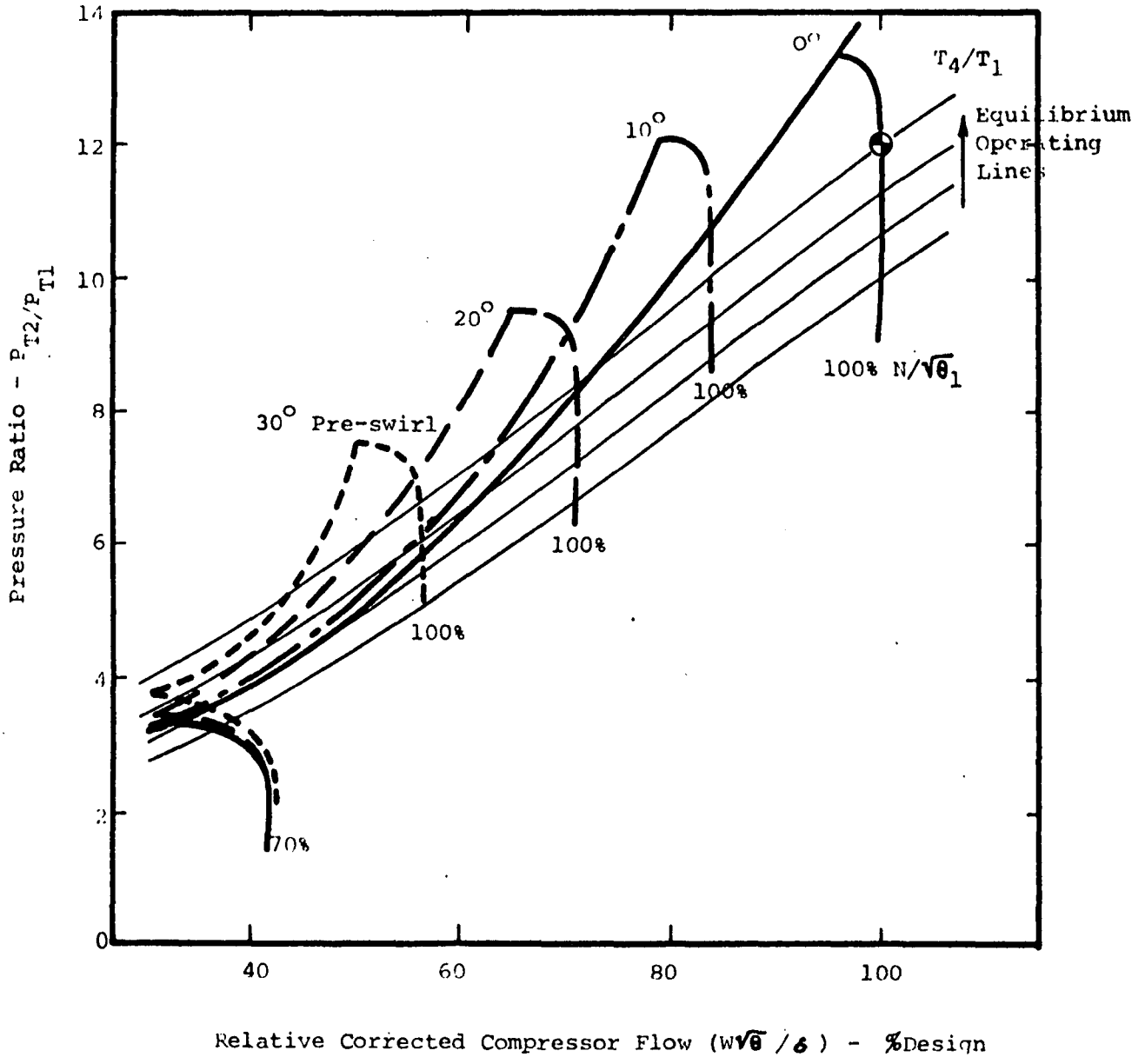


FIGURE 5.6 Typical Compressor Characteristics of High Pressure Ratio Centrifugal Compressor with Variable Inlet Guide Vane Pre-Swirl.

Size effects of compressors with flow rates of 1-5 lbs/second do not appear to represent a significant barrier with respect to performance. UACL has published results of tests conducted on compressors sized for 2-2.5 lb/second (throttled to correspond to airflow sizes of 1.2 - 1.4 lb/second) which meet part speed efficiency predictions. More information is required, however to establish the effect of correct airflow sizes, clearances (which cannot be scaled to small machinery sizes), and the effects of very high rotational speeds on bearing requirements. In conventional aircraft gas turbines the power consumption of roller bearings is a direct function of speed, and for speeds over 100,000 rpm (as required for high pressure ratio compressors) this loss can be significant. Development of low-friction sliding type bearings for small turbomachines is required.

5.2.3 Turbines

Turbines in common usage for small turbomachinery include both axial and radial inflow types. Originally the axial flow turbines were preferred for most applications, whereas radial turbines were utilized primarily in turbochargers where efficiency requirements were relatively modest. However, current trends indicate a changing preference for the structurally superior radial turbines in applications where low cost, configuration compatibility, or particular performance characteristics favor this type (18).

As has been mentioned previously, the economic barrier is the most formidable obstacle to mass production of turbine powered cars. Low cost manufacturing requirements dictate that turbine rotors be cast or forged with minimum machining, and for low cost mass production something other than investment casting will be needed. Radial turbines have demonstrated less sensitivity to casting tolerances and vibration than axial turbines. Cast axial turbines can experience losses of the order of 5 to 10% of efficiency potential if exact tolerances are not adhered to, which makes efficiency of radial and axial stages competitive on an equal cost basis for the range of specific speeds of interest in small vehicular gas turbines.

In addition, radial stages are structurally and aerodynamically capable of higher tip speeds and stage pressure ratios, which is important for the higher pressure ratio cycles now considered for automotive engines

due to the added cost of multi-axial stages necessary to accomplish the same task. Axial turbines typically show linear torque characteristics over a wider range and less of a degradation in efficiency with reduced tip speeds at constant pressure ratio (18). These characteristics, along with flowpath compatibility and lower stage work requirement, are important factors in the choice of axials for free power turbines..

In general, the efficiency of radial gasifier turbines is largely unaffected by variation in speed, and the use of a constant gasifier turbine efficiency in preliminary cycleanalyses can be shown to be essentially correct. The stagnation enthalpy rise across the impeller of a radial compressor without pre-swirl is:

$$\Delta h_{o_c} = \frac{m\, U_2^2}{g_c J}$$

where: m = slip factor

U_2 = impeller tip blade speed (fps)

g_c = constant in Newton's second law (ft-lbm/lbf-sec^2)

J = energy conversion factor (778 ft/lbf/BTU)

The gasifier turbine output head is:

$$\Delta h_{o_t} = \frac{Co^2}{2 g_c J}\, \eta_t$$

where: Co = isentropic spouting velocity (fps)

η_t = gasifier turbine adiabatic efficiency (total to total)

Now,

$$\Delta h_{o_c} = \Delta h_{o_t}$$

or

$$\frac{m\, U_2^2}{g_c J} = \frac{Co^2}{2 g_c J}\, \eta_t$$

and for a 30-32 blade compressor and a radial turbine of equal tip speed,

$$m \approx .9 \text{ and } \eta_t \approx .9$$

so

$$(U/Co)^2 = 1/2(\eta_t/m) = 1/2$$

and

$$U/Co = .707$$

which is approximately the best efficiency design point of a zero-swirl radial turbine and is relatively constant over the range of speeds and pressure ratios of radial gasifier turbines.

In contrast to the operation of the gasifier turbine or single-shaft turbines which operate over a limited output speed range (approximately 60-100% of design), the free power turbine can operate almost anywhere on its performance map in terms of pressure ratio and speed. A derivation of the axial power turbine relative efficiency relationship at any operating point is possible and it is convenient to present the results in terms of the hub spouting velocity or velocity ratio parameter (U_h/C_o).

The linear torque characteristics of axial turbines is shown in Figure 5.7, where torque at design turbine inlet temperature and 100% gasifier speed is plotted as a function of hub velocity ratio. The stall torque is shown to be twice the design point torque and hub velocity ratio is generally equal to .45 at the best efficiency point. Writing the equation of a straight line for this relationship gives:

$$\tau = \frac{\tau_D}{.45}\,(U_h/C_o) + 2\tau_D$$

where: τ = torque (ft-lbf) and subscript D refers to the design point

U_h = hub wheel speed, fps

C_o = isentropic enthalpy drop from inlet total to outlet static, $\sqrt{2g_c J\, h_{o_{isentropic}}}$

From the definition of efficiency and horsepower/torque relationship, we get:

$$\eta = \frac{HP}{HP_{ideal}} = \frac{\tau\omega}{\tau_D\omega_D/\eta_D} = \frac{\tau U}{\tau_D U_D} = \frac{\tau(U_h/C_o)\eta_D}{\tau_D (U_h/C_o)_D}$$

By combining with the linear relationship from above,

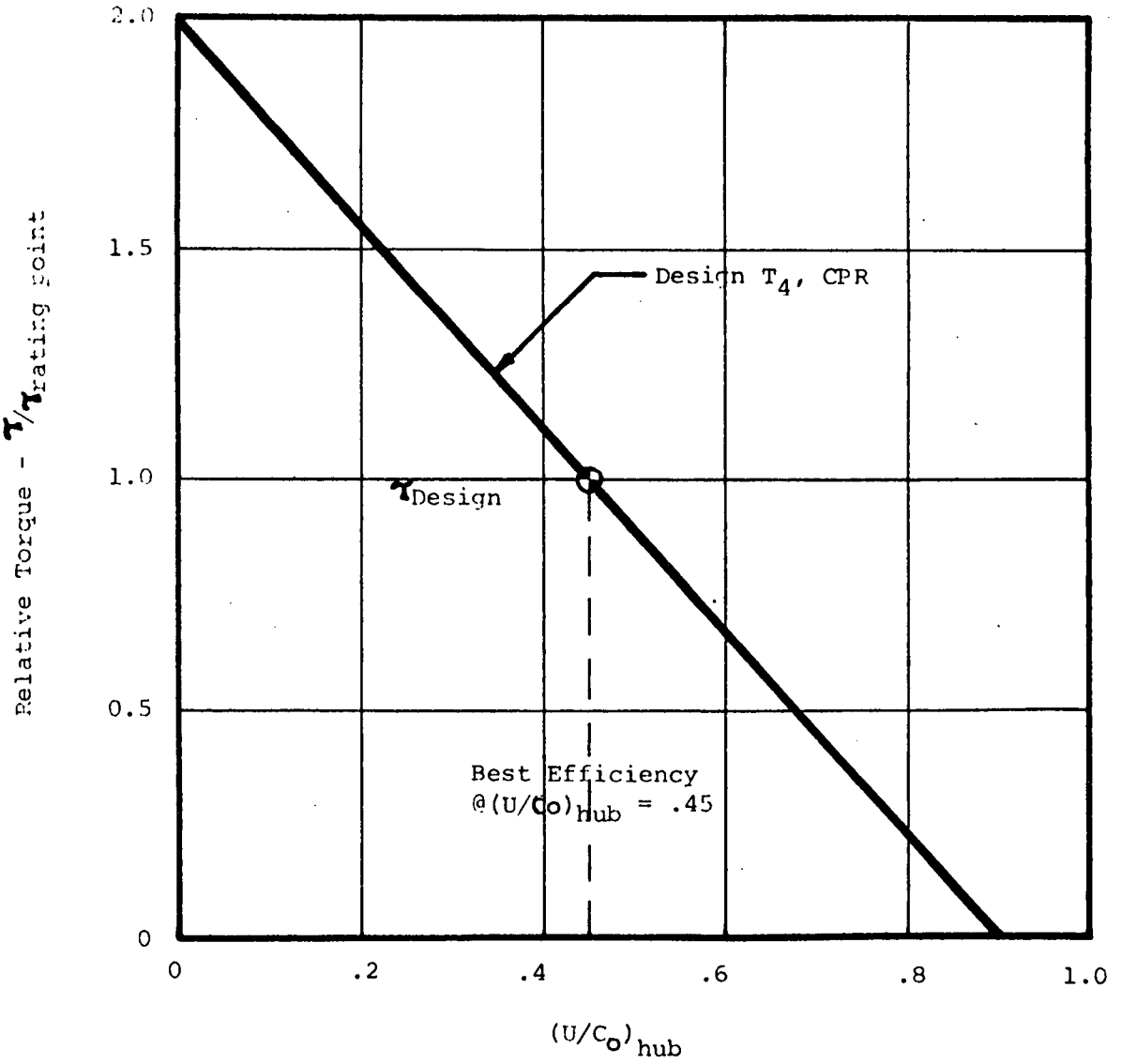


FIGURE 5.7 Axial Power Turbine Torque Characteristics vs. Hub Spouting Velocity at Design Turbine Temperature and Compressor Pressure Ratio.

$$\frac{\eta}{\eta_{Design}} = \frac{\tau(U_h/C_o)}{\tau_D(U_h/C_o)_D} = \frac{\left[2\tau_D - \frac{\tau_D}{.45}(U_h/C_o)\right](U_h/C_o)}{\tau_D\,(.45)} =$$

$$= \left[\left(\frac{2}{.45}\right) - \frac{1}{(.45)^2}(U_h/C_o)\right](U_h/C_o)$$

and

$$\eta/\eta_{Design} = \left(\frac{2}{.45}\right)(U_h/C_o) - \left(\frac{1}{.45}\right)^2 (U_h/C_o)^2$$

This equation gives the relationship of relative efficiency as a function of velocity ratio for an axial free-turbine and is shown in Figure 5.8.

In this study radial turbine types have been used for gas generators and single-shaft applications while axial turbines have been used for power turbine stages. Several of the possible disadvantages of radial turbines are increased weight, larger outer diameter, and generally higher polar moment of inertia, which should be minimized in an effort to reduce acceleration lag between idle and maximum gasifier speeds. However, weight and frontal area are not generally critical considerations for vehicular applications, and unless acceleration lag becomes an insurmountable operational problem with radial turbomachinery, it is concluded that the advantages of radial turbines (particularly cost, system compatibility, strength, and simplicity) outweigh the disadvantages. The radial turbines chosen for analysis throughout this study have been designed for a total-to-static adiabatic efficiency of about 85%. Figure 5.9 illustrates the variation in the increment of BSFC caused by incremental change of turbine efficiency for single shaft engines of 2000°F turbine inlet temperature, turbine and compressor efficiency at 86 and 80%, and a range of compressor pressure ratios. The effect of a two percent loss of turbine efficiency has approximately twice the effect of a similar loss in compressor efficiency (shown in Figures 5.5 and 5.6), which was previously noted to be comparable to a decrease in T_4 of approximately 100°F. Choice of a relatively conservative value of turbine efficiency (85%) reflects the consideration of the importance of this effect. Axial power turbine efficiency was also assumed to be 85%, and a discussion of the effects of variable turbine nozzle geometry on efficiency is included in the analysis of free turbine engines in Section 5.5.

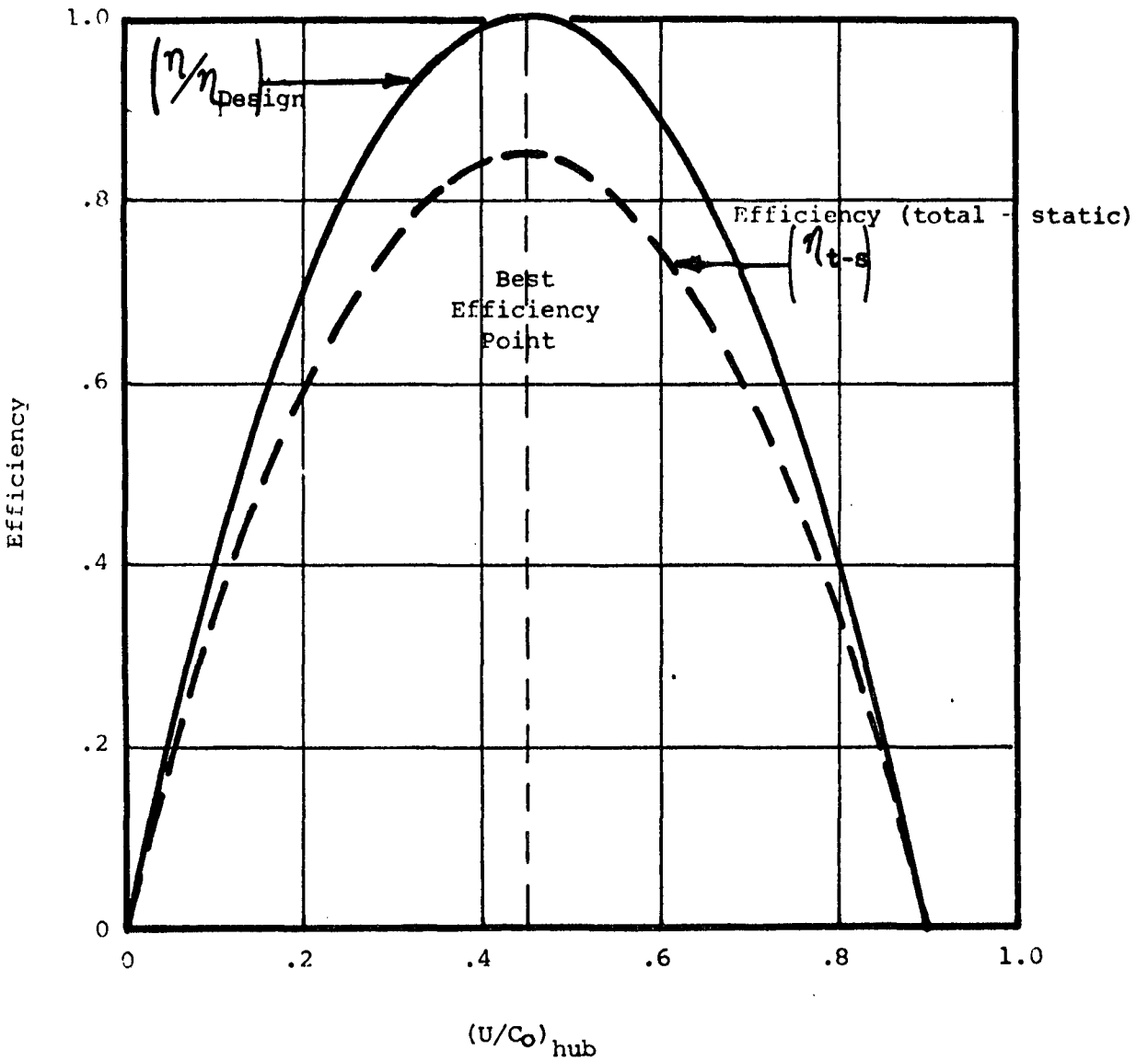


FIGURE 5.8 Axial Power Turbine Relative Efficiency vs. Hub Spouting Velocity.

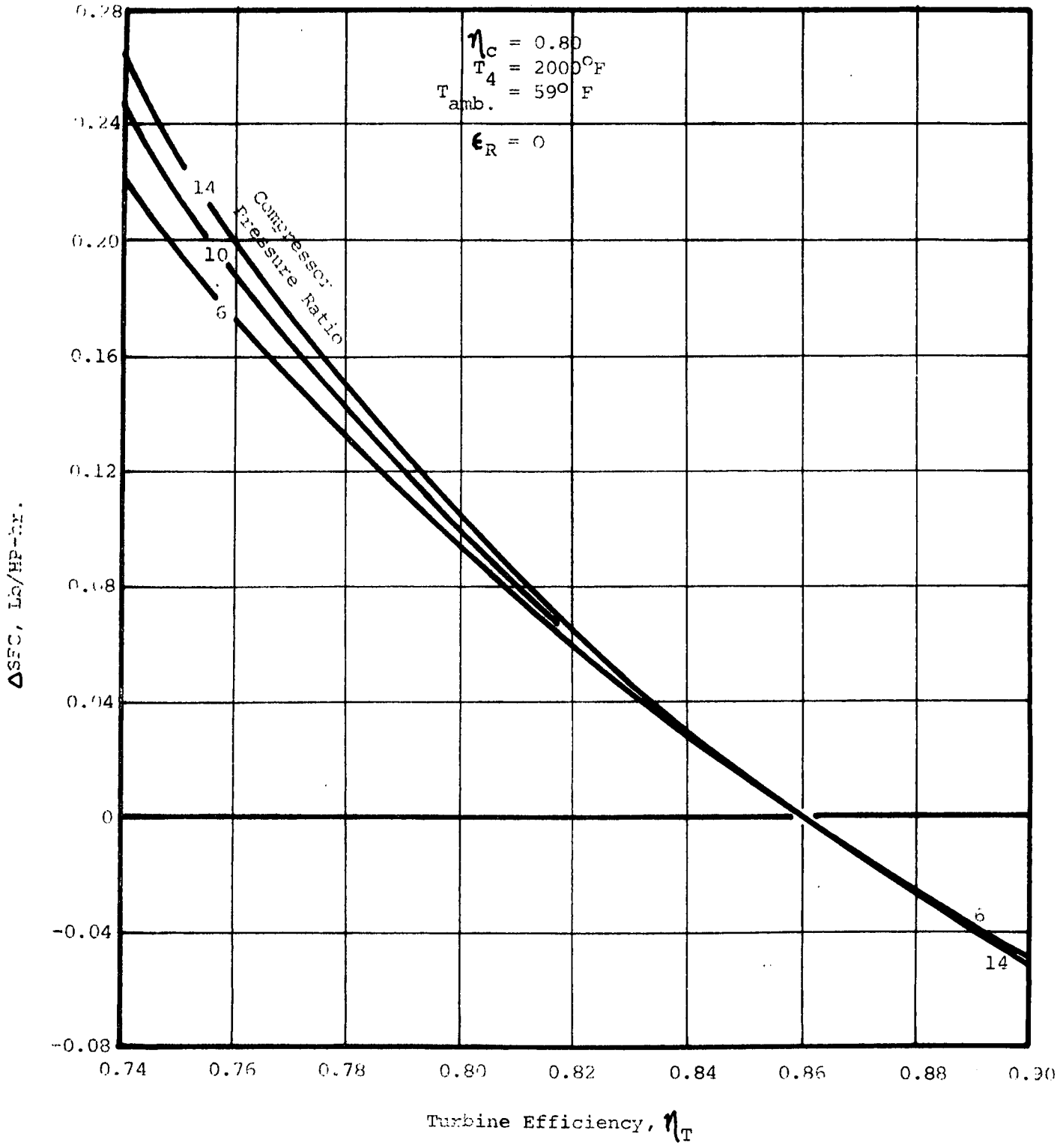


FIGURE 5.9 Effect of Change in Turbine Efficiency on Gas Turbine Incremental Change in Specific Fuel Consumption (ΔSFC).

The turbine temperature limit for uncooled blades and vanes has been determined to be about 1800°F, a temperature considered representative of turbine superalloy materials currently available. Cooled turbine blades are generally not considered for automotive application due to the additional cost and complexity and the lack of experience by the gas turbine industry in cooled radial turbine parts. Cooling of turbines may prove attractive, however, as an alternative to the use of critical materials in the turbine hot section, where a large percentage of the total quantity of cobalt and nickel are required. One of the primary benefits of the high pressure ratio, non-recuperative cycles is the miniaturizing of components resulting from the high specific air consumption and the subsequent reduction of critical material weight (16). A moderate pressure ratio regenerative engine with cooled radial turbine blades might be a better solution to both the critical materials problems and the poor low speed fuel consumption; however, this type powerplant will require additional development. For this reason, air-cooled turbine designs are not considered in this study.

5.2.4 Cycle Losses

Analysis and comparison of gas turbine cycles requires that careful consideration be given to cycle pressure, leakage, and mechanical losses appropriate to the particular design. For preliminary performance analysis of various cycles it has been convenient to account for intake pressure losses (assumed less than 1%) in compressor efficiency and to assume that exhaust pressure losses are a constant 1-1/2% for all cycles. This implies a rather careful component installation, which will be a must for any high efficiency gas turbine automotive powerplant. In heat recovery cycles the pressure losses and leakage losses through the heat exchangers are a function of heat exchanger type, size, and engine operating conditions. Careful attention has been given to this aspect of cycle analysis in evaluating recuperative or regenerative engines.

Mechanical efficiency has been determined to be about 98% for all cycles and is considered constant over the range of speed and power conditions. It is known that frictional losses are a direct function of speed for engines with conventional roller bearings, and the wide variations in design rotational speed between low pressure recuperative cycles (approximately

60,000 rpm) and high pressure simple cycles (120,000 rpm) suggests a two-fold increase in frictional losses with the latter type of engine. Low-friction slider bearings can alleviate this problem in the high speed design, but these bearings typically have large lubrication requirements (whether it be air or oil), can be costly, and have high starting friction. Slider bearings are used where necessary for competitive performance, thus making the somewhat favorable value of 98% mechanical efficiency possible.

5.2.5 Exhaust Heat Recovery

Recuperative and regenerative engines are presently dominant in automotive gas turbine powerplant applications due to the necessity for achieving acceptable part load and idle fuel consumption. A variety of heat recovery means have been used or are currently under study and development for automotive and truck-type applications, most of which favor the rotary regenerator with either metallic or ceramic matrices. Several varieties of stationary recuperators have also progressed to field and/or flight development status, but neither heat exchanger type has been field or customer proven to the degree necessary to assure operational feasibility.

The rotary disk or drum regenerator usually uses a high surface density matrix, and effectiveness values of the order of 90% are commonly attained. The major problems associated with these units are thermal distortion and sealing. The loss of seal integrity causes rapid deterioriation in engine performance due to the cycle losses caused by the leakage from hot and cold gas paths. The high surface density of this type of heat exchanger results in laminar flow which permits favorable effectiveness at low power and a less pronounced pressure drop (and thus power reduction) at higher power levels. Higher effectiveness also results in a lower optimum design compressor pressure ratio for the cycle, simplifying the development of this component and allowing the use of less costly materials. Matrix costs for rotating heat exchangers are typically low but the seals and rotational apparatus add to the total system cost.

Stationary heat exchangers or recuperators can range from tubular construction to intricate compact surfaces. A typical tubular 3-pass design represents a well proven heat exchanger design concept and should present a minimum of fabrication and development problems. Large bulk

and excessive pressure drop characterize this type of heat exchanger. More advanced recuperator types such as compact plate/split-fin designs must also be considered due to the much smaller size and reduced pressure losses at high power levels with these types of designs.

Recuperators have certain advantages over rotary regenerators, such as smaller leakage and no carryover losses, but they are often more difficult to manufacture and are subject to burnout and fouling problems. The regenerator experiences a periodic reversing of flow which purges the matrix and reduces the fouling and burnout problems.

A feasibility study has also been performed with respect to a part-time recuperator or regenerator system which utilizes a bypass concept. This concept developed from the fact that heat recovery (regeneration or recuperation) is really only required at the low power levels since for the automotive application the basic simple cycle has acceptable fuel consumption at the high power levels. A variety of heat recovery types were examined, including rotary regenerators and tubular and split/plate-fin recuperators, to determine the size, cost, and performance advantages of the bypass system. This concept is described in more detail in the heat recovery analysis of open cycle gas turbines, Section 5.5.

5.2.6 Operational Characteristics

Operation of the gas turbine powered vehicle is not expected to be greatly different from the conventional reciprocating engine powered automobile in service today. Engine power output is essentially determined by the turbine inlet temperature (T_4), which is controlled by increasing or decreasing the fuel flow to the combustor by means of a standard accelerator pedal. Fuel controls can be either electronic or hydromechanical, although electronic controls are favored for reasons of low unit cost and durability unless trouble is experienced in field operation. The current Volkswagen electronic fuel injectors combine almost all of the elements required for a gas turbine fuel control. Hydromechanical controls such as used on the GM and Chrysler gas turbine engines are also available and in higher production rates the cost can be brought down to a practical value.

Because low cost high temperature sensing devices do not have durability or quick response, the turbine outlet temperature (TOT) for

single-shaft engines (or inter-turbine temperature for two-shaft engines) is usually chosen as a more suitable control parameter. This is true because of the reduced gas temperature at this station and the fact that a unique TOT exists for any operating point. The TOT schedule at any compressor speed is made to conform to the value ofTIT corresponding to optimum cycle operating conditions. For any particular engine cycle the turbine match, the surge margin, and the turbine outlet hub temperature restrictions will determine the operating temperature at any output power.

Additional engine controls are necessary for utilization of various variable geometry systems. Vane settings are automatically scheduled as a function of engine and ambient temperature conditions. No driver response is necessary for variable geometry actuation, but such additional systems add complexity and are to be avoided unless absolutely necessary for performance or operational stability.

Engine braking in gas turbines is dependent upon the particular design type and the braking features included in the design. The single shaft turbine engine has superior braking capability compared to any of the two-shaft versions as a result of immediate absorption of excess power by the compressor component upon release of the accelerator pedal. In two-shaft designs the power turbine is in a free-wheeling condition rather than a retard position, and some technique of engine braking is required. Variable power turbine geometry generally required for optimum part-load performance in free-turbine engines is helpful in regard to braking because nozzle vanes can be rotated to admit flow to the back side of the turbine blades. Immediate power reduction is provided with bypass features such as the waste gate on the Continental J-65 or windows immediately upstream of the power turbine in the Williams Research engine, both of which divert gasifier turbine exhaust gases away from the power vehicle. Another aid to braking incorporated into both the Williams engine and the General Motors GT-309 truck turbine is the differential clutch arrangement which couples the gasifier and output shafts for braking and power turbine overspeed protection. Two-shaft engines withoutsuch braking aids fail to provide any retardation when the driver releases the throttle. Driver technique is affected by this lack of retardation such that there is a tendency to spend more time in free-wheeling and also very high throttle

settings than would normally occur with engines with good braking characteristics (17).

5.2.7 Power Transmission

The choice of specific engine type (single-shaft or free-turbine) influences the transmission requirements for providing torque at the wheels of the automobile. The primary types of transmissions suitable for vehicular gas turbine powerplants include:

1. Mechanical or hydromechanical
2. Electrical
3. Hydrostatic.

The single-shaft engine has a narrow useful speed range (approximately 60-100%) and an unfavorable torque-speed relationship (in contract to the free-turbine engine which produces maximum torque at stalled output speed). Vehicle acceleration considerations require more sophisticated transmission output characteristics for both regenerative and non-regenerative single-shaft engines. The torque-speed characteristics of the medium (6-8) pressure ratio regenerative cycle are somewhat more favorable because the torque and power at any speed are generally only temperature limited whereas the torque of the simple cycle engine is probably surge margin limited. However, it is almost certain that either an infinitely variable hydrostatic transmission or an 8 to 10 speed mechanical transmission with controlled slipping clutch or torque converter is necessary to provide reasonable acceleration characteristics. Unfortunately, the development status of these more sophisticated transmissions is not at the level where a definite judgement can be made regarding a choice of transmission for a particular signle-shaft engine design. Prototype demonstration will ultimately be required to establish the transmission choice. The hydrostatic transmission is infinitely variable, using positive displacement hydraulic components and technology developed for aircraft hydraulic units. This type of transmission has relatively constant but low efficiency over a wide delivery range. A transmission concept rapidly gaining favor is one which combines the best elements of hydrostatic and hydromechanical transmission, namely the dual mode transmission. In this concept torque is delivered entirely through the hydraulic loop at low speeds where the infinitely variable hydraulic loop

provides more favorable efficiency characteristics than the hydromechanical type. At higher output speeds a reduced percentage of power is transmitted through the hydraulic loop and the more efficient mechanical drive take over.

Pure hydrostatic transmissions are typically 10 to 15 percent less efficient than hydromechanical types, resulting in higher fuel consumption and a larger, more expensive engine to delivery the required power. A hydrostatic transmission does offer potential for transmission weight and volume savings, and the infinitely variable feature can be effectively used to provide improved response or to allow single-shaft or free-turbines to run at their most economical speeds for a given output power.

Electrical transmission, which consist of a d-c generator or a high speed alternator coupled to the engine and electric motors attached to the wheels or drive shaft, can be used with gas turbine engines. Gas turbine powered hauling trucks now use electric transmissions, but the major drawbacks to their use in automobiles is the high initial cost and heavy weight. Efficiency, however, is higher than hydrostatic and comparable to hydromechanical.

Because of their superior torque-speed characteristics, free-turbine engines probably require only a conventional 3 or 4 speed automatic gear shift without the torque converter used in present I.C. engines. Elimination of the torque converter reduces the size and cost of this component and partially accounts for the increased cost of the power turbine, which is really no more than a compressible fluid torque converter. Both of these devices multiply torque through slip, which means an inefficiency is introduced to the system. The power turbine component is lighter, smaller, and more efficient than the converter machinery required for single-shaft engines, and the choice ultimately depends upon weighing these factors against additional cost and system complexity of the free-turbine engine with variable turbine geometry features.

5.2.8 Gas Turbine Concepts

Having reviewed the basic characteristics of gas turbine components, it is now pertinent to investigate the gas turbine concepts which are being studied for vehicular application. Various combinations of compressors,

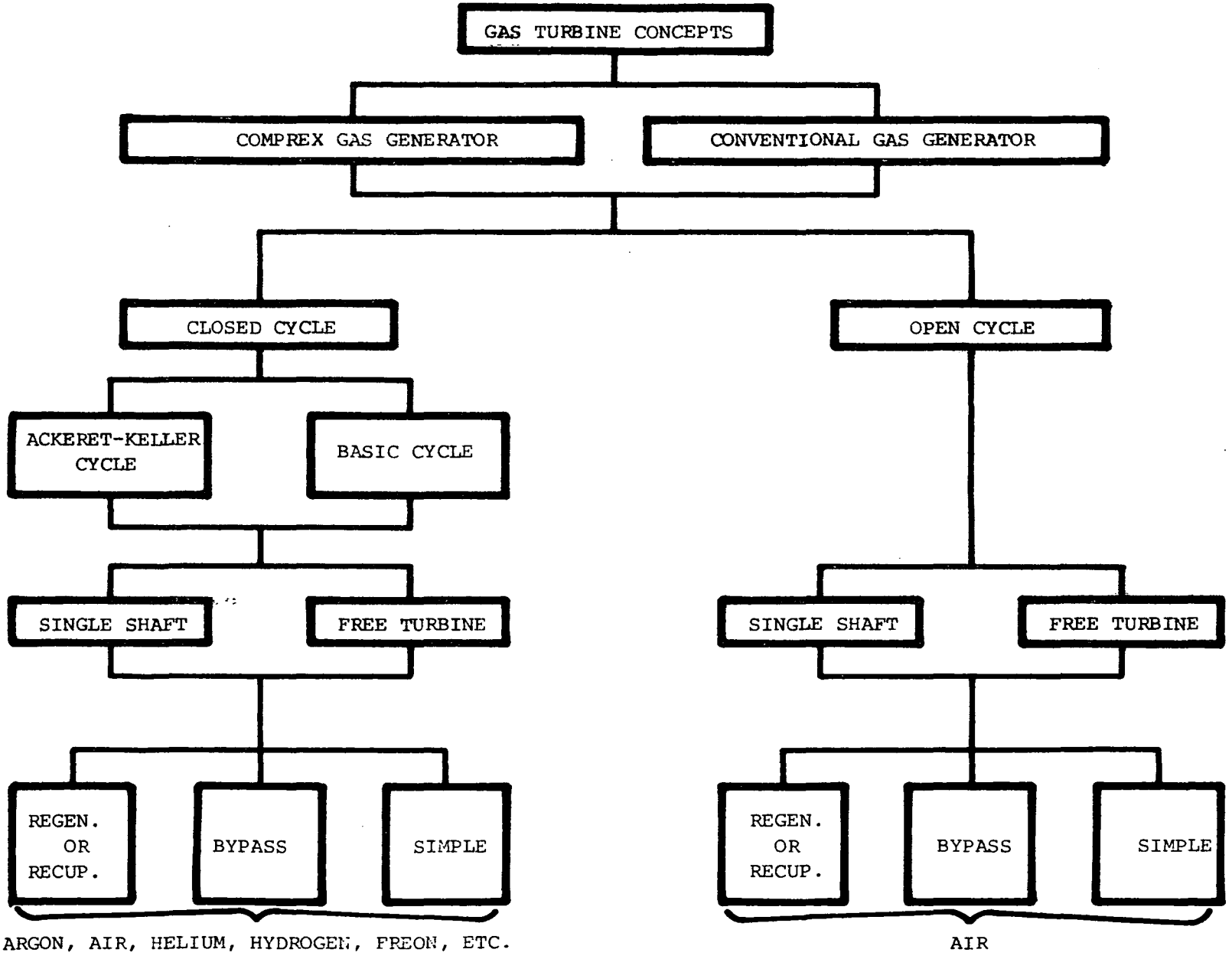
turbines, heat exchangers, and drive systems have been proposed for gas turbine powerplants, however, suitable systems for automobiles require a simple, low cost system with good efficiency over a broad power range. The various concepts investigated under this study are presented in Figure 5.10. The chart is divided to include both conventional (radial or axial compressors) and unconventional (comprex) gas generator types. The chart of Brayton concepts is divided into essentially two sections, namely, closed (and semi-closed) and open cycles. Open cycle Brayton, or gas turbines, using atmospheric air as a working fluid are currently the most popular choice for passenger car applications due to the relative simplicity of this type of system. Further breakdown of the open cycle gas turbine concepts into systems with and without heat recovery and for single or two-shaft designs is shown in Figure 5.10. These concepts are fully discussed in Sections 5.4 and 5.6. Particular emphasis has been given to the heat recovery concepts due to the lack of agreement between various gas turbine designers concerning the most suitable heat recovery means.

Several types of closed cycles have also been examined to determine the feasibility of such systems for automobile powerplants. These include the basic simple and recuperated closed cycles and the more complex Ackeret-Keller cycles with intercooling between compressor stages and reheat between turbine stages. The closed Brayton cycle study (Section 5.3) also includes an investigation into the possibility of utilizing working fluids other than air.

5.3 CLOSED CYCLE BRAYTON

In the closed cycle Brayton powerplant the working fluid is continuously recycled. Heat is supplied to the cycle by means of an external burner through a heat exchanger and waste heat is removed from the working fluid just prior to its return to the compressor inlet to bring its temperature back to the initial cycle starting point temperature. A schematic arrangement of the basic recuperative closed cycle system is presented in Figure 5.11. The basic recuperative system requires four heat exchangers, which include the waste heat exchanger, burner loop air pre-heater, combustion heat exchanger, and recuperator, in addition to an accumulator/compressor for working fluid density control. Of these components only the recuperator is required in a comparable open cycle system.

FIGURE 5.10 Gas turbine concepts for vehicular propulsion systems.



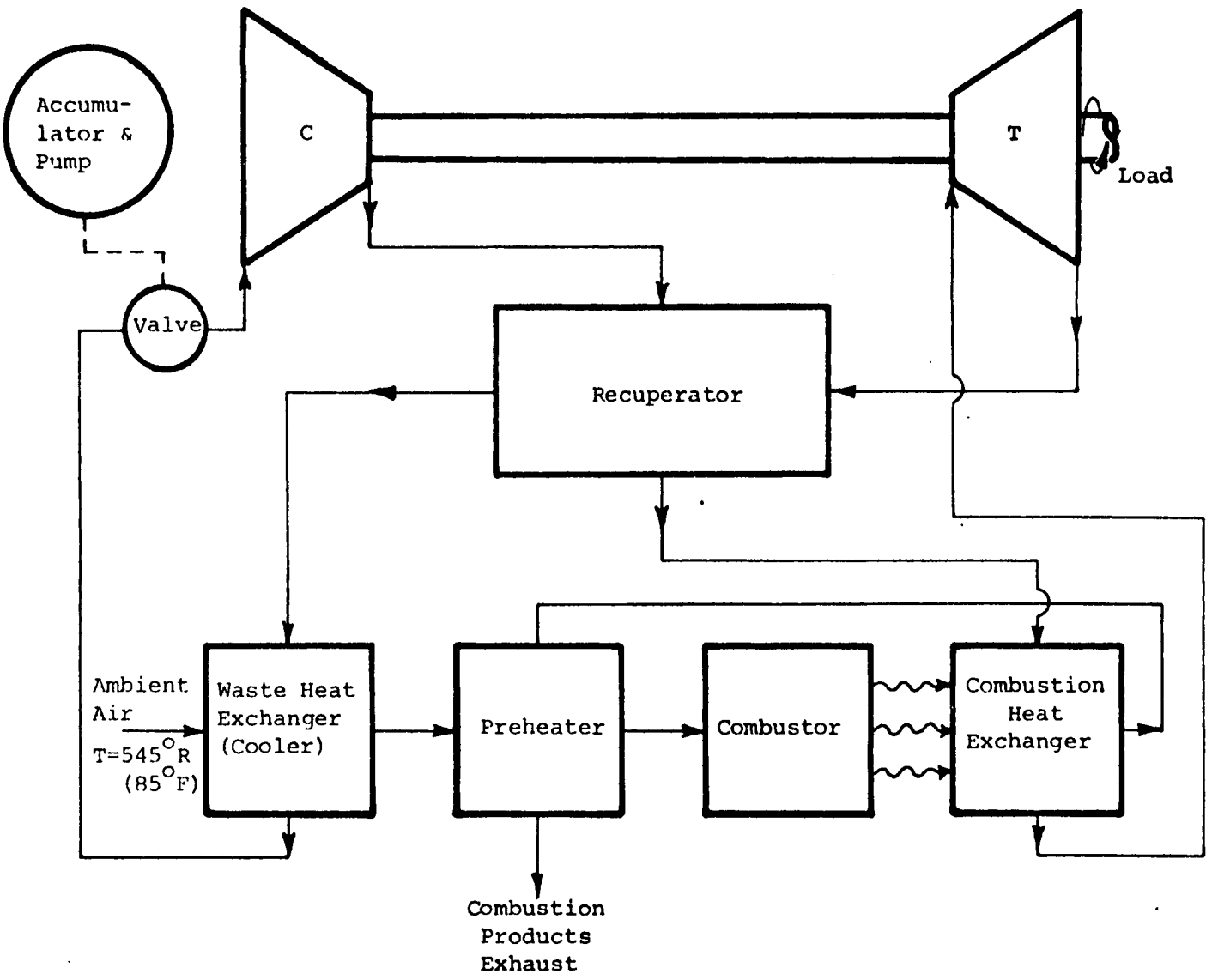


FIGURE 5.11 Schematic of Arrangement of Recuperative Closed Cycle Brayton.

Regardless of the added complexity of even the most basic closed cycle, this concept has developed interest largely due to the following potential advantages of the closed cycle:

1. Favorable emission characteristics of external burners designed for closed cycles as opposed to the less favorable open cycle burner emissions.
2. Relatively constant efficiency over a broad load range by regulating control system density to control power output.
3. Elimination of fouling of machinery and closed cycle heat exchangers due to continuous recirculation of clean working fluid.
4. Reduced size of all system components as a result of the increased cycle base pressure.
5. Increased heat transfer coefficients and lower pressure drops for same size heat exchangers.
6. Possibility of using a greater variety of fuels.
7. Possibility of using a working fluid other than air if found desirable.
8. Reduced noise due to isolation of turbomachinery in closed loop.

The following closed Brayton cycles have been evaluated to determine their feasibility in vehicular powerplant system:

1. Simple closed cycle.
2. Recuperative closed cycle.
3. Ackeret-Keller closed cycle with one stage intercooling and one stage of reheat.
4. Recuperative closed cycle with one stage of intercooling.
5. Recuperative closed cycle with one stage of reheat.

5.3.1 Simple and Recuperative Closed Cycles

The simple (non-recuperative) closed Brayton cycle system requires all of the components shown in Figure 5.11 except for the recuperator (or regenerator) used for turbine exhaust heat recovery. Control of base cycle pressure (density) is the primary mode of power regulation and allows the design efficiency to be maintained over a broad power range. Because of this characteristic, the cycle design parameters should be optimized to

provide best efficiency at the design point. Optimum design compressor pressure ratio for minimum fuel consumption is presented in Figure 5.12 for simple and recuperative cycles (75% recuperator effectiveness) with design turbine inlet temperatures ranging from 1300 to 2000°F. This figure illustrates that optimum compressor pressure ratio for non-recuperative closed cycles is much higher, necessitating either the development of very high pressure ratio single stage compressors or dual compressor stages, which necessarily add to system cost. In addition, these high pressure ratios increase the cycle specific air consumption, which when combined with the utilization of higher working fluid density results in miniaturizing of compressor and turbine component size below practical limits set by speed, clearance, and Reynold's number considerations. Whereas this miniaturization effect is beneficial in regard to large gas turbine units designed for central station power supply, there is no apparent advantage in the case of automotive powerplants where lower limits of size are easily reached or exceeded.

The other consideration which virtually eliminates simple system or the bypass recuperative cycle is the level of design BSFC attainable with these cycles as compared to the conventinal heat recovery type. Figure 5.12 illustrates that the simple system is approximately 25% higher in fuel consumption than a comparable recuperative cycle (ϵ_R=.75) over the range of turbine temperatures available with present materials. Both simple and bypass concepts can be eliminated from further consideration for closed cycle powerplant systems due to performance and size limitations.

It has been noted that external heat is supplied to the closed system through a heat exchanger to bring the closed cycle gases up to maximum operating temperature. Because of the stresses resulting from both the high closed cycle working pressures and from the high combustion product and working fluid temperatures, a tube-type design is suggested. Closed cycle maximum turbine inlet temperatures (TIT) exceeding 1300°F require utilization of stainless steel alloy materials with high chromium, nickel, or cobalt contents for the combustion heat exchanger. The critical nature of these materials in high production quantities with respect to both cost and availability suggests that closed cycle turbine inlet temperatures for large-scale vehicle powerplant production be limited to 1300°F. The importance of reducing critical materials in the combustor/heater component is

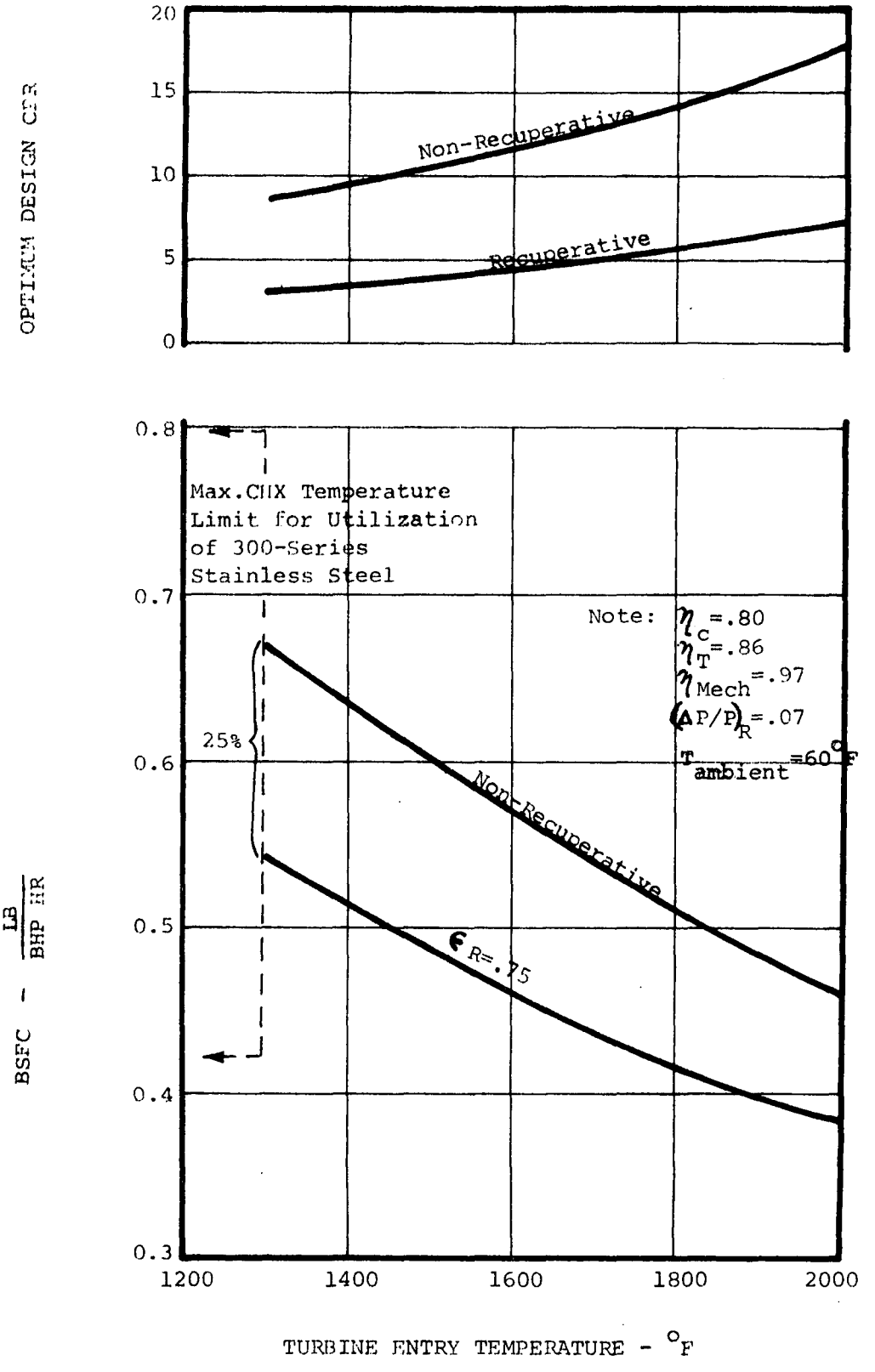


FIGURE 5.12 BSFC at Optimum Design Compressor Pressure Ratio for Recuperative and Simple Cycle Gas Turbines.

reflected in the relative component weights of a closed cycle automotive powerplant. Such a powerplant is proposed (26) to have a total engine weight of 547 lbs and a volume of 10.31 cu. ft., of which 29% of the weight and 35% of the volume are attributed to the combustion heat exchanger. The maximum cycle temperature of this unit (1500°F) requires extensive usage of stainless steel alloy materials in the combustion heat exchanger, and the total consumption of nickel (assuming this engine were mass produced for automotive applications) would be prohibitive.

Cycle parameters for a typical recuperative closed Brayton cycle with the 1300°F maximum turbine inlet temperature limitation are presented in Table 5-II. Design compressor pressure ratio was optimized for the cycle temperatures, assumed levels of component efficiency, and degree of recuperation (75%) to provide maximum thermal efficiency at the design point. Compressor efficiency (78%) was de-rated several points to account for expected losses resulting from corrected airflows less than 1 lb/sec. clearance effects and casting tolerance effects caused by the small rotating machinery size. Consideration of flow rate limits determined the maximum practical base cycle pressure of 75 psia.

At the design power the engine produced 150 HP at a total thermal efficiency of 15%. By reducing the base pressure down to ambient level, the design efficiency is retained down through 20% power. Further reduction of power is possible by either operating the closed cycle in a partial vacuum or reducing turbine inlet temperature. The latter method of power control is illustrated by the part load fuel consumption characteristic represented by the solid line in Figure 5.13. The part load BSFC (dashed line) of a recuperative open cycle engine representative of present technology is also shown. This engine is a fixed-shaft version with a maximum turbine inlet temperature of 1800°F, CPR = 6.3, and plate-fin recuperator designed for 70% effectiveness at approximately 60% output speed.

The comparison between open and closed cycles demonstrates that the recuperative closed Brayton cycle (with air as a working fluid) exhibits no efficiency or fuel consumption advantage over open cycle versions, even at part load. A slight advantage in near-idle BSFC could have been shown for the closed cycle if power were regulated by reducing cycle pressure below 14.7 psia. This would require increased accumulator size and pump

TABLE 5-II

CLOSED BRAYTON CYCLE DESIGN PARAMETERS

	Recuperative Closed Cycle	A-K Cycle 1 stage intercooling and reheat	Recuperative closed cycle with 1 stage intercooling	Recuperative closed cycle with 1 stage reheat
Turbine Inlet Temperature (TIT) ^{o}F	1300	1300	1300	1300
Compressor Pressure Ratio (CPR)	2.9	8.2	4.5	5.0
Compressor Efficiency	.78	$\eta_{c1} = .78$ $\eta_{c2} = .78$	$\eta_{c1} = .78$ $\eta_{c2} = .78$	.78
Turbine Efficiency (Constant)	.85	$\eta_{t1} = .85$ $\eta_{t2} = .85$	.85	$\eta_{t1} = .85$ $\eta_{t2} = .85$
Burner Combustion Efficiency	.99	.99	.99	.99
Recuperator Effectiveness (ϵ_R)	.75	.75	.75	.75
ϵ_{CHX}	.75	.75	.75	.75
$\epsilon_{PRE-HEATER}$	.75	.75	.75	.75
ϵ_{WHX}	.82	.82	.82	.82
Maximum Combustion Temp. ^{o}F	2540	2540	2540	2540
$(\triangle P/P)_R$	.007 (hot side) .002 (cold side)	.007 .002	.007 .002	.007 .002
$(\triangle P/P)_{CHX}$	.015	.015	.015	.015
$(\triangle P/P)_{WHX}$	.005	.005	.005	.005
$(\triangle P/P)_{TURBINE\ EXHAUST}$	.015	.015	.015	.015
Net Brake Horsepower	150	150	150	150
Thermal Efficiency - %	15.0 (air)	17.4	17.0	15.5
Working Fluids Examined	Air, Argon, CO_2, NH_3, Freon, Helium, Hydrogen, etc.	Air	Air	Air

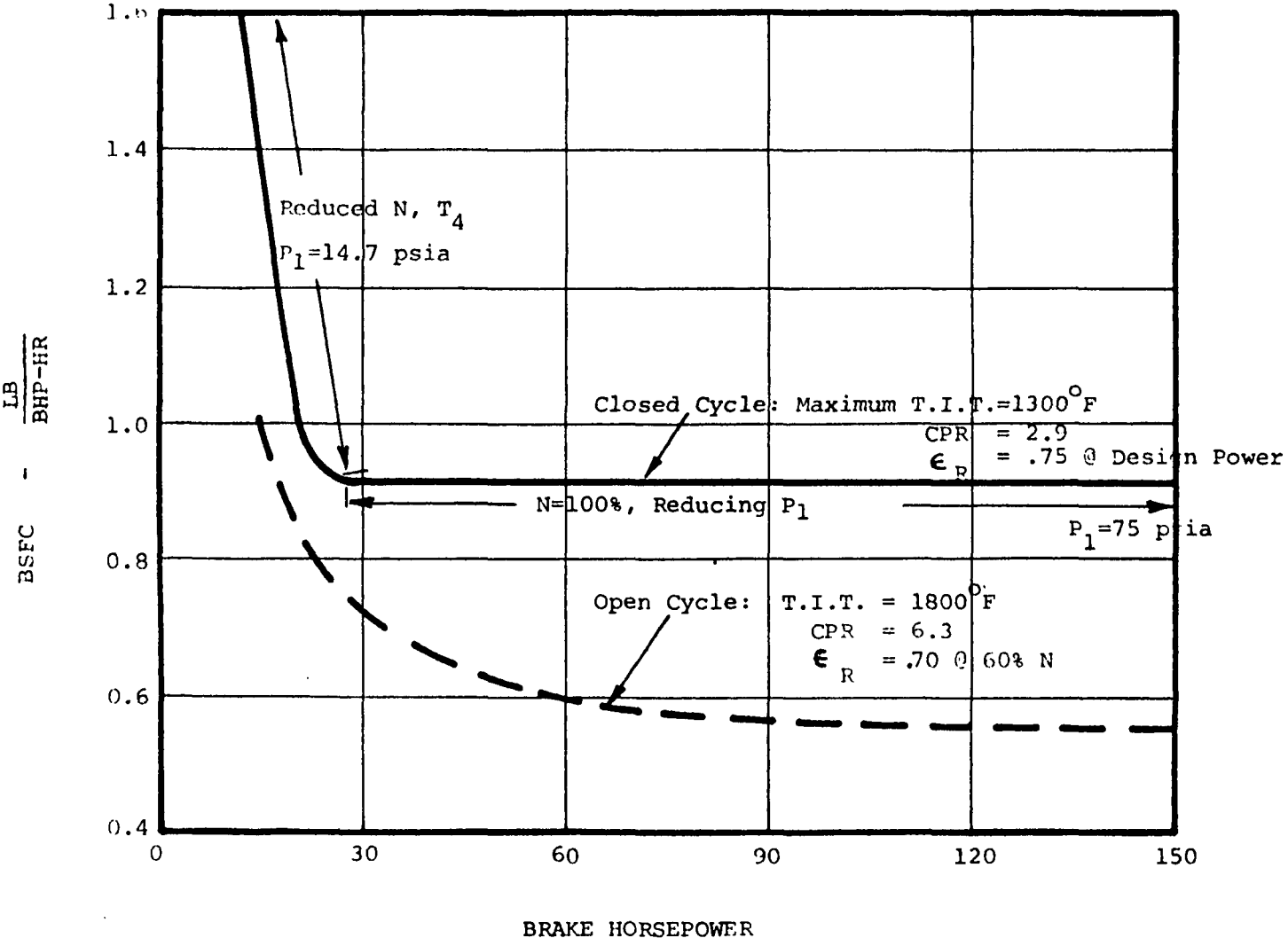


FIGURE 5.13 Fuel Consumption of Open and Closed Cycle Brayton Powerplants at Full and Part Load.

capacity, however, and result in higher engine speeds at low power levels, complicating the already difficult problem of low-power speed reduction in the transmission.

Two factors are largely responsible for the generally lower efficiency of the closed cycle unit:

1. Turbine inlet temperature was reduced to 1300°F to limit the critical materials content of the combustor/heater to a practical minimum.
2. Closed cycle working fluid temperature at the compressor inlet is necessarily higher than ambient air temperature when ambient air is used as the cooling medium for the waste heat exchanger (WHX).

For the cycle described above the WHX effectiveness was 82% and the starting temperature was 165°F. The effect of inlet air temperature on thermal efficiency is presented in Figure 5.14 for several typical Brayton cycles. Although open cycles are shown, the effect of inlet temperature would be similar for closed cycle versions. A 75% recuperative cycle with the design parameters shown is found to suffer a thermal efficiency loss from 25.8% to 20.9% as a result of an inlet air temperature increase from 85°F to 165°F. This amount of efficiency lost represents approximately 20% in increased fuel consumption.

Although the closed Brayton cycle (with air as the working fluid) was shown to be considerably less efficient than the open cycle, it does have the potential of functioning with certain gases which have somewhat better heat transfer properties than air and which might be used to improve the thermal efficiency of the closed cycle powerplant. It is more likely on a cost-effective basis that these fluid properties (high specific heat or fluid density, or both) could be used to more advantage by reducing the size of costly and bulky heat exchangers. Only a limited improvement in thermal efficiency (compared to the air cycle) is generated by the higher effectiveness and lower pressure drops in heat exchangers, not nearly enough to allow closed cycles to be competitive with the open cycle. A detailed cost and performance study is required to determine whether heat exchanger sizes can be reduced sufficiently by utilizing a working fluid with good heat transfer properties to permit use of alloys with higher

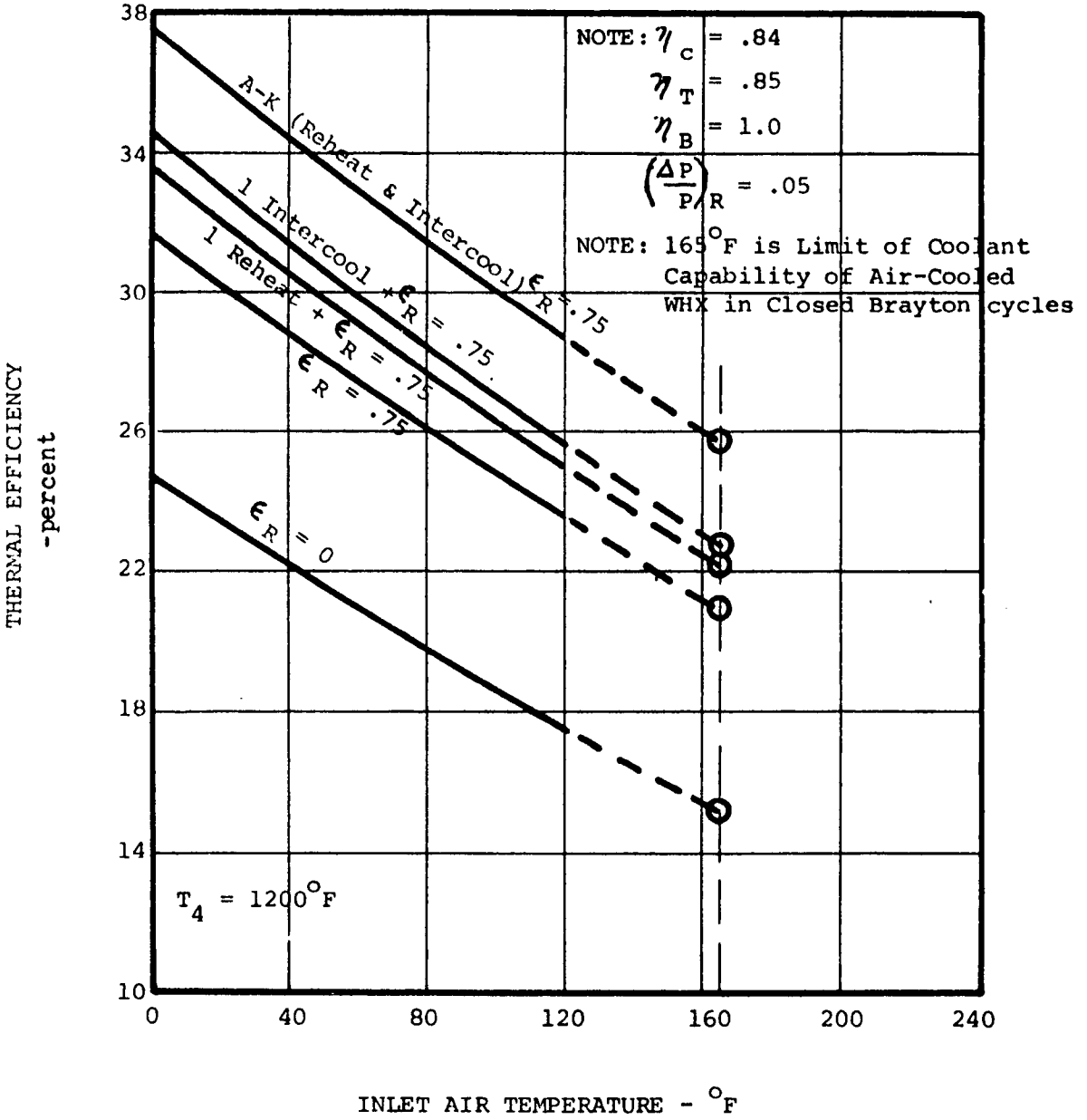


FIGURE 5.14 Effect of Inlet Temperature on Thermal Efficiency of Brayton Cycle Engines. (Ref. 2)

critical material content in the CHX. It would then be possible to increase maximum cycle temperature and allow closed cycle efficiency to be nearly competitive with the best open cycles. This work, however, was not considered in this study.

Some low density gases like helium have high specific heats and provide small heat exchanger components along with much smaller turbomachinery components, thereby requiring more stages of compression to achieve the same pressure ratio as the comparable air cycle. The problems associated with such very small machinery, including the requirement of delicate manufacturing techniques and extremely high rotational speeds, would seem to eliminate these gases from consideration regardless of the other benefits accrued with their usage. Higher density gases such as carbon dioxide or freon provide good heat transmission coefficients, which tends to reduce the size of the heat exchangers; however, this benefit is largely erased by the higher pressure drops which result from the increased density. In addition, some gases (freon, for example) require excessively small turbomachinery sizes and others (like CO_2) do not behave like an ideal gas at high pressures. These are some of the reasons why air is used almost exclusively as the working fluid in closed cycle central station powerplants.

The thermal efficiency of the recuperative closed cycle with a variety of working fluids is presented in Figure 5.15. Design compressor pressure ratio was optimized for each fluid, and other design parameters were comparable to those selected for the air cycle (Table 5-II). Design point efficiencies varied from 14.0 to 16.2%. Although no significant efficiency is evident with working fluids other than air, there are benefits directly attributable to improved heat transfer properties (HX size, cost, etc.). Characteristics of the various possible working fluids for a closed Brayton system are summarized in Table 5-III.

5.3.2 Ackeret-Keller Cycles

Assuming that the emission characteristics of open cycle gas turbines can be solved, it appears that the closed cycle gas turbine, because of its high initial cost, is justified from an economic standpoint only if it can significantly improve thermal efficiency over a broad operating range. An analysis of the recuperative Ackeret-Keller (A-K) cycle, shown schematically

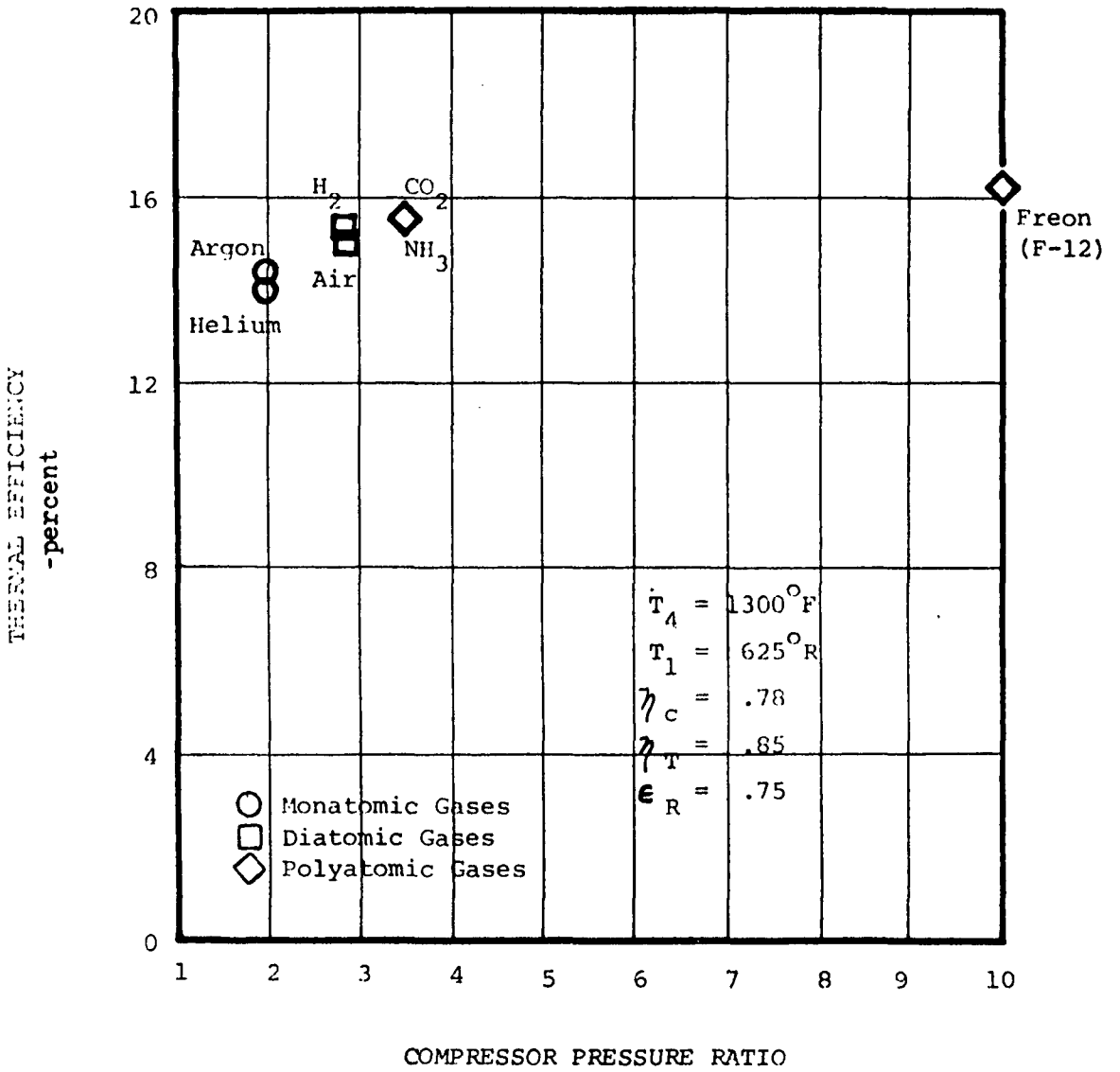


FIGURE 5.15 Thermal Efficiency of Closed Cycle Brayton Engines at Design Compressor Pressure Ratio for Various Working Fluids.

TABLE 5-III

CHARACTERISTICS OF DIFFERENT WORKING FLUIDS IN CLOSED BRAYTON CYCLE SYSTEM

	Specific Heat Cp-BTU/lb ^{o}F (@ 68°F and 14.7 psia)	$k = \frac{Cp}{Cv}$	Specific Gravity (Air=1.00)	Advantages	Disadvantages
Air	.241	1.40	1.00	Readily available; non-hazardous; turbo-machinery design problems minimal	Heat exchangers large in size due to mediocre heat transmission qualities
Argon	.124	1.67	1.379	High density fluid promises compact rotating unit size	Low specific heat counter-balance effect of high density on HX component size; high pressure drops in HX; costlier than air
Ammonia (NH_3)	.523	1.32	.596	High specific heat gives good heat transmission qualities; low density means low HX pressure drops	More compression stages required for same compression pressure level and velocity triangles; low density counter-balances effect of high Cp; costlier
Carbon Dioxide (CO_2)	.205	1.30	1.529	Higher density than air gives good heat transfer coefficients	Low specific and high pressure drops heat counter-balance beneficial effects of high density; costlier; does not behave like ideal gas at higher base pressures (higher BSFC)
Freon (F-12)	.244	1.13	4.520	Very dense and good heat transfer qualities of fluid promise compact unit size	Higher optimum design compressor pressure ratio (10) due to extremely low k value; turbomachinery size very small even without high base pressure; costlier
Helium	1.25	1.66	.138	Small heat exchanger size; inert gas so no oxidation danger exists for turbines or HX's at high operating temperatures	Rotating machinery and apparatus very small; more stages of compression required (21) or higher wheel speeds of turbomachine for same aerodynamic design as air; costlier; high leakage rates
Hydrogen	3.42	1.41	.0695	Promises very compact heat exchangers	Similar to helium; very hazardous even more difficult to control leakage
(He + CO_2)	.755	-	.28	Similar to helium	Similar to helium

in Figure 5.16 with one stage of intercooling and reheat, as well as recuperated cycles with (a) intercooling only, and (b) reheat only, was performed in an effort to document the extent of possible performance advantages of these more complex systems over the recuperative closed cycle and the open Brayton cycle systems. Design cycle parameters for these three variations of the Ackeret-Keller cycle are presented in Table 5-II. The cycle characteristics, including temperature and efficiencies, are the same as noted for the recuperative closed cycle previously noted. Compressor pressure ratios optimize at different values for each of the cycles, as illustrated in Figure 5.17, where thermal efficiency is plotted as a function of compressor pressure ratio (8). It should be pointed out that the relatively high levels of thermal efficiency (32-36%) indicated by Figure 5.17 are not representative of closed cycles operating at 165°F compressor inlet temperatures and somewhat reduced flowpath efficiencies. This plot is useful, however, in illustrating that an optimum compressor pressure ratio exists for each type of cycle. In this manner it was determined that the optimum compressor pressure ratios for the closed A-K cycle of Table 5-II and versions with intercooling and reheat only were 8.2, 4.5, and 5.0 respectively. The design point thermal efficiency of the A-K cycle with intercooling and reheat was 17.4%, only marginally better than that of the basic recuperative closed cycle (15.0%).

Full and part load fuel consumption of the recuperative closed Brayton cycles is presented in Figure 5.18. The reduction in BSFC made possible by addition of intercooling and reheat was approximately 14%. Fuel consumption of a comparable open cycle is an additional 30% lower in BSFC than the A-K cycle.

It is concluded that the closed Brayton cycles (including the A-K cycle) are impractical automotive powerplants for the following reasons:

1. Reduced efficiency relative to open cycle types due to lower maximum cycle temperature (1300°F) and higher compressor inlet temperature (165°F).
2. Increased cost and complexity of the closed system over the open cycle system resulting from addition of at least three heat exchangers plus a working fluid control system (accumulator/compressor).

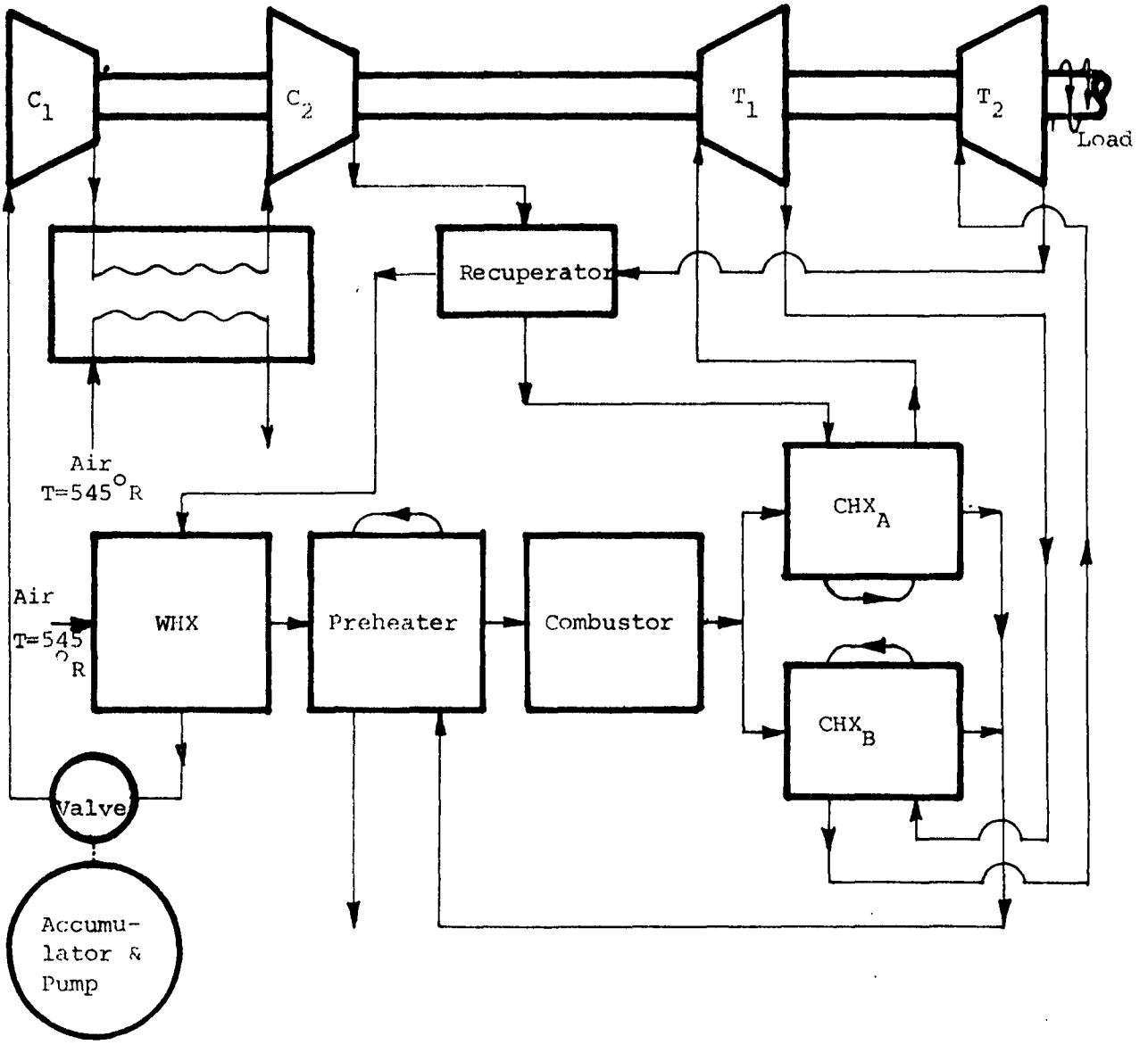


FIGURE 5.16 Ackeret-Keller Type Closed Brayton Cycle with One Stage of Intercooling and Reheat.

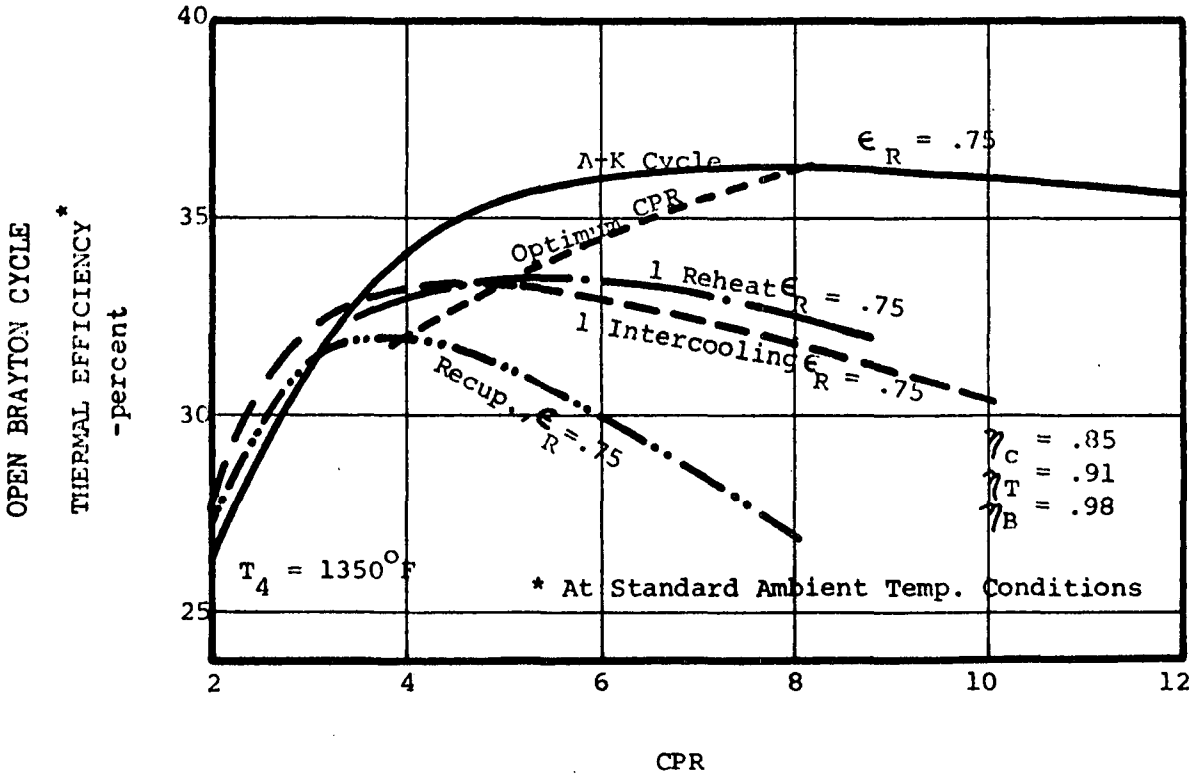


FIGURE 5.17 Thermal Efficiency of Recuperative Brayton Cycles vs. Compressor Pressure Ratio for a Turbine Inlet Temperature of 1350°F and a 75% Effective Recuperator (Reference 8).

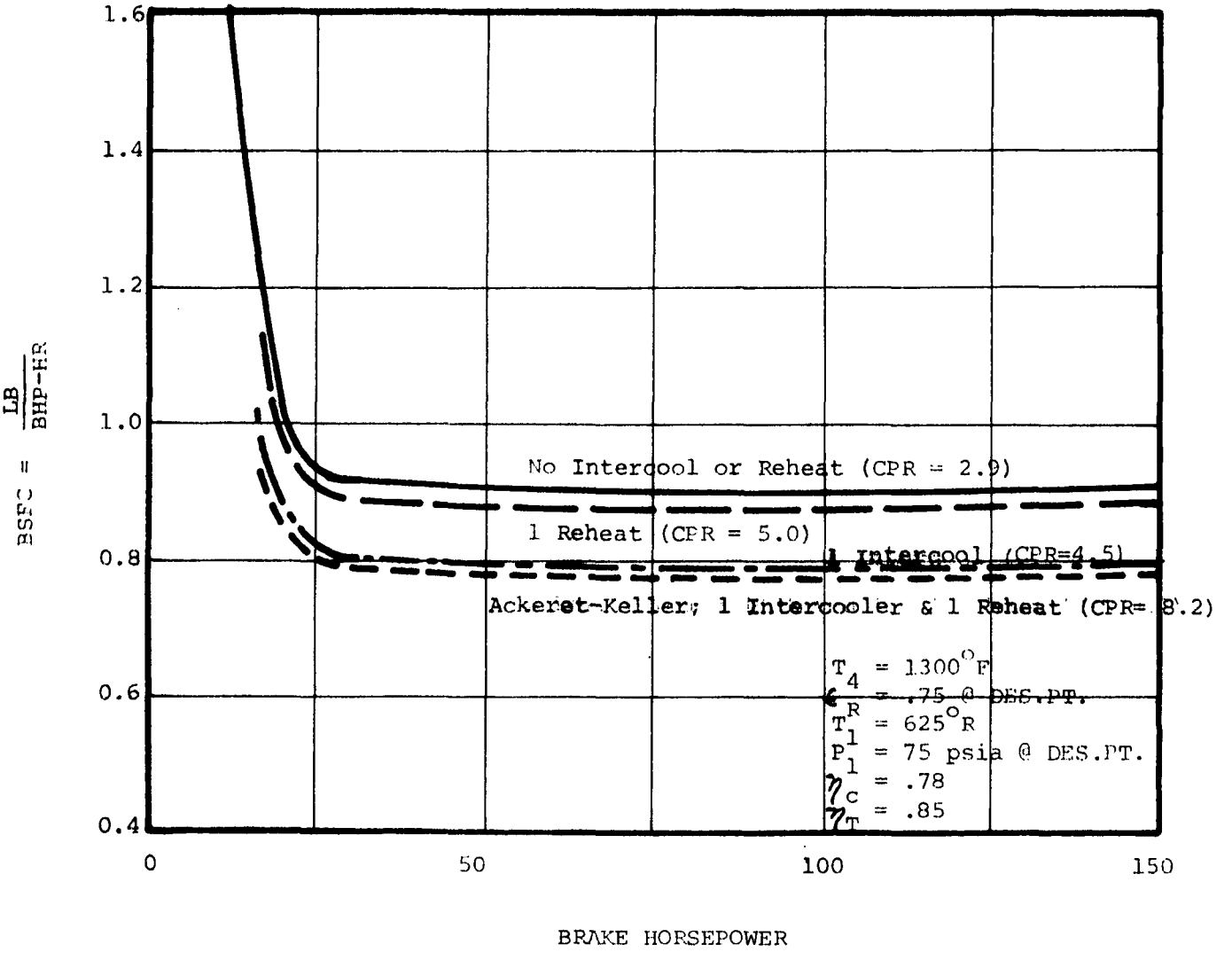


FIGURE 5.18 Full and Part Load Fuel Consumption of Recuperative Closed Brayton Cycles with Maximum Turbine Entry Temperatures of 1300°F and 75% Recuperator Effectiveness.

3. Reduced rotating machinery sizes and the high rotational speeds which are due to the high base cycle pressures.

5.4 OPEN CYCLE GAS TURBINE

Sections 5.1 and 5.2 gave a comprehensive review of automotive gas turbine characteristics in general, while 5.3 discussed the characteristics of the closed cycle gas turbine concept as an automotive powerplant. It was shown in this latter section that the lower efficiency and increased complexity of the closed cycle as compared to that which is realizable with an open cycle essentially eliminates the closed cycle as a contender for an automotive type powerplant. This section will now investigate the various open cycle gas turbine concepts, in particular, those concepts which use various types of heat recovery system to improve the part-load fuel consumption.

Various combinations of compressors, turbines, heat exchangers and drive systems have been proposed for gas turbine powerplants in order to attain improved efficiency over a broad operating range. This is especially important for automotive vehicle applications where efficient operation is required over a wide range of speeds and power levels. During the past few years considerable effort has been directed towards establishing those gas turbine parameters which will provide the best combination of performance and fuel economy. Figure 5.19 shows the fuel consumption characteristics for a simple cycle gas turbine (no heat recovery) as a function of compressor pressure ratio. As is well known, the fuel consumption decreases both as turbine inlet temperature and compressor pressure ratio increase. As was mentioned in Section 5.2, the maximum state-of-the-art turbine inlet temperature for uncooled turbine blades is about 1800°F (due to sulfidation limits imposed by the turbine blade material) and Figure 5.19 shows that the optimum pressure ratio for this turbine inlet temperature is about 15. Therefore, if 15 is chosen as the design compressor pressure ratio with an 1800°F maximum turbine inlet temperature, the specific fuel consumption woudl be approximately .52 lbm/hp-hr. More important, however, is the fact that at lower power levels and therefore lower speeds both the compressor pressure ratio and the turbine inlet temperature decrease resulting in much higher fuel consumption values. The major point to be noted here is that if the simple cycle is designed for minimum fuel consumption at maximum power

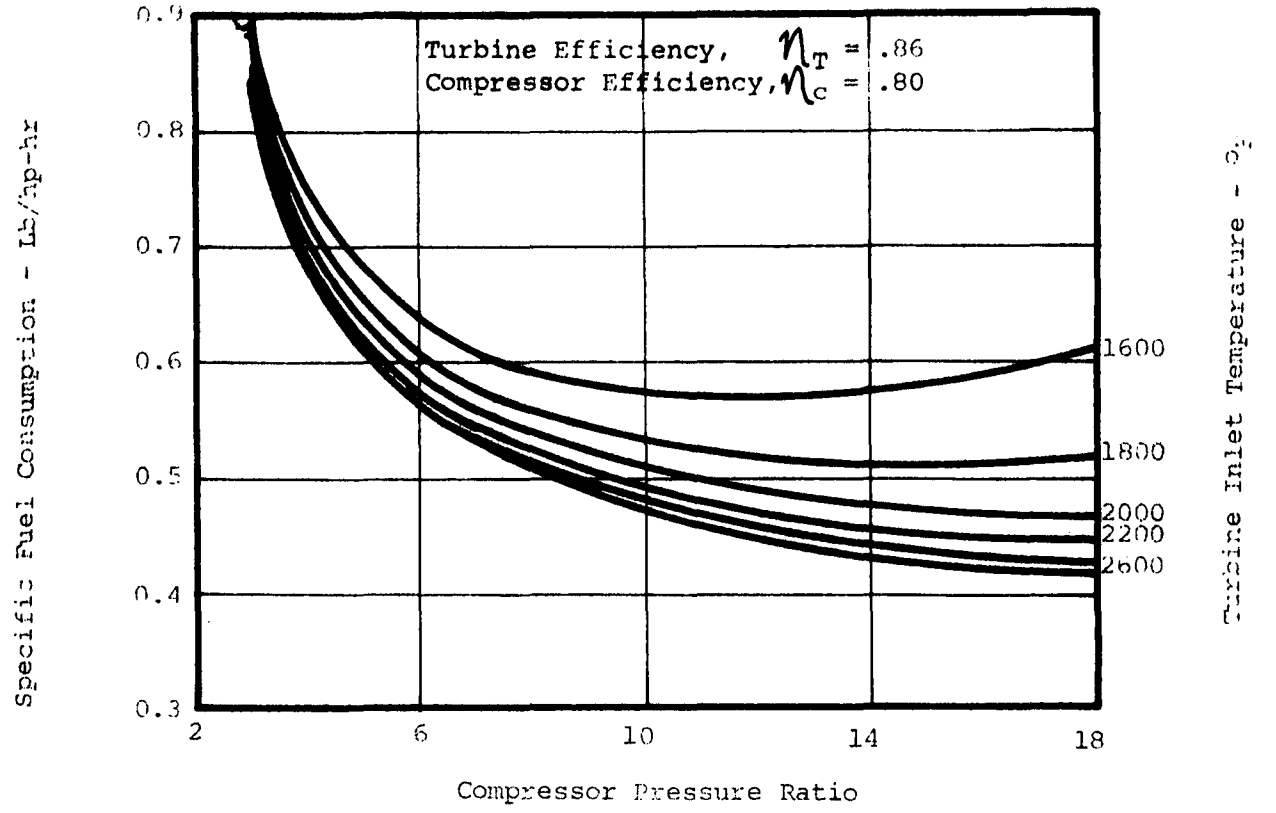


FIGURE 5.19 Generalized Simple Gas Turbine Performance Characteristics.

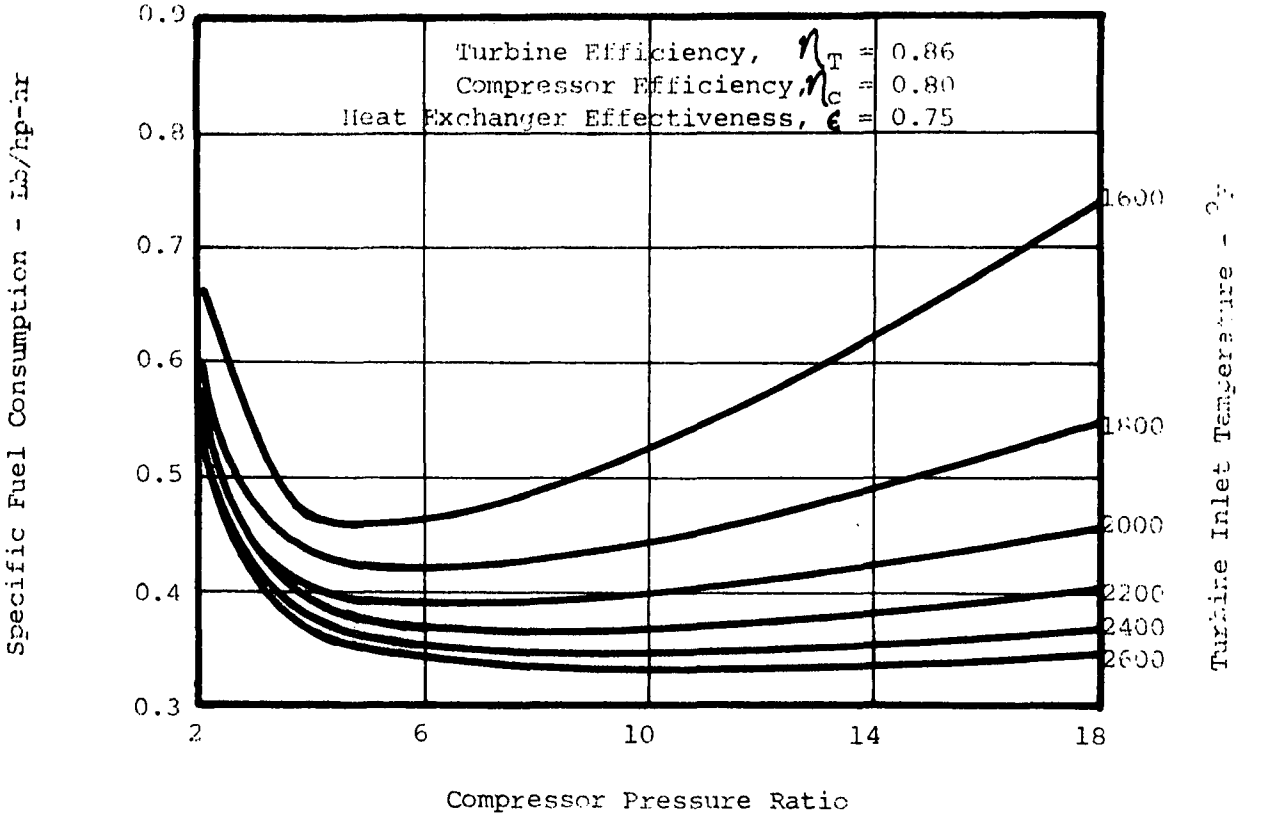


FIGURE 5.20 Generalized Regenerative Gas Turbine Performance Characteristics.

then the fuel consumption at part speed and power will be very poor. In contrast to Figure 5.19, Figure 5.20 shows the fuel consumption characteristics for a regenerative (or recuperative) gas turbine also as a function of compressor pressure ratio. It can be seen that the regenerative cycle has greatly different fuel consumption characteristics than the simple cycle in that the minimum fuel consumption occurs at a much lower compressor pressure ratio. For the 1800°F turbine inlet temperature cycle, and 75% regeneration, the minimum fuel consumption is about .42 lbm/hp-hr at a compressor pressure ratio of about 6. Unfortunately, as in the simple cycle, the fuel consumption of the regenerative cycle also increases at lower power levels and speeds where both the compressor pressure ratio and turbine inlet temperature decrease. However, as can be seen in Figures 5.19 and 5.20, fuel consumption for the regenerative cycle is much less than that for the simple cycle at both full and part power levels.

Therefore, from a fuel consumption standpoint, it appears that the medium pressure ratio, regenerative (or recuperative) gas turbine cycle is superior to the simple cycle. In addition, as was pointed out in Section 5.2, the present state-of-the-art maximum pressure ratio for one stage radial compressors (in the size required for automotive gas turbines) is about 7:1 (with or without pre-swirl). Furthermore, flow path, cost, and complexity considerations have indicated the use of a one stage radial compressor design for the most attractive gas turbine automotive powerplant. Therefore, due to the above characteristics, the one stage radial compressor, regenerative (or recuperative) cycle was chosen as the basis for the most feasible automotive gas turbine powerplant.

The cost and fuel economy advantage of the regenerative cycle over the simple cycle, however, can only be realized with the proper design and matching of the heat exchanger to the turbomachinery. Recent developments in heat exchanger materials have done much to decrease both the size and cost of these components for gas turbine applications. The many types of heat exchangers and their areas of application for various gas turbine powerplants are represented in Figure 5.21. This curve gives the approximate cycle parameter boundaries for the various gas turbine heat exchanger selections. It can be seen that at very high turbine inlet temperatures a ceramic type regenerator (rotary heat exchanger) is required while at very

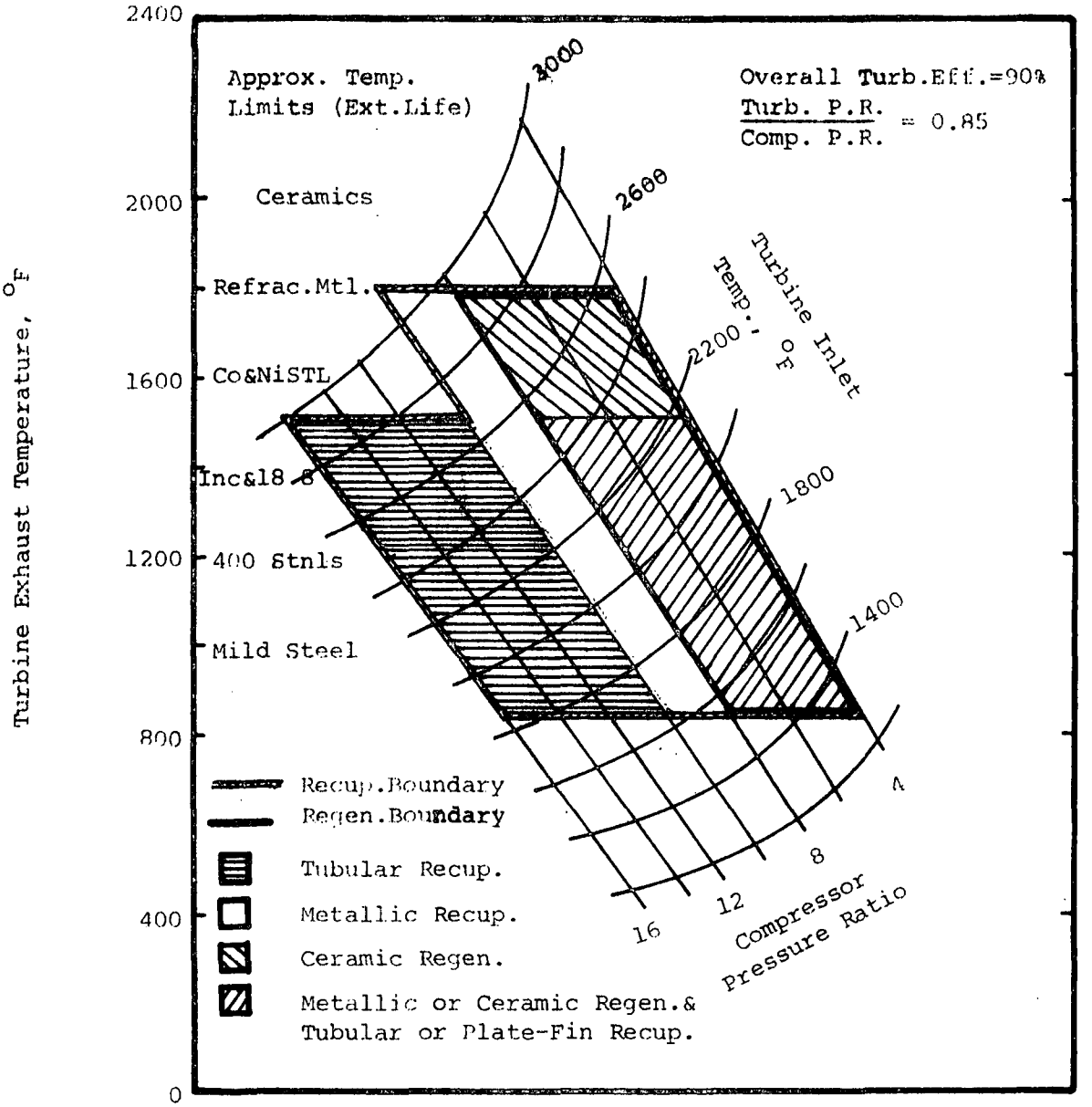


FIGURE 5.21 Approximate Boundaries for Gas Turbine Heat Exchanger Selection (18)

high pressure ratios a metal tubular recuperator (stationary heat exchanger) is required. For the cycle parameters discussed above (turbine inlet temperature = 1800°F, maximum pressure ratio = 7) any of the heat exchanger types are possible, that is, a metal or ceramic regenerator or a tubular or plate-fin recuperator.

Thus, it appears that there are many heat exchanger-gas turbine combinations possible and the problem now becomes one of determining the heat exchanger which, when integrated with the turbomachinery, forms the most compact, reliable, and low cost package. In attempting to design a heat exchanger that was compact, light weight, and low in cost, and which also provides the high effectiveness and low pressure drop required by the cycle thermodynamics, the concept of the bypass gas turbine was developed. This concept developed from the fact that regeneration is really only required at the low power levels since the basic simple cycle has acceptable fuel consumption at the high power levels. This is illustrated in Figures 5.19 and 5.20, where the specific fuel consumption of the simple cycle is about .57 lbm/hp-hr and that of the 75% regenerated cycle about .42 lbm/hp-hr at a compressor pressure ratio of 7 and turbine inlet temperature of 1800°F. In addition, for the same turbomachinery or air-flow size a simple cycle engine will be about 10% higher in maximum output power than the regenerative cycle due to the heat exchanger pressure losses for the regenerative cycle (48). Thus, bypassing the heat exchanger at full power appears to be a most promising concept for reducing turbomachinery size and cost while retaining acceptable fuel consumption at part load. Further, the smaller size regenerator designed specifically for operation at moderate speed road-load levels, combined with its lower cost and weight, should make the overall powerplant more suitable in an automotive application.

Having laid the groundwork for the regenerated gas turbine cycle analysis, the difficult and time consuming task of matching the many possible heat exchangers with the turbomachinery was undertaken. Heat exchanger size, weight, and cost, and powerplant output and fuel consumption were the major characteristics to be considered. The following heat exchanger types were examined for both full time and part time (bypass) heat recovery systems:

1. Tubular recuperator
2. Plate-fin recuperator
3. Ceramic regenerator.

The metallic regenerator was not investigated in any detail due to the fact that its performance is comparable to the plate-fin recuperator while its cost and complexity are somewhat greater. Both recuperators and regenerators were studied because both approaches are currently under development by companies actively engaged in the automotive gas turbine field. It should be emphasized, however, that none of the heat exchanger concepts, except for the tubular recuperator, have been field or customer proven to the degree where operational feasibility is assured.

The rotary disc or drum regenerator usually uses a high surface density matrix, either metallic or ceramic. The major problems associated with these units are thermal distortion and sealing, the loss of seal integrity causing rapid deterioration in engine performance. The high surface density results in laminar flow which permits favorable effectiveness at low power and a less pronounced pressure drop (and thus power reduction) at higher power levels. The ceramic matrix costs are typically low but the seals tend to be relatively expensive. The recuperators (stationary heat exchangers) examined were of the tubular and plate-fin types and counterflow, crossflow, and multi-pass arrangements were examined for both types. The tubular approach represents a well proven heat exchanger design concept. However, as will be shown, large bulk and excessive pressure drops characterize this type of heat exchanger. For the plate-fin design, several compact matrices were examined with the plate/split-fin design proving best. This type of recuperator is very compact, has relatively high heat transfer characteristics, and does not produce excessive system pressure drops. The recuperators (both plate-fin and tubular) have certain advantages over the rotary regenerators, such as no carryover losses, smaller leakage, and no requirement for the complex and power absorbing rotary drive mechanism. However, the rotary has the advantage of a certain degree of purging due to the periodic reversing of flow without which, as in the recuperator, burnout and fouling may occur. Having discussed the pertinent heat exchanger characteristics, it is now necessary to determine the turbomachinery parameters which, when combined with these heat exchangers, will optimize the powerplant design.

The regenerative (or recuperative) cycle performance is a strong function of pressure ratio and temperature. The maximum turbine inlet temperature was determined to be 1900°F and was set by sulfidation limits

of the turbine blade material. The maximum turbine outlet temperature was determined to be 1100°F and was set by the Larson Miller stress rupture characteristics of a 400 series stainless steel. The design point compressor pressure ratio was determined from Figure 5.2 where the locus of optimum pressure ratios for lowest fuel consumption was determined for various turbine inlet temperatures. For the design point (turbine inlet temperature = 1800°F) the optimum compressor pressure ratio was about 6.0 (with 75% regeneration); lower design pressure ratios than this will produce less feasible engines for automotive gas turbines. A single stage radial compressor was chosen and the compressor map shown in Figure 5.22 represents a scaled version of that reported by Kenny-Morris (9). The compressor efficiency has been discounted by 2 points to reflect diffusion and dump losses. This map has turbine inlet temperature lines overplotted and the design power output (160 BHP) was established at a pressure ratio of 6.3 (on an 85°F day) with a turbine inlet and outlet temperature of 1800°F and 1100°F respectively. The part load power control schedule is assumed to keep the turbine outlet temperature constant at 1100°F down to about 60% speed, which permits adequate surge margin and prevents turbine hub and heat exchanger over temperatures during acceleration. Operation along this schedule (represented by dots on Figure 5.22) results in the most efficient engine operation.

The compressor outlet, turbine inlet and turbine outlet temperatures which are characteristic of this steady state operating schedule are shown in Figure 5.23. At 100% power and speed, the turbine inlet temperature is 2260°R and as power and speed are decreased, the turbine inlet temperature must be decreased to keep the turbine outlet temperature constant at 1560°R and also to prevent the compressor from surging. Both compressor and turbine are limited to operation between 60 and 100% rotor speed, as speeds lower than this result in poor acceleration characteristics. As shown in Figure 5.23, the turbine outlet temperature is constant down to 60% speed (about 15% power) at which time it drops quite rapidly due to the rapid drop in turbine inlet temperature. The turbine inlet and outlet temperatures at idle (about 5 HP) are approximately 1200°R and 1000°R respectively. These temperature paths will be used to determine heat exchanger performance characteristics as a function of both speed and power.

The difference in performance between single-shaft and free-turbine engines or other variations do not significantly affect the results of the

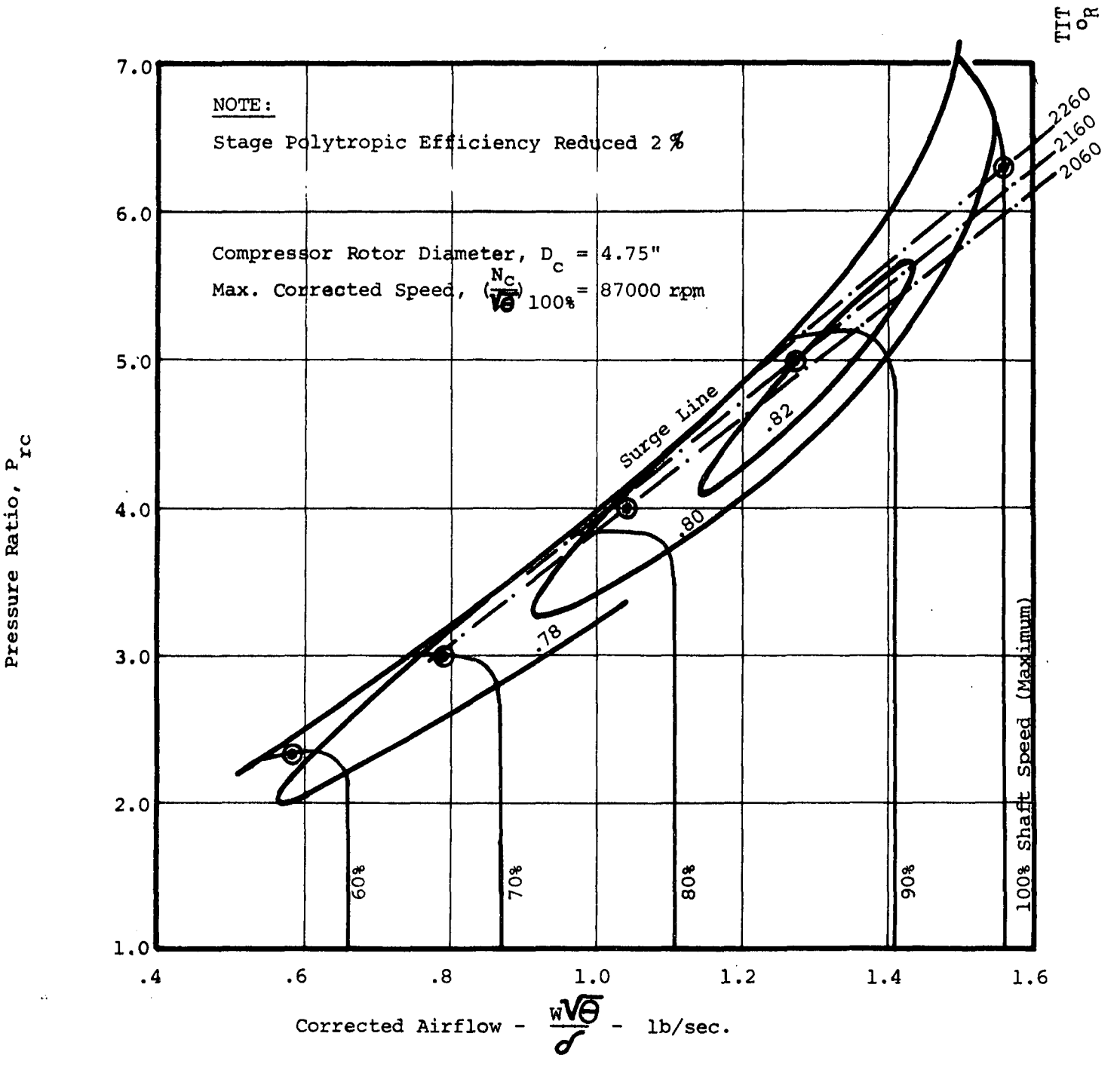


FIGURE 5.22 Single Stage Centrifugal Compressor Estimated Performance.

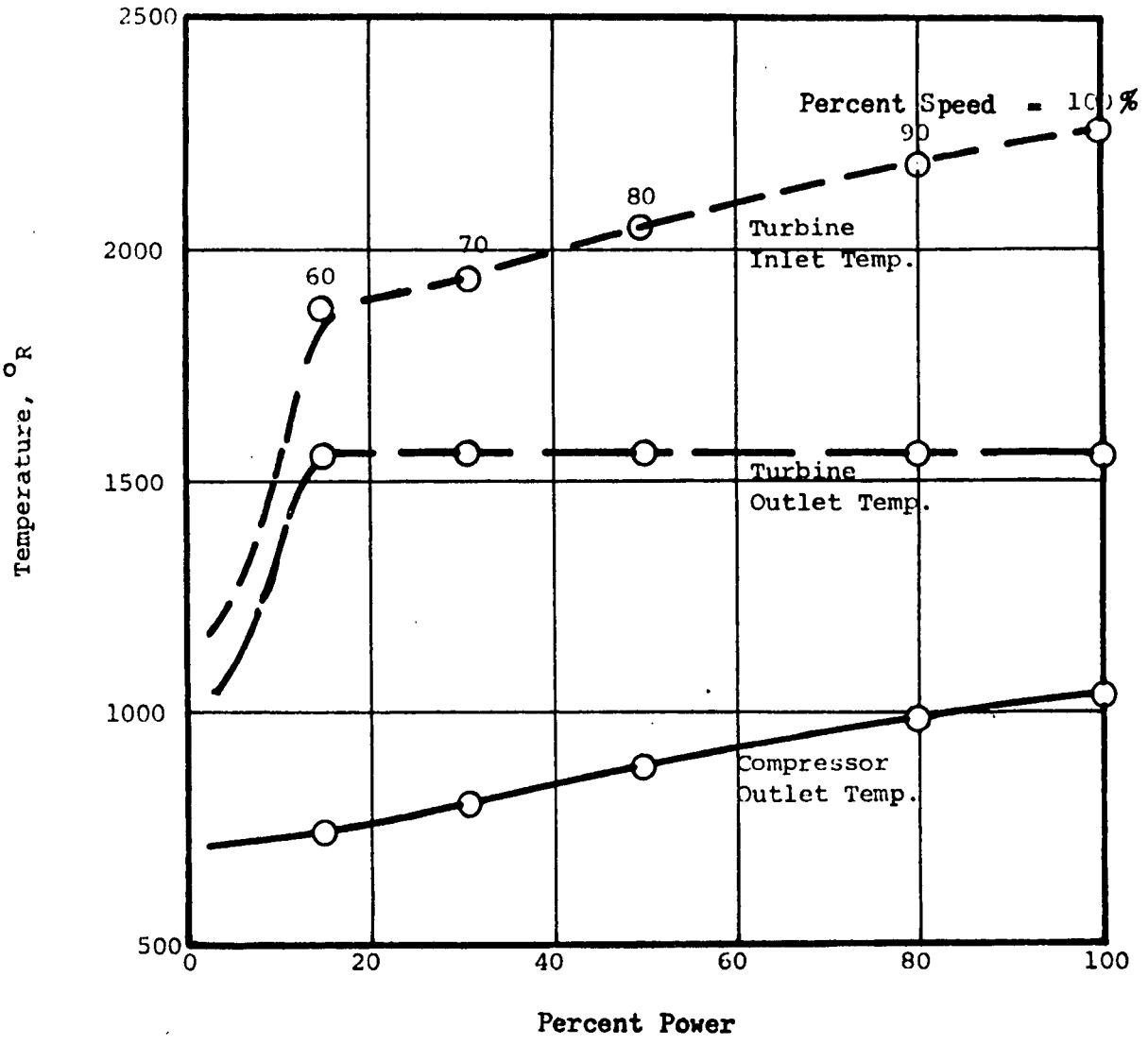


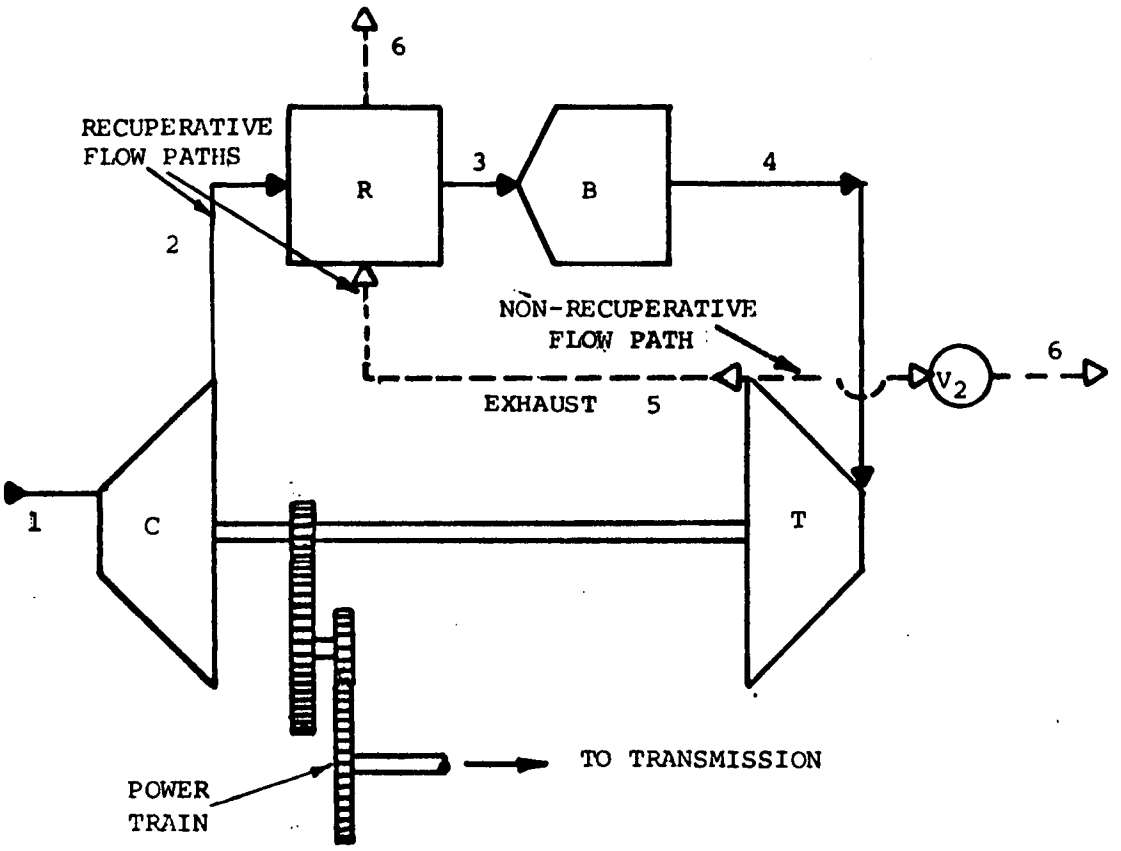
FIGURE 5.23 Compressor Outlet, Turbine Inlet and Turbine Outlet Temperatures as a Function of Power for the Single-Shaft Turbine.

regenerative (or bypass) study presented here. Of course, the differences do affect the economic feasibility of the engines, as will be shown in the next section. Thus, for simplicity, a single-shaft, one stage radial turbine has been assumed for this analysis.

For the bypass concept, the valve is envisioned as being similar to a waste gate butterfly valve used in certain turbochargers. Valve operating temperatures of over 1000°F will be encountered, which will require the use of high alloy steels. Valve leakage will not be a serious problem due to the fact that the gases which leak are merely routed through the alternate passage (bypass duct or heat exchanger) with only slight effects on powerplant performance. The most probable method of actuating the bypass valve will be by a hydraulic servo associated with the vehicle transmission, although solenoid or torque motor actuation is also a possibility. A schematic of the bypass engine concept is shown in Figure 5.24. This sketch does not show a bypass valve on the high pressure regenerator path and, as will be shown later, this is due to the fact that the pressure drop through this path of the regenerator remains relatively constant for all power levels. The three heat exchanger designs and the resulting powerplant performance and design characteristics will not be presented as evaluated both with and without the bypass concept.

5.4.1 Tubular Recuperator Concept

The first heat exchanger type evaluated was the well-proven tubular design using a multipass crossflow configuration. It was determined early in the study that a full time recuperator of this type designed for operation at all power levels (160 BHP maximum) would be prohibitively large. To keep the heat exchanger effectiveness and pressure drops at an acceptable level required a heat exchanger with a total volume of about 3 ft^3. In addition, since the heat exchanger cost is related to its size, weight, and complexity, the cost of this full time recuperator would certainly prohibit its use in an automotive application. Therefore, the only solution seemed to be to design a compact tubular recuperator for operation only at the low power levels and to bypass it at the high power levels. In this way, the powerplant becomes a combination of a simple cycle gas turbine for operation at high power levels with a regenerative gas turbine cycle for operation at low power levels.



C - compressor
R - recuperator

B - burner
T - turbine
V_2 - exhaust bypass valve

FIGURE 5.24 Schematic Flow Path Diagram for the By-Pass Turbine Engine.

The tubular recuperator that was found to provide the best powerplant performance in the most compact package is shown in Figure 5.25 with the pertinent heat transfer characteristics given in Figure 5.26. This recuperator, a 3-pass crossflow arrangement, was designed for the flow rate corresponding to 60% speed and an effectiveness of about .70. A trade-off between size and performance resulted in hot and cold side pressure drops ($\triangle P/P$) of about .05 at the design point. To determine the recuperator and hence powerplant performance under all power conditions it was necessary to determine the recuperator effectiveness and pressure drops as a function of power output. These characteristics are shown in Figure 5.27. It becomes evident from this figure that the bypass valve is only required on the hot (or low pressure exhaust) side. This is due to the fact that as the mass flow rate through the system goes up, so does the cold side absolute pressure, thereby keeping $(\triangle P/P)_C$ relatively constant. However, the absolute pressure on the hot side remains the same and consequently the $(\triangle P/P)_H$ on that side goes up almost as the square of the flow rate. These recuperator characteristics were incorporated into the turbomachinery cycle analysis and the resulting powerplant performance characteristics are presented in Figures 5.28 and 5.29.

The curves on these figures represent the brake horsepower and brake specific fuel consumption values for the simple cycle, the recuperative cycle and the combination simple-recuperative, or bypass, cycle. The advantage of the bypass cycle is evident from the fact that the fuel consumption for this cycle at 10% power is about 1.0 lbm/hp-hr, while at 100% power it is about .67 lbm/hp-hr. The fuel consumption for the simple cycle at 10% power is almost 2.0 lbm/hp-hr, or about twice that of the bypass recuperative cycle, while the fuel consumption at 100% power is about the same. As mentioned previously, the bypass valve would be similar to a waste gate butterfly and it is apparent from Figure 5.29 that the best operating schedule would be for the valve to begin to open at about 50% power and be completely open at approximately 85% power. Only the exhaust gas side of the recuperator is bypassed and the resultant power loss due to the full power flow through the air side of the recuperator is about 7 HP (Figure 5.28).

These results indicate that the tubular recuperator bypass concept forms the basis for a compact, low-cost powerplant with good performance

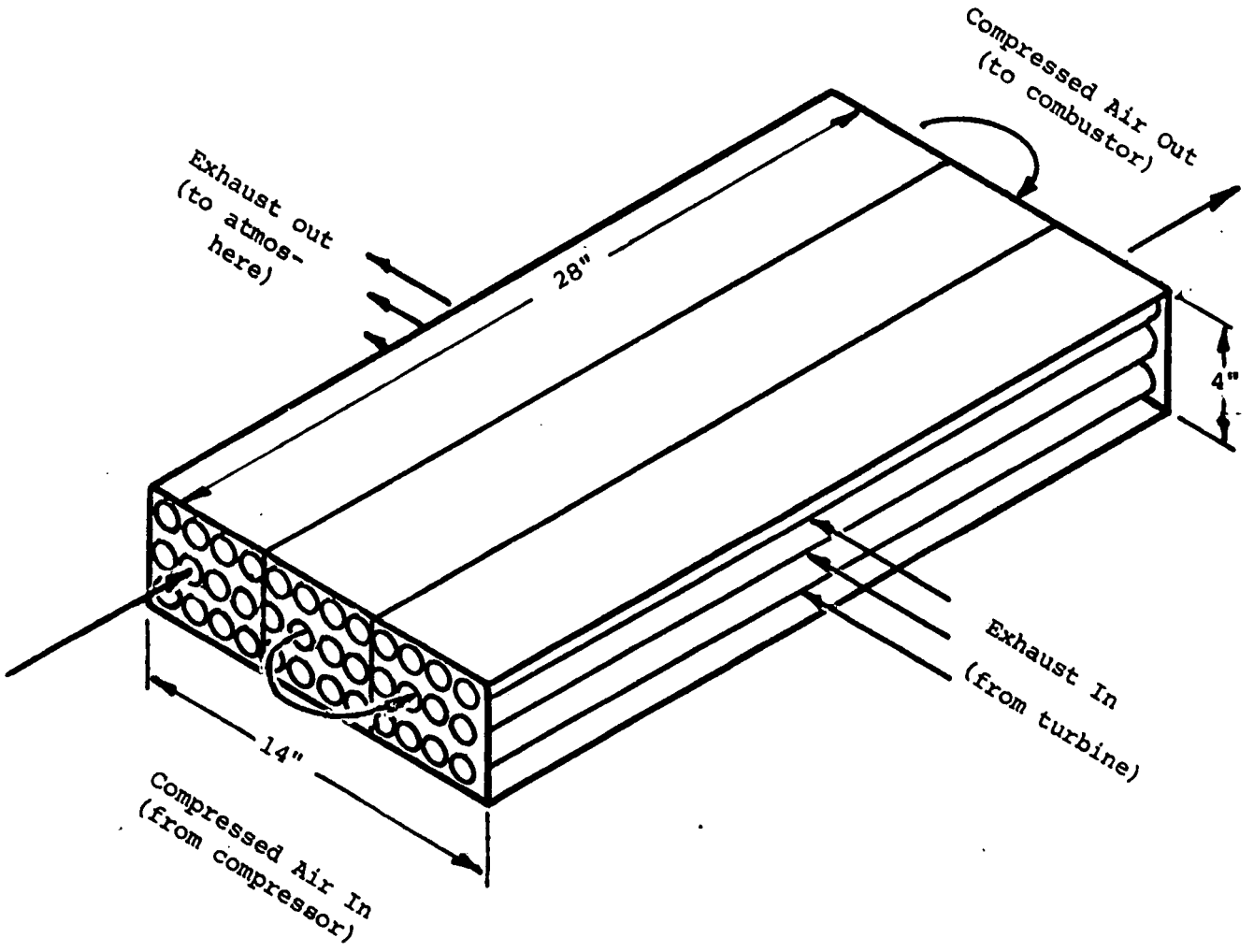


FIGURE 5.25 Bypass Gas Turbine Recuperator

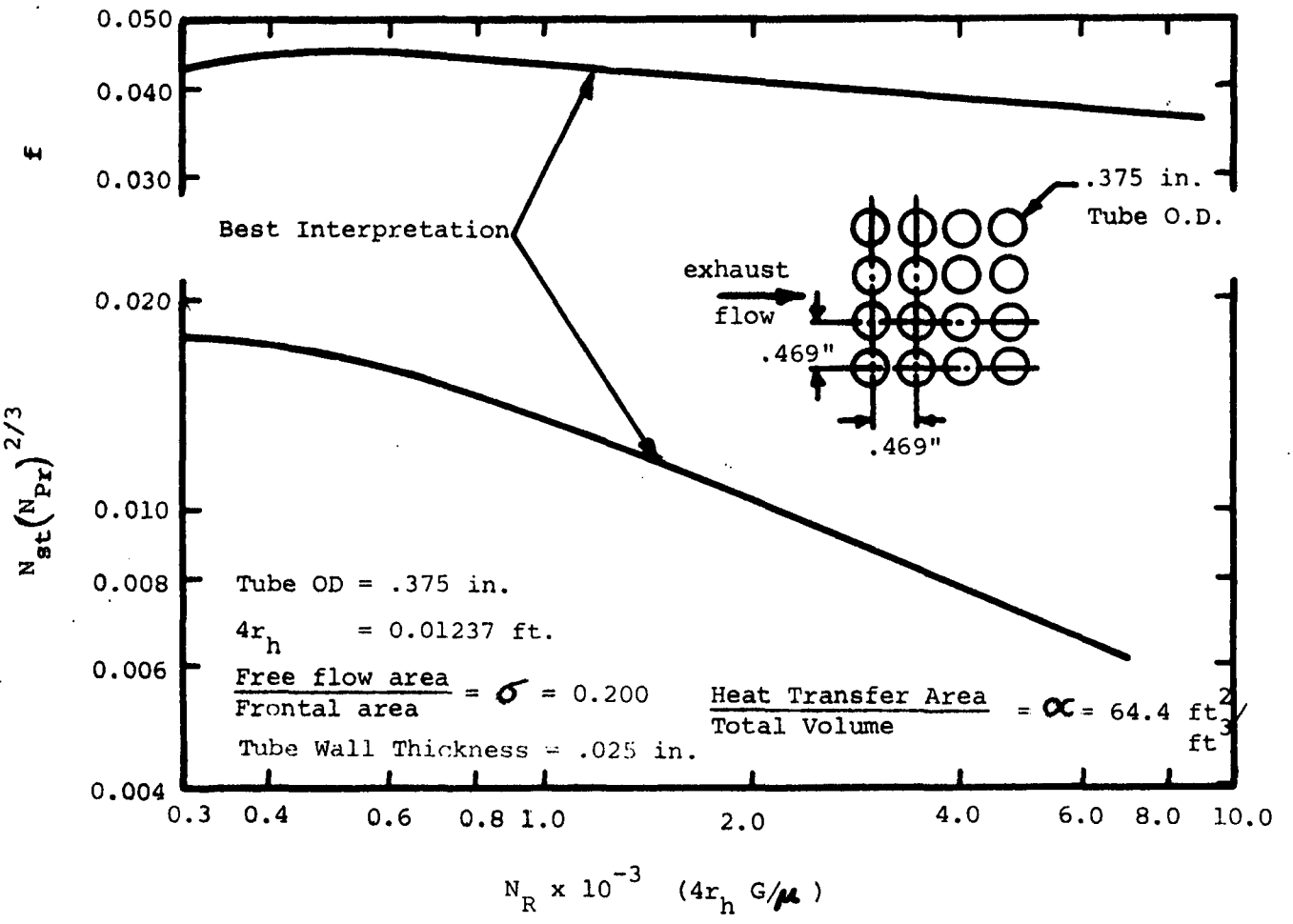


FIGURE 5.26 Bypass Gas Turbine Recuperator Characteristics.
Exhaust Gas Side Heat Transfer Characteristics.
(Ref.: Fig. 10-14, Kays and London).

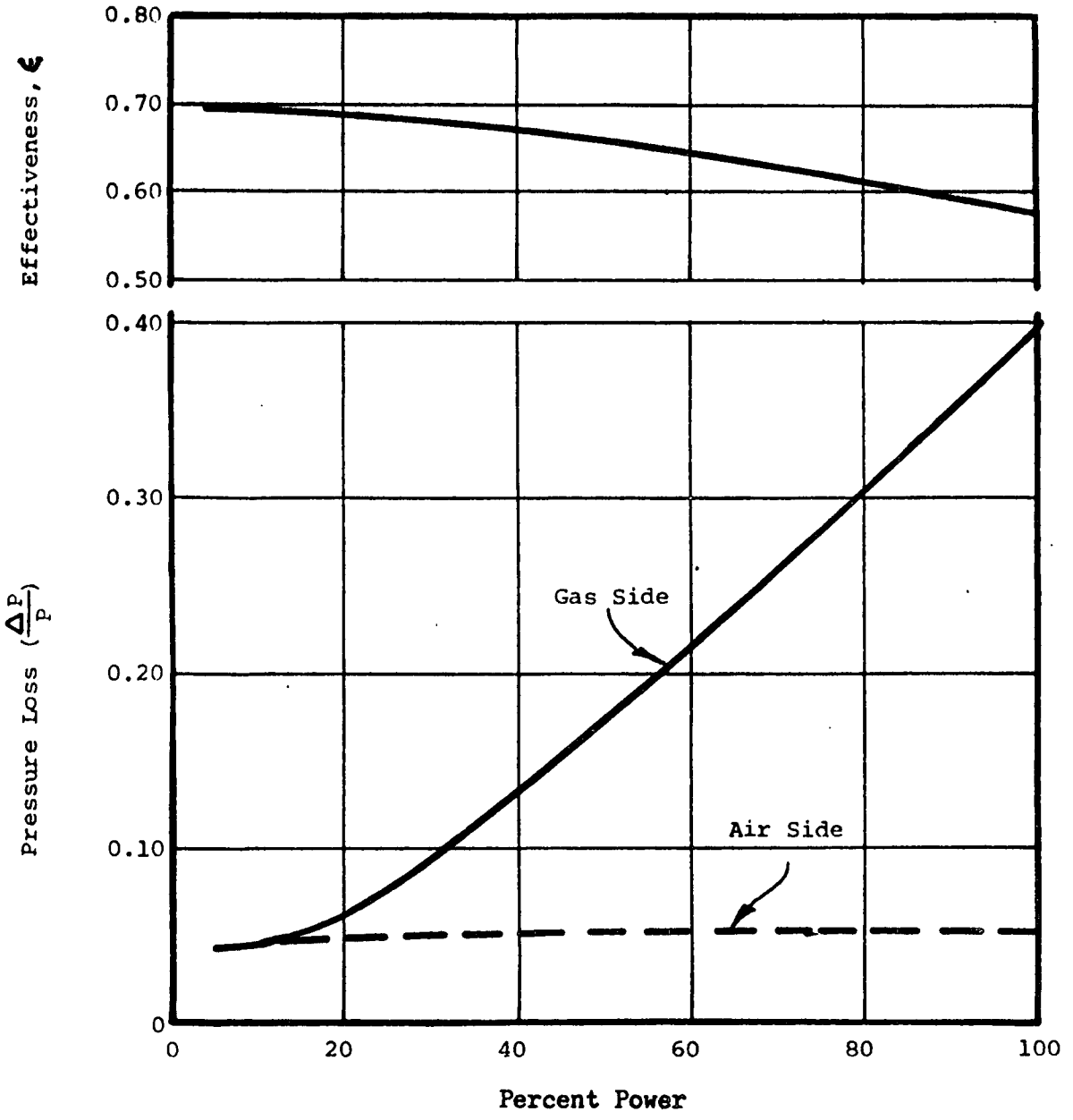


FIGURE 5.27 Pressure Drop and Effectiveness Characteristics for Tubular Recuperator as a Function of Power Output.

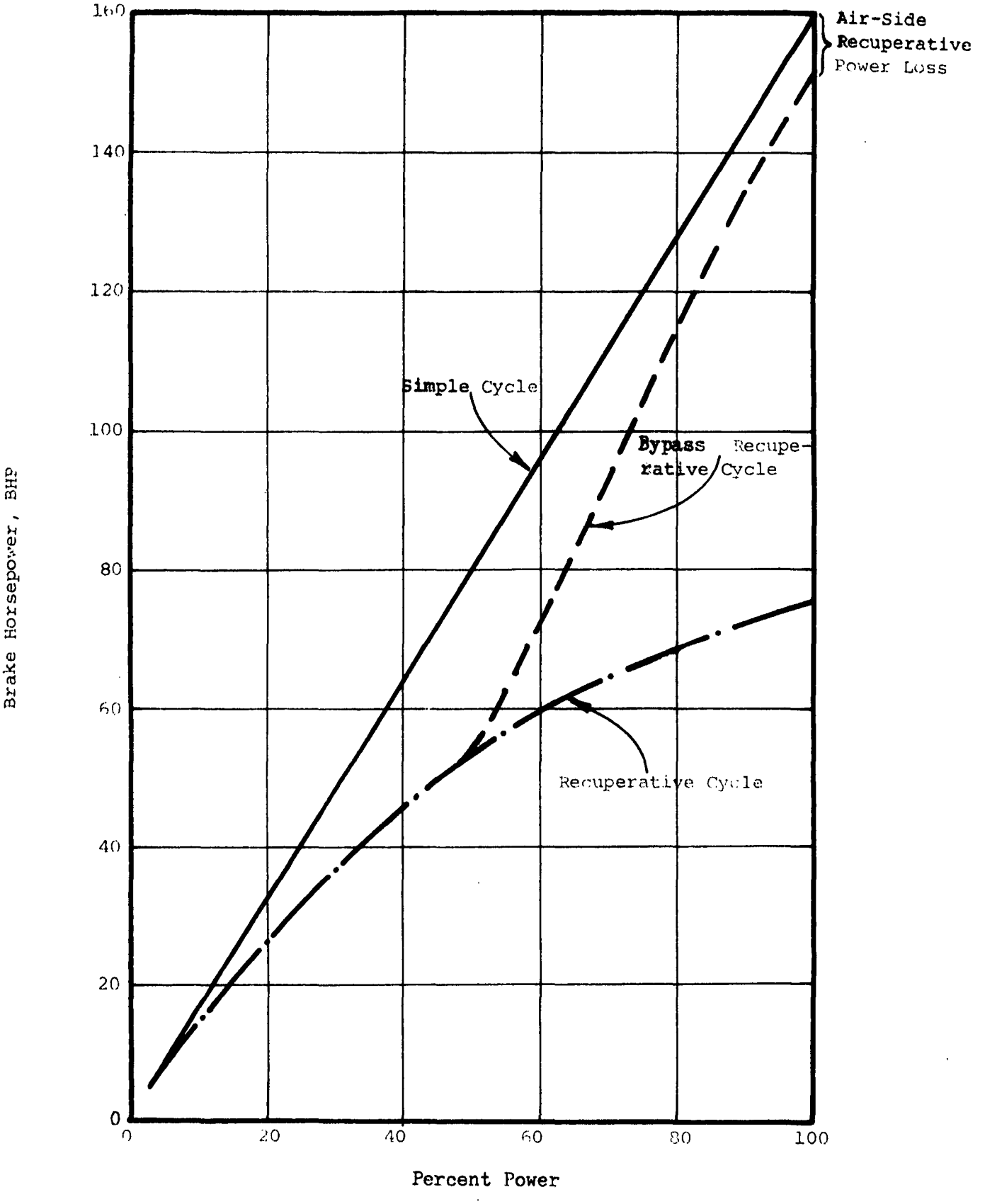


FIGURE 5.28 Power Output Characteristics for Gas Turbine Powerplant with Tubular Recuperator.

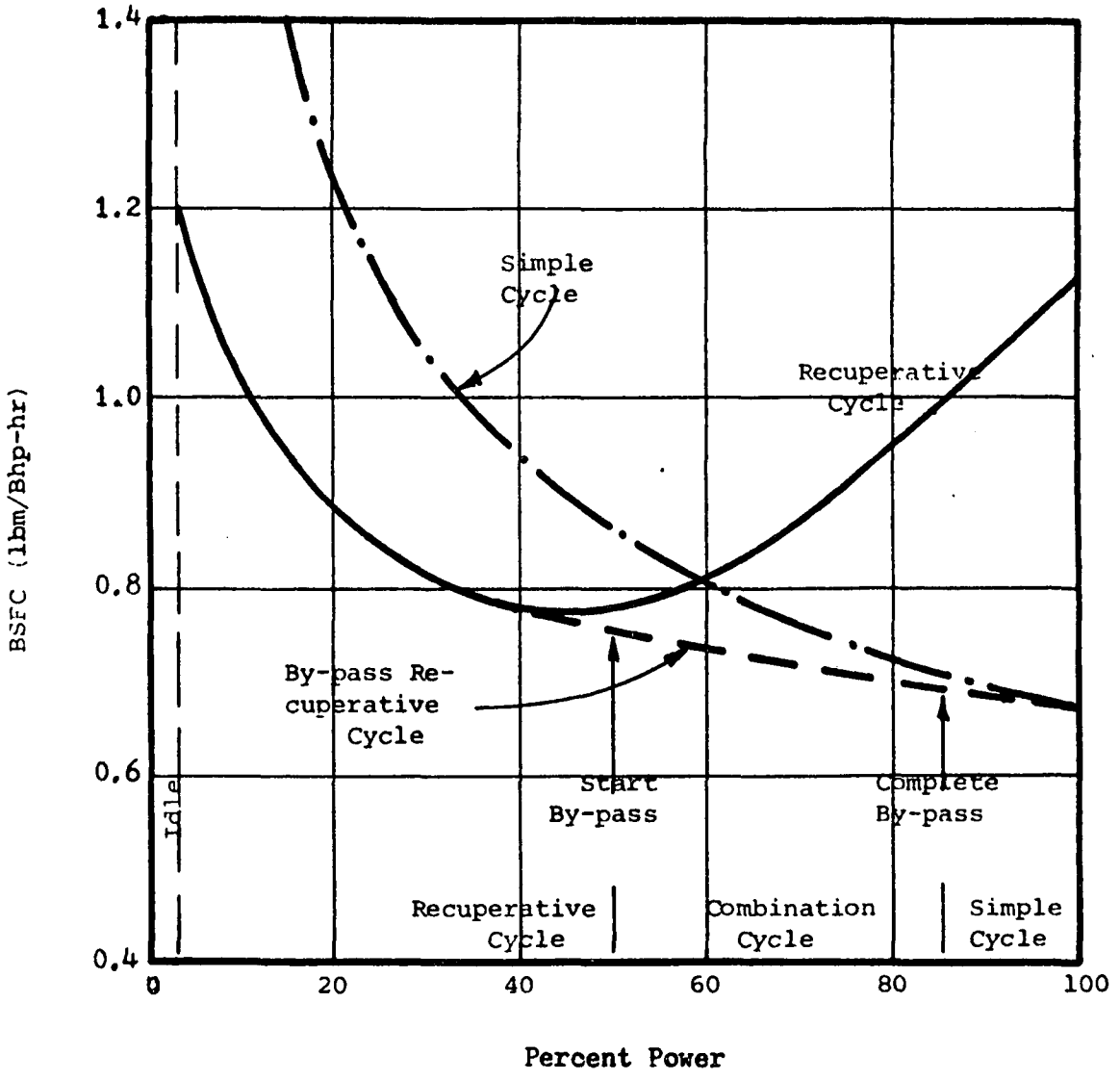


FIGURE 5.29 BSFC Characteristics for Gas Turbine Powerplant with and without Tubular Recuperator.

for all power conditions. It should be noted here that the cycle calculations were based on conservative, present state-of-the-art turbomachinery parameters as described in Section 5.2, and that improved BSFC characteristics are possible by using a less concervative approach. For example, improved fuel consumption characteristics are realizable by increasing either the turbine inlet temperature or the maximum compressor pressure ratio. However, it was felt that 1800°F for maximum turbine inlet temperature was a realistic value and that the maximum compressor pressure ratio of 6.3 was near enough to the present state-of-the-art for a one stage radial compressor. For the bypass concept the argument can also be made for using a higher design compressor pressure ratio. By assuming a two-stage radial compressor with backward swept blades a compressor with improved surge margin can be obtained. Such a compressor with improved surge margin would allow part power operation at higher turbine inlet temperatures than the single stage design, and these higher temperatures require higher pressure ratios for optimum cycle performance. This higher pressure ratio design would provide better fuel consumption at both full power (without the recuperator) and part power (with the recuperator). The bypass tubular recuperator design with an 8:1 compressor pressure ratio showed a BSFC of about .60 lbm/hp-hr for power levels ranging from 40% to 100% and a very attractive BSFC of about .70 lbm/hp-hr at 10% power. However, preliminary investigations indicated that the one stage radial (with maximum pressure ratio of between 6 and 7) provided the lowest cost, most compact powerplant design. Therefore, a selection of the single stage radial design was made in which advantages in powerplant size and cost were obtained at the expense of fuel consumption.

5.4.2 Plate-Fin Recuperator Concept

The next heat exchanger type to be evaluated was the metal, plate-fin recuperator. This recuperator concept required extensive investigation due to the large number of matrix types available and the corresponding large differences in friction factor and heat transfer coefficients. Plain, louvered, strip-fin, wavy-fin, and pin-fin surfaces were investigated with respect to overall effect on powerplant size and performance; the results indicated the strip-fin surface to be somewhat superior to the others. This surface, therefore, was used as the basic matrix for both full-time and part-time (bypass) recuperators for the gas turbine cycle. All cycle parameters

(turbine inlet temperatures, compressor pressure ratios, etc.) were the same as those used for the tubular recuperator analysis.

The strip-fin, plate-fin recuperator matrix is based on the data of Figure 10-61 of Reference 2, presented here in Figure 5.30. This particular strip-fin matrix was chosen due to its characteristic fin pitch (18.82 fins/inch) and plate spacing (.205 inch) which result in a favorable heat transfer coefficient and an acceptable friction factor. Using this type matrix the arrangement which gives the highest effectiveness is a straight-through counterflow as shown in Figure 5.31. As shown there, the hot side free flow area is twice as large as the cold side area which results in more equalized pressure drops for both sides of the heat exchanger. This flow distribution maintains Reynolds numbers in the laminar (or sometimes transition) regime for all operating conditions for both streams.

Several size heat exchangers using the aforementioned counterflow matrix were examined. As for the tubular recuperator analysis, the main objective here was to determine the most compact heat exchanger-turbomachinery arrangement having the best performance. To do this, several heat exchanger sizes were investigated and the resulting powerplant performance calculated. To determine the recuperator, and hence, powerplant performance under all operating conditions it was necessary to determine the recuperator effectiveness and pressure drops as a function of power output. These characteristics are shown in Figures 5.32 and 5.33 for a representative selection of possible recuperator designs (for both full time and bypass operation). These recuperator characteristics take into account the varying compressor and turbine outlet temperatures as a function of power output as shown in Figure 5.23. Once again it is evident from the pressure drop characteristics that the bypass valve is only required on the hot (or low pressure exhaust) side. These recuperator characteristics were then incorporated into the turbomachinery cycle analysis and the resulting powerplant performance characteristics are presented in Figures 5.34 and 5.35.

Examining Figures 5.32 through 5.35 in some detail brings out some of the more important characteristics of the plate-fin type of heat exchanger. First of all, Figure 5.32 shows that as the heat exchanger frontal area is increased, the exhaust side pressure drop decreases in the same proportion (for heat exchangers having the same length). This is due to the fact that the pressure drop is a function of the volume flow rate, which in turn is

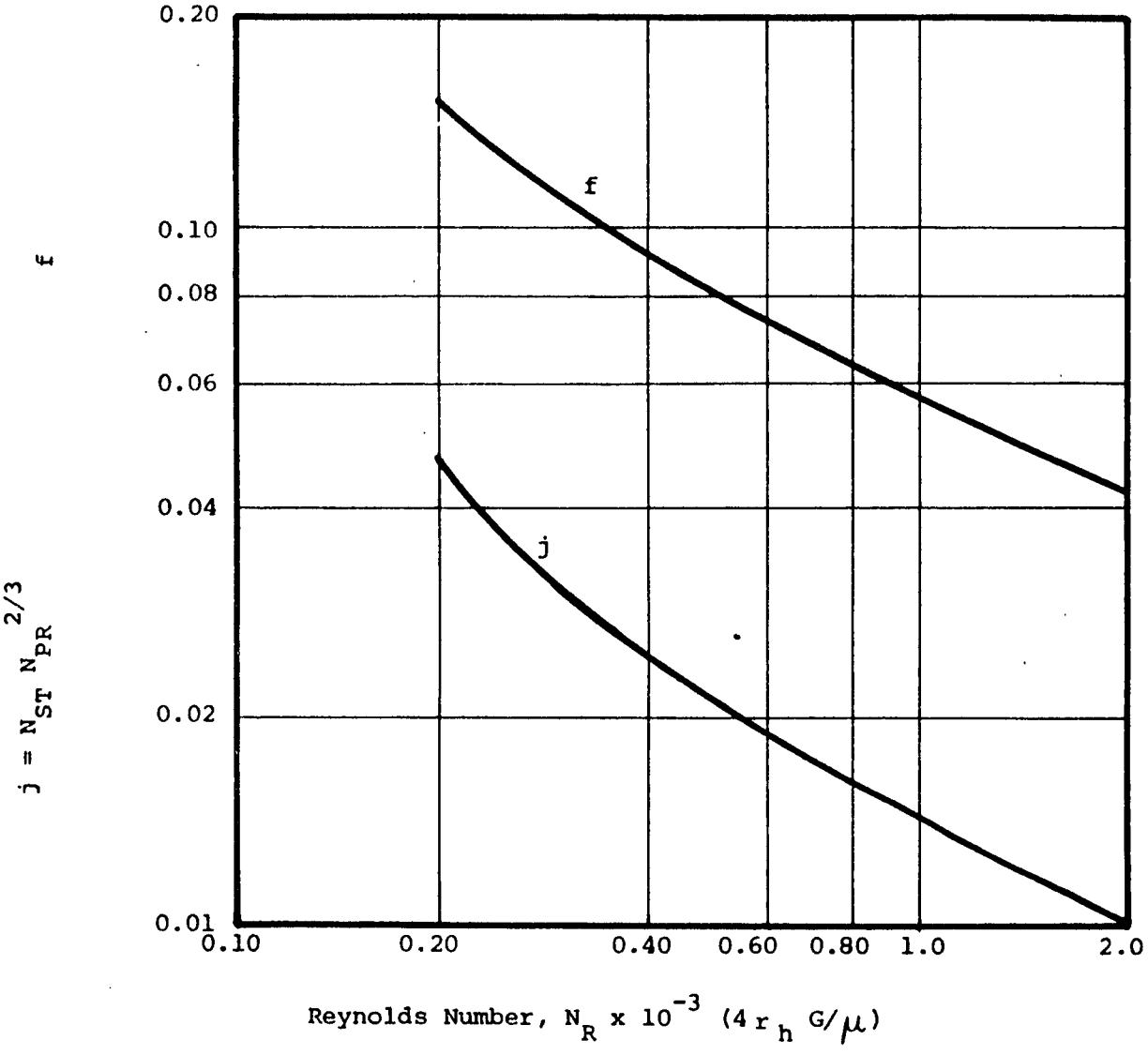


FIGURE 5.30 Heat Transfer and Friction Characteristics for Plate fin Recuperator Matrix.

Fin pitch = 19.82/in.

Plate Spacing, b = 0.205 in.

Symmetrical Splitter

Fin Length flow direction = 0.125 in.

Flow passage hyd. diam., $4 r_h$ = 0.005049 ft.

Heat transfer area/Vol. between plates, B=680 ft^2/ft^3

Fin thickness = 0.004 in., nickel

Splitter thickness = 0.006 in.

Fin area/total area = 0.841

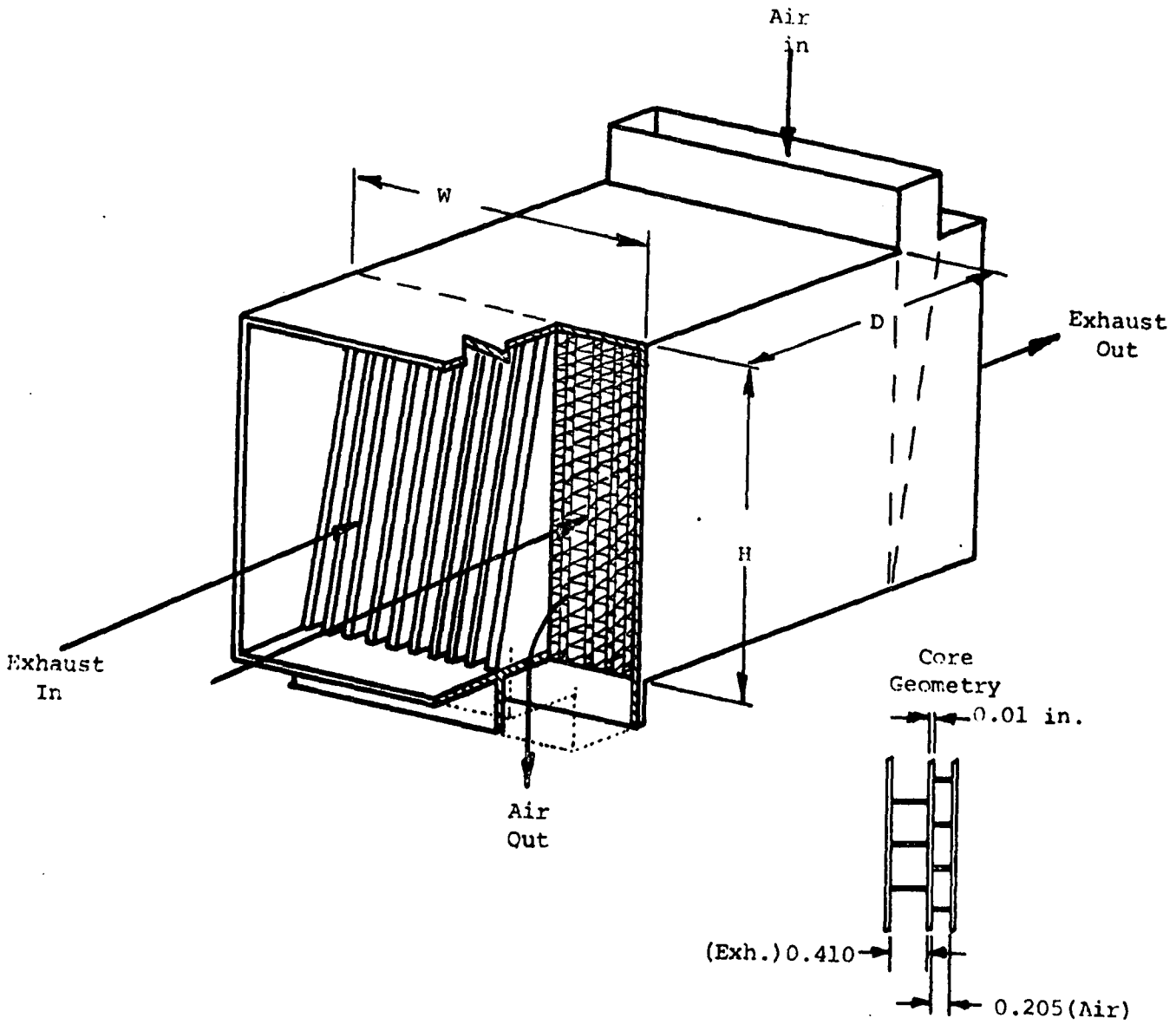


FIGURE 5.31 Gas Turbine, Plate-fin Recuperator.
Counterflow, Strip-fin, Plate-fin Heat Exchanger.
Surface = 1/8 - 19.82 D

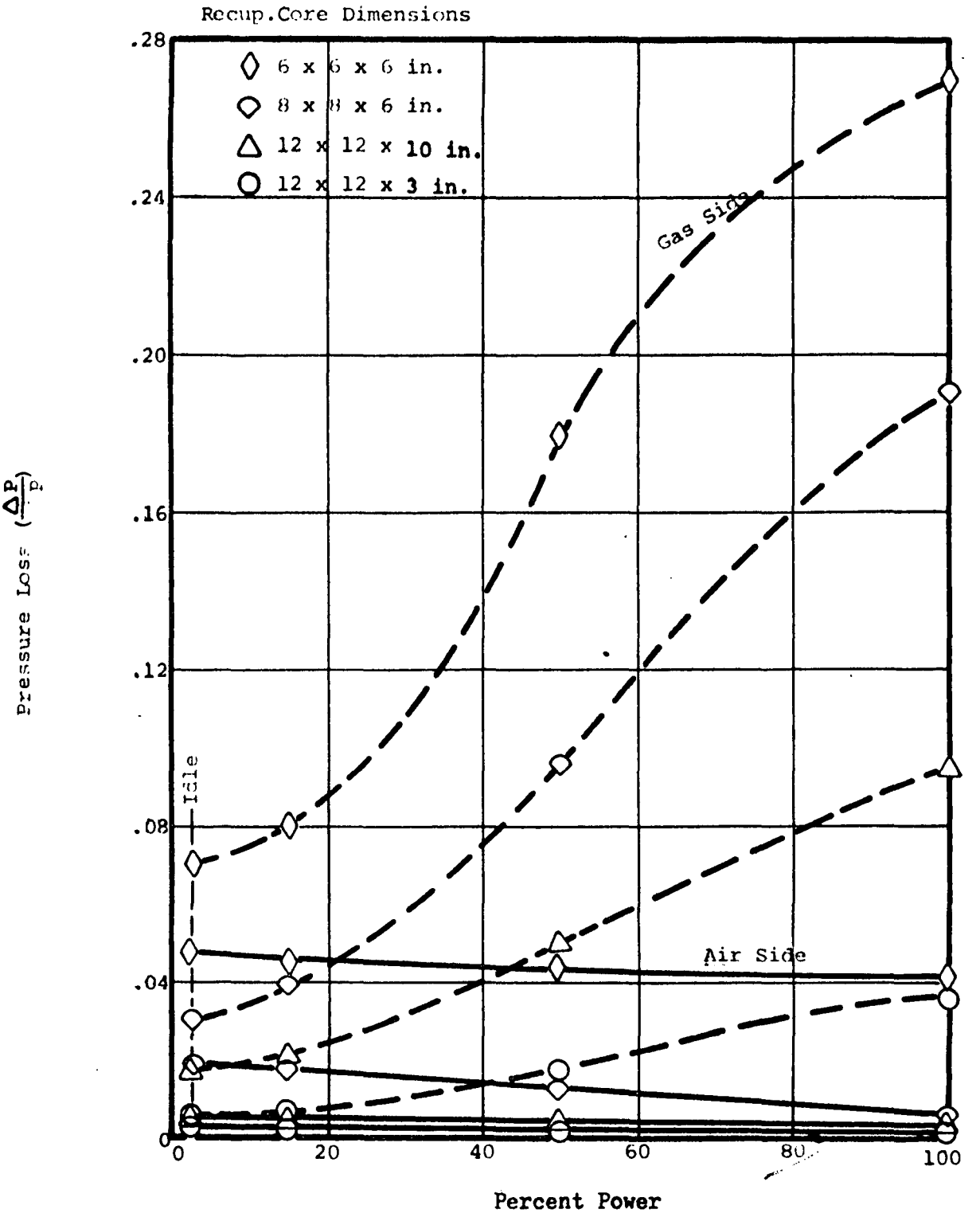


FIGURE 5.32 Pressure Loss Characteristics for Plate-fin Recuperators as a Function of Power Level.

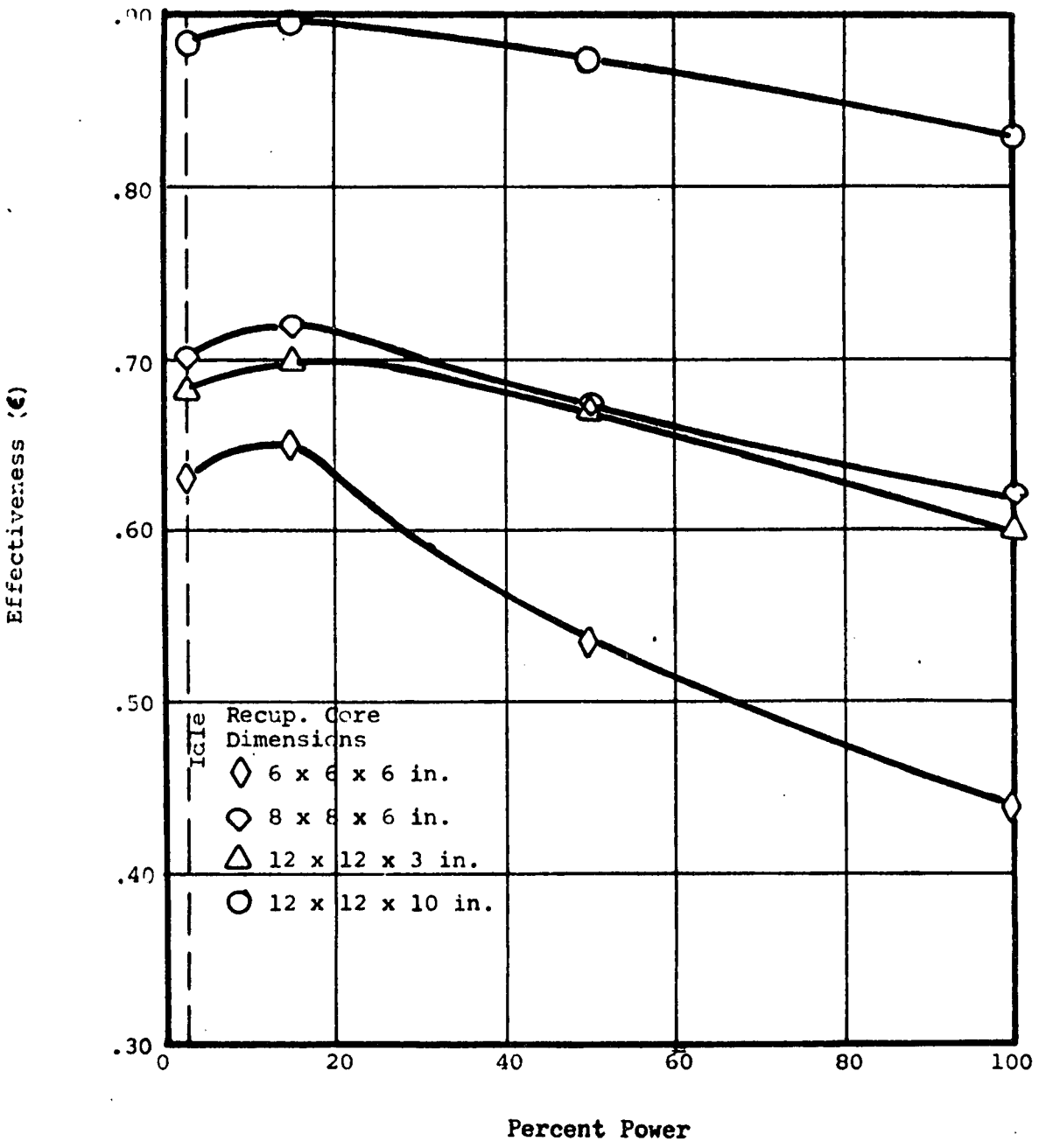


FIGURE 5.33 Effectiveness Characteristics for Plate-fin Recuperators as a Function of Power Level.

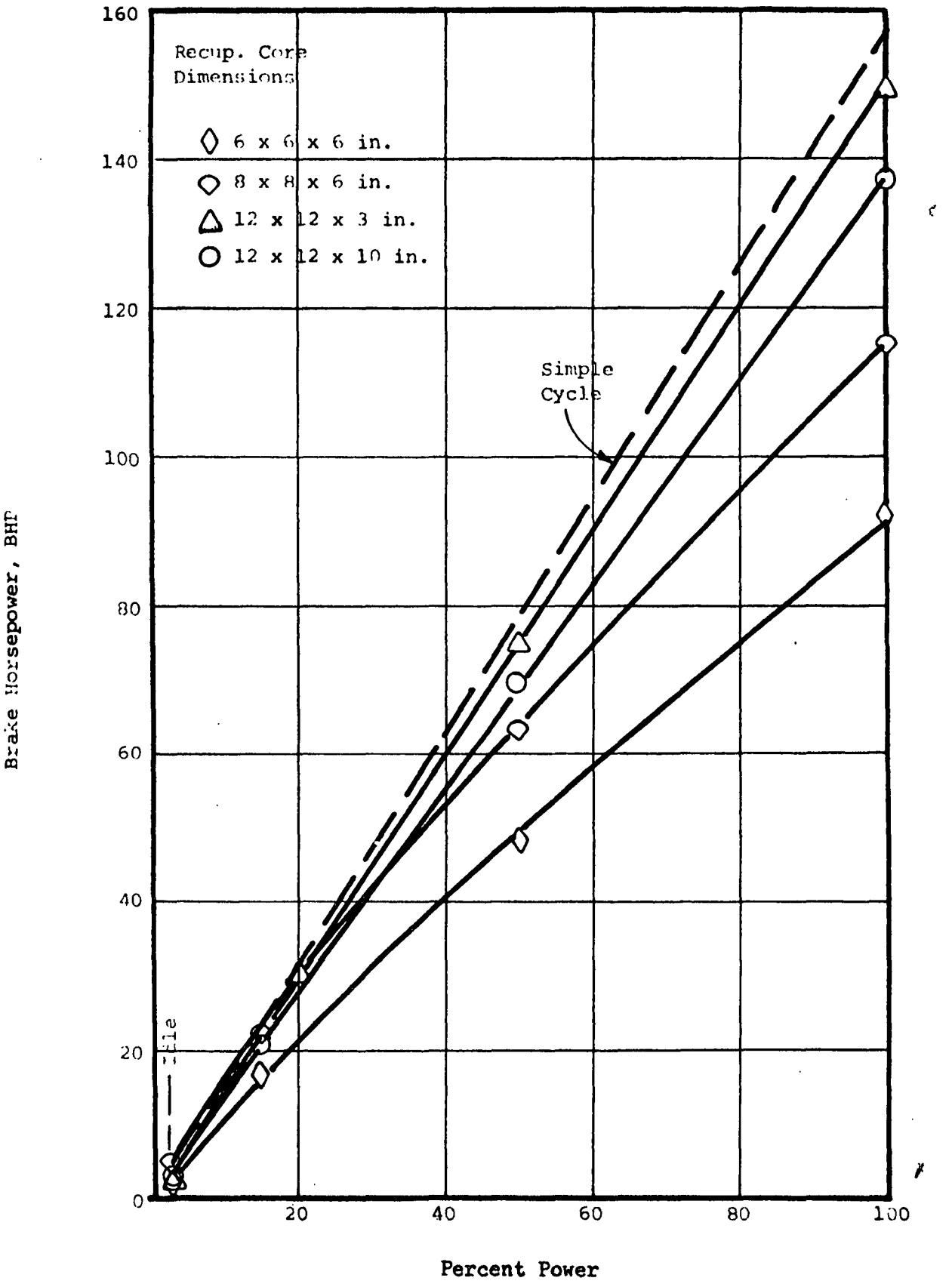


FIGURE 5.34 Power Output Characteristics for Gas Turbine Powerplant with Various Plate-fin Recuperators.

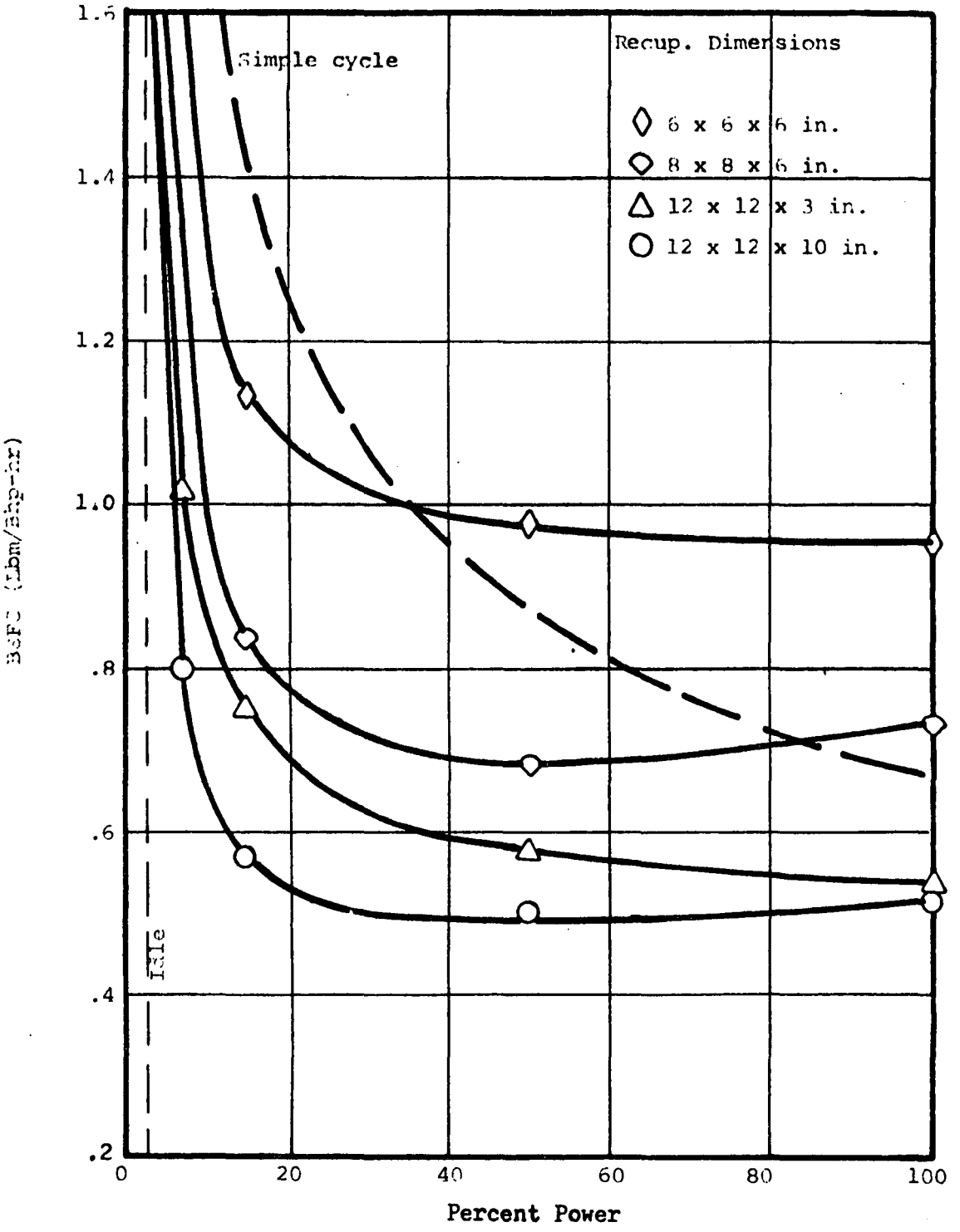


FIGURE 5.35 BSFC Characteristics for Gas Turbine Powerplant with Various Plate-fin Recuperators.

a function of the frontal cross sectional area. Figure 5.33 shows that the larger the volume of the recuperator (all having the same strip-fin matrix), the greater is the resulting effectiveness. Thus, the 12" x 12" x 10" recuperator has the highest effectiveness value (.90 at 15% power). The recuperators chosen given a wide spread of powerplant performance characteristics as shown in Figures 5.34 and 5.35. Figure 5.34 shows the power penalty associated with the various recuperator designs for the entire power range. The recuperators having the largest pressure drops experience the greatest power losses. It is apparent that to obtain full power with any of the more compact recuperators (6" x 6" x 6", 8" x 8" x 6") requires bypassing the recuperator at or near the 100% power level. In addition, Figure 5.35 shows the fuel consumption penalty due to operating the more compact recuperators at or near full power.

To examine the results presented in Figures 5.34 and 5.35 and to conclude that one recuperator design is superior to the others for all gas turbine powerplants would be a very meaningless task. One must consider not only the core size of the recuperator itself, but the size of the overall powerplant in conjunction with that recuperator. Such things as manifolding, diffuser lengths, control valves, bypass ducts, etc., will greatly affect both the size and cost of the total powerplant. For example, the turbine diffusion ducting required for the 12" x 12" face area recuperator (144 sq. in.) is necessarily wider and somewhat longer than that required for the 8" x 8" face area recuperator (64 sq. in.). It is these things which must be weighed against the performance curves of Figures 5.34 and 5.35 to determine the best powerplant with respect to size, cost, and performance. With these factors in mind, the various recuperator-turbine powerplant combinations were analyzed and found to exhibit the following characteristics.

The 6" x 6" x 6" recuperator powerplant experiences a large power loss throughout the entire power range due to the large pressure drops on both the air and gas side of the recuperator. These high power losses in combination with the recuperators relatively low effectiveness result in very poor BSFC characteristics. The BSFC characteristics are only slightly better than the simple cycle up to about 35% power and then become much worse. It would appear that this very compact recuperator does not improve

the simple cycle fuel consumption enough to warrant its use with or without the bypass concept.

The 8" x 8" x 6" recuperator powerplant experiences a moderate pressure loss on the gas side at the high power levels while the air side experiences an insignificant loss. The total pressure loss at full power (without the bypass) is such that the simple cycle power output is decreased by about 45 HP. The effectiveness of this recuperator is high enough to produce favorable BSFC characteristics at all power levels. The BSFC for the bypass recuperative concept is significantly better than the simple cycle for the lower power range (100% better at 10% power), and approximately equal to the simple cycle for the higher power range. Therefore, it appears that the bypass concept is attractive for this recuperator-powerplant combination. A minimum BSFC of about .70 lbm/hp-hr could be realized from about 30 to 100% power with a full power capability of about 155 HP. Only the exhaust gas side of the recuperator would be bypassed (at about 80% power), the power loss on the air side being only about 2 HP at full power. The 75% increase in frontal area (and total volume) of this recuperator over the previous one (6" x 6" x 6") has increased the powerplant performance significantly while increasing the total powerplant volume (and cost) only slightly.

The 12" x 12" x 3" recuperator powerplant exhibits very favorable performance characteristics and does such without the need for a bypass. The power loss at full power is only about 7 HP which could be regained most easily by slightly increasing the size of the turbomachinery. The BSFC is below .60 lbm/hp-hr from about 40% to 100% power and is only .85 lbm/hp-hr at the 10% power level. This is a great improvement over the simple cycle performance; however, this improvement is paid for by the larger powerplant volume. The increase in volume is due not only to the core volume of the recuperator but also the necessary counterflow manifolding and even more important the required diffusing duct from the radial turbine to the recuperator exhaust inlet. The outlet area from the turbine is only about 15 sq. inches and the exhaust must pass through this area, then through the necessary diffusing duct, and finally be spread out over the face of the recuperator (144 sq. in.) to insure against hot spots and uneven flow through the recuperator. It seems likely that to obtain full advantage from the recuperator,

vanes would be required to direct the exhaust flow to the outermost corners of the face. Of course, if the exhaust is diffused properly (to a Mach number of about .2) the exhaust would tend to fill the entire recuperator face area due to the back pressure caused by the recuperator. However, at low speeds where the back pressure is not large, the flow might not distribute itself evenly throughout the recuperator and the heat exchanger effectiveness could suffer severely. These are considerations which must be taken into account as they may significantly influence both the size, performance, and cost of the powerplant.

Finally, the 12" x 12" x 10" recuperator powerplant undoubtedly gives the best fuel consumption due to its high effectiveness; however, it suffers about a 20 HP loss at full power as compared to the simple cycle due to its length and the resulting pressure losses. This recuperator is much larger than the others examined and the good performance is obtained at the expense of size and cost. This design exhibits an almost constant BSFC of .50 lbm/hp-hr from about 20% to 100% power and increased to only .7 lbm/hp-hr at 10% power. The bypass concept would probably not be attractive for this design as the horsepower lost could most easily be regained by merely increasing the airflow size of the turbomachinery slightly. Both full time and bypass concepts would require an exhaust diffusion duct and flow guiding vanes similar to the 12" x 12" x 3" recuperator design.

In general, therefore, it appears that the recuperative, plate-fin gas turbine exhibits the potential for a compact, high efficiency automotive powerplant. Any of the recuperators examined (except for perhaps the 6" x 6" x 6" design) are feasible and the particular choice depends on the trade-off between size, cost and performance. For best fuel economy, the obvious choice would be the 12" x 12" x 10" design with no bypass, however, this would dictate a larger powerplant volume, weight, and cost. The most compact design with the lowest initial cost would be the 8" x 8" x 6" recuperator with an exhaust gas bypass. This combination offers advantages with respect to size, weight, and cost while giving somewhat poorer fuel consumption than could be obtained with the larger heat exchanger designed for full power operation without the bypass feature.

A schematic design of a single-shaft, bypass recuperative turbine employing the 8" x 8" x 6" plate-fin recuperator is shown in Figure 5.36.

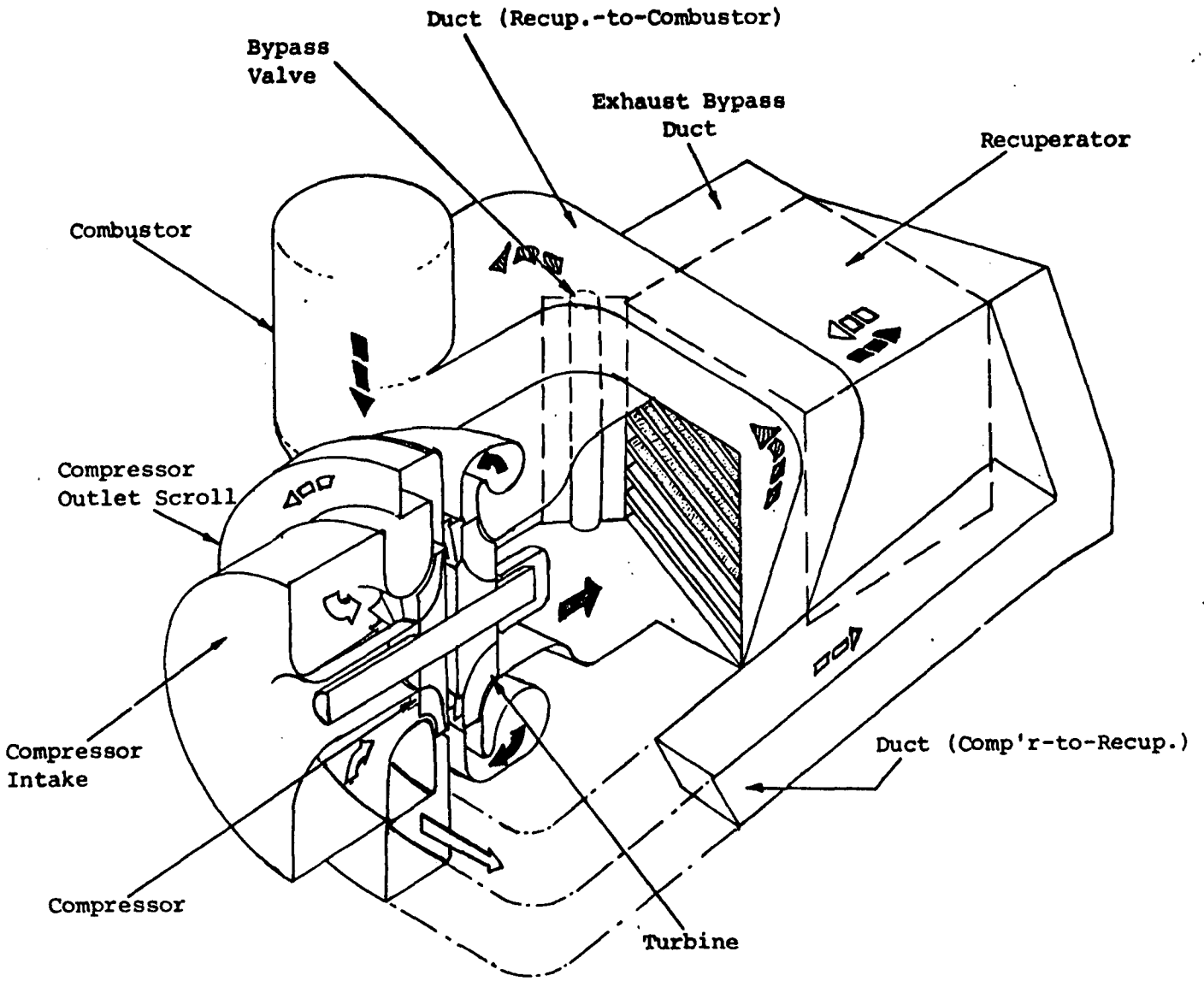


FIGURE 5.36 Schematic of a Single-Shaft Bypass-Recuperative Gas Turbine Engine.

This engine would require about 2.50 cubic feet of envelope volume with a one speed reduction gear box. The components and assemblies included in this volume are:

1. Radial compressor (4.75 in. tip dia., 6.3:1 max. press. ratio)
2. Radial inflow turbine (5.25 in. tip dia.)
3. Combustor
4. Speed reduction gear box (18:1)
5. 8" x 8" x 6" recuperator core
6. Exhaust bypass circuit
7. All scrolls and ducting.

As mentioned previously, the bypass valve is a simple "butterfly" valve which would probably be operated as a function of throttle pedal position or could be actuated by automatic controls as a function of shaft speed or pressure level.

The schematic design of the free-turbine bypass recuperative engine employing the 8" x 8" x 6" recuperator is shown in Figure 5.37. In this design, the bypass recuperator has been offset from the engine centerline to simplify the power takeoff. The addition of the axial flow free-turbine assembly and the necessity of two speed reduction gear boxes increases the envelope volume to about 3.60 cubic feet. The components and assemblies included in this volume are:

1. Radial compressor (4.75 in. tip dia., 6.3:1 max. press. ratio)
2. Radial inflow gasifier (4.75 in. tip dia.)
3. Axial flow free turbine (6.6 in. tip dia.)
4. Combustor
5. Speed reduction gear box-drive (18:1)
6. Speed reduction gear box accessory (18:1)
7. Recuperator (8 x 8 x 6 in. core)
8. Exhaust bypass circuit
9. All scrolls and ducting.

To demonstrate the volume advantage of the bypass recuperative gas turbine powerplant, it is compared with an internal combustion engine of comparable output in Figure 5.38. A speed reduction gear box has been included for the single-shaft turbine which reduces the output shaft speed to about 4500 rpm. The six-cylinder internal combustion engine occupies

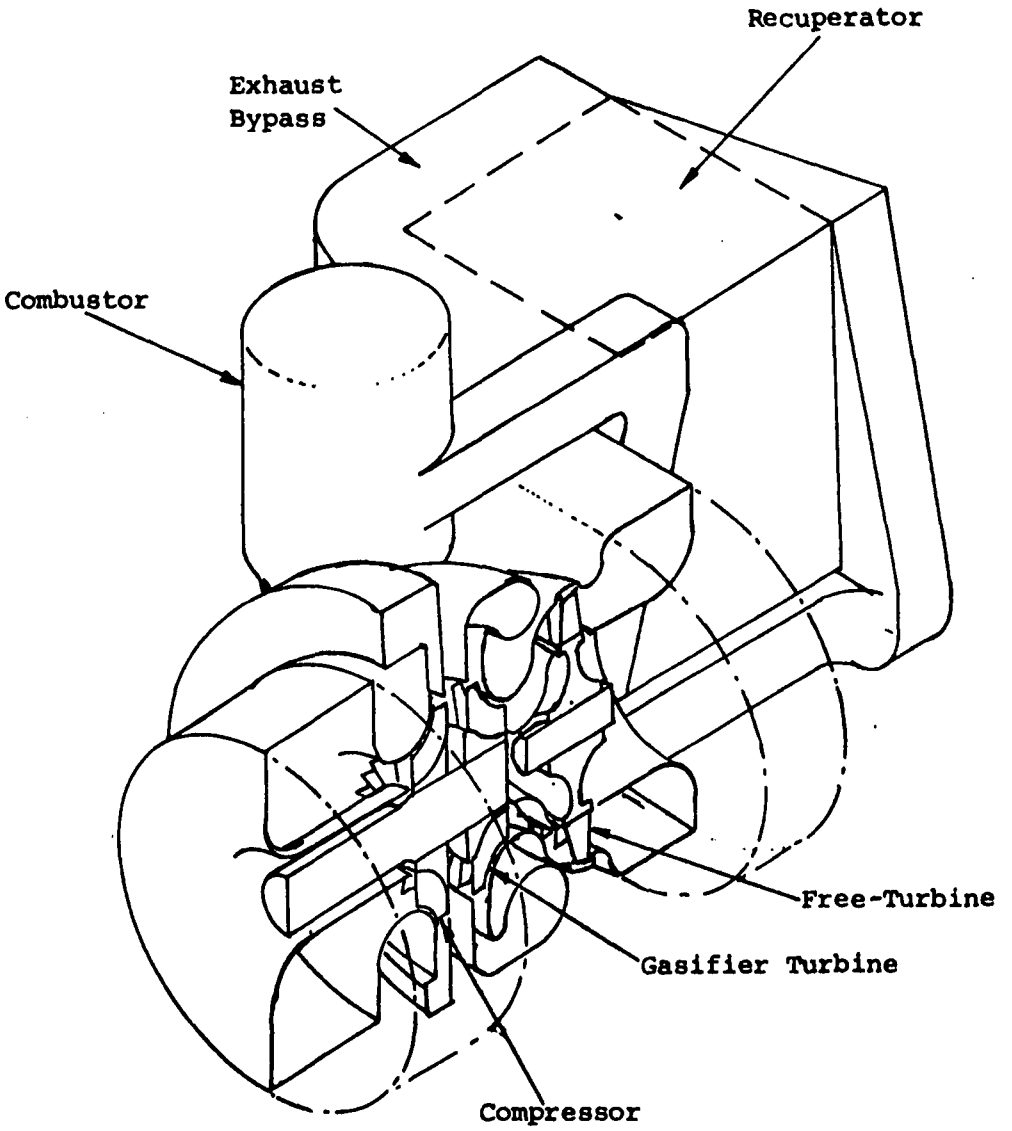


FIGURE 5.37 Schematic of a Bypass-Recuperative Free-Turbine Engine.

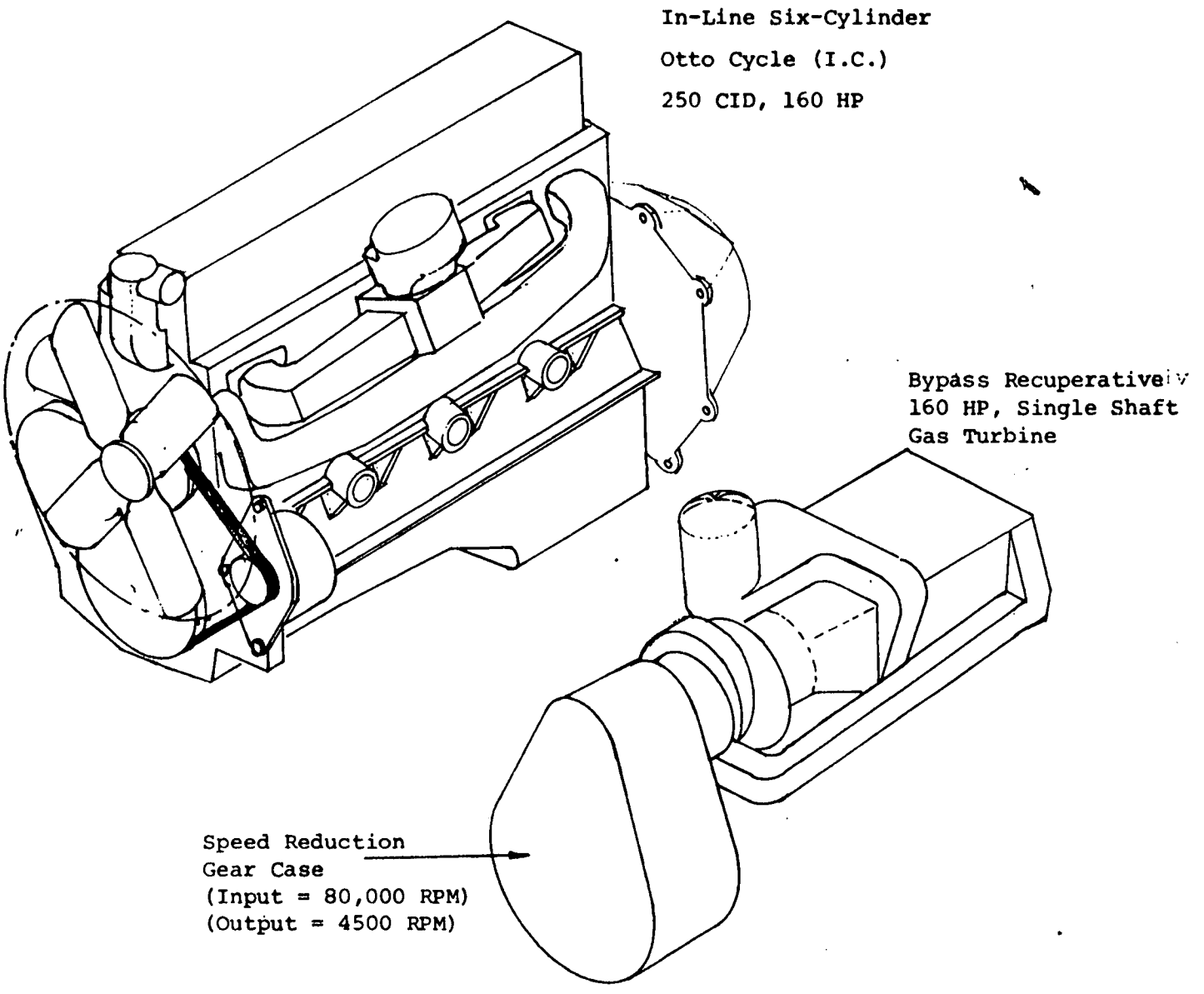
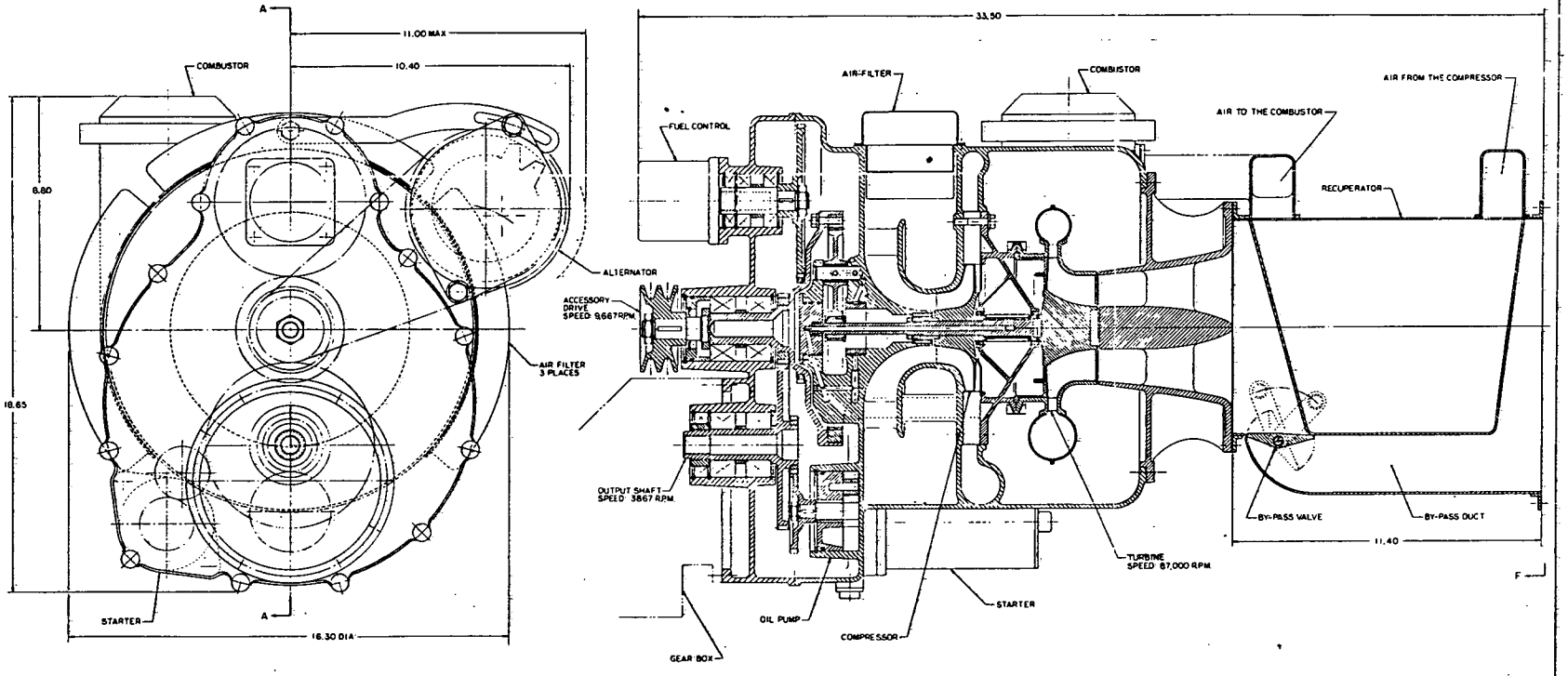


FIGURE 5.38 Comparison of 160 HP Bypass Gas Turbine & Internal Combustion Engines.
(Transmission for the Turbine Engine will be Approximately 1.5 times the Size of that for the Reciprocating Engine.)

approximately 9 cubic feet (at the minimum) as compared to about 2.50 cubic feet for the single-shaft bypass recuperative turbine. The transmission for the single-shaft turbine, however, will be approximately 1.5 times the size of that for the reciprocating engine.

Figures 5.39 and 5.40 are the photo-reductions of the full scale preliminary design layouts for the single-shaft bypass recuperative engine. It should be noted that the scope of the program does not permit detailed designs and therefore prime consideration was given to indicate the application of the bypass recuperator concept. Liberties were taken in turbine designs with respect to actual blade optimization, bearings, lubrication, fuel control systems and any other items which may be considered peripheral to the presentation of the bypass concept. The intent is to indicate a method of improving the broad range performance of those turbines currently under consideration for automotive application, without the addition of excessive heat exchanger envelope volume.

In this, the single-shaft turbine, all power take-off drives have been located forward of the compressor section as indicated in the left side sectional view of the engine in Figure 5.39. This arrangement facilitates the mounting of the 8 x 8 x 6 inch bypass recuperator directly in the exhaust gas flow path without the complication of turning this flow. While the air cleaners, starter and alternator are the only accessories indicated on this drawing, it is clear from the dimensions (33.5 in. length, 18.65 in. height, and 19 in. beam, and the resultant 5.5 ft^3 envelope volume) that the bypass-recuperative, single-shaft turbine does not represent a problem as regards engine compartment space in the standard six-passenger vehicle. A preliminary weight investigation prepared for this compact single-shaft bypass-recuperative design has indicated that the basic engine will weigh about 300 lbs and the total propulsion system weight, including accessories, drive line, and transmission, should not exceed 975 lbs. This weight figure is in line with the weight estimated for a similar advance recuperative design proposed by United Aircraft (16). The United Aircraft design assumed a higher turbine inlet temperature (2000°F) and resulted in a total propulsion system weight of 905 lbs. A comparable Otto cycle powerplant is estimated to be about 100 lbs heavier than the bypass-recuperative design, as shown by the 1090 lb weight estimation of the 250 CID six-cylinder Chevrolet powerplant presented in Table 4-II (ECPE powerplant analysis). It appears, therefore,



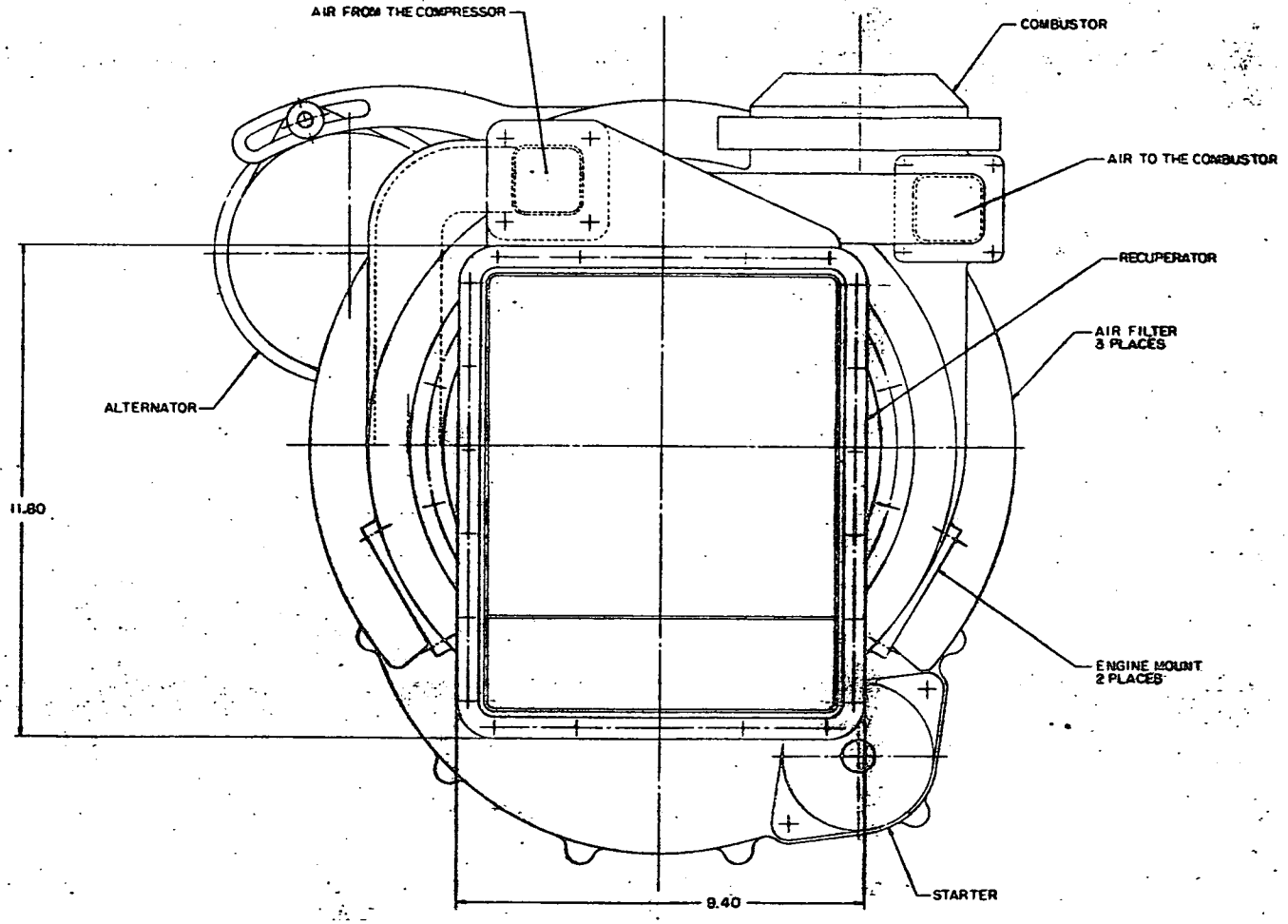
SECTION A-A

THERMO MECHANICAL SYSTEMS INC.
CANOGA PARK, CALIF.

SCALE FULL
DATE 10-21-71

GAS TURBINE-SINGLE SHAFT

150HP/67,000 RPM, BY-PASS RECUPERATOR



VIEW F-F

FIGURE 5.40

THERMO MECHANICAL SYSTEMS INC.
CANOGA PARK, CALIF.

021-71

GAS TURBINE - SINGLE SHAFT

that a small weight advantage is possible with an automotive gas turbine powerplant, although in 20 years of development of an automotive gas turbine no such advantage has been demonstrated. In addition, automotive powerplant weight is not of large concern in evaluation of competing systems except as it affects cost and critical materials consumption.

The geometry of the bypass recuperator and associated ducting is denoted in the side section of Figure 5.39 and the aft end view of Figure 5.40. Compressed air is ducted back along the left upper side of the engine to the aft end inlet header of the recuperator where it is turned 180° and flows forward through alternate plate-fin passages in the recuperator. At the forward end of the recuperator, the preheated high pressure air is collected in the outlet header and ducted forward to the combustor. The exhaust gases are diffused and flow aft to the recuperator where they are allowed to either bypass the recuperator via the duct along its underside or, with the bypass valve closed as indicated in Figure 5.39, flow aft through alternate plate-fin passages in the recuperator. Aft of the recuperator the exhaust gases, whether bypass or recuperated, dump into one manifold to be finally ejected to the atmosphere.

It should be noted that for the design shown the compressor air and some portion of the exhaust gases always pass through the recuperator. The bypass duct insures negligible exhaust system pressure drop at high power level conditions. The bypass is not necessary on the compressor air side due to low pressure losses at any power condition.

Generally, it does not appear that the bypass recuperator concept would involve the degree of complication or bulkiness (ducting, etc.) which might negate the favorable performance advantage over the simple cycle turbine engine. Relative to a fully recuperative turbine engine of comparable power output, the bypass concept does offer potential size and cost advantages, particularly if tube type recuperators are utilized, while providing comparable performance.

5.4.3 Ceramic Regenerator Concept

The last heat exchanger type evaluated was the ceramic rotary regenerator. The engine turbomachinery and operating schedule are identical to that

used for the tubular and plate-fin recuperator analysis. A Cercor (trademark of Corning Glass Works) disk matrix was used having a frontal area of 1 ft^2 and axial length of 2 inches, as shown in Figure 5.41. The regenerator design speed is 35 rpm and it exposes two-thirds of the face area to the low pressure flow while the high pressure flow occupies the remainder of the face area. The regenerator seals are of the rubbing variety and are spring and pressure loaded. Leakage for this type of recuperator was determined to be about three percent; however, this may increase with time. The heat transfer and friction characteristics of the Cercor regenerator matrix were obtained from reference 7 and are illustrated in Figure 5.42. The regenerator matrix characteristics were combined with the turbomachinery characteristics (see Figure 5.23) to give the regenerator effectiveness, leakage loss, and pressure loss as a function of power output, as shown in Figure 5.43. A cursory study of the effect of leakage was made and it revealed that BSFC increases on the order of 2% for each percent leakage and BHP decreases at about 4% for each percent leakage. These values were determined at intermediate power levels. These regenerator characteristics were then incorporated into the turbomachinery cycle analysis and the resulting powerplant performance characteristics are shown in Figure 5.44.

The regenerative engine performance is quite similar to the performance shown for the recuperative plate-fin engines. There is only about a 17 HP loss associated with using the regenerator at full power and BSFC values of less than .60 lbm/hp-hr are realizable from about 20 to 100% power level. The relatively small power loss renders the bypass feature undesirable for this engine configuration. The engine performance is quite comparable to the performance of the 12 x 12 x 3 inch plate-fin recuperative engine (without the pass) of the previous section. To further illustrate how regenerator size could influence the desirability of using a bypass, Figure 5.45 shows the effect of regenerator size on engine performance. As can be seen, the larger the regenerator face area, and consequently the closer one approaches complete regeneration, the lower the engine BSFC and the higher the power output. In fact, in the limit, for complete regeneration, an almost constant BSFC of .4 lbm/hp-hr is obtainable throughout the entire power regime. As the regenerator face area is made more realistic for an automotive application, however, the BSFC gradually climbs and the horsepower decreases. At a face area of

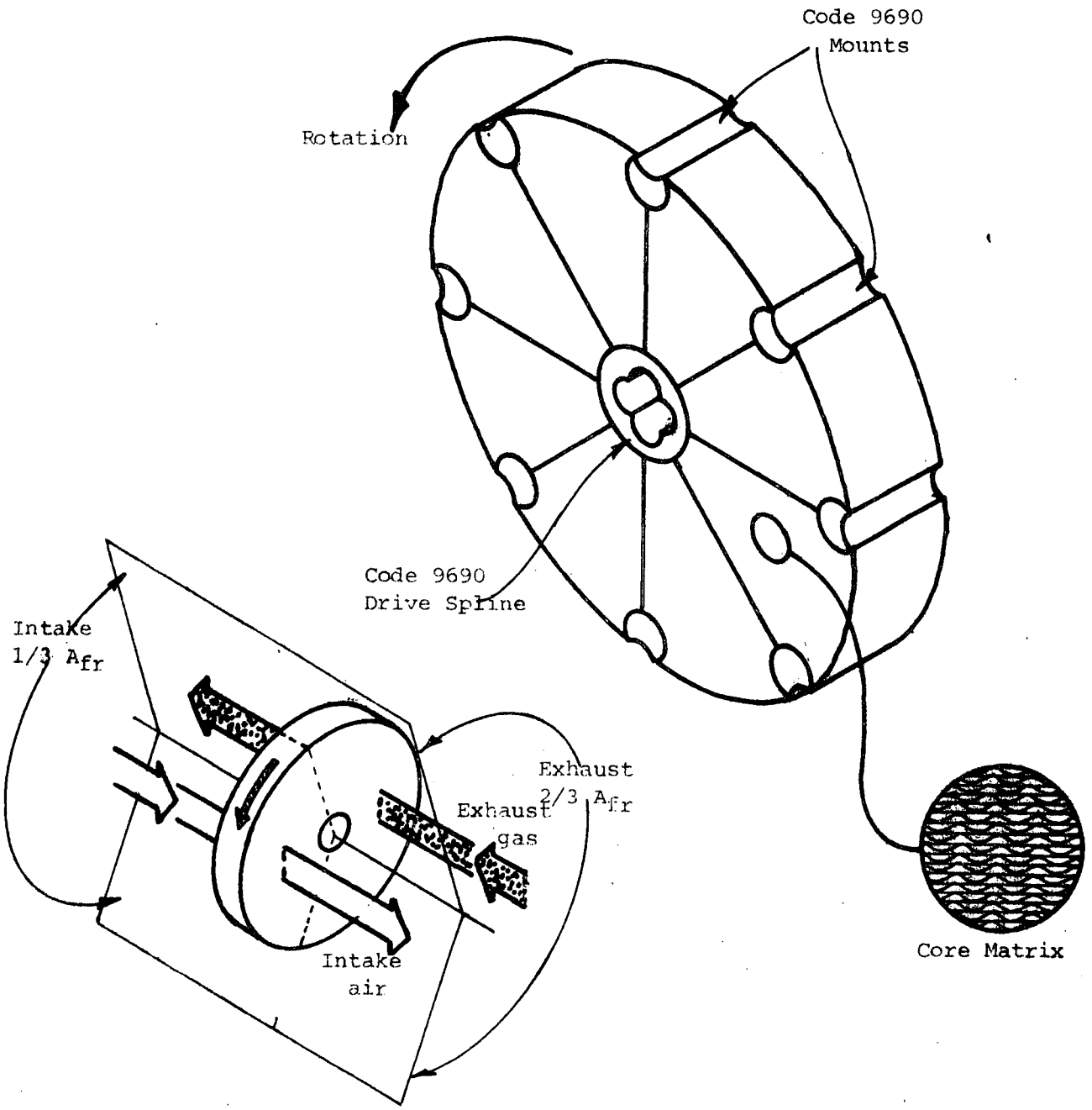


FIGURE 5.41 Gas Turbine Counterflow Ceramic Rotary Regenerator.

Constant Speed = 35 rpm

Frontal Area, A_{fr} = 1 ft^2 (2/3 exh., 1/3 intake)

Diameter, O.D. = 13.5 in.

Length, L = 2 in.

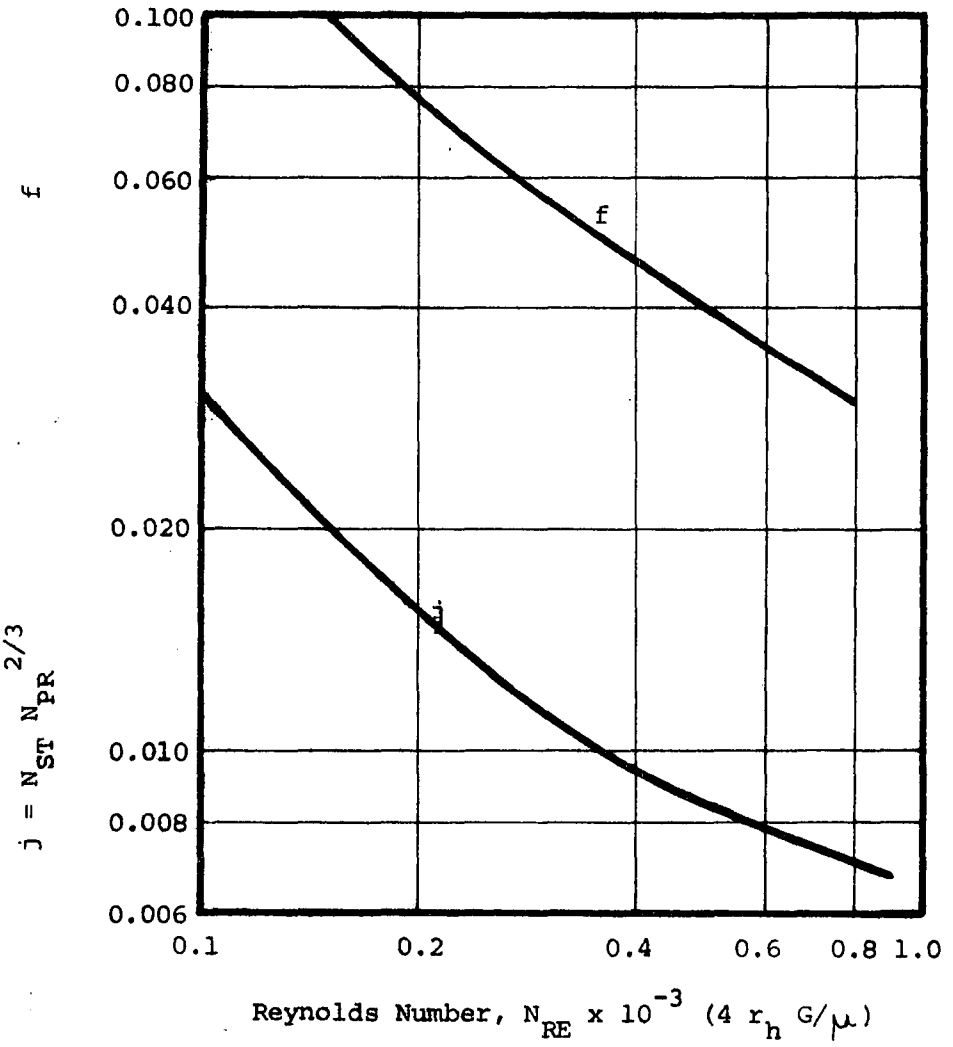


FIGURE 5.42 Glass Ceramic Regenerator Matrix Heat Transfer and Friction Characteristics.
(Ref.: Figure 6, Bayley and Rapley)

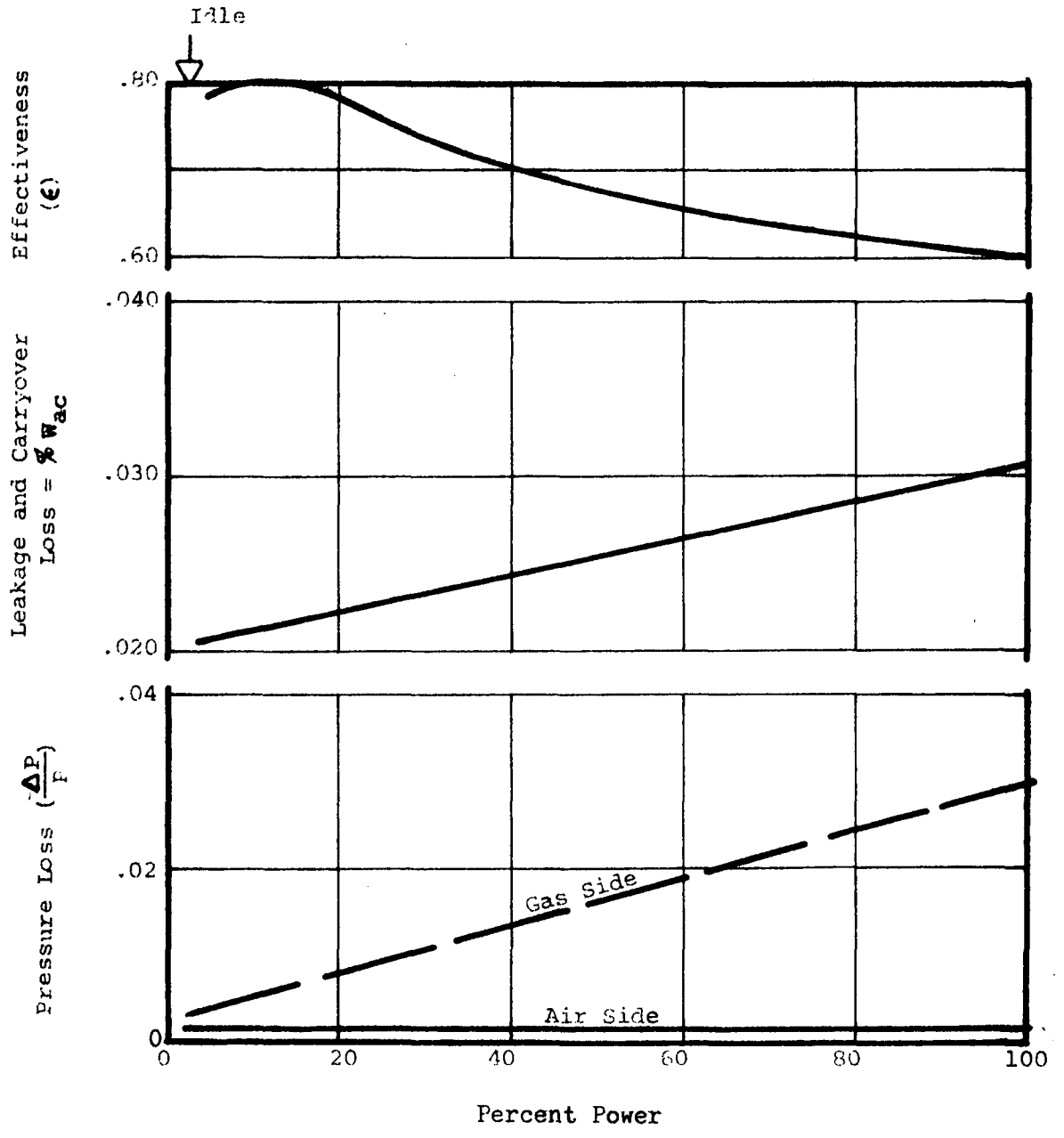


FIGURE 5.43 Characteristics of Ceramic Regenerator as a Function of Turbine Power Output.

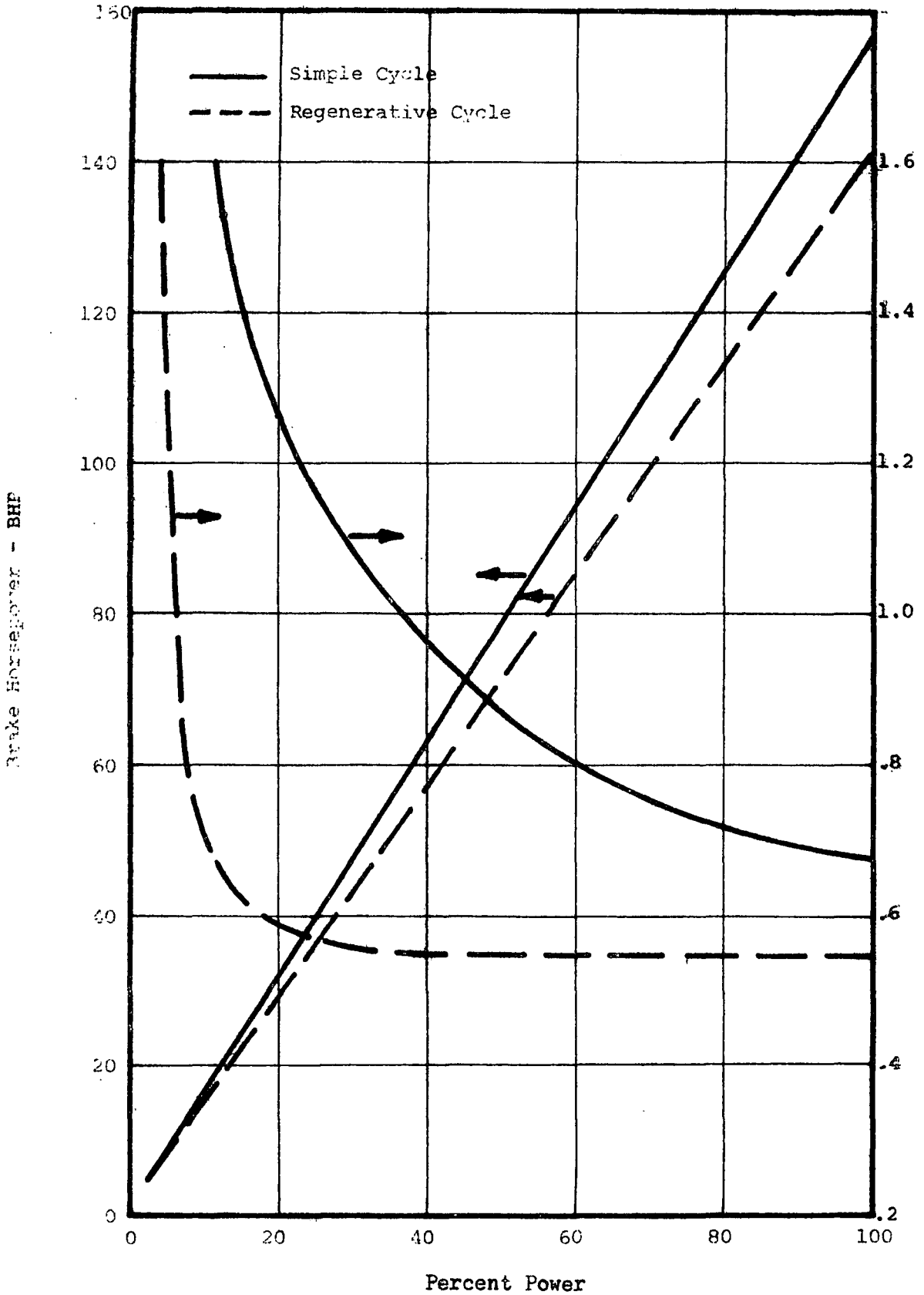


FIGURE 5.44 Power Output and BSFC Characteristics for Gas Turbine Powerplant with a Rotary Ceramic Regenerator. (1 ft^2 face area; 2 inch length, 35 rpm).

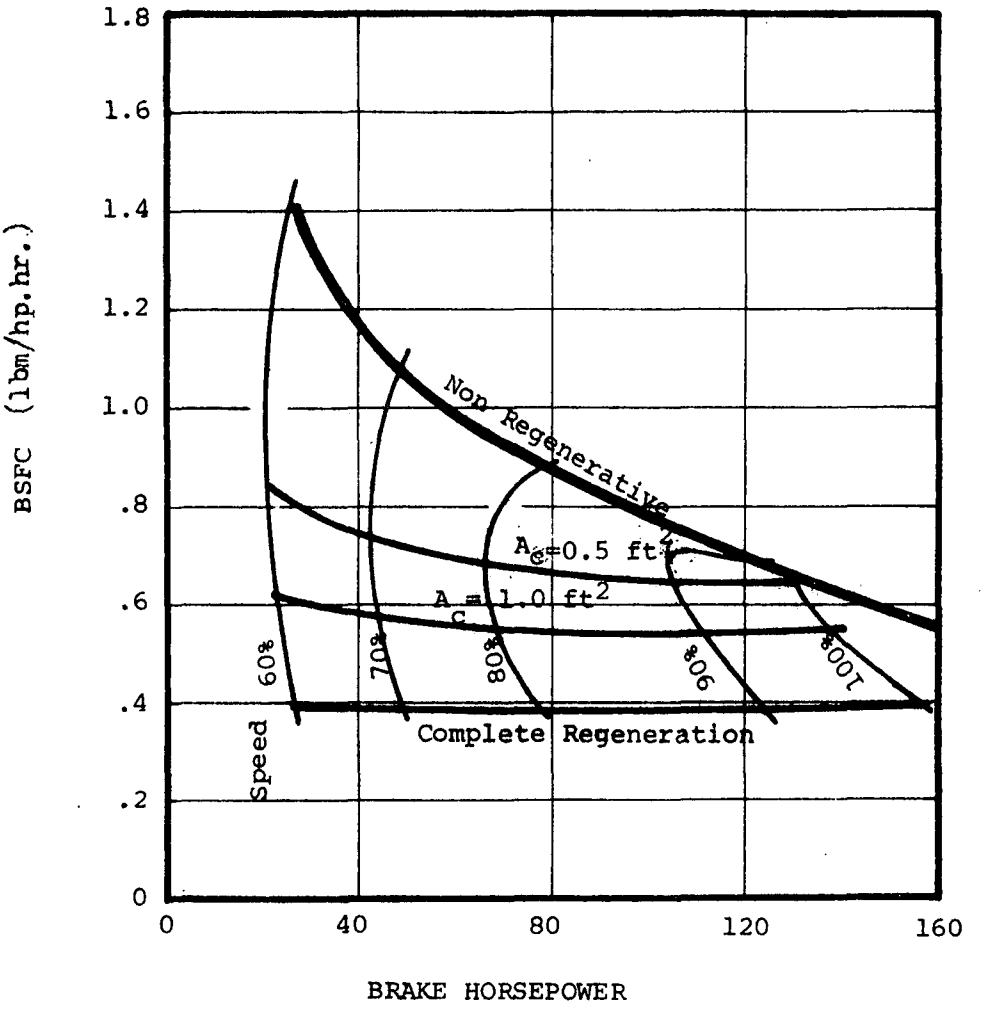


FIGURE 5.45 Effect of Regenerator Face Area on Performance Characteristics of Gas Turbine.

.5 ft^2 (diameter = 10 inches) the BSFC is about .70 lbm/hp-hr from 25 to 100% power level and the engine experiences about a 30 hp power loss at full power. This compares favorably with the 8 x 8 x 6 inch plate-fin recuperative engine and exhibits the same potential for using the bypass concept. However, it appears that the better choice here would be to use the 1.0 ft^2 face area regenerator, and eliminate the need for bypass, obtaining more favorable engine performance without a significant increase in total volume. The horsepower loss (17 HP) for the 1.0 ft^2 face area regenerator at full power could be most easily regained by increasing the size of the turbomachinery as necessary.

Further effort (engine schematics, design layouts, etc.) into the regenerative engine concept was deemed unnecessary as several well known automotive gas turbine research organizations have, and are now, devoting much time and money to this effort. From the results presented here, it appears to be a very practical gas turbine powerplant for automotive application, showing good performance and fuel consumption in a rather compact package. However, further development of the regenerative gas turbine is required to attain a suitable automotive powerplant.

5.5 SINGLE-SHAFT AND FREE-TURBINE ANALYSIS

The majority of experimental automobile gas turbine engines built to date have been of the free-turbine type. However, the failure of this type of engine to make any real progress toward the objective of a mass-produced turbine powered vehicle has induced gas turbine manufacturers to re-examine the myriad of possible cycle arrangements for a solution which may better satisfy both the operational and economic requirements of a successful powerplant system. Other arrangements which are primarily considered are the basic single-shaft and various differential shaft cycles, all of which are simply variations of the fixed or two-shaft types.

It has previously been noted that the fixed-shaft and free-turbine engines have greatly different torque-speed characteristics. This is illustrated in Figure 5.46 where the maximum torque of typical free-turbine and single-shaft engines is compared. Whereas the single-shaft engine has negligible torque below 60% output speeds and develops maximum torque only at rated speed, the free-turbine develops maximum torque at stalled output speed equal

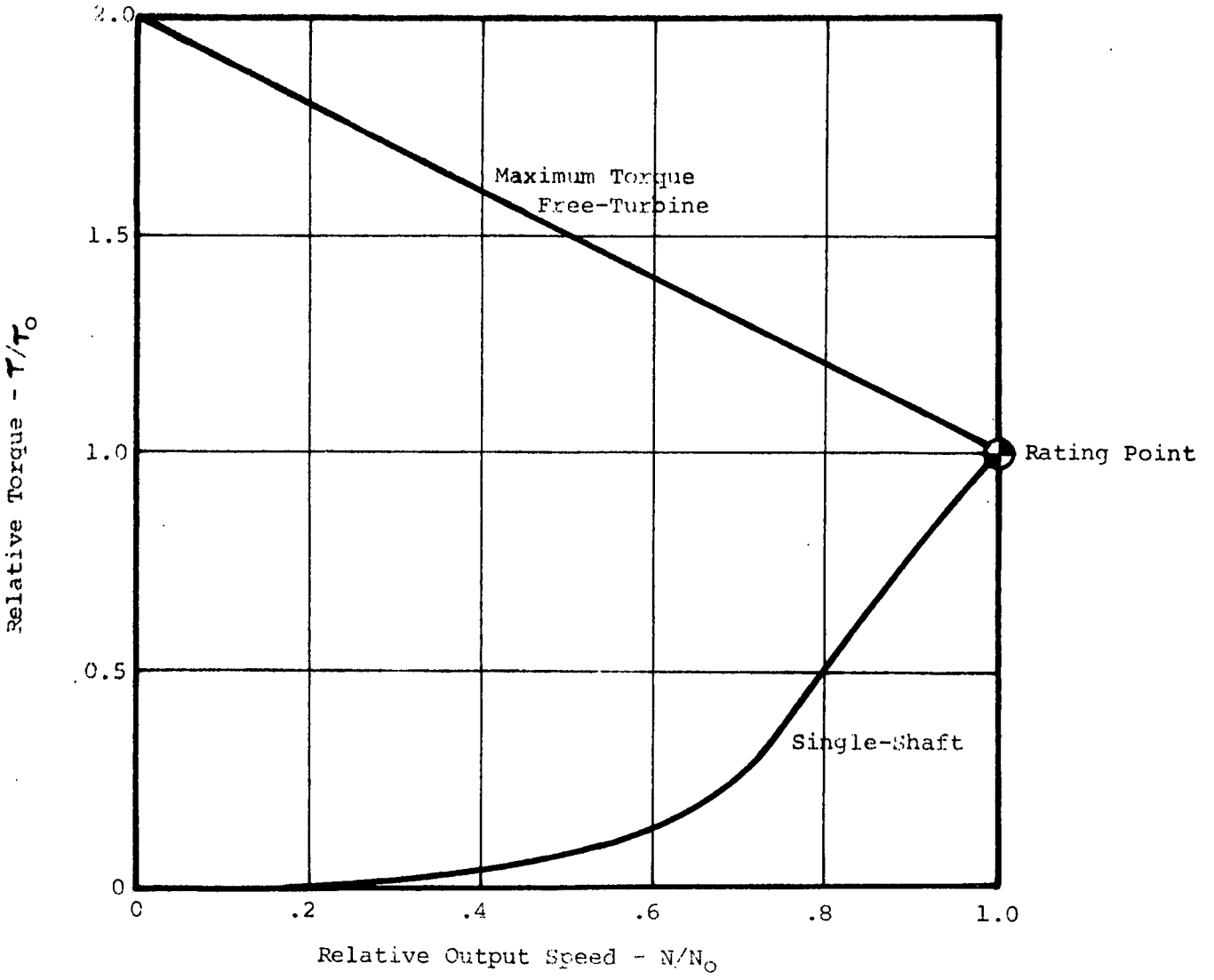


FIGURE 5.46 Comparison of Maximum Torque at Speed Relationships of Typical Free-Turbine and Single-Shaft Engines (19).

to twice its torque at the rating point. At 60% of rated output speed, the free-turbine maximum torque is approximately 10 times the torque available from the single-shaft engine.

Because of the limited torque at low speeds and the excessive time required to accelerate the compressor from these low speeds (low power) to full speed (where adequate power for acceleration is available), it is necessary to operate the fixed-shaft engine in a speed range from about 60 to 100%. The necessity of idling at high output speeds results in high idle air and fuel consumption and therefore poor powerplant efficiency. These are characteristics shared by all gas turbine types, including the two-shaft versions, where the gasifier compressor must also operate between 60 and 100% speed. The problem of low torque at near-idle output speeds is, however, unique with the fixed-shaft engine and a more sophisticated transmission (see Section 5.2) is required to provide acceptable vehicle acceleration characteristics.

The free-turbine engine may be thought of as a fixed-shaft engine with a compressible fluid torque converter attached to its own output shaft. Output speeds from zero to 100% are possible and axial power turbine efficiency varies with the hub spouting velocity according to the relationship developed in Section 5.2 (Figure 5.8). The efficiency of this form of torque converter is somewhat higher than the fluid torque converter used with conventional automatic transmissions, although it is doubtful if a power turbine assembly can be manufactured for the same cost. In addition, the free-turbine engine has certain matching characteristics which require some form of variable geometry to provide optimum performance.

In Figure 5.47 the operating schedules of comparable free and fixed-shaft engine designs are superimposed on the compressor map used for the performance analysis of each. As power is reduced, the operating line for the free-turbine engine (with fixed geometry) is shown to drift farther away from the surge line than that of the single-shaft engine. This results from the characteristic of the free-turbine engine to match to a lower cycle temperature and pressure at any particular off-design compressor speed, and lower cycle temperatures at part power result in lower efficiency. Thus, it is the primary interest of the various free-turbine matching schemes to increase the part-load turbine inlet temperature and optimize the fuel con-

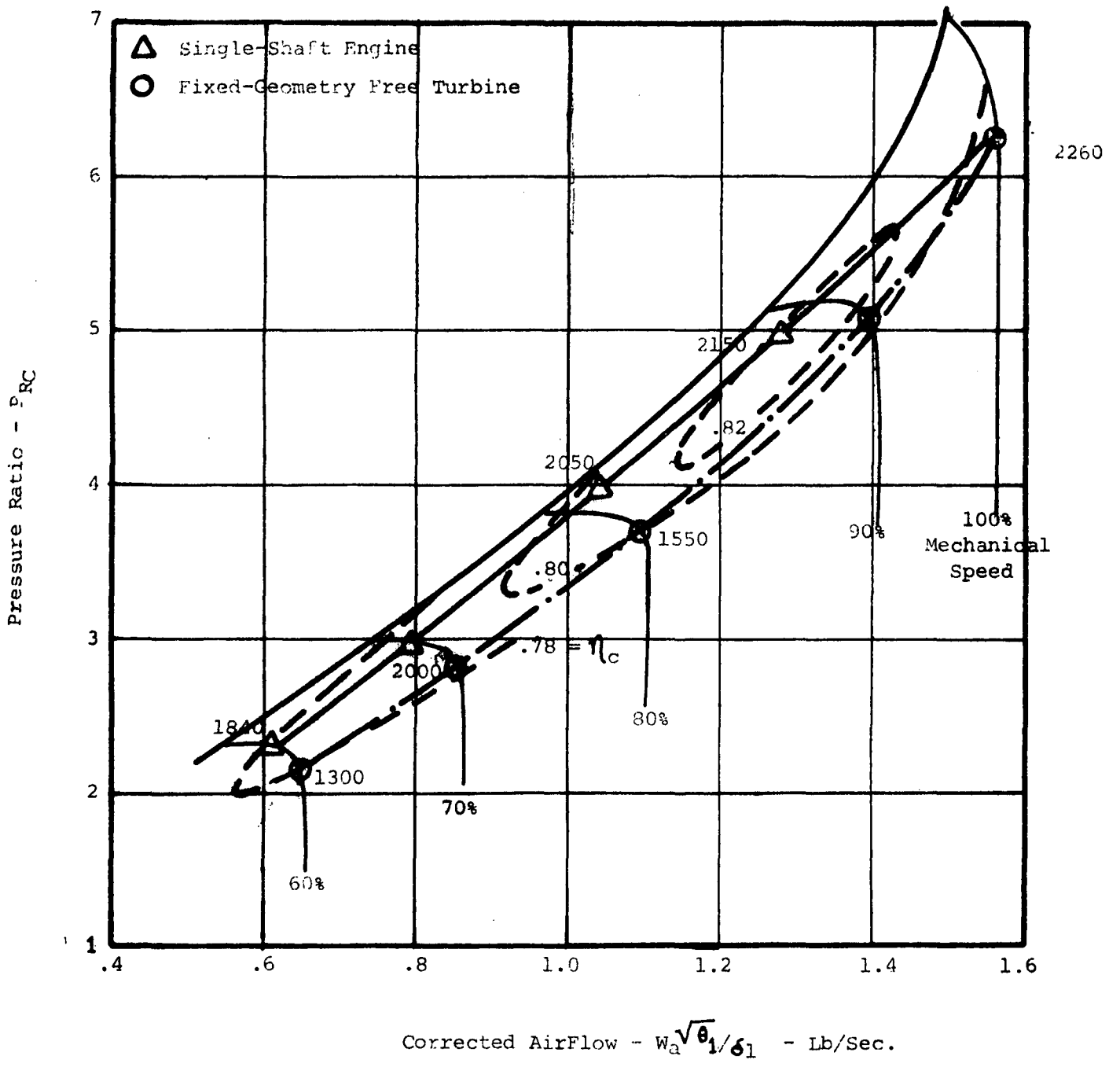


FIGURE 5.47 Comparison of Operating Characteristics of Comparable Free-Turbine and Single-Shaft Engines.

sumption characteristics of the free-turbine engine.

Several means of accomplishing this goal are available, including:

1. Variable geometry power turbine nozzles (VTN)
2. Differential or slipping clutches connecting the power turbine shaft to the gasifier shaft
3. Variable compressor inlet pre-swirl.

Variable geometry turbine nozzles are proposed for many advanced engine designs. This method of operation is illustrated by the schematic of Figure 5.48 where axial power turbine nozzle vanes are shown to rotate about a spanwise axis to set the proper stagger angle for any particular operating regime (22). The vanes can be rotated from the design position toward the optimum part power settings or in the opposite direction as shown to admit flow to the back side of the blade for a measure of braking. For economy or part-power operation the vane is rotated toward the closed position, effectively reducing the nozzle flow area and allowing the power turbine to match to a higher pressure ratio than the fixed geometry power turbine could, thereby permitting higher turbine inlet temperatures (TIT) and increased efficiency.

The effect of this power-turbine nozzle area reduction is illustrated in Figure 5.49. As the load is reduced, the power turbine nozzle unchokes, riding down the turbine flow characteristic quite rapidly as both pressure ratio and corrected flow decrease. During this period the gasifier turbine remains choked, for the most part, since at lower power settings (lower TIT) the gasifier turbine pressure ratio must be held nearly constant to supply the power demanded by the compressor. By reducing the power turbine nozzle flow area 20% it is possible to match the power turbine to a higher pressure ratio at a given corrected flow than that possible with fixed nozzles. This increase in power turbine pressure ratio is gained at the expense of gasifier turbine pressure ratio. Thus, it is necessary to increase the fuel flow and TIT to raise the enthalpy of the combustion products entering the gasifier turbine to match the turbine/compressor work. The effect of increasing TIT at any off-design compressor speed is, as previously stated, to increase the part-load efficiency of the free-turbine engine.

The operating characteristics of similar free-turbine engines with fixed and variable nozzle geometry features is presented in Figure 5.50. At

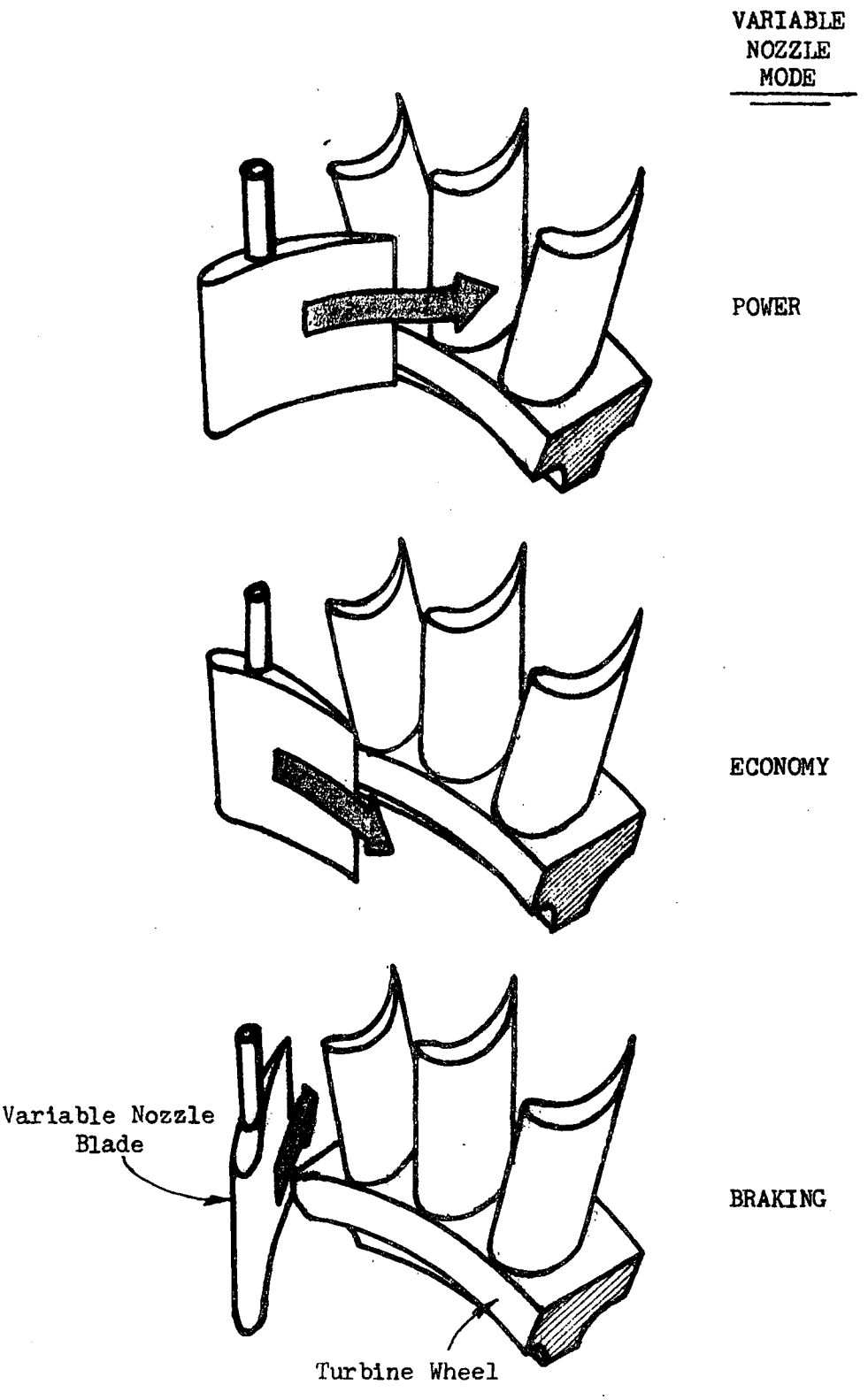


FIGURE 5.48 Schematic of Variable Turbine Nozzle Vane Settings at Different Operational Modes. (25)

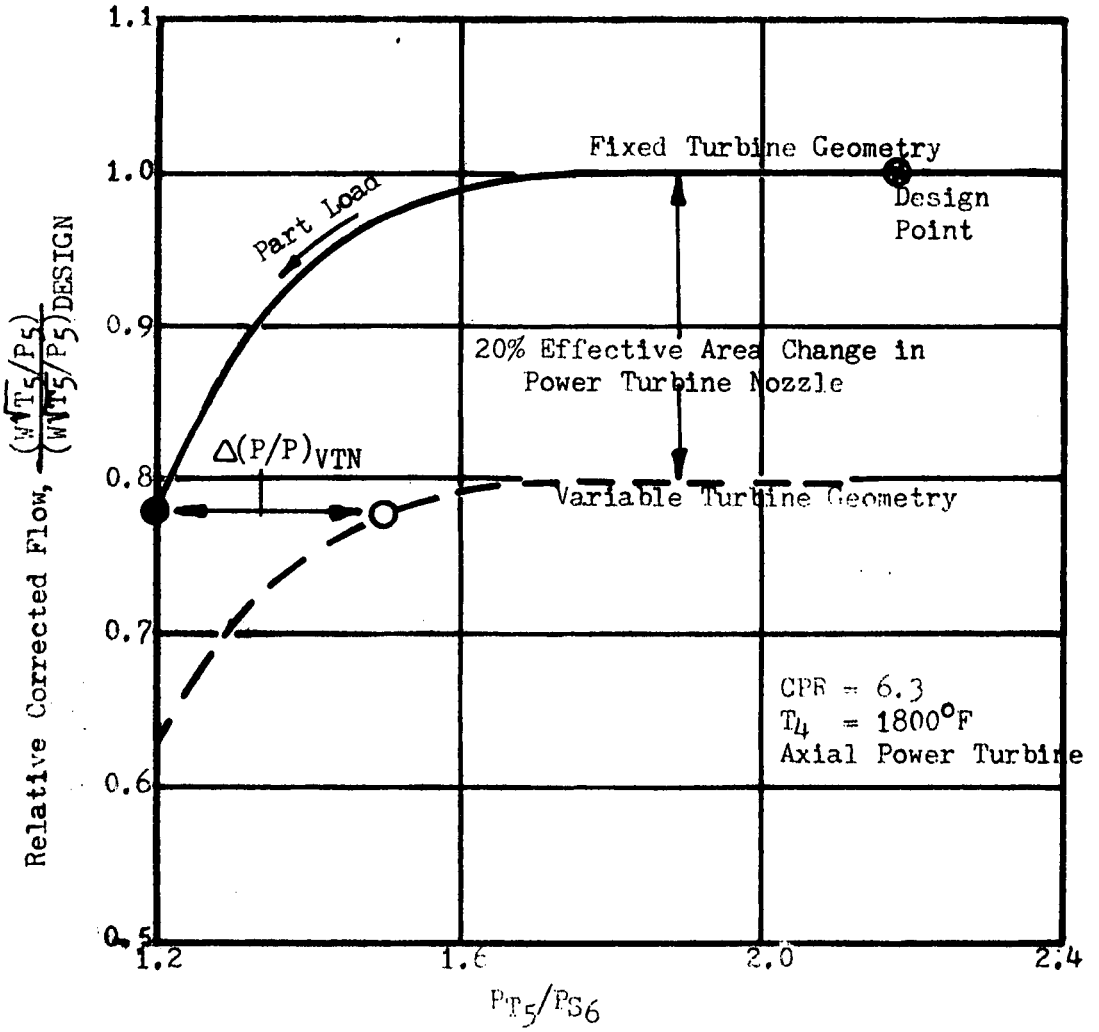


FIGURE 5.49 Effect of 20 percent reduction in power turbine nozzle area on power turbine pressure ratio at part load condition.

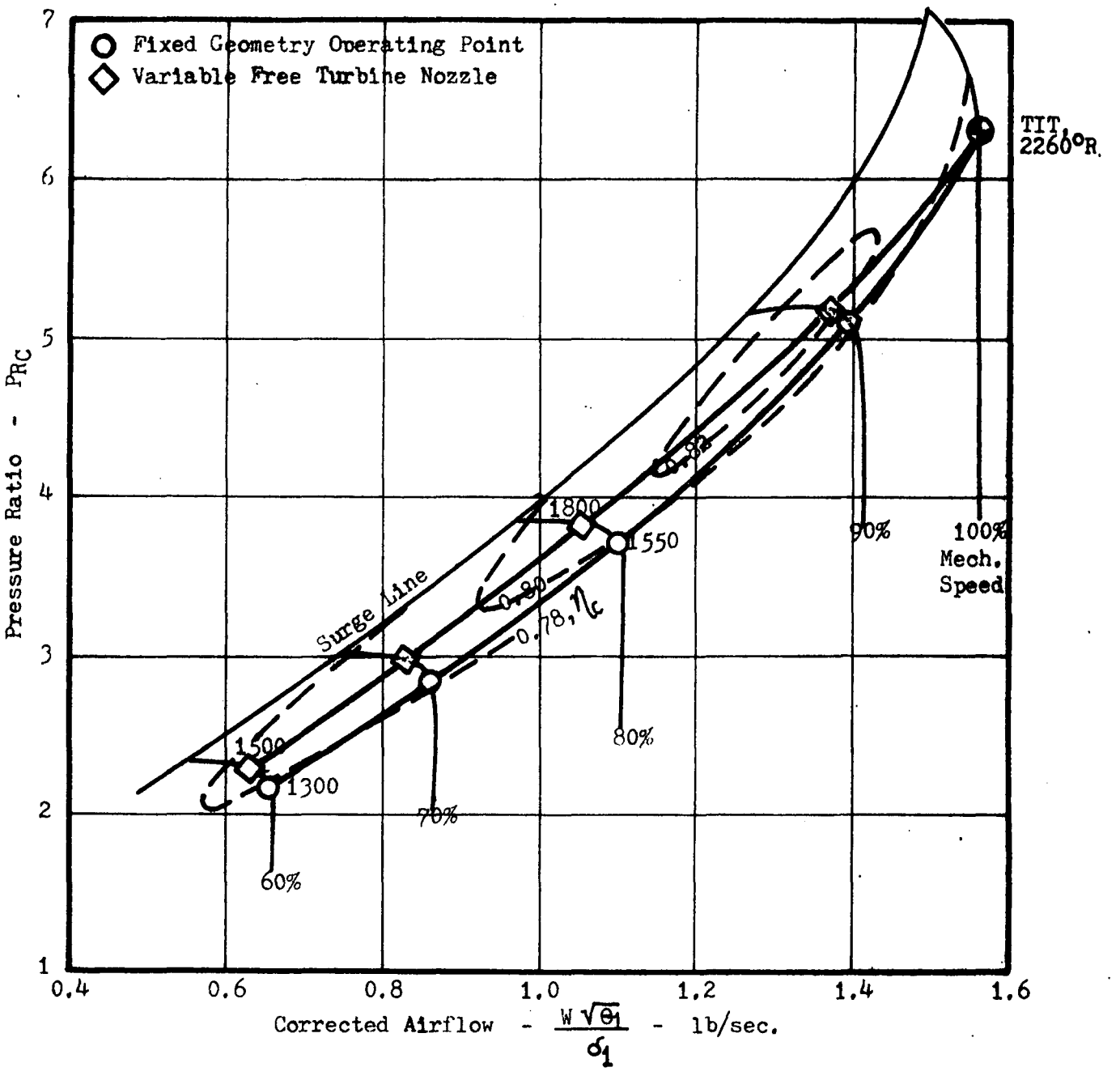


FIGURE 5.50 Comparison of typical operating characteristics of comparable Fixed and Variable Geometry Free-Turbine engines.

any particular compressor speed other than design, the variable geometry engine operates at a higher turbine inlet temperature and pressure ratio. The VTN characteristics shown are indicative of effective area reductions of up to 30%. By choice of a larger effective area reduction than 30%, a free-turbine operating schedule could be obtained which was very similar to that shown for the single-shaft turbine (Figure 5.47). In a practical sense this is not possible, however, since very large nozzle area reductions cause a significant reduction in power turbine efficiency. In addition, the interstage turbine temperature limitations could be exceeded if the single-shaft TIT schedule were followed.

Although it is obvious that the variable turbine geometry features are useful in optimizing performance, there are also some inefficiencies and other undesirable characteristics of variable turbine nozzle systems which also should be mentioned. The peak power turbine efficiency with variable nozzle vanes is generally stated to be one of three percent lower than the efficiency of the fixed geometry type. The amount of loss is greatly dependent upon the ability to minimize end-wall clearance effects through careful mechanical design. This is often difficult because of the curvature of the wall. More important, however, is the increase of optimum turbine pressure ratio as nozzle area is reduced, as illustrated in Figure 5.51. Peak efficiency is shifted to a higher pressure ratio as the nozzle area is reduced, resulting in significant efficiency losses at the low pressure ratios where reduced area is necessary for efficiency operation. In this manner a significant portion of the benefit from improved cycle matching is lost, since gas turbines are quite sensitive to turbine inefficiency.

In addition, the inclusion of variable geometry in the hot turbine section detracts from the inherent reliability of the turbine powerplant and adds to its cost and complexity. Because of these reasons, alternatives to variable turbine nozzle geometry are worthy of consideration for free-turbine engines.

Differential free-turbines, which use a variable coupling or clutch between the gasifier and power output shaft, have been demonstrated in vehicular gas turbines (General Motors GT-309 with Power Transfer). The Power Transfer feature used in the GM turbine improves the part-load fuel consumption in essentially the same way that the variable nozzles do; however,

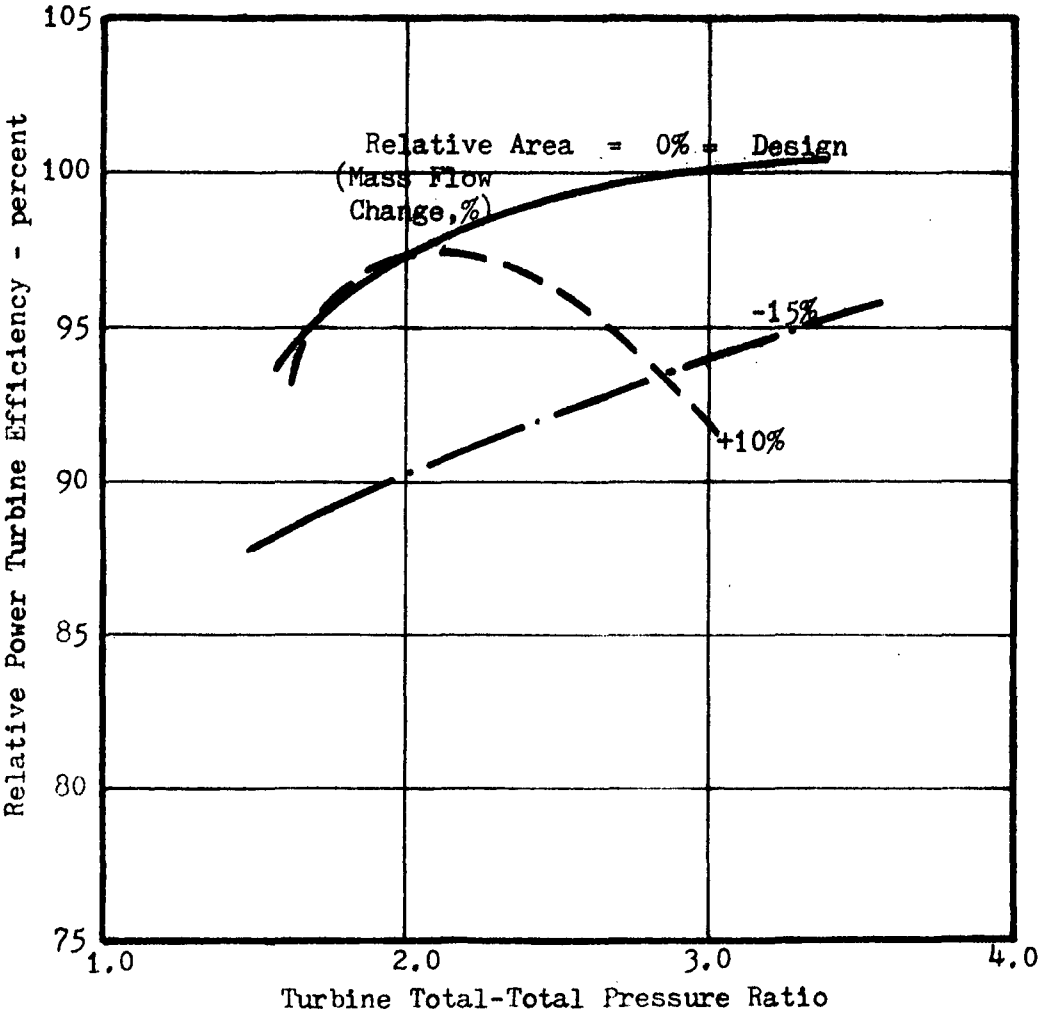


FIGURE 5.51 Effect of Variable Geometry on two-stage axial turbine performance (18).

it accomplishes this by means of well known and proven mechanical components (24). Basically, the variable coupling is used to transfer controlled amounts of power from the gasifier shaft to the output shaft. The gasifier speed governor increases fuel flow to maintain relative constant compressor speed, and the result is an increase in turbine inlet temperature at any particular gasifier speed and therefore, an increase in cycle efficiency.

The effect of variable turbine nozzles (VTN) or differential clutching on fuel consumption is basically the same. In properly designed engines the efficiency losses attributable to variablenozzles is approximately equivalent to the losses resulting from the hydromechanical power transfer system. Figure 5.52 shows the fuel consumption comparisons between fixed nozzle and two VTN designs, one of which retains constant turbine inlet temperature and the other constant exhaust gas temperature (22). At approximately 30% power (40 HP) the fuel consumption of the constant TIT engine is shown to be approximately 35% lower than the SFC of the fixed geometry system, whereas a 25% improvement is indicated for the engine which controls power turbine exhaust temperature. The latter is more indicative of the improvement possible in a practical system, where at low compressor pressure ratios the constant TIT engine would result in radial gasifier turbine hub overtemperatures (radial turbines are stress limited at hub temperatures around 1300°F). Realistic improvement expected from variable geometry is illustrated in Figure 5.53, which shows that the SFC of the variable nozzle design is approximately 20% lower in fuel consumption than a comparable fixed geometry design (15) at the 30% load condition.

In short, variable geometry or similar matching methods are desirable if not necessary free-turbine engine features. The choice between variable nozzles and variable coupling ultimately depends upon the success in developing variable nozzle systems competitive in the all important areas of cost and reliability. In addition, the ability to lock the shafts of differential free-turbines provides favorable engine braking, power turbine overspeed protection, and a means of handling gasifier turbine-driven accessory loads (24). Variable nozzle systems are less bulky and lighter in weight, although it has not been established that these are critical considerations for automobile powerplant installations.

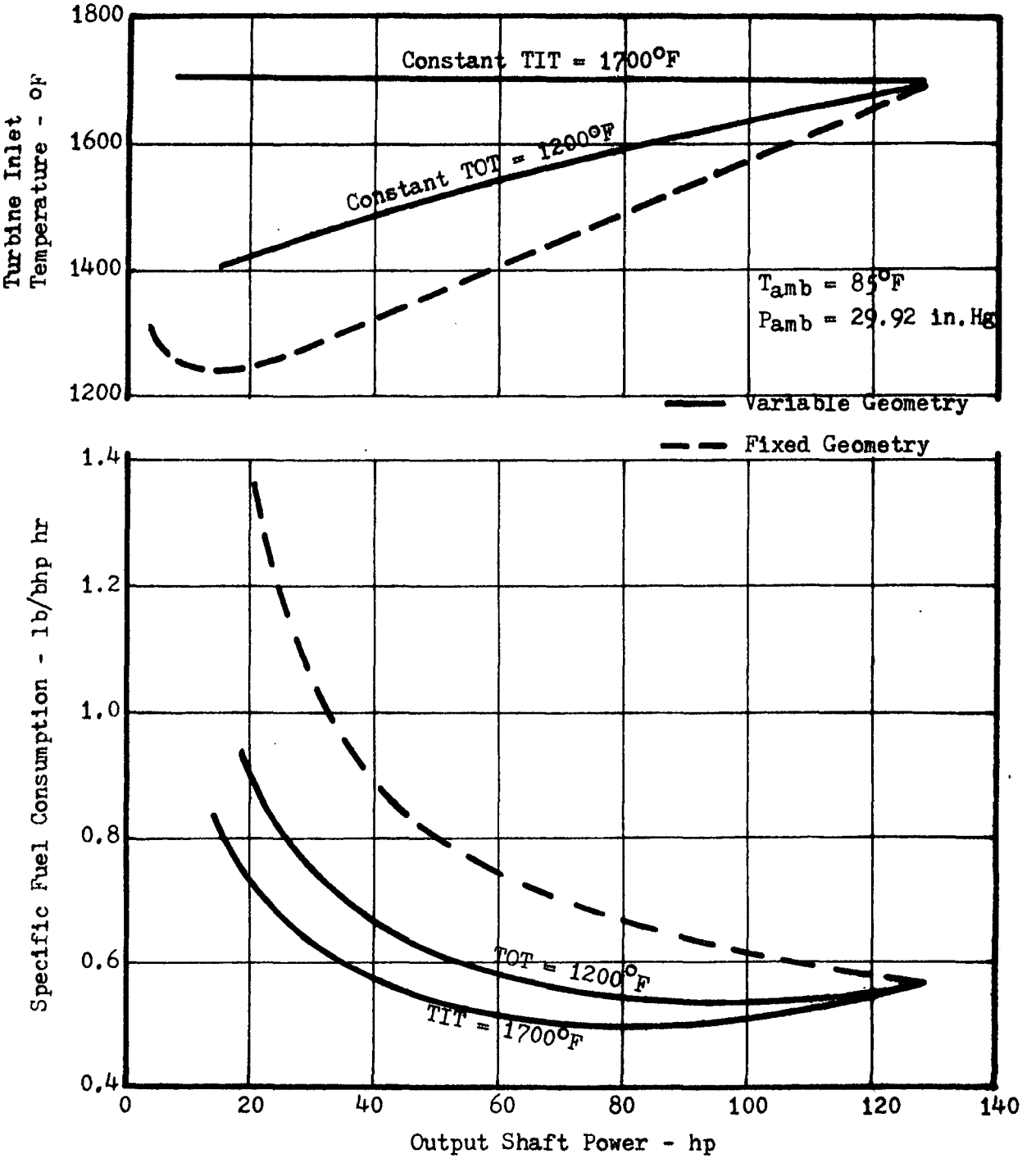


FIGURE 5.52 Fuel consumption characteristics for Free-Turbines with Variable and Fixed Geometry (22).

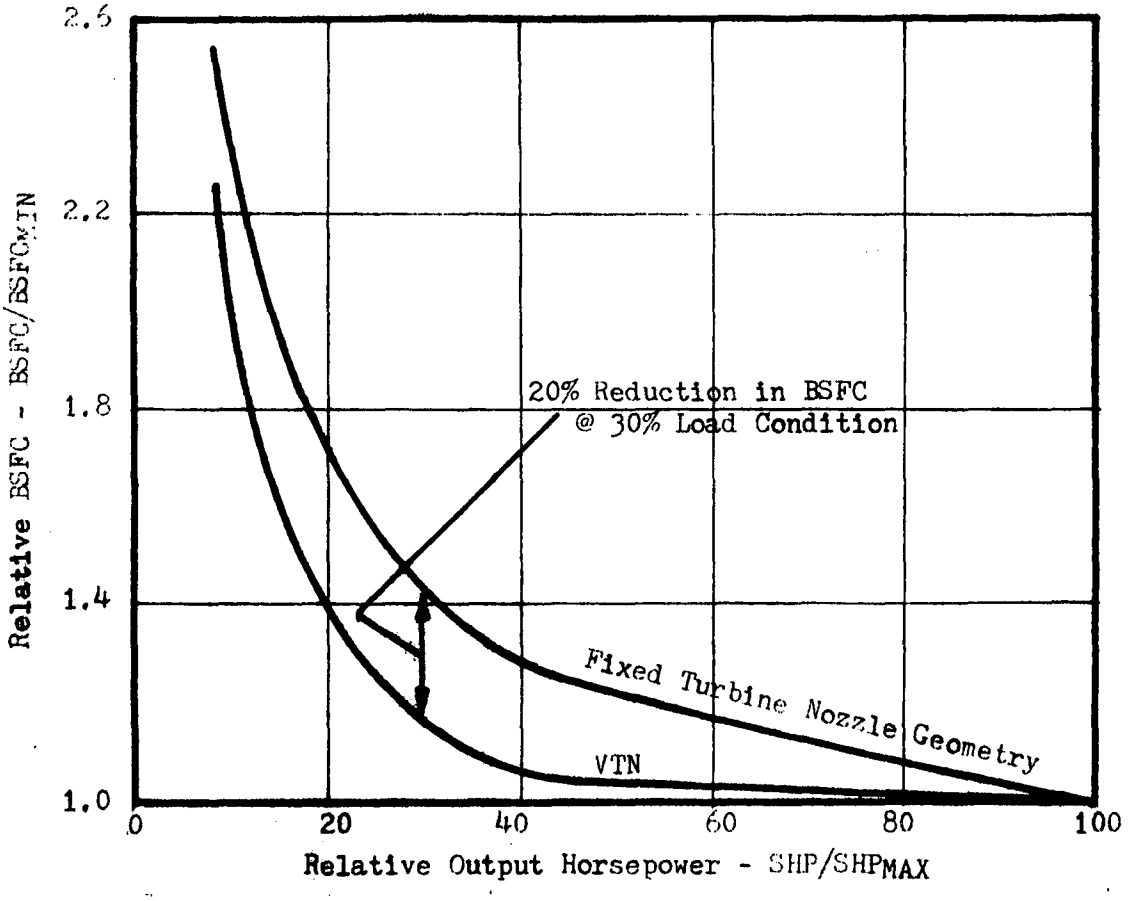


FIGURE 5.53 Effect of Variable Turbine Nozzle Geometry on fuel consumption characteristics of Free-Turbine engine. (15)

Because power turbine characteristics approximate those of the torque converter in single-shaft engine systems, it can be stated that the performance of optimized single and two-shaft engines is comparable. For this reason it was convenient to limit cycle analyses performed in this study (primarily Sections 5.3 and 5.4) to fixed-shaft engines, although this choice was not intended to imply that the free-turbine has less merit. Rather, this decision was made to allow greater emphasis to be placed on gas turbine heat recovery (recuperative or regenerative) analysis in general and the feasibility of the bypass/recuperative or regenerative system in particular. Choice of fixed-shaft or free-turbine types does not significantly affect this analysis. For this study it is sufficient to say that although comparable in performance, the choice in design type allows for trade-offs in areas of weight, size, cost, and operational characteristics.

The single-shaft engine appears to offer potential benefits in the area of cost. It seems reasonable to assume that an acceptable transmission can be provided well within the added cost of the power turbine and the engine matching features required in free-turbine systems. UARL (16) estimates the direct cost of an 8 or 10 speed hydromechanical transmission with controlled slipping clutch for the single-shaft engine to be no more than $50 more than that of the 3 or 4 speed (without a torque converter) transmission required for the free-turbine engine. The cost of the power turbine assembly for the free-turbine is likely to surpass this figure several times.

The weight of the VTN free-turbine powerplant is potentially lower than that of the single-shaft powerplant. This results not only from the more complex transmission required for the single-shaft system but also from the necessity for a higher rated power to provide comparable vehicle acceleration (due to the slope of the torque-speed curve). Much of this weight advantage is lost, however, if the free-turbine is of the differential-clutch type. Due to the fact that the free-turbine has superior torque at low speeds, it is expected that vehicle acceleration goals can be met with a somewhat less powerful engine than required with a single-shaft powerplant. Therefore, the vehicle fuel economy with the free-turbine engine is expected to be favorable, since road-load power requirements will be a larger percentage of maximum output power for the smaller powerplant and the gas turbine engine is more efficient at higher power levels.

In summary, the characteristics of single-shaft and free-turbine vehicle powerplants are compared in Table 5-IV.

5.6 COMPREX GAS TURBINE

The usual gas turbine is based on the compression, heat addition, and expansion of a gas flowing through a compressor, combustor, and turbine. Such engines deliver the maximum amount of power per unit volume and weight of machinery. While this approach is most advantageous for high output engines for stationary power systems and for aircraft power systems where high power output per unit weight and volume are essential, the use of corresponding levels of design sophistication for smaller automotive gas turbines (150-160 HP) is not as feasible as the larger aircraft types. The size of the rotating machinery is too small and correspondingly the speeds and the resulting gear ratios are prohibitively high. As an alternative, it would appear desirable to consider a form of elastic wave machine in which compression and partial expansion are carried out in a single chamber, a machine previously titled the Comprex. The Comprex is basically composed of several air passages and cells with shutters regulating the path of air flow and consequently the power output. It is proposed that the Comprex gas generator would replace the conventional compressor and gasifier turbine as the power source for a free-turbine type automotive powerplant. Such a machine was invented by Brown-Boveri, the manufacturer of turbomachinery, diesel engines, compressors, etc., approximately 25 years ago. It was investigated for application in the aircraft engine field but the details of the performance were such that the unit was sensitive to rotor speed. Recently Brown-Boveri has visited several diesel engine manufacturers to suggest its use as a turbocharger, since they had solved or alleviated the sensitivity to rotor speed.

The name Comprex originates from the fact that the same rotating machine compresses air and expands gas resulting in a much lower metal temperature than is experienced in the turbine of the conventional gas generator. Descriptions of Comprex operation are given in References 10-13. A sketch of the rotating element of the Comprex is shown in Figure 5.54. The hot expanding gas acts directly on the air to be compressed. The action occurs within the straight passages of the rotor. The rotation serves only as a means of connecting each channel with the desired port at the desired time.

TABLE 5-IV

COMPARISON OF FREE-TURBINE AND SINGLE-SHAFT GAS TURBINE POWERPLANTS

System Characteristics	Fixed Geometry Free-turbine	Variable Nozzle Free-turbine (VTN)	Differential Free-turbine	Single-shaft Turbine
Torque at low output speeds	good	good	good	unfavorable
Transmission Required	3 or 4 speed gearbox; no torque converter required			8 or 10 speed with infinitely variable hydrostatic or controlled slipping cluth
Maximum Power Required for Vehicle Acceleration	comparable			higher
Part Load Fuel Economy	low		comparable	
Vehicle Braking	none	fair	good	good
Overspeed Protection	none	fair	good	excellent
Size (including transmission)	small	small	larger	largest
Weight	lowest	low	higher than VTN	highest
Cost	lowest cost of free-turbines	highest	lower than VTN	lower than free-turbine
Use of Critical Material	comparable			lowest

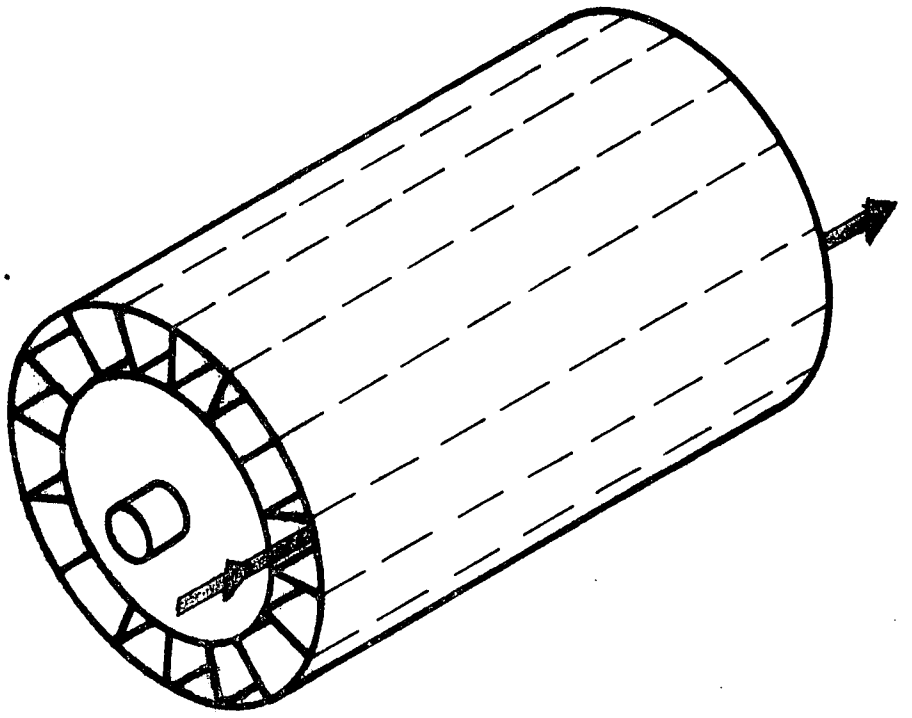


FIGURE 5.54 Sketch of Comprex Rotor

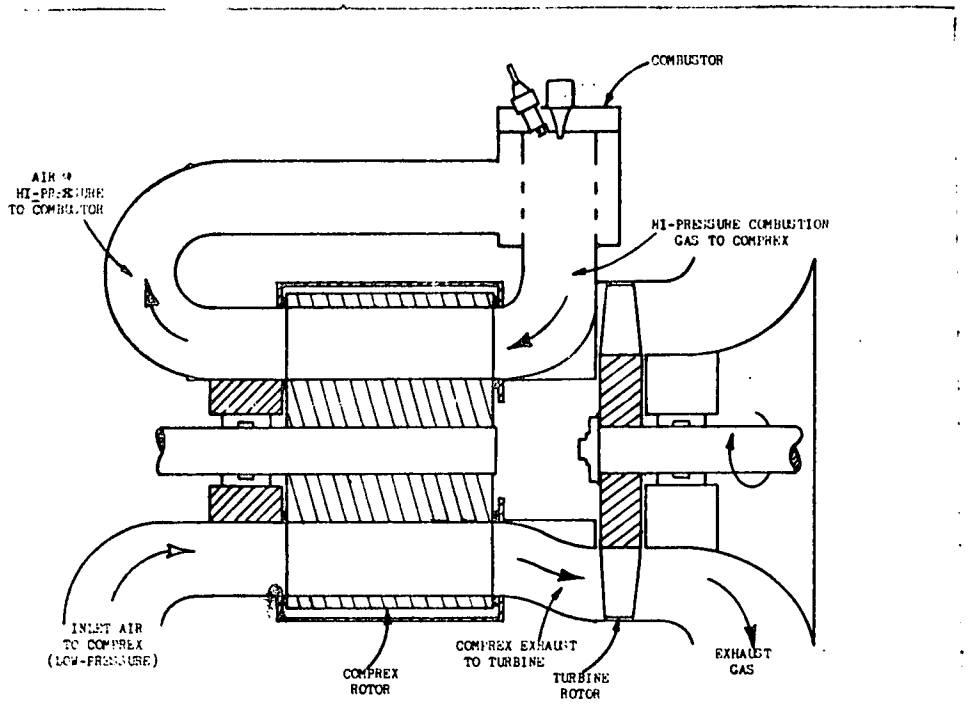


FIGURE 5.55 Schematic of Automobile Powerplant Using Comprex Gas Generator and Free Turbine.

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A schematic diagram of an automobile powerplant using a Comprex gas generator is shown in Figure 5.55. Hot gas from theccombustor is first used to compress the ambient pressure intake air and is then discharged into a free turbine to supply automotive power. Valve timing is accomplished by the motion of the rotor between two stator plates which have ports so arranged as to control conditions of the cycle. There are a large number of passages in the rotor, all of which participate simultaneously in different phases of the cycle. The multitude of channels results in a continuous flow of compressed air through the system.

Figure 5.56(a) shows a wave-flow diagram of an unwrapped rotor (11). In order to follow the wave and flow phenomenon through one complete cycle, consider first the channels at the top of the diagram. These channels are filled with ambient air at ambient pressure and temperature. As the channels move downward, the high pressure hot gas port on the right side is uncovered and a shock wave followed by the interface is driven to the left. The shock wave travels much faster than the air and interface which follow. The air behind the shock is compressed to a level higher than ambient and this air also possesses a velocity head,which if recovered, will provide additional compression.

As the channels move downward, the high pressure air outlet port opens at about the time the normal shock reaches the left end of the channel. Then, with both ends of the channel open, the air and gas continue to move to the left carried by the momentum already established. Next, the high pressure gas inlet port on the right closes causing an expansion wave to move to the left, reducing the pressure of the hot gas and reducing its leftward velocity to zero. At the time the expansion wave and the interface reach the left end of the channel, the left port closes. At about the same time, the low pressure gas outlet port opens on the right permitting the hot gas to expand at high velocity out of the right end of the channel. At this time, an expansion wave moves through the hot gas towards the left end of the channel. When the expansion wave reaches the left end of the channel, the low pressure air inlet port opens and the momentum of the hot gas moving at high velocity to the right causes a fresh charge of ambient air to be sucked into the channel. When the channel becomes filled with ambient air, the ports close and a new cycle is initiated. Figure 5.56(b) shows how the compression and expansion waves and particle paths move back and forth across the channels.

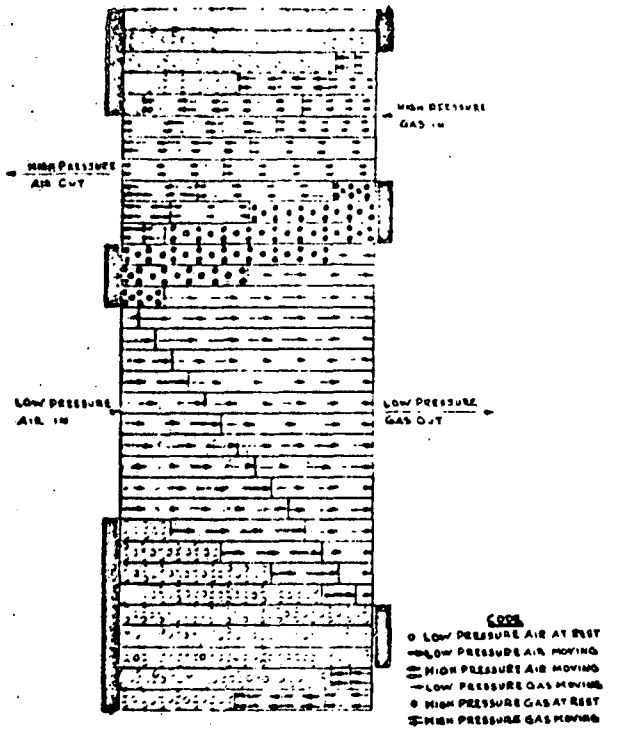


FIGURE 5.56(a) Comprex Wave-Flow Diagram.

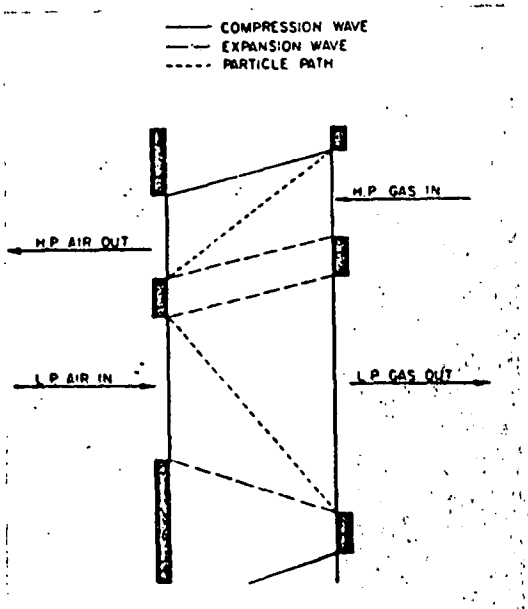


FIGURE 5.56(b) Simplified Wave-Flow Diagram.

P - pressure (Upper case - total cond.)
T - temperature (Lower case - static cond.)
S - Speed of sound
M - Mach no.
V - velocity

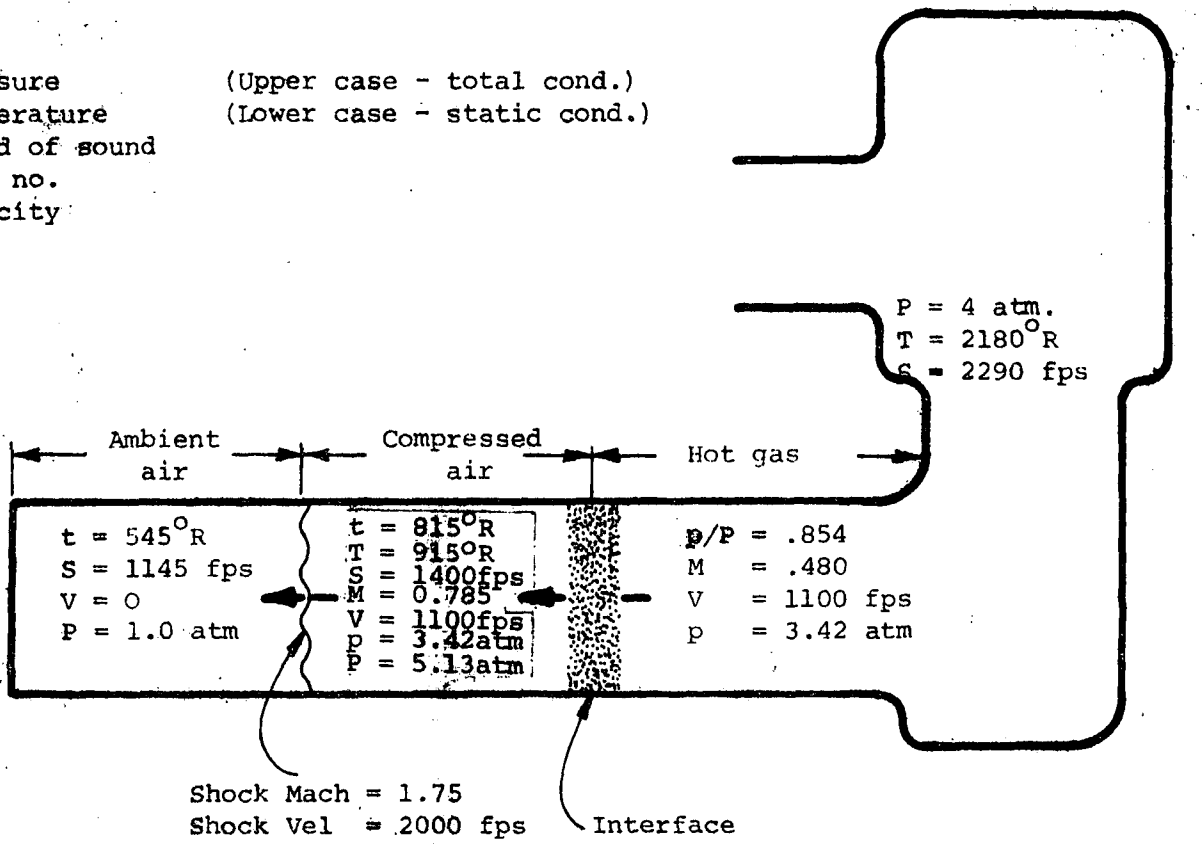


FIGURE 5.57 Results of Calculation to Determine Total Pressure of the Compressed Air in Channel (All calculations assume $k = 1.4$).

A preliminary calculation has been made to determine if the hot compressed gas could, in fact, sufficiently compress ambient air to keep the cycle in operation. The compressible flow charts of Reference 4 were used in making the calculations, and the results are presented in Figure 5.57. It was assumed that the pressure downstream of the combustion was 4 atm. and the temperature 2180°R. The initial conditions in the ambient air channel were 1 atm. and 545°R. The calculations indicate that when the hot gas port is opened, a shock wave will move to the left at a speed of about 2000 fps followed by the compressed gas, interface, and hot gas moving at a velocity of about 1100 fps. The static pressure in the compressed gas is 3.42 atm., but the total pressure is 5.13 atm. which should be more than enough to offset duct losses and combustor losses.

Potential merits of an automotive powerplant using a Comprex gas generator and free turbine are as follows:

1. Gas generator is a low speed machine (on the order of 5000 rpm), therefore, stresses are much lower than in conventional gas turbines. Also, a conventional transmission could be used.
2. Channels in gas generator alternately flow hot and cold gases, so that high temperature alloys are not required.
3. Cycle pressure ratio is not a function of rotational speed as with gas turbine so that part-load efficiency could remain high.
4. System could be designed to "over scavenge" to provide cooling for free turbine so that high temperature alloys for free turbine are not required.
5. System using high pressure ratio might not require heat exchangers, recuperators, regenerators, etc.
6. System is far simpler than Rankine system and eliminates many of the disadvantages of the current gas turbines.

It therefore appears that the Comprex gas generator has the potential for being incorporated into a successful automotive powerplant.

5.7 CONCLUSIONS

The preceding investigation into various gas turbine powerplant concepts for automotive application has generated the following conclusions:

1. In general, it appears that any automotive gas turbine powerplant will be limited to certain maximum temperatures and pressures as determined by the availability and cost of materials. The number and sizes of compressor, turbine and heat exchanger components must be kept to a minimum to reduce the very important initial cost. For this reason, a gas turbine with the following characteristics (representing present state-of-the-art technology) was determined to be best suited for an automotive application:

   a. A maximum turbine inlet temperature of 1800°F.
   b. A one-stage radial compressor and turbine (with present state-of-the-art maximum pressure ratio of about 7).

2. In general, the closed cycle gas turbine concepts do not appear as feasible as the open cycles due to their lower efficiencies, increased complexity, and cost. This is true for all the closed cycles using both air and other exotic working fluids and also includes the sophisticated Ackeret-Keller cycle.
3. For the open cycle concepts, the simple cycle employing a single-stage radial compressor and limited to 1800°F maximum turbine inlet temperature was found to have unacceptable fuel consumption. This led the way for the three heat recovery concepts, including both recuperative and regenerative heat exchangers. It was shown that all three heat recovery concepts can offer similar fuel consumption from idle to full power; however, trade-offs are possible with regard to size, cost, and reliability in choosing a system which best matches the particular powerplant application.

   The engine with the compact tubular recuperator requires a scheduled bypass to achieve optimum performance. This concept provides good reliability and has been field tested and proven. Its drawbacks are its large size and supposedly higher cost than the plate-fin recuperator or ceramic regenerator.

   The plate-fin recuperator and ceramic regenerator examined showed similar performance characteristics, both giving very good powerplant fuel consumption. The cost of the compact plate-fin recuperators and ceramic regenerators are comparable

and the differences in cost in mass production is merely speculation at the present time. The plate-fin recuperator is preferred over the ceramic regenerator due to the fact that the plate-fin type does not require the complex and costly drive system of the regenerator and is less susceptible to seal and bearing problems. The particular recuperator chosen will depend on the available space and the associated heat exchanger manufacturing cost; however, a very compact, low initial-cost arrangement appears to be an 8 x 8 x 6 inch recuperator operating with an exhaust-side bypass valve. This powerpalnt package will provide the power of a medium pressure ratio non-recuperative cycle for acceleration and high speed operation, while retaining the efficiency of a low pressure ratio recuperative cycle at the lower road-load poer levels. Although smaller and less costly, this bypass arrangement offers somewhat poorer fuel consumption than can be obtained with a larger heat exchanger designed for full power operation without the bypass feature. In effect, it appears that an attractive automotive gas turbine powerplant can be developed based upon the plate/split-fin recuperator (either bypass or full-time) which minimizes the very important initial cost while retaining acceptable fuel economy at road-load power levels.

4. Subject to the demonstration of (a) a suitable transmission for the single-shaft engine and (b) a reasonably priced power turbine assembly, the choice between single-shaft or free-turbine types allows for additional trade-offs in the areas of weight, size, cost, and operational characteristics. The single-shaft engine with an appropriate transmission should be less costly, though larger and heavier than the free-turbine. The free-turbine, which has good torque at low speeds, should be smaller in size and power for the same vehicle acceleration characteristics.
5. Finally, a very preliminary investigation of the concept of a Comprex gasifier indicated that it has generally favorable characteristics for application to a free-turbine type of automotive installation. More analytical studies are required to adequately evaluate its relative merit compared to other turbine power systems before consideration can be given to development of prototype hardware.

6.0 STIRLING ENGINE

6.1 INTRODUCTION

One of the engines to be considered for low emission automobile powerplants is the advanced design Stirling Engine. The basic concept for this powerplant dates back to various types of hot air engines which were used extensively in the nineteenth century. However, these early engines did not mature into a practical powerplant which could be considered to be competititve for stationary and mobile vehicle applications until the Philips Research Laboratories in Eindhoven, Netherlands, initiated a scientific and engineering program shortly before World War II. Through this project, the Stirling engine progressed to a high degree of refinement in the 1950's and 1960's.

Philips has developed several varieties of Stirling engines, and several companies throughout the world have been licensed for design, development, and manufacture of the powerplant. Licensees include:

1. General Motors Corporation in the United States
2. United Stirling in Sweden
3. M.A.N. in Germany
4. Ford Motor Company

At one time Japan was interested in the powerplant, but no activity has been reported in recent years. General Motors was extremely active in the research and development of Stirling engines, however, since 1970 General Motors converted their license from an exclusive agreement for North America into a non-exclusive agreement. Ford Motor Company has recently obtained a license for work on the Stirling engine.

Probably the most active organization (in addition to Philips in the Netherlands) at the present time is United Stirling. This company has an aggressive program which is generally directed to heavy duty vehicle and stationary Stirling engines. One of their most promising versions is a V-8 engine. This engine is effectively adapted for power levels higher than the 150-160 HP class which is required for the OAP representative automobile.

Of the several configurations of the modern Philips Stirling engine, all have about the same thermodynamics, friction losses, and performance with a given working fluid. Engines have been fabricated for a wide range of power levels, varying from single cylinder engines operating at 3 KW (Allison and General Motors Research), to multi-cylinder engines operating at 30, 40, 200, 400 HP (Philips).

Three working fluids have been used for the Stirling engines: air, hydrogen, and helium. Air was used in pre-1940 era (hot air engine) and, in general, the efficiency was of the order of 10% at low operating speeds. One innovation introduced by Philips concerned the recognition of the importance of (1) maintaining high conductivity and specific heat of the working fluid, (2) minimizing the void volume of the heater, the regenerator, and the cooler, and (3) minimizing the fluid pressure losses in the closed thermodynamic circuit. For vehicle applications, either helium or hydrogen can be used as the working fluid, with hydrogen having the distinct advantage with respect to high speed operation, specific power output and efficiency. At a gross efficiency of 40% (which is often quoted as a practical operating value) the engine attains almost 60 HP/liter on efficiency. For automobile engines, where size is of prime importance, hydrogen should be favorably considered as the working fluid since a given power level can be achieved in an engine having only 64% of the displacement and, therefore, about 64% of the engine bulk volume of that required with helium. The weight of the engine should decrease correspondingly. The effect of working fluids on performance will be discussed at greater length in Section 6.4.

6.2 THE PRINCIPLE OF THE STIRLING ENGINE

The Stirling engine is a hot gas external combustion engine which compresses a quantity of gas at low temperature and expands it at high temperature. Theoretically, the Stirling engine has the same ideal efficiency as a Carnot cycle operating between the maximum and minimum gas temperature. The cycle consists of the following four processes:

1. Constant temperature compression, during which waste heat is rejected.
2. Constant volume addition of heat which was stored in the regenerator from the preceding cycle.
3. Constant temperature addition of heat as the piston moves from minimum to maximum displacement.
4. Constant volume storage of heat as the working fluid returns to the initial low temperature condition.

The gas temperature is changed periodically by causing a "displacer piston" (hereafter simply called "displacer") to transfer the gas back and forth between two spaces, one at a fixed high temperature, and the other at a fixed low temperature (see Figure 6.1). If the displacer is raised, the gas will flow from the hot space via the heater and cooler ducts into the cold space; if the displacer is moved downwards, the gas will return to

the hot space along the same path. During the first transfer stroke the gas has to yield a large quantity of heat to the cooler; an equal quantity of heat has to be taken up during the second stroke from the heater. The regenerator shown in Figure 6.1 is inserted between the heater duct and cooler duct in order to prevent unnecessary wastage of this heat. It is a space filled with a compact, high density, porous material to which the hot gas yields heat before entering the cooler; when the gas streams back, it takes up the stored heat again prior to its entry into the heater.

The displacer system, which serves to move the gas through the heating and cooling elements periodically, is combined with a power piston (hereafter simply called the "piston") which compresses the gas while it is in the cold space and allows it to expand while in the hot space (all dead spaces in cooler, heater, etc. being disregarded). Since compression takes place at a lower temperature than expansion, a surplus of work results. Figure 6.2 shows four states of the cycle through which the working fluid passes. The fluid is assumed to undergo displacement as a function of time as plotted in Figure 6.3. The ordinate in band (E) represents the variation in the volume of the hot space, and the ordinate in band (C) represents the variation in the volume of the cold space. The volume variations are plotted separately in the lower part of the diagram. Figure 6.4 shows the pressure volume diagram of the cycle represented by the cyclic motion of Figure 6.2.

Referring to the T-S diagram in Figure 6.4, the constant volume heat input and output to and from the regenerator is shown to be equal for the case where constant specific heat is assumed. Thus, for heat input to the regenerator,

$$Q_{IV-I} = C_v (T_3 - T_1)$$

and for heat recovered from the regenerator, where $T_4 = T_3$

$$Q_{III-I} = C_v (T_3 - T_1) \qquad \text{where} \quad T_2 = T_1$$

The net cycle efficiency, then, becomes

$$\eta = \frac{T_3 S_3 - T_1 S_1}{T_3 S_3}$$

or

$$\eta = 1 - \frac{T_1}{T_3} \qquad \text{where} \quad S_3 = S_1$$

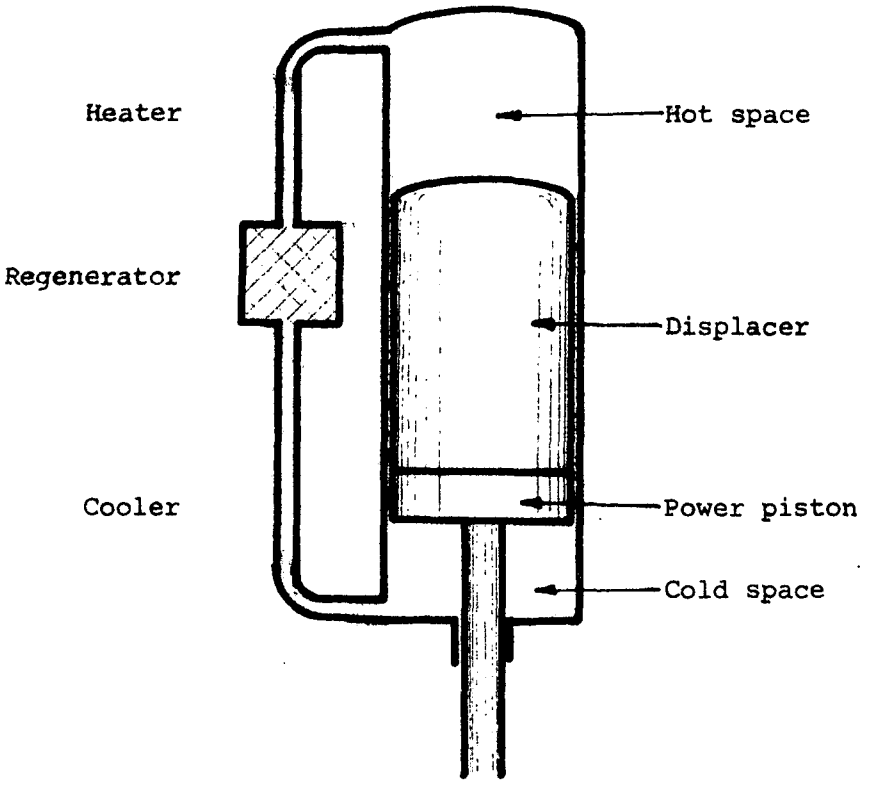


FIGURE 6.1 Principle of Displacer System. As displacer strokes, gas moves from hot to cold spaces via heater, regenerator, and cooler and back.

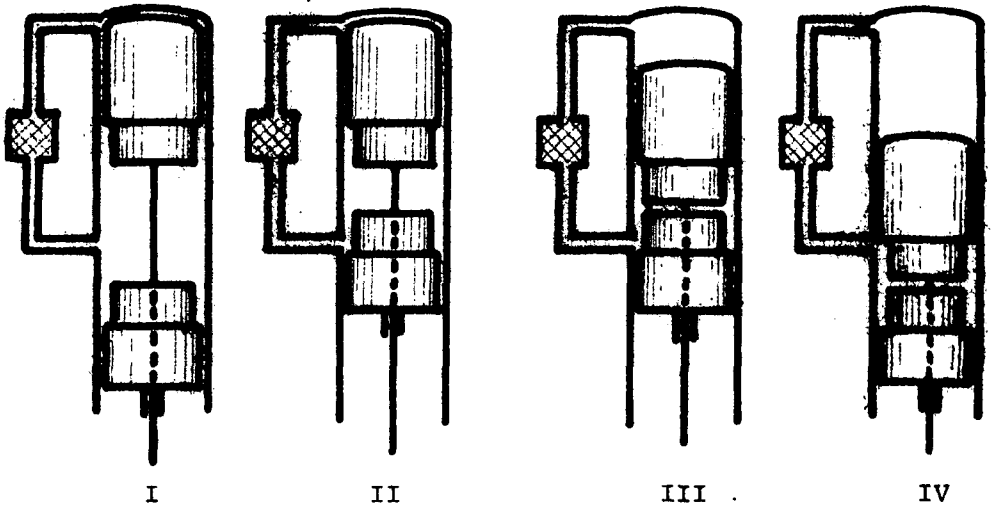


FIGURE 6.2 Principle of hot-gas Process. Assume Piston and Displacer move discontinuously, then divide cycle into four stages.

I - Piston BDC, displacer TDC, gas in cold space

II - Displacer TDC, piston TDC, gas compressed @ low temp.

III - Piston TDC, displacer mid-stroke, gas displaced to hot space

IV - Piston BDC, displacer BDC, gas expanded.

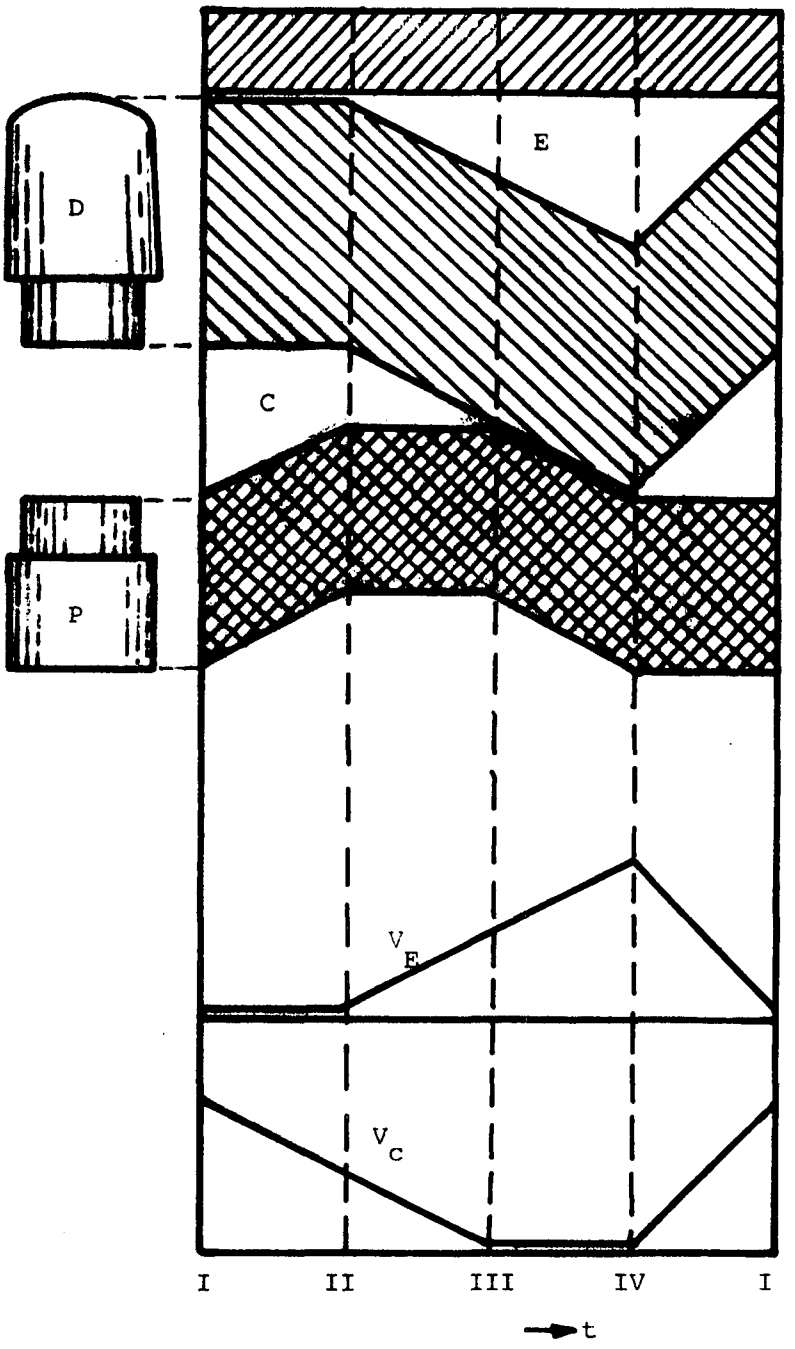


FIGURE 6.3 Discontinuous displacement of Piston (P) and Displacer (D) plotted as function of time.
Band E = volume variation of hot space (V_E)
Band C = volume var. of cold space (V_C)
Both are plotted separately at bottom.

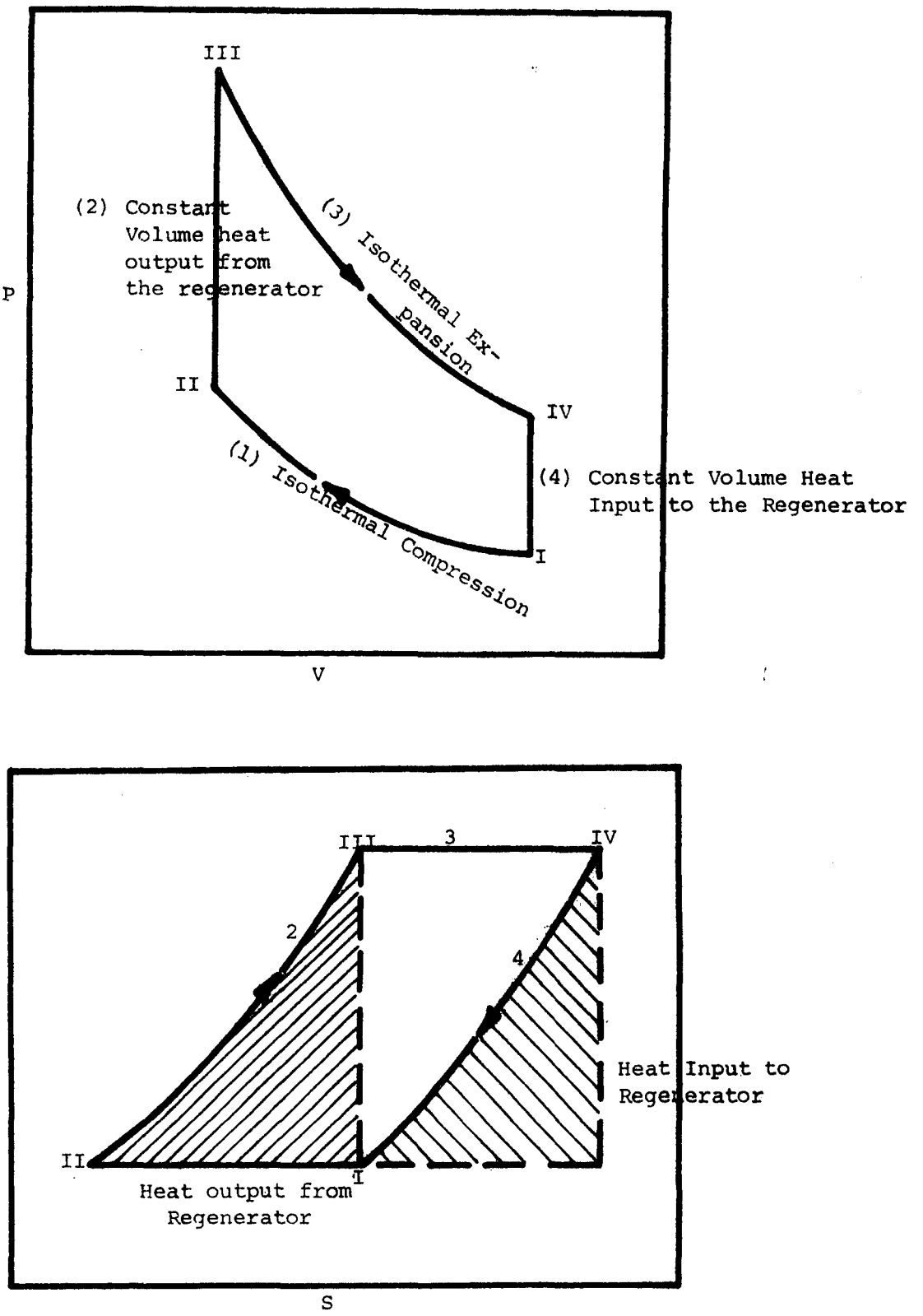


FIGURE 6.4 P-V and T-S of Ideal Stirling Cycle.

This is the expression for Stirling and Carnot efficiency. They are equivalent in the ideal case. For the temperatures used by Philips (1300°F and 60°F) the ideal efficiency is of the order of 71%. Including friction and heater losses, actual brake efficiencies of 40% are achieved, hence the actual performance is about 57% of the ideal. This is approximately equivalent to the relative cycle efficiency of a diesel engine.

In a practical version of the engine, the movement of the piston and displacer must be continuous, (not discontinuous, as they have been assumed to be in Figure 6.4) with the continuous movements being obtained with the aid of some kind of crank and connecting rod mechanism. It will then be impossible to distinguish any sharp transitions between the four stages but this will not alter the principle of the cycle (or detract from its efficiency). These movements of piston and displacer are indicated in Figure 6.5, in which the volume variations of the cold and hot spaces have again been plotted separately. The only essential condition for obtaining a surplus of work is that the volume variation of the hot space should have a phase lead with respect to that of the cold space. This is equivalent to requiring that the appropriate P-V diagram, shown in Figure 6.6 should be traced out in the clockwise direction.

6.3 DRIVE MECHANISM

There are various ways of making the piston and displacer perform the desired movements. Only those which are suitable for application to automobiles will be discussed at this time. Since 1953, most of the Stirling engines have been based on the rhombic drive system which is shown schematically in Figure 6.7 (Ref. 28). This system was found to work quite satisfactorily for the engines developed at Philips and at General Motors Research. It is somewhat complex and the engine has a high specific weight. However, it did achieve major objectives for the engine, among these being a mechanism coordinating the two pistons by a linkage which resulted in essentially linear motion, and a method of minimizing the need for lubrication. The linkage mechanism eliminated vibration, even in the case of a single cylinder engine, and although complex, achieved 90% mechanical efficiency. Engines using this mechanism were built for both single cylinder and four cylinder versions. They were fairly quiet and they ran satisfactorily, at least for the development phases.

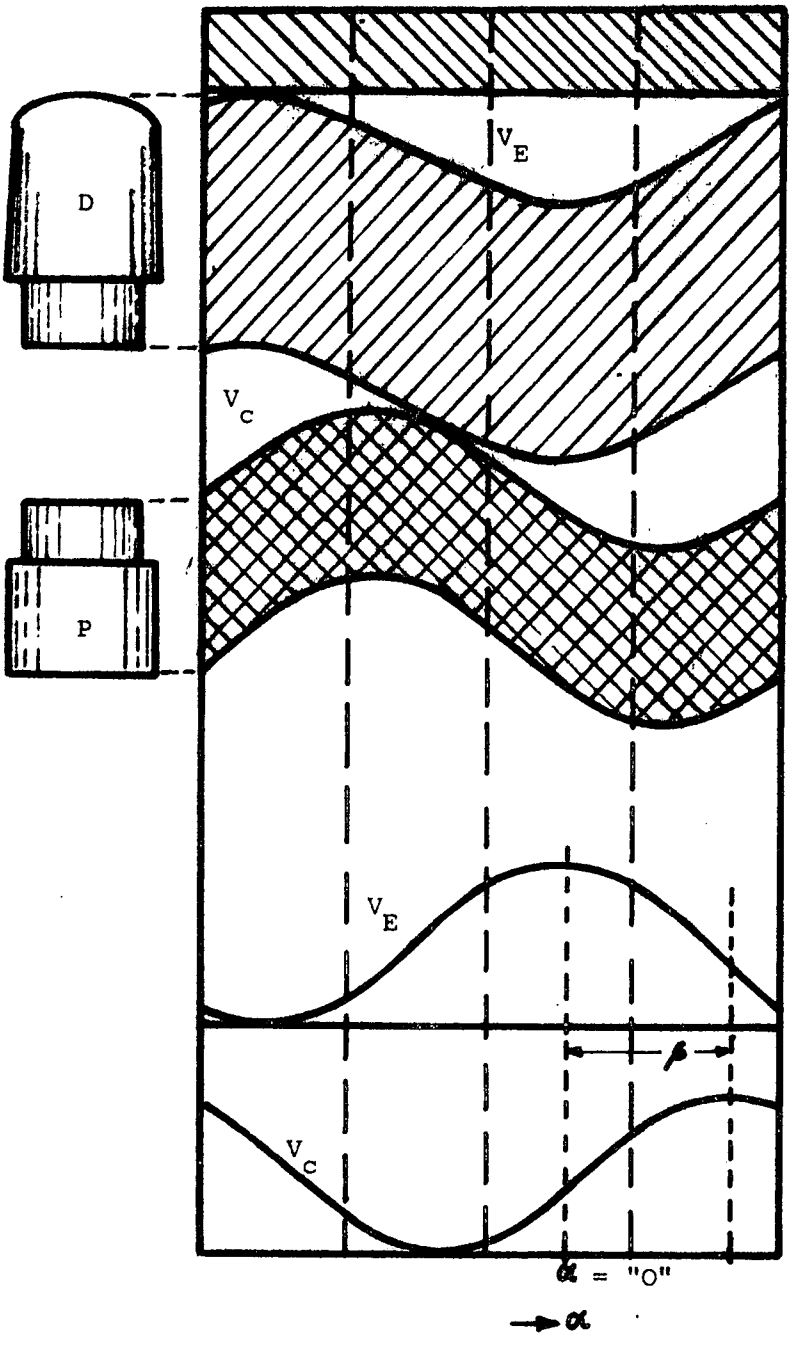


FIGURE 6.5 Continuous Motion of Piston and Displacer with Displacements Plotted as Function of Crank-angle (α). Stages of cycle now Indistinguishable.

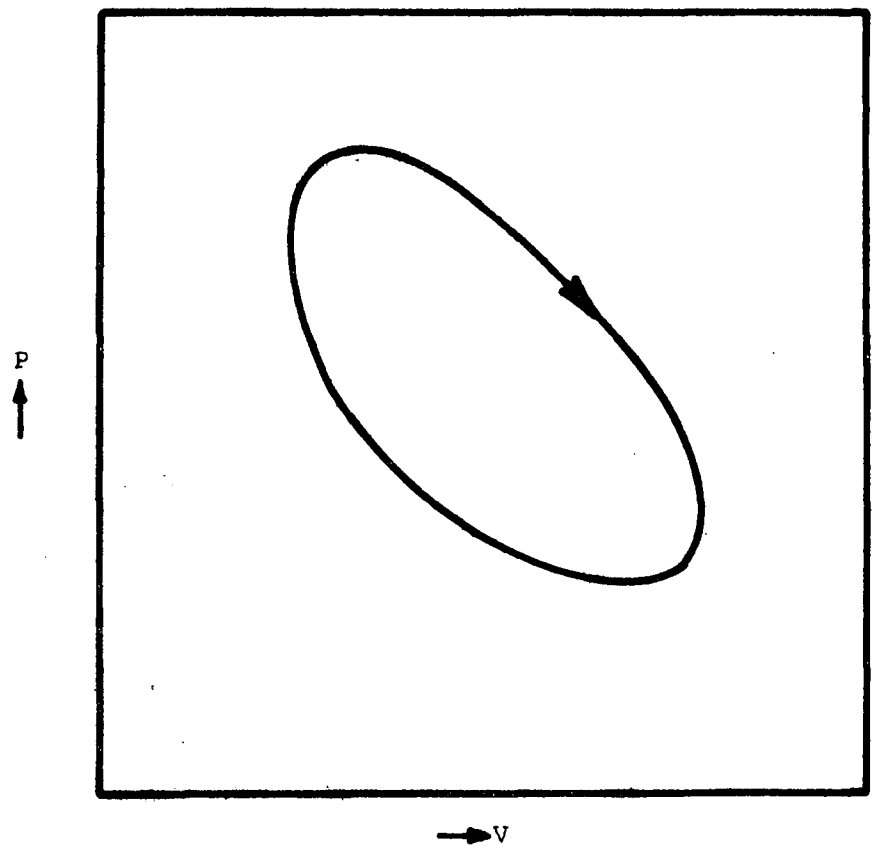


FIGURE 6.6 The P-V Diagram for the Cycle Represented in Figure 6.10.

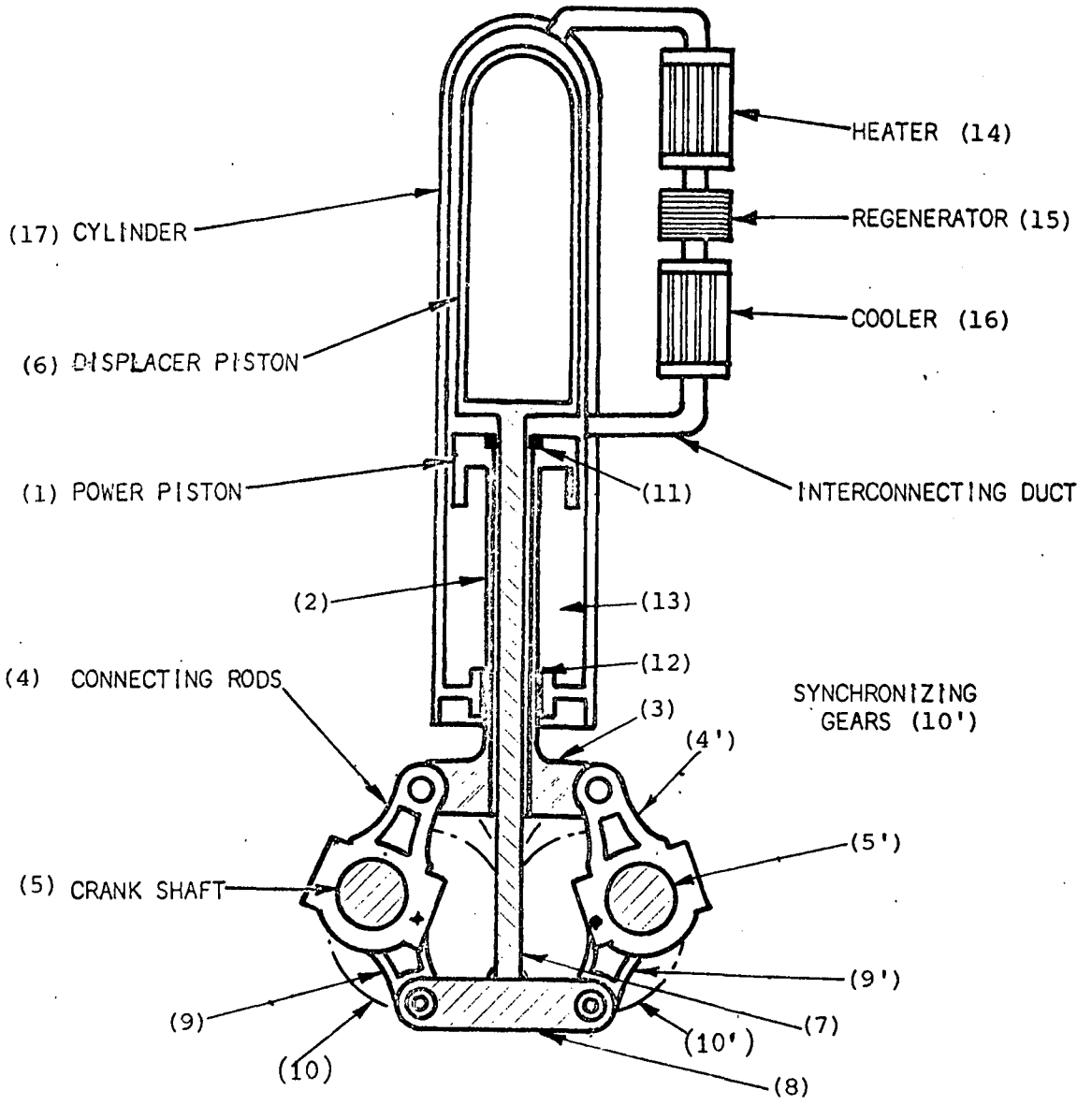


FIGURE 6.7 Schematic of Basic Rhombic Stirling Cycle Engine.

The rhombic drive mechanism allows the buffer pressure to be applied, while keeping the crankcase at atmospheric pressure. This drive mechanism also has the advantage of allowing even a single cylinder engine to be completely balanced. It consists of twin cranks and connecting rod mechanisms offset from the central axis of the engine; the cranks rotate in opposite directions and are coupled by two gears.

Fixed to the piston (1) by way of piston rod (2) is a yoke (3). One end of the yoke is linked by connecting rod (4) to crank (5), the other end by connecting rod (4') to crank (5'). The displacer is actuated by a precisely similar arrangement: the displacer rod (7), which passes through the hollow rod (2), has fixed to it a yoke (8), which is linked to cranks (5) and (5') by connecting rods (9) and (9') respectively. If (9) and (9') are given the same length as (4) and (4'), the two pairs of connecting rods will form a rhombus, of which only the angles vary when the system is in motion; it is for that reason that the name "rhombic drive" has been adopted. Gears (10) and (10') ensure exact symmetry of the system at all times. Since the two crankshafts are geared together, the entire shaft output can be taken off either.

The symmetry of the system and the coaxial arrangement of piston and displacer rods make it an easy matter to avoid pressurizing the crankcase. The seal (11) for the displacer rod is inside the hollow piston rod. One more seal located around the piston rod (12) is all that is necessary to form a comparatively small cylindrical chamber (13) under the piston, which is separate from the crankcase. This "buffer space" can be filled with gas at the desired buffer pressure. The minimum permissible volume of the buffer space is determined only by the range within which it is desired to limit the pressure variations inside the chamber. In a multi-cylinder engine the buffer chambers can be inter-connected; this allows the volume of the individual spaces to be made even smaller. The piston and displacer movements of the rhombic drive are displayed graphically in Figure 6.8.

There are several mechanisms in addition to the rhombic drive which can be used to achieve the proper phasing of the gas flowing through the heat exchangers. One of these consists of a 4-cylinder, double-acting engine using a swash-plate to convert reciprocating motion into rotary motion. For this design, used by G. M. Research, one double-acting piston is used for each of the four cylinders, and the heater tubes for all cylinders are heated by the flame of a common combustor as shown in Figure 6.9. Each of the cylinders has a separate regenerator and waste heat exchanger (cooler), as indicated schematically in Figure 6.10. The single piston is hollow

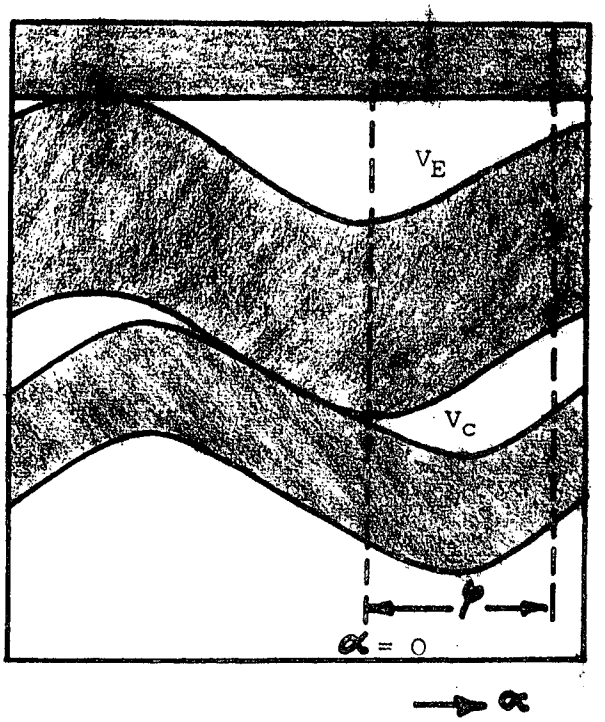


FIGURE 6.8 The Volume Variations of V_E, Hot Space, and V_C, Cold Space, of a Hot-Gas Engine Using a Rhombic Drive Mechanism.

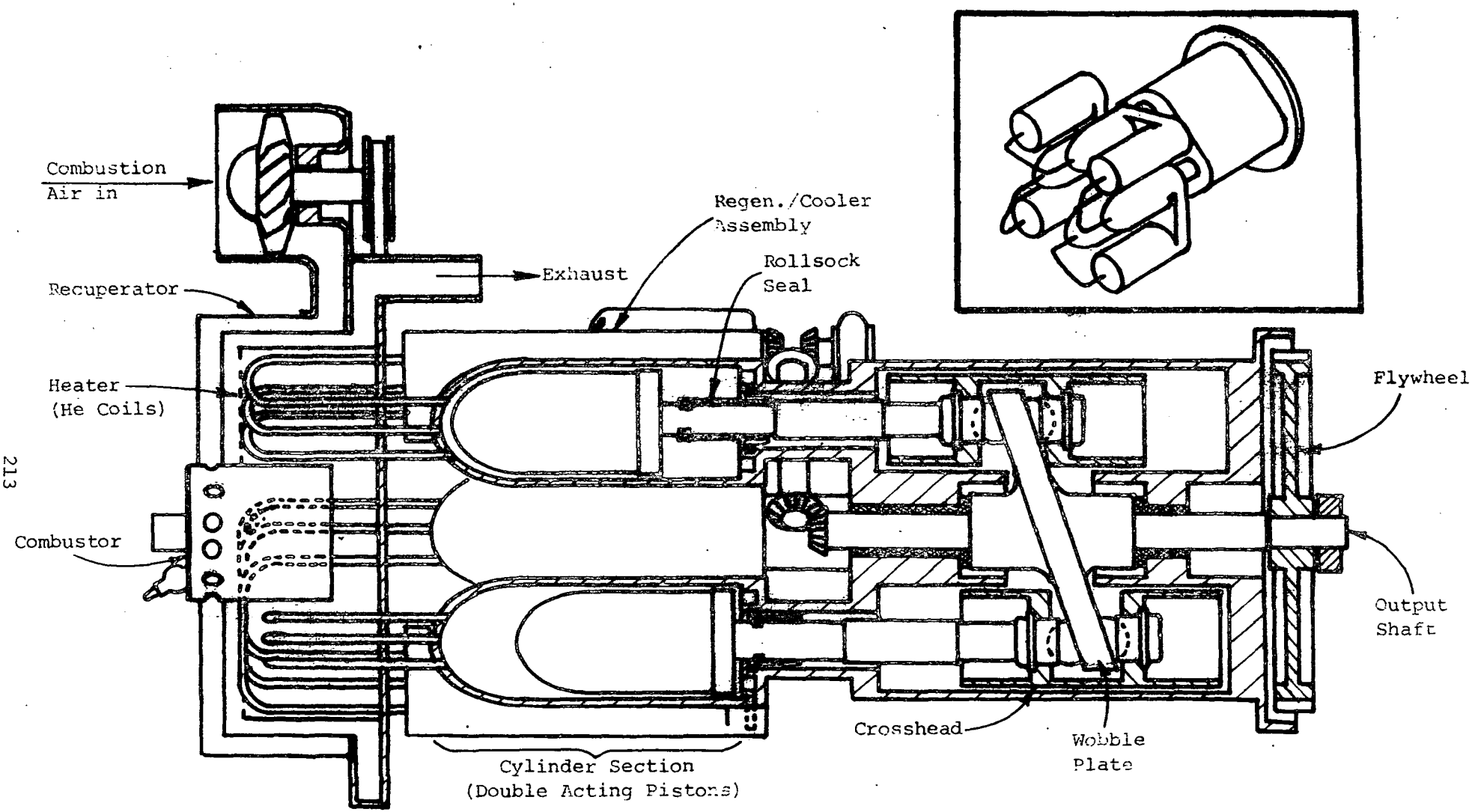


FIGURE 6.9 Stirling Cycle Swash-Plate Drive Engine.
3.1 lb/Hp, 160 HP.

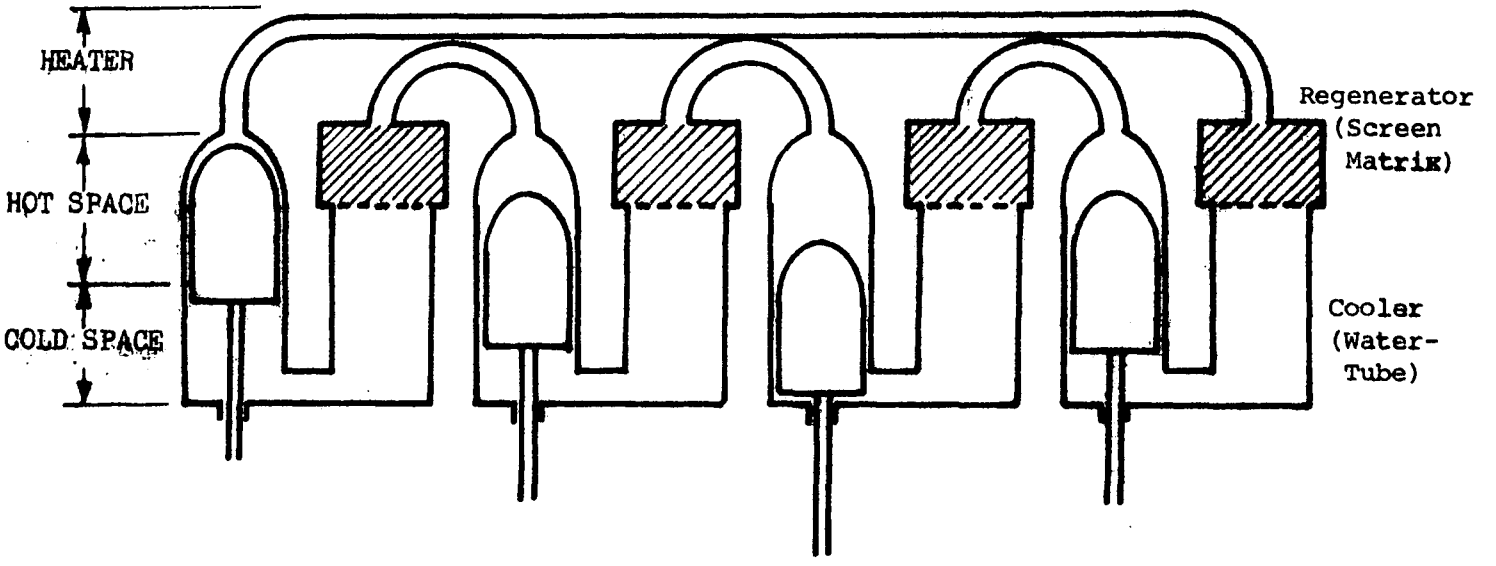


FIGURE 6.10 Schematic Representation of Double Acting Stirling Engine Flow.

to minimize heat transfer from the hot space to the cold space. A "roll sock" seal is located on the piston rod below the piston to eliminate flow of gas to the crankcase or lubricant to the under-side of the piston.

It is essential that the pistons and rods be restrained to provide essentially true axial motion so that, at the lower end of the pistons, leakage can be controlled by using a close-clearance gas seal between the pistons and cylinders, instead of the conventional piston ring and oil lubricated bearing surface.

The desired linear motion is achieved by attaching the connecting rod to a linear bearing device (the crossheads) which brackets the outer diameter of the swash-plate and includes two socketed hemispherical bearings for load transmission to the fore and aft faces of the swash-plate. With the roll-sock seal on the piston rod (below the piston), the linear bearings can be lubricated effectively with no lubricant contaminating the upper gas circuit.

A 4-cylinder swash-plate of the type shown in Figure 6.9, designed for a 160 HP automotive engine using hydrogen or helium as the working fluid would have a length, of the basic engine, of 36 inches, with a diameter of approximately 18 inches. (With accessories, the engine height and width are 24 x 22 inches.) In comparison, General Motors designed a 4-cylinder, 200 HP in-line Stirling engine with the dimensions of the cylinder block being approximately 47.2 inches long, 47 inches high, and 24.5 inches wide.

An advantage of the swash-plate configuration is that one combustor can be designed to provide uniform distribution, simple preheater design, and ease of gas recirculation. Contrasted with this is the problem of transferring heat to a multi-cylinder in-line engine without using individual combustors for each cylinder. Clearly, the system integration problem is more complex in the case of the in-line engine.

6.4 GENERAL PERFORMANCE

The available data on the Stirling engine must be abstracted from numerous documents in which the data are primarily used to illustrate characteristics rather than to present any one specific complete design. There appears to be no single source of data which can be adapted to an engine for direct application to an automotive powerplant. While data presented here are qualitative in the sense of illustrating principles and characteristics, extrapolated data will be presented, wherever possible, to document the performance of a specific automotive engine.

The Philips Stirling engine achieves a level of performance which approximates that of a Diesel engine with respect to (1) the maximum cycle efficiency, and (2) the generally flat efficiency characteristic relative to load and speed. The Stirling engine does not use valves and controls the power output by manipulation of the pressure level of the working fluid. To increase working fluid pressure for higher power, or to decrease working fluid pressure for lower power, working fluid is injected from or discharged into an external reservoir. A similar control system must be used for any closed cycle (including both reciprocating and turbo-machinery).

The optimum compression ratio can be shown to be strongly influenced by the practical limits of maximum and minimum cycle temperatures, and the practical limit is mostly determined by the capacity of the heater tubes and other hot sections of the engine to withstand the combustor temperature and the pressure of the working fluid inside the heater tubes. Philips and General Motors Research have chosen 1300°F (approx.) as the practical operating limit with heater tubes fabricated from heat and creep resistant stainless steels. Perhaps heater metal temperatures up to, say, 1500°F can be used for the present types of engines, but that level is generally considered to be beyond present metallurgical limits and will require considerable research and testing before this level can be considered.

With a limiting gas temperature of 1300°F, the compression ratio can be established and with the maximum and minimum gas temperatures the ideal cycle can be calculated. For the temperatures currently used, the compression ratio is found to be about 2:1. At significantly higher compression ratios, the effectiveness of the regenerator and the cycle efficiency both decrease.

For the greater part, the following data in this section is representative of experimental laboratory Stirling engines using hydrogen as the working fluid, which is typical of Philips and General Motors engine designs. Under the optimum cycle conditions, the efficiency is about 37 to 38%, rather than the value of 40% which is often quoted in many general discussions in the technical literature.

As shown in Figure 6.11, the design point performance of a laboratory engine operating with hydrogen at a coolant temperature of 60°F and a heater temperature of 700°C (1292°F) is 31 BHP with 37% efficiency. The power output can be reduced by 50%, to 15.5 BHP by reducing the heater temperature to 785°F, with a corresponding engine efficiency of 24%.

The Stirling engine must be operated at relatively high heater temperatures and relatively low cooler temperatures if high efficiency is to be achieved. Whereas the maximum cycle temperature of an Otto cycle engine will reach values somewhere of the order of 3500-4000°F, the Stirling engine operates at a maximum cycle temperature of about 1300°F. It can do this only because it is a regenerative engine which operates at low pressure ratio.

If the cycle pressure is used as the primary control method, a broad power range can be achieved with only a modest effect upon cycle efficiency. This is shown in Figure 6.12 where efficiency is plotted as a function of engine speed for several levels of gas pressure. As the operating gas pressure is decreased from a value of a 1991 psi down to 1000 psi at 1500 RPM the peak efficiency drops from approximately 38% down to 34%. For speeds higher and lower than 1500 RPM the efficiency drops further. The relation of speed and maximum cycle pressure to the power output of a laboratory type engine is presented in Figure 6.13. These data show that, for this engine operating with hydrogen as the working fluid, about 40 BHP was attained at 1991 psi maximum gas pressure.

The 40 BHP laboratory engine is specifically designed with heaters and coolers for the approximate speeds shown. Higher power levels at higher speeds have been projected for Stirling engines operating at 10,000 RPM, but the efficiency suffers, dropping down to about 27%. Within the limitations of the overall thermodynamics, many changes are possible depending on the heater and cooler tubes, regenerators, working fluid, type of heater, etc.

In all of these data, the performance is recorded at 60°F coolant temperature. While such performance is representative of engines which have abundance of cooling water, as would be the case for a ship, a much higher temperature would be required for an engine which is used in an

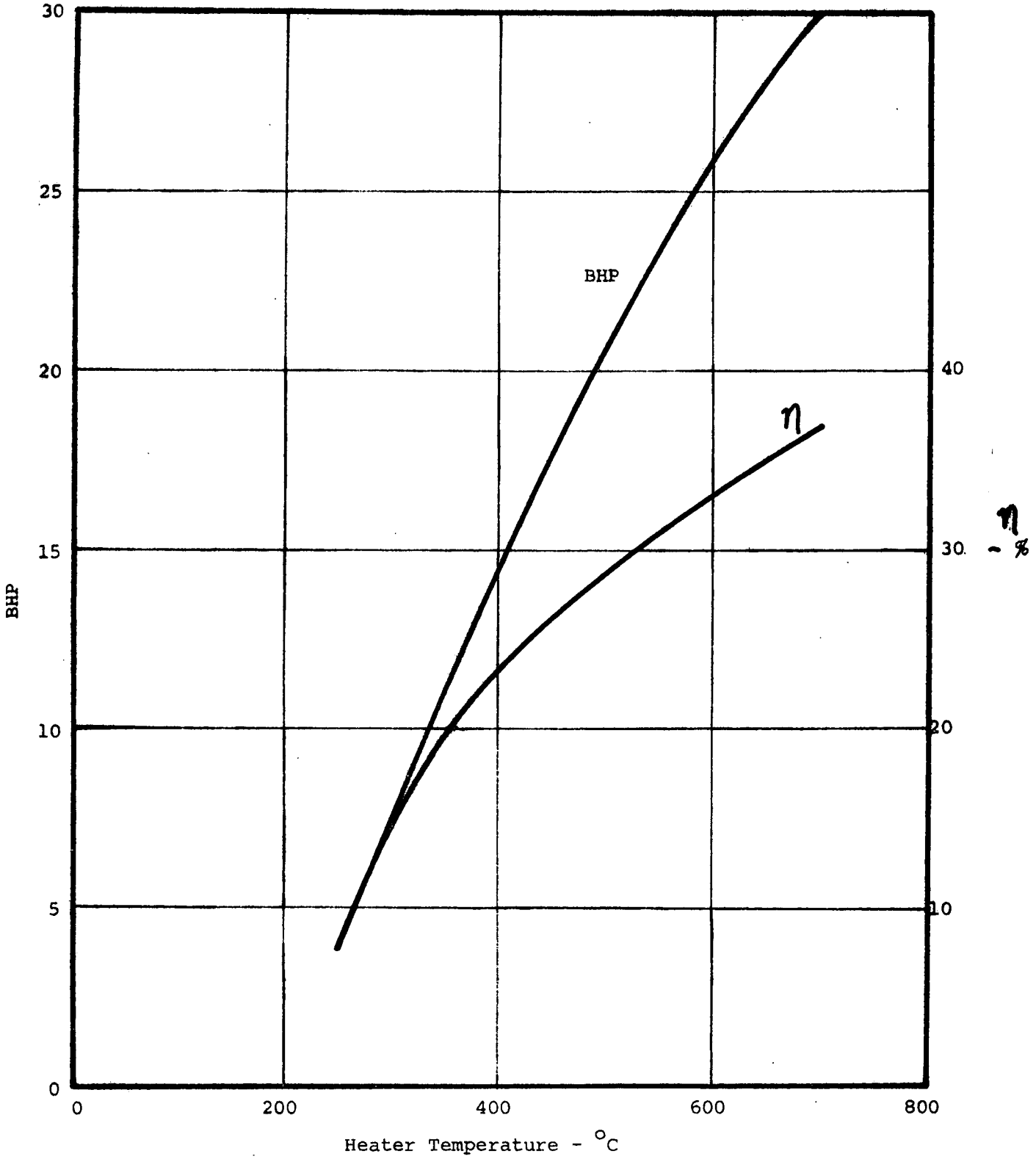


FIGURE 6.11 Brake HP and Efficiency plotted as Functions of the Heater Temperature, for Stirling laboratory engine. (N = 1500 rpm, Max. Press. = 140 kg/cm^2)[32]

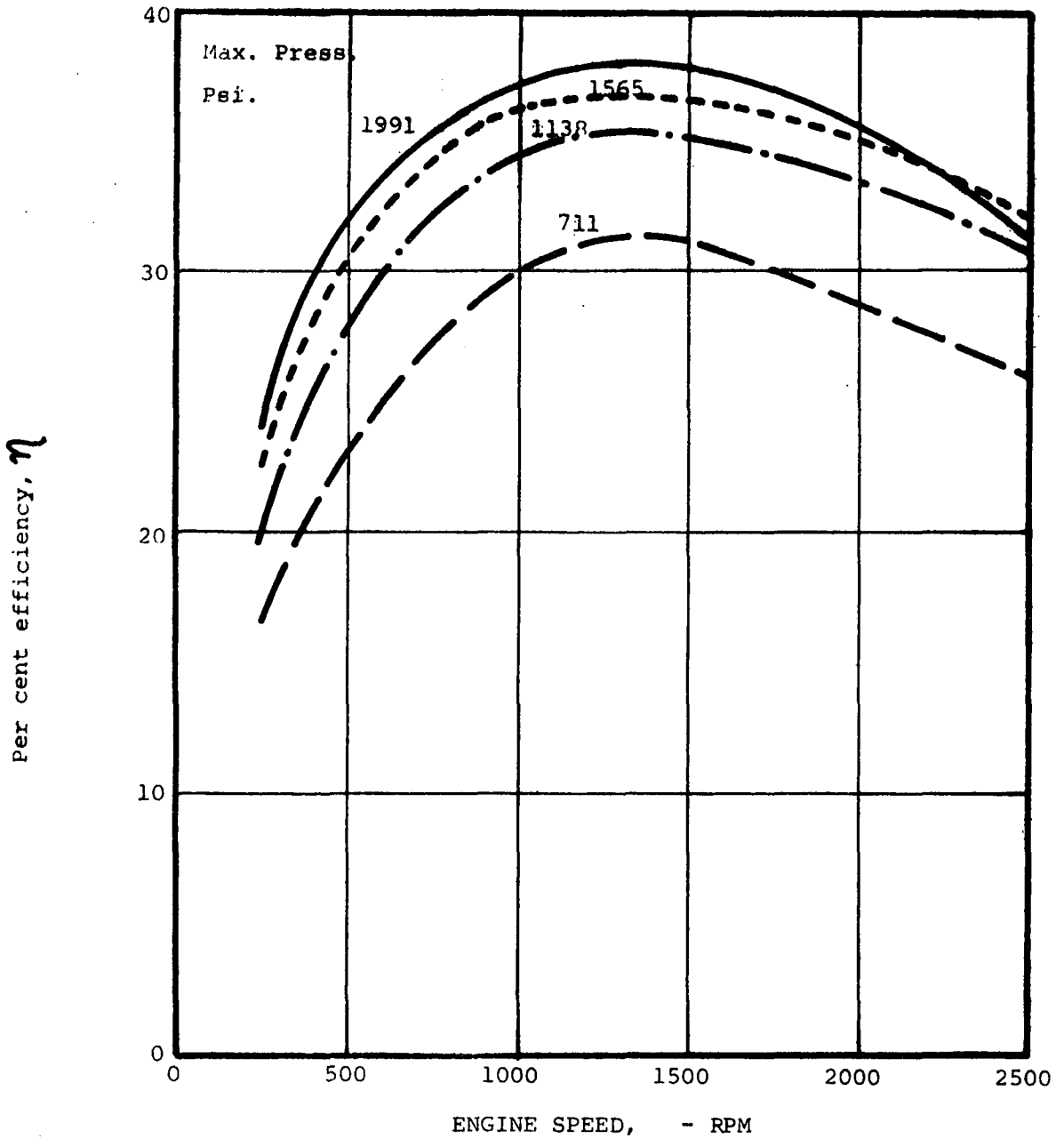


FIGURE 6.12 Efficiency of a small test engine as a function of engine speed at various values of maximum pressure.[32]

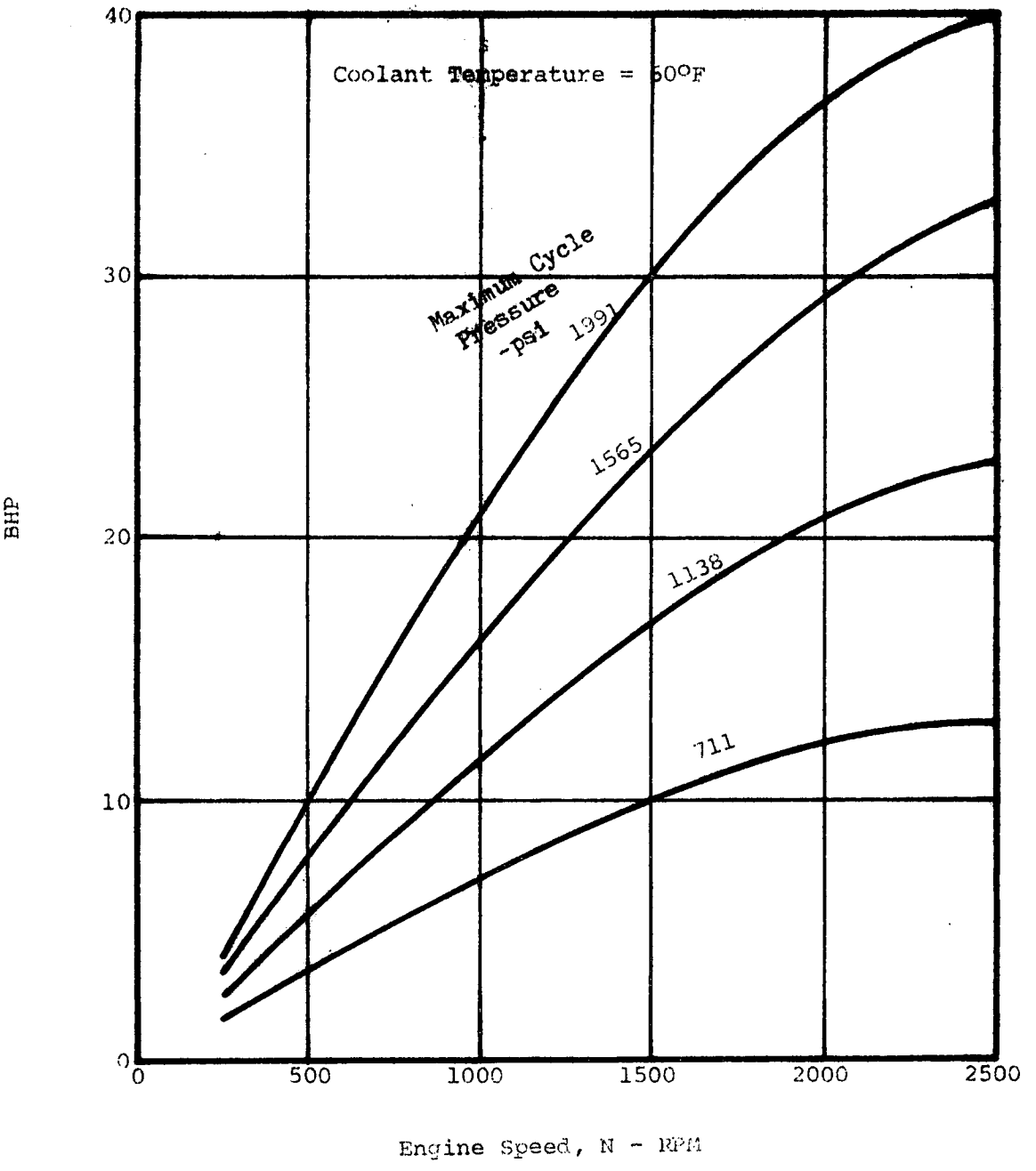


FIGURE 6.13 Brake HP of a Stirling Test Engine Plotted as a Function of Engine Speed for Various Values of Maximum Pressure with Hydrogen as the Working Fluid. (45)

automobile because the cooling water must be cooled by a radiator. In order to obtain an effective temperature difference in a radiator, a coolant temperature of the order of 130°F is required. The effects of coolant temperature on both efficiency and power are significant. At 20°C (68°F) an efficiency on the order of 38% is indicated in Figure 6.14. When the temperature is increased to 54.5°C (130°F) the efficiency drops to about 34%. On this basis, the overall engine efficiency drops about 4 points from the laboratory engine performance level to that practical in an automobile engine. This corresponds to a loss in power of 10%.

Typical heat balance data are plotted in Figures 6.15 and 6.16 for a representative laboratory engine. Figure 6.15 shows the effect of BMEP on power output. From 30% to 100% BMEP, the relative power increases from 30% to 38%, respectively. Cooling water heat rejection, heater exhaust gas and radiation losses, and friction losses also increase with power, although the relative percentage of the first two remains about the same while the latter decreases. The power output increases with BMEP up to, and presumably, beyond 26 kg/sq. cm., or 370 psi BMEP.

The overall performance of a representative laboratory Stirling engine is shown in Figure 6.16. The map presents BMEP as a function of engine speed for a range of BSFC and BHP values. It is to be noted that the minimum BSFC (.369 lb/Bhp-hr) occurs in the region of 1100 to 1500 rpm for this engine. As this chart is more or less representative of the operation of the Stirling engine, performance data for a 160 BHP Stirling engine will be extrapolated from these data.

6.4.1 Working Fluid

The working fluid which is used for the Stirling engine has a significant effect on the power output and efficiency. In general, the low molecular weight gases are far superior, especially when it is desirable to operate the engine at high speeds. While heat exchanger characteristics and the heat transfer properties are very important in themselves, the fluid dynamics and pressure drop of the gas flowing through the fluid passages predominate for a high performance engine.

While any gas can be used, low fluid friction and high heat transfer properties are so critical that most common working fluids are eliminated. The original Stirling and all designs up to the period that the modern Stirline engine was developed by Philips, used air as the working fluid. In the 1950's, a thorough fluid-dynamic and thermodynamic analysis resulted in the realization that hydrogen had the best combination of properties. Helium, having the next lowest molecular weight to hydrogen, was less

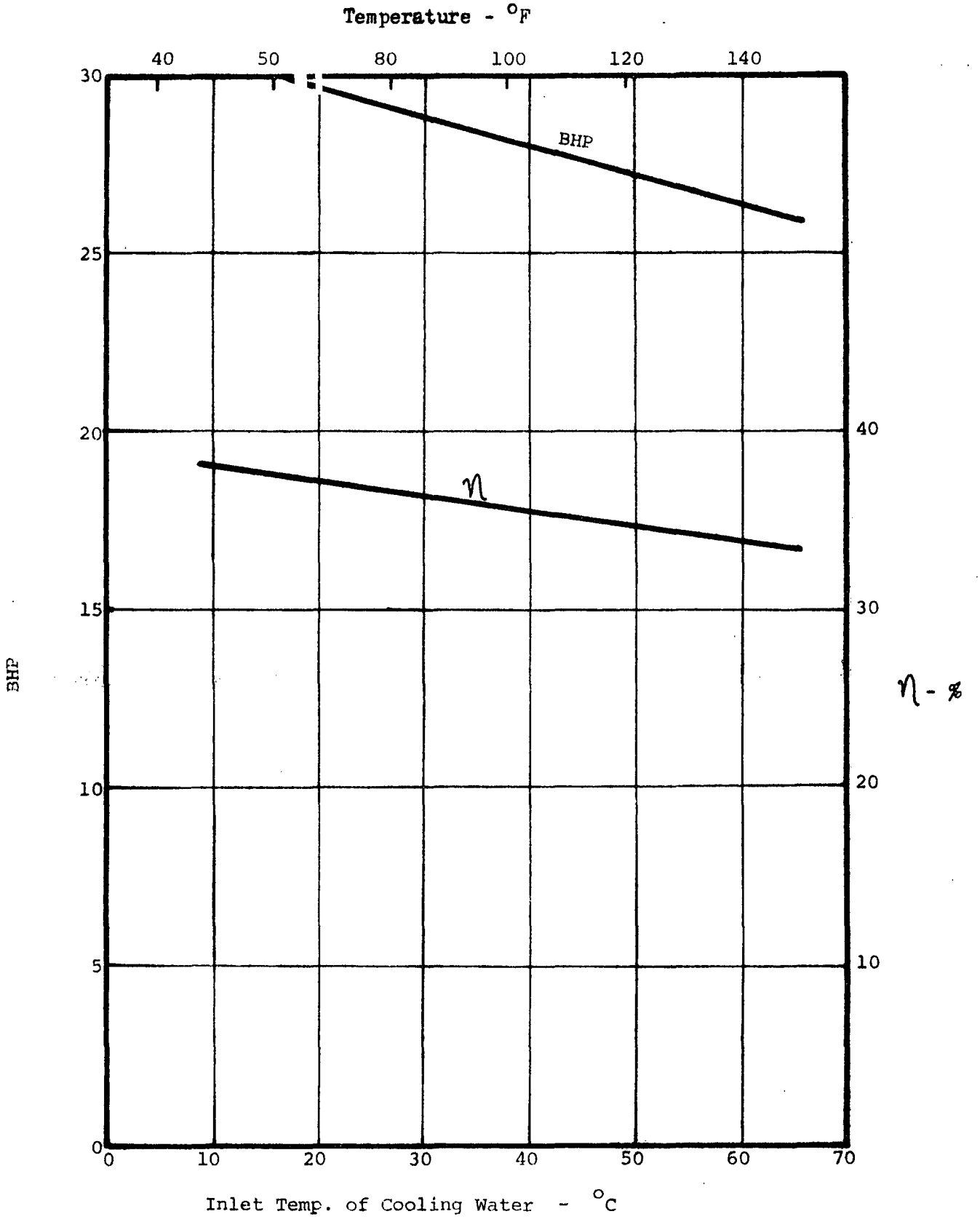


FIGURE 6.14 Brake Horsepower and Efficiency Plotted as Functions of Inlet Temperature of Cooling Water.
(N = 1500 RPM, Max. Press. = 1991 psi)
Ref. 28.

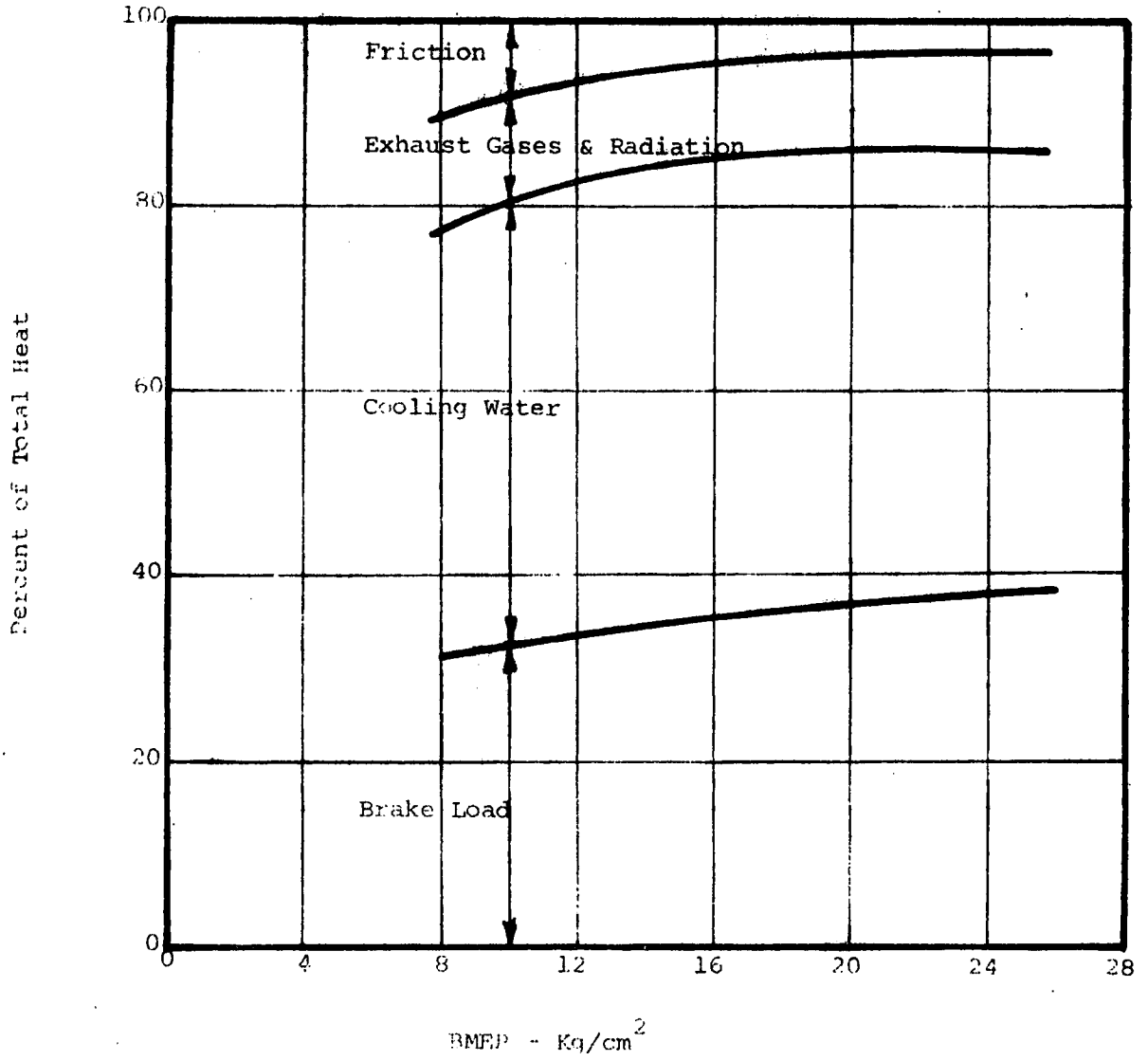


FIGURE 6.15 Single Cylinder Test Engine Heat Balance as a Function of Mean Effective Pressure (N = 1500 RPM).[45]

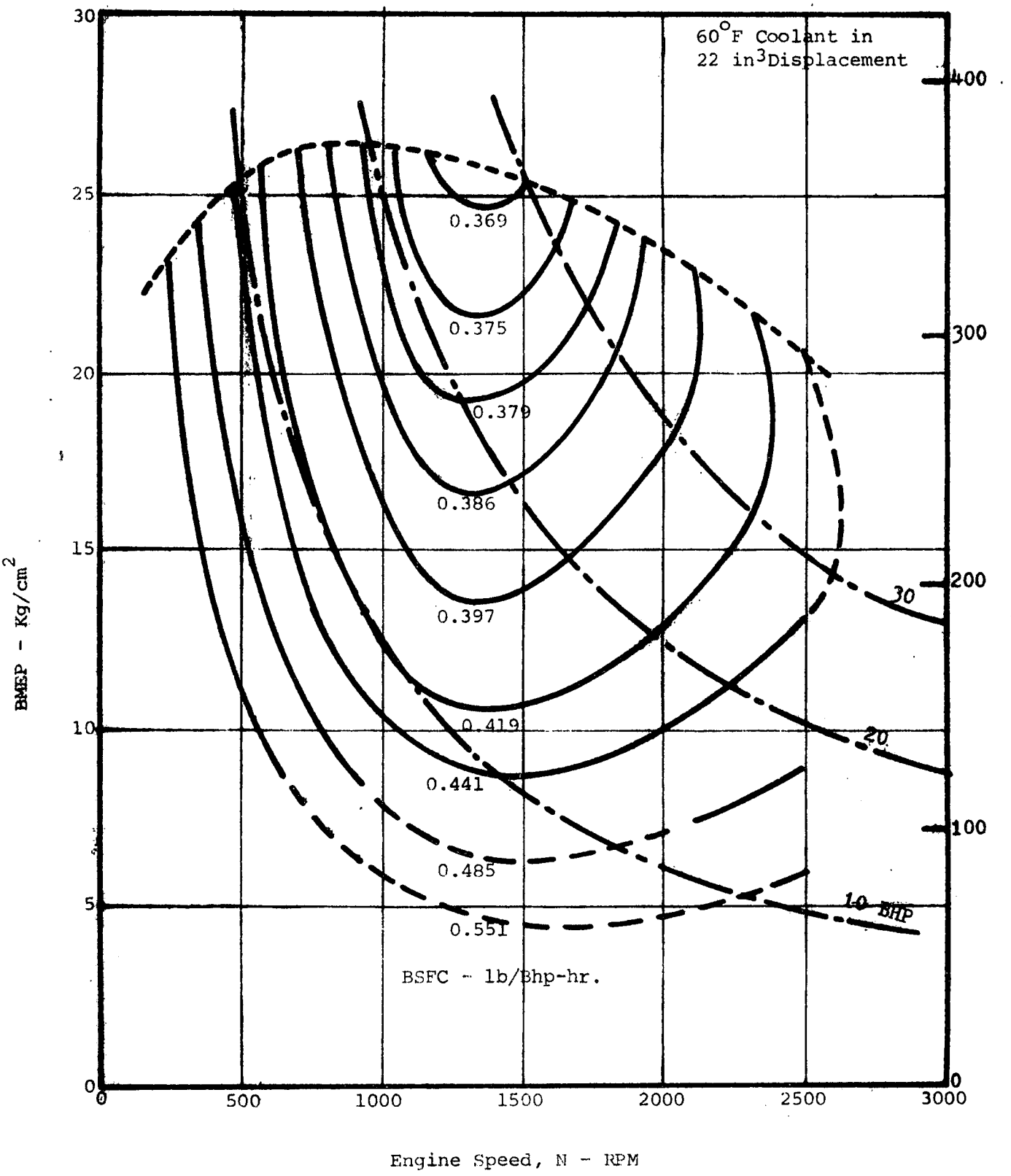


FIGURE 6.16 Lines of Constant Specific Fuel Consumption and Constant Brake Horsepower as a Function of Mean Effective Pressure and Engine Speed. (Fuel Consumption based on a Lower Heat Value of 10,000 Kcal/Kg). (45)

favorable but was superior to other gases.

Figure 6.17 compares the performance of a small, single cylinger, laboratory engine employing air, helium and hydrogen as the working fluids. As shown, air is almost as favorable as the other two in terms of maximum efficiency, but the maximum efficiency with air occurs at so low a speed that an engine so designed results in unacceptable horsepower-to-weight ratios. Helium is more effective, but hydrogen is definitely the superior with respect to efficiency and horsepower per unit volume displacement.

At the elevated temperatures experienced in a Stirling cycle, hydrogen diffuses through high temperature metals, and, while the rate is not intolerable for most applications, it cannot be used wherever a leakage cannot be resupplied. This is not a problem with helium or nitrogen. To date, no practical diffusion inhibitor has been used, but it is not inconceivable that one can be found. The engine with hydrogen as the working fluid has been chosen by Philips as the preferred working fluid for an automobile powerplant.

Considering only the performance factors, it is difficult to come to any other conclusion. However, the safety and leakage loss factors which are present with a hydrogen working fluid cannot be ignored. Those data available do not indicate what increase in fluid loss rate one should expect as the mean pressure is doubled. Nor is there any indication of the relative degree of complexity associated with safety devices which may be required to reduce the hazard in general public usage of hydrogen. Indeed, the hazard, if it exists, does not seem to have been very thoroughly investigated.

Helium has been selected as the tentative working fluid for the automotive Stirling engine analysis which follows. This is done with full recognition of the performance-weight penalty which results and is justifiable only in terms of least potential hazard.

6.4.2 Direct and Indirect Heating

The extrapolated performance characteristics for a 4-cylinder automotive Stirling engine with either direct or indirect heating are indicated in Figure 6.18. These data, generated by Philips, indicate reductions in efficiency levels as compared with the Philips data of Figure 6.17, which are based upon laboratory tests of small single cylinder engines. It should be noted that there are, in addition to configuration differences, significant differences in maximum engine speed, pressure levels and temperature levels.

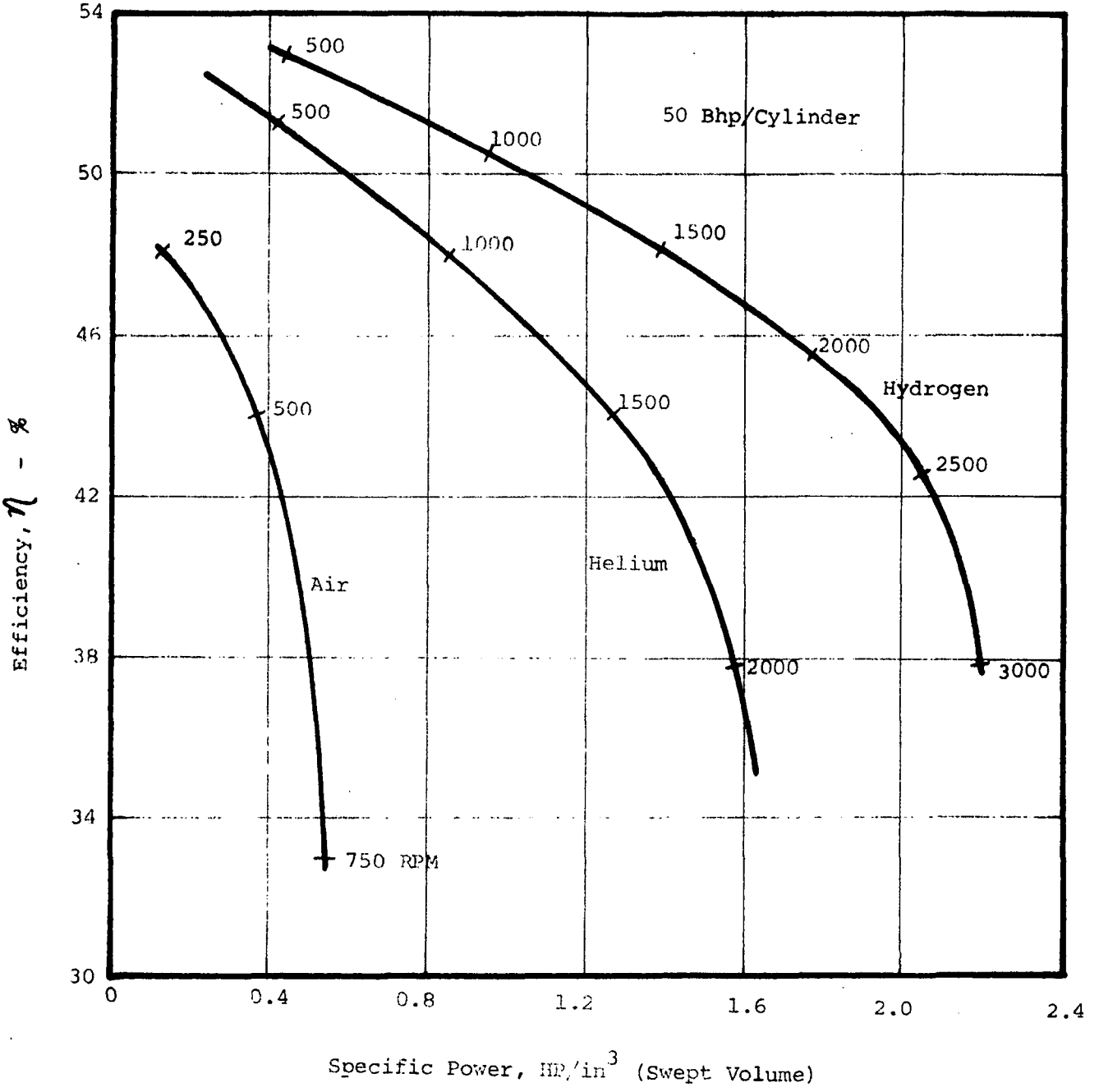


FIGURE 6.17 Relation of Single Cylinder Laboratory Stirling Engine Performance.[28]

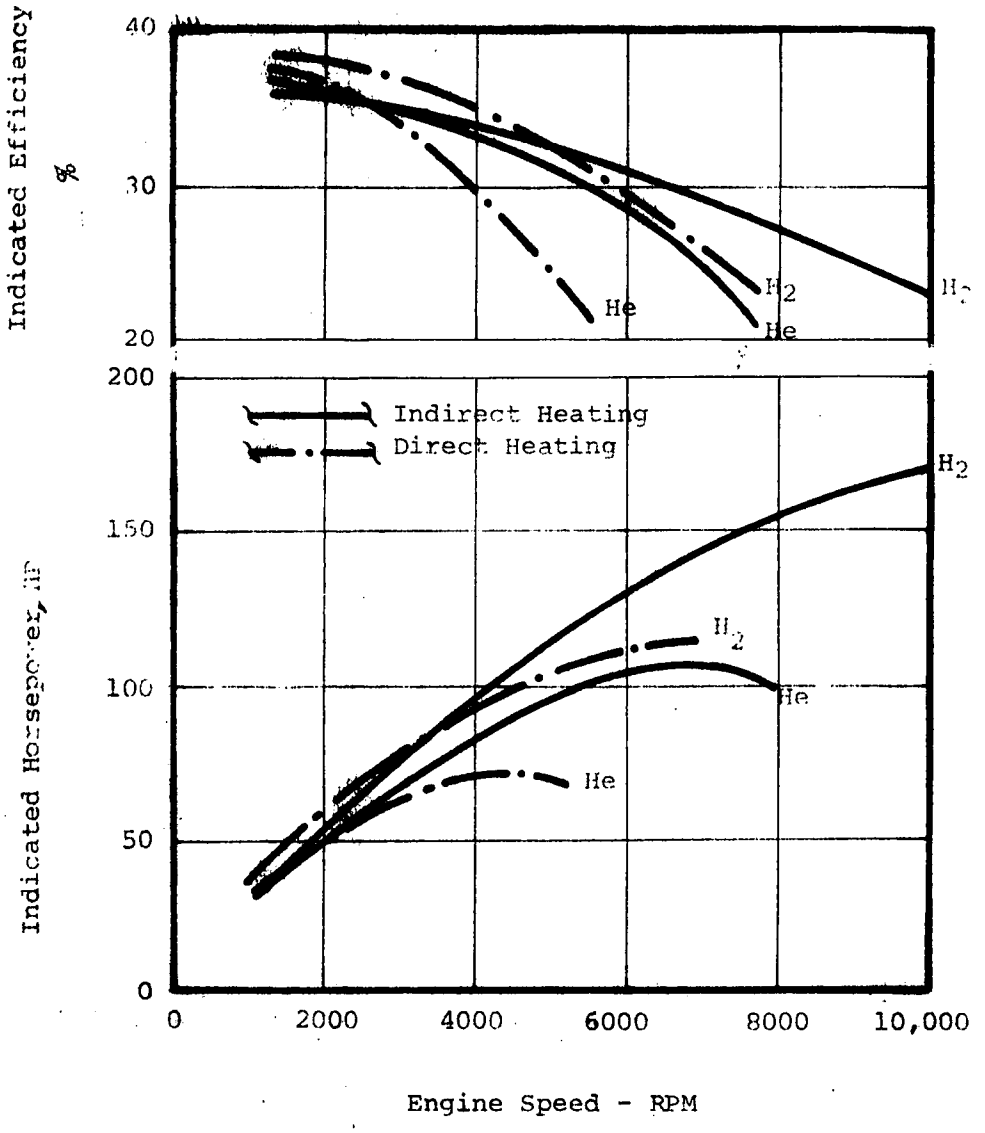


FIGURE 6.18 Predicted Four Cylinder Automotive Stirling Engine Performance with Helium or Hydrogen Working Fluid.
(Ref. 28)

It is clear that the heat pipe displays an advantageous heat transfer rate as compared to the design where the external tube surfaces are heated with direct combustion gases. A heat pipe employing liquid metal, NaK or sodium as a heat transfer medium, not only increases the overall heat transfer rate significantly but also results in less dead volume inside the tubes and, therefore, improves the performance of the cycle. The improvement in efficiency and allowable operating speed with indirect heating is significant.

The problem of pumping the liquid metal has been conveniently solved by Philips by using a heat pipe which transfers the heat through a combination of convection and capillary pumping of the liquid metal rather than using a mechanical pump. The liquid metal heat pipe has several times the conductivity of an equivalent diameter of pure copper and should be far superior to any other heat transfer device. However, in terms of performance, it seems best applied where the need for high specific power outweighs economy. It should also be noted that in addition to the problems of stress reliability associated with the elevated pressures (3000 psi mean), there will very likely be some degradation of seal life, particularly if the higher pressures are combined with very high speeds.

6.5 AUTOMOTIVE ENGINE PERFORMANCE

The power and efficiency characteristics of an engine which is suitable for use in an automobile is shown in Figure 6.19. These data were extrapolated from available sources and are modified to meet the performance of the vehicle. Design parameters are itemized in Table 6-I. This engine operates at high peak pressures (3500 psi) over a pressure range of 2:1 with a mean cycle pressure of approximately 3000 psia. Helium is used as the working fluid due to the potential element of hazard with hydrogen, but the expected performance levels associated with hydrogen are noted where appropriate. This results in an increase in the peak pressure level of about 50% in order to attain the equivalent power output and to compensate for the low waste heat exchanger temperature ($60^{\circ}F$) used in the Philips data.

From the extrapolated performance data of Figure 6.19 the road load fuel economy characteristics were approximated for the 160 HP Stirling engine in a 4300 lb automobile. The road load power requirement for this 4300 lb vehicle/engine combination is presented in addition to the fuel mileage in Figure 6.20. On the basis of performance the modern Stirling engine is definitely competitive.

TABLE 6-I

DESIGN PARAMETERS FOR THE STIRLING ENGINE
FOR THE EPA 6-PASSENGER AUTOMOBILE

Power	160 Bhp
Engine Speed (rated)	3000 RPM
Maximum Cycle Temperature	1300° F
Working Fluid	Helium
Minimum Cycle Temperature (Coolant water)	130° F
Mean Cycle Pressure	3000 psi
Compression Ratio	2:1
Number of Cylinders	4
Bore	2.85 inches
Stroke	1.825 inches
Displacement	46.57 in^3
Cylinder Arrangement	Swash Plate/Barrel Config.

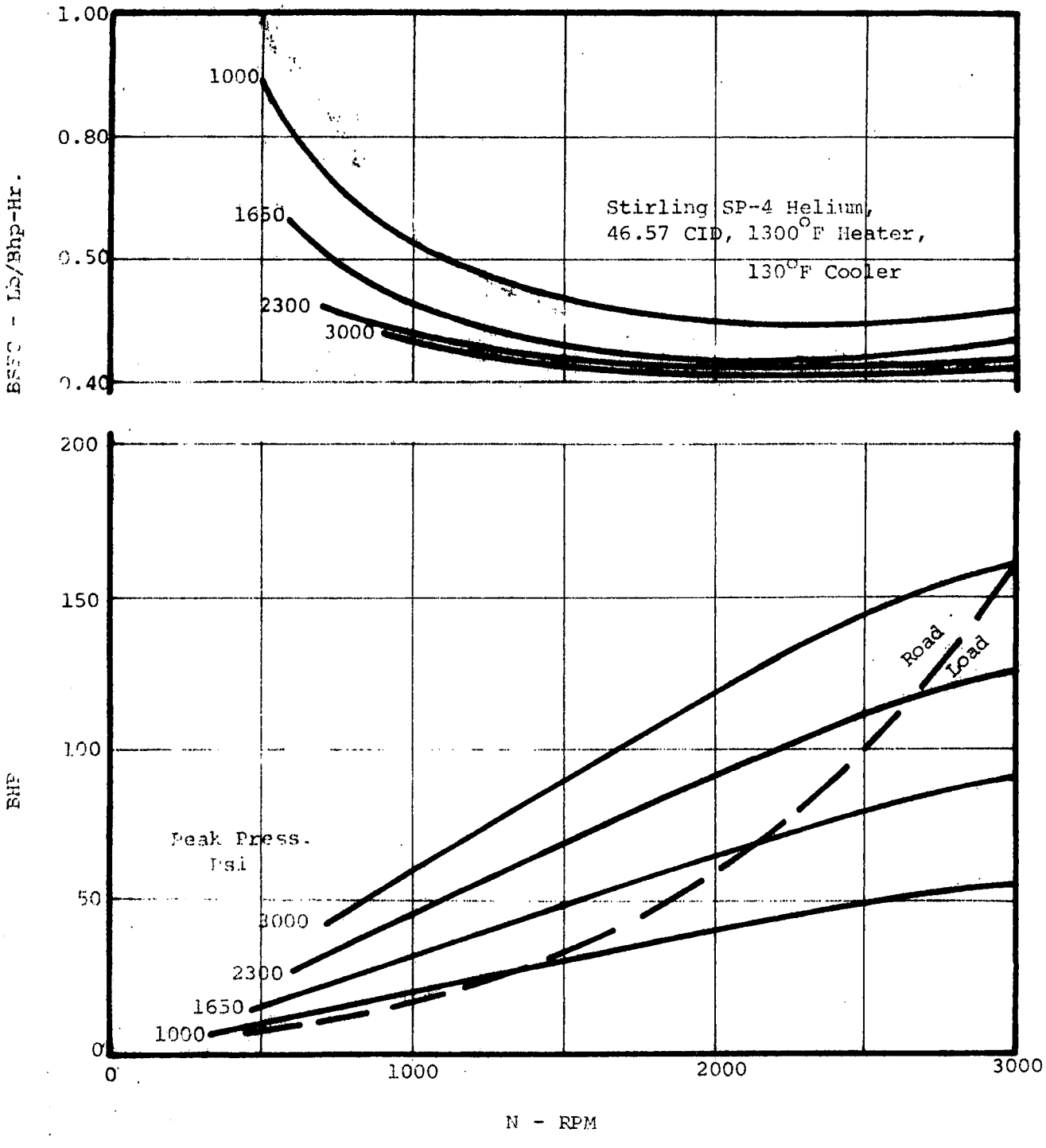


FIGURE 6.19 Estimated Performance for an Automotive Stirling Engine.

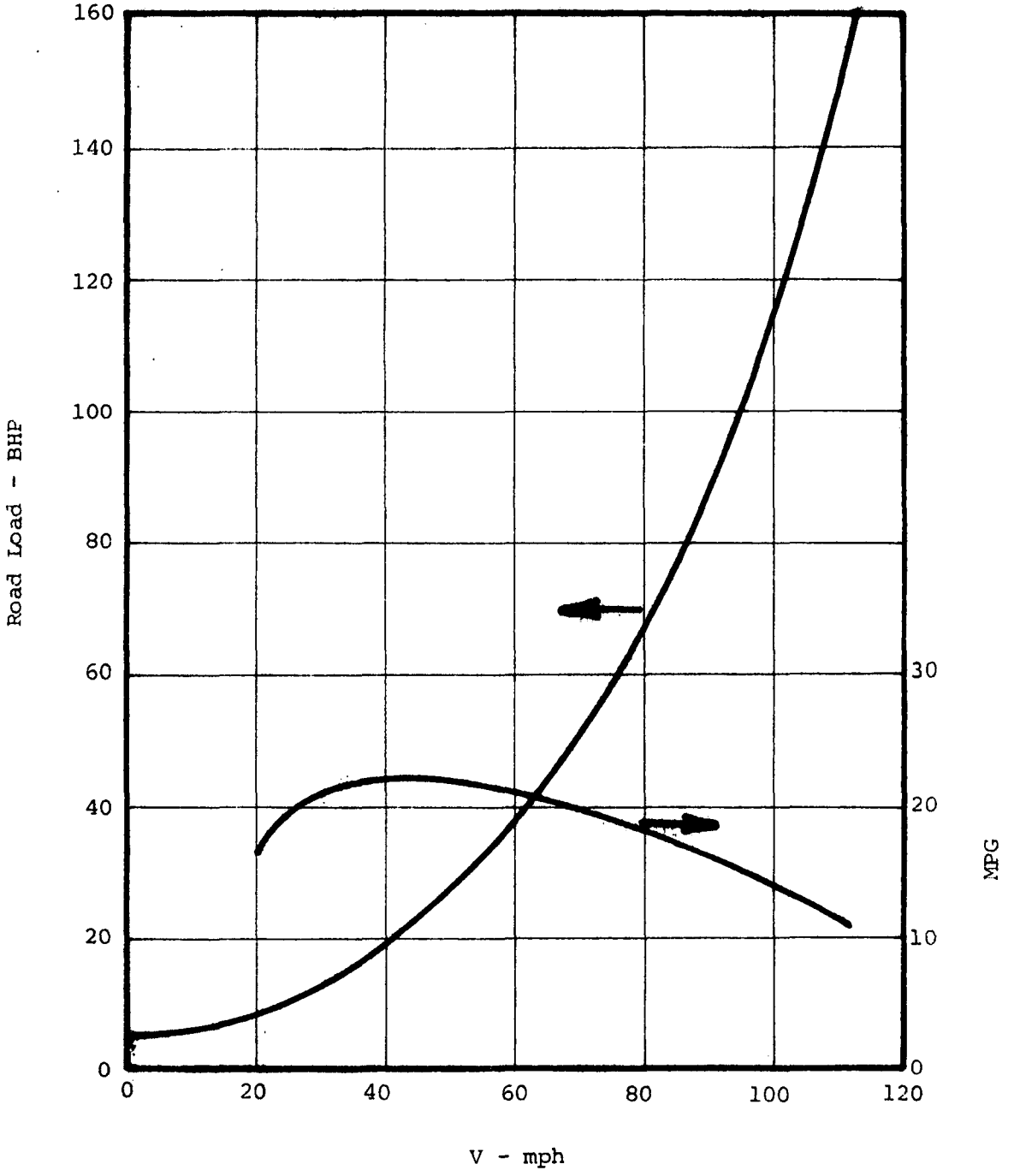


FIGURE 6.20 Automotive Stirling Engine Road-Load Fuel Mileage, at 130°F Cooling Water Inlet Temperature. (47 CID, 160 BHP @ 7000 RPM, He Working Fluid, Direct Heating.)

The reader may, at first, be surprised at the low efficiencies, as compared to previously presented Stirling performance, indicated in Figures 6.19 and 6.20. It must be kept in mind at all times that even the most modern Stirling engines which were actually tested were far from acceptable for those conditions peculiar to automotive application. Changes dictated by safety and operational practicality (working fluids, engine speeds, and minimum cycle temperatures) substantially reduce the efficiency of the automotive Stirling engine in comparison to laboratory and other special applications.

6.5.1 Stirling Engine Emission Characteristics

The Stirling engine is an external combustion engine and has the general characteristics of all external combustors. In this sense, it performs in a manner similar to the Rankine cycle, the closed cycle Brayton cycle, and the External Combustion Piston Engine powerplants. Differences in the characteristics of one versus another can be traced to the specific characteristics of a particular design and to the state-of-the-art of combustor design at the present time.

It is desireable to control smoke, odor, and emissions to a minimum level over the operating spectrum. The data presented are typical of the General Motors Research Models 1036, 1036R, and GPU engines which are rated at approximately 10 HP, the Philips Models S1050 and S1210 (which are rated at 80 to 90 HP per cylinder) and the Model 3015 (which is rated at 30 HP/cylinder). Detailed discussion relative to these engines are presented in Reference 30.

Both the Philips and General Motors combustor designs use conventional technology in the approach to combustor design. A single fuel nozzle and igniter is located centrally and kerosene, JP, or No. 2 Diesel fuel is sprayed into the combustor. A preheater using heater exhaust air is incorporated into the combustor design to provide primary air, dome cooling air, and body cooling and secondary combustor air. The dome cooling air uses a series of tangential cooling slots which provide a tangential swirl to improve combustion. The maximum-to-average temperature ratio was found to be approximately 1.04, indicating good distribution of fuel and air.

In the Stirling combustion system, burning takes place continuously in a lean overall fuel ratio (30% or more excess air). High peak temperatures, excess oxygen, and adequate resident time provide for essentially complete combustion of carbon compounds from carbon monoxide. These high peak temperatures also promote the formation of nitric oxide. A

flame ionization detection (FID) meter was employed to measure total exhaust hydrocarbons and continuous sampling techniques were used for measuring the exhaust gas CO, NO, and NO_2 concentrations.

The exhaust emissions results are presented in the following three forms:

1) As functions of air-fuel ratio with a constant burner inlet air temperature.
2) As functions of burner inlet temperature at constant fuel-air ratio, and
3) As functions of engine load at constant air-fuel ratio.

The air-fuel ratio dependence was determined at both half and full loads. The concentrations of NO, CO, and hydrocarbons were converted from the volume (ppm) to a mass basis (lbs/1000 lbs/fuel) by using 28.8 units as the average molecular weight of the exhaust gas. All measured hydrocarbon concentrations were considered to be ppm of n-hexane (C_6H_{14}). The three emissions of major concern, at half load, are expressed as a function of air-fuel ratio in Figure 6.21 (Reference 30). Measurements both upstream and downstream of the preheater (hot side) are presented for CO and NO. The NO and CO were affected favorably by the introduction of the preheater, to the extent of approximately 100 ppm at 20:1 air-fuel ratio, however, this effect decreased to lesser values as the air-fuel ratio increased. Although no downstream hydrocarbon measurements were taken, it is highly probable that the hydrocarbon measurements would follow the same trend as CO concentrations--namely, a decrease with passage through the preheater due to the added residence time. The same data are presented in terms of pounds of each component per 1000 lbs of fuel in Figure 6.22.

As the air-fuel ratio increases, the temperature of the combustor decreases and results in the corresponding decrease in the heat transferred to the heater tubes of the engine. This is clearly shown as a decrease in efficiency as the air-fuel ratio increases from 20 to 40:1. Whereas the efficiency was 23% at 20:1 air-fuel ratio, it drops to 20% at 40:1 air-fuel ratio. However, since the combustor should be operating at approximately 25:1 air-fuel ratio most of the time, it appears that the efficiency at half-load is about 23%. Similar data are shown for full-load

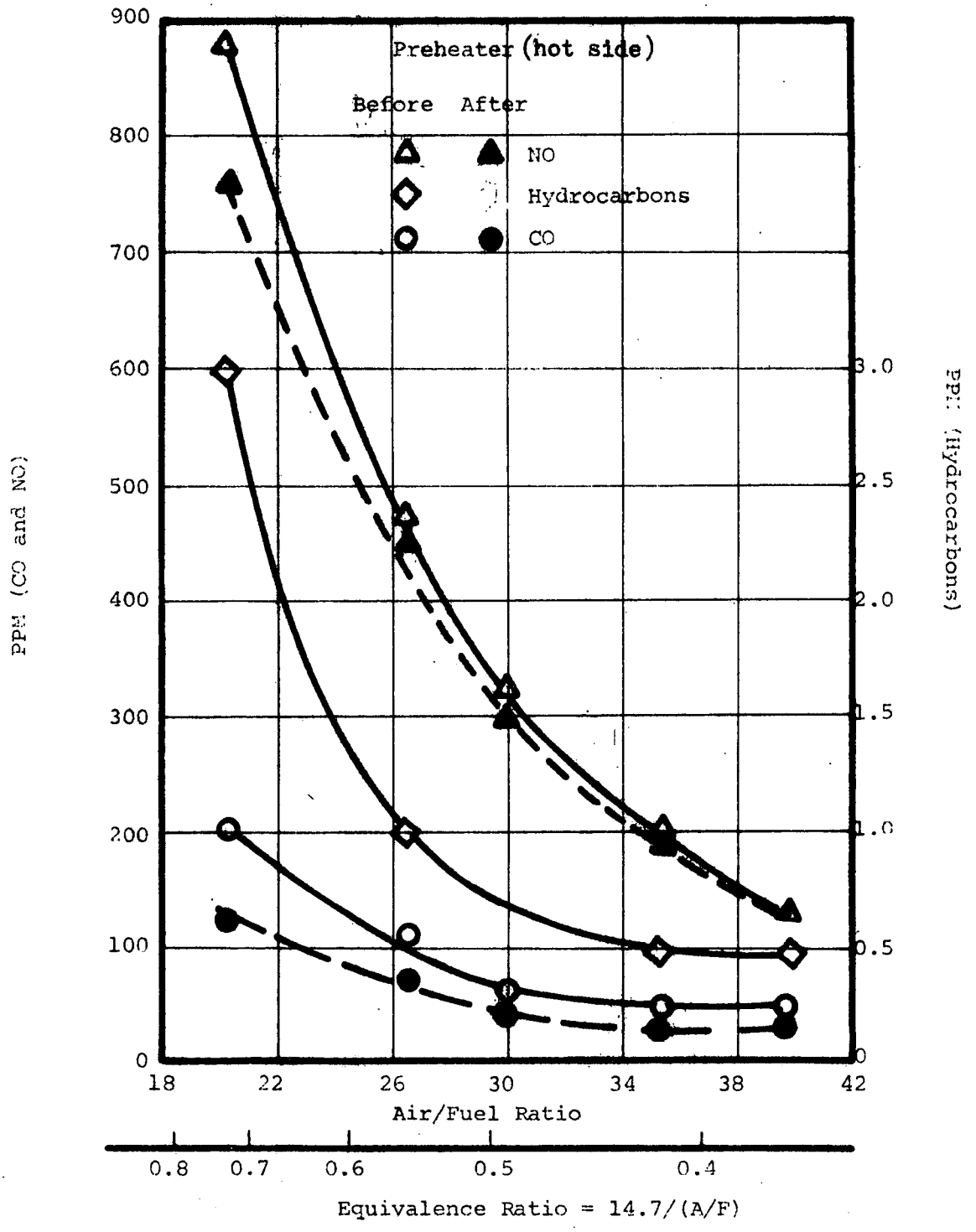


FIGURE 6.21 Effect of Mixture on Exhaust Concentrations at Half Load.[30]

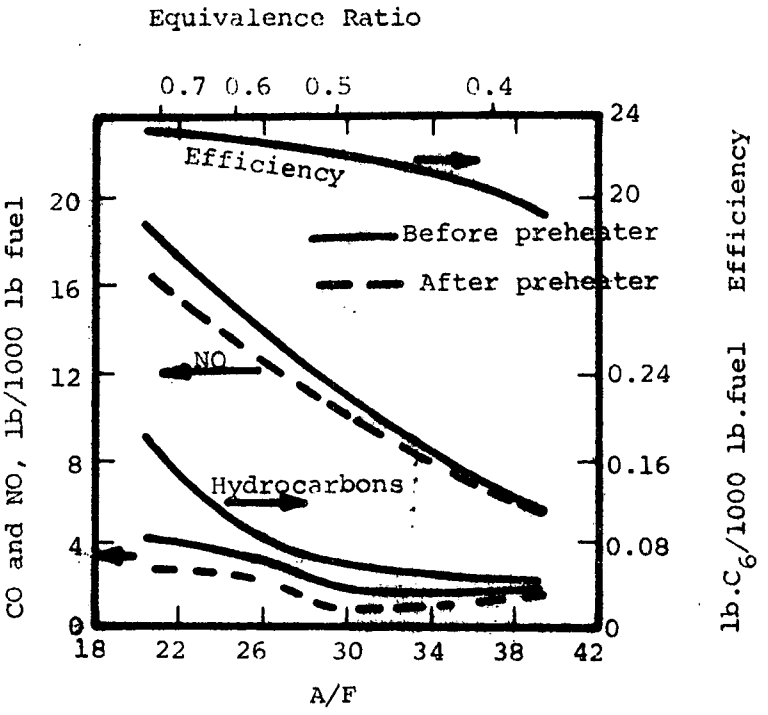


FIGURE 6.22 Effect of Mixture Ratio on Half Load Emissions at approximately 1200°F burner inlet temperature.(30)

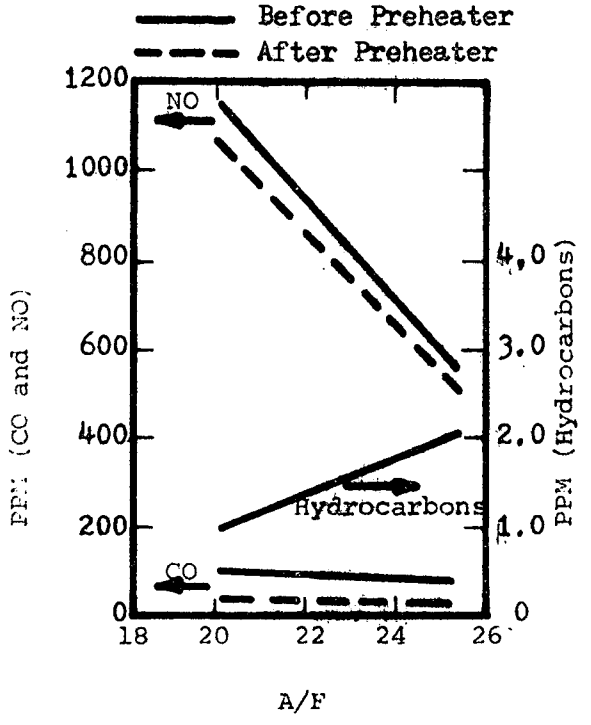


FIGURE 6.23 Effect of Mixture Ratio on Full Load Exhaust Concentrations at approximately 1355°F burner inlet temperature.(30)

over a lesser air-fuel ratio. While there is some scatter in the data taken at 20 and 25:1 air-fuel ratio, there tends to be a somewhat higher level of NO emissions at the lower and upper limits of air-fuel ratio, and correspondingly higher numbers of hydrocarbons as would be expected. These data are shown in Figure 6.23 (Reference 30).

The effect of combustor air inlet temperature on the three emissions of interest are shown in Figure 6.24 (Reference 30). As would be expected, the hydrocarbon and NO quantities decrease as inlet temperature increases and the NO concentration increases quite drastically at the higher temperatures. Values of approximately 450-500 ppm are measured at 1200°F.

Similar data were obtained on a 30 HP Philips Stirling engine. The results are shown in Figure 6.25. With no exhaust, i.e., circulation, the Philips burner achieves comparatively lower emission levels than the GM Research combustor, when both are operating at air-fuel ratios of 20:1. While the GM Research data indicated approximately 900 ppm, the Philips combustor shows about 300 ppm without circulation.

The effect of adding an inert diluent to the combustion air by recirculating exhaust gases was determined for both GM Research and Philips combustors. In both cases, part of the exhaust from the pre-heater was returned to the suction side of the combustion air blower. Because the recirculation line was not insulated, the exhaust gas inside it was cooled several hundred degrees and resulted in descreased efficiency with increasing circulation. This is shown for the louvered combustion and for the slit type combustor in Figure 6.25.

There is some difference between the NO and NO_x characteristics of the two combustors which can be related to a basic characteristic of re-circulation combustors. The louvered combustor shows a relatively high CO level, varying from 800 ppm at zero recirculation to a little below 400 ppm at 33% recirculation, increasing to about 800 ppm at maximum recirculation. The dip in CO emissions at 33% recirculation is probably in error and the curve very likely is nearly flat. The NO_x decreases from about 300 ppm at zero recirculation to about 40 ppm at close to 100% recirculation. The slotted combustor shows much better emission characteristics. The CO increases from 80 ppm at zero recirculation to about 170 ppm at 100% recirculation, representing only 20% of that encountered with the louvered combustor at the highest point. Similarly, the slotted combustor shows a decreasing NO_x emission level with increasing recirculation, with values dropping from about 110 ppm at zero recirculation to less than 40 ppm at maximum recirculation. At low levels of recirculation, the

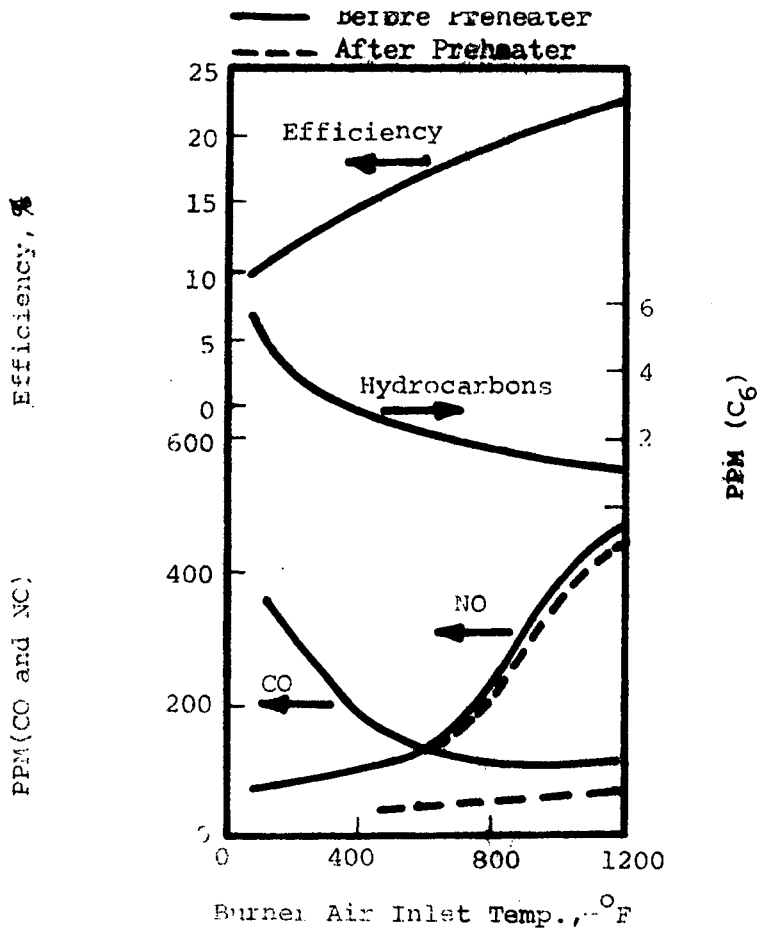


FIGURE 6.24 Effect of Combustion Air Temperature on Exhaust Concentrations at A/F = 25.(30)

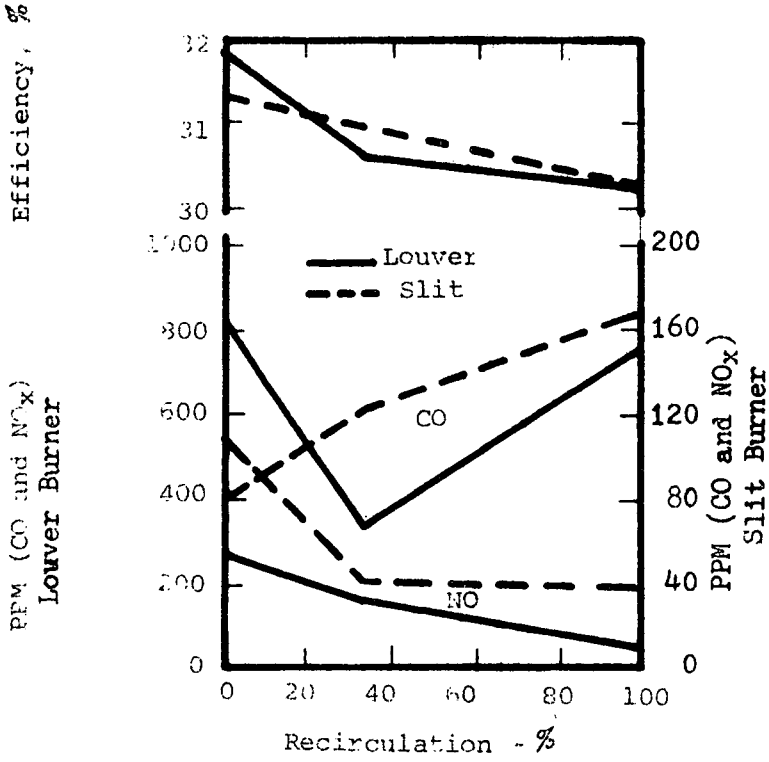


FIGURE 6.25 Effect of Exhaust Recirculation on Emissions of Slit and Louver Burner Designs.(30)

slotted combustor records about half the values encountered with the louvered design. At maximum recirculation, both combustors have about the same level of NO_x. In overall terms, the recirculation has a generally favorable effect on NO_x. Very likely further improvements in emission performance can be achieved by further research in combustor configuration. Such improvements will be required to achieve the levels projected for the future gasoline engine.

The cooling effect of exhaust gas dilution results in about a 1% loss in engine efficiency at 100% dilution. This effect can be traced to the lower combustor temperatures and reduced transfer of heat to the heater tubes.

The highly efficient combustion in the Stirling burner has been characterized by high temperatures which accelerate formation of nitrogen oxides. Because of this, the available data (circa 1968) on Stirling emissions indicated relatively high oxides of nitrogen levels. Three methods of NO_x control are outlined below. They all achieve lower NO_x emissions by lowering the peak combustion temperatures at varying penalties to overall thermal efficiency. These methods are:

1. Reducing the combustion air inlet temperature.

   A direct method of lowering the flame temperature is to reduce the air preheat. This is accompanied by a decrease in the theoretical equilibrium nitric oxide concentration for adiabatic combustion produces (29). Although not in equilibrium, the concentration of NO in the exhaust from a Stirling burner exhibits a similar percentage decrease as the flame temperature is reduced. A substantial loss in efficiency occurs with the preheater removed from the circuit as well as an increase in hydrocarbon and carbon monoxide emissions.

2. Leaning the mixture ratio at full preheat.

   The decrease in NO concentration with progressively leaner mixtures results primarily from the NO formation rate temperature dependence. Lean air-fuel ratios produce lower peak combustion temperatures and the arguments for lower NO concentrations from method 1 follow directly. A secondary effect arises from the increased flow rates through the burner at lean mixture ratios. Because the NO formation process is not in equilibrium, these high

mass flows shorten the residence time in which NO can be formed. These two effects are somewhat offset by the greater oxygen concentration in the combustion products with leaner mixtures, thereby tending to increase the NO concentration.

Although the exhaust temperature is essentially unchanged, thermal efficiency is adversely affected (Figure 6.24) because the higher flow rates result in reduced preheater performance and higher total exhaust enthalpy.

3. Recirculating part of the cooled exhaust gases.

This method has a three-fold reducing effect on NO formation. First, the dilution effect lowers the flame temperature and, consequently, the NO formation rate. Secondly, the increased flow rate through the burner decreases the residence time in a manner similar to operating with a lean overall air-fuel ratio. Finally, the exhaust diluent, while not inert, contains a lower percentage of oxygen than air. The combustion products, then, contain a smaller concentration of O_2, further reducing the formation rate of NO. Because of the added heat losses in the recirculation lines, engine efficiency decreases with increasing recirculation.

6.5.2 Noise Characteristics

In general, the automobile engine is sufficiently quiet for normal operating requirements. At idle, the sound level is relatively low. Even at high speed operating conditions, the engine noise tends to be overshadowed compared to tire noise, fan and exhaust noise. The Stirling engine is relatively quiet in comparison.

First, it is a closed cycle engine, hence, a very little internal aerodynamic noise excapes. Practically all aerodynamic noise is generated by the fan for the external combustor. Second, the elimination of abrupt changes in cylinder pressure permits the design of an engine with a minimum of mechanical noise.

Several Stirling engines have been made for terrestrial and space applications and the experience with these engines is presented very well

in the referenced reports. One 4-cylinder 400 BHP engine, which was tested by the Naval Engineering Station at Annapolis, showed a significant reduction in structure and airborne noise as compared to the Diesel engine. For roughly comparable power levels and engine speeds, the marine application showed a structure borne noise level of 96 dbA (RE: 1×10^{-3} cm/sec^2) as compared with about 135 dbA for the equivalent Diesel (Figure 6.26). For submarines, for which the tests were conducted, and for other vehicles which are sensitive to a structure-borne noise, this is a significant difference and can be traced to a large degree to the inherently balanced "Rhombic" drive which was used in this engine. The swash-plate drive should show similar characteristics. Figure 6.27 shows that the Diesel had a peak sound-pressure-level of about 112 db whereas the Stirling engine had a peak of only 95 db, for a decrease of 17 db.

Sound measurements were made on a selfcontained ground power unit (GPU-3) rated at three kilowatts continuous output (Figure 6.28). The engine design in this unit differs slightly from that in the dynamometer installation previously described. The GPU-3 has an integral preheater and drives all its own accessories, including an oil pump, nozzle atomizing air pump, radiator fan and combustion air blower.

Sound data were taken at the United States Army Mobility and Equipment Research and Development Center (MERDC) in Stafford, Virginia. The GPU-3 was operated at rated speed and load (3000 rpm, 3 KW) with no acoustic baffling of any kind placed around it. General Radio equipment was used to make sound intensity measurements at varying distances from the GPU-3 package. Because noise level is a relative matter, the Stirling engine test results are compared with results from an internal combustion 3 KW Military Standard unit tested in the same manner. Noise measurements of the two ground power units were made on different days at the same location. Since ambient sound levels change from hour to hour, the ambients for both test days were averaged for this comparison.

The results are shown in Figure 6.28, a plot of sound intensity in decibels versus frequency in hertz at a radial distance of 100 feet. The bars constitute an average sound level measured over a given octave pass band. (Each band or octave on the abscissa consists of a range of

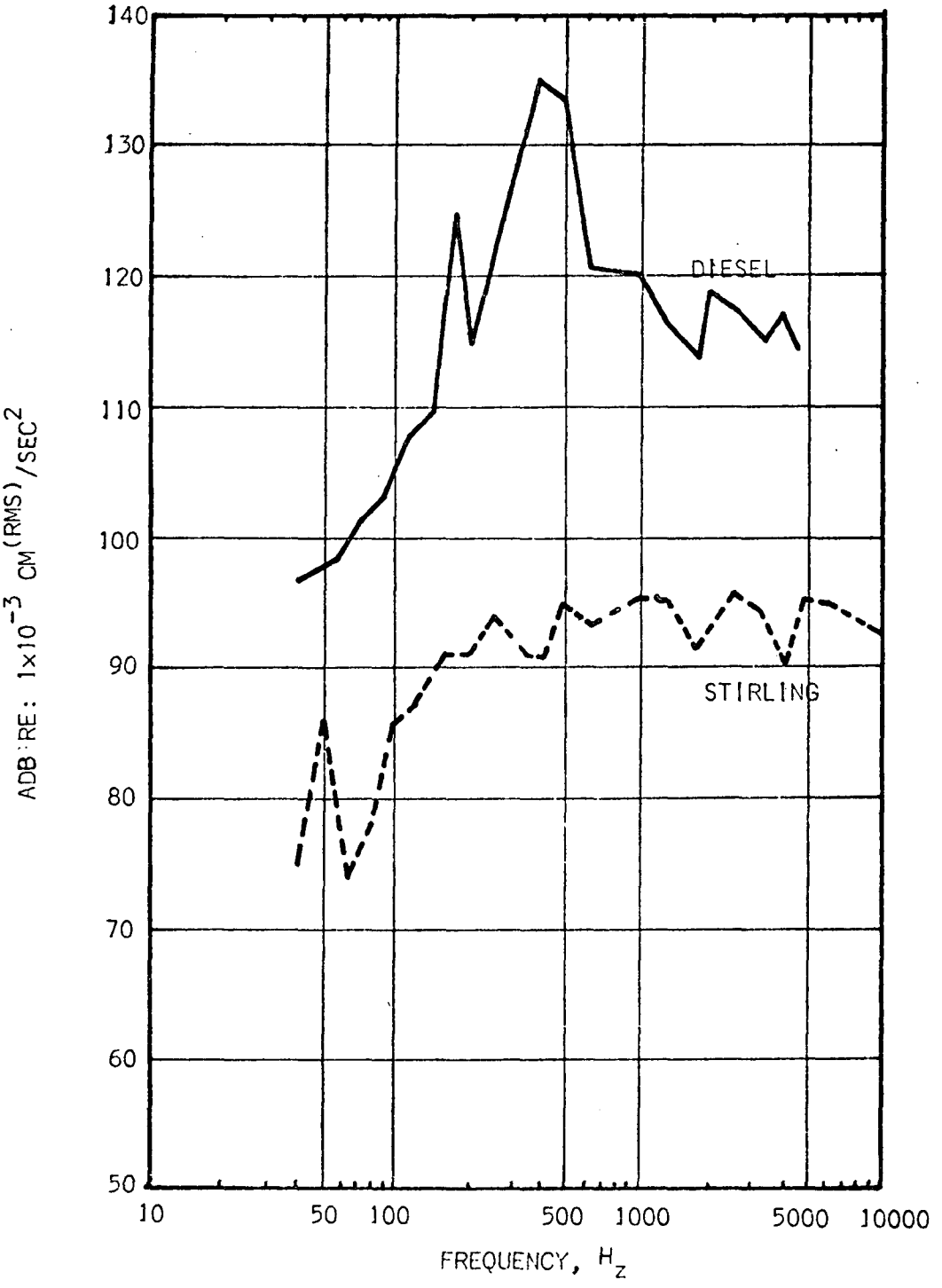


FIGURE 6.26 Results of Naval Engineering Station Tests on a 4-cylinder Stirling engine. 300 HP at 1200 rpm. (Reference 31)

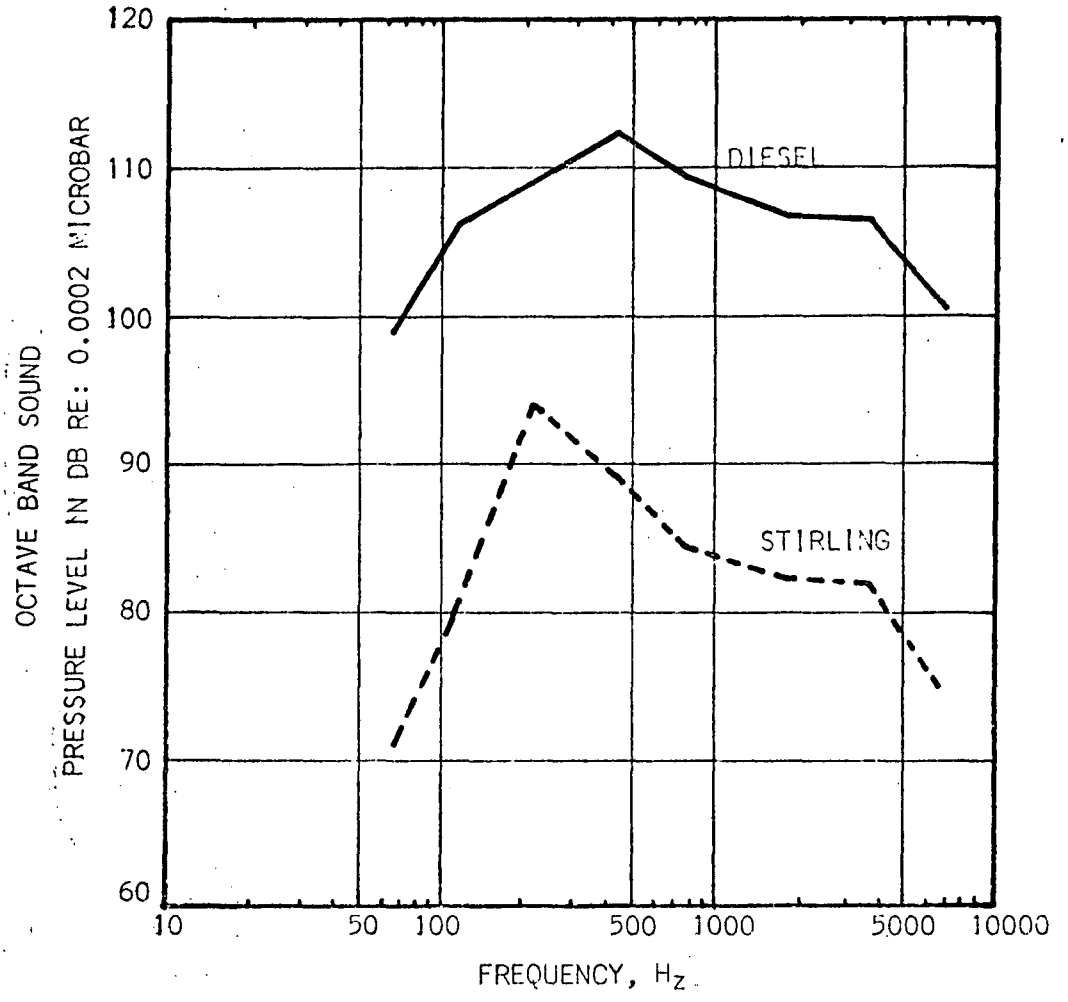


FIGURE 6.27 Airborne noise spectrum of Naval Engineering Station tests on a 4-cylinder Stirling engine. 300 HP at 1200 rpm. (Reference 31)

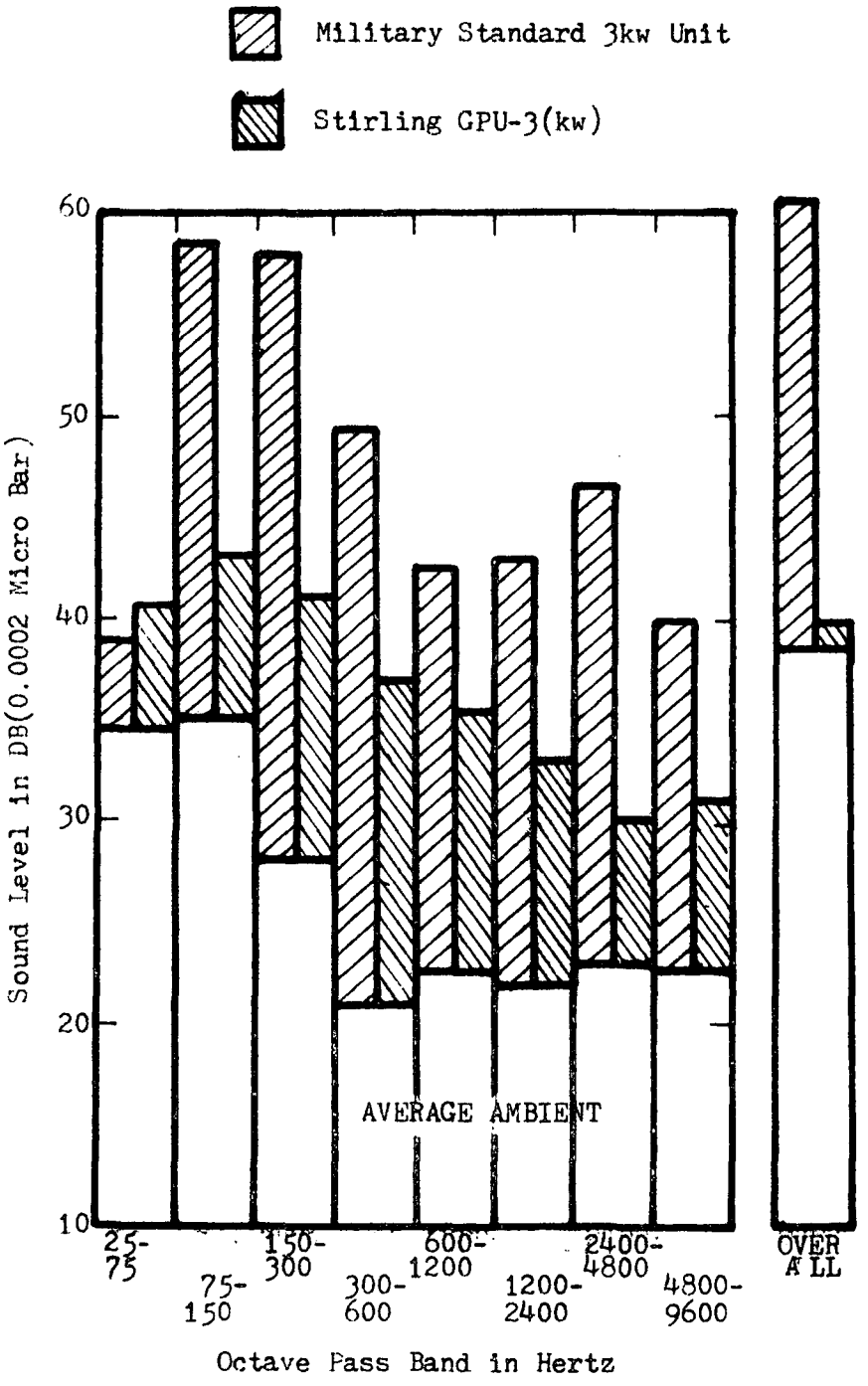


FIGURE 6.28 Noise level of Stirling and Military Standard (internal combustion) ground power units at 100 feet distance.[30]

frequencies from a given value to twice that value. All other frequencies are filtered out.) The column on the extreme right constitutes an "all pass" reading in which the entire frequency range is monitored. This figure shows the Stirling ground power unit to be 21 db quieter than the Military Standard unit when observed at 100 feet. It is apparent that the Stirling power plant enjoys a significant advantage over more conventional engines in the area of airborne sound.

6.5.3 Hazards

From a thermodynamic standpoint, the best working fluid for the Stirling engine is gaseous hydrogen. The next best fluid is gaseous helium, but the use of helium results in an increase in powerplant size and weight on the order of 30 to 40%. Whereas helium is a safe and inert working fluid, hydrogen is a highly flammable gas which requires precautionary measures.

Characteristics that make hydrogen hazardous include the following:

1. The fuel-air flammability limits of hydrogen are far broader than those of the petroleum fuels.
2. The energy required to ignite hydrogen-air mixtures is significantly less than gasoline-air mixtures. The energy of static electricity sparks is quite sufficient to ignite hydrogen-air mixtures.
3. The flame speed in hydrogen-air mixtures is far higher than in gasoline-air mixtures. It could potentially result in much greater explosive forces upon the initiation of hydrogen air fires.

An additional factor to be considered is that hydrogen, at the peak cycle temperature (1300°F) in the Stirling engine, is above the auto-ignition temperature of hydrogen-air mixtures. If a rupture should occur in the high temperature portion of the system, it is possible that the hydrogen would immediately ignite.

It can be argued that Philips and General Motors have had experience with hydrogen in Stirling engines, generally without incident. (There is one exception which occurred at the U.S. Army Research and Development Laboratories at Ft. Belvoir, Va. where a 10 HP General Motors Research engine had a hydrogen-air explosion in the crankcase due to static electricity.) Nevertheless, it is questionable whether engines operated under controlled laboratory conditions can be compared to putting 100,000,000 vehicles containing hydrogen in the hands of the general public.

One method of reducing the hazard associated with using hydrogen in the Stirling engine is to reduce the quantity of hydrogen by reducing the size of the plumbing, manifolds, pumps, reserve bottle, etc., to a minimum. Calculations indicate that for the General Motors 4L23 150 HP Stirling powerplant, the quantity of hydrogen required is as follows:

	Weight, lbs.	Free Volume @ 70°F, cu.ft.
Hydrogen in engine only	0.030	5.77
(at normal full load mean press. of 1500 psi)		
Plumbing, manifolds, pumps, etc.	0.012	2.31
Reserve bottle	0.008	1.54
Total	0.050	9.62

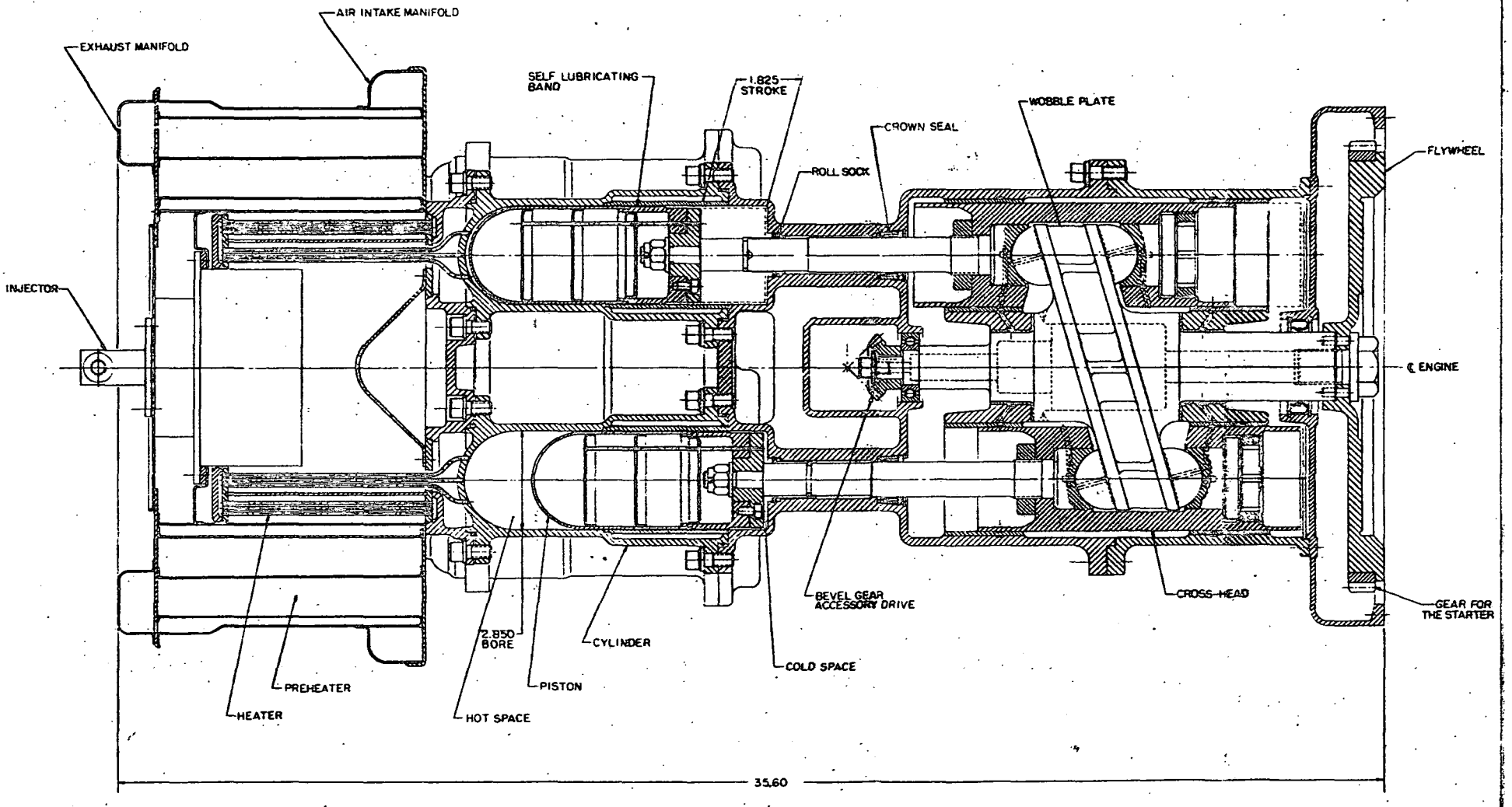
If 0.050 lb of hydrogen were completely burned, the total heat release would be about 2500 BTU. Although this amount of heat could cause a rather violent explosion if mixed with a stoichiometric quantity of air, no lasting fire could result. Also, the probability of having a stoichiometric mixture and an ignition source occur simultaneously appear somewhat remote. Therefore, in the case of the Stirling engine, it may be possible to design a relatively safe powerplant using a hazardous working fluid. If the Stirling engine were to be chosen as the Optimum powerplant for meeting future emission goals, it would probably be desirable to limit production until fire and explosion hazards can be determined based on actual operating experience of Stirling engine powered motor vehicles in the hands of the public.

Another safety precaution that would be necessary would be to vent the ceilings of any garage structures housing vehicles containing hydrogen, so that in case of a leak, the hydrogen would rise, pass through the ceiling, and dissipate into the atmosphere.

6.6 AUTOMOBILE ENGINE DESIGN

Preliminary design studies of a 160 HP Stirling engine for use in an automobile are shown in Figures 6.29, 6.30, 6.31, and 6.32. The engine has four cylinders arranged around and parallel to a central driveshaft with a swash-plate to convert the reciprocating motion of the pistons into rotary motion. This mechanical design is similar to that used frequently in hydraulic pumps and compact air compressors.

NOTE: ACCESSORY NOT SHOWN



SECTION A—A

FIGURE 6.29

THERMO MECHANICAL SYSTEMS INC.
CANOGA PARK, CALIF.

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STIRLING ENGINE STUDY — VERTICAL SECTION

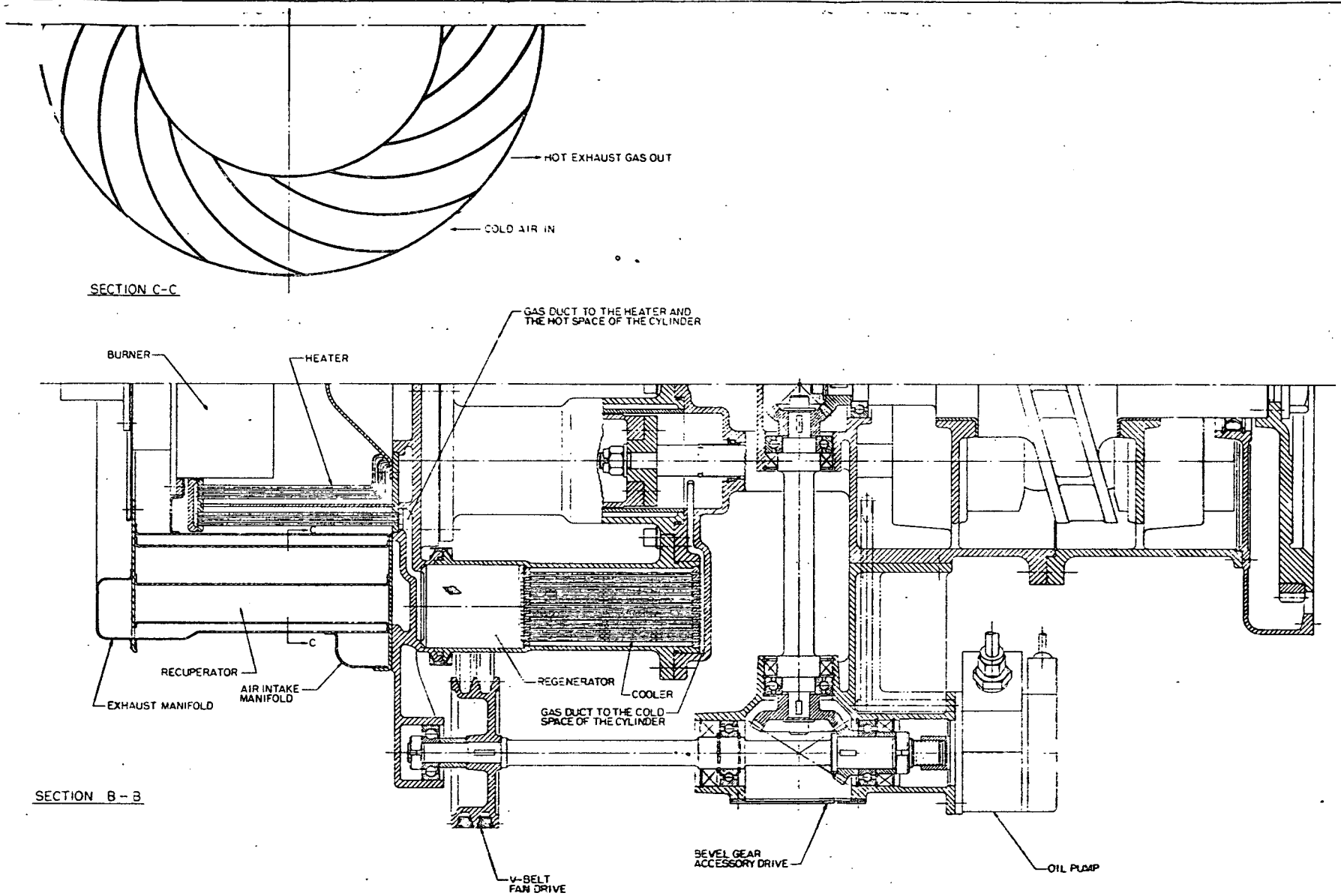

FIGURE 6.30

THERMO MECHANICAL SYSTEMS INC.
CANOGA PARK, CALIF.

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STIRLING ENGINE, STUDY-HEAT EXCHANGERS

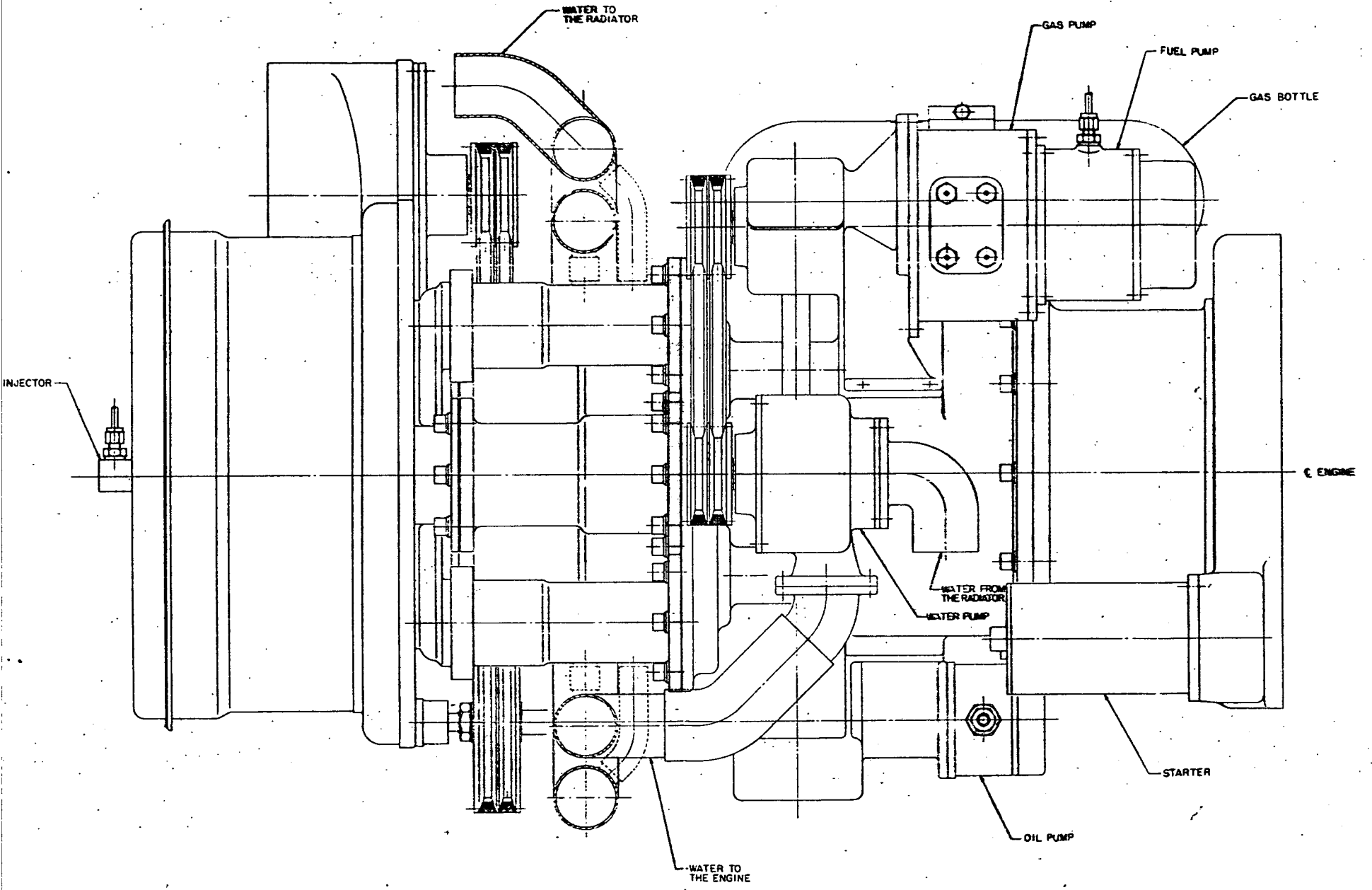

FIGURE 6.[illegible]

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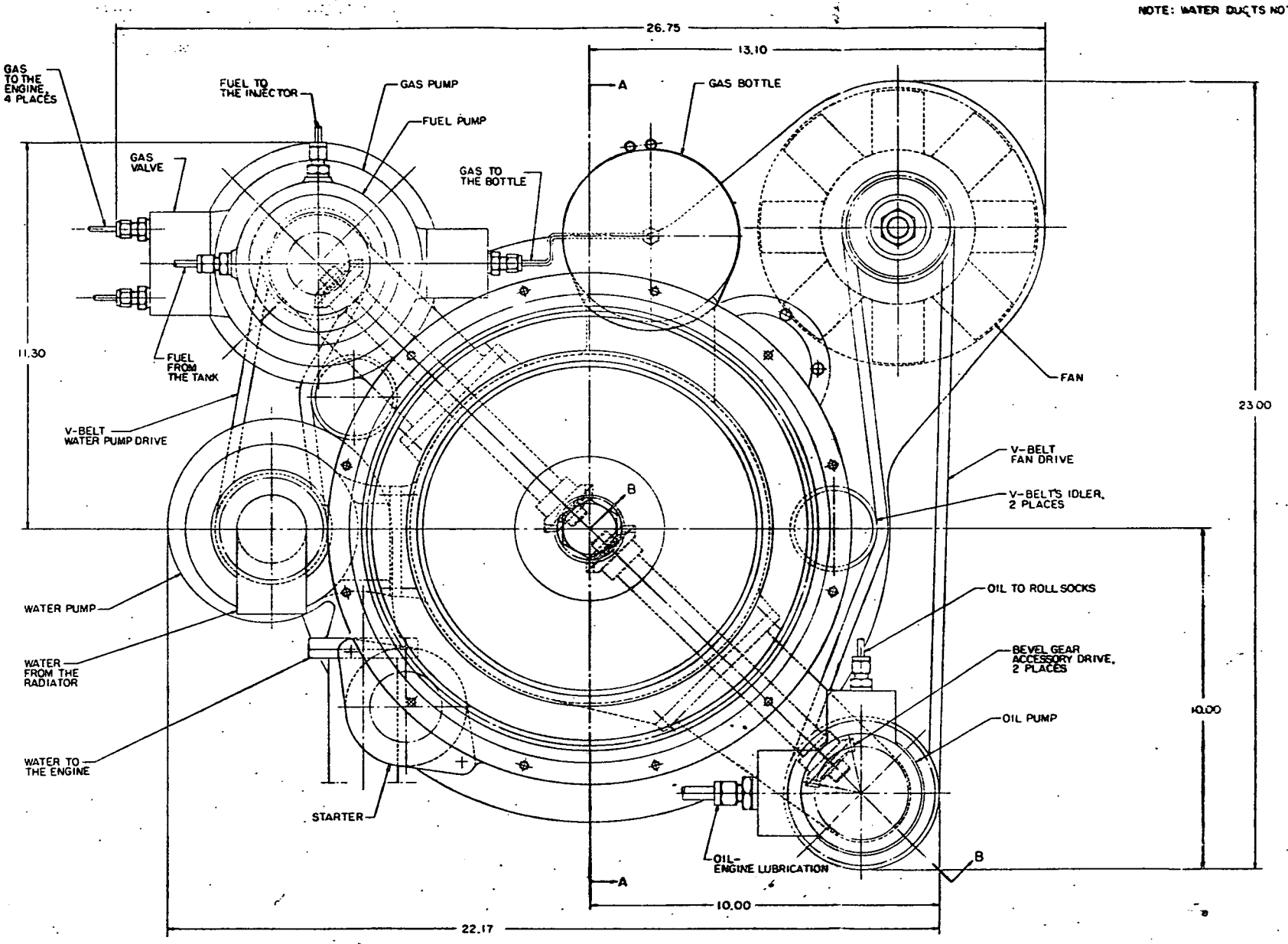
STIRLING ENGINE SIDE VIEW



REAR END VIEW

FIGURE 6.32

THERMO MECHANICAL SYSTEMS INC.
CANOGA PARK, CALIF.

STIRLING ENGINE – REAR END VIEW

4 CYL. V-SQUARE – 46 CUIN

Each cylinder has a cluster of small diameter heater tubes (0.10 in.) projecting forward from the head of each cylinder into a single combustor which heats all four sets of tubes. The regenerator and waste heat exchanger elements of the circuit, shown in Figures 6.29 and 6.30, are located in four containers which are positioned slightly outboard and between alternate cylinders. The hot gas is discharged from the heater section into the hot space above the piston as the piston moves downward. As the swash-plate continues to rotate, the piston moves upward, the hot gas is again discharged through the heater tubes and into the regenerator, which conserves cycle heat, after which it flows down into the waste heat exchanger, or cooler, and the cooled gas flows into the adjacent cylinder cold space under the piston, and returns via the same path. Waste heat is transferred to water or another suitable cooling fluid.

The circuit from one cylinder to the adjacent cylinder is shown schematically in Figure 6.10. Both tubular heater and cooler are designed for high effectiveness; the regenerator is similarly designed, but is shown to be a capsule of closely compacted screen in an assembly which has welded perforated plates at top and bottom to provide suitably low pressure drops.

The piston of the Stirling engine is hollow with three dams to minimize internal gas circulation which reduces the flow of heat from the hot space to the cold space. A small diameter bleed tube is inserted through the three baffles to balance the pressure in the cold space and the dome of the piston to inhibit collapse of the high temperature cylinder due to the combination of temperature and pressure (difference between peak and average pressure). Effectively all of the useful work results from the pressure difference acting on the top and bottom of pistons. A small clearance is maintained between the piston and cylinder since there is essentially no lubrication between the surfaces but there may be some residual misalignment. Since the region near the bottom of the piston is relatively cool, a low temperature material can be used; such as a "white metal", which is sufficiently soft to smear under contact, especially when impregnated with carbon or graphite; or a form of teflon.

One of the reasons that a swash-plate design was chosen was that it permitted a concept which facilitates linear motion of the piston. The four pistons are attached to corresponding piston rods which, in turn, are attached to cross-heads which restrain piston motion to essentially pure linear motion. Two hemispherical bearings are located in each cross-head and ride on opposite sides of the swash-plate, and are free to tilt and rotate in the cross-head socket to permit proper

contact pressure on the swash-plate. Each hemispherical bearing is lubricated on both sides and extensive tests at Philips and GMR have demonstrated the suitability of such bearings for operation at high loads for extended periods of time. (Thermo Mechanical Systems Company also demonstrated the suitability of bearings of a similar design under high bearing loads with friction coefficients of the order of .0015 to 0.002.) The ball socket operates as a "squeeze"bearing, much like the piston pin for a conventional reciprocating engine. The flat on the opposite side of the hemispherical bearing operates as a combination squeeze type bearing and a sliding Kingsbury type thrust bearing, since the swash-plate provides the relative motion corresponding to 4000 (or less) rpm.

While the lubricated contact between the hemispherical bearings and the swash-plate results in forces acting at an angle, thus tending to introduce radial forces which are alternately inward and outward, the lubricated cross-head, with very small clearances, inhibits transmission of other than linear movement to the pistons. Additional support for each piston rod is gained from the bearings acting within the structural housing.

The inhibition of lubricant flow into the thermodynamic section is accomplished by a combination of two seals per piston rod. One is a positive seal which is called a "roll-sock"; the other is a hydrodynamic crown type seal. The roll-sock and rod are designed such that as the piston rod moves linearly, the space between the two sealing components is maintained at a constant volume (Reference 32). The crown seals, which are located at the forward end of the crankcase, consist of a slotted assembly which is coated with babbitt on the inner surface and forced by a crown, or slotted, spring to maintain contact with the connecting rod in such a manner that the seal ring is slightly distorted, forming a very small wedge. Experiments have shown that the lubricant is pumped upward by the wedge, as would be expected by lubrication theory, from the crankcase into the constant volume space maintained between the two seals. While not shown in Figure 6.29, details are shown in Figure 18 of Reference 32, showing a small diameter bleed port to drain the excess lubricant, which is pumped upward from the crankcase, from the constant volume space between the two seals. This approach has been found to be successful for Philips but there is some evidence that the "roll-sock" encountered fatigue failures, thus making the seal ineffective.

An alternate design is used in which an external pump circulates pressurized oil through the space between seals, thus providing a more "elastic" liquid cushion, and for most conditions, a less uncertain upper seal. Theoretically, the original "crown" and "roll-sock" design requires filling only once, and maintains the lubricant in the constant volume space after the engine is stopped. It should always remain completely filled and, therefore, adequately resists the gas pressure at all times.

The engine has a flywheel at the back end of the crankcase and the power from the cylinders is taken out at this point through a clutch or hydraulic transmission of conventional design. The overall length of the engine, including the diesel fuel (or kerosene) heater assembly is around 36 inches and the maximum burner width and height are 22 inches, respectively. The accessory drive bevel gear is located on the forward end of the drive-shaft, and two accessory drive shafts project radially outward between the connecting rods to provide the necessary drive system.

The general arrangement of essential accessories is indicated in Figures 6.31 and 6.32. Gear driven accessories include: (1) oil pump, (2) starter, and (3) helium pump-fuel pump. Belt driven accessories include (1) water pump, and (2) blower. Neither the helium control system, the air conditioner, nor the power steering is indicated.

Considerable flexibility in incorporating the accessories can be exercised, and it appears that the entire 16 ft^3 powerplant assembly can be fitted in the engine envelope space available in a medium size vehicle (about 21 ft^3).

The preheater for the burner is shown schematically in Figure 6.30. It is of the slotted type and consists of a series of involute plates. The burner exhaust flows outward through alternate passages while the cold air flows inward through the intervening passages, picking up waste heat and increasing the air temperature into the burner. The effectiveness of this heat exchanger is about 85%, which contributes significantly to the overall thermal efficiency of the engine.

Materials and design specifications are tabulated in Table 6-II.

6.6.1 Weight Comparison

The swash-plate engine design generally has greater compactness than most other reciprocating powerplants of the same displacement, which results in lower weights. As illustrated in Table 6-III, the Stirling swash-plate design should weigh approximately 700 lbs, including all of

TABLE 6-II

GENERAL AUTOMOTIVE STIRLING ENGINE DESIGN CRITERIA

TMSCO SWASH-PLATE

General	
Rated power	160 HP
Speed	3000 rpm
Working fluid	helium
Cylinder bore diameter	2.850 inches
Stroke	1.825 inches
Piston swept volume	46.57 cu. inch
Mean cycle pressure	3000 psi
Heater Section	
Tube material	347 stainless
Heater temperature	1300°F
Regenerator-Cooler Section	
Regenerator Sections (4)	
Number of cartridges	1 (per section)
Matrix material	SS 310 screen
Cooler Sections (4)	
Tube material	AMS 5577
Coolant type	Water
Coolant flow	2 gpm
Coolant in temperature	150°F
Coolant pressure	75 psi
Housing Material	AMS 5651 (310 SS)
Main Cylinder	
Bore material	347 stainless
Drive Mechanism	
Wobble Plate and Shaft	
Main bearings journal OD	8.275/2.000 inches
Connecting rod crosshead OD	3.000 inches
W.P. pitch diameter	6.500 inches

TABLE 6-II (Continued)

W.P. bearing material	25% leaded bronze
Hemispherical ball (8) OD	1.750 inch
Ball material	AISI 8620 steel
Shaft material	AMS 6415 (alloy steel)
Counterweight material	AMS 4674
Connecting Rods	
Length (crosshead-to-piston)	8.000 inches
Material	AMS 6415 (alloy steel)
Piston	
Piston OD	2.850 inches
Piston length	6.500 inches
Seals	
Shaft seal type	Ring loaded by spring
Shaft seal material	SAE No. 12 babbitt
Piston seal type	Roll-sock
Piston seal material	Viton (Dupont Fluoro-Elastomer, -20 to +400°F)
Crankcase	
Main case	
OD	10.300 inches
Wall	0.225 inch
Material	AMS 6350
Main Bearing Supports	
Number required	2
Material	AMS 6350
Oil Pump	
Type	Full pressure, gear
Lubricant	SAE No. 10 oil
Pressure	200 psi
Flow	2.5 gpm
Filter type	Full flow
Drive gear material	AMS 6415 (alloy steel)
Pump gear material	AMS 6272
Body material	AMS 5316

TABLE 6-II (Continued)

Oil Helium Separator	
Type	Static
Filter material	Wound cotton cord
Body material	AMS 5316
Oil Cooler	
Type	Tube
Material	AMS 4015 (52S-O Aluminum)

TABLE 6-III

STIRLING ENGINE WEIGHT APPROXIMATION:

(Assuming Steel Throughout: 0.286 lb in^3)

Engine Weight (including helium bottle, compressor and controls)	700 lbs
Specific Weight	4.38 lb/hp
Envelope (including fan, starter, pumps, accelerator, gear, case and engine)	16 ft^3

COMPARABLE (160 HP) OTTO CYCLE ENGINE:

Bare Engine Weight	400 lbs
Specific Weight	2.50 lb/hp
Envelope (bare engine)	6 ft^3

those components peculiar to the Stirling cycle (helium bottle, helium compressor, and controls). One GM 3-cylinder Stirling engine has a specific weight of 6.6 lb/HP (using hydrogen), compared to 4.38 lb/HP for the helium swash-plate engine for the same power level (150-160 HP).

Also indicated in Table 6-III is the bare engine weight of a 160 HP six-cylinder spark ignition engine. Considering all components which are not included in the indicated weights of the Stirling and the S.I. engines (starter, alternator, etc.) to be common to both, the S.I. engine has a 300 lb weight advantage.

Inspection of Figure 6.33, which represents the most recent data available from Philips, reveals that, perhaps with hydrogen as the working fluid the specific weight of the engine should be about 3.5 lb/HP for the same maximum power and with some enhancement of overall efficiency. This would reduce the bare engine weight to 560 lbs.

The weights projected for this Stirling engine design are reasonably consistent with the data of Figure 6.33. The general design of the Stirling automotive engine is critically dependent upon the solution of primarily structural problems. Failure to resolve this difficulty, that of pressure and heat resistant metals, would result in prohibitively high specific weight. As indicated in Figure 6.33, the best extrapolated specific weight for a Stirling 160 HP engine using helium, with working pressures acceptable to current metallurgy, is about 8.6 lb/HP.

6.7 Conclusions

1. The Stirling engine is a relatively high efficiency powerplant capable of overall performance close to that of the Diesel engine.
2. The Stirling engine external combustor has demonstrated very low emission levels. CO and HC levels are presently below the 1976 standards and various methods, such as exhaust gas recirculation, are being tested to reduce the NO_x emissions.
3. Both hydrogen and helium are acceptable working fluids with hydrogen giving somewhat high efficiency (helium = 32% at 3000 rpm design speed, direct heating) while helium is significantly less hazardous. Helium is the preferred working fluid until the hazards associated with the hydrogen Stirling engine can be more clearly defined.
4. The cost and complexity of the present Stirling engine make

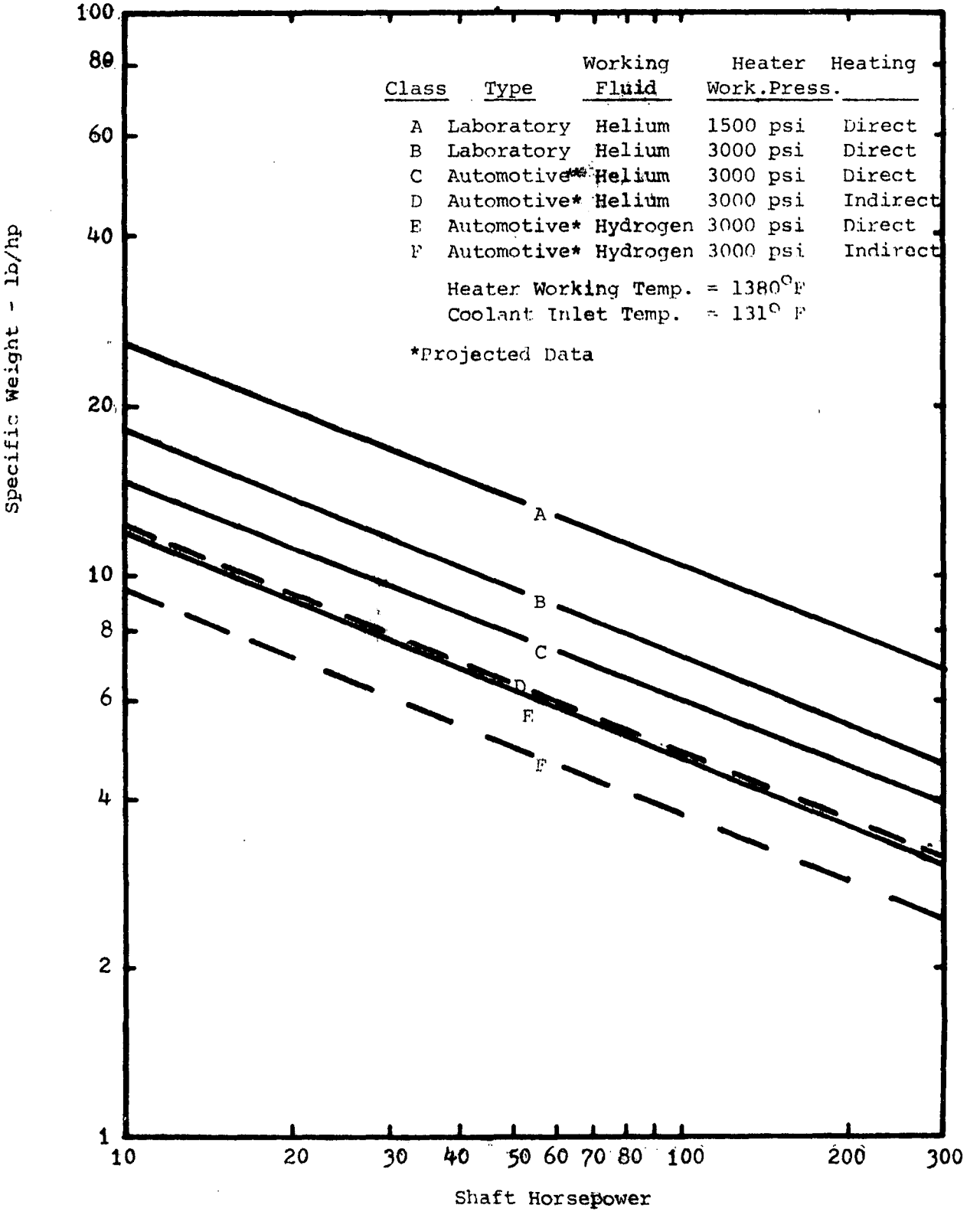


FIGURE 6.33 Stirling Engine Specific Weight.[28]

it impractical for an automobile powerplant. The use of hydrogen working fluid, higher engine speeds, and a barrel engine configuration are all means by which the Stirling engine automotive characteristics can be improved.

7.0 RANKINE AND OTHER CLOSED CYCLE WORKING FLUID HAZARDS

7.1 INTRODUCTION

In recent years emphasis has been placed upon the Rankine cycle (and other closed cycle systems such as the Stirling engine) as prime candidates for reducing the emissions of vehicle powerplants. The prime reason that the closed cycle systems merit serious consideration is that they use external combustors in which the combustion process can be controlled to produce relatively low values of emissions.

The specific objective of the work reported in this section was to evaluate the potential hazards associated with Rankine cycle organic working fluids, and other candidate liquid and gaseous working fluids for low emission closed cycle powerplants.

7.2 HAZARDS ASSOCIATED WITH CLOSED CYCLE WORKING FLUIDS

With the present interest in closed cycle powerplants (Rankine, Stirling, etc.) for low emission automotive propulsion systems, consideration must be given to the hazards associated with the working fluids used in these new systems. Working fluids will normally pass through several environmental phases during the manufacture and use in the standard passenger automobile. These environmental phases include:

1. Shipment of fluid raw materials
2. Manufacture of the fluid
3. Shipment of the fluid
4. Manufacture of powerplant system and vehicle assembly
5. Passenger vehicle under normal operation
6. Loss of working fluid during vehicle accidents
7. Maintenance and repair of powerplant and vehicle
8. Scrapping of vehicle or engine.

7.3 VEHICLE ACCIDENTS

The National Safety Council publishes a comprehensive tabulation of

of how people die in vehicle related accidents in the United States. Pertinent statistics from Reference 38 are included in Attachment I. During 1969, accidents involving motor-vehicles can be summarized as follows:

	Numbers of Accidents	Drivers (Vehicles) Involved	Deaths and Injuries
Fatal	47,000	70,000	56,000
Nonfatal Injury (disabling)	1,300,000	2,400,000	2,000,000
Property damage	14,200,000	24,300,000	-
Total (rounded)	15,500,000	26,800,000	2,060,000

As seen above, there were 26,800,000 vehicles involved in accidents in which property damage or injury resulted. With 107,000,000 registered vehicles in the U.S., this means that approximately 25 percent are involved in accidents each year. With this many vehicles involved in accidents, it appears worthwhile to examine the hazards associated with an engine system using a flammable and/or toxic working fluid.

7.4 FLUID FLAMMABILITY HAZARD

About 600-700 persons die annually in motor-vehicle fires resulting from accidents, according to the experience in Illinois tabulated by the Illinois Department of Public Health (39). National tabulations are not made of deaths, and no records are kept of injuries or the total number of fires; however, of the passenger cars involved in injury accidents, an estimated 10,000 had fires, based on a study made by Cornell Automotive Crash Injury Research. This is a relative low incidence of fires considering the total number of vehicles involved in accidents. However, there is perhaps only 1/2 pound of fuel in the engine compartment (carburetor, fuel pump, etc.) where there are a large number of potential ignition sources in case of accident. The remainder of the fuel is generally located beneath the car and aft of the rear axle where there is a minimum number of ignition sources in case of crash.

In a closed cycle alternate powerplant using a flammable working fluid, the hazards involved might be likened to using a flammable liquid

in the radiator and cooling system of the present automobile in addition to the fuel tank and combustion system. Instead of 1/2 lb. of flammable liquid in the engine compartment, there would be 20-30 pounds. Portions of the fluid would exist under relatively high pressures and temperatures, and the cooling system is frequently punctured in relatively minor crashes. Even with added crash protection, it is quite likely that the death toll due to crash-fires would increase manyfold if a flammable working fluid was used in a closed cycle automotive powerplant.

The relationship of volatility and fire hazard has been the subject of much controversy (1). There is no simple scientific measure for fire hazard, because it depends on particular circumstances. However, the following may be stated as facts which have a bearing on the subject:

1. The atmosphere over fuel in tanks may be explosive with either gasoline, jet fuel, or diesel oil, depending on temperature. For example, the explosive range (lean and rich limits) for gasoline is from $-80^{\circ}F$ to $0^{\circ}F$, for jet fuel (kerosene) $60^{\circ}F$ to $110^{\circ}F$, and for light diesel oil $140^{\circ}F$ to $180^{\circ}F$. These are approximate figures assuming the air space in the tank to be at sea-level pressure (40).

2. In partly ventillated enclosed spaces, such as boat bilges and aircraft-wing interiors, gasoline is usually more likely to furnish a combustible atmosphere than are the heavier fuels.

3. Gasoline fires develop to great intensity almost immediately after ignition. Fires with heavier fuels develop much more slowly (41).

4. The heavier fuels, having lower ignition temperatures, ignite more readily than gasoline when spilled on hot surfaces. On the other hand, the vapor from spilled gasoline are more likely to ignite from electric sparks than is the vapor from heavier fuels, because of the greater likelihood of a combustible mixture at the spark.

From the aforementioned considerations it would appear that, under most circumstances, increasing volatility tends to increase fire hazard, and therefore, the theory that the use of diesel oil in place of gasoline reduces fire hazard is generally correct. However, experience with jet airplanes, which use a fuel less volatile than gasoline, has not shown any noticeably smaller incidence of fire after a crash than has been experienced with gasoline-powered aircraft (42).

Because of the large number of automobiles involved in accidents each year (26,800,000), and because of the large number which are serious enough to involve personal injury (2,400,000), it is believed that the use of a flammable working fluid in a closed cycle engine would be inviting health hazards enormously greater than the current automotive exhaust emission problem. One possible exception might be where the working fluid is a gas and the toral weight of fluid is only a fraction of a pound as in the Stirling engine. In this case, the total amount of heat available by burning the working fluid might be sufficiently small as to make the fire and explosion hazard an acceptable risk.

7.5 FLUID TOXICITY HAZARD

Before evaluating the toxic hazards associated with candidate closed cycle working fluids, it is believed worthwhile to look at the hazards associated with current motor fuels. The two principal hazards are oral intake and inhalation of gases and vapors.

Attachment I shows that about 2500 persons died in 1969 from poisoning from solids and liquids. Details of the deaths in 1969 are not available, but 2506 such deaths in 1967 involved the following:

Sleeping pills	744
Barbiturates	506
Morphine	176
Alcohol	158
Aspirin	153
Other and unspecified drugs	281

Lead	61
Industrial solvents	36
Petroleum products	31
Arsenic	31
Other and unspecified solids and liquids	329

Inasmuch as only 31 persons died from the oral intake of petroleum products, and the number from motor fuels is probably less, it would appear that the hazards associated with the oral intake of current motor fuels is very small.

Attachment I also shows that about 1700 persons died from poisoning by gases and vapors. Details of the deaths in 1969 are not available, but 1574 such deaths occurred in 1967 as follows:

Motor vehicle exhaust gas while vehicle was standing	819
Other carbon monoxide gas, largely due to defective heating equipment	318
Utility gas	280
Other specified gases and vapors	144
Unspecified	13

The above tabulation does not indicate that any deaths occurred from the inhalation of motor fuel gases or vapors. However, 819 persons accidentally died from breathing motor vehicle exhaust gas. It could well be that a major contribution of the Federal Exhaust Emission Program will be to reduce the number of these deaths as the emission standards for carbon monoxide become more stringent.

Attachment II presents the toxicity characteristics of some common petroleum products as tabulated by the Shell Oil Company. It appears that common motor fuels are not a significant toxic hazard to the motoring public.

If a toxic working fluid were used in closed cycle motor vehicle powerplants it again appears that vehicle accidents would be the most critical environmental phase. With 26,800,000 vehicles involved in ac-

cidents, if they could be designed so that only one in ten encountered a break in the closed loop system, there would still be something on the order of 67,000,000 pounds of working fluid dumped on the ground and into the atmosphere annually. With a toxic working fluid, many persons trapped in vehicle wreckage might perish that would otherwise survive. In addition, there would probably be much less willingness on the part of bystandards to help the injured if there was fear of toxic poisoning. It therefore appears that a working fluid with any greater toxicity than that of gasoline should not be considered for automotive use.

Earlier in this discussion on hazards a list of environmental phases were set forth through which closed cycle working fluids would pass. It appears that hazards associated with vehicle accidents are the most critical phase and this phase dictates that the fluid be nonflammable and nontoxic. It is therefore concluded that a nonflammable, nontoxic working fluid would not present any significant hazards in any of the other environmental phases previously listed.

7.6 HAZARDS OF CANDIDATE CLOSED CYCLE WORKING FLUIDS

Candidate closed cycle Rankine working fluids include CP-27 (monochlorobenzene), CP-34 (thiophene), Fluorinol 85 (trifluoroethanol), P-1D, and AEF-78. Data on these fluids are included in Attachments III, IV, V, and VI. It is recommended that the definition of fluid flammability and toxicity be included to mean whether or not a fluid will burn or produce toxic liquids or vapors if a system rupture occurs in either the hot or cold portion of the closed loop. Data from Aerojet (Attachment VI) on AEF-78 indicate the toxicity hazard associated with the escape of the high temperature fluid is not known. Therefore, it is not yet known if AEF-78 would be an acceptable working fluid. Likewise, the Halocarbon Products Corporation (Attachment IV) indicates that Fluorinol 85 has an observable flash point but a very low heat of combustion (less than 3500/ BTU/lb). This indicates that the fluid could add some heat to an existing fire, but could not by itself sustain combustion. It would therefore appear that this fluid would be safe for use in a closed cycle system.

Data on the flammability and toxicity characteristics of the various candidate working fluids and other possible fluids and gases (for both the Rankine and other closed cycles) are tabulated in Table 7-I.

From the safety standpoint, it is concluded that water, Fluorinol 85, P-1D, Freon, and the nonflammable gases all qualify as candidate working fluids (Freon, upon contact with hot surfaces, can break down into phosgene, but there is no evidence that the rupture of an automotive air conditioning system adds a hazard in automobile accidents). The acceptability of AEF-78 is unknown at this time. CP-27, CP-34, and hydrogen (for the Stirling cycle) should be eliminated as candidate working fluids.

7.7 CONCLUSIONS

From the results of the study reported herein, the following conclusions are drawn:

1. Each year approximately 26,000,000 vehicles are involved in accidents resulting in property damage, injury, or fatalities.

2. In a closed cycle powerplant, the working fluid must necessarily circulate through various components of the engine system where there is a relatively large potential for closed system rupture and fire in case of a crash. It is therefore concluded that any candidate working fluid should not support combustion nor be of an explosive nature.

3. Toxicity to a large extent is relative. Statistics of the National Safety Council indicate that there is no significant health hazard associated with the use of common motor fuels. It is therefore concluded that any candidate closed cycle working fluid possess no greater toxicity than current motor fuels.

4. The use of a flammable or toxic working fluid would be inviting a health hazard enormously greater than the current automotive exhaust emission problem.

5. The National Safety Council reports that about 800 persons accidentally die annually from breathing motor vehicle exhaust fumes from non-moving vehicles. It could well be that a major contribution of the Federal Exhaust Emission Program will be to significantly reduce the number of these deaths as the emission standards for carbon monoxide become more stringent.

TABLE 7-I

HAZARDS OF CANDIDATE CLOSED CYCLE WORKING FLUIDS

FLUID	FLAMMABLE	TOXIC	SUITABLE
Water	no	no	yes
CP-27 (Attachment III)	yes	?	no
CP-34 (Attachment III)	yes	yes-vapor	no
Fluorinol 85 (Attachment IV)	apparently no	possible eye hazard	probably yes
P-1D (Attachment V)	no	apparently no	probably yes
AEF-78 (Attachment VI)	no	unknown at high temp.	unknown
Freon	no	no	yes
Hydrogen	yes	no	no
Air, N_2, A, H_e, etc.	no	no	yes

7.8 ATTACHMENTS

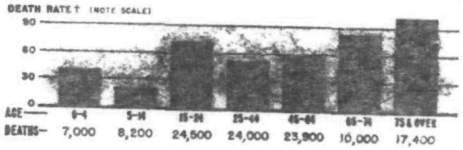
ATTACHMENT I

1969 NATIONAL SAFETY COUNCIL ACCIDENT FACTS

How people died accidentally in 1969

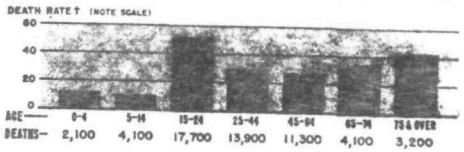
Type of accident and age of victim

All accidents — 115,000 — 0% — 57.0



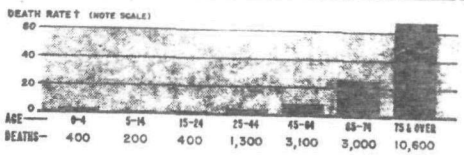
The term "accidents" covers most deaths from violence, but specifically excludes homicides, suicides and deaths in war operations.

Motor-vehicle accidents — 56,400 — +2% — 27.9



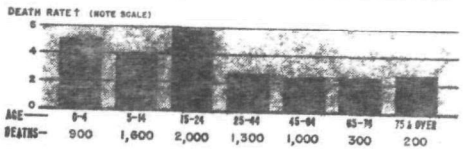
Includes deaths involving mechanically or electrically powered highway-transport vehicles in motion (except those on rails), both on and off the highway or street.

Falls — 19,000 — −5% — 9.4



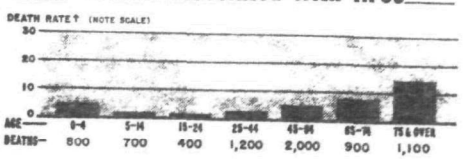
Includes deaths from falls from one level to another or on the same level, except falls in or from railway-, road-, water-, or air-transport vehicles, or those occurring in cataclysms.

Drowning — 7,300 — −1% — 3.6



Includes all drownings (work and nonwork) in boat accidents and those resulting from swimming, playing in the water, or falling in. Excludes drownings in floods and other cataclysms.

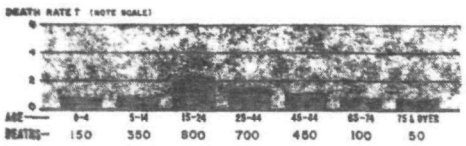
Fires, burns, and deaths associated with fires — 7,100 — −5% — 3.5



Includes deaths from fires, burns, and from injuries in conflagrations—such as asphyxiation, falls, and struck by falling objects. Excludes burns from hot objects or liquids.

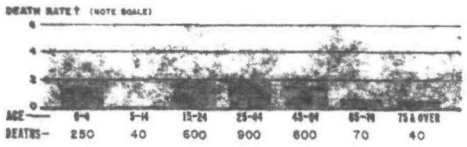
	Death Total	Change from 1968	Population Death Rate‡
Firearms	2,600	0%	1.3
Poisoning by solids and liquids	2,500	+4%	1.2
Machinery accidents	2,000	−5%	1.0
Poisoning by gases and vapors	1,700	+6%	0.8
All other types	16,400	+1%	8.1

Firearms



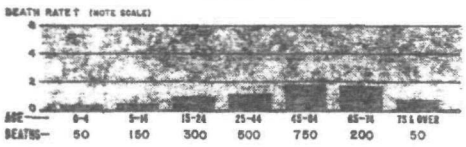
Includes deaths in firearms accidents in recreational activities or on home premises and a small number (less than 3 per cent) from explosions of dynamite, bombs, grenades, etc. Excludes deaths in war operations.

Poisoning by solids and liquids



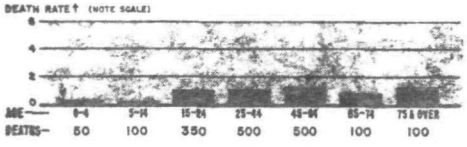
Includes deaths from medicines, as well as from commonly recognized poisons. Mushroom and shellfish poisoning deaths are included, but fatal poisoning from spoiled foods—botulism, etc.—are classified as disease deaths.

Machinery accidents



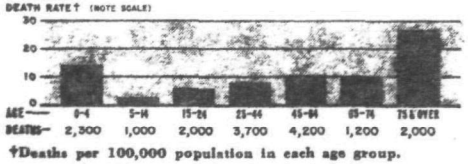
Includes deaths involving all types of machinery. Nearly half occur on farms in the course of work; of these, approximately three-fourths involve tractors. About one-third occur in industry. Five per cent occur in the home.

Poisoning by gases and vapors



Principally carbon monoxide due to incomplete combustion, involving cooking stoves, heating equipment and standing motor vehicles. Excludes deaths in conflagrations, or associated with transport vehicles in motion.

All other types



Most important types included are: inhalation or ingestion of food or other object, mechanical suffocation, air transportation, blow by falling object, electric current, railroad, excessive heat or cold, cataclysm.

†Deaths per 100,000 population in each age group.

‡Deaths per 100,000 population.

MOTOR-VEHICLE ACCIDENTS, 1969

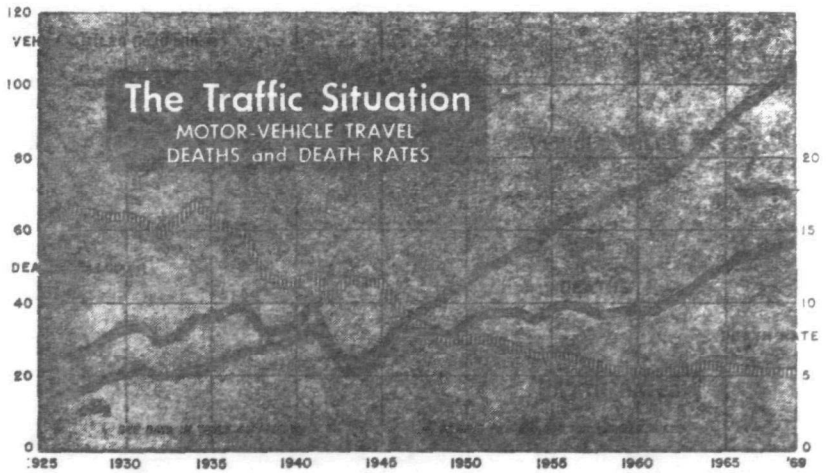
(See also page 2 for National Health Survey totals)

Between 1912 and 1969, motor-vehicle deaths per 10,000 registered vehicles were reduced 85 per cent, from 33 to about 5. (Mileage data were not available in 1912.) In 1912, there were 3,100 fatalities when the number of registered vehicles totalled only 950,000. In 1969, there were 56,400 fatalities, but registrations soared to 107 million.

Deaths	(See page 12 for effect of ICD Eighth Revision)	**56,400**
Injuries	(Disabling beyond the day of accident)	**2,000,000**
Costs	(For certain details of cost see page 5)	**$12,200,000,000**
Motor-vehicle mileage		**1,065,000,000,000**
Death rate (per 100,000,000 miles of travel)		5.30
Registered vehicles in the U.S.		107,000,000
Licensed drivers in the U.S.		107,500,000

Accident totals

		Number of Accidents	Drivers (Vehicles) Involved
Fatal		47,600	70,700
Nonfatal injury	(disabling)	1,300,000	2,400,000
Property damage	(including nondisabling injury accidents)	14,200,000	24,300,000
Total (rounded)		15,500,000	26,800,000



MOTOR-VEHICLE ACCIDENTS, 1950-1969

Motor-vehicle death and injury totals increased by more than one-fourth in the decade of the sixties as compared to the fifties. The annual death toll increased almost without interruption to a high of 56,400 from 34,763 in 1950.

	1950-1959	1960-1969
Deaths	**375,000**	**475,000**
Injuries	**13,350,000**	**17,200,000**
Costs	**$45.5 billion**	**$89.6 billion**
Motor-vehicle mileage	**5,800 billion**	**8,700 billion**

Accident costs nearly doubled in the sixties, due to more accidents and higher costs per accident.

Motor-vehicle mileage increased fifty per cent in the sixties and the mileage death rate declined by sixteen per cent. The death rate per 100 million vehicle miles was 7.59 in 1950 and declined steadily to 5.16 in 1961; it then increased to 5.70 in 1966, after which it declined to 5.30 in 1969.

Registered vehicles more than doubled over the two decades to 107 million in 1969 from about 49 million in 1950.

Licensed drivers increased nearly three-fourths over the two decades to nearly 108 million in 1969 from more than 62 million in 1950.

	Number of Accidents		Drivers (Vehicles) Involved	
Accident Totals	1950-1959	1960-1969	1950-1969	1960-1969
Fatal	**325,000**	**400,000**	**450,000**	**580,000**
Nonfatal injury	**8,900,000**	**11,000,000**	**14,600,000**	**19,800,000**
Property damage	**88,000,000**	**115,000,000**	**150,000,000**	**200,000,000**
Total (rounded)	**97,250,000**	**126,400,000**	**165,000,000**	**220,000,000**

The total number of accidents increased nearly one-third in the decade of the sixties as compared to the fifties. The more serious accidents, involving fatal and nonfatal disabling injuries, increased about one-fourth. The less serious accidents, involving property damage (including nondisabling injuries), increased nearly one-third. Similar increases occurred in the number of drivers (vehicles) involved.

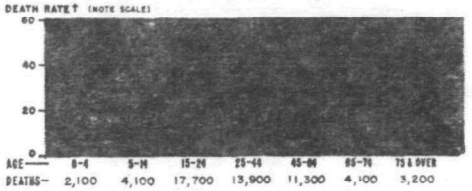
Changes in death rates, 1968-1969

Motor-vehicle deaths increased 2 per cent in 1969 over 1968 while vehicle mileage increased 5 per cent, the number of vehicles increased 4 per cent, and population increased 1 per cent. The rates for 1969 and 1968 were as follows: **Mileage death rate—5.30** down from 5.43; **Registration death rate—5.27** down from 5.35; **Population death rate—27.9** up over 27.6. **See page 59 for all rates.**

How people died in motor-vehicle accidents, 1969

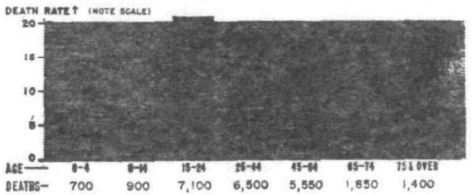
Type of accident and age of victim

	Death Total	Change from 1968	Population Death Rate†
All motor-vehicle accidents	**56,400**	**+2%**	**27.9**
Urban	18,000	+ 3%	
Rural	38,400	+ 2%	



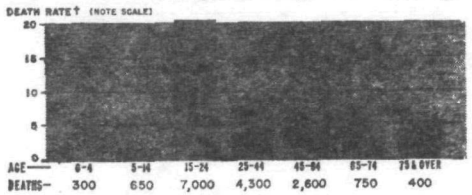
Includes deaths involving mechanically or electrically powered highway-transport vehicles in motion (except those on rails), both on and off the highway or street.

Collisions between motor vehicles	**24,000**	**+7%**	**11.9**
Urban	6,000	+ 9%	
Rural	18,000	+ 6%	



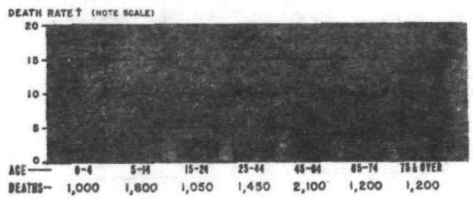
Includes deaths from collisions of two or more motor vehicles. Motorized bicycles and scooters, trolley buses, and farm tractors or road machinery travelling on highways are motor vehicles.

Noncollision in roadway, overturning, running off roadway	**16,000**	**††**	**7.9**
Urban	3,200	††	
Rural	12,800	††	



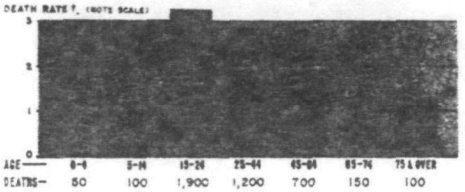
Includes deaths in all types of noncollision accidents. Classification is according to first event. If car runs off roadway and then strikes fixed object, death is charged as run off road accident.

Pedestrian accidents	**9,800**	**0%**	**4.9**
Urban	6,400	0%	
Rural	3,400	0%	



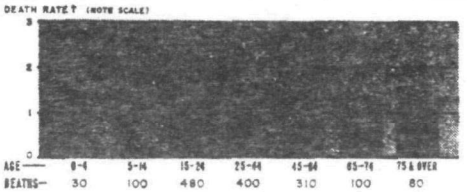
Includes all deaths of persons struck by motor vehicles, either on or off a street or highway, regardless of the circumstances of the accident.

	Death Total	Change from 1968	Population Death Rate†
Collisions with fixed objects	**4,200**	**††**	**2.1**
Urban	1,550	††	
Rural	2,650	††	



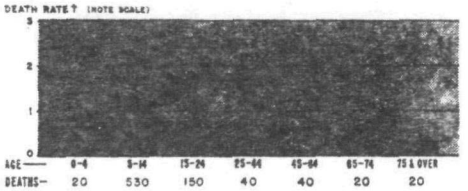
Includes deaths from collisions with fixed objects such as walls and abutments, where the collision occurred while all wheels of the vehicle were still on the road. (See comment on noncollision.)

Collisions with railroad trains	**1,500**	**—6%**	**0.7**
Urban	430	—10%	
Rural	1,070	— 4%	



Includes deaths from collisions of motor vehicles (moving or stalled) and railroad vehicles at public or private grade crossings. In other types of accidents, classification requires motor vehicle to be in motion.

Collisions with bicycles	**820**	**+3%**	**0.4**
Urban	420	0%	
Rural	400	+ 5%	



Includes deaths of bicyclists and motor-vehicle occupants from collisions between bicycles and motor vehicles on streets, highways, private driveways, parking lots, etc.

Other collisions (animals, animal-drawn vehicles, street cars)	**80**	**—20%**	**

Includes deaths from motor-vehicle collisions not specified in other categories above. Most of the deaths arose out of accidents involving animals or animal-drawn vehicles. Deaths from accidents involving street cars are not yet known for 1969.

†Deaths per 100,000 population in each age group.
‡Deaths per 100,000 population.
††Totals are not comparable to previous year. See page 48.
**Death rate was less than 0.1.

ATTACHMENT II

HEALTH HAZARDS OF PETROLEUM PRODUCTS

INDUSTRIAL HYGIENE BULLETIN

HEALTH HAZARDS OF PETROLEUM PRODUCTS

Fuels, Lubricants, Solvents

Waxes, Asphalts

Second Edition
January 1969

SHELL OIL COMPANY
PRODUCTS APPLICATION DEPARTMENT
50 West 50th Street
New York, New York, 10020

INDUSTRIAL HYGIENE BULLETIN

INTRODUCTION

Petroleum products are used for a variety of purposes. In order to insure optimum safety it is essential that the user be aware of the potential hazards involved and that he use the proper precautions.

The fire and explosion hazard, particularly in the case of such products as gasoline, liquefied petroleum gas, and hydrocarbon solvents, is generally quite well understood. However, hazards associated with toxicological properties often appear to be somewhat confusing. One reads, for example, of children who have died following the accidental swallowing of a small amount of kerosine, although kerosine is rated as "slightly toxic" or "practically non-toxic" in acute toxicity evaluations. Certain household labels required by the Federal Hazardous Substances Act may appear to the uninitiated to include statements which are inconsistent with each other. Labels on household products containing ten per cent or more of "petroleum distillates" must, for example, include both of the following statements.

"Harmful or fatal if swallowed" and "If swallowed, do not induce vomiting".

The purpose of this bulletin is to provide the reader with a clearer concept of the toxicological properties of petroleum products and the hazards associated with their use.

ACUTE TOXICITY

Acute toxicity, i.e., the effect of a single dose, is of primary importance as applied to petroleum products, since very few of these products give rise to chronic toxicity on repeated administration. A notable exception is benzene, which is unique among petroleum products in that it is capable of destroying blood-forming tissues.

The acute toxicity of a material is usually expressed as the LD-50, which is the minimum dosage required to kill fifty per cent of a group of test animals. The LD-50, which is usually expressed in milligrams or grams per kilogram of body weight, is dependent upon the method of application (usually either orally or through the skin) and upon the test animal used. Acute inhalation toxicity is normally expressed as a lethal concentration (LC-50), which is the minimum concentration which will kill fifty per cent of the test animals in a stated period of exposure, for example, 4 hours.

A classification known as the Hodge-Sterner scale is in common use in the industrial field. It is a six-step scale ranging from "extremely toxic" to "relatively harmless". A somewhat different scale, known as the Gleason, Gosselin, Hodge, is preferred by the Poison Control Centers (see later section). Both scales are described in Table 1. It will be observed that the former includes percutaneous and inhalation as well as oral toxicity, while the latter covers only oral. Unfortunately, a given descriptive rating, e.g., "practically non-toxic" does not correspond to the same LD-50 on both scales. In describing acute toxicity, except in very broad terms, it is therefore necessary to designate which scale is being used.

Petroleum products as a general rule fall either in the "practically non-toxic", the "relatively harmless" or the "slightly toxic" category (Hodge-Sterner). Most of the highly paraffinic or naphthenic products are "relatively harm-

less" or "practically non-toxic", while those containing high concentrations of aromatics are more likely to be rated "practically non-toxic" or "slightly toxic". Thus, regardless of whether administration is by month (oral toxicity) or through the skin (percutaneous toxicity), the effects of petroleum products are not particularly alarming. However, "light petroleum distillates" (gasoline, kerosine, naphtha, mineral spirits, etc.), while relatively inert when applied to the skin or introduced into the digestive tract, are much more dangerous when aspirated in liquid form into the lungs. The so-called aspiration hazard is discussed below.

Acute inhalation toxicity is likewise seldom a problem. Except in the case of the more volatile products (those with initial boiling points of about 300°F or lower) it is generally impossible to determine the LC-50, since the saturated vapor at room temperature is not sufficiently concentrated to be toxic. A value which is of greater practical significance than the LC-50 is the Threshold Limit Value, or Maximum Allowable Concentration as it was formerly called. Threshold Limit Values represent concentrations which can be tolerated by workmen throughout their working hours, day after day, without adverse effect. They are discussed in greater detail in another section.

ASPIRATION HAZARD

The most serious toxicological hazard associated with petroleum products, particularly with respect to the ingestion of light petroleum distillates such as kerosine, charcoal lighter fluid, etc., by children, is the aspiration hazard. Products which have little toxicological effect in the digestive tract can, if aspirated in liquid form into the lungs, cause a chemical pneumonitis which is often fatal. The greatest danger in connection with accidental ingestion of such products is not the effect in the stomach, but the subsequent effect which may follow if, during gastric lavage or induced vomiting, the patient aspirates some of the product into the lungs. Gerarde* has determined, on the basis of tests on rats in which kerosine was aspirated directly into the lungs, that the ratio of the intratracheal LD-50 to the oral LD-50 is approximately 1 to 140. Thus, kerosine is 140 times as toxic in the lungs as in the digestive tract of this particular test animal.

Gerarde has studied the aspiration toxicity of a large number of petroleum products, and found that the mortality rate decreases with increasing viscosity. Products having a viscosity less than 50 Saybolt Universal seconds at 100 degrees Fahrenheit possess the greatest hazard in this respect. Those with viscosity values between 50 and 80 may be considered to be borderline, and those with a viscosity above 80 are free from the aspiration hazard.

Thus, products such as gasoline, kerosine, lighter fluid, mineral spirits, naphtha, Stoddard solvent, and mineral seal oil are all capable of causing serious effects if introduced in liquid form into the lungs. This is the basis for the requirement in the Federal Hazardous Substances Act that household containers containing ten per cent or more of such products bear the following the following statements on their labels:

"DANGER!"
"Harmful or fatal if swallowed"
"If swallowed, do not induce vomiting"
"Keep out of reach of children"

IRRITATION

Some degree of irritation may be experienced when liquid petroleum products are splashed into the eyes. Aromatics are appreciably

H. W. Gerarde, Architeves of Environmental Health, March 1963, Volume 6, pages 329-341.

more irritating than the naphthenes and paraffins of comparable boiling range. Any petroleum product accidentally introduced into the eyes should be removed by washing with a generous quantity of water.

Because of their ability to dissolve natural fats from the skin, most petroleum products can cause some degree of skin irritation, particularly if the exposure is prolonged. Aromatics are somewhat more irritating to the skin than the non-aromatic products, but the difference between aromatics and saturates is much less pronounced than in the case of eye irritation. Prolonged skin contact with all liquid petroleum products should, therefore, be avoided, and any product which gets on the skin should be removed by wiping, followed by washing with soap and water.

THRESHOLD LIMIT VALUES

The American Conference of Governmental Industrial Hygienists (ACGIH) is a committee of experts, currently fourteen in number, which meets annually to determine recommended Threshold Limit Values (TLV's) and to prepare a report for publication. TLV's are time-weighted average concentrations believed to represent conditions which can be tolerated by nearly all workers, throughout the working week, without adverse effect. In a few instances, indicated in the list by a "C" designation, the value represents a "ceiling" not to be exceeded.

The 1968 list, adopted at the 30th annual meeting, contains TLV's for approximately 400 materials. Table 2 (shown on pages 7–10 contains that portion of the preface of TLV's for 1968 which explains the meaning of TLV's. It also shows all of the 1968 values on petroleum products plus several others of general interest for purposes of comparison. It will be observed that the TLV's of light paraffinic, naphthenic and olefinic products (e.g., propane, hexane, cyclohexane and cyclohexene) are generally in the 300 to 1000 ppm range. Those of aromatics, on the other hand, range from 0.2 ppm for diphenyl, through 25 ppm for benzene, to 200 ppm for toluene.

FEDERAL HAZARDOUS SUBSTANCES

The Federal Hazardous Substances Act, originally called the Federal Hazardous Substances Labeling Act, was passed in 1960, largely to protect children from injury or death through misapplication of household products. It had been estimated, prior to passage of the law, that 600,000 children swallowed household aids annually, and that about 500 of these died.

Household products which are either "corrosive", "extremely flammable" (flash point below 20°F) or "highly toxic" (oral LD-50 of 50 milligrams or less per kilogram; skin (LD-50 of 200 milligrams or less per kilogram; LC-50 of 200 ppm or less) must bear the signal word "DANGER", together with the appropriate statements describing the material and the hazards involved.

Products which are either "toxic" (oral LD-50 between 50 milligrams and 5 grams per kilogram; skin LD-50 between 200 milligrams and 2 grams per kilogram; LC-50 between 200 and 20,000 ppm), "irritating" or "flammable" (flash point between 20 and 80°F) must bear the signal word "WARNING" or "CAUTION" together with a description of the material and the hazard. The Federal Act describes test procedures for determining acute toxicity and skin and eye irritation.

A product which "generates pressure through heat, decomposition or other means", or which has been designated as a strong sensitizer or a radioactive material also requires labeling. This category is of little concern as far as petroleum products are concerned except for components of

products contained in pressurized containers, e.g., aerosols which contain propane or other petroleum fractions.

In addition, the law requires labels on a number of specific products, "based on human experience". Products containing carbon tetrachloride, diethylene glycol, ethylene glycol, methyl alcohol, and turpentine are included. The section of particular interest to the petroleum industry is the one which includes, under "products requiring special labeling", the following:

> *"Products containing five per cent or more of benzene and products containing ten per cent or more of toluene, xylene, or petroleum distillates such as kerosine, mineral seal oil, naphtha, gasoline, mineral spirits, Stoddard solvent, and related petroleum distillates".*

The reason for this is the aspiration hazard discussed earlier. The labels on household products covered in the "petroleum distillates" statement must include "DANGER", "Harmful or fatal if swallowed", "If swallowed, do not induce vomiting", and "Call physician immediately", those with a flash point of 80°F or less must also bear labels describing the fire hazard.

As a general rule, petroleum products are not "extremely toxic", "toxic", "corrosive" or "irritating" as these terms are defined in the Federal Act.

POISON CONTROL CENTERS

There are in the United States and its possessions 558 Poison Control Centers, whose facilities make available to the medical profession information on the prevention and treatment of accidents involving poisonous and potentially poisonous substances. This information is available to doctors, by telephone, on a round-the-clock basis. The Centers are kept informed of up-to-date information on various products, obtained from manufacturers on a voluntary basis, by the National Clearinghouse for Poison Control Centers, which is a part of the Public Health Service of the U.S. Department of Health, Education and Welfare. These Centers are located in 48 of the 50 states (all except Montana and Vermont) and in the Canal Zone, District of Columbia, Guam, Puerto Rico, and the Virgin Islands. Table 3 shows the location and telephone number of twelve strategically located Centers.

Reports were received by the Clearinghouse from 395 of these Centers in 1967. The number of cases of accidental ingestion reported was 83,704, and almost 87% of these involved children under 5. Medicines accounted for more than half of the cases, with aspirin responsible for 21.5%. Petroleum products accounted for 4.7% in the "all age group", and 4.6% among the children under 5 years of age. Specific petroleum products mentioned included gasoline, kerosine, lighter fluid, solvents, thinners, and furniture polish.

Mortality data, which are more complete than the figures quoted above since they are derived from death certificates submitted to the U.S. Public Health Service, showed 2,283 deaths due to accidental poisoning in 1966. Aspirin and petroleum products were involved in 7.7% and 1.8%, respectively. Children under 5 accounted for only 15.1% of the deaths, but petroleum products were cited in 10.1% of the 15.1%.

The Appendix contains the information on Shell's petroleum products which has been sent to the various Centers by the Clearinghouse. It will be noted that the various products have been classified into fifteen categories. This was done for purposes of expediency, since it would have been impractical for the National Clearinghouse to issue a page for each individual product. There is some variation in toxicity within a given category, and the descriptions

shown refer to the more toxic members. The solvents, high aromatic, low flash, for example, are shown as having an acute oral rating of "slightly toxic" (Hodge-Sterner). This corresponds to a range of LD-50 of 500 mg to 5 g per kg. However, certain members of this category are known to have LD-50 values greater than 5. Toluene, for example, has an LD-50 of 7. The ratings shown in the Appendix can therefore, in general, be regarded as conservative. More precise information is available on many of the products listed in the Appendix.

The Poison Control Centers have also been supplied with data on the various Shell branded specialty and anti-freeze products sold in service stations, e.g., brake fluids, furniture polish, SHELLZONE®, etc.

EMERGENCY

In the event of an emergency call for information involving a Shell product, it is recommended that the problem be referred to a Poison Control Center, preferably through a call by the attending physician, since the information contained in the Appendix contains medical terms which could be misinterpreted by a layman. The physician should be given the exact name of the Shell product and the telephone of a Poison Control Center (Table 3, page 11).

NOTE: Additional information may be obtained from:

Shell Oil Company
Products Application Department
Toxicology Section
50 West 50 Street
New York, New York 10020

Emergency information of a medical nature is available from:

C. H. Hine, MD.
San Francisco, California

Telephone: 415 UNderhill 1-5494

TABLE 1
ACUTE TOXICITY SCALES

RATING	LD-50 RATS - ORAL (PER KG BODY WEIGHT)	LD-50 RABBITS - SKIN (PER KG BODY WEIGHT)	LC-50 RATS - 4 HR. INHALATION (PARTS PER MILLION)	PROBABLE LETHAL ORAL DOSE FOR MAN
Extremely toxic	1 mg or less	5 mg or less	Less than 10	A taste; 1 grain
Highly toxic	1 to 50 mg	5 to 43 mg	10 to 100	A teaspoonful; 4 ml.
Moderately toxic	50 to 500 mg	44 to 340 mg	100 to 1000	An ounce; 30 g.
Slightly toxic	500 mg to 5g	350 to 2810 mg	1000 to 10,000	A pint; 250 g.
Practically non-toxic	5 to 15 g.	2.82 to 22.59 g.	10,000 to 100,000	A quart
Relatively harmless	15 g. or more	22.6 g. or more	Greater than 100,000	More than a quart

**Hodge and Sterner, American Industrial Hygiene Association Quarterly, December 1949.*

RATING	ORAL LD-50 (PER KG BODY WEIGHT)	PROBABLE LETHAL DOSE FOR 150 LB. MAN
Super toxic	Less than 5 mg	A taste (less than 7 drops)
Extremely toxic	5 to 50 mg	Between 7 drops and 1 teaspoonful
Very toxic	50 to 500 mg	Between 1 teaspoonful and 1 ounce
Moderately toxic	500 mg to 5 g.	Between 1 ounce and 1 pint (or 1 pound)
Slightly toxic	5 to 15 g.	Between 1 pint and 1 quart
Practically non-toxic	More than 15 g.	More than 1 quart

***Clinical Toxicology of Commercial Products, by Gleason, Gosselin and Hodge, second edition, Williams and Wilkins, Baltimore, 1963.*

NAME:	*TYPE OF PRODUCT:*
SHELL OIL COMPANY PRODUCTS	**Gasoline and related products**

LIST OF SPECIFIC PRODUCTS:

AEROSHELL® Turbine Fuel, JP-4
Aviation Gasoline
Gasoline
Marine Gasoline
Outboard Motor Fuel
Premium Gasoline
Shell Gasoline
Super Shell Gasoline
Tractor Fuel
White Gasoline

DESCRIPTION:

Gasolines are complex mixtures of hydrocarbons, ranging from C_4 (butane) through C_{11}. Most gasolines also contain additives (anti-knock, anti-oxidant, etc.), but the concentration is too low to contribute significantly to the acute toxicity of the product.

TOXICITY:

The vapor pressure of gasoline is high enough to create a hazard if used in open containers in a poorly ventilated area.

Acute Oral: Moderately toxic (Gleason, Gosselin, Hodge); slightly toxic (Hodge-Sterner)
Aspiration: Highly hazardous.
Inhalation: Slightly to moderately toxic.
Eye Irritation: *Vapors:* absent to moderately irritating, depending on concentration. *Liquid:* moderately irritating.
Skin Irritation: *Vapors:* non-irritating. *Liquid:* moderately irritating.

SYMPTOMS AND FINDINGS:

Oral: Irritation of mucous membranes of throat, esophagus, and stomach. Stimulation followed by depression of the central nervous system. Cardiac irregularities of rhythm.

Aspiration: Severe lung irritation with coughing, gagging, dyspnea, substernal distress, and rapidly developing pulmonary edema. Later, signs of bronchopneumonia and pneumonitis. Acute onset of central nervous system excitement followed by depression.

Inhalation: Irritation of upper respiratory tract. Central nervous system stimulation followed by depression of varying degrees ranging from dizziness, headache, and incoordination to anesthesia, coma, and respiratory arrest. Cardiac arrhythmias are a dangerous complication.

TREATMENT:

Oral: Do not induce vomiting. Lavage carefully if appreciable quantity was ingested. Guard against aspiration into lungs. Avoid use of epinephrine or related sympathomimetic amines. Administer 2 to 4 oz. of olive oil and 1 to 2 oz. of activated charcoal. Render supportive treatment if central nervous system depression occurs Watch for possible bronchopneumonia or pulmonary edema.

Aspiration: Enforced bed rest. Administer oxygen under slight positive pressure with antifoaming agent. Keep air passages open. Administer broad spectrum antibiotics prophylactically, if indicated. See card on Kerosine.

Inhalation: Maintain respiration. Perform cardiac resuscitation (e.g., external cardiac massage) if cardiac arrest occurs; defibrillation if ECG indicates ventricular fibrillation. Avoid administration of sympathomimetic amines. Prevent self-injury if patient convulses or is disoriented.

Eyes: Wash with copious quantity of water.

Skin: Remove gasoline by wiping, followed by washing with soap and water.

® *Registered trademark.*

NAME:	TYPE OF PRODUCT:
SHELL OIL COMPANY PRODUCTS	Solvents, high aromatic, high flash (see list below)

MANUFACTURER: Shell Oil Company, New York, New York

DESCRIPTION:

These products have aromatic contents greater than 33% and flash points of 80°F or greater. They are similar to the corresponding low flash solvents except that their vapor pressure is somewhat lower. Diesel fuel and furnace oil are included in this category because they often contain high concentrations of aromatics.

TOXICITY:

Acute Oral: Moderately toxic (Gleason, Gosselin, Hodge); slightly toxic (Hodge-Sterner)
Aspiration: Highly hazardous.
Inhalation: Moderately toxic.
Eye Irritation: Slightly to moderately irritating.
Skin Irritation: Moderately irritating.

SYMPTOMS AND FINDINGS:

Oral: *Gasrointestinal irritation:* nausea, vomiting, and cramping. Central nervous system depression ranging from mild headache to anesthesia, coma, and death. Pulmonary irritation secondary to exhalation of solvent. Delayed effects may include anuria, dysuria, hematuria, and laboratory evidence of kidney damage; hepatic tenderness, jaundice, liver enlargement, and other evidence of liver damage.

Aspiration: Severe lung irritation with coughing, gagging, dyspnea, substernal distress, and rapidly developing pulmonary edema. Later, signs of bronchopneumonia and pneumonitis. Acute onset of central nervous system excitement followed by depression.

Inhalation: Central nervous system depression of low grade characterized by headache and slight giddiness.

TREATMENT:

Oral: Do not induce vomiting. Lavage carefully (avoid aspiration) if appreciable quantity was ingested. Administer 2 to 4 oz. of olive oil and 1 to 2 oz. of activated charcoal. Render supportive treatment if central nervous system depression occurs. Watch for delayed liver and kidney damage. Start prophylactic liver and kidney protection program.

Aspiration: Enforced bed rest. Administer oxygen under slight positive pressure with antifoaming agent. Keep air passages open. Administer broad spectrum antibiotics prophylactically, if indicated.

Inhalation: Analgesics for headache. No specific treatment required.

Eyes: Wash with copious quantity of water.

Skin: Remove solvent by wiping, followed by washing with soap and water.

LIST OF SPECIFIC PRODUCTS:

Aromatic Solvent 29, 306
CYCLO-SOL®, 31, 32, 35 through, 38, 40 through 74, S-33, 6960
Diesel Fuel
DIESELENE®
ENSIS® Fluids (see lubricating oil section for ENSIS oils)
Form Oil J-11
Furnace Oil
Heavy Aromatic Solvent
No. 2 Fuel Oil
Mineral Spirits 80
NONA-SOL 85
Stove Oil
TS-28 (also 28M, 28R, 28W)
Xylene

® *Registered Trademark.*

NAME:	*TYPE OF PRODUCT:*
SHELL OIL COMPANY PRODUCTS	**Kerosine and related products (see list below)**

MANUFACTURER: Shell Oil Company, New York, New York

DESCRIPTION:

This group includes kerosine and other fuels such as aviation turbine fuel. Products of similar viscosity and boiling range, such as certain insecticide bases and light lubricating oils, are also included.

TOXICITY:

Vapor pressures are sufficiently low so that vapor exposure is not normally a problem unless the liquid product is at an elevated temperature. Some of the products contain additives, but these have little effect upon toxicity in the concentrations used.

Acute Oral: Slightly toxic (Gleason, Gosselin, Hodge); practically non-toxic (Hodge-Sterner).
Aspiration: Highly hazardous.
Inhalation: Due to low volatility, saturated vapors at room temperature are not toxic.
Skin & Eye Irritation: Minimally irritating.

SYMPTOMS AND FINDINGS:

Oral: Gastrointestinal tract irritation. Pulmonary tract irritation secondary to exhalation of vapors.
Aspiration: Severe lung irritation with coughing, gagging, dyspnea, substernal distress, and rapidly developing pulmonary edema. Later, signs of bronchopneumia and pneumonitis. Minimal central nervous system depression.
Inhalation: Olfactory recognition, no significant nasal or respiratory tract irritation. No systemic effects at room temperature.

TREATMENT:

Oral: Do not induce vomiting. Do not lavage. Administer 2 to 4 oz. of olive oil and 1 to 2 oz. of activated charcoal.
Aspiration: Enforced bed rest. Administer oxygen under slight positive pressure with antifoaming agent. Keep air passages open. Administer broad spectrum antibiotics prophylactically, if indicated.
Inhalation: No treatment required.
Eyes: Wash with copious quantity of water.
Skin: Remove product by wiping, followed by washing with soap and water.

LIST OF SPECIFIC PRODUCTS:

AEROSHELL®, Turbine Fuel, JP5, JP5A, JP6, JP7
Base Oil 3, 40, 185
Deodorized Spray Base
DISPERSOL®
Egg Coating Oil
No. 1 Fuel Oil
FUSUS® Oil A
Heavy Solvent No. 1, WTC
Honex Oil J-11
Hydraul Oil 11
Insecticide Base
Kerosine
Nylon Yarn Oil 11
Odorless Kerosine
Paraffin Base
PELLA® Oil 911
Range Fuel
Rollex Oil 111, 211, 311, 411
Shelldraw Oil 11
1300 Solvent
Special Fuel LF1-A
TEGULA® Oil 11
TELLUS® Oil, 11
Textile Process Oil 1
Turbine Fuel 650, 650A, 640
UMF® Grade C
38S Oil

® *Registered trademark.*

ATTACHMENT III

CHARACTERISTICS OF CP-27 AND CP-34

Monsanto

ORGANIC CHEMICALS DIVISION

Monsanto Company
800 N. Lindbergh Boulevard
St. Louis, Missouri 63166
Phone: (314) 694-1000

December 15, 1970

Mr. Harvey W. Welsh
Thermo Mechanical Systems Co.
18345 Ventura Blvd.
Tarzana, California 91356

Dear Mr. Welsh:

Enclosed are data bulletins and information on toxicity and safe handling of two Monsanto candidate thermodynamic fluids:

CP-27 - monochlorobenzene thermodynamic grade
CP-34 - thiophene thermodynamic grade

This should provide the information requested in your letter of December 1, but if you find you need any further information on Monsanto fluids, please feel free to contact me.

Very truly yours,

R. Davis

Richard Davis
Manager
Commercial Development

RD:ms

Encs.

CP-27

THERMODYNAMIC GRADE

MONOCHLOROBENZENE

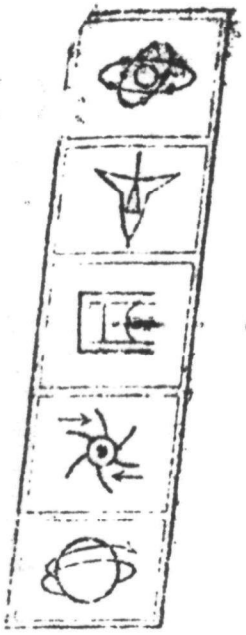
DEVELOPMENTAL SYNTHETIC FLUIDS

Formula	C_6H_5Cl
Molecular Weight	112.56
Boiling Point, °C @ 760 mm	132
@ 30 mm	44.8
Melting Point, °C	- 45.2
Density, g/ml @ 20°C	1.11
Latent Heat of Evap., cal/g @ 130°C	74
Liq. Spec. Ht., cal/gm °C @ 27°C	0.32
Viscosity, cp @ 0°C	1.06
100°C	0.37
200°C	0.19
Thermal Conduct., cal/cm sec °C x 10^6	
Vapor @ 130°C	47
Liquid @130°C	294
Flash Point, °C (closed cup)	32
Surface Tension, dyne/cm @ 20°C	33
Critical Temperature, °C	359.2
Critical Pressure, mmHg abs	33900
Critical Density, g/ml	0.365

MONSANTO COMPANY
9/28/70

MONSANTO COMPANY • 800 NORTH LINDBERGH BLVD. • ST. LOUIS, MO. 63166

TO ASSIST YOU IN THE EVALUATION OF THE ABOVE PRODUCT, WE HAVE OBTAINED THE INFORMATION CONTAINED IN THIS BULLETIN FROM LITERATURE SOURCES AND PRELIMINARY INVESTIGATION IN OUR OWN LABORATORIES. WHILE WE BELIEVE THE INFORMATION IS ACCURATE [illegible]



DEVELOPMENTAL SYNTHETIC FLUIDS

CP-27

Chemistry: Monochlorobenzene (treated)

Lubricity: No data

Thermal Stability: Monsanto thermodynamic grade monochlorobenzene has a predicted lifetime of several thousand hot hours when liquid temperatures never exceed 333°C and moisture and oxygen are excluded. This is based on Monsanto static tests conducted at temperatures to 343°C and times to 2000 hours. Commercial monochlorobenzene is far less stable.

Toxicity: The toxicology of monochlorobenzene is readily available in standard texts and compilations.

MONSANTO COMPANY
9/28/70

TO ASSIST YOU IN THE EVALUATION OF THIS ALDRICH PRODUCT, WE HAVE OBTAINED THE INFORMATION CONTAINED ON THIS LABEL FROM VARIOUS SOURCES AND PUBLISHED LITERATURE IN OUR OWN

CP 34

THERMODYNAMIC FLUID CANDIDATE (THIOPHENE)

DEVELOPMENTAL SYNTHETIC FLUIDS

CP 34 is a stable organic fluid which has been found to possess unique thermodynamic properties potentially useful in a simple Rankine cycle machine. The compressed vapor of CP 34 is capable of expansion without becoming excessively saturated or superheated. Using a hypothetic, non-regenerative cycle, CP 34 gave a predicted efficiency 40% greater than a commonly used Rankine cycle fluid, Freon 12. While the maximum permissible system temperature with CP 34 is as yet undetermined, it appears to be of the order of 200°F greater than that of the Freon fluids. Pertinent information concerning CP 34 is tabulated below.

Physical Properties

Temp. (°F)	Viscosity (cs)	Density (g/ml.)	Specific Heat (BTU/lb.)
-40	melting pt.		
0	0.90	1.106	0.302
60	0.67	1.069	0.326
100	0.53	1.044	0.342
150	0.42	1.013	0.360
200	(0.33) ext	0.981	0.384
300	(0.24) ext	0.920	0.426
400	(0.19) ext	0.859	0.460

Vapor Pressure (mm)	Temperature (°F)
760	183
100	86
10	10
1	-40

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TO ASSIST YOU IN THE EVALUATION OF THE ABOVE PRODUCT, WE HAVE OBTAINED THE INFORMATION CONTAINED IN THIS BULLETIN FROM LITERATURE SOURCES AND PRELIMINARY INVESTIGATION IN OUR OWN LABORATORIES. WHILE WE BELIEVE THE INFORMATION IS ACCURATE, WE ASSUME NO RESPONSIBILITY AND DISCLAIM ANY LIABILITY INCURRED IN USING THESE DATA OR SUGGESTIONS. ONLY QUALIFIED TECHNICAL PERSONNEL SHOULD WORK WITH THIS PRODUCT UNTIL SUBSEQUENT STUDIES JUSTIFY ITS USE IN YOUR PRODUCTS OR PROCESSES. OBVIOUSLY NOTHING IN THIS BULLETIN SHOULD BE CONSTRUED AS A RECOMMEN-

Riedel reduced vapor pressure equation -

$$\ln P = 6.031 \ln T - (0.1911)\left(\frac{36}{T} - 35 - T^6 + 42 \ln T\right)$$

- Surface Tension

33.89 dynes/cm at 69°F
32.37 dynes/cm at 86°F
30.89 dynes/cm at 104°F

- Refractive Index

$n_D^{20} = 1.5298$

$n_D^{25} = 1.5257$

- Flash Point (COC) ~~60°F~~ 20°F

- Autogenous Ignition Temperature 755°F

- Thermodynamic Properties

Critical Constants

$T_c = 584.6°F$ $V_c = 0.04341$ ft^3/lb.

$P_c = 791.73$ psia $Z_c = 0.258$

- Estimated Thermal Conductivity and Viscosity

Temp. (°F)	Thermal Conductivity (BTU/hr.ft.)		Viscosity (lb/hr.ft.)	
	Liquid	Vapor	Liquid	Vapor
200	0.0795	0.0188	0.7320	0.0594
300	-	0.0242	-	0.0678
400	0.0771	0.0299	0.2332	0.0758
500	-	0.0357	-	0.0836
600	0.1019	0.0417	0.1259	0.0910

(Calculated from Lennard-Jones parameters for 14.7 psia)

Fluid Stability

Based upon a series of bulk thermal stability tests, the life predictions for CP 34 are as follows in terms of time to reach a given level of conversion to decomposition products.

Temperature	Time (days) 0.5%	50%
600°F	17	170
550°F	62	910
500°F	240	1,200
450°F	920	180,000

These are conservative predictions and are based on the fluid remaining at that temperature constantly; a situation more severe than encountered in the actual practice.

Compatibility

Metal Corrosion Evaluation
(Test in confined vessel for 144 hours)

Metal	Corrosion Rate (calc.mils/yr) Liquid	Vapor
@ 300°F		
Aluminum 61S	0.21	<0.07
Magnesium	1.39	0.30
Copper (oxygen free)	0.17	0.18
Mild Steel	0.09	0.07
@ 500°F		
Titanium	0.09	0.04
Inconel	<0.08	0.15
@ 650°F		
Stainless 316	<0.10	-

- Elastomers
(FTS 791A Method 3603.4 - 168 hrs. at 158°F)

	Hardness		Swell (Vol. Percent)	
	Initial	Final		
1. Viton	65	51(53)	23.0	(8.97)
	65	49(53)	24.5	(10.4)
2. Buna N	60	46(56)	140.1	(1.99)
	60	45(57)	141.1	(2.12)
3. Neoprene	61	44(67)	127.0	(13.9)
	61	45(68)	127.0	(13.4)
4. Butyl	61	38(49)	109.0	(1.03)
	61	39(50)	111.6	(0.11)
5. Silicone	65	60(63)	42.3	(3.21)
	65	63(64)	42.9	(3.22)
6. EPR	66	52(72)	74.0	(8.22)
	66	52(73)	70.3	(8.31)

(Numbers in parentheses indicate data obtained after samples were air-dried for 24 hours at room temp.)

- Fluids

Completely Soluble	Limited Solubility	Insoluble
Acetone	Freon 11	Water
Benzene	Freon 12	
Chlorinated aliphatics	Freon 114	
	Freon 22	
Heptane		
Hydrocarbon oils		

- Lubricity Tests

- Falex Test

	CP 34	Water
Failure load (lbs.)	none up to 4400 lbs.	500 lbs.

- Shell Four-Ball Wear

Scar Diameter
(conditions: 10 kg load, 1200 rpm, one hr.)

	Steel-on-Steel	Steel-on-Bronze
CP 34	0.48 mm.	1.13 mm.
Water	1.20 mm.	1.53 mm.

- Mollier Diagram

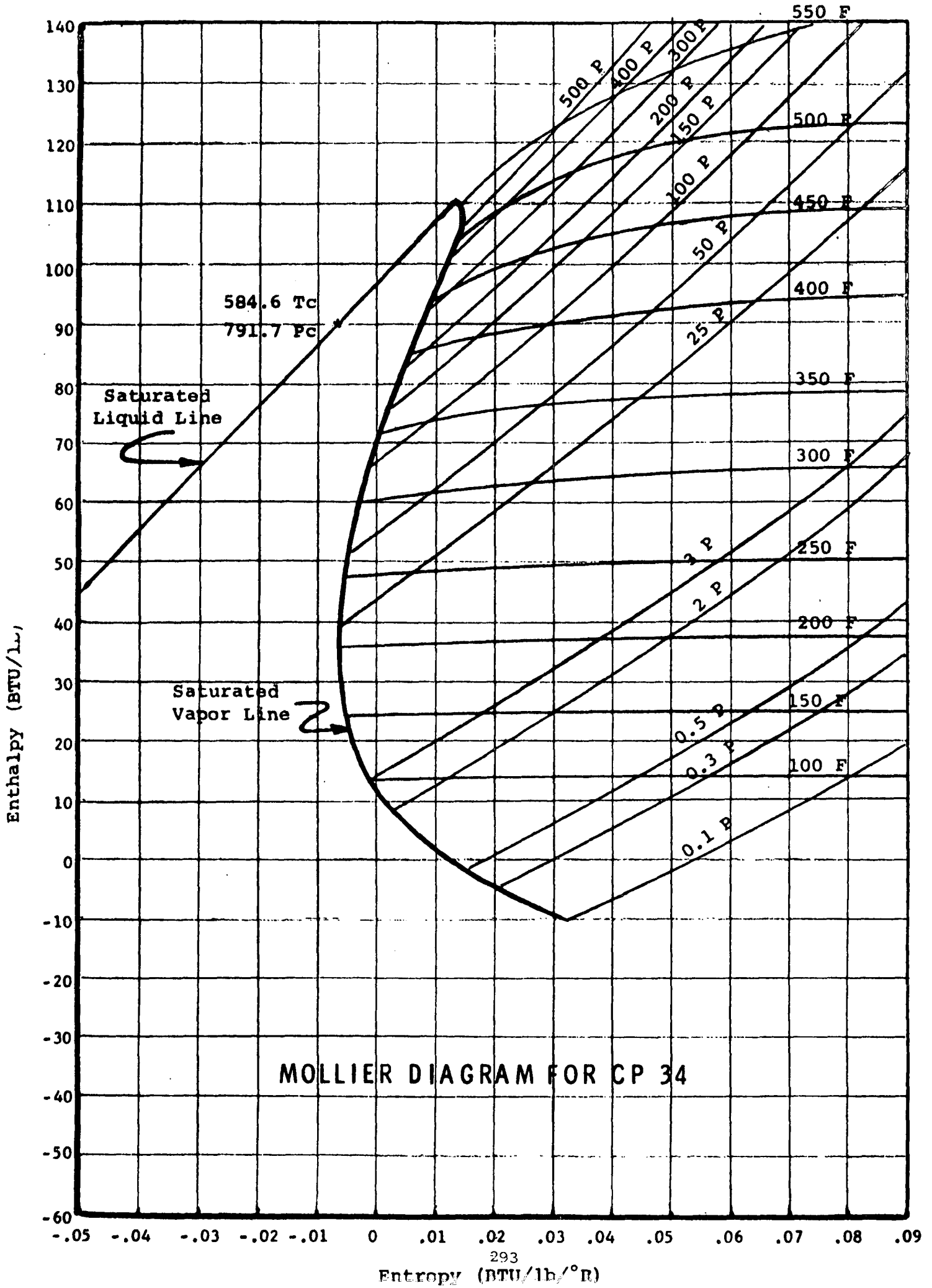
The Mollier Diagram on the next page has been drawn using entropy and enthalpy data derived from the critical constants and measured pressure-volume-temperature data. The most outstanding characteristic of the curve is the almost vertical saturated vapor portion.

- Precautions

Although CP 34 is a flammable material, its use is not considered hazardous if appropriate handling procedures are observed. The liquid has a flash point of 60°F and its vapors should not be exposed to open flame, spark or other ignition sources. In the absence of an ignition source, proper air-vapor mixtures would not be expected to undergo autoignition below 750°F.

On the basis of preliminary evaluation, CP 34 is tentatively classed as being only slightly toxic when ingested orally or absorbed through the skin. It is considered to be a moderate skin and eye irritant. Gross effect studies indicate that the vapors of CP 34 may represent a severe hazard when inhaled. Consequently, it should be used only in well ventilated areas and care should be exercised to avoid breathing the vapors.

August, 1968


550 F
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300 P
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100 P
50 P
25 P
3 P
2 P
0.5 P
0.3 P
0.1 P
584.6 Tc
791.7 Pc
Saturated Liquid Line
Saturated Vapor Line
MOLLIER DIAGRAM FOR CP 34
Enthalpy (BTU/lb)
Entropy (BTU/lb/°R)
140
130
120
110
100
90
80
70
60
50
40
30
20
10
0
-10
-20
-30
-40
-50
-60
-.05
-.04
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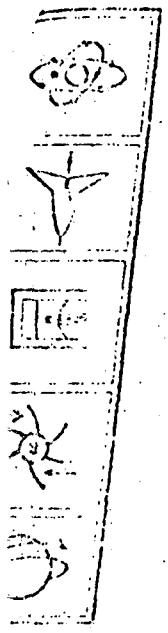


DEVELOPMENTAL SYNTHETIC FLUIDS

CP-34

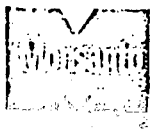
Chemistry: Thiophene (treated)

Lubricity: See CP-34 bulletin. No additional data available.

Thermal Stability: Monsanto thermodynamic grade thiophene has a predicted lifetime of several thousand hot hours when liquid temperatures never exceed 288°C and moisture and oxygen are excluded. This is based on Monsanto static tests conducted at temperatures to 399°C and times to 1500 hours. Commercial thiophene is far less stable.

Toxicity: See attachments.

Flash Point: Important. The flash point listed in the August 1968 CP-34 Bulletin (60°F by Cleveland Open Cup) is in error. It should be revised to read 20°F by Tag Open Cup.



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FOR PERSONAL FILE OF

Harvey Walker

API TOXICOLOGICAL REVIEW

THIOPHENE AND DERIVATIVES

SEPTEMBER 1948

Note: This review summarizes the best available information on the properties, characteristics, and toxicology of *thiophene and derivatives.* It offers suggestions and tentative recommendations pertaining to medical treatments, medical examinations, and precautionary measures for workers who are exposed to thiophene and derivatives. It was prepared at the Harvard School of Public Health, Boston, Mass., under the direction of Professor Philip Drinker. The review has been accepted for publication by the Medical Advisory Committee of the American Petroleum Institute. Anyone desiring to submit additional information or proposed changes for consideration prior to re-issuance of this review is requested to send them to the American Petroleum Institute.

This review was prepared by Marshall Clinton, M. D.

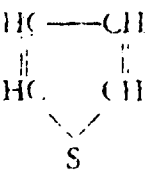
AMERICAN PETROLEUM INSTITUTE
DEPARTMENT OF SAFETY
50 WEST 50TH STREET
NEW YORK 20, N. Y.

Price 25 C.

TOXICOLOGICAL REVIEW OF THIOPHENE AND DERIVATIVES

I. Substance

Thiophene.
Formula: C_4H_4S.
Structural formula:



Molecular weight=84.13.
Synonym: thiofuran.

II. Properties and Characteristics [1, 2, 3a]

Melting point =minus 38 deg C (−36.4 deg F).
Boiling point =84 deg C (183.2 deg F).
Refractive index=1.5285.
1 mg per liter =291 ppm; 100 ppm=0.344 mg per liter.

Thiophene is a clear colorless heterocyclic compound encountered as an important contaminant of benzene. It is insoluble in water, but is readily soluble in alcohol, ether, benzene, and most hydrocarbons. Thiophene is difficult to separate from benzene by physical means because of their similar boiling points, but can be separated fairly readily after reaction of the more reactive thiophene with other substances, such as mercury. Thiophene can be obtained from crude benzene, and now can be synthesized without excessive difficulty. Thiophene is highly reactive, and is readily nitrated, sulfonated, halogenated, or mercurated. It can be made to undergo ketone formation or aminomethylation without difficulty. It is usually removed from benzene by sulfuric-acid treatment.

III. Probable Sources of Contact

Contact with thiophene may occur due to leaks occurring in the course of its handling or manufacture. Contact may result from handling crude coal-tar benzene, as this contains up to 0.5 per cent thiophene. It is not possible to state the most probable sources of exposure, as this substance is, commercially, relatively new, and its uses are ill-defined but growing.

Prepared under the auspices of the Subcommittee for Permissible Concentrations of Toxic Substances in the Petroleum Industry.

Figures refer to bibliography on p. 3.

IV. Toxicology

a. General Considerations

A considerable amount of study has been devoted to investigations of the toxicology and pharmacology of thiophene and its derivatives. Most of the latter studies have been comparisons of the thio-homologues of organic substances of known and, usually, fairly marked pharmacological activity. Thiophene in fairly high concentrations has, according to most authors,[4, 5] an acute narcotic effect greater than equal concentrations of benzene. Flury and Zernik report that the inhalation by mice of 2,900 ppm of thiophene results in loss of consciousness and, in some instances, death; whereas similar concentrations of benzene can be tolerated without difficulty. Concentrations of 8,700 ppm of thiophene caused death of mice in 20 min to 80 min; whereas benzene produced no such effect.

The acute toxic action of thiophene appears to be exerted primarily on the central nervous system. It has a selective action on the equilibrium centers of the cerebrum and cerebellum, producing severe ataxia following repeated injections.[5, 6] Thiophene produces fairly diffuse changes in the cerebellum, particularly the vermis, characterized by degeneration of nerve cells in these areas. There may be superimposed vascular changes.[7, 8] The metabolism of thiophene is poorly understood, although it is stated that 5 to 12 per cent is recovered in the urine in conjugated form. However, there is no increase in the conjugated sulfates, and the total sulfate excretion decreases after administration of thiophene.[6]

b. Acute Effects

As already noted, acute exposure to high concentrations of thiophene results in nervous-system depression. Repeated daily injection of 2 g of thiophene in dogs results in locomotor ataxia and paralysis,[5] rather similar to those noted in severe CS_2 poisoning. The effects of thiophene on humans has not been described.

c. Chronic Effects

No reports on the chronic effects of repeated or prolonged exposure to low concentrations of thiophene are available.

d. Safe Limits

Extreme concentrations of thiophene are obviously intolerable, as they produce acute poisoning. No information is available, however, on the effects of repeated exposure to lesser concentrations, such as 100 to 1,000 ppm. Therefore, no safe limits have been or can be promulgated at present.

V. Treatment

No information on the possible therapy of thiophene poisoning is available.

VI. Examinations

The present state of knowledge concerning thiophene does not permit the establishment of any special pre-employment or periodic examinations. It appears sensible to employ only men in good health to work with thiophene and to re-examine them frequently for possible evidences of blood dyscrasia or neurologic disturbances, but these measures may prove unnecessary.

VII. Precautionary Measures

Thiophene should be handled with extreme care in closed systems or with adequate ventilation, until its chronic toxicity is established or shown to be absent.

VIII. Bibliography

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4. F. Flury and F. Zernik, "Toxicity of Thiophene," *Chem. Ztg.* 56, 149 (1932).
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6. A. Christomanos, "Action of Organic Sulfur Compounds on the Dog Organism: Action and Fate of Thiophene in the Metabolism of the Dog," *Biochem. Z.* 229, 248 (1930).
7. T. Upners, "Experimental Studies Concerning the Local Action of Thiophene on the Central Nervous System," *Z. ges. Neurol. Psychiat.* 166, 623 (1939).
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ATTACHMENT IV

CHARACTERISTICS OF FLUORINOL 85

Halocarbon
PRODUCTS CORPORATION
82 BURLEWS COURT • HACKENSACK, N. J. 07601 • TELEPHONE 201-343-8703 • TELEX: 133469

April 28, 1971

Thermo Mechanical Systems Company
7252 Remmet Avenue
Canoga Park, California 91303

APR 30 1971

Attention: Mr. Harvey W. Welsh

Gentlemen:

Enclosed are data sheets on Fluorinol 85 as requested in your letter of April 22.

This material is basically a fluorinated alcohol to which about 3% by weight, of water has been added.

We offer other Fluorinol fluids containing greater percentages of water. These fluids are designated by number representing approximately the mol % of the fluorinated alcohol contained in each fluid. As an example, Fluorinol 85 contains 85 mol % of the fluorinated alcohol.

We will be glad to send samples and data on any of the Fluorinols you may want to investigate. We have data on the following at this time:

	Boiling Point	Freezing Point
Fluorinol 100	164°F	-48.9°F
Fluorinol 85	166°F	-82 °F
Fluorinol 61	170°F	-26 °F
Fluorinol 51	172°F	- 9 °F
Fluorinol 33	176°F	9 °F
Fluorinol 21	178°F	18 °F

All Fluorinols containing more water than Fluorinol 85 have a melting point (ice point) of -82°F due to selective freezing out of ice particles between the observed freezing point and the eutectic formed at -82°F. In that temperature range a slurry is formed.

All Fluorinols also have an observable flash point but a very low heat of combustion (less than 3500 BTU/LB). As a result, these materials will not sustain combustion. We have not found an auto-ignition temperature in any of these fluids, but we feel that it might be well to repeat these tests if you are equipped to do so.

Toxicity data on trifluoroethanol (Fluorinol 100) is attached hereto. As water is added, it is assumed that inhalation and dermal toxicity is lowered.

Please let us know if we can be of further assistance.

Very truly yours,

HALOCARBON PRODUCTS CORPORATION

Rex C. Conner

RCC:js
encl.

UNIVERSITY OF MARYLAND SCHOOL OF PHARMACY
636 West Lombard Street, Baltimore, Maryland 21201

DEPARTMENT OF PHARMACOLOGY AND TOXICOLOGY

EVALUATION OF TRIFLUOROETHANOL (TFE) TOXICITY AND HAZARD

Introduction

TFE is toxic after oral ingestion and is an eye irritant. Although its dermal and inhalation toxicities are significantly less, inhalation and skin contact should be minimized. If accidently ingested, it should be removed from the stomach by careful lavage with water. (Emesis should be avoided because of possible pulmonary aspiration). There may be a latent period prior to the manifestation of symptoms during which time ethanol may be administered to prevent conversion to toxic metabolites. The symptoms and treatment should parallel those of other metabolic uncoupling agents.

No human exposures to TFE which resulted in fatalities have been reported, however, the lethal action of TFE has been extensively studied in animals. Table 1 shows the LD_{50} data for various routes of administration.

TABLE 1

LETHAL DOSE LEVELS OF TRIFLUOROETHANOL

ROUTE	SPECIES	LD_{50}	REFERENCE
Oral	Rat	240 mg/kg	Hazleton (1965)
	Mouse	366 mg/kg	Blake, et al. (1967)
Intraperitoneal	Mouse	350 mg/kg	Blake, et al. (1965)
	Mouse	195 mg/kg	Airaksinen, et al. (1970)
Dermal	Rabbit	1680 mg/kg	Hazleton (1965)
Inhalation	Mouse	1.6 ml% 85.1 ppm /10 min.	Blake, et al. (1967)
	Rat	762 ppm/6 hr.	Hazleton (1965)

In dogs, an intravenous dose of TFE of 400 mg/kg was lethal within 24 hours. (Blake, et al. 1967). According to standards established by the Manufacturing

Chemists Association (1970) the acute oral LD_{50} falls in the category of a toxic substance. Therefore if ingested, efforts should be made to remove it from the stomach. A series of subhypnotic doses of ethanol is antidotal for trifluoroethanol toxicity in animals (Blake et al., 1967) and may be in humans.

Trifluoroethanol is not classified as toxic via the dermal or inhalation pathways (MCA, 1970). Nor is it a primary skin irritant, but contact with the skin should be avoided to prevent dermal absorption. Spills are readily washed off with water. Inhalation of the vapors should be avoided by working in a well ventilated area.

Animal studies have shown that trifluoroethanol causes severe eye damage similar to many chemicals (such as isopropyl alcohol and toluene). Immediate washing of the eye with water should be used following any splashes.

Inhalation Toxicity

Based on the inhalation toxicity of TFE in mice and rats it appears that a maximum atmospheric concentration in human environments should be suggested. According to the generally recognized concept of TLV (Threshold Limit Value for 8 hour exposures during a 5 day week-40 hours) it is suggested that concentrations in excess of 50-100 ppm be avoided. This level allows a safety factor of 10 in translating results of animal studies to possible human exposures. For the purpose of comparison the TLV values for some other common volatile liquids are listed below:

	TLV (ppm)
Ethanol	1000
Trichloroethylene	100
CCl_4	10
Benzene	25

Eye Irritation

When 0.1 ml of TFE was applied to the eye of rabbits it produced severe irritation, corneal opacity, iritis, marked conjunctivitis and corneal damage.

The effects were only slightly reduced by washing with water immediately after the application (Hazleton, 1965). Four human eye splashes, followed by immediate wash caused no discernable damage (Halocarbon Labs., personal communication, 1971).

Skin Exposure

There is no information available on the percutaneous absorption of TFE in humans but considering its animal toxicity and other properties, the development of systemic toxicity after external exposure is quite likely with sufficient amounts. TFE was found to produce death after high dose dermal application (24 hour) to abraded skin of rabbits and caused mild erythema on intact and abraded skin (Hazleton, 1965). Rapid decontamination should be advised in the event of skin exposure.

Symptoms of Acute Poisoning

TFE produced rapid depression of the central nervous system at high doses however its lethal action can be produced at sub-depressant dose levels. There is a latent period of 5 to 12 hours before overt symptoms appear. This delay in onset of signs increases the hazard associated with TFE usuage. Animals become markedly depressed and weak with tremors and rapid labored breathing. Salivation, vomiting, lacrimation and fever also occur. Bloody diarrhea has been observed.

Mechanism of Poisoning

Histopathologic findings at autopsy have not been conclusive. The evidence of liver toxicity (Rosenberg and Wahlstome, 1970) and pulmonary congestion (Hazleton, 1965; Blake, et al., 1967) that has been found does not appear to explain the observed symptoms.

The symptoms are similar to those seen with agents uncoupling oxidative phosphorylation such as dinitrophenol (Blake, et al., 1967). Moreover, TFE has been found to lower the ATP/ADP ratio in mouse liver (Airaksinen et al., 1970). The toxicity does not appear to be the result of conversion to monofluoroacetate as the trifluoromethyl group is refractory to dehalogenation and sodium acetate

affords no protection (Blake, et al., 1967).

There is extensive evidence that the toxicity of TFE is at least partially the result of its metabolites. Inhibitors of the oxidation of alcohols such as: disulfiram (Antabuse), 3-amino-1,2,4-triazole, allopurinal (Zyloprim), ethanol, 4-iodo-pyrazole and isoniazid have all been shown to decrease the toxicity of TFE. Allopurinol, an inhibitor of xanthine oxidase, has also been found to synergize the lethal action of TFE. A possible explanation for these findings is that trifluoroacetaldehyde, (which is also toxic-Airaksinen et al., 1970) is formed from TFE and is responsible for the toxicity.

Treatments and Antidotes

Lacking an actual poisoning experience in humans, there is no established antidote for TFE poisoning however on the basis of animal studies it would appear that the administration of inhibitors of TFE oxidation may be beneficial. Considering human safety and experimental efficacy, ethanol in sub-depressant doses appears to be the best choice. The antidoting should be continued until TFE and its metabolites are excreted which may take several days. Because of the evidence that TFE is an uncoupling agent it is also recommended that the usual treatment for Dinitrophenol poisoning be considered. In general, this consists of measures to reduce body temperature, assist respiration and maintain acid-base balance. Since human experience with TFE poisoning is lacking, treatment should be primarily symptomatic and supportive and victims should be closely observed for several days.

D. A. Blake, Ph.D. and D. R. Brown, D. Sc.
April, 1971

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Manufacturing Chemists Association (1970) Guide to Precautionary Labeling of Hazardous Chemicals, 7th Ed. Manual L-1, Washington, D.C.

Hazleton Laboratories (1965) Acute Dermal Applications - Rabbits Trifluoroethanol. Final Report to Halocarbon Products Corporation, Hackensack, New Jersey.

Hazleton Laboratories (1965) Acute Oral Administration - Rats Acute Inhalation. LC 50-Rats Trifluoroethanol. Ibid.

Halocarbon Products Corporation

82 Burlews Court, Hackensack, N. J. 07601

FLUORINOL 85

A Fluorocarbon Rankine Cycle Working Fluid

Molecular Weight	87.74
Boiling Point at 14.7 psia	166°F
Freezing Point	-82°F
Ice Point (Melting Pt.)	-82°F
Flash Point	109°F
Fire Point	None
Critical Pressure	930 psia
Critical Temperature	465.5°F

Toxicity (MCA Standards)
Inhalation - non-toxic
Dermal - non-toxic
Oral - toxic

Flammability - will not sustain combustion

Corrosion - non-corrosive with most metals

Probable upper temperature limit 600°F

ATTACHMENT V

CHARACTERISTICS OF P-1D

Specialty Chemicals Division
ALLIED CHEMICAL CORPORATION
P.O. Box 1087R, Morristown, New Jersey 07960 (201) 538-8000

August 11, 1971

Mr. Harvey W. Welsh
Thermo Mechanical Systems Co.
7252 Remmet Avenue
Canoga Park, California 91303

Dear Mr. Welsh:

In response to your request of June 9, I am enclosing various technical data on our P-1-D fluid.

As you will see, the toxicology work has not been extensive and if your application would involve unusual release, this area would need additional work.

We apologize for not replying sooner and hope that this information is sufficient. If not please advise us of your precise needs.

Very truly yours,

N. C. Fritzinger
Manager, Planning & Evaluation

NCF/ndr
Atts.

PHYSICAL AND THERMODYNAMIC PROPERTIES OF FLUOROCARBON P-1D

Allied Chemical Corporation
Specialty Chemicals Division
Morristown, New Jersey

SELECTED PHYSICAL PROPERTIES OF P-1D

Molecular Weight	570
Chemical Formula	$C_{10}F_{22}O_s$
Normal Boiling Point	135°C 275°F
Freezing Point	-85°C -121°F
Critical Temperature	244°C 470°F
Critical Pressure	172 psia
Critical Volume	0.027 cu. ft.
Liquid Density at 75°F,g./cc.	1.75
Liquid Viscosity at 75°F	1.2 c.s.
Liquid Viscosity at -40°F	11.5 c.s.
Liquid Specific Heat @ 75°F	0.24 BTU/lb./°F
Heat of Vaporization @275°F	28.6 BTU/lb.
Liquid Thermal Conductivity @75°F	0.066 BTU/hr./ft./°F
Vapor Pressure @75°F	0.103 psia
Surface Tension @77°F	13.8 dynes/cm.
Solubility of Water @75°F	45 p.p.m.
Flammability	non-flammable

TOXICITY

The following is part of a report from Rutgers University on toxicity tests on P-1-D.

PROCEDURE: Male Sprague-Dawley rats weighing 250-300 grams were exposed in a 10-liter glass disiccator containing soda-lime for carbon dioxide removal. The pressure within the desiccator was reduced to about 660 mm Hg and then brought to atmospheric pressure with oxygen; the pressure was now reduced to 520 mm Hg, the calculated amount of sample vaporized in a carburetor external to the desiccator and the vaporized sample in air slowly admitted into the evacuated chamber. By this means the concentration of oxygen in the chamber was 25 to 28% and the concentration of P-1-D about 1.5%. The closed system was then put under a positive pressure of about 1 mm Hg of oxygen, and the oxygen thereafter admitted into the desiccator upon demand, that is, in response to its use by the rats, the desiccator pressure falling by virtue of the use of oxygen and the removal of carbon dioxide.

TEST PRODUCT P-1-D: This high boiling compound was vaporized in an amount which saturated the atmosphere of the exposure chamber, yielding a concentration of approximately 1.5%. Rats were exposed for 6.3 hours; killed with ether 3 days later, no pathology was seen; the lungs were normal in appearance and weight.

CONCLUSIONS: These initial results indicate the essential physiological inertness, at the concentrations tested, of the perfluorinated ether. Apart from the apparent lack of activity on the central and autonomic nervous systems, there is also the fact that, at least for the times of exposure used, these materials do not appear to be pulmonary irritants.

Given concentrations and times which are unlikely to be exceeded in practice, this compound would appear to present no inhalational toxic potential and no danger to personnel.

STABILITY

High temperature stability tests were made on P-1-D using FC 75 as a control. The results of these tests are summarized below. These show that P-1-D has excellent stability at 700°F. under the test conditions and is considerably better than FC-75.

TABLE I :

RESULTS OF THERMAL STABILITY TESTS BASED ON THE FLUORIDE ION ANALYSES OF P-1D AND FC-75 IN THE PRESENCE OF VARIOUS METALS

Temperature: 371°C (700°F), Time: To 126 Days

Sample	Days	Copper		Stainless Steel		Aluminum		Cold Rolled Steel	
		in Fluid	on Metal	in Fluid	on Metal	in Fluid	on Metal	in Fluid	on Metal
P-1D	7	4	710	5	203	ND	190	9	338
P-1D	20	--	67	51	51	--	190	--	1020
P-1D	84	81	149	203	176	14	406	203	6500
P-1D	126	40	740	5	176	14	338	108	2300
FC-75	8	15	46	181	570	70	---	460	878
FC-75	20	610	139	540	408	810	148	1070	1430
FC-75	84	1460	4150	7800	9500	8300	6300	3860	7500
FC-75	126	----	3860	18,500	6100	12,000	3700	21,000	4630

Notes:

1. Tests were performed in an evacuated Pyrex tube containing 0.74g P-1D or 0.54g FC-75 in the presence of a metal rod of 1.26 in^2 area. Fluoride ion is expressed in ppm - parts by weight of fluoride found per part by weight of fluid.
2. Tests at 371°C of P-1D only gave 460 ppm of F^- after 126 days and 162 ppm of F^- after 213 days.
3. ND = non-detectable.

Some preliminary data was developed at 1000°F. which indicated substantial decomposition after 30 hours. Although all the decomposition products were not identified, the fragments found indicated rupture of the C-O bond.

ATTACHMENT VI

CHARACTERISTICS OF AEF-78

AEROJET LIQUID ROCKET COMPANY
P. O. BOX 13222 SACRAMENTO, CALIFORNIA 95813 • TELEPHONE (916) 355-0222

28 July 1971
9612:0098

AUG. 5 1971

Mr Robert A Yano
Project Engineer
Thermo Mechanical Systems Company
7252 Remmet Avenue
Canoga Park, California 91303

Dear Mr Yano:

Mr Alan H Kreeger has asked me to respond to your letter of 2 July 1971 in which you requested various properties of two Rankine cycle fluids: P-1D and AEF-78. Fluid AEF-78 is a recent selection for our low-pollution organic Rankine cycle engine program.

Table I summarizes some of the available properties of P-1D and AEF-78. I would suggest that you contact Allied Chemical Company, Buffalo, New York, if you require more complete data on P-1D. We are currently planning to determine several additional properties of AEF-78, particularly in the general area of health hazards and fluid/material compatibility. Table I does not contain any specific information on performance characteristics (Item 5), but perhaps we could supply required input if you would be a bit more specific about the information you require.

Table II contains some of our preliminary thoughts on fluid environmental considerations. We would be interested in any comments you might have in this area which might then enable us to respond more completely to your needs.

We are very much interested in your company's role in the demonstration of the feasibility of the Rankine cycle engine as a safe, low emission automotive power-plant and look forward to further interaction with you. Please do not hesitate to communicate with us should you require additional information.

Very truly yours,

R. Garcia

Dr Ramon Garcia
Sr Engineering Specialist

Encl: (1) Table I
(2) Table II

TABLE I - FLUID PROPERTIES

Property	Fluid AEF-78	Fluid P-1D
1. Thermodynamic Characteristics		
Boiling point	189°F	275°F
Freezing point	-103°F	-121°F
Critical temperature	484°F	470°F
Critical pressure	443 psia	172 psia
Heat of combustion	N/A	N/A
2. Fire Hazards		
Flash point	None	None
Fire point	N/A	*
Autoignition temperature	N/A	*
Explosive range	N/A	*
3. Health Hazards	N/A (to be measured)	*
4. Volume availability	Present: 5-10 gal Future: 100,000 - 1,000,000 lb/yr	Present: 5-10 gal Future: *
5. Performance Characteristics	High thermal stability	High thermal stability
6. Corrosive Characteristics	Currently being measured	*
7. Cost	Present: $800/gal Future: $100/gal	Present: $1100/gal Future: *

Notes: N/A - Not Available
* - Contact fluid manufacturer (Allied Chemical Company) for data

TABLE II - NORMAL OR ABNORMAL ENVIRONMENT LIMITS FOR FLUID AEF-78

Environmental Phases	Normal Environment	Abnormal Environment
1. Shipment of fluid raw materials	Manufacture of fluid is covered by patents held by supplier. We are not aware of fluid raw materials required or detailed manufacturing procedure at this time.	
2. Manufacture of fluid		
3. Shipment of fluid	Fluid at ambient temperature is sealed in a metal container with air occupying space above fluid.	In the event of a leak, fluid at ambient temperature would not react with air (air is already present). The fluid does not have a flash point, and there would be no fire hazard. The fluid has a low level of toxicity comparable to "FREON" refrigerant fluids and would not pose a serious problem.
4. Manufacture of engine and vehicle assembly	Fluid passages are flushed and cleaned. Fluid is introduced into system and vacuum condition is established.	
5. Normal passenger vehicle operation	Fluid is contained in a hermetically sealed system under vacuum. Fluid temperature ranges from approximately 150°F to 700°F, while fluid pressure level is 10 psia to 1000 psia.	In the event of a leak, fluid at high temperature and pressure could be released to the atmosphere. The fluid has no flash point, and there is no fire hazard. Toxicity and possible health hazards associated with the high temperature fluid are not known. Danger of personnel coming into contact with hot fluid is present.

Table II continued

Environmental Phases	Normal Environment	Abnormal Environment
6. Vehicle accidents	Fluid is contained within system. There is no hazard to personnel.	Fluid could leak from system at high temperature. There is no fire hazard. Toxicity of high temperature fluid is unknown. Danger of personnel coming into contact with hot fluid is present.
7. Garage maintenance or repair	Drain fluid at ambient temperature. Flush system. Perform repair and recharge fluid. Establish vacuum.	
8. Vehicle or engine scrapping	Drain fluid at ambient temperature. Flush system. Engine can be scrapped. Fluid may be saved for further use if decomposition is slight or reprocessed for future use.	

8.0 POWERPLANT COMPARISON

8.1 INTRODUCTION

In any study of this nature, it is desirable to finally make a comparison between the various concepts investigated along with conventional and Rankine cycle systems. However, a rigorous comparison poses significant problems. For example, it is frequently argued that the automotive gas turbine is lighter than the conventional spark ignition engine, therefore placing the gas turbine in an advantageous position. On the other hand, the prime criteria in the manufacture of the conventional spark ignition engine is minimum cost. If weight was of particular importance, the block could be made of aluminum instead of cast iron, or the engine could be made air-cooled as are engines for light aircraft. Thus, in this case of the conventional spark ignition engine, engine weight has been traded to gain minimum cost. Therefore, this section on powerplant comparisons will attempt to compare weight, cost, efficiency, etc. of each powerplant on a generalized basis rather than trying to provide specific quantities for each parameter.

8.2 COMPARISONS

Table 8-I presents a comparison of the characteristics considered most important in evaluating the various powerplants with respect to automotive feasibility. The spark ignition internal combustion (Otto) engine, without extensive emission controls, is used as the reference powerplant and the other candidate powerplants are compared to it.

The diesel, Wankel, stratified charge, and Otto cycle engines are all considered to be generally comparable as a class, all being internal combustion engines. The Wankel has the advantage of being smaller and lighter, whereas, the diesel trades off weight and cost to gain fuel economy. On the other hand, the ECPE, Rankine, and Stirling, all being external combustion engines*, will be somewhat worse with respect to specific weight, cost, and complexity as compared to the Otto cycle while the gas

*External combustion engine, as used here, refers to a powerplant in which the air and fuel mixing and combustion occurs outside or external to the power producing machinery.

TABLE 8-I

COMPARISON OF VARIOUS AUTOMOTIVE POWERPLANTS

Engine Type	Specific Weight/lb/HP	Cost $/HP	Complexity	Full-load Thermal Eff.	Road-load Thermal Eff.	Drive-ability
Otto (SI-IC)	Ref.	Ref.	Ref.	Ref.	Ref.	Ref.
Stratified Charge	Worse	Worse	Worse	Better	Better	Equal
Diesel	Worse	Worse	Worse	Better	Better	Equal
Wankel	Better	Equal	Better	Equal	Equal	Equal
Gas Turbine (Regen)	Equal	Worse	Equal	Equal	Worse	Worse
Rankine	Worse	Worse	Worse	Equal	Equal	Equal
ECPE	Worse	Worse	Worse	Equal	Equal	Worse
Stirling	Worse	Worse	Worse	Better	Better	Worse

turbine should be comparable with respect to weight and complexity, but significantly worse with respect to cost. The complexity of the gas turbine was considered to be comparable to the Otto engine, even though there are far fewer moving parts in the gas turbine, due to the fact that the high operating speeds and required close bearing, nozzle, and rotor tolerances demands somewhat more complexity for the fewer moving parts that it does have. The fuel economy for the external combustion cycles is shown to be equal to or better than that for the Otto cycle except for the part load fuel consumption of the gas turbine. Of course, there are indications that some of the advanced gas turbine concepts, as studied in this report (such as the plate-fin recuperative gas turbine), are comparable to the Otto cycle at both full and part power. However, in general, the part load fuel consumption of the turbine is worse than the Otto cycle and is shown in Table 8-I to indicate an area where continued improvement, both analytical and experimental, is needed. With respect to driveability, the external combustion engines, in general, exhibit greater acceleration lag than internal combustion engines except for the Rankine engines, which, after startup, miantain a constant source of high temperature, high pressure working fluid in the boiler. With respect to startup, however, the Rankine engine should require somewhat more time than any of the other engines due to the requirement to fill the high temperature, high pressure boiler.

Thus, it appears that there is no single powerplant which is clearly superior to the spark ignition internal combustion engine. It would appear that a strong case for any of the alternate powerplants can be made only if such powerplant(s) can show significant advantages with respect to exhaust emissions over that possible with the conventional spark ignition engine.

8.2.1 Spark Ignition (Otto Cycle) Engine

The spark ignition engine has been the prime mover for automobiles for many years. It has many advantages. It provides immediate starting and rapid response. The weight, volume, and cost are reasonable, and the demonstrated thermal efficiency is exceeded only by that of the diesel and Stirling engines. The primary construction material is cast iron which is both abundant and inexpensive.

Table 8-II presents demonstrated exhaust emissions of various automotive powerplants. To date, the only engine that has met the 1976 emission standards, is the Ford stratified charge engine. Thermal Electron Corporation (TECO) has demonstrated that they can meet the 1976 emission standards with their Rankine burner, however, only the burner has been tested, and not the burner-powerplant combination. Whether the burner will perform as well when combined with the powerplant, installed in a vehicle, and subjected to the various driving cycles, is merely speculations at the present time. The Ford stratified charge engine is still in the R&D stage and it may be some time before it could go into mass production. Nevertheless, the potential for low emissions, with an internal combustion engine, has been proven.

DuPont has supplied the State of California with six Chevrolet Impalas equipped with exhaust gas recirculation and exhaust gas thermal reactors, and six standard Impalas for reference use. The emission levels for the cars equipped with controls are shown in Table 8-II. As can be seen, the unburned hydrocargon and NO_x emissions approach the 1976 standards while the CO emissions are significantly higher. These are full-sized conventional passenger automobiles operating in every day service, and demonstrate the emission gains which have been made with the conventional Otto cycle engine.

A group of students from Wayne State University entered a vehicle in the 1970 Clean Car Race which showed significant reduction in emissions. The vehicle was a modified 1971 Ford Capri powered with a 302 cubic inch V-8 engine. The engine was fueled with unleaded gasoline and the car was altered to reduce its weight and aerodynamic drag. The emissions system consisted basically of an exhaust gas recirculation system (EGR) and a catalytic converter system. A low overlap camshaft was installed in the engine to improve the effectiveness of the EGR system. Included in the dual exhaust system were two pairs of catalytic converters to chemically remove NO_x, carbon monoxide, and unburned hydrocarbons. A modified air cleaner, carburetor and fuel system, along with a catalyst-temperature modulated air injection system, were used to maintain maximum catalyst efficiency at all engine operating conditions. The vehicle performance and measured emissions were as follows:

TABLE 8-II

DEMONSTRATED EMISSION LEVELS OF VARIOUS AUTOMOTIVE POWERPLANTS

Engine & Reference	Emissions, gm/mile HC	CO	NO_x
1976 Federal Standard	0.41	3.40	0.40
SPARK IGNITION I.C. ENGINE			
1. Chevrolet Impala with DuPont Reactor (49)	0.46	10.05	0.60
2. Wayne State Clean Car Entry	0.49	5.38	0.43
3. Liq. Pet. Gas, Thompson, NAPCA (50)	1.0-2.4	1.2-4.2	1.7-3.3
4. Ford Stratified Charge (EPA Release Sep. '71)	0.37	0.93	0.33
DIESEL			
1. Mercedes 220D, (51)	0.30	1.62	1.83
GAS TURBINE			
1. Chrysler Gas Turbine Car (52)	0.91	7.04	1.86
2. Williams Research (53)	0.34	4.5	2.15
RANKINE			
1. 1923 Doble (43)	1.4	3.9	2.0
2. SE-101 Grand Prix (43)	1.0	8.0	2.2
3. SE-124 Chevelle (43)	0.3	1.0	1.7

Acceleration, 0-60 mph:	7.5 sec.
Turnpike fuel economy:	24.5 mpg
Exhaust emissions (Test per Fed. Reg. 11/10/70)	
- hydrocarbons:	0.49 gm/mi.
- carbon monoxide:	5.38 gm/mi.
- nitrogen oxides:	0.43 gm/mi.

It is seen that the exhaust emissions are very close to the 1976 standards while maintaining outstanding vehicle performance.

Another spark ignition engine which is a strong contender as an automotive powerplant is the rotary engine. Dr. Felix Wankel and a team from NSU Notorenwerke A.G. Nekarsulm first publicly demonstrated the Wankel rotary engine on January 19, 1960. Since that time substantial progress has been made in the development of this unique engine. Advantages of the rotary engine over the reciprocating engine are as follows:

1. less vibration
2. simpler structure
3. ports replace valves eliminating the need for valve operating gear
4. less weight and space are required for equal horsepower output.

Some of the disadvantages currently facing the rotary engine are as follows:

1. the configuration of the casing makes it difficult to manufacture
2. there is a complex problem of sealing between the rotor and the casing
3. combustion and lubrication problems are more severe than with reciprocating engines
4. because of the large combustion chamber surface to volume ratio, the engine emits relatively higher quantities of unburned hydrocarbons and carbon monoxide.

There is much optimism in the automotive industry that the above disadvantages will be overcome, and the smaller, lighter engine permits more room in the engine compartment for exhaust emission controls. If the

disadvantages listed above are, in fact, overcome, the Wankel rotary engine may well be the engine of the future.

8.2.2 Compression Ignition (Diesel Cycle) Engine

As shown in Table 8-I, diesel engines weigh more and cost more per horsepower, and are more complex, than Otto cycle engines. However, diesels provide significantly better fuel economy than Otto cycle engines and have enjoyed wide acceptance in trucks and other vehicles where fuel costs are considered a major operating expense. Table 8-II presents the results of constant volume sampling (CVS) emission tests by Southwest Research Institute on the Mercedes 220D diesel powered automobile. The HC and CO emissions of 0.30 and 1.62 gm/mile are well within the 1976 standards while the NO_x emission of 1.83 gm/mile is still substantially higher than the standard. The CO emissions are low because diesels inherently run at lean mixtures while the HC emissions are low for that reason and because the combustion process is usually completed before the fuel-air mixture can reach the cylinder wall and be quenched. In the future it may be possible to reduce the oxides of nitrogen by using exhaust gas recirculation, lower compression ratios, and prechamber combustion. Diesels have the disadvantage of forming aldehydes which create an obnoxious odor.

8.2.3 Gas Turbine

The gas turbine has proved to be an ideal propulsion system for high speed aircraft, but as an automotive powerplant, it has not come into general use for a number of reasons. Basically, there are three problem areas. First, the gas turbine powerplant is characterized by acceptable fuel consumption only at high turbine temperatures and pressure ratios characteristics of high rotor speeds and power output. To obtain acceptable fuel consumption at part power and speed, high turbine temperatures must be maintained in combination with a very effective heat recovery system. Second, high operating temperatures require turbine and combustor materials having a high content of critical metals. And third, the torque speed characteristics of the automobile and the gas turbine engine are such that either a complicated turbine arrangement or a complicated transmission are required. Perhaps the most critical of the above problem areas is the

requirement for high temperature turbine materials. Reference 16 indicates that a 4:1 pressure ratio gas turbine using a ceramic regenerator would require about 2.03 lb of cobalt per engine and 12.2 lb of nickel per engine. For an annual production of 10 million engines, the cobalt required would be 156% of current U.S. consumption and the nickel required would be 38% of current U.S. consumption. For this many engines, the cost of manufacture per engine would go down significantly but the cost of material would increase. The ultimate cost per engine is difficult to estimate.

From the HC and CO emission standpoint, the gas turbine has some definite advantages (Table 8-II). The Williams Research engine in a Volkswagen squareback vehicle produces 0.34 gm/mile of unburned hydrocarbons, which is within the 1976 standards, and CO emissions of 4.5 gm/mile, which is very close to the 1976 standard. However, the 2.15 gm/mile of NO_x is above the standard by a factor of 5. It is difficult to speculate when a solution to the NO_x problem will be found, but it is anticipated that eventually the problem will be solved.

The Williams Research engine produces 80 HP, weighs 400 pounds, and completely fills the engine compartment of the AMC Hornet in which it is installed (53). The fuel economy is less than that of a comparable piston engine, and because the vehicle is underpowered, it requires 16 to 20 seconds to accelerate from 0 to 60 mph. However, advanced gas turbine concepts (such as the higher pressure ratio plate-fin recuperative cycle shown in this study) have, on paper, shown much improved performance and efficiency levels over that demonstrated in the Williams engine. The Williams engine itself can be improved by incorporating variable power turbine nozzles and by increasing turbine inlet temperature. It appears that continued development is required on various gas turbine powerplant concepts before the gas turbine can be considered to be an acceptable automotive powerplant.

8.2.4 Rankine Cycle

The steam engine was a well established predecessor to the internal combustion engine and was superceded by the internal combustion engine because the latter engine was more efficient, more compact, less complicated, and less expensive. The advent of organic working fluids has improved the fuel economy of this type of closed cycle powerplant, but it has not significantly improved the complexity, compactness, cost or driveability

of the system. On cold starts, warmup time is required to build up a head of steam, and to heat up steam pipes and expander to preclude condensation in them. In addition, many of the working fluids now being considered are either toxic, flammable, freezable, unstable, corrosive, or very expensive (several hundred dollars per charge).

Demonstrated emission levels of the Rankine powered automobiles are shown in Table 8-II. Again, the NO_x emission levels are higher than the 1976 standard by a factor of about 5. Recent combustion research sponsored by EPA has indicated emission levels from the Rankine burner can be made to meet the 1976 standard. However, even if the burner-powerplant combination can be made to meet the 1976 standards, it is doubtful that it can be made into an acceptable automotive powerplant, for the reasons stated above.

8.2.5 External Combustion Piston Engine (ECPE)

As an automotive powerplant, the ECPE is similar in many respects to the Rankine cycle engine. Both will cost more, weigh more, and are more complex than the conventional Otto cycle engine. Both use external combustors and have about the same potential for meeting the 1976 emission standards. Because of the similarity of the operational characteristics and the fact that comparable components exist in each of the ECPE and Rankine systems, it is concluded that both are equally feasible as low emission automotive powerplants. The Rankine engine may be slightly superior with regard to the problem of acceleration lag due to a continuous source of high energy working fluid in the boiler. However, the ECPE does not have the freezing, lubrication, and separation of fluid problems of the Rankine engine. It is judged that their overall costs should be comparable. In comparison to the gas turbine engine, the ECPE promises to have favorable part load fuel consumption and a lesser requirement for critical materials (due to a much lower cycle temperature), although total system weight is higher.

8.2.6 Stirline Engine

The Stirling engine is a relatively high efficiency powerplant, with an efficiency falling close to the diesel cycle engine. From the emissions

standpoint, it has the advantage of an external burner so emissions should be similar to the Rankine cycle engines. On the other hand, the Stirling engine introduces a degree of complexity far beyond that required for the internal combustion engine. It is estimated that the size, cost, and weight of such an engine would be at least twice that of the existing gasoline internal combustion engine.

Hydrogen is currently used as a working fluid in most Stirling engines. Hydrogen is a higly flammable gas and may pose a problem insofar as hazards are concerned. The fact that there is only a small quantity of hydrogen used in each engine, however, tends to reduce the hazard. Helium can be used but the working pressure must be increased from about 2000 psi to about 3000 to 3500 psi to maintain equivalent power output. This higher pressure involves major structural problems. Although the Stirline engine does have potential as a low-emission powerplant, it is believed that other engines being considered offer greater merit as an acceptable automotive powerplant.

8.3 CONCLUSIONS

Based upon the comparisons of various powerplants reported in this section, the following conclusions are drawn:

1. The gas turbine inherently produces low values of HC and CO emissions. It does use significant quantities of critical alloys and currently produces NO_x emissions higher than the 1976 standard by a factor of 4 or 5. Major advances in gas turbine technology will have to be made for this engine to become an acceptable automotive powerplant.
2. EPA burner/vapor generator development programs have demonstrated that the Rankine engine can meet the Federal 1976 NO_x emission standards. This remains to be demonstrated in a vehicle. The Rankine engine is, however, costly and complex, and it may not be an acceptable automotive powerplant.
3. The External Combustion Piston Engine eliminates some of the undesirable Rankine engine working fluid problems. Because the ECPE concept, like the Rankine, is more complex than Otto cycle and gas turbine engines, it may not prove to be

as acceptable a powerplant for vehicle applications. Further design and development work is required to ascertain the ECPE potential.

4. The Stirling engine has the potential of being a low-emission powerplant, but it is complex and costly. It is believed that other engines being considered offer greater merit.
5. Based on demonstrated emission levels, various versions of the internal combustion engine (Wayne State Clean Car entry, diesel engines, stratified charge concept, DuPont reactor system, etc.) have significant leads over other proposed powerplants. It could well be that this lead will be difficult to overcome.

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59. New Mach Number Tables for Ram-Jet Flow Analyses, Ordnance Aerophysics Laboratory, August, 1955.

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10.0 APPENDIX I

OAP Vehicle Design Goals and Road Load Characteristics for the Six-Passenger Automobile.

Thermo Mechanical Systems Co. has adhered as closely as possible to the OAP design goals for the standard six-passenger vehicle (Revision C - May 28, 1971). These goals are presented at the end of this section. For this study, less attention was directed towards the specific power ratings of the various powerplants examined since they were not (and not intended to be) designed in the detail necessary to accurately define the weight, torque curves, etc., to determine acceleration rates, parasitic drive losses, etc. Usually, such characteristics are determined only after the hardware is fabricated and a basic engine calibration can be measured. Therefore, with respect to adhering to the design goals, considerable latitude exists for the various powerplants such that the design parameter of each can be "scaled" to satisfy all the design goals. For example, if the parasitic drive train losses for the ECPE vehicle should be larger than those assumed, then the mean cycle pressure need merely be increased over the nominal value to enable the vehicle to meet the design goals. In addition, the acceleration required for various power systems will vary depending on its particular torque characteristics. For instance, a high torque Rankine cycle engine will require lower power levels to meet the vehicle acceleration requirement than a low torque single-shaft gas turbine engine. However, it has been determined that there is only a slight weight penalty, for say, an 8-speed transmission relative to a 3-speed transmission, and therefore, the torque curve of the engine can be made to satisfy the acceleration requirements by using the required multi-speed transmissions. Therefore, it was decided to neglect secondary differences in power train characteristics and assume similar drive train losses, transmission characteristics, and acceleration rates for each of the powerplants examined, which results in one peak power value.

The above approach is further justified as shown by Figure I-1 where the road load power as a function of speed is shown for various weight vehicles. The weights vary from a minimum of 3700 lbs to a maximum of 5300 lbs which includes up to 2600 lbs for the powerplant, power

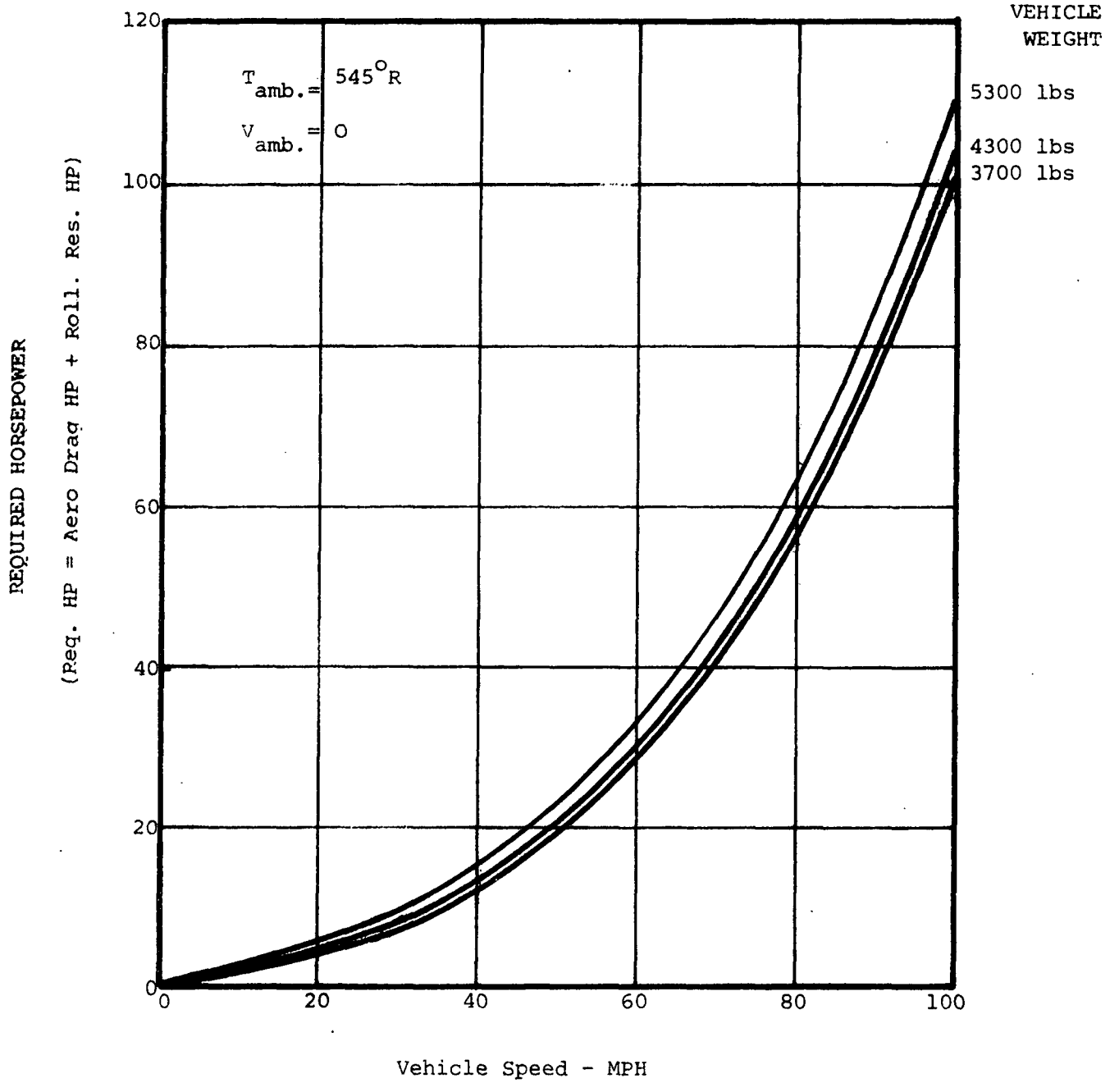

FIGURE I.1 Total Power Required due to Rolling Resistance and Aerodynamic Drag as a Function of Vehicle Speed.

train, payload, and test load (see "Vehicle Design Goals"). It can be seen that the powerplant weight will have little effect on road load power. From the "Vehicle Design Goals" it was determined that for a full size vehicle (2700 lbs) with an engine plus power train weight of 1300 lbs and a standard passenger and test load of 300 lbs, the design weight is 4300 lbs. The road load horsepower (aero + rolling) for this 4300 lb automobile is also shown in Figure I.1.

The power losses for the transmission, drive train, fans, and intake and exhaust ducting were assumed to be as follows:

Fan and auxiliary losses	6%
Transmission losses	7%
Rear axle losses	4%
Intake and Exhaust losses	3%
	20%

Therefore, the total estimated engine and drive-line power losses were 20%. Combining this with the average 4 HP accessory load (as given in the design goals) and the aero and rolling HP requirement as shown in Figure I.1, gives:

$$P_V = P_{RR} + P_{AD} + P_{ACC} + 0.20\ P_{RL}$$

or

$$P_V = 1.25\ (P_{RR} + P_{AD}) + 4.9$$

where:

P_V = total vehicle power requirement

P_{RR} = road rolling resistance

$= W/65 \left[1 + (1.4\text{x}10^{-3})\ V + (1.2\text{x}10^{-5})V^2\right]/550$

P_{AD} = road aerodynamic resistance

$= C_D A_f \left[\rho(V^3/2g)/550\right]$

P_{ACC} = average accessory load = 4 HP

W = vehicle weight = 4300 lbs

V = vehicle speed (ft/sec)

$C_D A_f$ = air drag = 12 ft^2

Figure I.2 shows the total vehicle power curve for the 4300 lb vehicle at ambient conditions of 14.7 psia and 545°R.

Finally, the peak power requirement for the vehicle was determined from the vehicle design goals. The most demanding maneuver was accelerating from a standing start to 440 ft in 10.0 seconds. The power required was approximately 150-160 HP for the 4300 lb vehicle (including all auxiliary and accessory requirements). Therefore, the powerplant design power level was determined to be 160 HP and, referring to Figure I.2, this gives a maximum speed of about 107 mph.

ROAD LOAD POWER REQUIREMENTS

APCO VEHICLE DESIGN GOALS

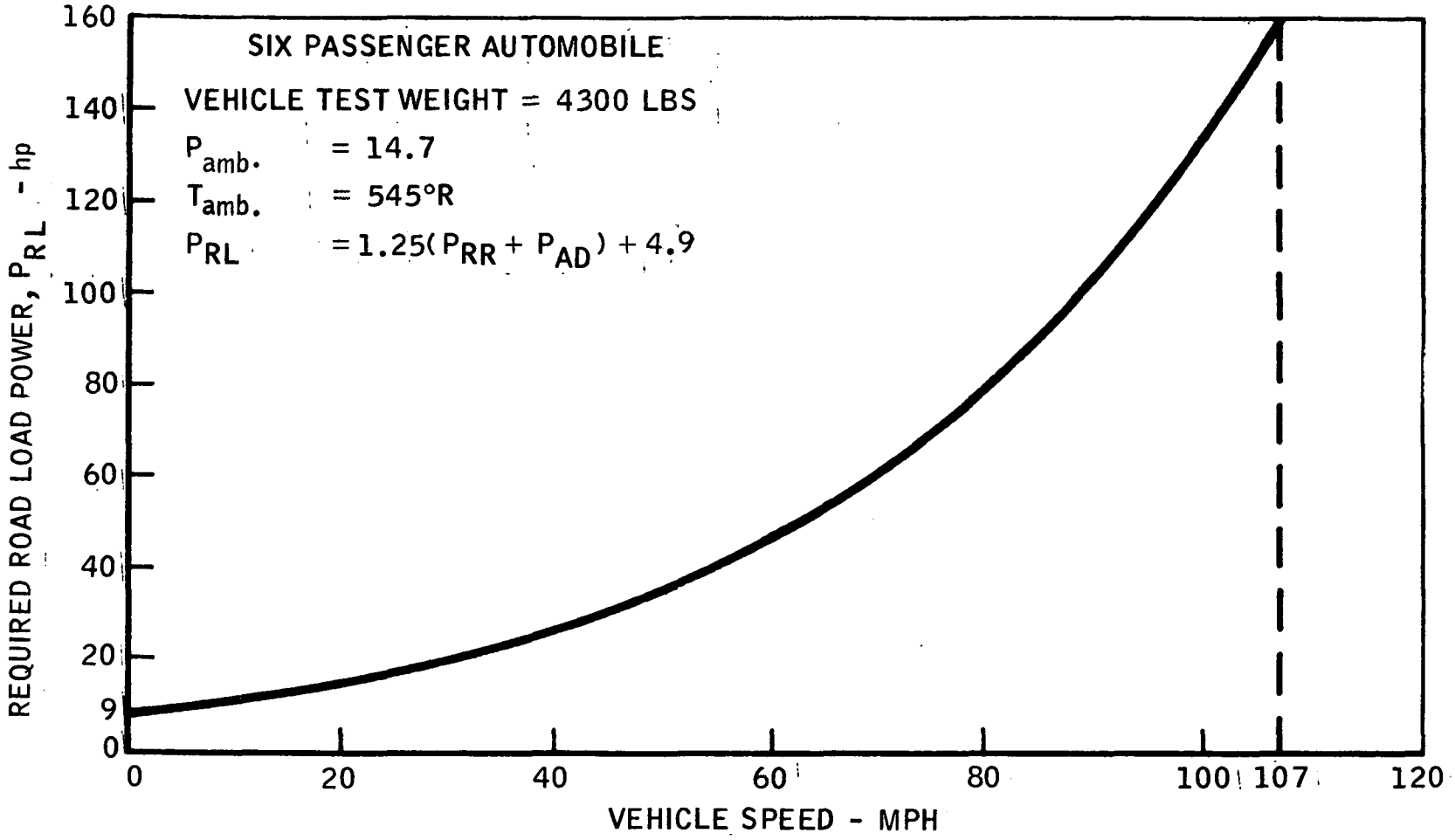

FIGURE I.2

AIR POLLUTION CONTROL OFFICE

ADVANCED AUTOMOTIVE POWER SYSTEMS PROGRAM

"Vehicle Design Goals - Six Passenger Automobile"

(Revision C - May 28, 1971 - 11 Pages)

The design goals presented below are intended to provide:

A common objective for prospective contractors.

Criteria for evaluating proposals and selecting a contractor.

Criteria for evaluating competitive power systems for entering first generation system hardware.

Advisory criteria in such areas as rolling resistance, vehicle air drag etc. are included to assist the contractor.

The derived criteria are based on typical characteristics of the class of passenger automobiles with the largest market volume produced in the U. S. during the model years 1969 and 1970. It is noted that emissions, volume and most weight characteristics presented are maximum values while the performance characteristics are intended as minimum values. Contractors and prospective contractors who take exceptions must justify these exceptions and relate these exceptions to the technical goals presented herein.

1. Vehicle weight without propulsion system - W_o.

   W_o is the weight of the vehicle without the propulsion system and includes, but is not limited to: body, frame, glass and trim, suspension, service brakes, seats, upholstery, sound absorbing materials, insulation, wheels (rims and tires), accessory ducting, dashboard instruments and accessory wiring, battery, passenger compartment heating and cooling devices and all other components not included in the propulsion system. It also includes accessories such as, the air conditioner compressor, the power steering pump, and the power brakes actuating device.

   W_o is fixed at 2700 lbs.

2. Propulsion system weight - W_p.

   W_p includes the energy storage unit (including fuel and containment), power converter (including both functional components and controls) and power transmitting components to the driven wheels. It also includes the exhaust system, pumps, motors, fans and fluids necessary for operation of the propulsion system, and any propulsion system heating or cooling devices.

The maximum allowable propulsion system weight, W_{pm}, is 1600 lbs. However, light weight propulsion systems are highly desired. (Equivalent 1970 propulsion system weight with a spark ignition engine is 1300 lbs.)

3. Vehicle curb weight - W_c

$W_c = W_o + W_p$

The maximum allowable vehicle curb weight, W_{cm}, is 4300 lbs. (2700 + 1600 max. = 4300)

4. Vehicle test weight - W_t.

$W_t = W_c + 300$ lbs. W_t is the vehicle weight at which all accelerative maneuvers, fuel economy and emissions are to be calculated. (Items 8c, 8D, 8e).

The maximum allowable test weight, W_{tm}, is 4600 lbs. (2700 + 1600 max. + 300 = 4600).

5. Gross vehicle weight - W_g

$W_g = W_c + 1000$ lbs. W_g is the gross vehicle weight at which sustained cruise grade velocity capability is to be calculated. (Item 8f). The 1000 lbs. load simulates a full load of passengers and baggage.

The maximum allowable gross vehicle weight, W_{gm}, is 5300 lbs. (2700 + 1600 max. + 1000 = 5300).

6. Propulsion system volume - V_p

V_p includes all items identified under item 2. V_p shall be packagable in such a way that the volume encroachment on either the passenger or luggage compartment is not significantly different than today's (1970) standard full size family car. The propulsion system shall not violate the vehicle ground clearance lines as established by the manufacturer of the vehicle used for propulsion system/vehicle packaging. Additionally, the propulsion system shall not violate the space allocated for wheel jounce motions and vehicle steering. Necessary external appearance (styling) changes will be minor in nature. V_p shall also be packagable in such a way that the handling characteristics of the vehicle do not depart significantly from a 1970 full size family car.

The maximum allowable volume assignable to the propulsion system, V_{pm}, is 35 $ft.^3$.

7. Emission Goals †

The vehicle when tested for emissions in accordance with the procedure outlined in the November 10, 1970 Federal Register shall have a weight of W_t. The emission goals for the vehicle are:

Hydrocarbons* †	-	0.14 grams/mile maximum
Carbon monoxide †	-	4.7 grams/mile maximum
Oxides of nitrogen**	-	0.4 grams/mile maximum
Particulates	-	0.03 grams/mile maximum

*Total hydrocarbons (using 1972 measurement procedures) plus total oxygenates. Total oxygenates including aldehydes will not be more than 10 percent by weight of the hydrocarbons or 0.014 grams/mile, whichever is greater.

**measured or computed as NO_2.

† For updated emission standards refer to EPA Prototype Vehicle Performance Specification, Jan. 3, 1972, item 9, "Propulsion System Emissions."

8. Start up, Acceleration, and Grade Velocity Performance.

a. Start up:

The vehicle must be capable of being tested in accordance with the procedure outlined in the November 10, 1970 Federal Register without special driver startup/warmup procedures.

The maximum time from key on to reach 65 percent full power is 45 sec. Ambient conditions are 14.7 psia pressure, 60°F temperature.

Powerplant starting techniques in low ambient temperatures shall be equivalent to or better than the typical automobile spark-ignition engine. Conventional spark-ignition engines are deemed satisfactory if after a 24 hour soak at -20°F the engine achieves a self-sustaining idle condition without further driver input within 25 seconds. No starting aids external to the normal vehicle system shall be needed for -20°F starts or higher temperatures.

b. Idle operation conditions:

The fuel consumption rate at idle operating condition will not exceed 14 percent of the fuel consumption rate at the maximum design power condition. Recharging of energy storage systems is exempted from this requirement. Air conditioning is off, the power steering pump and power brake actuating device, if directly engine driven, are being driven but are unloaded.

The torque at transmission output during idle operation (idle creep torque) shall not exceed 40 foot-pounds, assuming conventional rear axle ratios and tire sizes. This idle creep torque should result in level road operation in high gear which does not exceed 18 mph.

c. Acceleration from a standing start:

The minimum distance to be covered in 10.0 sec. is 440 ft. The maximum time to reach a velocity of 60 mph is 13.5 sec. Ambient conditions are 14.7 psia, 85° F. Vehicle weight is W_t. Acceleration is on a level grade and initiated with the engine at the normal idle condition.

d. Acceleration in merging traffic:

The maximum time to accelerate from a constant velocity of 25 mph to a velocity of 70 mph is 15.0 sec. Time starts when the throttle is depressed. Ambient conditions are 14.7 psia, 85° F. Vehicle weight is W_t, and acceleration is on level grade.

e. Acceleration, DOT High Speed Pass Maneuver:

The maximum time and maximum distance to go from an initial velocity of 50 mph with the front of the automobile (18 foot length assumed) 100 feet behind the back of a 55 foot truck traveling at a constant 50 mph to a position where the back of the automobile is 100 feet in front of the front of the 55 foot truck is, 15 sec. and 1400 ft. The entire maneuver takes place in a traffic lane adjacent to the lane in which the truck is operated. Vehicle will be accelerated until the maneuver is completed or until a maximum speed of 80 mph is attained, whichever occurs first. Vehicle acceleration ceases when a speed of 80 mph is attained, the maneuver then being completed at a constant 80 mph. (This does not imply a design requirement limiting the maximum vehicle speed to 80 mph.) Time starts when the throttle is depressed. Ambient conditions are 14.7 psia, 85° F. Vehicle weight is W_t, and acceleration is on level grade.

f. Grade velocity:

The vehicle must be capable of starting from rest on a 30 percent grade and accelerating to 15 mph and sustaining it. This is the steepest grade on which the vehicle is required to operate in either the forward or reverse direction.

The minimum cruise velocity that can be continuously maintained on a 5 percent grade with an accessory load of 4 hp shall be not less than 60 mph.

The vehicle must be capable of achieving a velocity of 65 mph up a 5 percent grade and maintaining this velocity for a period of 180 seconds when preceded and followed by continuous operation at 60 mph on the same grade (as above).

The vehicle must be capable of achieving a velocity of 70 mph up a 5 percent grade and maintaining this velocity for a period of 100 seconds when preceded and followed by continuous operation at 60 mph on the same grade (as above).

The minimum cruise velocity that can be continuously maintained on a level road (zero grade) with an accessory load of 4 hp shall be not less than 85 mph with a vehicle weight of W_t.

Ambient conditions for all grade specifications are 14.7 psia 85° F. Vehicle weight is W_g for all grade specifications except the zero grade specification.

The vehicle must be capable of providing performance (Paragraphs 8c, 8d, 8e 8f)within 5 percent of the stated 85° F values, when operated at ambient temperatures from -20° F to 105° F.

9. Minimum vehicle range:

Minimum vehicle range without supplementing the energy storage will be 200 miles. The minimum range shall be calculated for, and applied to each of the two following modes: 1) A city-suburban mode, and 2) a cruise mode.

Mode 1: Is the driving cycle which appears in the November 10, 1970 Federal Register. For vehicles whose performance does not depend on the state of energy storage, the range may be calculated for one cycle and ratioed to 200 miles. For vehicles whose performance does depend on the state of energy storage the Federal driving cycle must be repeated until 200 miles have been completed.

Mode 2: Is a constant 70 mph cruise on a level road for 200 miles.

The vehicle weight for both modes shall be, initially, W_t. The ambient conditions shall be a pressure of 14.7 psia, and temperatures of 60° F, 85° F and 105° F. The vehicle minimum range shall not decrease by more than 5 percent at an ambient temperature of -20° F.

For hybrid vehicles, the energy level in the power augmenting device at the completion of operation will be equivalent to the energy level at the beginning of operation.

10. System thermal efficiency:

System thermal efficiency will be calculated by two methods:

A. A "fuel economy" figure based on 1) miles per gallon (fuel type being specified) and 2) the number of Btu per mile required to drive the vehicle over the 1972 Federal driving cycle which appears in the November 10, 1970 Federal Register. Fuel economy is based on the fuel or other forms of energy delivered at the vehicle. Vehicle weight is W_t.

B. A "fuel economy" figure based on 1) miles per gallon (fuel type being specified) and 2) the number of Btu per mile required to drive the vehicle at constant speed, in still air, on level road, at speeds of 20, 30, 40, 50, 60, 70, and 80 mph. Fuel economy is based on the fuel or other forms of energy delivered at the vehicle. Vehicle weight is W_t.

In both cases, the system thermal efficiency shall be calculated with sufficient electrical, power steering and power brake loads in service to permit safe operation of the automobile. Calculations shall be made with and without air conditioning operating. The ambient conditions are 14.7 psia and temperatures of 60° F, 85° F and 105° F. Calculations shall be made with heater operating at ambient conditions of 14.7 psia and 30° F (18,000 Btu/hr).

11. Air Drag Calculation:

The product of the drag coefficient, C_d, and the frontal area, A_f, is to be used in air drag calculations. The product C_dA_f has a value of 12 ft^2. The air density used in computations shall correspond to the applicable ambient air temperature.

12. Rolling Resistance:

Rolling resistance, R, is expressed in the equation $R = W/65\ [1 + (1.4 \times 10^{-3}V) + (1.2\ 10^{-5}V^2)]$ lbs. V is the vehicle velocity in ft/sec. W is the vehicle weight in lbs.

13. Accessory power requirements:

The accessories are defined as subsystems for driver assistance and passenger convenience, not essential to sustaining the engine operation and include: the air conditioning compressor, the power steering pump, the alternator (except where required to sustain operation), and the power brakes actuating device. The accessories also include a device for heating the passenger compartment if the heating demand is not supplied by waste heat.

Auxiliaries are defined as those subsystems necessary for the sustained operation of the engine, and include condensor fan(s), combustor fan(s), fuel pumps, lube pumps, cooling fluid pumps, working fluid pumps and the alternator when necessary for driving electric motor driven fans or pumps.

The maximum intermittent accessory load, P_{aim}, is 10 hp (plus the heating load, if applicable). The maximum continuous accessory load, P_{acm}, is 7.5 hp (plus the heating load if applicable). The average accessory load, P_{aa}, is 4 hp.

If accessories are driven at variable speeds, the above values apply. If the accessories are driven at constant speed, P_{aim} and P_{acm} will be reduced by 3 hp.

14. Passenger comfort requirements:

Heating and air conditioning of the passenger compartment shall be at a rate equivalent to that provided in the present (1970) standard full size family car.

Present practice for maximum passenger compartment heating rate is approximately 30,000 Btu/hr. For an air conditioning system at 110° F ambient, 80° F and 40% relative humidity air to the evaporator, the rate is approximately 13,000 Btu/hr.

15. Propulsion system operating temperature range:

The propulsion system shall be operable within an expected ambient temperature range of -40° to 125° F.

16. Operational life:

The mean operational life of the propulsion system should be approximately equal to that of the present spark-ignition engine. The mean operational life should be based on a mean vehicle life of 105,000 miles or ten years, whichever comes first.

The design lifetime of the propulsion system in normal operation will be 3500 hours. Normal maintenance may include replacement of accessable minor parts of the propulsion system via a usual maintenance procedure, but the major parts of the system shall be designed for a 3500 hour minimum operation life.

The operational life of an engine shall be determined by structural or functional failure causing repair and replacement costs exceeding the cost of a new or rebuilt engine. (Functional failure is defined as power degradation exceeding 25 percent or top vehicle speed degradation exceeding 9 percent).

17. Noise standards: (Air conditioner not operating)

a. Maximum noise test:

The maximum noise generated by the vehicle shall not exceed 77 dbA when measured in accordance with SAE J986a. Note that the noise level is 77 dbA whereas in the SAE J986a the level is 86 dbA.

b. Low speed noise test:

The maximum noise generated by the vehicle shall not exceed 63 dbA when measured in accordance with SAE J986a except that a constant vehicle velocity of 30 mph is used on the pass-by, the vehicle being in high gear or the highest gear in which it can be operated at that speed.

c. Idle noise test:

The maximum noise generated by the vehicle shall not exceed 62 dbA when measured in accordance with SAE J986a except that the engine is idling (clutch disengaged or in neutral gear) and the vehicle passes by at a speed of less than 10 mph. the microphone will be placed at 10 feet from the centerline of the vehicle pass line.

18. Safety standards:

The vehicle shall comply with all current Department of Transportation Federal Motor Vehicle Safety Standards. Reference DOT/HS 820 083.

19. Reliability and maintainability:

The reliability and maintainability of the vehicle shall equal or exceed that of the spark-ignition automobile. The mean-time-between failure should be maximized to reduce the number of unscheduled service trips. All failure modes should not represent a serious safety hazard during vehicle operation and servicing. Failure propagation should be minimized. The power plant should be designed for ease of maintenance and repairs to minimize costs, maintenance personnel education, and downtime. Parts requiring frequent servicing shall be easily accessable.

20. Cost of ownership:

The net cost of ownership of the vehicle shall be minimized for ten years and 105,000 miles of operation. The net cost of ownership includes initial purchase price (less scrap value), other fixed costs, operating and maintenance costs. A target goal should be to not exceed 110 percent of the average net cost of ownership of the present standard size automobile with spark-ignition engine as determied by the U.S. Department of Commerce 1969-70 statistics on such ownership.