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Final Report
Hybrid Heat Engine / Electric Systems Study
Volume I: Sections 1 through 13

71 JUN ∅1

Prepared for DIVISION OF ADVANCED AUTOMOTIVE
POWER SYSTEMS DEVELOPMENT
U. S. ENVIRONMENTAL PROTECTION AGENCY
Ann Arbor, Michigan

Contract No. F04701-70-C-0059

Office of Corporate Planning
THE AEROSPACE CORPORATION
El Segundo, California

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FOREWORD

Basic to analyzing the performance of the hybrid vehicle was the importance of understanding the characteristics of each major component since each would be operating in a nonstandard mode required by the hybrid arrangement. In addition, the potential for improvement had to be understood to predict the performance of advanced designs. This report, therefore, contains two types of information: (a) hybrid system analysis and results; and (b) major component state-of-the-art discussions, characteristics used in this study, and advanced technology assessments. Heat engine operating characteristics, mechanical parameters, and exhaust emissions are covered extensively because of both their primary importance and the difficulty involved in collecting a reliable comprehensive set of data; this should relieve future investigators making studies of nonconventional propulsion systems of the necessity of repeating the burdensome task of assembling a data bank.

It should be recognized that calculated results are based on data compiled in this study. The magnitude and trends were established on the basis of a comprehensive survey and evaluation of the best data from both the open literature and current available unpublished data sources. These data are considered suitable for use in the feasibility study conducted under this contract. However, for further detailed design a substantial refinement of the data base would be necessary.

he report is organized to give a logical build-up of information starting with study specification, analytical techniques, and component characteristics and concluding with system performance results and recommendations for development. However, selective reading of major systems performance results is possible and to assist those so interested, the following brief guide is presented:

Section 1	Summary of study results and recommentations
Sections 2, 3, 10, and 11	Presentation of study objectives, design specifications, and results
Sections 3 and 4	Description of computational techniques and performance requirements
Sections 6 through 9	Review of contemporary and projected technology of major components
Section 12	Cost estimates for high-volume production of hybrid cars
Section 13	Presentation of a technological plan for component and system development

This report is published in two volumes for convenience; however, separation of the material is made with due regard to organization. Volume I consists of Sections 1 through 13 and presents the essential study information, while Volume II consists of Appendices A through F and presents supplementary data.

The period of performance for this study was June 1970 through June 1971.

ACKNOWLEDGMENTS

The extensive diversity in technological capabilities necessary for a thorough evaluation of the hybrid electric vehicle has required the reliance for support and expertise on select members of The Aerospace Corporation technical staff as well as members of the national technical community. Recognition of this effort is expressed herewith:

<u>The Aerospace Corporation</u>

Mr. Dan Bernstein	Electrical System-Control System
Mr. Lester Forrest	Heat Engines (Internal Combustion)
Mr. Gerald Harju	Programming for Computations
Mr. Merrill Hinton	Vehicle Specifications/Conceptual Design and Sizing Studies
Dr. Toru Iura	Heat Engines (Internal Combustion) Heat Engine Exhaust Emissions Vehicle Exhaust Emissions Test Program
Mr. Dennis Kelly	Electrical System - Motor and Generator
Mr. Jack Kettler	Electrical System - Batteries Heat Engines (External Combustion)
Mr. Harry Killian	Computational Techniques Electrical System - Batteries
Mr. Robert La France	Electrical System - Motor, Generator, Control Systems
Mrs. Roberta Nichols	Vehicle Exhaust Emission Test Program
Mr. Wolfgang Roessler	Heat Engine Exhaust Emissions Vehicle Exhaust Emission Test Program
Dr. Henry Sampson	Vehicle Specifications Computational Techniques Vehicle Power Requirements
Mr. Raymond Schult	Electrical System - Motor, Generator, Control Systems

<u>University of California, Berkeley</u>

Dr. Robert Sawyer	Heat Engine Exhaust Emissions

<u>University of California, Irvine</u>

Dr. Robert M. Saunders	Electrical System - Motor Generator, Control Systems

It is to be noted that considerable data of great value to this study were kindly provided by individuals in industry, universities, and government agencies. Acknowledgment of these data sources is given in Appendix F to this report.

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Director, Pollution and Resources Programs
Office of Corporate Planning

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SUMMARY

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SECTION 1

SUMMARY

1.1 INTRODUCTION

This report contains the results of a comprehensive study aimed at determining the feasibility of using a hybrid heat engine/electric propulsion system as a means of reducing exhaust emissions from street-operated vehicles. In this hybrid concept, the source of power is a combination of heat engine and batteries (in essence, the heat engine supplies steady state power and the batteries supply transient power demands). The study examined -- for several classes of vehicles -- many types of heat engines, batteries, and other major components, as well as several design configurations. Following a review of the associated technologies, hybrid performance, exhaust emissions, and major component requirements were determined. Based on these results, recommendations are formulated to ensure the development of critical powertrain components for an early demonstration of prototype vehicles.

1.2 STUDY GROUND RULES AND PROCEDURES

In the propulsion of the hybrid heat engine/electric vehicle, the ultimate source of all energy to be expended is the heat engine. The key to success in reducing exhaust emissions is good part-load and full load efficiency of powertrain components, and the ability to restrict operational requirements for the heat engine to those of supplying road load power and (in conjunction with a generator) recharging advanced high power/high energy density batteries that supply acceleration power. With this idea in mind, the study was tailored to examine six classes of vehicles: the 4000-lb family car, 1700-lb commuter car, low- and high-speed postal/delivery van, and low- and high-speed intracity bus. For each class of vehicle, five engines were included in the powertrain: spark ignition, compression ignition, gas turbine, Rankine cycle, and Stirling cycle. Lead-acid, nickel-cadmium, and nickel-zinc batteries were studied for adequacy in supplying acceleration power to each vehicle. Also, a wide range of AC and DC motors, generators, and power conditioning and control systems were evaluated for performance, efficiency, weight, simplicity, and cost.

Throughout the study, the following ground rules prevailed:

1. Conventional automotive vehicles are to be matched in acceleration, speed, gradeability, curb weight, range, and powertrain weight.
2. The battery is not to require external recharge. Therefore, the range of the vehicle is not dependent on the installed battery capacity. This requirement was simulated in computations by requiring that the heat engine-driven generator recharge the battery to the original state-of-charge prior to the end of the emission driving cycle.
3. The battery is to discharge only when the vehicle is undergoing acceleration, not on a smooth grade or at cruise conditions.
4. The heat engine is to supply steady road load power and is not required to undergo rapid acceleration.
5. Only design concepts compatible with near term (1972-1975) prototype vehicle development are to be considered.

With the establishment of the ground rules, the study was executed in the following manner:

1. Formulate quantitative specifications based on current conventional vehicle performance and design data. These values were coordinated with the Air Pollution Control Office (APCO), Environmental Protection Agency (EPA). *
2. Review contemporary and projected technology for powertrain components and determine performance, design, and cost characteristics.
3. Evaluate conceptual designs and select a series and a parallel powertrain configuration for further analysis. The series

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* Reference is made throughout this report to the DHEW (Department of Health, Education and Welfare) Driving Cycle. The sponsoring research and development office was formerly the Air Pollution Control Office and a part of the DHEW.

configuration is characterized by the principle that all power is transmitted to the rear wheels by an electric drive motor. The parallel configuration is characterized by the principle that the heat engine is mechanically linked to the drive wheels to supply a portion of the power required, while the electrical system supplies the remainder.

4. Calculate component and vehicle power and energy requirements for acceleration and steady road load.

5. Determine battery power density and energy density requirements based on realistic component weights and powertrain weight allocations.

6. Calculate vehicle fuel consumption and exhaust emissions, based on the energy expended by the heat engine for the vehicle operating over the emissions driving cycle. For the family and commuter car, the 1972 DHEW emission driving cycle was used.

7. Determine the trade-off between vehicle exhaust emissions and such factors as engine and battery type, battery recharge efficiency, electric motor efficiency, regenerative braking efficiency, vehicle weight, and parallel and series powertrain configurations.

8. Recommend viable configurations for further study and propose a program designed to ensure component development for early demonstration of a hybrid heat engine/electric vehicle; in this regard, estimate both development and high rate production costs.

1.3 SUMMARY OF RESULTS

So many different types of vehicle/configuration/heat engine combinations were studied that it is difficult to highlight every result shown in the body of the report; therefore, only the most important are enumerated in the following paragraphs.

It should be recognized that the calculated vehicle exhaust emission results are based on measured engine exhaust emission data compiled during the course of this study. Engine exhaust emission magnitudes and trends were established on the basis of a comprehensive survey and evaluation of the best data from both the open literature and current available unpublished engine data sources. However, it was found that very little emission data were available for the hybrid type of operation and especially for part-load engine operating conditions and for the cold start requirement consistent with the 1972 Federal Test Procedure. The resulting data are considered suitable for use in an initial feasibility study as conducted under this contract. However, in further detailed design studies, a substantial increase in the data base will be necessary for powertrain optimization. The current study data base is fully discussed in Appendix B.

In addition to reflecting the engine emissions data base, the study results also reflect the use of selected battery models. The charge-discharge characteristics for lead-acid, nickel-cadmium, and nickel-zinc batteries were based on available data but modified on the basis of projections for future near-term capability. These battery models are discussed in Section 7.3 of the report.

1.3.1 Family Car and Commuter Car

The following observations can be made about these classes of vehicles:

1. For the available powertrain weight and volume and vehicle performance specified for this study, only the spark ignition internal combustion engine (both reciprocating and rotary) and the gas turbine engines can be practically packaged into the hybrid heat engine/electric vehicle. These engines impose realistically achievable goals on the battery specifications for power and energy density.
2. All hybrids examined showed marked calculated emission reductions over current conventional vehicles. This is illustrated by the results shown in Figure 1-1. In this figure, measured cold start emission data available for a 1970 conventional spark ignition engine automobile is compared with calculated hot start emission levels for several development stages of a spark ignition engine in a hybrid powertrain automobile. In the first emissions comparison, a small conventional engine is used in the hybrid vehicle; the second comparison is for the same engine but operating over the restricted air/fuel ratio range noted and with exhaust recirculation; the third comparison is for an advanced technology engine operating at very high air/fuel ratio with exhaust gas recirculation and incorporating catalytic converters.

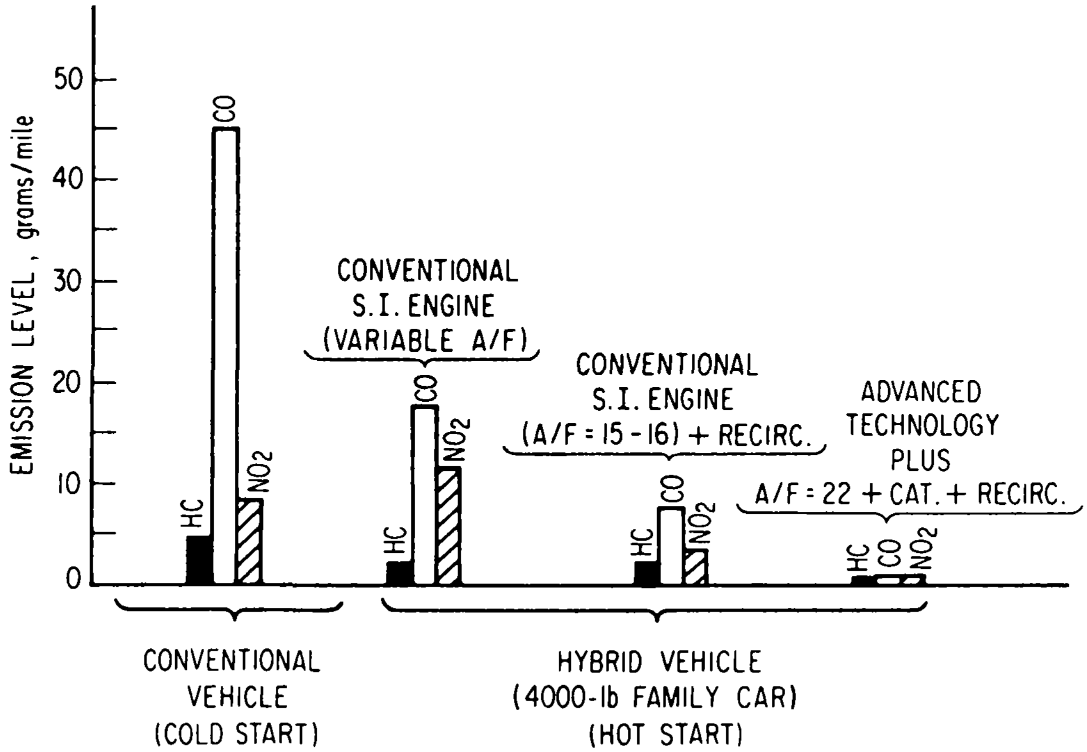

Figure 1-1. Vehicle Emission Comparison, Conventional Operation Versus Hybrid Operation (Spark-Ignition Engine, DHEW Driving Cycle)

3. Based on analysis, if currently available engine technology is used, no version of the family car could meet 1975/76 emission standards. No catalytic converters or thermal reactors were added to the powertrain for this case.

4. Calculations based on hot start with advanced engine technology indicates that all versions could meet 1975/76 standards except for the NO_2 excess for the spark ignition family car version (discussed in item 6) and the NO_2 excess for the diesel. Potential diesel engine improvements that might reduce the NO_2 emission level are discussed in Appendix B.

5. Commuter car emissions are less than one-half of those for the family car and with advanced technology easily meet the 1975/76 standards as shown in Figure 1-2. (The commuter car weighs only 1700 lb and has reduced acceleration and maximum cruise speed capabilities.)

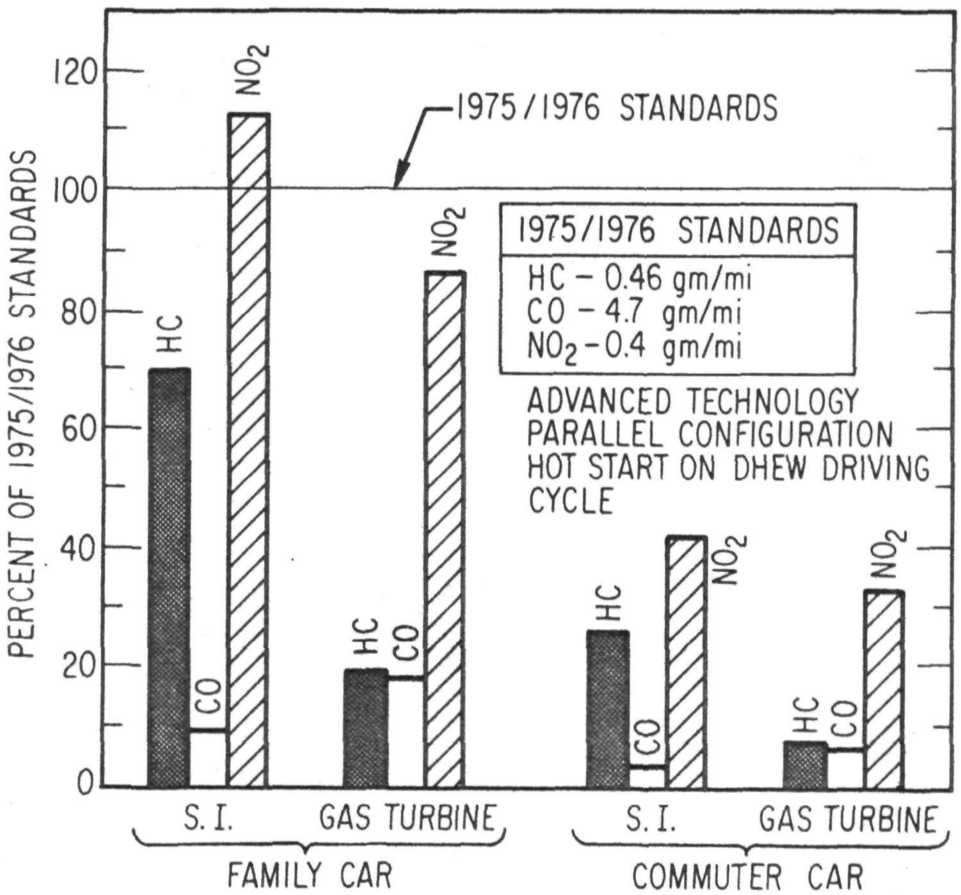

Figure 1-2. Comparative Emission Levels of the Family and Commuter Cars

6. Calculated hot start emission results for family and commuter cars using spark ignition and gas turbine engines are summarized in Figure 1-2 and compared to the 1975/76 standards. The values shown are for vehicles incorporating advanced technology components and using the parallel powertrain configuration. All values meet the numerical values of the 1975/76 standards (cold start), except for NO_2 in the spark ignition family car, and even this value is very close. This standard could be met if vehicle specifications were revised to permit a slight reduction in vehicle performance and approximately a 10 percent reduction in family car weight specifications.

7. Emissions are sensitive to: (a) heat engine class and assumed engine emission part-load characteristics; (b) driving cycle characteristics selected for evaluation; (c) the engine operating mode used over the cycle; (d) the battery discharge and charge characteristics assumed for the analysis; and (e) electric drive motor efficiency and part load characteristics.

8. Only spark ignition and gas turbine engine versions warrant intensive near term effort when availability, weight, emissions, and cost are considered.

9. Emissions are approximately 10 and 15 percent lower for the parallel powertrain configuration as compared to the series configuration in the family and commuter cars, respectively. However, the parallel powertrain is more complex. Descriptions of the powertrains analyzed can be found in Section 10. 1.

10. As noted earlier, study results are based primarily on hot start data. Incorporation of cold start effects, based on the limited amount of cold start data available, would still allow the advanced technology engine (very lean with exhaust treatment) versions of the hybrid vehicle to meet 1975 HC and CO standards. The NO_2 emission values are reduced when cold start effects are incorporated. Cold start effects are discussed in Section 9.

11. An improved lead-acid battery is needed which provides increased power density capabilities under shallow discharge operation to be used in near term hybrid applications. The near term application will not quite meet vehicle specifications for vehicle performance due to an exceeding of the powertrain weight allocation or due to insufficient battery lifetime. In order to meet all specifications, the nickel-zinc battery looks promising for the post-1975 period. Production costs for both types of batteries must be carefully considered in selection of a suitable battery design.

12. Based on the powertrain and battery models assumed and the two driving cycles used in analysis of the family car (Section 3. 3), lead-acid battery development goals were generated. The analysis results in the goal of a 38 amp-hr battery which operates at less than 4 percent depth-of-discharge. Normal vehicle operation over the DHEW Driving Cycle requires up to 260 peak amperes for acceleration with an average discharge current of about 50 amperes and a maximum energy drain of 0. 3 kw-hr (which is replenished by the generator before the end of the cycle). During occasional maximum vehicle acceleration to 80 mph, about 460 peak amperes and 0. 5 kw-hr are withdrawn from the battery. For a design life of 5000 hr of operation or

about 100,000 vehicle miles of city driving, between 900,000 and 1,000,000 charge/discharge cycles occur (Section 7.6).

13. In Figure 1-3, battery power density and energy density capabilities are compared with installed battery requirements for a spark ignition, series powertrain version of a family car. The installed requirements for energy density are based on the battery charge/discharge characteristics assumed for this study and may vary somewhat depending on actual test data from a particular advanced battery design. The intersection of the battery capability and vehicle-required installed densities gives the power and energy density compatible with vehicle weight (and battery weight) allocation. For the lead-acid case shown, the maximum power density requirement ranges from 118 to 150 watt/lb and the installed energy density ranges from 11 to 14 watt-hr/lb. The vehicle weight ranges from 4200 to 4400 lb, which represents 600 to 800 lb of batteries; this vehicle would have reduced road performance. With the nickel-zinc battery, a 4000-lb car could be built which meets the performance specifications of this study. Nickel-zinc power density and energy density values would be approximately 230 and 20, respectively.

14. Battery charge acceptance characteristics play an extremely important role in determining resultant vehicle exhaust emissions (Section 7).

15. Regenerative braking has essentially no effect on emissions for the hybrid heat engine/electric vehicle due to battery charge acceptance limitations that preclude the ability to store the braking energy. Hence, the expected advantage in reduced generator output for recharging batteries (and therefore reduced engine power and emissions) did not materialize.

    Charge acceptance improvement goals should be at least 40 amperes at over 95 percent state-of-charge without regenerative braking and as high as 400 amperes at over 95 percent state-of-charge with regenerative braking to minimize emissions.

16. Battery lifetime and charge acceptance are important areas for battery improvements.

17. Vehicle weight increases of several hundred pounds to accommodate additional battery or engine weight have a minor effect on exhaust emissions, but the heavier vehicles would have reduced road performance.

18. Realistically varying the battery recharge efficiency (to account for resistive losses and incomplete chemical reactions) has little effect on emissions.

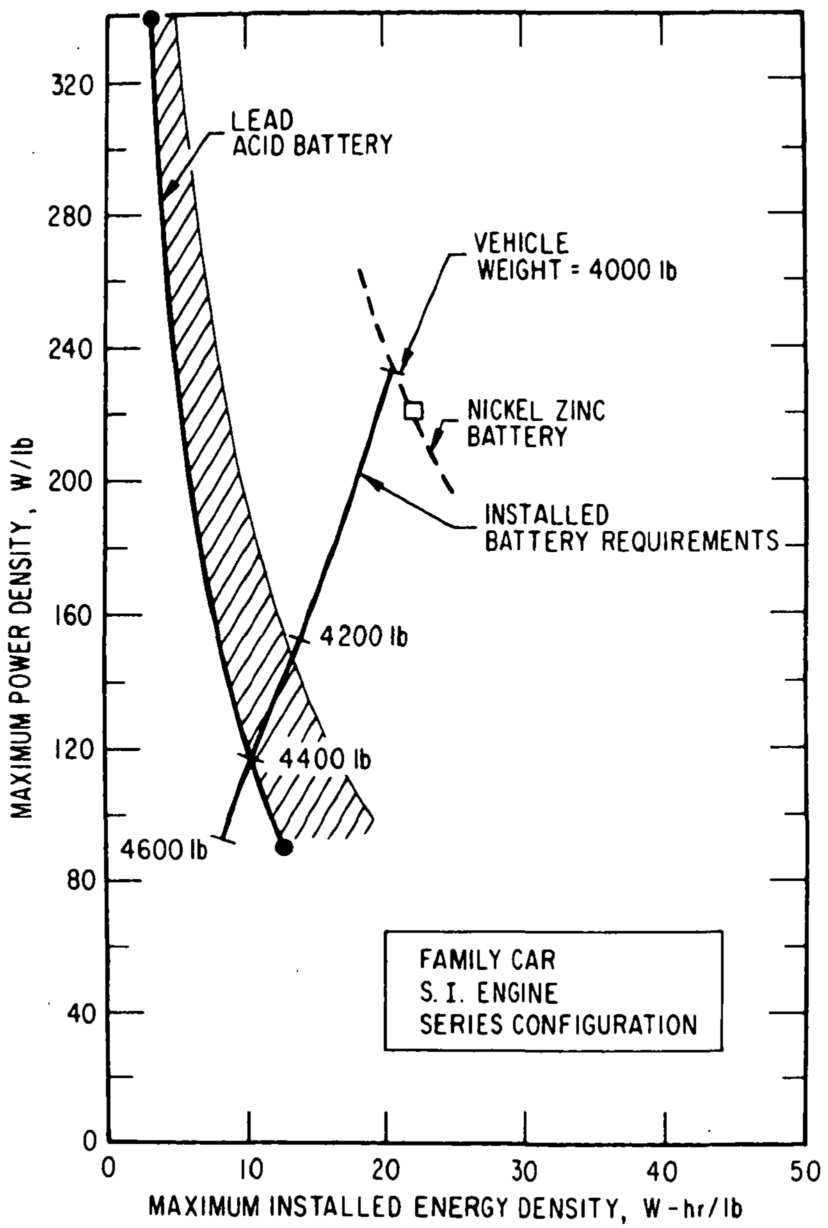

Figure 1-3. Installed Battery Requirements and Projected Battery Capabilities

19. Fuel consumption values for the spark ignition engine are summarized in the following table for all vehicles operating over their emission driving cycles (the 1972 DHEW Driving Cycle for the commuter car and the family car). The levels shown for the family and commuter cars are competitive with equivalent 1970 conventional vehicles.

Vehicle	Series Configuration (mi/gal)	Parallel Configuration (mi/gal)
Commuter Car	26	30.5
Family Car	11	12.5
Low-speed Van	3.75	----
High-speed Van	4	5
Low-speed Bus	1.25	----
High-speed Bus	1.5	2

These results were developed using specific fuel consumption characteristics based on the minimum SFC/rated horsepower correlation presented in Section 8. The data here are representative of current carbureted spark ignition engines operating at air/fuel ratios of from 14 to 16. No adjustment in SFC was made for the lean air/fuel ratio regimes adopted for hybrid operation because there is every reason to expect that appropriate modifications in the design of advanced engine systems (viz. stratified charge) will permit operation at high air/fuel ratios without serious degradation in fuel consumption. If no improvement were made, the miles per gallon at the very lean air/fuel ratios would be approximately 20 percent lower than those shown.

20. Estimates of consumer costs for the major subsystems of an advanced hybrid vehicle in large volume production were prepared by judging system complexity and performance requirements using current hardware cost data wherever available. The powertrain and vehicle component cost estimates were then used to construct a total-vehicle-cost comparison between hybrid system designs for the family car and current (1970) conventional family cars. As shown in the following table, the hybrid costs range from 1.4 to 2.25 times higher than conventional cars. However, it is expected that the conventional car meeting the 1975 emission standards will be more expensive than today's version. It should also be noted that the hybrid using the Diesel, Rankine, and Stirling engines would not meet the powertrain weight allocations or the performance specifications.

The tabulation results should be approached with caution, giving due regard to the preciseness of the assumptions made in the cost analysis. The hazard of assigning significance to the relative magnitudes of the cost ratio is apparent when it is recognized that to arrive at production costs it has been necessary to estimate figures for a number of critical components which at present may be barely classified as being in a conceptual design phase. The basis of these estimates are presented in Section 12.

Vehicle	Relative Costs
Current Conventional Car	1
Hybrid Car	
Spark Ignition	1.4 - 1.6
Diesel	1.5 - 1.7
Gas Turbine	1.6
Rankine	2+
Stirling	2.25+

1.3.2 Busses and Vans

Extensive investigation was also conducted on busses and vans in this study. This included analysis of component requirements, vehicle performance and exhaust emission levels. The information generated on busses and vans can be found throughout this report.

The following limited observations can be made about these classes of vehicles:

1. Relative evaluations were not possible since emission standards, vehicle emissions test data, and realistic driving cycle data were not available.
2. Emission data to be used in future hybrid evaluations were generated over a representative driving cycle.
3. For the bus, battery power density and energy density requirements are such that batteries could be readily made with current technology.

1.4 SUMMARY OF RECOMMENDATIONS

The intent of the recommended programs presented in this report is to provide the EPA with a planning document for ensuring the early availability of a low emission, viable alternative to the conventional automotive passenger car. In this regard, a development effort has been formulated in three phases. In brief, the first phase should be aimed at a finer definition of important hybrid parameters through both expanded analysis and data collection. A study should be performed to define in greater detail the hybrid vehicle production and operating costs since costs are an important parameter in determining if the hybrid is a viable competitor to the conventionally powered automobile. In addition to the cost analysis, a performance analysis should be performed to a level of depth greater than was performed in this feasibility study. Acquisition of component test data is needed to support this analysis. A very important area for expanded data collection is in the engine emission area. Here, data on engines operating in the hybrid mode are needed to strengthen the data base used for analysis. Comparative analysis between cars using hybrid heat engine-electric powertrains and those using advanced engines should be made to determine the advantages or disadvantages of the hybrid concept as a means of reducing auto pollution. Recommendations for additional work effort in Phases II and III are of course highly dependent on the results of studies conducted in Phase I.

The second phase should consist of an intensive effort to develop critical powertrain components destined for a prototype vehicle. This would include advanced technology work on engines, batteries, motors/generators and control systems designed to operate in the hybrid mode.

The third phase encompasses the hardware definition and development necessary for an early test bed vehicle as well as for a later prototype vehicle. The details of each phase of the recommended work effort are summarized in the subsequent discussion.

Figure 1-4 shows a schedule of activity for the three phases of recommended hybrid heat engine/electric system efforts. More information on these recommendations can be found in Section 13 and also in Sections 6 through 9 for individual components

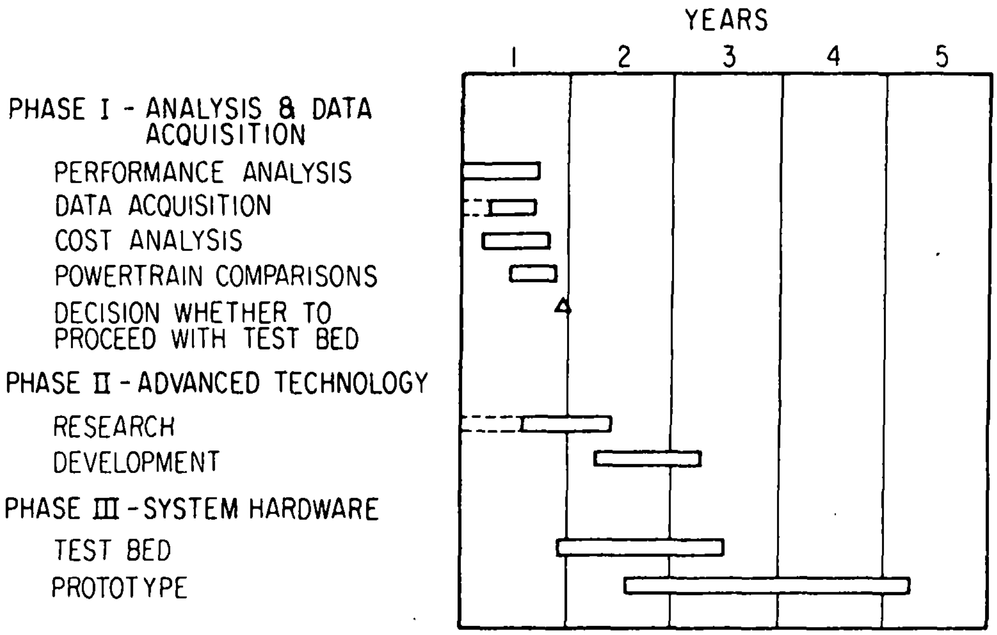

Figure 1-4. Hybrid Electric Recommended Development Schedule

1.4.1 Phase I - Detailed Hybrid System Analysis and Expanded Data Base

A logical progression from the current feasibility study would be a study directed at an in-depth analysis of the hybrid vehicle powertrain in a passenger car application. Thus, in a study narrowed in scope, the more intricate details of component operation and installation in the vehicle can be examined. The analysis is fundamental to establishing a firmer basis for objective evaluation of the hybrid electric vehicle in terms of exhaust emissions and costs when compared to present and projected versions of the engine-driven passenger car.

A major effort in the study program should be the establishment of an expanded data base for the powertrain components. This could be accomplished in two ways: (1) through planning and conducting of tests on specific component hardware to define performance maps over the entire operating range, and (2) through consultation with component manufacturers and reliance on their existing and projected data. These discussions with manufacturers should also provide a means of assessing the cost factors associated with variations in component operation.

Three major subsystems appear to need markedly increased scrutiny before a major funding effort for hybrid vehicle hardware can be initiated. These are: (1) heat engines (advanced internal combustion engines and gas turbines), (2) motor/generator control systems, and (3) batteries. The variation in heat engine emissions at part-load conditions can be very critical in ultimately determining vehicle exhaust emissions. Hence, these data are needed for the following engines operating in the hybrid mode:

1. Advanced internal combustion engines operating in the lean regime
    a. Spark ignition engines

        modified conventional engine
        stratified charge engine
        pre-chamber engine
        modified rotary (Wankel) engine

    b. Compression ignition engine (cursory examination)

        modified diesel engine (low NO_x, lightweight)

2. Gas Turbine

        single and dual shaft
        recuperated and non-recuperated

The complete operating maps for these engines should be compared with the operating maps of the electrical components in order to define the interface relationships of power and rpm that are crucial for maintaining low emissions and high overall efficiency resulting in low fuel consumption.

Through discussions with hardware manufacturers and the further clarification of electrical component operation in the hybrid car that will be accomplished in the Phase I effort, the electrical and electronic elements of the subsystems in the overall vehicle control system can be defined. This step is necessary in order to confidently predict the production costs associated with the entire electrical system. The control system circuit design should also be examined from the viewpoint of reliability and maintainability as well as first costs, and the complexity should be evaluated in terms of heat engine operating modes and the degree of manual control that could be realized.

As part of the Phase I effort to improve the data base, performance of the latest lead-acid batteries should be documented. Test data should include charge/discharge characteristics, temperature effects, and in particular cycle lifetime at shallow discharge. These data should be supplemented with test results for high power density cells that are under laboratory development. If control system operation induces transient currents at the battery terminals, the resultant effects on battery lifetime should be ascertained.

To provide an expanded critique of the hybrid electric system one further evaluation merits inclusion in Phase I studies. This relates to comparing the advanced version of the hybrid electric passenger car with advanced versions of engine-driven passenger cars. Because of near-term potential for use in cars, only the spark ignition and gas turbine engines are recommended for powerplants to be included in each vehicle's powertrain. For equivalent performance in terms of acceleration, cruise speed, and gradeability, the respective systems should be compared on the basis of production cost, exhaust emissions, and fuel consumption.

Finally, in addition to establishing a solid basis for estimating comparative hybrid passenger vehicle emission levels and production and operating costs, the proposed work effort should also provide a definitive package of information that is required prior to implementation of hardware assembly for a test bed vehicle and prior to implementation of fully funded development programs for a prototype vehicle. This information package should consider such items as:

1. Recommended powertrain design and vehicle weight and powertrain weight allocations
2. Performance specifications for each major component in the powertrain for the test bed and prototype vehicles based on vehicle specifications to be defined for acceleration, cruise speed, and gradeability
3. Rationale for powertrain design and component selection including trade-offs between cost, exhaust emissions, fuel consumption, and reliability
4. Vehicle performance capabilities including the effect of various driving cycles and cold-start on exhaust emissions

1.4.2 Phase II - Component Advanced Technology

A research and development program is recommended to provide powertrain components with performance markedly improved over contemporary hardware. The effort should lie predominantly in the areas of heat engine emissions and battery lifetimes. Initially, the program emphasis should be on research with limited funding until the Phase I study results in the form of comparative vehicle performance and cost as well as component specifications are available for review. Should these Phase I results still favor the development of a hybrid electric automobile, then the Phase II effort should be expanded rapidly with increased funding and eventual initiation of the hardware development portion of the program. The required work effort is summarized as follows:

1. Internal Combustion Engine

   Design for low specific mass emissions at part-load engine operation. Lean air/fuel ratio engines should be evaluated to select the best approach towards achieving low emission goals consistent with fuel economy. Approaches to be evaluated should include the stratified charge engine, pre-chamber engine, and engines with optimized induction system design. The rotary combustion (Wankel) engine, because of its low weight and volume and its potential for operating in the lean air/fuel ratio regime, should also be investigated. Diesel engine technology should be investigated to assess its potential for reducing NO_x emissions and engine weight. A two-year engine research and development program should be conducted with efforts also directed towards incorporating efficient catalytic converters, thermal reactors, and exhaust gas recirculation.

2. Advanced Gas Turbine Engine

   Design a burner to minimize the NO_2 emissions of the gas turbine. Studies should be conducted to select an optimized gas turbine and to plan its development to meet the requirements of the prototype vehicle. The gas turbine should be developed with the hybrid vehicle in mind and have good part-load emission characteristics and provide optimum matching of the heat engine with the electrical drive system.

3. Batteries

   The battery research and development program should consist of parallel laboratory studies of a lead-acid battery and a nickel-zinc battery optimized to the hybrid vehicle requirements in terms of power density, energy density, lifetime, and charge acceptance. It is anticipated that nickel-zinc batteries will demonstrate superior performance characteristics than lead-acid but will be more expensive. It is also anticipated that selection of an optimum battery for the prototype vehicle will be made at the end of two years.

Design concepts generated in this Phase II program should eventually be introduced into the hybrid vehicle test bed program for evaluation, and field test results should be used to tailor the later development work effort. The test bed program is discussed next in Phase III of the overall development effort.

1.4.3 Phase III - Test Bed and Prototype Vehicle Development

The following recommendations are based on results of the completed feasibility study on hybrid electric vehicles, and should be considered solely as generalized planning information at this time. If results from the Phase I program are favorable for continued development of the hybrid electric automobile, then the available detailed design information from the expanded analysis and data base can be used to refine the plans formulated in the subsequent discussion. Detailed plans for the prototype vehicle should also be dependent on the success in improving component performance demonstrated in the Phase II research effort.

1.4.3.1 Recommended System Development

A 2-1/2 yr program is recommended for development of two mobile test beds for the hybrid electric vehicle. The intent of developing these instrumented test vehicles is to permit early evaluation of system integration in the automotive environment presented by actual urban driving situations not readily simulated in the laboratory.

The test vehicle is expected to demonstrate marked improvements in exhaust emissions, but will likely not meet the 1975 emission standards. That goal is expected to be fulfilled by a prototype hybrid electric vehicle planned for completion in the 1974-1975 time period -- a vehicle which will largely benefit from the experience and component development accrued within the test bed vehicle and advanced component technology programs. It is expected that specifications can be released for component development bids nine months after Phase I initiation, and completely assembled vehicles will be available for a road test program within 21 months after Phase III initiation.

1.4.3.2 Recommended Hybrid Vehicle System Design

Only two heat engines offer the combination of near term availability with low emissions and also provide acceptable vehicle performance without requiring unreasonable battery power/energy density goals. These are the spark ignition engine (with exhaust catalytic converters and/or thermal reactors) and the gas turbine. The rpm range for a spark ignition engine is compatible with transmission/wheel rpm and this engine should be considered for use in the parallel mode configuration. The gas turbine, however, is more suitable for the series configuration because it can operate at the normally high rpm without requiring a gear reduction system. Hence, for the test bed vehicle development program, two system designs (incorporating the two configurations previously outlined) are recommended at this time in order to most effectively utilize each of these heat engines. It is expected that both configurations will have received sufficient evaluation in the test bed program to permit the choice to be narrowed to just one configuration for the prototype vehicle program.

Both configurations should use DC traction motor(s) for acceleration because of past experience with this equipment in vehicular applications and because the torque characteristics are well matched to vehicle needs over a wide speed range.

An SCR-augmented control system designed for varying motor voltage and use of separately excited field power is recommended. This system offers considerable flexibility in design which is essential to solution of design problems that may arise once all powertrain elements are integrated on the test bed vehicle.

Alternators are generally recommended for providing battery recharge power, but, because of the rpm range of the spark ignition engine and the restricted electric generator power output range required in the single motor parallel configuration, a DC generator may prove to be acceptable.

Lead-acid batteries are suggested for both configurations since they have the greatest experience factor, are not costly, and appear to have the best near-term potential for marked increases in performance. Nickel-zinc batteries, because of their current underdevelopment but future potential for even greater increases in performance, might eventually replace lead-acid batteries.

1.4.3.3 Recommended Component Development

A well-planned and executed component development program is essential for ensuring the vehicle performance intended for the hybri electric prototype vehicle. Because of the inf'uence on vehicle performance, all components and subsystems are to be designed for low weight and volume with due regard for effects on part-load to full load efficiency. They are also to be designed to operate acceptably under the environmental conditions expected during final evaluation in the test bed vehicle and prior to introduction into the prototype vehicle.

The following brief comments serve to highlight those essential design goals that are peculiar to the hybrid electric vehicle.

1. Motor/Generator

   Design for low cooling requirements, for non-steady operation, and for an optimized balance between weight, part-load efficiency and efficiency achievable at full load.

2. Control System

   Design for simplicity, reliability, and low audible noise and vibration.

3. Batteries

   Design for lead-acid batteries with high power density, long life, high charge acceptance, minimum (or zero) maintenance, and low production costs. For charge/discharge characteristics similar to those assumed for this study, high energy density is also a design requirement.

4. Heat Engine

   Design for low emissions at full load and part-load consistent with good fuel economy. Application of catalytic converters and/or thermal reactors, as well as exhaust gas recirculation should be considered for the internal combustion engine case.

SECTION 2

INTRODUCTION

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SECTION 2

INTRODUCTION

2.1 PURPOSE OF STUDY

In search of alternative vehicular power sources that are expected to offer substantial reductions in exhaust pollution of the atmosphere over that produced by the internal combustion engine, the Air Pollution Control Office (APCO), Environmental Protection Agency (EPA), has ranked the hybrid heat engine/electric system high on the list of candidates. However, before it could commit funds for promoting early development of this power source, APCO required detailed substantiation of its design feasibility and potential for major reductions in exhaust emissions. Hence, the present study was directed at providing an analysis of the hybrid heat engine/electric system by examination of the performance and exhaust emissions resulting from computer simulation of vehicles operating over select driving cycles. Establishment of general design goals for components and subsystems in the vehicle powertrain was also included. In addition, a technical development plan was formulated for defining the schedule and allocation of resources by APCO to accelerate the development of critical components so that viable production vehicles could be expected by 1975.

2.2 THE HYBRID VEHICLE CONCEPT

There are numerous types of hybrid powerplants which conceivably could be considered in a search for low-pollution systems, e.g., flywheel/heat engine hybrid or battery-battery hybrid. This study was limited solely to the hybrid heat engine/electric vehicle concept in which the power for vehicle propulsion is supplied by two specific sources: a heat engine and a set of batteries. The heat engine is designed to supply cruise power while the batteries are designed to supply power for acceleration. In this combination the heat engine can be reduced markedly in size compared to a conventional propulsion system and, by restricting the rpm range and permitting only slow acceleration of the engine, its design and operation can be optimized to substantially reduce exhaust emissions and enhance longevity. (Detailed discussion of engine operating modes and power distribution between engine and batteries can be found in Section 10.) Moreover, the benefits accruing from restricted operation of the engine may also carry over to ancillary equipment such as catalytic converters. Furthermore, since the heat engine can be linked to an electric generator, the batteries can be recharged in transit and the complexities associated with providing area-wide electric power sub-stations are avoided. Thus, the hybrid vehicle represents an intermediate step between current internal combustion engine-powered vehicles and the practical all-electric vehicle of 1985-1990.

2.3 ORGANIZATION OF STUDY

To fulfill the objectives of this study, the work effort was divided into four major interrelated study tasks (Fig. 2-1). The first task -- Systems Synthesis and Preliminary Sizing -- was designed to establish at an early date the powertrain configurations to be investigated and the general range of power requirements for each component and subsystem in the powertrain. The second task -- Subsystem/Component Data Acquisition and Technology Assessment -- involved the polling of acknowledged experts in industry, universities, and government on the state of the art in specific technical areas and the merits or deficiencies of proposed methods of operation for each component or subsystem. The third task -- Systems Evaluation and Comparison -- consisted of collation of empirical data, formulation of computational procedures, selection of the most promising combinations of powertrain components and subsystems, and detailed evaluation of the resulting vehicle exhaust emissions and design requirements. The fourth task -- Technology Development Program Plan -- covered the required planning and funding for research and development efforts necessary to upgrade the performance of critical components.

2.4 SCOPE OF STUDY

This study did not encompass a detail design effort. Rather, its scope was limited to an evaluation of contemporary technological capabilities as well as

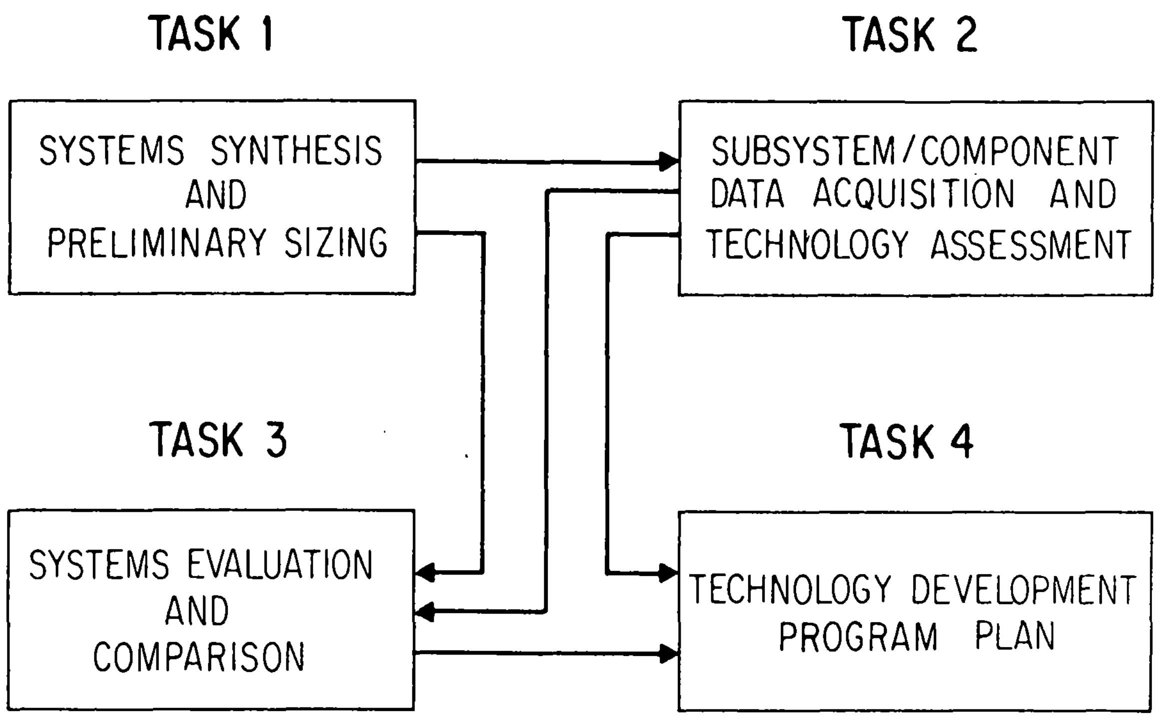

Figure 2-1. Major Study Tasks

near term advancements in technology for powertrain component development, illumination of potential problem areas, and establishment of some exemplary powertrain systems for hybrid vehicles with potential for low exhaust emissions.

Although major emphasis in the study was placed on evaluation of the hybrid concept for the full-size family car, three other classes of vehicles were considered: the two-passenger commuter car, delivery/postal van, and intra-city bus. Within each class, several different types of heat engine (ranging from spark-ignition to Stirling cycle) were considered. In addition, a large variety of batteries, motors, generators, etc., were studied for incorporation into the vehicle powertrain.

The performance of a baseline configuration for each class of vehicle was established for each type of heat engine. Since a specific detail design was not a goal in this study (rather design feasibility), the effect on vehicle exhaust emissions and battery design goals of such diverse factors as component efficiencies, regenerative braking, vehicle weight, powertrain weight, types of driving cycle, and types of battery was also covered. Hence, the relative importance of these factors in establishment of vehicle design goals can be readily assessed.

It should be noted that because of the multiplicity of vehicle classes and engines considered in this study, it was considered prudent to constrain the investigation to vehicles of fixed curb weight and powertrain weight. However, for the detailed design of a specific vehicle class, the vehicle and powertrain weights should be allowed to vary in order to diminish the severity of battery design requirements without compromising the goal of maintaining low vehicle exhaust emissions.

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VEHICLE SPECIFICATIONS AND STUDY METHODOLOGY

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VEHICLE SPECIFICATIONS AND STUDY METHODOLOGY

3.1 VEHICLE SPECIFICATIONS

The APCO specifications for the four vehicles to be examined for potential applicability of the heat engine/electric hybrid powerplant concept are shown in Table 3-1. Significant vehicle design point conditions most likely to affect powerplant sizing and operational capability include vehicle top speed, gradeability (in terms of percent grade, velocity on the grade, and grade length), vehicle weight, and aerodynamic drag area and drag coefficient. The only limitations imposed upon the power train were the assigned power train weights and volumes. A final requirement was that the acceleration capability of each vehicle with a hybrid powerplant installed was to be equal to that of a contemporary automotive vehicle. Therefore, as stipulated by the APCO specifications of Table 3-1, any resulting hybrid vehicle must match the acceleration, speed, and gradeability characteristics of conventional, contemporary vehicles. The rationale for this requirement is that such performance will enhance public acceptance of the hybrid vehicle and will also avoid the prospect of poor traffic safety.

In addition to the above constraints, certain criteria in the areas of (a) battery state-of-charge condition and (b) vehicle characteristics were adopted by Aerospace. Since a prime objective of a hybrid-electric powerplant is an inherent capability to recharge the batteries used with the incorporated heat engine (i.e., no external source require ` to recharge batteries for operational readiness), it was further defined that: (a) when the vehicle is operated at cruise conditions under the gradeability requirements of Table 3-1, the installed heat engine power output shall be sufficient to prevent the battery from discharging; and (b) when the vehicle is operated over a representative driving cycle for emission calculations, the hybrid-electric powerplant

Table 3-1. APCO Hybrid Vehicle Specifications

Vehicle Characteristics	Family Car	Commuter Car	Intracity Bus		Delivery/Postal Van	
			Low Speed	High Speed*	Low Speed	High Speed*
Maximum cruise velocity (mi/hr)	80	70	40	60	40	65
Cruise velocity for maximum range (mi/hr)	66.5	59.4	40	60	40	65
Velocity on grade at grade (mi/hr at percent)	40 at 12	33 at 12	6 at 20	10 at 10	8 at 20	8 at 20
Grade length (mi)	8	4	0.5	0.5	0.5	0.5
Range (mi)	200	50	200	200	60	60
Curb weight (lb)	3,500	1,400	20,000	20,000	4,500	4,500
Loaded weight (lb)	4,000	1,700	30,000	30,000	7,000	7,000
Assigned power train weight (lb)	1,500	600	6,000	6,000	1,700	1,700
Assigned power train volume (ft^3)	28	16	175	175	42	42
Aerodynamic drag area (ft^2)	25	18	80	80	42	42
Drag coefficient, C_d	0.5	0.35	0.85	0.85	0.85	0.85
Acceleration	** Equal to contemporary automotive vehicle					

*Recommended Aerospace values

**See Section 5.4.3 and Figs. 5-1 and 5-2 for values used.

shall have the batteries fully recharged at the end of the driving cycle. Consideration was also given to operation of the battery at shallow discharge with the anticipated goal of improved battery lifetime associated with shallow discharge. Under these conditions, the vehicle range is not dependent on installed battery capacity.

It was recognized that the requirement for the battery full state-of-charge at the end of the driving cycle might be extreme for such vehicles as delivery vans and buses which are garaged in facilities which could be readily modified to provide recharging capability at the end of a prescribed work cycle or day. However, personal transit vehicles (i.e., family cars, commuter cars, etc.) present a more stringent requirement in that recharging facilities are not readily available to them, at least at the present time. Therefore, it was felt more reasonable to adopt the "fully-recharged at end of driving cycle" design criterion for all vehicles as a baseline requirement; appropriate tradeoffs for the delivery van and bus might be made in subsequent studies to assess the importance of this requirement.

It was further felt appropriate to add two more vehicle classes to be examined, in addition to the four classes specified by APCO. As can be noted in Table 3-1, both the delivery/postal van and the intracity bus have very low (40 mph) top speeds and severe (20 percent) grade requirements. While these characteristics may be very adequate for many municipalities (e.g., San Francisco), they would not appear to be most appropriate for urban areas with large freeway networks on which these vehicles are required to operate (e.g., Los Angeles). Therefore, a high-speed version of the delivery van and bus was added to the basic group of vehicles listed in Table 3-1. The top speeds of the delivery van and the bus were selected as 65 mph and 60 mph, respectively. No gradeability requirement was set for these two additional vehicles; the resulting gradeability was determined from sizing for maximum velocity. Aside from top speed and gradeability, the other specifications of Table 3-1 apply to the two additional vehicles.

3.2 VEHICLE ACCESSORY POWER REQUIREMENTS

Conventional vehicles normally are provided with accessory features such as air conditioning, lights, instrumentation power, and power steering by engine-driven accessory units. When the vehicle is driven by conventional internal combustion engines, the maximum accessory power load varies as a function of engine rpm, which in turn is a function of vehicle velocity and transmission gear ratio.

As the hybrid heat engine/electric powerplant is currently conceptual in nature, the variation of heat engine rpm and power output capability versus vehicle velocity is not definitized. Therefore, the accessory power requirements for the various vehicle classes were stipulated by APCO to be of constant value over the operational speed range of each vehicle, as shown in Table 3-2. It is recognized that this approach will likely result in exhaust emission calculations being based upon a greater heat engine power output at low vehicle speeds than would be required if variable accessory power requirements were used (See Section 8). However, this leaves some margin available for electronic and electrical cooling power requirements which will rise in the low-speed range where free convection air flow rates are low.

3.3 DRIVING CYCLE SPECIFICATIONS

3.3.1 Emission Comparison Driving Cycles

A number of driving cycles have been used/proposed by various municipal, state, and federal agencies. The current test procedures utilized by the Federal Government (and California) to enforce existing automotive emission standards are conducted with the "seven-mode driving cycle." The seven-mode cycle is an abbreviated test cycle intended to simulate urban driving requirements as exemplified by the ~22-min. LA-4 traffic route pattern. For this purpose, the seven-mode test results are modified by weighting factors, as set forth in the test procedures published in the Federal Register (Ref. 3-1).

Table 3-2. Engine-Driven Accessory Power Requirements

Vehicle	Power Requirement, hp Air Conditioning With	Without
Family Car	12.6	6.7
Commuter Car	5.7	1.7
Delivery/Postal Van	----	2.3
Intracity Bus	39.3	12.3

Note: Includes cooling fan, air conditioning, lights, instrumentation power, etc. Power steering also included on Family Car and Intracity Bus.

It is currently proposed to modify the federal test procedures to require emission testing over the complete ~22-min. urban traffic route (LA-4). This new test cycle is called the DHEW urban dynamometer driving schedule and hereafter will be referred to as the DHEW urban driving cycle (Ref. 3-2). Figure 3-1 illustrates the salient features of the seven-mode and DHEW urban driving cycle in terms of vehicle speed versus time. For comparison purposes, an urban driving cycle characteristic of New York City is also shown (Ref. 3-3).

For the present study, the proposed DHEW urban driving cycle was selected as the baseline driving cycle for emission calculations and battery comparisons for the family car and the commuter car. For the family car, selected comparisons with the New York City driving cycle were also made.

Similar driving cycles for such vehicles as delivery vans and buses do not currently exist. Therefore, based upon available nominal work or duty cycle data, emission driving cycles were postulated for these vehicles. These cycles are shown in Fig. 3-2, along with the DHEW urban driving cycle for comparative purposes.

3.3.2 Design Driving Cycles

The vehicle specifications of Table 3-1 and the emissions-related driving cycles do not in themselves afford a basis for completely comparing the performance of a vehicle with a conventional powerplant against a similar vehicle with a hybrid-electric powerplant. Therefore, design driving cycles were postulated for each vehicle which contain the criteria of Table 3-1 and which afford a definitive basis for comparison. The design driving cycle for the family car is shown in Fig. 3-3, and includes the performance phases of maximum acceleration, maximum high-speed cruise, high-speed cruise for range, and the gradeability requirement.

The design driving cycles for the other vehicles are presented in tabular format in Appendix D.

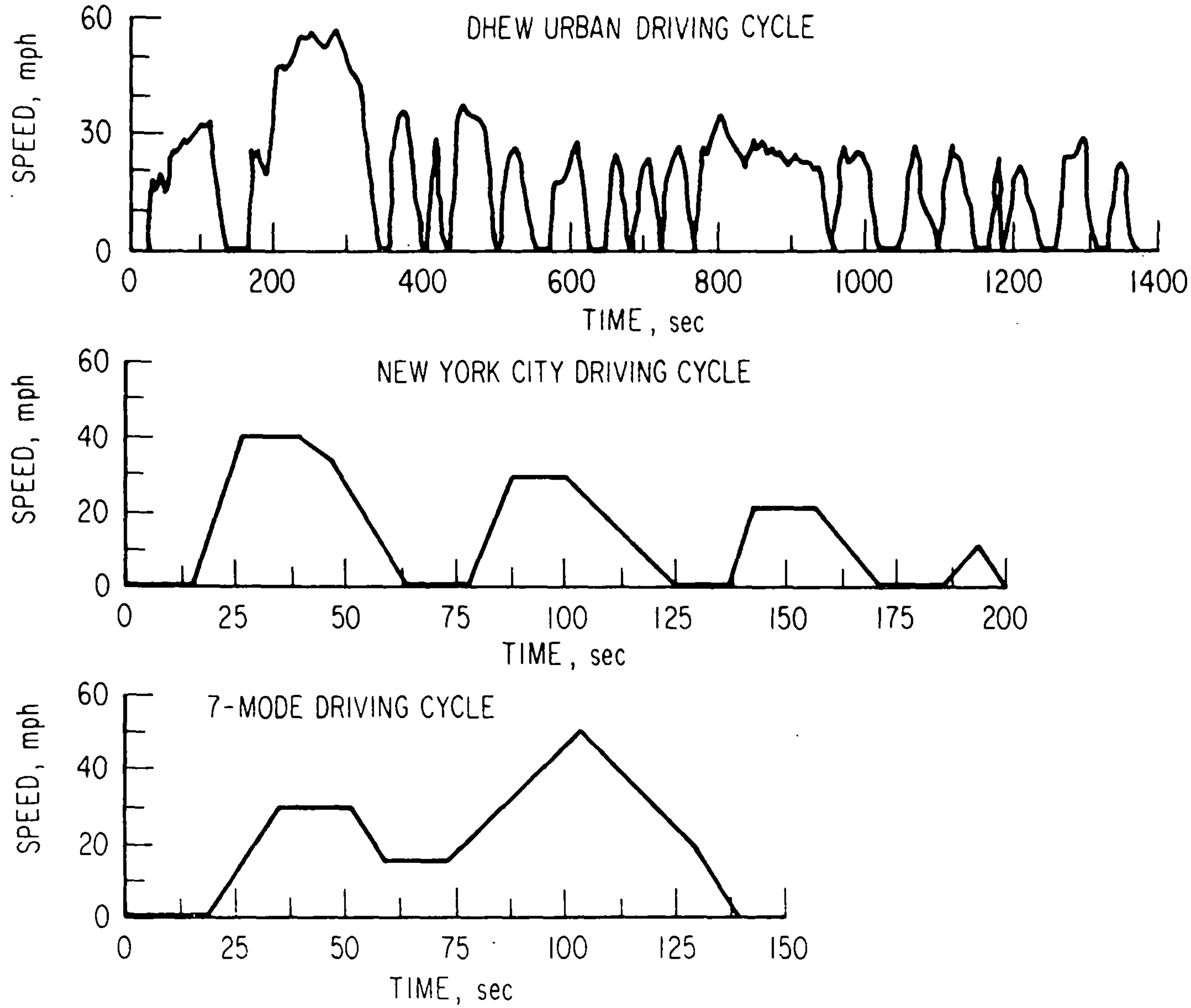

Figure 3-1. Driving Cycles for Emission Comparisons
Family and Commuter Cars

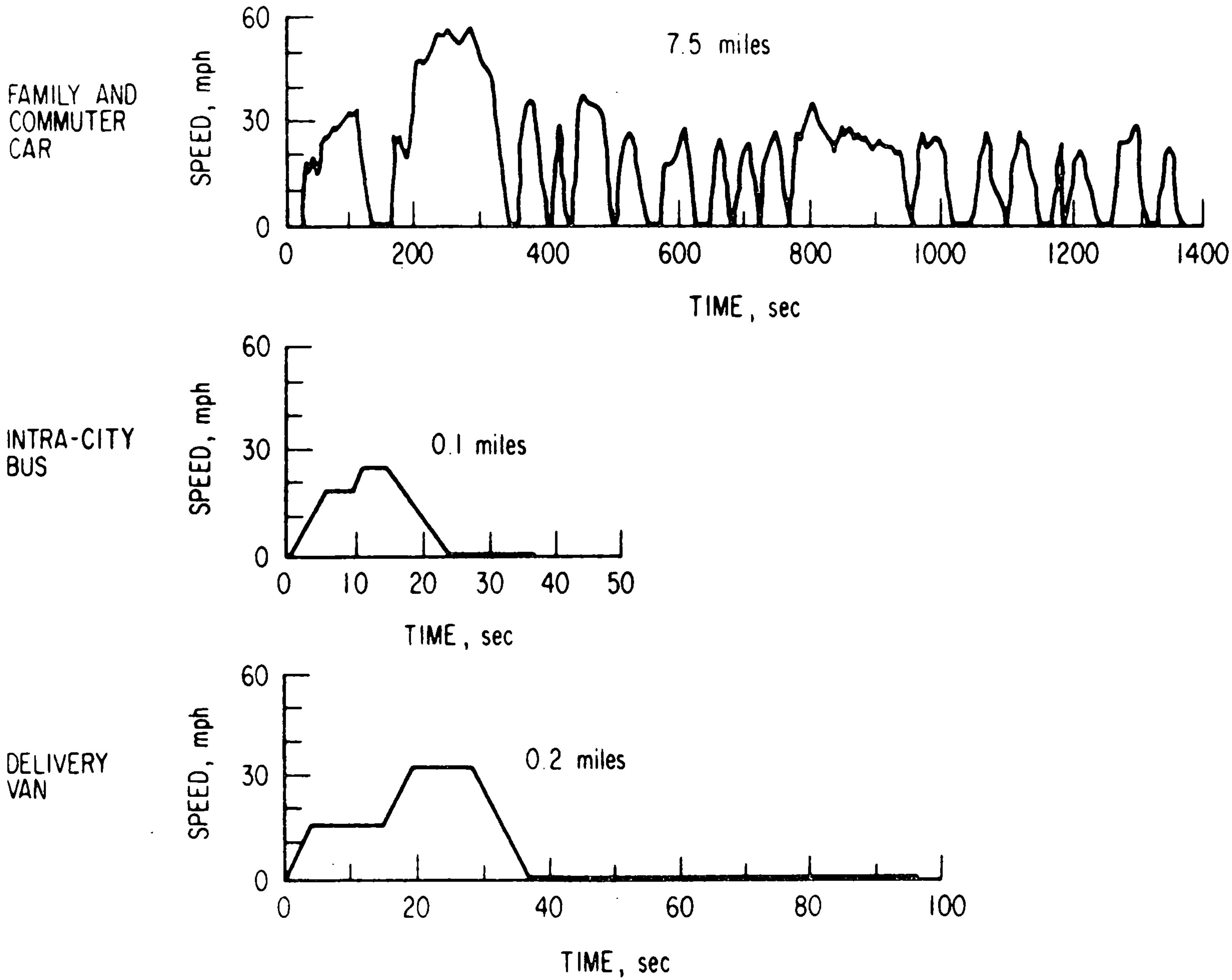

Figure 3-2. Driving Cycles for Emission Comparisons

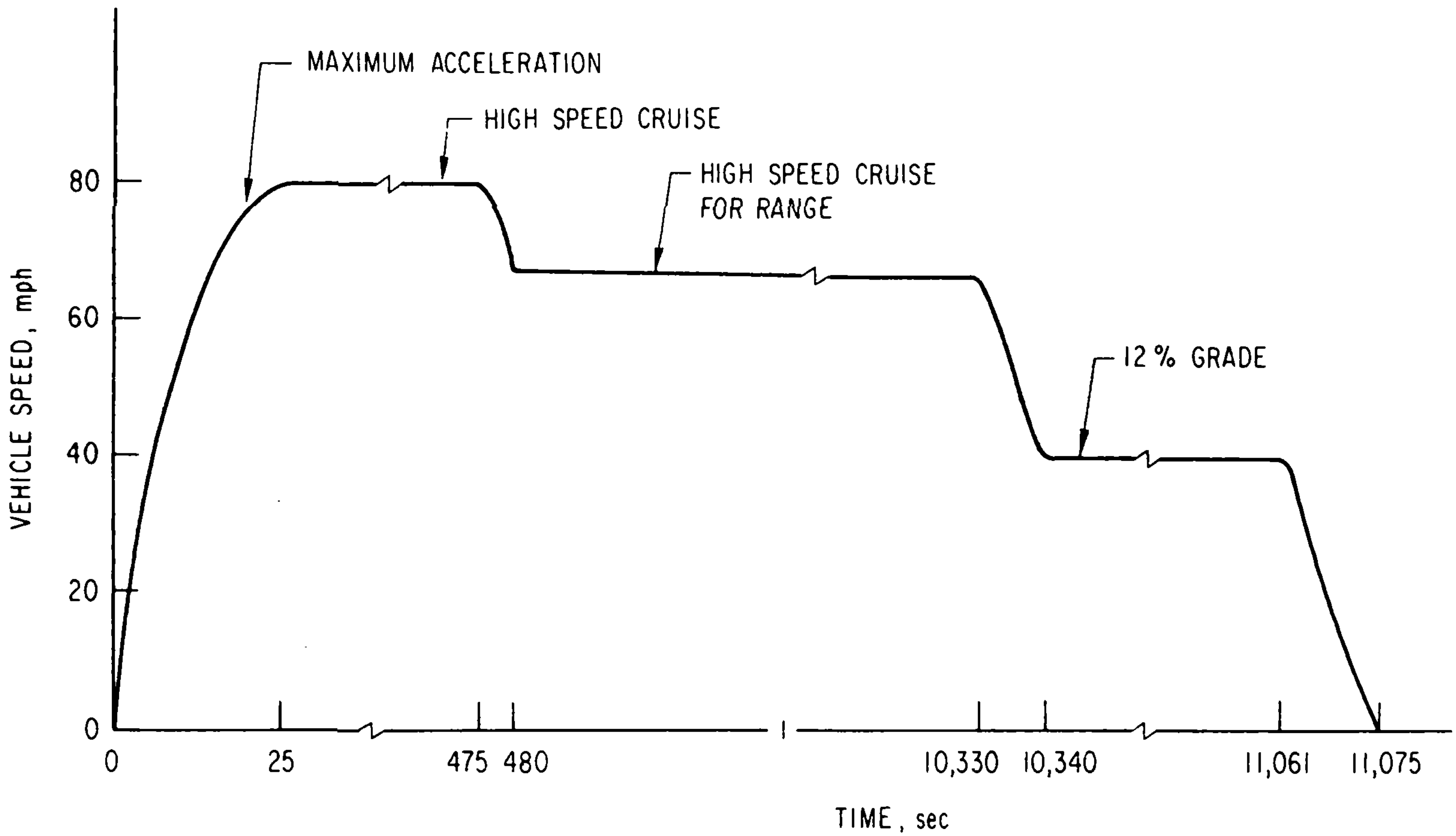

Figure 3-3. Design Driving Cycle,
4000-lb Family Car

3.4 STUDY METHODOLOGY

The methodology selected for the conduct of the study consisted of the following essential steps:

1. Preliminary system synthesis
2. Development of computational techniques
3. Subsystem technology evaluation
4. Power train conceptual design synthesis and sizing
5. Performance and tradeoff studies
6. Technology development program plan

3.4.1 Preliminary System Synthesis

A preliminary system synthesis was performed to (a) identify reasonable hybrid-electric powerplant concepts and (b) identify reasonable subsystem performance and technology requirements. In this effort, material readily available in the literature was reviewed to ascertain the type and depth of information pertaining to hybrid vehicles and powerplants. Specific materials used in this task are listed in Refs. 3-4 through 3-11.

Hybrid-electric powerplant concepts can be grouped into two broad classes as shown in Fig. 3-4. The first class, series configuration, is characterized by the principle that all power is transmitted to the rear wheels via an electric drive motor which receives electrical energy either from a generator, a battery, or both, depending upon the electric motor power demand and the generator output at the time of demand. The heat engine drives the generator mechanically; however, all other elements of the powerplant system are electrical in nature. In the series configuration, the heat engine is decoupled from the drive wheels. The fact of decoupling enables a wide variety of heat engine/generator operational modes to be envisioned as possible. Several of these operational modes and their attendant ramifications are discussed in more detail in Section 10.

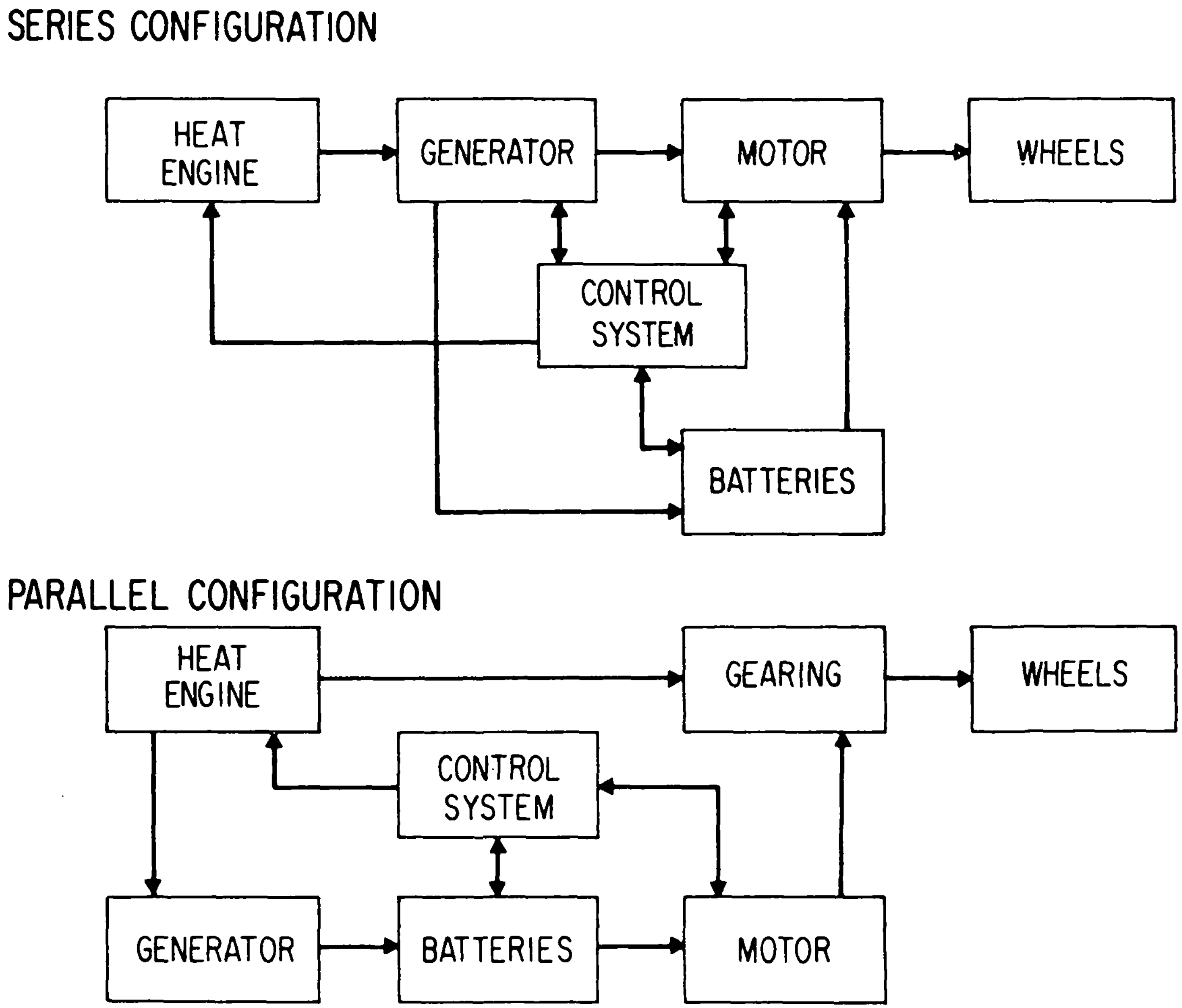

Figure 3-4. Schematic of Hybrid - Electric Powerplant Concepts

The second class, parallel configuration, is characterized by the principle that the heat engine is mechanically linked to the drive wheels to supply all or a portion of the power required there. The mechanical link can be one of several gearbox/transmission arrangements. It is a further principle of the parallel configuration that the power mechanically transmitted from the heat engine to the drive wheels be sufficient only to maintain vehicle cruise speeds, and that power required for acceleration of the vehicle be supplied by an electric drive motor which derives its energy source from a battery and/or a generator, also driven by the heat engine. There are many specific parallel arrangements which can be envisioned; some of these will be more thoroughly discussed in Sections 6 and 10.

Various subsystems/components considered for use in the hybrid-electric powerplant, whether of series or parallel configuration, are shown in Table 3-3. In the heat engine area, an investigation of conventional spark-ignition engines, diesels, gas turbines, Rankine cycles, and the Stirling cycle was conducted. In the battery area, recent assessments of capability indicated that for near-term application (circa 1975), the lead-acid, nickel-cadmium, and nickel-zinc were of prime importance; advanced batteries with better power density and energy density characteristics would enhance the capability of a hybrid-electric powerplant, but would require extensive development funding and would not be available for production by 1975. Electric drive motors of specific types (i.e., AC induction, DC shunt-wound, DC brushless) have been shown to have specific advantages in specific installations. Both DC and AC generators/alternators had been shown to be promising, depending upon the specific vehicle/powerplant configuration. In the power conditioning and control area, a wide range of types from silicon-controlled rectifiers (SCRs) to relays/switches had been shown to be reasonable/attractive, depending again upon the application and method of control selected.

Table 3-3. Vehicle Component Array

Engines

- IC Spark
- Diesel
- Gas Turbine
- Rankine Cycle
- Stirling Cycle

Batteries

- Lead-Acid
- Nickel-Cadmium
- Nickel-Zinc

Motors

- AC Induction
- DC Shunt Wound - Externally Excited
- DC Series Wound
- DC Compound Wound
- DC Brushless

Generators

- DC
- AC (Alternator)

Power Conditioning and Control

- Silicon Controlled Rectifiers
- Inverters
- Solid State Integrated Circuits
- Cycloconverter
- Relays/Switches
- Resistors/Inductors

In addition to the components outlined in Table 3-3, mechanical gearboxes, differential drive units, and transmissions (of several varieties) are also required to complete the basic elements of the hybrid-electric powerplant.

3.4.2 Development of Computational Techniques

An essential step in the evaluation and capability assessment of the various heat engine/electric hybrid powerplant concepts is the development of computational techniques adequate to:

1. Accommodate the various heat engine/electric hybrid concepts
2. Determine the characteristics of various vehicle classes and the subsystem/component requirements for vehicle operation over
   a. Design driving cycles
   b. Emission driving cycles
3. Determine heat engine exhaust emission levels
4. Determine battery charge/discharge characteristics over various driving cycles
5. Determine distribution of useful and dissipated energy throughout the system

The specific details of the computer program developed for these purposes and its use in subsequent analyses are described in Section 4. The use of other existing computer programs to determine the various vehicle power requirements (i.e., acceleration, torque, power, etc.) over the different driving cycles is described in Section 5.

3.4.3 Subsystem Technology Evaluation

The efforts devoted to determining subsystem performance, weight, and design characteristics are presented in Sections 6 through 9, for both current state-of-the-art and future projections of technology. The data presented have been carefully constituted to provide practical, contemporary information as confirmed by an intensive in-depth survey of acknowledged experts in specific technical areas. (Sources of data may be found in Appendix F.)

3.4.4 Conceptual Design and Sizing Studies

Those conceptual design studies made to select heat engine/electric power train combinations for detailed analysis are treated in Section 10, together with the resultant subsystem/component sizing necessary to meet vehicle performance specifications. This section also develops battery weight allocations as a function of powerplant installed weight for the various vehicles and various heat engines within a vehicle class.

3.4.5 Performance and Tradeoff Studies

Section 11 presents the results of the study, in terms of vehicle exhaust emissions levels and battery design goals for the various vehicle/powerplant combinations discussed in Section 10. Also presented are the results of various tradeoff studies to assess emission/vehicle performance sensitivity and battery design goals sensitivity to:

1. Effect of regenerative braking
2. Effect of battery recharging efficiency
3. Effect of vehicle weight (other than baseline)
4. Effect of drive motor efficiency
5. Effect of type of battery
6. Effect of emission driving cycle (DHEW cycle versus New York driving cycle)
7. Series versus parallel mode of operation

3.4.6 Technology Development Program Plan

Section 1 delineates a recommended technology development program plan, based on the results given in Section 11 and the technology capability projections of Sections 6 through 9. The program plan is directed toward defining a technology development program for the most promising systems and is designed to enhance the probability of viable prototype hardware in the near future and production hardware in the 1975-1980 time period.

3.5 REFERENCES

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SECTION 4

COMPUTATIONAL TECHNIQUES

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COMPUTATIONAL TECHNIQUES

4.1 INTRODUCTION

The evaluation of the performance of the various hybrid vehicle power train concepts considered in this study is performed with the aid of a digital computer program. This program, entitled Hybrid Electric Vehicle Performance Evaluation Computer Program (HEVPEC), was developed specifically for this study. A computer simulation technique is used to determine the performance of hybrid vehicles over a specified driving cycle. The overall objective of these calculations is to identify those design approaches which give low exhaust emission, to determine the sensitivity of emission levels to changes in the operating characteristics of various power train components, and to determine the resulting battery requirements. This section contains a discussion of the analytical model, a brief description of the computer program, followed by an explanation of how the program was used to achieve the desired results.

4.2 ANALYTICAL MODEL OF HYBRID VEHICLE POWER TRAIN

An analytical model of a hybrid vehicle power train was derived and programmed for the Aerospace CDC 6600 digital computer. The basic equations for the rectilinear motion of a rigid body were combined with Newton's Second Law of Motion to establish the basic link between the velocity of the vehicle as specified by the driving cycle and the net driving force at the wheels. Detailed models of several of the major components of the power train were incorporated in the computer program to determine the response of power train elements to instantaneous power demands associated with vehicle operation on a given driving cycle. The major components of the hybrid power trains were considered to be the heat engine, a generator, a secondary battery, an electric traction motor, an electrical control package, and, in the parallel configuration, a power transmission. Models of two different power train configurations designated as "series" and "parallel" were derived. In both configurations the heat engine provides all energy expended for vehicle operation, and the secondary battery provides most of the power required for vehicle acceleration.

4.3 DESCRIPTION OF COMPUTER PROGRAM

4.3.1 Program Logic Elements

The computer program includes not only the basic mathematical expressions associated with the analytical model discussed above, but also the logic* required to regulate the power and energy flow from each component during vehicle operation over the driving cycle. A simplified version of the basic program is presented in Fig. 4-1.

There are sets of logic elements built into the program that warrant special mention. The first controls battery charge and discharge. If the power demanded by the motor exceeds generator output power level, an amount of power equal to the deficiency is directed from the battery to the electric motor. Another logic element tracks the state-of-charge and voltage of the battery and terminates the calculation procedure if the latter falls below some specified value. Power for battery charging is available if the power demand by the motor is less than the output of the generator. The maximum charging power the battery can accept is constrained by the battery charge characteristics and by a specified maximum allowable voltage level of the battery. Excess power that the battery cannot accept is assumed to be dissipated in resistive load. The cumulative amount of energy so dissipated is calculated and included in the output data.

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*Refer to Appendix A for a detailed description of the computer program logical structure.

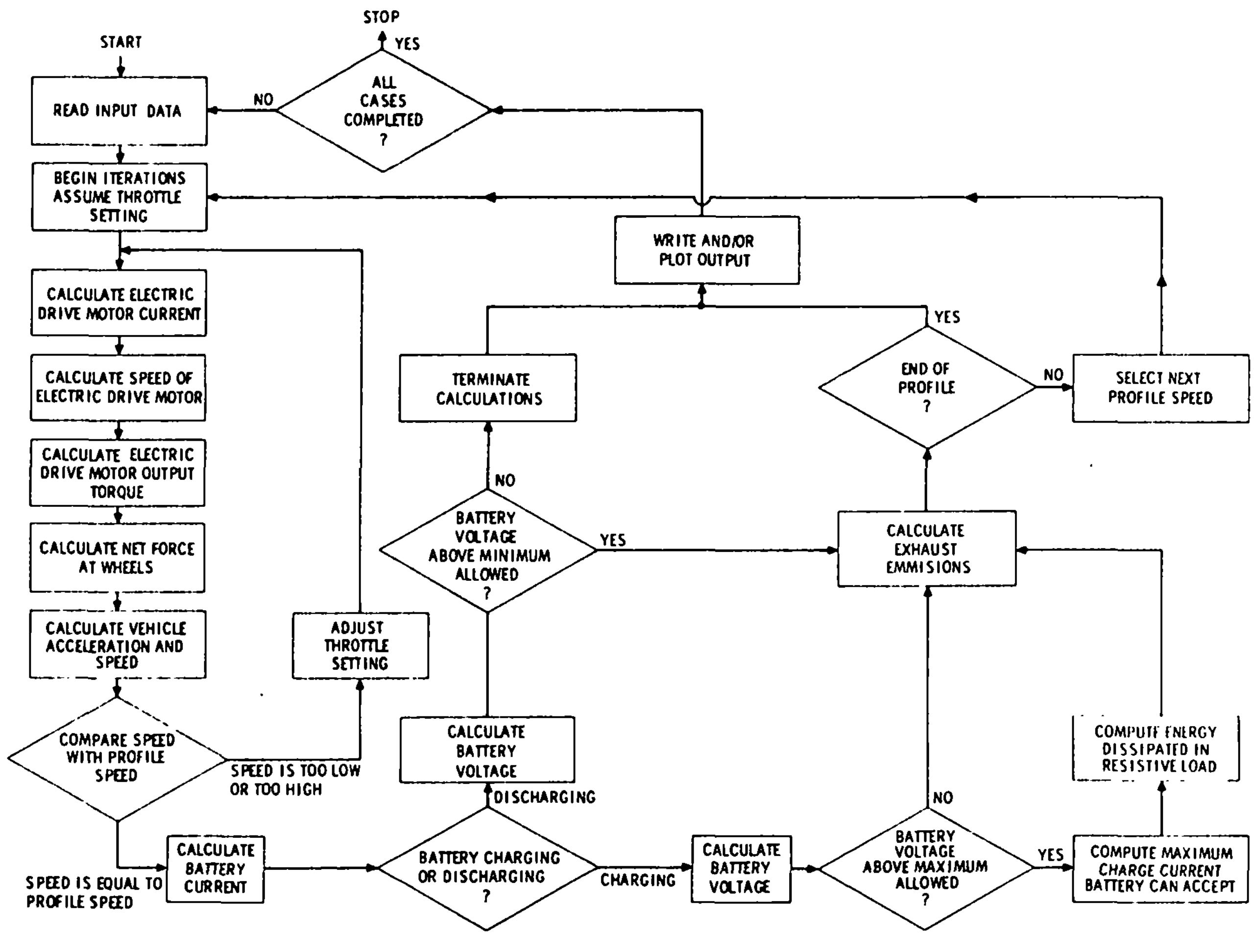

Figure 4-1. Simplified Flow Chart of HEVPEC Program

The second special set of logic elements controls the mode of operation of the heat engine.** A simulation can be run with the heat engine operating at constant power output; i.e., independent of vehicle speed or power demand at the wheels. This mode of operation is applicable only to a series-configured power train. In an alternate mode of operation applicable to both the series and parallel configurations, the heat engine power output varies with total road load horsepower down to a minimum specified value below which the engine output is constant.

4.3.2 Program Input Data Requirements

The input data required for a simulation run on the computer are presented in detail in Appendix A and include tables of battery charge and discharge data, heat engine emission characteristics, driving cycle data (velocity, time, road grade), vehicle characteristics (weight, frontal area, rolling resistance coefficients, aerodynamics drag coefficients, gear ratio, tire radius, etc.).

4.3.3 The Handling of Output Data†

Output from a typical run for each version of the program includes the following:

1. Vehicle Status
    a. Profile time
    b. Speed
    c. Acceleration
    d. Wheel horsepower
    e. Total road resistance

**Refer to Section 10 for a full discussion of heat engine operating modes.

†Sample printouts from typical simulation runs are presented in Appendix A.

2. Heat Engine-Generator Status

   a. Power output

   b. Generator Current

   c. Emissions (CO, HC, NO_2)

3. Electric Motor Status

   a. Speed

   b. Input current

   c. Output torque

4. Battery Status

   a. State of charge

   b. Discharge current

   c. Maximum discharge current available

   d. Total charge current available

   e. Maximum acceptable charge current

   f. Cell voltage

The user has the option of obtaining the above information at each time point of the driving cycle or at only the last time point. In addition to the digital output, a graph plotting routing was added to the program and allows the user to obtain the results printed out in graphical form. Samples of both the digital printout and plotted output are included in Appendix A.

4.4 APPLICATION OF PERFORMANCE EVALUATION COMPUTER PROGRAM

Two types of simulations were required to obtain a complete evaluation of a particular hybrid power train. The first type involved simulated operation of a vehicle over a design driving cycle and was performed to verify sizing of the heat engine and battery. The electric drive motor was sized using the results of the analysis presented in Section 5 of this report. The design driving cycle was synthesized from APCO specifications defining the required vehicle maximum performance. Although a different design cycle was required for each vehicle considered in this study, they were similar in organization because they each contained maximum acceleration to maximum cruise speed, cruise at constant speed, and operation on a grade at constant speed.

Having verified adequate sizing for the hybrid power train, the emissions (HC, CO, and NO_2) were then determined by simulating vehicle operation over an emission driving cycle. A more detailed discussion of the emission driving cycles used in this study is presented in Section 5, and speed-time plots of each cycle are presented in Appendix D.

SECTION 5

VEHICLE POWER REQUIREMENTS

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NOMENCLATURE

A	frontal area of vehicle, ft^2
a	acceleration of vehicle, mph/sec
C_d	aerodynamic drag coefficient
HP	wheel horsepower, hp
F_d	road resistance force, lb
F_p	propulsive force (at wheels), lb
G	gear ratio
m	actual vehicle mass, lb-sec/mph
m^*	effective vehicle mass = 1.1 m
n	electric motor speed, rpm
P	tire pressure, psi
R_r	tire radius, ft
TE	tractive effort (force), lb
T_m	electric motor output torque, ft-lb
V	vehicle speed, mph
W	loaded vehicle weight, lb
η	driveline efficiency, percent
θ	grade angle

SECTION 5

VEHICLE POWER REQUIREMENTS

5.1 INTRODUCTION

This section presents a discussion of the development of power and torque requirements for the drive train of the hybrid engine. These requirements were developed for each class of vehicles studied. The following criteria were used as the principal guidelines in establishing the vehicle performance requirements upon which the power and torque requirements are based:

1. Vehicle performance must meet all specifications of the Air Pollution Control Office (APCO), Environmental Protection Agency*
2. Vehicle performance must be adequate to meet the acceleration requirements of emission driving cycles
3. Vehicle performance must be comparable to that of present-day vehicles of the same class.

The third criterion appears reasonable for at least first-generation hybrid vehicles since they would share the roads with present vehicles. Incorporating hybrid vehicles having a significantly less performance capability than conventional vehicles into metropolitan traffic would probably not only increase congestion but might also promote unsafe driving conditions. As discussed in Sections 5.5 and 5.6, the incorporation of high speed versions of the intracity bus and delivery van is justified on the basis of this criterion.

The procedure used to establish baseline vehicle performance requirements/ capabilities and to develop hybrid drive-train power and torque requirements was not the same for each vehicle class. Published road test data were used to define the basic acceleration characteristics for the family and commuter cars. The unavailability of similar test data for the van and bus made it necessary to use either manufacturer acceleration performance

*This agency was formerly designated as the National Air Pollution Control Administration.

calculations or to derive vehicle acceleration performance using vehicle specification and engine performance data. Hybrid drive-train power and torque requirements were derived from the acceleration curves for each vehicle.

Analysis of each emission driving cycle was performed to determine the peak acceleration, peak power, and total energy requirements for vehicle operation. The results of these analyses were used as minimum design constraints for hybrid drive-train components. Additional use of emission driving cycles and design driving cycles is discussed in Section 4.4 of this report.

5.2 SUMMARY OF VEHICLE SPECIFICATION

Section 3.1 summarizes the basic specifications that were used in establishing power requirements for each class of vehicle.

5.3 EMISSION DRIVING CYCLE ANALYSIS

5.3.1 Objectives of Driving Cycle Analysis

The primary objective of this work was to compute drive-train output requirements for vehicle emission driving cycle combinations listed in Table 5-1.

The low-speed delivery van and low-speed bus emission cycles were developed using the basic velocity mode characteristics presented in Section 3. The development of an emission cycle for the high-speed bus is discussed in Section 5.6.4. It was not necessary to derive separate cycles for the high- and low-speed van (See Section 5.5.2).

A revised version of the Department of Health, Education, and Welfare (DHEW) cycle has recently been published (Ref. 5-3). This new cycle has peak acceleration requirements which are significantly lower than the previous DHEW version. The potential impact of the revised cycle on vehicle performance requirements could not be assessed due to the time limitations on this study.

Table 5-1. Vehicle/Driving Cycle Combinations

Vehicle Type	Weight, lb	Driving Cycle
Family Car	4,000	DHEW* (Ref. 5-1)
Family Car	4,000	New York (Ref. 5-2)
Commuter Car	1,700	DHEW
Delivery Van (Low Speed)	7,000	Van Emission Cycle
Delivery Van (High Speed)	7,000	Van Emission Cycle
High-speed Intracity Bus	30,000	High-speed Bus Emission Cycle
Low-speed Intracity Bus	30,000	Low-speed Bus Emission Cycle

*DHEW - Department of Health, Education, and Welfare

5.3.2 Results of Driving Cycle Analysis

A driving cycle analysis computer program was used to determine power, energy, and torque requirements for each vehicle. Output from the program included plots of:

1. Vehicle speed versus time
2. Vehicle acceleration versus time
3. Cumulative distance traveled versus time
4. Wheel horsepower versus time
5. Cumulative energy delivered to wheels versus time
6. Wheel torque versus time
7. Vehicle acceleration versus vehicle speed
8. Wheel horsepower versus vehicle speed

A set of these plots for each vehicle/driving cycle combination is presented in Appendix D. A summary of the principal results is presented in Table 5-2.

5.4 HYBRID DRIVE-TRAIN POWER AND TORQUE REQUIREMENTS FOR THE FAMILY AND COMMUTER CARS

5.4.1 Computational Procedure

The following procedure was used to establish baseline vehicle performance and to define drive train torque and power output requirements:

1. Acquire speed-time (maximum acceleration) curves and physical characteristics for each class of conventional vehicles from published road test results

2. Select reference speed-time curves from these data which define the required performance for each class of hybrid vehicle

3. Use the reference acceleration curves to compute (a) acceleration versus speed, (b) power output (at the wheels) versus speed, (c) gradeability versus speed, and (d) required motor torque versus motor speed

4. Verify that the selected (reference) performance for each hybrid vehicle is adequate to meet the performance specifications and the requirements of the DHEW driving cycle which was used for subsequent emission calculations

Table 5-2. Summary of Vehicle Performance Requirements Derived From Emission Driving Cycle Analysis

Vehicle Type	Cycle	Peak Acceleration, mph/sec	Peak Power, hp	Average Power, hp	Total Energy at Wheels, hp-hr/Cycle
Commuter	DHEW	5	26	2.7	1.03
Family	DHEW	5	61	6.2	2.34
Delivery Van (Low and High Speed)	APCO	4	98	6.56	0.17
Intracity Bus					
Low speed	APCO	4	265	39	0.39
High speed	Aerospace Corporation	2.5	105	27	0.28

5.4.2 Vehicle Speed-Time Characteristics

Figures 5-1 and 5-2 show speed-time curves for several makes of conventional cars in the 4000-lb and 1700-lb classes, respectively. The baseline curve for the family car was selected to represent a performance intermediate between high and low performance vehicles.

The baseline speed-time curve shown in Fig. 5-2 for the commuter car was selected primarily to satisfy requirements of the DHEW driving cycle as discussed in the following section.

5.4.3 Acceleration Performance

An acceleration-speed relationship was obtained for each vehicle by graphically differentiating the results shown in Figs. 5-1 and 5-2. The results are plotted in Fig. 5-3. Acceleration-time points computed from DHEW driving cycle data are also plotted in Fig. 5-3. These results show that hybrid powerplants sized on the basis of the reference performance curves will satisfy all points and are therefore capable of operating over the DHEW driving cycle. Figure 5-3 also shows that the reference commuter car curve has adequate performance for the DHEW cycle.

5.4.4 Horsepower Requirements of Hybrid Drive Train

The required wheel horsepower that must be supplied by a hybrid drive train as a function of vehicle speed, weight, and drag characteristics was derived from the reference plots shown in Fig. 5-3. The instantaneous propulsive force required to accelerate a vehicle can be expressed as

$$F_p(V) = m^{*}a(V) + 0.002558\, C_d\, AV^2 + \frac{W}{2000}\left(10.0 + \frac{300}{P} + \frac{0.070V^2}{P}\right) \tag{5-1}$$

where the acceleration, a, is expressed as a function of vehicle speed, and the first, second, and third terms on the right of the equation represent the inertia, aerodynamic drag, and rolling resistance forces, respectively. The rolling resistance is given in Ref. 5-4.

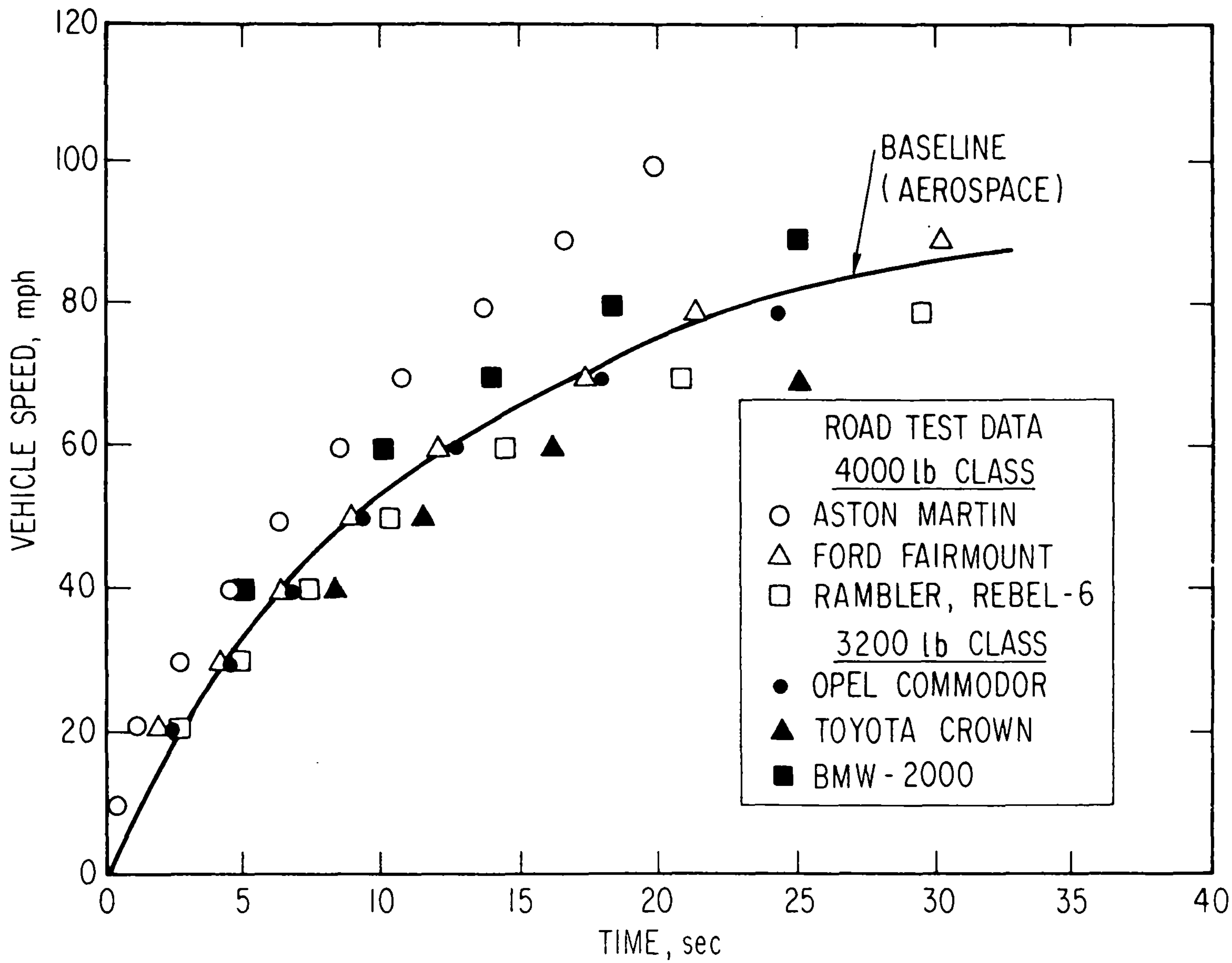

Figure 5-1. Speed Versus Time Curves for Family Cars

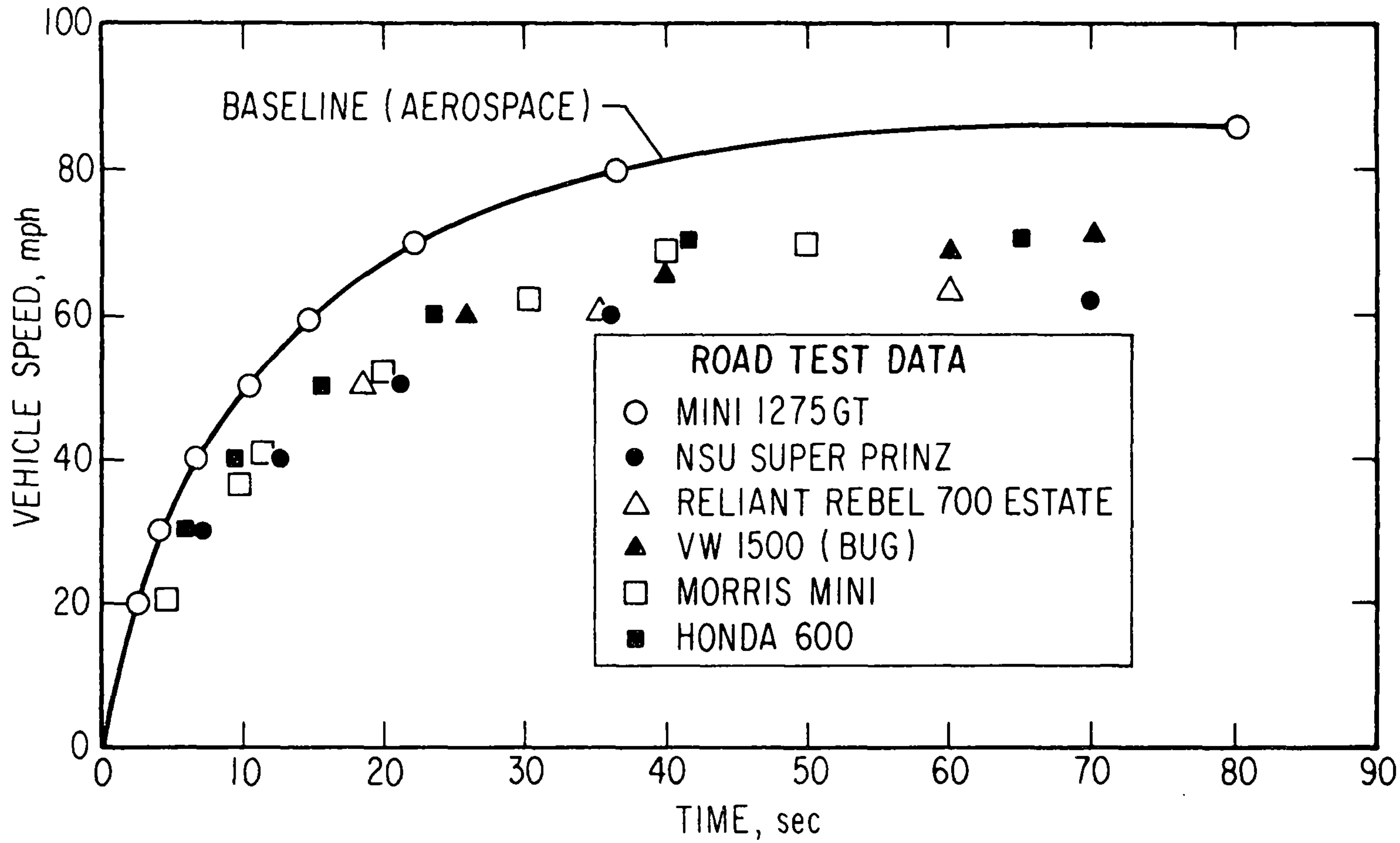

Figure 5-2. Speed Versus Time Curves for Commuter Car

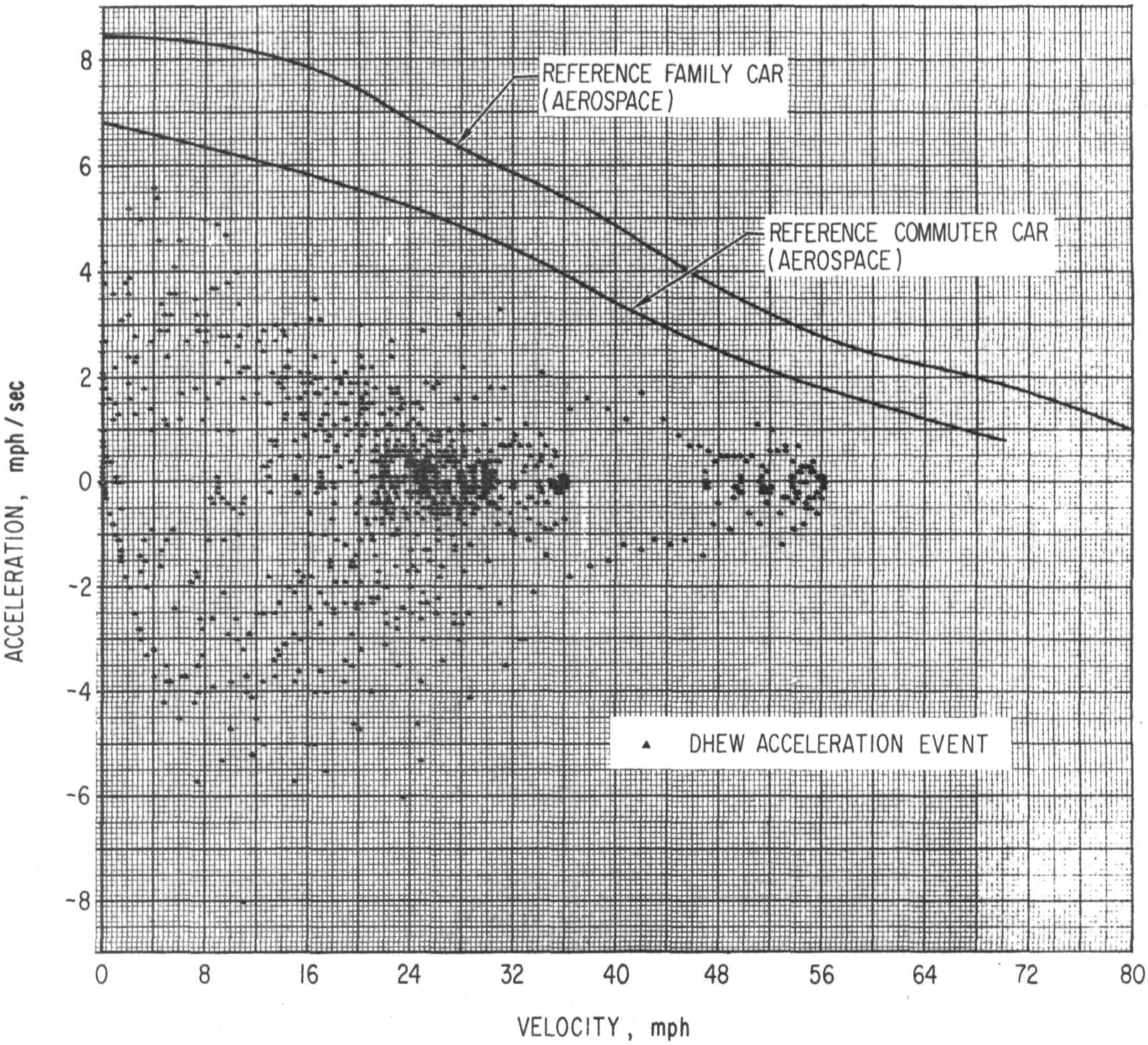

Figure 5-3. Comparison of Vehicle Velocity - Acceleration Characteristics to DHEW Driving Cycle Requirements

The instantaneous horsepower during maximum acceleration can be expressed by

$$HP(V) = 0.00267\ F_p(V)V \tag{5-2}$$

With the exception of the tire pressure, P, values of the parameters used to numerically evaluate Eqs. (5-1) and (5-2) are presented in the vehicle specifications.

The resulting equations for each vehicle are

Family Car:

$$HP(V) = 0.534\ a(V) + 0.106\ V + 9.88 \times 10^{-5}\ V^3 \tag{5-3}$$

Commuter Car:

$$HP(V) = 0.227\ a(V)\ V + 0.0455\ V + 4.83 \times 10^{-5}\ V^3 \tag{5-4}$$

Cruise and maximum acceleration power requirements at the wheels are depicted in Figs. 5-4 and 5-5.

5.4.5 Gradeability Performance Requirements

The basic specifications give a gradeability requirement for each vehicle. Gradeability is defined as the maximum grade on which a specific velocity can be maintained. Mathematically, it can be expressed by

$$\sin\theta = \frac{F_p(V) - F_d(V)}{W} = \frac{m^* a(V)}{W} \tag{5-5}$$

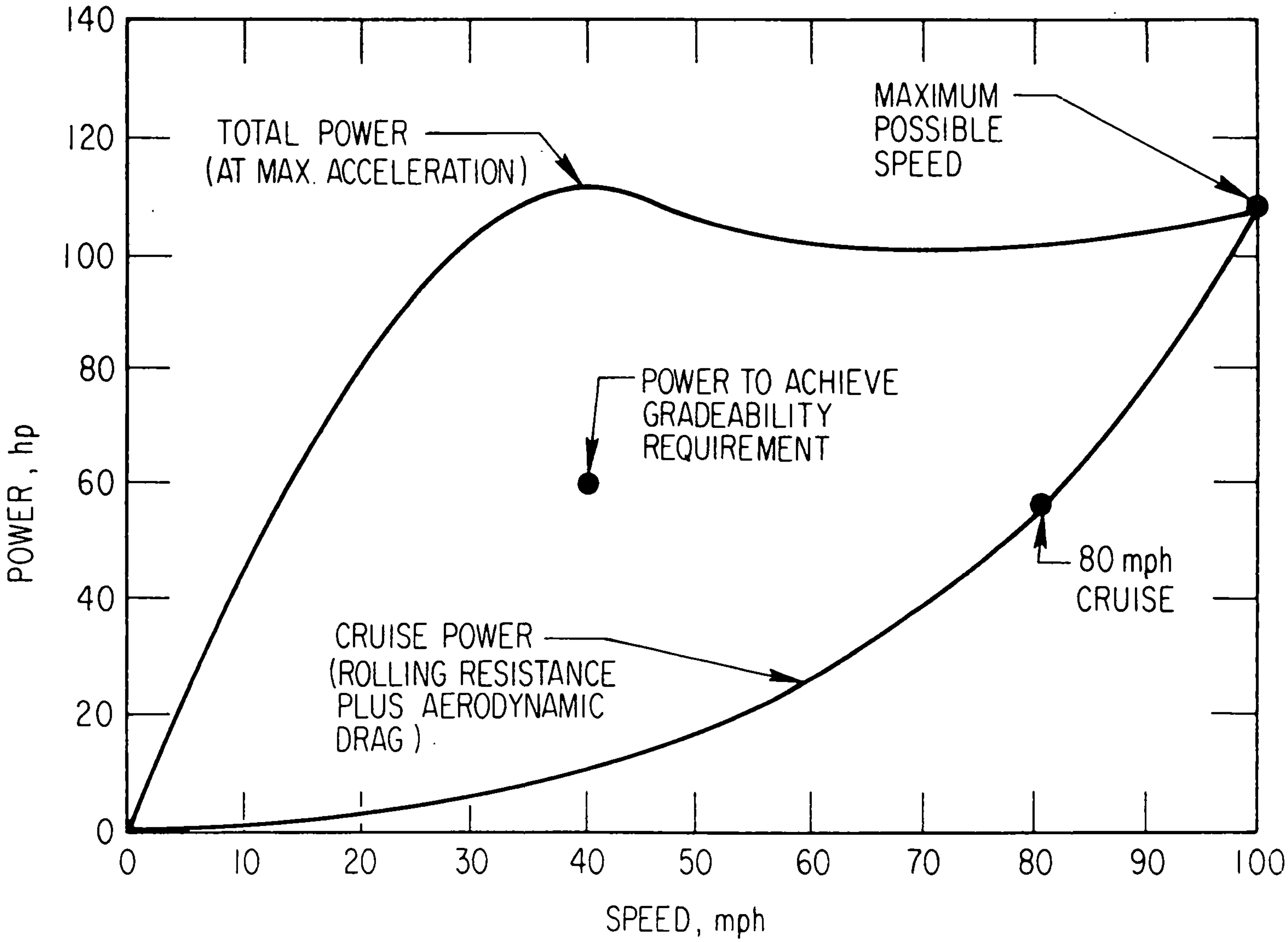

Figure 5-4. Power Requirements for Family Car

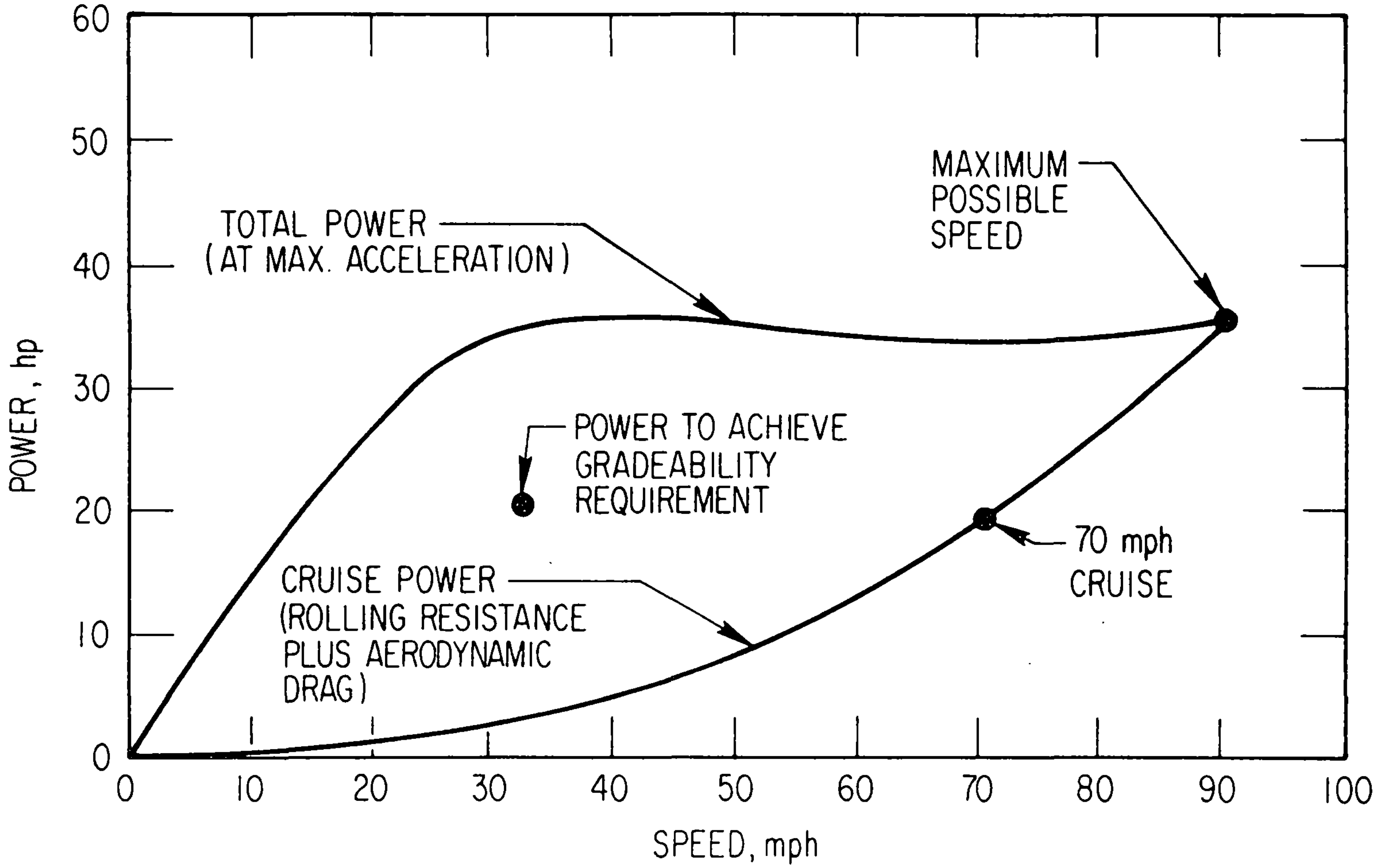

Figure 5-5. Power Requirements for Commuter Car

So that if $a(V)$ is the maximum acceleration capability at any velocity, then θ is the gradeability at that velocity. Substituting for m^* and adjusting units gives

$$\sin \theta = \frac{(1.1)\ W\ a(V)}{(32.2)\ (0.6818)\ W} = 0.05\ a(V) \tag{5-6}$$

Percent grade is equal to 100 x tan θ. The data in Fig. 5-3 were used to obtain tan θ as a function of vehicle speed and the results for each vehicle are shown in Fig. 5-6. It can be seen that both vehicles have a maximum gradeability performance which exceeds the specification requirement. Maximum gradeability is that which could be achieved with full power train output. It is more than could be achieved using, for example, only the power output of the heat engine. Output horsepower at the wheels required to achieve a certain gradeability performance can be expressed as

$$HP = 0.00267\ [W \sin \theta + F_d(V)]\ V \tag{5-7}$$

The APCO gradeability specifications require output horsepowers of 61 hp and 21 hp for the family car and the commuter car, respectively. Each of these specifications is well below the maximum power required for acceleration performance.

5.4.6 Torque Requirements for the Electric Drive Motor

Torque requirements for an electric drive motor (assuming that the motor is the only source of drive torque) were obtained using the relation

$$T_m = \frac{F_p(V)\ R_r}{\eta G} = \frac{R_r}{\eta G}\ [m^*\ a(V) + F_d(V)] \tag{5-8}$$

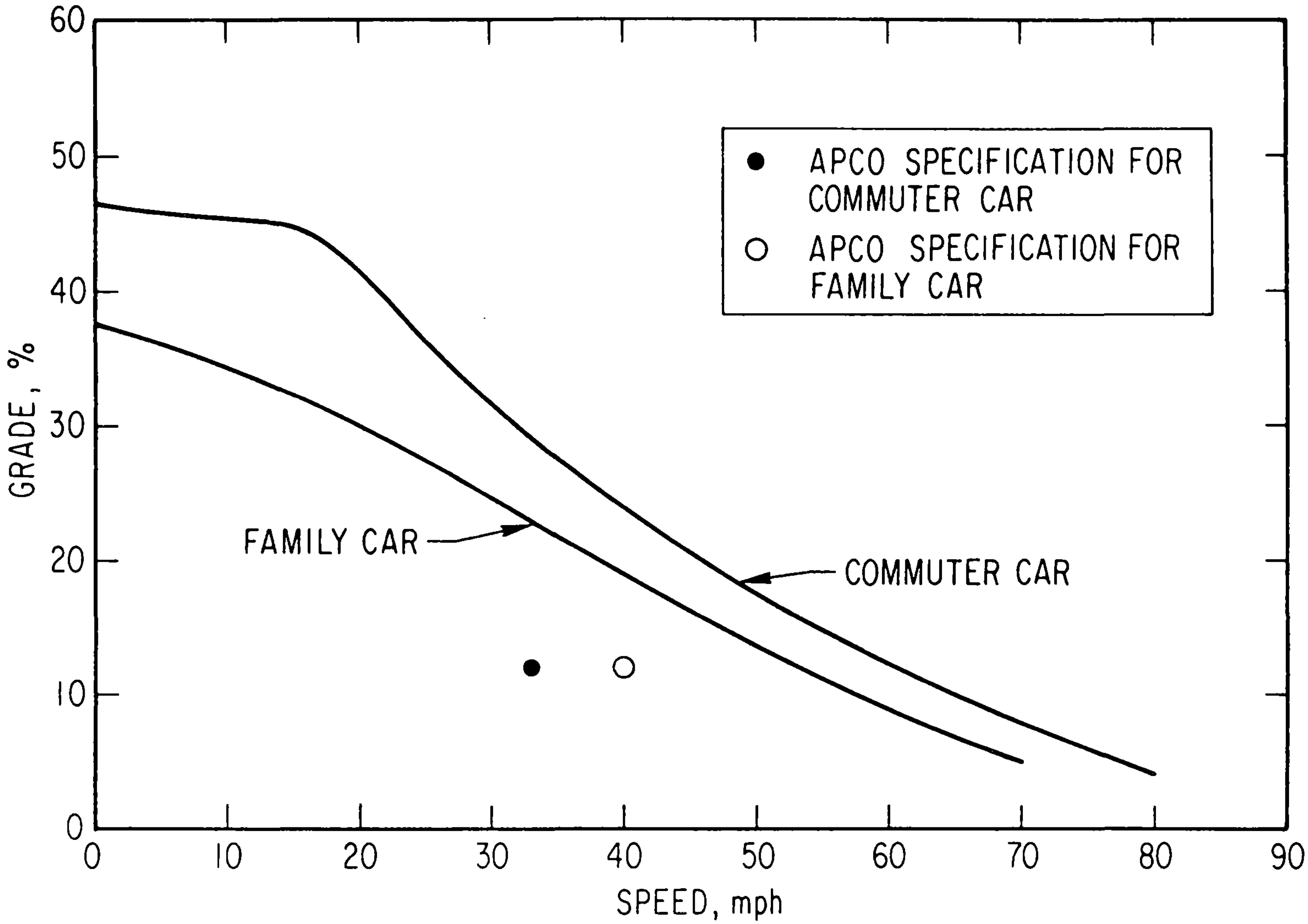

Figure 5-6. Gradeability of Family and Commuter Cars

A driveline efficiency of 90 percent and a tire radius of 1.1 ft was assumed for the family and commuter car. Calculations were performed for gear ratios of 1.0, 2.5, 5.0, and 10. In order to express torque as a function of motor speed, n, the following expression was used:

$$n = \frac{G\ V}{0.0715\ R_r} = 12.75\ V\ G \qquad (5\text{-}9)$$

The results are plotted in Figs. 5-7 and 5-8.

5.5 HYBRID DRIVE-TRAIN POWER AND TORQUE REQUIREMENTS FOR THE DELIVERY VAN

5.5.1 Computational Procedure

The following procedure was used to compute hybrid drive-train power and torque requirements for the delivery van:

1. Compute tractive effort-speed relationship for a typical van using manufacturers' engine performance and body data
2. Simulate vehicle acceleration and calculate wheel power, wheel torque, and gradeability performance
3. Compare calculated performance with APCO specifications and emission cycle acceleration requirements

5.5.2 Acceleration and Gradeability Performance

Tractive effort versus speed was computed using engine torque and transmission data supplied by the manufacturer (Ref. 5-5). It was found that the following expression gives a good approximation of van tractive effort (i.e., propulsive force)

$$TE = 2511.8 - 46.1V + 0.26\ V^2 \qquad (5\text{-}10)$$

The coefficients in Eq. (5-10) were used as inputs for a vehicle performance computer program (Ref. 5-6) which was developed to determine vehicle speed, acceleration, drag force, gradeability, and total output horsepower at the wheels as a function of time.

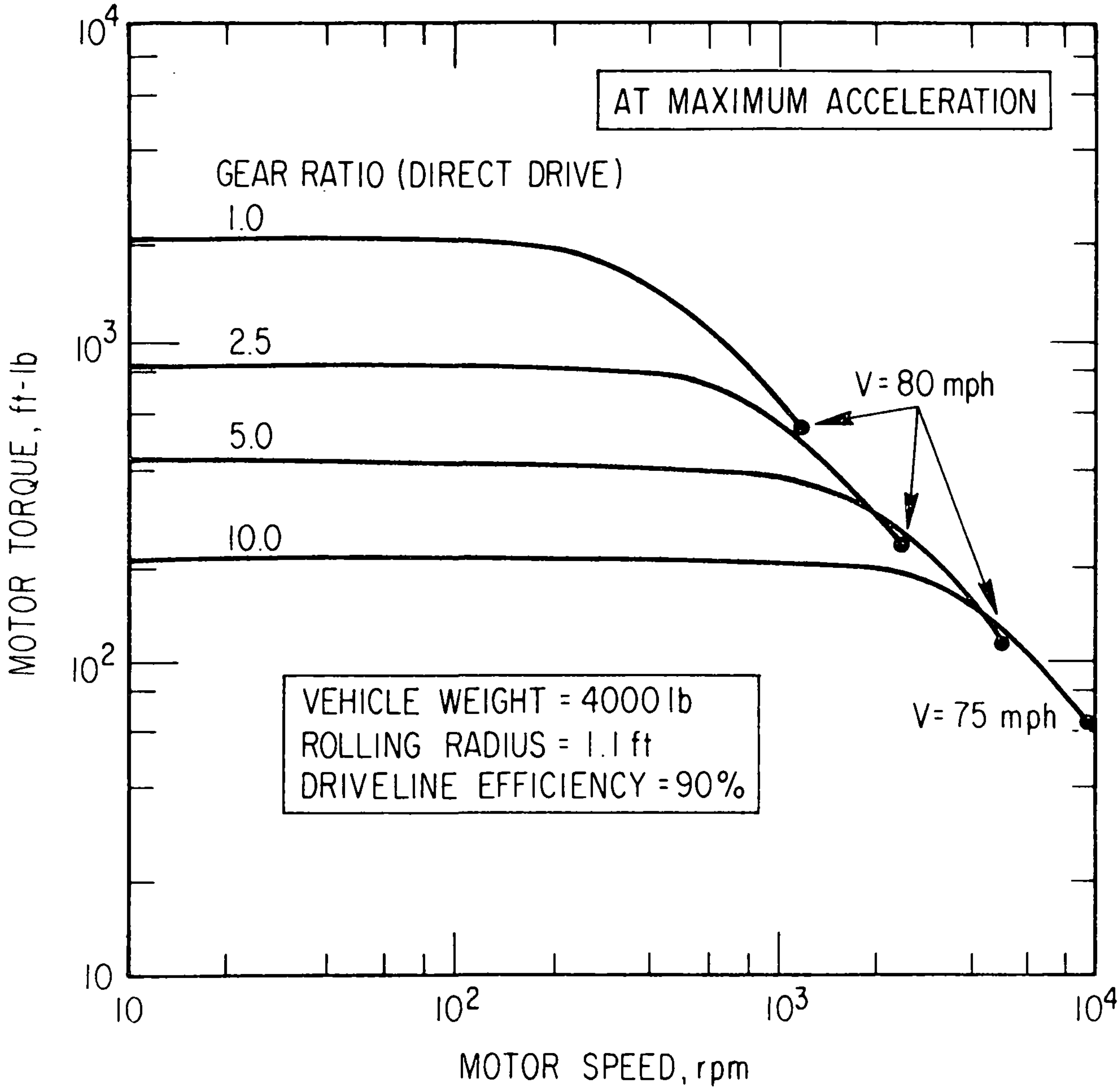

Figure 5-7. Required Motor Torque for Family Car

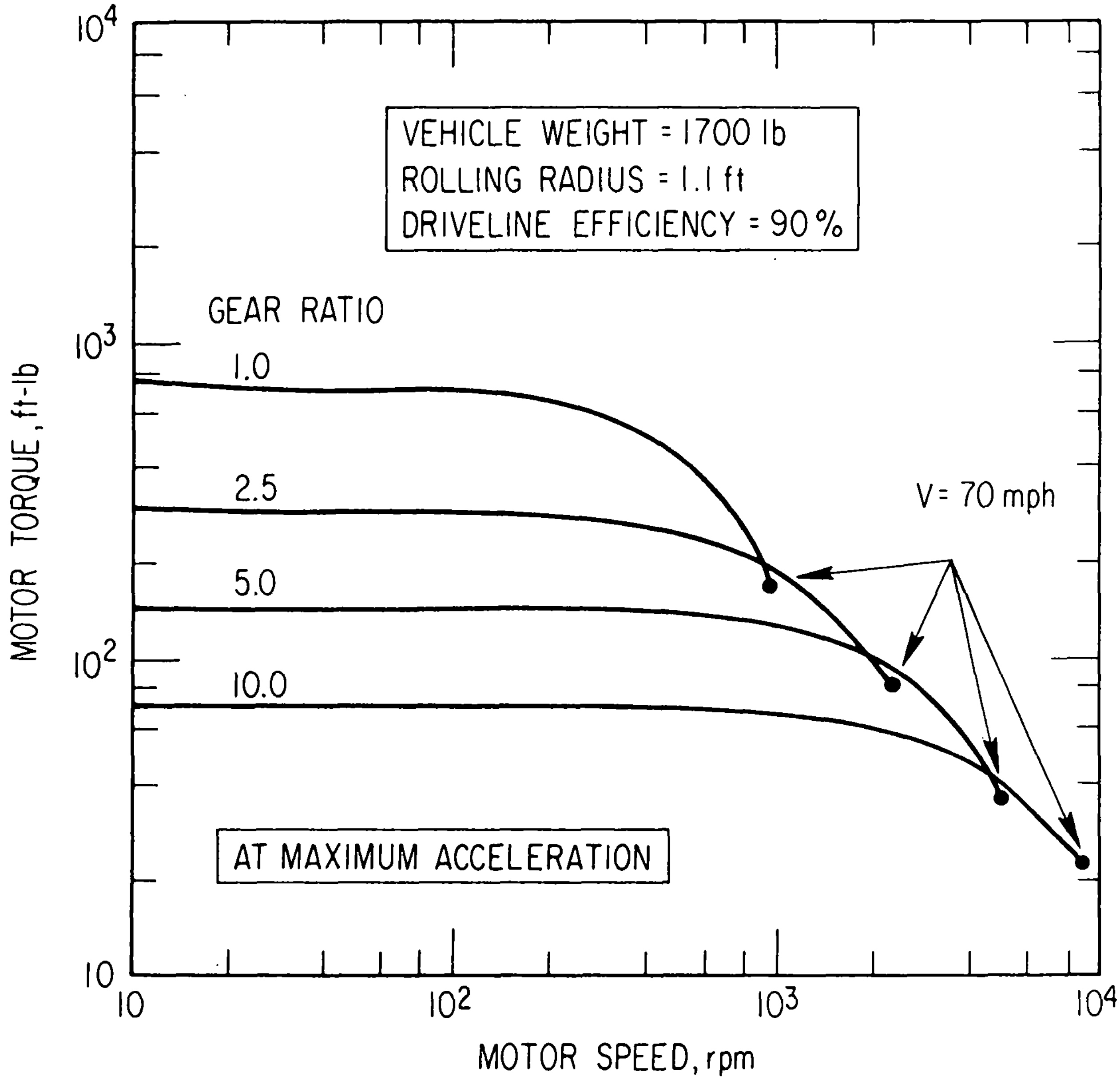

Figure 5-8. Required Motor Torque for Commuter Car

Table 5-3. Summary of Body and Chassis Characteristics for a Delivery Van

Body	
Type	Forward Control
Wheel Base	137 in.
Load Length	12 ft
Body Weight	1675 lb
Frontal Area*	46 ft^2
Chassis	
Manufacturer	Ford Motor Co.
Engine Type	6 Cylinder
Engine Displacement	240 $in.^3$
Rated Horsepower	150 at 4000 rpm
Transmission Type	Three-speed Manual
Gear Ratios	3.77, 1.87, 1.0
Axle Ratio	4.58
Chassis Weight	2900 lb
Maximum Payload Weight	3240 lb
Maximum Loaded Vehicle Weight**	7815 lb
Approximate Maximum Speed	70 mph

*Performance requirements based on frontal area of 42 ft^2

**A loaded weight of 7000 lb was assumed for performance requirements calculations

Acceleration performance of the delivery van is shown in Fig. 5-9 and is comparable to performance of vans which have been in use since 1965 (Refs. 5-7 and 5-8). Basic body and engine data for a typical van are summarized in Table 5-3. Results from Fig. 5-9 were used to obtain acceleration and gradeability versus vehicle speed as shown in Figs. 5-10 and 5-11.

Figure 5-11 shows that van performance is more than adequate to meet the APCO gradeability specification. Also, Table 5-4 shows that the acceleration capability for both the low- and high-speed versions of the delivery van is sufficient to meet the acceleration requirements of a proposed delivery van emissions driving cycle presented in Appendix D. Therefore, the same emission cycle could be used for both versions of the van.

Table 5-4. Comparison of Low- and High-speed Van Acceleration Performance and Emission Driving Cycle Acceleration Requirements

Velocity Mode, mph	Time, sec	
	Proposed Cycle Requirement	Calculated Performance
0 - 16	4	3.5
16 - 32	4	3.5

The performance of the low- and high-speed vans is based on the same vehicle-engine combination. It is assumed that, for a conventional vehicle, a governor would be used to limit the rpm of the engine in the low-speed vehicle to a value consistent with a maximum vehicle speed of 40 mph. The maximum attainable velocity of the high-speed van was found to be approximately 75 mph and was determined by plotting tractive effort versus speed and the sum of the road resistance forces versus speed on the same graph. Maximum vehicle speed is the point at which the curves cross. The 75-mph calculated maximum theoretical speed is consistent with a maximum speed estimate of 70 mph for existing vans of similar size and weight.

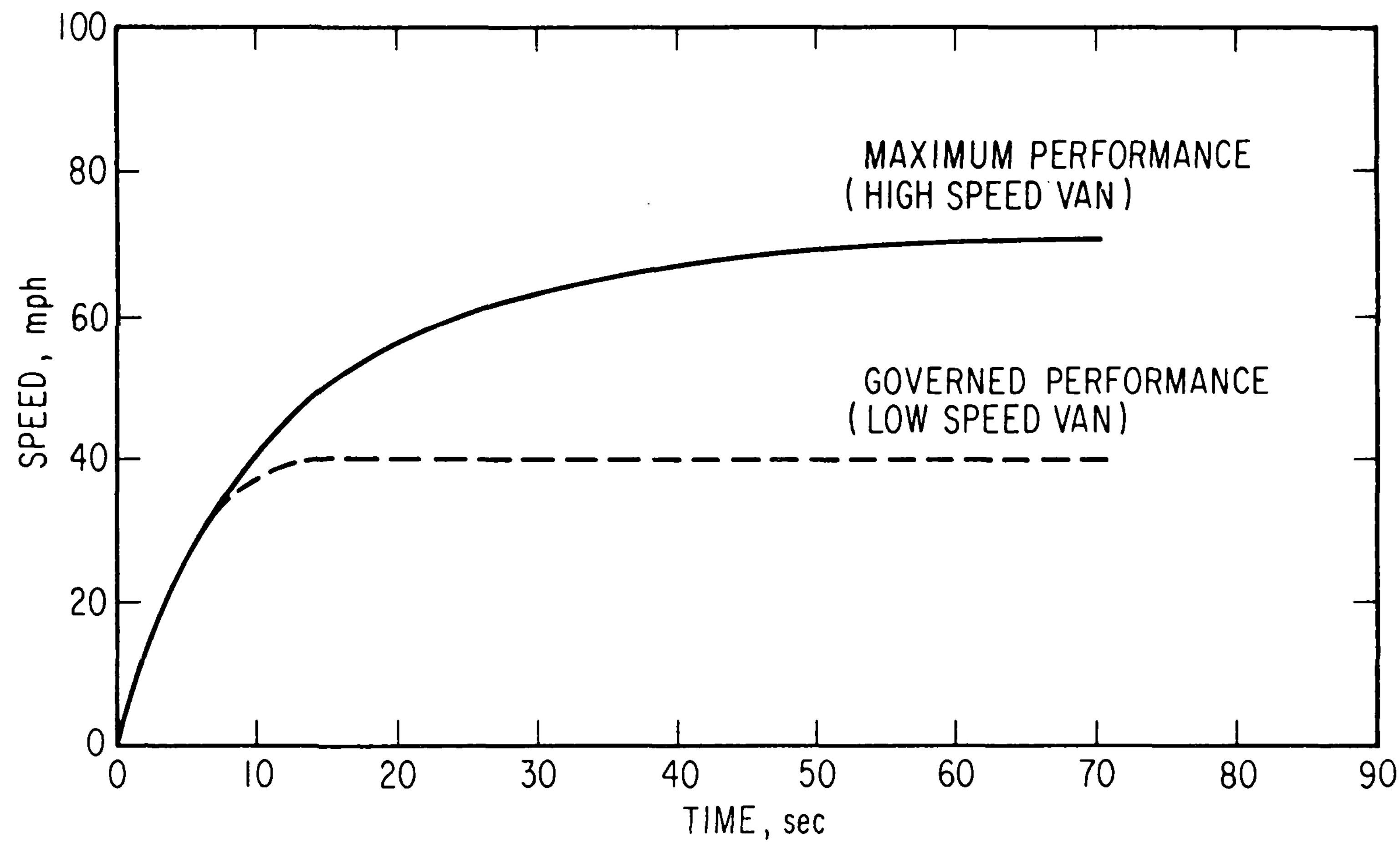

Figure 5-9. Speed Versus Time Curves for Delivery Van

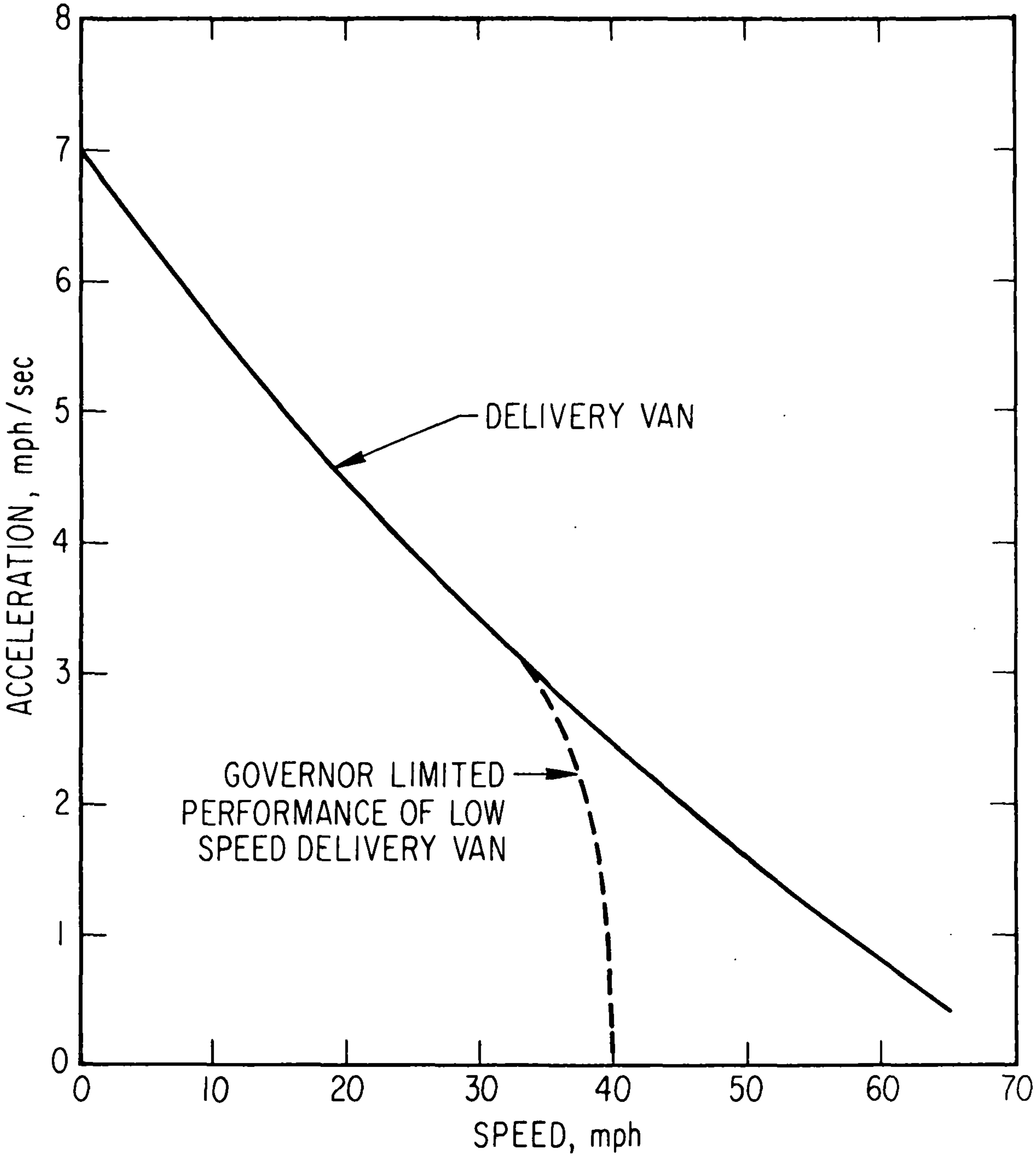

Figure 5-10. Acceleration Performance Requirements for Delivery Van

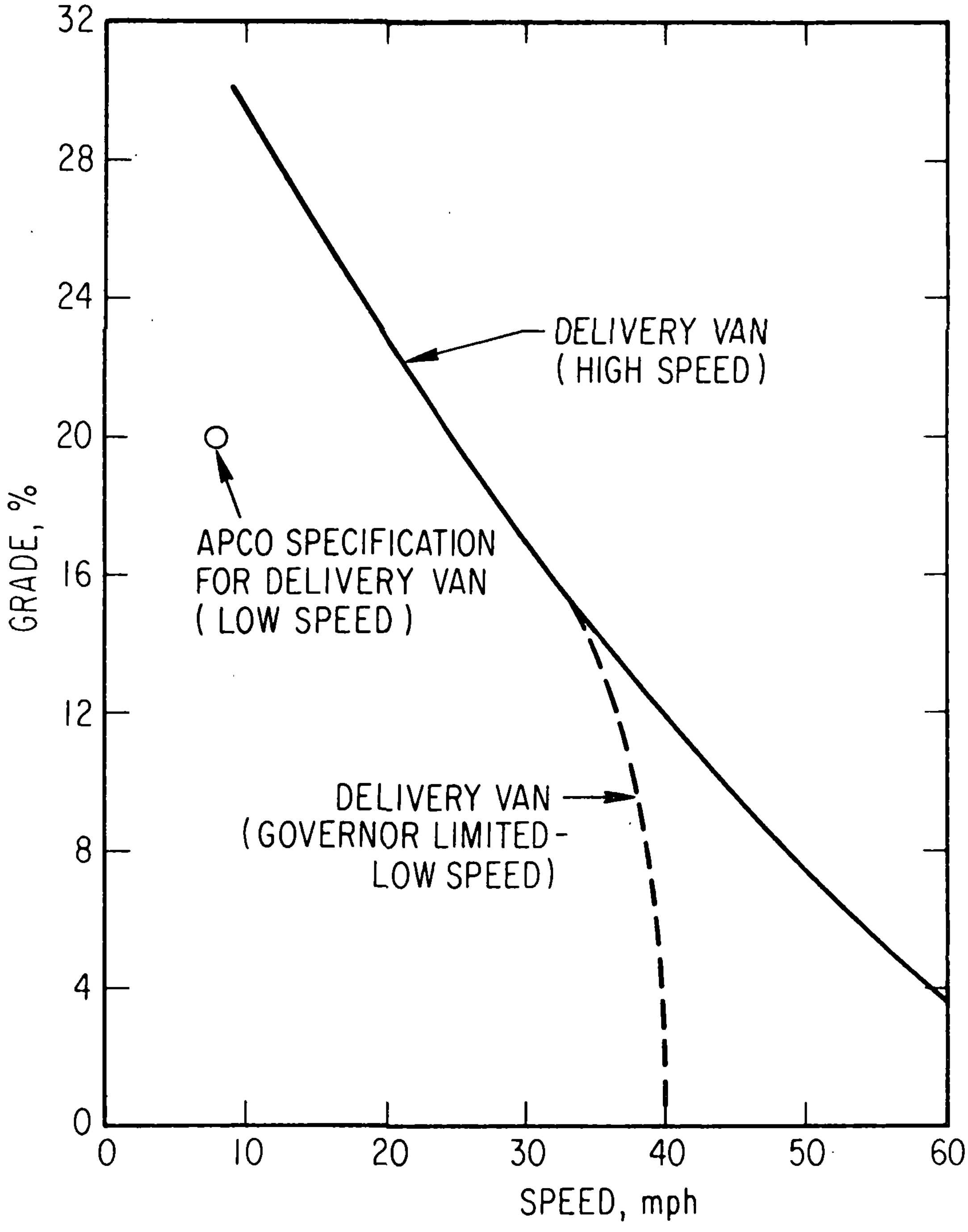

Figure 5-11. Maximum Gradeability Performance of Delivery Van

5.5.3 Hybrid Drive-Train Power and Torque Requirements

The power that must be delivered to the wheels in order to achieve the acceleration performance presented in Fig. 5-9 is shown in Fig. 5-12. The total power requirement is approximately the same for the low- and high-speed versions in the speed range between 0 and 40 mph.

Electric motor torque requirements are shown in Fig. 5-13.

5.6 HYBRID DRIVE-TRAIN POWER AND TORQUE REQUIREMENTS FOR INTRACITY BUS

5.6.1 Computation Procedure

The following procedures were used to compute hybrid drive-train power and torque requirements for the low- and high-speed versions of the intracity bus:

1. Computation Procedure for Low-speed Bus

   a. Use APCO gradeability and maximum speed specifications and emission cycle accelerations to calculate tractive effort/speed relationship

   b. Using the tractive-effort/speed relationship as input to the computer program, calculate acceleration, wheel horsepower, wheel torque, and gradeability as a function of vehicle speed

2. Computation Procedure for High-speed Bus

   a. Use manufacturers' published speed-time data to calculate vehicle acceleration as a function of speed

   b. Using the results of the preceding calculations, calculate wheel horsepower, torque, and gradeability as a function of speed, and synthesize the exhaust-emission cycle

5.6.2 Acceleration and Gradeability Performance

The acceleration curves for the low- and high-speed versions of an intracity bus are shown in Fig. 5-14. Only guidelines (1) and (2) discussed in Section 5.1 were used to estimate the required performance for the low speed bus

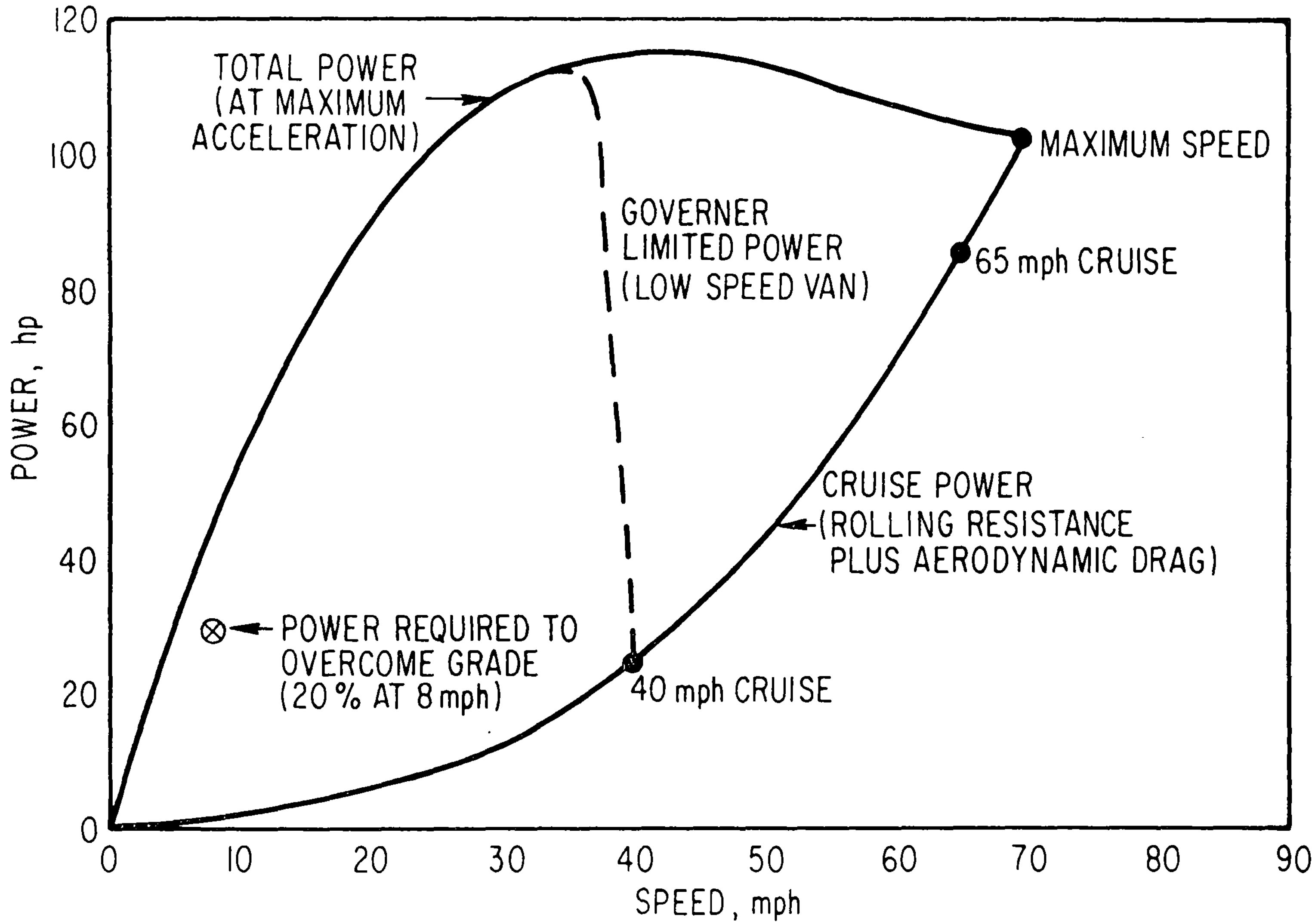

Figure 5-12. Power Requirements for Delivery Van

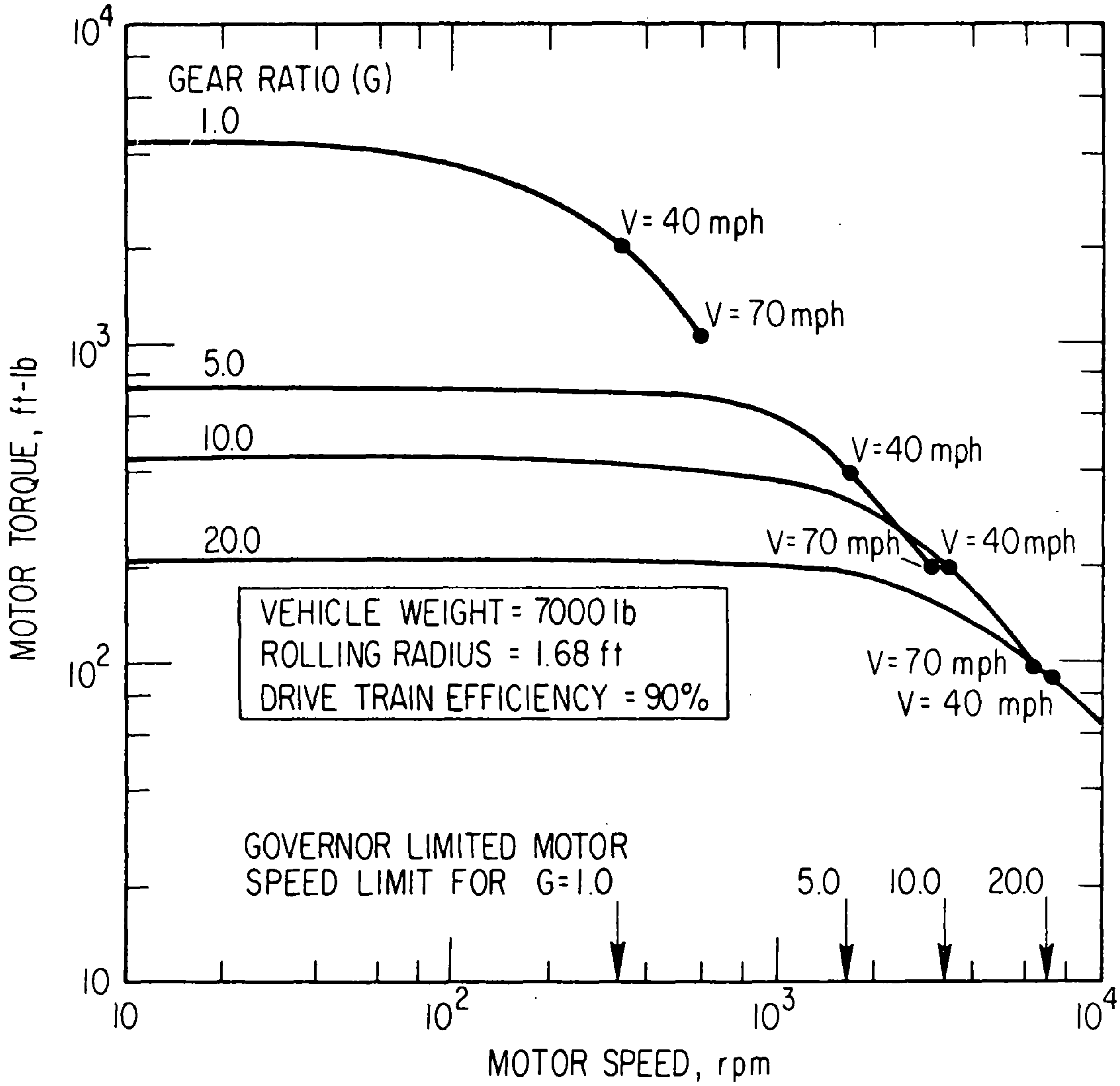

Figure 5-13. Electric Motor Torque Requirements for Delivery Van

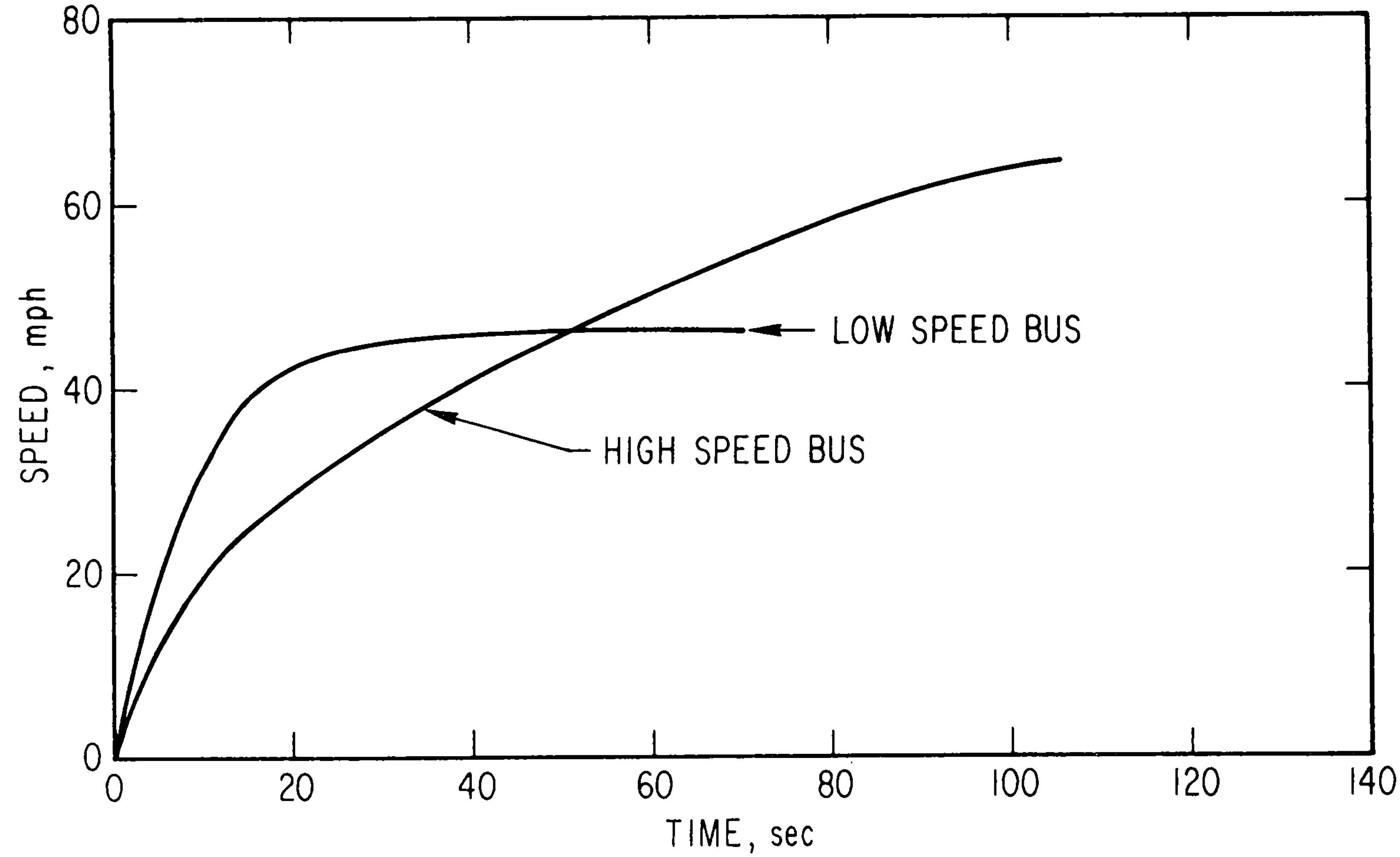

Figure 5-14. Speed Versus Time Curves for Intracity Bus

because data obtained from transportation agencies indicate that the vast majority of vehicles in use today have a maximum speed capability in excess of 40 mph (Refs. 5-9 and 5-10). The acceleration curve for the low-speed bus was derived by first estimating the tractive force required to (a) satisfy the APCO gradeability specification (20 percent grade at 6 mph), (b) provide the acceleration required by a proposed low-speed bus driving cycle in the speed ranges 0-19.2 mph and 19.2-25.6 mph, and (c) cruise at a constant speed of 40 mph. The lower boundary envelope of these tractive effort-speed points was then fitted to a quadratic function using the least-square fitting technique. The expression that resulted is

$$TE = 7234 - 85V - 1.2\,V^2 \tag{5-11}$$

The above coefficients were used as input to a computer program (Ref. 5-6) and velocity, acceleration, gradeability, wheel power, wheel torque, and load resistance forces were computed as a function of time. Acceleration and gradeability performance for the low-speed bus are shown in Figs. 5-15 and 5-16, respectively. A comparison of the acceleration rates in Table 5-5 verifies that the proposed low-speed bus has an acceleration capability sufficient to meet emission driving cycle requirements.

Table 5-5. Comparison of Intracity Bus Acceleration Capability with Emission Driving Cycle Acceleration Requirements

Driving Cycle Requirement		Maximum Performance Capability			
		Low-speed Bus		High-speed Bus	
Velocity Mode, mph	Time, sec	Velocity Mode, mph	Time, sec	Velocity Mode, mph	Time, sec
0 - 19.2	6	0 - 19.3	5	0 - 12	5
19.2 - 25.6	2	19.3 - 25.5	2	12 - 15	2

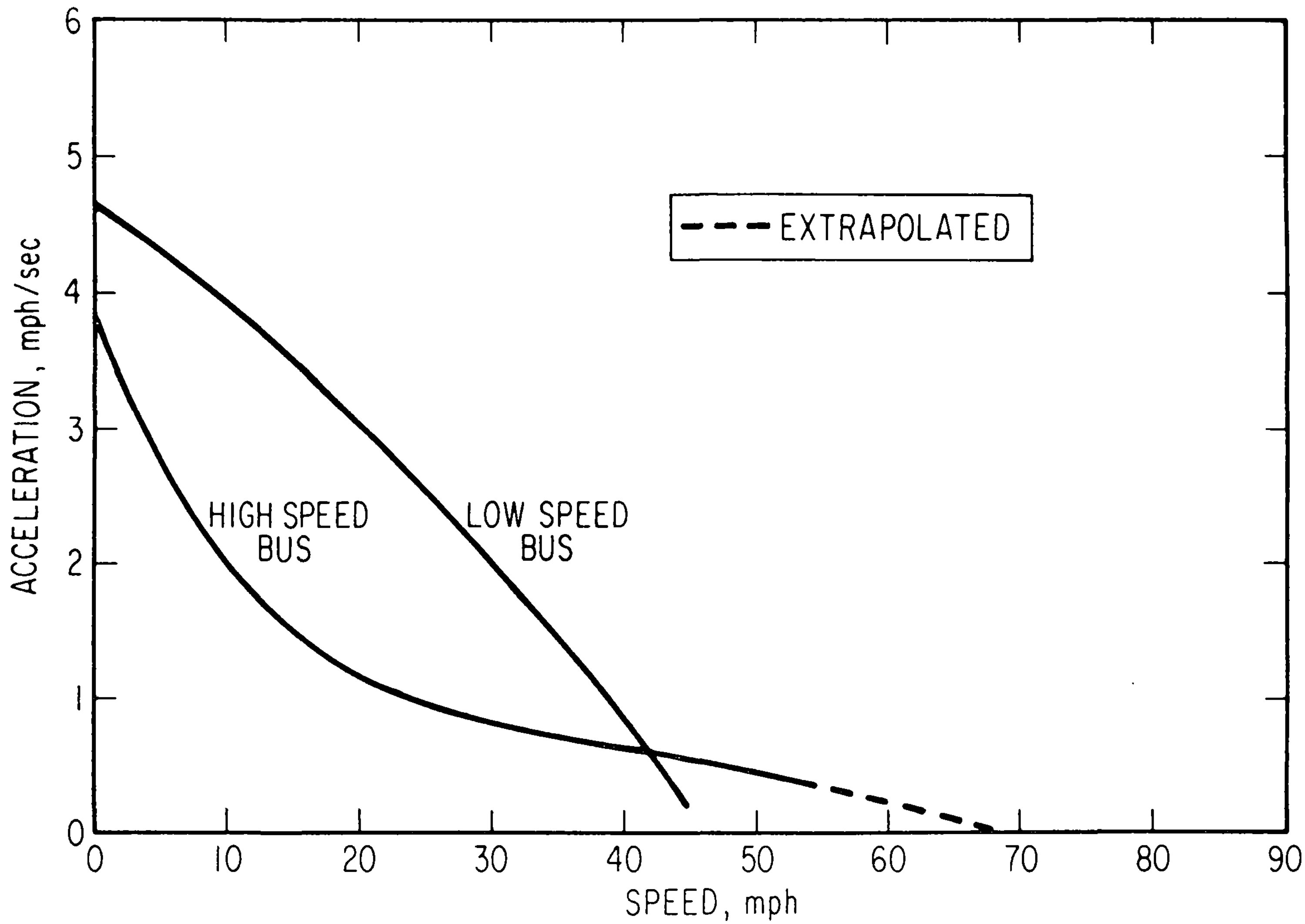

Figure 5-15. Acceleration Performance Requirements for Intracity Bus

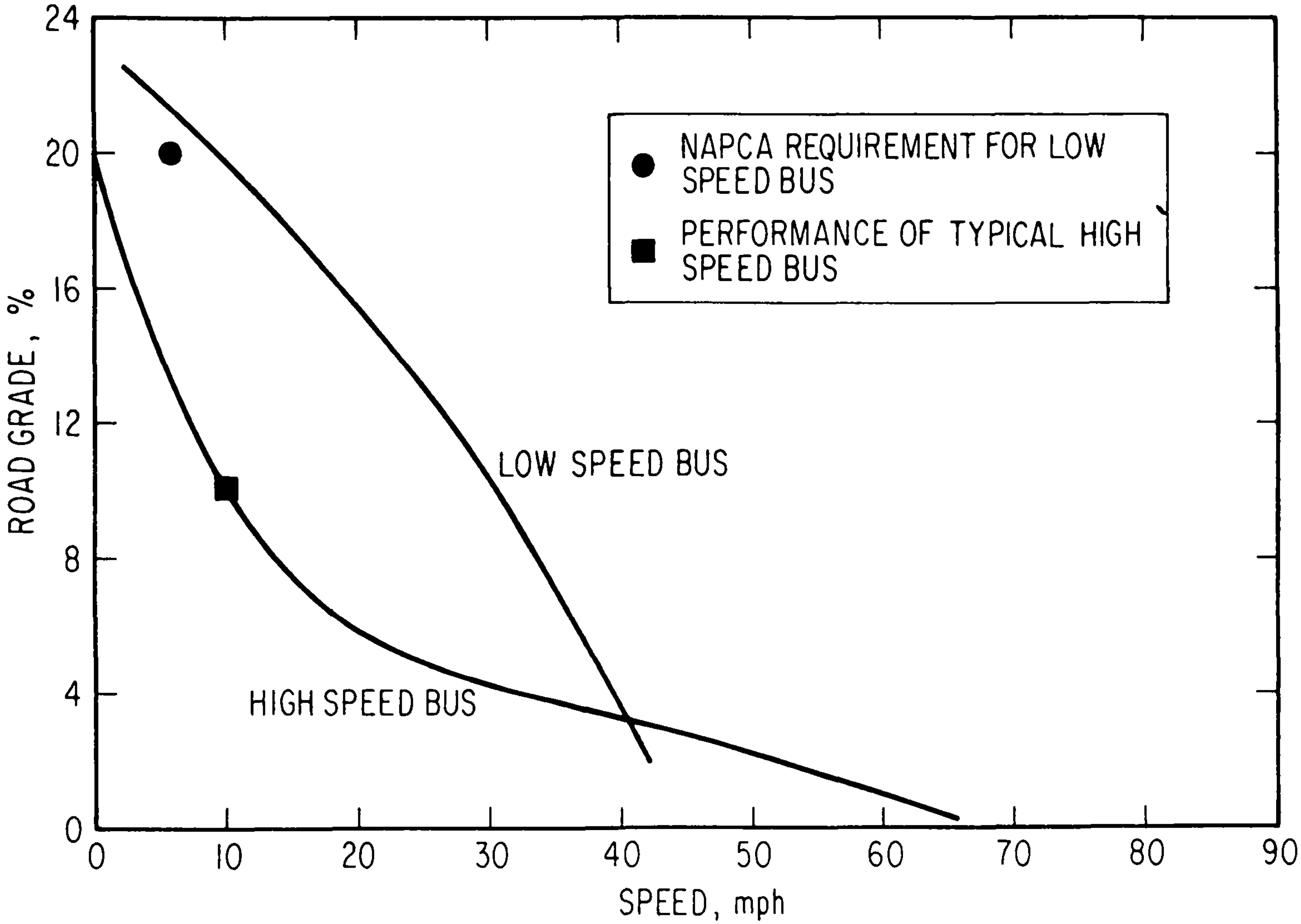

Figure 5-16. Maximum Gradeability Performance of Intracity Bus

The acceleration curve for the high-speed bus shown in Fig. 5-14 is typical of the performance of buses currently in operation in Los Angeles and Chicago (Refs. 5-9 and 5-10). Technical specifications supplied by the manufacturers are listed in Table 5-6.

Table 5-6. Specifications for High-speed Bus

Engine Type	Diesel
Manufacturer and Model No.	Cummins, V8-265 (special)
Displacement	785 $in.^3$
Rated Horsepower	210 hp
Torque Converter	Spicer 184-A
Axle Ratio	4.625
Rolling Radius	1.68 ft
Curb Weight	22,600 lb
51 Passenger + Driver	7,800 lb
Maximum Loaded Weight*	30,400 lb

*30,000-lb loaded weight was used in the performance calculations.

Table 5-5 also shows that a typical high-speed bus does not have the acceleration capability to meet bus driving cycle requirements. Therefore, it was necessary to synthesize a new cycle for the high-speed bus (See Section 5.6.4). A graphical procedure was used to determine the variation of acceleration with speed using the data from Fig. 5-14 for the high-speed bus and the results are shown in Fig. 5-15. Since the data shown in Fig. 5-14 do not show a limiting speed, it was necessary to extrapolate the acceleration-speed curve to zero slope to obtain an estimate of the maximum speed. The zero acceleration point, i.e., maximum speed point, is approximately 67 mph. The gradeability-speed relationship for the high-speed bus is shown in Fig. 5-16.

5.6.3 Hybrid Drive-Train Power and Torque Requirements

Hybrid engine drive-train power requirements for the low- and high-speed versions of the intracity bus are presented in Fig. 5-17. The total power curve of the low-speed bus peaks at approximately 295 hp and this peak stems directly from the acceleration requirements imposed by the bus driving cycle (See Table 5-5). For the high-speed bus, the following relationship was derived to express instantaneous wheel horsepower as a function of vehicle speed

$$HP(V) = 4.01\ a(V)\ V + 0.8\ V + 5.58 \times 10^{-4}\ V^3 \qquad (5\text{-}12)$$

A plot of the above expression is also shown in Fig. 5-17. The maximum practical speed was considered to be 60 mph for this study so that the peak power output of a hybrid drive train for the high-speed bus is approximately 230 hp. The power requirement for steady cruise, however, is only 170 hp.

Torque curves for the low-speed bus are presented in Fig. 5-18 and similar data for the high-speed bus are presented in Fig. 5-19. Torque-motor speed relationship was computed using Eqs. (5-8) and (5-9) for a rolling radius of 1.68 ft, drive line efficiency of 90 percent, and gear ratios of 1, 5, 10, and 20.

5.6.4 Emission Driving Cycle for High-speed Bus

Because of reasons discussed in Section 5.6.2, an emission driving cycle was developed for the high-speed bus. This cycle was used for subsequent exhaust emission calculations. The cycle was designed on the basis of the following criteria:

1. Cycle acceleration requirements must be compatible with bus performance
2. Distance traveled in each velocity mode must be approximately equal to the respective distances in the low-speed bus cycle
3. Rest time must be the same as that for low-speed bus cycle

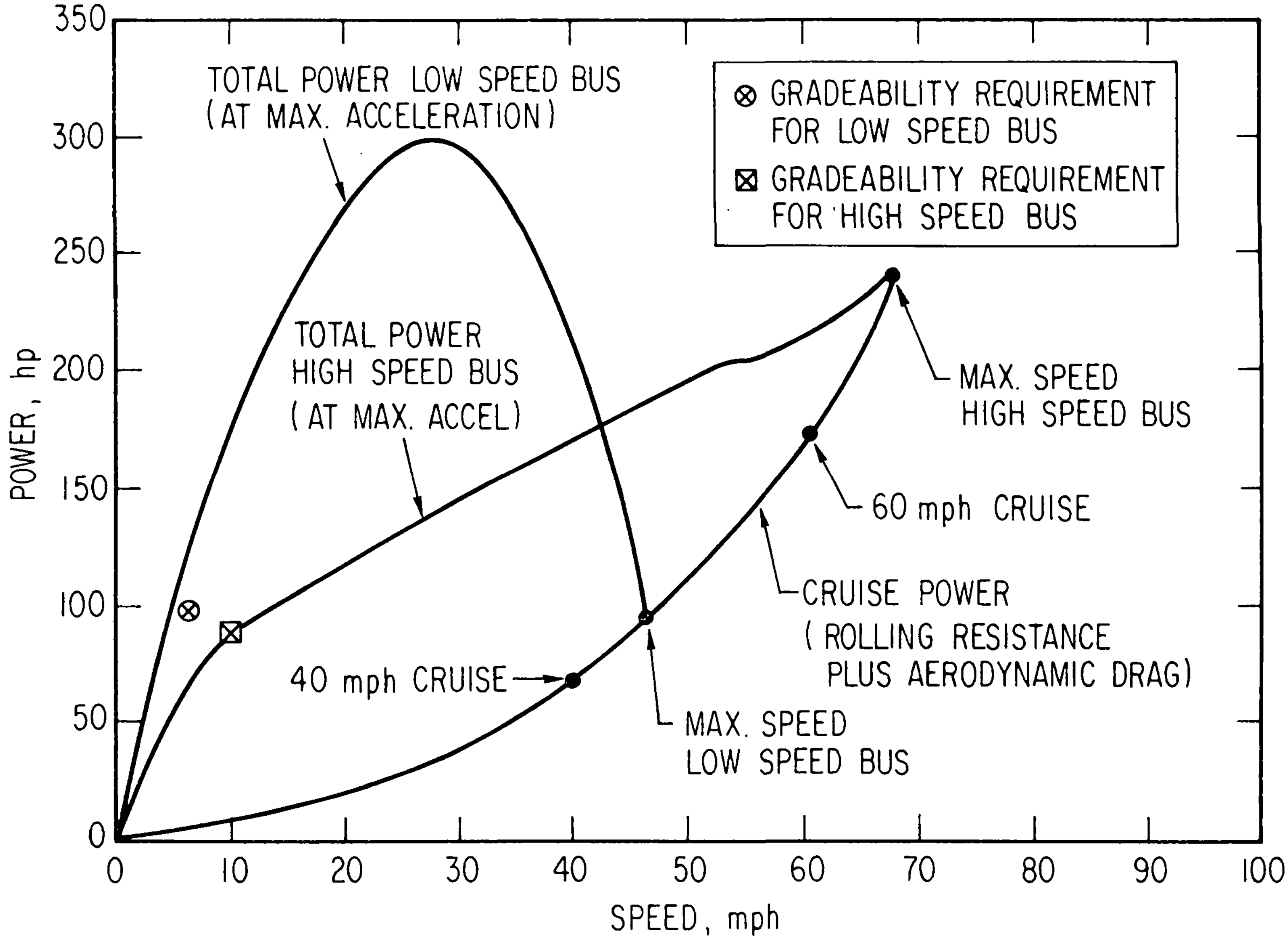

Figure 5-17. Power Requirements for Intracity Bus

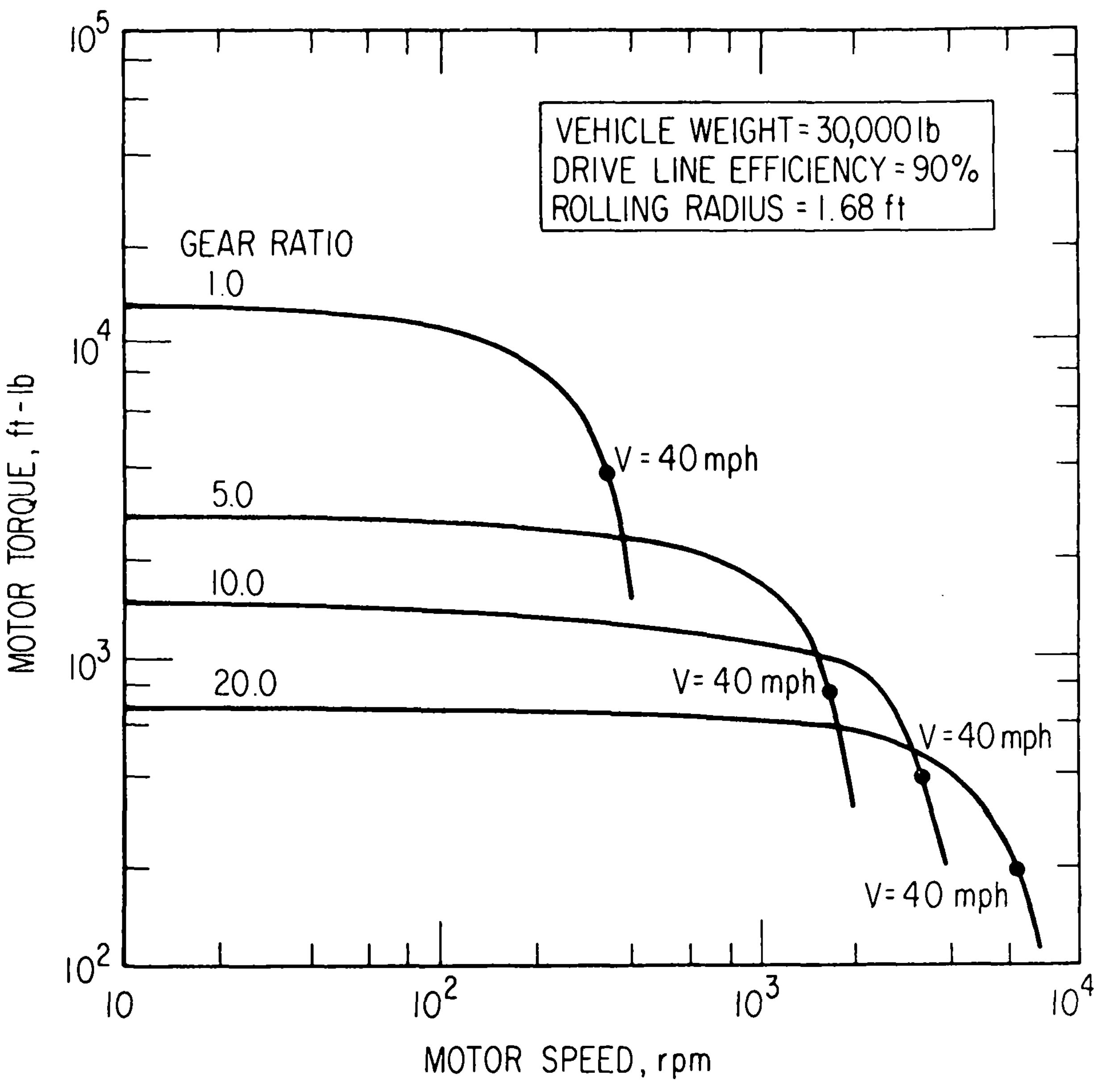

Figure 5-18. Electric Motor Torque Requirements for Low-speed Intracity Bus

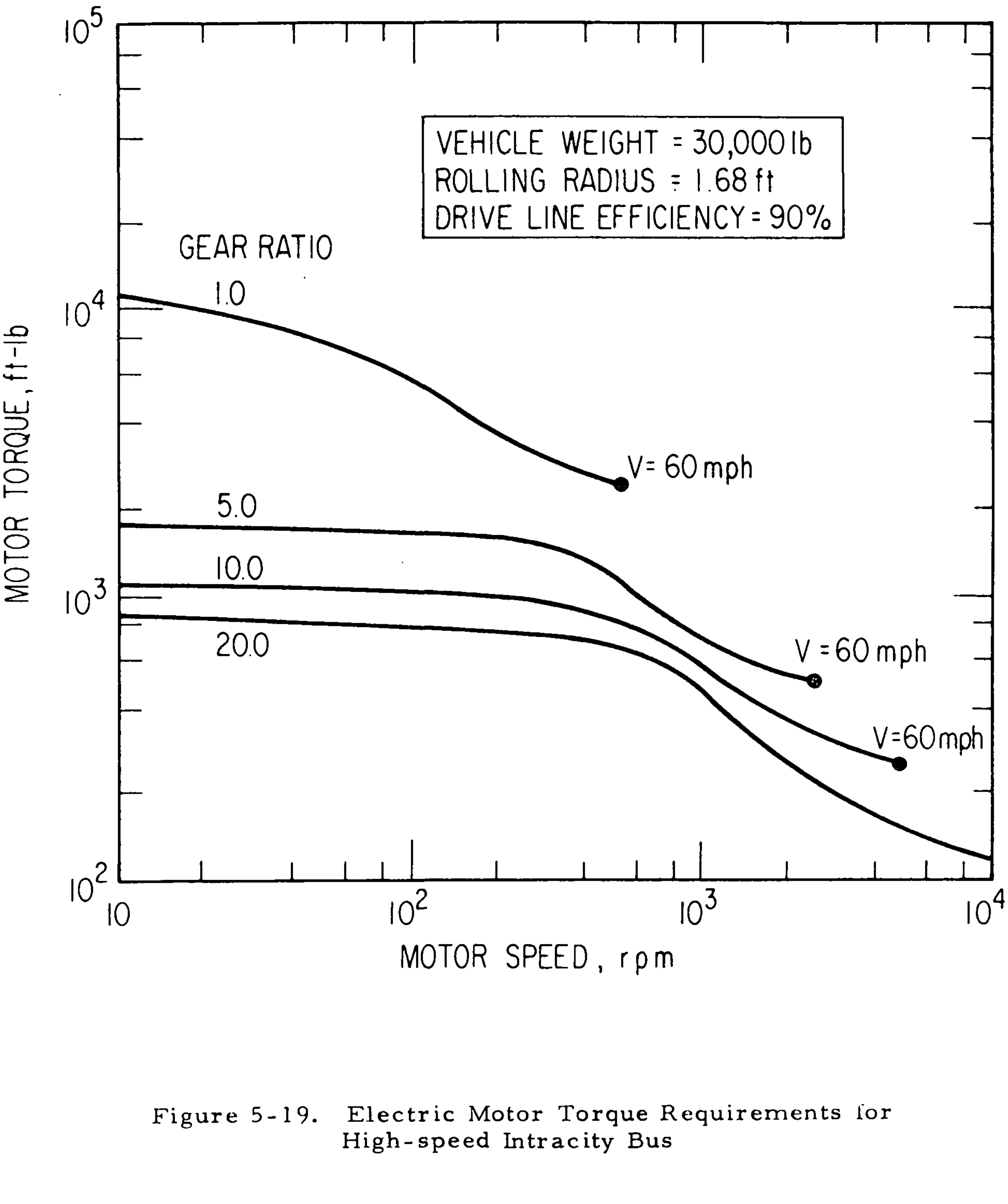

Figure 5-19. Electric Motor Torque Requirements for High-speed Intracity Bus

Adoption of the second criterion means that both buses can be compared over the same route, i.e., distance between stops is the same for both cycles. The two driving cycles are compared in Table 5-7.

Table 5-7. Comparison of Intracity Bus Emission Cycles

Low-speed Bus			High-speed Bus		
Velocity Mode, mph	Time, sec	Distance, ft	Velocity Mode, mph	Time, sec	Distance, ft
0 - 19.2	6.0	99	0 - 16.2	8.0	99
19.2	4.0	112	16.2	4.7	112
19.2 - 25.6	2.0	71	16.2 - 20	3.0	71
25.6	3.0	112	20	3.8	112
25.6 - 0	8.0	132	20 - 0	9.0	132
0	13.0	0	0	13.0	0
Total	36	526		41.5	526

Effective Speed = 10 mph | Effective Speed = 8.6 mph

Distance Between Stops = 0.1 mile | Distance Between Stops = 0.1 mile

Braking time for the high-speed bus cycle is based on a constant braking rate of 2.2 mph/sec.

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SECTION 6

ELECTRICAL SYSTEM - MOTOR, GENERATOR, AND CONTROL SYSTEM

CONTENTS

TABLES

FIGURES

SECTION 6

ELECTRICAL SYSTEM - MOTOR, GENERATOR, AND CONTROL SYSTEM

6.1 INTRODUCTION

The electrical system is composed of the electric traction motor, generator, control system, and batteries. Batteries are discussed in Section 7, but the remainder of the items will be treated here. First, the electrical system parameters or characteristics that have the greatest impact on the total system are considered. Next, details of the advantages and disadvantages of various approaches are summarized, and, finally, development efforts are recommended.

6.2 SYSTEMS SYNTHESIS

There are many different design approaches to the development of an electrical system for the hybrid vehicle. The series versus the parallel power-train configuration is a major division of the concepts. One form of the series configuration is shown in Fig. 6-1, in which all of the energy of the heat engine flows through the generator and the motor to the wheels. Part of the energy used for peak power requirements flows through the battery. The battery is then recharged during cruise and its energy utilized for starting and acceleration. In Figs. 6-2, 6-3, and 6-4, three different design approaches are shown for the parallel configuration. The first two have been built and tested with varying degrees of success. In addition, the electric motor, battery, and control system portions of the third approach have been built and tested in a prototype wheelchair propulsion system.

Figure 6-2 is a block diagram similar to the TRW parallel configuration. The power from the heat engine is transferred to a planetary differential gearing arrangement, which transmits a portion of the energy directly to the wheels. The remainder, not required for propulsion, is converted to electrical energy in the generator and stored in the battery. During periods of

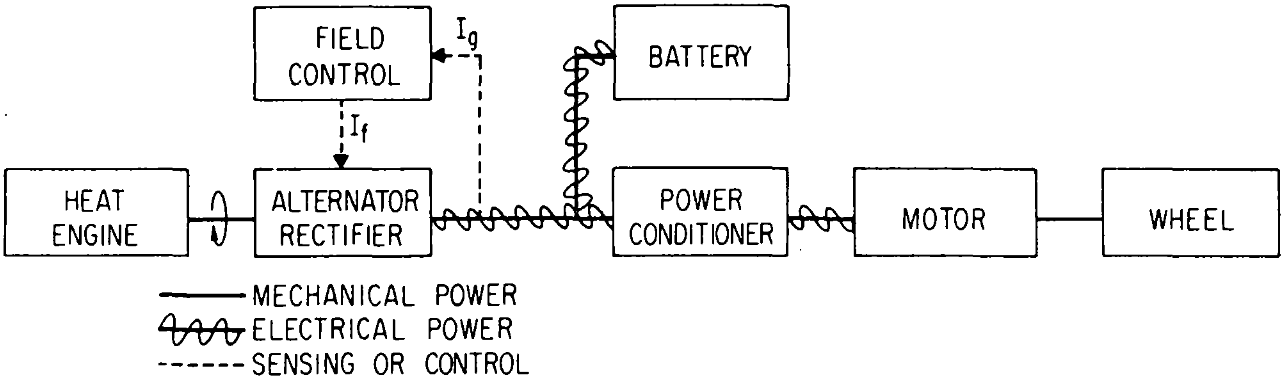

Figure 6-1. Electrical Control Schematic, Series Configuration

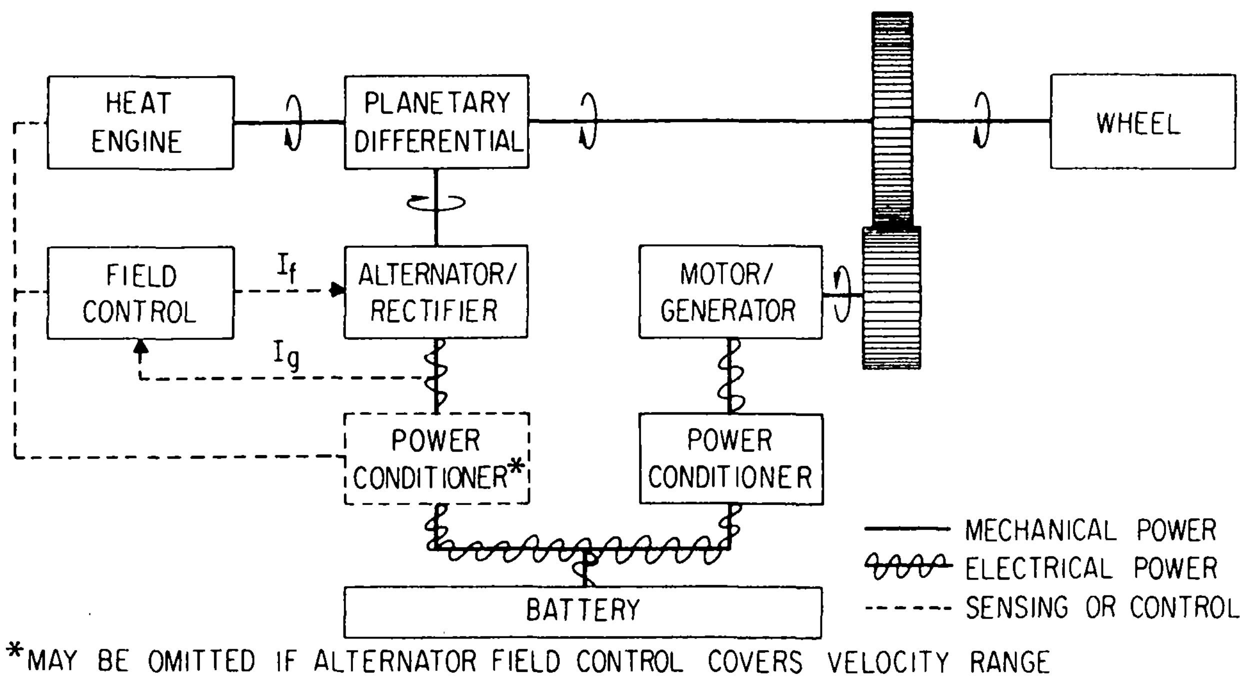

Figure 6-2. Single Motor Parallel Configuration

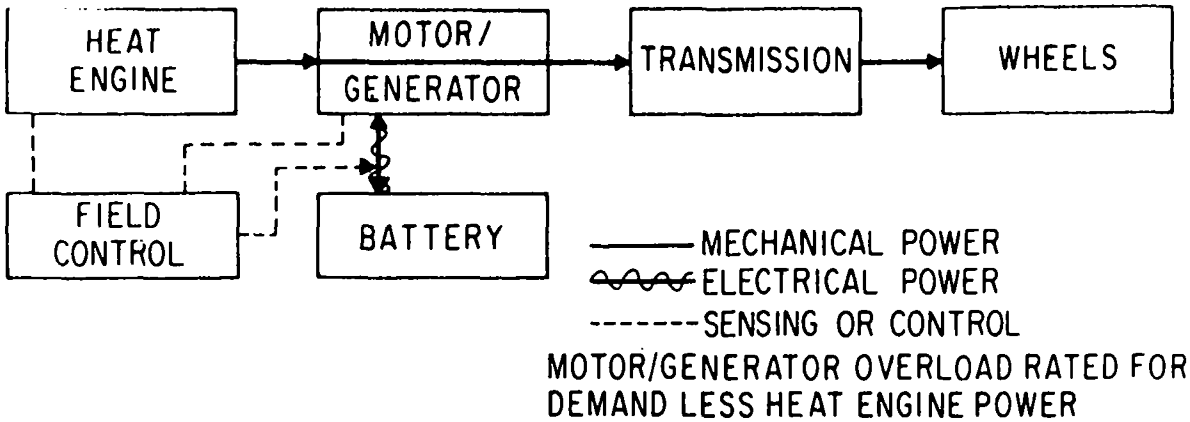

Figure 6-3. Single Motor Parallel Configuration Concept - Variable Velocity Heat Engine with In-Line Augmenting Electric Motor/Generator

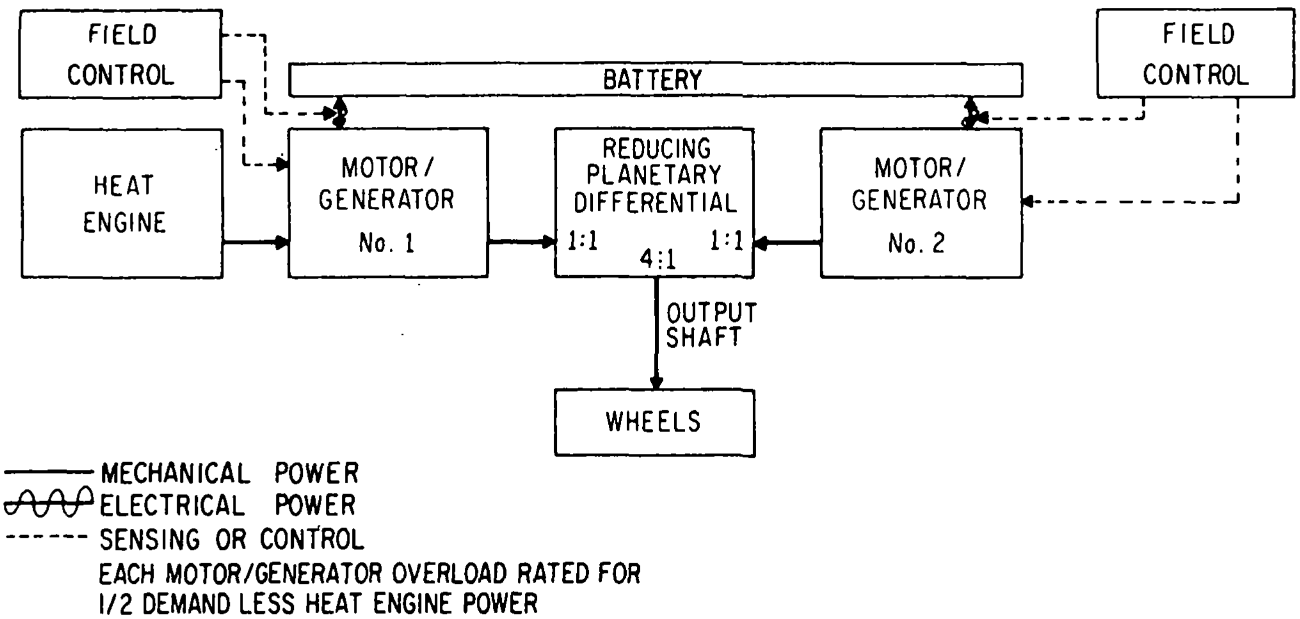

Figure 6-4. Dual-Motor Parallel Configuration Concept

start and acceleration, power is drawn from the battery for the motor/generator to help the heat engine drive the wheels. During deceleration, the motor/generator becomes a generator and feeds energy back into the battery. At low vehicle speeds, the heat engine operates at very nearly constant power and speed. For more details, see Ref. 6-3.

In Fig. 6-3 (Refs. 6-1 and 6-2), the Minicar approach to a parallel system design is shown. It also has a direct drive to the wheels but not through a differential. The motor/generator is mounted on the same shaft as the heat engine, with its rotor a part of the drive shaft. During periods of start and acceleration, the motor/generator acts as a motor and assists the heat engine in driving the wheels. During cruising and deceleration, the motor/generator operates as a generator and feeds energy back into the battery. Gear shifting is required and the heat engine must operate at variable speed and power output.

Figure 6-4 is a dual motor concept that has interesting possibilities. One form of this system was developed by Electric Motion Control Corporation as a completely electrical drive for wheelchairs, fork lift trucks, golf carts, etc. Portions of the system are proprietary pending patent agreements, so it cannot be discussed in complete detail. However, suffice to say that each motor/generator can and does help drive the wheels under heavy loading situations, such as start and acceleration. Electric Motion Control Corporation states that the control system complexity is considerably reduced when compared to other parallel configurations.

In summary, it can be said that the electric motors can be smaller in the parallel configuration than in the series configuration, due to the fact that they are used only during peak loading situations. The parallel arrangement appears to have the potential of greater efficiency because a large portion of the energy does not flow through the lossy electrical system but is channeled directly to the wheels. The principal loss is friction in the mechanical system, and the electrical loss is reduced below the series configuration of Fig. 6-1.

Another major division in design concepts is the AC versus DC motor approach. The AC motors are smaller, lighter, and easier to cool, but they require a variable-frequency power supply that must be derived from DC power if a storage battery is to be used in the system. If an AC generator is used in the system, regardless of the type of motor used, rectification of this power source is necessary for recharging the battery.

The AC power can be made available by passing battery power through an inverter. Either induction or synchronous motors can be used, and being considerably smaller than equal-power-output DC machines, they are more easily adaptable to mounting as a part of the wheel assembly.

If DC motors are utilized, a decision must be made relative to the field configurations, that is, the manner of separately exciting the main field current and the relative benefits derived from compensating and interpole windings. The operating characteristics must be analyzed to determine specific design details affecting overall efficiency, weight, size, complexity, cost, development-status, etc. Low-weight components are important, since the power required to propel the vehicle depends on its weight.

The selection of system voltage is primarily based on the weights of the electrical components. Higher voltages result in lower weights for distribution wiring, motors, generators, and controls. For the commuter and family cars, a 220-V system was selected to limit currents so that they do not exceed 500 amp. For the bus and van, a 440-V system was recommended to supply higher power requirements and still not exceed 500 amp. The voltages and currents are based on the case of the series powertrain configuration where the motor provides total power to the wheels. The motor under consideration has active independent field control for meeting power requirements over the entire vehicle speed range. In the parallel powertrain configuration, requirements are somewhat lower since this motor will deliver only acceleration power.

6.3 SUBSYSTEM TECHNOLOGY

6.3.1 Motor Characteristics and Control

Designs for electric motor drive systems should have the following goals for performance characteristics:

a. High starting torque
b. Sufficient accelerating torques over the specified speed range
c. High overall operating efficiency
d. Simple, inexpensive speed control
e. Simple, inexpensive, efficient regenerative braking

The most common approach to the design of electrically propelled vehicles in the United States has been to use DC series motors utilizing chopper circuits for their control, either pulse frequency or pulse duration. Although this approach appears reasonable for some classes of vehicles and driving cycles, it is not optimum for all types. Hence, this section will be devoted to an analysis of the motor characteristics for a number of different design approaches that could be used for the hybrid vehicle electric propulsion system.

The motor-induced voltage varies with speed; it is very low at zero speed and increases as the motor speed increases. Exceeding this voltage results in high currents leading to overheating of the motor. The armature applied voltage must be varied to match the induced voltage of the motor at all speeds. This can be achieved in several ways:

1. Chopper circuit
2. Variable resistance in armature circuit
3. Step voltage change and field control

The chopper circuit provides an efficient means for transforming a fixed battery voltage to a smoothly variable effective voltage matching the requirement of the motor at all speeds of operation and providing a smoothly variable speed. Also, while the chopper sees a varying impedance from the motor (depending on motor speed), it presents a relatively constant high impedance to the battery when used with proper filtering elements. This allows the reduction of high current pulses in the battery. The main disadvantage of the chopper is the high cost of the power switching components and the associated control circuitry. Also, compared to pure DC control, the chopper introduces losses due to high frequency operation. This can be partially reduced by special motor design and adequate filtering.

The use of variable resistance in the armature circuit introduces high losses associated with the voltage drop in the resistance, and is an inefficient method of voltage control for a vehicle required to operate over a wide speed range.

Motor voltage control can also be achieved by step voltage switching of the battery cell groups from parallel to series as the vehicle speed is increased. However, to provide adequate voltage matching over a wide speed range several stages of voltage switching are required to obtain reasonable motor efficiency and avoid excessive loading of the battery. The number of switching steps may be reduced by combining field control with voltage switching. This method is schematically illustrated in Fig. 6-5 showing how voltage switching is accomplished by speed sensing and relays (Ref. 6-10).

A comparison of operational characteristics utilizing the three types of controllers are given in Table 6-1. Additional comparisons can be found in Ref. 6-11 which discusses actual field testing of electric cars.

The general motor equation used for the transformation of electrical to mechanical power in the computer study is

$$K = IV/\tau N$$

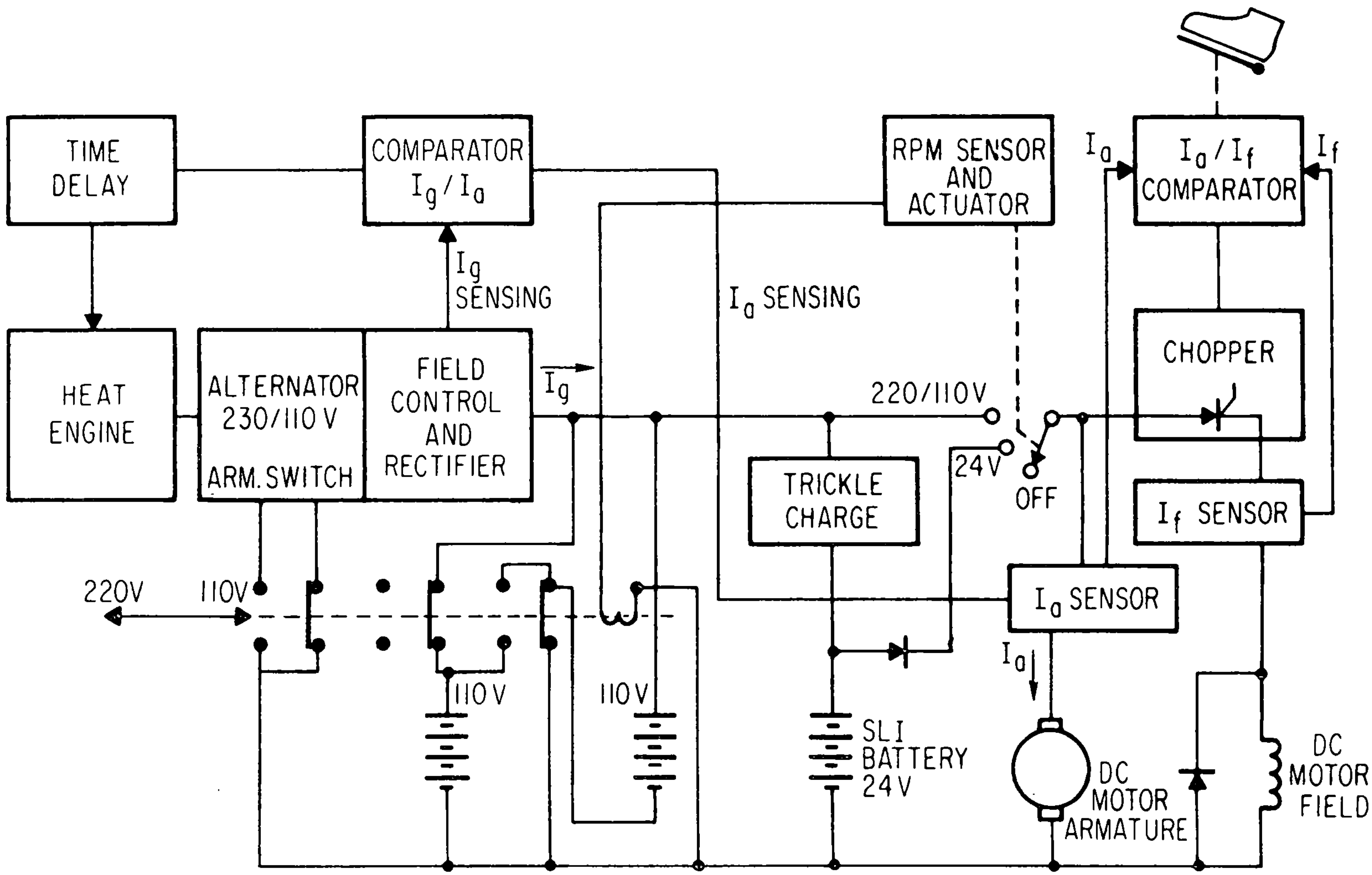

Figure 6-5. Separately Excited DC Motor with Step Voltage Control

Table 6-1. Comparison of Motor Controllers

Item	DC Chopper	Variable Resistance	Step Voltage with Field Control
Types of Motor Controlled	All DC brush motors	All DC brush motors	Only separately excited, stabilized or compound wound
Velocity Range	Zero to maximum speed	Start only	Wide with three steps or more
Smoothness of Velocity Change	Very smooth	Jumpy	Initial jump 0-5 mph then smooth
Controller Protection	Solid state only - circuit breakers and fuses too slow	Circuit breakers and fuses sufficient	Circuit breakers and fuses
Controller Cost (1975)	High	Low	Medium
Controller Efficiency	Medium	Very low	High with controller logic
Special Sensors and Control Logic	Complex	Simple	Complex
External Smoothing Filter	Heavy filter required	Not required	Not required
Starting Torque	High but inefficient	Medium and very inefficient	High with inefficient over-excitation
Velocity Stability	Stable with shunt motor, decreasing with load on series motor	Somewhat unstable varying with load	Stable up to torque limit
Torque at High Speed	High	Low	Medium-limited by field weakening ratio

Table 6-1. Comparison of Motor Controllers (Continued)

Item	DC Chopper	Variable Resistance	Step Voltage with Field Control
Power Conditioning Characteristics	Modulation of full power used by motor	High switching currents with much dissipation	With small signal field control, high contactor currents at switch closing but zero contactor currents on switch opening*

*Before a change of armature voltage takes place, the field is momentarily increased to the point where armature current reaches zero. The feedback from the current sensor then allows the armature relay to open. The usual problem of interrupting DC current is thus avoided.

where

K = constant $\cong 0.142$

I = amperes

V = volts

τ = torque in lb-ft

N = rpm

This equation is applicable to all types of motors, both AC and DC, but does not include the efficiency of transformation of electrical into mechanical power. Efficiency was included in the computer program as a separate item.

6.3.1.1 Near Term Motor Application - 1972 to 1975

6.3.1.1.1 DC Series Motor

For this application, the DC series field motor with chopper control is the most highly developed approach (Fig. 6-6) attained in this country (Refs. 6-3 through 6-9). It has the advantage of a high starting torque and smooth power control.

The main disadvantage of the chopper control is the present high cost of SCR's. These costs have been coming down in recent years, however, and can be expected to continue to do so. A cost-versus-current rating for 300, 600, and 1000 V SCR's, given in 1970 catalogues for quantities up to 1000 is shown in Fig. 6-7.

Since starting current surges can be several times larger than the normal cruising power, rather large current SCR ratings are required. The current and voltage ratings of SCR's (allowing for safety factors) are on the order of 1000 amp at 1000 V for the bus, and 500 amp at 500 V for the family car vehicles. Thus, the present catalogue cost of the power SCR's is approximately $50 per SCR for the smaller vehicles, and in the neighborhood of $200 each for the larger vehicles in quantities of 1000.

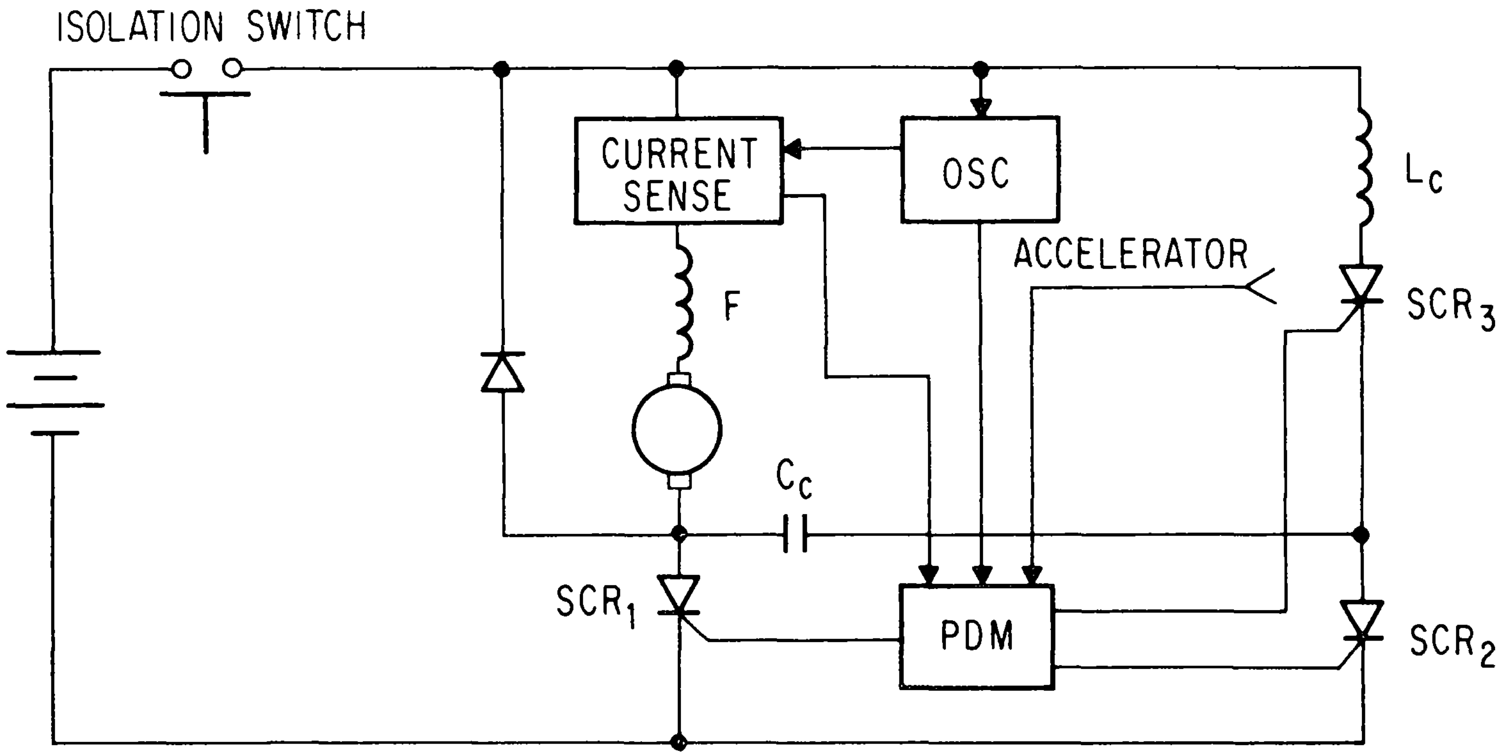

Figure 6-6. Series Motor Controller

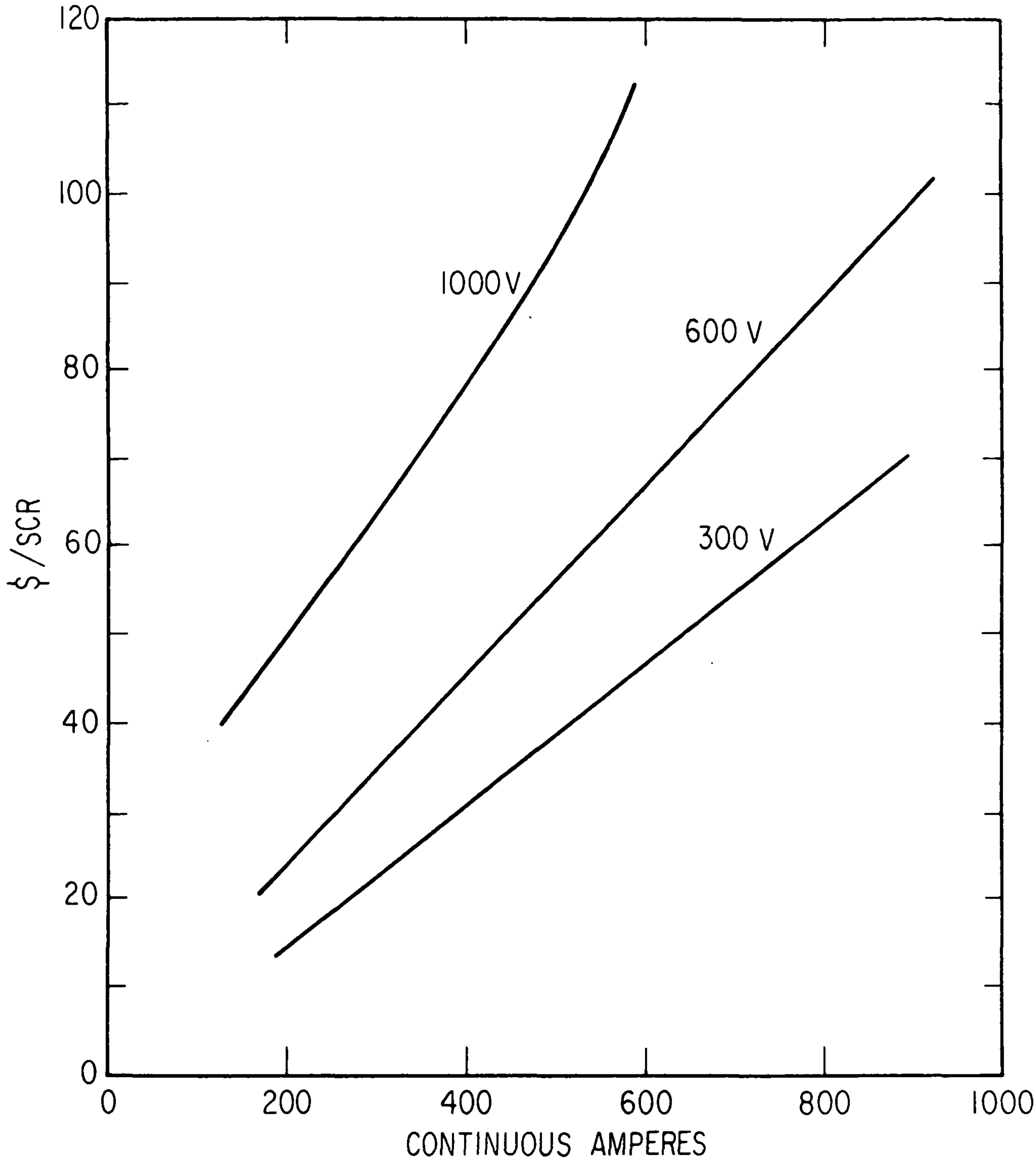

Figure 6-7. Cost of High Current Rating SCRs 1970 Catalogue Prices in Quantities of 1000

6.3.1.1.2 Separately Excited DC Motors

Separately excited field control (sometimes called shunt field control) combined with voltage switching for DC motors was investigated and appears to have considerable advantage from the standpoint of cost. Figure 6-8 is a diagram of this approach. The starting current can be limited to safe values by the application of low voltage steps for starting.

As normally used, a shunt field DC motor also has a few shortcomings, the principal ones being low starting torque and limited speed control range. It has many advantages, however, such as good accelerating torque at high speed and high operating efficiency (see Fig. 6-9).

A scheme where the advantages of both the series and shunt motor can be realized is shown in Fig. 6-10. In this scheme the armature is energized from a chopper and the field is separately excited through a field controller. The armature and field current are independent, thus, by proper control of the field current, the high torque characteristics of the series motor at low speeds can be combined with the high efficiency of the shunt motor at high speeds. This scheme offers the following advantages and disadvantages:

1. During the generating mode the induced voltage can be varied to match the battery voltage (except at very low speeds) so as to improve the efficiency of charging the battery and increase the effectiveness of electrical braking.

2. Compared to the series motor, the field current is low thus minimizing the switching problems involved in reversing the direction of motor torque.

3. It has a more complex and costly control system when compared to the chopper-controlled DC series motor or when compared to step voltage with field control of the separately excited motor.

4. A larger size external series choke is required compared to other methods of control.

Figure 6-11 illustrates the relationship of torque to speed for various values of field excitation β, which is defined as the ratio of armature current I_a to field current I_f (Ref. 6-10). There is an optimum value of β that will produce

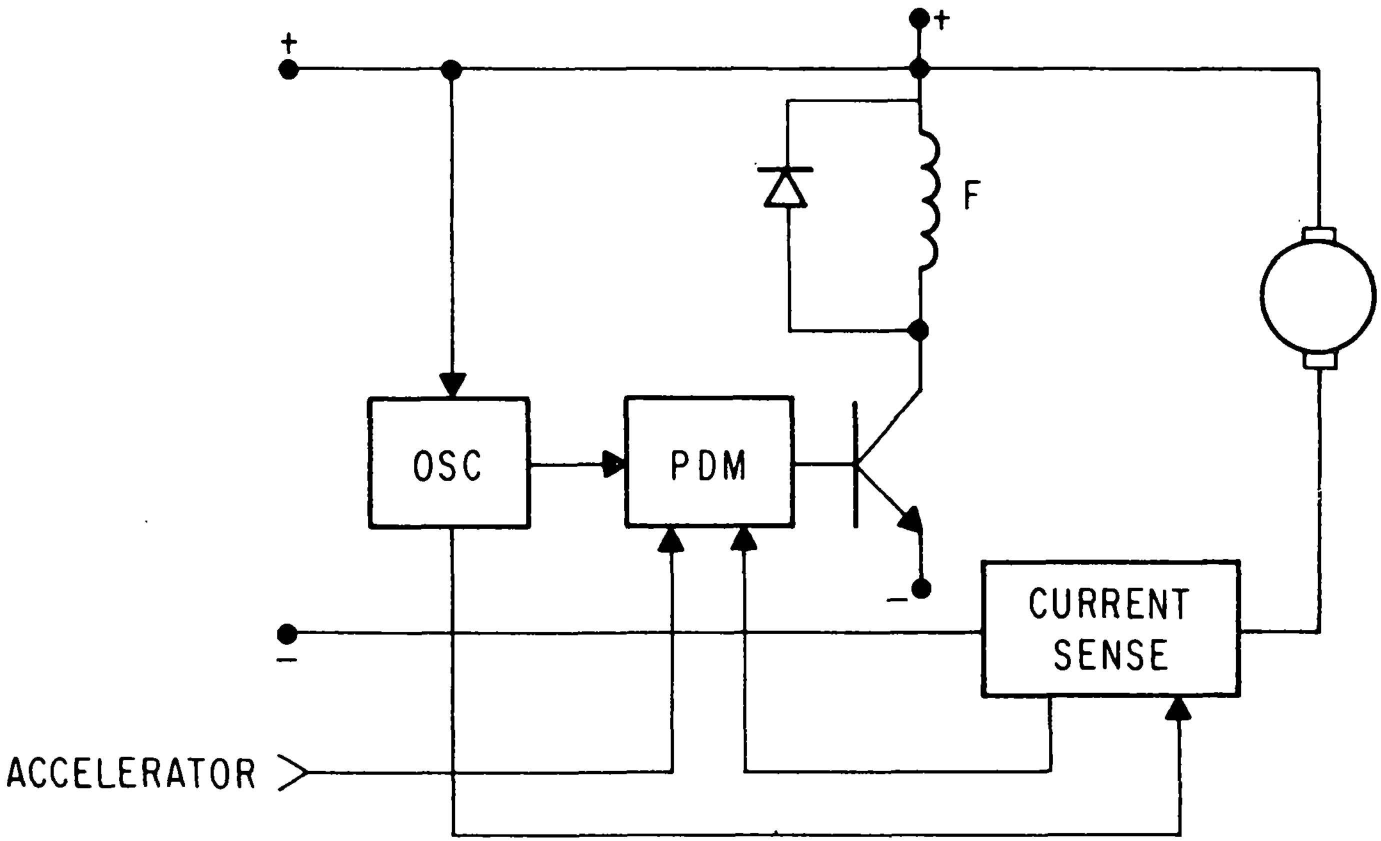

Figure 6-8. Separately Excited Field Motor Controller

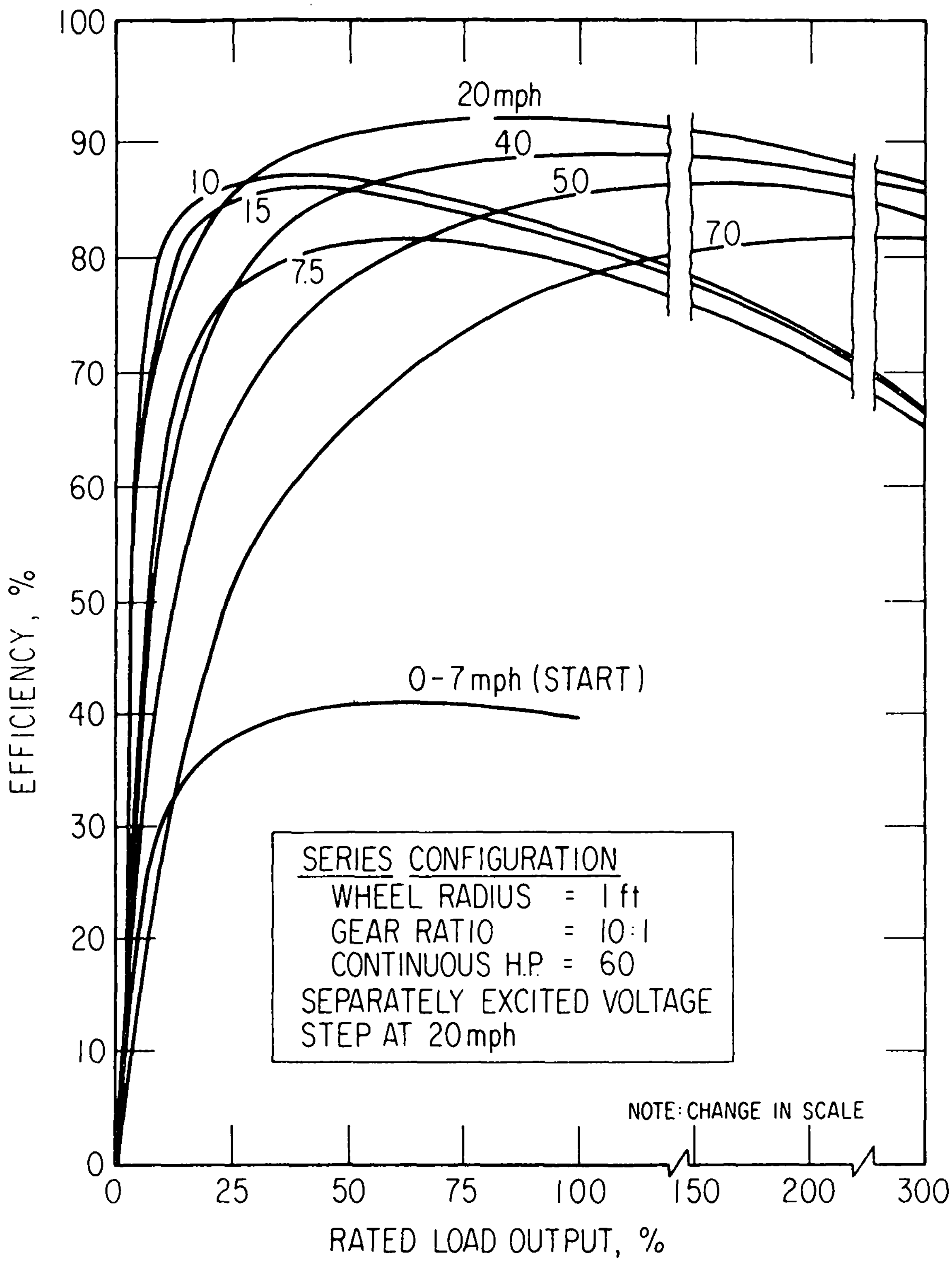

Figure 6-9. Typical DC Motor Efficiency

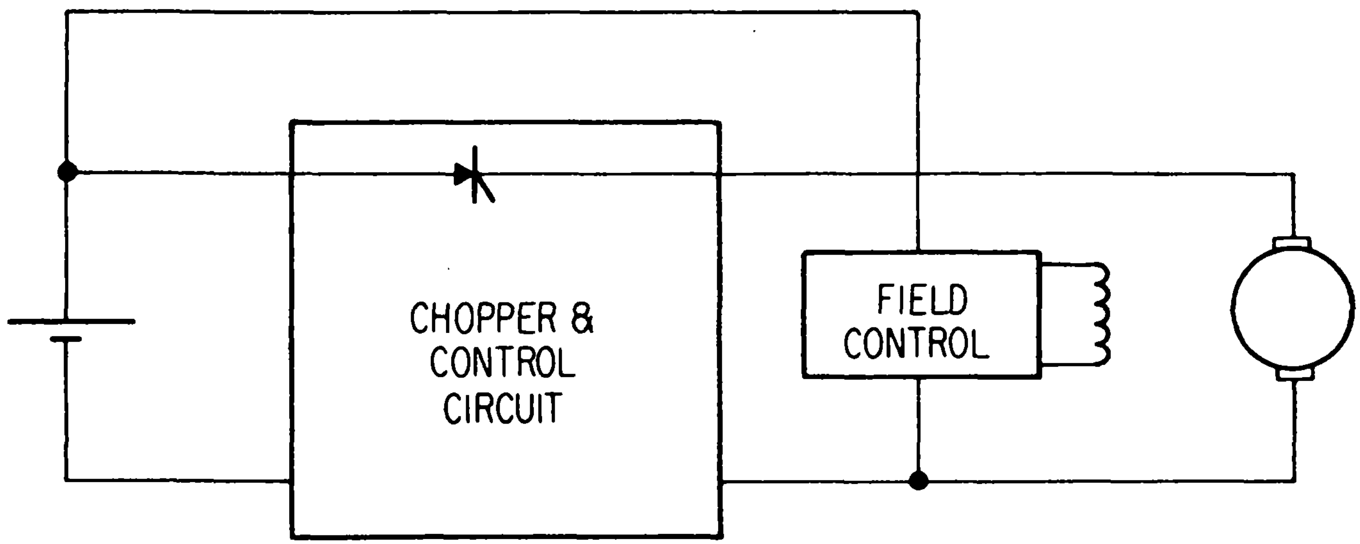

Figure 6-10. Separately Excited Motor Controlled by
Field Control and Chopper

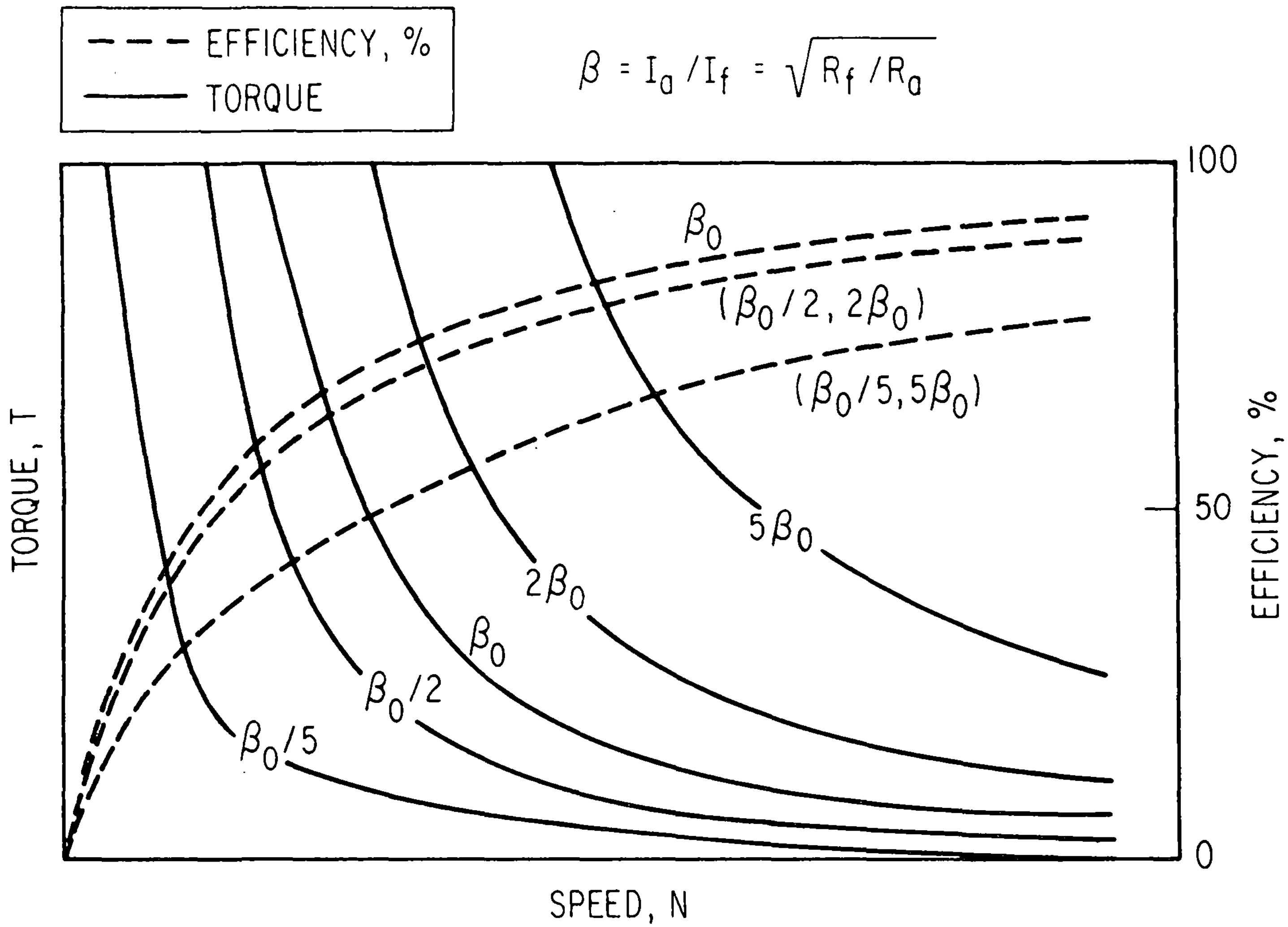

Figure 6-11. Torque and Efficiency Characteristics of Separately Excited DC Motor (from Ref. 6-10).

the maximum electrical efficiency of a separately excited machine at any speed. This optimum value β_0 can be expressed as

$$\beta_0 = I_a/I_f = \sqrt{R_f/R_a}$$

Also, one finds that

$$\tau = (K/2\pi)I_a I_f = (K/2\pi\beta)I_a^2$$

where

τ = torque	R_a = armature resistance
K = constant	R_f = field resistance

and

$$E = KNI_f = KNI_a/\beta$$

where

E = induced armature voltage

N = rpm

Now it is seen that

$$V_B - E = I_a R_a$$

and

$$\tau = (V_B^2 \beta)/(2\pi K N^2)$$ (this is an approximation where the $I_a R_a$ product is neglected)

where

V_B = applied voltage

From the above equation, it is evident that torque and speed can be controlled by varying β (field excitation) and/or V_B. The curves in Fig. 6-11 and the above equations neglect the effect of friction, windage, and core losses, which could be similar for series and separately excited machines. Figure 6-11 also shows efficiency versus speed for various values of β. Overexcitation reduces efficiency primarily due to the increased field losses; however, the increased excitation is required only for starting and low-speed operation. As speed increases, the excitation is reduced until, at rated speed, normal excitation is applied and normal β_0 efficiency is obtained. Figure 6-12 presents test data taken from a series motor and a separately excited motor. It shows that for $\beta = 1$ the machines are comparable. Hence, β can be increased to obtain better high-speed performance for the separately excited motor.

6.3.1.1.3 AC Induction Motor/Inverter

Recently, a great deal of development work has been done in the area of AC induction motors for the propulsion of vehicles by General Motors Research Laboratories and the U.S. Army Equipment Research and Development Center at Fort Belvoir, Virginia (Refs. 6-12, 6-13, and 6-17). The concept involves an alternator driven by a gas turbine or other engine, and a cycloconverter (variable-frequency) driving induction motors on the individual wheels. This concept was developed primarily for high-traction vehicles utilizing electric motors on each wheel. One General Motors passenger car concept uses an AC generator, a rectifier, a battery, and a variable-frequency inverter driving an AC induction motor. For larger vehicles, this approach has merit, primarily due to the small lightweight motor. The variable-frequency inverter is heavy, relatively large, and would be extremely complex and expensive, however. Induction motor weights on the order of 1.1 lb/hp have been achieved with oil-cooled motors, and variable frequency inverters have been built that weigh about 1.7 lb/peak hp using oil cooling (Ref. 6-17). This could be reduced with more effective cooling. Peak efficiency of the motor and the inverter system is approximately 85 percent. This corresponds

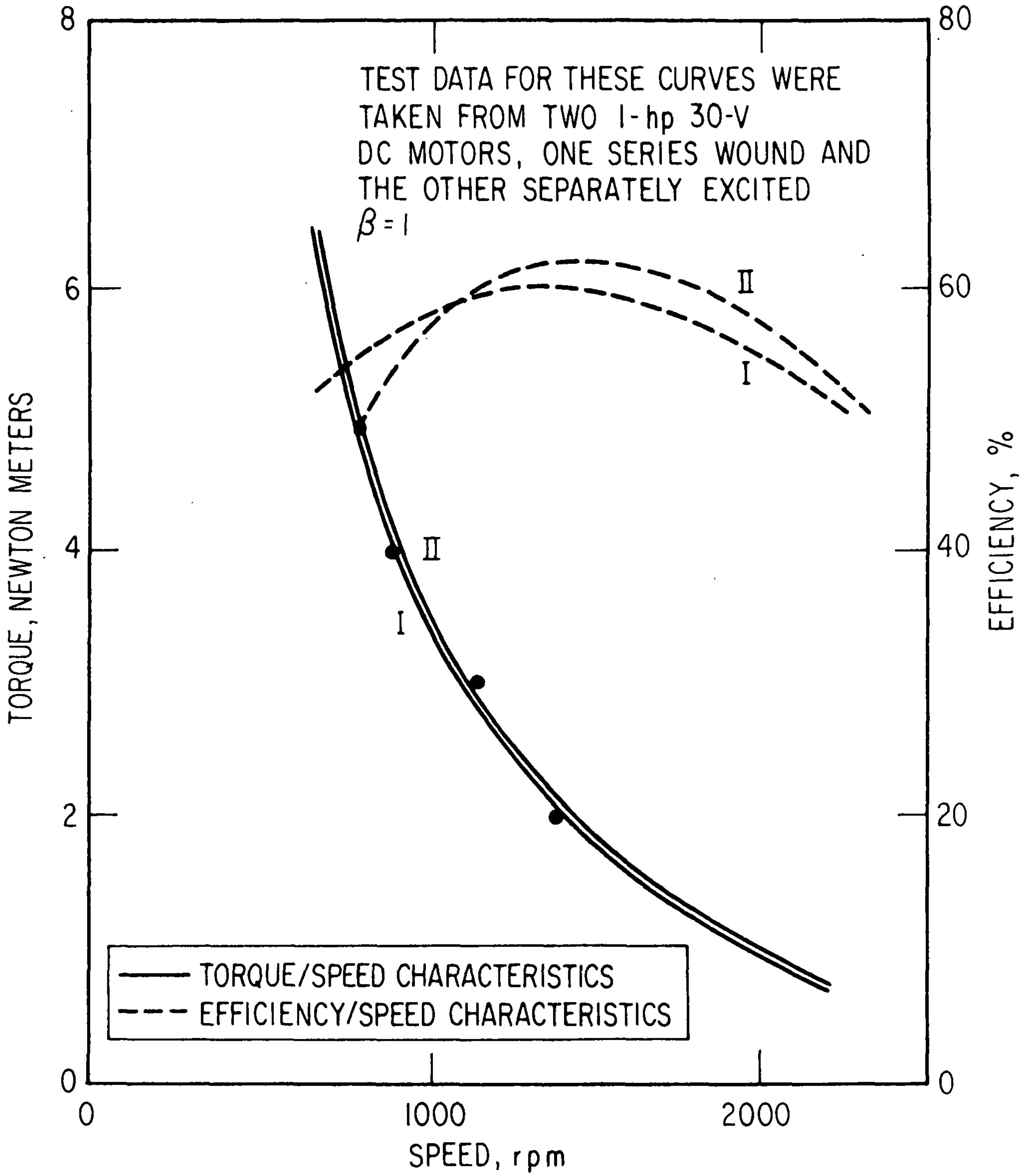

Figure 6-12. Performance Characteristics — Separately Excited DC Motor (I) and Series DC Motor (II) (see Ref. 6-10).

to an efficiency of 91 percent for the same motor driven by a sine wave (Ref. 6-18). If the weight of the oil-cooling system is added to it, the motor and inverter system still weighs less than a DC motor and control and is particularly advantageous for larger power requirements as shown in Fig. 6-13.

At present this method is more costly than previously mentioned approaches; however, it may be feasible for larger vehicles.

6.3.1.2 Long Term Motor Application - Beyond 1975

6.3.1.2.1 Synchronous Motors

Great advances have been made in the use of ceramic magnets for motor applications in recent years. Barium ferrite is used in place of Alnico as a permanent magnet material for DC as well as AC synchronous motors. This material is very cheap and economically magnetized, and the new types are not easily demagnetized. Permanent ceramic magnet synchronous AC motors are feasible for electric drives for vehicles, although they are expected to be heavier and more costly than induction motors of the same size. A variable-frequency inverter is also required as the basic power controller. This approach appears attractive for large high-traction vehicles.

6.3.1.2.2 Brushless DC Motors

Brushless DC motors have been built in small sizes at great cost for limited applications. They have been developed primarily for space vehicles where brush-type motors would not be suitable due to the limited life of the brushes in the vacuum environment. Additional development and cost reduction could make this type competitive with existing motors and provide the advantage of longer life (Ref. 6-14).

6.3.2 Generator Characteristics and Control

Both AC and DC generators have been highly developed for automotive and aircraft applications, and considerable data exist on expected performance except for those that operate at very high speeds, i.e., 50,000 rpm and

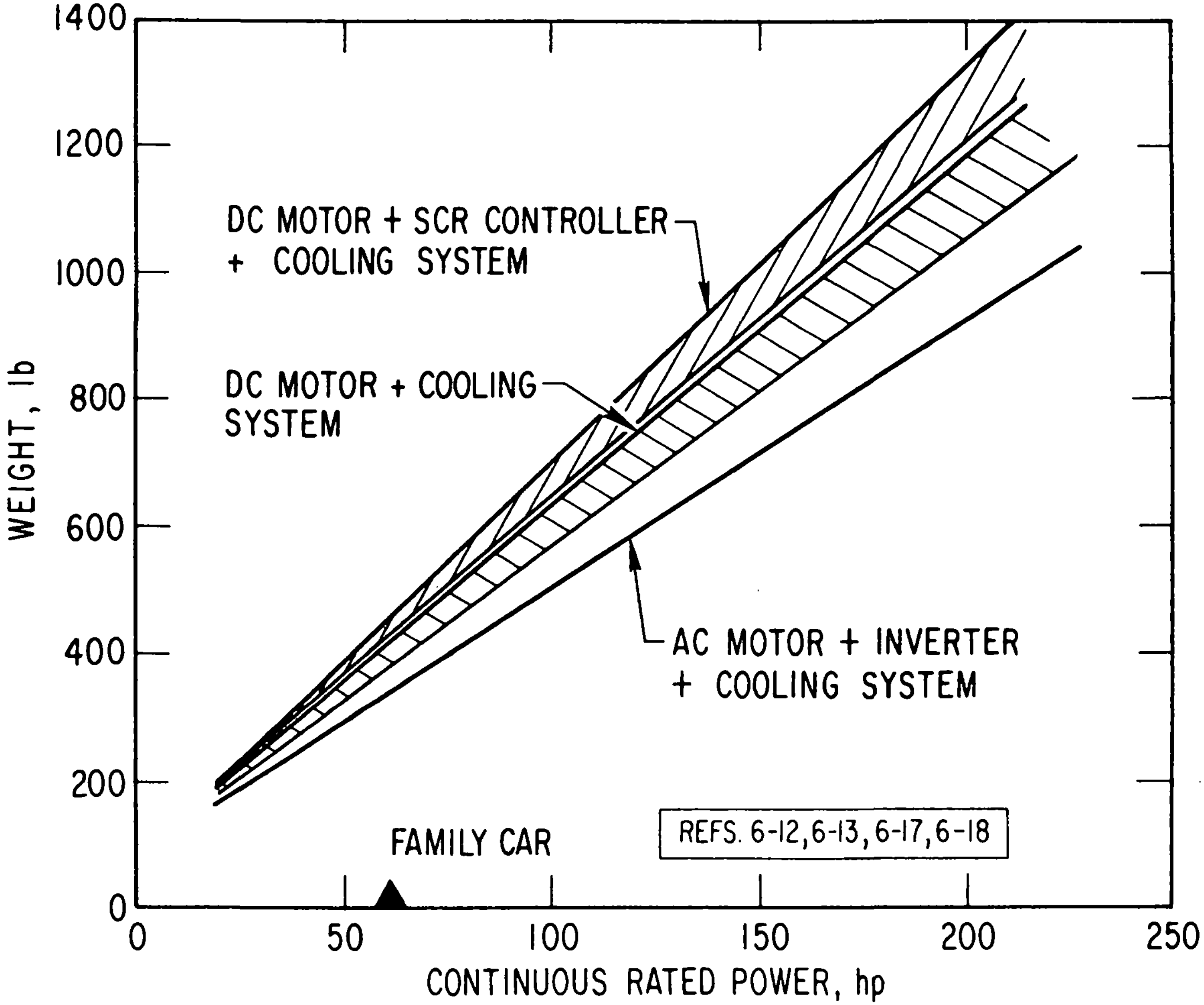

Figure 6-13. Weight Comparison for Electric Motors with Overload Capability

above (Refs. 6-15 and 6-16). The efficiency of DC generators for aircraft runs on the order of 80 percent at full load and rated rpm, and about 90 percent for AC generators of equal weight. For equal efficiency, the DC generator is heavier due to the added weight of commutation and, in some cases, interpole compensation. Figure 6-14 gives efficiency curves for AC and DC machines. The numbers are nominal values and increased efficiencies can be obtained by adding weight (iron and copper) to reduce the losses. Power rectifiers are required if AC generators are used; however, the peak efficiency of these rectifiers is very high, exceeding 99 percent, and their weight and size are very small.

Of great interest for the automotive application is the variation of efficiency with the load and speed of the machine. Since the generator on a hybrid vehicle operates at part-load for a large part of the time, part-load efficiency is very important. The results of calculations of efficiency are presented in Fig. 6-15.

Lastly, a weight comparison of AC and DC generators is shown in Fig. 6-16. Overall, it is clear that the AC generator has distinct advantages over the DC from the standpoint of efficiency and weight (and volume). The use of AC generators is recommended for all classes of vehicles.

The problem of controlling the generator was analyzed and it was found that field excitation controlled by a switching transistor circuit would be the least costly, simplest, and most efficient method available today. Efficiencies of 99 percent or better can be achieved when controller losses are compared to generator output power. The reason is that in controlling the field, only a small portion of the generator output power goes through the regulator circuitry. Figure 6-17 is a block diagram of a controller that could be used for all sizes of vehicles.

6.4 SUBSYSTEM EVALUATION AND COMPARISON

6.4.1 Electric Drive Motor Systems

A comparison of electric motor systems must include:

a. Operating characteristics and suitability to demand requirements

b. Control system complexity and cost

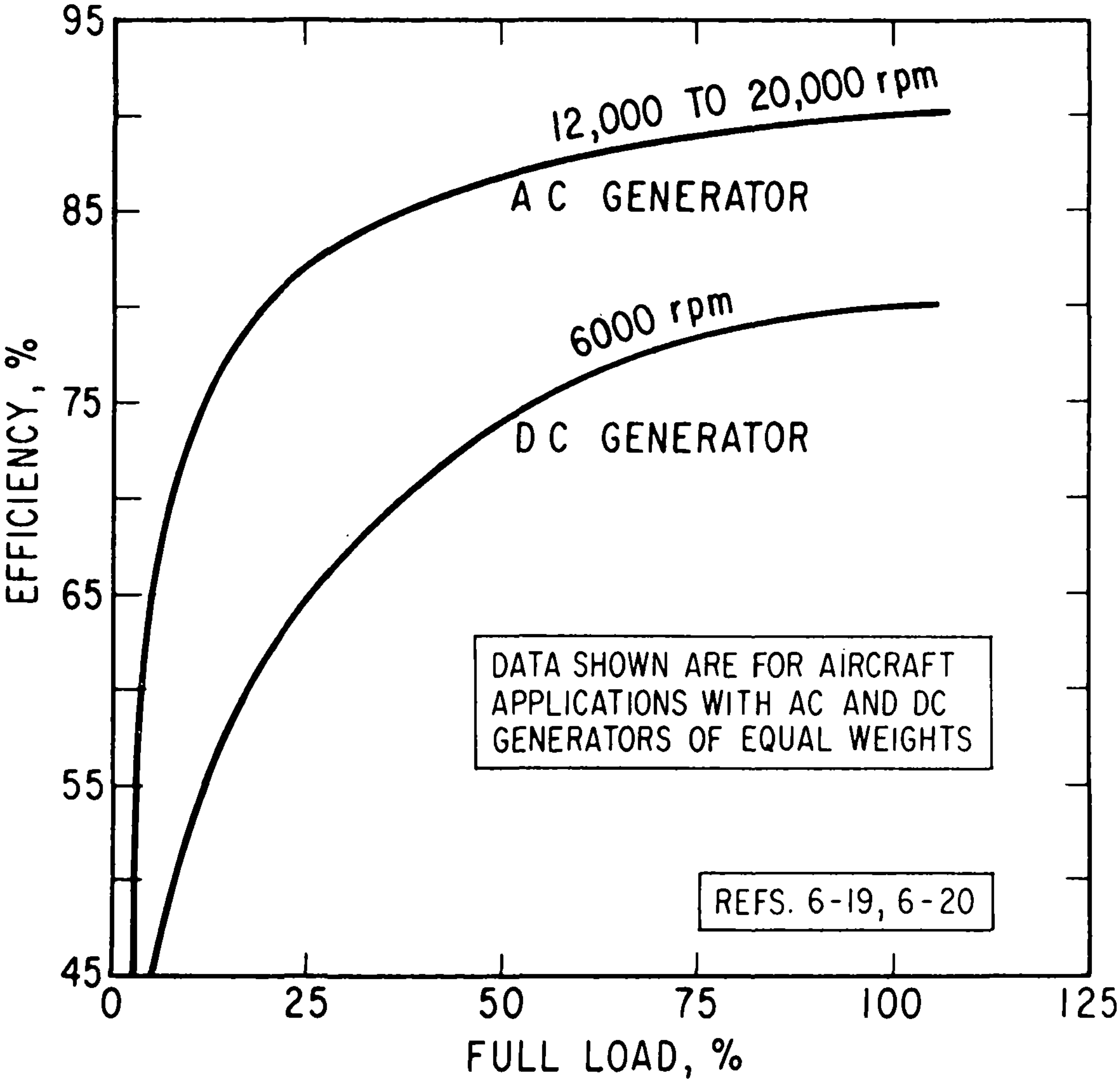

Figure 6-14. Generator Efficiency

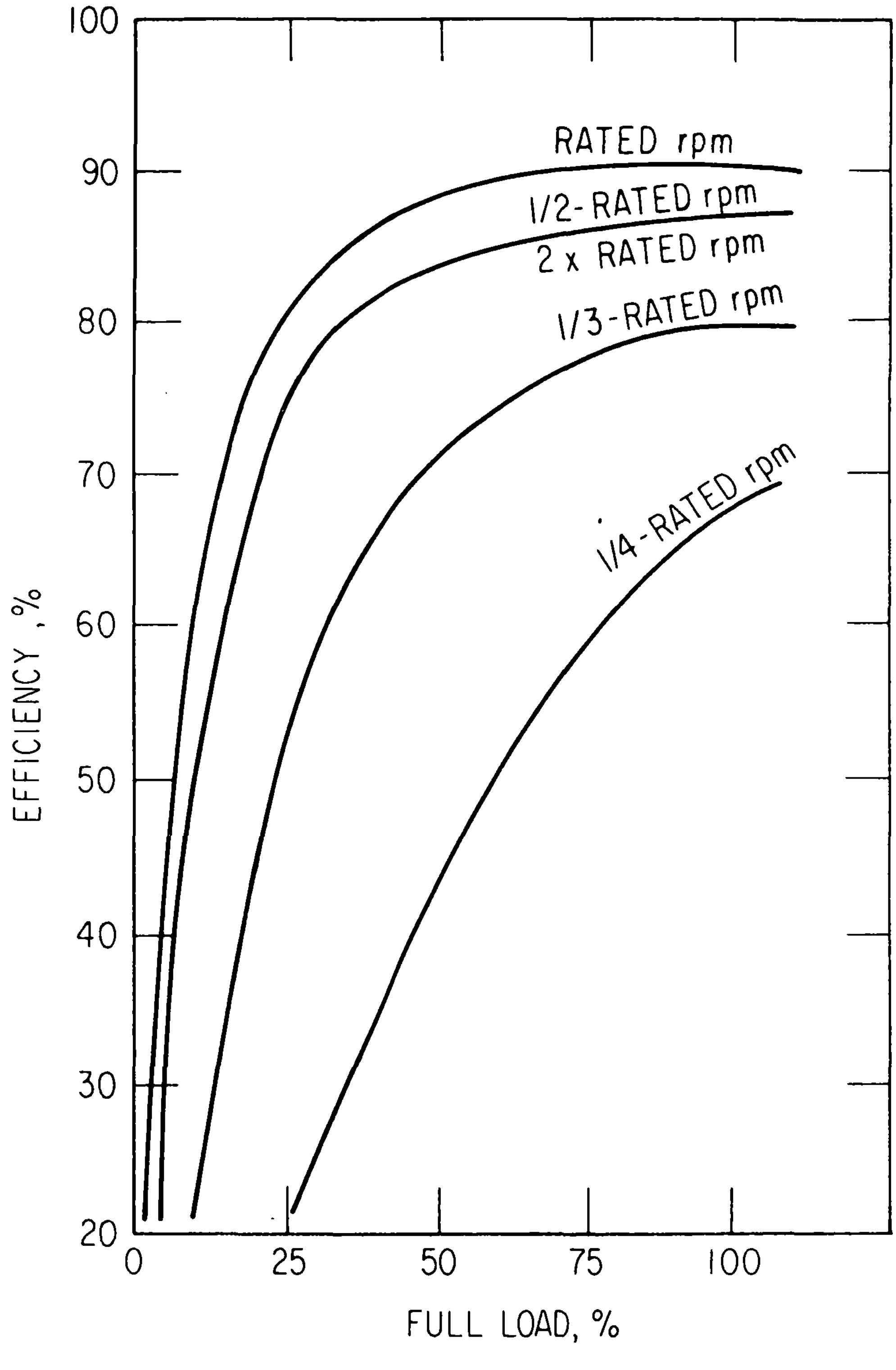

Figure 6-15. Generator Efficiency, AC
(Calculated Data)

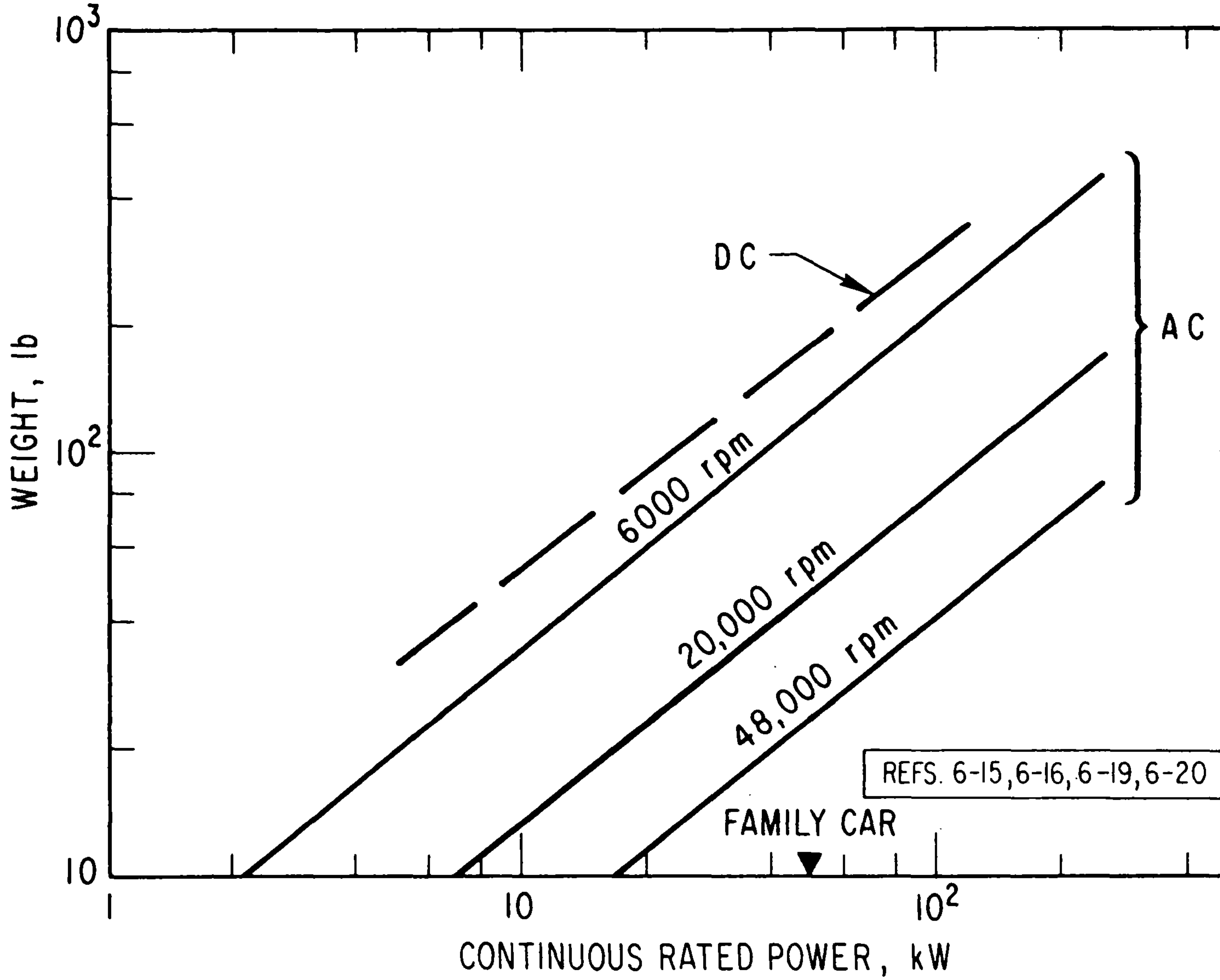

Figure 6-16. Weight Comparison for Electric Generators
Not Designed for Overload

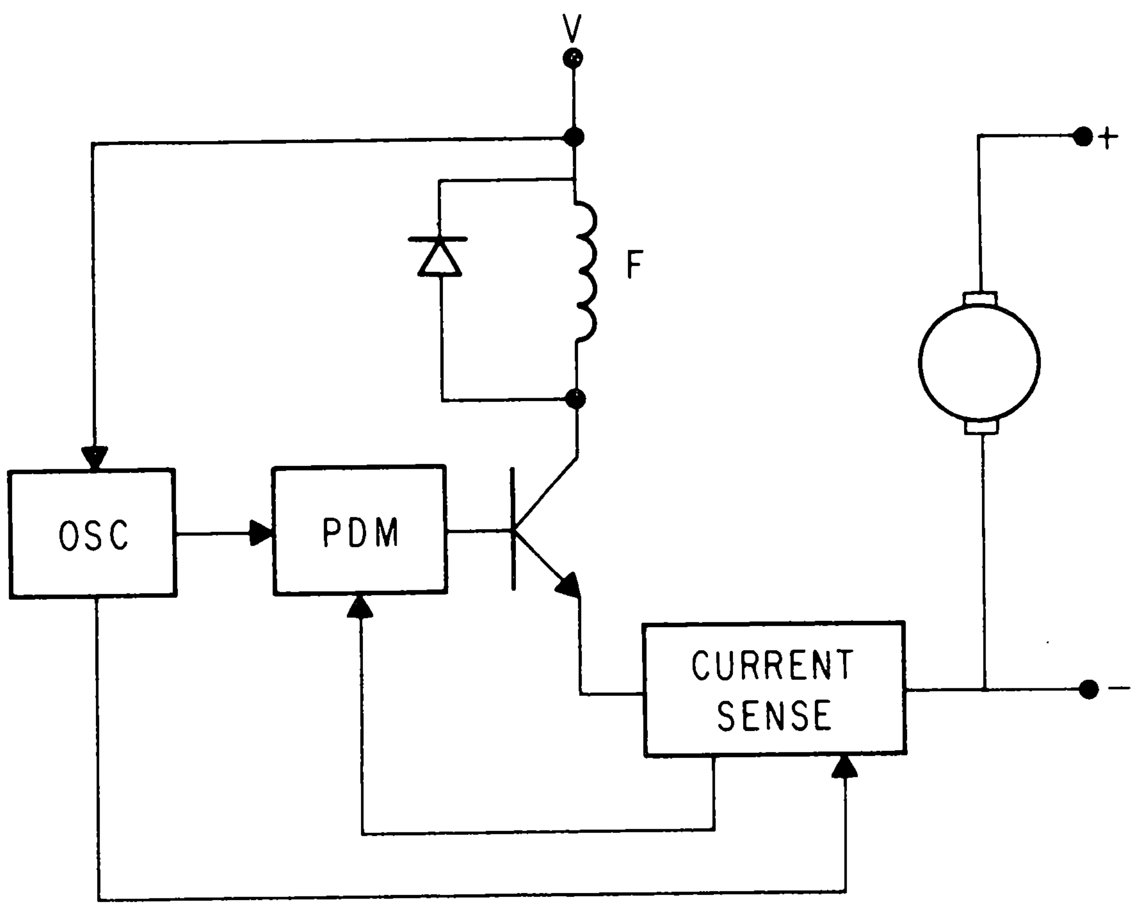

Figure 6-17. Generator Controller

c. Operating limits including:

   1. Surge currents
   2. Commutating current limit
   3. Temperature rise limit
   4. Velocity limit set by centripetal strength and commutation speed

d. Power density (lb/hp) and efficiency

e. Motor cost and availability

f. System weight

g. Reliability and maintainability

Series, separately excited, compound and brushless DC motors as well as AC induction types were evaluated on the basis of the above mentioned parameters. Detailed explanations follow.

6.4.1.1 Operating Characteristics Compared

The series-wound DC motor is basically a torque demand system with velocity as a function of both applied voltage and load (torque applied). At a given voltage, a change in velocity will occur with a change of load. At a constant load, the velocity will vary approximately in proportion to the applied voltage. Its use is principally in applications requiring high starting torque.

The separately excited DC motor (shunt-wound but not necessarily shunt-connected) is basically a velocity demand system. With constant field excitation, torque demanded at a given velocity will automatically be met up to the commutation limit. It is used extensively in industry where starting loads are not high, but where relatively high constant speed is required under varying loads (or at constant load). Velocity can be widely varied, however, by changing the applied voltage and/or by changing the field excitation. Velocity variation through "field weakening" in a standard motor is limited to about 3:1; but in conjunction with changing the applied voltage to the armature, this variation can be extended to a wider speed range.

The compound motor has both the series and shunt field windings to provide the automatic high starting torque of the series motor and high-speed constant velocity with variable load that is characteristic of the shunt motor. A small penalty of increased weight and increased complexity of the control system results from use of this motor.

The torque motor is listed as a candidate where low-speed, precise velocity control is important. The large number of commutation segments and large diameter yield a weight penalty that makes it non-competitive at this time.

The brushless DC motor is listed as a candidate but it is presently constrained by cost and further development required in the power ranges of the hybrid vehicles. To date, it is known to have been built by Aeroflex Corporation up to only 20 kW, but present technology in SCR's makes this motor practical for any size of vehicle. It is created essentially from the redesign of a shunt motor by providing brushless commutation with armature position sensors driving SCR switches. Its operation is thus similar to that of the AC synchronous motor controlled with a variable-frequency feedback system.

The AC induction motor operating at a fixed frequency is not practical for variable speed applications. However, when driven by a variable frequency inverter or cycloconverter, torque-speed characteristics similar to DC motors can be obtained. Controlled slip mode of operation is described in Ref. 6-21. The induction motor has the advantage in specific power compared to DC motors at the vehicle horsepower levels. Its overwhelming utilization in industry and its simple construction make it a strong contender for future vehicles.

6.4.1.2 Control System Complexity and Cost

In a comparison trade-off of the drive train, it is very important to also include the costs and weight of the control system. In terms of performance and versatility, the selection of an adequate control system is as significant as the selection of the motor. All of the contending electric motors need variable voltage applied. Step voltage has been used by which multiple-pole relays switch batteries from parallel to series in steps. This is undesirable because: velocity increments may prevent a vehicle from following another vehicle at the same velocity; the relays are constantly working under load, thereby shortening their lives and producing some ozone from the arcs; and the generator must feed constantly changing voltage levels, which complicates its control logic.

A step-voltage system augmented by field control can be made more desirable by using armature current sensing to provide feedback information for controlling the field. This would control current surges and the corresponding jerks.

A much improved variable-voltage system is being used more generally for low-speed vehicles whereby smoothly varying effective voltage may be applied to the motor. This DC chopper system provides pulse frequency, pulse width, or a combined pulse width and frequency modulation. The result is smooth control of power supplied to the motor. At present, the cost is high. However, if industry has the incentive for high production levels, it is estimated that at some period beyond 1975 the price of the high-current, high-voltage SCR should be reduced sufficiently to make it economically viable. But, the SCR protection circuit and the current smoothing filters will still remain as significant cost factors.

Since the forward voltage drop of the high-current SCR is about 1 V, it can be seen that at 500 amp, 0.5 kW would be lost in the SCR which represents a heat dissipation problem. The higher the maximum voltage of the system, the lower the proportionate loss of the SCR controller system at a given motor power.

There is one other problem with the SCR effective voltage controller when it is used with a series field winding. The field magnetic material must be laminated with higher permeance steels to prevent relatively high core losses at the chopper frequency. An attempt to reduce motor losses by decreasing the chopper frequency can cause motor noise and vibration if the size of the current smoothing filters is not allowed to increase.

A fully compensated motor has very low inductance in the armature. Therefore, an external inductor filter is needed to smooth the motor current at the chopper frequency.

Of greater complexity are the controllers for the brushless DC and the AC induction motors since they must provide not only variable voltage but also variable-frequency to the motor. At present, three-phase, variable-frequency and variable-voltage inverters at power levels associated with the hybrid vehicles are very expensive, since they are complex (12 SCR's or more are needed with at least six having high current ratings). The voltage control may be incorporated into the inverter or a separate chopper may be used.

The more complex and expensive controllers have not yet been fully developed; therefore, an engineering risk still appears at this time. Following is the estimated order of increasing complexity and cost.

a. Step voltage, relay-operated controller

b. Step voltage augmented with field control and armature current sensing

c. Pulse-width modulation SCR chopper

d. Chopper with both frequency and pulse width modulation

e. SCR controller with position sensor for DC brushless motor

f. Multiphase inverter with variable frequency and voltage control.

6.4.1.3 Motor Size and Comparison of Operating Limits

The frame size of a motor is determined by several factors such as torque and speed, thermal characteristics, the type of motor, and the efficiency required. The duty cycle of operation for which the motor is designed is also an important factor in determining the motor size. This provides the weighting factor necessary to determine the instantaneous and average loss in the motor during the duty cycle for which the motor is used. These data combined with the thermal characteristics of the motor can be used to determine the motor temperature under various ambient conditions.

Generally speaking, the higher speed motors are lighter in weight, and less expensive than low speed motors. The implication is that, other things being equal, higher speed motor should be chosen. However, besides the mechanical limitations there are other factors limiting the increased speed such as poor commutation.

To prevent excessive losses and brush wear, as well as flashover at weak field, some form of compensation for armature reaction must be built into the DC motor. Variable pole-face air gaps and pole-face windings are possible. Interpoles, commutation segment resistance, or diodes have also been used to reduce circulating currents and arcing. For given operational characteristics a fully compensated motor need weigh no more than its uncompensated counterpart (Ref. 6-8), however, it is more expensive.

For standard applications such as fan drive, pump drive, etc., the horsepower rating and size of the motors have been established over the years and can be obtained from manufacturers. The requirements imposed on motors suitable for electric drive are new and different. Considerable research and manufacturing efforts are needed for developing and designing motors optimized for such applications. It is hoped that this report provides a basis for further investigation.

6.4.1.3.1 Surge Current Limit

Some small motors (under 2 hp) can tolerate the sudden application of full voltage if the motors are completely unloaded, since the current surge is for a short duration; but most DC motors would suffer catastrophic failure under full load without an armature current limiter. In the series powertrain configuration, the electric motor is gear-linked directly to the wheels and is thus required to start under possibly heavy loads (up to the traction limit of the tires). The current must therefore be limited to a value required to provide the maximum low speed torque. Due to absence of brushes, the AC induction motor is not as susceptible to immediate damage under heavy surge conditions.

All motors will need a device to disconnect them from the power source in the event of overload. Remotely reset current cut-out relays, magnetic trip circuit breakers, magnetic blow-out arc extinguishers, and redundant fuses will have to be investigated further to determine the best means of protection.

6.4.1.3.2 Commutating Current Limit

Once a DC motor starts turning, the overload current is limited by the maximum amount that the brushes can commutate to the bar segments. A compensated motor is superior in this respect.

6.4.1.3.3 Thermal Temperature Rise Limit

The type and quality of insulating material constrains the temperature rise capability. The continuous duty rated current is established by the thermal limit. Overloads may be tolerated for short durations at spaced intervals. For example, compensated DC motors are usually capable of the following at low speeds (Ref. 6-22):

Rated Current, %	Duration	Repeated Less Than
800	0.5 sec	1 per min
550	5.0 sec	1 per 5 min
350	1.0 min	1 per 20 min

These overload capabilities gradually decrease with increased velocity. It is significant that these overload torque values (approximately proportional to current) are not absolute limits. Therefore, acceleration frequently repeated and lasting less than 1 min allows 300 percent or more overload capability, compared to the continuous rating. In this study, overload current for acceleration was allowed to reach three times the continuous current.

A cooling system increases the load ratings for a motor of given size, since considerable heat losses can be transferred out, with the monitor remaining within the thermal rise limit. If the appropriate surfaces are coated black, heat transfer through radiation is improved somewhat. Conduction of heat to the vehicle frame is desirable, but vibration-suppressing rubber shock mounts may impede conduction. Convection transfer of heat remains the method that can be well controlled and is very effective.

Future large motors for the buses may justify the use of cryogenic cooling, such as liquid nitrogen, to provide higher power density. The Fort Belvoir Research Center is investigating this cooling system for large trucks and off-the-road vehicles that use electric drive systems.

Currently, it is practical to cool brush-type DC motors with forced-air systems only. Self-cooling is not effective at low speeds where considerable loading occurs, and the windage loss becomes excessive at very high speeds. Air vanes should be limited, therefore, to self-ventilated motors that operate at only one velocity. Hence, for automotive vehicle applications, a forced-air system, capable of supplying sufficient cubic feet per minute at the proper pressure, is a definite requirement. Table 6-2 gives some current data points for the forced-air cooling systems of typical DC motors.

Table 6-2. Standard Ventilation for DC Motors and Manufacturers Data Points

Efficiency Range, %	Continuous Range, hp	Forced Air, ft^3/min	Static Pressure Drop, in. H_2O
90 up	5 to 20	150	1.00
	25 to 60	350	1.25
	75 to 150	800	1.90
75 to 80	10	350	1.00
	20	350	6.00

A recycling oil-cooled system can be used for large AC induction motors. Since the squirrel cage, or solid rotor, can withstand extremely high temperatures, the oil cooling is constrained to the wire-wound stator. Such a system allows a continuous power density capability to 1 lb/hp.

6.4.1.3.4 Velocity Limit

Most motors show a marked improvement of power density with higher velocities. Three factors constrain the application of very high rpm to DC motors: the safe velocity beyond which there is danger of centrifugal forces causing catastrophic failure, such as pulling armature wires out of their slots; commutation speed; and the increase of power losses. Core losses increase with motor speed. This limits the highest speed for a given efficiency of the motor. Wind and friction losses can be minimized by a smooth armature surface and quality bearings.

The AC motor can achieve much higher velocities since the solid or squirrel cage rotor can be built with greater centripetal strength, and brush friction is eliminated.

6.4.2 Method of Sizing of Motor and Generator

A trade-off can be made between size and weight, or between a motor that meets a certain torque and power requirement, and its efficiency. The weight of the mounting frame must be minimized without compromising structural rigidity. Next, the weight of core material can be reduced by using core stock that is more expensive but of higher permeability. If the same core stock is used but reduced in size, then there will be more core losses for obtaining the same magnetic flux magnitude. Another way to save weight is to reduce the copper cross section, which results in more copper loss (I^2R). Since all this increased loss is in the form of heat, more energy is required to operate the cooling system, and this results in still less overall efficiency.

Power densities as low as 1 lb/hp can be obtained with DC motors but with such low efficiency that it is impractical for continuous duty. Figure 6-18 shows the relationship between efficiency and power density for DC motors currently available in the speed range of 4000 to 8000 rpm.

For a given duty cycle, a given speed/range, and a given type of motor, the weight in pounds per horsepower decreases with increase in horsepower (Fig. 6-19). Hence, at a given efficiency, a large motor will have better power density. The usual large motor thus provides greater efficiency as well as improved power density. Unfortunately, this advantage is offset by the part-load penalty in efficiency. For example, the peak efficiency of a certain DC motor, rated at 60 hp continuous duty, is approximately 92 percent (see Fig. 6-9). However, when the part-load and high-speed efficiency penalties were included and efficiency, velocity, and load were integrated throughout the DHEW Driving Cycle, it was determined that the average efficiency is 80 percent.

In this study, the weight and efficiency of the DC motor for a given speed range was determined from Figs. 6-18 and 6-19. For example, for the series powertrain configuration for the family car, the continuous power rating is 61 hp. This corresponds to a power density of 6.4 lb/hp determined from Fig. 6-19 and a corresponding motor weight of 390 lb. The corresponding efficiency determined from Fig. 6-18 is 92 percent. These data are applicable to standard DC motors currently available in the range of 4000 to 8000 rpm. It is anticipated that by improving motor design and raising the speed of operation that the weight can be reduced.

Based on the data of Fig. 6-16, the weight of a 12,000 rpm alternator providing 51 kw is about 70 lb. Allowing a derating factor of 15 percent for possible variation in heat engine speed, the size of the alternator required is 80 lb. For the hybrid mode of operation, the generator is released from providing the acceleration power and no derating factor is necessary for overload capability.

Figure 6-16 also presented data on high speed, low weight generators that have been developed for space vehicles. These types of generators have not been produced in large quantities. If a 20,000 rpm generator of this type were used for the family car, the generator would weight about 48 lb. High speed generators are readily adaptable to turbine drive systems. For spark ignition engines, gearing would be required to match the engine to the generator.

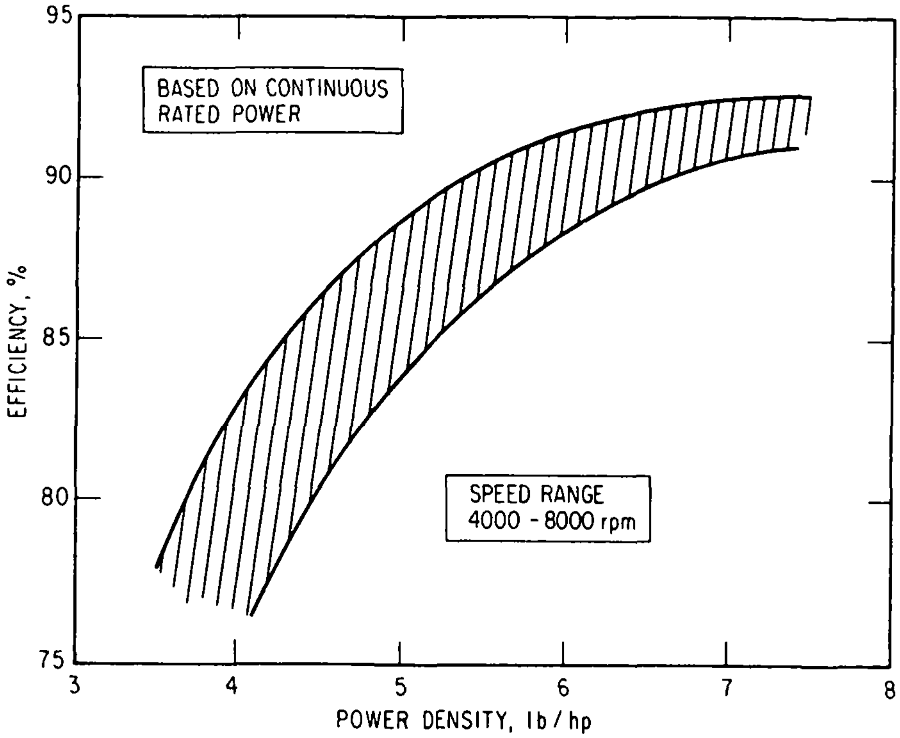

Figure 6-18. Typical Power Density vs Maximum Efficiency, DC Motors - Family and Commuter Cars

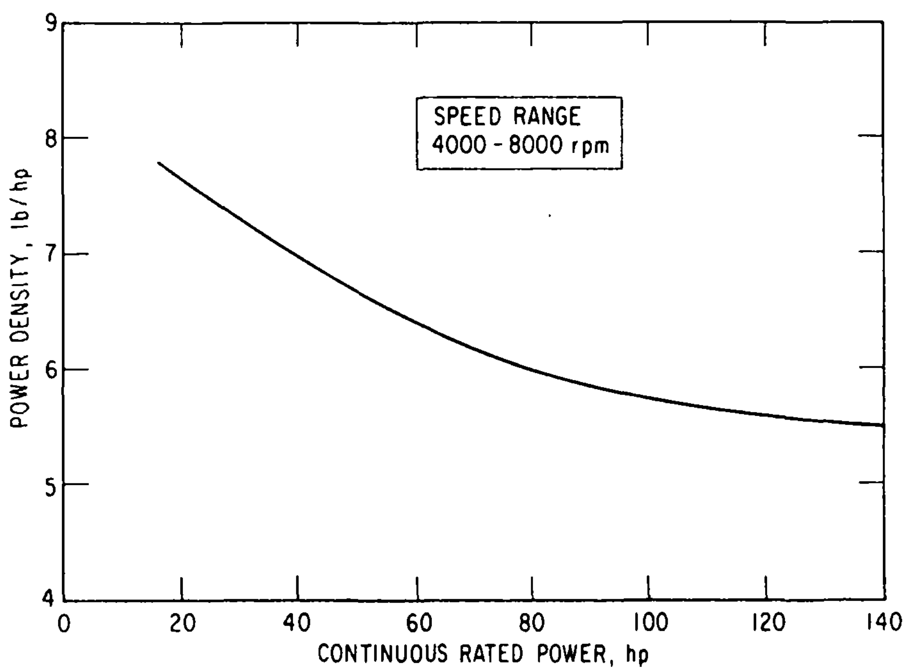

Figure 6-19. Typical Density vs Horsepower, DC Motors Including Forced Air Cooling - Family and Commuter Cars

6.5 DESIGN GOALS

As indicated, AC and DC generators have been developed to a fairly high degree for aircraft and automotive applications, and it is apparent that little development work is required in this area. The AC generator (with rectifiers) is preferred for the hybrid vehicle because of higher efficiency, lighter weight, and low cost. Some effort should be expended, however, to improve the part-load efficiency.

Electric motors, on the other hand, particularly DC motors, have not been developed to optimize efficiency, weight, size, and cost for vehicle propulsion. It is believed that DC electric motors can be designed with higher efficiencies and lighter weights than those on the market today, and with equal reliability and lifetimes. Reasonable weight and efficiency goals, which it is expected can be achieved for the various classes of vehicles, are tabulated in Table 6-3 for the series configuration. Efficiency can be traded off against motor weight as has been indicated in Fig. 6-18.

Table 6-3. DC Electric Motor Weights (Including Forced Air Cooling)

Vehicle, hp	Weight, lb	Power Density, lb/hp	Maximum Efficiency, %
Commuter, 21	160	7.6	92
Family, 61	390	6.4	92
Van			
Low Speed, 30	180	6.0	92
High Speed, 80	430	5.4	94
Bus			
Low Speed, 100	870	8.7	94
High Speed, 175	1050	6.0	94

For a given design power level, the weight per unit horsepower can be decreased if the efficiency is allowed to decrease. Power densities of from 5.5 to 8 lb/hp should be achievable at reasonable cost by merely optimizing the design for the particular application and utilizing lightweight materials whenever possible. The efficiencies of these devices would range between 90 and 94 percent at design load depending on the size of the motor. The weight per unit of horsepower may be further reduced by the use of AC motors, inverters, and liquid cooling. Part-load efficiency is also very important because during a typical driving cycle, the motor operates at part-load most of the time.

It is further estimated that the efficiencies of the controllers and the motor in the regenerative mode (with the motor acting as a generator when the vehicle is decelerating) can be improved quite markedly, thus increasing the overall efficiency of the vehicle. The field power of the separately excited motor is typically 5 percent of full load power of the motor. Since the controller directly changes only the field current, its efficiency is thus high when compared to the total motor power being altered. It appears reasonable to believe that regenerative efficiencies on the order of between 25 and 40 percent should be achievable. These values represent the combined efficiencies of: the drive motor acting as a generator, the battery charge, mechanical friction, and the effect of the driving cycle on battery charge acceptance.

Overall system efficiencies of the different parallel powertrain approaches for various vehicles and driving cycles should be investigated to determine which one will have the greatest possibilities for high efficiency and low pollution levels. One such system, not analyzed in depth nor tested, is the one shown in Fig. 6-4. It is recommended that this system be analyzed and tested, and then compared with the other two parallel systems as well as with the series approach. The parallel system is most likely to achieve higher efficiencies since a considerable portion of the energy does not pass through the generator, motor, and battery, thus eliminating the attendant losses.

6.6 RECOMMENDED SUBSYSTEM DEVELOPMENT

As already discussed in detail in this section, certain areas require further development effort in the electrical system (exclusive of batteries, which are covered in Section 7). These efforts consist of the following:

a. Develop lightweight, efficient DC motors optimized for efficiency and weight for the automotive application. Both shunt and series types are required.

b. Develop lightweight, efficient controllers for shunt motors. Very little development appears necessary in the area of series choppers.

c. Develop small and compact, vehicle-borne logic and control circuits to optimize electrical/heat engine performance. Inputs to the logic circuit would be generator current, battery charge current, motor armature current, engine speed, battery voltage, and accelerator pedal position. Based on these inputs the logic circuit would determine the desired optimum heat engine power setting. Under these conditions, maximum utilization of energy available from regenerative braking could be achieved.

d. Investigate the various parallel system approaches to the design of hybrid cars. Two parallel concepts have been evaluated, the TRW and Minicar types; however, the efficiency of a third type (Fig. 6-4) using two motors requires further evaluation. Two versions of this configuration need to be investigated, one using armature voltage and/or external excitation as a speed control mechanism, and the other using a proprietary scheme proposed by Electric Motion Control Corporation of Pasadena, California.

e. Determine and compare the efficiencies and heat rejection systems of DC and AC motors and associated control systems, particularly for the large vehicles.

f. Compare in more detail the cost of various approaches.

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SECTION 7

ELECTRICAL SYSTEM - BATTERY CHARACTERISTICS AND OPERATION

CONTENTS

CONTENTS (cont.)

TABLES

FIGURES

FIGURES (cont.)

SECTION 7

ELECTRICAL SYSTEM - BATTERY CHARACTERISTICS AND OPERATION

7.1 INTRODUCTION

Next to the heat engine and its associated emission characteristics, the battery is the most important element in the hybrid electric vehicle powertrain since it has a marked effect on vehicle exhaust emissions. It is the intent of this section to elaborate on this point by showing the rationale for battery selection, the assumed battery characteristics, how the battery operates in the hybrid vehicle, the resulting battery design goals, and, finally, a recommended program for battery development. In addition to hybrid electric vehicle-oriented battery considerations, an assessment is made of the state of the art for general battery technology to enhance evaluation of expected progress in developing high-performance batteries.

7.2 BATTERY SELECTION CRITERIA

Because of the large battery capacity required in the hybrid electric vehicle, battery cost will represent a significant fraction of overall vehicle cost. Hence, only reasonably priced batteries could be considered for personal transportation vehicles, leaving the more expensive batteries for commercial vehicles where first costs are more readily depreciated. Besides the initial costs, the cost of battery replacement over the vehicle lifetime must be considered (even for commercial vehicles). Therefore, battery lifetime becomes a major evaluation factor in battery selection and in permissible cyclic operation in the hybrid vehicle.

Battery maintenance can also be viewed as a cost factor. Inspection and/or test of a large complex of individual cells would be time consuming for the user of personal transportation vehicles and would possibly be a major cost factor for commercial vehicle operators in view of the added labor and time out of service. Therefore, battery selection should consider low maintenance or (preferably) no maintenance batteries.

To achieve superior battery lifetime by avoiding deep discharge, the battery energy capacity must be reasonably large relative to the maximum energy demand. At the same time, to supply the large currents required by the electric motor for producing maximum vehicle acceleration, the battery must be capable of delivering high power for short periods. For fixed battery weight, these are conflicting requirements since energy capacity is usually traded to enhance power capability and vice versa. Thus, there is no perfect battery system and a compromise must be reached between acceptable vehicle acceleration and acceptable battery lifetime.

One other major factor not to be overlooked is availability. There are some very advanced battery concepts being examined and tested in both research and early development programs. However, with the possible advent of a prototype hybrid electric vehicle within a few years and with a need to ensure production of a viable hybrid electric vehicle in the 1975 to 1980 period, attention must be focused on those systems that are readily available or have short term development potential.

With the recognized importance of the battery system to the feasibility and success of the hybrid electric vehicle, the Air Pollution Control Agency (APCO) (at the time, NAPCA) and The Aerospace Corporation convened a meeting at Argonne National Laboratories, Argonne, Illinois, of government and university battery experts to decide which batteries should receive primary emphasis in this study. The discussion included a review of preliminary study results depicting the type of battery operation to be expected in the hybrid electric vehicle. After evaluating performance, cost, and availability (with availability the dominant factor), the final consensus of both the invited attendees and members of The Aerospace Corporation was that only two types of batteries were suitable for application to a hybrid electric vehicle at this time: lead-acid and nickel-zinc. A third battery, nickel-cadmium, was added because of its high state of development, high power density, and ability to accept rapid charge; however, the cost and limited availability of cadmium restricts the consideration of this battery primarily to commercial hybrid electric vehicles of low levels of production and to test bed hybrid vehicles for prototype evaluation programs.

7.3 MODELS OF BATTERY CHARACTERISTICS

Battery characteristics in terms of voltage, current, and state of charge were simulated in the Hybrid Electric Vehicle Performance Computer Program by linearized approximations for each of the three batteries. The charge-discharge models shown in Figs. 7-1, 7-2, and 7-3 for lead-acid, nickel-cadmium, and nickel-zinc batteries, respectively, were entered in tabulated form into the computer program and used for all analyses documented in this report. These characteristics were based on available data and projections of future capability. They have a marked effect on selection of heat engine/generator operating mode and power output level as well as on battery design goals for energy density and lifetime. Hence, final selection of hybrid vehicle design specifications and operation will be influenced by the test data emanating from battery development programs.

The characteristics of the advanced lead-acid battery shown in Fig. 7-1 are generally beyond any which have now been demonstrated, but they are based on extensions of the data of Ref. 7-1 and follow the extrapolations indicated in Ref. 7-2. The nickel-cadmium battery model in Fig. 7-2 represents advanced characteristics of a sintered plate prismatic design. For the nickel-zinc battery model (Fig. 7-3), characteristics of all the batteries described in Refs. 7-3 through 7-8 were examined to determine if any similarities existed, but no consistent form was shown. Therefore, the characteristics of the cells described in Ref. 7-6 were generally used in establishing the model.

The characteristics of the battery cell models are summarized in Table 7-1. These characteristics are based upon specific batteries, but within any battery type any number of variations can exist. The maximum and minimum allowable voltage for each battery model was established to limit outgassing during charge and to limit energy drain during discharge; in both cases the objective of imposing these constraints is to prolong battery life. On a

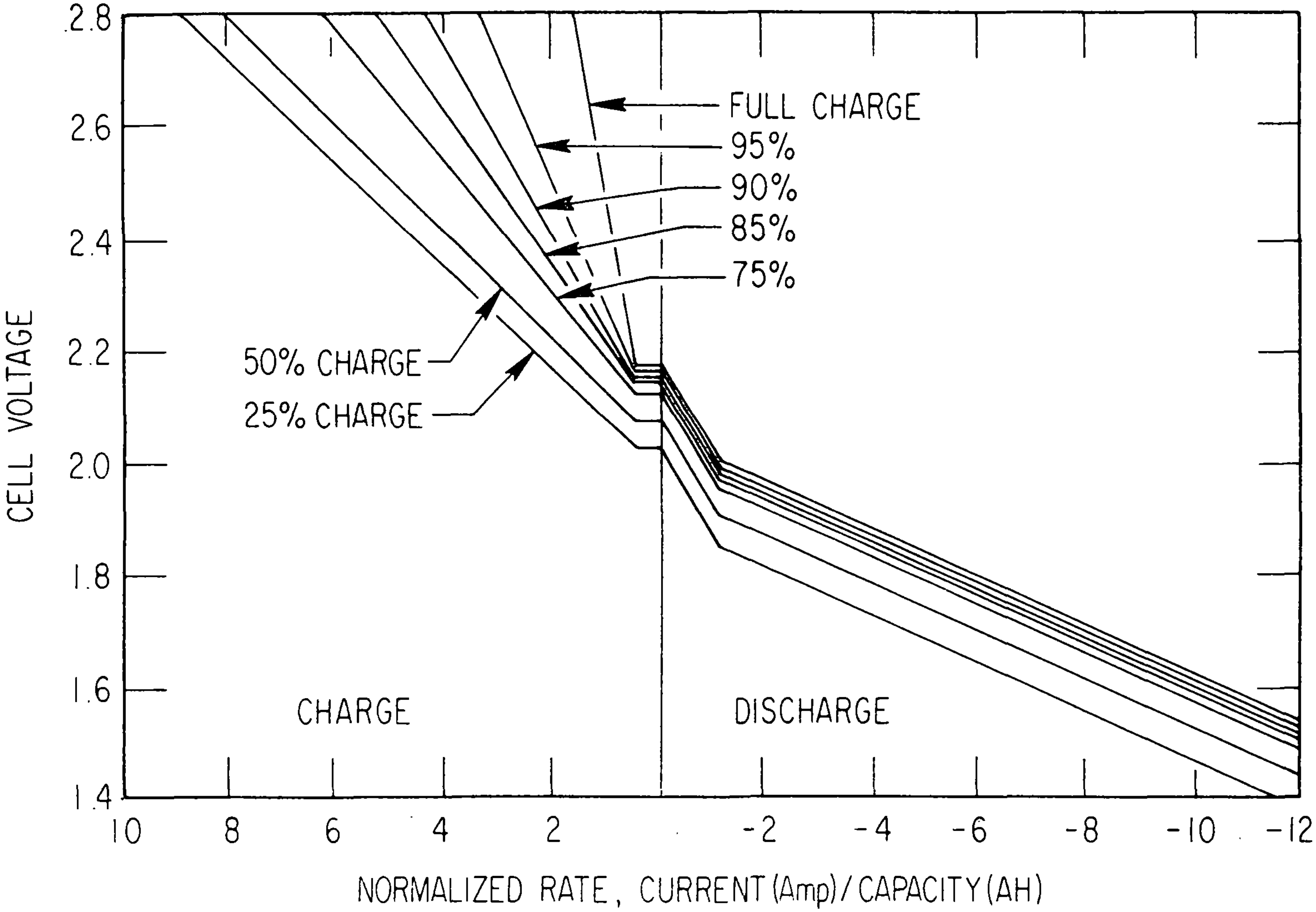


Figure 7-1. Computer Program Model of Advanced Lead-Acid Battery

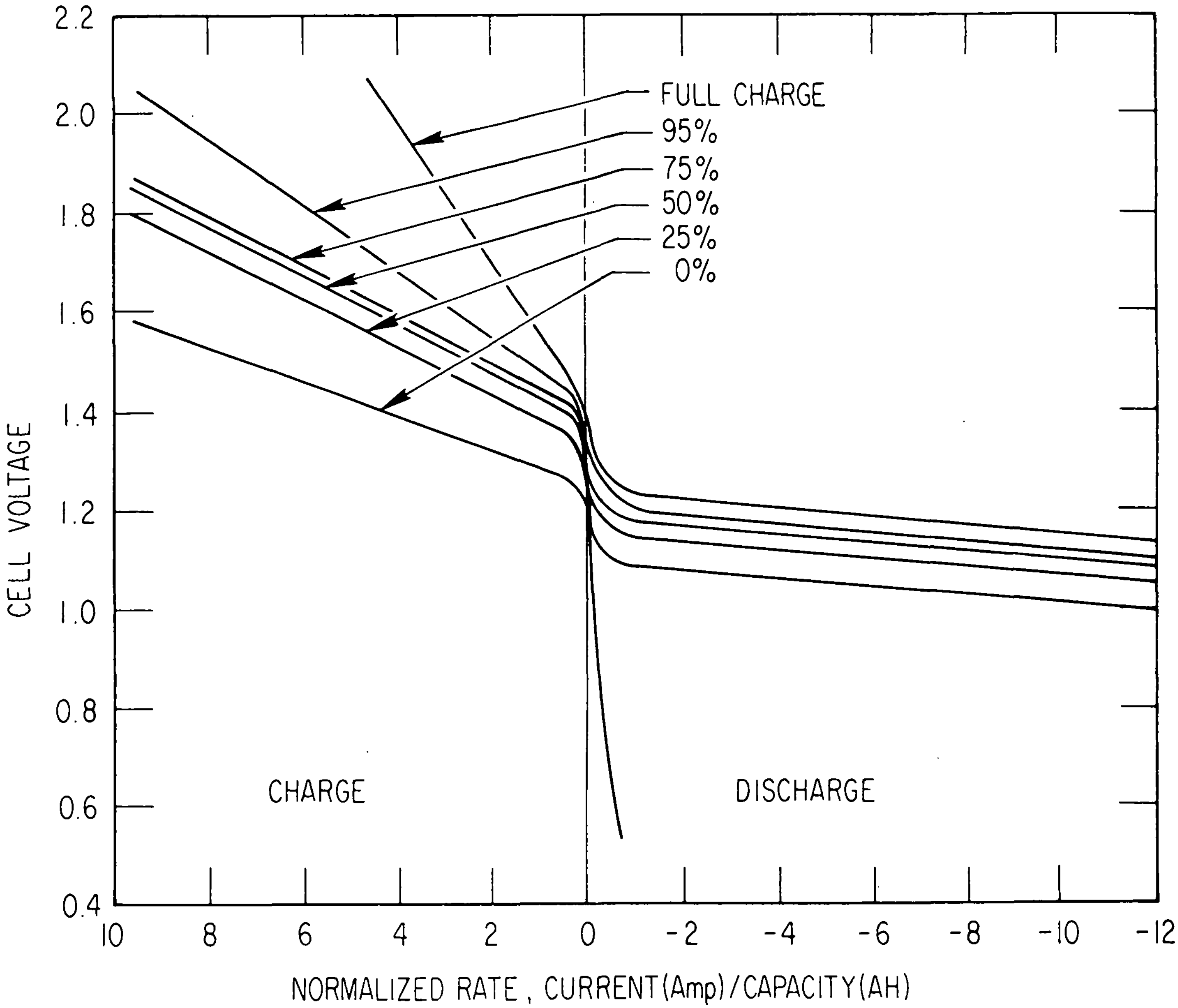


Figure 7-2. Computer Program Model of Advanced Nickel-Cadmium Battery

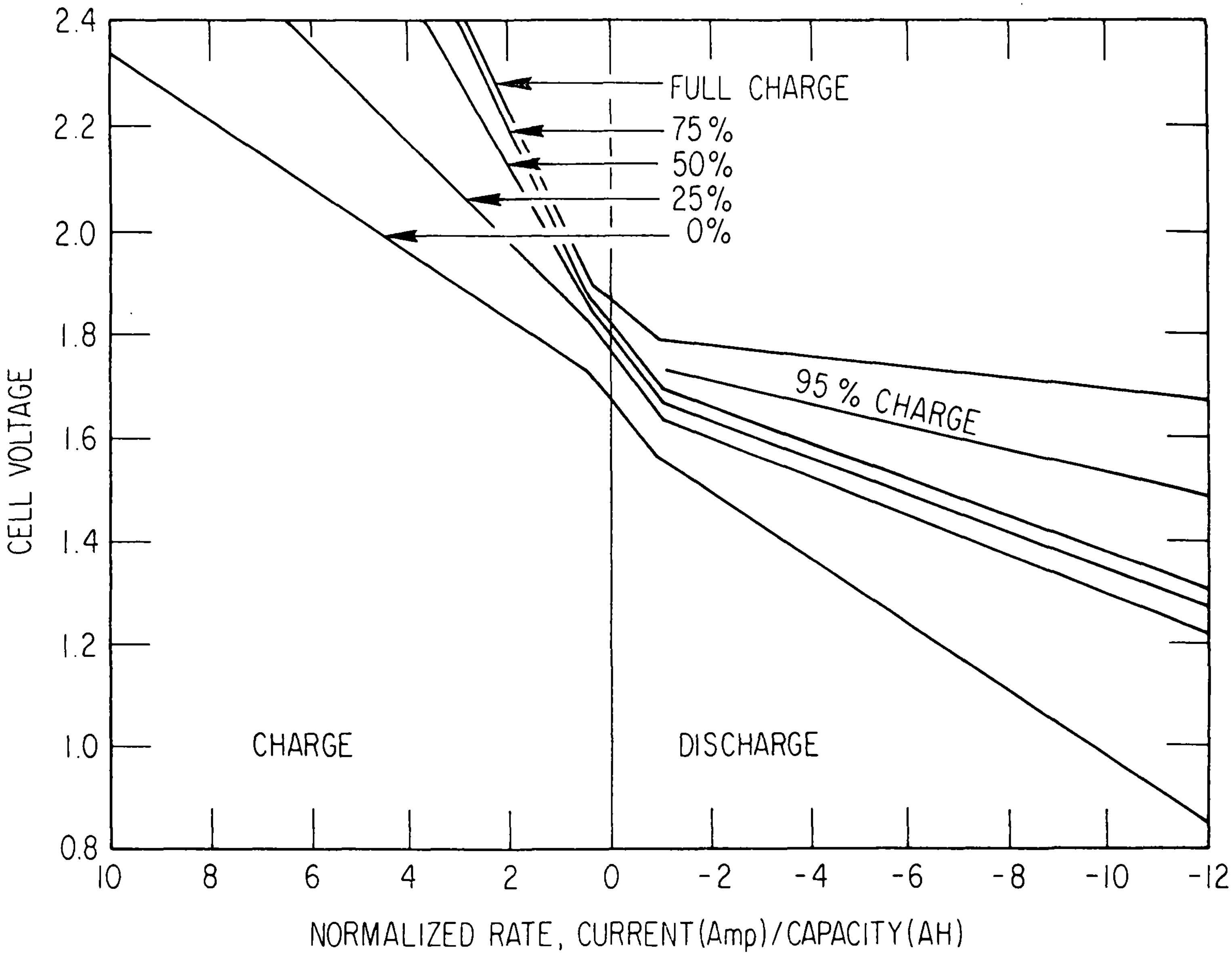


Figure 7-3. Computer Program Model of Nickel-Zinc Battery

normalized basis, the ratio between the minimum and maximum voltages is about the same for all of the cells. This ratio is important in establishing the operating range of the generator and drive motors and might indicate that batteries of different types could be used interchangeably in a hybrid vehicle.

Table 7-1. Cell Model Characteristics

CHARACTERISTICS	CELL MODEL		
	Lead-Acid	Nickel-Cadmium	Nickel-Zinc
Open Circuit Voltage	2.1	1.35	1.8
Minimum Allowable Discharge Voltage	1.50	1.10	1.30
Maximum Allowable Charge Voltage	2.40	1.55	2.10
Maximum I/C* Rates			
Discharge, 90% DOD**	11.7	12.1	17.8
Charge, 100% Charge	1.0	1.1	1.35

*I/C = Current/Rated Capacity

**DOD = Depth of Discharge

The cell models indicate a definite advantage for the nickel-zinc battery in terms of high current capability. Whether this model can be achieved in a high-cycle life cell or whether the other cells would be improved is another factor which must be determined.

In the computer program, at the start of each emission driving cycle, the battery is set to full charge. As explained further in Section 4 and Appendix A, the voltage, current, and state of charge are calculated at each one second time step throughout the cycle; during discharge, the state of charge is determined by the equation

$$S_{i+1} = S_i - \frac{i_B (t_{i+1} - t_i) 100}{C_R}$$

(discharge current efficiency assumed = 100%)

and during charge by the equation

$$S_{i+1} = S_i + \frac{\eta_{RB}\, i_B\, (t_{i+1} - t_i)100}{C_R}$$

where

S = battery state of charge, percent

i_B = battery charge or discharge current, amp

t = time, hr

C_R = battery rated capacity, amp-hr

η_{RB} = battery recharge current efficiency

The recharge efficiency is used to account for both internal and external resistive losses as well as for deviations from complete theoretical chemical conversion between electrodes and electrolyte.

All calculations were conducted for the battery initial and final states-of-charge of 100 percent; somewhat different results may have resulted by using a lesser value for the state-of-charge. This would improve charge acceptance but diminish discharge capacity. The overall effect on vehicle operation may show improvement and warrants further study.

With the magnitude of current drained given by the difference between electric traction motor demand and generator current, and following calculation of the state of charge, the battery voltage is then uniquely determined from tabular reproductions of Figs. 7-1, 7-2, and 7-3. If at any time during the emission cycle the lower voltage limit is passed, computation is halted. With regard to the upper voltage limit, it can be seen by reference to the aforementioned figures that the maximum allowable charge current is established by the battery capacity and the state of charge. Since the available charge current can readily exceed the allowable value, the excess current is shunted to a load resistor and accounted for in the computer program.

7.4 BATTERY SIZING AND OPERATION OVER DRIVING CYCLES

Fundamental sizing of the battery is established by the ability to furnish power and energy as required by the vehicle during operation over design and emission driving cycles. In establishing battery capacity, a minimal value was selected to fulfill the more restrictive of two requirements by:

a. Ensuring that the voltage does not fall below minimum allowable voltage for maximum power demand during acceleration on the design driving cycle.

b. Ensuring that required* generator power output for emission driving cycles is established at low levels. (It has been determined that as battery capacity was decreased the required generator output remained at low levels until a critical point was reached where the required output rose sharply. Since a higher generator output power level is reflected directly in higher exhaust emissions, this operating region is to be avoided.)

Table 7-2 shows the required lead-acid battery capacity for each class of vehicle along with system voltage and minimum generator current. With these capacities, battery discharge current during vehicle operation over the design driving cycle is shown for several representative cases in Figs. 7-4 through 7-9. Peak currents range from a high of ~500 amp for the low-speed bus to a low of ~150 amp for the commuter car; the family car required ~460 amp.

With battery capacity established, the operating characteristics can next be established for the emission driving cycles. Typical battery operation is illustrated for the family car using a lead-acid battery in Figs. 7-10 through 7-14. A total of 73 battery discharge cycles was noted during the driving cycle as shown in Fig. 7-10. Total duration of discharge operation was 406 sec, leaving 964 sec available for recharge.

Peak discharge currents for each of the discharge cycles are described in Fig. 7-11. The peak current was 261 amp, and the next highest 145 amp. Average of all peak currents for all of the cycles is about 49 amp with a median current of under 40 amp. These high power discharge cycles are also the

*to return battery to original state of charge

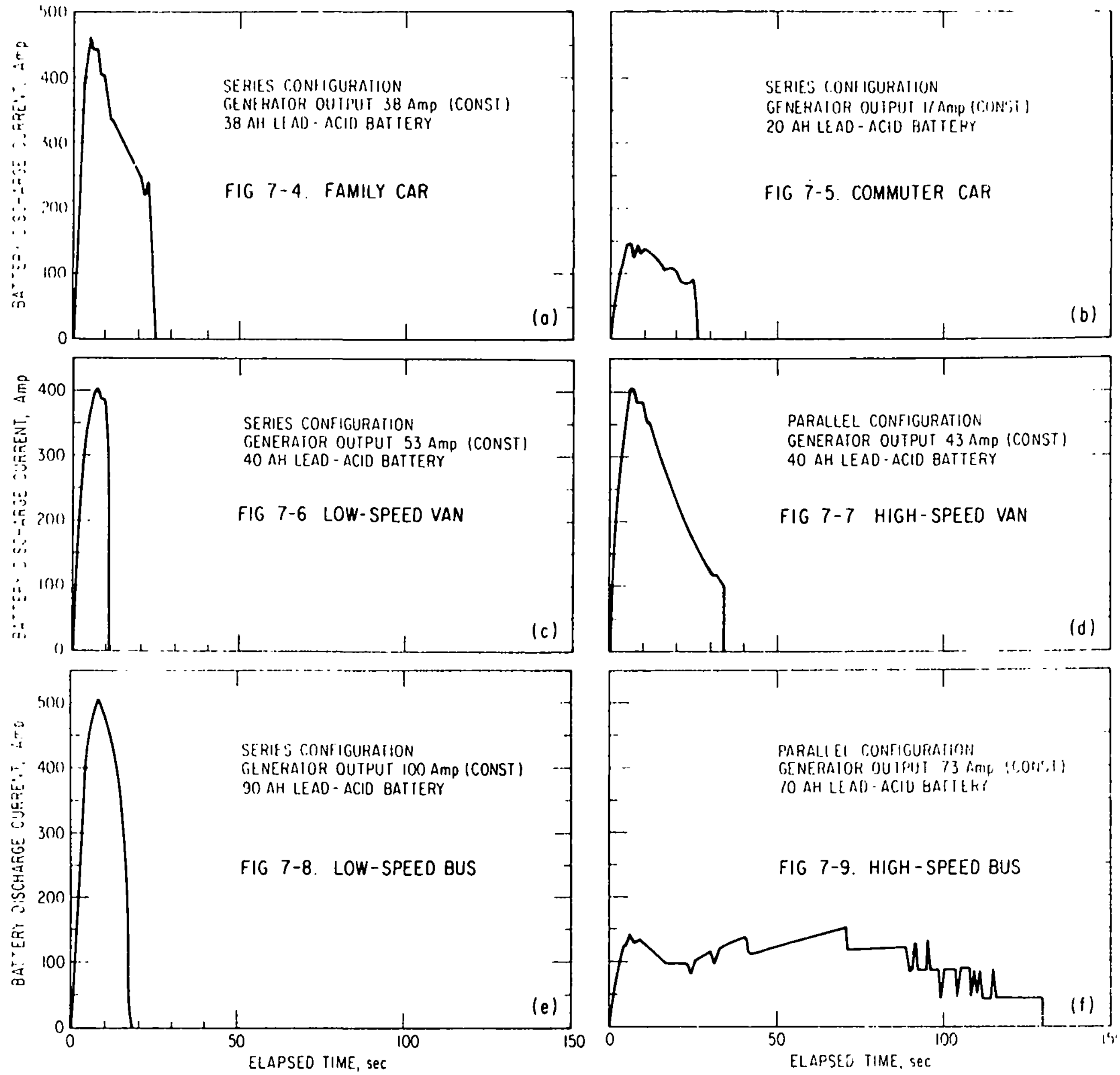
Table 7-2. Baseline Design Energy Expenditures Over Emission Driving Cycles (Advanced Lead-Acid Battery)

Configuration	Vehicle	Time To Traverse Cycle (sec)	I_{Min}[1] (amp)	System Voltage, V_s (volt)	Minimum[2] Heat Engine Power Output (hp)	Rated Heat Engine Power Output (hp)	Installed Battery Capacity (amp-hr)	Heat Engine Energy Output (hp-hr)	Overall[3] Efficiency, $\overline{\eta}$ (%)	Heat Engine Energy Per Mile (hp-hr/mi)
Series	Family Car	1370	38	220	20.7	92	38	8.37	28.6	1.12
Series	Commuter Car	1370	17	220	7.97	33	20	3.19	32	0.425
Series	Low-Speed Van	96	53	220	22.4	42.3	40	0.602	29.2	3.05
Series	High-Speed Van	96	54.5	220	22.4	107	40	0.602	29.2	3.05
Series	Low-Speed Bus	36	100	440	86	168	90	0.885	44	8.95
Series	High-Speed Bus	42	79	440	70.6	257	79	0.843	32.8	8.10
Parallel	Family Car	1370	34	220	19.2	84.2	30	7.32	31.9	0.975
Parallel	Commuter Car	1370	14.5	220	7.1	30.3	20	2.7	38	0.36
Parallel	High-Speed Van	96	43	220	18.1	102	40	0.481	36.6	2.43
Parallel	High-Speed Bus	42	73	440	66.2	236	70	0.751	36.8	7.22

Notes: (1) $I_{Min} \times V_s$ = electrical equivalent of sum of mechanical and electrical power delivered by heat engine. For parallel configuration, generator current decreases as vehicle speed (road load) increases (see Section 10).

(2) With accessories but no A. C.

(3) $\overline{\eta} = \dfrac{\text{energy required at vehicle wheels}}{\text{energy output from heat engine}}$



Figs. 7-4 through 7-9. Battery Discharge Characteristics, Design Driving Cycle

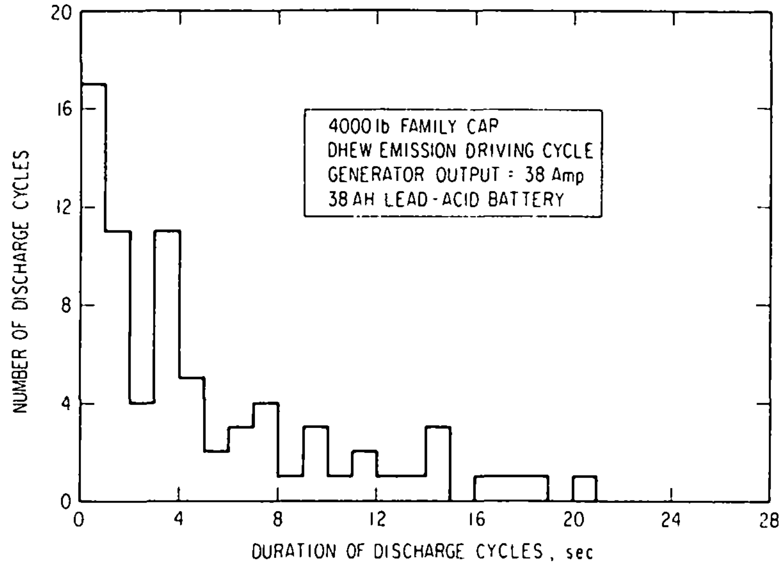


Figure 7-10. Duration Distribution of Battery Discharge

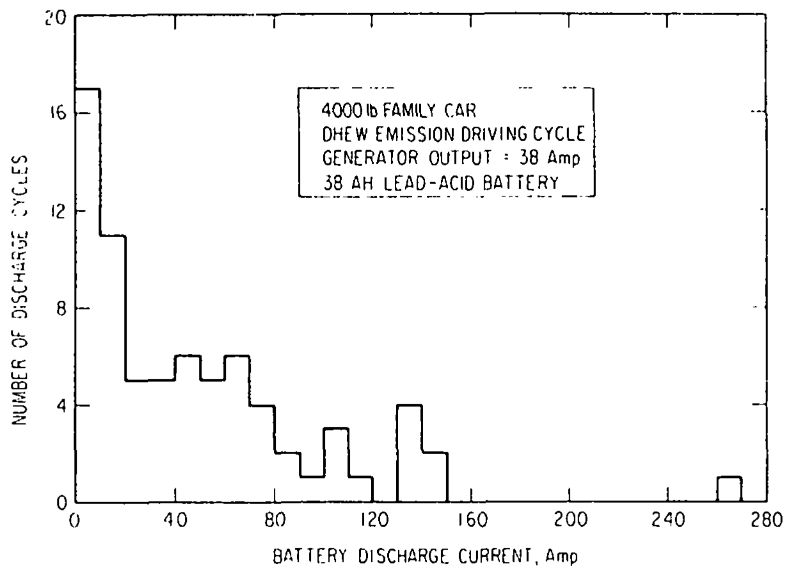


Figure 7-11. Battery Discharge Current Distribution

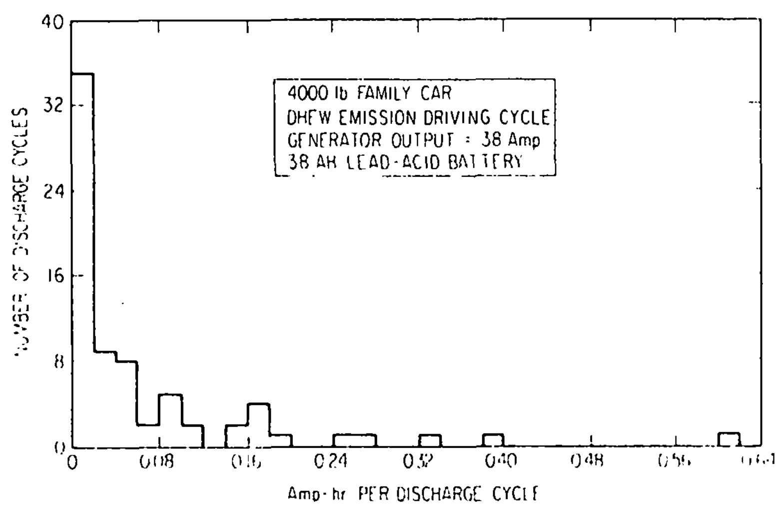


Figure 7-12. Amp-hr Distribution During Battery Discharge

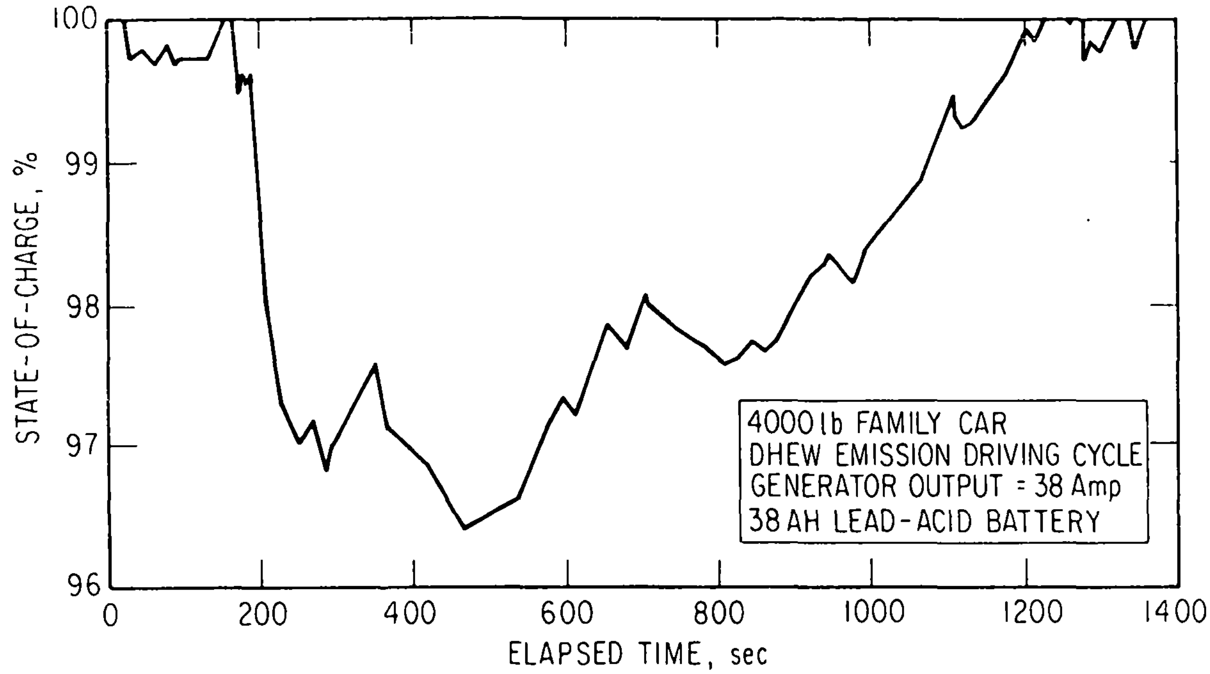


Figure 7-13. Battery Discharge Characteristics

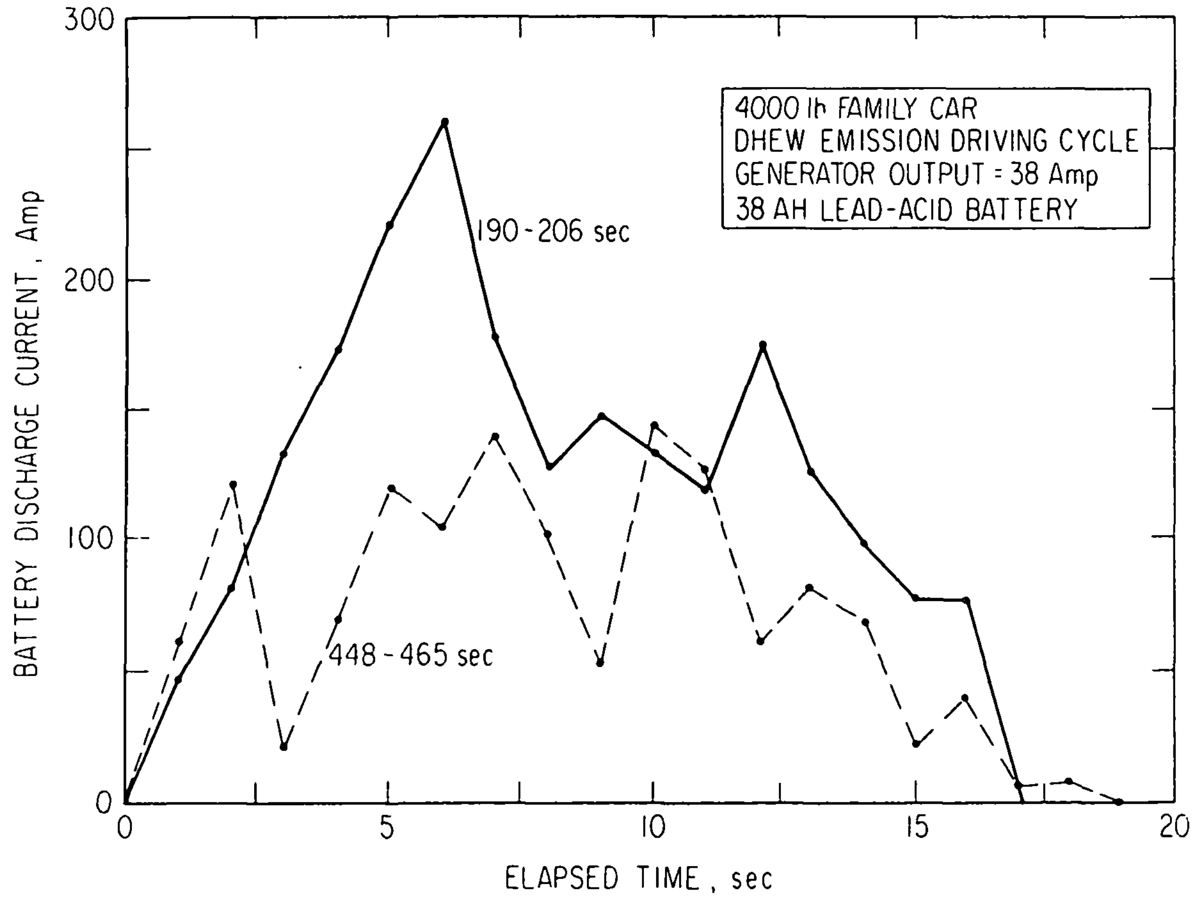


Figure 7-14. Battery Peak Discharge Currents

highest amp-hr discharge cycle and are 17 and 18 sec in duration with an average current during the two discharges of 129.2 and 76.0 amp, respectively.

The amp-hr contained in each discharge cycle is shown in Fig. 7-12. The total energy is 4.76 amp-hr with the maximum during any discharge of 0.61 amp-hr; the average is 0.065 amp-hr. Median discharge is about 0.02 amp-hr.

The state of discharged battery capacity for the family car during the DHEW Emission Driving Cycle is given in Fig. 7-13. In this case, the battery reached a minimum charge level of 96.47 percent state of charge which corresponds to 1.34 amp-hr. This form of shallow discharge was demonstrated by all batteries and is considered to be characteristic of how batteries will operate when continually recharged by a generator/alternator on all classes of hybrid electric vehicles.

The current traces for the two maximum power (also maximum energy) cycles are shown in Fig. 7-14. It is to be noted that the high currents exist for only about a second.

The emphasis to this point has been upon the family passenger car and its requirements. For the commuter car and vans which were operated over their respective emission driving cycles, the results were almost identical in terms of number of cycles and duration of cycles, but the power requirements were different. In all cases, the maximum power demand occurred during the design driving cycle - nearly twice that resulting from the emission driving cycle.

Some general comments should be made concerning how changes in operating parameters affect the characteristics shown in Figs. 7-10 through 7-14. The first change which might be considered is an increase in generator output. This will restore the battery to full charge more rapidly, will reduce the average battery current, and may change the number of discharge cycles. Raising the generator current, however, has the disadvantage of increasing vehicle exhaust emissions since the average engine output is increased. By operating the battery more fully charged battery life could increase, but the amount of energy dissipated through the external resistor circuit or in the battery is also increased and this could have the opposite effect of decreasing battery life. Nonetheless, in this study, the objective was to decrease emissions and for that reason increasing generator output is not an attractive alternative.

Another method of decreasing the number of discharge cycles might be to increase battery capacity. This is accomplished with a proportional increase in battery weight. An increase in battery capacity will cause improved charge acceptance and will lower the maximum depth of discharge. The larger batteries would provide a smaller depth of discharge which is ordinarily beneficial to battery life.

The emission driving cycles utilized in this study are considered to be typical of driving conditions expected for each vehicle. However, based upon the approach of maintaining the battery state of charge after each driving cycle, an important factor affecting design of the battery system may be the type of driving cycle employed. Hence, for prototype vehicles operated in various cities, a more sophisticated battery control system may be necessary; e.g., one which senses battery voltage and possibly other parameters continuously, and regulates charge current levels according to need rather than in a predetermined manner.

One other important factor describing battery performance is the percentage of recharge energy shunted to the load resistor because available recharge current exceeded allowable recharge current. Since this represents wasted energy (i.e., generator output must be increased to overcome this loss and still ensure a fully charged battery at the end of the emission driving cycle), the resultant effect is increased vehicle exhaust emissions for batteries with poor charge acceptance.

The charge acceptance characteristics also have a marked effect on the utilization of energy from regenerative braking. This utilization was shown to be negligible for all batteries in this study since the regenerative braking charging currents generally exceeded the allowable charging current by up to an order of magnitude. Some improvement might be possible by starting the battery on emission driving cycles at less than 100 percent state-of-charge and likewise the goal would be to return to the original state-of-charge by resetting generator output at a lower value. This technique would be evaluated in a prototype vehicle. If this condition were carried over to the design driving cycle, however, a greater battery capacity might be needed to maintain battery voltage above the minimum allowable level.

The battery models employed in the analysis generally had acceptable charge acceptance as shown typically by the charge energy utilization factors for the advanced lead-acid battery given in Table 7-3. Here it can be seen that for the family car only 1.4 percent of available charge energy was shunted to the load resistor. Similarly low values are apparent for the commuter car, the van, and the bus. Although not shown, far worse performance was given by contemporary lead-acid batteries; they require that the generator current level be about twice that needed for the advanced lead-acid battery. The result is a significant increase in vehicle exhaust emissions (See Section 11). Hence, battery development programs should be directed toward achieving at least the charge acceptance capabilities of the battery models used in this study. To utilize energy from regenerative braking, however, charge acceptance would have to be improved significantly further. This is considered to be a very important aspect in assuring a viable hybrid electric vehicle.

7.5 REVIEW OF BATTERY STATE OF THE ART

The characteristics of a large number of batteries are presented in Table 7-4. While these are of interest in establishing the state of the art for the general battery field, the succeeding discussions will be limited to those batteries considered for near term use with the hybrid electric vehicle: the lead-acid, nickel-cadmium, and nickel-zinc batteries.

Table 7-3. Advanced Lead-Acid Battery Energy Utilization Over Emission Driving Cycles (Recharge Efficiency = 70%)

CONFIGURATION	VEHICLE	BUF[1]	E_S[2]	BLF[3]
Series	Family Car	0.582	0..127	0.014
Series	Commuter Car	0.597	0.0033	0.008
Series	Low-Speed Van	0.541	0.0246	0.0261
Series	High-Speed Van	0.541	0.0246	0.0261
Series	Low-Speed Bus	0.700	0.0001	0.0002
Series	High-Speed Bus	0.611	0.034	0.057
Parallel	Family Car	0.628	0.243	0.029
Parallel	Commuter Car	0.638	0.0001	0.00003
Parallel	High-Speed Van	0.689	0.022	0.030
Parallel	High-Speed Bus	0.696	0.023	0.0435

Notes:

(1) $\text{BUF} = \dfrac{\text{amp-hr delivered by battery}}{\text{amp-hr available for charging battery}}$

(2) E_S = amp-hr shunted to load resistor

(3) $\text{BLF} = \dfrac{\text{amp-hr shunted to load resistor}}{\text{amp-hr available for charging battery}}$

Table 7-4. Characteristics of Batteries

CELL	ANODE	CATHODE	ELECTROLYTE	THEORETICAL		REPORTED	
				Cell Voltage	Energy Density (W-hr/lb)	Energy Density (W-hr/lb)	Energy per Unit Volume (W-hr/in.^{3})
LEAD-ACID	Pb	PbO_2	H_2SO_4	2.04	74	20-30	2.0
NICKEL-CADMIUM	Cd	NiOOH	KOH	1.30	96	12-20	1.1
NICKEL-IRON (EDISON)	Fe	NiOOH	KOH	1.58	142	10-15	1.2
NICKEL-ZINC	Zn	NiOOH	KOH	1.74	170	15-30	2.0
SILVER-CADMIUM	Col	Ag_2O/AgO	KOH	142/1.15	122/82	20-30	2.9
SILVER-ZINC	Zn	Ag_2O/AgO	KOH	1.85/1.59	220/130	25-110	4.5
MERCURY-CADMIUM	Cd	HgO	KOH	0.907	67	18-35	6
LALANDE	Zn	CUO	NaOH	–	109	20	0.9
LE CLANCHE (DRY CELL)	Zn	MnO_2	NH_4Cl	–	153	25-30	2.5
ALKALINE	Zn	MnO_2	NaOH	–	149	30-45	2.2
EDISON AIR CELL	Zn	O_2	KOH	1.65	671	50-55	2.5
MERCURY (RUBEN)	Zn	HgO	KOH	1.34	116	45-52	8
MAGNESIUM DRY CELL	Mg	MnO_2	$MgBr_2$	–	247	45-50	2.5
MAGNESIUM-CHLORINE	Mg	Cl_2	$MgCl_2$	–	954	–	–
SODIUM-CHLORINE	Na	Cl_2	NaCl	3.98	849	–	–
SODIUM-SULFUR	Na	S	$Na_2O \cdot HAl_2O_3$	–	346	148	8.1
ALUMINUM-FLUORINE	Al	F_2	Na_3AlF_6	–	1,940	–	–
LITHIUM-CHLORINE	Li	Cl_2	LiCl	–	990	125-250	–
LITHIUM-FLUORIDE	Li	NiF_2	PC	–	626	100	–
LITHIUM-CHLORIDE	Li	C-$TeCl_4$	KCl/LiCl	3.25	–	60	5
LITHIUM-SELENIUM	Li	Se	–	2.2	–	130	–
LITHIUM-SULFUR	Li	S	LiI/LiCl/KI	2.25	-700	70	6.7
ALUMINUM-CHLORINE	Al	Cl_2	$AlCl_3$	3.02	828	–	–
H_2-O_2 FUEL CELL	H_2	O_2	KOH	1.23	17,875	45	2.8

7.5.1 The Lead-Acid Battery

The lead-acid battery is of primary interest because of its low cost, reliability, and availability. The battery can be mass produced easily with inexpensive tooling, and the formation procedure is simple. In 1967, there were 233 companies producing lead-acid batteries (Ref. 7-9) and, of these, 120 had more than 20 employees. These companies did 580 million dollars worth of business, with 260 million dollars of this being value added by manufacture and the rest being cost of materials. In 1967, the average cost of lead was 14 cents/lb whereas the current cost is 16.5 cents/lb. Retail cost of starting-lighting-ignition (SLI) batteries is 40 to 80 cents/lb, and at open circuit voltage, from 2.2 to 4.0 cents/watt-hr. Industrial lead-acid batteries cost about twice as much.

7.5.1.1 Present Battery Characteristics

Characteristics under discharge and charge for SLI batteries are given in Fig. 7-15 and at various rates in Fig. 7-16. The tendency for capacity to decrease with discharge rate is indicated by this curve. It is interesting to note, as shown in Fig. 7-17, that the capacity of a lead-acid battery increases with temperature. In Fig. 7-18 is shown the trend of cycle life with temperature.

7.5.1.2 Battery Failure Modes

Three modes of battery failure are generally considered. The first type involves the gradual dissolution of the positive grid. Under charge, a small layer of the grid may become oxidized, and this layer is then stripped off during discharge. Eventually, as the grid is eaten away, the conductive path is broken and the active material may lose contact with grid. Industrial batteries and, to a certain extent, golf cart batteries avoid or delay onset of this problem by using thicker grids.

A second type of failure, not entirely dissociated from the first, is the sloughing off of material from the plates and the accumulation of this material

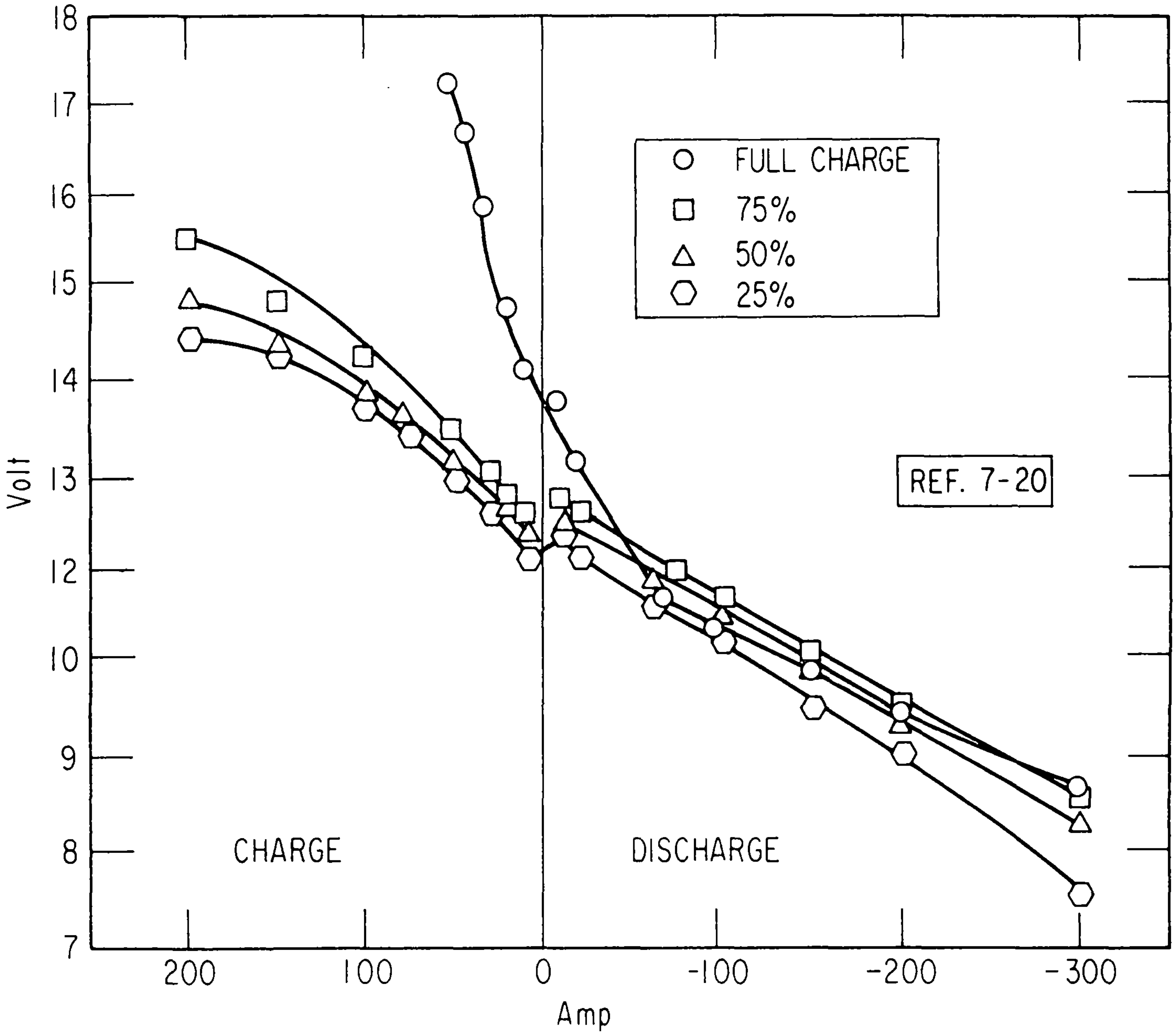


Figure 7-15. SLI Lead-Acid Battery Charge/Discharge Characteristics

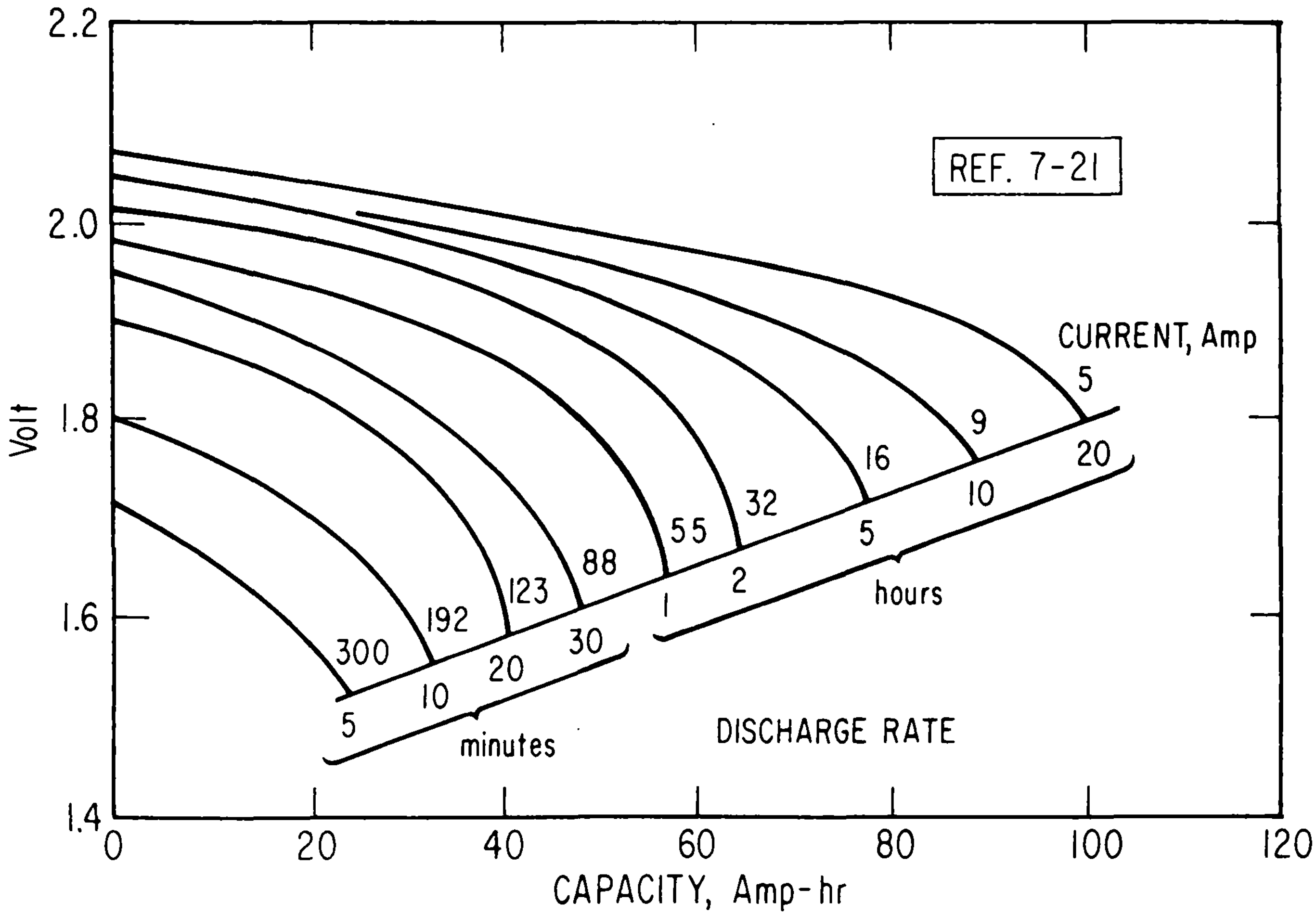


Figure 7-16. Discharge-Voltage Curves and Number of Ampere-Hours Available at Various Rates of Discharge

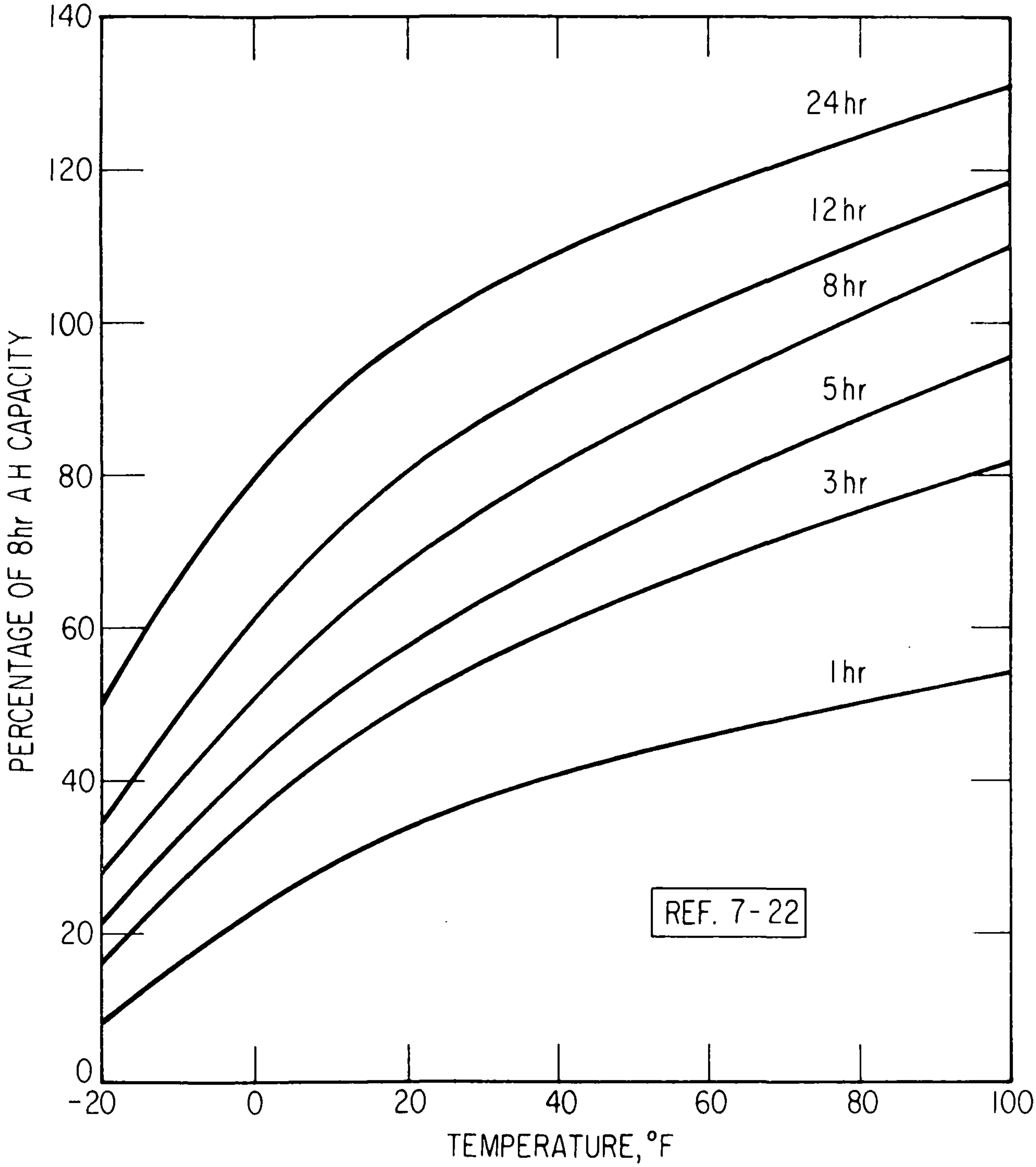


Figure 7-17. Temperature Correction Curve for Stationary Batteries

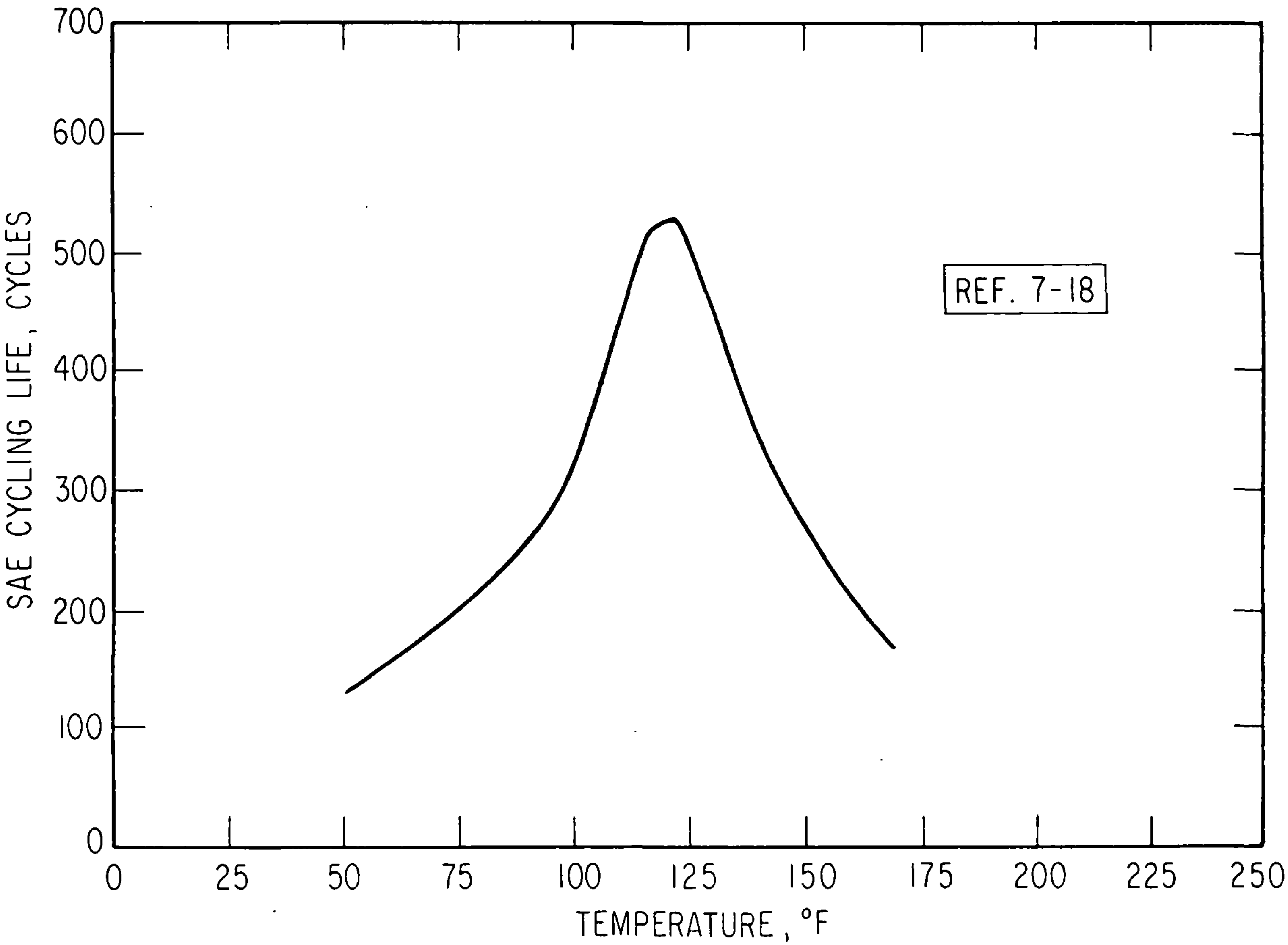


Figure 7-18. Effect of Temperature on Cycling Life

in the bottom of the battery. With time the materials can build up to the point where the plates become shorted across the bottom edges.

The third type of failure mode involves separator failure. This failure mode is normally associated with service where considerable shock and vibration are present.

The venting of hydrogen with the carryover of sulfuric acid, while not truly a failure mode, presents an operational problem. Sulfuric acid produces corrosion of wiring and metal parts located in the vicinity of the battery. Hydrogen represents an explosive hazard which must be considered in large capacity or enclosed installations, and, as a further problem, represents a loss of water from the electrolyte.

7.5.1.3 Battery Advancements

7.5.1.3.1 Low Maintenance

Recently, there has been progress in overcoming the shortcomings of the SLI battery and in producing a maintenance-free or low-maintenance battery in which water addition will no longer be necessary and there is no longer any acid carryover. Several approaches have been followed in achieving the low-maintenance battery. Generally, the new batteries use grids of calcium or pure lead instead of antimony and this reduces the tendency to gas. By having the voltage regulator maintain charge voltage below 2.3 volt, gassing is reduced or eliminated.

7.5.1.3.2 Battery Design Changes

Adoption of calcium grids, while causing the lead-acid battery to be more expensive, does allow for thinner plates so that more surface area can be packed into the same volume. It has been estimated that surface area can be increased by a factor of three. Fig. 7-19 shows the effect of increasing surface area upon both the power and energy density of a battery with comparison to an SLI battery.

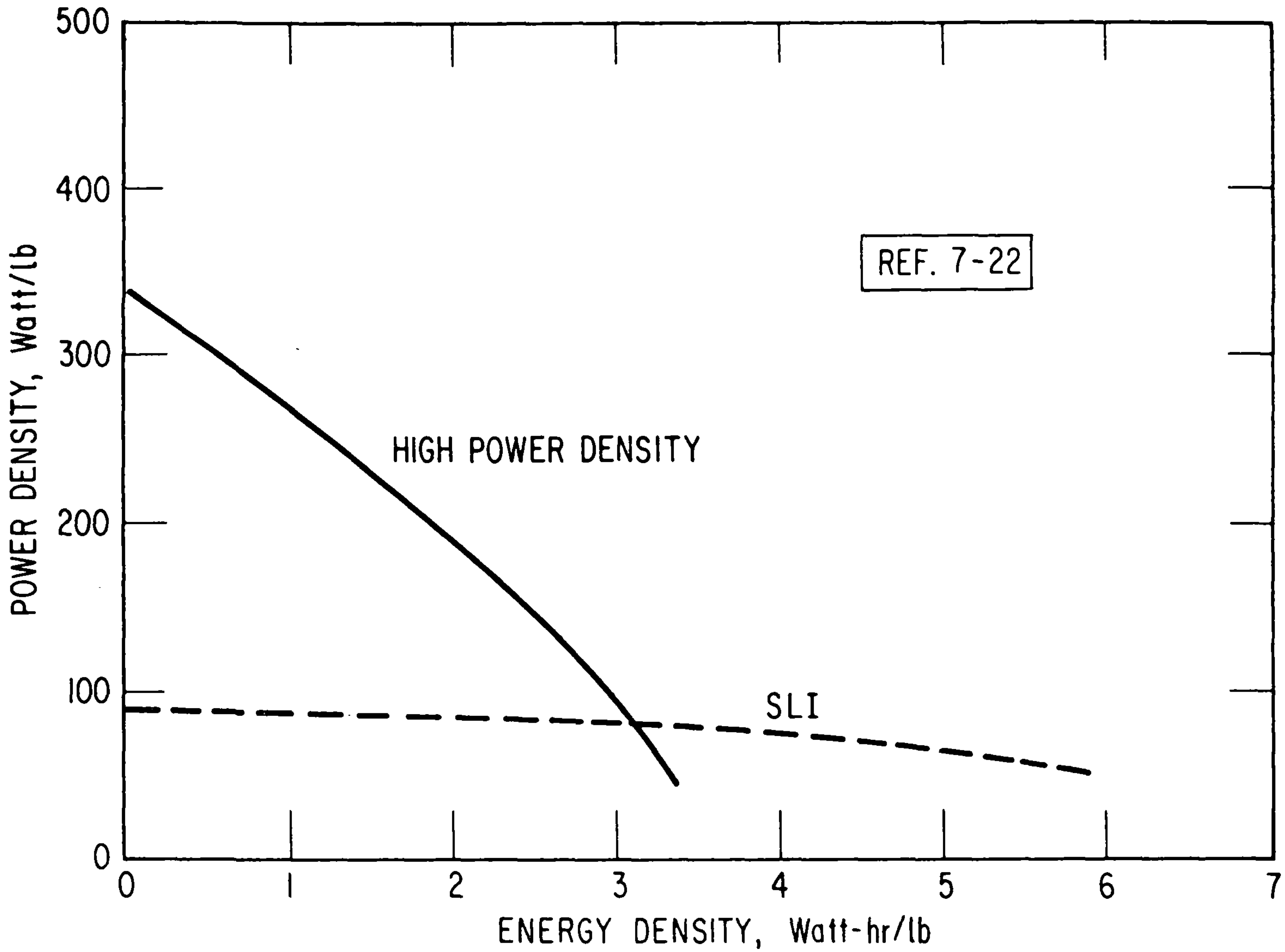


Figure 7-19. High Power Density Battery Compared to SLI Battery

The dead short circuit current of a common SLI battery is above 1000 amp with the instantaneous power density approximately 200 to 250 watt/lb. The limitation to higher currents is the resistance of the terminals and posts which eventually melt. It is also probably true that much of the heating in the lead-acid battery is caused by the resistance of the posts and terminals, so it would be recommended that the battery be redesigned to provide a lower resistance current path. At the same time, redesign of the case to use plastics rather than hard rubber should provide a savings in weight and volume which can be reflected as higher power or energy density. Several companies have designed semi-transparent plastic cases which can be used to indicate electrolyte level.

As is well known, the active lead oxide paste is retained within an inactive lead grid which accounts for about 30 percent of the positive plate weight. Several new developments indicate that the grid weight might be decreased substantially and provide other beneficial effects. Dr. Samuel Ruben of Ruben Laboratories has recently patented (Ref. 7-10) a titanium nitride grid which provides a stronger grid with good conduction, thermal, and electrical characteristics. Besides reducing weight, thinner plates (to 20 mils thick) have been fabricated using this new grid material. To make the grids, titanium sheet is nitrided and then expanded to form the grid which is then pasted. Dr. Ruben has indicated (Ref. 7-11) that although he is having problems in retaining the paste at high temperatures (140°F) development prospects remain encouraging.

Another development by L.D. Babusic and others at Bell Telephone Laboratories (BTL) (Ref. 7-1) uses a pure lead grid which is shaped to maximize strength, and electrical and thermal conductivity. The BTL battery, which uses cone-shaped grids with a concentric and radial spike pattern, is claimed to increase battery life to over 30 years.

The lead industry is continuously investigating new types and forms of lead oxide, some of which show promise in increasing the specific power and energy density of the lead-acid battery.

7.5.1.3.3 Battery Control

Successful long term battery operation depends upon adequate voltage control. Generally, battery charge voltage must be maintained below 2.3 to 2.5 volt in order to prevent gassing and the consumption of water. However, these low voltages will reduce the rate at which the battery can be recharged.

Recently, it has been announced that rapid recharge of lead-acid batteries is possible; reports are made that batteries can be fully recharged at high voltage in under 15 min with low temperature rise and no water consumption. This procedure initially causes electrolysis of the electrolyte and the formation of oxygen and hydrogen bubbles on the cathode. Then, current reversal causes the bubbles to be driven off the electrode into the electrolyte where they rise to the volume maintained above the cell. At this location, a small tungsten electrode causes the gases to recombine and form water (Ref. 7-12). High-rate recharge would be a desirable feature for the hybrid vehicle battery since this might allow recovery of the energy produced by regenerative braking and improved use of energy produced by the heat engine/generator.

Thermal control of the batteries should be considered in any battery system. Because of the high current drain and recharge rates, it may be necessary to examine the use of active coolant systems.

7.5.2 The Nickel-Cadmium Battery

7.5.2.1 Present Battery Characteristics

A curve showing the discharge characteristics of a nickel-cadmium cell is shown in Fig. 7-20. Temperature effects upon the nickel-cadmium cell are shown in Fig. 7-21 for both charge and discharge. The effect of temperature and depth of discharge on cycle life are shown in Fig. 7-22 while Fig. 7-23 shows the trend in cycle life with depth of discharge at constant temperature.

7.5.2.2 Advanced Battery Characteristics

For high-rate applications the bipolar battery, illustrated in Fig. 7-24, is of interest. In this battery, the nickel of one cell and the cadmium of the next cell are plated onto the two sides of a conducting thin sheet or substrate. These cell elements are stacked together, with a suitable separator and electrolyte in between to form a pile or stack. The current enters the battery at one end and exits at the other, and because of the direct current path the internal impedance is low (accounting for the high-rate capability). With adequate voltage control, cell gassing can be minimized so that sealed construction can be used.

Power densities as high as 1000 watt/lb on a microsecond basis and 300 watt/lb for minutes have been described for bipolar cells. Voltage-current characteristics for a 100 in.^{2} cell are given in Fig 7-25.

It has been estimated that a battery capable of 450 amps at 70 volt (31,500 watt) with 10 amp-hr capacity would weigh 125 lb and would occupy a volume of 24 x 12 x 8 in. Since the energy density of this battery at low rate is only about 7 watt-hr/lb, the rating appears conservative.

7.5.2.3 Industrial Capability

The following concerns are, or have been, involved in the fabrication and manufacture of nickel-cadmium batteries.

Bright Star Industries, Clifton, New Jersey

Catalyst Research Corp., Baltimore, Maryland

Eagle Picher Corp., Joplin, Missouri

ESB/Ray-O-Vac, Madison, Wisconsin

General Electric, Gainesville, Florida

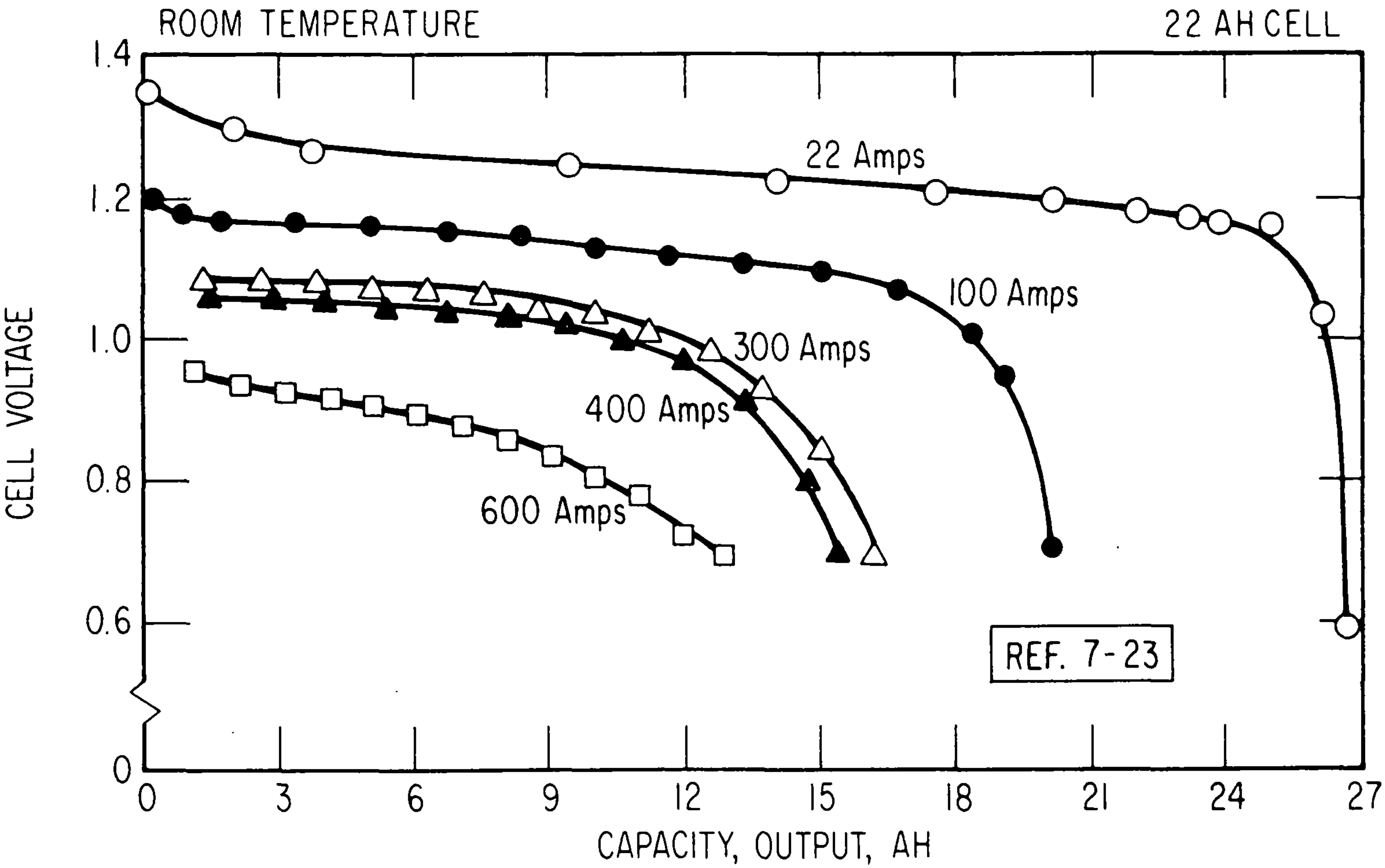


Figure 7-20. Cell Voltage vs AH Capacity at Various Discharge Rates for Nickel-Cadmium Cell

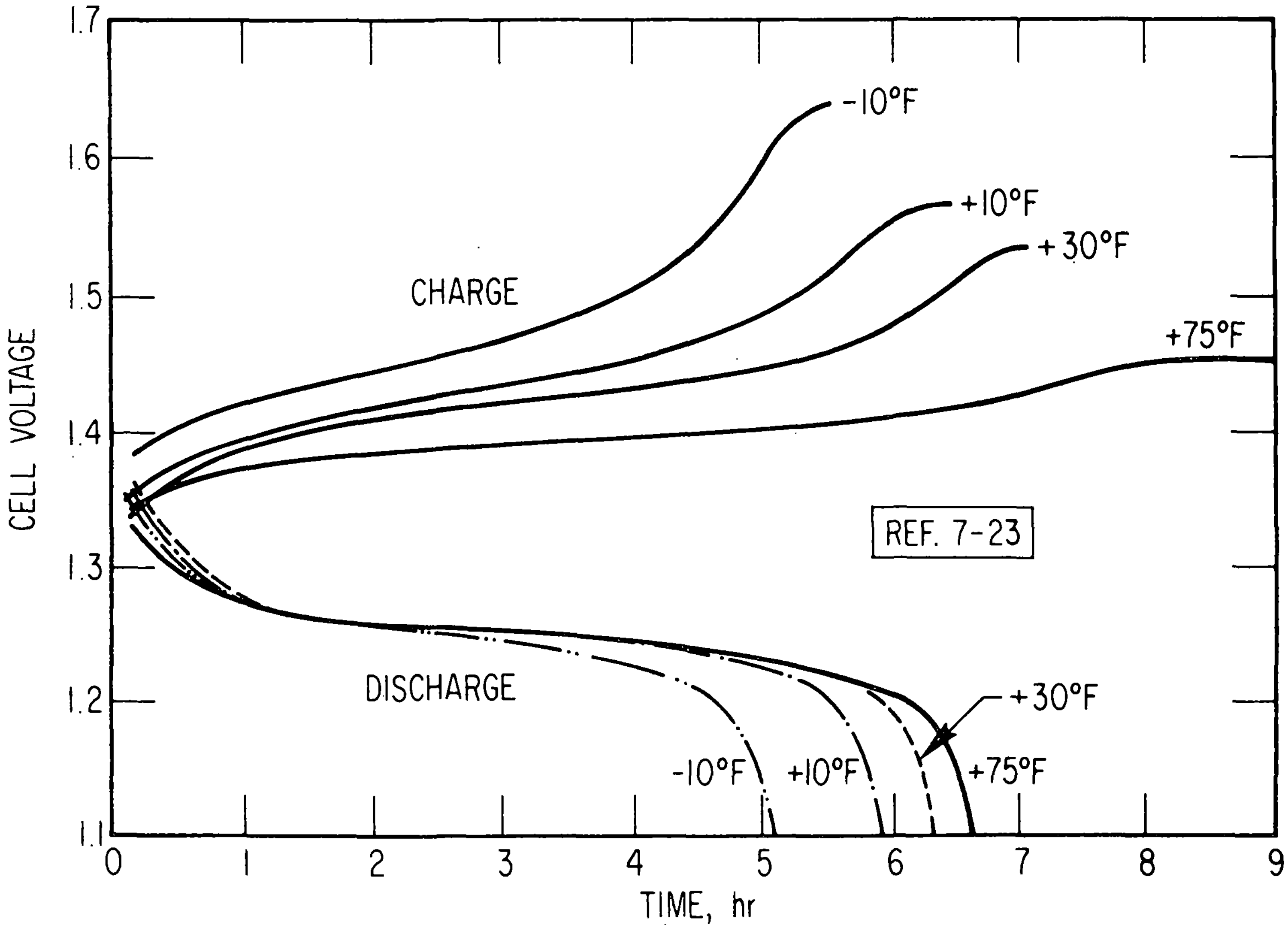


Figure 7-21. Typical Voltage Characteristics at C/6 Charge and Discharge Rates for Sealed Nickel-Cadmium Cells

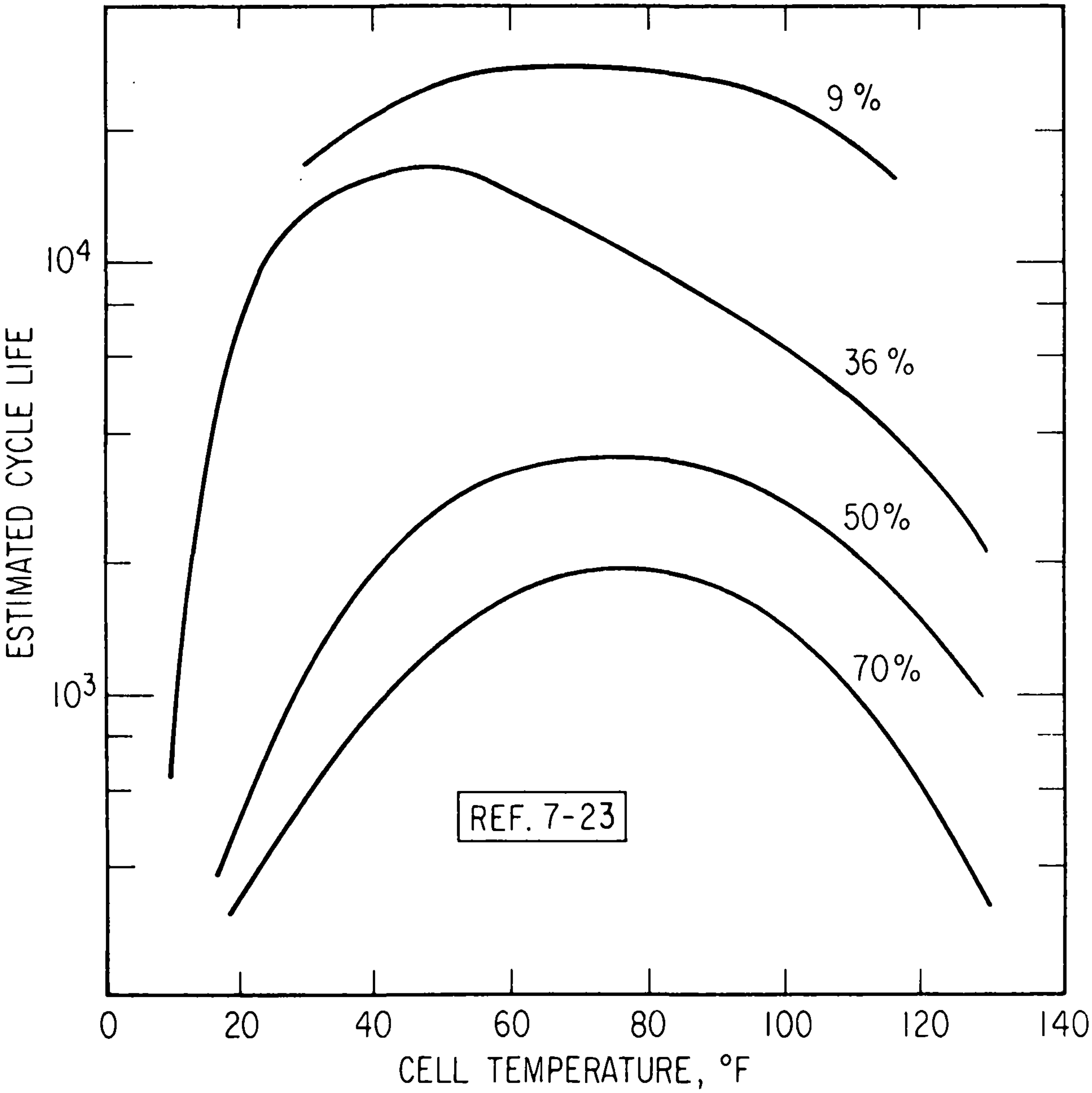


Figure 7-22. Estimated Cycle Life of Sealed Nickel-Cadmium Cells as a Function of Temperature for Various Depths of Discharge

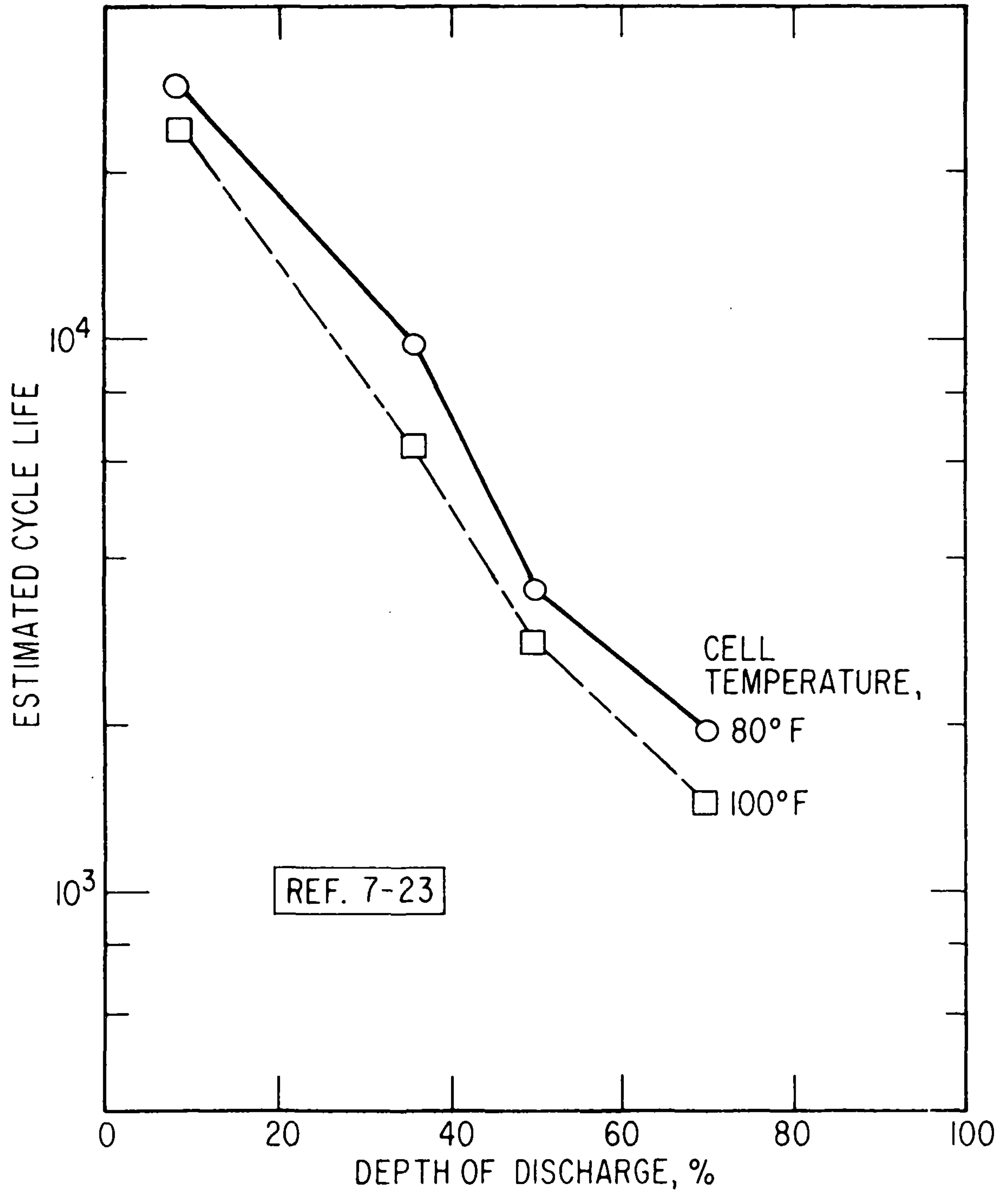
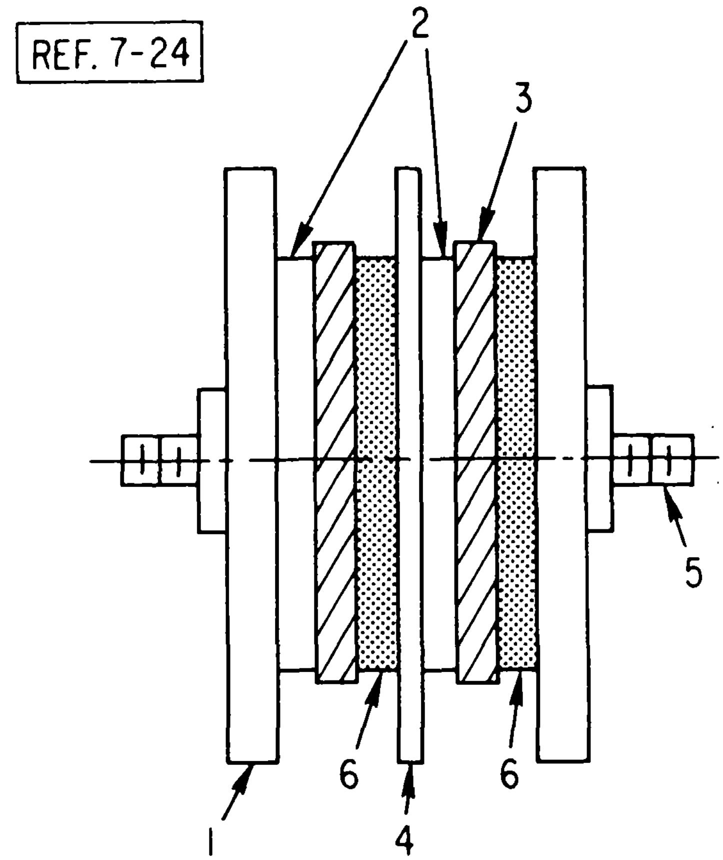


Figure 7-23. Estimated Cycle Life of Sealed Nickel-Cadmium Cells as a Function of Depth of Discharge for Various Temperatures



1. SUBSTRATE FOR END PLATE
2. SINTERED MATRIX - NiOXIDE POS. PLATES
3. SEPARATOR
4. SUBSTRATE FOR INTERIOR PLATE
5. END TERMINAL
6. SINTERED MATRIX - CADMIUM NEG. PLATES

Figure 7-24. Construction of a Bipolar Battery

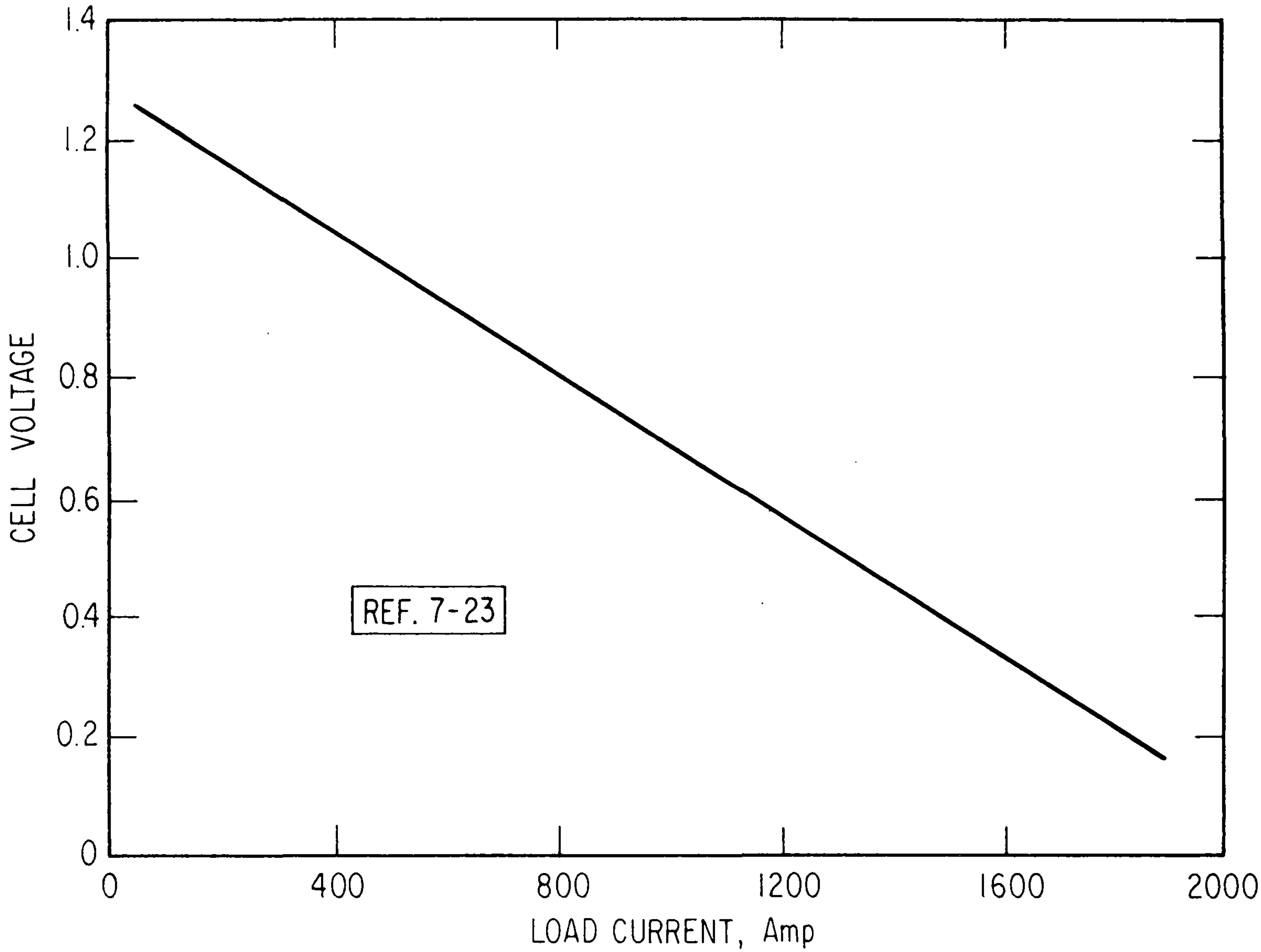


Figure 7-25. Voltage-Current Relationship for a 100 in.^{2} Electrode Area Nickel-Cadmium Bipolar Battery

Gould-National, St. Paul, Minnesota (Ni-Cd Division)

Gulton Industries, Metuchen, New Jersey

NIFE, Copiague, New York

Marathon (Sonotone Corp.), Elmsford, New York

Sprague Electric, North Adams, Massachusetts

Sylvania Electric Products, New York

Union Carbide/Eveready, New York

7.5.2.4 Availability

The availability of cadmium is the principal problem affecting use of nickel-cadmium batteries in the hybrid vehicle. Some thought has been given to the possibility of overcoming the scarcity of cadmium by recycling nickel-cadmium batteries. The nickel electrode is little affected during battery usage so that it can be reused simply by washing and rewrapping with new separators. It thus is possible to rebuild a nickel-cadmium battery by replacing the cadmium electrode, the separators, and the electrolyte and, since the remainder of the battery is reusable and the cadmium replaced is recoverable, the cost might not be prohibitive. This would imply that once a stable relation between new batteries being sold and old batteries being turned in or scrapped was reached there would be only a limited demand upon the primary sources of cadmium metal.

This argument has several weaknesses. If it is assumed that roughly fifty million hybrid vehicles are on the road, each containing 200 lb of batteries, the total cadmium supply requirement would be about one billion pounds or about 30 times the annual world production of cadmium. Also, the cost of nickel-cadmium batteries for a vehicle will be prohibitive. Based upon material costs, a nickel-cadmium battery will be about 10 to 15 times more expensive than a comparable lead-acid battery or about $10 to $15/lb; therefore, batteries for a hybrid car would cost between $2000 and $3000.

It is therefore judged that nickel-cadmium batteries, while attractive in an engineering sense, should not be considered further for widespread use in the personal family or commuter car versions of the hybrid vehicle so long as suitable alternate battery systems exist or can be developed. However, the nickel-cadmium battery may be considered for special situations where cost and availability are not of concern. Such applications may be found with the limited production levels associated with intracity buses and delivery/postal vans.

7.5.3 The Nickel-Zinc Cell

The nickel-zinc cell was first described as early as 1898, appeared briefly as an experimental railway battery (Drumm Cell) in the 1930's, and, in the early 1950's, the Russians became active in its development. About 1966, the U.S. Army Electronic Components Laboratory began an investigation of the battery as a replacement for the nickel-cadmium battery in field radio equipment. At the same time, industry had become interested in the battery because of success with zinc electrodes in orbiting spacecraft batteries and the fact that this battery, among the batteries composed of silver and nickel, cadmium and zinc, was the least developed and should exhibit many desirable characteristics as listed below:

a. High Energy Density — The theoretical energy density of the nickel-zinc cell is 50 percent greater than the cells for either nickel-cadmium or lead-acid. As high as 30 watt-hr/lb might be anticipated.

b. Discharge Voltage — The 1.71 discharge voltage of the nickel-zinc battery is high compared to the 1.30 voltage of the nickel-cadmium cell. The lower voltage compared to the lead-acid cell reduces the tendency towards gassing and makes hermetic sealing feasible.

c. Stable Voltage — Voltage is stable with depth of discharge since the electrolyte is not involved in the reaction.

d. Temperature — Electrode characteristics are favorable under high rate and low temperature conditions.

e. Deep Discharge — Fairly good cycle life has been demonstrated under complete or high-discharge cycling. Complete discharge is not injurious to further operation.

f. Cost — The cost is expected to be about twice that of lead-acid and considerably lower than nickel-cadmium batteries.

g. Availability — No materials are used which are, or are likely to be, in critical supply.

h. Safety — The materials used in the battery are relatively nonhazardous.

7.5.3.1 Performance Characteristics

The theoretical energy density of the nickel-zinc cell is 146 watt-hr/lb compared to 95.3 for the nickel-cadmium battery. Some difference of opinion exists in the literature over the voltage of the zinc half-cell reaction so that the theoretical energy density for the nickel-zinc cell can be as high as 148 watt-hr/lb. Since the practical density for a vented battery is about 20 percent of theoretical, it would be expected that 30 watt-hr/lb is a realistic goal.

In Fig. 7-26, test results for a nickel-zinc battery are shown in comparison to those of a high power density lead-acid battery. At 150 watt/lb, the energy density is 17 watt-hr/lb rather than the 2.5 watt-hr/lb of the lead-acid battery which indicates that the nickel-zinc battery might be designed for even higher power densities or that improved cycle life might be expected because of a lower depth of discharge.

Figure 7-27 shows the characteristics of a five amp-hr cell at 75°F and Fig. 7-28 shows the effect of ambient temperature upon performance. Data are lacking on the characteristics of the battery at high ambient temperatures. Tests have been conducted where a battery was soaked at 300°F followed by recharge and discharge. After 149 hr at 300°F, battery capacity after recharge was only 40 percent lower than its initial capacity, indicating low permanent damage.

Charge characteristics for the nickel-zinc battery are plotted in Fig. 7-29. Below 1.88 to 1.92 charge voltage, there is no evidence of gassing. At higher charge voltages, the characteristics are nearly the same as those of the

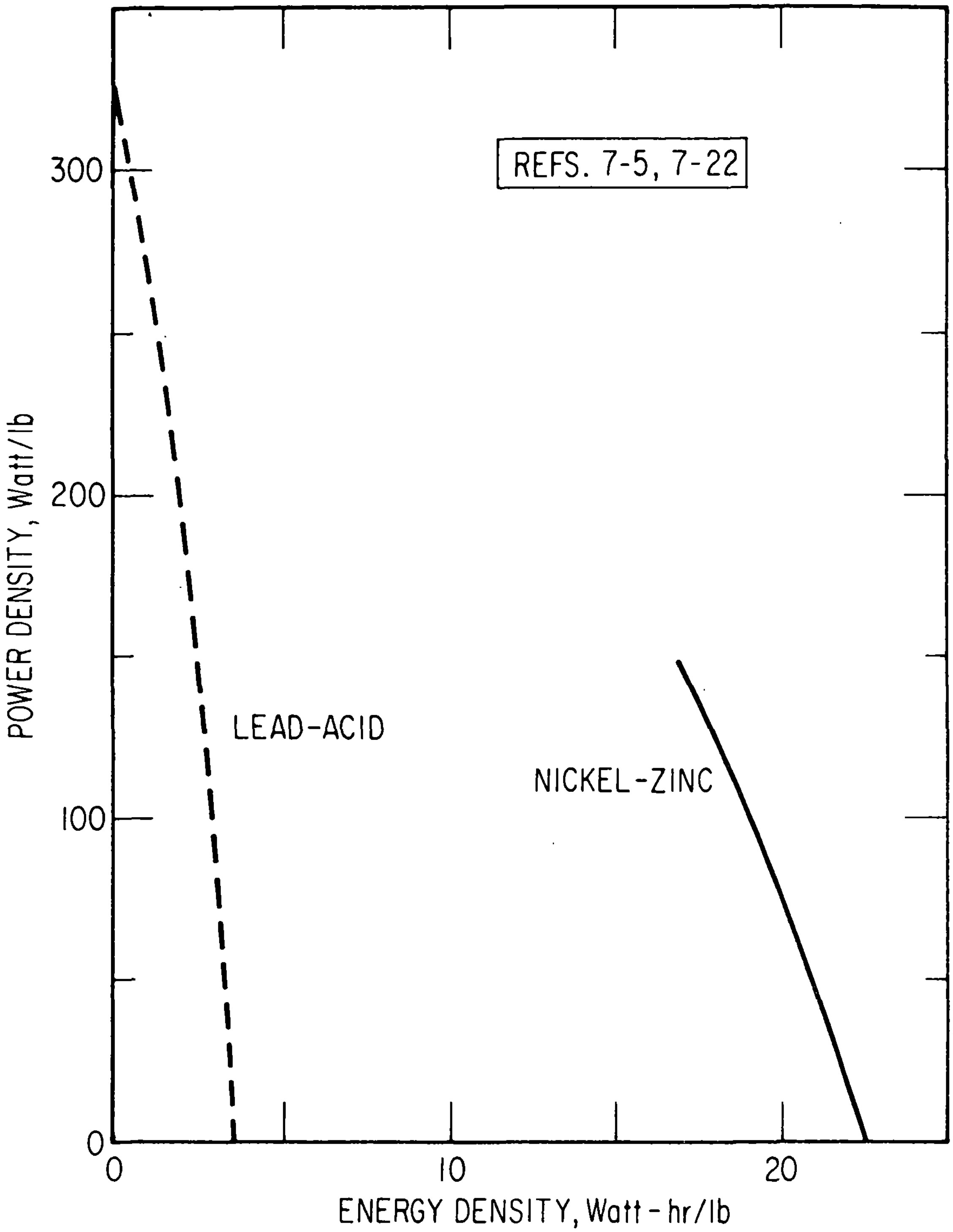


Figure 7-26. Comparison of Energy/Power Density Characteristics of High Power Density Lead-Acid and Nickel-Zinc Batteries

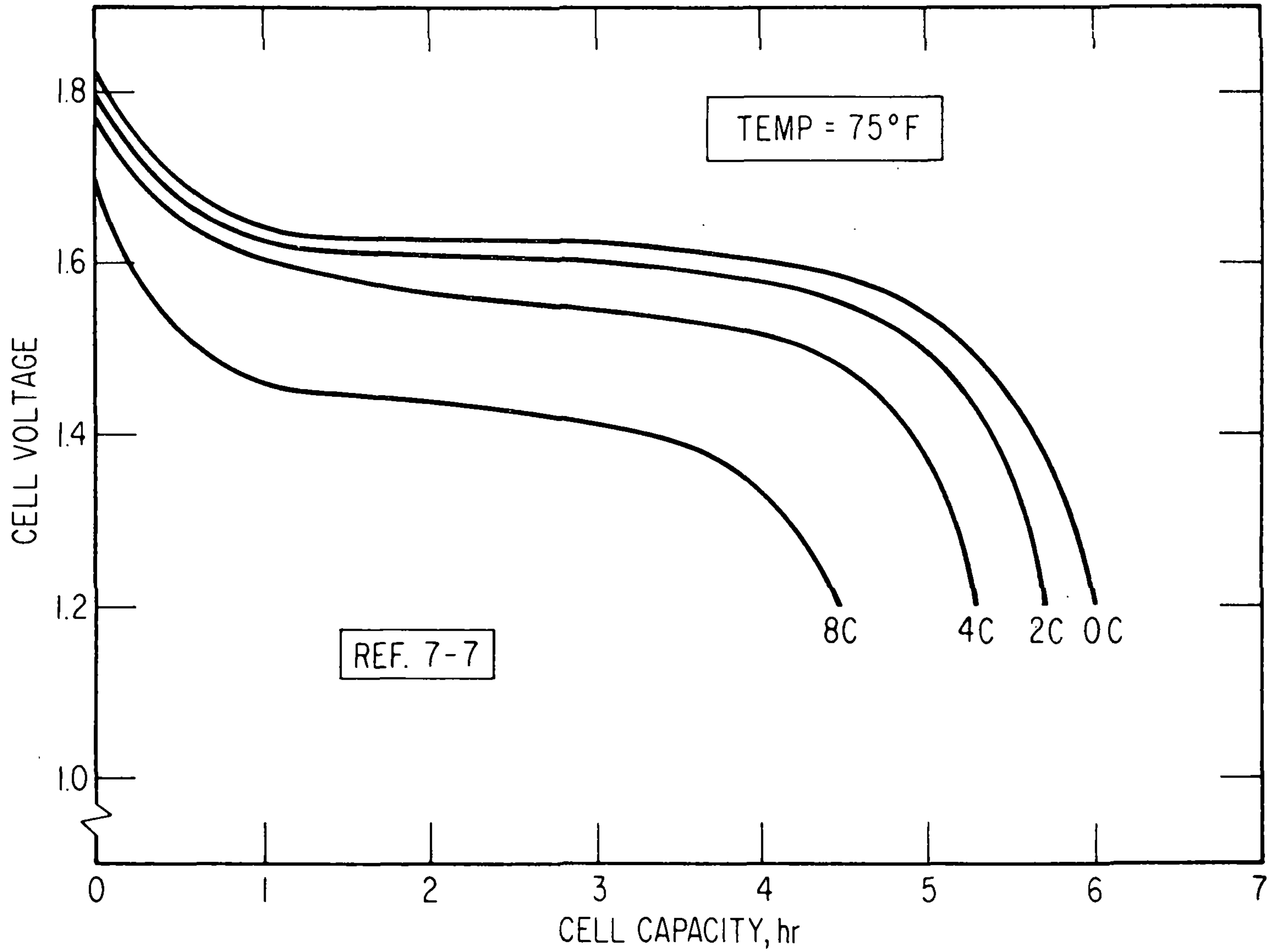


Figure 7-27. Nominal Discharge Characteristics of 5 AH Nickel-Zinc-Cell at 75°F

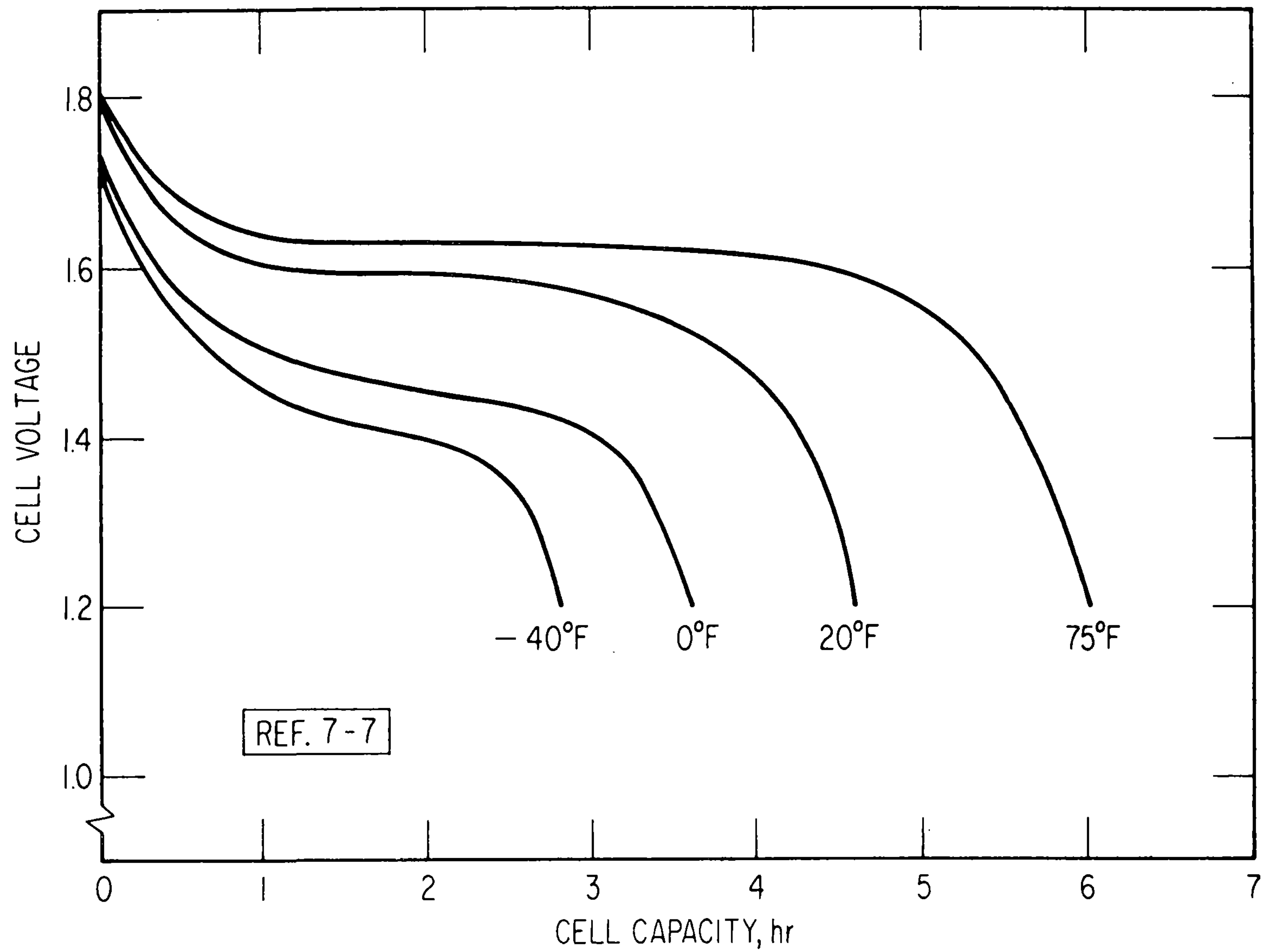


Figure 7-28. Nominal Discharge Characteristics of 5 AH Nickel-Zinc Cell at Various Temperatures

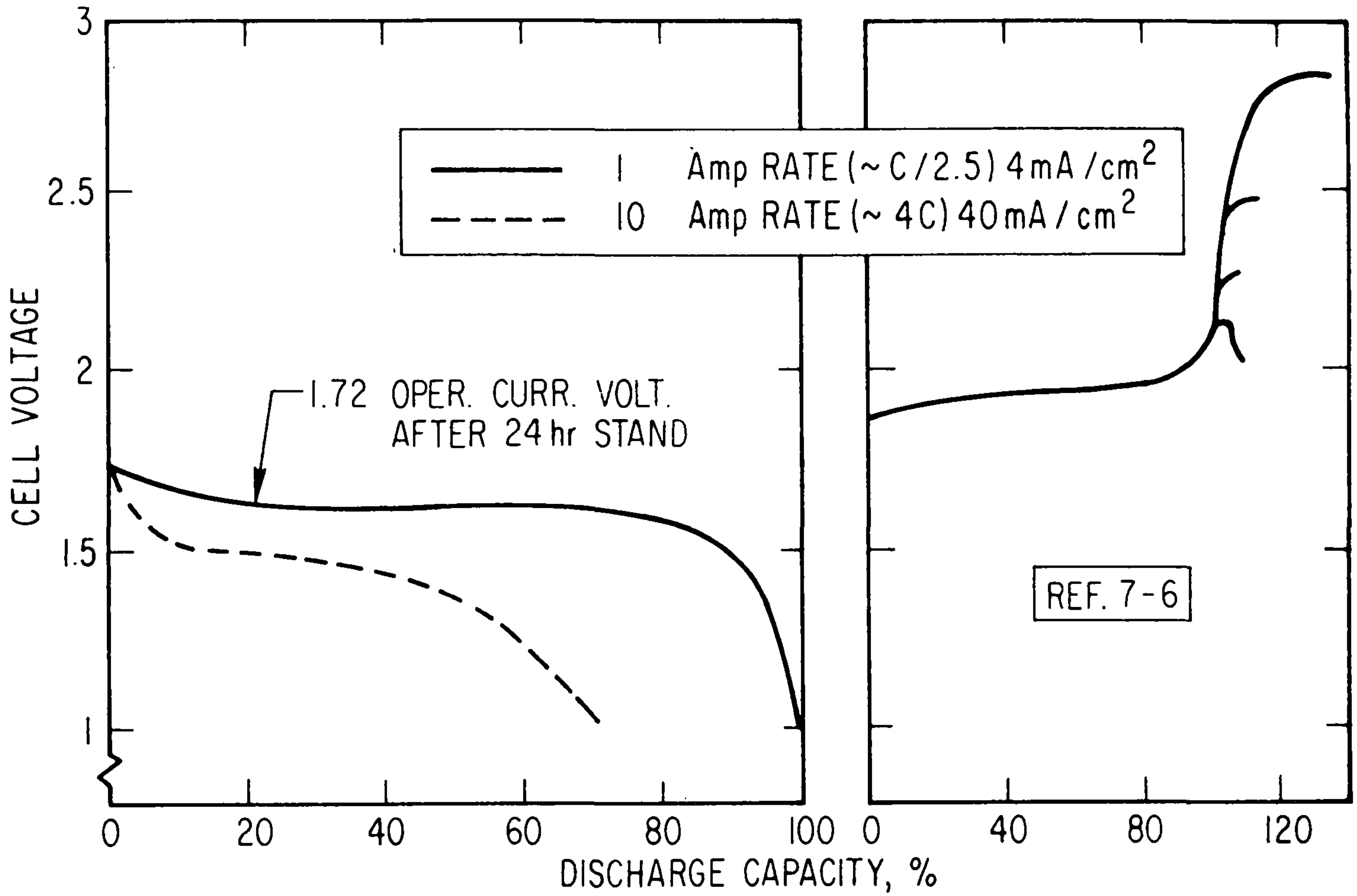


Figure 7-29. Representative Charge-Discharge Characteristics of 2- 2.5 AH Nickel-Zinc Cells

nickel-cadmium cell in that an equilibrium of generation and recombination is achieved so that pressure rise is limited and hermetic sealing is possible.

At present, the nickel-zinc cell characteristically degrades with number of operating cycles as indicated in Figs. 7-30 and 7-31. Tendency of the zinc to move around and loss of electronic contact are given as reasons for the loss of capacity. Dendritic shorting has not been a problem with the nickel-zinc cell.

A repeatable life of 150 complete discharges is currently possible. This is comparable to the capability of the lead-acid cell and is perhaps one-fifth that of the nickel-cadmium cell. Atomics International (Ref. 7-13) and Professor Edwin Gilliland of the Chemical Engineering Department at MIT, (Refs. 7-3 and 7-14) report a deep depth cycle life of greater than 500. Little is known of the cycle life under float operation and at low depths of discharge.

7.5.3.2 Industrial Capability

Government support of nickel-zinc battery development has been confined to the U.S. Army Electronic Components Laboratory, Fort Monmouth, New Jersey, with Mr. Martin Sulkes the cognizant project officer. Since the success of the nickel-zinc battery is associated with development of reliable zinc electrodes and separators, the directed effort of the NASA Lewis Research Center and the Air Force Aero Propulsion Laboratory in these areas should also be considered. Principal industrial efforts have been centered among the following organizations:

Eagle Picher, Joplin, Missouri

Energy Research, Bethel, Connecticut

E.S.B., Raleigh, North Carolina

General Electric, Gainesville, Florida

General Telephone and Electronics, Bayside, New York

Gould Battery Company, Minneapolis, Minnesota

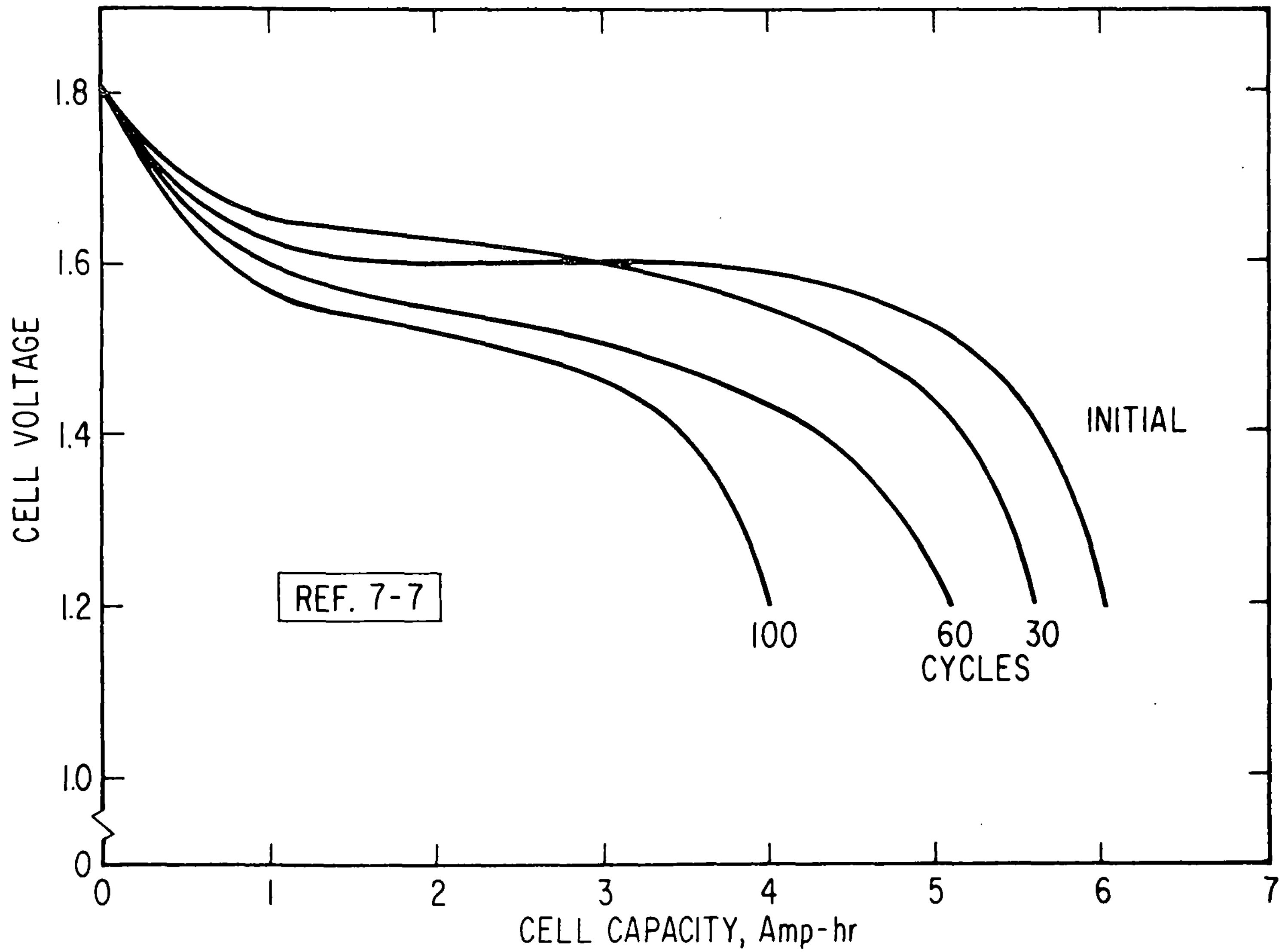


Figure 7-30. Nominal Discharge Characteristics of 5 AH Nickel-Zinc Cell at 75-F After Cycling Tests to 75% DOD at C/2.5 Rate

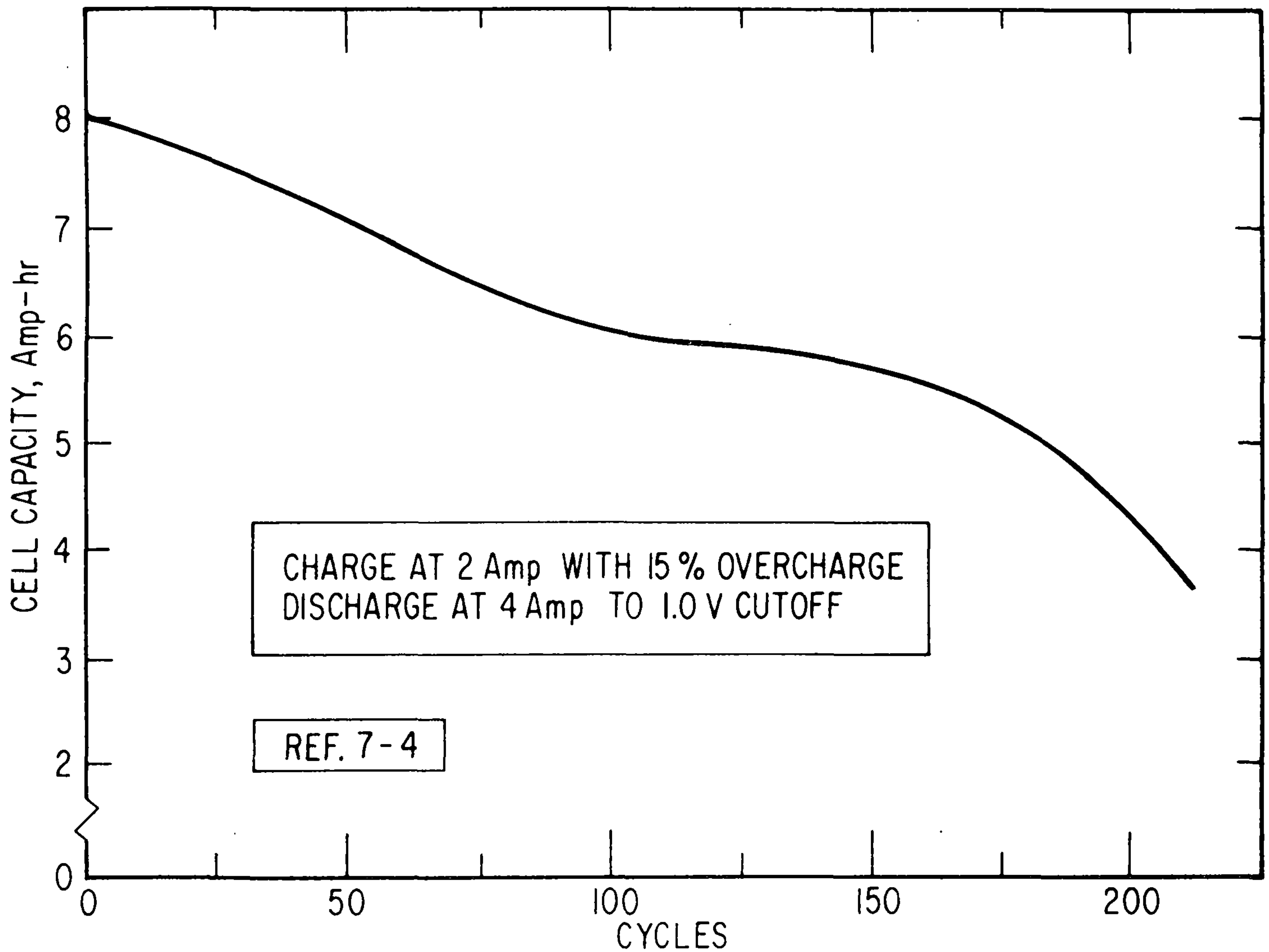


Figure 7-31. Capacity vs Cycles for Nickel-Zinc Cells

Gulton Industries, Inc., Metuchen, New Jersey

NAR-Atomics International, Canoga Park, California

Whittaker Corporation, Denver, Colorado

Yardney Electric, Pawcatuck, Connecticut

Other organizations such as Delco-Remy and Leesona-Moos might be involved in any long range development program due to their experience with zinc electrodes.

7.5.4 Summary of Battery State of the Art

Characteristics of the three candidate batteries are summarized in Table 7-5. Based on the theoretical voltages, it is apparent that the lead-acid battery would have one advantage over the others in that the number of cells would be lower to achieve a given voltage. This is particularly important for the hybrid vehicle in that the assigned voltages are on the order of 200 volt for the passenger cars and vans and 400 volt for the buses. (The checking of fewer cells should reduce the maintenance time for the battery system.)

In terms of theoretical energy density, it appears that the nickel-zinc battery has a definite potential advantage compared to the other two.

Relative costs as indicated on the chart are based upon raw material costs of the electrodes. Construction of a lead-acid battery is relatively simple compared to the other two battery types so the differences could be even more pronounced. It is expected that construction of the nickel-zinc battery will follow closely the configuration of the nickel-cadmium battery.

Demonstrated power density of the three battery types is perhaps inconclusive because of the limited test time at these values. They are shown only to indicate that some rather high power densities have been obtained. Those listed have been achieved for times of only seconds; even higher power densities have been demonstrated on a microsecond basis. The nickel-zinc battery seems to demonstrate an encouraging combination of both high energy and

Table 7-5. Characteristics of Secondary Batteries Selected for Investigation

CHARACTERISTICS	BATTERY TYPE		
	Lead-Acid	Nickel-Cadmium	Nickel-Zinc
Voltage, Theoretical	2.041	1.299	1.735
Theoretical Amp-hr, AH/lb	37.8	73.3	85.4
Theoretical Energy Density, W-hr/lb	77.1	95.3	148.1
Relative Cost*	1.0	11.1	2.5
Demonstrated Capability			
Power Density, W/lb			
Bipolar	226	450	-----
Prismatic	328	100	180
Energy Density, W-hr/lb			
Bipolar	-----	8.3	-----
Prismatic	23.3	18	22
Volumetric Power Density, W/in.^{3}			
Bipolar	22.5	33	-----
Prismatic	1.82	1.5	1.6
Cycle Life at Depth of Discharge			
Bipolar	200,000 at 2.0% at 14C rate**	-----	-----
Prismatic	19,000 at < 5% †	34,000 at 25% at C/10 rate ††	2,300 at 30% §

Based on Active Material Cost

Ref. 7-15

† Ref. 7-16

†† Ref. 7-17

§ Ref. 7-18

power density. (Except for the bipolar nickel-cadmium and the nickel-zinc batteries, the power and energy densities were measured under different conditions and, hence, are not consistent with one another.)

Volumetric power density indicates a very strong advantage to bipolar type construction. No strong advantage is indicated for any battery type.

There are few test results which would indicate the potential cycle life of the candidate batteries at low depths of discharge. Since high-cycle life has been achieved with what might be considered a highly experimental lead-acid bipolar cell, the more highly developed prismatic cell could prove to perform even better. Figure 7-32 is an attempt based on rather limited data to define the relationship between cycle life and depth of discharge. While a straight line on this chart may not represent a true situation, it is generally accepted in predicting cycle characteristics of nickel-cadmium batteries. Other observers, however, feel that a line of constant energy may be more significant and one is shown plotted in this figure. It is obvious that such a result, if true, would indicate reduced cycle life characteristics for a hybrid electric vehicle.

7.6 DESIGN AND DEVELOPMENT GOALS

Because of the problem of cost and availability associated with nickel-cadmium batteries, they have been excluded from consideration in the remainder of the discussion in Section 7.

7.6.1 Vehicle Battery Requirements

For each class of hybrid vehicle, battery requirements are influenced by numerous factors. Furthermore, once battery power and energy requirements are defined, the power and energy density requirements will be established by available powertrain weight less the weight of all other powertrain components and subsystems. Since component and subsystem weights will increase with the severity of imposed vehicle specifications (e.g., acceleration, peak cruise speed), the resulting reduction in allocated battery weight will correspondingly increase the severity of battery design requirements.

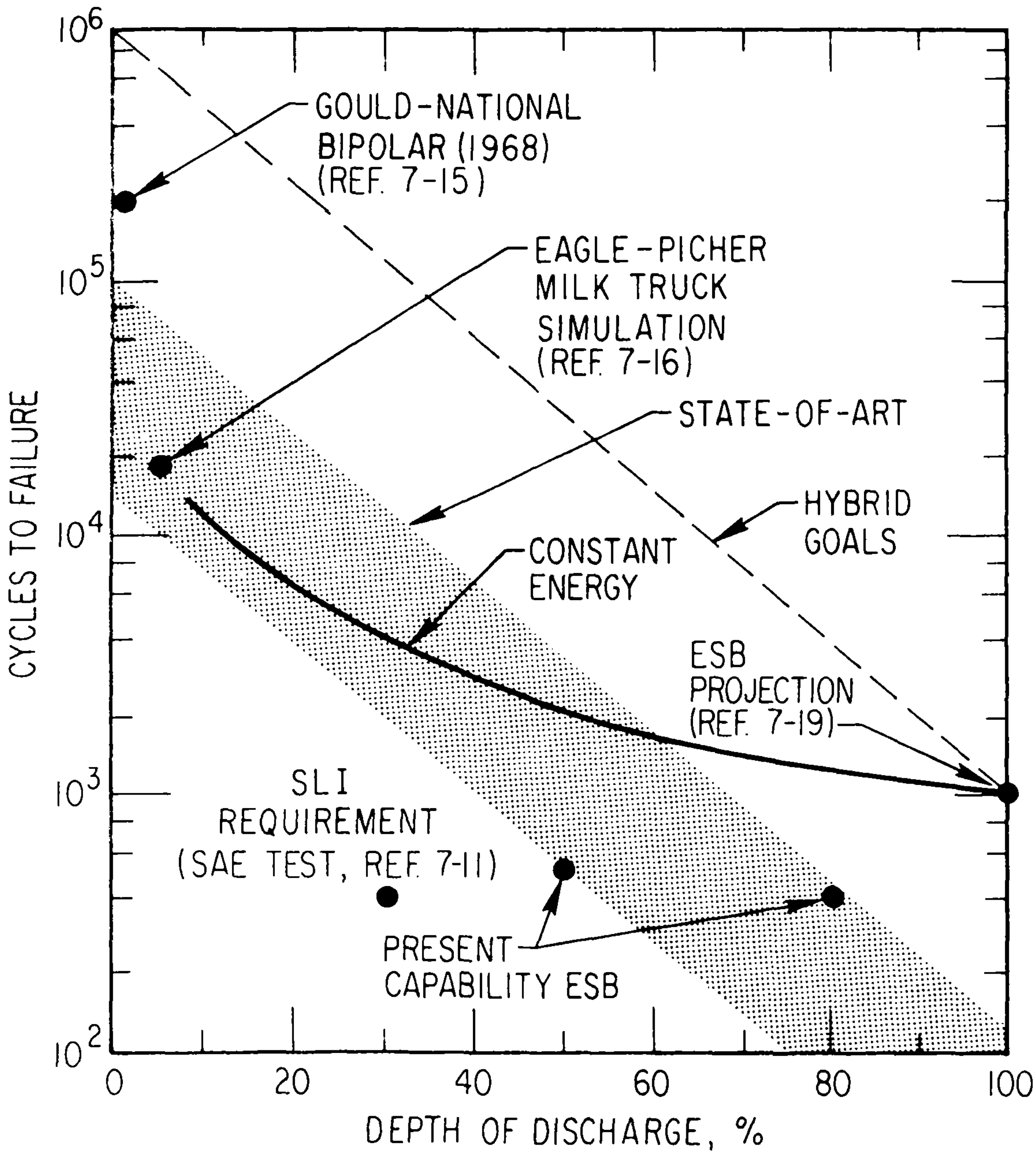


Figure 7-32. Cycle Life of Lead-Acid Batteries

As indicated in Section 7.4, the hybrid vehicle battery must be capable of a large number of cycles and of sustaining high currents for a period of up to 20 sec. As an additional constraint the battery should be capable of accepting all the charge current delivered by the generator. An indication of requirements for the family car is given by the results of analyses using the DHEW Emission Driving Cycle and Design Driving Cycle (Table 7-6).

The development goals of 500-hr design life or 100,000 mi of service are arbitrary but in keeping with a requirement that the hybrid vehicle be economical and not present any major impact upon vehicle operation. This development goal would correspond roughly to the design life of the engine.

Table 7-6. Battery Development Goals, Family Car (Lead-Acid Battery)

DHEW Emission Driving Cycle	Hybrid Vehicle-Family Car Development Goals
38 AH battery operating for 1370 sec	38 AH battery at less than 5% depth of discharge to deliver up to 500 amp
7.5 vehicle miles with 73 battery charge/discharge cycles and 1.34 AH depth of discharge	5000 hr of operation and 100,000 vehicle miles with 975,000 charge/discharge cycles
Design driving cycle	
38 AH battery delivered 462 amp	

The life of 975,000 cycles is well beyond any demonstrated performance and thus becomes a critical development goal. If the DHEW Driving Cycle is representative of average driving requirements (a conservative assumption), then each event in the cycle would occur over 13,000 times during the life of the batteries.

The data in Table 7-6 can also be used to provide additional information on battery operation. With the battery capacity established to provide low emission levels and the maximum current established from the Design Driving Cycle, an assumed power density for the lead-acid battery of 150 watt/lb would result in a requirement for 680 lb of batteries. In a similar fashion for the nickel-zinc battery, with a power density of 250 watt/lb and 30 AH of capacity needed, the battery weight would be 410 lb. At these weights and capacities the energy density of the lead-acid and nickel-zinc batteries would be 16.5 and 22.5 watt-hr/lb, respectively.

In a similar fashion the weights of batteries and their requirements for the other vehicles can be obtained and these are indicated in Table 7-7. Use of the parallel configuration will reduce the battery capacities shown by about ten percent.

7.6.2 Battery Development

7.6.2.1 Cell Capacity

Required battery capacities as determined in this study are preliminary and further battery testing and design analysis are needed before the final battery capacities can be specified. These capacities have been based upon cell models which must be verified; furthermore, information is needed on such considerations as degradation with time. Changes to the system as well as differences between actual and design efficiencies will also have a bearing upon the final battery capacity which might be established.

Table 7-7. Summary of Battery System Design and Operating Characteristics, Series Configuration

CHARACTERISTICS	PASSENGER CAR				VAN				BUS			
	Commuter		Family		Low-Speed		High-Speed		Low-Speed		High-Speed	
	Lead-Acid	Nickel-Zinc	Lead-Acid	Nickel-Zinc	Lead-Acid	Nickel-Zinc	Lead-Acid	Nickel-Zinc	Lead-Acid	Nickel-Zinc	Lead-Acid	Nickel-Zinc
Generator Current, amp	17	17	38	38	53	53	54.5	54.5	100	100	79	79
Maximum Charge Voltage	340	340	340	340	340	340	340	340	680	680	680	680
Battery Capacity, AH	20	16	38	30	40	32	40	32	90	70	79	65
No. of Cells	147	170	147	170	147	170	147	170	294	340	294	340
Maximum Discharge Current, amp	148	148	462	462	402	402	402	402	506	506	142	142
Minimum Battery Voltage	220	220	220	220	220	220	220	220	440	440	440	440
Battery Weight, lb	300	170	680	410	720	440	720	440	2700	1500	2400	1400
Battery Volume, ft^3	2.3	1.5	4.3	2.7	4.6	2.9	4.6	2.9	20.4	12.5	18.0	11.6
Battery Power Density												
W/lb	109	192	150	250	127	211	127	211	83	149	27	45
$W/in.^3$	8.3	13.3	15.0	24.4	11.3	18.1	11.3	18.1	6.3	10.4	2.0	3.1
Battery Energy Density												
W-hr/lb	20	29	16.5	22.4	16.4	22.3	16.4	22.3	20	29	20	29
$W\text{-}hr/in.^3$	1.5	2.0	1.5	2.0	1.5	2.0	1.5	2.0	1.5	2.0	1.5	2.0

Battery capacity is established by the required charge acceptance to produce low emission levels and the current discharge capability to achieve designated vehicle accelerations. A reduction in the acceleration requirements would have an extremely pronounced effect upon battery capacity. For the family car the installed battery capacity could be reduced by one third if the DHEW Emission Driving Cycle acceleration levels formed the specifications.

7.6.2.2 Power and Energy Density

Based on the previous discussions, it appears that power densities of 150 and 250 watt/lb are reasonable objectives for the lead-acid and nickel-zinc batteries, respectively. It has been determined that on the hybrid vehicle the battery depth of discharge is 5 percent or less; therefore, these power densities need only be achieved for shallow discharge. To the power density requirements must be added the other constraints of energy density, cycle life, and degradation. The power and energy densities as determined by available battery weight in the powertrain for each class of vehicle can be higher or lower than these values depending on the heat engine used and the allowable powertrain weight (see Section 11).

7.6.2.3 Hybrid Battery Life

As indicated for the family car about one million charge/discharge cycles would be expected for each 100,000 mi of operation. Even if the design life were to be decreased to 50,000 mi, the resulting number of cycles would be well beyond any existing demonstrated capability. It is therefore important that this criterion be prominent in any development program. Furthermore, the battery should be exposed to a reproduction of the current levels and charge/discharge periods resulting from operation over the emission driving cycle in a hybrid vehicle, rather than to the normally accepted procedure of repetitive cycles having constant magnitude and duration.

7.6.2.4 Charge Acceptance

An important finding in the computer studies to determine minimum emission designs has been the importance of charge acceptance characteristics indicated by the cell model. Charge acceptance is not ordinarily treated with enough importance in any battery development program and care should be taken that the charge characteristics are fully explored and acceptance rates maximized; charge acceptance will be particularly important if regenerative braking is utilized. Laboratory studies might be conducted to determine whether the maximum allowable voltage limits can be raised in order to improve charge acceptance.

7.6.2.5 Thermal Control

A desirable characteristic of any cell or battery is that the dominant reaction proceed reversibly. Also, the current efficiency of the battery should be as near 100 percent as possible. In the absence of any side reactions, efficiency of a reversible battery may be described in terms of voltages. Therefore, charge efficiency, η_C, of a battery is given by

$$\eta_C = \frac{E_R}{E_C}$$

where E_C is the charge voltage and E_R is the reversible or theoretical cell voltage and the discharge efficiency, η_D, is given by

$$\eta_D = \frac{E_D}{E_R}$$

where E_D is the discharge voltage. Overall battery efficiency, η_B, can be expressed as the product of the two efficiencies using the average values

$$\eta_B = \eta_C \bullet \eta_D = \frac{E_R}{E_C} \bullet \frac{E_D}{E_R} = \frac{E_D}{E_C}$$

In the analyses of this study the battery current efficiency will be nearly 100% since the limiting charge voltage has been established low enough to avoid the normal side reaction, the electrolysis of water. There will be some thermal flux created in the battery due to discharge inefficiency, but this should not be too much of a problem since the battery will normally be operating in a range close to the reversible voltage.

A more critical thermal problem associated with the battery will be that due to overcharge. The charge control system should include provisions to prevent the continued charging of a fully charged battery. To accomplish this, the battery will need a control system based upon sensors which measure battery voltage and possibly temperature. While a sensor which could measure battery state-of-charge would be ideal for this purpose, such sensors are not yet sufficiently accurate to serve as a control input.

It is quite important that battery thermal control be an integral part of any battery development program. Adequate thermal control may be as important to the successful performance of a hybrid vehicle battery as any other factor.

7.6.3 Summary of Development Goals

A preliminary listing of the development goals for the hybrid batteries is given in Table 7-8. Battery weights shown result from stipulating that for lead-acid and nickel-zinc batteries, respectively, the power densities not exceed 150 watt/lb and 250 watt/lb and that energy densities not exceed 16.5 watt-hr/lb and 22.5 watt-hr/lb; similarly the volumes shown were dictated by limits of 1.5 and 2.0 watt-hr/in.^{3}.

It should be emphasized that these specifications are preliminary since a more detailed system analysis of a specific vehicle might indicate that certain requirements should be relaxed such as that of acceleration since this factor has dominated battery sizing for the family car. Another factor which has influenced the battery sizing has been the cell modeling. It might be possible to develop better batteries or perhaps the combination of life and cycle requirements may not cause a relaxation in requirements. It might also be that the combination of energy and power density along with the life and cycle requirements may not be achieved which could also reflect on these development goals.

7.7 RECOMMENDED BATTERY DEVELOPMENT PROGRAM

With battery design and development goals tempered by current and projected technology, a development program oriented toward the hybrid electric vehicle can be evolved. A suggested program is schematically described in Fig. 7-33.

7.7.1 General Battery Development (Phase I)

7.7.1.1 Development of Lead-Acid Battery for Hybrid Electric Vehicle

First, it is apparent that lead-acid battery technology exists which could provide an acceptable interim battery for the hybrid electric vehicle by 1973. While the technology exists, the hardware does not, so it will be necessary in a Phase I Program to redesign and repackage the battery into a form more suited for the hybrid vehicle and to make such minor improvements which

Table 7-8. Summary of Battery Design Specifications, Series Configuration

CHARACTERISTICS	PASSENGER CAR		VAN		BUS	
	Commuter	Family	Low-Speed	High-Speed	Low-Speed	High-Speed
Battery Capacity, AH:						
Lead-Acid	20	38	40	40	90	79
Nickel-Zinc	16	30	32	32	70	65
Maximum Battery Weight, lb:						
Lead-Acid	300	680	720	720	2700	2400
Nickel-Zinc	170	410	440	440	1500	1400
Maximum Battery Volume, ft^3:						
Lead-Acid	2.3	4.3	4.6	4.6	20.4	18.0
Nickel-Zinc	1.5	2.7	2.9	2.9	12.5	11.6
Minimum Battery Voltage, v	220	220	220	220	440	440
Maximum Charge Voltage, v	340	340	340	340	680	680
Life, yr	5	5	5	5	5	5
Charge/Discharge Cycles	500,000	1,000,000	1,000,000	1,000,000	1,000,000	500,000
Cycle Distribution	1% OF CYCLES, 5% DOD, 12C* (5 sec) & 5C 2% OF CYCLES, 2% DOD, 6C (5 sec) & 3C 7% OF CYCLES, 1% DOD, 2C (2 sec) & C 10% OF CYCLES, 0.5% DOD, C (2 sec) & C/2 80% OF CYCLES, 0.25% DOD, C/2				3% OF CYCLES, 5% DOD, 6C (5 sec) & 3C 7% OF CYCLES, 1% DOD, 2C (2 sec) & C 10% OF CYCLES, 0.5% DOD, C (2 sec) & C/2 80% OF CYCLES, 0.25% DOD, C/2	
Maximum Rate	C/10* (60 min) + 12C (5 sec)				C/10 (60 min) + 6C (5 sec)	
Temperature, °F:						
Normal	30-120					
Capability	0-160					
Maximum Charge Rate	>C (lead acid) >1.3C (nickel-zinc)					
Maintenance	2 yr minimum					

*
$$C/N = \frac{\text{Capacity}}{\text{Time to 100\% Discharge}} = \frac{\text{Ampere Hours}}{\text{Hours}} = \text{Amperes}$$

DOD — Depth of discharge

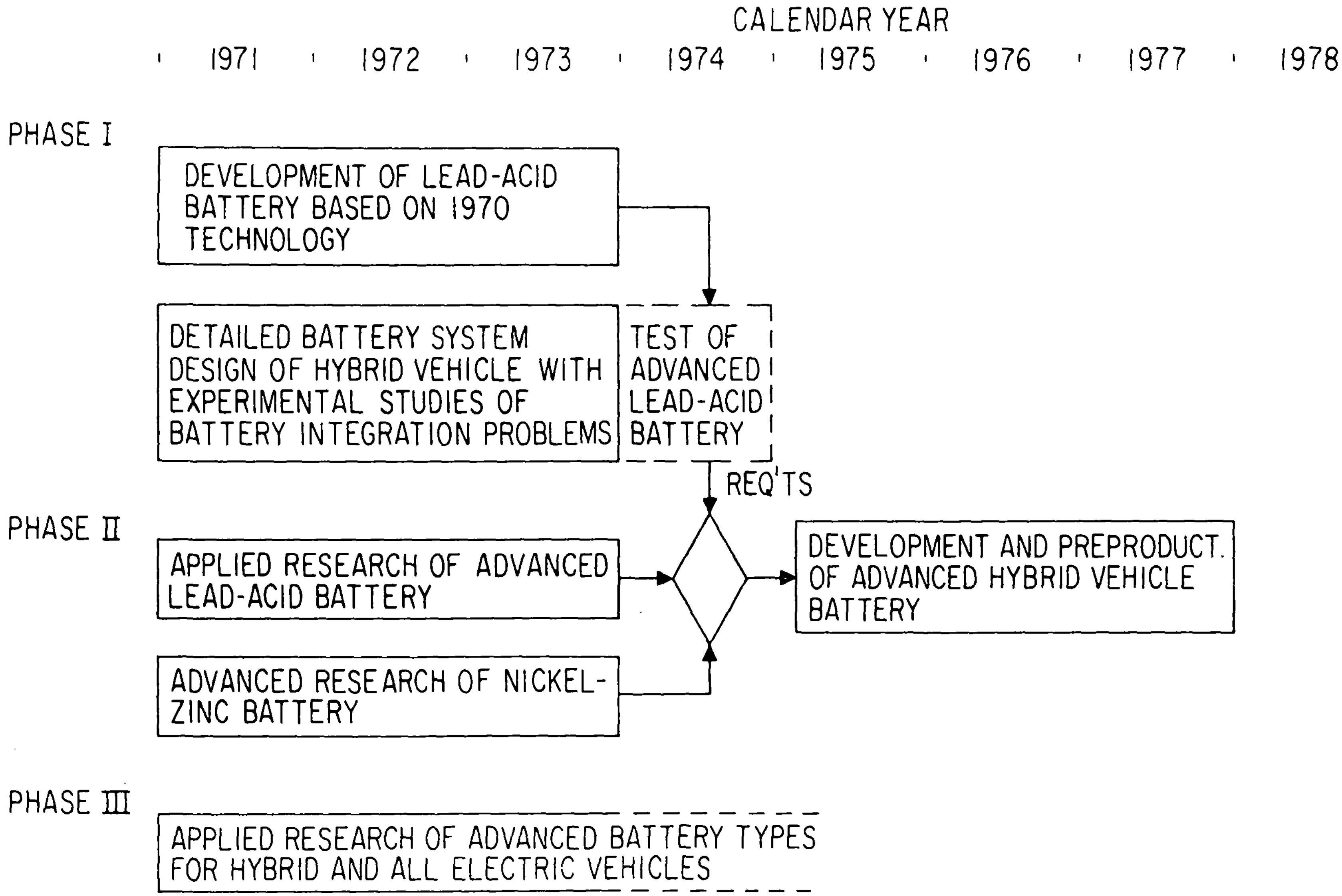


Figure 7-33. Battery Development Program Schedule

will not add undue risk or expense. An objective of this program would be to provide battery performance equivalent to or better than that characterized in Fig. 7-1.

7.7.1.2 Hybrid Electric Vehicle Battery Simulation and Analysis

At the same time it would be worthwhile to conduct additional studies of the hybrid electric vehicle using more accurate battery simulations, and driving profiles more representative of the wide variety of vehicle usage throughout its lifetime. This would allow a better evaluation of charge acceptance, thermal effects, charge control, cycle life, and other potential problems. As the more advanced cells and batteries become available, these could then be tested and compared with the results obtained in these more detailed studies.

7.7.2 Advanced Battery Development (Phase II)

As the second phase of the battery development program, advanced studies should be undertaken for both the lead-acid and nickel-zinc batteries. Following is a list of possible and suggested tasks which might be included in the Phase II effort.

7.7.2.1 Lead-Acid Battery Development

From available data and through discussions with lead-acid battery manufacturers and cognizant government personnel it is apparent that there is a possibility for much improvement of the lead-acid battery and optimization for the hybrid electric vehicle. Some of these areas where development could be productive are listed below.

a. Increase electrode area per unit volume
   1. Use thinner plates
   2. Use corrugated plates

b. Decrease internal resistance
   1. Redesign posts and internal collectors
   2. Optimize grids for current collection

3. Use bipolar design
4. Decrease electrode spacing
5. Develop new grid alloys
6. Consider stirred electrolyte

c. Investigate new concepts

1. Lightweight grids
2. Plastic cases
3. Low maintenance design
4. Improved lead oxide
5. Stirred electrolyte
6. Cylindrical packaging

d. Improve charge control

1. Use rapid charge systems
2. Increase charge acceptance

7.7.2.2 Nickel-Zinc Battery Development

Most development work accomplished so far with the nickel-zinc battery has been directed towards utilization of its good energy density characteristics and so there has been relatively little effort devoted to determination or development of nickel-zinc battery characteristics for low-energy discharge under controlled float operation. Studies of this nature should be pursued along with the development and determination of optimum charge control methods.

The U.S. Army, which has been developing nickel-zinc batteries, is primarily interested in low-rate, high-depth-of-discharge batteries to replace nickel-cadmium batteries. In battlefield situations, the long cycle life of the nickel-cadmium cell is not needed so the good energy density, limited cycle life, and relatively low cost of the nickel-zinc battery are attractive. As a result, present cell designs may not be configured towards hybrid vehicle characteristics. In like manner, the Army is not interested in systems employing such innovations as a stirred electrolyte battery, even though the performance of the electrolyte is significant, since this kind of battery must sacrifice some energy density. However, the zinc electrode is especially sensitive to concentration polarization effects, so some experimental work with stirred electrolyte systems should be conducted.

The zinc electrode and its separator system are the keys to successful development of the nickel-zinc battery. While some performance improvement can be made in the nickel electrode, the major development emphasis should be on the zinc electrode.

Funding of nickel-zinc batteries may have important consequences elsewhere. Whether the nickel-zinc cell may directly replace nickel-cadmium or LeClanche cells is questionable because of the different voltage range. But the energy and power density capabilities of the nickel-zinc battery are, as a minimum, about 50 percent greater (with possibly reduced cycle life, however) than either nickel-cadmium or lead-acid systems, and this along with attractive cost would provide a significant market incentive.

The two major shortcomings of the nickel-zinc battery are its cost, perhaps two to three times that of a lead-acid battery, and a questionable cycle life. A feature of the nickel electrode is that it is relatively unaffected during life of the battery, and in nickel-cadmium or zinc batteries it is the cadmium or zinc electrodes which degrade. Provided that reasonable development objectives can be met during early development phases, the final nickel-zinc battery design might consider periodic maintenance of the battery. The design should probably allow replacement of the zinc electrode, the separator system, and the electrolyte. Since the major and expensive components of the battery are reusable, the cost of a nickel-zinc battery could possibly be competitive with the lead-acid battery.

Development funding of nickel-zinc batteries should be directed as described above to the following areas in the approximate order of priority shown:

a. Zinc electrode
b. Zinc separator system
c. Stirred electrolyte systems
d. Float characteristics
e. Charge control
f. Low maintenance, salvageable design

7.7.2.3 Pre-Production Phase of Advanced Hybrid Electric Vehicle Battery

After completion of Phase I and Phase II Programs, a better indication of battery requirements will be available and it should be possible to make a decision as to which battery type merited further effort. Development of the selected battery would then proceed into pre-production.

7.7.3 Battery Applied Research (Phase III)

As a concurrent task to Phases I and II, it would be desirable to maintain an applied research program for advanced types of batteries. This effort should be broad in scope and should be directed to develop batteries which might be useful to either the hybrid electric or all-electric vehicle systems.

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SECTION 8

HEAT ENGINE PERFORMANCE CHARACTERISTICS AND OPERATION

CONTENTS

CONTENTS (Continued)

TABLES

FIGURES

FIGURES (Continued)

SECTION 8

HEAT ENGINE PERFORMANCE CHARACTERISTICS AND OPERATION

8.1 INTRODUCTION

The hybrid propulsion concept treated in this study comprises a heat engine/generator power set, electric drive motor(s), and batteries. The basic function of the heat engine is to provide rotational power to the generator. In the parallel configuration, which incorporates a direct mechanical link to the drive wheels, the heat engine may simultaneously provide power to the vehicle drive shaft.

The five candidate thermal engine systems which were examined for possible use in the hybrid concept are:

1. Otto Cycle (Spark Ignition Engine)
2. Diesel Cycle (Compression Ignition Engine)
3. Brayton Cycle (Gas Turbine Engine)
4. Rankine Cycle ("Steam" Engine)
5. Stirling Cycle

Sections 8.2 through 8.6 of this report provide a general description of each system and its design options (where applicable), a discussion of considerations pertaining to heat engine operation in the hybrid mode, and a characterization of engine fuel consumption, weight, and volume properties. Section 8.7 compares and evaluates the relative merits of the alternative engine types. Guidelines for future engine development efforts are discussed in Section 8.8.

8.2 OTTO CYCLE (SPARK IGNITION) ENGINE

8.2.1 General Description

Most spark-ignition engines operate on a reciprocating piston principle in which a piston sliding back and forth in a cylinder transmits power through a connecting rod and crank mechanism to the drive shaft. The Wankel engine substitutes a rotary member for the reciprocating piston, resulting in definite engine weight and volume advantages (See Section 8.2.3). The following remarks on engine operation will be addressed to the four-stroke reciprocating system which is typical of automotive designs worldwide.

The engine thermodynamic cycle is illustrated in Fig. 8-1. The familiar four-stroke sequence of engine operations consists of an intake stroke (terminating at Point 1), a compression stroke (Point 1 to Point 2) followed by ignition and combustion of the charge (Point 2 to Point 3), an expansion or power stroke (Point 3 to Point 4), and an exhaust stroke (Point 4 to Point 1). The fuel charge enters and the exhaust products leave the cylinder through poppet valves operated by a cam mechanism driven by the crankshaft. The charge is ignited by an electric spark, which is timed in relation to the top dead center piston position by speed and manifold pressure controls to ensure maximum performance at different engine rpm and load conditions.

The charge mixture is controlled in conventional engines by a carburetor consisting basically of a venturi, fuel nozzle, and throttle valve. The nominal air/fuel ratio is about 15, but values from 12 to 16 are developed over the normal operating range of the engine. Speed and load control is achieved by altering the position of the throttle valve to restrict the flow of air through the carburetor. A number of foreign manufacturers, including Volkswagen, Opel, Mercedes, Porsche, Volvo, and Triumph, have recently converted to port-type fuel injection systems in their production engines, primarily to avoid characteristically high HC emissions produced by conventional carburetor systems during periods of acceleration and deceleration.

Representative performance curves for an automotive S. I. engine are shown in Fig. 8-2 (Ref. 8-1). Typically, the power output peaks at about 65 percent of maximum rpm and the rated performance of the engine is quoted at this point. Also, the torque curve peaks at about half the speed of the horsepower peak, while the lowest value of specific fuel consumption (SFC) occurs near the midrange of speed.

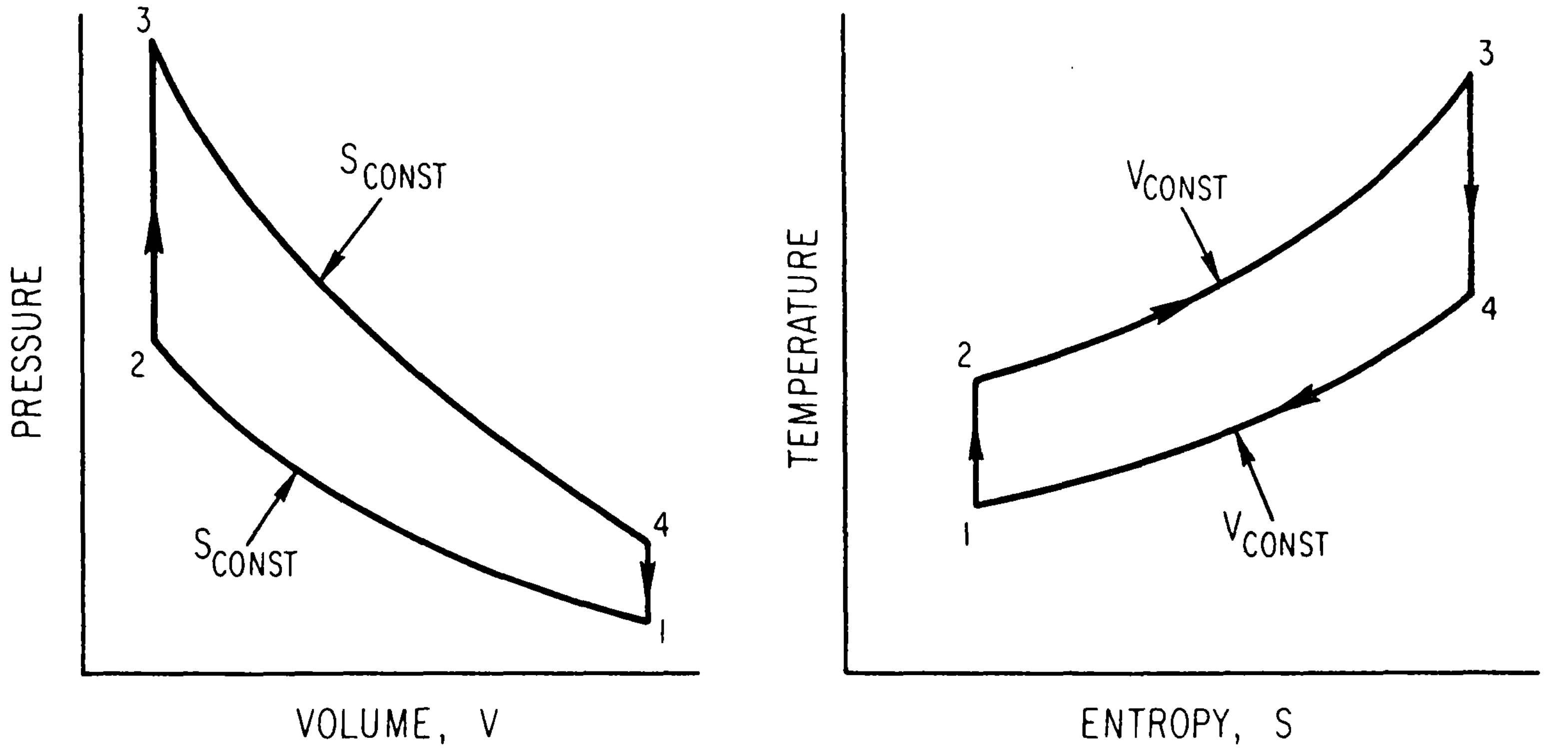


Figure 8-1. The Otto Cycle

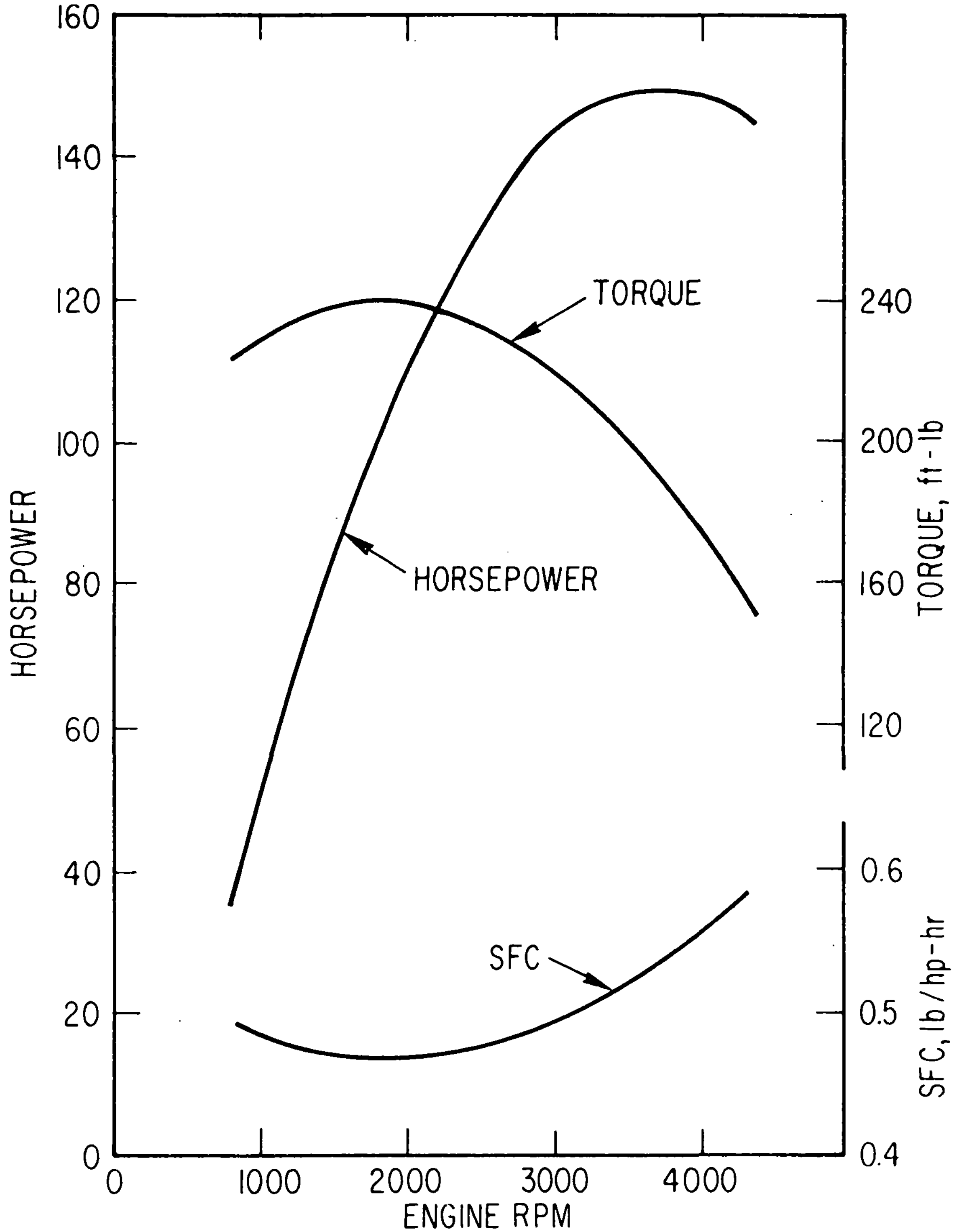


Figure 8-2. Spark Ignition Engine Performance Characteristics (258-CID American Motors Ambassador)

It should be noted that the power curve shown in Fig. 8-2 is based on dynamometer test results obtained with an engine stripped of normal running equipment and accessories, including fan, pump, generator, air cleaner, and conventional exhaust system. When these as well as convenience accessories such as power steering and air conditioning are added to the engine, the peak power output at the flywheel may be reduced by 25 to 30 percent. Auxiliary power requirements for a 135 hp, 232-CID engine are shown in Fig. 8-3 (Ref. 8-2). The air cleaner/exhaust system curve is an estimate based on data from Ref. 8-3. At a vehicle cruise speed of 80 mph, which in this system occurs at an engine rpm of 3450, the loss due to accessories and running equipment is 26 hp. At rated conditions, the loss is 33 hp, or 25 percent of peak rated power.

8.2.2 Hybrid Operation

The curves shown in Fig. 8-2 are typical of the limited performance data normally supplied by the engine manufacturer. They represent S.I. engine performance solely under the conditions where the throttle is fixed in its wide open position and rpm is varied by an adjustment of external load. Normally, the automotive engine operates to accommodate load and speed changes by varying the throttle setting; therefore, a knowledge of engine characteristics over the complete spectrum of throttle settings is essential to the present investigation of hybrid vehicle potentialities.

A complete performance map for a small automotive engine such as might be used in the hybrid family car is presented in Fig. 8-4 (Ref. 8-4). The SFC is plotted versus engine gross horsepower output at constant rpm. The term "gross horsepower" refers to the sum of the flywheel and accessory power quantities. The near-closed throttle position appears at the upper left hand corner of the plot. This region is characterized by high specific fuel consumption, due partly to pumping losses created by throttling the incoming air charge. As the throttle plate is opened at constant rpm, the pressure in the intake manifold increases, the pumping loss decreases, the net engine power output increases, and the SFC declines accordingly.

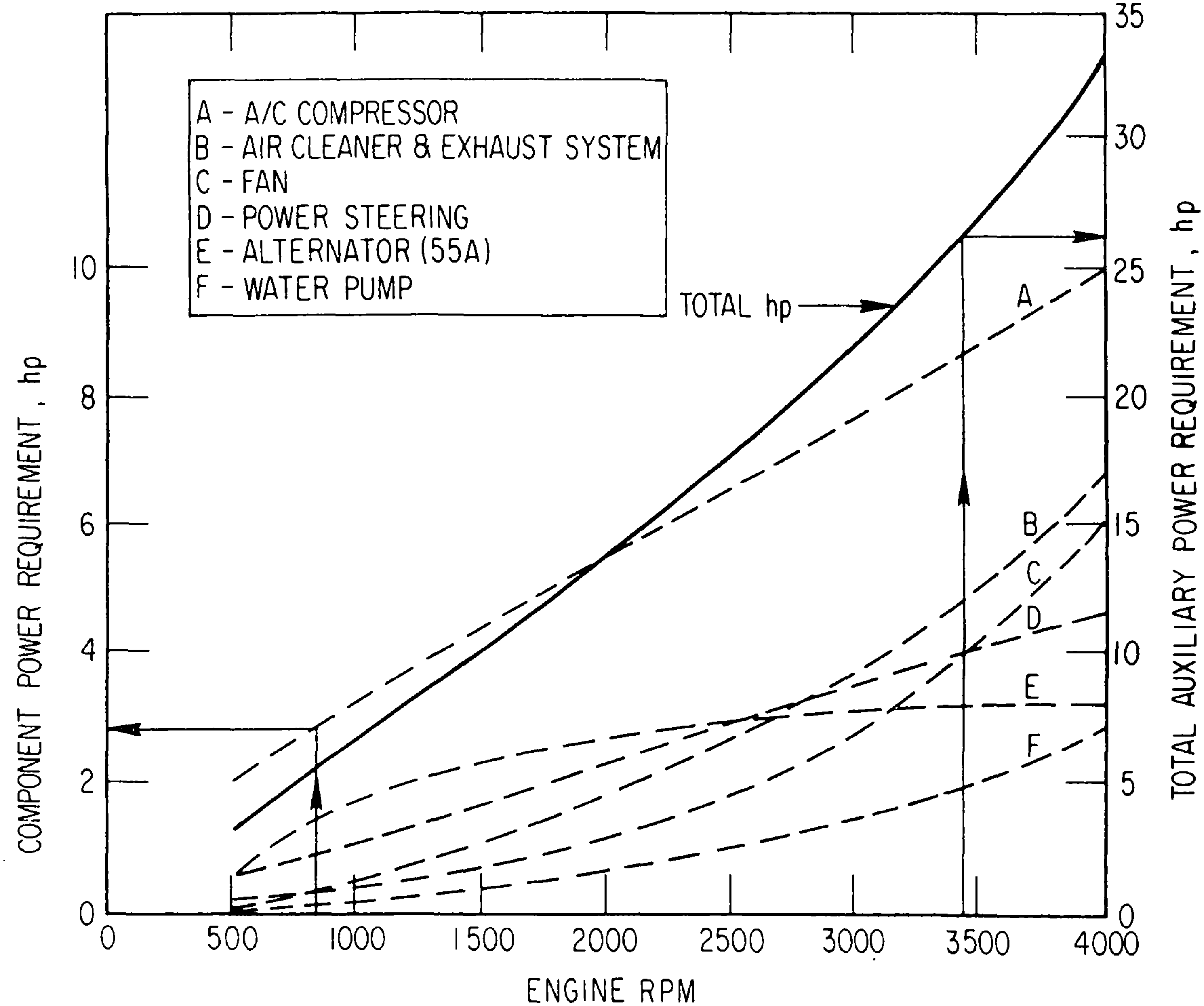


Figure 8-3. Auxiliary Power Requirements (232-CID American Motors Hornet)

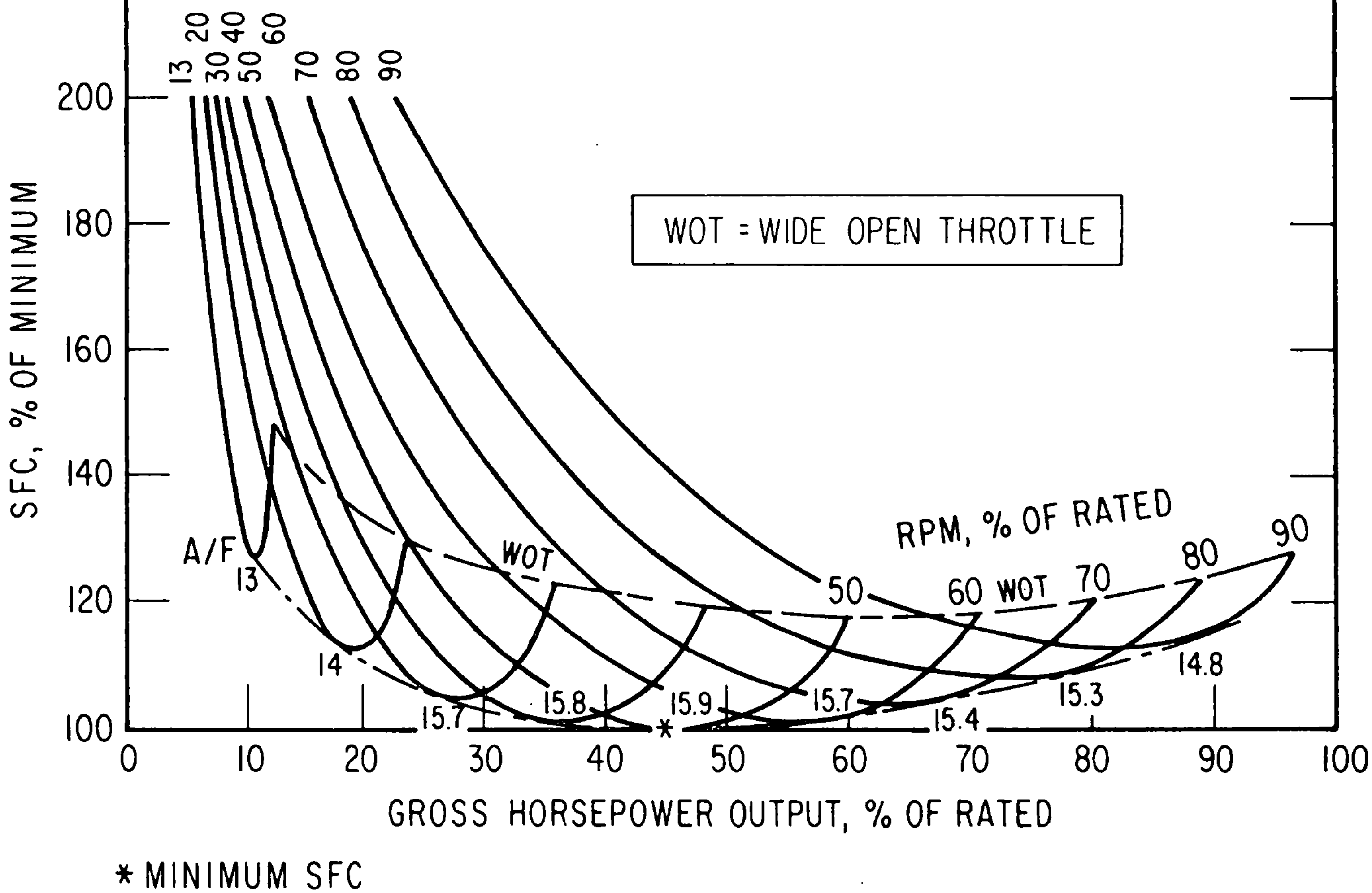
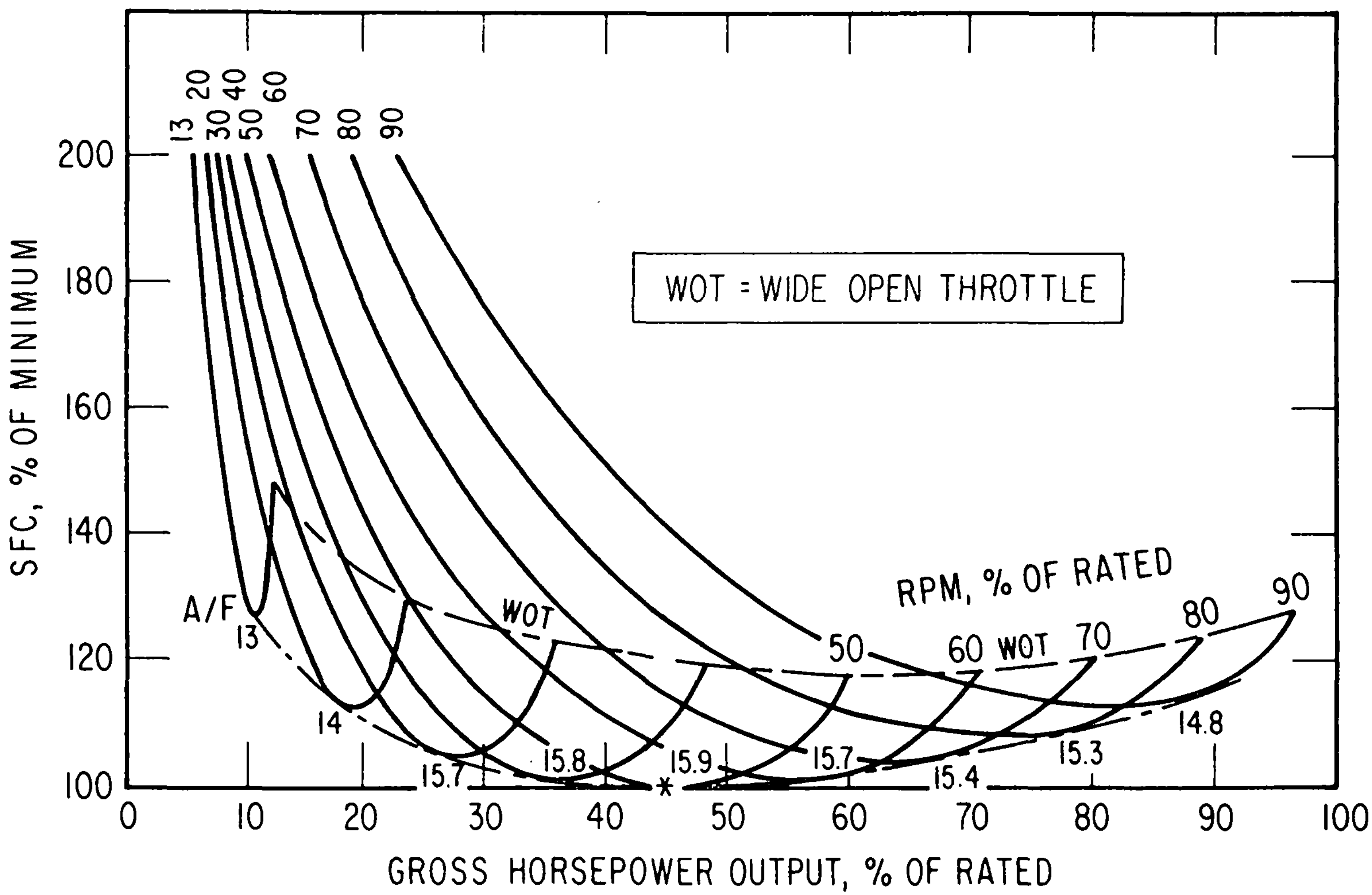


Figure 8-4. S. I. Engine SFC Map (Normalized)

During the above process the air/fuel ratio remains essentially constant. At near full-throttle opening, a mixture enrichment device in the carburetor goes into operation to permit maximum power to be obtained from the engine. Thus, the SFC passes through a minimum and begins to increase with enrichment of the charge as the throttle further moves toward the wide open position. Additional information on air/fuel effects related to throttle position is provided in Fig. 8-5.

A number of different engine operating modes may be postulated for the hybrid vehicle application. For the series propulsion configuration, in which the coupling between engine and drive wheels is purely electrical, the engine may operate at fixed speed and fixed power output, at fixed rpm and variable power output, and at variable rpm and variable power output. Studies show that the fixed power output mode does not match a number of vehicle duty cycle energy requirements and/or may severely limit the maximum top speed of the vehicle (See Section 10). Therefore, this mode, which is represented simply as a single point on the Fig. 8-4 operating map, will not be discussed further at this time.

The fixed rpm and variable power output mode is frequently used in engine/generator power units and may also be applied to the hybrid vehicle. Here the engine rpm is held constant by the action of a governor (mechanical or otherwise) which operates to adjust the engine throttle setting to accommodate changes in loads imposed by the generator. Current industrial practice suggests that 80 percent of rated rpm may be taken as a limiting speed level for operation in this mode. Then, based on the 80 percent rpm characteristic given in Fig. 8-4 and on engine SFC data developed in Section 8.3, the SFC/power output relationship for hybrid vehicle engines would appear as shown in Fig. 8-6. The dashed portion of the SFC characteristic represents the region of power output that is not presently attainable with conventionally carbureted engines except through rich mixtures. If one elects not to operate in this zone (in consideration of its impact on emissions), a 15 percent loss of potentially available power at 80 percent rpm is incurred.



* MINIMUM SFC

Figure 8-4. S. I. Engine SFC Map (Normalized)

During the above process the air/fuel ratio remains essentially constant. At near full-throttle opening, a mixture enrichment device in the carburetor goes into operation to permit maximum power to be obtained from the engine. Thus, the SFC passes through a minimum and begins to increase with enrichment of the charge as the throttle further moves toward the wide open position. Additional information on air/fuel effects related to throttle position is provided in Fig. 8-5.

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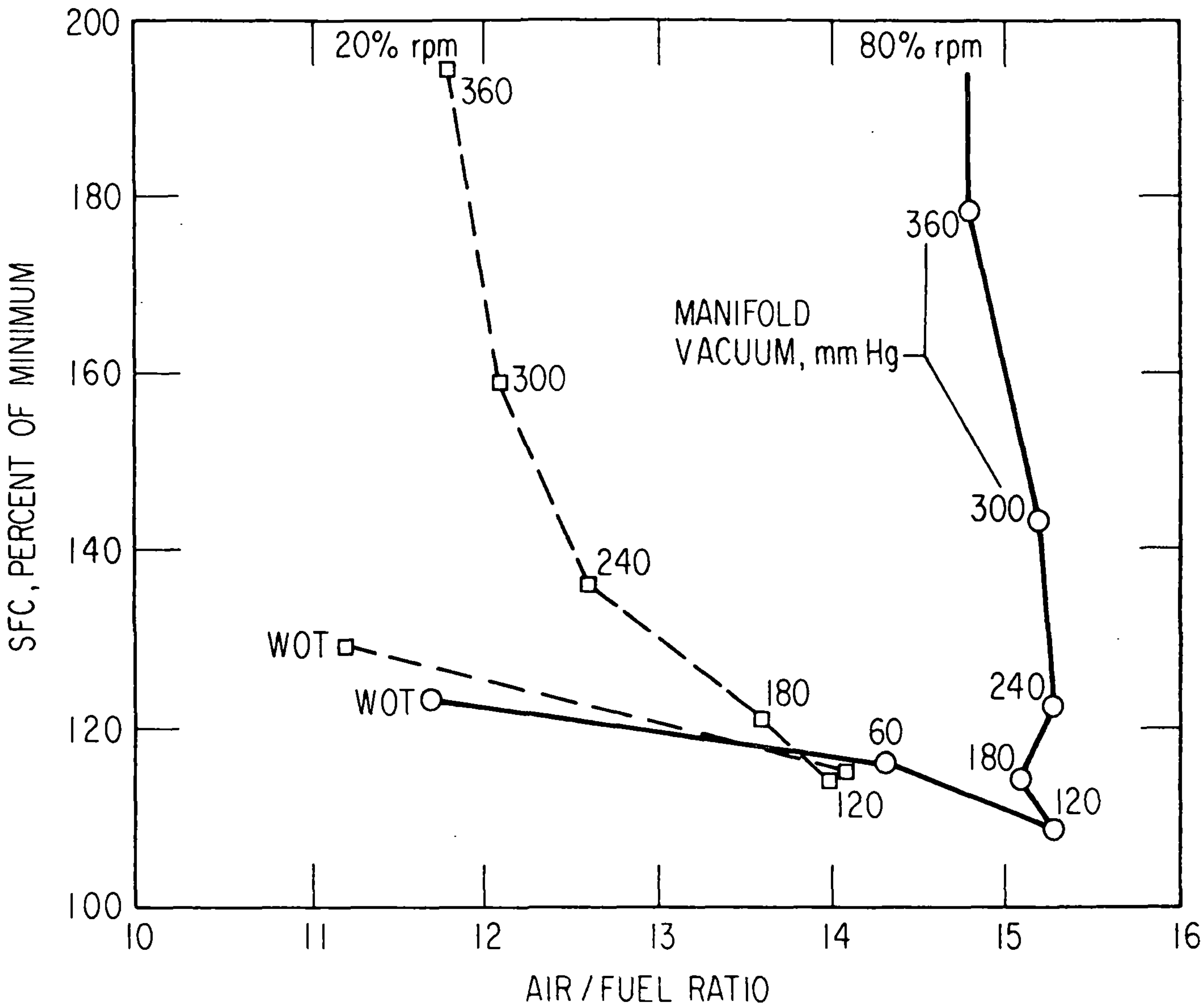


Figure 8-5. S. I. Engine Specific Fuel Consumption - Air/Fuel Characteristics at Constant RPM

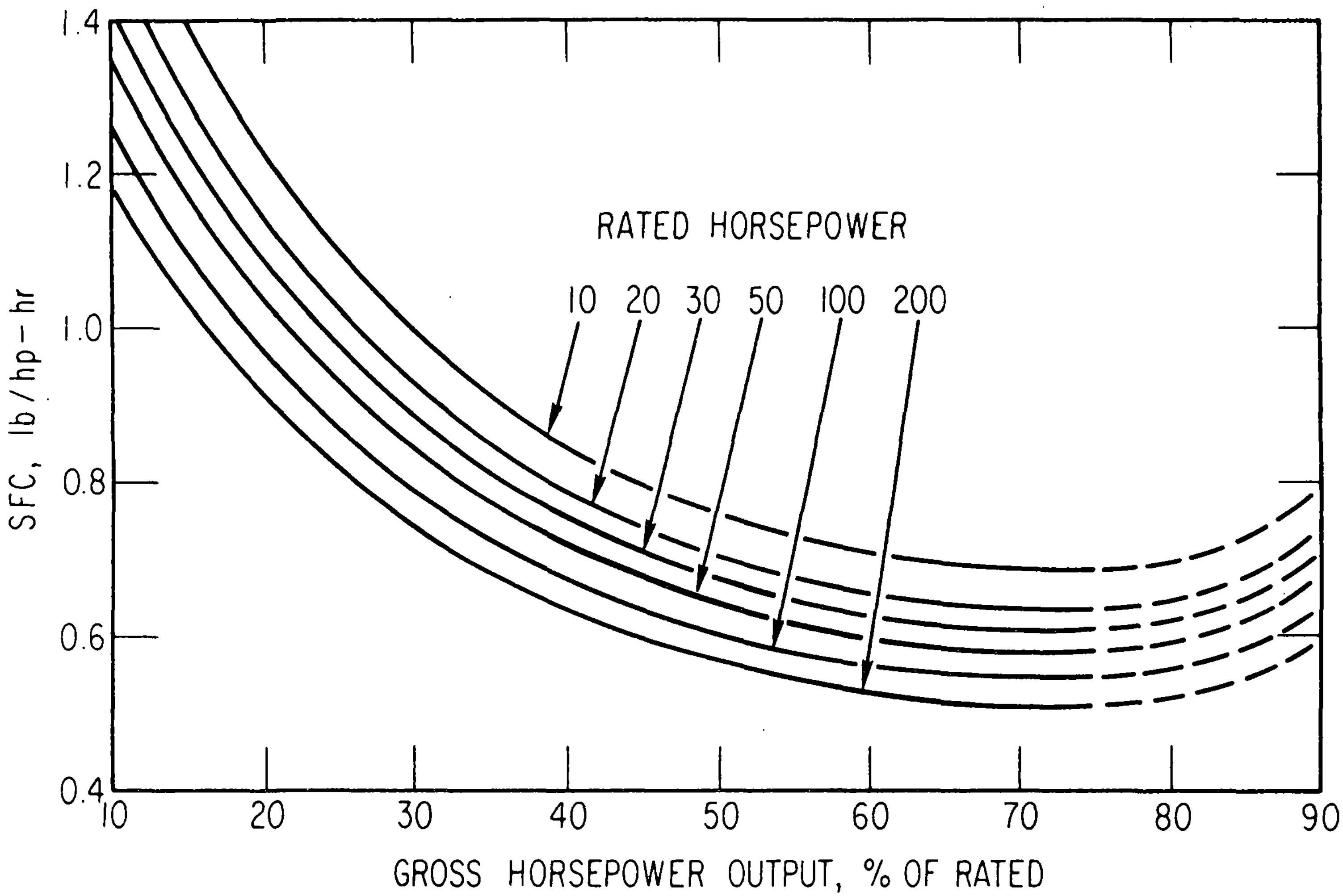


Figure 8-6. S.I. Engine Part Throttle Specific Fuel Consumption (RPM Constant at 80-Percent Rated Horsepower)

For the variable rpm and variable power output mode, operation at optimum throttle setting (optimum SFC) suggests itself as an interesting possibility. The air/fuel characteristic for this mode is indicated in Fig. 8-4 by the numbers spotted along the rpm envelope curve. Operation along this curve can be approximated by removing the carburetor power enrichment device and holding the throttle at a fixed position at or near the wide open setting. The deletion of carburetor enrichment results in a 7 percent loss of available power at 100 percent of rpm. However, some form of speed governing may actually be desirable to limit rpm at minimum load conditions. In this mode, the SFC/power output relationship for hybrid vehicle engines would appear as shown in Fig. 8-7.

In general, low SFCs favorably influence vehicle exhaust emissions. By comparing the two postulated series-configuration operating modes on this basis, it is evident from Fig. 8-4 that the optimum throttle mode is preferable since it provides significantly lower SFCs over a substantial portion of the power range. The performance advantage is particularly apparent in the low range of power output where the heat engine will frequently operate. It seems possible that the elimination of throttle travel may facilitate carburetor design improvements which could further enhance the SFC characteristic for operation in this mode.

The parallel-propulsion hybrid vehicle configuration features a direct mechanical link from heat engine to drive wheels. The heat engine operates in a quasi-steady-state manner to provide sustaining power for cruise at any given vehicle speed, while power demands for acceleration are met by the electric drive motors using battery and/or generator current. In the TRW Systems design, the heat engine is operated at constant speed and power output over the speed range up to 40 mph and at variable speed and power output over 40 mph. Other transition speeds and other direct drive systems are possible and are currently being studied. Lacking definite design details at this time, it may be adequate to describe the fuel consumption characteristic for this mode of operation by simply defining the

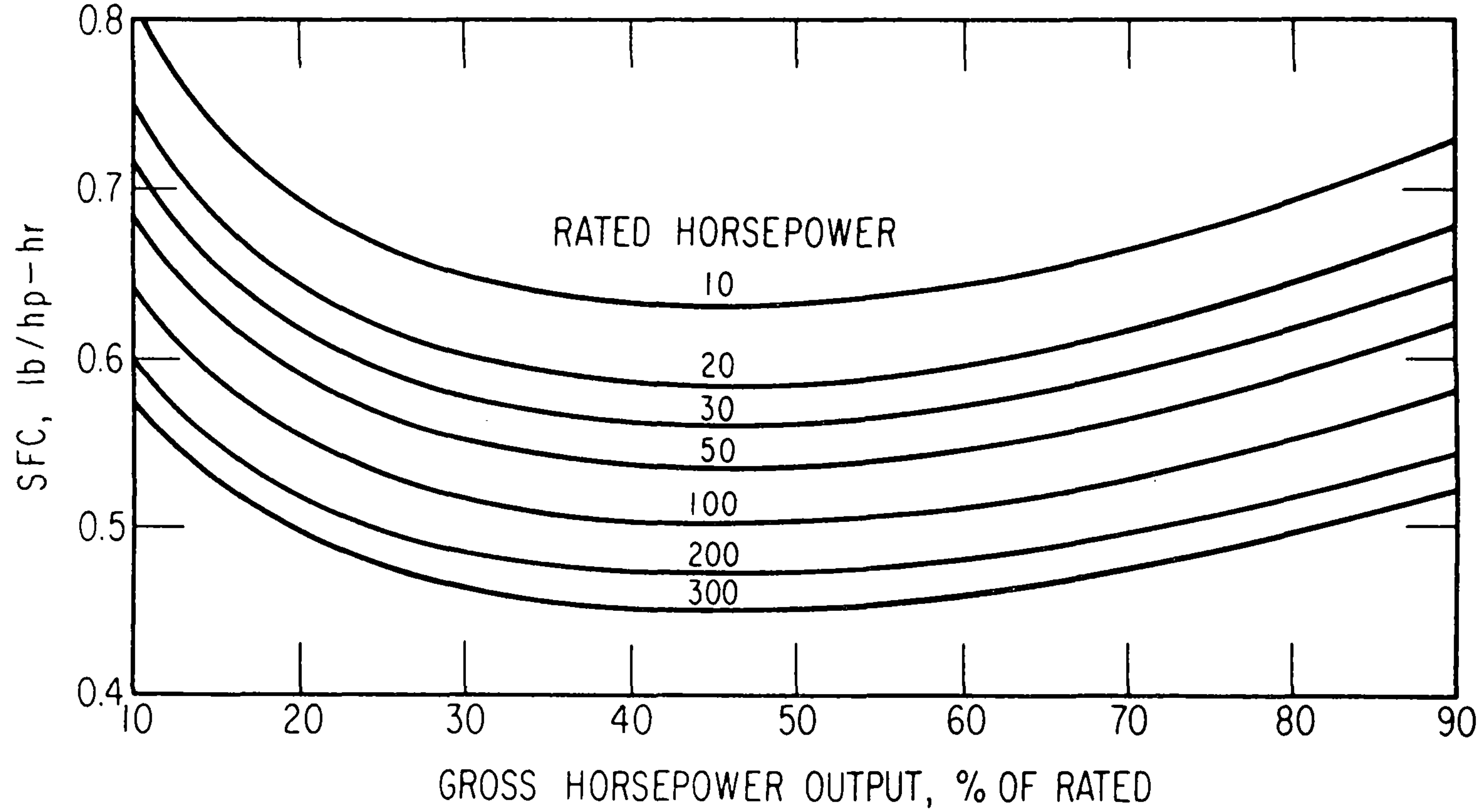


Figure 8-7. S. I. Engine Part Throttle Specific Fuel Consumption (Optimum Throttle Setting)

requirements for cruise over the complete range of vehicle operating speeds. This may be done for a given vehicle (car, van or bus) by the use of the engine performance map together with relationships linking wheel speed and road power with engine rpm and gross power output. For example, Fig. 8-8 shows the cruise SFC profile for the 4000-lb hybrid family car superimposed on the S. I. engine SFC map (Fig. 8-4). The cruise profile for other vehicles will vary depending upon road load and auxiliary power requirements.

It should be noted that the cruise curve depicted in Fig. 8-8 for the family car was constructed using the auxiliary power characteristics shown in Fig. 8-3 (excluding alternator). Based on these data, an engine rated at 4000 rpm, which is typically in the speed range of most U.S. designs, would require 98 hp in order to meet the 80-mph cruise speed requirement for the hybrid family car (A value of 92 hp was obtained in Section 5 using accessory power data supplied by the APCO.).

8.2.3 Engine Characteristics

8.2.3.1 Specific Fuel Consumption

The lowest value fuel consumption in the engine operating map was identified earlier as "minimum SFC." In addition to its utility as an index of optimum performance, this parameter also serves to identify characteristic performance trends related to heat engine size or rated horsepower.

A correlation of minimum SFC data for various industrial and automotive reciprocating spark ignition engines (identified in Appendix E) is shown in Fig. 8-9. The horizontal scale, Rated Horsepower, refers to the bare engine peak power output at the engine flywheel. A negative trend of minimum SFC with rated horsepower is indicated, with SFCs ranging from 0.58 to 0.47 over the rated horsepower band from 20 to 200. This correlation was used in conjunction with the Fig. 8-4 performance map to develop the spark ignition engine SFC/power output characteristics shown in Figs. 8-6 and 8-7.

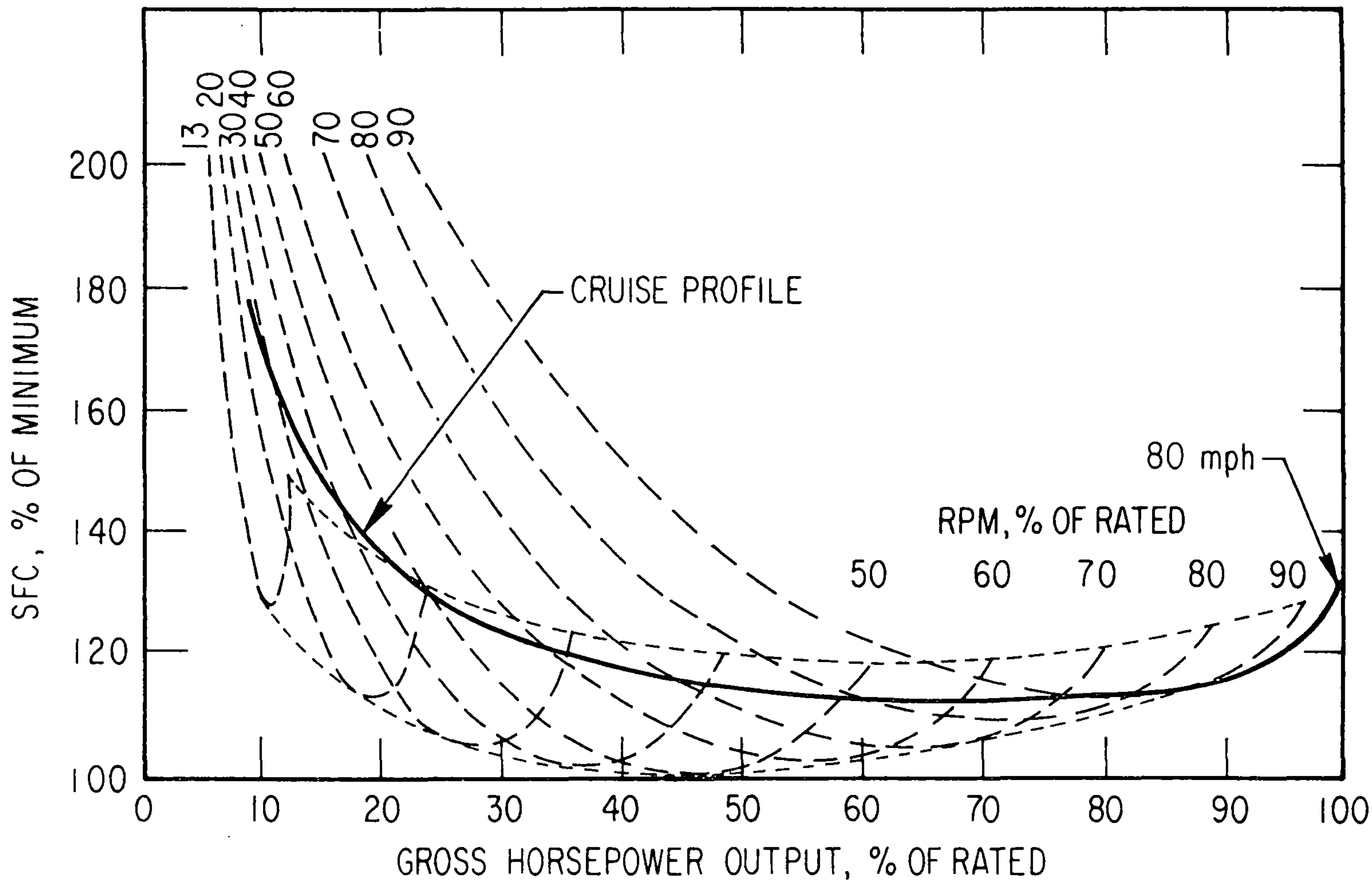


Figure 8-8. S. I. Engine SFC Cruise Profile
(Parallel Hybrid-Vehicle Operation)

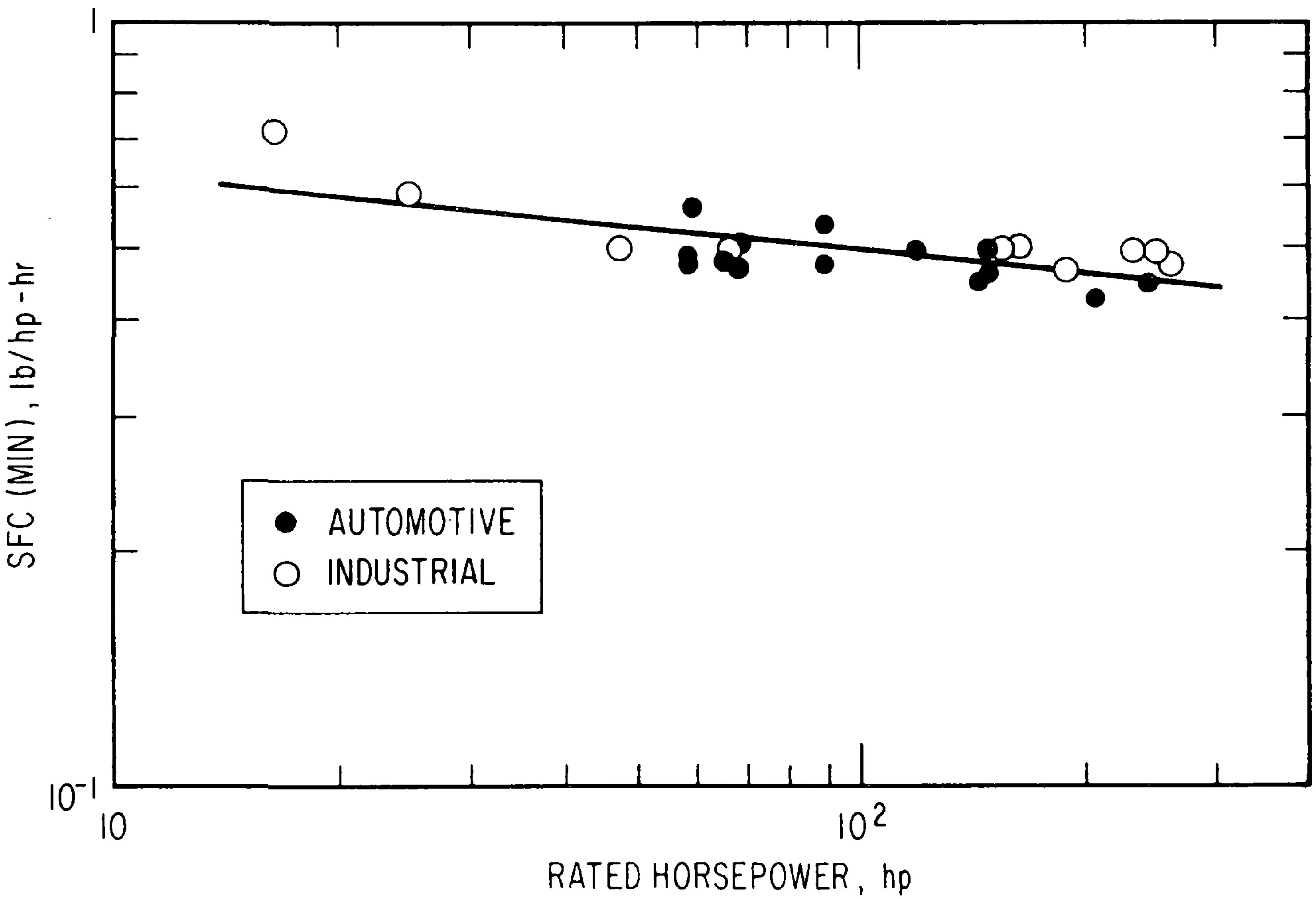


Figure 8-9. Minimum Specific Fuel Consumption - Reciprocating Piston SI Engine

For comparison, minimum SFC data for Wankel-type rotary piston spark ignition engines (Ref. 8-5) are displayed with the SFC correlation for reciprocating engines in Fig. 8-10. The Wankel data are derived from production engines manufactured by Curtiss Wright. The industrial engines are air-cooled; the automotive engines are water-cooled. The 20-hp, air-cooled engine is currently in use in two U.S. manufactured snowmobiles (Arctic Cat and Polaris). Two automotive engines are shown. One of these, the RC 2-30 (128.5 hp at 5500 rpm), is used in the German NSU RO-80 and the Japanese Mazda 110 S automobiles. The other automotive engine, the RC 2-30-10A (110 hp at 7000 rpm), is used in the Japanese Mazda R-100 automobile.

It may be concluded from Fig. 8-10 that the minimum SFC characteristic for the Wankel engine is very similar to the reciprocating engine. We note that the data shown represent all versions of the Wankel engine currently in production. Engines currently under development include a Mercedes-Benz three-rotor, 110-CID, 335-hp engine and NSU engines ranging from 3 to 800 hp. No additional information on these models is available.

8.2.3.2 Specific Weight

Although much information on industrial engines was acquired, sufficient data for automotive-type spark ignition engines were accumulated to permit an accurate weight correlation to be made without the necessity of using the industrial data. This arrangement is preferred primarily because the auxiliary equipment on industrial engines may differ somewhat from automotive engines (e.g., fan, flywheel) and also because the peak power ratings of industrial engines vary, depending on service or application. Thus, the data points plotted in Fig. 8-11 exclusively represent automotive engine designs and equipment. The weight indicated is based on a complete engine, including starter, alternator and flywheel, but excluding radiator, oil, and water. Twenty-five data points are shown (See Appendix E for identification) of which 23 are cast-iron block engines and two are aluminum block (Vega) engines. A least-squares fit has been drawn through the cast-iron data set

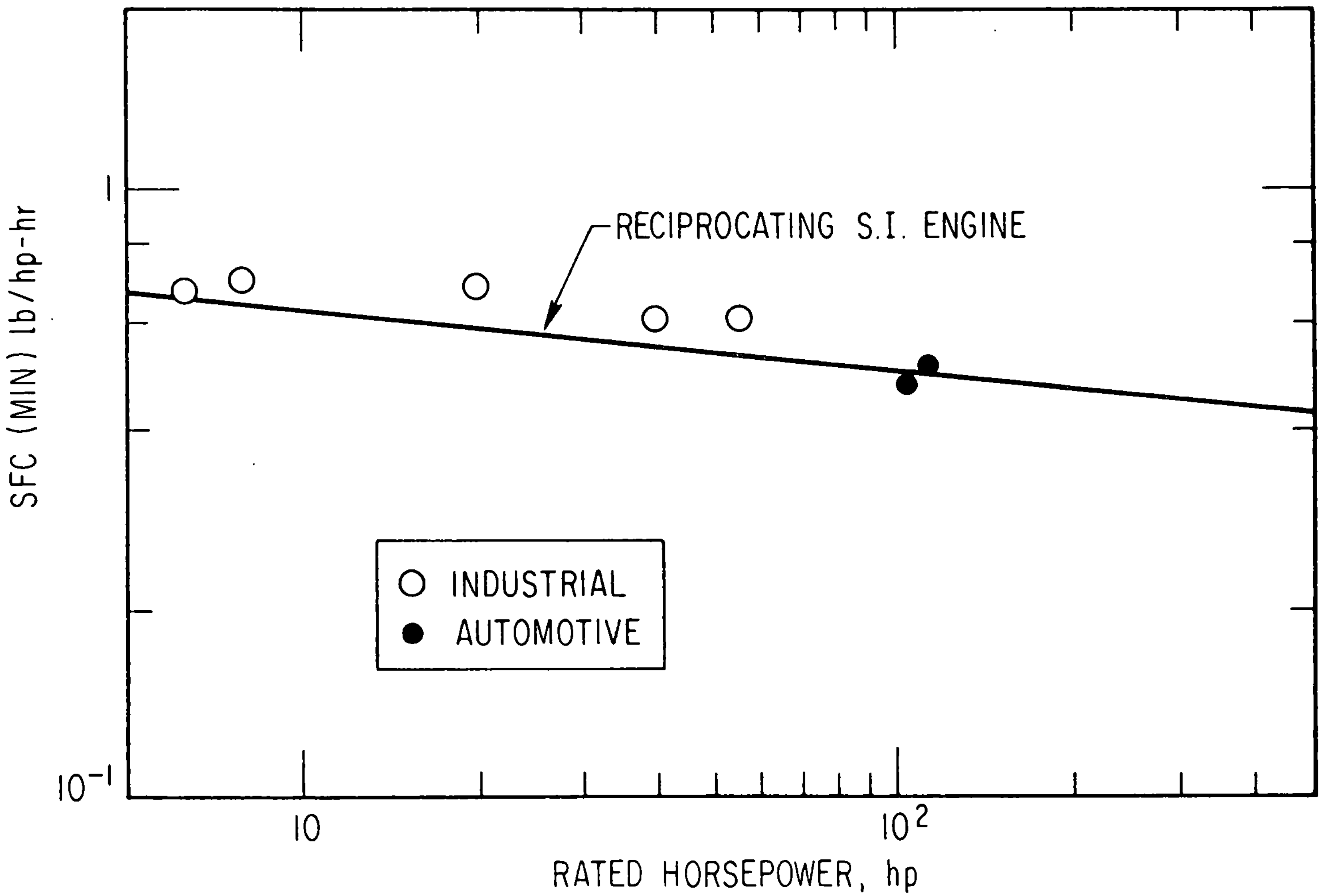


Figure 8-10. Minimum Specific Fuel Consumption - Rotary Piston SI Engine

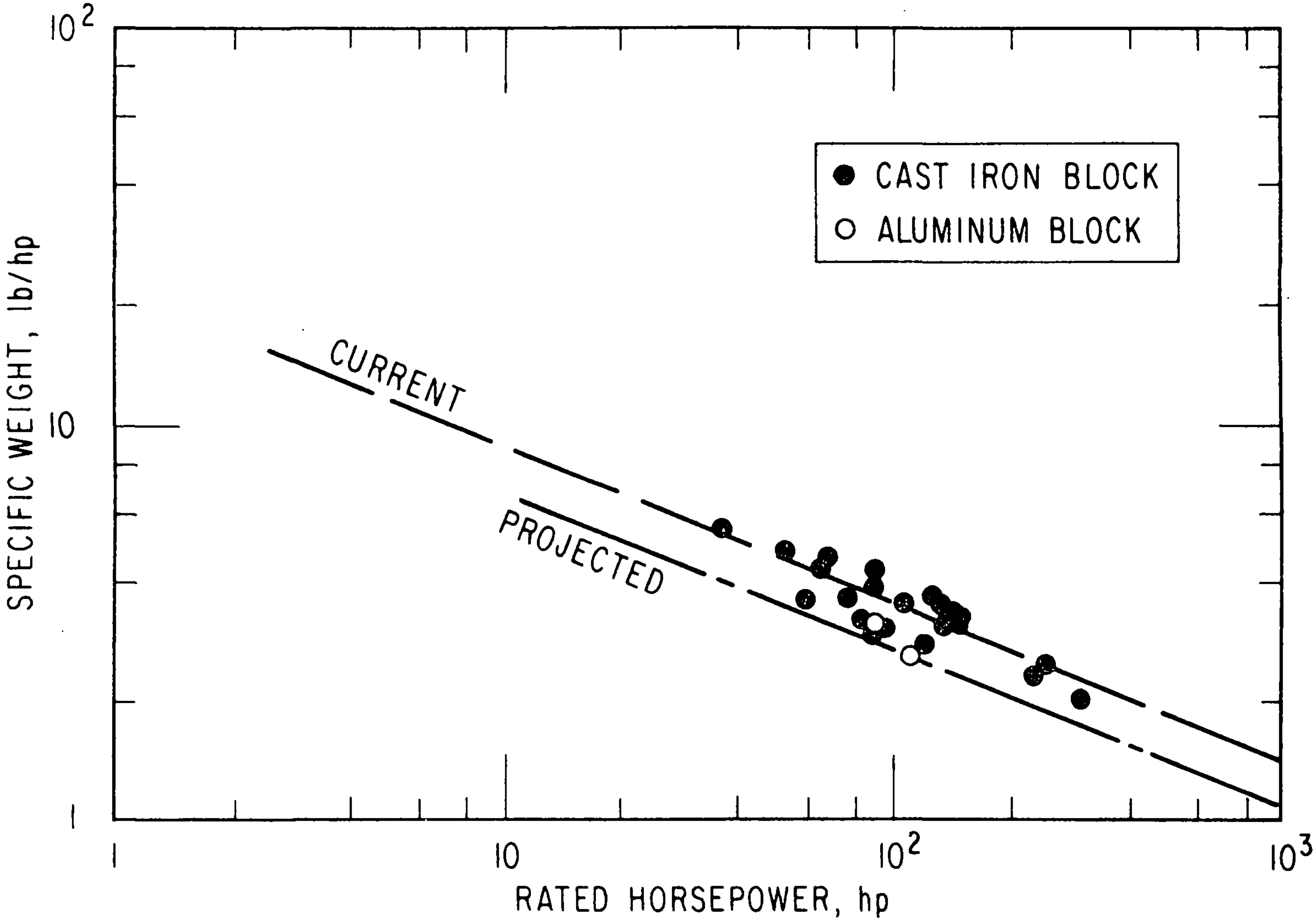


Figure 8-11. Specific Weight - Reciprocating Piston SI Engine

and this line may be interpreted as representing the mean for current state-of-the-art automotive engine designs. The line labelled "projected" has been drawn through the lighter of the two aluminum block engines and is proposed to represent a mean characteristic for the year 1975.

For comparison, weight data for the Wankel engines discussed under Section 8.2.3.1, Specific Fuel Consumption, are displayed with the current weight correlation for reciprocating engines in Fig. 8-12. The automotive Wankel engines average about 35 percent lighter than the reciprocating engines.

8.2.3.3 Specific Volume

Specific volume data (ft^3/hp) for current automotive and industrial reciprocating spark ignition engines are correlated with rated horsepower in Fig. 8-13. Wherever necessary, the industrial data were adjusted to reflect a bare-engine peak horsepower rating equivalent to the rating for an automotive engine. The volume represented in the plot is the engine envelope from fan to flywheel and from air cleaner to crankcase pan. Representative dimensions for the volume envelope are characterized by the following ratios:

Length/Length	1.0
Width/Length	0.8
Height/Length	0.9

The least-squares data fit shown in Fig. 8-13 is reproduced in Fig. 8-14 for comparison with the Wankel data. Note that a significantly steeper trend with engine rated horsepower is indicated for the Wankel engines. At the 115-hp rating, the Wankel volume is indicated to be 6.5 ft^3 compared with 14.6 ft^3 for the reciprocating piston design, or less than 50 percent the size. Representative dimensions for the Wankel engine volume envelope are characterized by the following ratios:

Length/Length	1.0
Width/Length	1.5
Height/Length	1.1

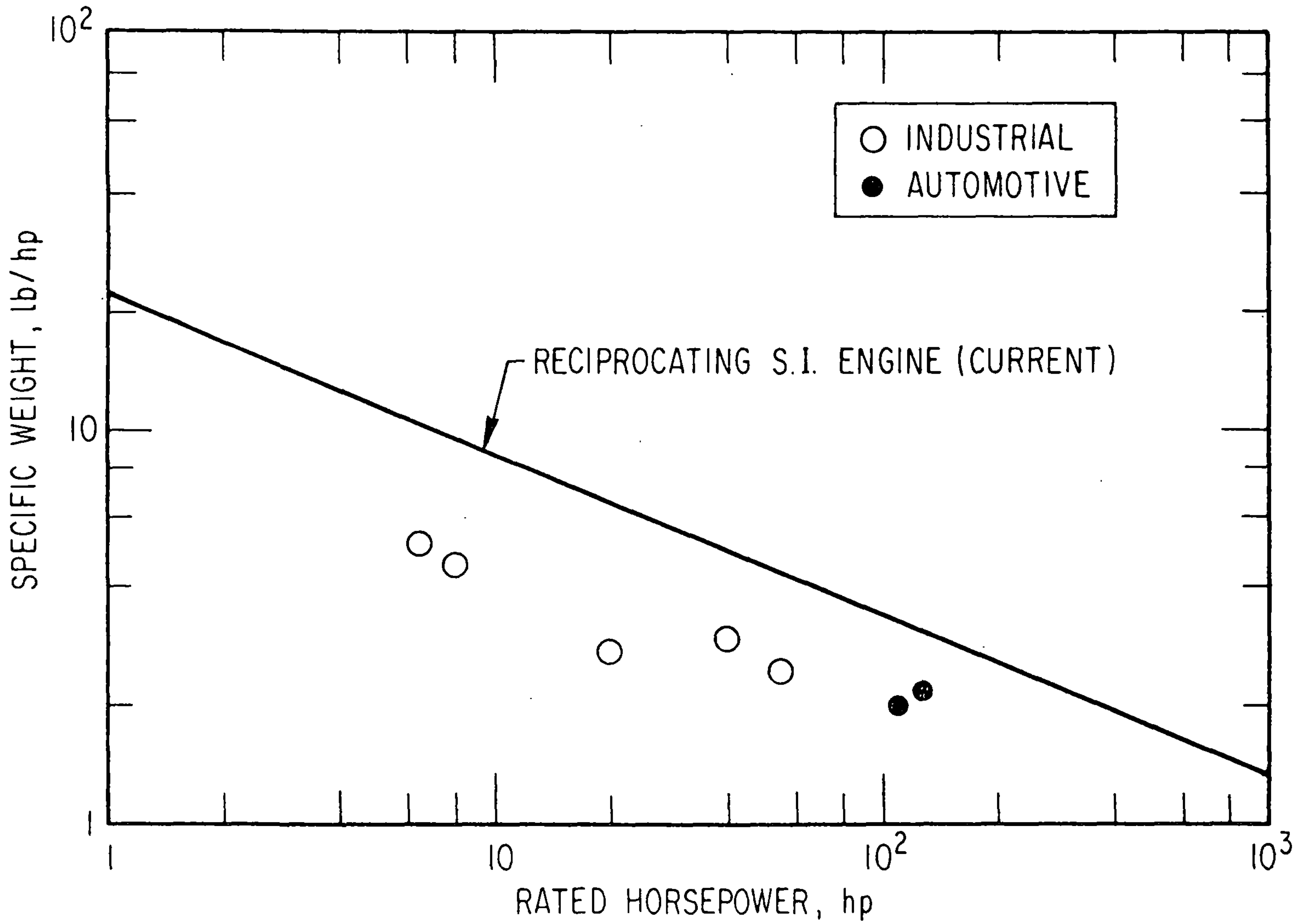


Figure 8-12. Specific Weight - Rotary Piston SI Engine

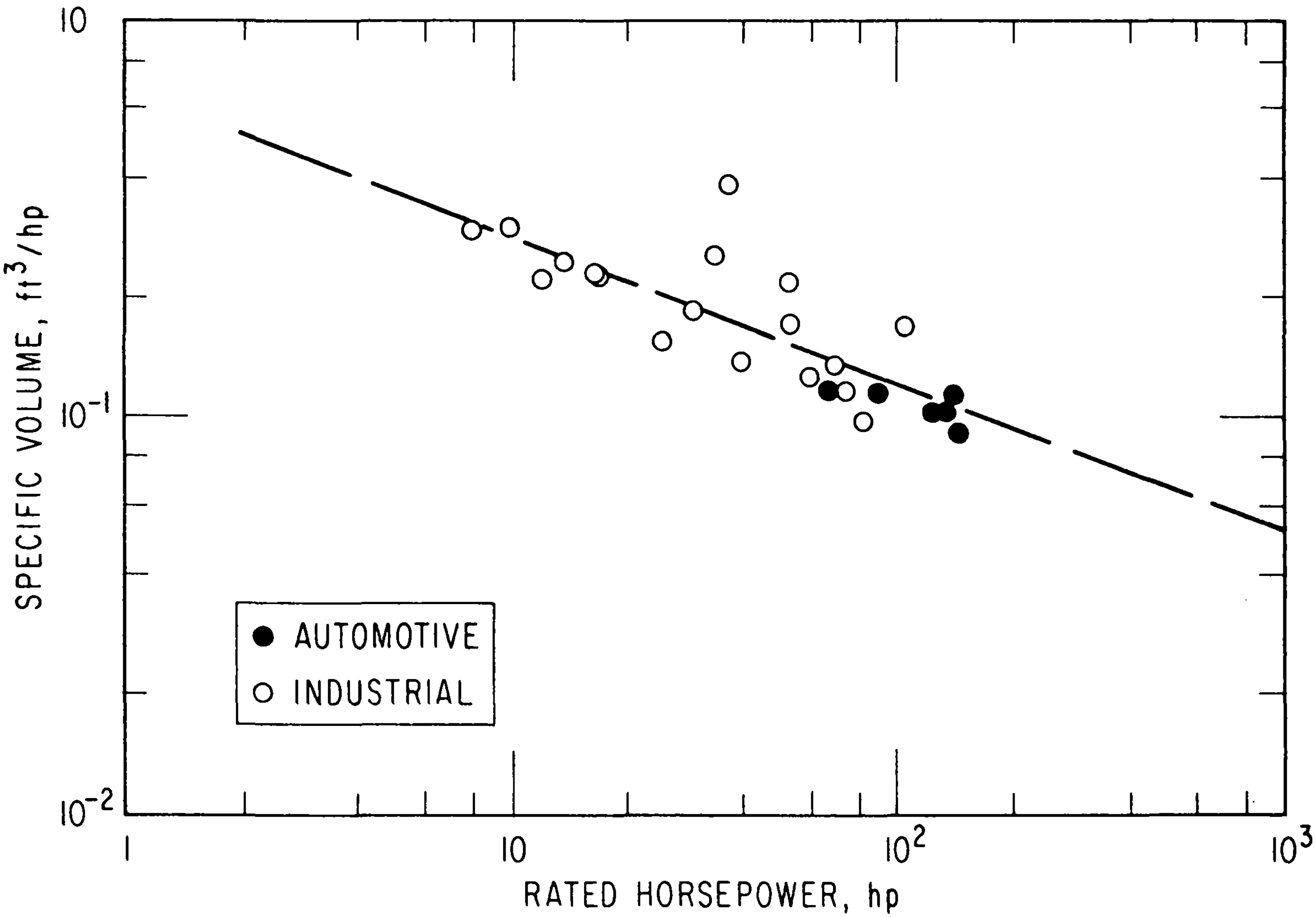


Figure 8-13. Specific Volume - Reciprocating Piston SI Engine

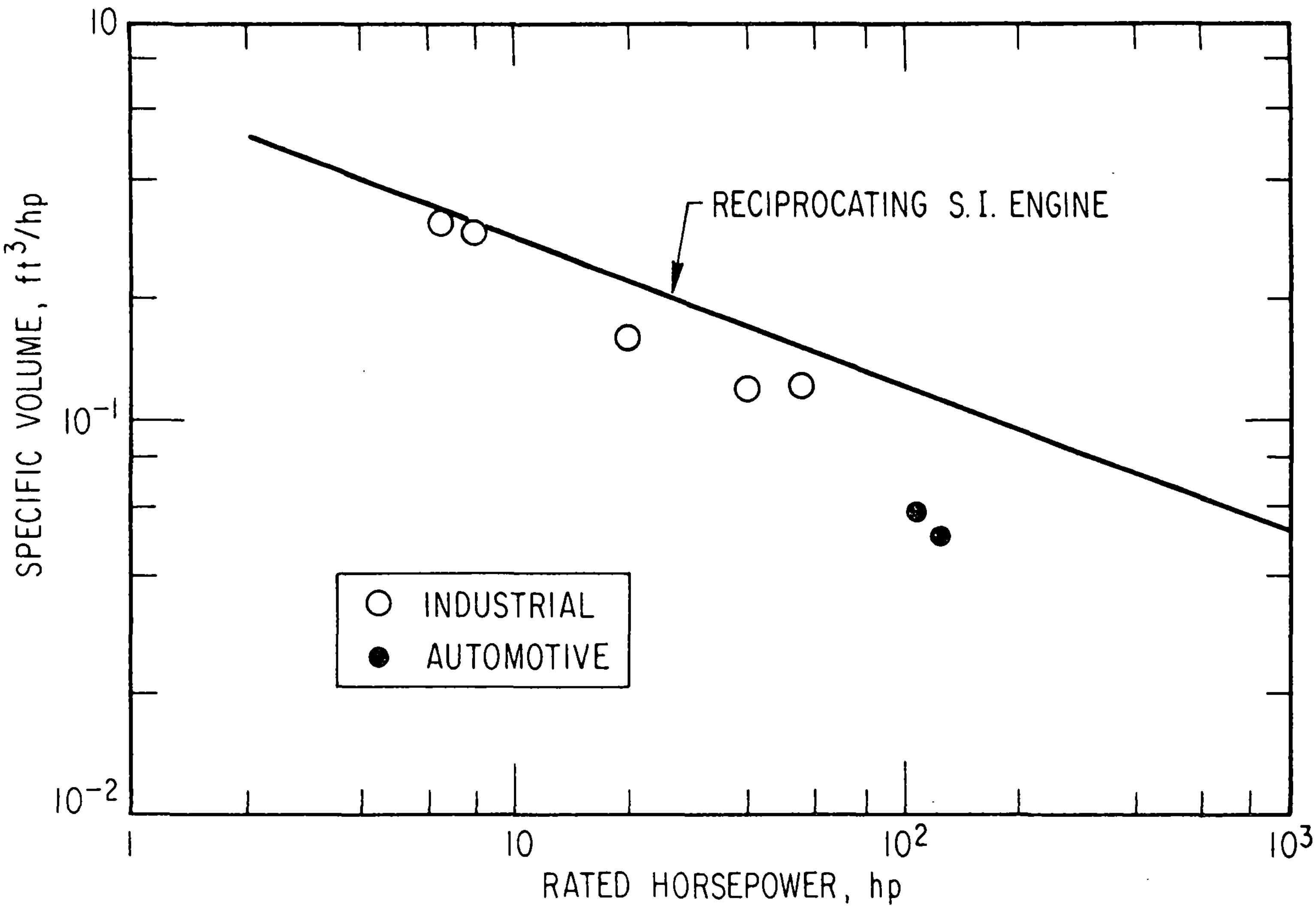


Figure 8-14. Specific Volume - Rotary Piston SI Engine

8.3 DIESEL CYCLE (COMPRESSION IGNITION) ENGINE

8.3.1 General Description

The thermodynamic cycle for this reciprocating piston engine is shown in Fig. 8-15. The four-stroke sequence of engine operations consists of an air-only intake stroke (terminating at Point 1), a compression stroke which raises the temperature of the air above the auto-ignition point of the fuel (Point 1 to Point 2) followed by combustion of the injected fuel charge (Point 2 to Point 3), an expansion or power stroke (Point 3 to Point 4), and an exhaust stroke (Point 4 to Point 1). The classical constant pressure combustion process illustrated in Fig. 8-15 is achieved by metering fuel into the cylinder during the expansion stroke. In fact, however, combustion in most modern compression ignition (CI) engines proceeds first at constant volume (the S. I. engine combustion process) and late burning occurs at constant pressure.

Fuel under high pressure (2000 to 20,000 psia) is delivered to the cylinder through individual-cylinder nozzle injection valves by an injection pump operated by the camshaft. Air enters and exhaust products leave the cylinder through intake and exhaust poppet valves also operated by the camshaft. Unlike the spark ignition engine, the charge mixture is not regulated and air/fuel ratios ranging from 20 to 75 or higher may be encountered over the normal operating range of the engine. Load and speed control is achieved by adjusting the amount of fuel injected during the combustion cycle. Maximum fuel delivery is fixed by control stops on the injection pump to limit the power output over the speed range to conform with specified smoke standards for operation on the road. A governing mechanism is also included to limit engine speed at predetermined minimum and maximum values.

Representative performance curves for an automotive CI engine (Ref. 8-6) are shown in Fig. 8-16. Typically, the power curve does not display a peak point because this region of CI engine operation usually is accompanied by heavy smoke which, if sustained, will cause the engine to foul. The rated

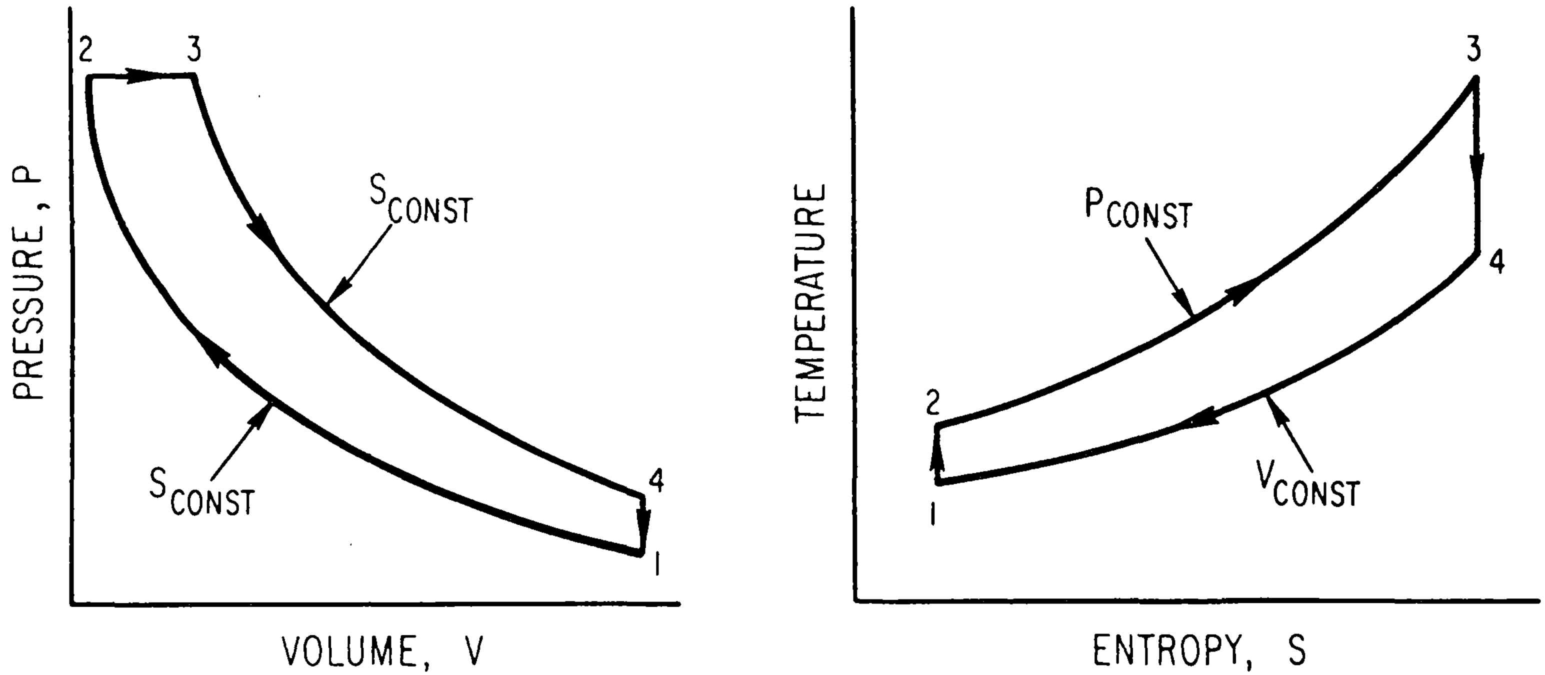


Figure 8-15. The Diesel Cycle

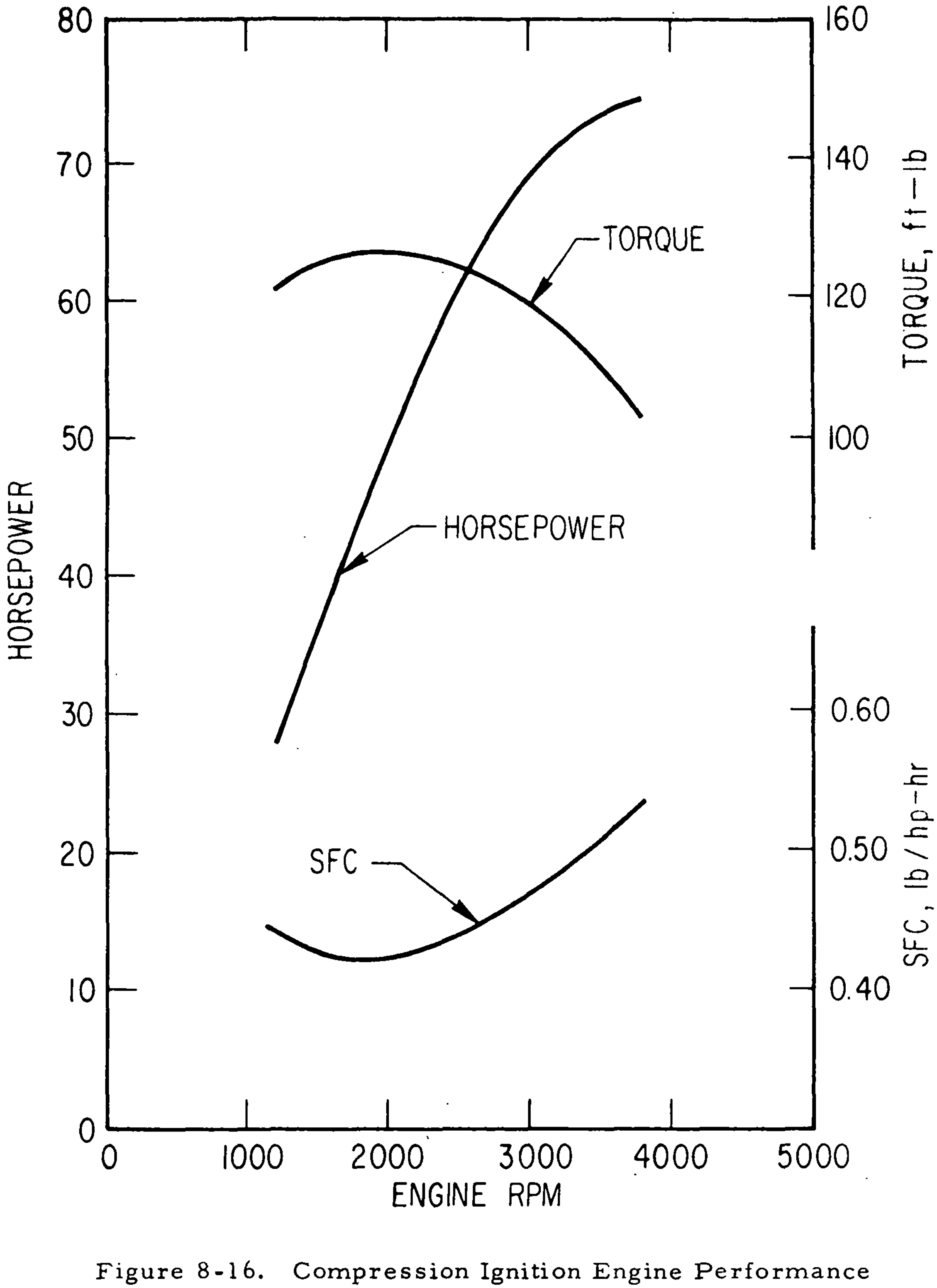


Figure 8-16. Compression Ignition Engine Performance Characteristics (154-CID Daihatsu)

power of the engine, therefore, is not sharply defined by an intrinsic upper bound on energy output [as the wide-open throttle (WOT) peak for spark ignition engines], but is based instead on some limiting condition of exhaust smoke, either defined by the manufacturer or stipulated by legislation prohibiting excessive smoke on the road. Typically, the torque curve peaks and the SFC curve bottoms at a somewhat lower percentage of rated rpm than the spark ignition engine. However, variations in injection system and combustion chamber design may alter this relationship significantly.

8.3.2 Hybrid Operation

Investigations discussed in Section 9, Heat Engine Exhaust Emissions, indicate that diesel engines with indirect injection (i. e., divided) combustion chamber designs [turbulence chambers (TC) or precombustion chambers (PC)] have emission characteristics that are superior to those for direct injection combustion chamber designs. For this reason, the divided chamber engine has been selected as the preferred diesel configuration for the hybrid vehicle application. A complete SFC performance map (normalized) for an engine of this type (Refs. 8-7 and 8-8) is presented in Fig. 8-17. The characteristics shown, though based specifically on a turbulence chamber design, are believed to be more or less typical of divided chamber (TC or PC) engines with rated speeds in the neighborhood of 3500 rpm.

In general, the map displays trends that are similar to those shown for the spark ignition engine. One difference that may be observed is that the SFC characteristic is relatively flat over a broad range of power output. This feature is frequently claimed to be typical of all diesel engines; actually, it is not readily distinguishable in some designs. Values for SFC at the left of the map approach infinity as a limit as the net engine power output diminishes.

The air/fuel ratio varies significantly with load as shown by the numbers spotted on the 80 percent rpm curve. This is because the CI engine takes in a full charge of air at each induction stroke and adjusts the amount of fuel injected to control power output. The SFC declines along with air/fuel ratio as

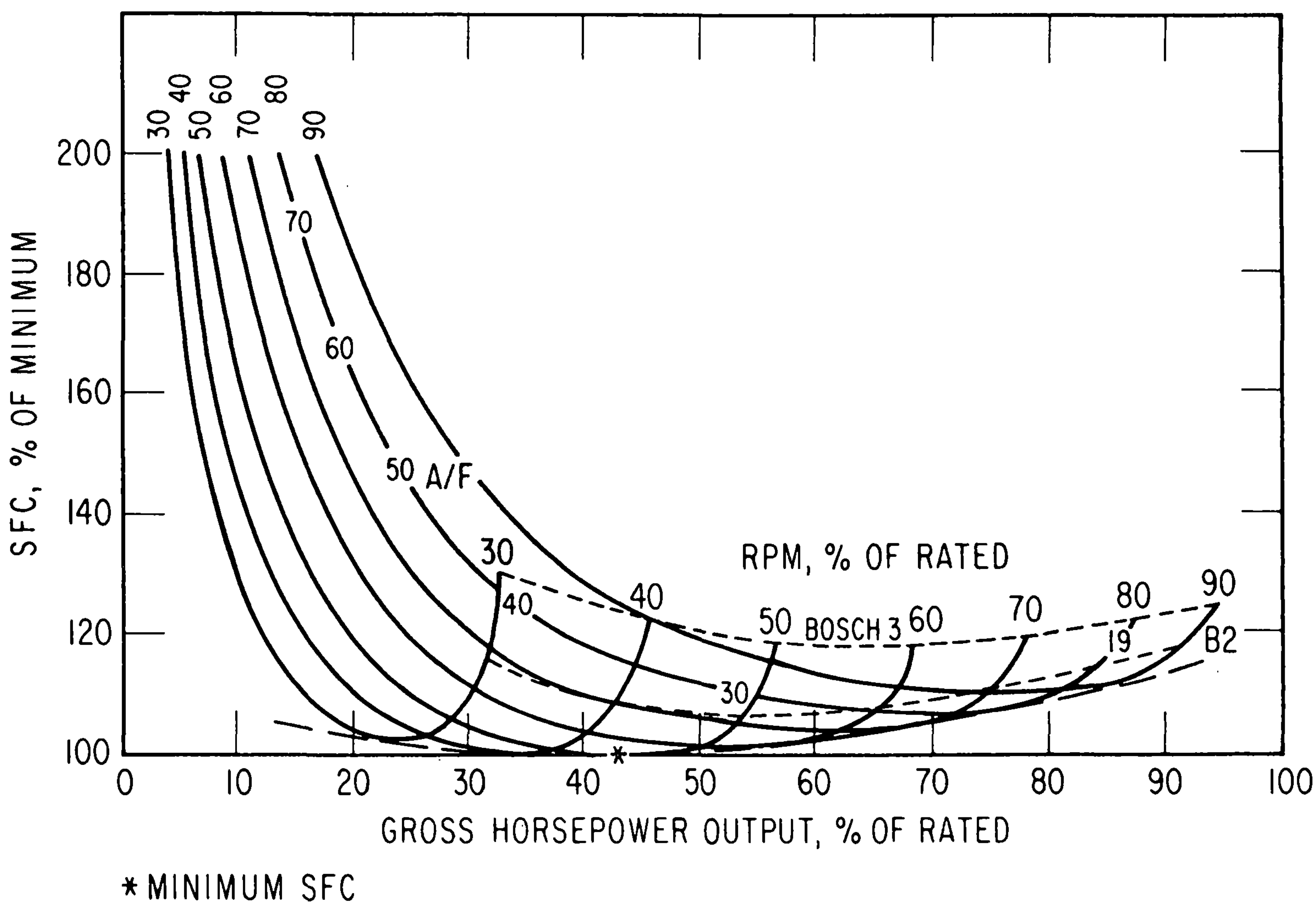


Figure 8-17. CI Engine SFC Map (Normalized)

brake mean effective pressure (BMEP) increases with increase of the fuel charge. Minimum SFC at fixed rpm is achieved at about 75 percent of load and with air/fuel ratio in the range from 24 to 22. At or near this point a haze appears in the engine exhaust because of failure of the fuel to find air. With further increase in load the haze darkens and the SFC begins to increase.

The Fig. 8-17 rpm curves, which are shown to terminate at a smoke rating of Bosch 3, have been extended beyond the normal operating limits of the engine to illustrate the relationship between exhaust smoke, fuel consumption, and power output at rich mixture conditions. The true load limit envelope is shaped by the setting of the fuel delivery stop and may cut across several smoke lines, depending on the speed characteristics of the injection system. Bosch 3 represents a slight discoloration of the exhaust equivalent to a U.S. smoke obscurity rating of about 4 percent in engines up to 100 hp (Ref. 8-9); Bosch 1.5 and lower is nearly invisible.

Following the procedure employed for spark ignition engines, the normalized SFC map for the diesel engine may be used to develop characteristic curves of fuel consumption for the hybrid vehicle. For the series-propulsion configuration we may again postulate the constant speed (80 percent rated rpm) mode of operation as one of several alternatives. Then, based on the 80 percent characteristic of Fig. 8-17 and on SFC data for divided chamber diesel engines developed in Section 8.3, the SFC/power output relationship for hybrid vehicle diesel engines would appear as shown in Fig. 8-18. The dashed portion of each curve identifies a zone of progressively darkening exhaust haze which, though innocuous, should perhaps be avoided for the hybrid application.

A variable rpm and variable power output mode which conforms to the optimum SFC envelope in Fig. 8-17 is another series-configuration operating mode which might be considered. Operation along the optimum curve can be approximated by fixing the control rod or rack at an appropriate maximum stop position or, more accurately, by speed governing the injection system to

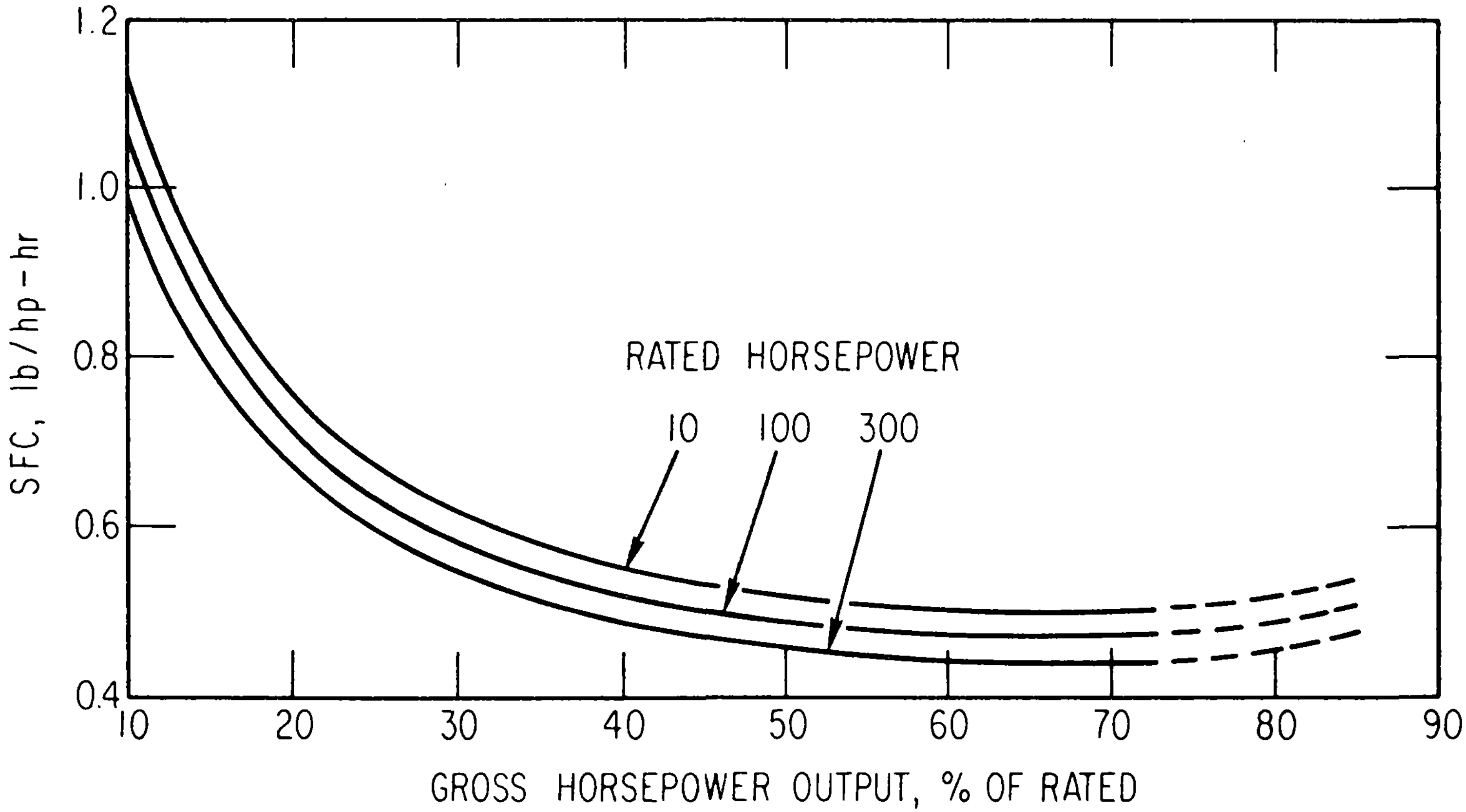


Figure 8-18. CI Engine Part-Load Specific Fuel Consumption (RPM Constant at 80-Percent Rated Horsepower)

an optimum fuel delivery profile. In this case, the SFC/power output characteristic for hybrid vehicle engines appears as shown in Fig. 8-19.

With regard to the parallel-propulsion hybrid configuration, the remarks under spark ignition engines concerning operation in this mode apply also to diesel engines. An SFC profile which matches engine rpm and flywheel brake horsepower output to vehicle cruise speed and road power requirements can be constructed for a vehicle of given weight, frontal area, and wheel diameter. This has been done for the 4000-lb hybrid vehicle family car using the engine auxiliary power data presented in Fig. 8-3. The resulting SFC characteristic is shown superimposed on the diesel SFC map reproduced in Fig. 8-20.

8.3.3 Engine Characteristics

8.3.3.1 Specific Fuel Consumption

A correlation of minimum SFC versus rated horsepower for compression ignition engines is shown in Fig. 8-21. The horizontal scale, Maximum Rated Bare Engine Horsepower, refers to a no-accessory power output equivalent to the automotive rating for a spark ignition engine. All of the data shown conform to this rating basis (See Appendix E for identification of data).

Figure 8-21 distinguishes the characteristics of divided chamber diesels from open chamber (direct injection) diesels, the former being preferred for the hybrid application because of lower emissions. Two major types of divided chamber designs which have been identified are (a) turbulence chamber (TC) designs as exemplified by Continental, Waukesha, Daihatsu, Peugot, and Ricardo diesels, and (b) precombustion chamber (PC) designs as exemplified by Caterpillar, Perkins, Onan, Mercedez-Benz, and Leyland diesels. Both systems operate on basically the same principle; that is, both develop swirl and initiate combustion in an antechamber separated from the main chamber by a restricted passageway. The TC antechamber is larger (50 percent of the clearance volume) than the PC chamber (20 to 30 percent of the clearance volume). Its principal distinction appears to be that it

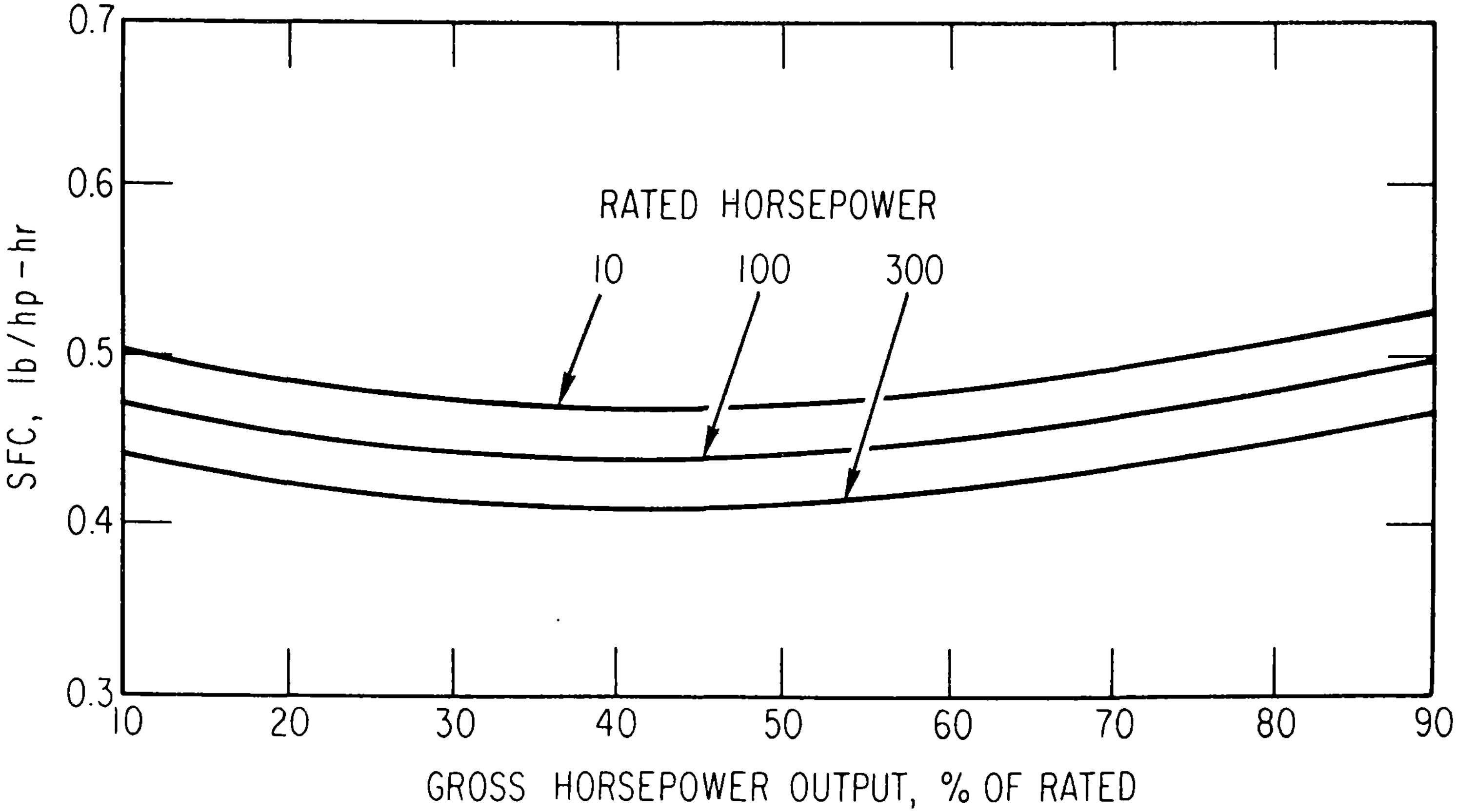


Figure 8-19. CI Engine Part-Load Specific Fuel Consumption (Optimum Throttle Setting)

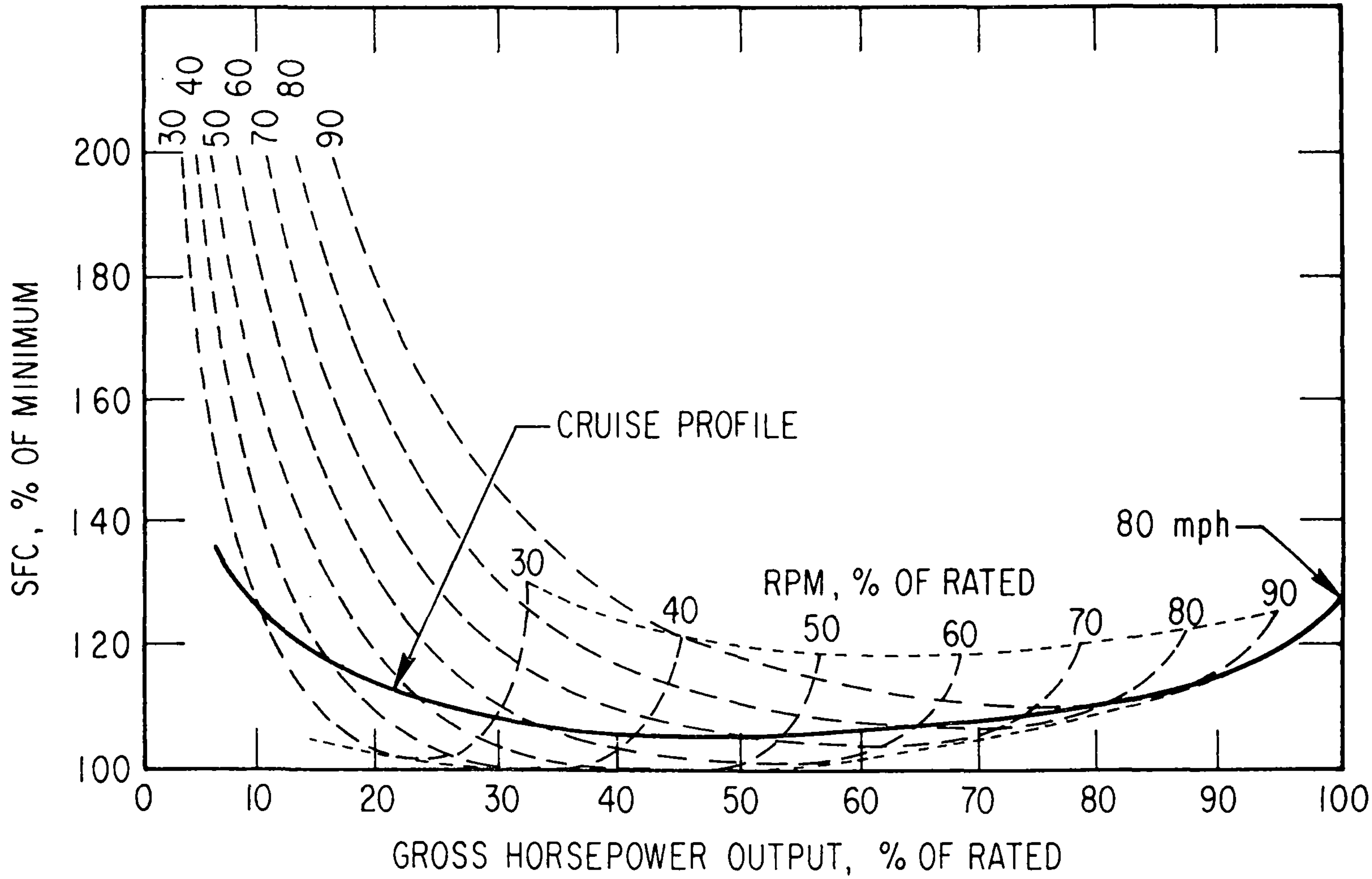


Figure 8-20. CI Engine SFC Cruise Profile
(Parallel Hybrid-Vehicle Operation)

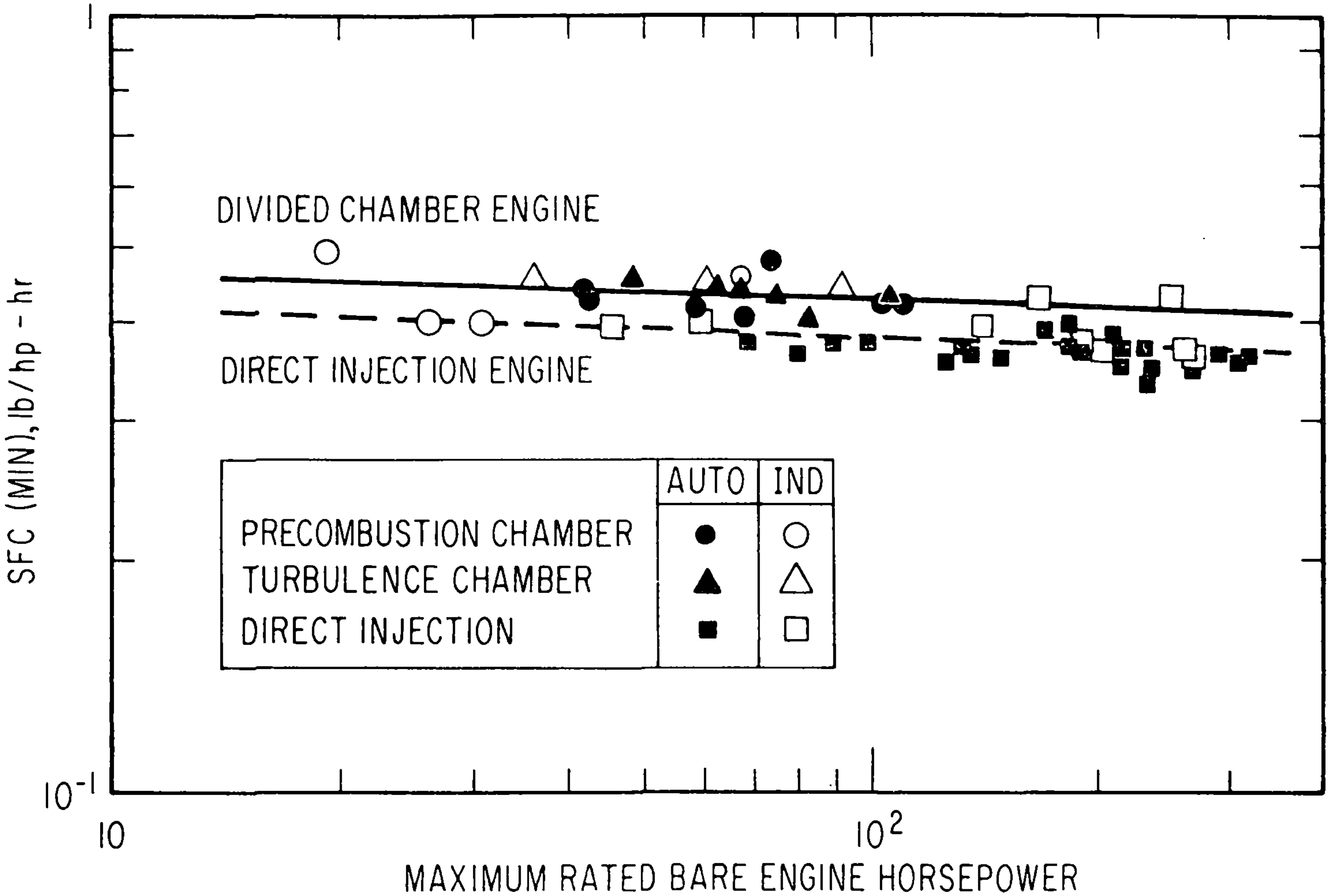


Figure 8-21. Minimum Specific Fuel Consumption - CI Engine

develops a higher degree of swirl than the PC type. For the purpose of this analysis (though not necessarily from the standpoint of emissions), it was assumed that PC and TC diesels could be lumped together and correlated under the single category of "divided chamber" engines. Figure 8-21 symbolically distinguishes the individual chamber types for information purposes only.

Fifty-two data points are plotted in Fig. 8-21; 32 for direct injection (DI) engines and 20 for divided chamber (DC) engines. The correlation approach first isolated the DI data, producing the least squares fit shown by the lower of the two lines. The correlation for DC engines (upper line) was then obtained by translating the DI slope to the mean level of the DC data set. This procedure was preferred to fitting a new slope to relatively few DC data points embracing a much smaller horsepower range. The upper line, or DC correlation, was used in conjunction with the Fig. 8-17 performance map to develop the diesel engine SFC/power output characteristics shown in Figs. 8-18 and 8-19.

Figure 8-21 indicates that the SFC for DC engines is higher (by about 10 percent) than the SFC for DI engines. This result is supported by the following considerations:

1. The DC engine has a greater heat loss due to (a) the higher surface-to-volume ratio of the divided chamber and (b) higher secondary turbulence. It also has a greater throttling loss due to the flow restriction created by the antechamber throat.

2. The DI engine has a higher maximum pressure and a higher pressure rate than the DC engine. Therefore, it should have a correspondingly higher indicated thermal efficiency.

It should be cautioned that Fig. 8-21 may not be freely translated as a statement on engine fuel economy. Duty cycle may have a significant differential effect on the overall performance of the two engine types.

The effects of turbocharging on minimum SFC were examined and were found to be negligible. For a moderate degree of turbocharging (e.g., 40 percent), the minimum SFC is reduced about 5 percent from the values shown in Fig. 8-21. If the degree of turbocharging involves a reduction in compression ratio to avoid excessive peak cylinder pressures, the improvement in SFC is smaller (or non-existent) and depends in part on the net change in compression ratio.

8.3.3.2 Specific Weight

Specific weight data for compression ignition engines are correlated with rated horsepower in Fig. 8-22. The correlation attempts to distinguish the weight characteristics of DC engines (PC and TC types) from DI engines. The theoretical justification for doing this is that DI engines develop higher peak combustion pressures and therefore it is reasonable to expect that certain engine components such as the cylinder head, connecting rods, crankshaft, and bearings will tend to be heavier than similar components in the DC engine.

Seventy-two data points are shown of which 45 represent DI engines. Because of their number and range of horsepower, the DI data were used exclusively to establish the trend of specific weight with rated horsepower. The correlation for DC engines was obtained by translating this slope to the mean level of the DC data set. The data shows that DC engines generally run 25 percent lighter than DI engines. Even so, a 100-hp DC diesel is 80 percent heavier than a spark-ignition engine with the same power rating.

The use of turbocharging as a means of increasing power output and therefore improving the specific weight characteristic for diesel engines was examined. The available turbocharged engine data, however, proved not to be useful since the data indicated a specific weight characteristic that falls on or above the naturally aspirated line. This may be related to the fact that turbocharging usually is accompanied by higher mechanical loads, and consequently, engines that are structurally overdesigned are most suitable for conversion.

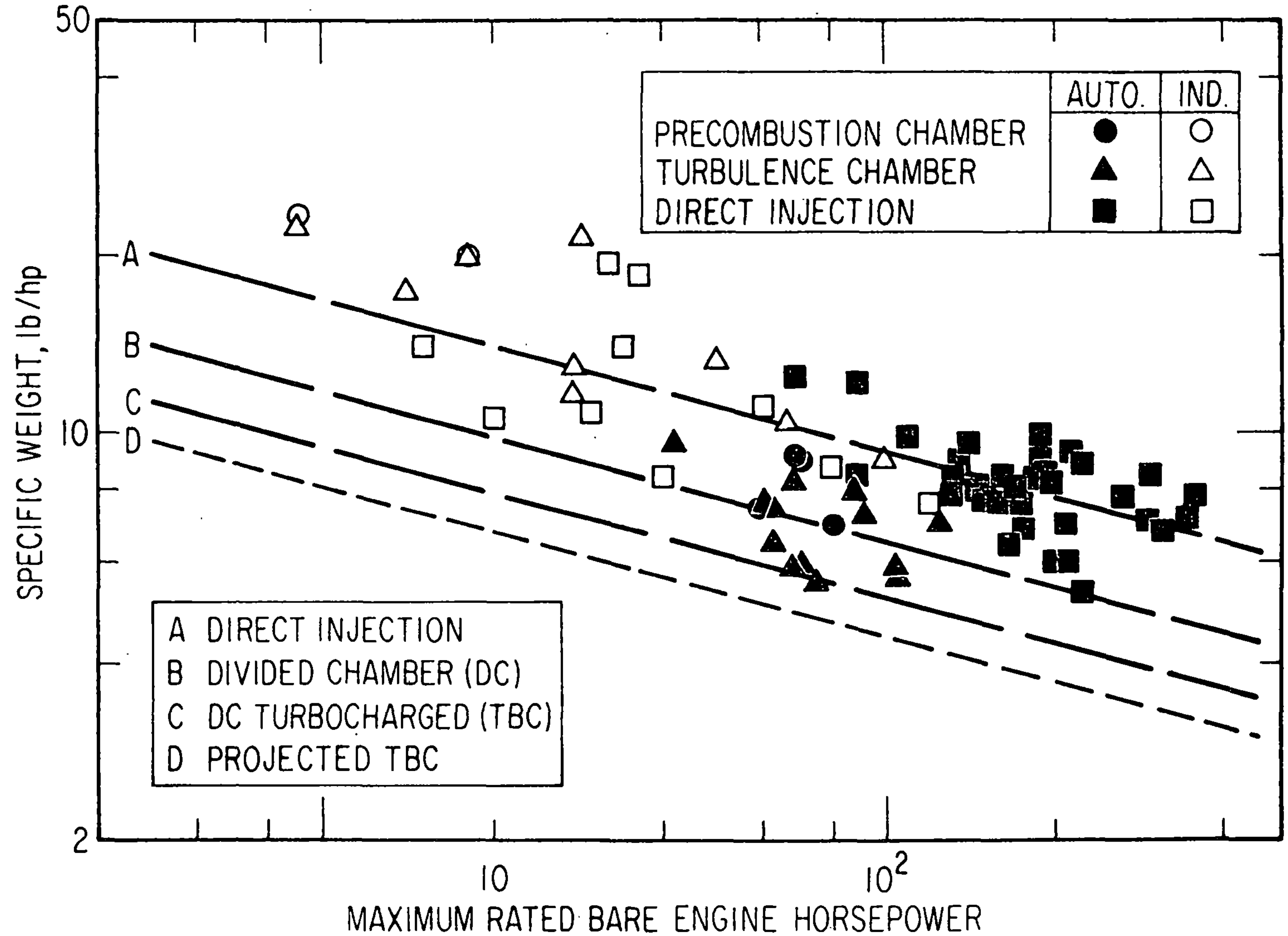


Figure 8-22. Specific Weight - CI Engine

The turbocharged characteristic shown in Fig. 8-22 was obtained by assuming that light automotive DC-type engines might be turbocharged to 40 percent above their rated power level (naturally aspirated) and that turbocharging components might add 4 percent to the engine weight. The power assumption is an extrapolation (down) of current commercial practice which limits turbocharging to give 50 to 75 percent power increase. However, the available data indicate that these levels are not attainable (from the standpoint of limiting peak pressures) with engine specific weights below about 10 lb/hp. With 40 percent turbocharging, the indicated weight of a 100-hp diesel engine is 520 lb, compared to 350 lb for the spark-ignition engine. The weight discrepancy tends to get larger at higher rated power.

The projected 1975 weight characteristic was obtained by assuming that new materials combined with higher operating piston speeds might bring the turbocharged engine specific weight to a level corresponding to 75-percent turbocharging of current DC engines.

8.3.3.3 Specific Volume

Specific volume data for automotive and industrial compression ignition engines are correlated with rated horsepower in Fig. 8-23. The volume represented is the engine envelope from fan to flywheel and from air cleaner to oil pan. There is no reason to expect biases between engines of different combustion chamber design, and no attempt has been made to isolate and correlate the data on this basis. While there is considerable scatter in the plot, the least-squares fit represented by the drawn line indicates a relatively high degree of correlation. The turbocharged characteristic shown in the figure was obtained on the basis of turbocharging to a power output 40 percent above rated horsepower. No additional volume allowance was made for turbocharging components, since these may easily be fitted within the naturally-aspirated engine envelope. Representative dimensions for the DI volume envelope are characterized by the same ratios given in Section 8.2.3 for spark-ignition engines.

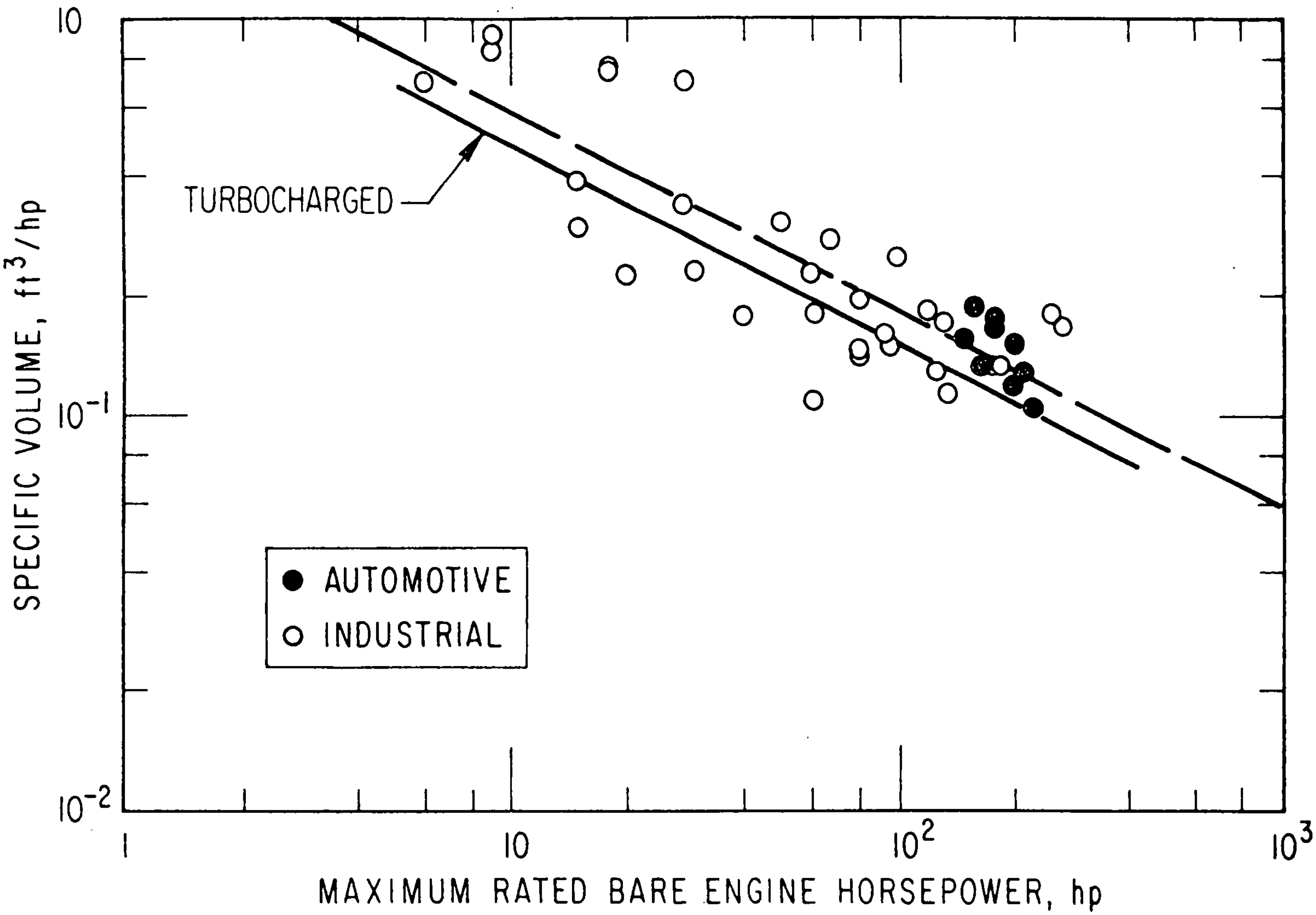


Figure 8-23. Specific Volume - CI Engine

8.4 BRAYTON CYCLE (GAS TURBINE ENGINE)

8.4.1 Thermodynamic Processes

The gas turbine (Brayton or Joule) cycle is illustrated in Fig. 8-24. In the basic cycle the inlet air is compressed (Point 1 to Point 2), heated at constant pressure (Point 2 to Point 4), expanded through a power turbine (Point 4 to Point 5), and discharged to the atmosphere (Point 5) where it eventually reaches equilibrium with the environment (Point 1). As will be shown, it is advantageous to use a regenerator in this cycle to recover some of the rejected heat (Point 5 to Point 6) and reintroduce it into the cycle (Point 2 to Point 3) to conserve input energy.

Gas turbine performance can be improved by increasing the maximum cycle temperature and by improving component efficiencies. Other methods which might be used involve the incorporation of additional components. In particular, intercooling can be added to multistage compressors and reheat can be added to multistage turbines. However, in many applications, including the hybrid vehicle, the complication of the additional ducting and components required by intercooling and reheating cannot be justified for the small performance gains realized. Improvement in efficiency is best obtained by improving component efficiencies and/or by raising temperature limits of the cycle as permitted by material advances. In Fig. 8-25, it is noted that the optimum value of the parameter $\Delta T_c/T_o$ decreases with increasing recuperator effectiveness, but is not affected, to any degree, by independent changes in turbine or compressor efficiency (ΔT_c is the actual temperature rise across the compressor and T_o is the compressor inlet absolute temperature). However, lower $\Delta T_c/T_o$ values require fewer compressor stages or lower stage pressure ratios, implying potentially higher compressor performance. Hence, in practice, high recuperator effectiveness is doubly desirable.

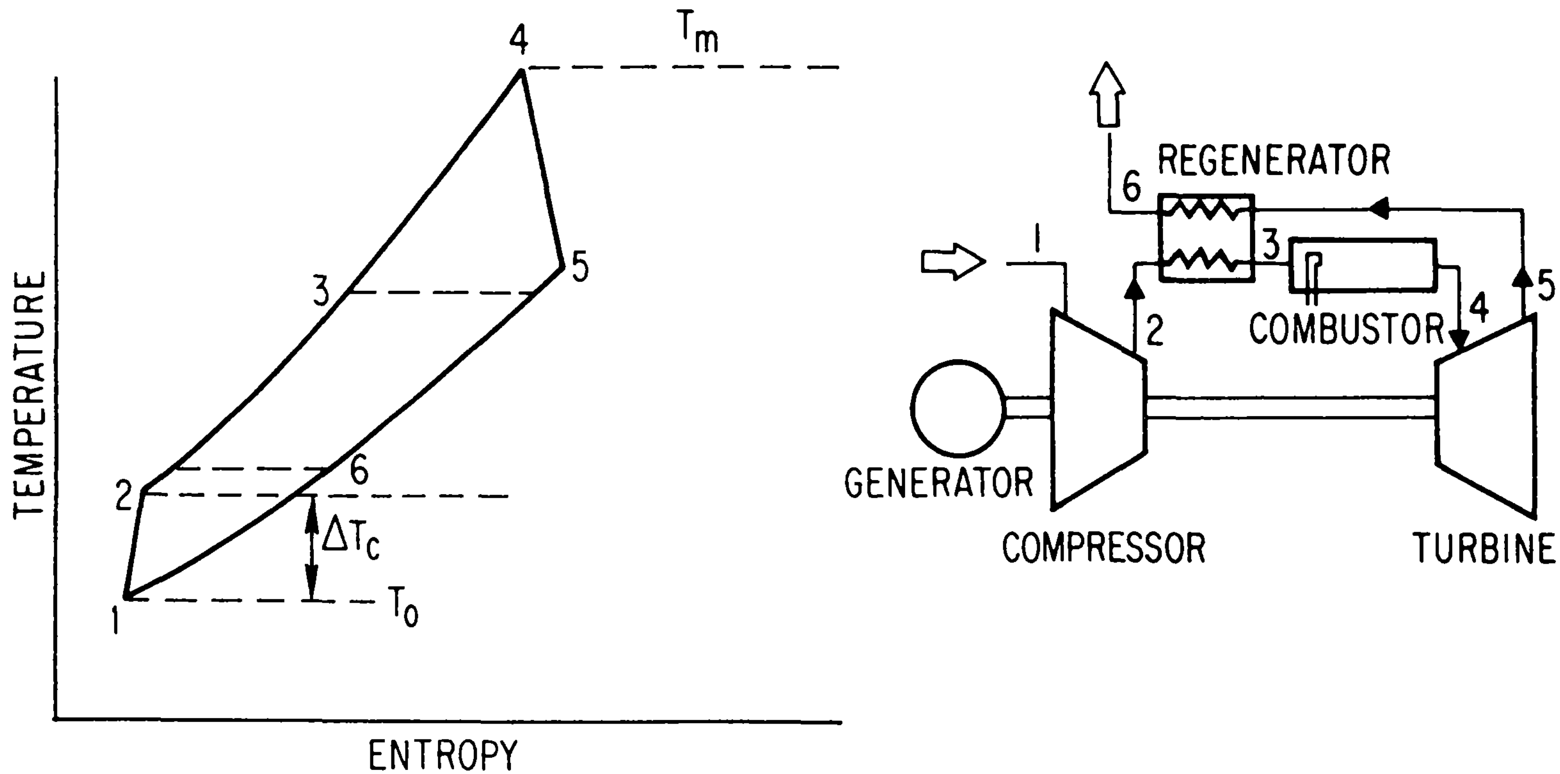


Figure 8-24. Gas Turbine Cycle and Schematic Arrangement

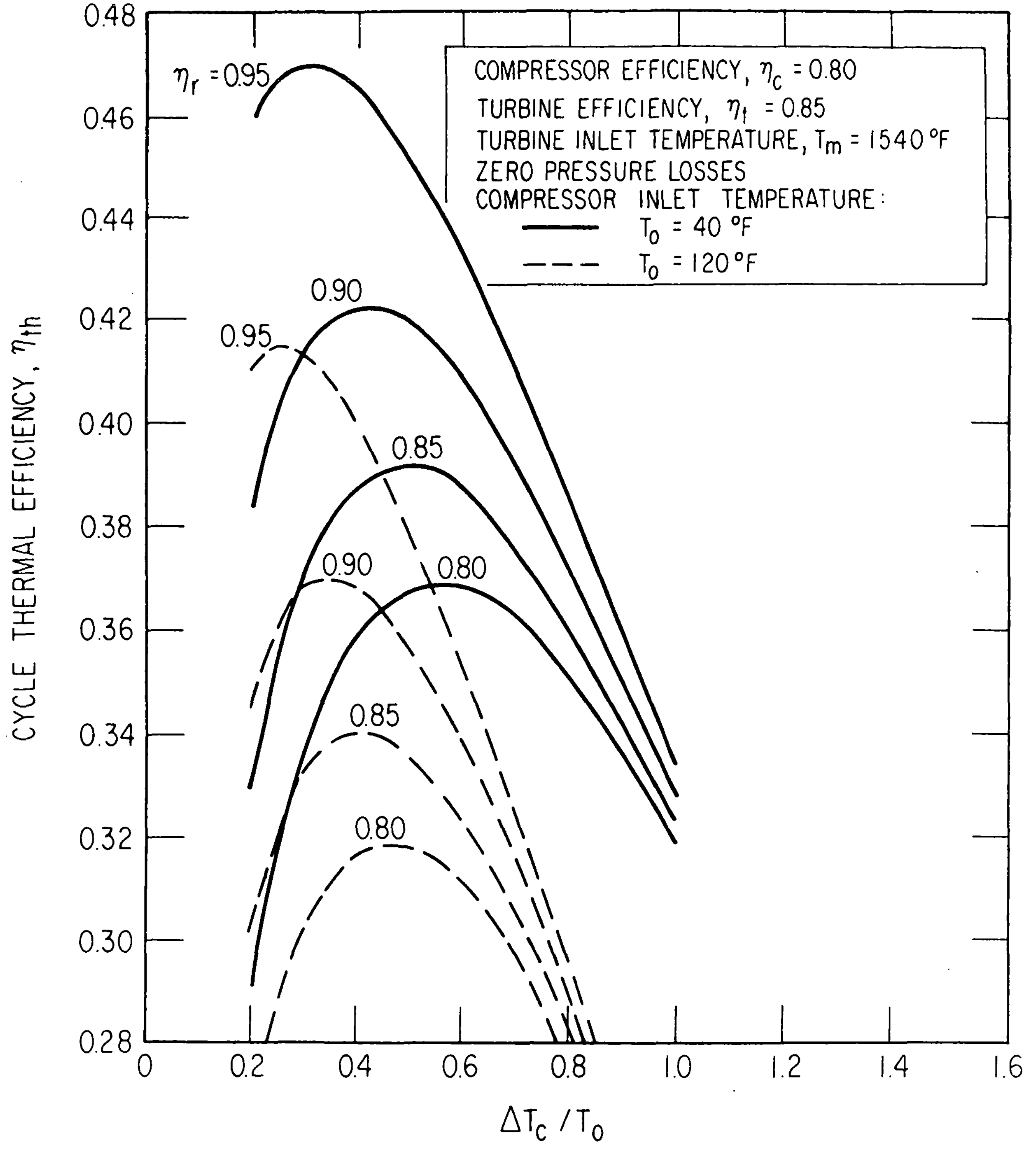


Figure 8-25. Effect of Recuperator Effectiveness on Recuperated Brayton Cycle Performance

8.4.2 Vehicular Design Considerations

Vehicular gas turbines under development as prime power sources all employ regeneration. As indicated in Fig. 8-26, the cycle arrangements can be complicated, even without such special features as variable turbine nozzles.

Because of the possibility of extended idle operation in the hybrid application, it is important to consider cycle arrangements which would provide lower idle fuel consumption than that normally encountered with constant speed gas turbines. In constant speed gas turbines, no load fuel consumption may be as high as 60 percent of the full load consumption. A major design consideration of vehicular gas turbines which has actually dictated the final design is the response to load change. Because of the differences between normal vehicular operation and hybrid operation it was considered advantageous to do some preliminary studies to determine possible configurations for the hybrid vehicle.

The gas turbine configurations investigated are as follows:

1. Simple
2. Simple with free turbine
3. Simple with regeneration
4. Simple with free turbine and reheat
5. Twin spool
6. Twin spool with regeneration
7. Twin spool with intercooling
8. Twin spool with reheat
9. Twin spool with regeneration and intercooling
10. Twin spool with regeneration and reheat
11. Twin spool with intercooling and reheat
12. Twin spool with regeneration, intercooling, and reheat

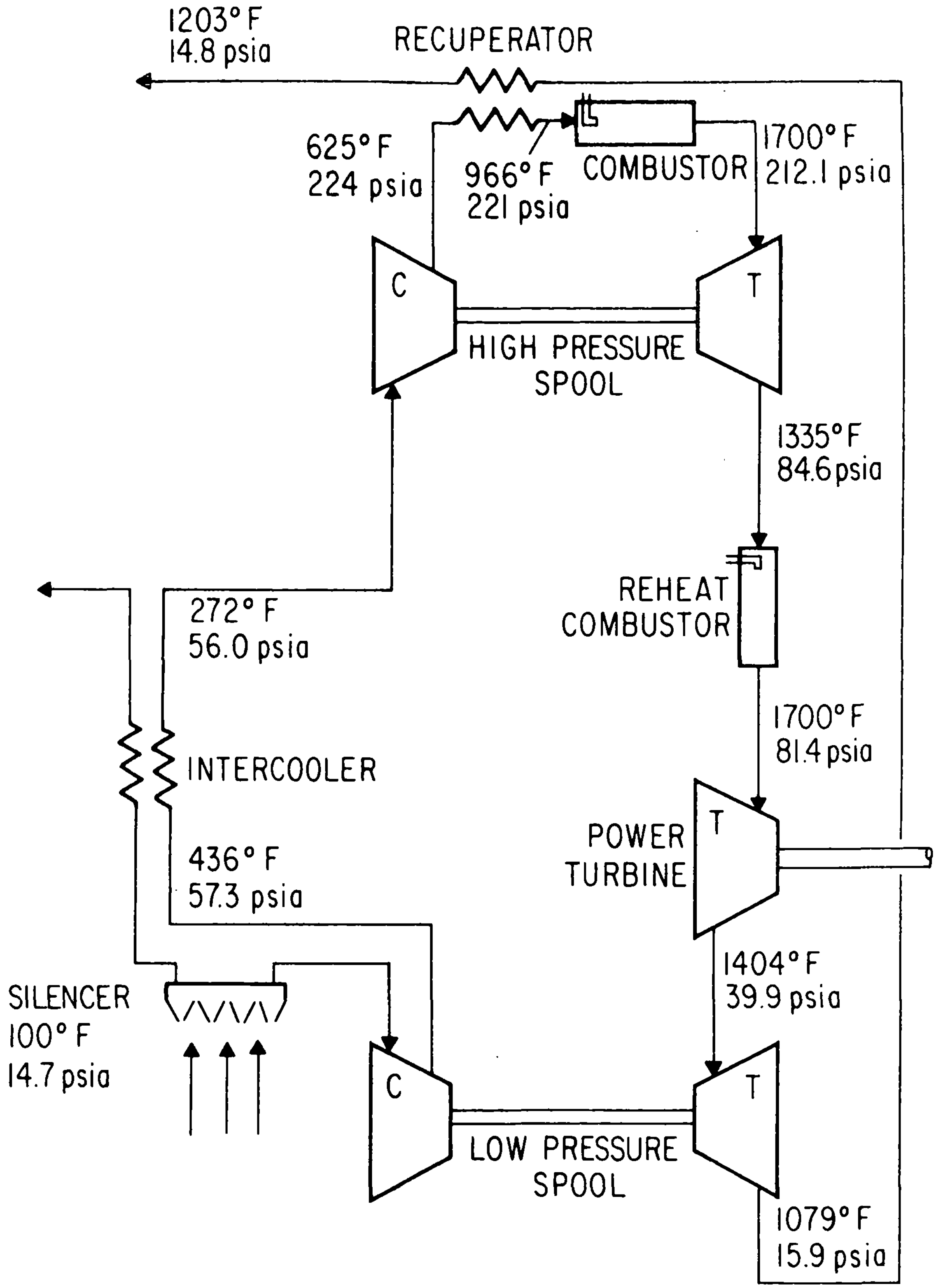


Figure 8-26. Cycle Diagram of Ford 704 Gas Turbine Engine

Configurations 1, 2, and 3 represent those in common use for vehicular applications. The other arrangements were examined to determine if their characteristics might be better suited to hybrid application. The interest in the free turbine and twin spool configurations stems from their good idle and acceleration characteristics. Intercooling and reheat are chiefly valued for their effect upon increasing the power density of the gas turbine. Regeneration is the most effective method of decreasing fuel consumption by increasing thermal efficiency.

The assumptions used in the analysis are given in Table 8-1. The stage pressure ratio was established at 2.8 to represent what a moderate performance centrifugal compressor might do. The other efficiencies and the turbine inlet temperature would be considered advanced over what might be considered for industrial purposes. They would not, however, be as high as those being achieved in aircraft gas turbines. The assumption of constant specific heat, while technically not exact, will lead to results which will be accurate within 10 percent and therefore usable for comparative purposes.

In this study, a low-speed, no-load condition of 45-percent rated speed was assumed. This assumption, based on a priori judgment, is well above the self-sustaining speed to assure stability and fast acceleration to rated speed, and yet it is low enough to cause a substantial air flow reduction in the machine in comparison to the rated speed air flow. Air flow reduction pays off directly in lower power losses in the cycle at no-load and, therefore, in lower fuel consumption at no-load. The results indicate that the no-load fuel consumption can be reduced from as high as 60 percent of rated fuel consumption at full-speed idle to under 20 percent of the rated full-load consumption by using a low-speed idle. Another method, not investigated, which would produce similar and possibly even lower idle fuel consumption would be to use variable turbine nozzle vanes. Results of the study are presented in Table 8-2.

Table 8-1. Gas Turbine Characteristics Assumed in Cycle Analysis

Compressor Inlet Pressure	14.7 psia
Inlet Temperature	80°F
Pressure Ratio/Stage	2.8
Stage Efficiency	0.80
Turbine Inlet Temperature	1600°F
Stage Efficiency	0.83
Mechanical Efficiency	0.95
Combustor Efficiency	0.98
Pressure Drop	5 Percent of Inlet Absolute Pressure
Regenerator Effectiveness	0.95
Pressure Drop	2.5 Percent Inlet Absolute Pressure/Leg
Intercooler Effectiveness	10°F Above Ambient
Pressure Drop	2.5 Percent Inlet Absolute Pressure
Turbine Weight Flow = Compressor Weight Flow	
Specific Heat	1.395
Fuel Lower Heating Value	18,700 Btu/lb

Table 8-2. Gas Turbine Cycle Analysis Results

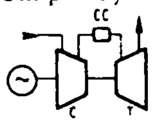
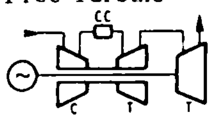
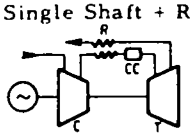
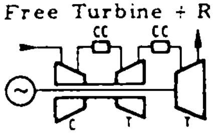
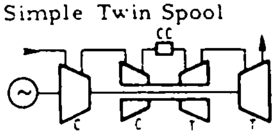
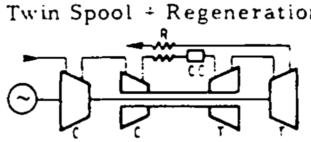
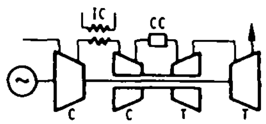
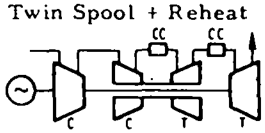
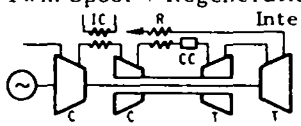
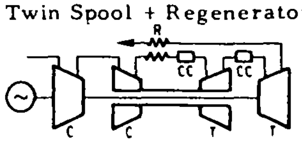
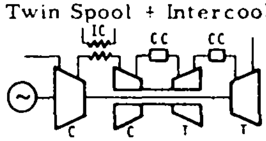
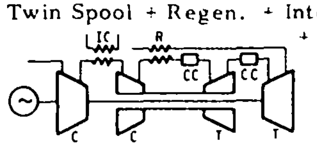
Cycle	Thermal Efficiency	BSFC, lb/bhp-hr	Specific Output, hp/lb (air) min	Exhaust Volume Flow, ft^3/hp (rated)	Low Speed Idle: Turb. Inlet Temp., °F	Low Speed Idle: Fuel Consumption, Percent of Maximum
1. Simple Cycle 	0.11	1.24	0.94	44.2	703	18.2
2. Free Turbine 	0.11	1.24	0.94	44.2	703	18.2
3. Single Shaft + Regeneration 	0.33	0.42	0.84	24.5	1035	11.9
4. Free Turbine + Reheat 	0.10	1.43	0.97	48.6	703	15.8
5. Simple Twin Spool 	0.15	0.93	0.97	34.2	703	22.0
6. Twin Spool + Regeneration 	0.16	0.84	0.89	30.6	1035	6.0

Table 8-2. Gas Turbine Cycle Analysis Results (Continued)

Cycle	Thermal Efficiency	BSFC, lb/bhp-hr	Specific Output, hp/lb (air) [illegible]	Exhaust Volume Flow, ft^3/hp (rated)	Low Speed Idle	
					Turb. Inlet Temp., °F	Fuel Consumption, Percent of Maximum
7. Twin Spool + Intercooling 	0.18	0.74	1.59	21.0	703	30.4
8. Twin Spool + Reheat 	0.15	0.88	1.37	28.9	703	25.6
9. Twin Spool + Regeneration + Intercooling 	0.31	0.44	1.52	18.8	1035	11.4
10. Twin Spool + Regenerator + Reheat 	0.21	0.64	1.23	23.0	1035	7.8
11. Twin Spool + Intercooling + Reheat 	0.17	0.80	1.74	21.7	703	28.1
12. Twin Spool + Regen. + Intercooling + Reheat 	0.29	0.47	1.65	12.6	1035	10.7

Key (~) Generator; C Compressor; CC Combustion Chamber; T Turbine; R Regenerator; IC Intercooler

As expected, the systems using regenerators had appreciably higher thermal efficiencies and therefore lower SFC than did the systems without regenerators. As is generally true, the regenerator is more effective at lower compressor discharge temperatures so that the single-stage compressor or the two-stage compressor with intercooling will have a higher thermal efficiency.

The use of intercooling and reheat increases the horsepower available from a given air flow rate. This implies that the compressor inlet area would be lower with reheat and intercooling and therefore the size of an inlet air or noise filter would be lower. On the other hand, regeneration increases air flow requirements slightly. This can be explained by the pressure loss in the regenerator which leaves less head available across the turbine and necessitates a higher air flow to compensate.

Somewhat the opposite trend is noticed in the exhaust volume flow of the regenerated engines since the temperature drop (density increase) of the gas through the regenerator reduces the size of the exhaust ducting needed.

A review of all these factors indicates that the simple cycle with regeneration (i. e., recuperation) would be preferred for use with the hybrid vehicle in comparison to the other cycles examined. The regenerated gas turbine could use either a single shaft or a free turbine, however, the use of multiple shafts do not offer any advantages in this application. An advantage, indicated by the results of the study, is that the simple regenerated cycle would operate at a lower speed than the other arrangements (because of higher air flow rates which would entail larger rotating machinery) and would therefore be more suited for a generator/alternator drive. As a further result of this study, it does not appear that intercooling or reheat would be advisable for a low-power gas turbine.

8.4.3 Engine Characteristics

Performance characteristics for gas turbine engines are presented in Table 8-3. Based on these figures, specific fuel consumption, specific weight and specific volume are presented in Figures 8-27 through 8-30.

Table 8-3. Gas Turbine Engine Characteristics

	Gas Turbine	Model	Rated HP	Weight (lb)	Volume (ft^3)	Rated SFC (lb/hp-hr)	Comp. Pressure Ratio	Turbine Inlet Temperature °F
1.	AiResearch	331	300	--	--	0.465	8.0	--
2.	AiResearch	331	400	--	--	0.465	8.0	--
3.	Chrysler	120	120	199	17.2	0.602	4.25	1507
4.	Chrysler	CR2A	140	445	20.0	0.515	4.0	1702
5.	Ford	704	300	651	18.3	0.544	4.0	1697
6.	General Motors	305	225	596	13.7	0.535	3.5	1597
7.	General Motors	--	175	825	--	0.586	--	--
8.	Rover	2S/140	150	470	18.3	0.549	3.92	1538
9.	Volvo	--	250	805	23.2	0.401	3.01	1562
10.	Williams	--	80	290	3.5	0.590	4.0	--
11.	Williams	--	180	550	6.05	0.470	4.0	--

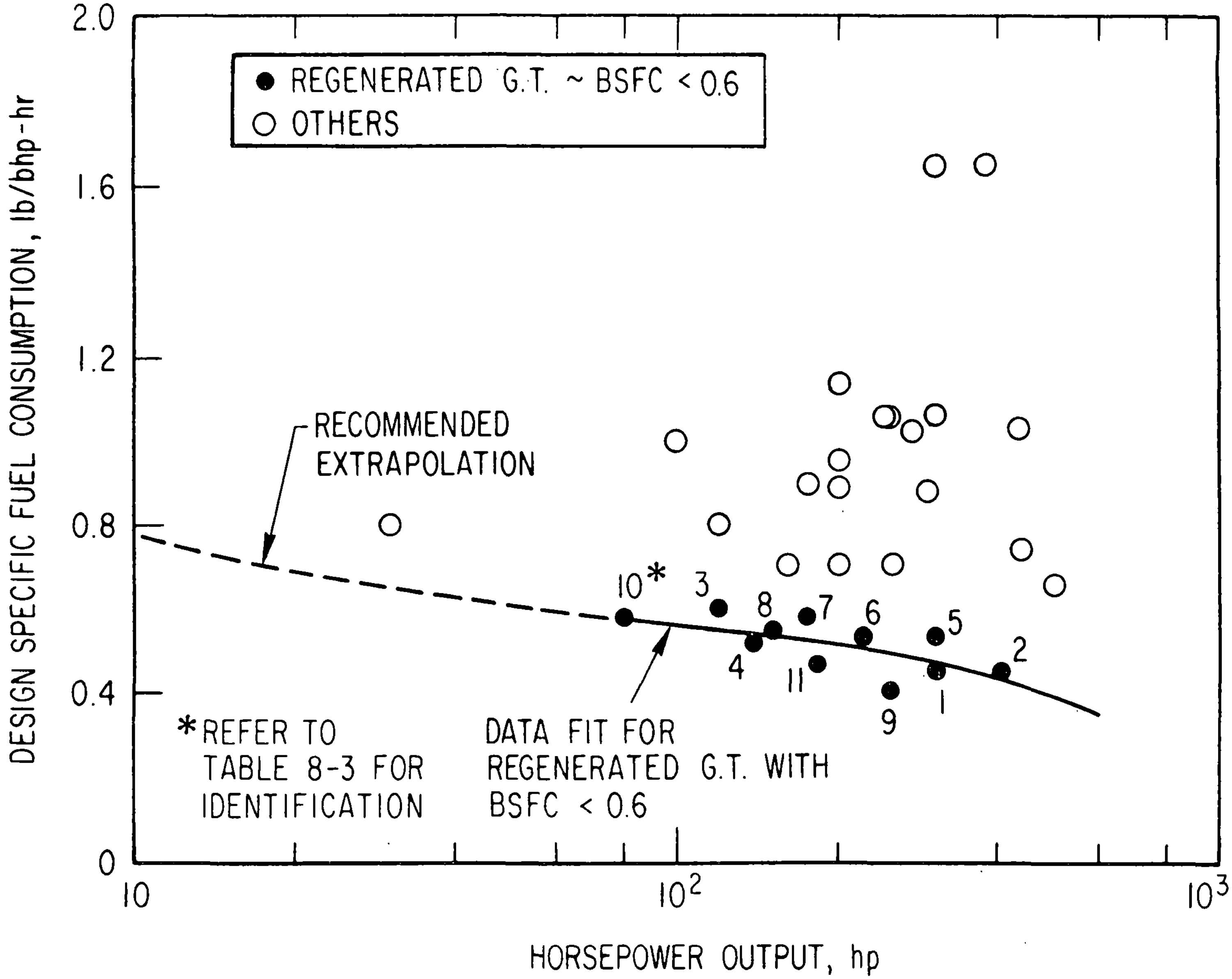


Figure 8-27. Design SFC of Automotive Gas Turbines

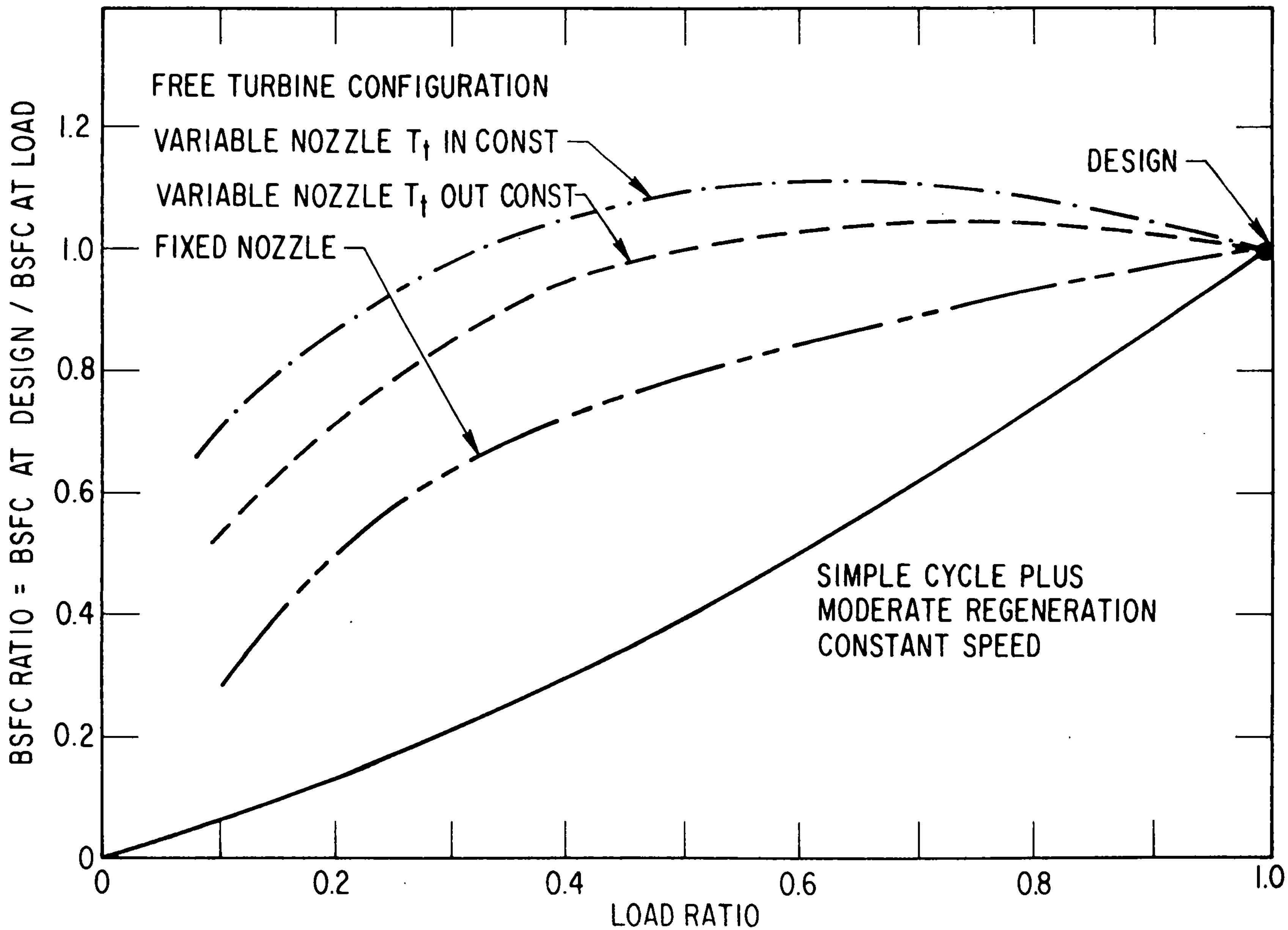


Figure 8-28. Part-Load BSFC Characteristics of Automotive Gas Turbine Engines

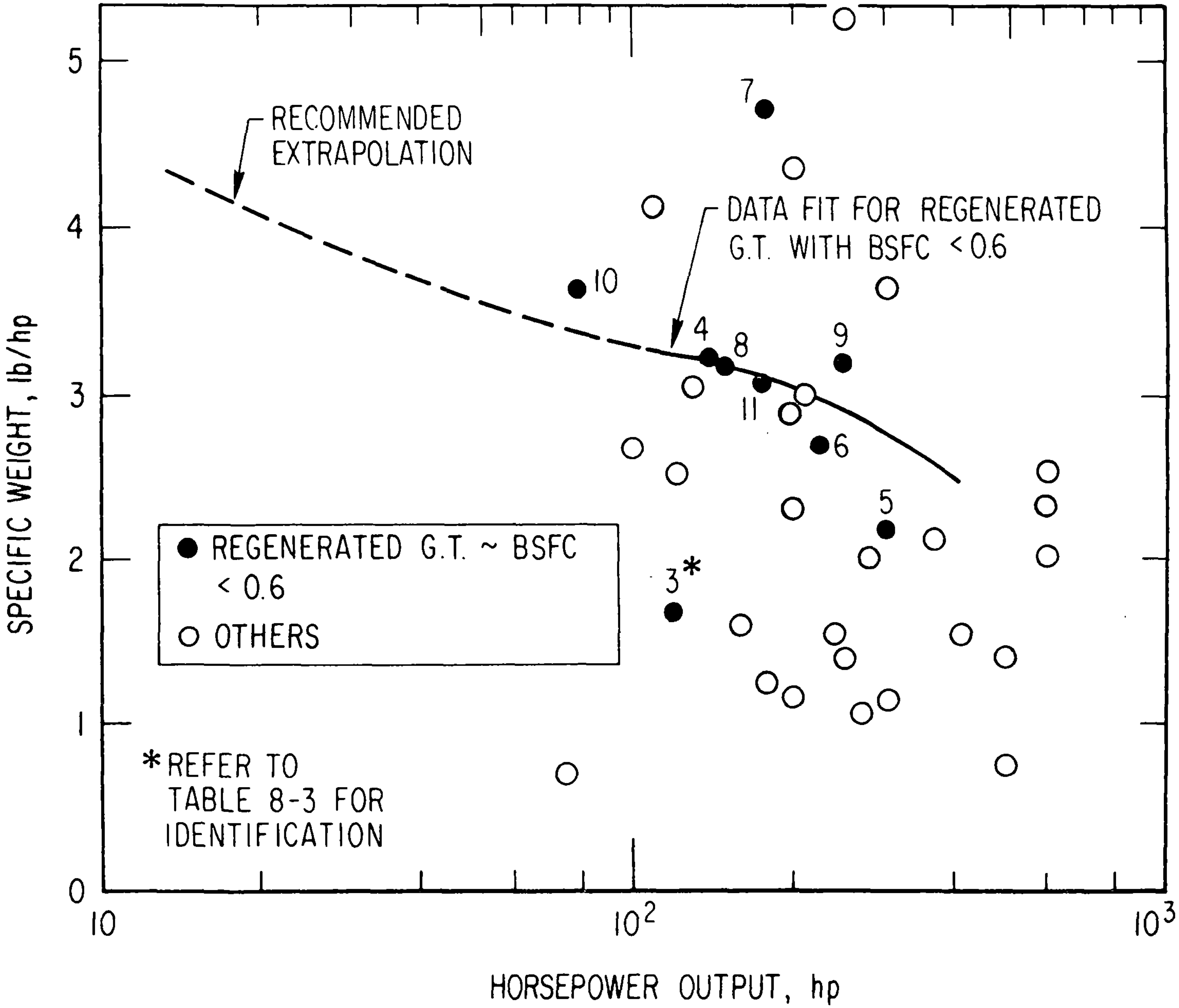


Figure 8-29. Specific Weight of Automotive Gas Turbines

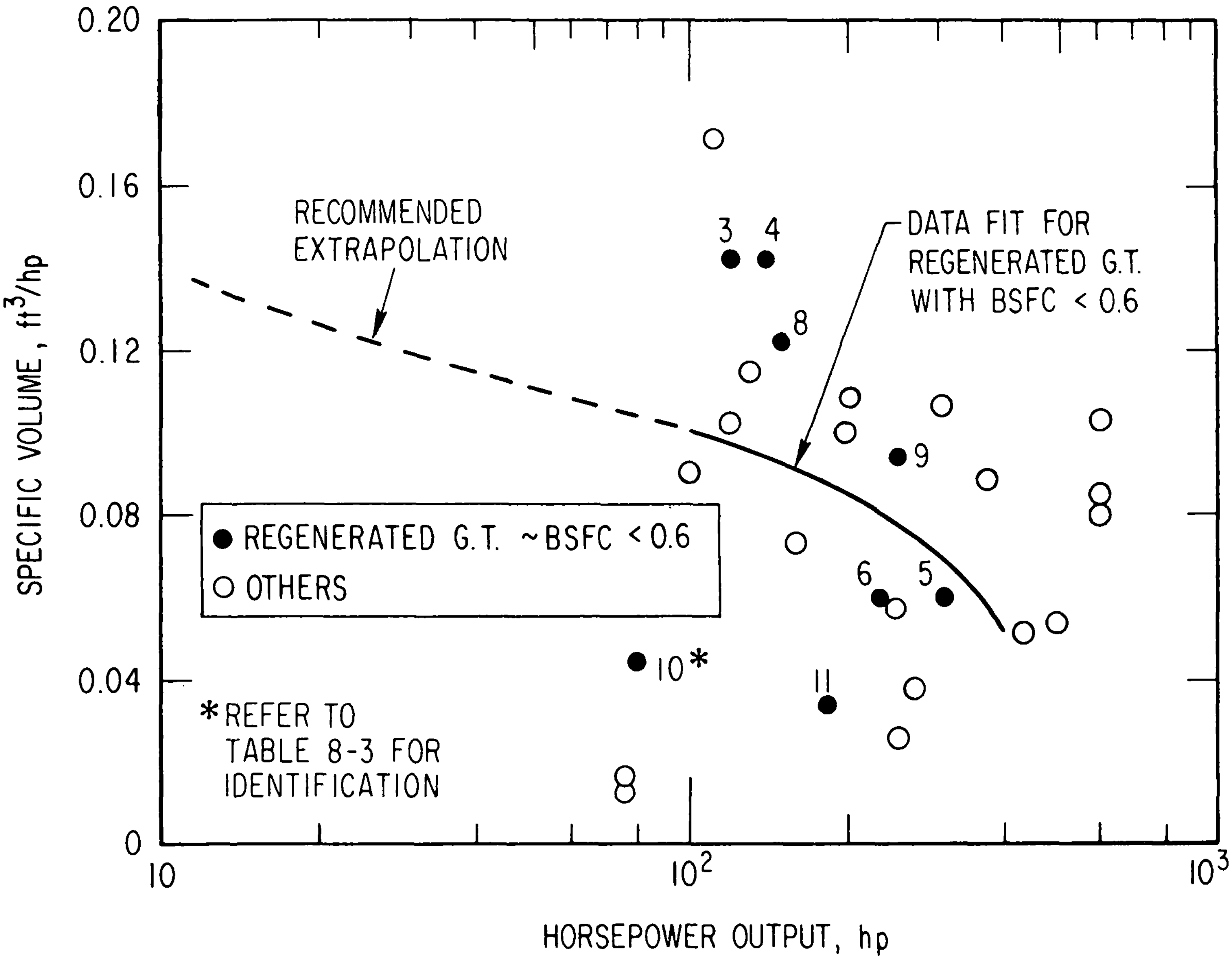


Figure 8-30. Specific Volume of Automotive Gas Turbines

8.4.3.1 Specific Fuel Consumption

Specific fuel consumption characteristics for a number of gas turbines have been plotted in Fig. 8-27. Since the gas turbines represented on the curve vary widely in arrangement and application, the characteristic performance has been based upon only those whose SFC is 0.6 lb/hp-hr or lower. The gas turbines in this set include all of the normal vehicular types. Those not included are, for the most part, aircraft and stationary auxiliary power units which are designed for characteristics other than fuel consumption. It is also to be noted that all of these contain regenerators which should provide a good basis for weight and volume characteristics.

Part-load characteristics for the gas turbine are shown in Fig. 8-28. Here the ratio of design SFC and part-load SFC is plotted. This, in effect, then shows the variation in efficiency with part-load. The upper curves are for a vehicular gas turbine with and without a variable nozzle turbine while the bottom curve is for a constant speed gas turbine with a fixed nozzle. In the upper curves, temperature of the turbine inlet is held constant in one case, while in the other, the turbine discharge temperature is held constant. For the design case used in this study the variable nozzle arrangement with a fixed turbine discharge temperature has been selected. This arrangement gives good part-load characteristics and is "easier" on the turbine since gas temperatures will be lower at part-load. In the case of a constant turbine inlet temperature, turbine discharge temperature would increase at part-load.

8.4.3.2 Specific Weight

Specific weights of the selected gas turbines have been plotted in Fig. 8-29 as a function of output horsepower. A characteristic curve for these data has been established and has been extrapolated into the lower horsepower region of interest. It is to be noted that the gas turbine, at least from these data, does not show as low a specific weight as might be expected. Instead of one pound per horsepower, the specific weight is closer to three pounds per horsepower. The difference is due to the inlet silencers, regenerators and heat exchangers, and the gearing needed in the vehicular design as compared to aircraft-type gas turbines.

8.4.3.3 Specific Volume

Specific volume characteristics for the gas turbine are shown in Fig. 8-30. A wide scatter of data is shown which reflects the type of regenerator used, whether rotary or stationary, and the amount of inlet silencing. Again, the volume is higher than anticipated. The energy conversion section of the gas turbine is usually small but, because of the high volume of gas flow, the inlet and exhaust gas handling sections are large.

8.5 RANKINE CYCLE

8.5.1 Thermodynamic Processes

A cycle diagram and schematic of the Rankine cycle engine is shown in Fig. 8-31. It is difficult to explicitly relate Rankine efficiency to Carnot efficiency because the Rankine cycle working fluid undergoes phase changes, and substantial real-gas effects accompany vapor expansion processes. It is known that cycle efficiency is closely related to the shape of the T-S diagram and, thus, will vary considerably from fluid to fluid. In general, it has been found that the Rankine cycle efficiency, η_{R_i}, can be expressed in terms of the Carnot efficiency, η_c, as

$$\eta_{R_i} = \eta_e \frac{T_m - T_o}{T_m} = \eta_e \cdot \eta_c$$

where

η_e = engine efficiency (constant for a given system design and set of operating conditions)

T_m = the maximum cycle temperature

T_o = the minimum cycle temperature

The term η_e approaches 0.9 for an ideal Rankine cycle.

In the actual cycle the expander will have an efficiency, η_{ex}, of generally 0.7 to 0.85 times that of an ideal isentropic engine. The reciprocating expander will usually have an efficiency of about 0.8. A single turbine

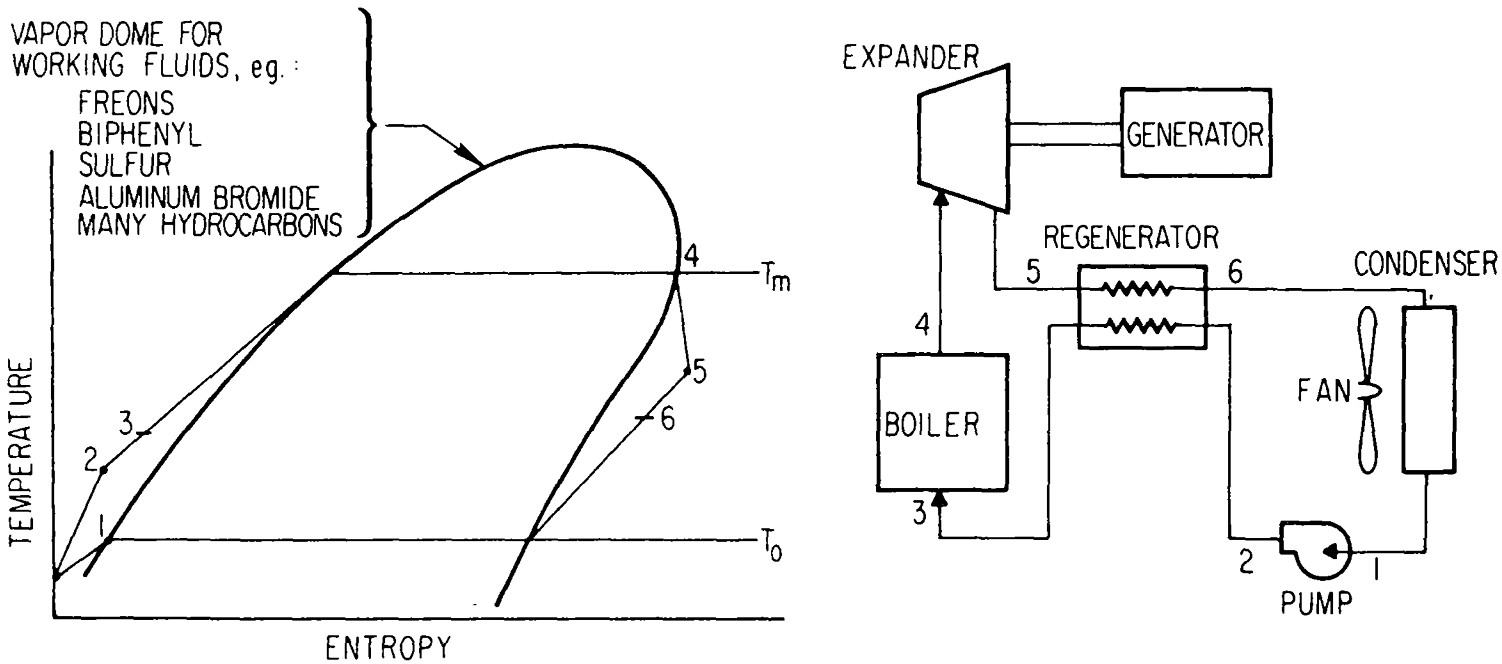


Figure 8-31. Schematic Diagram of Rankine Engine
Using Type B Working Fluid

stage cannot generally utilize effectively the pressure head available and will tend towards lower efficiencies as shown in Fig. 8-32.

In the Rankine cycle, the temperature rise of the working fluid is low compared to other heat engines. Due to incomplete combustion and because it is not practical to exhaust the burner gases at the minimum cycle temperature, the burner efficiency, η_B, will be approximately 0.8.

The Rankine cycle employs or needs considerable auxiliary equipment. In order to keep condenser size low, the condenser fans will use considerable power. The pumps for working fluid and lubricant and the blower fans for the combustor, will also require considerable shaft power so the efficiency in the utilization of shaft power, η_s, will be about 0.7. Mechanical efficiency of the expander, η_m, due to friction losses in bearings and seals will be about 0.95. If all of these efficiencies are combined, then

$$\eta_{Ra} = \eta_e \, \eta_{ex} \, \eta_b \, \eta_s \, \eta_m \, \frac{T_m - T_o}{T_m}$$

$$= 0.9 \times 0.8 \times 0.8 \times 0.7 \times 0.95 \, \frac{T_m - T_o}{T_m}$$

$$= 0.383 \, \frac{T_m - T_o}{T_m}$$

Figure 8-33 illustrates a plot of Carnot efficiency as a function of the maximum and minimum cycle temperatures. Spotted on the curve are the Carnot efficiencies for different engine cycles, including three Rankine engines.

8.5.2 Vehicular Design Considerations

Characteristics of a number of proposed and actual Rankine cycle systems are presented in Table 8-4. In all of the reciprocating engines, except for the Kinetics and the Thermo Electron engines, steam is used. Since none

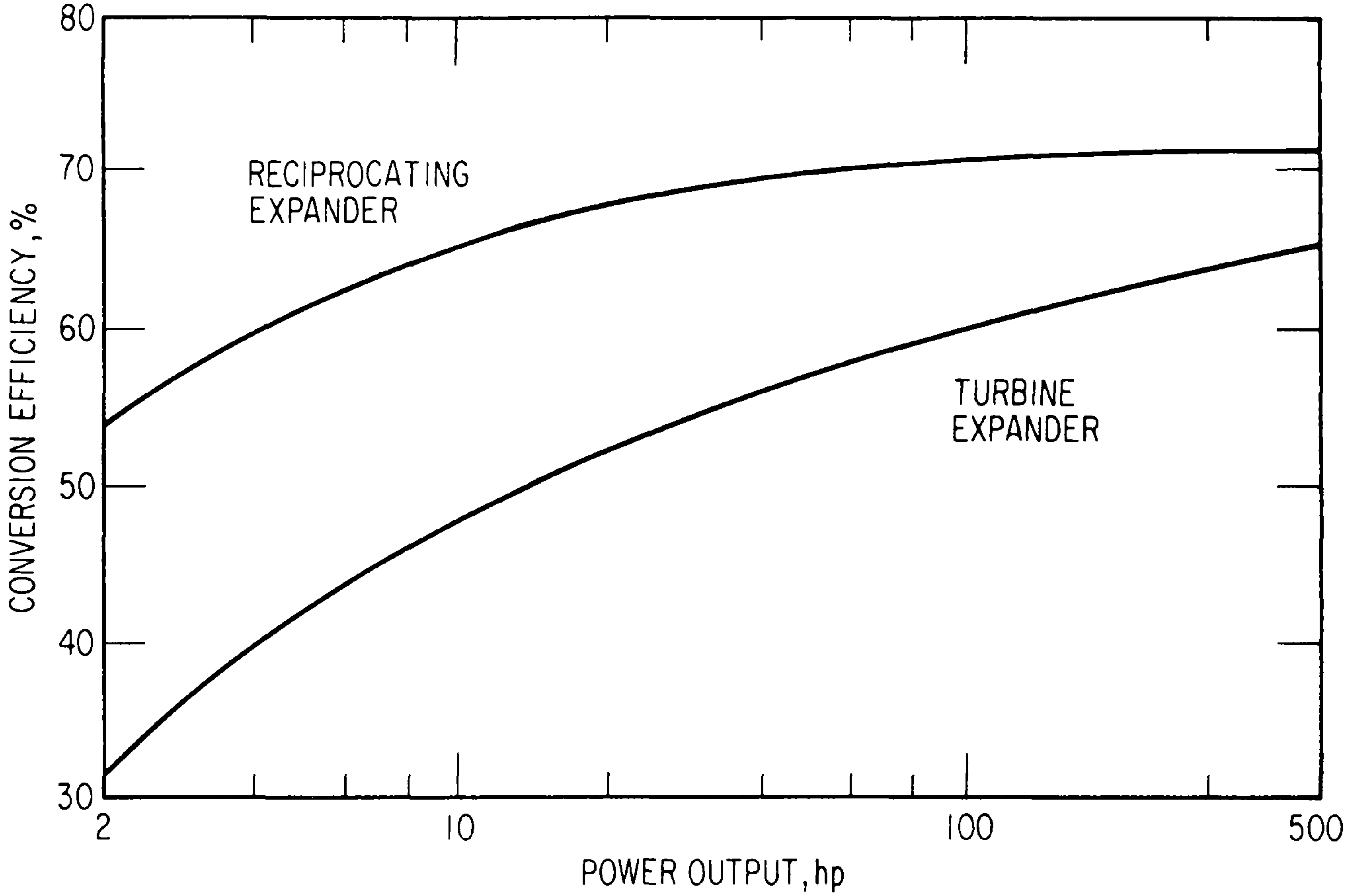


Figure 8-32. Efficiency of Steam Turbine and Reciprocating Expanders as a Function of Power Output (1200°F, 1200 psia)

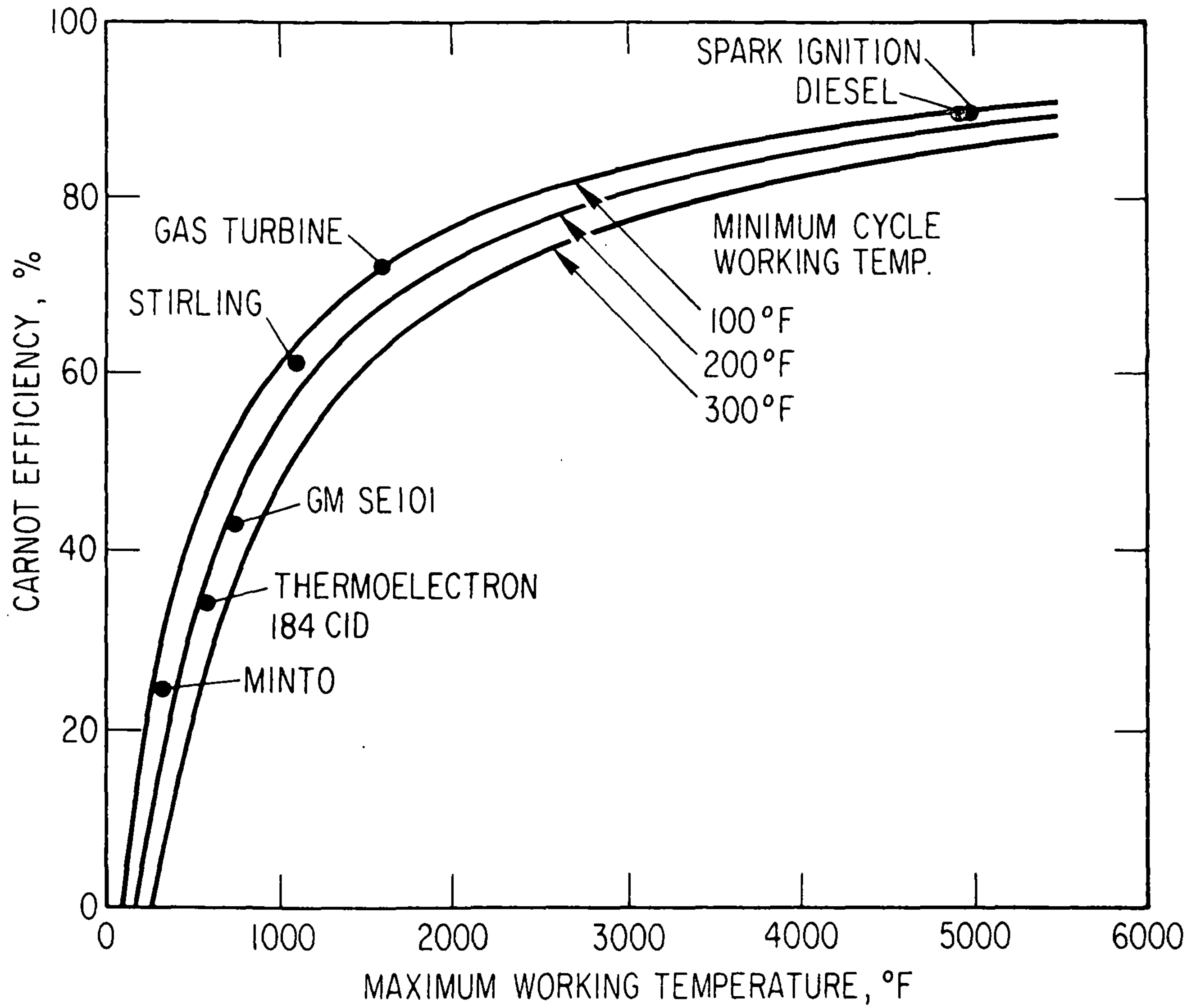


Figure 8-33. Carnot Efficiency of Various Engine Types

Table 8-4. Rankine Engine Characteristics

Positive Displacement Expanders	Power, hp	Working Fluid	Weight, lb	Working Temperature, °F	Working Pressure, psia	Thermal Efficiency, Percent	References
1. Empirical Eng. Co. Midway City, Calif.	250	Steam	100	700	1000	28	8-10
2. Energy Systems Cambridge, Mass.	200	Steam	---	1000	1000	26	8-11
3. General Motors Warren, Mich.	160	Steam	1140	700	800	--	8-12, 8-13
4. Gibbs & Hosick Greensboro, N.C.	1000	Steam	300	1000	2000	--	8-11, 8-14
5. Kinetics, Inc. (Minto) Sarasota, Fla.	350	F 113 $CClF\text{-}CClF_2$	340	390	500	17	8-15, 8-16, 8-17
6. McCulloch Corp. Los Angeles, Calif.	120	Steam	953	900	2000	23	8-18
7. Pritchard Steam Power Melbourne, Australia	50-75	Steam	450	870	1000	--	8-11
8. Thermal Kinetics Rochester, N.Y.	80	Steam	800	850	1200	--	8-11
9. Thermo Electron Eng. Co. Waltham, Mass.	103	Thiophene SCH:CHCH:CH	957	550	500	13.7	8-19
10. Williams Eng. Co. Ambler, Pa.	125	Steam	500	1000	1000	26	8-11, 8-20, 8-21
	300	Steam	650	1000	1000	--	
	400	Steam	800	1000	1000	--	

Table 8-4. Rankine Engine Characteristics (Continued)

Rotary Displacement Expanders	Power, hp	Working Fluid	Weight, lb	Working Temperature, °F	Working Pressure, psia	Thermal Efficiency, percent	References
11. Aerojet Nuclear Azusa, California	8.05	Dowtherm A 26.5% $C_{12}H_{10}$ 73.5% $C_{12}H_{10}O$	---	700	145	17.7	8-22, 8-23, 8-24
12. Fairchild-Hiller Bay Shore, N. Y.	2.0	Perfluoro-2- Butyltetrahydro- foran	145	428	206	14.6	8-25
13. General Dynamics/ Convair San Diego, Calif.	500	Steam	---	1000	1200	22	8-26
14. Kinetics, Inc. (Minto) Sarasota, Fla.	-----	R 113 $CClF\text{-}CClF_2$	---	325	220	----	8-15, 8-16, 8-17
15. NAR/Rocketdyne Canoga Park, Calif.	2.0	Monoisopropyl Biphenyl $C_{15}H_{16}$	74	750	---	7.2	8-27
16. Paxve Costa Mesa, Calif.	160	Carbon Tetra- chloride CCL_4	647	---	---	17.9	8-28

of the efficiencies projected have been substantiated (except for that quoted for the Thermo Electron engine), it is believed that burner efficiency and allowances for auxiliary horsepower have not been considered.

Of the rotary expander systems listed, only the Kinetics and Paxve designs are specifically designed for vehicle application. The other systems have been designed for use in powering generator sets and, in one case, a battle tank.

8.5.2.1 Expander

A variety of expanders have been proposed for use in Rankine engines. Partly because of engineering evolution and also because of specific speed considerations, the majority of Rankine automotive engines have used reciprocating expanders. Generally, a Rankine expander will develop several times the power per unit volume that a corresponding spark ignition or diesel engine can. This is due to the use of a higher average BMEP and the use of a two-stroke rather than a four-stroke design.

Another advantage of the Rankine engine compared to the gasoline engine is its high-stall torque. Because of the torque characteristics, it is possible to eliminate the need for a gear box.

Due to the inherently small size of the reciprocating expander, as well as its relatively high efficiency and simple throttle control, a strong argument for turbomachine expanders does not exist. In central station powerplants, thermal efficiency now approaches 45 percent, but this is accomplished with considerably more complexity than is possible for a small automotive engine. The heat exchanger size, especially that of the condenser, is critical to vehicle installation and since size is directly related to thermal efficiency, it is advisable to use the most efficient expander.

8.5.2.2 Burner

Two types of burners have been used in Rankine cycle engines. Most commonly used is the vortex burner which was used in the Doble steam car and has been adapted to many of the new systems. Because of their relatively long flame path and high volume, these burners provide sufficient time for near complete oxidation of hydrocarbons and carbon monoxide. These burners are capable of heat release rates up to 10^6 Btu/hr-ft^3-atm.

A more compact burner design results from the gas-turbine-type annular burner in which the fuel is injected into a linearly flowing airstream. At Rankine cycle conditions, burner heat release rates of up to 4 x 10^6 Btu/hr-ft^3-atm can be obtained. Since the flame is quenched more rapidly, these burners will produce more hydrocarbons and carbon monoxide than the vortex burners.

The Rankine burner operates at essentially atmospheric pressure and therefore it is relatively easy to introduce recirculation for control of nitrogen oxides. The vortex burner can provide such natural recirculation so that this capability along with the normally low flame temperature results in low nitrogen oxide formation.

8.5.2.3 Boiler

Most Rankine cycle systems have used monotube boiler designs even though more compact systems could be obtained if flash evaporators were used. Safety and ease of control are prime considerations in selecting the monotube design. It would be possible, with the hybrid vehicle, to consider use of an electric heater to assist the initial warmup of the engine. Currently, it is possible to obtain about 1 million Btu/ft^3 in boiler designs.

8.5.2.4 Condenser

There were no major problems with condenser frontal area because most of the early steam cars used engines having less than 75 horsepower and steam venting was allowed. However, if higher horsepower and complete condensing is needed (when working fluids other than water are used), there can be problems in obtaining sufficient condenser area.

Specific output for condensers is about 150,000 to 175,000 Btu/hr-ft^2 of frontal area, 350,000 to 700,000 Btu/ft^3 of core volume, and one horsepower per each 70,000 to 100,000 Btu of condensing capacity.

8.5.2.5 Regenerator Economizer

In the steam cycle with expansion into the wet region, it is not possible to use regeneration. Regeneration should be used, however, when using a working fluid which allows expansion into a superheated region. An economizer can be used which preheats the water by using the boiler flow gases; where even higher efficiency is desired, an air preheater may be used to preheat the combustor inlet air also using flue gases.

8.5.3 Engine Characteristics

8.5.3.1 Specific Fuel Consumption

Specific fuel consumption for Rankine engines as a function of horsepower is given in Fig. 8-34 with a characteristic line estimated. As with specific weight, the estimate is based largely upon the data of the General Motors Corporation SE101 and the Thermo Electron 184-CID engines. The other values appear overly optimistic or have failed to include proper allowances for burner efficiency and auxiliary equipment.

Part-load fuel consumption characteristics for a reciprocating and a rotary expansion Rankine cycle system are presented in Fig. 8-35.

8.5.3.2 Specific Weight

The specific weight characteristics of the Rankine engines listed in Table 8-4 have been plotted in Fig. 8-36. An estimate of engine specific weight as a function of horsepower is also provided. The estimate is mostly based upon the characteristics of the Thermo Electron 184-CID engine and the General Motors SE101 engine, both of which are fully documented. Allowance has been made for some improvement to these and a scale factor equivalent to a 5-percent decrease in specific weight for each doubling of power level has also been included.

Documentation of most other Rankine engine systems is too inaccurate to be used in establishing their characteristics. In the cases where very low specific weights are indicated, the weights are based upon only the weight of the bare expansion engine and do not include provisions for heat exchangers or auxiliary equipment.

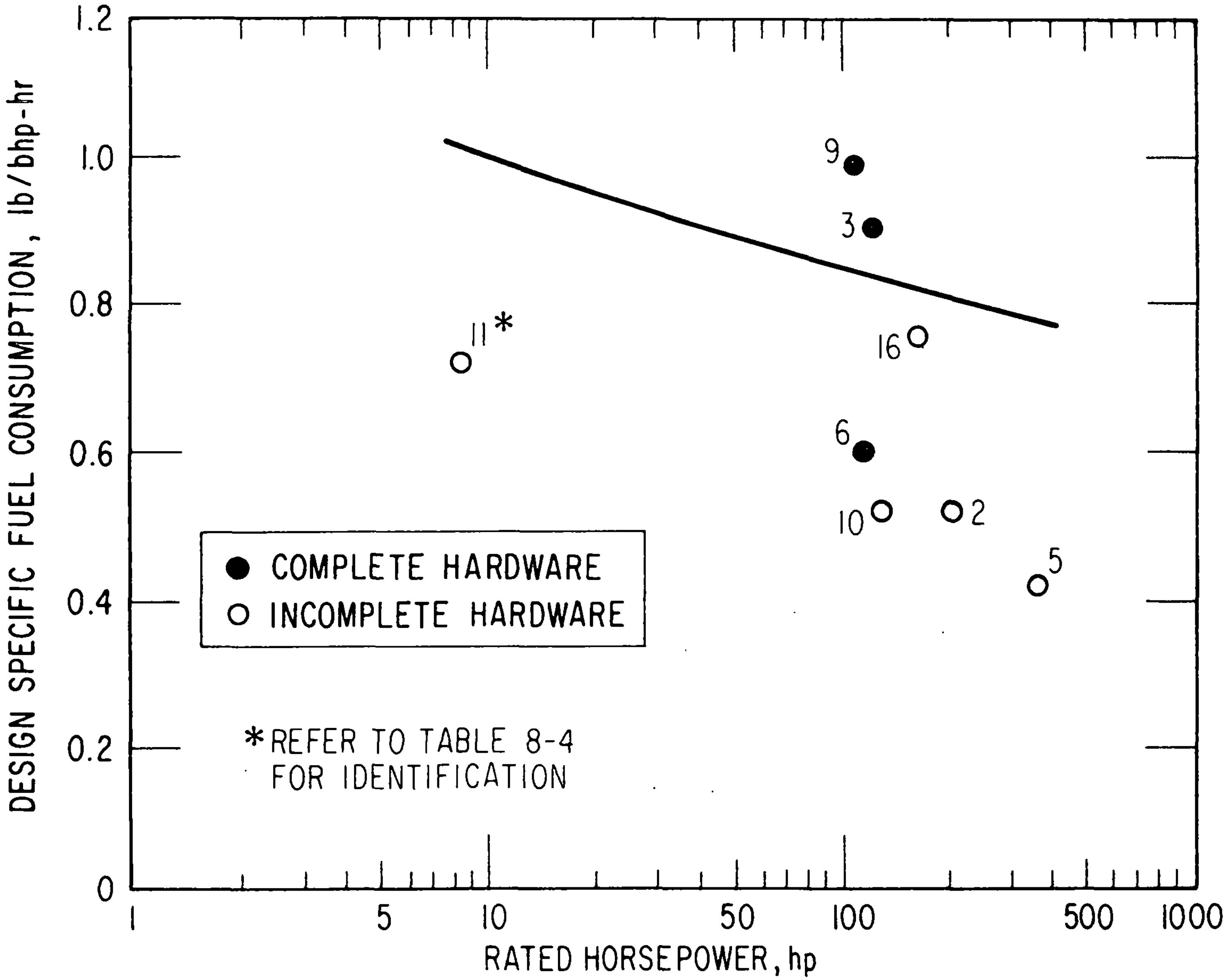


Figure 8-34. Design SFC of Automotive Rankine Engines

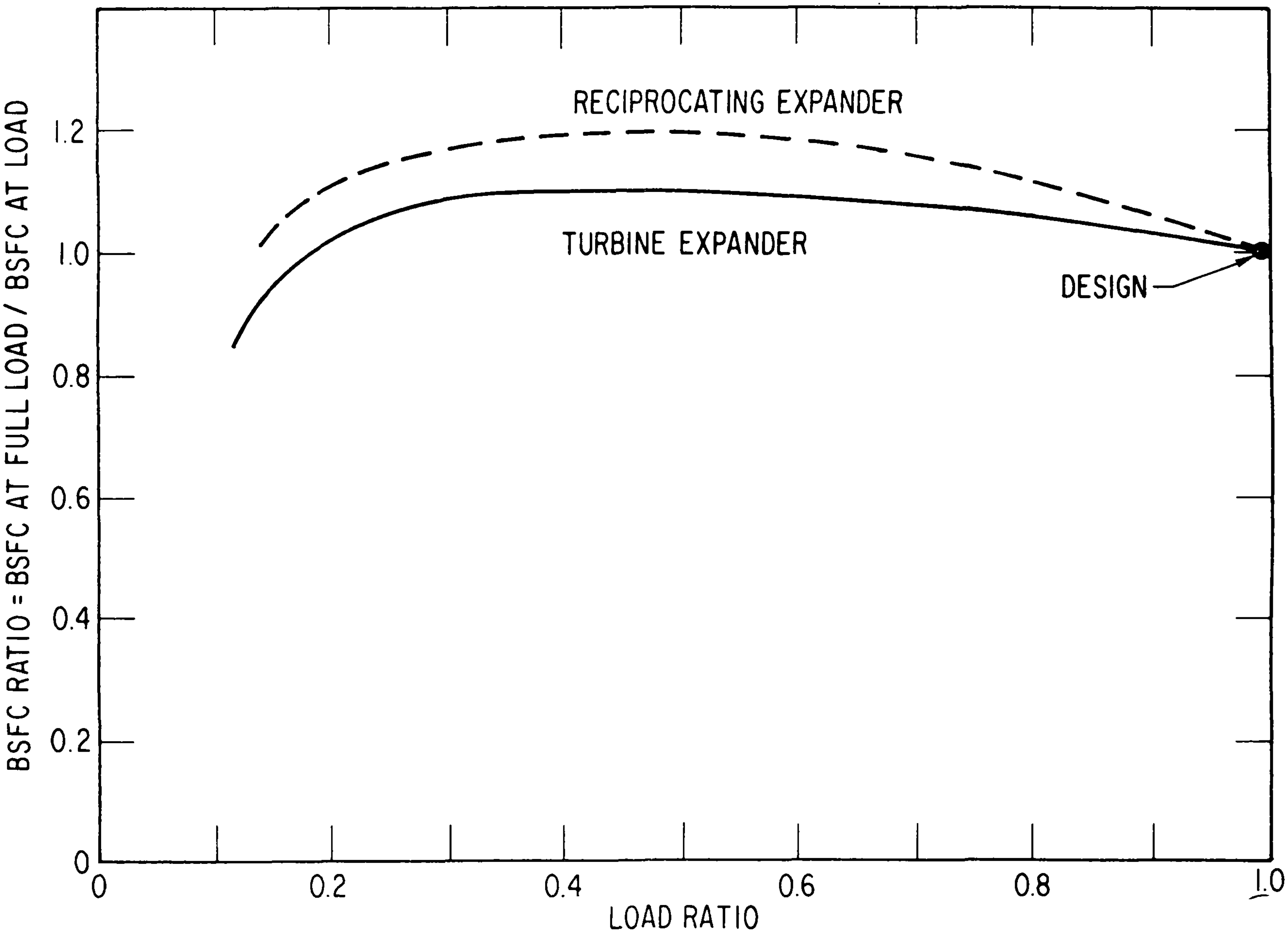


Figure 8-35. Part-Load BSFC Characteristics of Automotive Rankine Engines

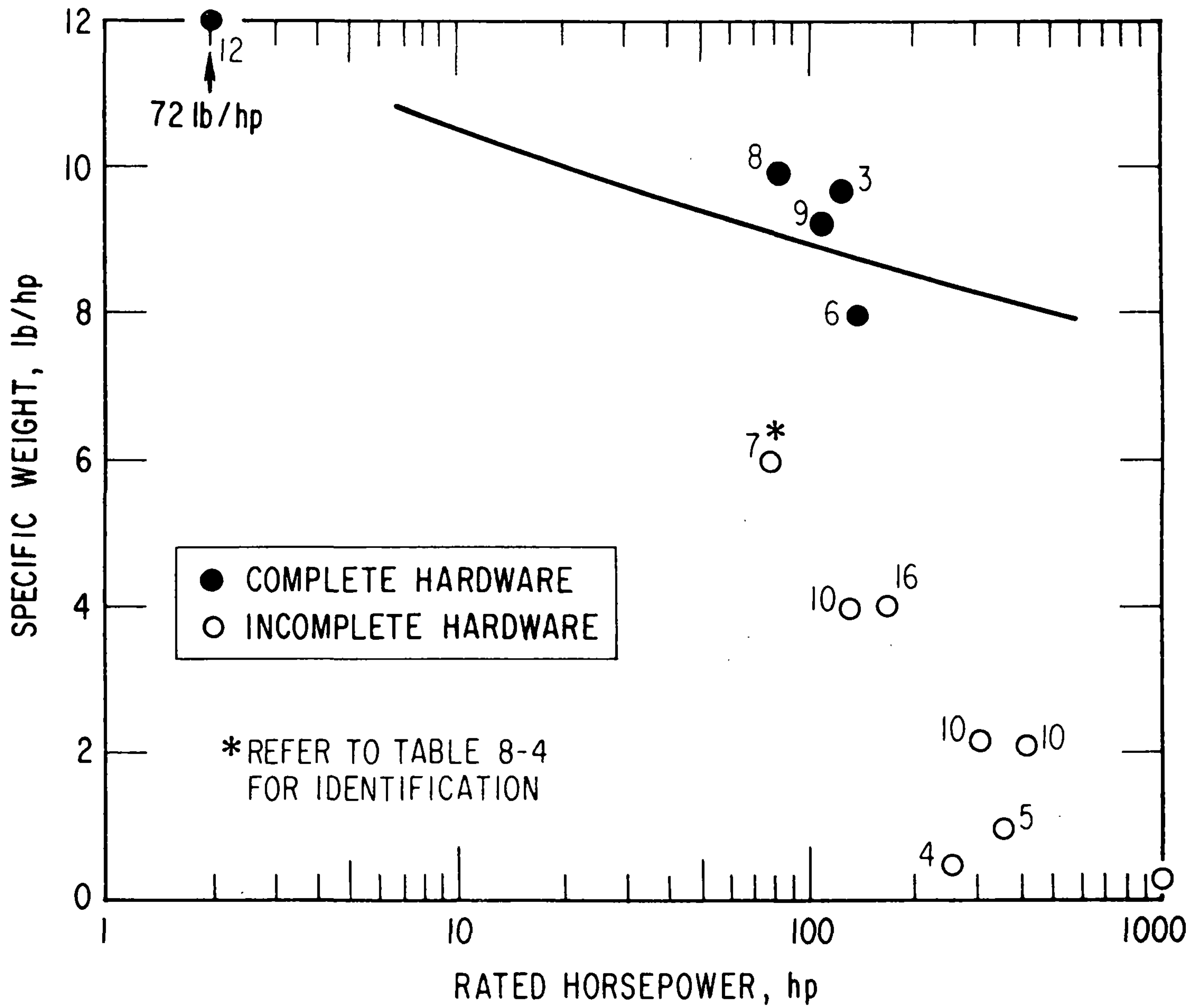


Figure 8-36. Specific Weight of Automotive Rankine Engines

8.5.3.3 Specific Volume

Specific volumes for Rankine cycle systems are presented in Fig. 8-37 with an estimated design trend again with high reliance upon the SE101 and 184-CID systems.

In Table 8-5 the volumes for the Thermo Electron 184-CID and the General Motors SE101 systems are presented in detail.

8.6 STIRLING CYCLE

8.6.1 Thermodynamic Processes

The pressure-volume and temperature-entropy diagrams for the Stirling cycle are shown in Fig. 8-38. The ideal Stirling cycle may be described as a constant volume-regenerative cycle consisting of two constant volume processes and two isothermal processes. It differs from the Carnot cycle in that it employs two constant volume processes in place of the Carnot's two isentropic processes. The practical cycle employs a reciprocating engine that functions with a fluid having a low molecular weight (e.g., helium). Torque, efficiency, and power characteristics of a Stirling engine are presented in Fig. 8-39.

8.6.2 Cycle Characteristics

The basic cycle efficiency equations are independent of working fluid. However, the design of the engine tends to be limited by heat transfer, and therefore, it is advisable to use low molecular weight gases such as hydrogen and helium. While in some cases hydrogen has been used, safety considerations have more often prompted the use of helium. It is also believed that the use of hydrogen at high temperatures should be avoided because of possible leakage. Loss of efficiency using helium rather than hydrogen is on the order of three percent.

Aluminum chloride has been suggested as a working fluid because of its apparently good heat transfer characteristics; however, no known work has been done with this gas. Since the boiling point of aluminum chloride is 361°F, such a system could be used only in special situations.

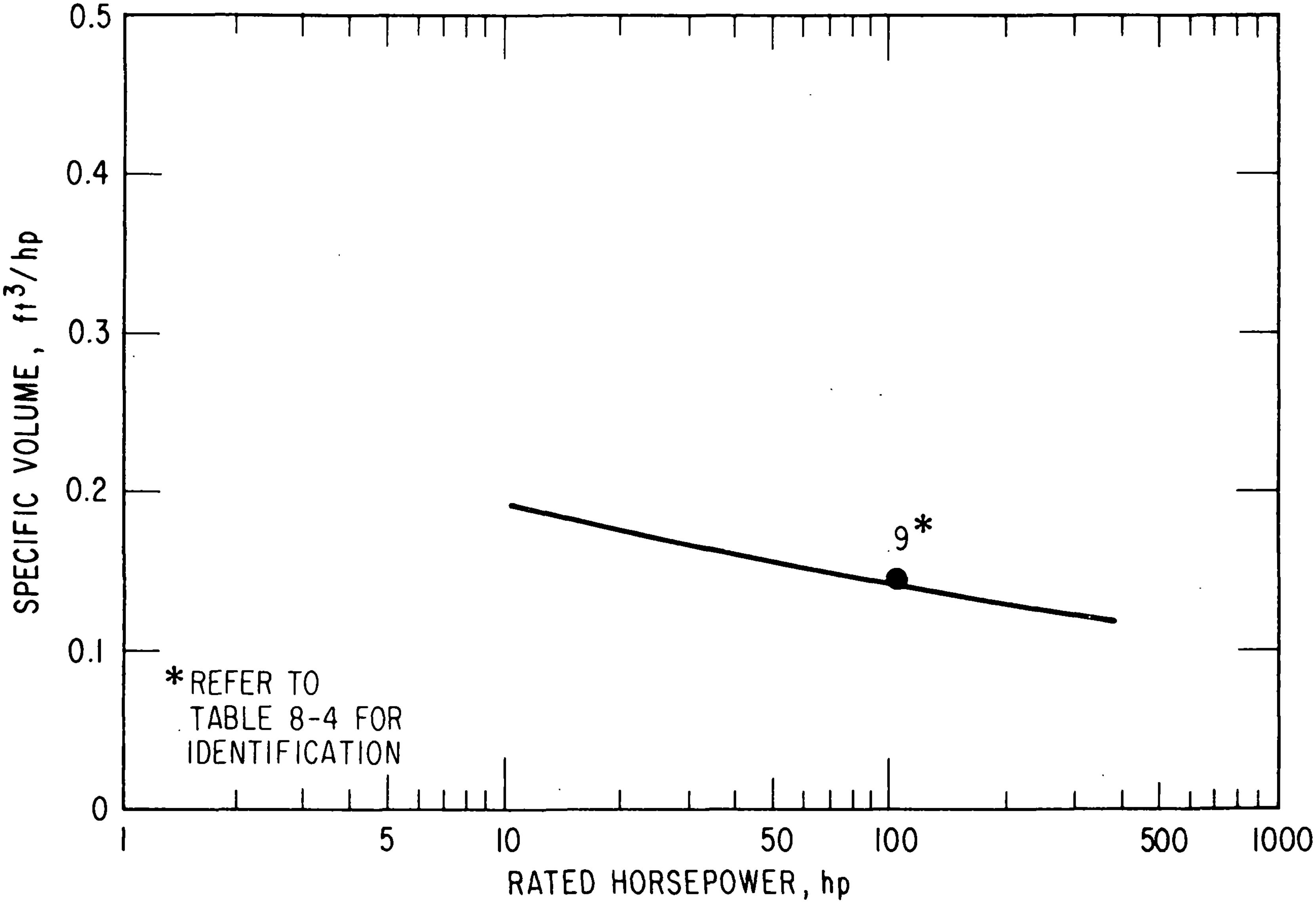
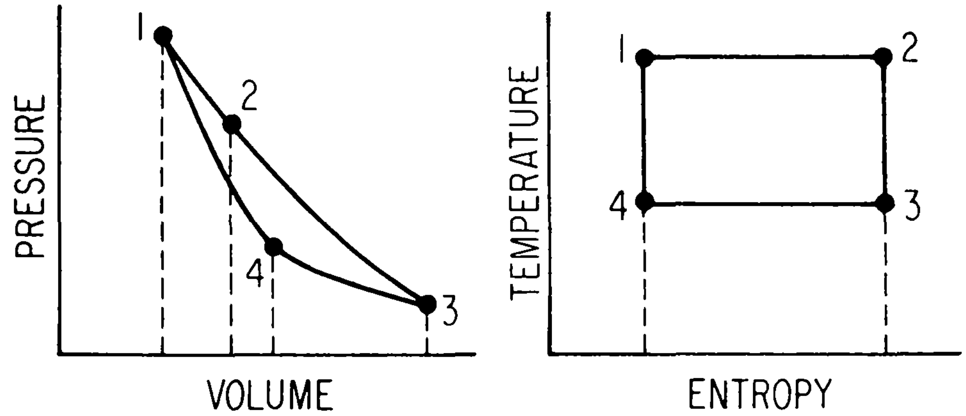


Figure 8-37. Specific Volume of Automotive Rankine Engines

Table 8-5. Performance of Rankine Heat Engines

System	Thermo Electron 184-CID	General Motors SE-101
Working Fluid	Thiophene	Water
Net Shaft Horsepower	103.2	113
Engine		
Dimensions, in.	19.7 x 19 x 27.2	------
Volume, ft^3	5.91	------
Specific Horsepower, hp/ft^3	17.5	------
Burner		
Dimensions, in.	20.2 x 21.3 x 8.5	10 dia x 7.5 (two)
Volume, ft^3	2.02	0.68
Btu Output, Btu/hr	2.06×10^6	4.2×10^6
Combustion Intensity $Btu/hr\text{-}ft^3\text{-}atm$	2.8×10^6	3.75×10^6
Boiler		
Dimensions, in.	18.2 x 21.3 x 14.5	32 x 15 x 11.5
Volume, ft^3	2.68	3.34
Btu Output, Btu/hr	1.7×10^6	3.4×10^6
Specific Output, Btu/ft^3	0.64×10^6	1.02×10^6
Condenser		
Dimensions, in.	50 x 19.9 x 3	19 x 42.2 x 5
Volume, ft^3	1.73	2.33
Frontal Area, ft^2	6.91	5.56
Capacity, Btu/hr	1.2×10^6	8.2×10^5
Specific Frontal Area, Btu/ft^2	1.7×10^5	1.5×10^5
Specific Capacity, Btu/ft^3	7.0×10^5	3.5×10^5
Regenerator		None
Dimensions, in.	28.6 x 7.7 x 6.8	
Volume, ft^3	0.87	
Capacity, Btu/hr	2.5×10^5	
Specific Capacity, Btu/ft^3	2.8×10^6	

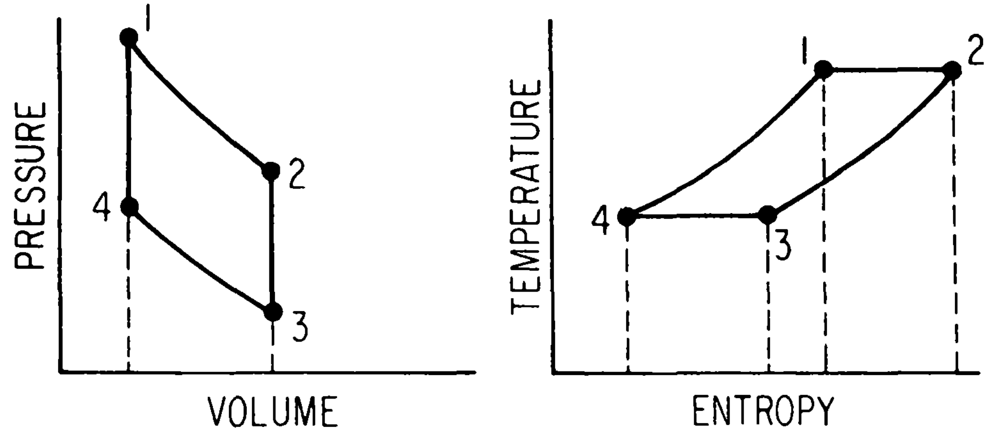
CARNOT CYCLE



EFFICIENCY

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

STIRLING CYCLE



MEAN EFFECTIVE PRESSURE

$$P_m = \frac{(T_1 - T_3)\, wR \ln (V_2/V_1)}{(V_1 - V_3)}$$

Figure 8-38. Pressure-Volume and Temperature-Entropy Diagrams for Carnot and Stirling Cycles

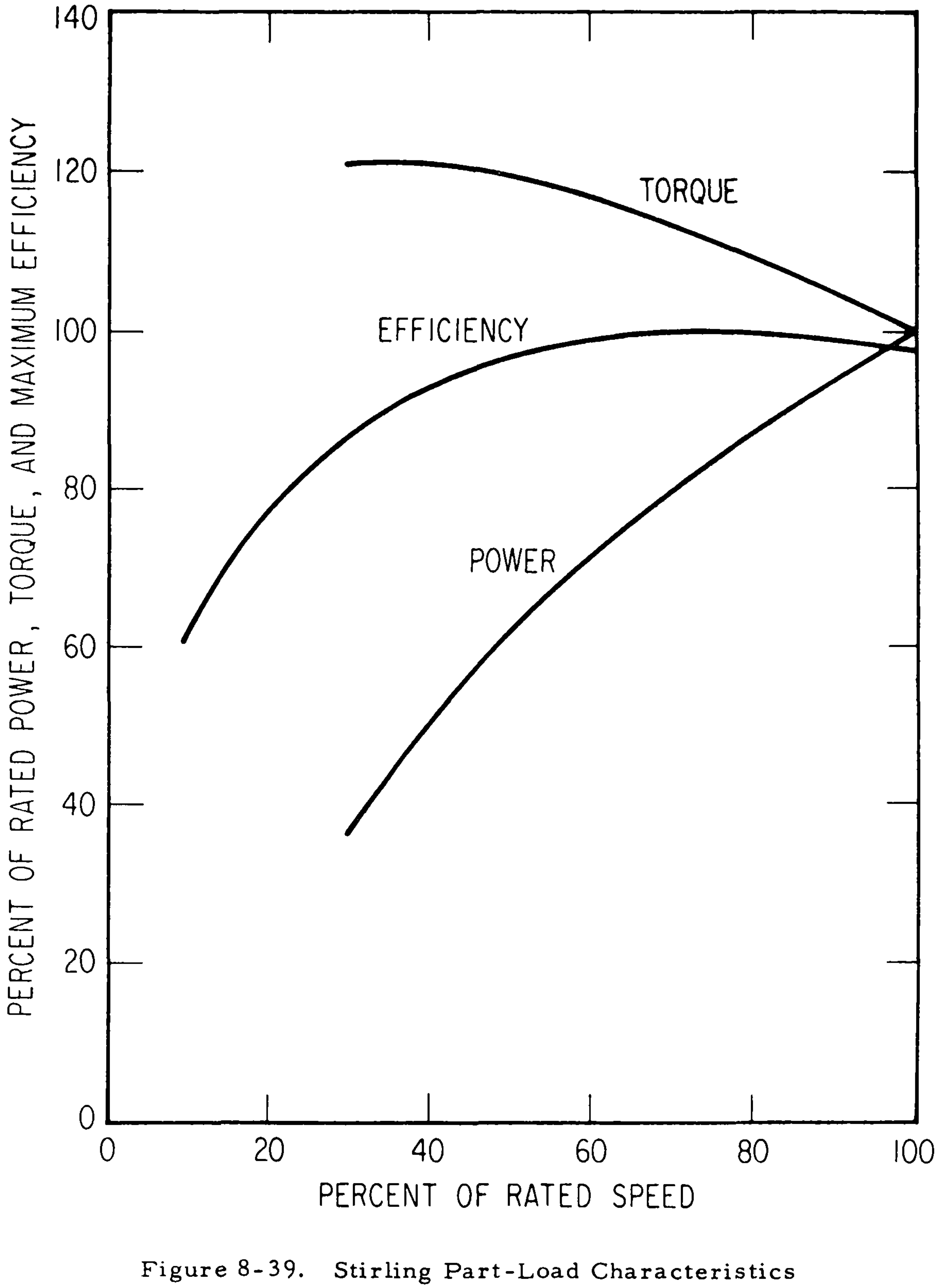


Figure 8-39. Stirling Part-Load Characteristics

8.6.3 Operating Considerations

The major factor creating interest in the Stirling cycle is its high thermal efficiency achieved with a relatively low working fluid temperature. The specific fuel consumption of the Stirling engine is approximately that of the diesel engine. Thermal efficiencies between 30 and 40 percent have been demonstrated. Also, since effectiveness of the regenerators and heat exchangers increases at low loads, part-load efficiency of the engine is good.

The operating temperature of the Stirling engine is on the order of 1200°F with a maximum burner temperature of about 1400°F. This can be contrasted to temperatures above 3000°F for diesel and Otto cycle engines. As a result the nitrous oxide emission is projected as being drastically reduced for the Stirling engine.

A Stirling engine without any acoustic treatment, tested in comparison to a standard military engine of the same size, indicated a sound level 21 db quieter than diesel and Otto cycle engines at 100 ft. The sound level was 40 db.

Since the engine uses an external burner system, any type of fuel can be used. It is likely that the engine could run interchangeably with several types of fuels. Use of non-leaded fuels allows the use of exhaust reactors to eliminate unburned hydrocarbons or nitrogen oxide emissions.

The prototype versions of high-performance Stirling cycle engines have not demonstrated good life capability with something less than 1000 hr being the best life period recorded in the literature. The high efficiencies which have been reported have been obtained with well-tuned, new engines operating at conditions which would not promote long life. One of the main problems with the Stirling engine is the fact that the working fluid does not contain any lubricating qualities so the life of seals and piston rings has been low. However, Philips reports rolling seals having been tested for 11,250 hr which may provide a longer life engine.

The Stirling engine has a sizeable thermal inertia due to the mass of the burner, the heat exchangers, the regenerator, and the engine itself. The engine is therefore sluggish to load changes. Figure 8-40 shows response times to full load from 0 and 50 percent initial loads by changing system pressure and by increasing temperature. Use of a working fluid accumulator has been suggested as perhaps the best method of achieving good response. However, such an approach would affect efficiency and would add additional bulk to the engine.

The engine is best suited as a constant-speed, constant-load device because of its poor response to load change due to the thermal inertia of the regenerator, heat exchangers, and the engine itself, and is thus best suited for the hybrid vehicle rather than as a primary powerplant. The control system would be elaborate to permit higher reaction rates and quite probably would reduce the engine's efficiency.

No one has proposed a multicylinder engine in which the critical functions could be combined. For example, in present designs each cylinder will have its own burner, heat exchangers, and regenerator. The number of elements in the crankshaft and drive mechanism would increase materially if a multicylinder engine were used since each cylinder would have to be controlled individually.

Starting may be a problem with the engine, especially in cold weather. The engine would have to be motored for a considerable period before it could sustain itself. The large number of bearings add to the problem. Also, the engine would tend to motor after the burner has shut down.

The Stirling engine has approximately the same thermal efficiency as an Otto or diesel cycle engine, but in the Otto and diesel engines, about one-third of the energy contained in the fuel goes out the exhaust with only about 17 percent

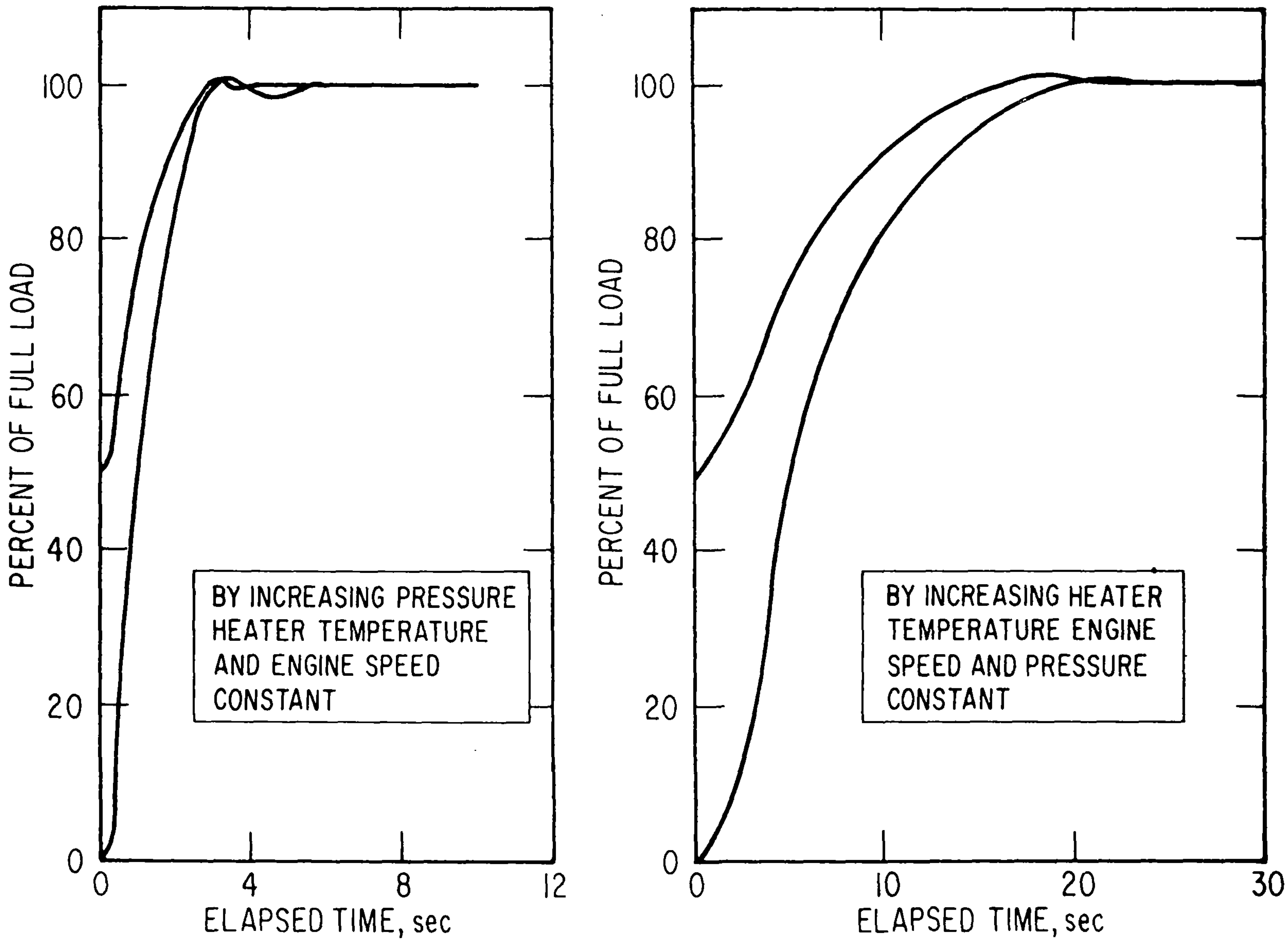


Figure 8-40. Estimate of Response Characteristics for Stirling Engine, Ref. 8-29

of the energy being rejected into the cooling water. In the closed-cycle Stirling engine, about 60 percent of the fuel's energy will be rejected into the cooling system provided an efficient burner system is used. The Stirling engine will therefore need a radiator area three to four times larger than those used in comparable present-day vehicles, and power for the cooling fan will be appreciably higher.

Modern development of the Stirling engine has practically been restricted to Philips in the Netherlands and to General Motors (licensed by Philips) in the United States. This has produced a limited amount of actual test information, and the number of specialists in Stirling engine technology is likewise limited. This lack of experience would increase the cost and time needed to bring the engine to vehicle readiness. Also, the licensing agreement could hinder any industry-wide program for development of the engine.

8.6.4 Engine Characteristics

8.6.4.1 Specific Fuel Consumption

A number of Stirling engines of both conceptual and prototype designs are listed in Table 8-6. Specific fuel consumption is plotted in Fig. 8-41. It should be noted that the data for conceptual engines indicate lower fuel consumption than that of the prototype engines. However, the difference is not large and may be achieved through growth and design refinements.

A curve showing anticipated part-load fuel consumption is given in Fig. 8-42. The curve is characteristic of regenerated engines with a slight improvement in efficiency at high part-loads with increased consumption at low part-loads as parasitic loads become prominent.

Table 8-6. Characteristics of Stirling Engines

	Engine	Model	hp	Volume, ft^3	ft^3/hp	Weight, lb	lb/hp	BSFC lb/bhp-hr	Reference
1	GM Research	GPU-2	7.5	3.04	0.40	90	12.0	0.595	1
2	GM Research	GPU-3	10	4.89	0.49	188	18.8	0.508	1
3	Philips	3015	40	6.4	0.16	550	13.8	0.351	1
4	GM Electromotive	8015	380	130	0.34	5000	13.2	0.457	1
5	Philips	Marine	120	23.20	0.19	725	6.0	0.343	1
6	GM Allison	PD-67	7	3.5	0.50	186	26.6	0.457	1
7	GM Research		8.63	-----	----	127	14.7	0.470	2
8	GM Research		40	-----	----	450	11.0	0.358	2
9	KB United Stirling		27	3.25	0.121	240	8.9	0.410	3
10	KB United Stirling		20	2.53	0.127	150	7.5	0.395	3
11	KB United Stirling		80	5.06	0.064	440	5.5	0.410	3
12	KB United Stirling		175	30.6	0.175	2200	12.6	0.365	3
13	KB United Stirling		200	30.6	0.153	2200	11.0	0.375	3
14	KB United Stirling		300	41.3	0.138	2900	9.7	0.375	3

References:

1. Battelle Memorial Institute, "Study of Unconventional Power Sources for Urban Vehicles," 15 March 1968.
2. Flynn, G., Jr., W. Percival, F.R. Heffner, "GMR Stirling Thermal Engine," SAE Trans. Vol. 68, 1960.
3. KB United Stirling, Brochure, 1970.

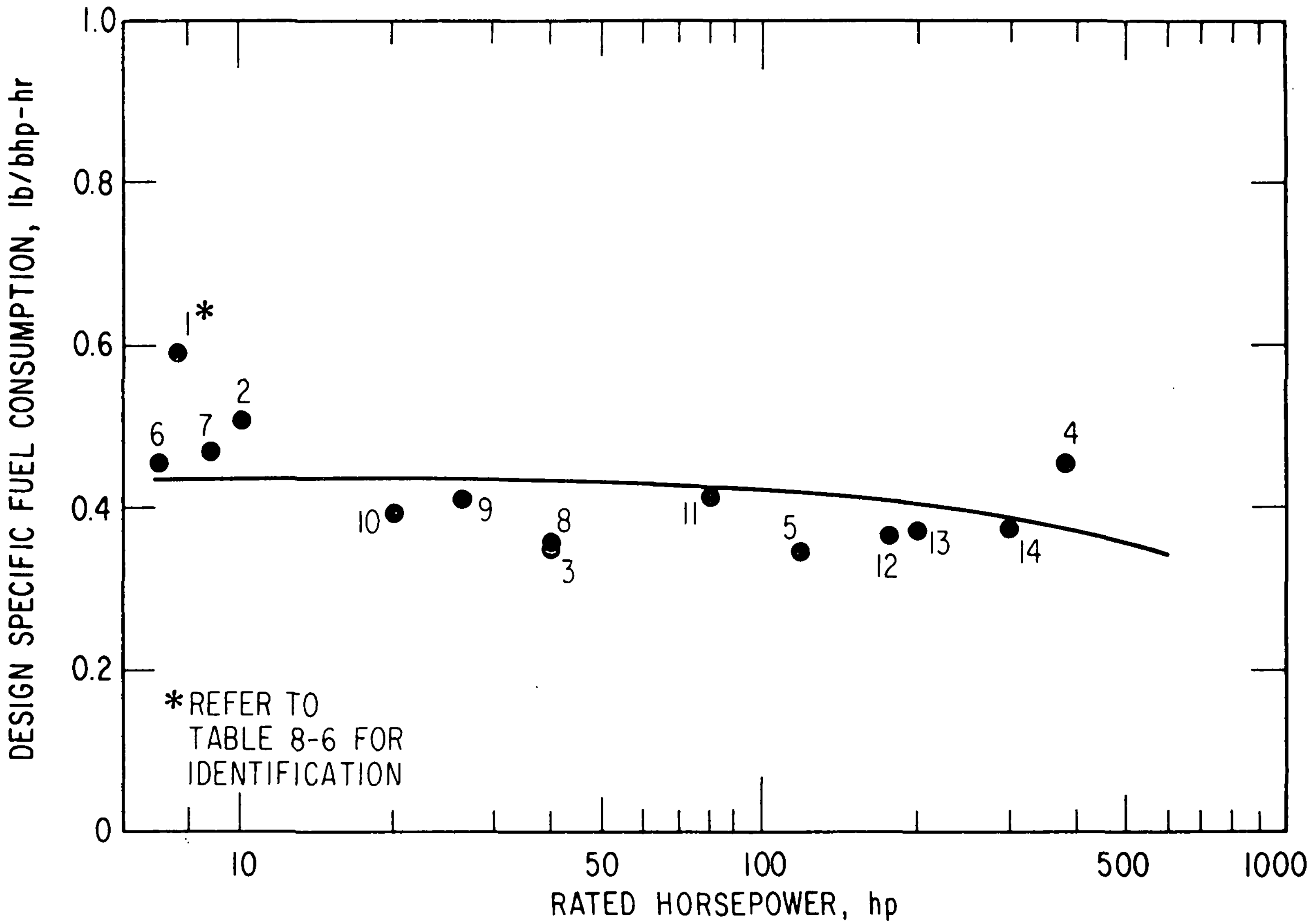


Figure 8-41. Design SFC of Automotive Stirling Engines

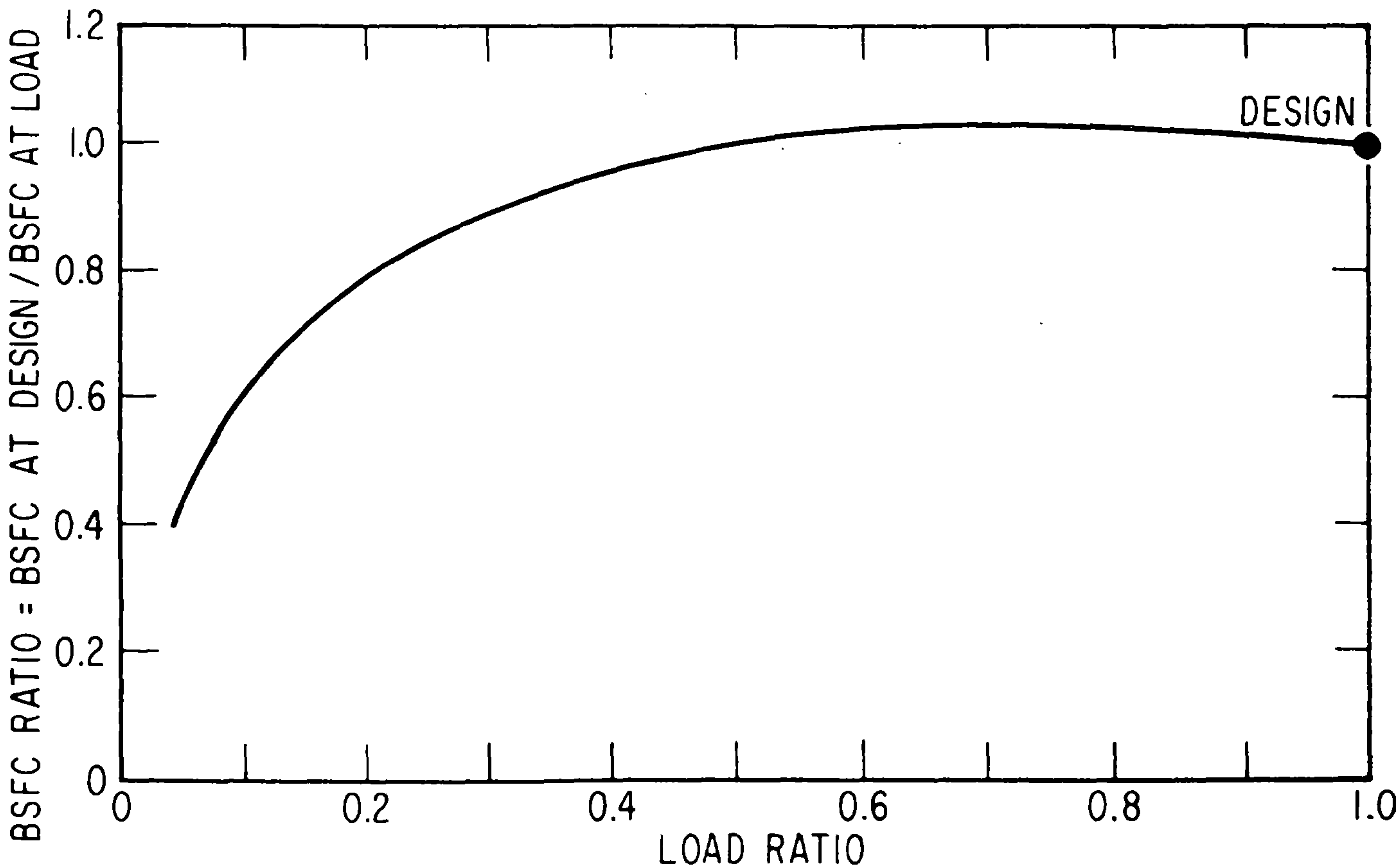


Figure 8-42. Part-Load BSFC Characteristics of Automotive Stirling Engines

8.6.4.2 Specific Weight

Specific weight characteristics are described in Fig. 8-43. From the raw data it is difficult to judge what the true potential of the Stirling engine might be. The list includes heavy laboratory-type engines and some optimistic conceptual designs, along with some marine systems which reduce the cooling system weight. However, it is quite probable that a detailed engineering design utilizing such advanced techniques as heat pipes and lightweight materials could reduce the weight of the Stirling system significantly. It does not appear realistic when considering the comparative thermal efficiencies that the Stirling engine should be larger and heavier than a Rankine engine.

8.6.4.3 Specific Volume

Specific volume characteristics for the Stirling engine are shown in Fig. 8-44.

8.7 COMPARISON AND EVALUATION OF HEAT ENGINES

Heat engine horsepower requirements for the various classes of hybrid vehicles are delineated in Section 10. The requirements for series and parallel operation are not significantly different. For simplicity in the discussion that follows, it will be assumed that the series configuration power requirement represents the engine size needed in each vehicle class. With this assumption, the heat engines for each of the hybrid vehicle classes may be described in single-valued terms of weight, volume, and specific fuel consumption as shown in Tables 8-7 through 8-12.

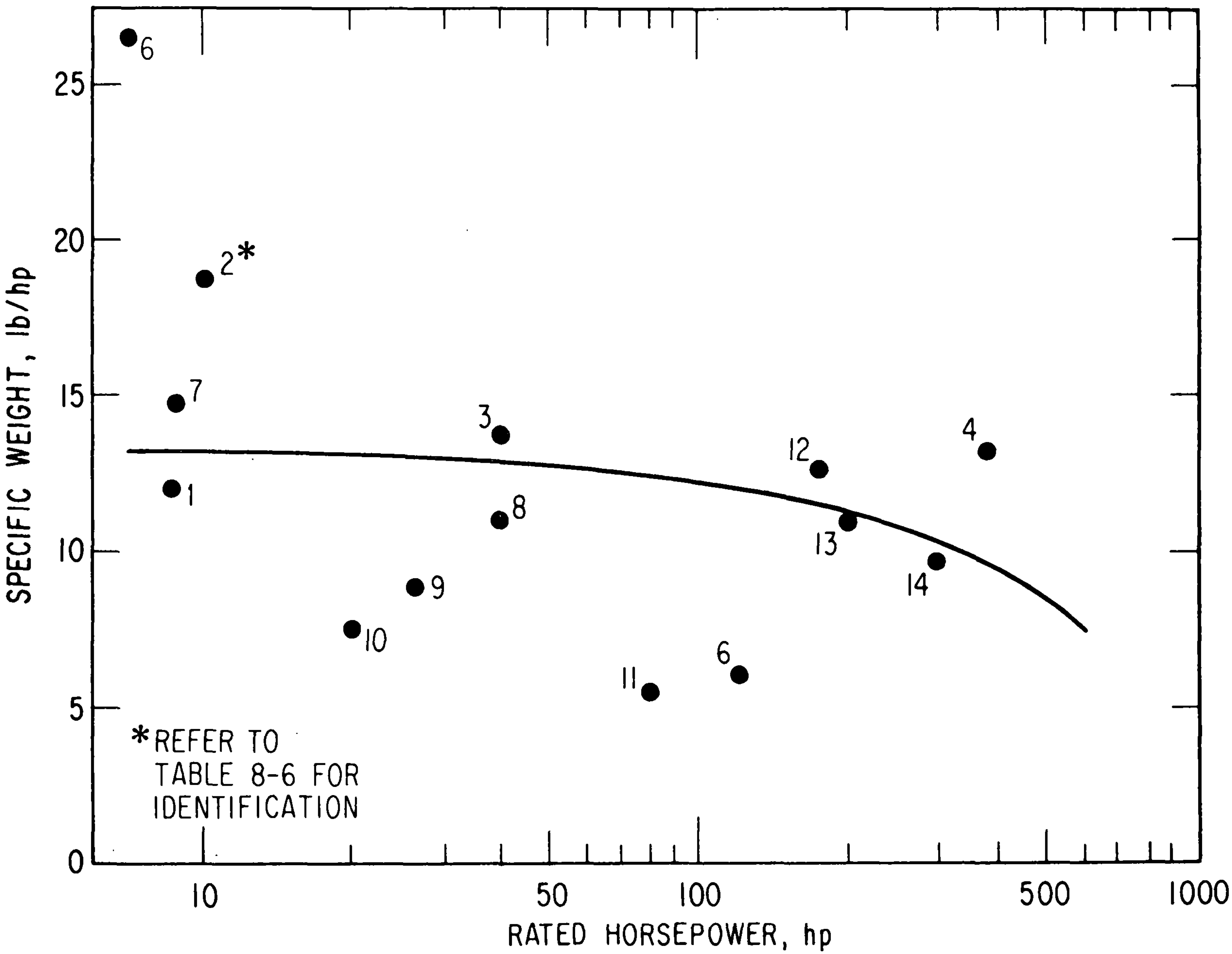


Figure 8-43. Specific Weight of Automotive Stirling Engines

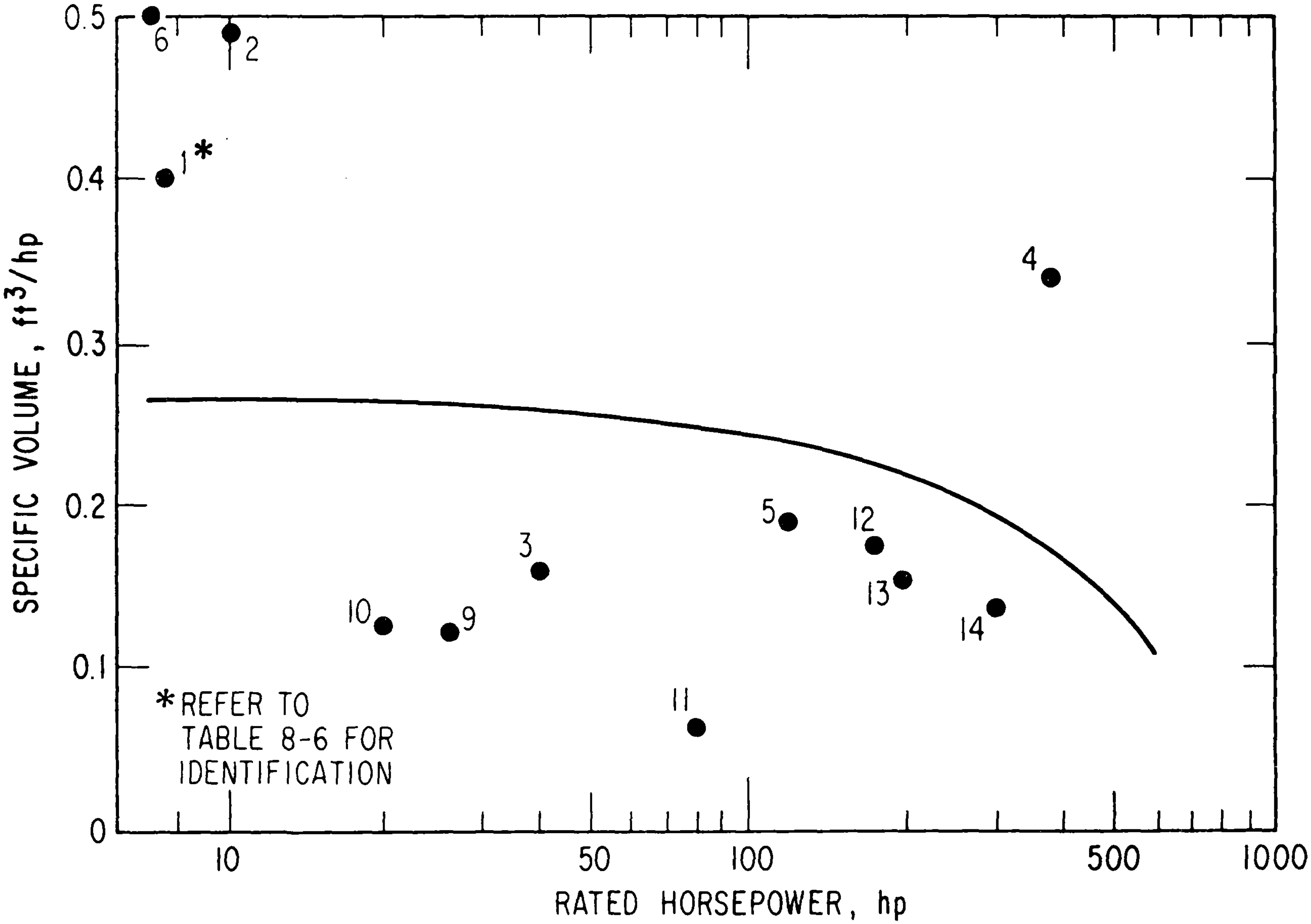


Figure 8-44. Specific Volume of Automotive Stirling Engines

Table 8-7. Family Car Heat Engine Characteristics

Heat Engine	SFC, lb/hp-hr	Weight, lb	Volume, ft^3
S. I. Engine - Reciprocating Piston	0.50	335	11.8
S. I. Engine - Rotary Piston	0.50	216	5.8
CI Engine	0.43	493	15.1
Brayton Cycle Engine	0.57	310	10.4
Rankine Cycle Engine	0.87	846	13.5
Stirling Cycle Engine	0.42	1153	22.8

Table 8-8. Commuter Car Heat Engine Characteristics

Heat Engine	SFC, lb/hp-hr	Weight, lb	Volume, ft^3
S. I. Engine - Reciprocating Piston	0.56	180	6.1
S. I. Engine - Rotary Piston	0.56	116	4.0
CI Engine	0.45	228	8.9
Brayton Cycle Engine	0.65	125	3.9
Rankine Cycle Engine	0.93	322	5.5
Stirling Cycle Engine	0.43	432	8.6

Table 8-9. Low-speed Van Heat Engine Characteristics

Heat Engine	SFC, lb/hp-hr	Weight, lb	Volume, ft^3
S. I. Engine - Reciprocating Piston	0.54	205	7.2
S. I. Engine - Rotary Piston	0.54	134	4.2
CI Engine	0.46	273	10.1
Brayton Cycle Engine	0.63	155	4.8
Rankine Cycle Engine	0.92	403	6.8
Stirling Cycle Engine	0.43	546	10.9

Table 8-10. High-speed Van Heat Engine Characteristics

Heat Engine	SFC, lb/hp-hr	Weight, lb	Volume, ft^3
S. I. Engine - Reciprocating Piston	0.50	357	12.8
S. I. Engine - Rotary Piston	0.50	236	6.1
CI Engine	0.43	545	16.1
Brayton Cycle Engine	0.56	350	11.8
Rankine Cycle Engine	0.86	963	15.3
Stirling Cycle Engine	0.42	1305	25.7

Table 8-11. Low-speed Bus Heat Engine Characteristics

Heat Engine	SFC, lb/hp-hr	Weight, lb	Volume, ft^3
S. I. Engine - Reciprocating Piston	0.48	478	17.3
S. I. Engine - Rotary Piston	0.48	311	7.2
CI Engine	0.42	755	20.3
Brayton Cycle Engine	0.52	521	15.0
Rankine Cycle Engine	0.83	1462	22.7
Stirling Cycle Engine	0.41	1949	38.1

Table 8-12. High-speed Bus Heat Engine Characteristics

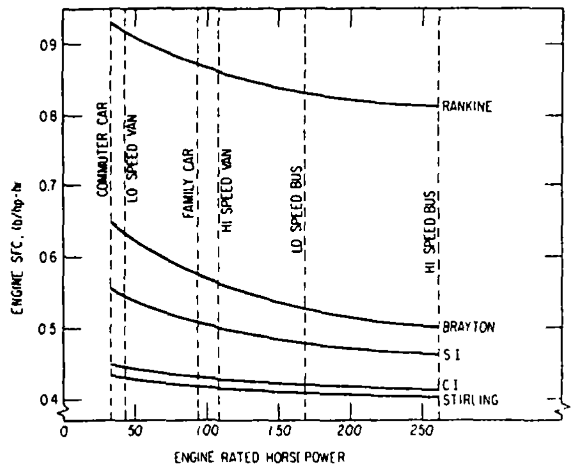
Heat Engine	SFC, lb/hp-hr	Weight, lb	Volume, ft^3
S. I. Engine - Reciprocating Piston	0.46	626	22.4
S. I. Engine - Rotary Piston	0.46	405	8.6
CI Engine	0.41	1050	25.3
Brayton Cycle Engine	0.50	744	19.8
Rankine Cycle Engine	0.81	2218	33.9
Stirling Cycle Engine	0.40	2793	53.0

The tabular data were developed from heat engine characteristics given in the preceding sections. The weights represent current technology for each engine type. Values shown for the rotary piston S. I. engine were estimated from the Curtiss Wright data, appropriately adjusted to reflect a consistent set of automotive accessories for all engines. Weights and SFCs for the CI engine are based on turbocharged, divided-chamber designs. The SFCs for spark and compression ignition engines represent the minimum point in the engine operating map; the SFCs for the Brayton, Rankine, and Stirling engines represent the design or full-load point in the engine operating map. Nevertheless, no great error is incurred by comparing these values on an equal basis, since the SFC characteristic for all (design-optimized) systems under consideration is relatively flat over a wide range of load.

A broader view of the tabulated results may be obtained by referring to the plots of Figs. 8-45, 8-46 and 8-47, where the data are grouped by heat engine characteristic and are plotted over the range of vehicle-class horsepowers. The SFC plot, Fig. 8-45, shows that the Stirling and compression-ignition engines provide the lowest fuel consumption for all hybrid vehicles. The spark-ignition engine ranks second on this basis, with SFCs ranging 25 to 12 percent higher for low to high horsepower applications.

The weight plot, Fig. 8-46, shows that the rotary-piston spark-ignition engine is the lightest of the heat engine candidates for all hybrid vehicles. The Brayton cycle is second best in this category for the commuter car and low speed van (i.e., the lower horsepower applications). The reciprocating-piston spark-ignition engine is (a) competitive with the Brayton cycle for the family car and high-speed van and (b) superior to the Brayton cycle for the two buses. The Stirling and Rankine cycles run significantly heavier than other heat engine types and, in view of the criticality of weight in relation to battery power density requirements, these systems appear not to be useful in their present form.

Figure 8-45.
Heat Engine
SFC Comparison



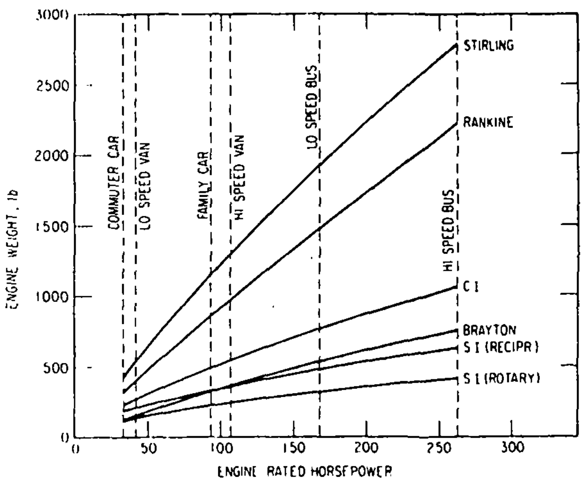
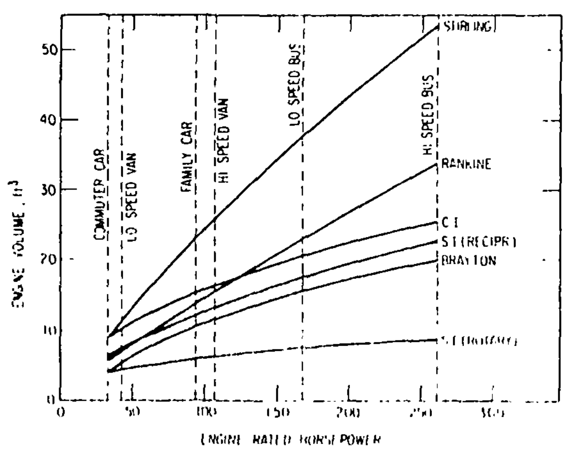


Figure 8-46.
Heat Engine
Weight Comparison

Figure 8-47.
Heat Engine
Volume Comparison



In addition, the latter two engine types make a generally poor showing in volume, as indicated by Fig. 8-47 (The volume improvement in the Rankine system at low rated horsepower levels is noted.). Volume, it may be observed, has a secondary impact on engine-assignable weight, since a bigger structure is required to support and enclose a larger engine envelope. The rotary-piston spark-ignition engine is best in the volume category for all vehicles, while the Brayton cycle is second best for all vehicles.

The practice of using numerical weighting factors to evaluate the relative merits of alternative systems frequently produces specious results which depend heavily on the influence assigned to the individual criteria of evaluation. For this reason, and because exhaust emissions are a primary concern of this study, a ranking of the alternative heat engine systems based on SFC, weight, and volume considerations will not be attempted. It is apparent, however, that (a) the rotary-piston engine is a prime candidate for further examination and evaluation in relation to the full spectrum of hybrid vehicle applications, (b) the Brayton cycle constitutes a promising alternative for applications involving low horsepower requirements, and (c) the reciprocating spark-ignition engine must be included for consideration as an alternative to the rotary-piston engine for applications involving intermediate to high horsepower requirements.

8.8 TECHNOLOGY GOALS

8.8.1 Spark-Ignition Engines

The impact of engine weight on battery power density requirements is discussed in Section 11. In general, a reduction of engine weight is desirable from the standpoint of relieving design constraints on other propulsion system components; in particular, a reduction of engine weight is an absolute necessity for the application of certain heat engines to specific hybrid vehicle classes.

Weight has never had a strong influence on automotive engine design practice except as it indirectly impacts production cost through raw material purchase expense. In this sense, the present specific weight status of automotive spark-ignition engines is largely artificial and can be significantly improved if requirements demand. There are a number of general approaches which can be taken to bring this about. One of these, not related to weight per se, is to increase engine power output by increasing engine speed, by increasing compression ratio, or by supercharging. The first technique results in reduced mechanical and volumetric efficiency with concomitant loss in engine fuel economy. Neither of the latter two techniques is recommended since they both raise combustion temperature and pressure, require higher octane fuels to prevent detonation, and may therefore add to, rather than improve, engine exhaust emissions.

Another approach is to reduce engine weight through the use of materials having maximum strength-to-weight ratios such as high tensile alloy steels and aluminum and magnesium alloys. This route is assuredly expensive, yet it is doubtful that significant advances can be made in this area without economic repercussions of some magnitude. The Vega aluminum block engine is an elementary example of what might be accomplished by this approach (the 105-hp Vega engine specific weight is 2.6 lb/hp, or 24 percent lower than the mean characteristic for cast iron block engines).

Design modifications in the direction of large bores and minimum number of cylinders offer the best hope for substantial gains and should be pursued. High-displacement, short-stroke, air-cooled designs with opposed cylinder arrangements (such as used by Volkswagen and Porsche) to effectively treat the reciprocating-mass balance problem appear attractive from the standpoint of both weight and volume.

New principles of operation, as embodied by the rotary-piston engine, clearly provide breakthroughs to uniquely low levels of weight and volume. These systems are attractive, provided that they do not introduce more problems than they solve.

The major emphasis in rotary-engine systems has been placed on the Wankel engine, which has been under development for barely over 10 years and is under limited production by NSU in Germany, Toyo Kogyo (Mazda) in Japan, and Citroen in France. In addition to its weight and volume features, the Wankel engine provides smoother operation (no reciprocating parts to contribute to unbalanced inertia forces), has fewer components, and (probably) can be mass-produced cheaper than the conventional S. I. engine. *Attendant with these advantages are a number of problems and disadvantages* which need resolution before the system can be regarded as being truly on a competitive developmental par with the reciprocating type. These problems include high HC and CO emission levels, difficulties in rotor lubrication, and poor durability of the rotor apex seals.

The emissions problem stems from wall quenching effects associated with the high surface-to-volume ratio of the combustion chamber, combined with the action of the trailing apex seal which scrapes off the quench layer into the exhaust port (Ref. 8-30). The use of a thermal reactor with air injection has enabled NSU and Mazda to meet current U.S. emission standards. However, it appears likely that drastic changes in engine design might be required in order to meet U.S. standards proposed for 1975.

Citroen has approached the lubrication problem by adding about one percent oil to the incoming gasoline. This has further complicated the problem of emission control, requiring the addition of an afterburner. The efforts of NSU to reduce seal wear has led to the development of a secondary combustion chamber in which tip seal blow-by is contained. The chamber retards further passage of combustion gases, preventing the intermingling of oil with blown-through volatile combustion residue. Seal life is thus claimed to be extended and the previously recommended 12,000-mi engine oil change has been eliminated (Ref. 8-31).

Other rotary-engine systems are under development and should be investigated as possible Wankel alternates for application to the hybrid vehicle family. One of these, the Walker engine (New Zealand) utilizes an elliptical rotor. A two-rotor version, now possibly in production, develops 60 to 100 hp and has only seven moving parts (Ref. 8-32). Production costs have been estimated at about $200. Unlike the Wankel, the seals on the Walker engine are mounted on the engine casing rather than on the rotor, permitting simpler adjustment or replacement (Ref. 8-33). The 90-hp, 135-lb English "Tri-Dyne" (Ref. 8-34) might also be mentioned, as well as the U.S. Tschudi, a torroidal rotary engine rated 88-hp at 1600 rpm. The latter engine employs conventional compression rings on the torroidal pistons, thereby avoiding the seal problems associated with the Wankel design (Refs. 8-31 and 8-34).

Aside from the general desirability of seeking means to reduce engine weight and volume (properties which appear not to be critical for spark-ignition engine application to the hybrid vehicle), every effort should be made to investigate, encourage, and/or support the development of techniques which may lead to the attainment of satisfactory engine operation at high air/fuel ratios. As shown in Section 11, an estimated air/fuel ratio of about 22 would permit the hybrid commuter car and family vehicle, equipped with appropriate controls, to meet or approach the emission goals set for 1975/1976. Current technology limits lean operation for conventional vehicles to an air/fuel ratio of about 17 and for hybrid vehicles to an estimated air/fuel ratio of about 19.

The general challenge in achieving extremely lean operation is to maintain normal vehicle driveability functions. Lack of throttle response, stalls, and unsteady vehicle forward progress are always encountered at the extreme limits indicated above, primarily as a result of deterioration in combustion. The driveability problem is more severe in conventional vehicles because the heat engine is the sole source of power needed for rapid response to acceleration demands; the problem is considered to be significantly more tractable in the hybrid vehicle since the heat engine is required only to provide steady or quasi-steady power output. High idle speeds in some of the proposed configurations may also serve to minimize the hybrid driveability problem.

The means to achieve satisfactory lean operation may lie in improvements or innovations in the engine induction and/or combustion systems. As a first step, improvements which lead to more uniform distribution of the air/fuel ratio among individual cylinders should be pursued in order to prevent local fuel starvation with a mixture which would otherwise be satisfactory if homogeneously dispersed. Changes in inlet manifold ducting or carburetor design would thereby be indicated. The use of liquid fuels which are inducted as a gas; Propane, for example, should be considered (Propane also provides the additional advantage of reduced exhaust HC reactivity).

The distribution problem might also be solved by the use of fuel injection systems, either port-type such as recently adopted by foreign engine manufacturers (e.g., Volkswagen, Opel) or, preferably, direct-cylinder-injection type. In this connection, the stratified charge concept, as exemplified by the Ford Combustion Process, might be mentioned as an injection technique which simultaneously treats the distribution problem and additionally promotes and improves combustion at lean mixtures by providing a localized-rich charge mixture in the vicinity of the spark electrodes (Ref. 8-35). Precombustion chamber designs which implement the same process by isolating the rich mixture zone in an external pre-chamber should also be investigated. In each innovative, lean-operating design examined, consideration must be given to the possible degradation of maximum power output for current engine designs. Satisfactory methods of treating this problem must be found in order to limit the growth in engine size required to offset the power loss. It is anticipated that in the future such methods can be achieved (viz., stratified charge engines) and, hence, the power loss effects were not included in the analysis.

8.8.2 Compression-Ignition Engines

Unlike the spark-ignition engine, the weight of the diesel engine is critical and, for the baseline propulsion system allocation, the battery requirements are excessive for certain hybrid vehicle applications such as the commuter and family cars.

The remarks made under spark-ignition engines with regard to engine design practice apply also to diesel practice but with considerably more emphasis. While it is unrealistic from the standpoint of the fundamental compression-ignition process to expect that the automotive diesel engine could rapidly overtake the spark-ignition engine weight advantage, it is nevertheless evident that the industry has in the past been guided by application requirements dictating durability and long life, and that significant improvements in weight and volume are achievable. As an example of what might be accomplished, the specific weight for a conceptual design intended for light aircraft application may be cited: 1.8 lb/hp at 180 hp (Ref. 8-36). This compares with 4.5 lb/hp indicated by the state-of-the-art characteristic for divided-chamber turbocharged engines shown in Section 8.3.3.

Diesel HC and CO emission characteristics look relatively good compared with S.I. engines, while NO_2 emissions, which are generally higher than S.I. engines, might be effectively treated by recirculation. The NO_2 problem is compounded, however, by the necessity of resorting to turbocharging in order to achieve reasonable values for engine specific weight. Therefore, the possibilities for weight improvement in naturally-aspirated designs should be pursued. New swirl and prechamber configurations which act to reduce peak cylinder pressures, thereby minimizing static and dynamic loads on the engine system, might be investigated for this purpose.

In further connection with innovative designs, the weight and volume properties of rotary-piston diesel-process machines look attractive and these should be examined in light of their potential for hybrid vehicle use. For example, the Rolls-Royce model 2-R6, a two-stage, two-bank diesel Wankel which is now being built for testing in early 1971, develops 350 hp at a weight of 929 lb and a volume of 19.3 ft^3 (2.7 lb/hp, 0.055 ft^3/hp). Diesel Wankels are currently being developed for Great Britain's Military Vehicle Engineering Establishment (Ref. 8-37).

8.8.3 Gas-Turbine Engines

As noted previously, the gas turbine engine for the hybrid vehicle can be simpler than an engine designed for prime power. Ordinarily, a regenerated gas turbine is sluggish in response to load or speed change due to the thermal inertia of the regenerator and thus additional complexity is needed to improve the engine's response capability. As a result, it has been estimated that in high production the automotive gas-turbine engine will cost more than a spark-ignition engine. This additional cost is estimated at anywhere from 10 to 50 percent more. However, the hybrid gas turbine will operate at a nearly constant speed for long periods and there is no need to provide for rapid response to speed or load change. Hence, added cost factors may be minimized.

The areas where effort is needed in the development of the hybrid gas turbine are described in the following paragraphs.

In order to improve response of the gas turbine, it has been necessary to decrease the inertia of the rotating assembly which necessitates increasing the specific output. Higher pressure ratios, reheat, intercooling, and other techniques are used to provide the high specific output. The hybrid engine does not need a high specific output. As a matter of fact, just the opposite might be true, for if a direct generator drive is used, lower specific output will provide greater compatibility between the gas turbine speed and the desired speed range for a generator.

In the previous section where gas turbine design arrangements were investigated, a moderate pressure ratio was assumed. This pressure ratio was selected because it is more compatible with that of gas turbines currently in production.

A turbocharger is a gas turbine which uses the diesel engine it serves as a source of energy to power the turbine. A turbocharger equivalent in air flow to a 40-hp gas turbine will cost under $50 [Original Equipment Manufacturer (OEM) Price] and run for 4000 hr under off-highway conditions with full warranty. To make a hybrid gas turbine from a turbocharger it is necessary to add all of the auxiliaries. Although it appears that the rotating assembly will not be too expensive, the regenerator, controls, and combustor will be the main sources of cost.

Therefore, a major study of the hybrid gas turbine should be undertaken. This study should concentrate on cost aspects and determine what the true high production costs of the gas turbine will be. An important output of this study will be the weight, volume, and configuration specifications for the hybrid gas turbine.

The hybrid engine could have as few as two operating points, full speed (design output) and idle speed (no output condition). This latter condition will be especially important to the commuter car, and the low-speed van and bus. Some additional thought should be given to the reduction of fuel consumption at idle. As mentioned earlier, the reduction of speed and the use of variable turbine nozzles would decrease fuel consumption at idle.

While the gas turbine has low emission characteristics, it should be possible to make some further significant reductions. The main considerations used in the design of present combustor systems have been low volume and high combustion efficiency with little or no thought given to emissions. It would appear that vaporizing injectors and recirculation could be integrated into the combustor to reduce emissions. Development studies of gas turbines should be performed to reduce emissions.

8.8.4 Rankine Engines

A wide number of concerns have examined the Rankine engine for application as a prime powerplant for vehicles. Considerable support to Rankine cycle development has been and is being provided by the Atomic Energy Commission and by the U.S. Army. Rankine cycle engines, by virtue of their external burner system, have better emission characteristics than any of the internal combustion engines and for this reason their use has been advocated in spite of their rather poor weight, volume, and specific fuel consumption characteristics. Unfortunately, the basic problem relative to any significant improvement in these items is the second law of thermodynamics. Due to the characteristics of working fluids there does not appear to be any room for revolutionary improvements to the present systems. However, because of the low emission characteristics, it might be advisable (although low in priority for hybrid application) to continue some work on Rankine engines with a view to refining the design of the components.

8.8.5 Stirling Engines

Available data on the Stirling engine indicate that it has, or should have, very good specific fuel consumption and low emission characteristics. The principal problems of concern are weight, volume, and life. Since none of the existing engines appears suited to hybrid vehicle application, a design study and analysis of the Stirling engine powerplant would have to be conducted with the objective of decreasing its weight and volume before it could be considered for use in the hybrid engine. With acceleration modes removed for hybrid operation, potential performance improvements should be examined. The characteristics of all auxiliary equipment such as radiator, fans, pumps, and burners should also be investigated. Consideration should be given to variations (e.g., the use of heat pipes) in the cycle which would permit consolidation of some of the engine processes so that the coolers and burners might be shared by the individual cylinders. The problem of increasing the life of seals and bearings as well as improving accessibility and maintenance should also be studied.

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SECTION 9

HEAT ENGINE EXHAUST EMISSIONS

CONTENTS

CONTENTS (Continued)

FIGURES

SECTION 9

HEAT ENGINE EXHAUST EMISSIONS

9.1 GENERAL

9.1.1 Introduction

Five heat engine concepts were investigated for the hybrid heat engine/electric vehicle systems considered in this study. They are the following:

Spark Ignition Engine (Otto Cycle)
Compression Ignition Engine (Diesel Cycle)
Gas Turbine Engine (Brayton Cycle)
Rankine Engine ("Steam")
Stirling Engine

This section discusses the heat engine exhaust emission characteristics utilized in the calculation of vehicle exhaust emissions. These characteristics were derived as a result of evaluating all available information in the open literature as well as much unpublished data obtained from various engine manufacturers. Discussion of the data and the various options of engine operation are contained in Appendix B. This section is devoted primarily to describing the engine exhaust emission characteristics that were selected as representative of state-of-the-art technology and projected technology, and were used in the vehicle emission calculations.

The major pollutants emitted from heat engines are hydrocarbons (HC), carbon monoxide (CO), oxides of nitrogen (NO_x), oxides of sulfur, aldehydes, and particulates. This study was limited to the HC, CO, and NO_x emissions since these are the pollutants of primary concern with respect to vehicle emission standards. Because of the lack of quantitative information, other pollutants will be discussed qualitatively in connection with the emissions of each type of engine.

Engine exhaust specific mass emissions for NO_x are reported on a nitric oxide (NO) basis. Nitric Oxide was used as a matter of convenience since the NO_x in the engine exhaust gases is predominantly NO (on the order of 95 percent or more). However, since Federal regulatory requirements will stipulate the vehicle NO_x emissions to be reported as NO_2, the vehicle emission data presented in this study have been calculated on the basis of NO_2. The factor for converting NO mass emission to NO_2 is 1.533, which is the ratio of the molecular weights.

9.1.2 Exhaust Emission Data Format

Inasmuch as the vehicle exhaust emissions are to be expressed on a mass basis in terms of grams per mile, it was found most convenient, especially for the hybrid mode of heat engine operation, to express emissions in terms of specific mass emissions having the units of grams per bhp-hr. Mass emission correlations were established for the five heat engines and curves were generated describing the exhaust emission characteristics of heat engines, at both design load (or full load) and part-load conditions. Thus, for each pollutant there will be two basic curves:

1. Design load specific mass emissions (grams/bhp-hr) as a function of engine design horsepower

2. Part-load emission factors (ratio of part-load specific mass emission to design load specific mass emission at rated speed) as a function of percent design load

The variation of the emissions with part-load conditions can be very critical in determining the exhaust emission characteristics of a hybrid vehicle. In conducting the data correlation, it was discovered that a severe shortage of steady-state mass emission data, particularly at part load, existed in the open literature. Considerable information is available on concentrations of pollutants, but usually without the information necessary for conversion to mass numbers. Although the curves presented here have been established from the best data available today, considerable effort is warranted to develop a more comprehensive and reliable data base for emissions. In particular, part-load emission data are inadquate for several types of engines.

9.1.3 Cold Start Emissions

For light-duty vehicles (under 6000-lb gross weight), the Federal test procedures specify that the vehicle be kept at ambient temperature for 12 hr prior to the test. The HC and CO emissions are generally higher when the engine is cold; thus, it is necessary to account for the emissions during this cold start period, which can be considerable, depending on engine operating conditions. In an engine equipped with a catalytic converter, there is an additional degradation of emission during the engine and catalyst warmup period.

The engine exhaust emission characteristics presented in this section are based on steady-state, hot engine data and the vehicle emission levels computed for the various options and configurations presented in this study are therefore hot start emission levels. The effects of engine cold starts on emissions are not included in the recommended correlations at this time because of the following factors:

1. Cold start data are not available for all the heat engines.
2. Inclusion of cold start effects are required at present only for the lightweight vehicles tested over the DHEW cycle (passenger car and commuter car).

To incorporate cold start effects, a cold start emission factor (ratio of cold start cycle emission to hot start cycle emission) can be applied to the vehicle emission levels computed from the hot engine data. Some cold start emission characteristics are available for conventionally powered automobiles. Additional data were generated during the period of this study that pertain to spark ignition engine cold starts in the hybrid mode of operation as well as in diesel engines. These cold start factors are summarized in Section 9.2.3.

For low-pollution engines, the effect of cold start can be very critical, since the emissions generated in the first minutes of warmup can overshadow the emissions generated during the rest of the driving cycle when the engine is hot. Much work remains to be done in this area on evaluation of data and investigation of techniques to minimize cold start effects.

9.2 SPARK IGNITION ENGINE EMISSIONS

9.2.1 Design Load Emissions

9.2.1.1 State-of-the-Art Technology

The design load specific mass emissions are presented in Figure 9-1. Selected air/fuel ratios are presented in this curve to represent regimes of operation with spark ignition engines. Other cases were covered and are described in Appendix B. The data for operation at air/fuel ratios between 15-16 were based on evaluation of characteristics from two basic engines and were used to represent emission levels from current engines operating at that particular air/fuel ratio. Since lean operation (high air/fuel ratio) appears to be attractive from the standpoint of minimizing CO and NO emissions, an air/fuel ratio of approximately 19 was selected to represent the present state-of-the-art technology. It would be difficult, of course, to operate such an engine in a normal automotive vehicle; however, the "driveability" problems normally associated with lean engine operation should be minimized in the hybrid because the engine can be designed for essentially steady-state operation over a restricted range of operating conditions. The assumed increase in HC specific mass emissions at an air/fuel ratio of 19 compared to an air/fuel ratio of 15 to 16 is due to the combined effects of power loss and increasing quench effects. The specific mass emissions are assumed to be constant for engine design power levels above approximately 50 hp. Below that point, the emissions were assumed to increase, reflecting the trend of decreasing engine efficiency with decreasing size.

9.2.1.2 Projected Technology

Many approaches are possible toward decreasing engine emissions. These are discussed in Appendix B and include variation of spark timing, chamber design, mixture preparation, manifold pressure, exhaust gas recirculation, water injection, and catalytic converters. For the projected technology spark ignition engine, an ultra-lean air/fuel ratio of approximately 22 was

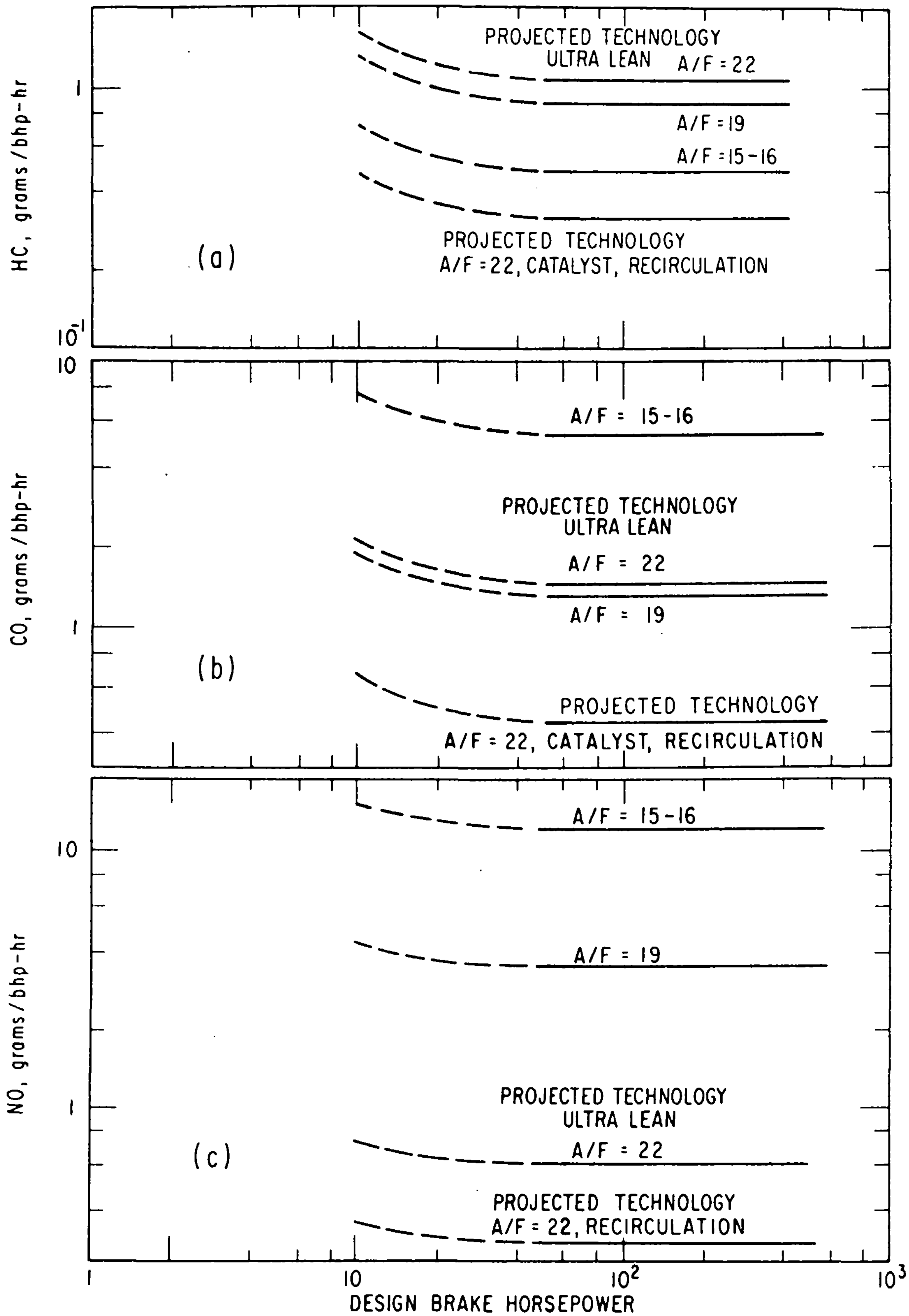

Figure 9-1. Spark Ignition Engine (Gasoline) Emissions, Steady State Design Load: (a) Hydrocarbon, (b) Carbon Monoxide, (c) Nitric Oxide

selected to indicate the potential offered by the spark ignition engine. The data base for such an operating point are the single-cylinder data obtained informally from the Bureau of Mines and the dual-chamber engine work of Newhall at the University of Wisconsin. Other lean-engine approaches which could potentially achieve the same results are the stratified charge combustion chamber, and utilization of improved carburetor/intake manifold configurations, and pre-heated or pre-mixed air/fuel charges. Further studies are required to determine the optimum lean air/fuel ratio by considering emissions as well as engine performance aspects. The lower projected technology curve is based on an air/fuel ratio of 22 and utilization of a catalytic converter for HC and CO reduction, and exhaust gas recirculation for further NO control. A catalyst conversion efficiency of 70 percent and an exhaust gas recirculation effectiveness of 50 percent were used to construct the projected technology curve.

9.2.2 Part-Load Emissions

The part-load emission characteristics of spark ignition engines operating at air/fuel ratios of 15-16 are presented in Fig. 9-2 in terms of the ratio of specific mass emissions at part-load to specific mass emissions at design load versus percent design load. These curves were derived for constant engine speed from the very limited data sample provided by Toyota Motor Company and another manufacturer. The two engines showed somewhat different part-load emission characteristics, and the curves presented in Fig. 9-2 are the average from the two engines. Part-load emission data for CO and NO recently received from General Motors and TRW Systems indicate similar trends.

It is realized that engine exhaust emissions are also a function of engine speed. As a result, lower part-load emission factors might be obtained by varying speed as load is varied. The optimum speed versus load schedule must be determined for each hybrid system application by considering heat engine emissions as well as the performance characteristics of other components such as the generator, motor, controls, etc. Presently, there

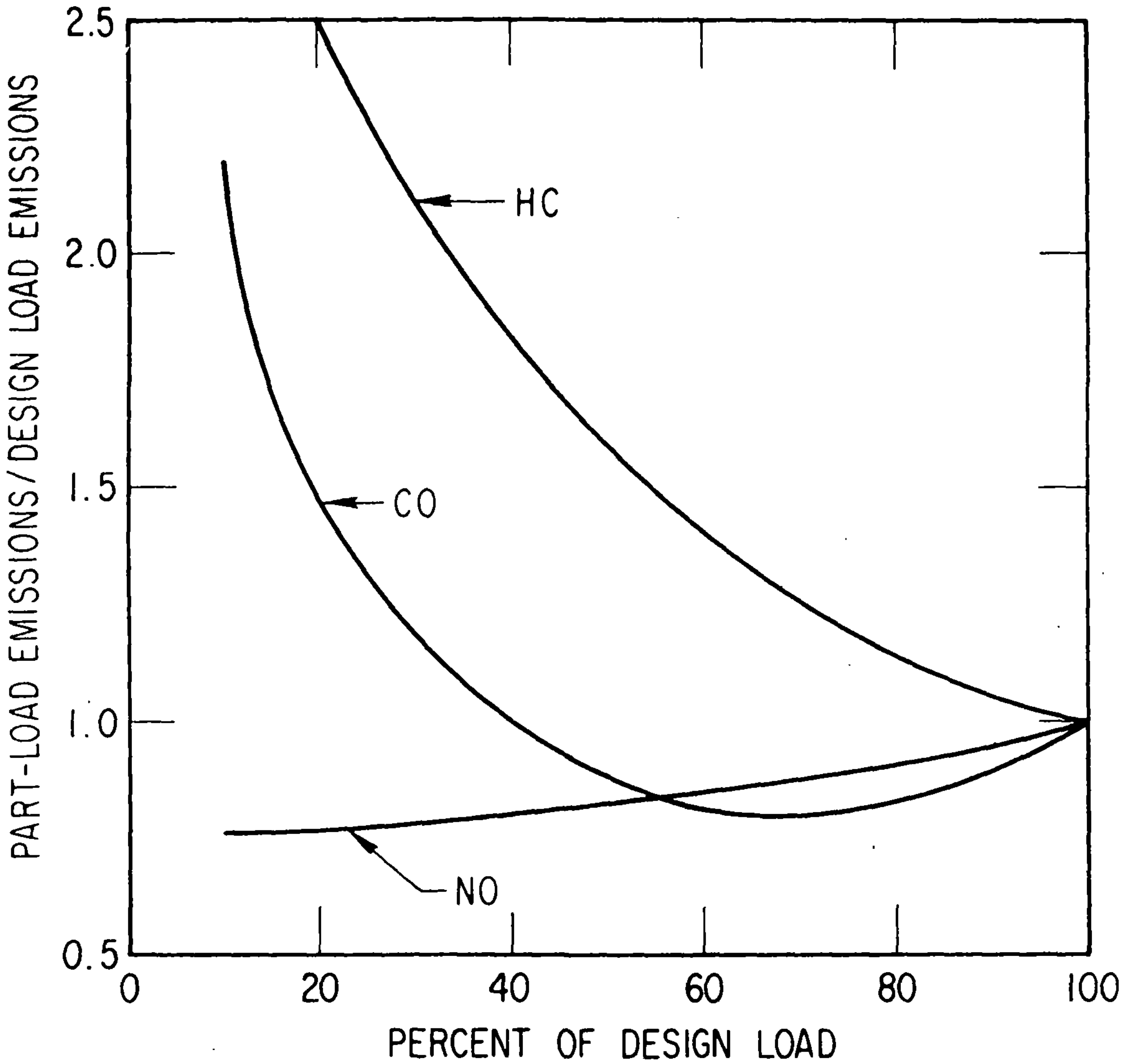

Figure 9-2. Spark Ignition Engines - HC, CO, NO, Emissions, Steady State Part-Load, Air/Fuel = 15-16

is a serious lack of applicable engine data, and thus speed could not be used as an emission correlation parameter. More work is required in this area before these questions can be adequately answered.

For the purpose of this study, HC, CO, and NO specific emissions were assumed to be constant for all load conditions for the lean air/fuel ratios of 19 and 22. This choice was made primarily because of the lack of applicable test data. Also, it appears that change in spark timing and engine design modifications, together with the addition of catalytic converters and exhaust gas recirculation, can result in considerably different part-load emission characteristics, and these changes cannot be anticipated at this time. It is most important that additional work be conducted to acquire data to resolve these questions.

9.2.3 Cold Start Emissions

Some cold start emission data for conventionally powered automobiles are available, and during the course of this study, additional data were made available by TRW Systems which are applicable to state-of-the-art lean engines operating in the hybrid mode. Cold start vehicle emission data were also obtained from General Motors. These are discussed in Appendix B. Based on these data, the following numbers were selected to be indicative of the cold start correction factors to be used for the spark ignition engines considered in this study.

	Cold Start Correction Factor	
Pollutant	State-of-the-Art Technology	Projected Technology
HC	1.30	1.20
CO	1.30	1.20
NO	0.95	0.95

The above factors apply only to the vehicle emissions computed over the DHEW driving cycle.

9.2.4 Other Pollutants

No quantitative data on other pollutants are available for spark ignition engines. Sulfur and lead oxides can be controlled by limiting the sulfur and lead content in the fuel. Smoke is generally not a problem in spark ignition engines except in poorly maintained engines.

9.3 DIESEL ENGINE EMISSIONS

9.3.1 Design Load Emissions

9.3.1.1 State-of-the-Art Technology

The design load specific mass emissions are presented in Figure 9-3. State-of-the-art technology emission characteristics are shown for three types of four-cycle diesel engines: (1) naturally aspirated, direct injection, (2) turbocharged, direct injection, and (3) turbocharged prechamber. These curves were based on emission test data obtained from the literature as well as directly from the manufacturer. Insufficient data were available to classify the two-cycle diesels. As shown, constant specific mass emissions were assumed for each engine type for design power levels above 50 HP. Below that point, the emissions are assumed to increase, primarily to reflect the lower engine efficiency and the higher wall quenching effects, resulting from less favorable cylinder surface area-to-volume ratios. The NO increases at a lower rate than HC and CO.

As indicated in Fig. 9-3a, the HC specific mass emissions of the four-cycle turbocharged, direct injection engine are the highest of the three engine types, although theoretically the HC emissions of turbocharged diesels should be lower than those of naturally aspirated engines. Additional emission data are needed for turbocharged, direct injection diesels to clarify this issue. The four-cycle turbocharged, prechamber diesel indicates the lowest HC emission level.

The design load emission characteristics of the turbocharged, direct injection and prechamber diesels were derived from a very limited data sample, and as a result the effects of manufacturing tolerances may not be adequately accounted for. This should be considered when using these curves.

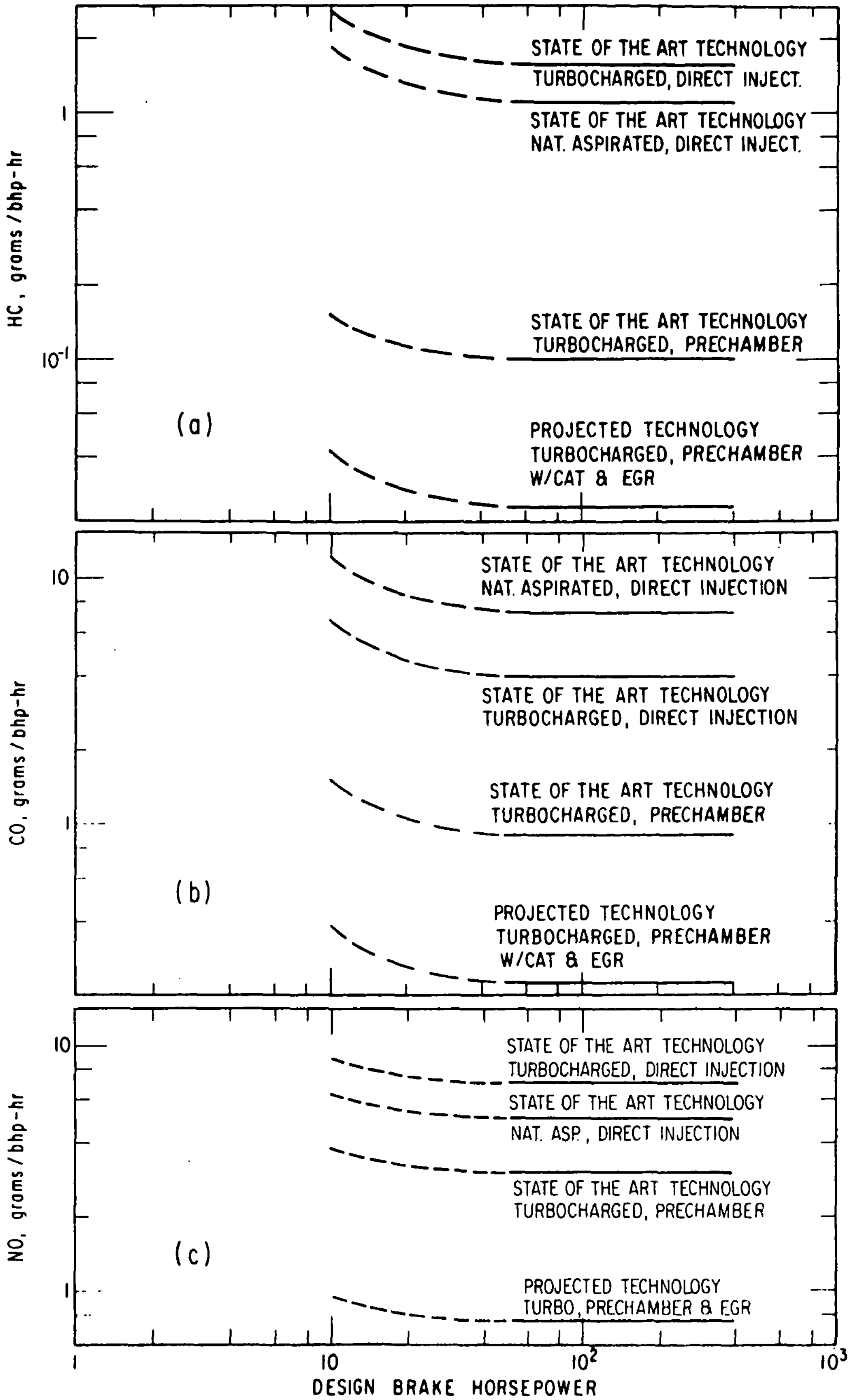

Figure 9-3. Four Cycle Diesel Engine Emissions, Steady State Design Load: (a) Hydrocarbon, (b) Carbon Monoxide, (c) Nitric Oxide

The CO specific mass emission data from various engines used to construct the curves of Fig. 9-3b showed considerably smaller variation than that shown by the HC data. This trend was expected, since CO concentration is determined primarily by air/fuel ratio and the design point air/fuel ratios of the diesels are quite comparable. In accordance with expectations, the highest CO emissions are obtained with the naturally aspirated, direct injection diesels and the lowest emissions with the turbocharged prechamber engines.

Nitric oxide represents the major emission problem in diesel engines. As indicated in Fig. 9-3c, the turbocharged, direct injection diesel has the highest NO emissions, and these values are comparable to the NO emitted from present spark ignition engines. The naturally aspirated, direct injection diesels show somewhat lower NO emissions. The NO emissions of turbocharged prechamber engines are even lower. As will be shown in Section 11, the NO emission levels of present diesels have to be reduced significantly before future emission goals can be met.

Because of its low specific mass emissions, the turbocharged, prechamber diesel curves were used to represent state-of-the-art technology diesel engines for the vehicle emission comparisons. Notice, however, that the specific fuel consumption of prechamber diesel engines is slightly higher than that of direct injection engines (see Section 8).

9.3.1.2 Projected Technology

The projected technology design load emission curves shown in Figure 9-3 are based on improvements to the turbocharged, prechamber diesel engine. An arbitrary reduction by a factor of 4 was applied to the state-of-the-art technology emission levels for all pollutants (HC, CO, and NO).

The improvements in HC and CO reduction are considered reasonable goals, achievable with modified injection systems, combustion chambers, injection timing, and catalytic converters. The effects of catalytic converters on diesel engine emissions have been investigated by Springer at Southwest Research Institute and more recently by Aerospace (Appendix C) and some reduction in HC and CO emission was achieved. In addition to reducing HC and CO, the odor level of the diesel exhaust was reduced. Further tests are needed to evaluate the catalyst performance as affected by operating time.

The projected NO emissions shown in Fig. 9-3c reflect the effects of exhaust gas recirculation, as well as chamber and injection system modifications, and possibly a catalyst. Since no data are available on diesel engine exhaust recirculation and its effect on NO emission, the selected reduction factor of 4 is approximate at best. Research work should be conducted, particularly in exhaust recirculation, to provide the parametric data required for a complete assessment of this concept, including the effects on engine performance and "driveability." At present, the prospects for NO catalysts are not bright, but such a device might become feasible in the future.

9.3.2 Part-Load Emissions

The part-load emission characteristics for the three types of diesel engines operating at rated speed are shown in Figure 9-4 in terms of the ratio of part-load specific mass emissions to the full load emissions at rated speed versus percent full load. The HC emissions increase with decreasing load for all engines. For the direct injection engines, the CO emissions decrease initially as a result of increasing air/fuel ratio, and then increase again as load is further reduced. This increase is the result of lower engine efficiency and some increase of CO concentration. The CO emissions of the turbocharged prechamber engine increase steadily with decreasing load.

The part-load NO emission characteristics showed rather distinct and differing trends for the three types of engines. The naturally aspirated, direct injection engine has a flat characteristic, whereas the turbocharged prechamber engine showed an increase and the turbocharged, direct injection engine a decrease in emissions with decreasing load.

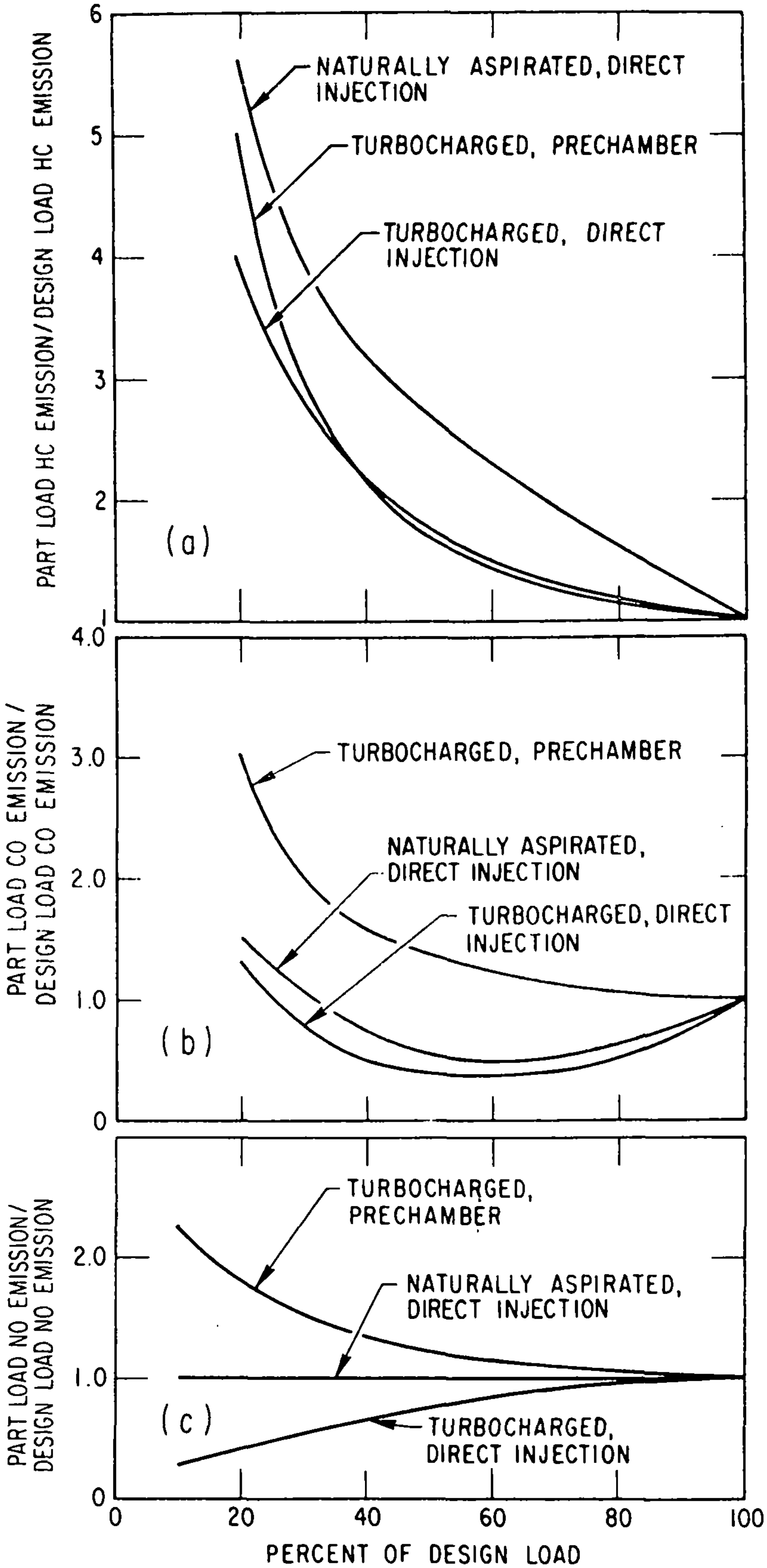

Figure 9-4. Four Cycle Diesel Engine Emissions, Steady State Part-Load, Constant Speed: (a) Hydrocarbon, (b) Carbon Monoxide, (c) Nitric Oxide

The part-load curves shown here are based on constant speed operation and were used for the state-of-the-art vehicle emission calculations. For the projected technology, because of the lack of data on catalytic devices and exhaust gas recirculation effects with load, a flat (invariant with load) characteristic was assumed for all pollutants.

A second set of part-load curves based on varying engine speed with load was constructed and is presented in Appendix B. In general, it is more desirable from an emission and SFC point of view to vary engine speed with load. However, in a hybrid vehicle, the operating characteristics (primarily efficiency) of the other system components must be considered in selecting the optimum engine speed schedule.

Owing to the lack of data, no attempt was made to establish part-load emission factors for two-cycle engines.

9.3.3 Cold Start Emissions

The only cold start emission data available for diesel engines are from the tests conducted by The Aerospace Corporation as a part of this study. The details of this program are contained in Appendix C of this report. Multiple bag vehicular tests indicate that there is no change in CO emissions from cold to hot start conditions. The HC emissions also appear to be the same for both cold and hot conditions, based on comparison of seven-mode data calculated from hot FID concentration data. The NO emissions, as expected, showed a decrease under cold start conditions.

9.3.4 Other Pollutants

The diesel engine can emit other pollutants, primarily odor and smoke. Much work has been conducted in the past to study the odor characteristics of diesels, and it is generally concluded that a relationship exists between odor and aldehydes. However, further study is required before this problem is completely understood. There are indications that the odor/aldehyde emissions from prechamber diesels are lower than those from naturally aspirated engines. Odor from diesel engines can be reduced by fuel injection system modification and by using a catalytic converter in the exhaust. This is discussed in Appendices B and C.

Smoke is largely dependent upon engine air/fuel ratio and load, but is also affected by combustion chamber and injection system design, type of fuel used, and engine maintenance. Formation of smoke has been reduced by using barium additives in the fuel.

9.4 GAS TURBINE EMISSIONS

9.4.1 Design Load Emissions

9.4.1.1 State-of-the-Art Technology

The design load specific mass emissions are presented in Figure 9-5. While these characteristics were based on evaluation of eight different engines, the curves representing the state-of-the-art technology were drawn through the data points from the General Motors GT-309 gas turbine. The HC and CO curves were derived from the basic GT-309 with the so-called standard burner, and the NO curve more nearly matches that indicated by the GT-309 with the modified burner designed for minimum NO.

The design load emission correlations are flat for engine design loads above 50 HP. Below that point, the specific mass emissions are assumed to increase as a result of lower turbomachinery efficiencies. In addition, wall quench effects may become increasingly important, resulting in higher HC and CO concentrations. The HC and CO specific mass emissions are assumed to increase similarly, while NO increases less.

9.4.1.2 Projected Technology

The projected technology emissions are also presented in Figure 9-5. Examination of current specific mass emission data indicates that nitric oxide poses the most serious emission problem in gas turbines, and significant improvements must be made before the future emission goals can be met.

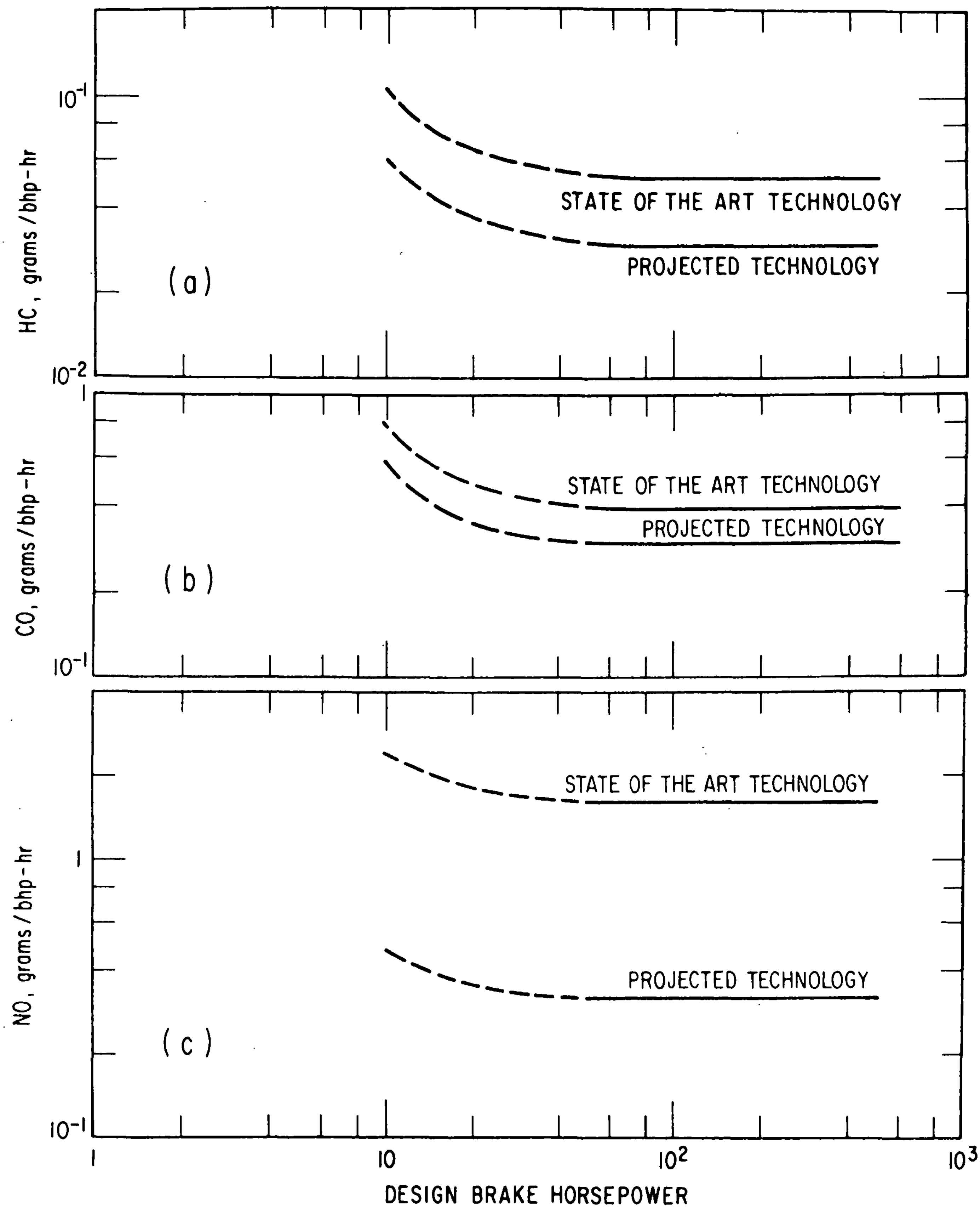

Figure 9-5. Gas Turbine Emissions, Steady State Design Load: (a) Hydrocarbon, (b) Carbon Monoxide, (c) Nitric Oxide

Since the formation of NO is kinetically controlled and is rather slow by comparison to other chemical reactions in the burner, it can be controlled by quenching the NO formation reactions immediately downstream of the primary zone of the burner. The addition of secondary air further upstream of the burner can accomplish this by reducing the residence time of the combustion gases in the primary zones. Experimental work indicates that the nitric oxide emissions are affected by primary zone air/fuel ratio. Reduction in NO by a factor of approximately 2 has been demonstrated experimentally without adversely affecting engine and burner operation and the emissions of hydrocarbon and carbon monoxide. Further improvements are believed to be possible through additional work on the burner, including optimization of primary and secondary zones, inlet air temperature, mixture and mass flow distribution, and addition of exhaust gas recirculation. Also, the feasibility of an exhaust gas reactor should be investigated, especially if significantly higher HC and CO emissions would be obtained as a result of the modifications required for control of nitric oxide emissions.

Based on these considerations, a reduction of the NO emissions by a factor of 5, compared to the present state-of-the-art values, appears to be feasible. The projected HC and CO emissions are reduced by a factor of 2.

9.4.2 Part-Load Emissions

The recommended part-load emission characteristics are presented in Fig. 9-6. Since the General Motors GT-309 gas turbine was designed for automotive use with exhaust emissions a design consideration, its part-load emission data were used as the basis for the part-load curves of Fig. 9-6. The HC and CO specific mass emissions of all engines increased with decreasing load. This is largely due to a reduction of turbomachinery and cycle efficiencies with decreasing load.

Single spool gas turbines show an increase in NO specific mass emissions with decreasing load, which is a direct result of lower thermodynamic cycle efficiency at part load. However, as shown in Fig. 9-6, the part-load NO

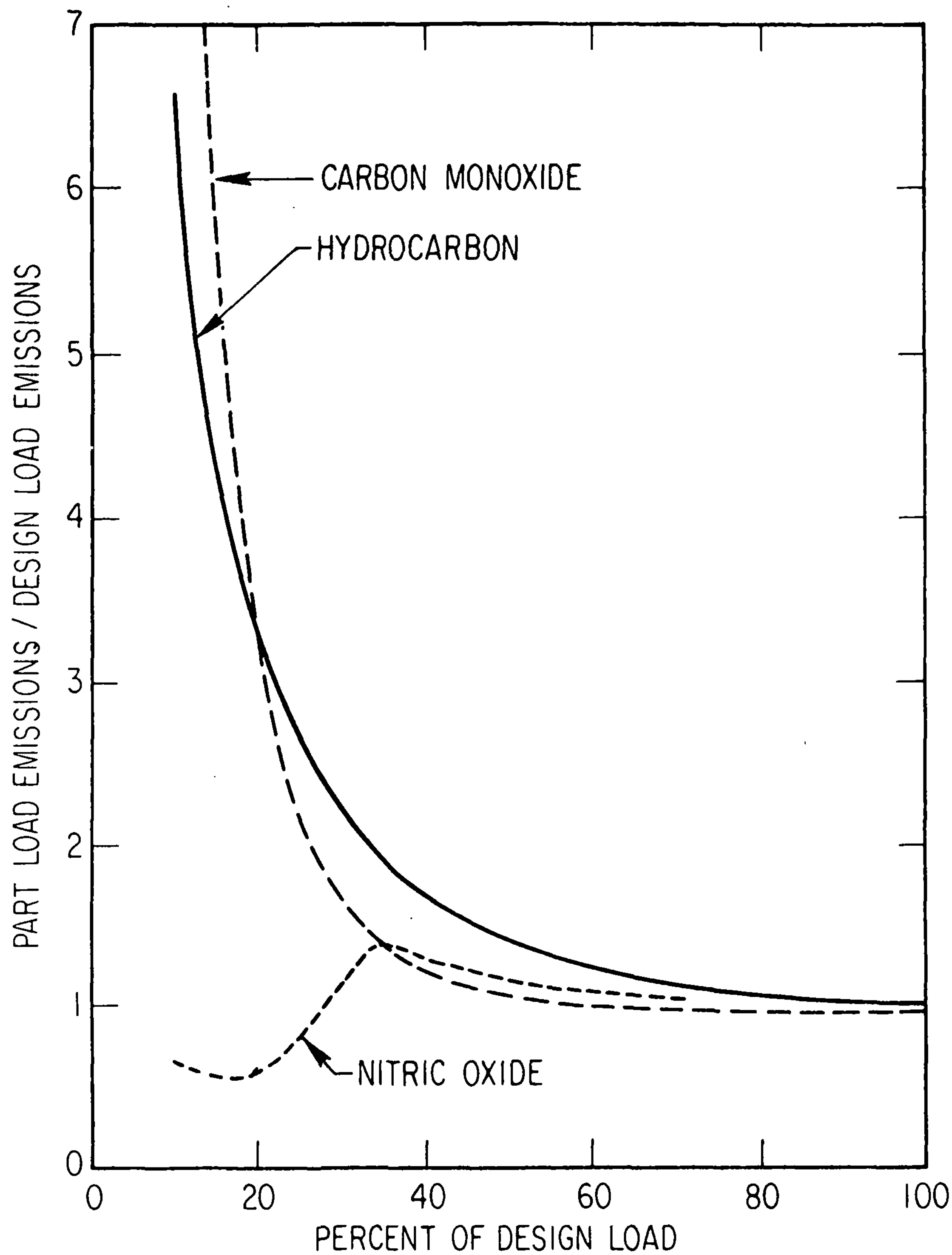

Figure 9-6. Gas Turbines - HC, CO, NO, Emissions, Steady State Part-Load

emissions of the General Motors automotive gas turbine increase initially with decreasing load as a result of increasing burner air inlet temperature. As the degree of power transfer is reduced, both burner and turbine inlet temperature decrease rapidly, resulting in lower specific mass emissions of NO. At very low part loads, the NO emissions increase again as a result of rapidly decreasing turbomachinery efficiencies.

The part-load emission characteristics for the projected technology are assumed to be identical to the present state-of-the-art characteristics. Thus the curves presented in Fig. 9-6 are considered applicable to both present and projected technologies.

9.4.3 Cold Start Emissions

Some cold start data are available for gas turbine automobiles. These indicate that the ratios of cold start versus hot start emissions for HC, CO, and NO are 1.21, 1.17, and 0.89, respectively. These ratios are in reasonable agreement with those determined for spark ignition engines. Additional experimental work is required before the question of cold start versus hot start emissions from gas turbines can be adequately answered.

9.4.4 Other Pollutants

As previously mentioned, this study is only concerned with the emission of HC, CO, and NO. However, a few comments on the other pollutants emitted from gas turbines are in order. Often smoke can be observed in the exhaust of gas turbines, primarily at high loads. This is the result of locally fuel-rich zones in the combustor. Aldehydes are believed to be related to lean combustion and to the temperature-time history of the combustion products. Sulfur dioxide emission is directly related to the sulfur content in the fuel and control is achieved by limiting the allowable sulfur content. Control of smoke and sulfur dioxide appears to be well in hand. However, additional work is required to characterize the emissions of aldehydes from gas turbines and to develop methods which will effectively reduce these products.

9.5 RANKINE ENGINE EMISSIONS

9.5.1 Design Load Emissions

9.5.1.1 State-of-the-Art Technology

The design load specific mass emissions are presented in Fig. 9-7. Data used to characterize the state-of-the-art curve were based primarily on information from the General Motors Research SE-101 and SE-124 steam engines, the Doble automobile tested by General Motors, and the Williams Steamer, as well as burner data from the Marquardt Corporation, Thermo Electron Corporation, and the University of California at Berkeley.

There was a very large scatter in the HC data used to arrive at the HC curve, but much better agreement is achieved in NO primarily, and in CO. Inadequate HC measuring techniques and differences in burner specific heat release rates (residence time) may partially explain the data scatter. In view of these uncertainties, it was decided to use the Stirling engine HC data (which was based on the more reliable hot FID instrumentation) discussed in Section 9.2.5 as a guideline to establish the design load emission characteristics of Rankine engines. This can be done because of the similarity in burners for these two engine types. The specific mass emission data are based upon the assumption of a constant engine efficiency of 15 percent. Changes in this parameter affect the calculated emissions.

The design load emission correlations are flat for engine design loads above 50 hp. Below that point, the specific mass emissions are assumed to increase because of lower engine efficiency.

9.5.1.2 Projected Technology

The projected technology emissions are also presented in Fig. 9-7. The most critical emission specie of the Rankine cycle is NO. Reduction of NO is believed to be possible by means of optimizing the primary and secondary zones of the burner. In addition, exhaust gas recirculation may be feasible. This technique has been used successfully on the Philips Stirling engine. Based on these considerations, the projected NO emission

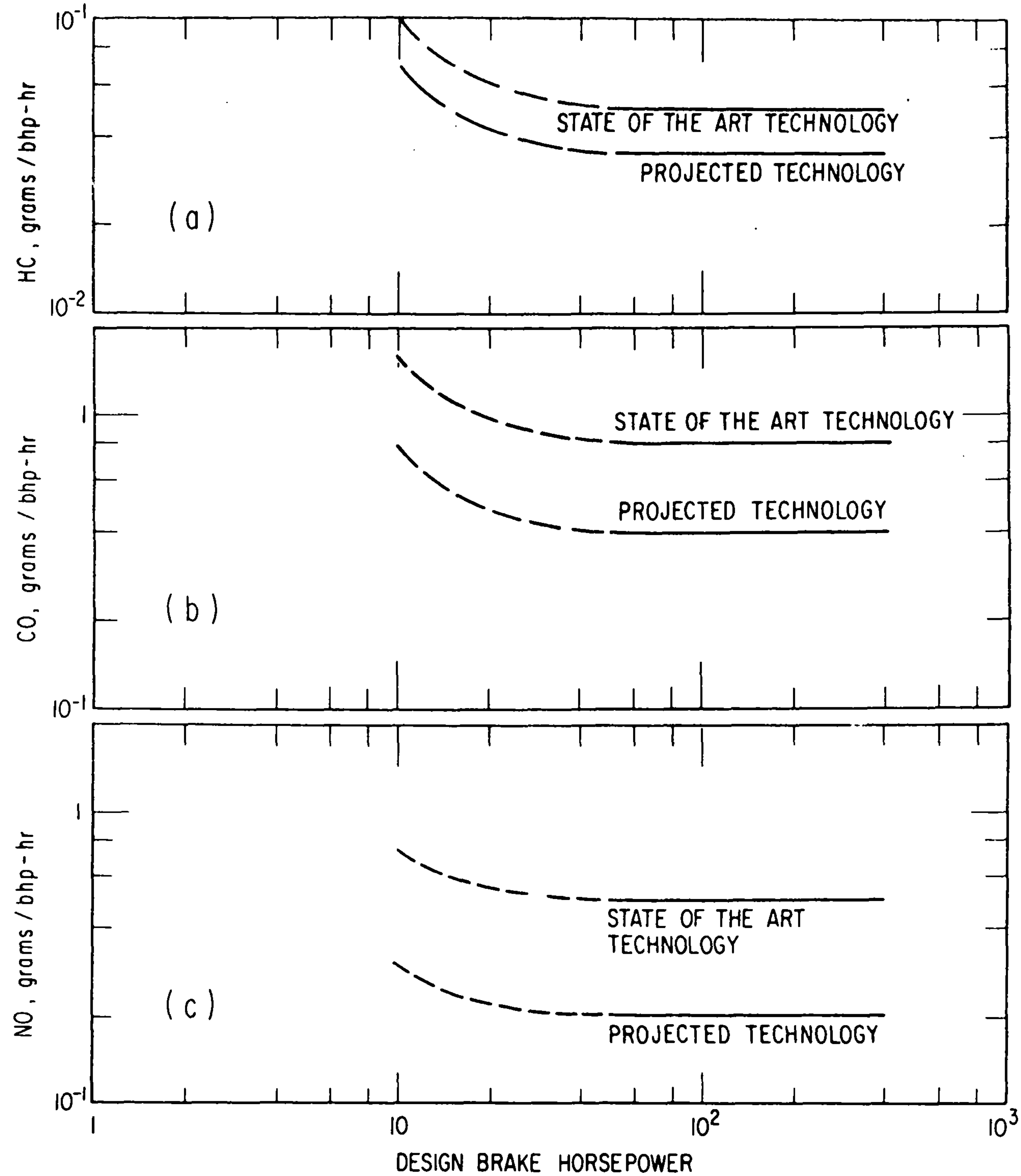

Figure 9-7. Rankine Engine Emissions, Steady State Design Load: (a) Hydrocarbon, (b) Carbon Monoxide, (c) Nitric Oxide

is estimated to be 40 percent of the present state-of-the-art technology value. The projected reduction in HC and CO is somewhat lower, reflecting the general tendency of obtaining higher HC and CO concentrations with the burner design modifications required for NO control.

9.5.2 Part-Load Emissions

There is little agreement in the part-load emission characteristics of the engines and burners tested. Some data indicated a reduction in the concentration of CO and HC with decreasing load, and other data showed little change over a wide range of part-load conditions. For NO, there is reasonable agreement and very little change in NO concentration with load is observed.

Considering the lack of a sufficiently large data sample and the contradictory trends observed in the data, it was decided to use engine efficiency as a measure of the part-load emissions for both present and projected technologies. The part-load factors presented in Fig. 9-8 reflect the variation of efficiency with load. Obviously, this is a crude assumption and points out the need of reliable Rankine engine part-load emission data.

9.5.3 Cold Start Emissions

The only cold start emission information available is the data published by General Motors for the G. M. SE-101, SE-124, and Doble steam cars. These are discussed in Appendix B. The warmup emissions have only a small effect on the total emissions of the SE-101 automobile, resulting from the fact that only a 2.8-min warmup period was required to achieve adequate steam pressure. In the SE-124 and the Doble, the warmup emissions represent a significant portion of the total emissions, primarily because a much longer warmup time was required.

9.5.4 Other Pollutants

In addition to measuring the emissions of HC, CO, and NO, General Motors has made attempts to determine the odor and smoke characteristics of their engine. No offensive odor was detected so long as air/fuel ratio was below 40:1. Smoke was never observed at air/fuel ratios of 25:1 or higher.

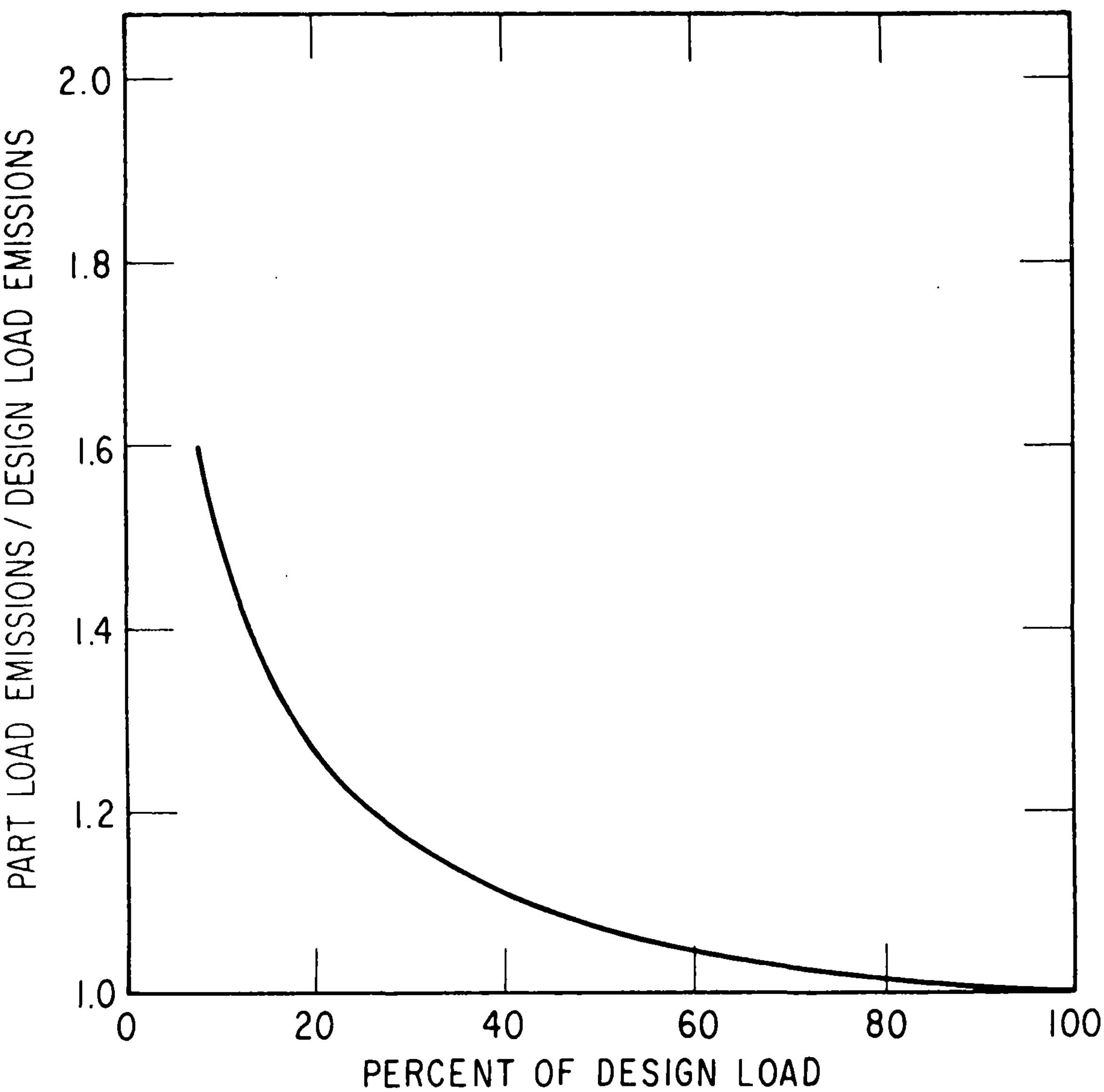

Figure 9-8. Rankine Engines - HC, CO, NO, Emissions, Steady State Part-Load

9.6 STIRLING ENGINE EMISSIONS

9.6.1 Design Load Emissions

9.6.1.1 State-of-the-Art Technology

Characteristics of the exhaust emissions of Stirling engines are based upon emission data from two engines built and tested by General Motors Research Laboratories and by Philips Research Laboratories of the Netherlands.

The design load specific mass emission characteristics are shown in Fig. 9-9. The upper curves in these figures are considered a reasonable representation of the state-of-the-art technology. As in the other heat engines the specific mass emissions are considered constant for design power levels above 50 hp. Below that point the specific mass emissions are assumed to increase to reflect the deterioration of engine efficiency. It should be pointed out that in arriving at the state-of-the-art curve, several HC and CO data points actually fell below the selected curve. However, the corresponding NO emissions for these points were excessive. This points out the importance of selecting the proper combination of engine operating parameters to minimize all emissions. Since HC and CO are inherently low, attention must be primarily focused on NO. The curves reflect this approach.

9.6.1.2 Projected Technology

Nitric oxide is the principal emission problem in Stirling engines. A number of approaches were considered to reduce the emissions, including burner modifications, exhaust gas recirculation, lower burner air inlet temperature and reduction of residence time of the gases in the primary zone of the burner.

With these considerations in mind the projected technology-specific mass emission curves were drawn in Figure 9-9. The projected NO emissions are lower than the corresponding present state-of-the-art values by a factor of three. Since the various approaches aimed at reducing NO have a tendency to increase the other emission species, it is assumed that the projected HC and CO emissions are reduced by only a factor of 1.5

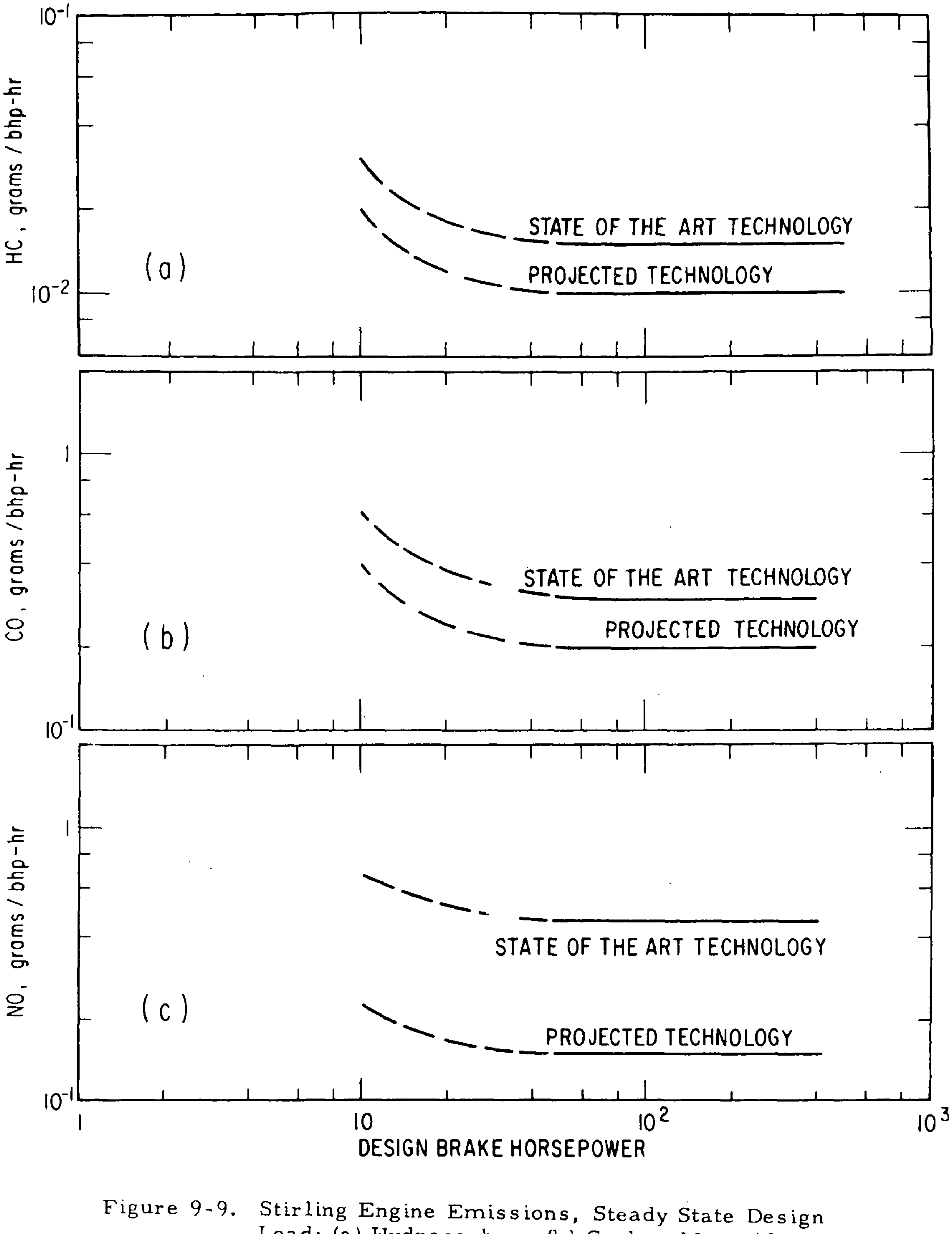

Figure 9-9. Stirling Engine Emissions, Steady State Design Load: (a) Hydrocarbon, (b) Carbon Monoxide, (c) Nitric Oxide

9.6.2 Part-Load Emissions

Lacking sufficient test data, a meaningful part-load emission study could not be conducted for the Stirling engine. To account for at least some of the part-load effects it was decided to use the cycle efficiency versus percent of design load correlation as the basis for estimating the emissions at part load. The same approach was used to characterize the part-load emissions of Rankine engines. The recommended part-load emissions are presented in Fig. 9-10 in terms of the ratio of part-load emission to design load emission versus percent of design load. These factors are applicable to HC, CO and NO, for both present state-of-the-art and projected technologies. This approach is approximate, at best, and points out the need of a comprehensive Stirling engine emission test program.

9.6.3 Cold Start Emissions

There is no information available to characterize cold start emissions of Stirling engines. Obviously these factors have to be resolved before a complete assessment can be made of the emissions from a Stirling engine.

9.6.4 Other Pollutants

Little information is available on smoke and odor of the Stirling engine exhaust. The General Motors engine was reported to be smoke free at all operating conditions, including cold start. Also no odor was detected. The Philips engine shows some smoke during warmup. However, it appears that this problem might be alleviated by means of burner modification and/or variation of the air/fuel ratio during warmup.

9.7 SUMMARY

The design load specific mass emissions for each of the five heat engines are summarized in bar-chart format in Fig. 9-11 for engines greater than 50 hp. These charts allow a relative comparison of the specific mass emissions for each engine category for both state-of-the-art and projected technologies. It is emphasized that the values indicated do not give a direct

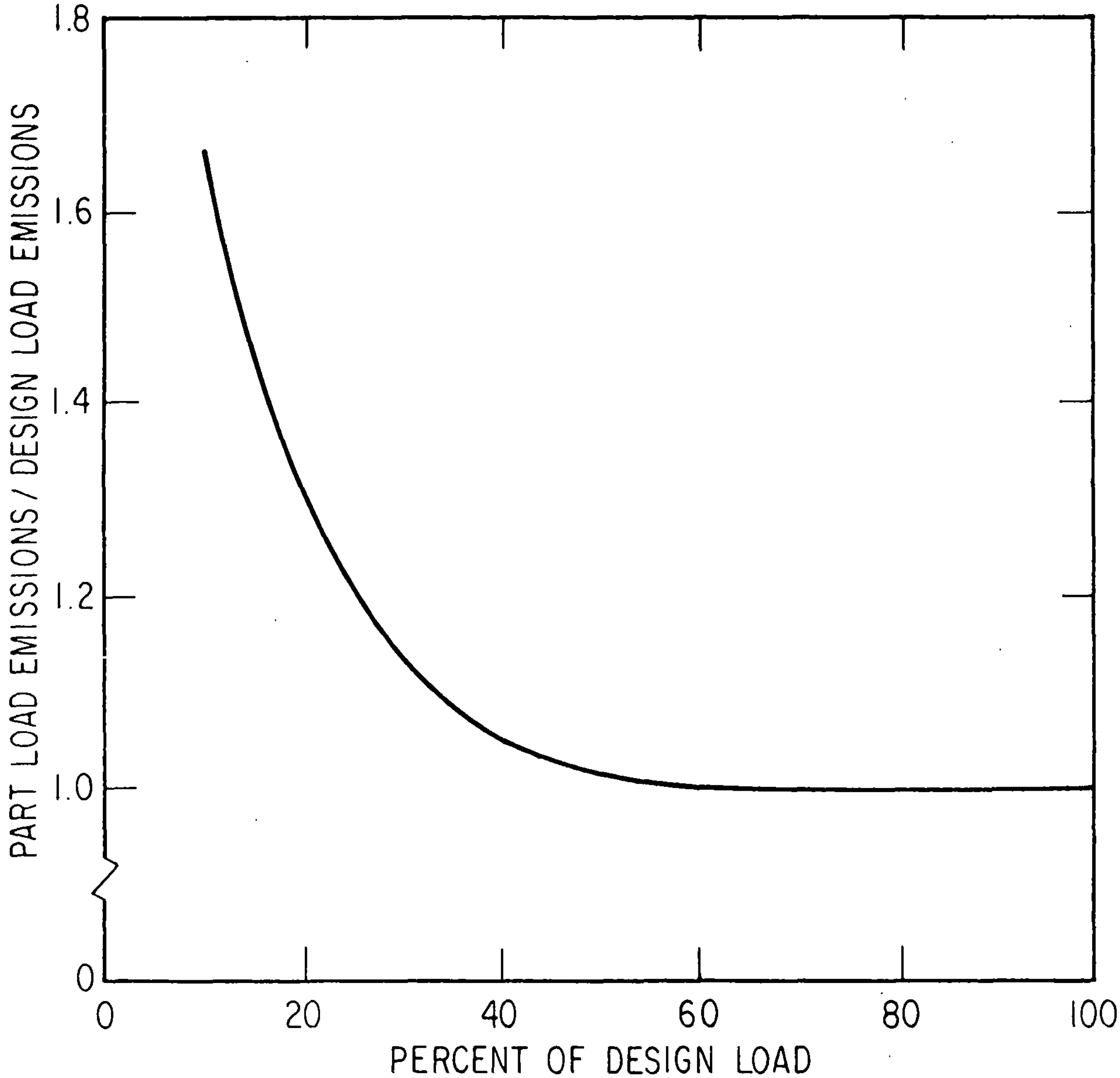

Figure 9-10. Stirling Engines - HC, CO, NO, Emissions, Steady State Part-Load

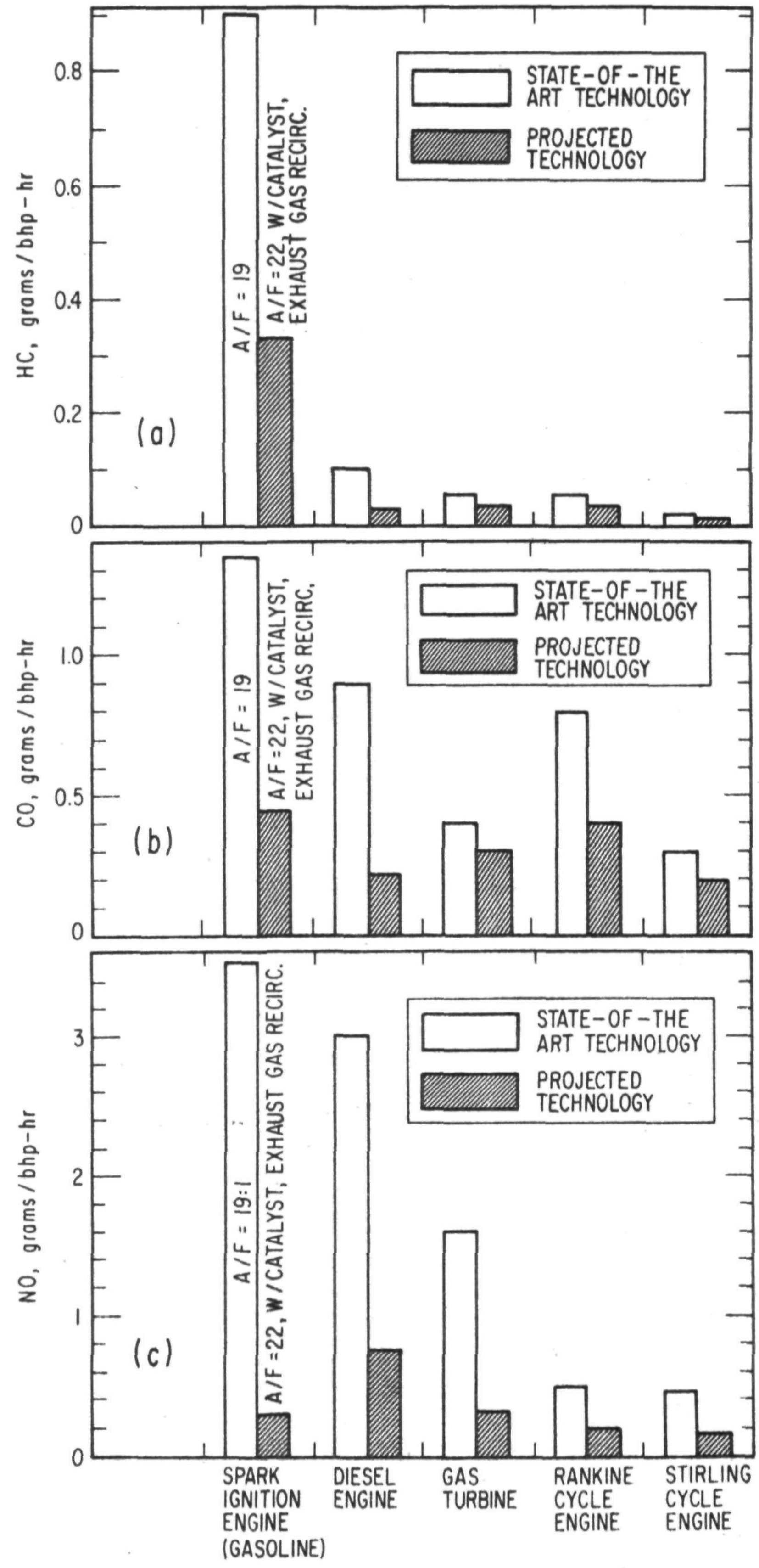

Figure 9-11. Heat Engine Exhaust Emissions, Large Engines (> 50 hp) Steady State Design Load: (a) Hydrocarbon, (b) Carbon Monoxide, (c) Nitric Oxide

correlation to vehicle emissions since the latter will be based on the part-load operating point and its attendant emission level.

Figure 9-11a compares the HC emissions for the five engines. The spark ignition engine is considerably higher than the other four engines, and even with a catalyst, the HC level of the projected technology engine is still a problem insofar as spark ignition engines are concerned. Figure 9-11b concerns the CO emissions for the five engines. Notice that the level indicated for the state-of-the-art technology spark ignition engine corresponds to operation at an air/fuel ratio of 19:1. The conventional spark ignition engines operating in a rich air/fuel ratio regime would more typically have CO specific mass emissions of 40 grams/bhp-hr or greater. Even for an air/fuel ratio of 15-16 the CO specific mass emission level is 5.5 grams/bhp-hr. Figure 9-11c compares the NO emissions, and it can be seen that for state-of-the-art technology, spark ignition engines, diesel engines, and the gas turbine all exhibit relatively high NO emissions. With projected improvements, these levels can be reduced considerably. As discussed in Appendix B, the possibility exists for lower NO emissions than indicated for the diesel engines; however, no supporting data were available during the course of the study.

Again, from the standpoint of the impact of the specific mass emissions on hybrid vehicle emissions, the part-load characteristics for each engine are critical, and more data are required to substantiate the projections made.

References for this discussion of heat engine exhaust emissions are provided in Appendix B, together with a collation of pertinent data.

SECTION 10

CONCEPTUAL DESIGN AND SIZING STUDIES

CONTENTS

TABLES

TABLES (Continued)

FIGURES

SECTION 10

CONCEPTUAL DESIGN AND SIZING STUDIES

10.1 CONCEPTUAL DESIGNS

10.1.1 Introduction

Heat engine/electric hybrid powerplant concepts can be grouped into two broad classes: series and parallel configurations, as previously defined in Section 3 and further discussed in Section 6.

In all cases, the difference between power required for vehicle propulsion and power supplied by the heat engine must be supplied by the batteries. Hence, at the outset, it should be recognized that once the vehicle maximum power requirements have been established, the battery design goals can be markedly influenced by the heat engine power output profile. This effect is shown in Fig. 10-1 where vehicle maximum power (for maximum acceleration) and cruise power requirements are illustrated, along with three different vehicle-velocity varying power profiles delivered by the heat engine. Profile #2 is defined as that power output profile which will result in the batteries being fully recharged at the end of the driving cycle.

It is clear that profile #1 imposes far less severe requirements on the battery (in terms of power demand) than profiles #2 and #3, but it also requires a higher level of sustained heat engine power output at lower vehicle speeds, and is in excess of that heat engine power level required to maintain the battery state-of-charge, thus resulting in higher heat engine exhaust emissions and increased fuel consumption.

An alternative type of power profile to those shown in Fig. 10-1 can be envisioned wherein the heat engine is required to accelerate (change power output) rapidly, as in conventional SI engine-powered vehicles. The heat engine could have a power output profile similar to profile #3 for constant velocity operation, and accelerate to a maximum power level (similar to the level of profile #1) during periods of vehicle acceleration, thus reducing battery peak demand requirements. However, for purposes of this study, it was assumed that this form of engine performance might not be attainable with low-pollution engines with possible "driveability" (i.e., smooth power-output profiles under instantaneous load changes) constraints.

Therefore, all subsequent discussion is directed toward conceptual approaches in which heat engines are not subjected to large instantaneous changes in power output. In the following illustrative cases which depict power output varying with time, vehicle velocity, or step changes, it is assumed that these power changes take place over finite time intervals commensurate with the acceleration capability of the engine under load.

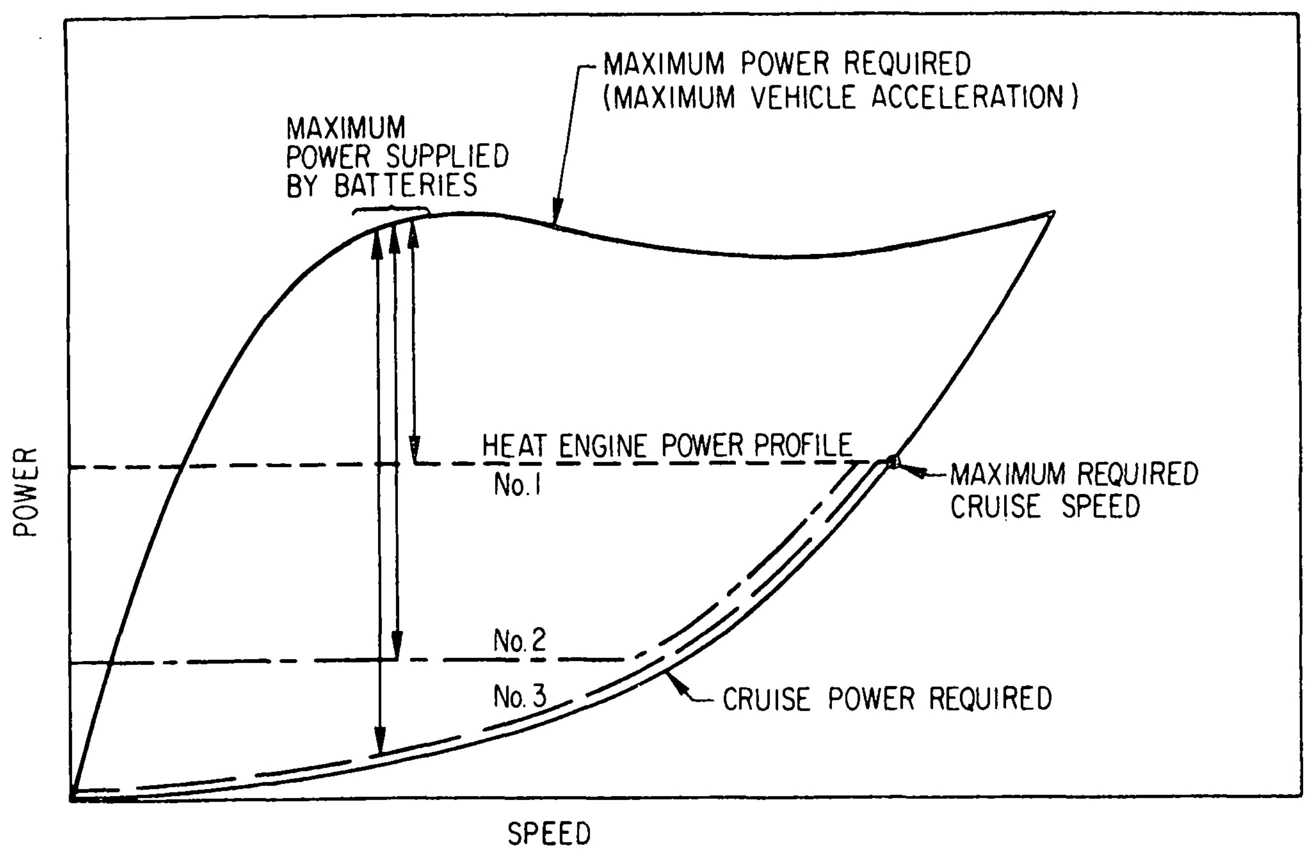

Figure 10-1. Effect of Heat Engine Power Profile on Required Maximum Battery Power

10.1.2 Series Configuration

10.1.2.1 Basic Subsystems/Components

A heat engine/electric hybrid powerplant configured for the series mode, as defined above, requires certain basic subsystems/components which can vary as to type and/or number according to designer's choice.

For example, a single electric drive motor can be utilized with a central differential drive to the driving wheels, or multiple drive motors could be used with a drive motor at each wheel, negating the need for the differential drive. However, aside from this configurational design option, the remaining options in the series configuration primarily center around selection of the specific type of subsystem/component to be used, and the control system to be used for the preferred mode of operation.

The selected series configuration used as a baseline in the present study for all vehicles is as shown in Fig. 10-2. A single electric drive motor is used to supply power to the rear wheels through a central differential drive unit. Where appropriate, the differential drive unit is envisioned to contain an overdrive unit to provide a step change in electric drive motor rpm to allow high-speed cruising at near-maximum drive motor efficiency levels.

The generator is mechanically-driven by the heat engine through a gearbox (speeder/reducer) which allows the generator (or alternator as the case may be) and the heat engine to operate at different rpm levels. The other two major subsystems (i.e., battery, control system) are then electrically-connected to the generator and drive motor as schematically represented in Fig. 6-1.

10.1.2.2 Operational Modes

With the foregoing series configuration arrangement, a number of modes of operation are conceivable. Several of the more significant modes are shown in Fig. 10-3 and discussed in the following paragraphs in terms of the mode of operation of the heat engine. The heat engine mode of operation was

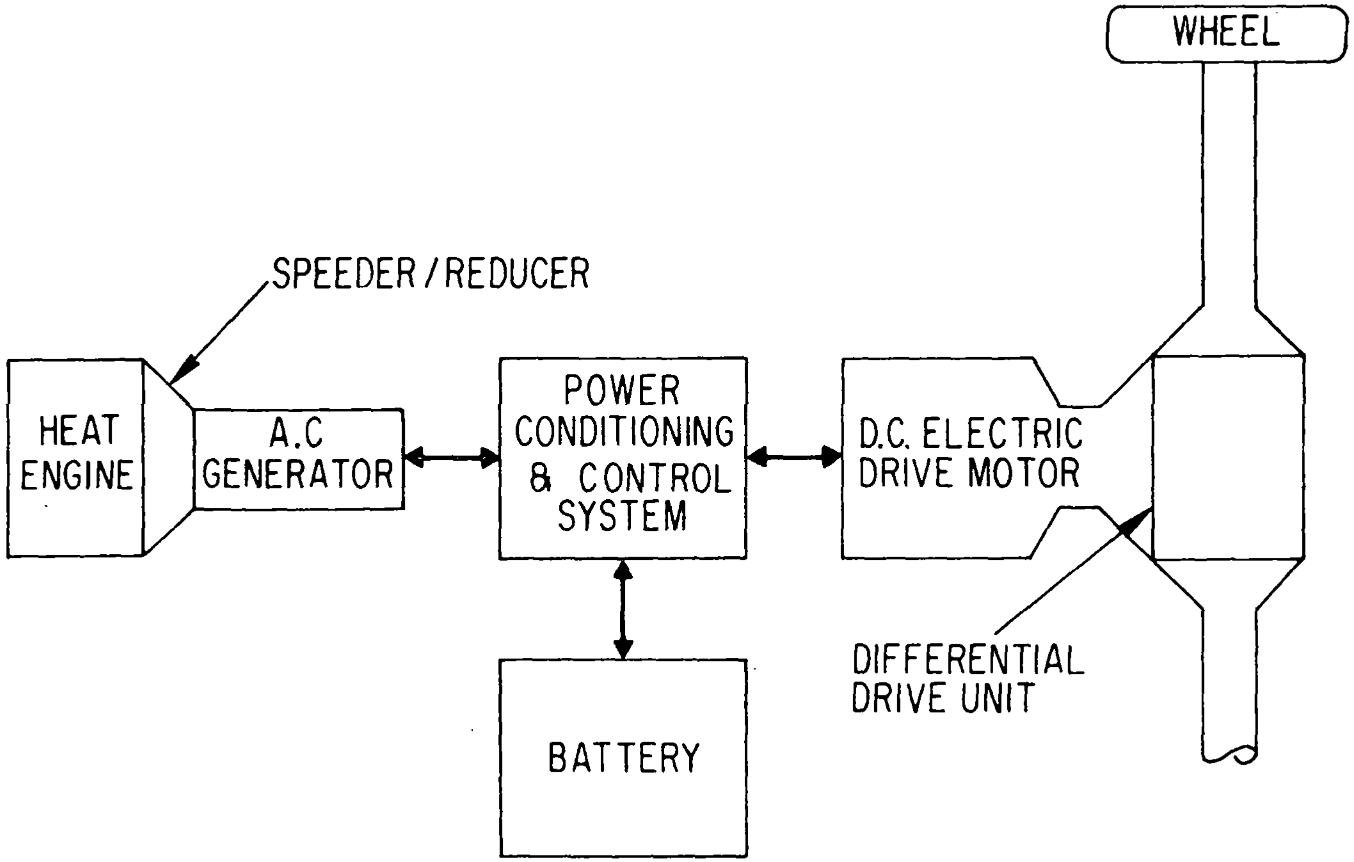

Figure 10-2. Selected Baseline Series Configuration

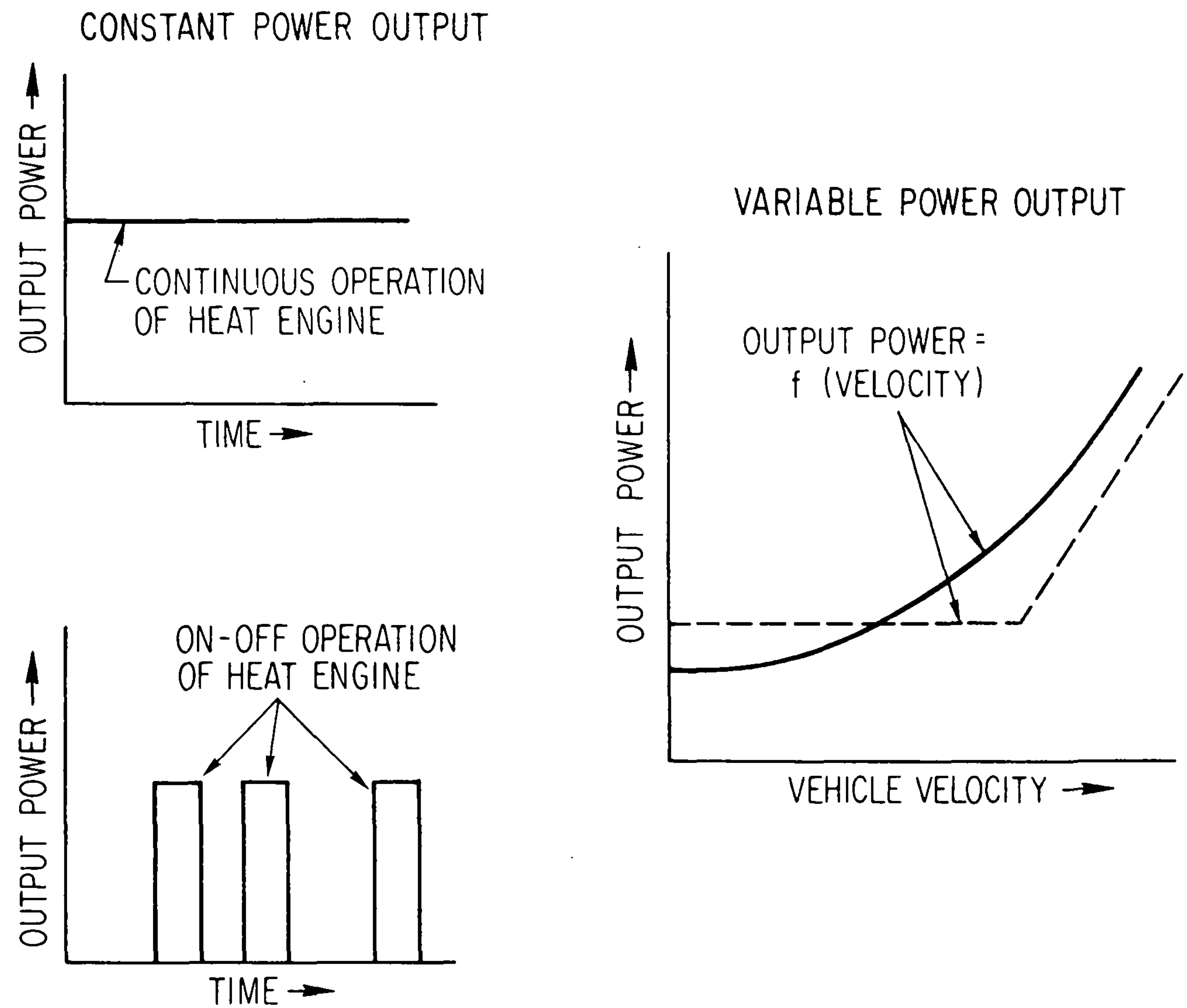

Figure 10-3. Various Heat Engine Operational Modes
Series Configuration

selected as the descriptor in that heat engine exhaust emission determination, a principal objective of this study, is more directly-relatable to this descriptor.

10.1.2.2.1 Constant Speed (rpm) and Power Output

10.1.2.2.1.1 Heat Engine Operated Continuously

In this mode of operation, a severe problem arises in relation to sizing the heat engine. If the heat engine is sized only to produce a total energy required in the time duration of the emission driving cycle (including inefficiencies of the powerplant system), then the heat engine may not provide the proper continuous high-speed power demand for highway operation. This results in discharge of the batteries at high speeds (if the heat engine size is too small). Conversely, if the heat engine is sized for the maximum continuous power demand for highway operation, excessive energy loss to a heat-dump can occur (if heat engine size is too large).

This mode of heat engine operation is of course attractive from the standpoint of heat engine exhaust emissions per se, in that it should be possible to select an operating point (i.e., rpm, air/fuel ratio, etc.) most amenable to reduced emissions. However, its apparent inflexibility with regard to heat engine sizing and meeting both design driving cycle as well as emission driving cycle vehicle performance led to its discard as a viable series mode of operation for the particular classes of vehicles under consideration. However, this mode may still be suitable for vehicles with reduced top speeds and/or revised specification requirements.

10.1.2.2.1.2 On-Off Operation of Heat Engine

As an alternative to continuous operation, it is possible to operate a constant power output heat engine in an "on-off" mode. Here, the heat engine would be sized to meet the continuous high-speed power demand for highway operation, and would operate intermittently during urban driving conditions. The heat engine could be turned on or off in response to (a) a battery voltage and/or state-of-charge signal, (b) a power demand from the electric drive motor, or (c) a combination of both.

In this manner, various total energy requirements can be matched by the "pulse" mode operation of the heat engine, which would allow the engine to be off more in low-energy portions of urban driving cycles.

Preliminary calculations, however, indicated that this mode of operation resulted in very high energy losses during those time periods when drive motor demand is low due to battery charge rate limitations, i.e., a good portion of heat engine power output must be dumped because the battery simply cannot accept the power at the rate being supplied. This same limitation applies to the continuous heat engine operation mode previously discussed.

There is one further disadvantage of intermittent or on-off operation. When operating continuously, power from the heat engine/generator can go directly to the drive motor during periods of power demand and bypass the battery loop entirely. When operating in the on-off mode, it would only be fortuitous if drive motor power demand occurred at the same time the engine was on. Therefore, more of the heat engine power output flows through the battery circuit in the "on-off" mode than in the continuous operation mode. Even if battery recharge efficiency is high, the on-off mode of operation would be less efficient than the continuous mode of operation. It was concluded that while on-off operation of the heat engine at constant power output was more flexible than continuous operation at constant power output, it was not adequate for the wide range of vehicle driving requirements under consideration.

10.1.2.2.2 Variable Power Output

10.1.2.2.2.1 Heat Engine Operated Continuously

Many of the deficiencies of the constant-power output mode of operation can be avoided by allowing the power output of the heat engine to vary. In this case, the heat engine can be sized for the maximum continuous power requirement and allowed to operate at lower power levels for those periods of vehicle driving cycles which require less power. If heat engine rpm is also allowed to vary to produce this variation in power output (as in conventional internal combustion engines), it is envisioned that the control system can effectively vary throttle setting response time constants so that engine rpm and power changes take place at a controlled rate in such a manner that no true vehicle acceleration demands are imposed on the heat engine in the conventional sense.

Such a mode of operation would allow the energy requirements of a variety of urban driving cycles to be more closely matched (than with constant power output mode), although the matching of all duty cycle energy requirements may not be possible. To overcome this difficulty, it has been suggested that the heat engine power output be scheduled as a function of vehicle velocity (heat engine produces more power as road load increases) with a throttle "bias" feature in the heat engine fuel control system to increase or lower the baseline heat engine power output schedule in accordance with an input signal related to battery voltage and/or state-of-charge as illustrated in Fig. 10-4.

With these features, the continuous operation of the heat engine on a variable power output basis appears to be a highly versatile and accommodating mode of operation for the series configuration of a heat engine/electric hybrid powerplant.

10.1.2.2.2.2 <u>"Step-Mode" Operation</u>

Another technique for varying heat engine power output is to schedule power output in discrete steps. Figure 10-5 illustrates one such approach, wherein three levels of power output are used. A "low" level would be scheduled for a low-velocity range (e.g., 0-30 mph), an "intermediate" level for velocities between the low-velocity range and vehicle top speed, and a "peak" level for cruising at maximum continuous power conditions.

Again, battery voltage and/or state-of-charge signals could be used to override the nominal schedule of power output versus velocity.

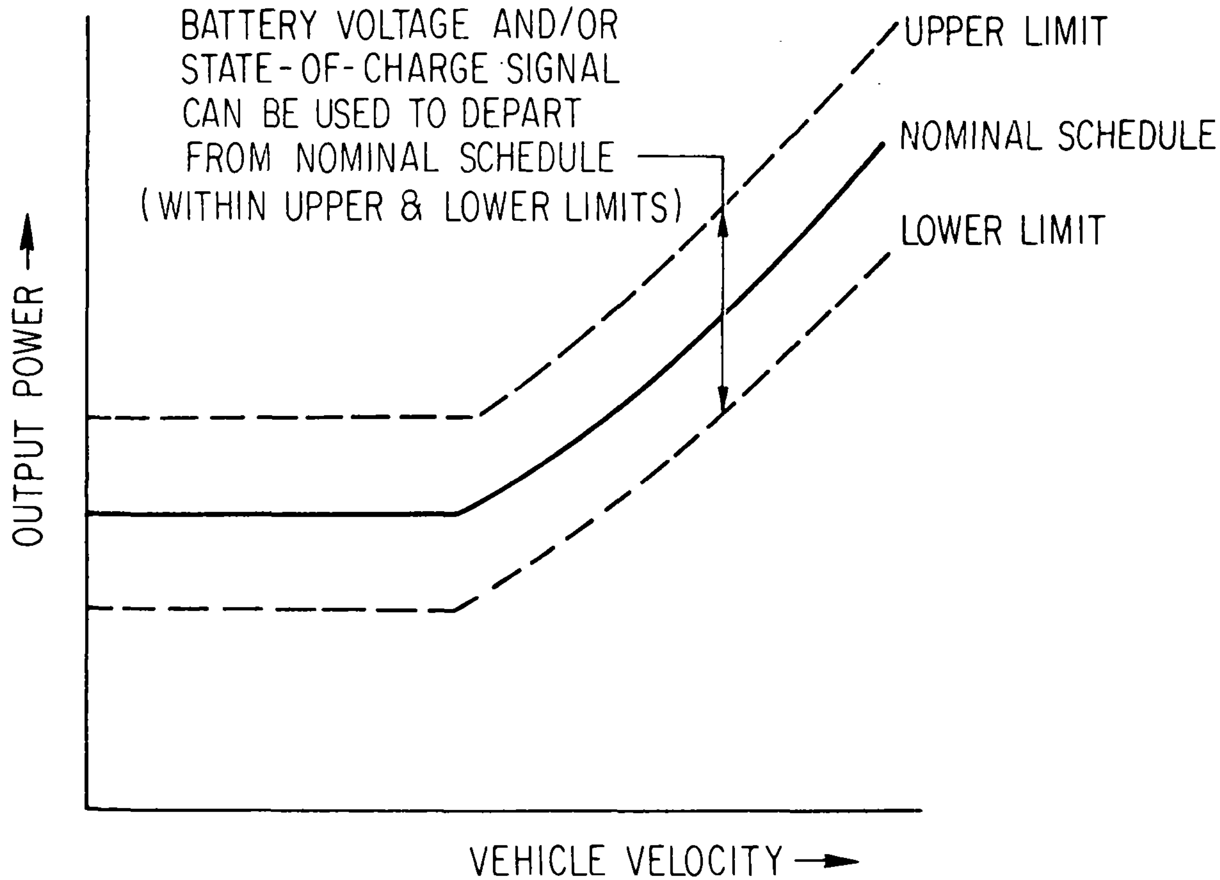

Figure 10-4. Heat Engine Variable Power Output Mode "Biased" Throttle Setting Feature

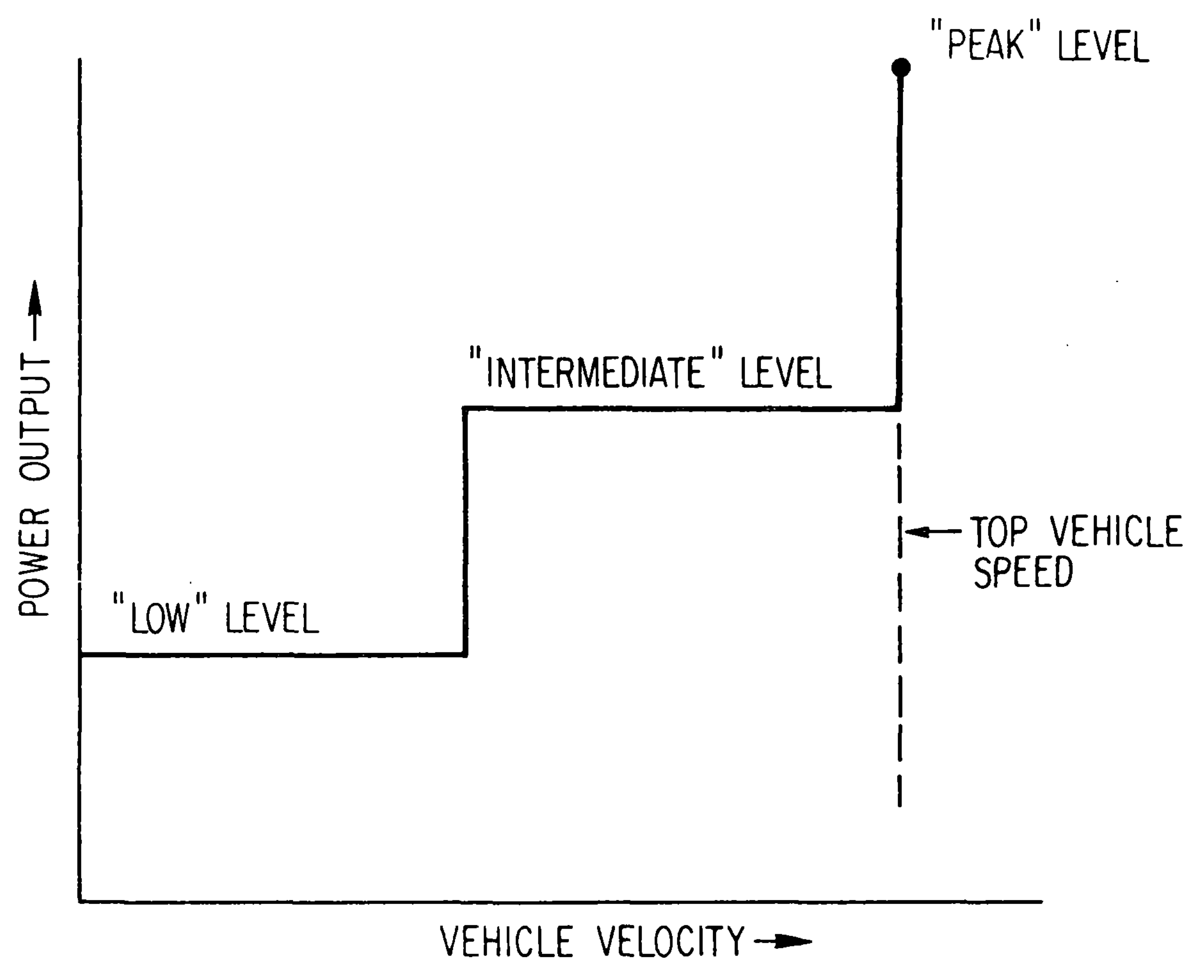

Figure 10-5. Heat Engine Variable Power Output Mode Step-Mode

10.1.2.2.3 Selected Baseline Operational Mode

On the basis of the discussion of characteristics, advantages, and disadvantages, the variable power output mode with the heat engine operating continuously throughout the driving cycle was selected for the series configuration. More specifically, the heat engine power output was tailored in accordance with vehicle velocity as shown in Fig. 10-6. The specific heat engine power output profile for each vehicle is a function of the power required for steady road load above a certain vehicle velocity. Below this velocity, the power output is at a constant value. This value is determined uniquely for each vehicle as the value required to result in the battery being returned to its initial state-of-charge at the end of the vehicle driving cycle. The more sophisticated approach of having battery voltage and/or state-of-charge override this value as depicted in Fig. 10-4, while offering greater system flexibility, falls outside the scope of the current study.

10.1.3 Parallel Configuration

10.1.3.1 Basic Subsystems/Components

A heat engine/electric hybrid powerplant configured for the parallel mode requires the same basic subsystems/components as the series mode plus the additional need for a transmission or gearbox for the mechanical drive from the heat engine to the differential drive and/or wheels. However, the sizing criteria for some subsystems are very different from those in the series mode. For example, the drive motor in the series case must be sized to provide all power required at the wheels. In the parallel case, the drive motor is supplementary to the mechanical power supplied by the heat engine, and is sized to provide acceleration torques on an intermittent basis, not continuous duty. The generator in the series case is sized to accommodate full power output of the heat engine, while in the parallel case the generator is sized on the basis of heat engine minimum operating power level. The size of the heat engine required can differ between the two concepts, depending upon the particular subsystem efficiencies assumed. The particular choice of mechanical arrangement of the heat engine, generator, transmission/gearbox, and drive motor can also result in the requirement for more than one drive motor, or for the drive motor to have the dual function of motor and generator (motor/generator).

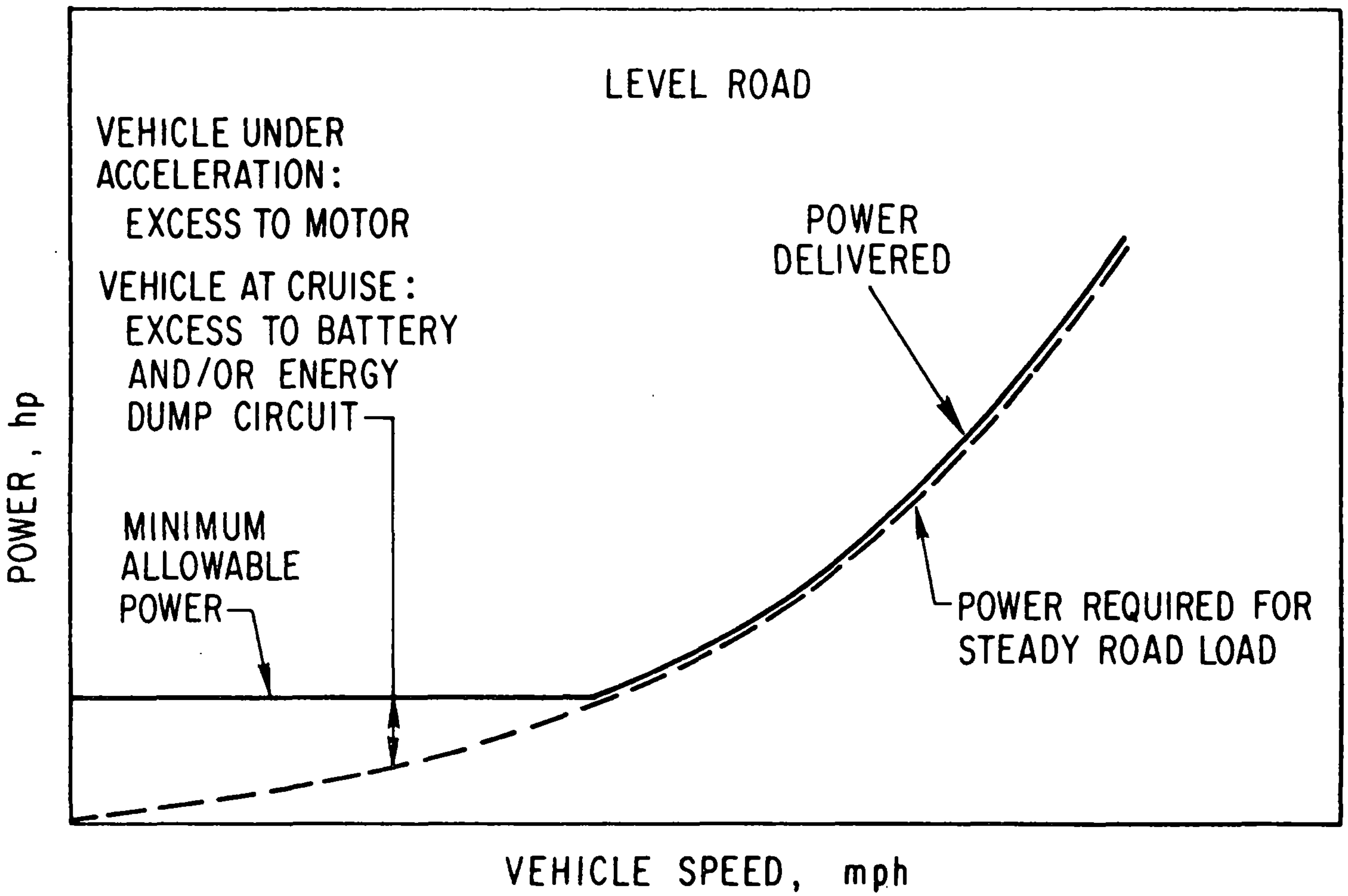

Figure 10-6. Series Configuration - Variation of Heat Engine Power with Vehicle Speed

Because of the wide variation in mechanical approaches possible for the parallel configuration, a simple concept was selected for the baseline parallel configuration in the present study which more readily allowed for a direct comparison of the inherent features of the parallel versus series approach in terms of heat engine power/energy requirements and resultant exhaust emissions over the emission driving cycles. A discussion of various parallel configurations can be found in Section 6.

As shown in Fig. 10-7, the baseline parallel concept utilizes an automatic transmission to provide the mechanical drive connection from the heat engine to a differential drive unit powering the drive wheels. The electric drive motor, used for vehicle acceleration torque demands, is geared to the output shaft of the transmission. The generator is similarly geared to the output shaft of the heat engine. The control system is postulated to have the capability to synchronize the input and output rpm's of the automatic transmission (by controlling heat engine rpm, generator load, and drive motor rpm) to the extent they are essentially equal and that fluid coupling losses are minimal (i. e., no torque amplification used).

In this concept, the generator can supply power to the batteries when heat engine power is in excess of wheel demand, and the drive motor can also function as a generator during periods of deceleration, if desired (regenerative braking). Additionally, the drive motor could also function as a generator during vehicle cruise periods if the heat engine power output was prescheduled or "biased" via the throttle schedule in the control system to provide more power at any given speed than required by the vehicle for road load power. Conceptually, the baseline parallel system provides all of the operational attributes postulated for the various single motor parallel concepts.

10. 1. 3. 2 Operational Modes

With the parallel configuration arrangement, a single mode of operation was selected as most compatible with the hardware arrangement and as providing an equitable comparison with the operational mode selected for the baseline series configuration. As shown in Fig. 10-8, the total power output of the heat engine is scheduled as a function of vehicle velocity, with that portion above a certain velocity equal to the steady road load power. Below this velocity, a minimum power level, constant with velocity, is selected. The

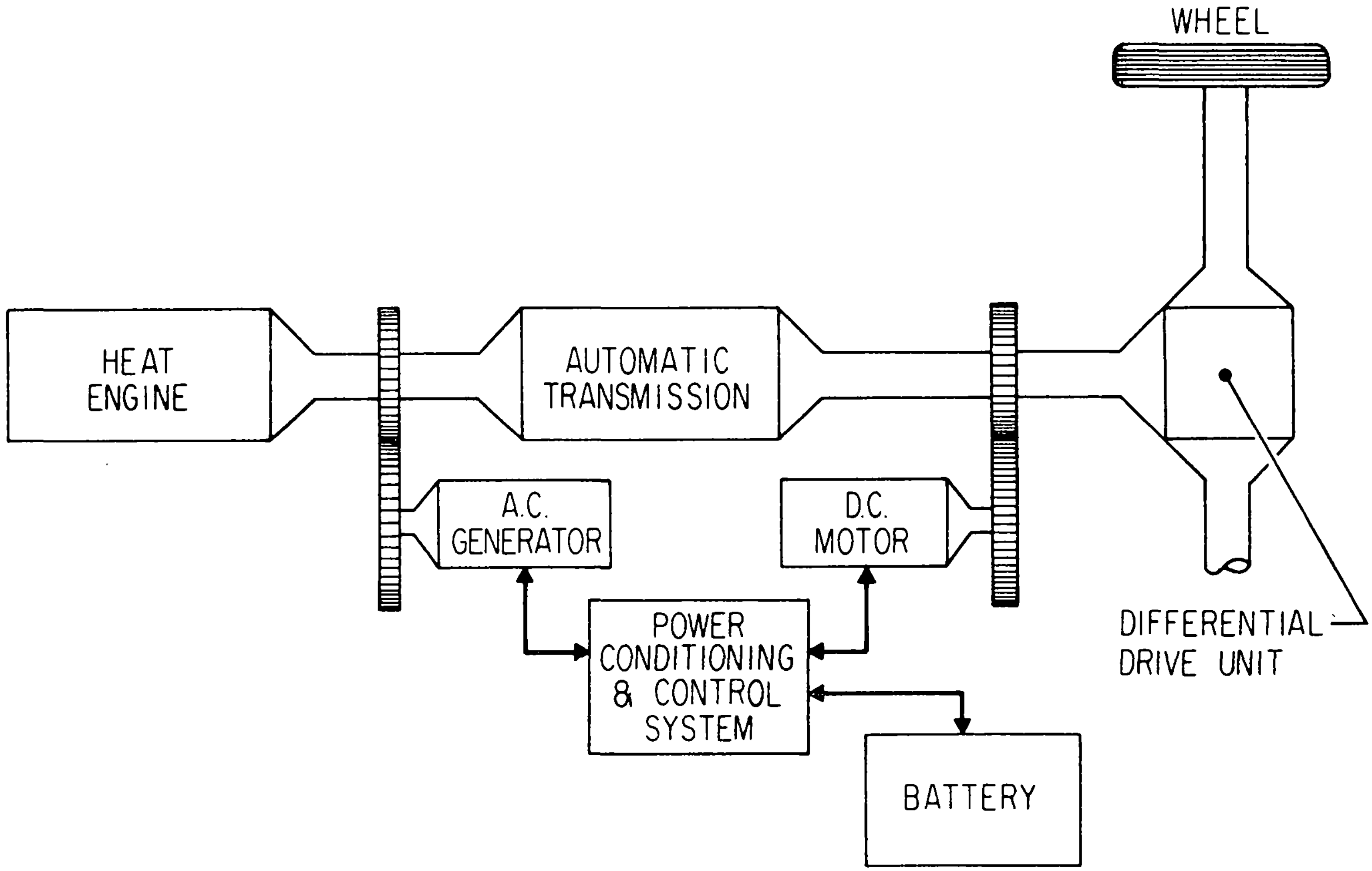

Figure 10- 7. Selected Baseline Parallel Configuration Concept

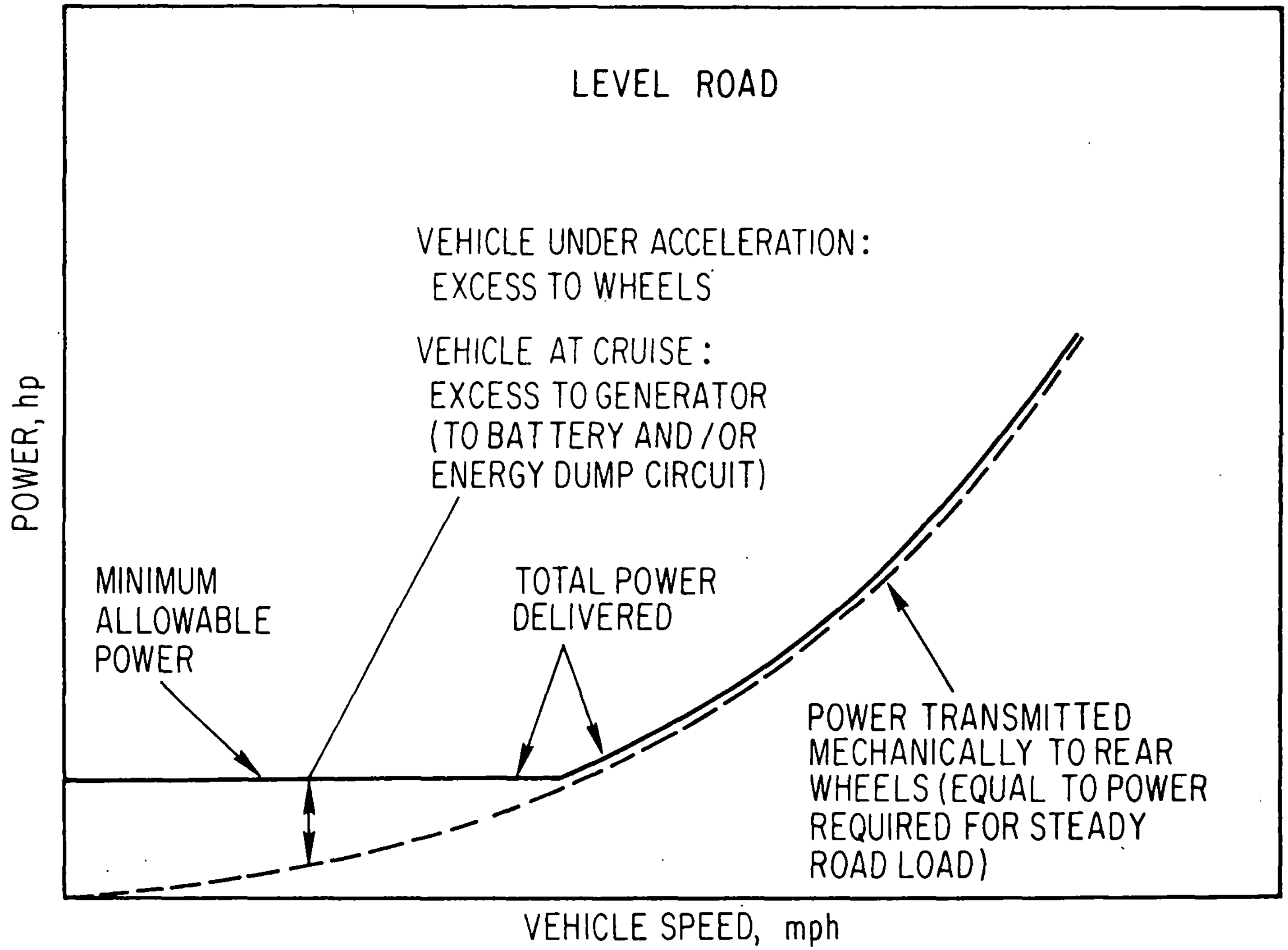

Figure 10-8. Parallel Configuration - Variation of Heat Engine Power with Vehicle Speed

portion of heat engine power transmitted mechanically to the rear wheels is shown by the dashed line and is just equal to that power required for constant velocity road load demand. The difference, then, between the selected minimum power level and road-load demand is available to the generator, and is either used to charge the battery, go to the drive motor during acceleration, or to an energy dump circuit as appropriate to the particular driving schedule/cycle. During periods of vehicle deceleration, the mechanical power is reduced to zero and the heat engine power output is reduced to the minimum level. The motor (as a generator) and/or the generator can then utilize the heat engine power output during vehicle deceleration for battery recharging.

10.2 SIZING STUDIES

10.2.1 Subsystem Sizing

In order to conduct the desired performance and tradeoff studies, it was necessary to select subsystems, define vehicle characteristics and establish a baseline for comparing various vehicle classes. It should be stressed that, due to the complexity of factors and problems involved in analyzing various hybrid systems during the short duration of this study, it was only possible to make limited, general investigations of the wide range of subsystems and alternative schemes possible. This report should therefore be considered in this context and as establishing the basis for more refined investigations.

Component characteristics for the electrical subsystem are merely initial selections, based on the limited scope of technology review of Section 6, and do not at this time represent either optimized systems or preferred approaches. Rather they are considered to be preliminary selections serving as a baseline for comparison of various vehicles.

In the case of the family car for example, to establish a baseline two types of motor voltage control were considered, namely:

- A solid state chopper control
- Voltage step switching combined with field control of the motor

Since step voltage switching combined with field control of the motor has not been extensively demonstrated for automotive application, this scheme would require thorough investigation to determine the feasibility of use in hybrid powertrains. If such a scheme is proven to be feasible, it offers the advantages of higher efficiency, lighter weight, and possibly lower cost.

A comparison of component weights for electrical subsystems in series and parallel powertrain configurations is shown in Tables 10-1 and 10-2 for each vehicle class. All data shown are based on a control scheme using step voltage switching combined with field control of the motor except for the first column in each table which is based on a chopper scheme for controlling motor voltage. The two types of electrical control systems have been presented here in the case of the family car solely for relative comparison of weight, volume, and efficiency. This shows that the difference in the total weight of electrical components in the two approaches is quite small compared to the overall family car weight.

The step voltage/field control scheme was chosen for the final analysis of component weights, power requirements, and costs, and was used as the basis for comparing the performance of various classes of vehicles. It is felt that the performance data obtained with this scheme applies approximately to the chopper approach.

10.2.1.1 Series Configuration

Table 10-1 denotes the characteristics of the electrical subsystems (drive motor, motor controller, generator, generator controller, AC rectifier) selected for the baseline series configuration for each of the six vehicle classes. Included in the table are such features as subsystem type, rating (where appropriate), volume, weight, and efficiency at rated load conditions.

As mentioned previously, the final drive for the series configuration is defined as a conventional differential drive unit, adapted to contain an overdrive mechanism for a step-change in gear ratio during high-speed cruise operation for increased drive motor efficiency. The heat engine, of course,

Table 10-1. Baseline Series Configuration Characteristics of Selected Electrical Subsystems

Subsystem \ Vehicle	Family Car		Commuter Car	Delivery/Postal Van		City Bus	
	DC Chopper	Step Voltage & Field Control (3)		Low Speed	High Speed	Low Speed	High Speed
Electric Drive Motor							
Type	DC Series	DC Shunt-Wound	DC Shunt-Wound	DC Series	DC Series	DC Series	DC Compound
Rated Voltage, volts	220	220	220	220	220	440	440
Rated HP, hp	61	61	21	30	80	100	175
Volume, ft^3	4.3	4.80	2.24	2.94	6.75	14.60	14.53
Weight, lb (5)	315	337	133	156	462	816	968
Efficiency @ Rated Load, %	90	92	92	92	94	94	94
Motor Controller							
Volume, ft^3	1.8	0.023	0.023	1.4	1.8	3.0	2.5
Weight, lb	120	12.5	9.5	64	84	135	112
Efficiency @ Rated Load, %	95	99	99+	97.4	97.4	97.7	97.6
Generator (1)							
Type	AC	AC	AC	AC	AC	AC	AC
Maximum-RPM	12,000	12,000	12,000	12,000	12,000	12,000	12,000
Rated Output, kw	54	51	18	28	71	88	154
Volume, ft^3	0.2	0.16	0.09	0.12	0.25	0.30	0.50
Weight, lb (4)	84	80	37	52	103	126	195
Efficiency @ Rated Load, %	90	90	90	90	90	90	90
AC Rectifier							
Volume, ft^3	0.15	0.15	0.05	0.1	0.1	0.1	0.2
Weight, lb	18	18	9	9	9	9	18
Efficiency @ Rated Load, %	99+	99+	99+	99.6	99.6	99.6	99.6
Generator Controller							
Volume, ft^3	0.02	0.02	0.009	0.02	0.02	0.02	0.02
Weight, lb	3	3	1	3	3	3	3
Cables, Low Level Electronics, Accessories, Cooling System & Miscellaneous							
Weight, lb (2)	55	50	38	45	80	120	150

(1) Gear weight accounted for in Tables 10-7 through 10-12.
(2) This weight accounted for as part of vehicle body weight.
(3) This column used for final analysis results (Sections 10 and 11).
(4) Allowing a derating factor of 15% for possible variation in heat engine speed.
(5) Without forced air cooling system.

Table 10-2. Baseline Parallel Configuration Characteristics of Selected Electrical Subsystems

Subsystem \ Vehicle	Family Car		Commuter Car	Delivery/Postal Van		City Bus	
	DC Chopper	Step Voltage & Field Control (3)		Low Speed	High Speed	Low Speed	High Speed
Electric Drive Motor							
Type	DC Series	DC Shunt-Wound	DC Shunt-Wound	DC Series	DC Series	DC Series	DC Compound
Rated Voltage, volts	220	220	220	220	220	440	440
Rated HP, hp	38	38	12	30	30	100	30
Volume, ft^3	3.0	3.4	1.2	2.95	2.95	14.66	2.95
Weight, lb (5)	232	250	83	170	170	831	170
Efficiency @ Rated Load, %	90	92	92	92	94	94	94
Motor Controller							
Volume, ft^3	1.5	0.023	0.023	1.4	1.4	3.0	1.4
Weight, lb	100	12.5	9.5	64	64	135	64
Efficiency @ Rated Load, %	95	99+	99+	97.4	97.4	97.7	97.4
Generator (1)							
Type	AC	AC	AC	AC	AC	AC	AC
Maximum RPM	12,000	12,000	12,000	12,000	12,000	12,000	12,000
Rated Output, kw	8.1	7.5	4.5	13	13	63	38
Volume, ft^3	0.08	0.07	0.06	0.08	0.08	0.19	0.14
Weight, lb (4)	19	18	12	27	27	95	65
Efficiency @ Rated Load, %	90	90	90	90	90	90	90
AC Rectifier							
Volume, ft^3	0.1	0.1	0.05	0.1	0.1	0.1	0.1
Weight, lb	9	9	5	9	9	9	9
Efficiency @ Rated Load, %	99+	99+	99+	99.6	99.6	99.6	99.6
Generator Controller							
Volume, ft^3	0.009	0.009	0.009	0.009	0.009	0.009	0.009
Weight, lb	2	2	2	1	1	1	1
Cables, Low Level Electronics, Accessories, Cooling System & Miscellaneous							
Weight, lb (2)	55	50	38	45	80	120	150

(1) Gear weight accounted for in Tables 10-13 through 10-18.
(2) This weight accounted for as part of vehicle body weight.
(3) This column used for final analysis results (Sections 10 and 11).
(4) Allowing a derating factor of 15% for possible variation in heat engine speed.
(5) Without forced air cooling system.

can be any one of the five classes under examination in the present study (i.e., S.I. engine, diesel, gas turbine, Rankine, Stirling). A small gearbox (speeder or reducer) is utilized between the heat engine and generator to produce the desired speed ratio between these subsystems.

The required batteries, in terms of power density and energy density, were not treated in this portion of the study effort except on the basis of power-plant weight available for battery use. Rather, the battery requirements were determined with the use of the computer program and the various driving cycles for each vehicle (See Section 11).

10.2.1.2 Parallel Configuration

Electrical subsystems with characteristics similar to those of Table 10-1 were also considered to be applicable to the parallel configuration, except for rated size, weight, and volume changes necessitated by the sizing requirements of the parallel mode of operation. Table 10-2 denotes the electrical subsystem characteristics selected for the baseline parallel configuration.

The same comments as to batteries, control system, generator, heat engine, speeder/reducer, and final drive that were made for the series configuration apply to the parallel configuration. In addition, a conventional automatic transmission was assumed in the driveline between the heat engine and the final drive unit.

10.2.2 Sizing Criteria

10.2.2.1 Series Configuration

The essential sizing criteria and significant operational efficiencies assumed for the baseline series configuration are shown in Table 10-3.

With regard to electric drive motor sizing, the family car, commuter car, low-speed van, and low-speed city bus were all sized for the continuous rated or 100 percent load condition to occur at the grade power and velocity conditions. The high-speed van and high-speed bus were conversely sized at the maximum cruise velocity power level.

Table 10-3. Baseline Series Configuration Subsystem Sizing Criteria

Sizing Criteria / Vehicle	Family Car		Commuter Car	Delivery/Postal Van		City Bus	
	DC Chopper	*Step Voltage & Field Control		Low Speed	High Speed	Low Speed	High Speed
Vehicle Specification Requirements							
Maximum Cruise Speed (V_{max}), mph	80	80	70	40	65	40	60
Velocity on Grade, mph @ %	40 @ 12	40 @ 12	33 @ 12	8 @ 20	-	6 @ 20	-
Road HP @ V_{max}, hp	58	58	20	24	80	70	170
Road HP @ V_{grade}, hp	61	61	21	30	-	100	-
Selected Baseline Subsystem Efficiencies for Design-Point Sizing							
Final Drive (Differential), %	95	95	95	95	95	95	95
Electric Drive Motor, %	90	90	90	90	90	90	90
Control System, %	95	99.5	99.5	99.5	99.5	99.5	99.5
Generator, %	90	90	90	90	90	90	90
Accessory Power Requirements							
All Accessories, hp	12.6	12.6	5.7	2.3	2.3	39.3	39.3
No Air Conditioning, hp	6.7	6.7	1.7	2.3	2.3	12.3	12.3
Maximum Heat Engine Power Output Required, hp	100	93	33	42	107	170	260
Selected Baseline Subsystem Efficiencies for Part-Load Operation During Emission Driving Cycles							
Final Drive (Differential), %	95	95	95	95	95	95	95
Electric Drive Motor, %	80	80	80	80	80	80	80
Control System, %	95	99.5	99.5	99.5	99.5	99.5	99.5
Generator, %	80	80	80	80	80	80	80

*This column used for final analysis results (Sections 10 and 11)

The various subsystem efficiencies used in determining the power output required from the heat engine (at the aforementioned design sizing points) are as shown in the table.

The accessory power requirements (previously given in Section 3) are repeated for clarity.

Thus the maximum heat engine power output shown in the table for each vehicle is the power required at the output of the heat engine to enable the drive motor to perform at the rated condition shown (grade or maximum velocity point) with the intervening subsystem efficiencies indicated, and to provide the maximum accessory power load.

Also shown in the table are those subsystem efficiencies assumed for the part-load operation of the series configuration under the various emission driving cycles.

10.2.2.2 Parallel Configuration

Table 10-4 contains similar assumptions for the baseline parallel configuration defined previously.

While the design points (grade or maximum velocity condition) are the same for each vehicle as in the series configuration, here it is the automatic transmission and final drive unit (and their efficiencies) which combine with accessory power requirements to define the maximum power output required of the heat engine. In either case, grade or maximum velocity sizing point, the heat engine is providing all required road power to the wheels mechanically. As no acceleration is involved, the motor is not under load, and battery and power conditioning control system are inactive.

For purposes of powerplant weight determination, the electric drive motor was assumed to have a continuous duty rated power level equal to one-third of the maximum power required from the motor during vehicle maximum acceleration. This was based on the criteria of Section 6 which indicated the drive motor was capable of 300 percent overload for short periods of time, such as those which occur during vehicle accelerations. The generator

Table 10-4. Baseline Parallel Configuration Subsystem Sizing Criteria

Sizing Criteria \ Vehicle	Family Car		Commuter Car	Delivery/Postal Van		City Bus	
	DC Chopper	*Step Voltage & Field Control		Low Speed	High Speed	Low Speed	High Speed
Vehicle Specification Requirements							
Maximum Cruise Speed (V_{max}), mph	80	80	70	40	65	40	60
Velocity on Grade, mph @ %	40 @ 12	40 @ 12	33 @ 12	8 @ 20	–	6 @ 20	–
Road HP @ V_{max}, hp	58	58	20	24	80	70	170
Road HP @ V_{grade}, hp	61	61	21	30	–	100	–
Selected Baseline Subsystem Efficiencies for Design-Point Sizing							
Final Drive (Differential), %	95	95	95	95	95	95	95
Automatic Transmission, %	90	90	90	90	90	90	90
Electric Drive Motor (Torquer), %	90	90	90	90	90	90	90
Control System, %	97	99.5	99.5	99.5	99.5	99.5	99.5
Generator, %	90	90	90	90	90	90	90
Accessory Power Requirements							
All Accessories, hp	12.6	12.6	5.7	2.3	2.3	39.3	39.3
No Air Conditioning, hp	6.7	6.7	1.7	2.3	2.3	12.3	12.3
Maximum Heat Engine Power Output Required, hp	84	84	31	38	96	156	240
Selected Baseline Subsystem Efficiencies for Part-Load Operation During Emission Driving Cycles							
Final Drive (Differential	95	95	95	95	95	95	95
Automatic Transmission	90	90	90	90	90	90	90
Electric Drive Motor (Torquer)	80	80	80	80	80	80	80
Control System	97	99.5	99.5	99.5	99.5	99.5	99.5
Generator	80	80	80	80	80	80	80

*This column used for final analysis results (Sections 10 and 11) was assumed to have a continuous duty power rating equal to the minimum power level of the heat engine, as defined in Section 10.1.3.2.

As can be noted by comparing Tables 10-3 and 10-4, the power train drive-line efficiency (from heat engine to drive wheels) at maximum continuous power demand conditions is higher for the parallel configuration than for the series configuration. Consequently, the heat engine size (HP) for the parallel configuration is smaller than for the series configuration in the order of 6 to 10 percent.

10.2.3 Powerplant Weight Analyses

10.2.3.1 Powerplant Elements

The APCO has defined (under their Advanced Automotive Power Systems Program) the powerplant or propulsion system weight (W_p) to include the energy storage unit (including containment), power converter (including both functional components and controls), and power transmission to the driven wheels. It also includes the exhaust system, pumps, motors, and fans necessary for operation of the propulsion system, as well as any propulsion system heating or cooling devices.

Based upon this definition, and adopting elements of nomenclature and weight apportionment after Hoffman (Refs. 10-1 and 10-2), the weight apportionment for conventional and hybrid vehicles of the personal transit type are shown in Table 10-5.

As can be seen, the vehicle weight without propulsion (W_o) includes tires, wheels, and brakes and equals 67.3 percent of the vehicle curb weight (W_c), where

$$W_c = W_o + W_p$$

Also, the powerplant or propulsion system weight (W_p) is then 32.7 percent of the curb weight for the conventional vehicle.

Table 10-5. Weight Apportionment in Conventional and Hybrid Vehicles

Component	Conventional Vehicle Component Weight/ Curb Weight	Hybrid Vehicle Component Weight/ Curb Weight
Vehicle Weight (No Propulsion), W_o		
Body	0.330	0.330
Trim	0.140	0.140
Glass	0.032	0.032
Suspension	0.060	0.060
Steering	0.016	0.016
Tires	0.032	0.032
Wheels	0.025	0.025
Brakes	0.038	0.038
Vehicle Weight/Component Weight, W_o/W_c	0.673	0.673
Power Train, W_p		
Heat Engine	0.150	A*
Fluid Systems		
Radiator (Full)	0.014	B
Fuel Tank (Full)	0.044	0.044
Exhaust	0.014	C
Electrical		
Battery	0.012	D
Generator and Controls	0.005	E
Starter	0.005	F
Transmission	0.040	G
Drive Line	0.020	H
Rear Axle Drive	0.023	0.023
Electric Drive Motor	0	I
Motor Controller	0	J
AC Rectifier	0	K
Gearing (H. E. to Generator)	0	L
Power Train Weight/Component Weight, W_p/W_c	0.327	0.067 Plus A-L

*See Table 10-6 for data applicable to "A" through "L".

The APCO has further defined a vehicle test weight (W_t) as

$$W_t = W_o + W_p + 300 \text{ lb}$$

The term W_t is the vehicle weight at which all accelerative maneuvers, fuel economy, and emissions are to be calculated by participants in their Advanced Automotive Power Systems Program.

Based upon these definitions and using the 4000-lb family car of the present study as an example, the 4000-lb weight corresponds to W_t. Subtracting the 300-lb allowance for passengers and/or baggage implies a 3700-lb curb weight and a 1210-lb weight allowance for the powerplant (W_p), based on the 32.7 percent allowance of curb weight.

Referring to the vehicle specifications previously outlined in Section 3, which were given as study guidelines prior to the current powerplant weight guidelines stated above, it can be seen that a 500-lb weight allowance was made for passengers and/or baggage (3500-lb curb weight) and that a 1500-lb allowance is stipulated for the powerplant. Therefore, it is apparent that the powerplant weight allocations given as guidelines for the present study are at variance with current APCO powerplant weight criteria. The recognition of this deviation between initial study guidelines and current APCO criteria came too late in the program to adjust the study guidelines. Consequently, the powerplant weights (less batteries) developed in the remainder of this section have been compared to the 1500-lb weight allowance required by the original vehicle specifications as stated in Section 3. This comparison was made to determine the amount of weight available for batteries (in any given vehicle and powerplant combination) and to initially assess the impact of such allowable battery weight on battery power density and energy density requirements.

However, in Section 11, parametric displays of the effect of powerplant weight allocation on battery power density are presented which afford the opportunity to observe the effect of changing the 1500-lb powerplant weight allocation of the family car (as an example) to the 1210-lb value mentioned previously, or any other reasonable value.

10.2.3.2 Scaling Assumptions

Again, referring to Table 10-5, the weight allocations for the various power train subsystems/components of the hybrid vehicle are represented by letters (A through L) except for the fuel tank and rear axle drive. It should be noted that the total power train system includes elements not often specifically considered (full fuel tank, full radiator, exhaust system, etc.). Even though departing from the current APCO criteria that power train weight is constrained to 32.7 percent of the vehicle curb weight, it was found extremely useful to adhere to various conventional-vehicle component weight characteristics developed by Hoffman (as shown in Table 10-5) for some components and also to use these characteristics as a basis of weight-scaling for other components.

In the present study, fuel tank (full) weights were maintained at 0.44 W_c for the family car, commuter car, and van. In the case of the bus, a 95-gallon tank was provided per current intracity buses.

The rear axle drive weight was calculated as 0.023 W_c for the family car and commuter car. For the van and bus, the family car rear axle drive weights were increased by the ratio of vehicle maximum acceleration power demand divided by family car maximum acceleration power demand.

The remaining powerplant subsystem/component weights were either the result of calculations performed for the hybrid study or were based upon conventional family car weight allocations modified by suitable power ratios, as above. The specific scaling/computational techniques are illustrated in Table 10-6.

Table 10-6. Weight Scaling/Computational Techniques (See Table 10-4)

A = Calculated heat engine weight/curb weight

B = 0 for gas turbine, Rankine, Stirling systems (engine weight includes radiators)

$$= 0.014 \times \frac{\text{hybrid vehicle heat engine rated hp}}{\text{conventional vehicle heat engine rated hp}^{*}}$$

(for S. I. engine and diesel)

$$C = 0.014 \times \frac{\text{hybrid vehicle heat engine rated hp}}{\text{conventional vehicle heat engine rated hp}}$$

(for all heat engines)

D = Calculated battery weight/curb weight

E = Calculated generator and control weight/curb weight

F = 0 (if generator can also be used to start heat engine)

$$= 0.005 \times \frac{\text{hybrid vehicle heat engine rated hp}}{\text{conventional vehicle heat engine rated hp}}$$

G = 0 (in series mode)

$$= 0.02 \times \frac{\text{hybrid vehicle heat engine rated hp}}{\text{conventional vehicle heat engine rated hp}} \quad \text{(in parallel mode)}$$

H = 0 (in series mode)

$$= 0.02 \times \frac{\text{hybrid vehicle heat engine rated hp}}{\text{conventional vehicle heat engine rated hp}} \quad \text{(in parallel mode)}$$

I = Calculated drive motor weight/curb weight

J = Calculated controller weight/curb weight

K = Calculated AC rectifier weight/curb weight

L = Calculated gearing weight/curb weight

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*Conventional vehicle heat engine rated hp = vehicle peak hp demand/0.75

10.2.3.3 Results

Using the powerplant element weight scaling techniques delineated above (Section 10.2.3.2 and Table 10-6) and the subsystem sizing criteria previously defined (Section 10.2.1 and Tables 10-1, 10-2, 10-3, and 10-4), the powerplant weight estimations for the various vehicle classes are shown in Tables 10-7 through 10-12 for the series configuration, and Tables 10-13 through 10-18 for the parallel configuration. It should be noted that the diesel system weights are based on the use of a divided chamber turbocharged engine as described in Section 8.

These results are summarized in Table 10-19 in terms of powerplant weights and volumes (less batteries) and weight and volume allowances for batteries under the vehicle specification criteria of Section 3.

10.2.3.3.1 Family Car

Only the S. I. engine and the gas turbine heat engines result in meaningful weight allocations for batteries. For these cases, the parallel configuration is lighter in weight than the series and allows 62 to 73 more pounds for batteries.

10.2.3.3.2 Commuter Car

Again, only the S. I. engine and gas turbine result in meaningful weight allocations for batteries (101 to 211 lb) with the parallel configuration allowing the highest battery weights.

10.2.3.3.3 Low-speed Van

The extremely low continuous power requirements of the low-speed van enable all heat engine classes to result in meaningful battery weight allocations. However, in this case, the electric drive motor and heat engine weights are nearly the same in the parallel configuration as in the series configuration. Therefore, the additional driveline and transmission weights of the parallel configuration make it the heaviest, thus being more restrictive in battery weight allocation.

Table 10-7. Preliminary Weight and Volume Summary of Power Train - Family Car Series Mode

Power Train Subsystems \ Heat-Engine Class	S.I. Engine		Diesel		Gas Turbine		Rankine		Stirling	
	Wt.[1]	Vol.[2]	Wt.	Vol.	Wt.	Vol.	Wt.	Vol.	Wt.	Vol.
Electrical Drive Motor	337.0	4.80								
Controller (Motor)	12.5	0.02								
Generator	80.0	0.16								
AC Rectifier	18.0	0.15								
Generator Controller	3.0	0.02								
Fuel Tank (Full)	154.0	3.08								
Rear Axle Drive	80.5	0.48								
Heat Engine	335.0	11.80	493.0	15.10	310.0	10.4	846.0	13.50	1153.0	22.80
Gearing (Heat Engine to Generator)	11.5	0.09	11.5	0.09	11.5	0.09	11.5	0.09	11.5	0.09
Radiator (Full)	29.7	0.39	29.7	0.39	0	0	0	0	0	0
Exhaust	29.7	0.48	29.7	0.48	29.7	0.48	29.7	0.48	29.7	0.48
Starter	10.7	0.09	10.7	0.09	10.7	0.09	0	0	0	0
Sub-Total	1101.6	21.56	1259.6	24.86	1046.9	19.77	1572.2	22.78	1879.2	32.08
Assigned Value	1500.0	28.0	1500.0	28.0	1500.0	28.0	1500.0	28.0	1500.0	28.0
Available for Batteries	398.4	6.44	240.4	3.14	453.1	8.23	0		0	

(1) Weight in lb
(2) Volume in ft^3

Table 10-8. Preliminary Weight and Volume Summary of Power Train - Commuter Car Series Mode

Power Train Subsystems \ Heat-Engine Class	S. I. Engine		Diesel		Gas Turbine		Rankine		Stirling	
	Wt.$^{(1)}$	Vol.$^{(2)}$	Wt.	Vol.	Wt.	Vol.	Wt.	Vol.	Wt.	Vol.
Electrical Drive Motor	133.0	2.24								
Controller (Motor)	9.5	0.02								
Generator	37.0	0.09								
AC Rectifier	9.0	0.05								
Generator Controller	1.0	0.01								
Fuel Tank (Full)	61.6	1.24								
Rear Axle Drive	32.2	0.15								
Heat Engine	180.0	6.03	228.0	8.90	125.0	3.90	322.0	5.50	432.0	8.60
Gearing (Heat Engine to Generator)	4.1	0.03	4.1	0.03	4.1	0.03	4.1	0.03	4.1	0.03
Radiator (Full)	13.3	0.15	13.3	0.15	0	0	0	0	0	0
Exhaust	13.3	0.19	13.3	0.19	13.3	0.19	13.3	0.19	13.3	0.19
Starter	4.8	0.04	4.8	0.04	4.8	0.04	0	0.04	0	0.04
Sub-Total	498.8	10.24	546.8	13.11	430.5	7.96	622.7	9.56	732.7	12.66
Assigned Value	600.0	16.0	600.0	16.0	600.0	16.0	600.0	16.0	600.0	16.0
Available for Batteries	101.2	5.76	53.2	2.89	169.5	8.04	0		0	

(1) Weight in lb
(2) Volume in ft^3

Table 10-9. Preliminary Weight and Volume Summary of Power Train - Low-speed Delivery Van Series Mode

Power Train Subsystems \ Heat-Engine Class	S. I. Engine		Diesel		Gas Turbine		Rankine		Stirling	
	Wt.(1)	Vol.(2)	Wt.	Vol.	Wt.	Vol.	Wt.	Vol.	Wt.	Vol.
Electrical Drive Motor	156.0	2.94								
Controller (Motor)	64.0	1.40								
Generator	52.0	0.12								
AC Rectifier	9.0	0.10								
Generator Controller	3.0	0.02								
Fuel Tank (Full)	198.0	3.98								
Rear Axle Drive	80.5	0.48								
Heat Engine	205.0	7.15	273.0	10.1	155.0	4.75	403.0	6.80	546.0	10.90
Gearing (Heat Engine to Generator)	5.0	0.04	5.0	0.04	5.0	0.04	5.0	0.04	5.0	0.04
Radiator (Full)	13.6	0.18	13.6	0.18	0	0	0	0	0	0
Exhaust	13.6	0.22	13.6	0.22	13.6	0.22	13.6	0.22	13.6	0.22
Starter	4.85	0.04	4.85	0.04	4.85	0.04	0	0	0	0
Sub-Total	804.6	16.67	872.6	19.62	741.0	14.09	984.1	16.10	1127.1	20.20
Assigned Value	1700.0	42.0	1700.0	42.00	1700.0	42.0	1700.0	42.0	1700.0	42.0
Available for Batteries	895.4	25.33	827.4	22.38	959	27.91	715.9	25.90	572.9	21.80

(1) Weight in lb
(2) Volume in ft^3

Table 10-10. Preliminary Weight and Volume Summary of Power Train - High-speed Delivery Van Series Mode

Power Train Subsystems \ Heat-Engine Class	S. I. Engine		Diesel		Gas Turbine		Rankine		Stirling	
	Wt.$^{(1)}$	Vol.$^{(2)}$	Wt.	Vol.	Wt.	Vol.	Wt.	Vol.	Wt.	Vol.
Electrical Drive Motor	462.0	6.75								
Controller (Motor)	84.0	1.30								
Generator	103.0	0.25								
AC Rectifier	9.0	0.10								
Generator Controller	3.0	0.02								
Fuel Tank (Full)	198.0	3.98								
Rear Axle Drive	80.5	0.48								
Heat Engine	357.0	12.80	545.0	16.1	350.0	11.8	963.0	15.3	1305.0	25.7
Gearing (Heat Engine to Generator)	27.0	0.22	27.0	0.22	27.0	0.22	27.0	0.22	27.0	0.22
Radiator (Full)	35.0	0.45	35.0	0.45	0	0	0	0	0	0
Exhaust	35.0	0.56	35.0	0.56	35.0	0.56	35.0	0.56	35.0	0.56
Starter	12.5	0.10	12.5	0.10	12.5	0.10	0	0	0	0
Sub-Total	1406.0	27.51	1594.0	30.81	1364.0	26.06	1964.5	29.46	2306.5	39.86
Assigned Value	1700.0	42.0	1700.0	42.0	1700.0	42.0	1700.0	42.0	1700.0	42.0
Available for Batteries	294	14.49	106	11.19	336	15.94	0		0	

(1) Weight in lb
(2) Volume in ft^3

Table 10-11. Preliminary Weight and Volume Summary of Power Train - Low-speed Intracity Bus Series Mode

Power Train Subsystems / Heat-Engine Class	S. I. Engine		Diesel		Gas Turbine		Rankine		Stirling	
	Wt.(1)	Vol.(2)	Wt.	Vol.	Wt.	Vol.	Wt.	Vol.	Wt.	Vol.
Electrical Drive Motor	816.0	14.60								
Controller (Motor)	135.0	3.00								
Generator	126.0	0.30								
AC Rectifier	9.0	0.10								
Generator Controller	3.0	0.02								
Fuel Tank (Full)	763.0	15.30								
Rear Axle Drive	212.0	1.26								
Heat Engine	478.0	17.30	755.0	20.3	521.0	15.0	1462.0	22.7	1949.0	38.1
Gearing (Heat Engine to Generator)	42.5	0.34	42.5	0.34	42.5	0.34	42.5	0.34	42.5	0.34
Radiator (Full)	55.0	0.71	55.0	0.71	0	0	0	0	0	0
Exhaust	55.0	0.88	55.0	0.88	55.0	0.88	55.0	0.88	55.0	0.88
Starter	19.7	0.17	19.7	0.17	19.7	0.17	0	0	0	0
Sub-Total	2714.2	53.98	2991.2	56.98	2702.2	50.97	3623.5	58.5	4110.5	73.90
Assigned Value	6000.0	175.0	6000.0	175.0	6000.0	175.0	6000.0	175.0	6000.0	175.0
Available for Batteries	3285.8	121.02	3008.8	118.02	3297.8	124.03	2376.5	116.50	1889.5	101.1

(1)Weight in lb
(2)Volume in ft^3

Table 10-12. Preliminary Weight and Volume Summary of Power Train - High-speed Intracity Bus Series Mode

Power Train Subsystems / Heat-Engine Class	S.I. Engine		Diesel		Gas Turbine		Rankine		Stirling	
	Wt.(1)	Vol.(2)	Wt.	Vol.	Wt.	Vol.	Wt.	Vol.	Wt.	Vol.
Electrical Drive Motor	968.0	14.53								
Controller (Motor)	112.0	2.50								
Generator	195.0	0.50								
AC Rectifier	18.0	0.20								
Generator Controller	3.0	0.02								
Fuel Tank (Full)	763.0	15.30								
Rear Axle Drive	141.0	0.84								
Heat Engine	626.0	22.40	1050.0	25.3	744.0	19.8	2218.0	33.9	2793.0	53.0
Gearing (Heat Engine to Generator)	67.0	0.54	67.0	0.54	67.0	0.54	67.0	0.54	67.0	0.54
Radiator (Full)	87.0	1.13	87.0	1.13	0	0	0	0	0	0
Exhaust	87.0	1.43	87.0	1.43	87.0	1.43	87.0	1.43	87.0	1.43
Starter	31.0	0.27	31.0	0.27	31.0	0.27	0	0	0	0
Sub-Total	3098.0	59.66	3522.0	62.56	3129.0	55.93	4572.0	69.76	5147.0	88.86
Assigned Value	6000.0	175.0	6000.0	175.0	6000.0	175.0	6000.0	175.0	6000.0	175.0
Available for Batteries	2902	115.34	2478	112.44	2871	119.07	1428	105.24	853	86.14

(1) Weight in lb
(2) Volume in ft^3

Table 10-13. Preliminary Weight and Volume Summary of Power Train - Family Car Parallel Mode

Power Train Subsystems \ Heat-Engine Class	S.I. Engine		Diesel		Gas Turbine		Rankine		Stirling	
	Wt.(1)	Vol.(2)	Wt.	Vol.	Wt.	Vol.	Wt.	Vol.	Wt.	Vol.
Electrical Drive Motor	250.0	3.40								
Controller (Motor)	12.5	0.02								
Generator	18.0	0.07								
AC Rectifier	9.0	0.10								
Generator Controller	2.0	0.01								
Fuel Tank (Full)	154.0	3.08								
Rear Axle Drive	80.5	0.48								
Heat Engine	319.0	10.90	445.0	14.30	280.0	8.65	755.0	12.20	1025.0	21.00
Gearing (Heat Engine to Generator)	2.0	0.02	2.0	0.02	2.0	0.02	2.0	0.02	2.0	0.02
Radiator (Full)	27.1	0.36	27.1	0.36	0	0	0	0	0	0
Exhaust	27.1	0.44	27.1	0.44	27.1	0.44	27.1	0.44	27.1	0.44
Starter	10.0	0.08	10.0	0.08	10.0	0.08	0	0	0	0
Transmission	59.0	0.42	59.0	0.42	59.0	0.42	59.0	0.42	59.0	0.42
Drive Line	70.0	0.15	70.0	0.15	70.0	0.15	70.0	0.15	70.0	0.15
Sub-Total	1040.2	19.53	1166.2	22.93	974.1	16.92	1439.1	20.39	1709.1	29.19
Assigned Value	1500.0	28.0	1500.0	28.0	1500.0	28.0	1500.0	28.0	1500.0	28.0
Available for Batteries	459.8	8.47	333.8	5.07	525.9	11.08	60.9	7.61	0	

(1) Weight in lb
(2) Volume in ft^3

Table 10-14. Preliminary Weight and Volume Summary of **Power** Train - Commuter Car Parallel Mode

Power Train Subsystems \ Heat-Engine Class	S.I. Engine		Diesel		Gas Turbine		Rankine		Stirling	
	Wt.$^{(1)}$	Vol.(2)	Wt.	Vol.	Wt.	Vol.	Wt.	Vol.	Wt.	Vol.
Electrical Drive Motor	83.0	1.20								
Controller (Motor)	9.5	0.02								
Generator	12.0	0.06								
AC Rectifier	5.0	0.05								
Generator Controller	2.0	0.01								
Fuel Tank (Full)	61.6	1.24								
Rear Axle Drive	32.2	0.15								
Heat Engine	171.0	5.70	217.0	8.70	117.0	3.60	300.0	5.30	390.0	7.80
Gearing (Heat Engine to Generator)	1.5	0.01	1.5	0.01	1.5	0.01	1.5	0.01	1.5	0.01
Radiator (Full)	12.1	0.14	12.1	0.14	0	0	0	0	0	0
Exhaust	12.1	0.17	12.1	0.17	12.1	0.17	12.1	0.17	12.1	0.17
Starter	4.3	0.04	4.3	0.04	4.3	0.04	0	0	0	0
Transmission	21.0	0.15	21.0	0.15	21.0	0.15	21.0	0.15	21.0	0.15
Drive Line	28.0	0.05	28.0	0.05	28.0	0.05	28.0	0.05	28.0	0.05
Sub-Total	455.3	8.99	501.3	11.99	389.2	6.75	567.9	8.41	675.9	10.91
Assigned Value	600.0	16.0	600.0	16.0	600.0	16.0	600.0	16.0	600.0	16.0
Available for Batteries	144.7	7.01	98.7	4.01	210.8	9.25	32.1	7.59	0	

(1) Weight in lb
(2) Volume in ft^3

Table 10-15. Preliminary Weight and Volume Summary of Power Train - Low-speed Delivery Van Parallel Mode

Power Train Subsystems \ Heat-Engine Class	S. I. Engine		Diesel		Gas Turbine		Rankine		Stirling	
	Wt.(1)	Vol.(2)	Wt.	Vol.	Wt.	Vol.	Wt.	Vol.	Wt.	Vol.
Electrical Drive Motor	170.0	2.95								
Controller (Motor)	64.0	1.40								
Generator	27.0	0.08								
AC Rectifier	9.0	0.10								
Generator Controller	1.0	0.01								
Fuel Tank (Full)	198.0	3.98								
Rear Axle Drive	80.5	0.48								
Heat Engine	198.0	6.65	255.0	9.50	143.0	4.40	360.0	6.1	494.0	10.00
Gearing (Heat Engine to Generator)	3.0	0.03	3.3	0.03	3.3	0.03	3.3	0.03	3.3	0.03
Radiator (Full)	12.3	0.16	12.3	0.16	0	0	0	0	0	0
Exhaust	12.3	0.20	12.3	0.20	12.3	0.20	12.3	0.20	12.3	0.20
Starter	4.4	0.04	4.4	0.04	4.4	0.04	0	0	0	0
Transmission	26.6	0.20	26.6	0.20	26.6	0.20	26.6	0.20	26.6	0.20
Drive Line	70.0	0.07	70.0	0.07	70.0	0.07	70.0	0.07	70.0	0.07
Sub-Total	876.4	16.35	933.4	19.2	809.1	13.94	1021.7	15.60	1155.7	19.50
Assigned Value	1700.0	42.0	1700.0	42.0	1700.0	42.0	1700.0	42.0	1700.0	42.0
Available for Batteries	823.6	25.65	766.6	22.80	890.9	28.06	678.3	26.40	544.3	22.50

(1) Weight in lb
(2) Volume in ft^3

Table 10-16. Preliminary Weight and Volume Summary of Power Train - High-speed Delivery Van Parallel Mode

Power Train Subsystems \ Heat-Engine Class	S. I. Engine		Diesel		Gas Turbine		Rankine		Stirling	
	Wt.[1]	Vol.[2]	Wt.	Vol.	Wt.	Vol.	Wt.	Vol.	Wt.	Vol.
Electrical Drive Motor	170.0	2.95								
Controller (Motor)	64.0	1.40								
Generator	27.0	0.08								
AC Rectifier	9.0	0.10								
Generator Controller	1.0	0.01								
Fuel Tank (Full)	198.0	3.98								
Rear Axle Drive	80.5	0.48								
Heat Engine	345.0	12.00	500.0	15.50	315.0	9.60	855.0	14.40	1150.0	23.70
Gearing (Heat Engine to Generator)	3.3	0.03	3.3	0.03	3.3	0.03	3.3	0.03	3.3	0.03
Radiator (Full)	31.1	0.40	31.1	0.40	0	0	0	0	0	0
Exhaust	31.1	0.50	31.1	0.50	31.1	0.50	31.1	0.50	31.1	0.50
Starter	11.1	0.09	11.1	0.09	11.1	0.09	0	0	0	0
Transmission	67.0	0.48	67.0	0.48	67.0	0.48	67.0	0.48	67.0	0.48
Drive Line	70.0	0.17	70.0	0.17	70.0	0.17	70.0	0.17	70.0	0.17
Sub-Total	1108.1	22.67	1263.1	26.17	1047.0	19.87	1575.9	24.58	1870.9	33.88
Assigned Value	1700.0	42.0	1700.0	42.0	1700.0	42.0	1700.0	42.0	1700.0	42.0
Available for Batteries	591.9	19.33	436.9	15.83	653.0	22.13	124.1	17.42	0	8.12

[1] Weight in lb
[2] Volume in ft^3

Table 10-17. Preliminary Weight and Volume Summary of Power Train - Low-speed Intracity Bus Parallel Mode

Heat-Engine Class / Power Train Subsystems	S.I. Engine		Diesel		Gas Turbine		Rankine		Stirling	
	Wt.[1]	Vol.[2]	Wt.	Vol.	Wt.	Vol.	Wt.	Vol.	Wt.	Vol.
Electrical Drive Motor	831.0	14.66								
Controller (Motor)	135.0	3.00								
Generator	95.0	0.19								
AC Rectifier	9.0	0.10								
Generator Controller	1.0	0.01								
Fuel Tank (Full)	763.0	15.30								
Rear Axle Drive	212.0	1.26								
Heat Engine	468.0	16.40	716.0	19.50	485.0	14.00	1350.0	21.00	2025.0	36.00
Gearing (Heat Engine to Generator)	15.7	0.13	15.7	0.13	15.7	0.13	15.7	0.13	15.7	0.13
Radiator (Full)	50.5	0.65	50.5	0.65	0	0	0	0	0	0
Exhaust	50.5	0.81	50.5	0.81	50.5	0.81	50.5	0.81	50.5	0.81
Starter	18.1	0.16	18.1	0.16	18.1	0.16	-		-	
Transmission	110.0	0.78	110.0	0.78	110.0	0.78	110.0	0.78	110.0	0.78
Drive Line	184.0	0.28	184.0	0.28	184.0	0.28	184.0	0.28	184.0	0.28
Sub-Total	2942.8	53.73	3190.8	56.83	2909.3	50.68	3756.2	57.52	4431.2	72.52
Assigned Value	6000.0	175.0	6000.0	175.0	6000.0	175.0	6000.0	175.0	6000.0	175.0
Available for Batteries	3057.2	121.27	2809.2	118.17	3090.7	124.32	2243.8	117.48	1568.8	102.48

(1) Weight in lb
(2) Volume in ft^3

Table 10-18. Preliminary Weight and Volume Summary of Power Train - High-speed Intracity Bus Parallel Mode

Power Train Subsystems \ Heat-Engine Class	S.I. Engine		Diesel		Gas Turbine		Rankine		Stirling	
	Wt.[1]	Vol.[2]	Wt.	Vol.	Wt.	Vol.	Wt.	Vol.	Wt.	Vol.
Electrical Drive Motor	170.0	2.95								
Controller (Motor)	64.0	1.40								
Generator	65.0	0.14								
AC Rectifier	9.0	0.10								
Generator Controller	1.0	0.01								
Fuel Tank (Full)	750.0	15.30								
Rear Axle Drive	208.0	0.84								
Heat Engine	614.0	21.80	985.0	24.0	710.0	18.30	2000.0	30.00	2700.0	51.50
Gearing (Heat Engine to Generator)	9.5	0.08	9.5	0.08	9.5	0.08	9.5	0.08	9.5	0.08
Radiator (Full)	79.5	1.03	79.5	1.03	0	0	0	0	0	0
Exhaust	79.5	1.31	79.5	1.31	79.5	1.31	79.5	1.31	79.5	1.31
Starter	28.4	0.25	28.4	0.25	28.4	0.25	0	0	0	0
Transmission	172.0	1.23	172.0	1.23	172.0	1.23	172.0	1.23	172.0	1.23
Drive Line	184.0	0.45	184.0	0.45	184.0	0.45	184.0	0.45	184.0	0.45
Sub-Total	2433.9	46.79	2804.9	49.09	2450.4	42.36	3712.0	53.81	4412.0	75.31
Assigned Value	6000.0	175.0	6000.0	175.0	6000.0	175.0	6000.0	175.0	6000.0	175.0
Available for Batteries	3566.1	128.21	3195.1	125.91	3549.6	132.64	2288.0	121.19	1588.0	99.69

[1] Weight in lb
[2] Volume in ft^3

Table 10-19. Summary of Powerplant Weights and Effects

Vehicle/Characteristic \ Heat Engine Class/Mode	S.I. Engine		Diesel		Gas Turbine		Rankine		Stirling	
	S	P	S	P	S	P	S	P	S	P
Family Car										
Powerplant Weight (Less Batteries), lb	1102	1040	1260	1166	1047	974	1572	1439	1879	1709
Powerplant Volume (Less Batteries), ft^3	21.6	19.5	24.9	22.9	19.8	16.9	22.8	20.4	32.1	29.2
Weight Available for Batteries, lb	398	460	240	334	453	526	0	61	0	0
Volume Available for Batteries, ft^3	6.4	8.5	3.1	5.1	8.2	11.1	-	7.6	0	-
Commuter Car										
Powerplant Weight (Less Batteries), lb	499	455	547	501	430	389	623	568	733	658
Powerplant Volume (Less Batteries), ft^3	10.2	9.0	13.1	12.0	8.0	6.8	9.6	8.4	12.6	10.9
Weight Available for Batteries, lb	101	145	53	99	170	211	0	32	0	0
Volume Available for Batteries, ft^3	5.8	7.0	2.9	4.0	8.0	9.2	-	7.6	-	-
Low-Speed Van										
Powerplant Weight (Less Batteries), lb	805	876	873	933	741	809	984	1022	1127	1156
Powerplant Volume (Less Batteries), ft^3	16.7	16.4	19.6	19.2	14.0	13.9	16.1	15.6	20.2	19.5
Weight Available for Batteries, lb	895	824	827	767	959	891	716	678	573	544
Volume Available for Batteries, ft^3	25.3	25.6	22.4	22.8	27.9	28.1	25.0	26.4	21.8	22.5
High-Speed Van										
Powerplant Weight (Less Batteries), lb	1406	1108	1594	1263	1364	1047	1965	1576	2307	1871
Powerplant Volume (Less Batteries), ft^3	27.1	22.7	30.4	26.7	25.6	19.9	29.0	24.6	39.4	33.9
Weight Available for Batteries, lb	294	592	106	437	336	653	0	124	0	0
Volume Available for Batteries, ft^3	14.9	19.3	11.6	15.8	16.4	22.1	-	17.4	-	-
Low-Speed Bus										
Powerplant Weight (Less Batteries), lb	2714	2943	2991	3191	2702	2909	3624	3756	4111	4431
Powerplant Volume (Less Batteries), ft^3	53.9	53.7	57.0	56.8	51.0	50.7	53.7	57.5	74.0	72.5
Weight Available for Batteries, lb	3286	3057	3009	2809	3298	3091	2648	2244	1888	1569
Volume Available for Batteries, ft^3	121.0	121.3	118	118.2	124.0	124.3	121.3	117.5	101.0	102.5
High-Speed Bus										
Powerplant Weight (Less Batteries), lb	3098	2434	3522	2805	3129	2450	3964	3712	5147	4412
Powerplant Volume (Less Batteries), ft^3	59.7	46.8	62.6	49.1	55.9	42.4	62.9	53.8	89.9	75.3
Weight Available for Batteries, lb	2902	3566	2478	3195	2871	3550	2036	2288	853	1588
Volume Available for Batteries, ft^3	115.3	128.2	112.4	125.9	119.1	132.6	112.1	121.2	86.1	99.7

10.2.3.3.4 High-speed Van

The higher continuous power requirements of the high-speed van again indicate that only the S. I. engine and the gas turbine systems afford meaningful battery weight allocations. The parallel configuration again is the lightest and allows more battery weight. Although the parallel configuration of the diesel and Rankine systems shows some battery weight allowance, they are definitely inferior to the S. I. engine and gas turbine in this respect.

10.2.3.3.5 Low-speed Bus

All heat engine classes result in substantial weight allocations for batteries, although the Stirling engine system is definitely inferior to the other classes. As in the case of the low-speed van, the parallel configuration is heavier in weight than its series counterpart, with a lower battery weight allocation.

10.2.3.3.6 High-speed City Bus

Again, all heat engine classes indicate meaningful battery weight allocations. The parallel configuration results in lighter powerplant weights (less batteries) in all cases.

The foregoing remarks are, of course, made with reference to the baseline powerplant weight and volume allocations for each vehicle as specified in Section 3 (and indicated on Tables 10-7 through 10-18). The effect of the resulting battery weight and volume allocations of Table 10-19 will be discussed further in Section 11 with regard to battery power-density and energy-density requirements.

10.3 SUMMARY

The conceptual design analyses and vehicle powerplant weight determinations have resulted in baseline series and parallel powerplant configurations for further analysis as to their relative value in terms of (a) vehicle emissions characteristics and (b) battery design goals and characteristics.

In the process of configuration selection, several important differences between the series and parallel configurational approaches were noted which will be further elaborated on in Section 11.

The first such difference is that the parallel configuration has superior high-speed cruise efficiency due to the direct mechanical transmission of power from the heat engine to the drive wheels at this operating condition. As previously noted, this reduces the heat engine size in the order of 6 to 10 percent. This higher high-speed cruise efficiency should also result in better fuel economy.

A second difference is that, in most cases, the parallel configuration results in a lighter powerplant weight (less batteries) which allows more battery weight for the same total powerplant (including batteries) installation weight. This lighter weight system results from the 6 to 10 percent smaller heat engine size, a reduction in electric drive motor weight, reduced generator and generator gearbox weights, and ancillary system weight reductions afforded by the above (i.e., radiator, etc.). These weight reductions offset the weight additions of the transmission and main driveline (heat engine to transmission to differential).

As noted previously, however, the specific vehicle classes of low-speed van and low-speed city bus show a higher powerplant weight (less batteries) for the parallel configuration than the series configuration. This is brought about by the fact that, regardless of whether series or parallel configuration is used, the low-speed design conditions result in very similar weights for the drive motor and heat engine. Thus, the weight of the additional transmission and driveline in the parallel configuration makes it a heavier installation for these two vehicle classes.

As opposed to the advantages described above for the parallel configuration, it should be mentioned that the series mode offers greater simplicity and flexibility in powerplant/vehicle design, and is more amenable to conversion to an all-electric powerplant system at a future date.

10.4 REFERENCES

10-1. G. A. Hoffman, Hybrid Power Systems for Vehicles, University of California at Los Angeles.

10-2. G. A. Hoffman, Automobiles - Today and Tomorrow, RAND Corporation, Report RM-2922-FF, November 1962.

SECTION 11

SUMMARY OF RESULTS

CONTENTS

CONTENTS (Continued)

TABLES

FIGURES

FIGURES (Continued)

FIGURES (Continued)

FIGURES (Continued)

FIGURES (Continued)

FIGURES (Continued)

FIGURES (Continued)

FIGURES (Concluded)

SUMMARY OF RESULTS

11.1 GENERAL STUDY RESULTS

This section is designed to summarize all of the computational results conducted under this study and to offer an interpretation of those results which can be used to help direct future APCO research and development programs associated with hybrid heat engine/electric vehicles. With so many different types of vehicle/configuration/heat engine combinations, it is difficult to highlight every result shown in the body of the report; however, the most important results for each vehicle class are delineated in the following subsections.

It should be recognized that the calculated vehicle exhaust emission results are based on measured engine exhaust emission data compiled in this study. The engine exhaust emission magnitudes and trends were established on the basis of a comprehensive survey and evaluation of the best data from both the open literature and current available unpublished engine data sources. However, it was found that very little emission data were available for the hybrid type of operation and especially for part-load engine operating conditions and for the cold start requirement consistent with the 1972 Federal Test Procedure. The resulting data are considered suitable for use in an initial feasibility study as conducted under this contract. However, in further detailed design studies, a substantial increase in the data base would be necessary for powertrain optimization. The current study data base is fully discussed in Appendix B.

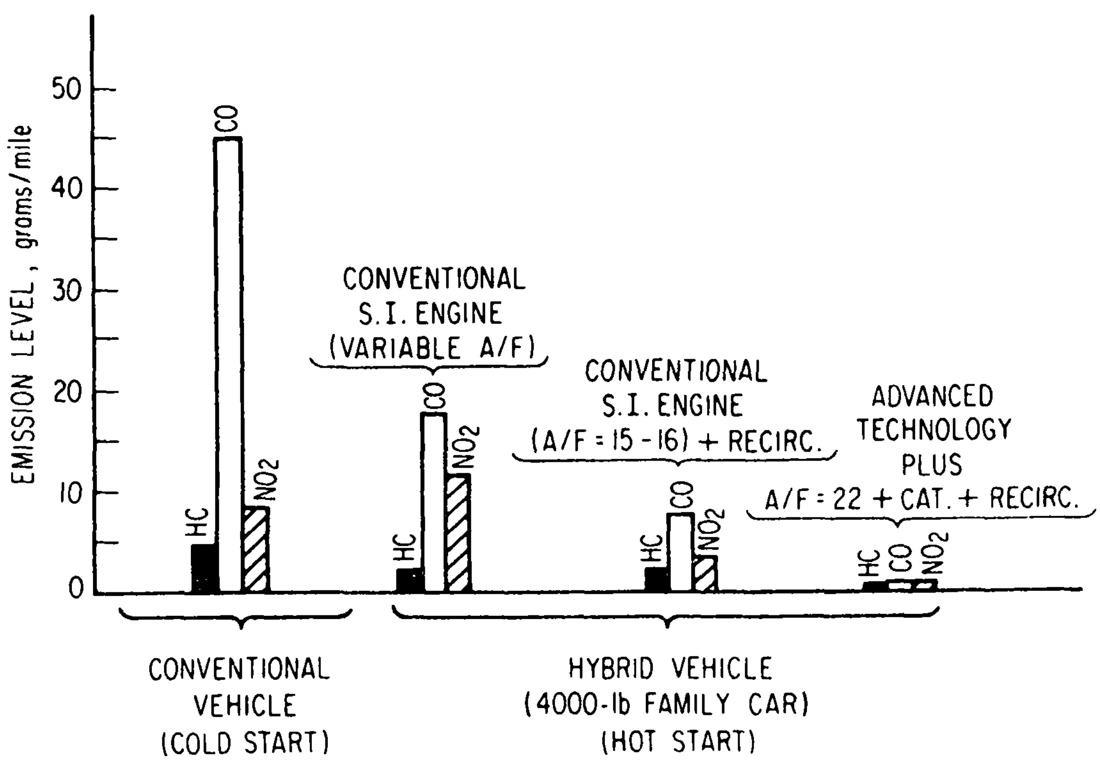
In addition to reflecting the engine emissions data base, the study results also reflect the use of selected battery models. The charge-discharge characteristics for lead-acid, nickel-cadmium, and nickel-zinc batteries were based on available data but modified on the basis of projections for future near-term capability. These battery models are discussed in Section 7.3 of the report.

11.1.1 Family Car and Commuter Car

The following observations can be made about these classes of vehicles:

a. For the available powertrain weight and volume and vehicle performance specified for this study, only the spark ignition internal combustion engine (both reciprocating and rotary) and the gas turbine engines can be practically packaged into the hybrid heat engine/electric vehicle. These engines impose realistically achievable goals on the battery specifications for power and energy density.

b. All hybrids examined showed marked calculated emission reductions over current conventional vehicles. This is illustrated by the results below, where measured cold start emission data available for a 1970 conventional spark ignition engine automobile is compared with calculated hot start emission levels for several development stages of a spark ignition engine in a hybrid powertrain automobile.



In the first emissions comparison, a small conventional engine is used in the hybrid vehicle; the second comparison is for the same engine but operating over the restricted air/fuel ratio range noted and with exhaust recirculation; the third comparison is for an advanced technology engine operating at very high air/fuel ratio with exhaust gas recirculation and incorporating catalytic converters.

c. Based on analysis, if currently available engine technology is used, no version of the family car could meet 1975/76 emission standards. No catalytic converters or thermal reactors were added to the powertrain for this case.

d. Calculations based on hot start with advanced engine technology indicate that all versions could meet 1975/76 standards except for the NO_2 excess for the spark ignition family car version (discussed in item f) and the NO_2 for the diesel. Potential diesel engine improvements that might reduce the NO_2 emission level are discussed in Appendix B.

e. Commuter car emissions are less than one-half of those for the family car and with advanced technology easily meet the 1975/76 standards. (The commuter car weighs only 1700 lb and has reduced acceleration and maximum cruise speed capabilities.)

f. Calculated hot start emissions for family and commuter cars using advanced spark ignition and gas turbine engines with the parallel powertrain configuration meet the numerical values of the 1975/76 standards (cold start), except for NO_2 in the spark ignition family car, and even this value is very close. This standard could be met if vehicle specifications were revised to permit a slight reduction in vehicle performance and approximately a 10 percent reduction in family car weight specifications.

g. Emissions are sensitive to: (1) heat engine class and assumed engine emission part-load characteristics; (2) driving cycle characteristics selected for evaluation; (3) the engine operating mode used over the cycle; (4) the battery discharge and charge characteristics assumed for the analysis; and (5) electric drive motor efficiency and part-load characteristics.

h. Emissions are approximately 10 and 15 percent lower for the parallel powertrain configuration as compared to the series configuration in the family and commuter cars, respectively. However, the parallel powertrain is more complex. Descriptions of the powertrains analyzed can be found in Section 10. 1.

i. As noted earlier, study results are based primarily on hot start data. Incorporation of cold start effects, based on the limited amount of cold start data available, would still allow the advanced technology engine (very lean with exhaust treatment) versions of the hybrid vehicle to meet 1975 HC and CO standards. The NO_2 emission values are reduced when cold start effects are incorporated. Cold start effects are discussed in Section 9.

j. Regenerative braking has essentially no effect on emissions for the hybrid heat engine/electric vehicle due to battery charge acceptance limitations that preclude the ability to store the braking energy. Hence, the expected advantage in reduced generator output for recharging batteries (and therefore reduced engine power and emissions) did not materialize.

k. Vehicle weight increases of several hundred pounds to accommodate additional battery or engine weight have a minor effect on exhaust emissions, but the heavier vehicles would have reduced road performance.

l. Battery power density requirement for a series powertrain family car with a spark ignition engine is 232 w/lb; the installed energy density is 20 w-hr/lb. The requirements for energy density are based on the battery charge/discharge characteristics assumed for this study and may vary somewhat depending on actual test data from a particular advanced battery design.

m. Realistically varying the battery recharge efficiency (to account for resistive losses and incomplete chemical reactions) has little effect on emissions.

n. Fuel consumption values for the spark ignition engine are summarized in the following table for all vehicles operating over their emission driving cycles (the 1972 DHEW Driving Cycle for the commuter car and the family car). The levels shown for the family and commuter cars are competitive with equivalent 1970 conventional vehicles.

Vehicle	Series Configuration (mi/gal)	Parallel Configuration (mi/gal)
Commuter Car	26	30.5
Family Car	11	12.5
Low-speed Van	3.75	----
High-speed Van	4	5
Low-speed Bus	1.25	----
High-speed Bus	1.5	2

These results were developed using specific fuel consumption characteristics based on the minimum SFC/rated horsepower correlation presented in Section 8.0. The data here are representative of current carbureted spark ignition engines operating at air/fuel ratios from 14-16. No adjustment in SFC was made for the lean A/F regimes adopted for hybrid operation because there is every reason to expect that appropriate modifications in the design of advanced engine systems (viz. stratified charge) will permit operation at high air/fuel ratios without serious degradation in fuel consumption. If no improvement were made, the miles per gallon would be approximately 20 percent lower than shown above.

11.1.2 Buses and Vans

a. Little comment can be offered regarding emissions for the other classes of vehicles (low and high-speed postal/delivery van and low and high-speed intracity bus) because there are no emission standards currently available to provide a reference comparison, nor are there measured emissions available from conventional versions of these vehicles driven over a representative driving cycle.

b. A comparison of hybrid versions of passenger cars with current conventional cars driven over the same cycle showed the hybrid to have significantly less emission. If similar comparisons of buses and vans could be made, commensurate reductions for the hybrid are anticipated.

c. In the case of the bus (with its generous weight allocation for the propulsion system), battery power and energy density requirements are quite low. These values are available today. For this case, battery life would be an area of concentration for future improvements.

d. A hybrid bus design could be formulated in the near future. The work in this study could be expanded in the bus to try and arrive at a firm conceptual design.

The following sections present the significant results of the present study for each reference vehicle class in terms of:

1. Driving cycle emission levels.

2. Battery power density and energy density requirements.

3. An assessment of the interactions and effects of significant variations from adopted baseline study assumptions via appropriate trade-off analyses.

Vehicle emission level determinations are summarized in Table 11-1 for those combinations of vehicle class, operational mode, and subsystem characteristic variations investigated in the present study.

For convenience of presentation and discussion, the various study results have been grouped as they pertain to either: (a) the baseline conceptual designs and operational conditions, or (b) a variation in subsystem characteristic/capability or operational mode (from the baseline case assumption).

Recommendations based on the study results are presented in Section 13.

11.2 BASELINE CONCEPTUAL DESIGNS

In this section, all results pertain to the baseline vehicles with powerplants as conceptually defined in Section 10 (series and parallel configurations), and as operated over the pertinent emission driving cycle (e.g., DHEW cycle) as set forth in Section 3. Similarly, all baseline battery characteristics are for the Pb-Acid batteries, as defined in Section 7.

Table 11-1 Summary of Baseline and Trade-off Areas Investigated for Vehicle Emission Effects

AREA \ VEHICLE CLASS/MODE	FAMILY CAR		COMMUTER CAR		DELIVERY/POSTAL VAN				INTRA-CITY BUS			
					Low Speed		High Speed		Low Speed		High Speed	
	Series	Parallel	Series	Parallel	Series	Parallel	Series	Parallel	Series	Parallel	Series	Parallel
BASELINE VEHICLE EMISSION DATA - Baseline Emission Driving Cycle - All Heat Engine Classes (Current and Advanced Technology) - Pb-Acid Batteries	√	√	√	√	√		√	√	√		√	√
EFFECT OF REGENERATIVE BRAKING - Pb-Acid Batteries	√	√	√	√								
EFFECT OF BATTERY RECHARGE EFFICIENCY - Pb-Acid Batteries	√	√	√									
EFFECT OF VEHICLE WEIGHT - Pb-Acid Batteries	√	√	√									
EFFECT OF BATTERY TYPE												
- Ni - Cd	√											
- Ni - Zn	√											
EFFECT OF EMISSION DRIVING CYCLE - New York City vs. DHEW	√	√										
EFFECT OF DRIVE MOTOR EFFICIENCY LEVEL - Pb-Acid Battery	√	√										

11.2.1 Heat Engine Minimum Operating Power Levels for Baseline Emission Driving Cycles

As defined in Section 10, the heat engines for all vehicle classes were constrained to operate with output power as a discrete function of vehicle velocity (See Fig. 10-6 for series configuration and Fig. 10-8 for parallel configuration). In all cases the output power was constant in the low-velocity (0 to 30, and 40 mph) region, at a "minimum operating power level" which was just sufficient to result in the batteries being fully charged at the end of the emission driving cycle.

These values were determined with the use of the computer program, as described in Section 4, and are listed in Table 11-2 for each vehicle class and powerplant configuration.

11.2.2 Resultant Vehicle Exhaust Emissions

Figures 11-1 through 11-30 summarize, for all vehicle classes and heat engines considered, the vehicle exhaust emissions which result from the use of a given powerplant over the baseline emission driving cycle using the part-load exhaust emission characteristics described in Section 9. It should be pointed out that these calculations are based on the baseline vehicle weights and therefore assume that the required battery weight (to fulfill the design driving cycle requirements of battery power and energy density) can be installed in any given powerplant within the baseline powerplant weight allocation. The implications arising from not having sufficient weight available for batteries are discussed separately in Section 11.3.1.3.

It should also be emphasized that the emission values shown do not include cold-start effects, as they are not known or defined for certain heat engine classes. The effect of cold starts on these baseline emission values were discussed in Section 9 based on existing data on spark-ignition, diesel, and gas turbine engines. Cold-start data are presented in Appendices B and C. A brief discussion of cold-start effects can also be found in Section 11.4.

Table 11-2. Heat Engine Minimum Operating Power Levels for Baseline Emission Driving Cycles

Vehicle Class	Heat Engine Minimum Operating Power Level*, hp
Family Car	
Series Configuration	20.70
Parallel Configuration	19.20
Commuter Car	
Series Configuration	7.97
Parallel Configuration	7.10
Low-speed Van	
Series Configuration	22.40
High-speed Van	
Series Configuration	22.40
Parallel Configuration	18.10
Low-speed Bus	
Series Configuration	86.00
High-speed Bus	
Series Configuration	70.60
Parallel Configuration	66.20

*Does not include air conditioning power requirements.

The values for 1975/76 emission standards used in this section are found on each of the curves in grams/mile; they are for HC = 0.46, CO = 4.7, and NO_2 = 0.4. In the case of NO_2 standards, the value of 0.4 grams/mile is not firm. Values ranging from 0.4 to 0.8 grams/mile have been discussed as possible standards. The use of the lowest estimated value for comparative purposes should be considered in evaluating the results.

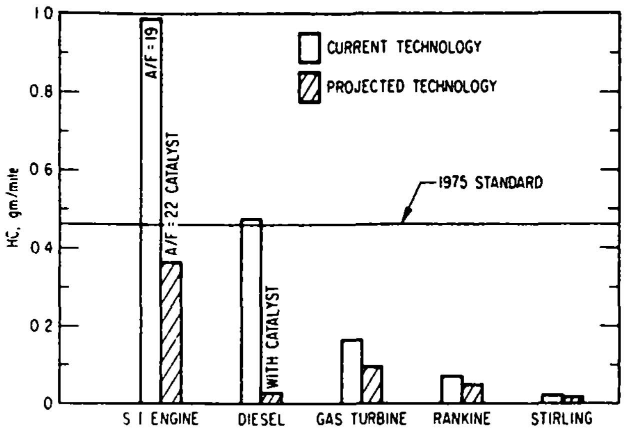


Figure 11-1. Family Car/DHEW Cycle - Series Configuration - HC Emissions

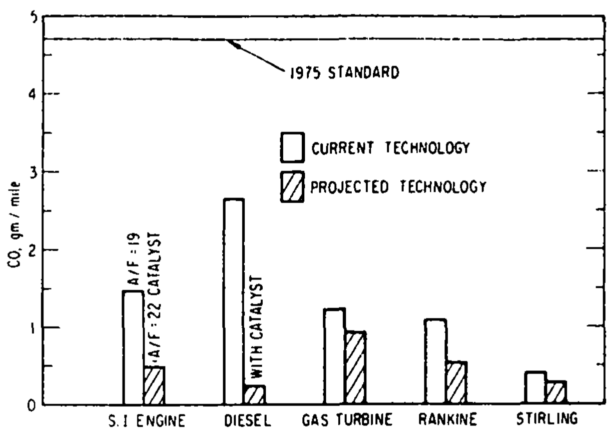


Figure 11-2. Family Car/DHEW Cycle - Series Configuration CO Emissions

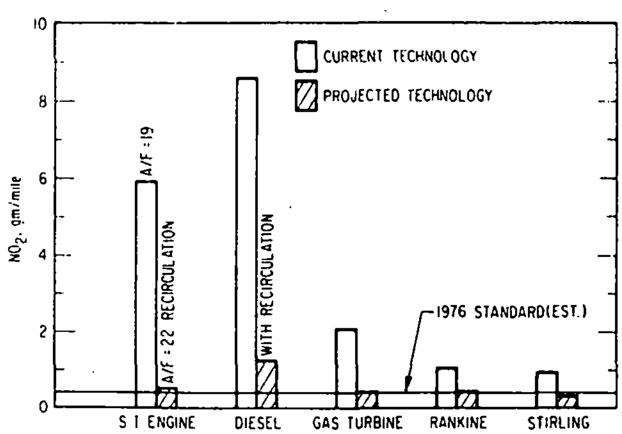


Figure 11-3. Family Car/DHEW Cycle - Series Configuration NO_2 Emissions

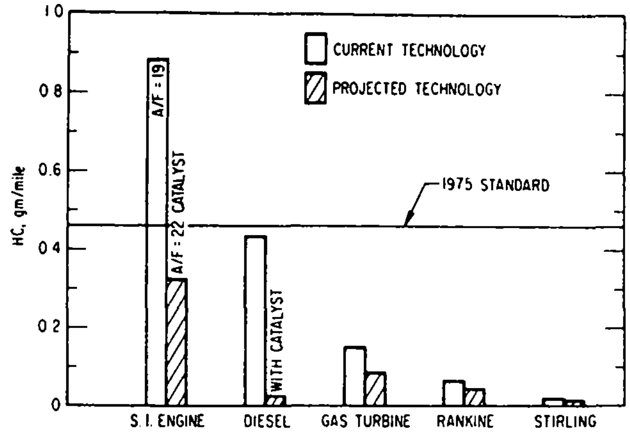


Figure 11-4. Family Car/DHEW Cycle - Parallel Configuration HC Emissions

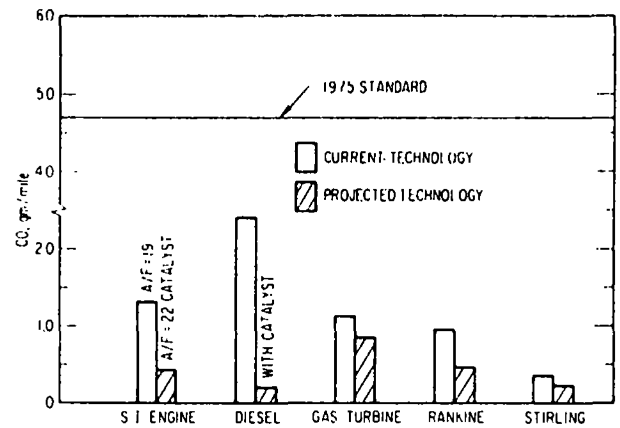


Figure 11-5. Family Car/DHEW Cycle - Parallel Configuration CO Emissions

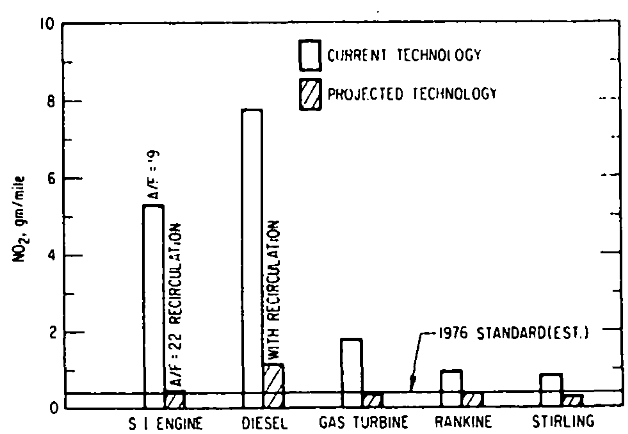


Figure 11-6. Family Car/DHEW Cycle - Parallel Configuration NO_2 Emissions

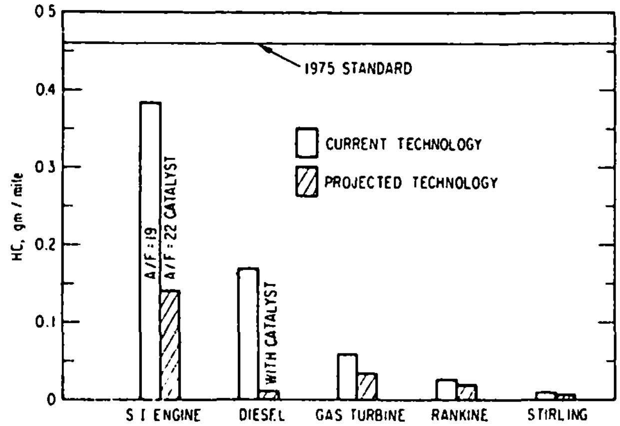


Figure 11-7. Commuter Car/DHEW Cycle - Series Configuration
HC Emissions

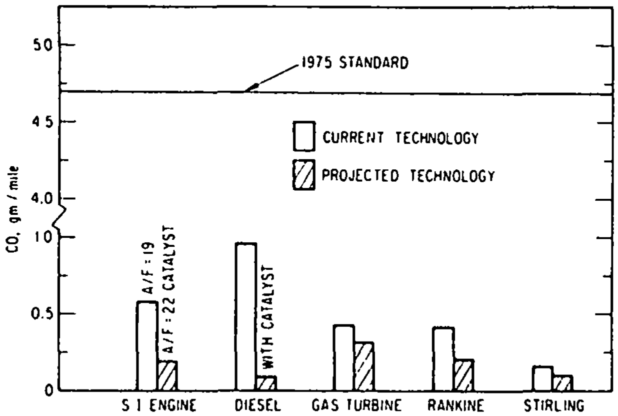


Figure 11-8. Commuter Car/DHEW Cycle - Series Configuration
CO Emissions

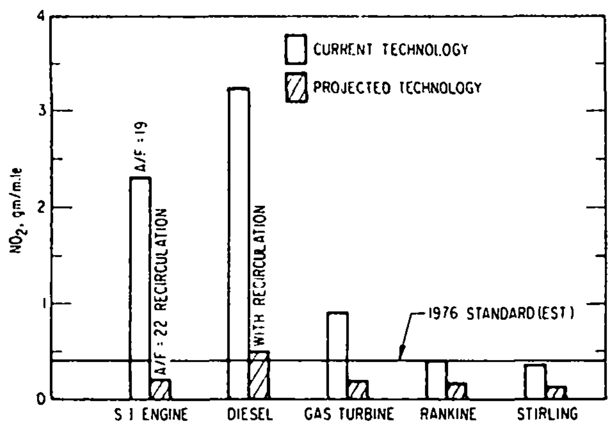


Figure 11-9. Commuter Car/DHEW Cycle - Series Configuration
NO_2 Emissions

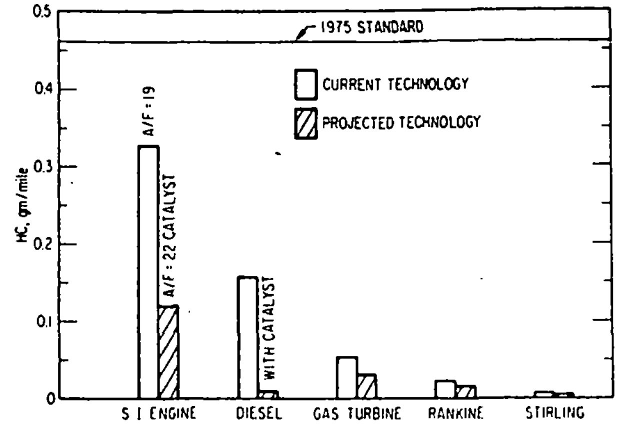


Figure 11-10. Commuter Car/DHEW Cycle - Parallel Configuration
HC Emissions

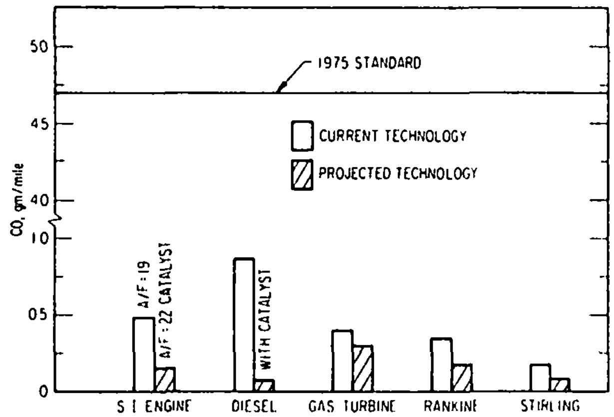


Figure 11-11. Commuter Car/DHEW Cycle - Parallel Configuration
CO Emissions

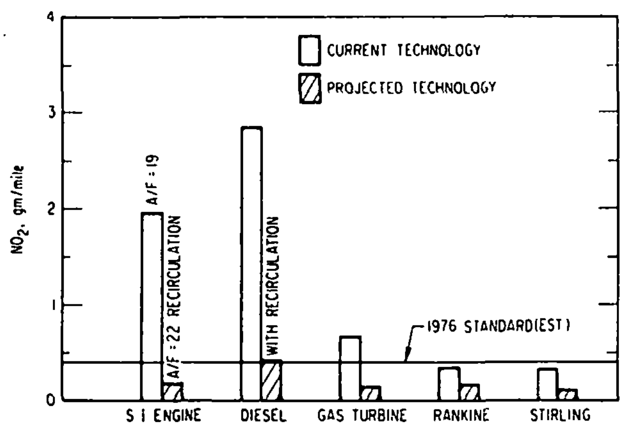


Figure 11-12. Commuter Car/DHEW Cycle - Parallel Configuration
NO_2 Emissions

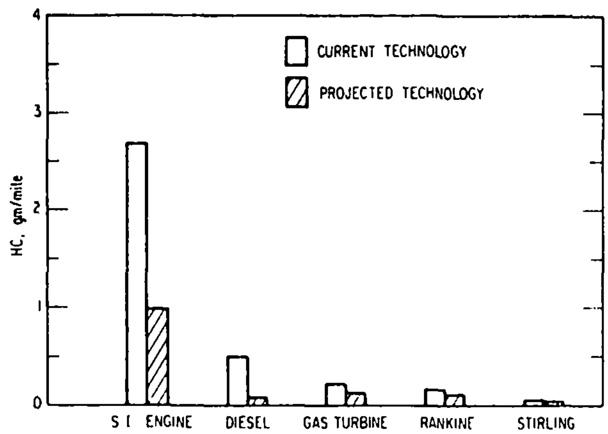


Figure 11-13. Low-Speed Van - Series Configuration
HC Emissions

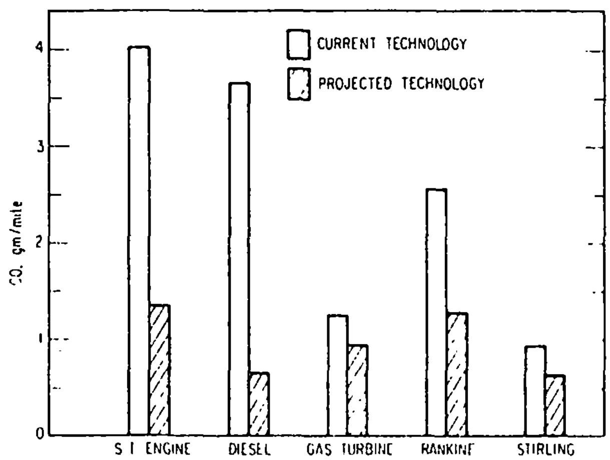


Figure 11-14. Low-Speed Van - Series Configuration
CO Emissions

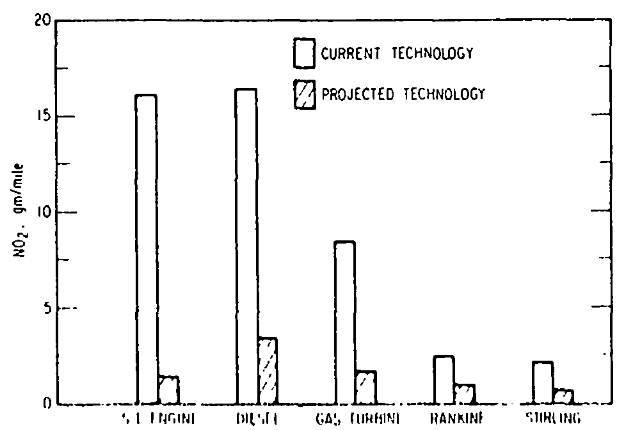


Figure 11-15. Low-Speed Van - Series Configuration
NO_2 Emissions

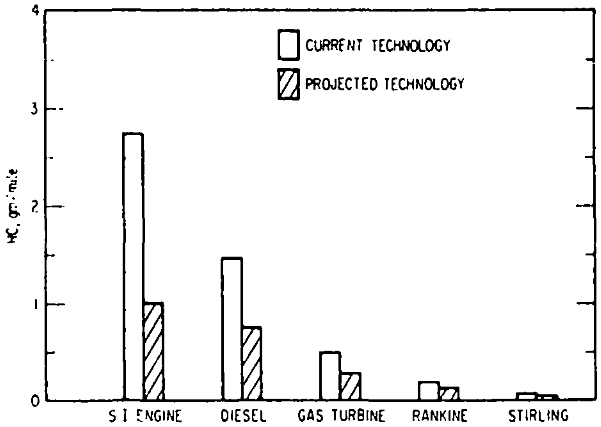


Figure 11-16. High-Speed Van - Series Configuration HC Emissions

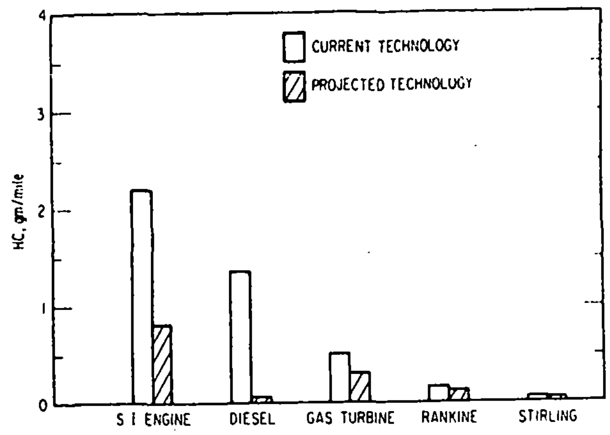


Figure 11-19. High-Speed Van - Parallel Configuration HC Emissions

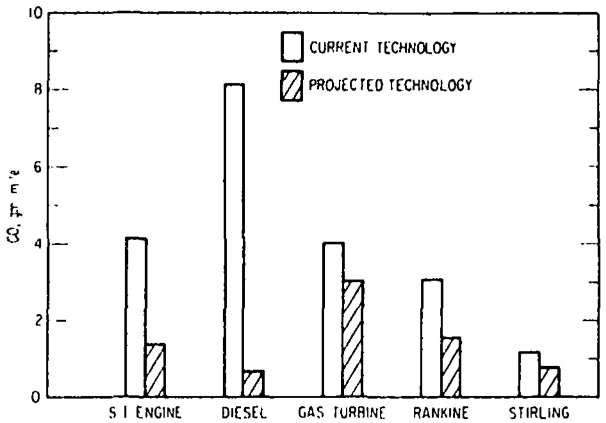


Figure 11-17. High-Speed Van - Series Configuration CO Emissions

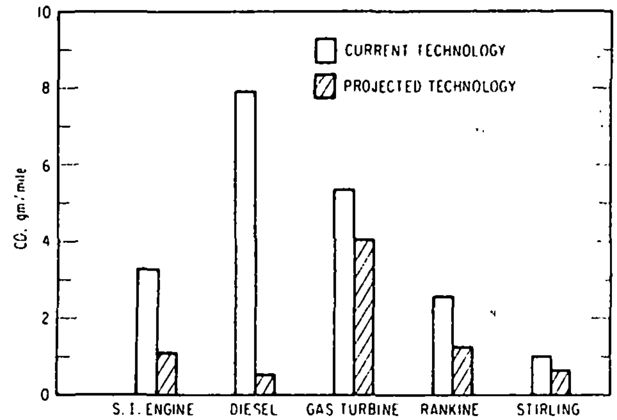


Figure 11-20. High-Speed Van - Parallel Configuration CO Emissions

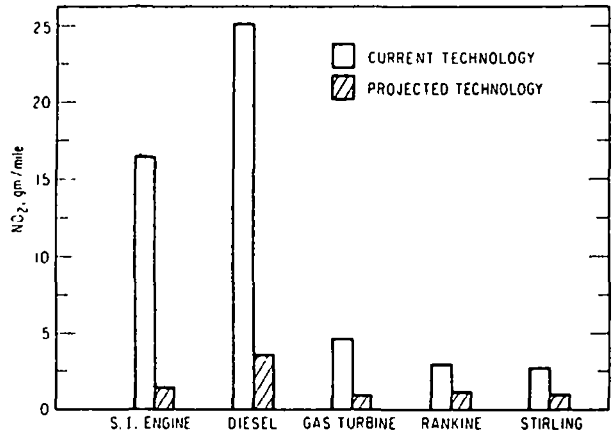


Figure 11-18. High-Speed Van - Series Configuration NO_2 Emissions

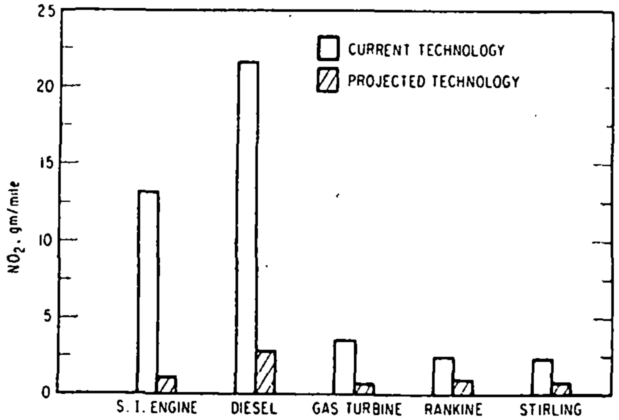


Figure 11-21. High-Speed Van - Parallel Configuration NO_2 Emissions

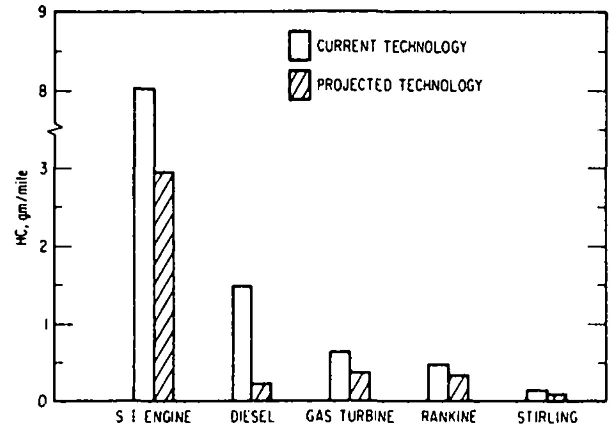


Figure 11-22. Low-Speed Bus - Series Configuration
HC Emissions

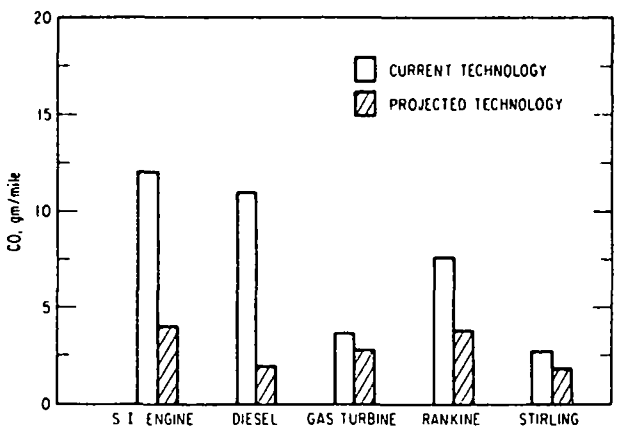


Figure 11-23. Low-Speed Bus - Series Configuration
CO Emissions

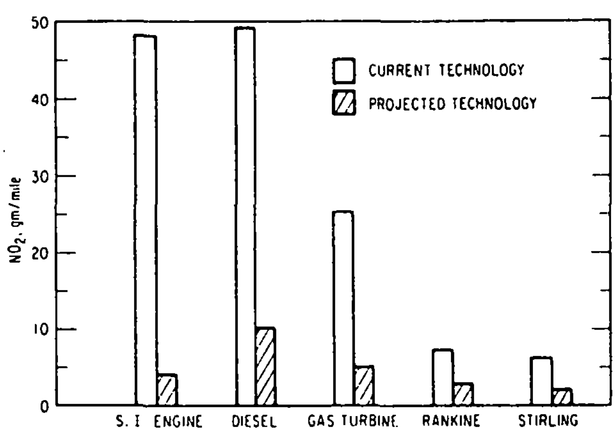


Figure 11-24. Low-Speed Bus - Series Configuration
NO_2 Emissions

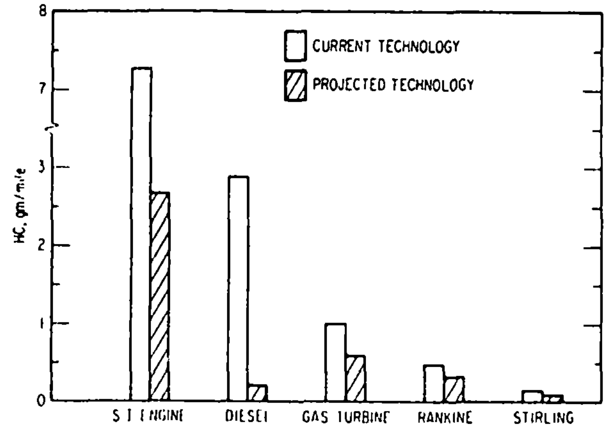


Figure 11-25. High-Speed Bus - Series Configuration
HC Emissions

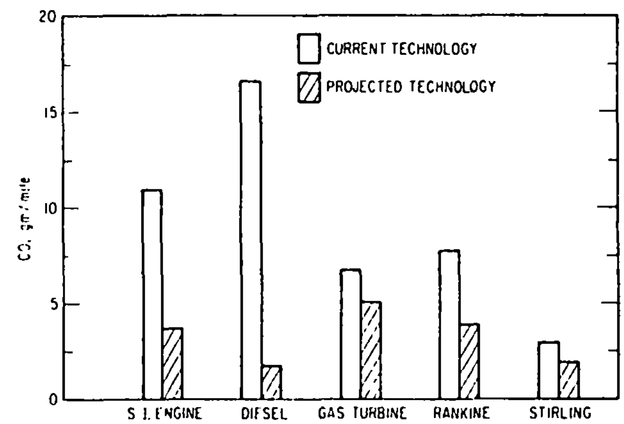


Figure 11-26. High-Speed Bus - Series Configuration
CO Emissions

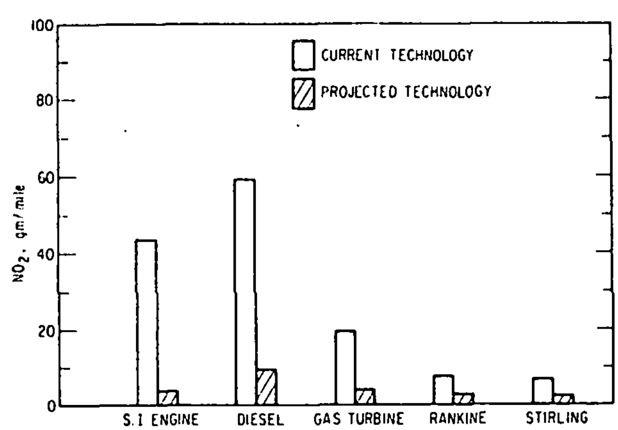


Figure 11-27. High-Speed Bus - Series Configuration
NO_2 Emissions

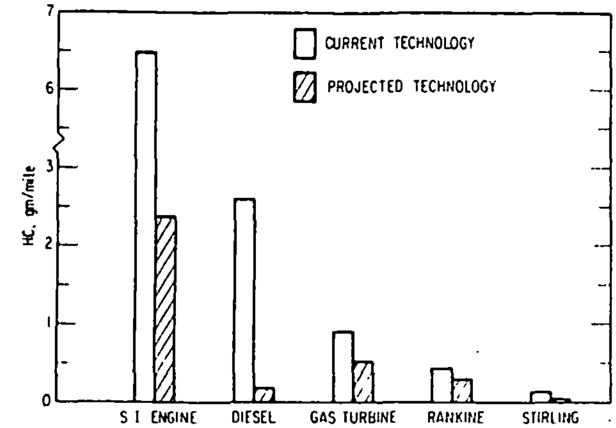


Figure 11-28. High-Speed Bus - Parallel Configuration
HC Emissions

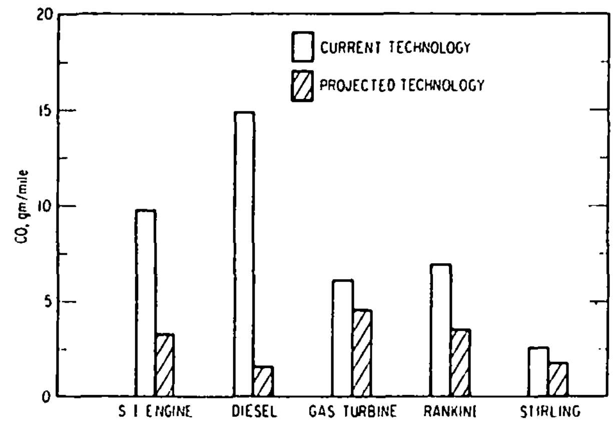


Figure 11-29. High-Speed Bus - Parallel Configuration
CO Emissions

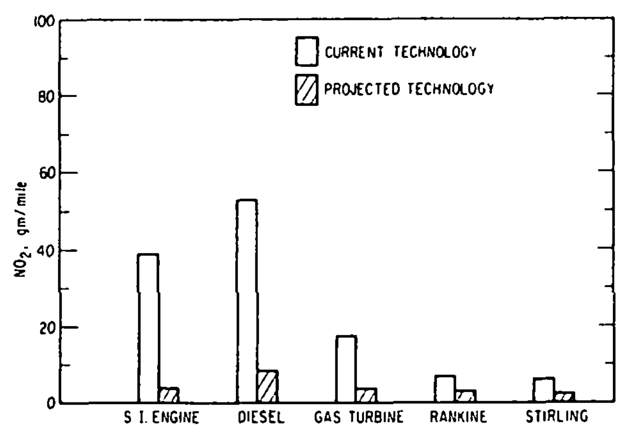


Figure 11-30. High-Speed Bus - Parallel Configuration
NO_2 Emissions

11.2.2.1 Family Car

Figures 11-1, 11-2, and 11-3 denote the resultant family car exhaust emissions in terms of hydrocarbons (HC), carbon monoxide (CO), and oxides of nitrogen (NO_2), respectively, for all five classes of heat engines examined incorporated in a series powerplant configuration. Figures 11-4, 11-5, and 11-6 contain similar results for the parallel powerplant configuration.

In the case of the series powerplant configuration, all heat engine classes, except for the spark ignition and diesel, meet the 1975 HC standards with current technology engine capability (Fig. 11-1). With projected technology all heat engines exceed the 1975 standard by considerable margins. The spark ignition engine HC emissions are 78.5 percent of the 1975 standard; the gas turbine HC emissions are 20.5 percent, while the diesel, Rankine, and Stirling engines are considerably lower, as shown.

With regard to CO emissions, all heat engine classes measurably exceed the 1975 standard with current technology capability (Fig. 11-2), while projected technology greatly reduces CO emissions (to 10.5 percent of the 1975 standard for the spark ignition engine and 19.8 percent for the gas turbine).

The estimated 1976 NO_2 emission standards are not met by any of the 5 heat engine classes with current technology capability (Fig. 11-3). The Rankine and Stirling engines come closest to the standard at approximately two times the required value. However, even with projected technology, only the Stirling engine is below the 1976 standard (78.5 percent). The spark ignition, gas turbine, and Rankine engines do not greatly exceed the standard (126 percent, 104 percent, and 104 percent, respectively), but they still do not meet the 1976 standards.

In the case of the parallel powerplant configuration, all heat engine classes except the spark ignition engine meet the 1975 HC standard with current technology (Fig. 11-4). With projected technology, the spark ignition engine is below the standard (70 percent), while all other heat engines are considerably lower (from 3 percent for the Stirling to 19 percent for the gas turbine).

The 1975 CO emission standards are met by all heat engines with current technology (Fig. 11-5). With projected technology, the CO emissions are reduced to 4.6 percent of the standard for the diesel and to 9.4 percent of the standard for the spark ignition engine.

Again, the 1976 NO_2 standards are not met by any heat engine class with current technology (Fig. 11-6). (The Rankine and Stirling engines are the lowest at approximately 2 times the standard value). With projected technology, the gas turbine, Rankine, and Stirling engines are below the 1976 standard (88.5 percent, 93 percent, and 70 percent, respectively). The spark ignition engine is 113 percent of the 1976 standard.

With reference to the spark ignition engine and gas turbine (those baseline powerplants resulting in a meaningful weight allocation for batteries in the family car), it can be seen that the spark ignition engine requires projected technology advancements to meet 1975 HC standards and does not quite meet 1975 NO_2 standards even with projected technology (113 to 126 percent). The CO standard is easily met with current technology.

The gas turbine, on the other hand, only requires projected technology capability to meet the 1976 NO_2 standard, and even then it slightly exceeds the 1976 value in the series configuration (104 percent).

The parallel configuration results in lower exhaust emissions in any heat engine class. The ratio of parallel configuration emissions divided by series configuration emissions (with projected technology) for spark ignition and gas turbine engines are:

	Spark Ignition	Gas Turbine
HC	0.895	0.913
CO	0.895	0.925
NO_2	0.895	0.858

Therefore, in general, the use of the parallel configuration results in an approximately 10 percent reduction in exhaust emissions for the family car operated over the DHEW driving cycle (over the series configuration).

11.2.2.2 Commuter Car

Figures 11-7, 11-8, and 11-9 illustrate the emission characteristics (HC, CO, NO_2, respectively) of the commuter car operated over the DHEW cycle with a series powerplant configuration. Figures 11-10, 11-11, and 11-12 contain similar results for the parallel powerplant configuration.

For the series configuration, all heat engine classes are below 1975 HC standards with current technology engine capability (Fig. 11-7). With projected technology, the HC emissions are greatly reduced. In this case the spark ignition engine HC emissions are 30.6 percent of the standard, while gas turbine emissions are only 7.4 percent. Diesel, Rankine, and Stirling HC emissions became miniscule.

The same situation is present with regard to CO emissions (Fig. 11-8). Current technology capability is below the 1975 standard (e.g., 12 percent for the spark ignition engine and 9 percent for the gas turbine) while projected technology further reduces these already low values (e.g., 4.1 percent for the spark ignition engine and 6.8 percent for the gas turbine).

The 1976 NO_2 emission standard is met only by the Stirling (90 percent) and Rankine (99 percent) engines with current technology capability. The spark ignition engine and gas turbine values are 576 percent and 226 percent, respectively (Fig. 11-9). However, with projected technology capability, all engines except the Diesel (123 percent) are below the standard (e.g., 49 percent for the spark ignition engine and 45 percent for the gas turbine).

In the case of the parallel powerplant configuration, again all heat engine classes are below the 1975 HC standard with current technology (Fig. 11-10). With projected technology, these emission values are greatly reduced (e.g., 26 percent for the spark ignition engine and 6.8 percent for the gas turbine).

The CO emission standard is also met by all heat engine classes with current technology (Fig. 11-11). With projected technology, CO emissions are further reduced (e.g., 3.5 percent for the spark ignition engine and 6.5 percent for the gas turbine).

The 1976 NO_2 standard is met by only the Rankine (85 percent) and Stirling (77.5 percent) engines with current technology (Fig. 11-12). With projected technology, all heat engines except the Diesel (104 percent) are lower than the 1976 standard. The spark ignition engine and gas turbine levels are 41.5 percent and 33.6 percent, respectively, of the 1976 standard.

With reference to the spark ignition engine and gas turbine (those baseline powerplants resulting in a meaningful weight allocation for batteries in the commuter car), it can be seen that both require projected technology only to meet 1976 NO_2 standards, the HC and CO standards being met with current technology.

The parallel configuration results in lower exhaust emissions in any heat engine class. The ratio of parallel configuration emissions divided by series configuration emissions (with projected technology) for spark ignition and gas turbine engines are:

	Spark Ignition	Gas Turbine
HC	0.85	0.915
CO	0.85	0.960
NO_2	0.83	0.745

Therefore, in general, the use of the parallel configuration results in an approximately 10 to 15 percent reduction in exhaust emissions for the commuter car operated over the DHEW driving cycle (over the series configuration).

As expected, the commuter car, with its lower weight and top cruise speed, has significantly lower emissions than the family car.

11.2.2.3 Low Speed Delivery/Postal Van

Figures 11-13, 11-14, and 11-15 illustrate the emission characteristics (HC, CO, and NO_2, respectively) of the low speed delivery/postal van operated over the selected driving cycle with a series powerplant configuration.

No applicable standards presently exist for this vehicle class and therefore no comparisons can be made in this regard.

The figures do serve to illustrate the variation of emission level with heat engine class and do indicate the reductions in emissions possible with projected technology capability. These analyses will become valuable when standards are established or can be used to help formulate feasible standards.

11.2.2.4 High Speed Delivery/Postal Van

Figures 11-16, 11-17, and 11-18 illustrate the emission characteristics (HC, CO, and NO_2, respectively) of the high speed delivery/postal van operated over the selected driving cycle with a series powerplant configuration.

Again, as no applicable standards for this vehicle class presently exist, no comparisons in this regard can be made.

The relative emission levels of the various heat engine classes are as shown, and the reductions in emissions possible with projected technology as delineated. Figures 11-19 through 11-21 give results for the parallel configuration.

Comparison of the series and parallel configurations indicates that the parallel configuration results in lower exhaust emissions in all heat engine classes except the gas turbine. The ratio of parallel configuration emissions divided by series configuration emissions (with projected technology) for spark ignition, gas turbine, and diesel engines are:

	Spark Ignition	Gas Turbine	Diesel
HC	0.80	1.02	0.80
CO	0.80	1.30	0.80
NO_2	0.80	0.76	0.80

11.2.2.5 Low Speed Intra-City Bus

Figures 11-22, 11-23, and 11-24 depict the emission characteristics (HC, CO, and NO_2, respectively) of the low speed intra-city bus over the selected driving cycle with a series powerplant configuration.

As no applicable standards presently exist for this vehicle class, no comparisons can be made in this regard.

Again, the variation of emission level with heat engine class and reductions in emissions possible with projected technology are evident from inspection of the figures.

11.2.2.6 High Speed Intra-City Bus

Figures 11-25, 11-26, and 11-27 show the emission characteristics (HC, CO, and NO_2, respectively) of the high-speed intra-city bus operated over the selected driving cycle with a series powerplant configuration. Figures 11-28, 11-29, and 11-30 depict similar results with a parallel powerplant configuration.

No comparisons with standards are shown since no standards presently exist for this vehicle class.

The relative emission levels of the various heat engine classes and the reductions in emissions possible with projected technology are evident from the figures.

Comparison of the series and parallel configurations indicates that the parallel configuration results in lower exhaust emissions in all heat engine classes. The ratio of parallel configuration emissions divided by series configuration emissions (with projected technology) for spark ignition, gas turbine and diesel engines are:

	Spark Ignition	Gas Turbine	Diesel
HC	0.89	0.895	0.89
CO	0.89	0.905	0.89
NO_2	0.89	0.885	0.89

11.2.3 Resultant Battery Requirements

Table 11-3 summarizes, for all vehicle classes and powerplant combinations, the power density and energy density requirements which result from either the design driving cycle or emission driving cycles applicable to each

Table 11-3. Resultant Battery Requirements (Baseline Cases)

AREA \ VEHICLE CLASS/MODE	FAMILY CAR		COMMUTER CAR		DELIVERY/POSTAL VAN				INTRA-CITY BUS			
					Low Speed		High Speed		Low Speed		High Speed	
	Series	Parallel	Series	Parallel	Series	Parallel	Series	Parallel	Series	Parallel	Series	Parallel
PEAK POWER DEMAND (kw) (From Design Driving Cycle)	92.5	92.5	28	28	90	90	90	90	210	210	136	136
INSTALLED ENERGY CAPACITY (kw-hr) (From Design and/or Emission Driving Cycle)	8.36	8.36	4.40	4.40	8.80	8.80	8.80	8.80	39.6	39.6	30.8	30.8
WEIGHT AVAILABLE FOR BATTERIES (lb) (From Baseline Weight Statements - Section 10)												
- SI Engine	398	460	101	145	895	824	294	592	3286	3057	2902	3566
- Diesel	240	334	53	99	827	767	106	437	3009	2809	2478	3195
- Gas Turbine	453	526	170	211	959	891	336	653	3298	3091	2871	3550
- Rankine	0	61	0	32	716	678	0	124	2377	2244	1428	2288
- Stirling	0	0	0	0	573	544	0	0	1890	1569	853	1588
POWER DENSITY REQUIRED (w/lb)												
- SI Engine	232	201	279	193	101	109	306	152	64	69	47	38
- Diesel	385	277	527	284	109	117	850	206	70	75	55	42
- Gas Turbine	204	176	165	133	94	101	268	138	64	68	47	38
- Rankine	-	1520	-	875	125	133	-	725	88	93	95	59
- Stirling	-	-	-	-	157	165	-	-	111	134	160	86
ENERGY DENSITY REQUIRED* (w-hr/lb)												
- SI Engine	20	18.1	43.8	30.3	9.9	10.7	30	14.9	12.1	13	10.6	8.6
- Diesel	35	25	83	44.5	10.6	11.5	83	20.1	13.2	14.1	12.4	9.6
- Gas Turbine	18.4	15.9	25.9	20.9	9.2	9.9	26.4	13.5	12.0	13	10.7	8.6
- Rankine	-	137	-	137	12.3	13	-	71	16.7	17.6	21.6	13.5
- Stirling	-	-	-	-	15.2	16.1	-	-	21	25.3	36.1	19.4

*See Section 7.4.

vehicle (see Section 3) and the weight available for batteries in each powerplant type, as delineated in Tables 10-7 through 10-18 of Section 10.

As can be seen in Table 11-3 (and discussed earlier in Section 10), certain powerplants are not applicable under the baseline powerplant weight allocation constraints as defined in Section 3 in that they simply do not allow enough (or any) weight for batteries.

Therefore the vehicle emission data presented for these vehicle powerplant combinations (Section 11.2.2) do not apply, and must be modified to reflect increased vehicle weights which do incorporate the required battery weights. Such increased vehicle weight effects are discussed later for the family car.

Brief comments pertaining to the more significant aspects of the data in Table 11-3 are summarized in the following sections. Energy density figures are based on the battery model characteristics assumed in this study.

11.2.3.1 Series Configuration

1. Family Car - Only the spark ignition engine and the gas turbine result in powerplant weights sufficiently less than the 1500-lb allocation to result in meaningful battery power density requirements (204 to 232 watts/lb). For the same battery, the energy density is 18 to 20 w-hr/lb.

2. Commuter Car - Again, only the spark ignition engine and the gas turbine afford any weight allowance for batteries, resulting in power density requirements of 165 to 279 watts/lb and energy density is 26 to 44 w-hr/lb.

3. Low-speed Delivery/Postal Van - The extremely low continuous power requirements of this vehicle enable all examined heat engines to result in powerplant weights allowing battery weights which indicate meaningful power density requirements of 94 to 157 watts/lb and energy density is 9 to 15 w-hr/lb.

4. High-speed Delivery/Postal Van - The much higher continuous power requirements of this vehicle again limit the heat engines to spark ignition engine and gas turbine, insofar as affording battery weight allowances (power density requirements of 268 to 306 watts/lb and energy density is 26 to 30 w-hr/lb).

5. Low-speed Intracity Bus - The generous weight allowance for the powerplant (6000 lb) allows all heat engine classes to indicate reasonable battery weight allowances resulting in lower easily achievable battery power density requirements (64 to 111 watts/lb) and energy density (12 to 21 w-hr/lb).

6. High-speed Intracity Bus - As in the case of the low-speed bus, all heat engine classes indicate reasonable battery weight allowances (power density requirements of 47 to 160 watts/lb and energy density is 10 to 36 w-hr/lb).

11.2.3.2 Parallel Configuration

1. Family Car - Four of the five heat engine classes (excluding Stirling) provide for some battery weight allowance; however, the Rankine value is so low it results in extremely high battery power density requirements (1520 watts/lb). Therefore only the spark ignition engine and gas turbine are regarded as realistic contenders, resulting in battery power density requirements of 176 to 201 watts/lb and energy density is 16 to 18 w-hr/lb. The diesel engine requires 277 watts/lb and 25 w-hr/lb; further engine weight reductions are required in order to make this engine a firm contender.

2. Commuter Car - Again, only the spark ignition engine and the gas turbine afford any weight allowance for batteries, resulting in power density requirements of 133 to 193 watts/lb and energy density is 21 to 30 w-hr/lb.

3. Low-speed Delivery/Postal Van - The extremely low continuous power requirements of this vehicle enable all examined heat engines to result in powerplant weights allowing battery weights which indicate power density requirements of 101 to 165 watts/lb and energy density is 10 to 16 w-hr/lb.

4. High-speed Delivery/Postal Van - The higher continuous power requirements of this vehicle limit the heat engine applicability to four classes (excludes Stirling); however, only the spark ignition engine and the gas turbine afford reasonable power density requirements (138 to 152 watts/lb) and energy density (13 to 15 w-hr/lb).

5. Low-speed Intracity Bus - Again the generous weight allowance for the powerplant (6000 lb) indicates all heat engine classes are feasible, from the standpoint of

battery power density requirement of 68 to 134 watts/lb and energy density is 13 to 25 w-hr/lb.

6. High-speed Intracity Bus - As in the case of the low-speed bus, all heat engine classes indicate reasonable battery weight allowances (power density requirements of 38 to 86 watts/lb and energy density is 8 to 20 w-hr/lb).

11.2.4 Vehicle Fuel Economy

The results of an analysis of fuel economy for hybrid vehicles equipped with gasoline-powered reciprocating spark ignition engines are shown in the table below. The levels shown for the family and commuter cars may be seen to be competitive with equivalent 1970 conventional vehicles.

Vehicle	Series Configuration (mi/gal)	Parallel Configuration (mi/gal)
Commuter Car	26	30.5
Family Car	11	12.5
Low-speed Van	3.75	----
High-speed Van	4	5
Low-speed Bus	1.25	----
High-speed Bus	1.5	2

These results were developed using specific fuel consumption characteristics based on the minimum SFC/rated horsepower correlation presented in Section 8.0, Fig. 8-9. The data here are representative of current (carbureted) SI engines operating at air/fuel ratios from 14 to 16. No adjustment in SFC was made for the lean A/F regimes adopted as goals for hybrid operation (19 for current technology and 22 for projected technology) because there is every reason to expect that appropriate modifications in the design of advanced engine systems will permit operation at high air/fuel ratios without serious degradation in fuel consumption.

According to Refs. 1, 2, and 3, the current technology goal of 19 A/F ratio is attainable with minimal design modifications to the conventional engine. Hansel (Ref. 1) shows a fuel economy characteristics for a conventional engine with standard spark timing that is essentially flat at an optimum level out to an air/fuel ratio of 19. The same general trend was achieved by Toyota (Ref. 2) with spark timing adjusted for best torque. Additionally, Ref. 3 provides substantial evidence indicating that the current lean limit for automotive engines can be extended significantly with modifications to the ignition system and control of mixture distribution.

With regard to the projected technology A/F goal of 22, a limited amount of data (e.g., Ref. 1) suggests that conventional engines with minor modifications to spark timing may suffer fuel economy losses of 25 percent or more at lean mixtures approaching 21 A/F. It is therefore anticipated that alterations in the design of the engine may be required in order to achieve far-lean operation with satisfactory fuel consumption. There is encouraging evidence that the 22 A/F goal might be achieved by use of the stratified charge concept or by a precombustion chamber design. Both of these approaches indicate the potential of low SFC at the lean operating condition. Reference 4 indicates that a 20 percent SFC improvement over the carbureted gasoline engine can be achieved with the stratified charge approach. Emission data from the single-cylinder prechamber concept of Newhall (Ref. 5) also looks promising with regard to satisfactory lean operating performance. Data obtained from Newhall's work shows that ISFC (indicated SFC) decreases as A/F ratio is increased. This trend suggests that the increase in BSFC at high air/fuel ratios may be minimal.

11.3 TRADEOFF STUDIES

A number of selected tradeoff studies were made to determine the sensitivity of vehicle emissions to a number of potential subsystem variables and operational variables (as shown in Table 11-1) as well as the effect of powerplant weight on battery requirements, as mentioned previously and discussed in Section 10.

These results are discussed in the following sections, with regard to the effect of the variable on vehicle emission levels and/or battery requirements.

11.3.1 Effect on Vehicle Emission Levels

11.3.1.1 Regenerative Braking

Computations for the family car and the commuter car indicated that varying regenerative braking efficiency from zero percent (as used in all baseline vehicle cases) to 100 percent had no effect on vehicle exhaust emissions. While contrary to expectations, analysis of the computer data indicates that this is the unique result of the heat engine power output schedule used as a baseline in the present study, coupled with charge-rate limitations of the batteries.

More specifically, as explained in Section 10, when the vehicle decelerates, the heat engine power automatically drops to the "minimum operating power level" during the entire deceleration time interval. Using the family car as an example (series configuration), the generator output current at this condition is ~38 amps. The maximum charge-rate of the battery, due to its relatively high state-of-charge throughout the DHEW cycle, is also ~38 amps. Therefore the battery is being supplied by the generator up to its full capacity and current generated by the regenerative-braking process simply cannot be accepted by the battery under these conditions.

These results clearly indicate that if regenerative braking is to be useful, the heat-engine power output schedule should be such that power output is reduced to a minimal value (idle power) during vehicle deceleration periods.

11.3.1.2 Battery Recharge Efficiency

A series of computer runs was made with the family car and the commuter car to determine the effect of battery recharge efficiency on vehicle exhaust emissions. Figures 11-31, 11-32, and 11-33 present the results for the various emissions (HC, CO, and NO_2, respectively) with all five classes of heat engines having current technology capability installed in the family car in a series configuration. Similar results are presented in Figs. 11-34, 11-35, and 11-36 wherein the heat engines incorporate projected technology. Figures 11-37, 11-38, and 11-39 contain similar information for the family car with current heat engine technology in a parallel configuration, while Figs. 11-40, 11-41, and 11-42 show similar results for the commuter car with current heat engine technology in a series configuration.

With the exception of the gas turbine and diesel engines, the remaining heat engines indicate a nearly linear relationship between battery recharge efficiency and exhaust emissions. Using the family car and the spark ignition engine as an example, the significant values are listed in the following table, where change in emissions is expressed as a percentage increase or decrease from that existing for the baseline recharge efficiency of 70 percent.

Compared to 70 Percent Recharge Efficiencies	Current Heat Engine Technology				Projected Heat Engine Technology	
	Series		Parallel		Series	
Recharge Efficiency, Percent	50	100	50	100	50	100
Change in Emissions, Percent:						
HC	+8	-12.5	+12.5	-12.5	+8.0	-12.5
CO	+8	-12.5	+13.0	-12.5	+7.5	-13.0
NO_2	+8	-12.5	+13.0	-12.5	+7.0	-12.5

Thus it can be seen that the parallel configuration is slightly more sensitive to reduced recharge efficiencies than the series configuration.

The diesel and gas turbine engines exhibit characteristics associated implicitly with the particular emission constituent, as a result of the assumed part-load emission characteristics as delineated in Section 9. Most noticeable is the *increase of CO occasioned by increasing recharge efficiency* (Figs. 11-32, 11-35, 11-38) with the gas turbine engine. In addition, the gas turbine exhibits more marked sensitivity of NO_2 emissions to recharge efficiency than the other heat engines (Figs. 11-33, 11-36, 11-39).

In the case of the commuter car (Figs. 11-40, 11-41, and 11-42), similar results are obtained except for the fact that the emission sensitivity to recharge efficiency is slightly greater than in the case of the family car.

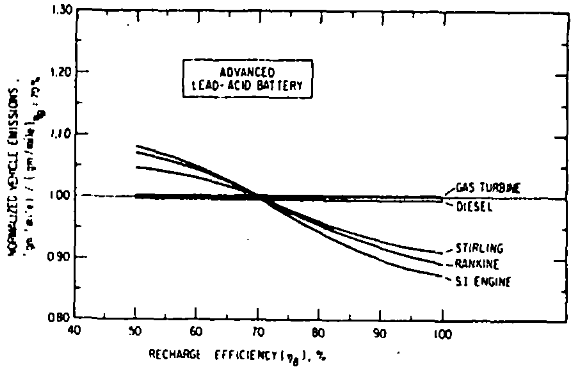


Figure 11-31. Effect of Battery Recharge Efficiency on HC Emissions - Family Car - Series Configuration - Current Technology

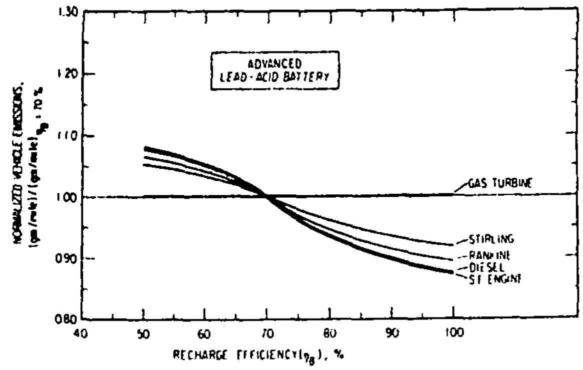


Figure 11-34. Effect of Battery Recharge Efficiency on HC Emissions - Family Car - Series Configuration - Projected Technology

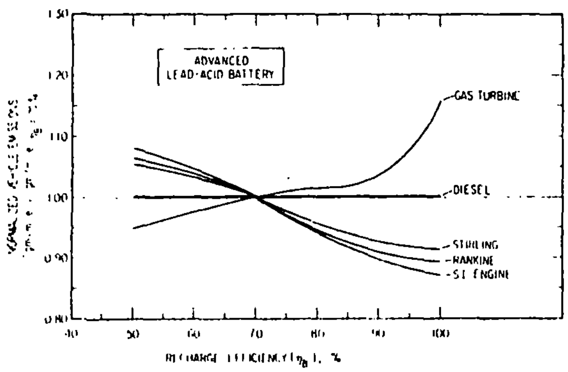


Figure 11-32. Effect of Battery Recharge Efficiency on CO Emissions - Family Car - Series Configuration - Current Technology

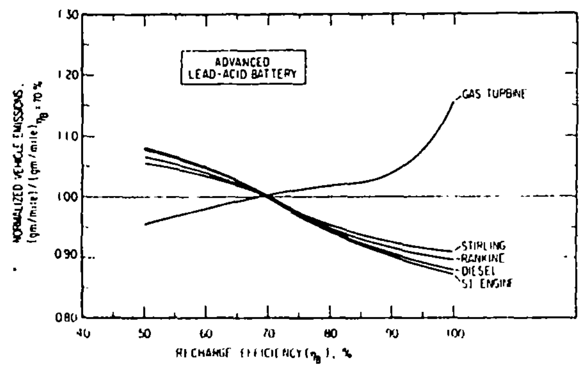


Figure 11-35. Effect of Battery Recharge Efficiency on CO Emissions - Family Car - Series Configuration - Projected Technology

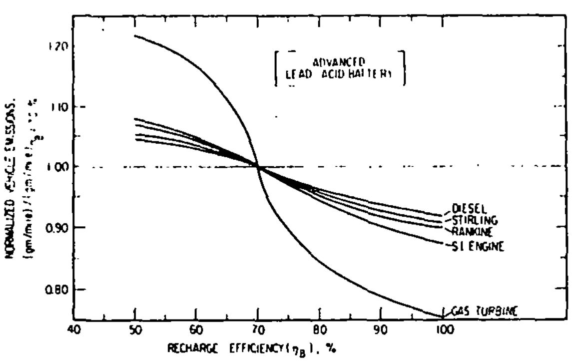


Figure 11-33. Effect of Battery Recharge Efficiency on NO_2 Emissions - Family Car - Series Configuration - Current Technology

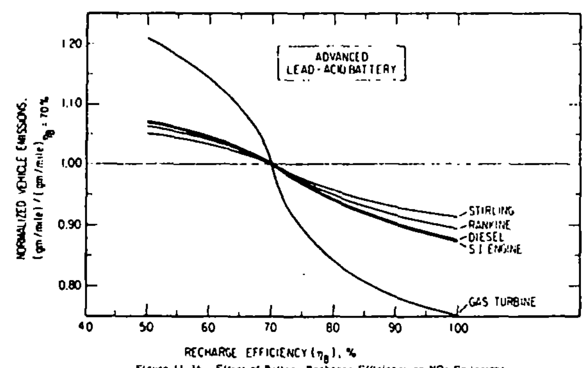


Figure 11-36. Effect of Battery Recharge Efficiency on NO_2 Emissions - Family Car - Series Configuration - Projected Technology

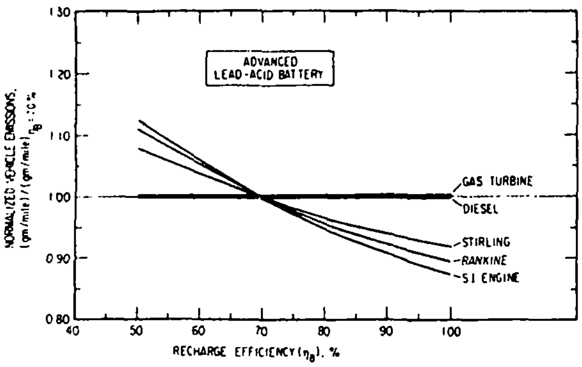


Figure 11-37. Effect of Battery Recharge Efficiency on HC Emissions - Family Car - Parallel Configuration - Current Technology

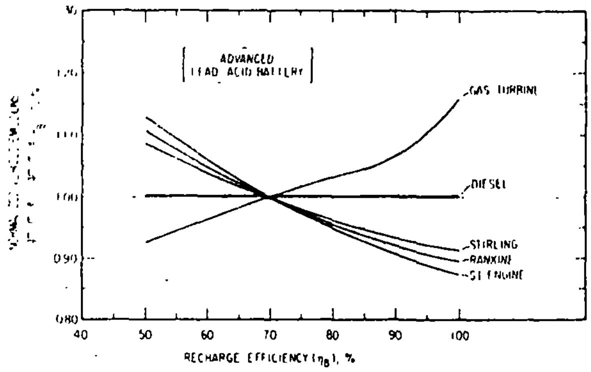


Figure 11-38. Effect of Battery Recharge Efficiency on CO Emissions - Family Car - Parallel Configuration - Current Technology

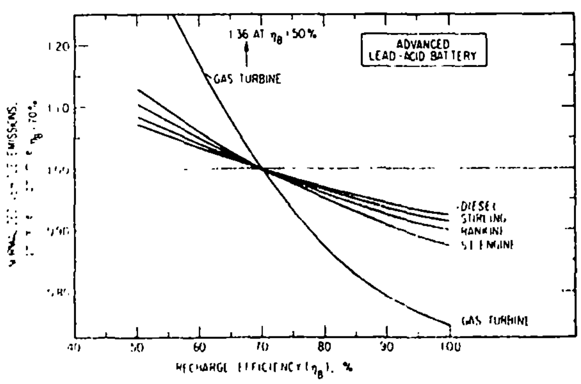


Figure 11-39. Effect of Battery Recharge Efficiency on NO_x Emissions - Family Car - Parallel Configuration - Current Technology

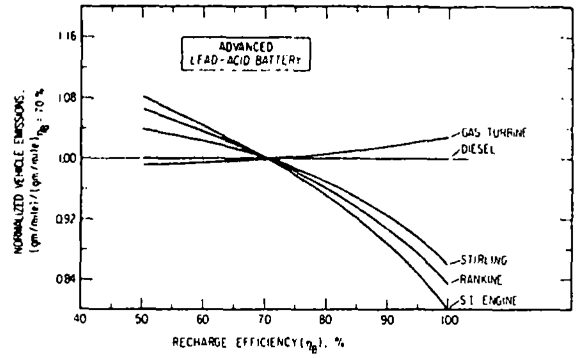


Figure 11-40. Effect of Battery Recharge Efficiency on HC Emissions - Commuter Car - Series Configuration - Current Technology

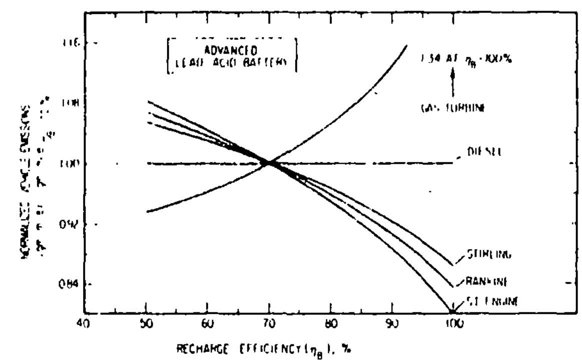


Figure 11-41. Effect of Battery Recharge Efficiency on CO Emissions - Commuter Car - Series Configuration - Current Technology

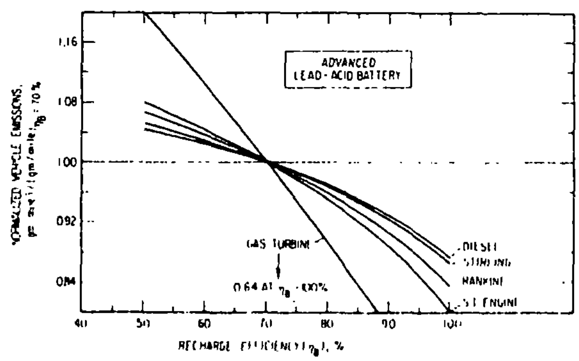


Figure 11-42. Effect of Battery Recharge Efficiency on NO_x Emissions - Commuter Car - Series Configuration - Current Technology

11.3.1.3 Vehicle Weight Effect

As mentioned previously, the baseline powerplant weight computations of Section 10 indicated that for several vehicle classes only the spark ignition and gas turbine engines afforded meaningful weight allocations for batteries, and that even in these cases the resultant battery power density and energy density requirements were very high (See Table 11-3).

Therefore computations were made for the family car and commuter car to determine the effect on exhaust emissions of increasing vehicle weight to allow more weight for batteries, thereby reducing their power density and energy density requirements.

These results are illustrated in Figs. 11-43 through 11-60 in the following sequence:

	Current Heat Engine Technology	Projected Heat Engine Technology
Family Car (Series Configuration)		
HC	Fig. 11-43	Fig. 11-46
CO	11-44	11-47
NO_2	11-45	11-48
Family Car (Parallel Configuration)		
HC	11-49	11-52
CO	11-50	11-53
NO_2	11-51	11-54
Commuter Car (Series Configuration)		
HC	11-55	11-58
CO	11-56	11-59
NO_2	11-57	11-60

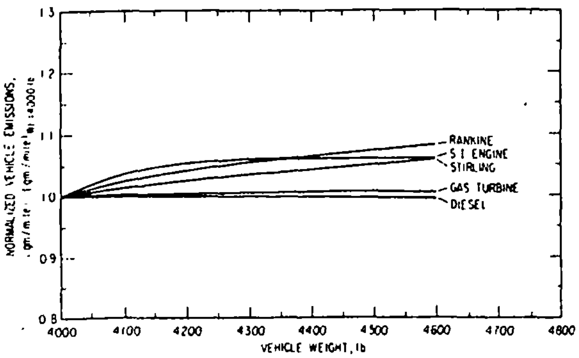


Figure 11-43. Effect of Vehicle Weight on HC Emissions - Family Car/DHEW Cycle
Series Configuration - Current Technology

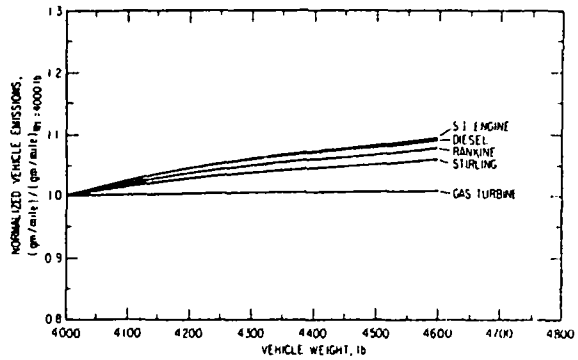


Figure 11-46. Effect of Vehicle Weight on HC Emissions - Family Car/DHEW Cycle
Series Configuration - Projected Technology

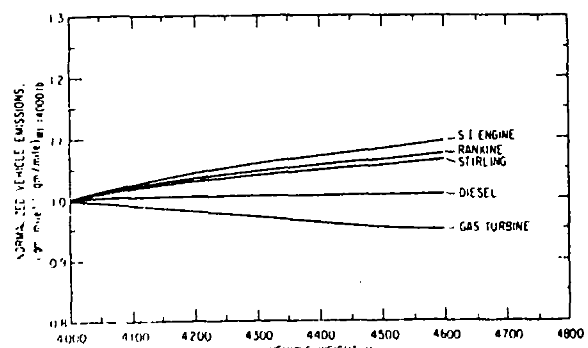


Figure 11-44. Effect of Vehicle Weight on CO Emissions - Family Car/DHEW Cycle -
Series Configuration - Current Technology

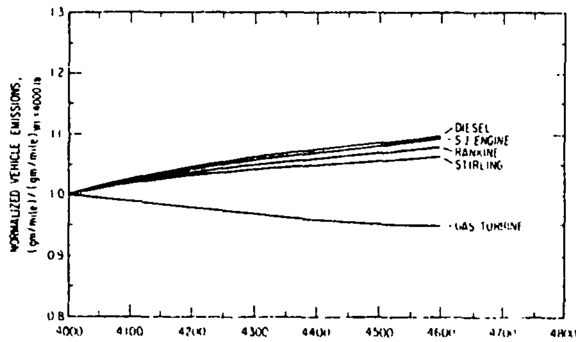


Figure 11-47. Effect of Vehicle Weight on CO Emissions - Family Car/DHEW Cycle
Series Configuration - Projected Technology

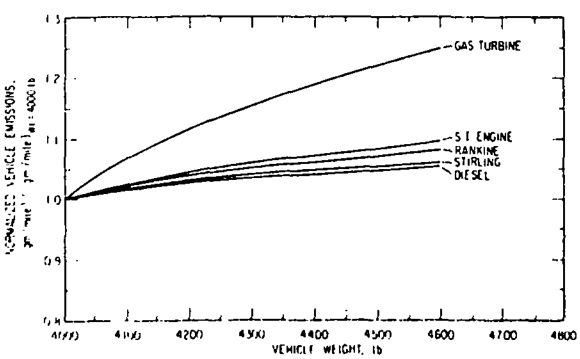


Figure 11-45. Effect of Vehicle Weight on NO_x Emissions - Family Car/DHEW Cycle
Series Configuration - Current Technology

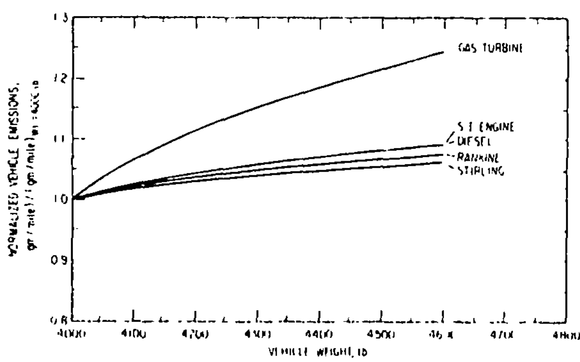


Figure 11-48. Effect of Vehicle Weight on NO_x Emissions - Family Car/DHEW Cycle
Series Configuration - Projected Technology

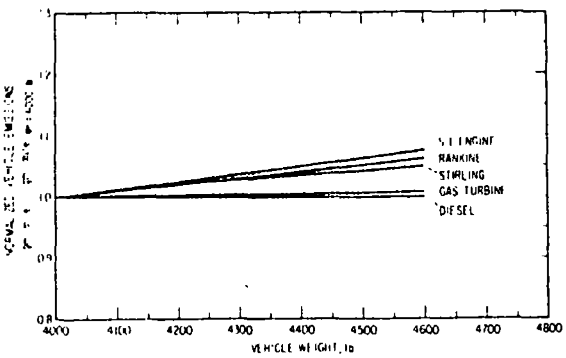


Figure 11-51. Effect of Vehicle Weight on HC Emissions - Family Car/DHEW Cycle - Parallel Configuration - Current Technology

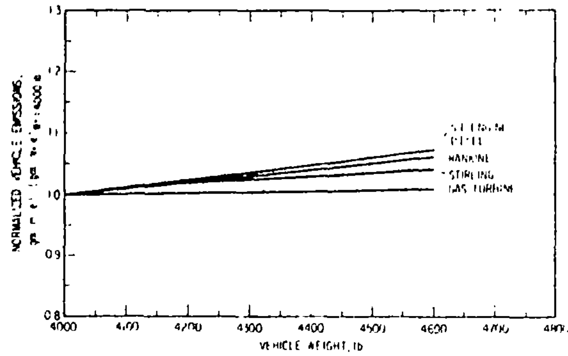


Figure 11-52. Effect of Vehicle Weight on HC Emissions - Family Car/DHEW Cycle - Parallel Configuration - Projected Technology

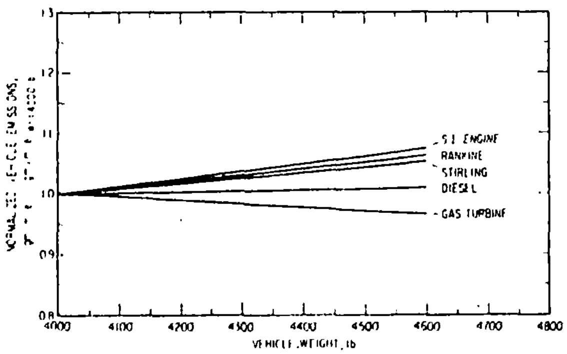


Figure 11-53. Effect of Vehicle Weight on CO Emissions - Family Car/DHEW Cycle - Parallel Configuration - Current Technology

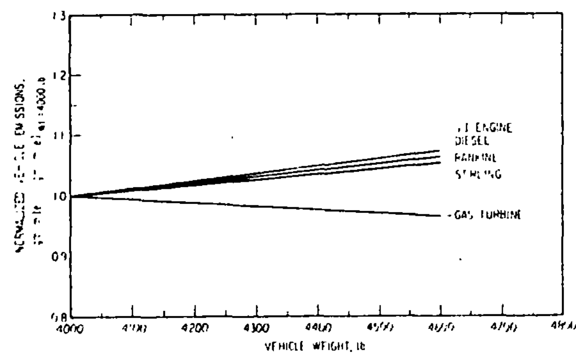


Figure 11-54. Effect of Vehicle Weight on CO Emissions - Family Car/DHEW Cycle - Parallel Configuration - Projected Technology

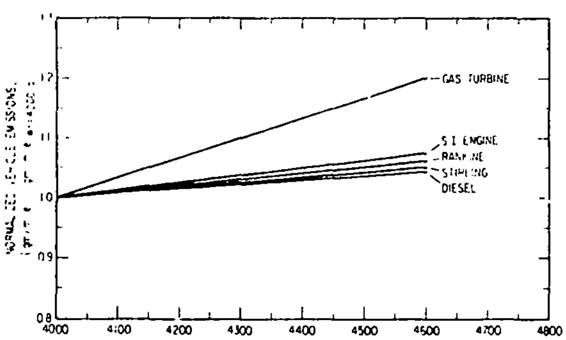


Figure 11-55. Effect of Vehicle Weight on NO_x Emissions - Family Car/DHEW Cycle - Parallel Configuration - Current Technology

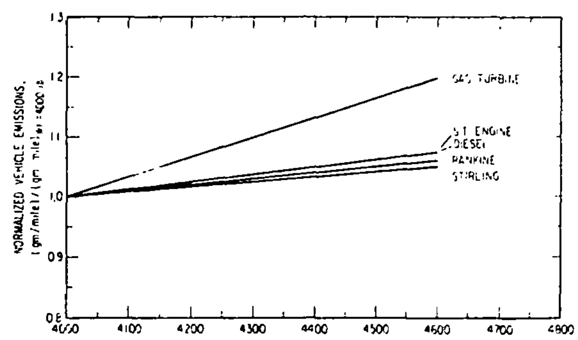


Figure 11-56. Effect of Vehicle Weight on NO_x Emissions - Family Car/DHEW Cycle - Parallel Configuration - Projected Technology

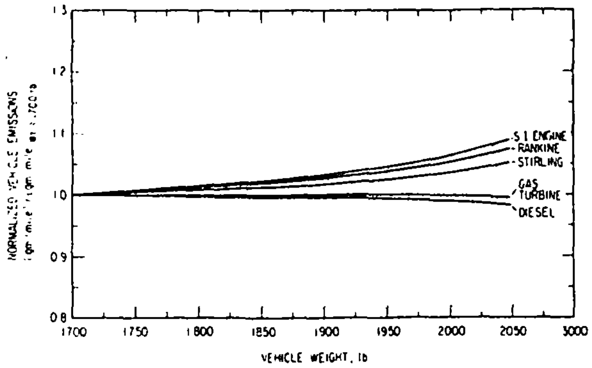


Figure 11-55. Effect of Vehicle Weight on HC Emissions - Commuter Car/DHEW Cycle - Series Configuration - Current Technology

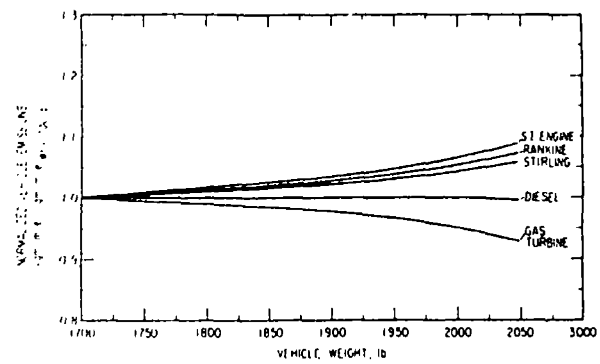


Figure 11-56. Effect of Vehicle Weight on CO Emissions - Commuter Car/DHEW Cycle - Series Configuration - Current Technology

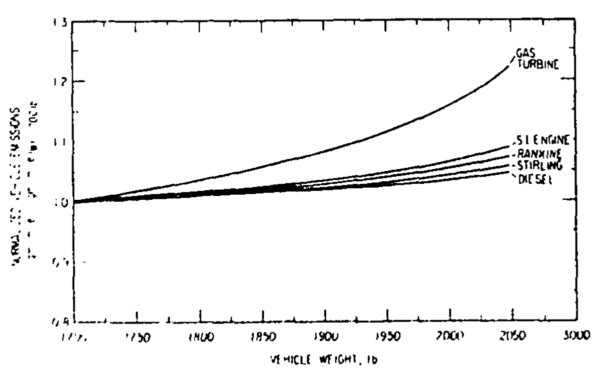


Figure 11-57. Effect of Vehicle Weight on NO_x Emissions - Commuter Car/DHEW Cycle - Series Configuration - Current Technology

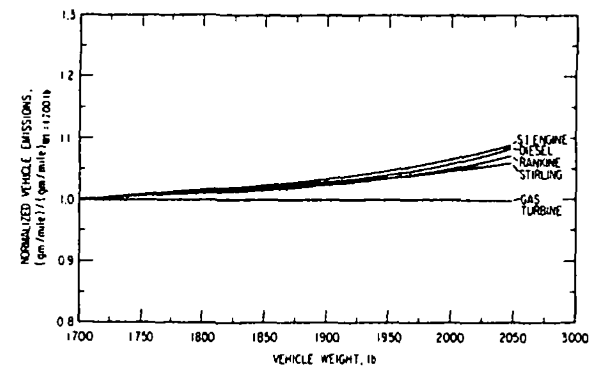


Figure 11-58. Effect of Vehicle Weight on HC Emissions - Commuter Car/DHEW Cycle - Series Configuration - Projected Technology

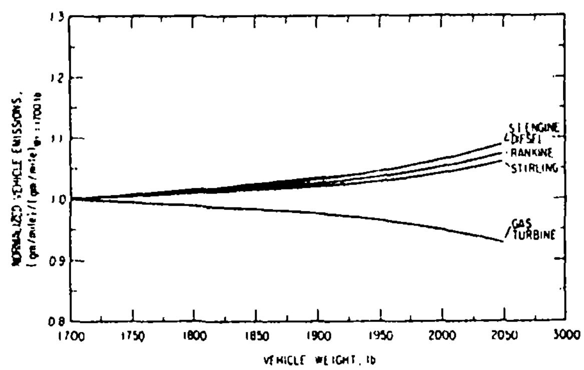


Figure 11-59. Effect of Vehicle Weight on CO Emissions - Commuter Car/DHEW Cycle - Series Configuration - Projected Technology

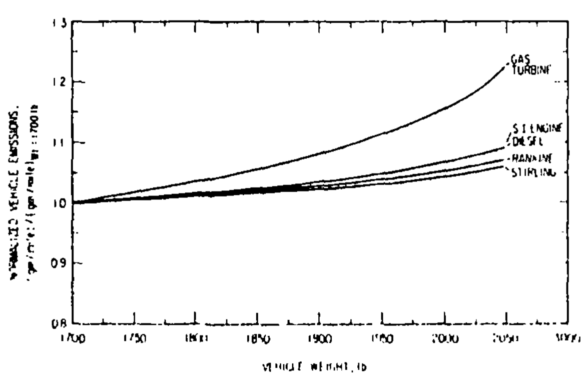


Figure 11-60. Effect of Vehicle Weight on NO_x Emissions - Commuter Car/DHEW Cycle - Series Configuration - Projected Technology

The family car results were obtained for vehicle weights up to 4600 lb (a 600-lb increase over the baseline vehicle weight) and the commuter car results for vehicle weights up to 2040 lb (a 340-lb increase over the baseline vehicle weight).

In the case of the family car, all heat engines except the diesel and gas turbine exhibited a fairly uniform rise in all species of emissions with increasing vehicle weight. Using the spark ignition engine as a representative example, the percentage increase in exhaust emissions at the 4600-lb weight level (over the emission levels at 4000 lb) are:

Percent of Change in Emissions	Current Technology		Projected Technology	
	Series	Parallel	Series	Parallel
HC	+6	+7.5	+9.5	+7.5
CO	+9.5	+7.5	+9.5	+7.5
NO_2	+9.5	+7.5	+9.0	+7.5

Thus, for a 15-percent increase in family car weight (to 4600 lb) the emissions were increased ~9.5 percent for the series configuration and ~7.5 percent for the parallel configuration.

Again, using the spark ignition engine as a representative example, the percentage increase in exhaust emissions for the commuter car (series configuration) at the 2040-lb weight level (over the emission levels at 1700 lb) are:

Percent of Change in Emissions	Current Technology	Projected Technology
HC	+9	+9
CO	+9	+9
NO_2	+9	+9

Thus, for the 12-percent increase in commuter car weight (to 2040 lb), the emissions were increased ~9 percent with a series configuration.

The variation of emissions for diesels and gas turbines is again a unique function of the particular emission constituent, whether family car or commuter car, due to the part-load emission characteristics delineated in Section 9. In the case of the gas turbine, CO emissions decrease (~3.5 percent for the family car and ~7 percent for the commuter car) at the maximum vehicle weights examined, while NO_2 emissions increase with vehicle weight at a greater rate than other heat engine classes (~20 percent for the family car and ~22 percent for the commuter car).

Aside from these unique variabilities of the gas turbine, it appears that vehicle weights could be increased ~15 percent to afford more weight for batteries (and thus reduce power density and energy density requirements) at a minimal (~10 percent) sacrifice in increased vehicle emissions.

It should be noted that these results were generated using the baseline propulsion system sized for a 1700-lb commuter car or a 4000-lb family car driven over the DHEW driving cycle. Although these vehicles with increased weight variations were not driven over the design driving cycle, it is most probable that the peak cruise speed and maximum acceleration capabilities were reduced.

11.3.1.4 Battery Capacity and Type

All baseline vehicle emissions shown in Section 11.2.2 were calculated with the baseline installed battery capacities delineated in Table 11-3. In the case of the family car, the installed battery had a capacity of 38 amp-hr, which was required to meet the design driving cycle requirements. At this installed capacity, all three battery types investigated (Pb-acid, Ni-Zn, and Ni-Cd) resulted in the same family car emissions over the DHEW cycle.

To investigate the effect of changing battery installed capacity and type, computer runs were made with the family car having Pb-acid batteries ranging from 20 to 70 amp-hr in capacity, and Ni-Zn batteries having capacities from 18 to 70 amp-hr. The results of these calculations are shown in Fig. 11-61, where the emissions (HC, CO, NO_2) are normalized by dividing the calculated results by the 1975/76 standards. As can be seen, the installed battery capacity of the family car can indeed be reduced for the DHEW cycle (at some sacrifice in maximum vehicle design acceleration capability) at the expense of increased exhaust emissions.

For example, if the 38-amp-hr capacity was reduced to 20 amp-hr, the Pb-acid battery results indicate a 36-percent increase in NO_2, a 33-percent increase in HC, and a 35-percent increase in CO. For the same reduction in capacity (to 20 amp-hr), the Ni-Zn battery results indicate a 15-percent increase in NO_2, a 21-percent increase in HC, and a 15-percent increase in CO.

Conversely, however, increasing battery capacity above the baseline requirement does not lead to decreased exhaust emissions.

Similar results pertaining to battery capacity effects on emission levels are shown in Fig. 11-62 for the high-speed intracity bus (spark ignition engine, projected technology, series configuration) with Pb-acid batteries. Reducing installed battery capacity in half (from 70 amp-hr to 35 amp-hr) increases HC, CO, and NO_2 by 51 percent each.

Another very significant effect was shown by using present day battery charge/discharge characteristics for a lead-acid battery rather than the advanced design characteristics presented in Section 7. Because of the reduced charge acceptance capabilities of the present battery, the generator power output level in the family car nearly doubled in order to return the battery state-of-charge to its initial value at the end of the DHEW cycle. Consequently, the emission levels increased over the baseline series powertrain configuration by the following factors for the current heat engine technology: HC, 1.35; CO, 1.33; NO_2, 1.28.

11.3.1.5 Drive Motor Efficiency

The baseline drive motor average operational efficiency in all vehicle classes was shown to be 80 percent (see Table 10-3). To assess the importance of this important subsystem efficiency on the baseline emissions, a number of computer runs of the family car on the DHEW cycle (series and parallel configurations) were made in which the drive motor average efficiency was varied from 50 to 100 percent. The results of these computer simulations are shown in Figs. 11-63 through 11-74 in the following order.

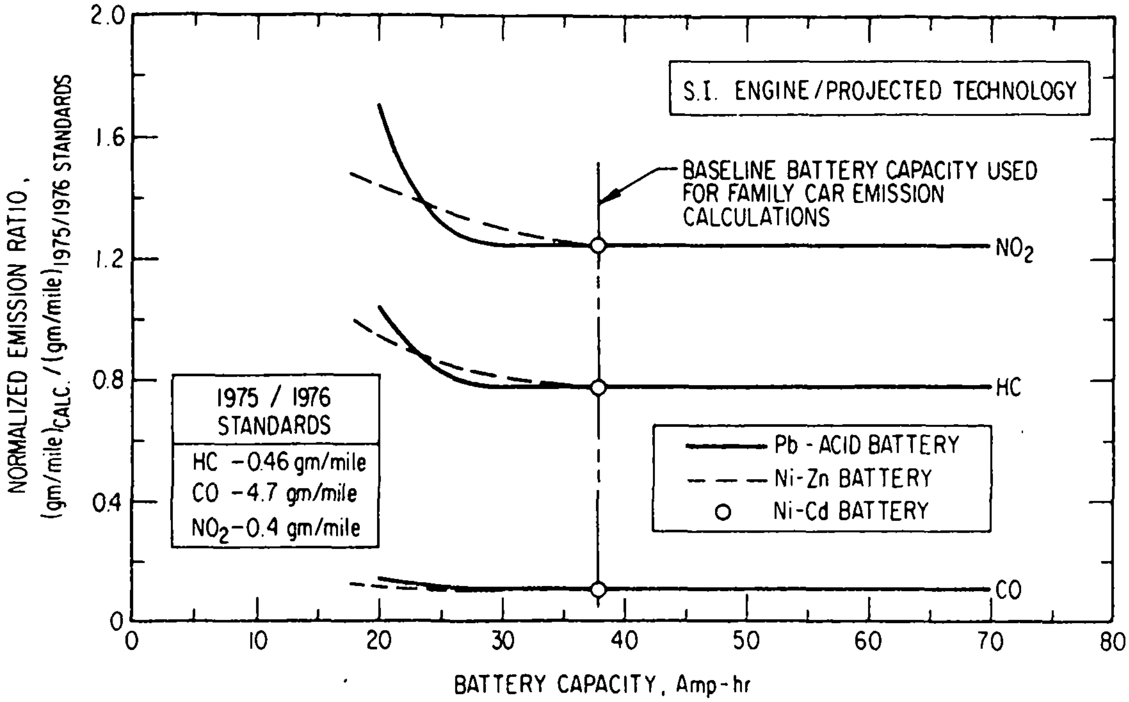


Figure 11-61. Effect of Battery Capacity and Type on HC, CO, and NO_2 Emissions - Family Car/DHEW Cycle - Series Configuration

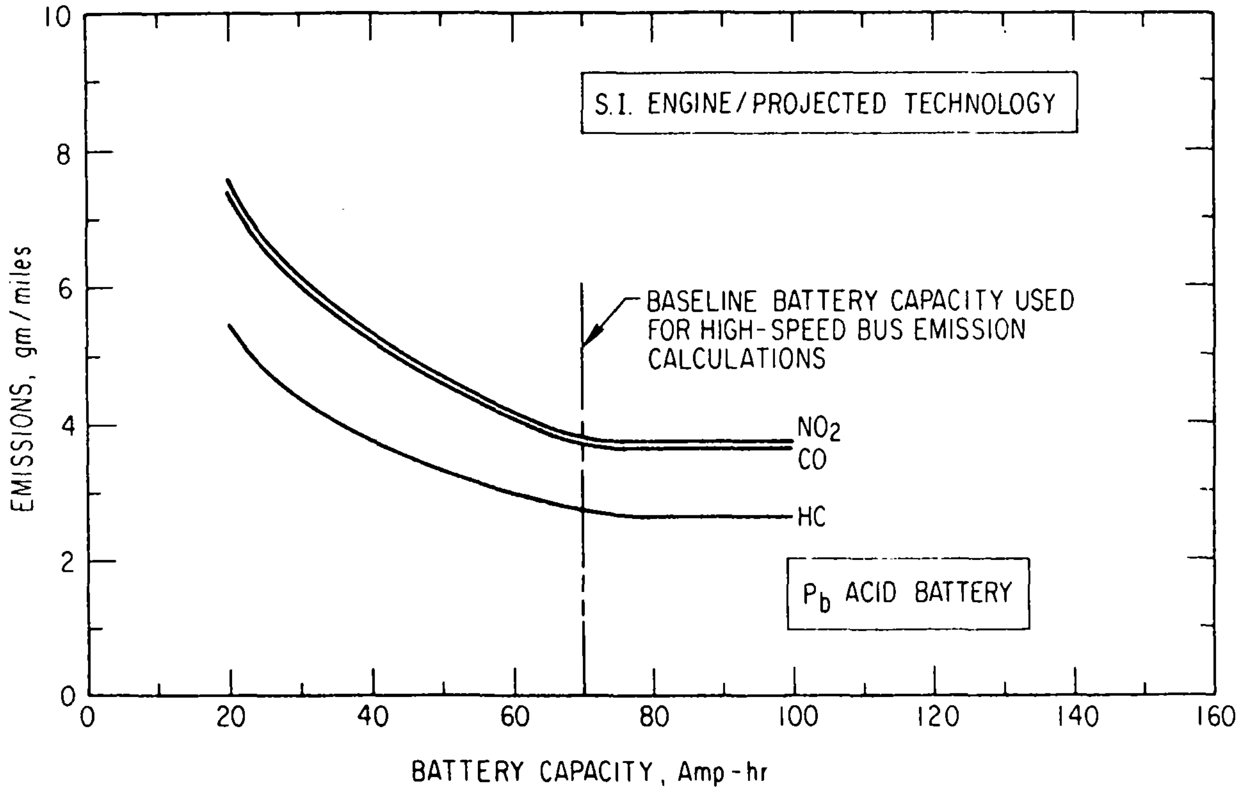


Figure 11-62. Effect of Battery Capacity on HC, CO, and NO_2 Emissions- High-speed Bus

	Series		Parallel	
	Current Technology	Projected Technology	Current Technology	Projected Technology
HC	Fig. 11-63	Fig. 11-66	Fig. 11-69	Fig. 11-72
CO	11-64	11-67	11-70	11-73
NO_2	11-65	11-68	11-71	11-74

In the case of the series configuration with current technology (Figs. 11-63 to 11-65), all heat engines except the gas turbine and diesel indicate a fairly uniform relationship between drive motor average efficiency and emission level. Using the spark ignition engine for illustration purposes, increasing the drive motor efficiency from 80 to 100 percent reduced all emissions (HC, CO, and NO_2) by 14 percent. Decreasing drive motor efficiency from the 80-percent baseline to 50 percent increased all emissions by 44 percent.

The gas turbine and diesel reflect emission-specie-dependent relationships between drive motor efficiency and emission level because of their unique part-load emission characteristics, as defined in Section 9. As can be seen in Fig. 11-64, gas turbine CO emissions actually increase with increasing drive motor efficiency. The NO_2 emissions for gas turbines (Fig. 11-65) are much more sensitive to drive motor efficiency than the other heat engine classes. Using projected technology heat engine capability (Figs. 11-66 to 11-68), the results are very similar to the current technology case just discussed.

When the parallel configuration is considered (Figs. 11-69 to 11-71 for current technology; Figs. 11-72 to 11-74 for projected technology), the results are the same except for absolute values. Again using the spark ignition engine for illustrative purposes (current technology), increasing the drive motor efficiency to 100 percent reduced all emissions 10 percent (as opposed to 14 percent in the series case), while decreasing the efficiency to 50 percent increased emissions 22.5 to 25 percent (as opposed to 44 percent for the series case). Therefore, the parallel configuration is less

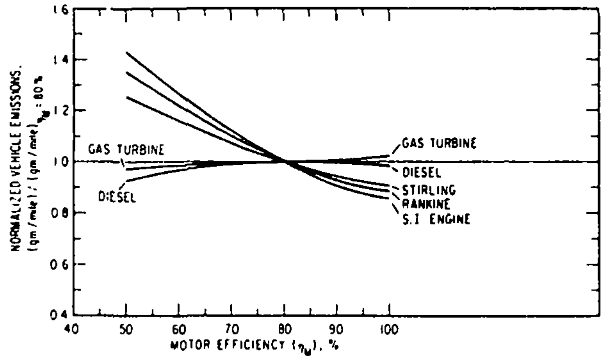


Figure 11-65. Effect of Drive Motor Efficiency on HC Emissions - Family Car/DHEW Cycle - Series Configuration - Current Technology

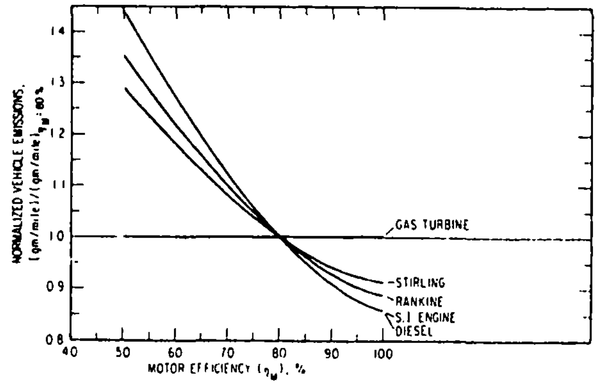


Figure 11-66. Effect of Drive Motor Efficiency on HC Emissions - Family Car/DHEW Cycle - Series Configuration - Projected Technology

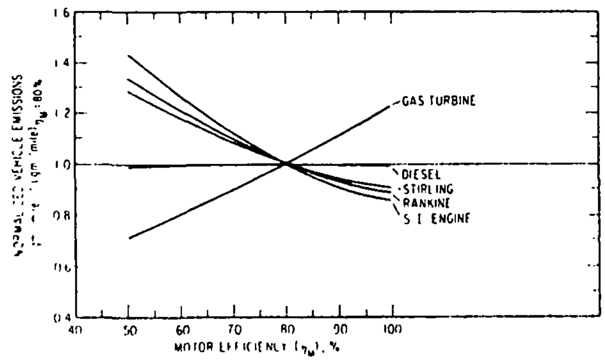


Figure 11-6[illegible]. Effect of Drive Motor Efficiency on CO Emissions - Family Car/DHEW Cycle - Series Configuration - Current Technology

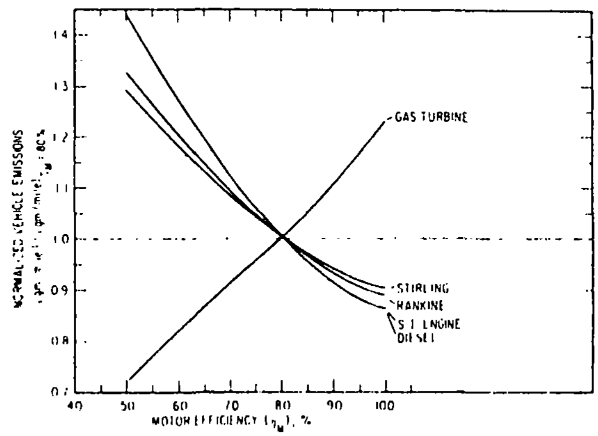


Figure 11-67. Effect of Drive Motor Efficiency on CO Emissions - Family Car/DHEW Cycle - Series Configuration - Projected Technology

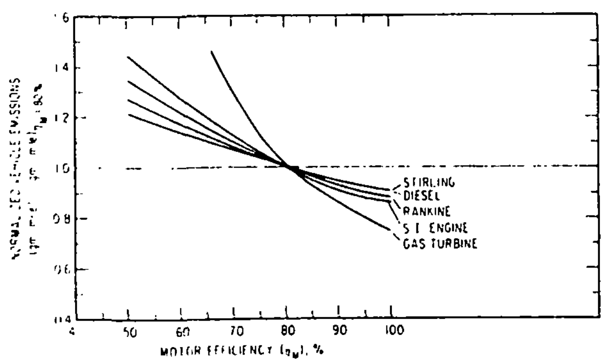


Figure 11-[illegible]. Effect of Drive Motor Efficiency on NO_x Emissions - Family Car/DHEW Cycle - Series Configuration - Current Technology

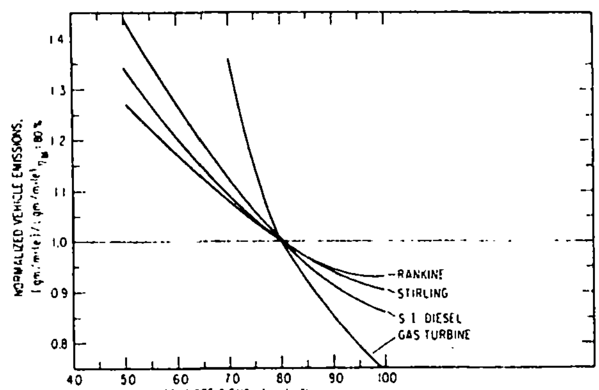


Figure 11-6[illegible]. Effect of Drive Motor Efficiency on NO_x Emissions - Family Car/DHEW Cycle - Series Configuration - Projected Technology

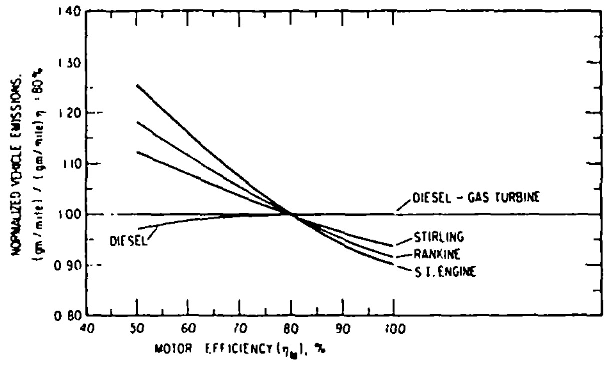


Figure 11-69. Effect of Drive Motor Efficiency on HC Emissions - Family Car/DHEW Cycle - Parallel Configuration - Current Technology

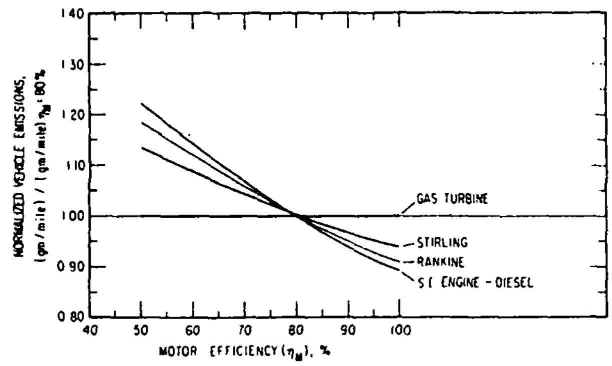


Figure 11-72. Effect of Drive Motor Efficiency on HC Emissions - Family Car/DHEW Cycle - Parallel Configuration - Projected Technology

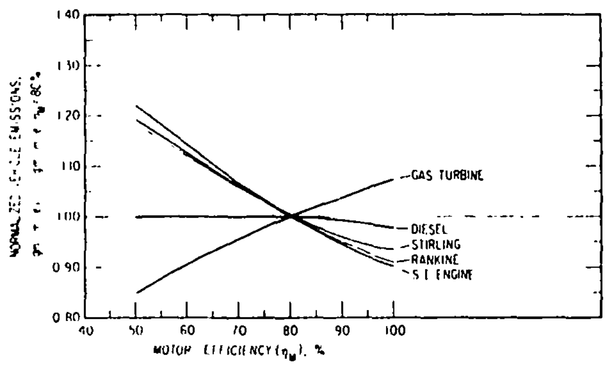


Figure 11-70. Effect of Drive Motor Efficiency on CO Emissions - Family Car/DHEW Cycle - Parallel Configuration - Current Technology

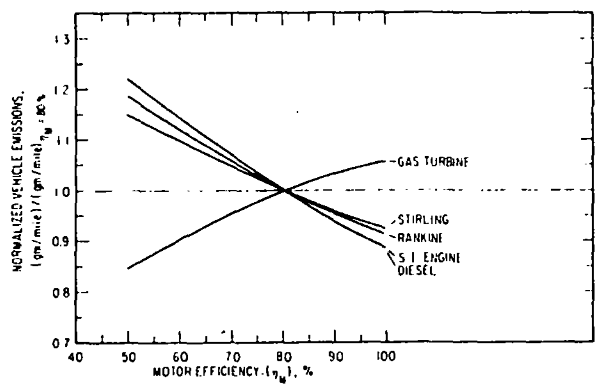


Figure 11-73. Effect of Drive Motor Efficiency on CO Emissions - Family Car/DHEW Cycle - Parallel Configuration - Projected Technology

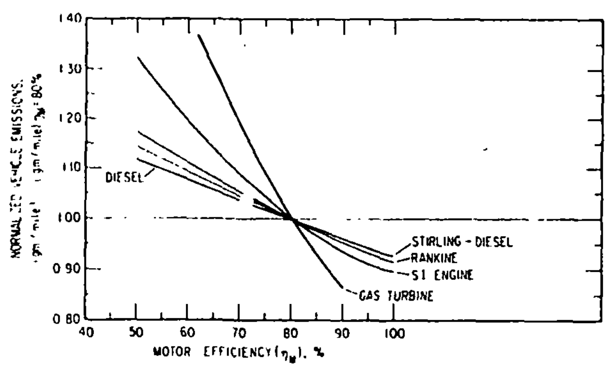


Figure 11-71. Effect of Drive Motor Efficiency on NO_2 Emissions - Family Car/DHEW Cycle - Parallel Configuration - Current Technology

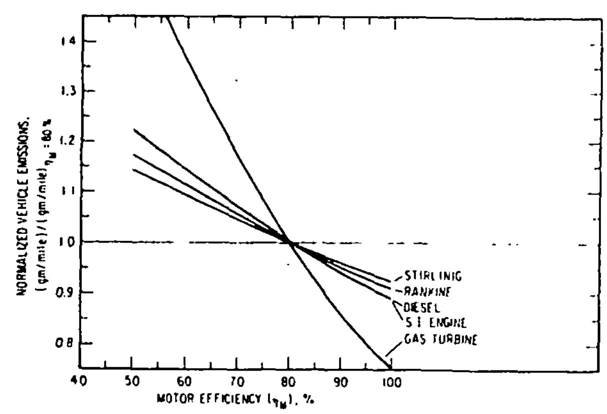


Figure 11-74. Effect of Drive Motor Efficiency on NO_2 Emissions - Family Car/DHEW Cycle - Parallel Configuration - Projected Technology

sensitive to drive motor efficiency effects. This is because only acceleration power is affected by motor efficiency in the parallel configuration whereas all power to the wheels is affected in the series configuration.

11.3.1.6 Type of Emission Driving Cycle

All of the baseline emission calculations for the family car and commuter car were made with computer runs simulating the DHEW driving cycle (See Section 3). While this is the proposed Federal test cycle requirement, it is known that some urban areas exhibit markedly different "average" or "typical" driving cycle profiles. One such different cycle is the New York City cycle (profile shown in Fig. 3-1). To assess the effects of such driving cycle profile variations, computer simulations were made with the family car driven over the New York cycle. As for the DHEW driving cycle, the heat engine power output was adjusted to assure that the battery final state-of-charge matched the initial state-of-charge. Figures 11-75 through 11-80 illustrate the results of these calculations in a comparative fashion, wherein the ordinate scale (normalized emissions) is the ratio of New York cycle emissions divided by DHEW cycle emissions. Figures 11-75, 11-76, and 11-77 present emission results (HC, CO, and NO_2, respectively) for current technology, while Figs. 11-78, 11-79, and 11-80 present similar values for the projected technology case.

Examination of the figures shows that driving profile of the New York cycle results in a 45- to 55-percent increase in vehicle emissions over those ocurring during the DHEW driving cycle. Although the various heat engine classes show minor deviations from one another (the gas turbine being the most singular in deviation), the series configuration (exclusive of the gas turbine) results in 45- to 54-percent increases, while the parallel configuration (exclusive of the gas turbine) results in 49- to 55-percent increases. In general, the parallel configuration, for any given heat engine, results in a greater sensitivity to the New York driving cycle. However, the differences are not sufficiently high to alter the fact that the absolute emission values for

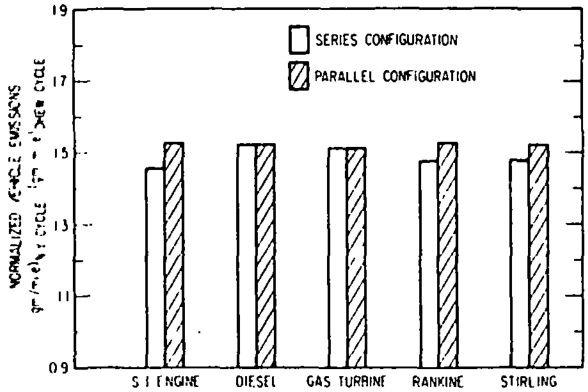


Figure 11-75. Effect of New York Cycle on HC Emissions - Family Car/DHEW Cycle - Current Technology

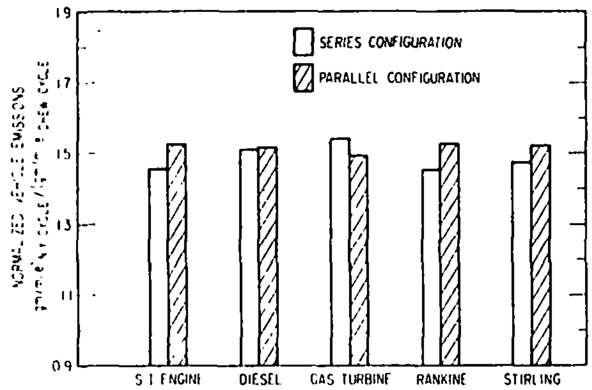


Figure 11-76. Effect of New York Cycle on CO Emissions - Family Car/DHEW Cycle - Current Technology

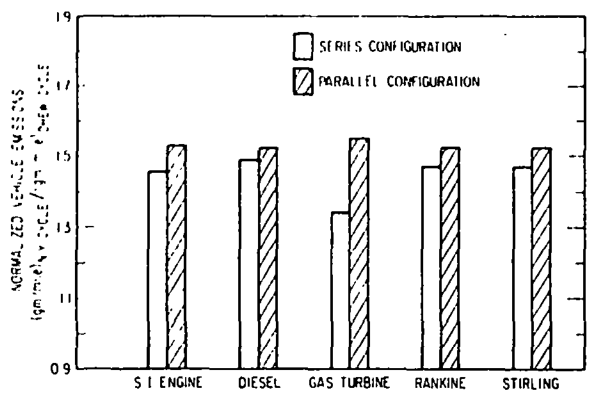


Figure 11-77. Effect of New York Cycle on NO_2 Emissions - Family Car/DHEW Cycle - Current Technology

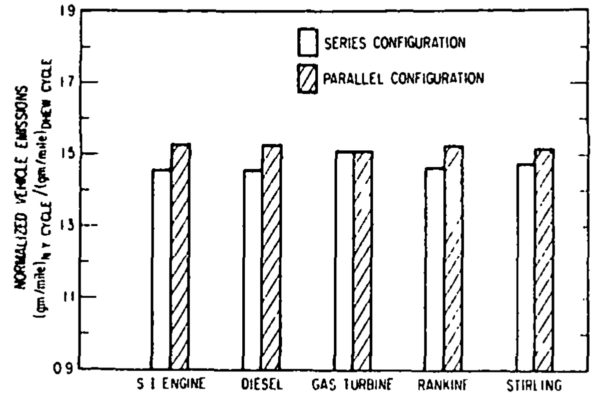


Figure 11-78. Effect of New York Cycle on HC Emissions - Family Car/DHEW Cycle - Projected Technology

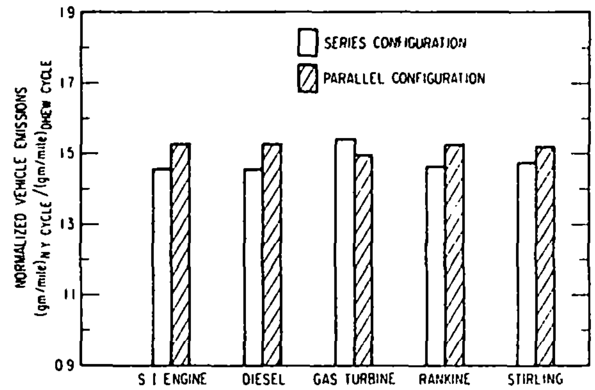


Figure 11-79. Effect of New York Cycle on CO Emissions - Family Car/DHEW Cycle - Projected Technology

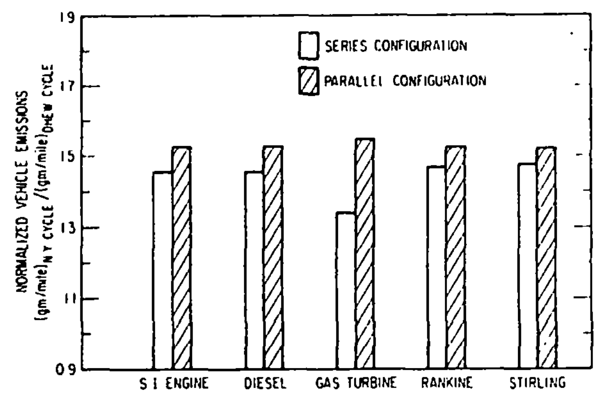


Figure 11-80. Effect of New York Cycle on NO_2 Emissions - Family Car/DHEW Cycle - Projected Technology

the parallel configuration are lower than for the series configuration on the New York cycle.

The increased emissions of the New York cycle apparently are the result of more accelerations, more decelerations, and more time stopped (than in the case of the DHEW cycle) which poses a greater energy demand per mile from the powerplant.

11.3.2 Effect on Battery Requirements

A number of battery requirement tradeoff analyses were made, utilizing the baseline series and parallel configurations and powertrain weights determined in Section 10. In terms of weight effects, two of the most meaningful parameters are the relationship between powertrain weight and required battery power density and energy density for a given vehicle. As shown in Section 10, baseline powertrain weights were determined for all subsystems/components except battery weights. For any given vehicle class and allowable weight for the powertrain, the weight available for batteries is uniquely established and the power density requirement determined by the peak power demand in the battery during vehicle maximum acceleration, and the energy-density requirement determined by the necessary installed capacity.

Utilizing this approach, the following tradeoffs were made.

11.3.2.1 Effect of Available Powertrain Weight on Required Battery Power Density

The baseline tabular data developed in Tables 10-7 through 10-18 for the powerplant weights of the various vehicle classes and heat engines (series and parallel configurations) were utilized to develop parametric displays of the effect of available powertrain weight on the battery power density required for each vehicle. These parametric results are shown in Figs. 11-81 through 11-86 for the series configurations and in Figs. 11-87 through 11-92 for the parallel configurations. In each case, the study baseline for allowable powertrain weight is indicated. Battery power density and energy density requirements for any of the five classes of heat engines at any other allowable powertrain total weight can be determined by inspection.

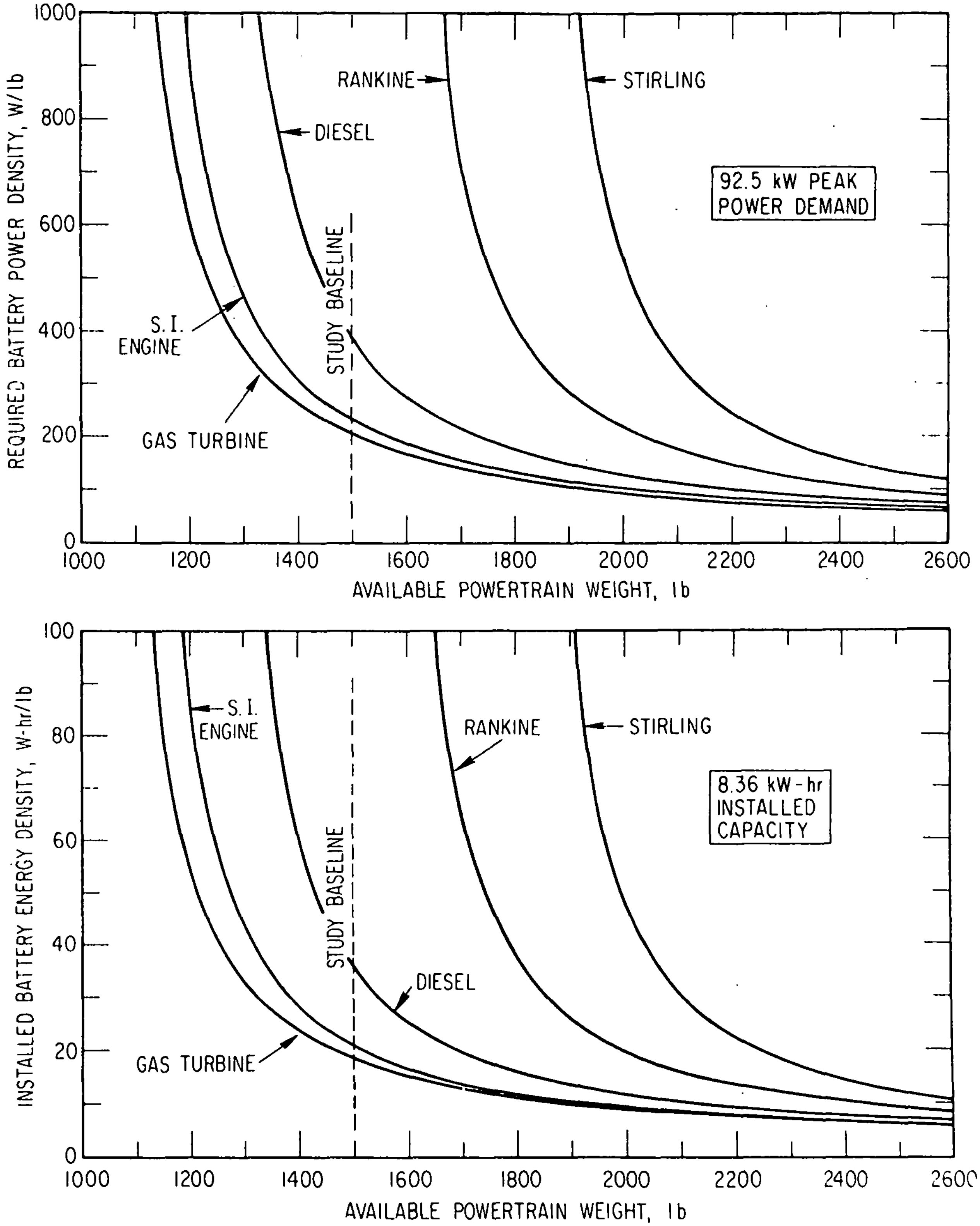
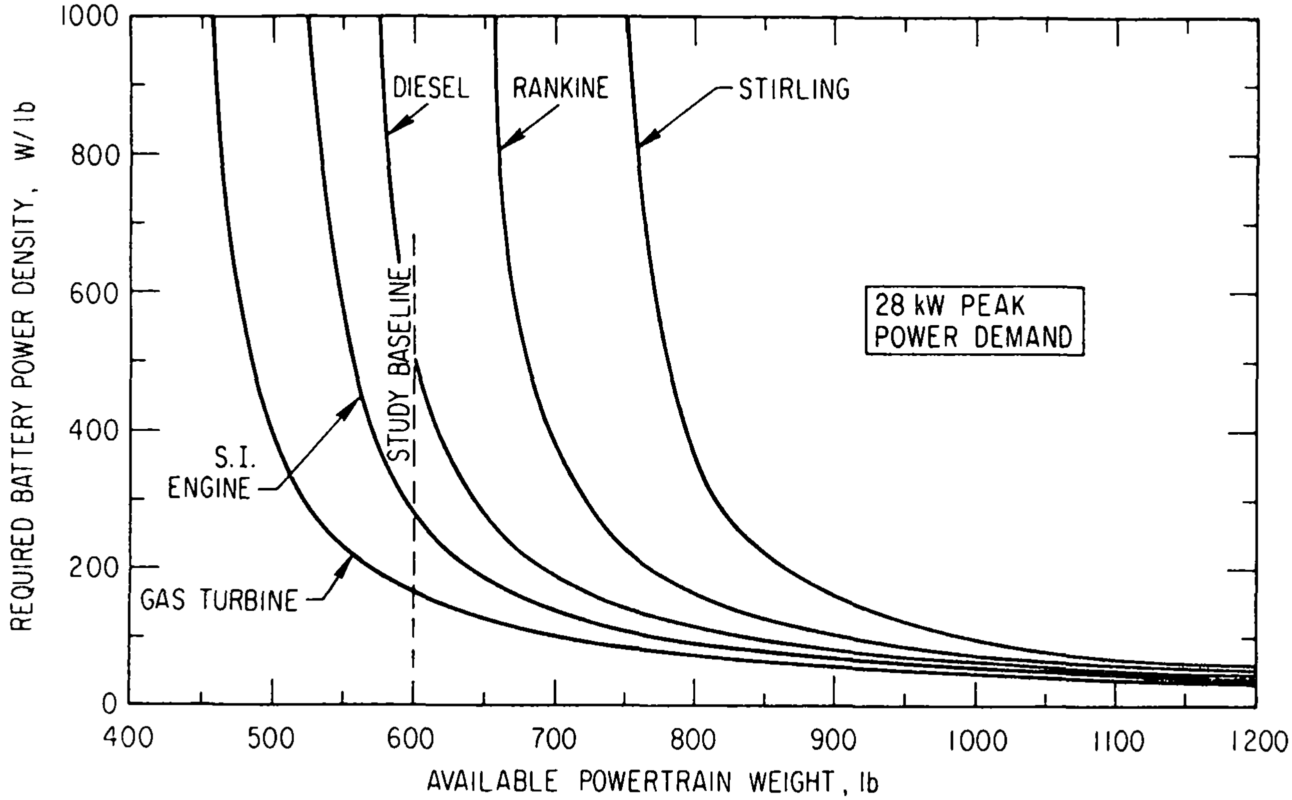


Figure 11-81. Effect of Powertrain Weight on Battery Requirements
Family Car - Series Configuration



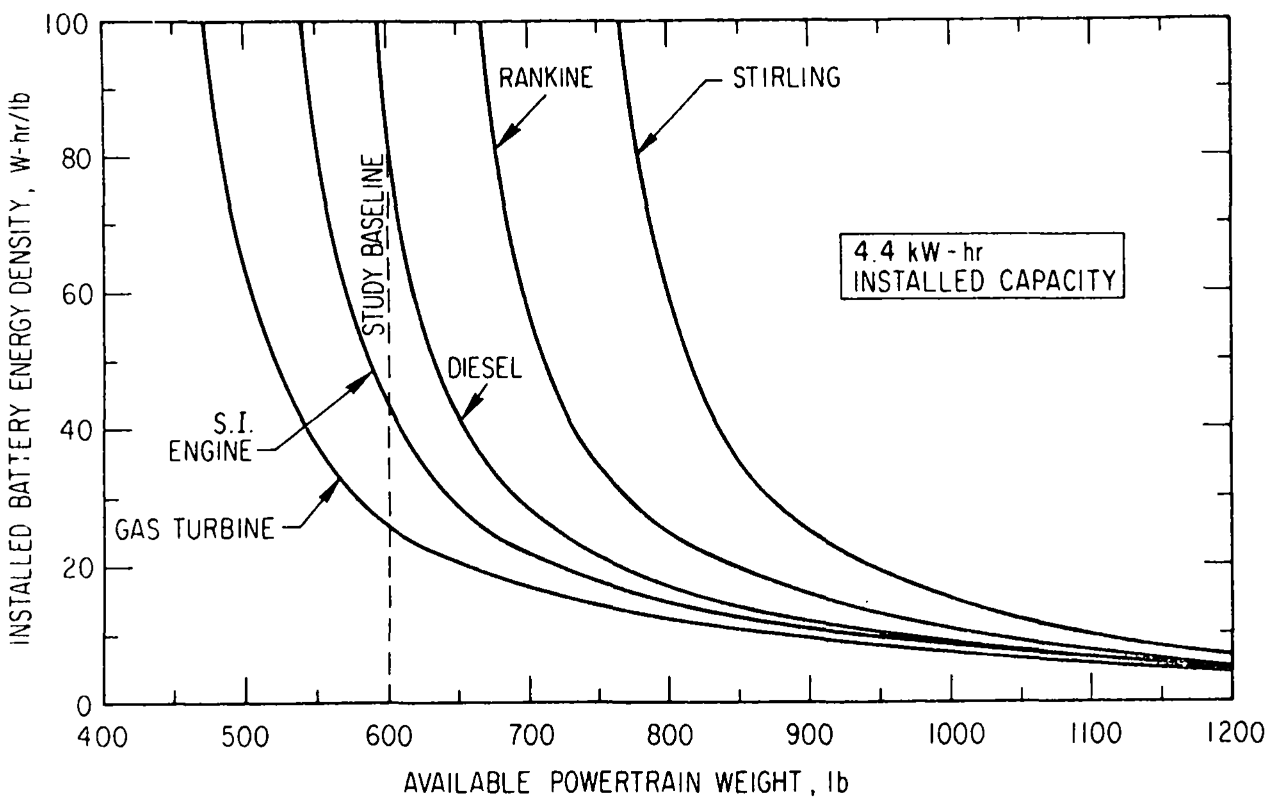


Figure 11-82. Effect of Powertrain Weight on Battery Requirements Commuter Car - Series Configuration

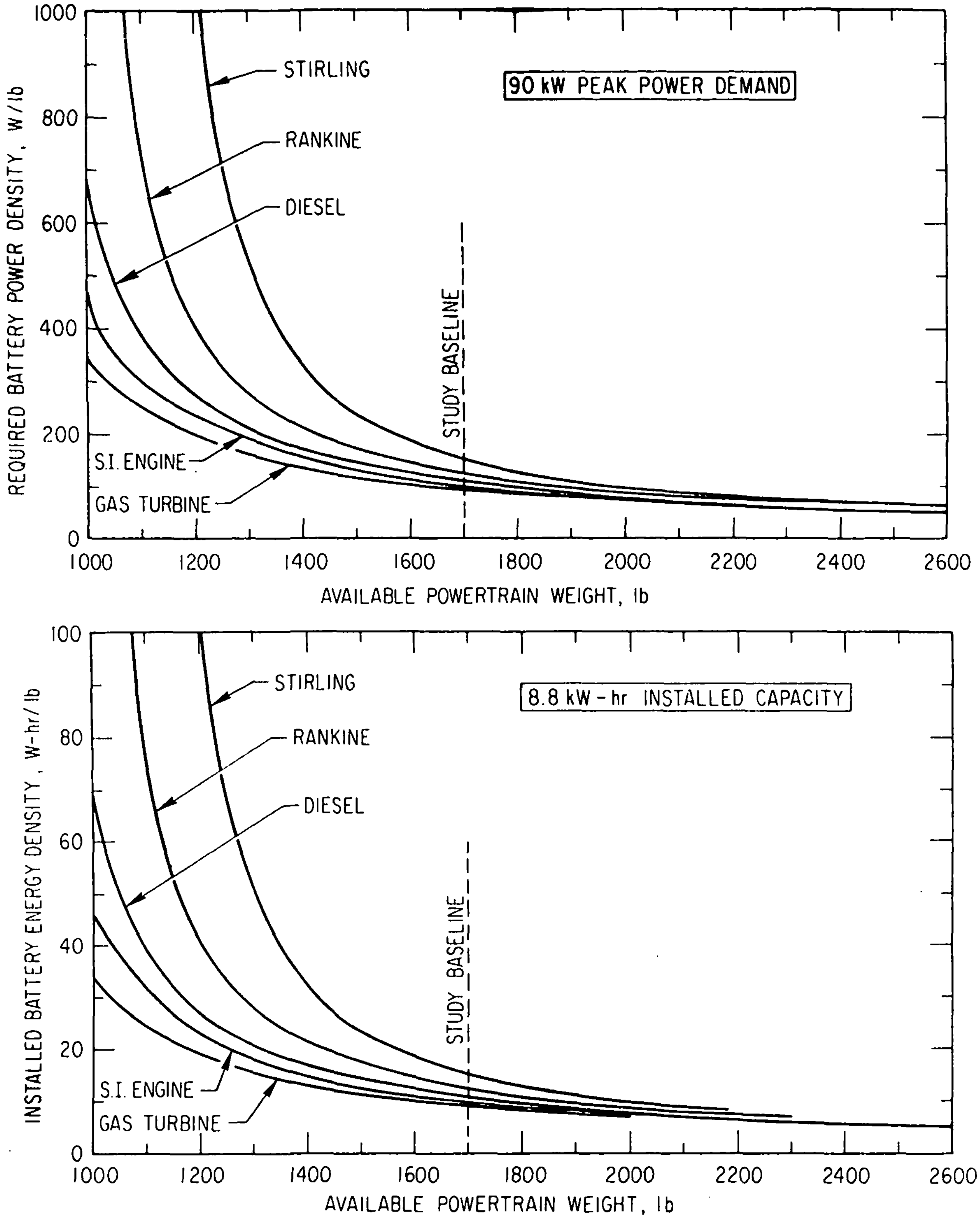


Figure 11-83. Effect of Powertrain Weight on Battery Requirements Low-Speed Van - Series Configuration

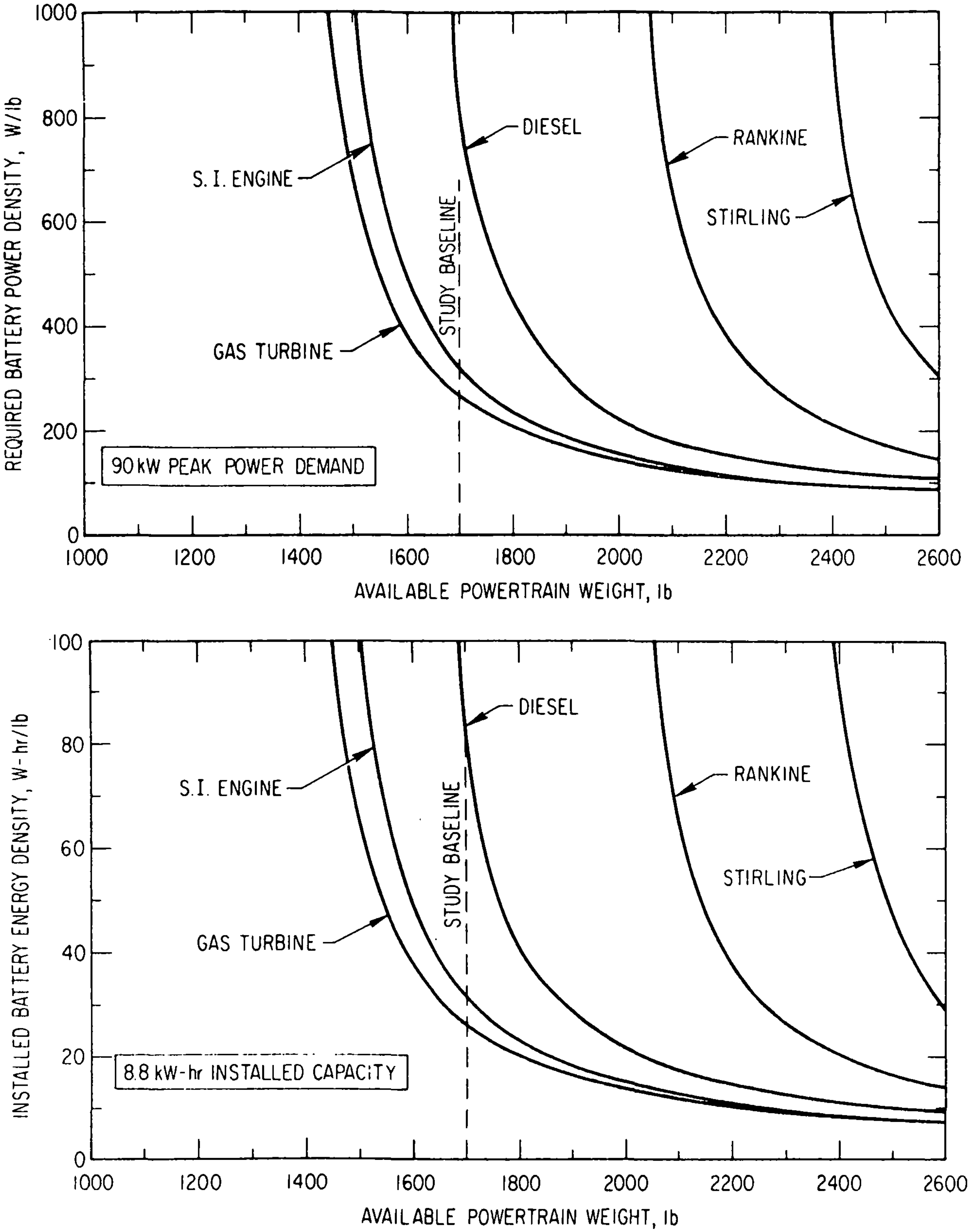


Figure 11-84. Effect of Powertrain Weight on Battery Requirements
High-Speed Van - Series Configuration

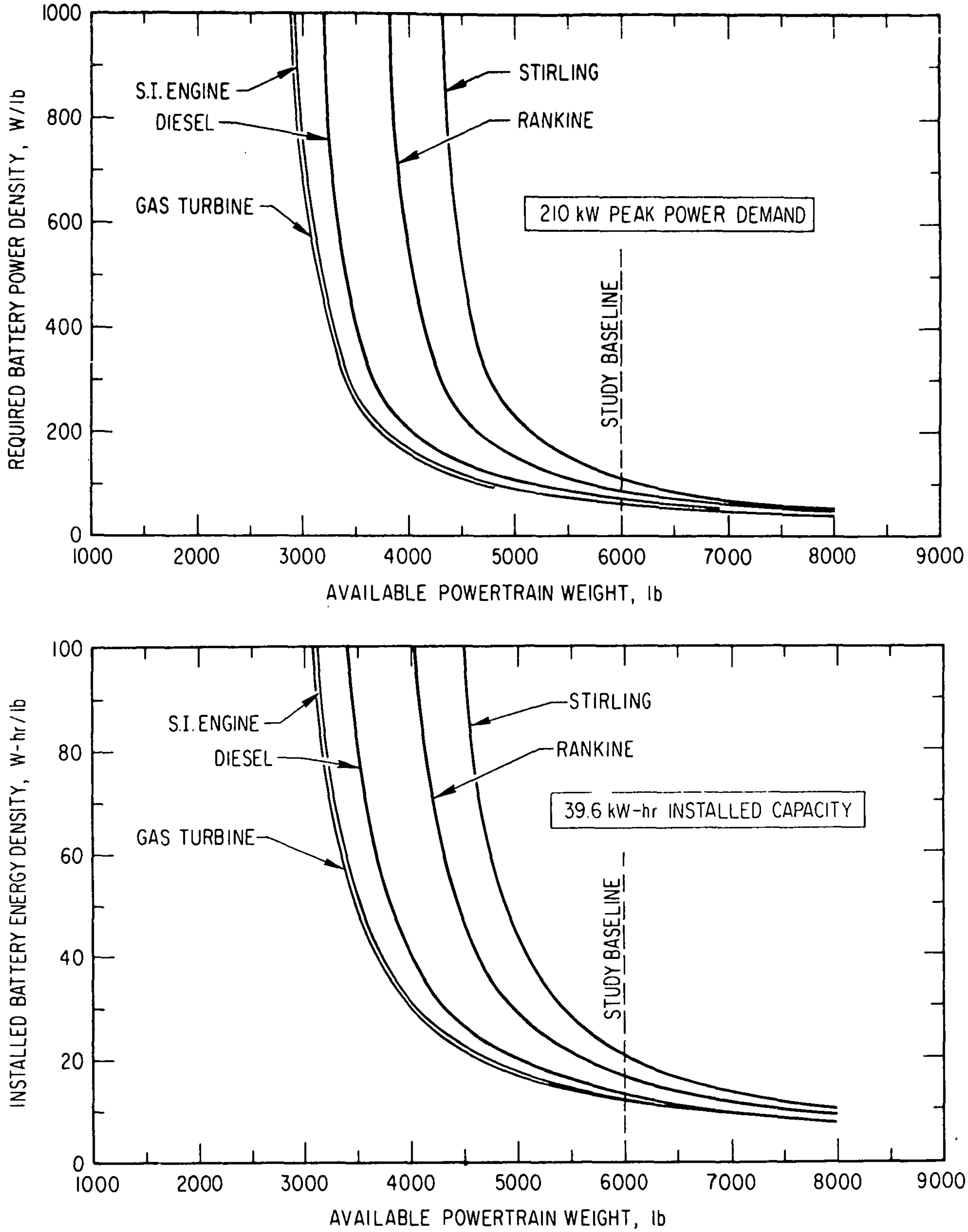


Figure 11-85. Effect of Powertrain Weight on Battery Requirements
Low-Speed Bus - Series Configuration

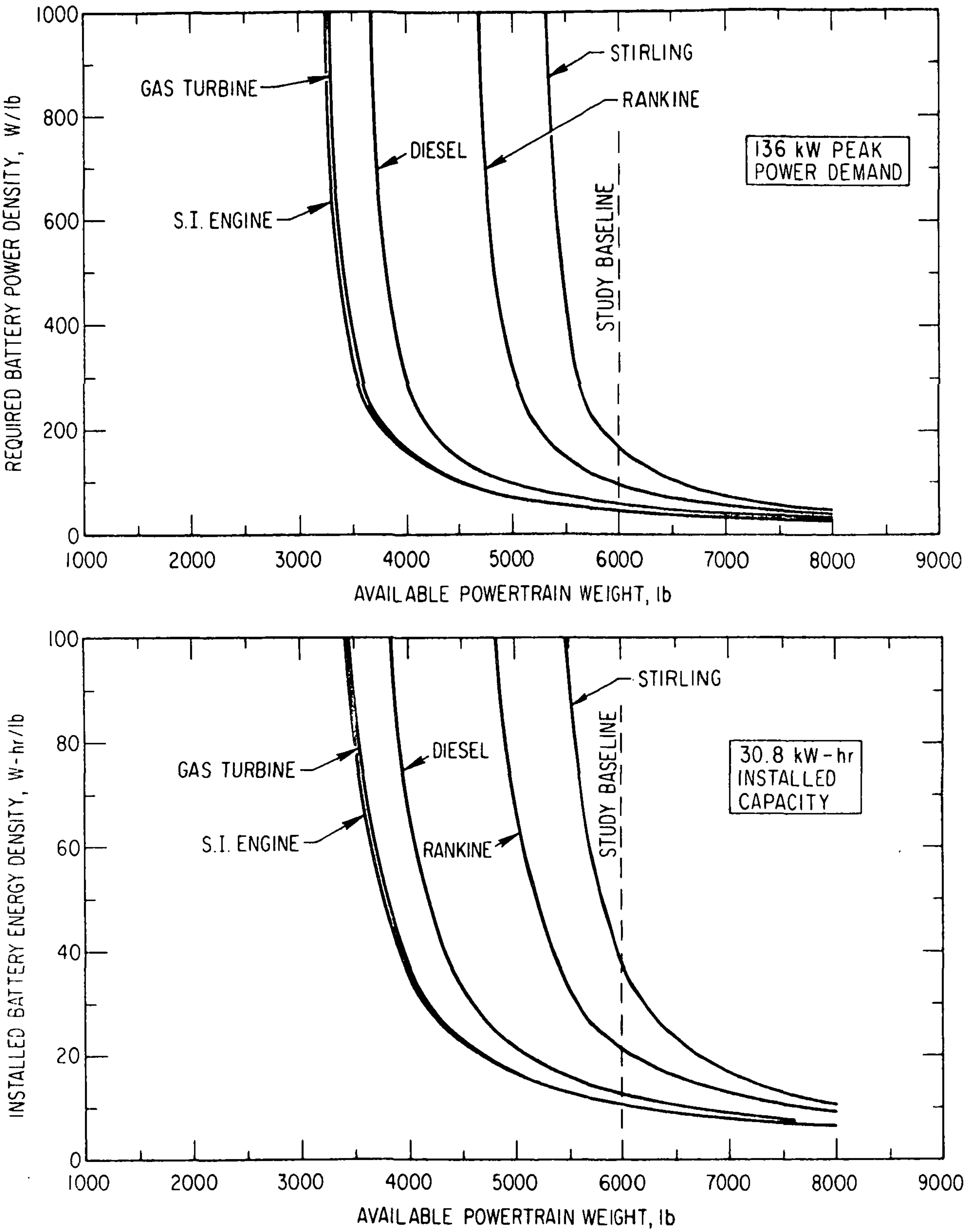


Figure 11-86. Effect of Powertrain Weight on Battery Requirements
High-Speed Bus - Series Configuration

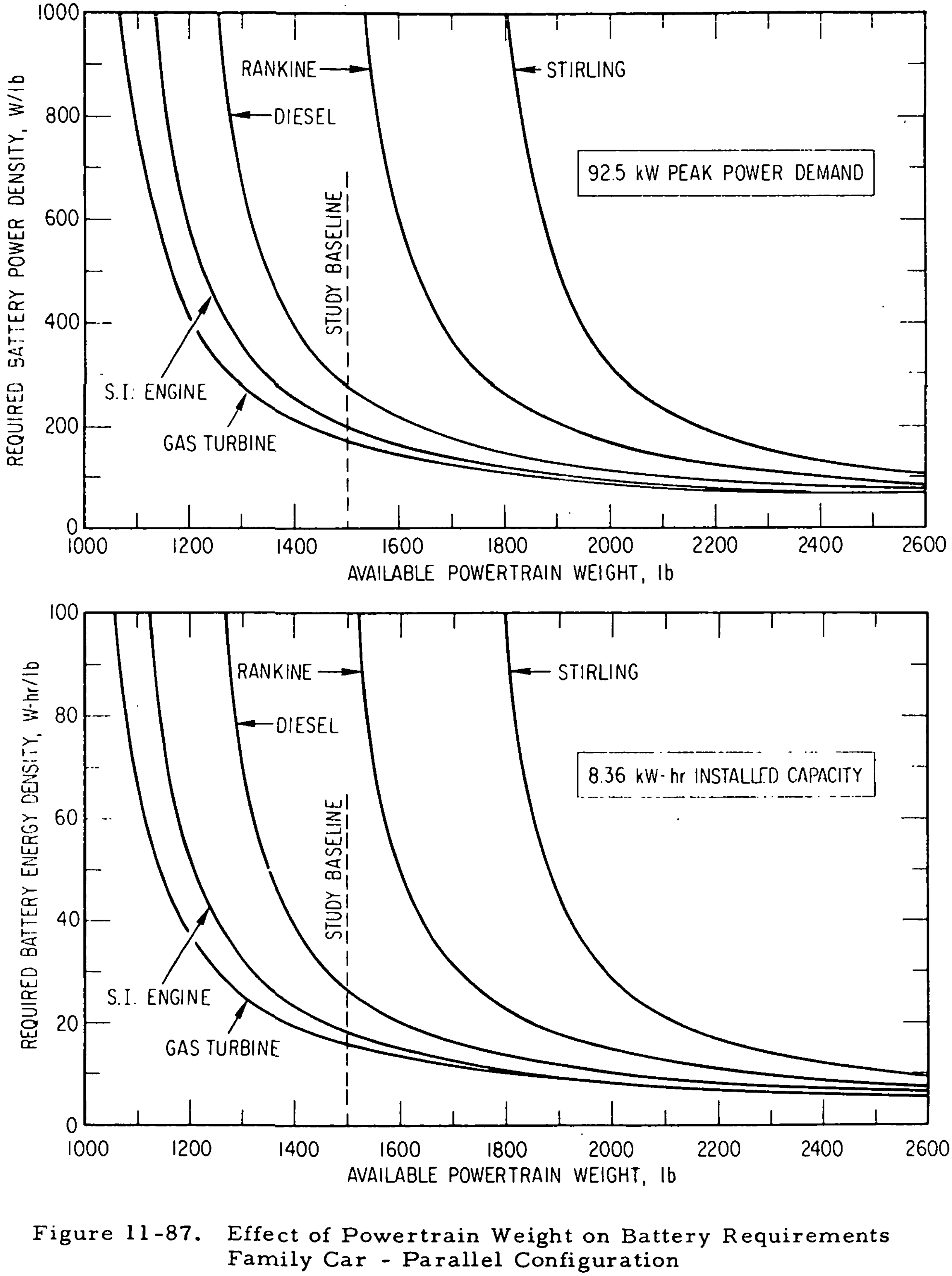
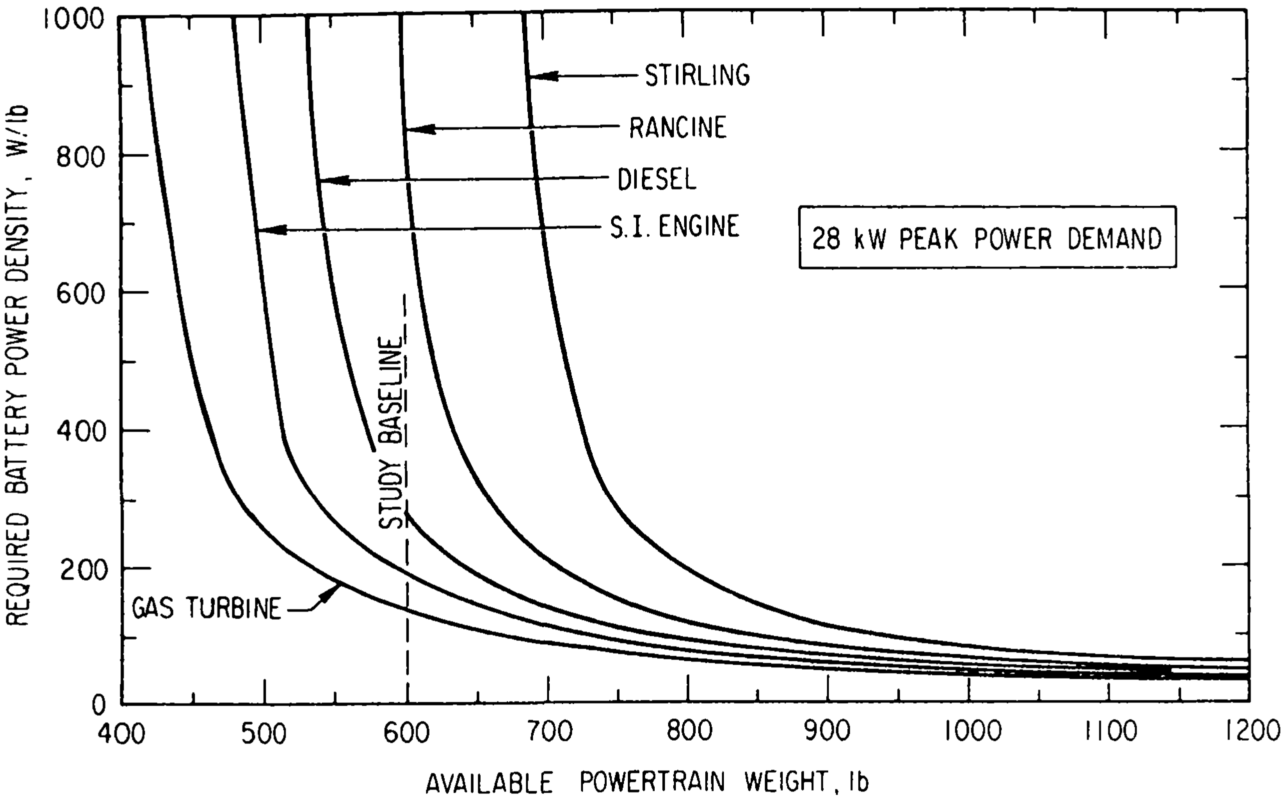


Figure 11-87. Effect of Powertrain Weight on Battery Requirements Family Car - Parallel Configuration



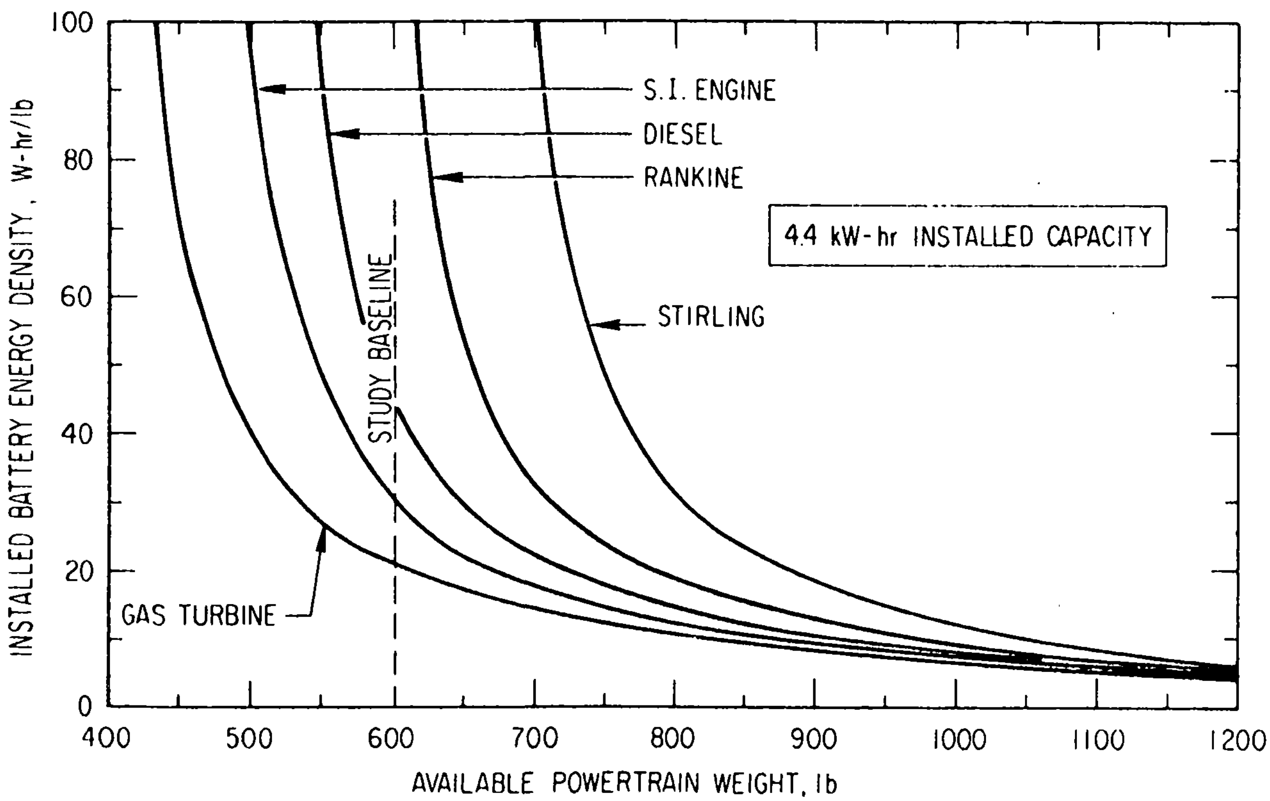
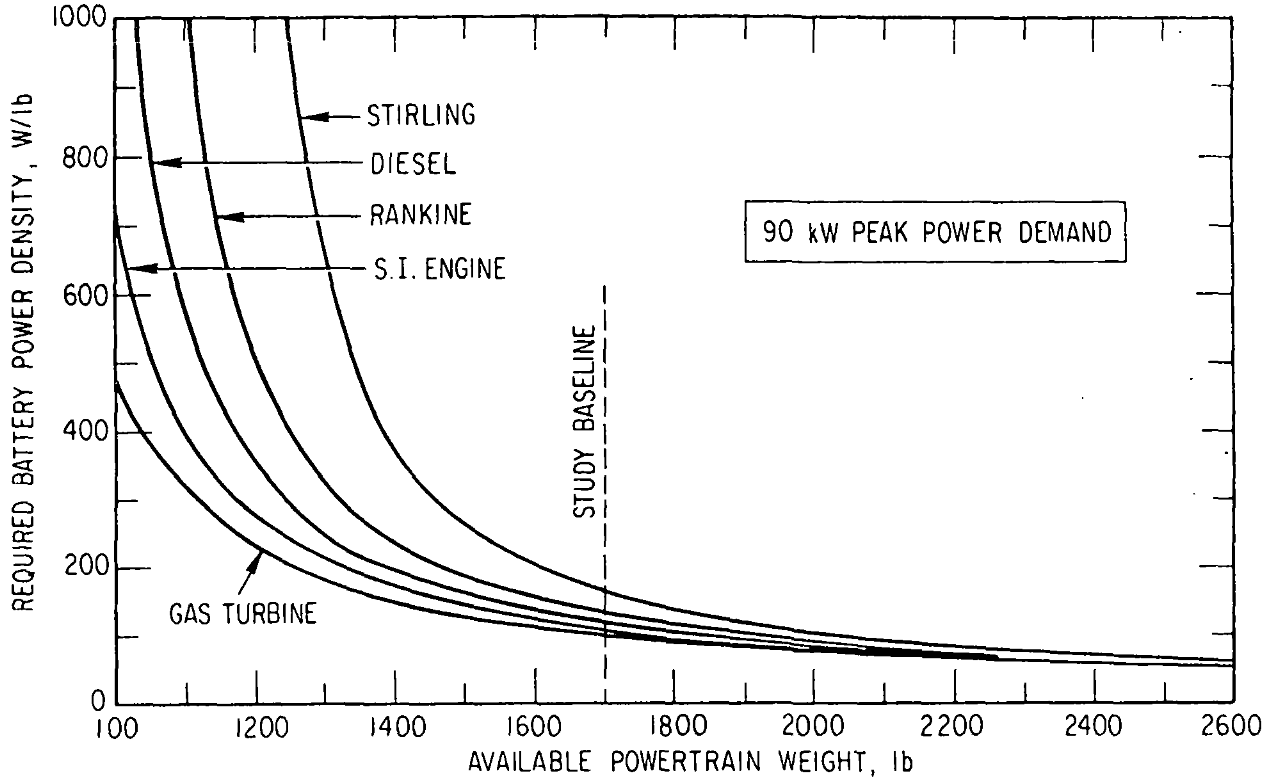


Figure 11-88. Effect of Powertrain Weight on Battery Requirements Commuter Car - Parallel Configuration



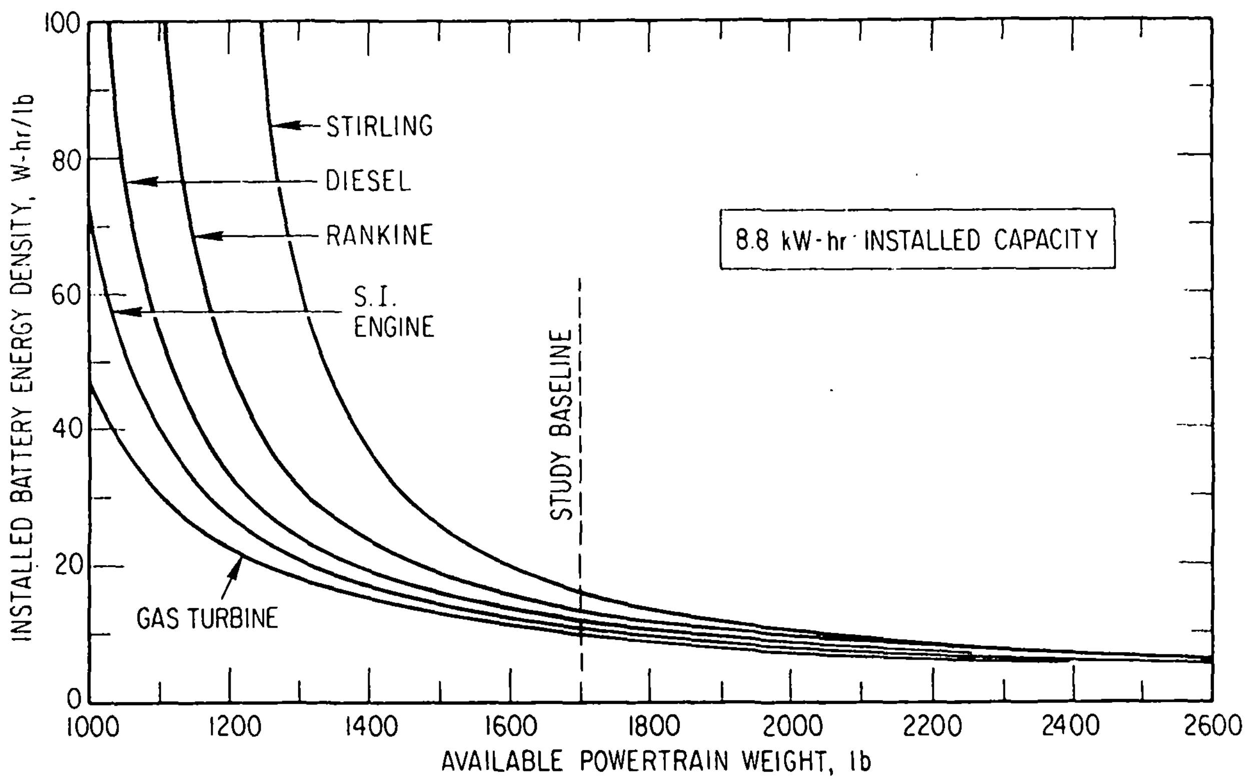


Figure 11-89. Effect of Powertrain Weight on Battery Requirements
Low-Speed Van - Parallel Configuration

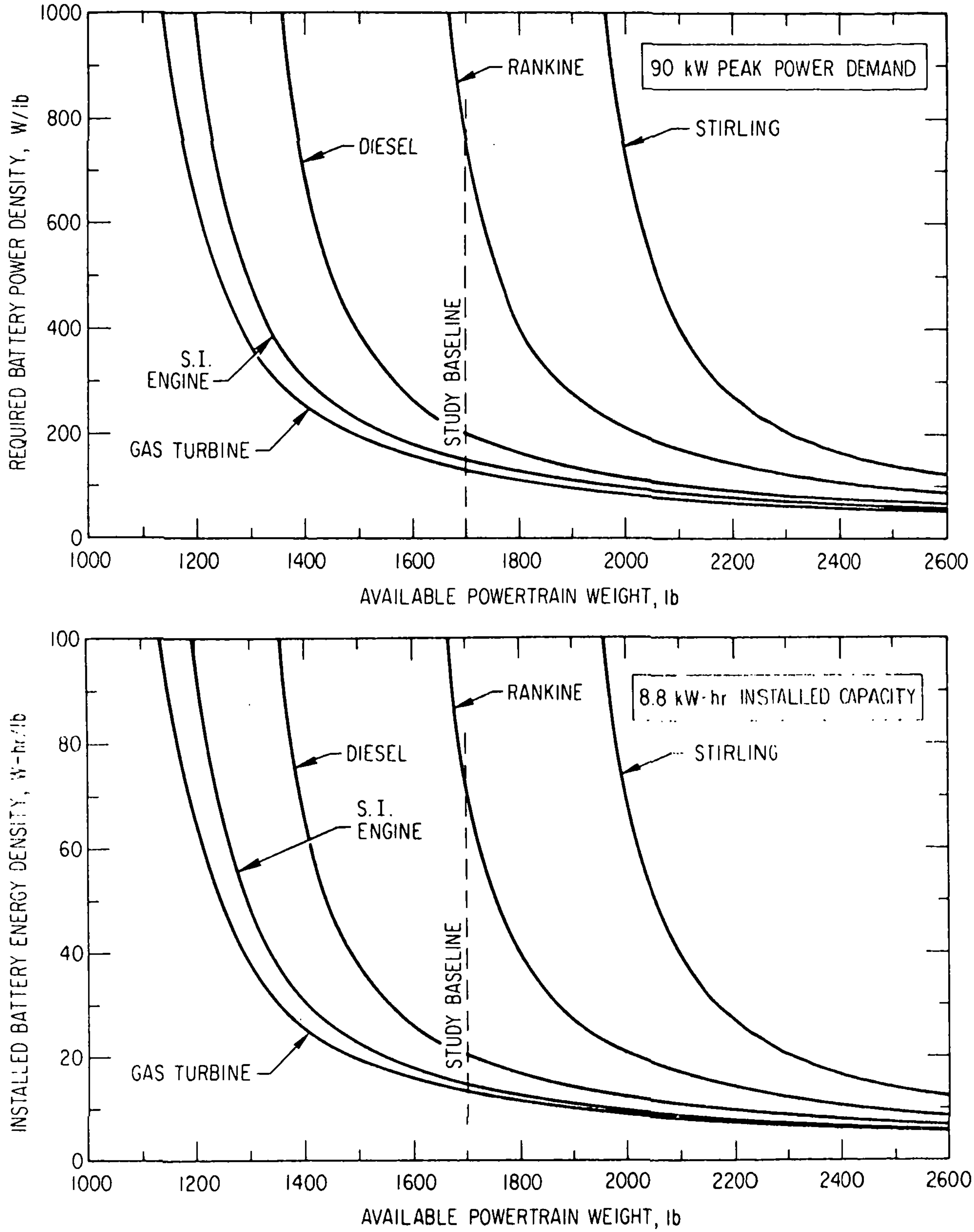


Figure 11-90. Effect of Powertrain Weight on Battery Requirements High-Speed Van - Parallel Configuration

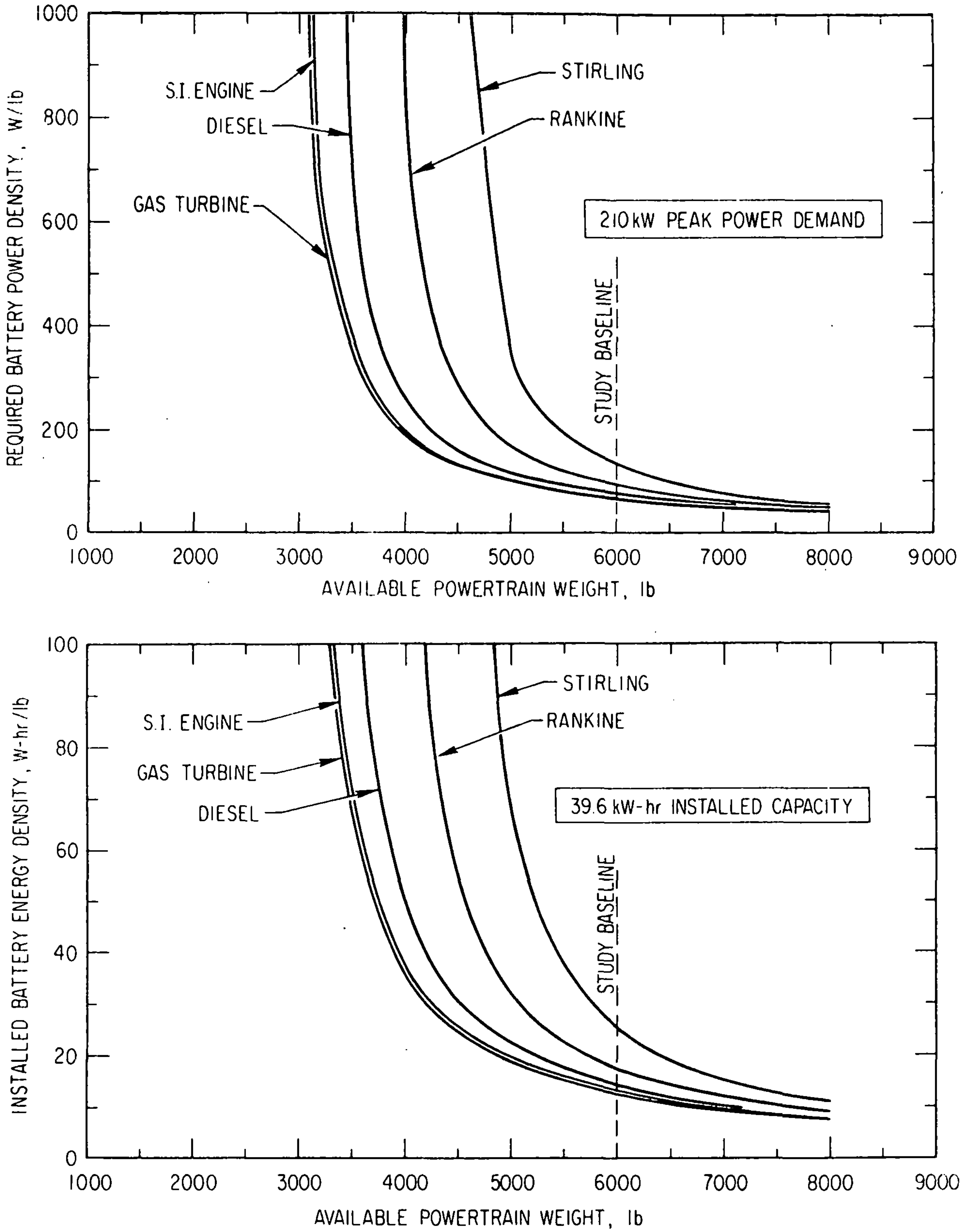
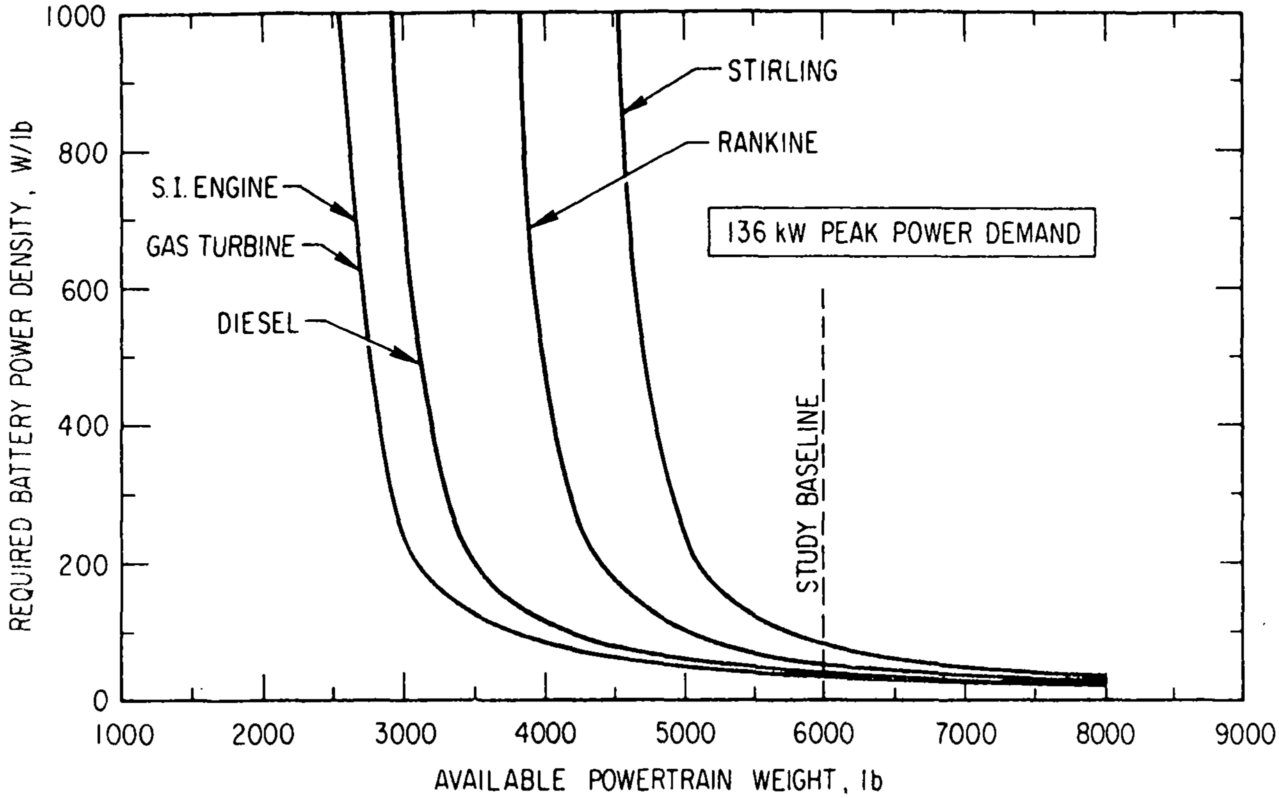


Figure 11-91. Effect of Powertrain Weight on Battery Requirements
Low-Speed Bus - Parallel Configuration



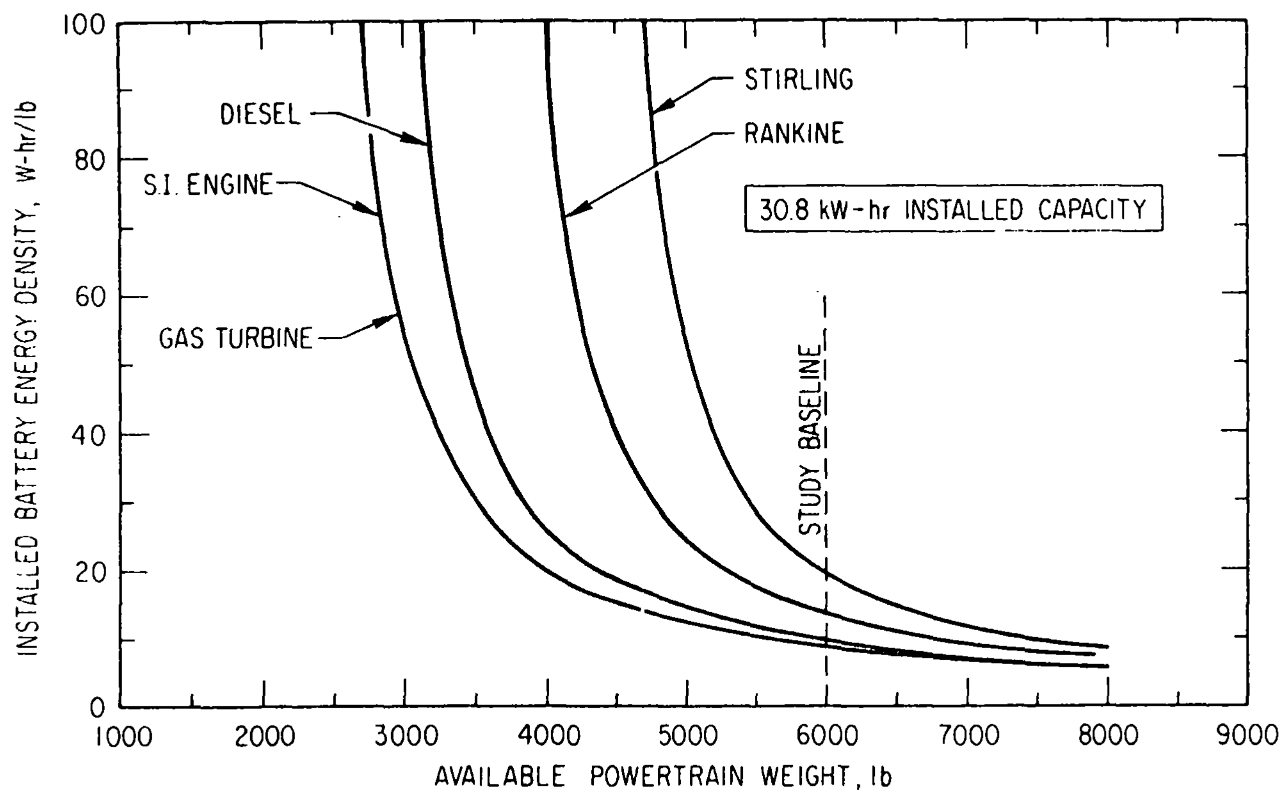


Figure 11-92. Effect of Powertrain Weight on Battery Requirements High-Speed Bus - Parallel Configuration

Alternatively, the curves can also be used to assess the impact on required battery power density and energy density of reducing powertrain subsystem weights below those baseline values presented in Tables 10-7 through 10-18. For example, in Fig. 11-81 (family car, series mode) the study baseline was 1500-lb allowable total powertrain weight. At that value, the battery power density requirement, if the spark ignition engine is assumed, is 232 watts/lb. Assuming a 200-lb weight reduction in the spark ignition engine weight itself (e.g., through use of the Wankel engine), this would be equivalent, as far as the battery is concerned, of having an additional 200 lb available for batteries. The effect of this 200-lb battery weight increase can be observed by entering the figure at 1700-lb available powertrain weight instead of 1500 lb, with a resultant power density requirement of 155 watts/lb.

Conversely, powerplant weight increase effects are determinable by entering the figure at an available powertrain weight commensurately less than the baseline value.

11.3.2.2 Comparison of Series Versus Parallel Configuration Effects on Battery Power Density Requirements

Because of the many variables involved in the cases discussed above (series mode, parallel mode, five heat engines, six vehicle classes), it is worthwhile to briefly compare the significant differences occasioned by the choice of series versus parallel configuration for the vehicle classes. For this purpose, the conventional spark ignition engine powerplant was selected as illustrative of the trends. Table 11-4 summarizes, from Tables 10-7 through 10-18, the baseline powertrain weight (less batteries), the allowable battery weight (baseline case), and the resulting required battery power density and energy density to meet the vehicle peak acceleration power demands and installed energy capacity requirements.

As can be seen, for all vehicles except the low-speed van and the low-speed bus, the use of the parallel configuration results in a reduction of power density requirement ranging from 13 to 52 percent, depending upon the specific vehicle class. Conversely, use of the parallel configuration in the

Table 11-4. Battery Requirements - Series Versus Parallel Configuration, S. I. Engine

VEHICLE CLASS/MODE / AREA	FAMILY CAR		COMMUTER CAR		DELIVERY/POSTAL VAN				INTRA-CITY BUS			
					Low Speed		High Speed		Low Speed		High Speed	
	Series	Parallel	Series	Parallel	Series	Parallel	Series	Parallel	Series	Parallel	Series	Parallel
BASELINE POWERTRAIN WEIGHT (lb) (Less Batteries)	1102	1040	499	455	805	876	1406	1108	2714	2943	3098	2434
NET WEIGHT ADVANTAGE (lb)		62		44	71			298	229			664
ALLOWABLE BATTERY WEIGHT (lb) (Baseline Case)	398	460	101	145	895	824	294	592	3286	3057	2902	3566
REQUIRED BATTERY POWER DENSITY (w/lb)	232	201	279	193	101	109	306	152	64	69	47	38
REQUIRED BATTERY ENERGY DENSITY (w-hr/lb)	20	18.1	43.8	30.3	9.9	10.7	30	14.9	12.1	13	10.6	8.6
ELECTRIC DRIVE MOTOR WEIGHT* (lb)	337	250	133	83	156	170	462	170	816	831	968	170
SI ENGINE WEIGHT (lb)	335	319	180	171	205	196	357	345	478	468	626	614

*Without forced air cooling system.

low-speed van and bus results in increases in battery power density requirements of 9 and 8 percent, respectively.

In all vehicle classes, the same basic weight tradeoff phenomena are involved, as far as series mode versus parallel mode is concerned. Use of the parallel configuration has two very significant weight-advantageous characteristics (as opposed to the series configuration):

1. The electric drive motor is sized (100-percent rating or design point) at one-third the peak acceleration power demand of the vehicle (in the case of the series configuration, the drive motor is sized at maximum continuous power demand of vehicle).

2. The heat engine size is reduced (as mentioned in Section 10) due to the efficiency advantage assumed for the mechanical power transmission feature of the parallel configuration at maximum continuous vehicle power demands.

Table 11-4 also lists the drive motor and heat engine weights (from Tables 10-7 through 10-18 for the spark ignition engine examples shown. As can be seen in all cases except the low-speed van and low-speed bus, the parallel configuration does have reduced drive motor and heat engine weights.

However, in the case of the low-speed van and bus, the electric drive motor sized for maximum continuous operation in the series configuration is essentially the same motor size required for the parallel configuration, when sized for one-third maximum acceleration power demand (300-percent overload rating of electric drive motor). Therefore, in these two cases, use of the parallel configurations affords only slight heat engine weight reductions which are more than offset by drive line and transmission weights of the parallel configuration (not used in series) and result in a net increase in powerplant weight and battery power and energy density.

11.3.2.3 Effect of Drive Motor and Heat Engine Weights on Required Battery Power Density for the Family Car

To further illustrate the various interactions between battery power density sensitivity and powertrain subsystem weights, Fig. 11-93 is included to show parametrically the effects of electric drive motor and heat engine weights on

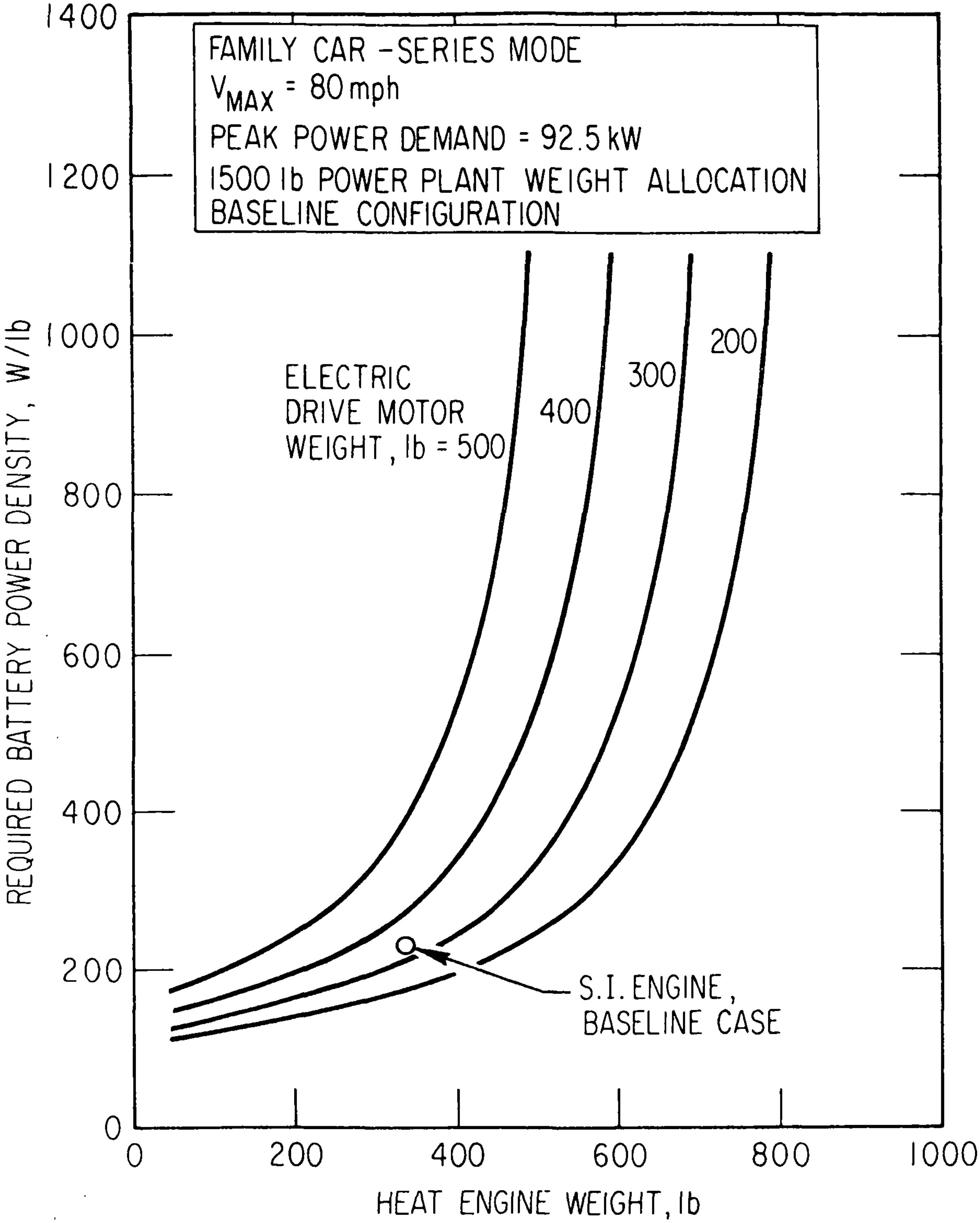


Figure 11-93. Effect of Drive Motor and Heat Engine Weights on Battery Power Density - Family Car - Series Configuration

power density requirements for the family car. The powertrain configuration is series and the figure is commensurate with the 1500-lb baseline powerplant weight allocation. All other subsystems are constant at the weight values given in Table 10-7.

The drive motor and heat engine (at the spark ignition engine baseline case point shown) represent two-thirds of the powerplant weight. Reducing both the drive motor and heat engine weights over the assumed baseline values by 50 percent results in decreasing the required battery power density by 48 percent.

11.3.2.4 Effect of Design Point Sizing on Battery Power Density Requirements for the Family Car

It has been recognized that the requirement, in the case of the family car, to meet the 40 mph/12-percent grade and 80-mph maximum cruise speed requirements is a most severe one and that substantial reductions in battery requirements would result from lowering the maximum speed requirements due to weight savings made by sizing the various subsystems at lower power ratings.

To illustrate such effects, Fig. 11-94 was prepared for the family car, with a series configuration incorporating a spark ignition heat engine. Figure 11-94 indicates that as the V_{max} condition is lowered from 80 to 60 mph, the total powertrain weight (less batteries) is reduced from 1102 lb to 806 lb (with air conditioner) and the battery power density requirement reduced from 232 watts/lb to 133 watts/lb. These decreases are occasioned by a 98-lb decrease in spark ignition engine weight, a 177-lb decrease in electric drive motor weight, and a 21-lb decrease in generator weight, for the total weight decrease of 296 lb. All other subsystems/components remain invariant, as shown in Table 10-7.

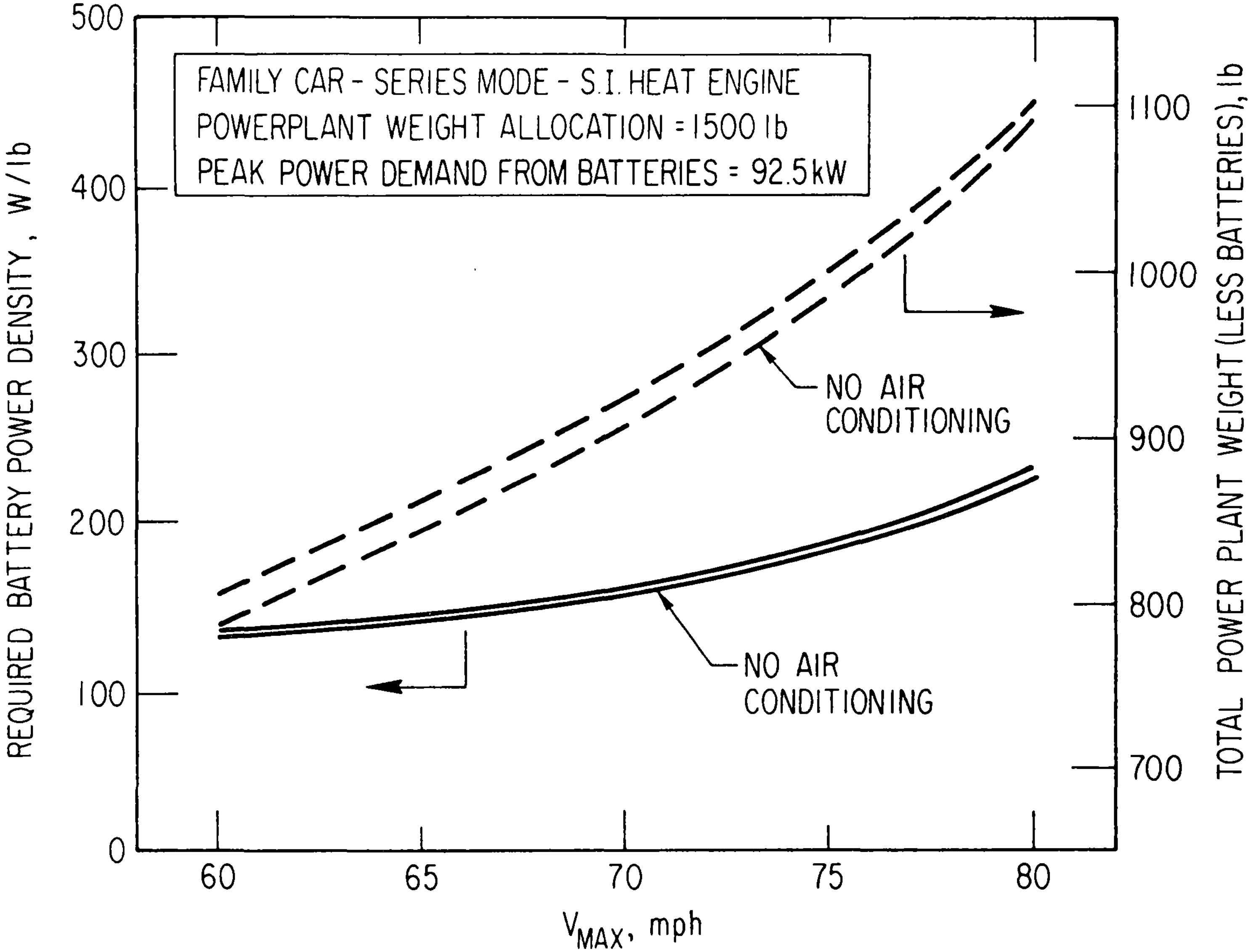


Figure 11-94. Effect of Design Point Sizing on Battery Power Density Requirements Family Car - Series Configuration

It should be mentioned that no specific gradeability requirement was stipulated for any V_{max} condition on Fig. 11-94 except at the 80-mph condition; i.e., a reduced sustained gradeability must be accepted along with the reduced size of heat engine. As long as the rated power output of the electric drive motor remains greater than one-third of the peak power output established for the baseline configuration, the acceleration capability of the vehicle will not have to be reduced.

11.3.2.5 Effect of Electric Drive Motor Efficiency on Battery Power Density Requirements for the Family Car

It was further recognized that significant electric drive motor weight savings could be made through use of motors incorporating advanced design techniques. However, for the same 100-percent or design point sizing condition, a reduced motor maximum efficiency is realized. This relationship is shown in Fig. 11-95 where motor efficiency is related to motor weight in terms of lb/hp.

Again, the family car with a series configuration incorporating the spark ignition heat engine was selected to illustrate the resulting vehicle characteristics, as shown in Fig. 11-96. Here the effect of reducing the drive motor maximum efficiency from 90 to 80 percent is shown. The total powertrain weight (less batteries) is reduced from 1102 lb to 983 lb (a 119-lb savings) with a decrease in battery power density from 232 to 179 watt/lb.

Here, the drive motor weight was reduced from 337 to 188 lb (a 149-lb savings). However, to overcome the reduced motor efficiency, more engine/generator power is required and the heat engine weight was increased from 335 lb to 359 lb (a 24-lb increase), and the generator weight also increased 6 lb. All other subsystem/components remained invariant, as in Table 10-7.

11.3.2.6 Effect of Spark Ignition Engine Air/Fuel Ratio

The baseline spark ignition engine weights used in all cases were consistent with the normalized variation of spark ignition engine weight with rated horsepower, depicted in Section 8, regardless of air/fuel ratio. It was recognized that for spark ignition engines lean air/fuel ratios such as selected for the purposes

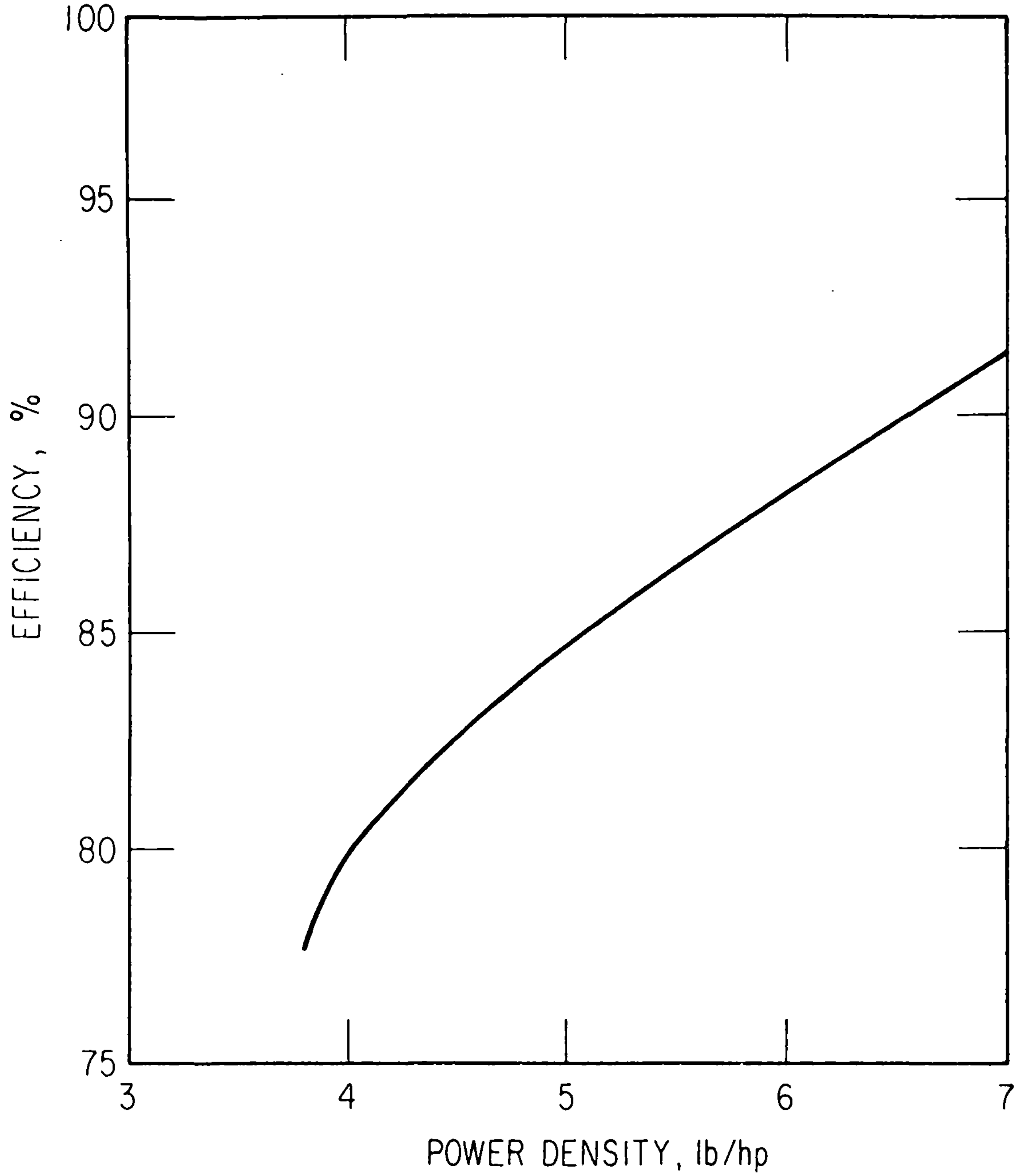


Figure 11-95. Power Density vs. Maximum Efficiency - DC Motors - Family and Commuter Cars

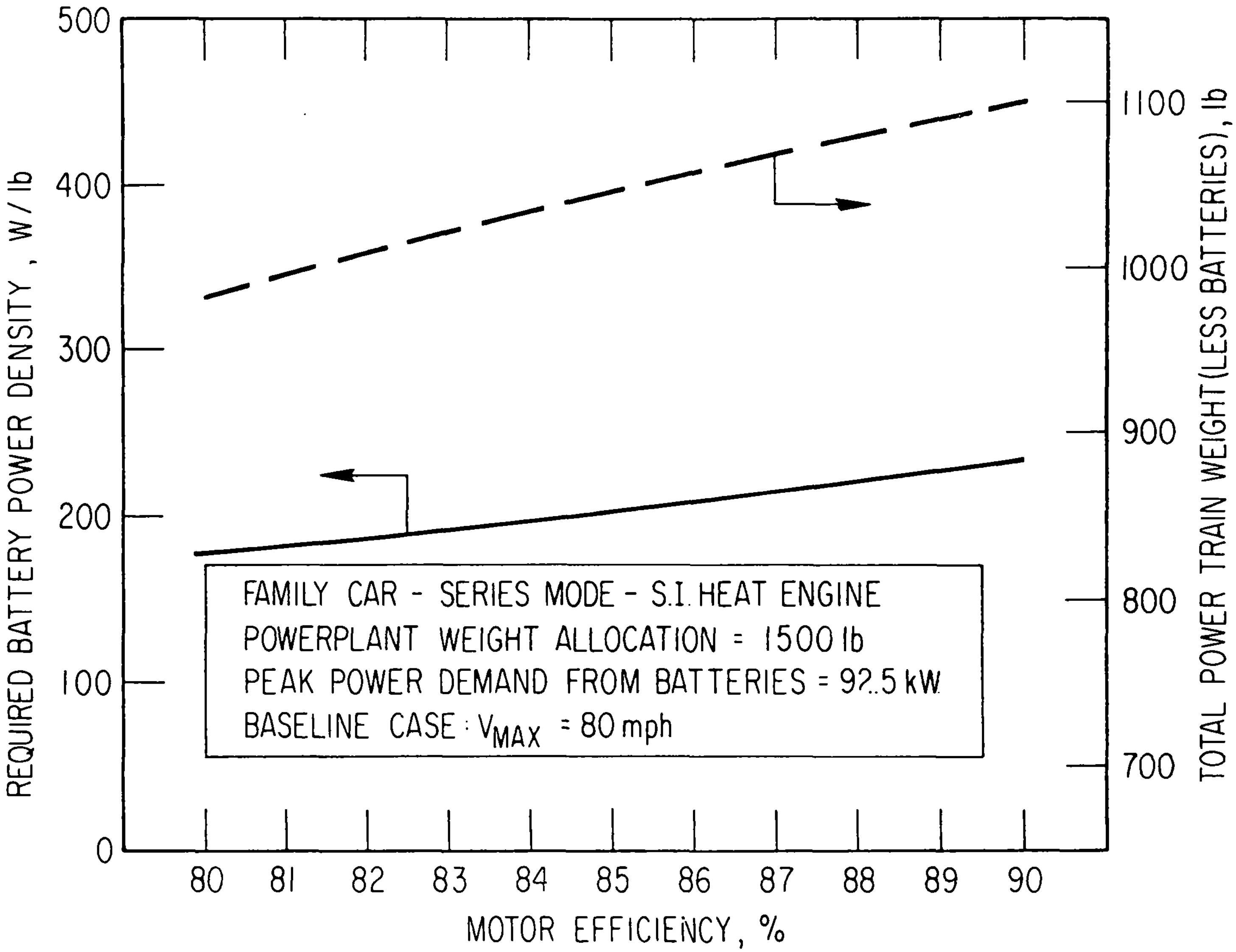


Figure 11-96. Effect of Drive Motor Efficiency on Battery Power Density Requirements Family Car - Series Configuration

of this study (a ratio of 19 for current technology and 22 for projected technology) would most likely result in a power loss (at the same displacement) over spark ignition engines having more conventional air/fuel ratios of 15 to 17. However, there was insufficient data to establish discrete variation from the "band" of spark ignition engine weight versus horsepower characteristics displayed in Section 8.

Assuming, for purposes of discussion, that the normalized variation of spark ignition engine weight versus rated horsepower used as a study baseline was strictly applicable to only nominal air/fuel ratios, and further assuming an ~15 percent loss in rated power output to occur at an air/fuel ratio of ~19, the increase in engine weight for the family car would be approximately 30 lb. Assuming an ~30-percent loss in rated power output to occur at an air/fuel ratio of ~22, the increase in engine weight for the family car would be approximately 59 lb. As noted previously, these weight increases are not expected in future engine designs and, hence, were not included in the powertrain weight tables.

11.4 COLD START EFFECTS

The vehicle emission levels computed for the various options and configurations presented previously in this section represent hot start cycle emissions since they are based on steady-state, hot engine exhaust emission data. For light-duty vehicles, the Federal test procedures specify that the vehicles be "cold-soaked" for 12 hours prior to the test. The HC and CO emissions are generally higher when the engine is cold; thus it is necessary to account for the emissions during this cold start period. In an engine equipped with a catalytic converter, there is an additional degradation of emission during the engine and catalyst warmup period. Figure 11-97 illustrates the effect of equivalent cold catalyst time (i.e., period during which the catalyst is ineffective) on effective catalyst efficiency over the DHEW cycle. For example, with a hot catalyst efficiency of 0.7, if the equivalent time period during which the catalyst is cold is 2 min (zero efficiency assumed) a hot catalyst efficiency of 78 percent is required to give the equivalent emission over the DHEW cycle. Conversely, the same cold catalyst time will result in a degraded value of catalyst efficiency over the DHEW cycle to a value of 0.63.

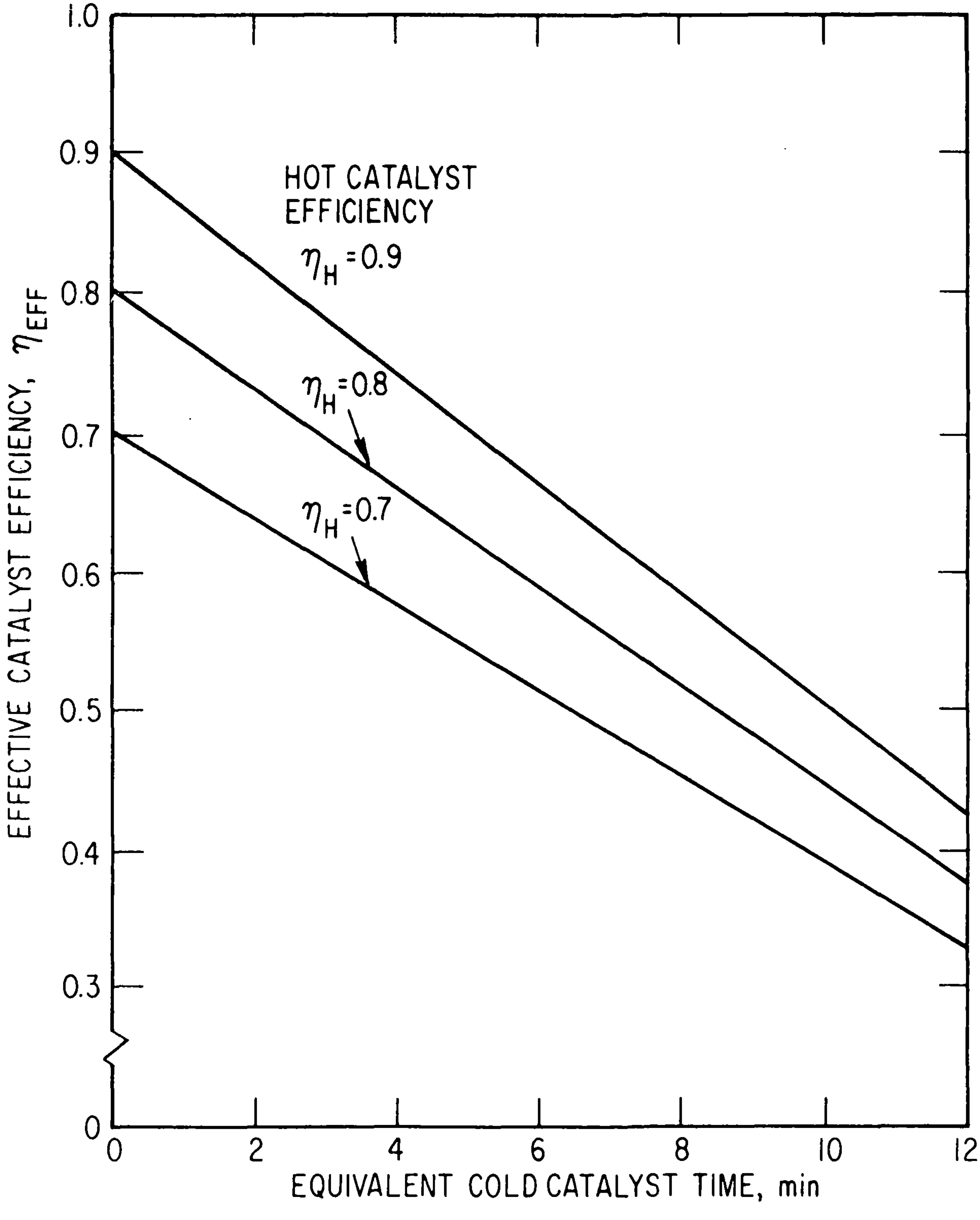


Figure 11-97. Effect of Catalyst Cold Time on Effective Catalyst Efficiency DHEW Cycle

In order to incorporate cold start effects, a cold start emission factor (ratio of cold start cycle emission to hot start cycle emission) can be applied to the vehicle HC and CO emissions computed from the hot engine data. Figure 11-98 illustrates the effect of equivalent cold start time on the emission correction factor for various values of catalyst efficiency and ratio of cold period to hot period engine emission level, X_C/X_H, where X_C is the representative engine emission level during the cold transient period and X_H is the steady-state hot emission level. It should be noted that the correction factors increase with increasing values of X_C/X_H and are lower with lesser values of catalyst efficiency. It should also be noted, however, that the higher correction factors associated with the more efficient catalyst would be applied to a lower hot engine emission level. Figure 11-98 is shown to illustrate the tradeoffs that can occur between equivalent cold start time, catalyst efficiency, and ratio of cold to hot engine emission level. It also illustrates that as catalytic converters are utilized, the cold start effect can become more pronounced unless considerable effort is expended to minimize these factors by decreasing effective cold start time.

Although cold start data are still scarce, and the factors affecting cold start merit considerable investigation, the following cold start correction factors were chosen to represent typical values that could be applied to the vehicle emission levels calculated. The improved correction factor shown for the spark ignition engine projected technology case (with catalyst) was based on minimization of engine cold start emissions through a programmed engine start as well as shortening of the catalyst warm-up time. Table 11-5 presents the values of the spark ignition engine, the diesel engine, and the gas turbine engine.

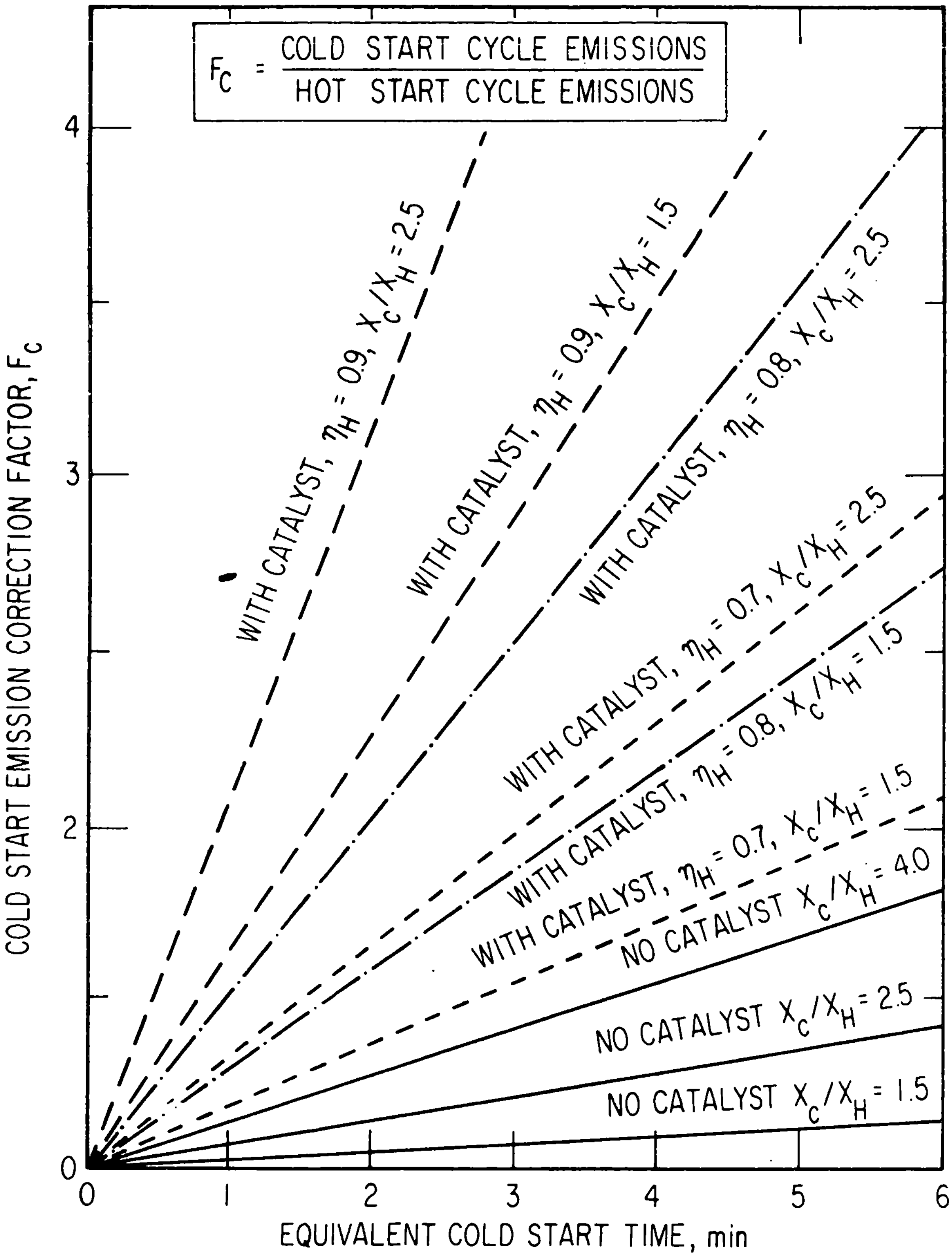


Figure 11-98. Effect of Equivalent Cold Start Time on Cold Start Emission Correction Factor - HC and CO Emissions, DHEW Cycle

Table 11-5

Cold Start Emission Correction Factors

	Spark Ignition		Diesel	Gas Turbine
	Current Technology (No Catalyst)	Projected Technology (With Catalyst)		
HC	1.3	1.2	1.0	1.2
CO	1.3	1.2	1.0	1.2
NO_2	0.95	0.95	0.95	0.90

Utilizing the factors listed in the table, the vehicle emission levels presented earlier can then be corrected for the cold start effect. As more information is obtained on cold start effects, these factors will undoubtedly change.

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SECTION 12

VEHICLE PRODUCTION COST COMPARISON

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SECTION 12

VEHICLE PRODUCTION COST COMPARISON

Cost estimates for the major subsystems of an advanced hybrid vehicle in volume production were prepared by judging system complexity and performance requirements using current hardware cost data wherever available. The powertrain and vehicle component cost estimates were then used to construct a total-vehicle-cost comparison between conventional and hybrid system designs for the family car. The results are presented in Table 12-1 and an explanation of these cost estimates is offered in the succeeding discussion.

Because the cost estimates reflect assumptions evolved from the current feasibility study, a more detailed cost analysis based on one specific hybrid vehicle design is necessary in order to refine the figures presented herein. A study incorporating this analysis should also link costs with exhaust emissions and fuel consumption. In this manner, the value of design trade-offs can be assessed directly in terms of reduced pollutants to the atmosphere as well as vehicle operating economy.

12.1 CONVENTIONAL CAR COST

The conventional family car is defined as a 3900-lb (curb weight), four-door sedan, equipped with a 230-hp engine, automatic transmission, air conditioning, power steering, and radio. These and other features are shown in Table 12-2. The curb weight is a mean quantity derived from the U.S. automobile population weight distribution (see Fig. 12-1), excluding imports, compact cars, station wagons, and prestige automobiles. The Plymouth Fury I, the Ford Custom 500, and the Chevrolet Bel Air are prime examples of U.S. automobiles which fit this weight category (when equipped with the accessories mentioned above). Each of these vehicles is offered with a choice of either a straight six (150 hp) or a V8 (230-265 hp) engine at a cost differential of about $90. The V8 was taken as the standard in consideration of the additional engine power required to operate the standard set of accessories.

Table 12-1. Cost Comparison of Conventional and Hybrid Family Cars, $

COMPONENT	CONVEN-TIONAL VEHICLE	SERIES HYBRID-BASELINE CONFIG. (S. I. ENGINE)	SERIES HYBRID-BASELINE CONFIG. (GAS TURBINE ENGINE)	PARALL. HYBRID-BASELINE CONFIG. (S. I. ENGINE)	PARALL. HYBRID-DUAL MOTOR CONFIG. (S. I. ENGINE)
• Vehicle					
Body					
Trim					
Glass					
Suspension					
Steering	1260	1300	1300	1300	1300
Tires					
Wheels					
Brakes					
Miscellaneous					
• Power Train					
Heat Engine	635	495	920	480	480
Fluid Systems					
Radiator	50	35	0	35	35
Fuel Tank	90	90	115	90	90
Exhaust	30	25	35	25	25
Electrical					
Battery	30	560	560	560	560
Battery Charge Control	10	125	125	125	125
Starter	45	30	30	30	0
Generator	55	250	250	200	500
Motor(s)	0	400	400	350	
Generator Control	0	50	50	200	250
Motor Control	0	350	350	275	
Ac Rectifier	0	30	30	30	0
Drive Train Logic	0	100	100	125	150
Electrical Cooling	0	50	50	50	50
Gearing (HE to Gen)	0	60	60	60	0
Transmission	205	0	0	205	0
Drive Line					
Differential(s)	245	245	245	350	350
Rear Axle					
• Hybrid Sensors & Display Instr.	0	30	30	30	30
• Convenience Accessories					
Air Conditioning					
Power Steering	545	545	545	545	545
Radio					
• Emission Control Equipment	50	125	50	125	125
• Total Cost	3250	4895	5245	5190	4615
• Cost Ratio	1.0	1.5	1.6	1.6	1.4

Table 12-2. Characteristics of the Conventional Family Car

Body Style	Four-door Sedan
Transmission	Automatic
Engine	230 hp, V8
Accessories	Air Conditioning, Power Steering, Radio
Shipping Weight, lb	3600
Accessory Weight, lb	130
Fluid Weight, lb	170
Curb Weight, lb	3900
Dealer Cost, $	2850
List Price, $	3650
Customer Price, $	3250
Federal Tax, $	158
Freight (average), $	130
Total Customer Price, $	3538

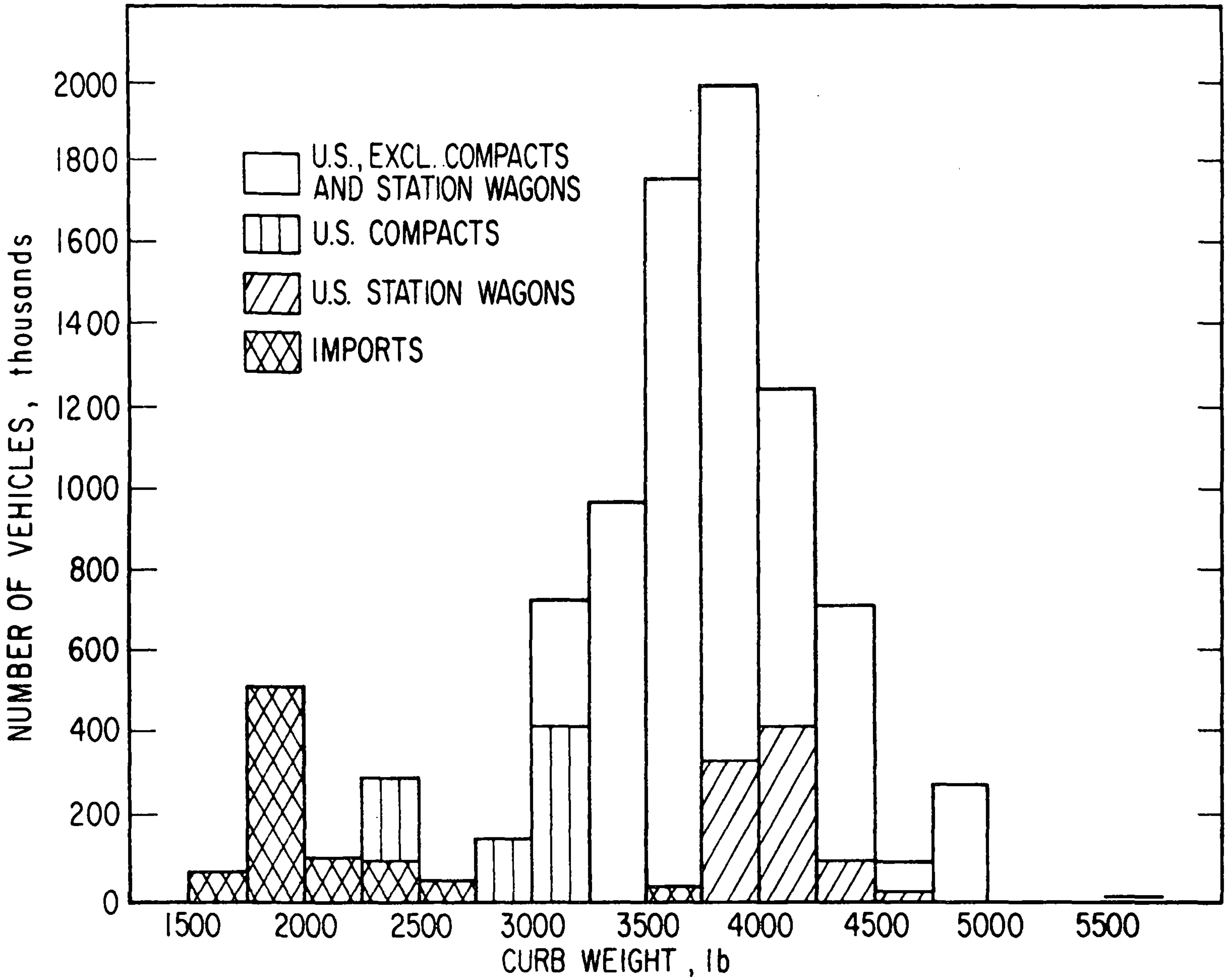

Figure 12-1. Automobile Population Weight Distribution (based on 1969 new car registrations)

The vehicle cost to the customer was set midway between the dealer cost and the retail price, thus allowing 12 to 15 percent for dealer profit. By this method, the cost of the conventional vehicle was found to be $3250 (exclusive of federal tax and freight charges). The cost of the components comprising the powertrain, accessory set, and emission control equipment for the conventional vehicle was set midway between dealer cost and the retail price. The resulting figure of $1990 when subtracted from $3250 yields $1260 for the cost of the remaining elements designated as vehicle components (body, suspension, wheels, etc.).

12.2 HYBRID CAR COST

12.2.1 Vehicle Component Costs

The costs of the vehicle components in the hybrid system (not including the powertrain) cannot be estimated accurately at this time, but it seems likely that these costs will not differ significantly from those of the conventional vehicle. Hence, a value of $1300, which provides a small allowance for hybrid-peculiar structural details, has been assigned to the vehicle-associated component set for each of the hybrid cost tabulations. Additional explanatory notes concerning the cost breakdown of the powertrain components follow.

12.2.2 Powertrain Component Costs

12.2.2.1 Heat Engine Costs

The series configuration engine is rated* at 92 hp while the parallel configuration engine is rated at 84 hp. The costs for the spark ignition engines were evaluated from the data given in Table 12-3 and plotted in Fig. 12-2 where specific cost data for automotive spark ignition engines are correlated with engine rated horsepower. The data fit is given by the line with the OEM (Original Equipment Manufacturer) designation, representing the cost of the engine as purchased by a dealer or distributor from the factory. As a rule of thumb, the list price can be taken as twice the OEM cost and this characteristic is shown in the figure for information purposes. The vehicle-installed purchase price is difficult to estimate. It is neither the OEM cost nor the list

*bare engine peak power output at the engine flywheel

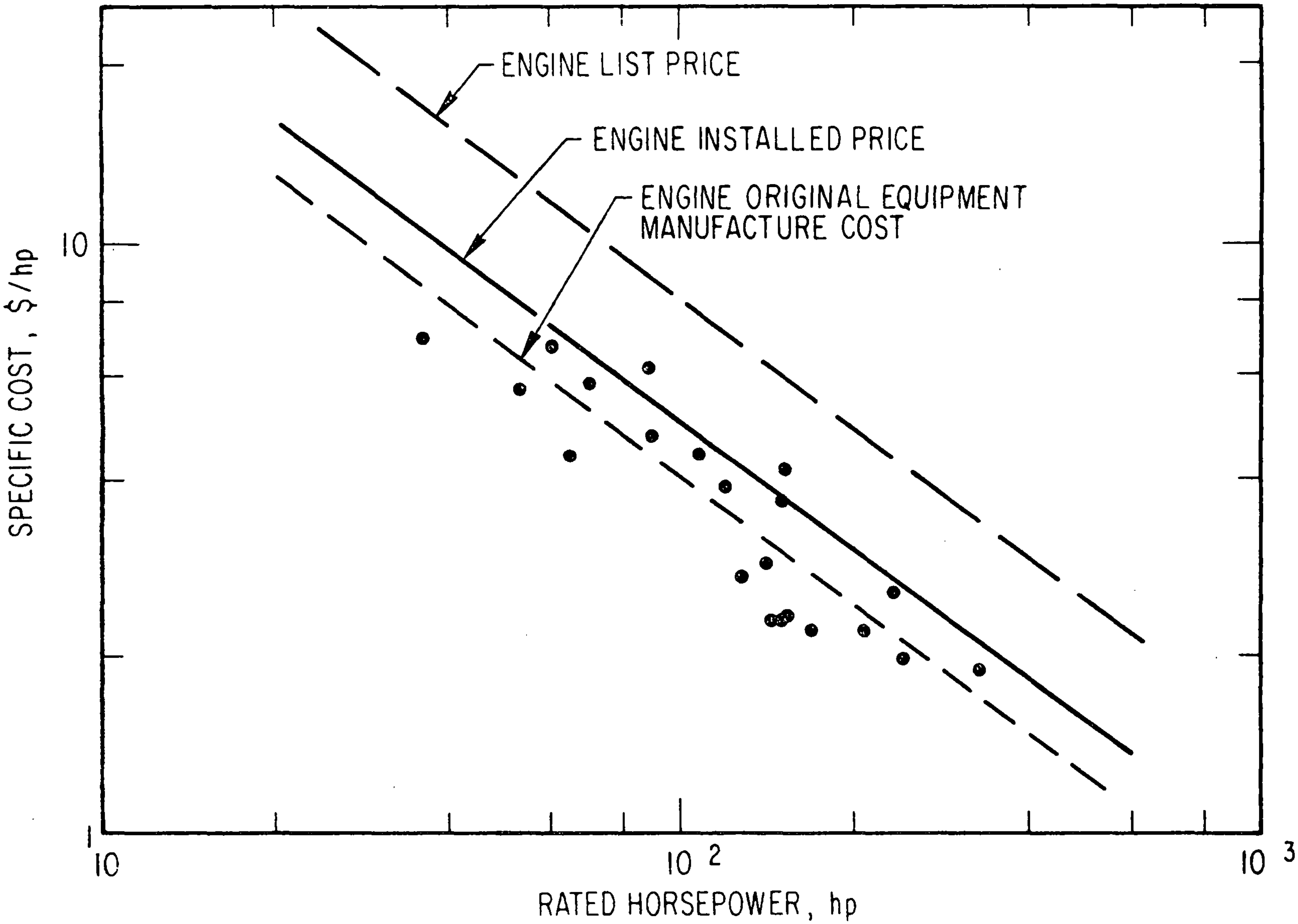

Figure 12-2. Spark Ignition Engine Cost Characteristics

price of the engine purchased as a component, but may lie between these two quantities. The installed-price characteristic shown in the figure is a crude estimate based on the OEM cost, to which has been added a 25 percent allowance for assembly-line installation expense and manufacturer/dealer profit.

For the gas turbine there is no production base from which high production engine prices may be estimated. Various estimates have been made which give relative costs between engine types, but these are of limited usefulness unless all of the hardware details are clearly established. In this report, costs for the gas turbine engine are based on a dual shaft, free turbine, recuperative engine design that does not require a transmission for delivering torque to the rear wheels. It has a bsfc under 0.6 lb/hp-hr at design rating. In the absence of any definitive study on the subject, the following technique was used in estimating costs. Costs were predicated on a dollars per pound basis with consideration of the type of materials being used. A factor was employed to account for the fixed per unit costs for such items as controls, ignition system, lubricant pumps, and other accessories. These costs do not include the amortization of the fixed cost of development and initial tooling, but are based on strictly the recurring costs of production.

The gas turbine costs are shown as a function of engine brake horsepower for three different production levels in Fig. 12-3. For purposes of estimating costs in Table 12-1, it was assumed that engine production rates exceeded 100,000 units per year, and a 25 percent allowance for assembly-line installation expense and manufacturer/dealer profit has been added on to the costs given in Fig. 12-3.

Although not pertinent to the cost breakdown given in Table 12-1, data for compression ignition engines were acquired in the course of this study and are included in the current discussion since future studies may benefit from this information. These costs were used to estimate rough comparative values shown in Table 12-4. The OEM cost and list price characteristics for compression ignition engines are shown in Fig. 12-4. As seen by comparing these data with those in Fig. 12-2, current cost/price levels for

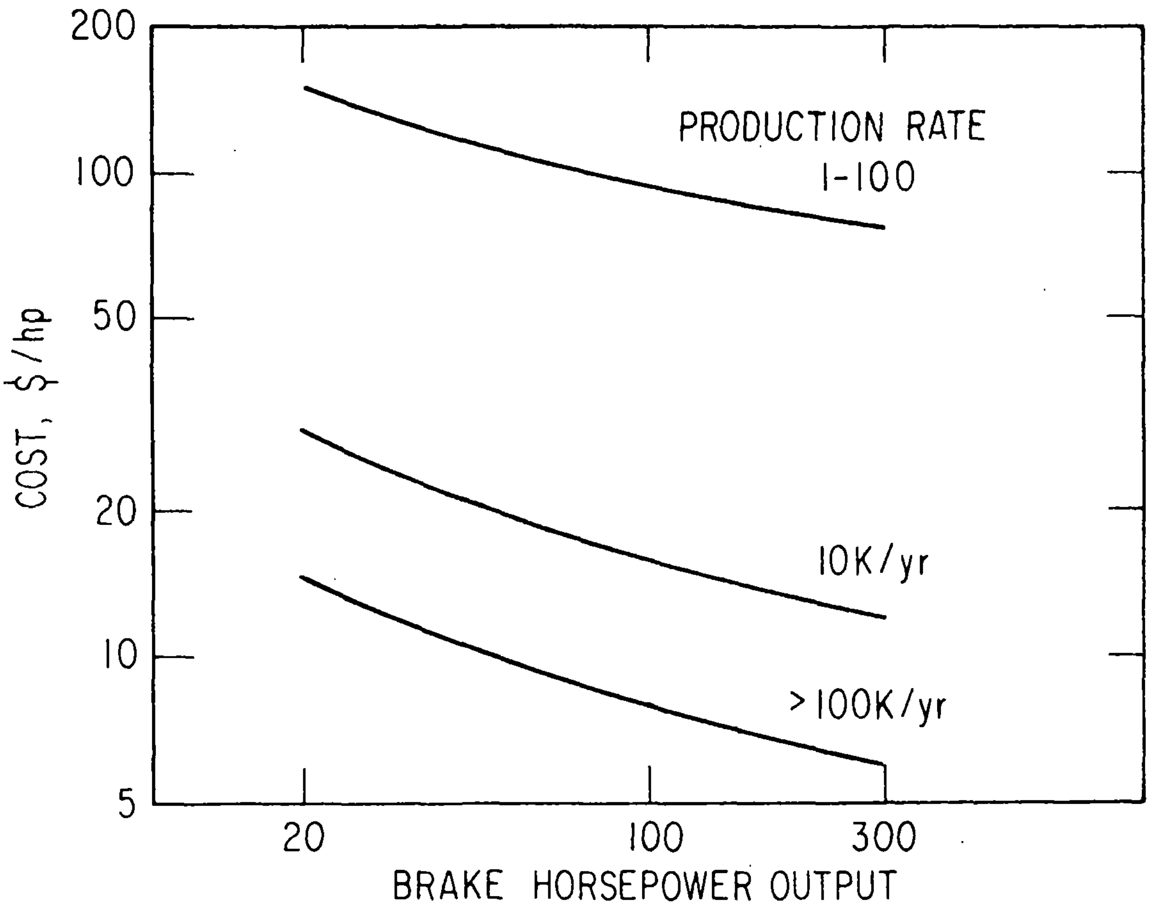

Figure 12-3. Estimated OEM Prices of Regenerated Gas Turbines

Table 12-3. Spark Ignition Engine Cost Parameters

Make	Model	Application	Rated HP [1]	Cost ($ List)	Cost ($ OEM)	$\frac{\$\ OEM}{\$\ List}$	$\frac{\$\ OEM}{HP}$ **
Alfa Romeo	Giula 1300	A	89	1100			6.19*
American Motors	Jeep 4L	A	60		403		6.72*
American Motors	Jeep 4F	A	70		407		5.82*
American Motors	Hornet	A	128	800	350	0.44	2.74*
American Motors	Jeep 155	I	136	750	412	0.55	3.02
American Motors	V6-225	A	142		412		2.90*
American Motors	Hornet SST	A	145	850	336	0.40	2.32*
American Motors	Ambassador	A	150		348		2.32*
American Motors	Ambassador SST	A	210		461		2.20*
American Motors	AMX	A	245		489		1.99*
American Motors	Rebel SST	A	330		625		1.90*
Briggs-Stratton	200-400	I	8	244			15.10
Briggs-Stratton	233-400	I	9	247			13.70
Briggs-Stratton	243-430	I	10	258			12.90
Briggs-Stratton	300-420	I	12	320			13.30
Briggs-Stratton	370-420	I	14	334			11.92

Table 12-3. Spark Ignition Engine Cost Parameters (Continued)

Make	Model	Application	Rated[1] HP	Cost ($ List)	Cost ($ OEM)	$ OEM / $ List	$ OEM** / HP
Ford	V4-91	I	58	818	409	0.50	7.05
Ford	4-172	I	68	918	459	0.50	6.75
Ford	V4-104	I	70	858	429	0.50	6.12
Ford	Mustang	A	120	936	468	0.50	3.90*
Ford	Galaxie	A	150	1110	555	0.50	3.68*
Ford	6-300	I	165	1144	572	0.50	3.47
Ford	330HD	I	190	1122	561	0.50	3.06
Ford	361EHD	I	200	1462	731	0.50	3.65
Ford	391EHD	I	235	1620	810	0.50	3.45
Ford	V8-477	I	253	2644	1322	0.50	5.06
Ford	V8-534	I	266	2704	1352	0.50	4.86
General Motors	Chev. 250L6	T, B	155	729			2.35*
General Motors	Chev. 292L6	T, B	170	753			2.21*
General Motors	Chev. 366V8	T, B	235	1196			2.54*

Table 12-3. Spark Ignition Engine Cost Parameters (Concluded)

Make	Model	Application	Rated[1] HP	Cost (\$ List)	Cost (\$ OEM)	\$ OEM / \$ List	\$ OEM / HP **
International	BO-308	T	154	1278			4.15*
Tecumseh	H4-120	I	12	252			10.50
Toyota	Corolla	A	60	819			6.80*
Toyota	Corona	A	90	858			4.77*
Toyota	Corona MkII	A	108	957			4.43*
Volkswagen	1200	A	36	500			6.95*
Volkswagen	1600	A	53	600			5.67*
Volkswagen	1500	A	65	575			4.40*

Key:

A Automobile
B Bus
I Industrial
T Truck
OEM Original Equipment Manufacturer
* Indicates data plotted
** Based on \$ OEM = 0.50 \$ List
(1) Bare engine peak power output at the engine flywheel

Table 12-4. C. I. Vs S. I. Engine Component Cost Differentials

Component	$ List	$ OEM	$ OEM, H. P.*	+ 25%	Notes - $ List
+ C. I. Components					
Injection System	440	220	110	138	Roosa-Master
Turbocharger	300	100	50	62	Garrett Corporation
Glo Plugs	12	6	3	4	Four Cylinder Engine
Speed Governor	----	----	----	----	Included in Injection Pump
TOTAL				+204	
- C. I. Components					
Carburetor	50	25		31	
Fuel Pump	8	4		5	
Distributor	25	12		15	
Coil/Spark Plugs	8	4		5	
TOTAL				-56	
Net C. I. Differential Cost = 204 - 56 = 150					

*H. P. = High Production Rate = 0.50 ($ OEM)

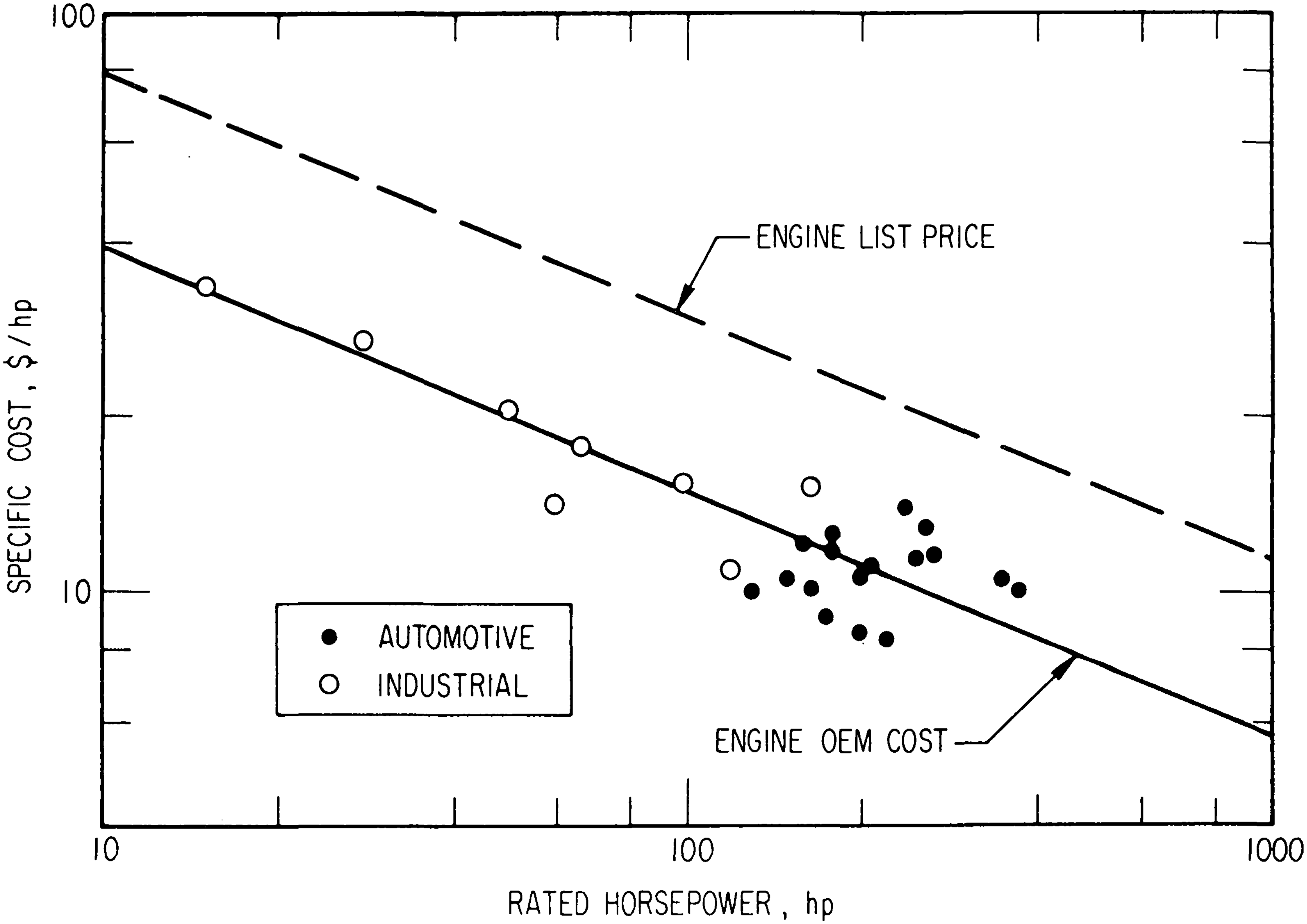

Figure 12-4. Compression Ignition Engine Cost Characteristics (Current)

diesel engines run 2.5 to 3.5 times higher than those for spark ignition engines (20 to 100 hp). This is largely due, it is felt, to the influence of production quantity on manufacturing cost. In this connection, it may be noted that the yearly production of diesel engines in the U.S. barely totals 360,000 units compared with 9 million units for spark ignition engines rated 20 hp and above (Refs. 12-1 and 12-2).

It is postulated that if the diesel industry were appropriately geared for high volume production, the cost of the basic engine (i.e., block, heads, pistons, crankshaft, camshafts, valves) might be reduced to a value that would be nearly (though perhaps not exactly) comparable to the cost of the basic S.I. engine. If this premise is accepted, then it is possible to define the differential cost of the engines (C.I. versus S.I.) in terms of the cost of system-peculiar auxiliary equipment required for engine operation. Table 12-4 attempts to develop a reasonable estimate of this differential cost for the hybrid family car application by comparing the auxiliary equipment requirements and costs for small (70-100 hp) diesel and spark ignition engine systems. It will be noted that a generous (50 percent) allowance has been made for diesel component cost improvements which might conceivably be brought about by the economics of high production automation techniques. Nevertheless, the diesel engine still shows a cost increment of $150 relative to the S.I. engine (this estimate may be low). Assuming a nominal horsepower requirement for the family car of, say, 80 hp, and, using Fig. 12-2 for S.I. engine installed price characteristics as a base, the C.I. engine would cost $625 installed, compared to $475 for the S.I. engine (30 percent higher).

Since there is very limited data available for assessing costs of a Rankine engine, the estimate must be considered very tentative. Compared to the gas turbine, the principal cost increase in the Rankine system will be found with the heat exchangers; e.g., the boiler will probably be constructed of stainless steel or material of similar cost. The cost will also be very much dependent upon the design bsfc specified since heat exchanger size will be a function of the desired bsfc. Rankine cycle estimated costs based on an engine bsfc of 0.85 are presented in Fig. 12-5 for three levels of production rate. These estimates are based on data presented in Refs. 12-3, 12-4, 12-6, 12-7, and 12-8.

Figure 12-6 shows estimated cost of a Stirling engine based on data in Refs. 12-4 through 12-8. Here, cost is figured at the same dollars per pound as the Rankine engine. However, there is an even higher degree of uncertainty in this case because of the less developed state of the engine. It is entirely possible that the engine can be reduced in weight below that of the Rankine engines and, since the ratio between heat exchanger and power generator weights is lower for the Stirling engine, the cost would be lower. Development costs might be higher because of the less advanced state of present development, but tooling costs might be slightly lower than for the Rankine engine.

12.2.2.2 Exhaust Emission Control Costs

To the heat engine costs must be added the additional costs associated with advanced emission control equipment. The emission control features of the spark ignition engine would include lean operation, exhaust gas recirculation, and catalytic conversion; the emission control features of the compression ignition engine would include exhaust gas recirculation and, possibly, catalytic conversion; and the gas turbine engine assumes the utilization of a thermal reactor for control of hydrocarbon emissions for costing purposes. At the present time, it is necessary to speculate that these costs will not differ significantly between the spark ignition and compression ignition engines. A value of $125 is assigned to the costs for the internal combustion engines and a value of $50 is assigned to the cost for the gas turbine engine.

12.2.2.3 Battery Cost

The battery cost for the hybrid vehicles is for a nickel-zinc system required to meet the baseline propulsion system weight allocation. The cost estimate was arrived at through consideration of contemporary lead-acid batteries and

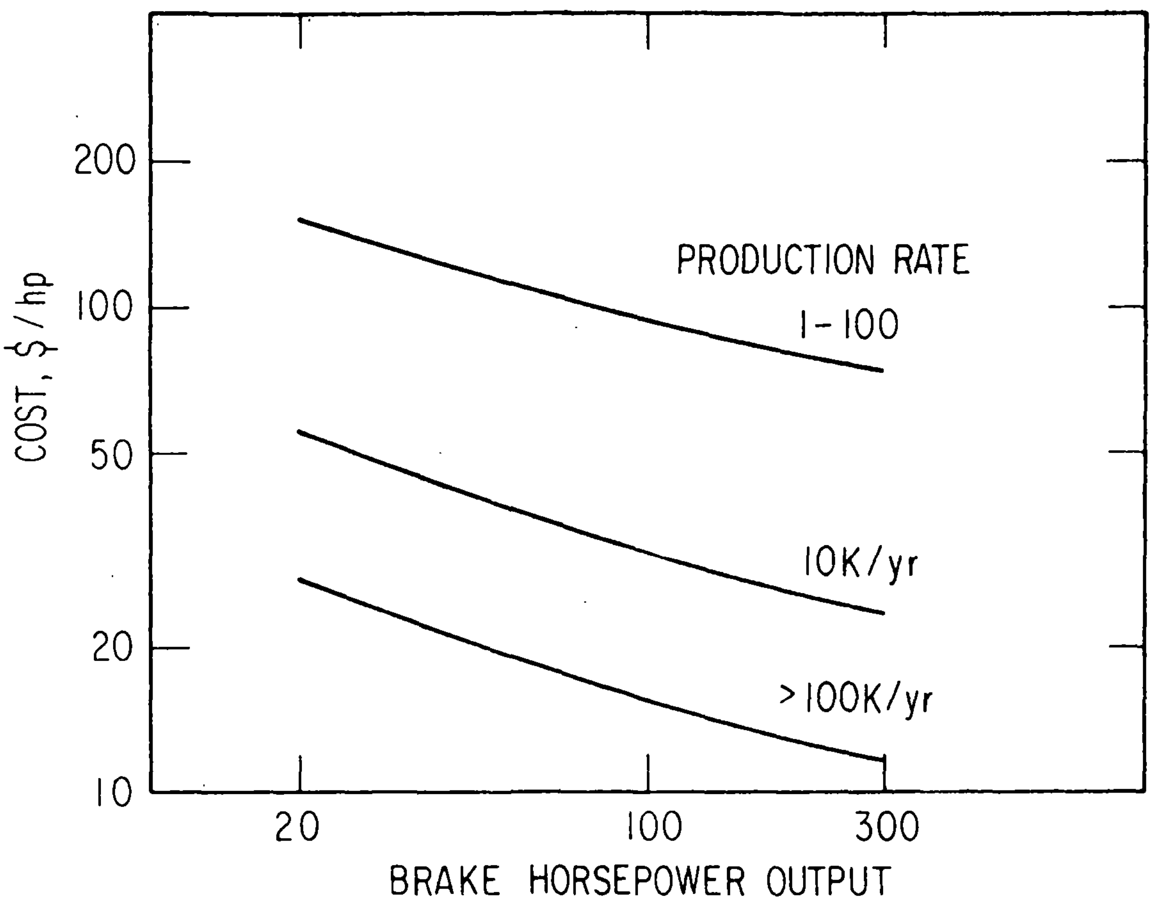

Figure 12-5. Estimated OEM Prices of Rankine Engines

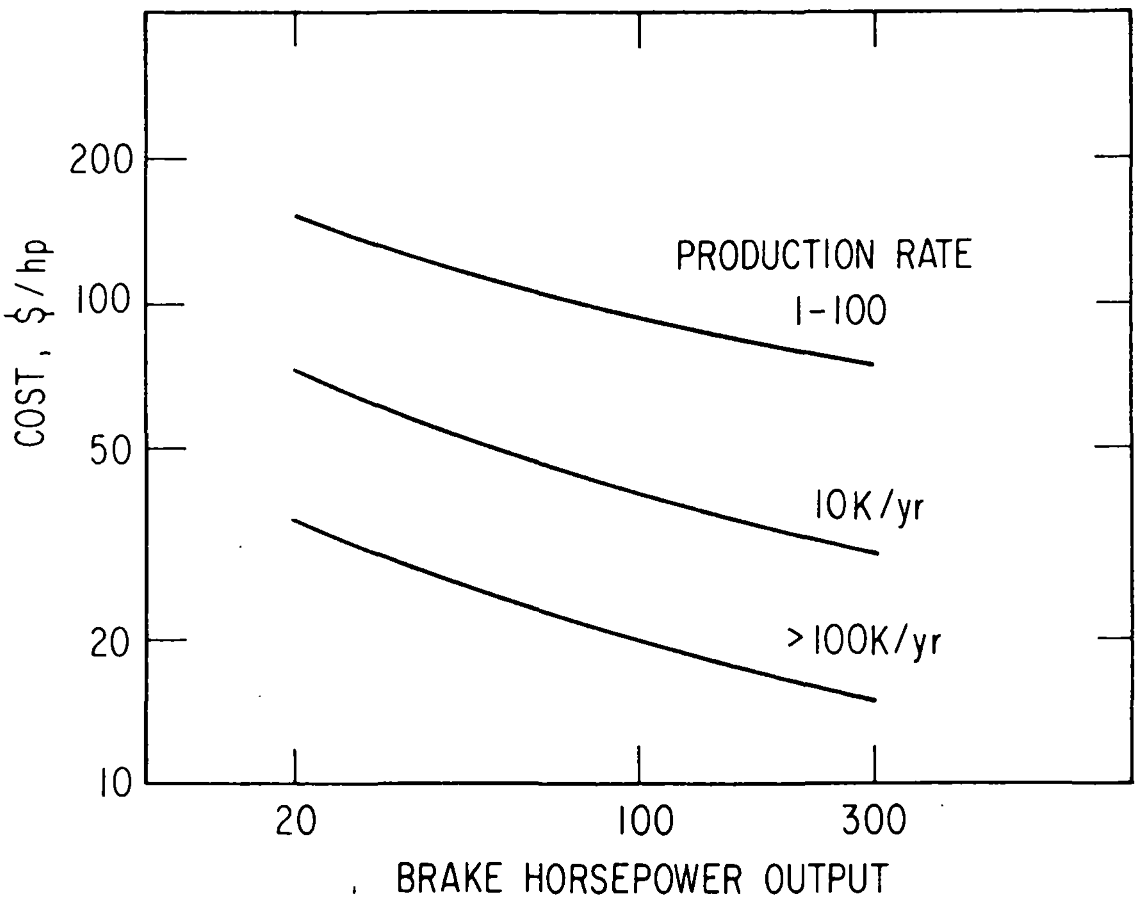

Figure 12-6. Estimated OEM Prices of Stirling Engines

the relative costs between different classes of batteries given in Section 7. At the present time, cost of a Sears Diehard lead-acid battery to the customer is about 2-1/2 cents per W-hr at the nameplate rating. Golf cart batteries will cost about twice as much. Because of refined manufacturing techniques and additional plate area, the battery for hybrid vehicle application will cost more than the present day SLI battery so a figure of 3.0 cents per W-hr was estimated on an OEM basis. Through ratio of the relative active material costs between battery systems, the cost of a nickel-zinc battery would then be 7.5 cents per W-hr and for 7470 W-hr installed battery capacity the cost is $560.

12.2.2.4 Electrical Component Costs

The costs of the DC electric drive motor ranges from $350 to $400. This cost quotation is based on an estimate obtained from the General Electric Company, D. C. Motor and Generator Department, for quantities of 100,000 or more. In contrast, present low production costs of a 60 hp, 2500 rpm base-speed motor of the type that would be used on a hybrid vehicle would cost approximately $3500 to $5000 in quantities of six or less on the present market, according to General Electric.

The AC generator costs are somewhat less than the DC motor costs of the same rating due to the fact that they are simpler in construction and contain less material because they are smaller. Also, no overload requirement exists for the generator. Based on these considerations, the cost range for generators is estimated to be from $200 to $250.

The electrical controls and logic are estimated to cost from $500 to $800 at present day prices in quantity production of 100,000 or more. These figures are probably the least reliable of all the cost estimates contained in the report. The reasons for this are the following:

1. A production cost estimate is not available at this time from industry for even small quantities of equivalent power handling devices. The closest device that was quoted for small quantities (six or less) was from General Electric, Automation Products, Salen, Virginia. They estimated a cost of $750 to $800 for a

72 volt maximum, 250 ampere rated chopper with pulse frequency control. This would be just for the motor control and would not include the generator control, battery charge controls, rectifiers, and drive train logic.

2. Cost estimates of future production are very difficult because costs of some parts have been coming down (such as SCR's) while others have been increasing and also the cost of labor is at best difficult to predict and is the greater part of the cost. The only alternative is to base a rough estimate on costs of labor, materials, overhead and profit, assuming present day prices. Materials costs were estimated on the basis of costs for similar circuits developed under U.S. Government military programs. Labor costs were estimated to be three times the material costs and a 25 percent factor was added to the combined labor and material costs to account for overhead and profit.

Motor/generator costs for the series configuration vehicles tend to be somewhat higher than those using the parallel configuration. This is because the drive motor and generator for series operation must be sized to provide or accommodate all the power required at the wheels, whereas, in the parallel case, the drive motor and generator need only be sized to supplement the mechanical power supplied by the heat engine. The separate-field-excitation feature of the dual motor configuration provides sufficient torque to start the heat engine, obviating the need for a separate starter. This feature also permits a simpler motor control circuit design and accounts for the lower motor control cost estimates relative to the baseline (single motor concept) parallel system which employs SCR's for this purpose. In the separately-excited field case, only the field current passes through the control circuit, reducing the power handling requirement. The transmission is eliminated in the dual motor design since the torque characteristic of an all-electric drive is provided by the second motor/generator operating through the planetary differential.

12.2.2.5 Drive Line and Fluid System Costs

The cost assigned to the drive line/differential for the parallel system vehicles is higher than that of the conventional vehicle design. In the baseline case, this is because additional gearing linking the motor/generator to the output shaft is required; in the dual motor case, the higher cost reflects the additional economic burden imposed by the requirement for two differential mechanisms.

Slight differences in the fluid system costs among the different vehicles primarily reflect the influence of such factors as heat engine size and specific fuel consumption (SFC) characteristics.

12.3 APPLICATION OF RESULTS FROM COST ANALYSIS

The tabulation results (Table 12-1) given by the Total Cost and Cost Ratio entries should be approached with caution, giving due regard to the preciseness of the assumptions made in the cost analysis. The numbers indicate that the series/gas turbine and baseline-parallel/spark ignition engine configurations are most expensive at a cost ratio of 1.6, while the series/spark ignition engine and dual motor/spark ignition engine configurations have cost ratios of 1.5 and 1.4, respectively. The hazard of assigning significance to the relative magnitudes of the cost ratio is apparent when it is recognized that to arrive at production costs it has been necessary to estimate figures for a number of critical components which at present may be barely classified as being in a conceptual design phase. Therefore, it is recommended that the indicated range of the cost ratio be regarded as the tolerance on a general estimate of 1.5 for the cost ratio of the several hybrid vehicles investigated.

The study indicates that only the spark ignition engine and the gas turbine engine offer reasonable weight margins for the battery system and, for this reason, only detail costing of these could be justified. The theoretical family car constructed using Diesel, Rankine, or Stirling engines would have higher weights and probably reduced performance. These car costs could not be realistically compared to those in Table 12-1. However, a relative vehicle cost estimate (Table 12-5) can be generated if an oversimplifying assumption were made that the cost differentials of these hybrids varied by virtue of relative heat engine costs (remembering that the specifications differ). Relative costs for the engines are compared with the spark ignition engine installed in the hybrid in the table below. In addition, relative vehicle costs are compared with those of a conventional spark ignition car.

Table 12-5. Heat Engine Cost Comparison

Heat Engine	Approximate Relative Engine Cost	Approximate Relative Hybrid Vehicle Cost
Conventional Car	----	1
Hybrid Spark Ignition	1	1.4 - 1.6
Hybrid Diesel	1.5	1.5 - 1.7
Hybrid Gas Turbine	2	1.6
Hybrid Rankine	3.75	2
Hybrid Stirling	5	2.25

12.4 REFERENCES

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SECTION 13

TECHNOLOGY DEVELOPMENT PROGRAM PLAN

CONTENTS

CONTENTS (Continued)

FIGURES

SECTION 13

TECHNOLOGY DEVELOPMENT PROGRAM PLAN

13.1 INTRODUCTION

The intent of presenting a development program in this report is to provide APCO with a planning document for ensuring the early availability of a low emission, viable alternative to the conventional automobile. This document defines the specific tasks to be accomplished and the corresponding schedule of activities. The entire three-phase program is directed toward the passenger car since this vehicle is by far the major contributor of air pollutants from mobile sources and is expected to receive the greatest emphasis from government and industry. The first phase covers a detailed performance analysis for providing a finer definition of vehicle operating and production costs; an expanded data base on heat engine emissions and battery characteristics forms a basic part of the effort which includes cost and performance comparisons with advanced heat engine-driven automobiles. The second phase entails the development of advanced versions of heat engines and batteries designed to operate in the hybrid mode. The third phase encompasses the hardware definition and development necessary for an early test bed vehicle as well as for a later prototype vehicle.

The development program recommended would result in a prototype vehicle that would meet performance goals and the 1975/1976 emission standards in approximately four years. An important step in this development is the construction of a test bed vehicle; it would be available for field testing in about two years. The reasons that a test bed-type vehicle, rather than a sophisticated prototype, is selected are twofold:

1. An instrumented vehicle with tailored components for early field testing is essential for resolving interfacing problems and for determining the response of all components and

subsystems to the automotive environment presented by actual urban driving situations.

2. Selection of a test bed concept allows for more flexibility of design goals and for gathering of more data on component and vehicle performance than a polished hybrid.

The test bed vehicle is expected to demonstrate marked improvement in exhaust emissions over current conventional cars, but will likely not meet all the 1975/1976 performance goals or emission standards; the HC and CO values are likely to be met, but NO_2 values may be exceeded by a factor of 2 to 3. Component advanced technology programs will be conducted concurrently with the test bed phase of the program for introduction into the later second generation vehicle. This vehicle, a prototype design, will have received sufficient design review to ensure that high production rates are both feasible and cost effective. It will also have received the benefits gained from experience with the environmental test bed vehicle.

The study indicates that the bus might make an attractive hybrid heat engine/electric vehicle (mainly because of the ease of attaining proper batteries). But for this case, only more study or analysis is warranted at this time. The major factors which prevented performing this analysis during the study were: (1) insufficient realistic driving cycle data, (2) lack of bus emission standards, and (3) inadequate emission data from current buses to be used for comparative evaluation. A bus study program is recommended which would collect or generate these three factors and use them in the computed analysis of performance and emissions.

13.2 RECOMMENDED HYBRID VEHICLE SYSTEM DESIGN FOR THE FAMILY CAR

Two system designs are recommended to provide greater assurance of early development of the hybrid prototype. Versions using the internal combustion spark ignition engine and the gas turbine are the only two heat engines that offer the combination of near term availability with low emissions and also provide acceptable vehicle performance without requiring unreasonable battery power/energy density goals. The spark ignition engine version will make the 1975/1976 goals, but with little margin for the family car as specified in this study. The turbine has the potential of exceeding the goals and might lead to more acceptable post-1975/1976 vehicles. However, the gas turbine engine for use in automobiles is not as advanced in technology as the spark ignition engine and thereby results in a higher risk program.

The recommended hybrid electric vehicle configurations are as follows:

1. A parallel mode configuration, powered by a spark ignition heat engine with lead-acid batteries for energy storage, dc traction motor(s) for acceleration power, an SCR-augmented control system designed for varying motor voltage and separately excited field power, and an engine-driven generator (or alternator) for recharging the batteries.

2. A series mode configuration, powered by a gas turbine heat engine with lead-acid batteries for energy storage, a dc traction motor for acceleration power, an SCR-augmented control system designed for varying motor voltage and separately excited field power, and an engine-driven alternator for recharging the batteries.

The rationale used in selecting the aforementioned configurations is as follows:

1. The rpm range for a spark ignition engine is compatible with transmission/wheel rpm and is merely another application of current design experience; no gear reduction system is required between the heat engine and the remainder of the powertrain.

2. Because the spark ignition engine operates at less than 5000 rpm, a dc generator may prove to be as light and

efficient as an alternator in recharging the batteries, particularly since, as vehicle speed increases, the generator output will be controlled down to lower output (almost zero); for the dual motor design, power generation is inherent in the motor/generator system and a separate generator no longer forms part of the design. A dc generator can also use battery power directly for starting the heat engine.

3. The higher rpm associated with a gas turbine should prove to be compatible with the rpm range for alternators and, with no mechanical link to the wheels in the series mode, gear reduction systems are not necessary.

4. Direct current motors appear to offer adequate performance for passenger car service although they are not the lightest motor available (considering motor control systems as part of the weight definition). However, the weight (and size) differential is not great enough to offset the gains from control system simplicity, past experience in vehicular applications, and torque characteristics that are well matched to vehicle needs over a wide speed range.

5. The control system specified offers considerable flexibility in application which is essential to solution of design problems that may arise once all powertrain elements are integrated on the test bed vehicle.

6. Lead-acid batteries are selected since they have the greatest experience factor, are not prohibitively costly, and appear to have the best near term potential for marked increases in performance. Nickel-zinc batteries, because of their current underdevelopment but future potential for even greater increases in performance, might eventually replace lead-acid batteries.

7. With respect to modes of operation, the parallel configuration appears to offer more promising performance (e.g., battery requirements, fuel consumption). However, it may present more problems in design and fabrication than the series configuration because of its greater complexity in some design areas (due, for example, to the direct mechanical link between the heat engine/generator and the rear wheels).

13.3 RECOMMENDED DEVELOPMENT PROGRAM

A development effort contingent on results from early analyses of the hybrid automobile has been formulated in three phases. In brief, the first phase should be aimed at a finer definition of important hybrid parameters both via expanded analysis and data collection. A study should be performed to define in greater detail the hybrid vehicle production and operating costs since costs are an important parameter in determining if the hybrid is a viable competitor to the conventionally powered automobile. Particular emphasis should be placed on defining in greater detail the operating requirements and costs for the vehicle control system. In addition to the cost analysis, a performance analysis should be conducted to a level of depth greater than was performed in this feasibility study. Acquisition of component test data is needed to support this analysis. A very important area for expanded data collection is in the engine emission area. Here, information on engines operating in the hybrid mode are needed to strengthen the data base used for analysis. A comparative analysis between cars using hybrid heat engine-electric powertrains and those using advanced internal combustion or gas turbine engines should also be made to determine the relative advantages or disadvantages of the hybrid concept as a means of reducing auto pollution. Recommendations for additional work effort in Phases II and III are of course highly dependent on the results of studies conducted in Phase I.

The second phase should consist of an intensive effort to develop critical powertrain components destined for a prototype vehicle. This would include advanced technology work on engines and batteries designed to operate in the hybrid mode.

The third phase encompasses the hardware definition and development necessary for an early test bed vehicle as well as for a later prototype vehicle. Figure 13-1 shows a schedule of activity for the three phases of recommended hybrid heat engine/electric system efforts. The details of each phase of the recommended work effort are discussed in the subsequent sections.

13.3.1 Phase I - Detailed Hybrid System Analysis and Expanded Data Base

A logical progression from the current feasibility study would be a study directed at an in-depth analysis of the hybrid vehicle powertrain in a passenger car application. Thus, in a study narrowed in scope, the more intricate details of component operation and installation in the vehicle can be examined. The analysis is fundamental to establishing a firmer basis for objective evaluation of the hybrid electric vehicle in terms of exhaust emissions and costs when compared to present and projected versions of the engine-driven passenger car. A four-part effort covers analysis of vehicle/powertrain/component performance, data acquisition for an enlarged data base, a component and system cost analysis, and a comparison of hybrid versus engine-driven cars based on costs, exhaust emissions, and fuel economy.

13.3.1.1 Vehicle Configuration

Since APCO has consistently emphasized the importance of reduced exhaust emissions for a general purpose passenger automobile, the study should be limited to examining a hybrid heat engine/electric version of this type of vehicle. The following components should be examined in depth in parallel and/or series powertrain configurations:

1) heat engine

2) batteries

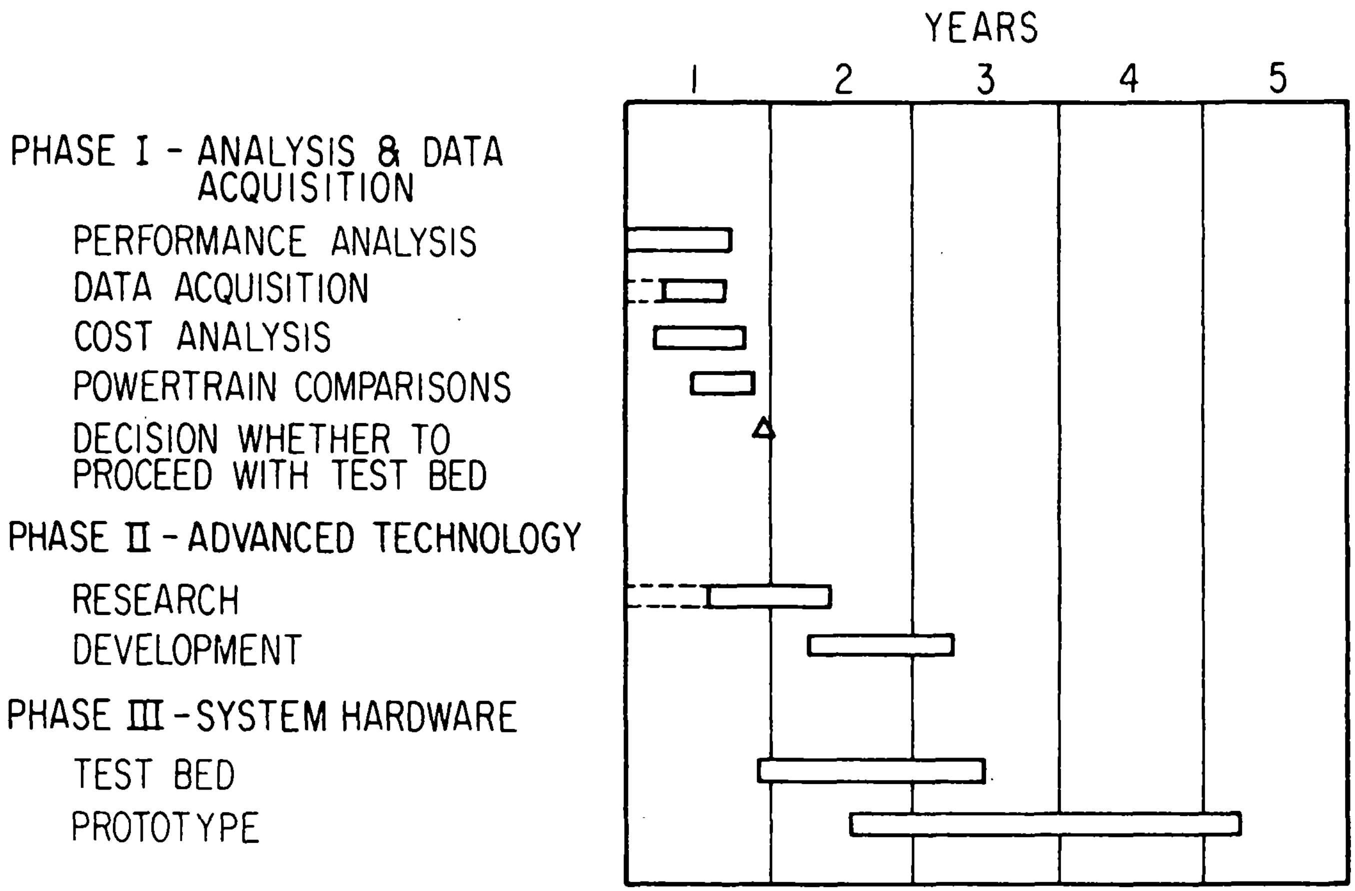

Figure 13-1. Hybrid Electric Recommended Development Schedule

3) generator

4) motor

5) transmission/gearing

6) control system

The complete powertrain and the vehicle system should also be analyzed. The parallel configuration should be examined for both the single motor and dual motor concepts because the dual motor concept offers superior operating flexibility but the efficiency remains to be defined.

13.3.1.2 Powertrain Elements

(1) Heat Engines

The following heat engines were selected for further examination in the study because of their near term potential for marked reduction in emissions as well as noted qualities.

(a) Internal combustion spark ignition

reciprocating conventional - best known and most research accomplished

reciprocating stratified charge - good potential for lean operation without weight growth

reciprocating dual chamber concept - good potential for lean operation

rotary Wankel - light weight and low cost potential

(b) Gas turbine

single vs dual shaft designs - cost vs operating flexibility

recuperating vs non-recuperating designs - cost and weight vs fuel economy

(c) Internal combustion compression ignition - cursory review of dual chamber concept to update cost, weight and emissions and re-establish ranking with regard to other engine classes.

(2) Batteries

Only the lead-acid battery should be considered with the charge and discharge characteristics varied in order to observe the effect on exhaust emissions and energy density. Thus, required operating characteristics can be specified in more detail.

(3) Generator

Only alternators should be considered, and their superior weight and efficiency to be re-examined in light of component cost.

(4) Motor

The following motors should be considered with one to be selected on the basis of a balance between cost, weight, and efficiency:

AC induction
DC shunt wound - externally excited field
DC compound wound
DC series wound

(5) Transmission/Gearing

A simple fluid coupling transmission should be utilized from a weight and cost standpoint for the single motor parallel configuration. Both differential and planetary gears should be considered for motor power transmission in the other configurations.

(6) Control System

The following elements should be considered in terms of effect on electric circuit cost, reliability, operating flexibility and efficiency:

silicon controlled rectifiers
resistors/inductors/capacitors
relays/switches

Wherever possible, manual control should be evaluated for cost savings potential.

13.3.1.3 Expanded Data Base

A major effort in the study program should be the establishment of an expanded data base for the powertrain components. This could be accomplished in two ways: (1) through planning and conducting of tests on specific component hardware to define performance maps over the entire operating range, and (2) through consultation with component manufacturers and reliance on their existing and projected data. These discussions with manufacturers should also provide a means of assessing the cost factors associated with variations in component operation.

Three major subsystems appear to need markedly increased scrutiny before a major funding effort for hybrid vehicle hardware can be initiated. These are: (1) heat engines (advanced internal combustion engines and gas turbines); (2) motor/generator control systems; and (3) batteries.

Assessing and developing the full potential of the hybrid vehicle with respect to meeting and surpassing future vehicle exhaust emission standards will require the acquisition of more engine exhaust emission data. The variation of emissions at part-load conditions can be very critical in determining exhaust emission characteristics of the hybrid vehicle. While a comprehensive evaluation of all available data was made in the present study, it was also discovered that a shortage of steady-state mass emission data, particularly at part-load conditions, existed in the open literature.

Due to driveability constraints in the case of the spark ignition engine-driven automobile, the requirement for transient acceleration power limits the extent to which some advantage can be gained from lean operation. But, since the engine runs at essentially steady-state for the hybrid vehicle, the limit of lean air/fuel ratios can be extended to take advantage of the reduction in CO and particularly the NO_x emissions. However, such data for the extreme air/fuel ratios are limited at present, and a more vigorous effort should be made to examine and evaluate the options available to the engine designer for meeting air/fuel ratio goals of about 19 and greater.

Furthermore, for all candidate heat engines, the techniques to minimize both exhaust emissions and fuel consumption at the desired part-load conditions should be examined. An emission data acquisition program should be instituted which would include the running of selected engines at the desired engine operating points. Acquisition of these data would provide a firmer base from which to determine the most suitable heat engine for the hybrid vehicle from the standpoint of potential for reducing atmospheric pollution.

Data are needed for the following engines operating in the hybrid mode.

(1) Advanced internal combustion engines operating in the lean regime

(a) Spark ignition engines

modified conventional engine
stratified charge engine
pre-chamber engine
modified rotary (Wankel) engine

(b) Compression ignition engine (cursory examination)

modified diesel engine (low NO_x, lightweight)

(2) Gas turbine

single and dual shaft
recuperated and non-recuperated

The complete operating maps for these engines should be compared with the operating maps of the electrical components in order to define the interface relationships of power and rpm that are crucial for maintaining low emissions and high overall efficiency resulting in low fuel consumption.

Through discussions with hardware manufacturers and the further clarification of electrical component operation in the hybrid car that will be accomplished in the Phase I effort, the electrical and electronic elements of the subsystems in the overall vehicle control system can be defined. This step is necessary in order to confidently predict the production costs associated with the entire electrical system. The control system circuit design should also be examined from the viewpoint of reliability and maintainability as well as first costs, and the complexity should be evaluated in terms of heat engine operating modes and the degree of manual control that could be realizeable.

As part of the Phase I effort to improve the data base, performance of the latest lead-acid batteries should be documented. Test data should include charge/discharge characteristics, temperature effects, and in particular cycle lifetime at shallow discharge. These data should be supplemented with test results for high power density cells that are under laboratory development. If control system operation induces transient currents at the battery terminals, the resultant effects on battery lifetime should be ascertained.

13.3.1.4 Comparative Evaluation of Hybrid Automobile

To provide an expanded critique of the hybrid electric system one further evaluation merits inclusion in Phase I studies. This relates to comparing the advanced version of the hybrid electric passenger car with advanced versions of engine-driven passenger cars. Because of near-term potential for use in cars, only the spark ignition and gas turbine engines are recommended for powerplants to be included in each vehicle's powertrain. For equivalent performance in terms of acceleration, cruise speed, and gradeability, the respective systems should be compared on the basis of production cost, exhaust emissions, and fuel consumption.

13.3.1.5 Hybrid System Performance and Cost Analysis

Finally, in addition to establishing a solid basis for estimating comparative hybrid passenger vehicle emission levels and production and operating costs, the proposed work effort should also provide a definitive package of information that is required prior to implementation of hardware assembly for a test bed vehicle and prior to implementation of fully funded development programs for a prototype vehicle. This information package should consider such items as:

(1) recommended powertrain design and vehicle weight and powertrain weight allocations,

(2) performance specifications for each major component in the powertrain for the test bed and prototype vehicles based on vehicle specifications to be defined for acceleration, cruise speed, and gradeability,

(3) rationale for powertrain design and component selection including trade-offs between cost, exhaust emissions, fuel consumption, and reliability,

(4) vehicle performance capabilities including the effect of various driving cycles and cold-start on exhaust emissions.

In compiling this information package, it is estimated that the following work effort will have to be accomplished: First, establish minimally acceptable vehicle operating specifications for cruise speed, acceleration and gradeability so that effect on reduced requirements for component performance can be assessed. Then, allow vehicle and powertrain weight to vary for establishment of an optimum configuration using complete component operating maps and link hybrid vehicle powertrain elements through combined factors of power, efficiency, and rpm.

Next, define the effect of variations in component weight and performance on system cost and review preliminary operational requirements in detail with component manufacturers to assess cost trade-offs and to acquire latest data on component operating maps; viz. emissions and fuel consumption versus power output and rpm for heat engines.

Next, vary part-load characteristics of each component in order to optimize overall powertrain efficiency and establish vehicle optimum weight for a general purpose passenger automobile. Also, evaluate the effect of different driving cycles on powertrain operation, fuel consumption, and exhaust emissions and determine if the control system demonstrates suitable flexibility.

Finally, calculate system performance in terms of exhaust emissions and fuel consumption for advanced heat engine-driven automobiles (spark ignition and gas turbine) and compare to results for the hybrid electric powertrain. Use the same heat engines and driving cycles as those used in analysis of the hybrid powertrain in providing this comparison between hybrid powertrain cars and advanced heat engine-driven cars with evaluation factors of cost, exhaust emission levels, and fuel consumption. This effort is necessary to establish whether the hybrid electric powertrain cost margin over the heat engine-driven car is adequately balanced by the performance delivered in terms of exhaust emissions and fuel economy.

Furthermore, establish prototype conceptual designs for two alternative hybrid heat engine/electric automobiles specifying the required operating map characteristics of each component in the powertrain, the exhaust emissions, and the fuel consumption. One vehicle shall use a selected "best" spark ignition engine while the other vehicle shall use a selected "best" gas turbine. Provide a detailed cost breakdown for each recommended hybrid vehicle design using improved component hardware and with an optimized powertrain and vehicle weight. Provide trade-off factors between costs, exhaust emission levels, and fuel consumption involved with the selected hybrid vehicle.

13.3.1.6 Program Schedule

The schedule of work effort for all activities associated with Phase I is given in Figure 13-2. Component specifications and production cost estimates are shown as being available within 9 months following program inception in addition to a comparative evaluation with advanced heat engine driven automobiles.

13.3.2 Phase II - Component Advanced Technology

A research and development program is recommended to provide power-train components with performance markedly improved over contemporary hardware. Because of the influence on vehicle performance, all components and subsystems are to be designed for low weight and volume with due regard for effect on part-load to full-load efficiency. In order to ensure that the 1975/76 emission goals are met or exceeded, effective research is needed in several areas, but the effort should lie predominantly in the areas of heat engine emissions and battery lifetimes.

Initially, the program emphasis should be on research with limited funding until the Phase I study results in the form of comparative vehicle performance and cost as well as component specifications are available for review. Should these Phase I results still favor the development of a hybrid electric automobile, then the Phase II effort should be expanded rapidly with increased funding and eventual initiation of the hardware development portion of the program. The required work effort is presented in the following discussion. Component development goals are discussed more extensively at the end of Sections 6 through 9 of the report.

13.3.2.1 Advanced Internal Combustion Engines

To select the optimum engine to be used in the advanced prototype vehicle, a state-of-the-art evaluation should be conducted of lean (high air/fuel ratio) engine technology. The current hybrid studies have indicated that this type of spark ignition engine shows promise; however, more data and

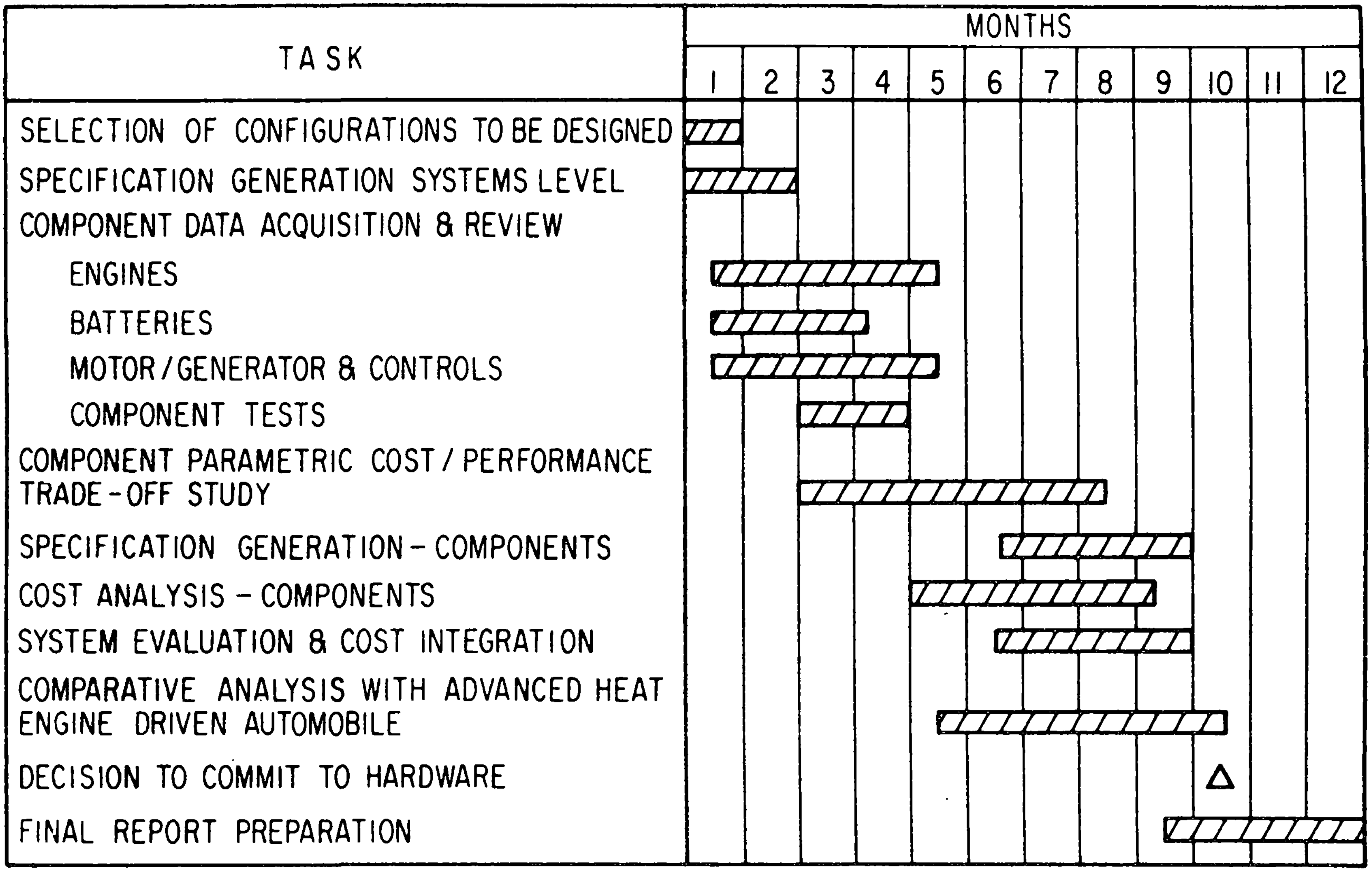

Figure 13-2. Schedule of Work Effort - 12-Month Program (Phase I)

evaluation of current developments must be accomplished to determine the best approach. In this regard, efforts should be made to design for low specific mass emissions at part-load engine operation. Lean air/fuel ratio engines should be evaluated to select the best approach towards achieving low emission goals consistent with fuel economy. Approaches to be evaluated should include the stratified charge engine, pre-chamber engine, and engines with optimized induction system design. The rotary combustion (Wankel) engine, because of its low weight and volume and its potential for operating in the lean air/fuel ratio regime, should also be investigated. Diesel engine technology should be investigated to assess its potential for reducing NO_x emissions and engine weight. A 2-year engine R&D program should be conducted with efforts also directed towards incorporating efficient catalytic converters, thermal reactors, and exhaust gas recirculation.

The lean engine evaluation program should delineate engine developments to be conducted as well as identify technology and data gaps. Following this development period, an engine should be selected for the hybrid system based not only on the results of this program, but on results from other, concurrent spark ignition engine programs that have been conducted for non-hybrid applications. Efforts should be also directed to developing efficient catalytic converters and techniques to reduce cold start emission effects.

13.3.2.2 Advanced Gas Turbine

A burner development program should be instituted to minimize the NO_2 emissions of the gas turbine by means of optimizing primary and secondary zone air/fuel ratios and the residence time of the gases in the primary zone. Studies should be conducted to select the optimum gas turbine design for the hybrid vehicle and to further the development necessary to meet the requirements of the prototype vehicle. The gas turbine developed with the hybrid vehicle in mind should have good part-load emission characteristics, and provide optimum matching of the heat engine with the electrical drive system.

13.3.2.3 Batteries

The battery research and development program should consist of parallel laboratory studies of a lead-acid battery and a nickel-zinc battery optimized to the hybrid vehicle requirements in terms of power density, energy density, lifetime, and charge acceptance. It is anticipated that nickel-zinc batteries will demonstrate superior performance characteristics than lead-acid but will be more expensive. It is also anticipated that selection of an optimum battery for the prototype vehicle will be made at the end of 2 years. Reduction in packaging volume is also necessary for realistic installation with other components in a restricted powertrain-allocated volume within the vehicle chassis/body combination.

Early implementation of this program is needed to determine whether such factors as increased plate area, thinner plates, stirred electrolyte, reduced internal resistance, and minimum (or zero) maintenance can be combined in a long life, low cost design that will be compatible with the automotive environment.

13.3.2.4 Component Design Evaluation

Design concepts generated in this Phase II program should eventually be introduced into the hybrid vehicle test bed program for evaluation, and field test results should be used to tailor the later development work effort. The test bed program is discussed next in Phase III of the overall development effort.

13.3.3 Phase III - Test Bed and Prototype Vehicle Development

The discussion that follows is highly dependent on the results of studies conducted in Phase I and the success of component research and development efforts in Phase II. If the cost and performance analyses indicate that the hybrid heat engine/electric automobile should remain as a strong contender in the APCO advanced powerplant program, then the philosophy guiding the formulation of the subsequent development program is to confidently provide as soon as possible a mobile test bed for the hybrid electric vehicle. This instrumented test vehicle will permit evaluation of the adequacy of the integrated system under combined environmental conditions that cannot be simulated either in the laboratory or by "breadboard" simulation systems. The resulting data will be reflected in realistic specifications being imposed upon components and subsystems that are in evolutionary stages and destined for the second generation vehicle (a prototype of the production passenger car). An approach of this nature will ensure that development funds are efficiently expended throughout the program and permit expenditures to be curtailed or expanded at critical evaluation points.

The recommendations are based on results of the just completed feasibility study on hybrid electric vehicles, and should be considered solely as generalized planning information at this time. As results from the Phase I program become available, (viz. detailed design information from the expanded analysis and data base) they can be used to refine the plans formulated in the subsequent discussion. In addition, refinement of plans for the prototype vehicle should be dependent on the success in improving component performance demonstrated in the Phase II research effort.

A 2-1/2 yr program is recommended for development of two mobile test beds for the hybrid electric vehicle. It is expected that specifications can be released for component development bids 7-1/2 months after Phase III initiation, and completely assembled vehicles will be available for a road test program within 21 months after Phase III initiation. The test vehicle is expected to demonstrate marked improvements in exhaust emissions, but will likely not meet the 1975/1976 emission standards. That goal is expected to be fulfilled by a prototype hybrid electric vehicle planned for completion in the 1974-1975 time period -- a vehicle which will largely benefit from the experience and component development accrued within the test bed vehicle and advanced component technology programs.

A logical division of the test bed vehicle program is presented herewith. First, the directed effort from analysis through design, development, fabrication, test, and specification release is discussed; second, a program schedule is presented for major tasks to be accomplished between 1971 and 1975.

13.3.3.1 Analyses

13.3.3.1.1 Design Factors Definition

Perform analyses for each configuration to define design factors in detail for the test bed vehicle including the following:

1. Vehicle and powertrain weight based on latest available component data.
2. Vehicle performance in terms of acceleration, maximum cruise speed, and gradeability.
3. Component and complete powertrain operating characteristics at all part-loads up to full load.
4. Structural loads and component/subsystem environment.

13.3.3.1.2 Component Data Evaluation

Evaluate test data on components being developed for the hybrid electric vehicle and factor into the vehicle performance analyses.

13.3.3.1.3 Test Bed Data Analysis

Analyze data acquired from vehicle test bed and use results to modify design and tailor future component development to prototype system needs.

13.3.3.2 System Design Detail

13.3.3.2.1 Structural Load Design

Establish adequacy of chassis and body design for static and dynamic structural loads.

13.3.3.2.2 Layout Design

Design component/subsystem layout, gearing, routing of lines, weight distribution, c.g. location, ...

13.3.3.2.3 Establishment of System Interfaces

Establish control system mechanical/electrical interfaces.

13.3.3.3 Specification Release and Contract Definition

13.3.3.3.1 Specification Evolution

Evolve final specifications for chassis, body, and powertrain components and subsystems from results of analysis and design.

13.3.3.3.2 Specification Release and Contract Award

Release specifications to vendors for bid and subsequently contract for development and fabrication.

13.3.3.4 Hardware Design, Development, and Fabrication

All components and subsystems are to be designed for low weight and volume with due regard for effects on part-load to full load efficiency. They will also be designed to operate acceptably under the environmental conditions expected for the test bed vehicle (e.g., shock, vibration, acceleration, temperature, moisture, dust) as delineated in the specifications. The following comments serve to highlight those design factors peculiar to the hybrid electric vehicle.

13.3.3.4.1 Motor/Generator

Design for low cooling requirements, for nonsteady operation, and for an optimized balance between high part-load efficiency and efficiency achievable at full load.

13.3.3.4.2 Control System

Design for simplicity, reliability, and low audible noise.

13.3.3.4.3 Batteries

Design for high power density, high energy density, long life, high charge acceptance, and minimum (or zero) maintenance.

13.3.3.4.4 Heat Engine

Design for low emissions at part-load up to full load and for application of catalytic converters and thermal reactors.

13.3.3.4.5 Body and Chassis

Design for weight balance, c.g. control, cooling provisions for electronics and batteries, and noise suppression.

13.3.3.5 Component Test and Evaluation

During component design and fabrication, test data are to be acquired for verification of adherence to specifications. Evaluation of these data should offer alternatives to design approaches if specifications are not met under all operating conditions. These results can then be factored into the design before hardware delivery.

13.3.3.6 Static Interfacing Tests

All components and subsystems will be assembled in a breadboard test set-up to provide initial evaluation of performance and interfacing problems. A dynamometer will be utilized to simulate road load, and exhaust emissions will be measured along with component performance. Control system modifications will be incorporated at this time if necessary to optimize reduction in exhaust emissions.

13.3.3.7 Vehicle Assembly and Final Component Integration

All components and subsystems will be installed in a conventional automobile chassis modified for the hybrid electric vehicle design and for use as an instrumented test bed. Body and interior arrangements are to be tailored for esthetic reasons as well as for functional operation in a test bed vehicle (i.e., the body shell should be easily removable to permit ready access to components or for structural alternations to the chassis).

13.3.3.8 Vehicle System Tests and Evaluation

13.3.3.8.1 Road Tests

Following initial checkout of the assembled test bed vehicle for handling, drivability, and response to power demand, an extensive series of tests are to be conducted for evaluating component, subsystem, and total system operation in the urban and open highway environment. Tests are to be run both at steady speeds as well as in a dynamic traffic-following situation. Data will be evaluated to: (a) determine how well the vehicle matches design performance goals, and (b) determine required design modifications to improve performance.

13.3.3.8.2 Laboratory Emission Tests

The vehicle is to be tested in an exhaust emissions test laboratory over the prescribed government test driving cycle, and measurements are to be made of unburned hydrocarbons, carbon monoxide, and oxides of nitrogen from exhaust gas sampling. Basic emissions data should also be acquired for various operating conditions.

13.3.3.9 Prototype Vehicle Program

Based on the analysis of the advanced technology program engine results and the test bed results, one of the two types of engines will be selected for prototype development. Development of the second generation vehicle will largely follow the plan given for the test bed in the preceding sections with the following exceptions:

1. Component design and development should be directed toward long-range ultimate improvements in performance.

2. Component qualification tests should be introduced prior to vehicle assembly.

3. Component and vehicle specifications should be more restrictive, particularly those associated with affecting exhaust emissions.

13.3.3.10 Test Bed Vehicle Program Schedule

A 29-month program has been scheduled by major task elements for the test bed vehicle as shown in the accompanying chart (Fig. 13-3). The basic design and release of final specifications for use in soliciting contract bids for hardware development is accomplished 7-1/2 months after program inception; a fully assembled test bed vehicle is ready for road tests 22 months after program inception. The prototype vehicle program is initiated 1 to 1-1/2 yr after inception of the test bed vehicle program and continues on for approximately 3 yr. Based on analysis, test bed data, and detailed costing, the more promising of the two hybrids should be selected for use in the prototype development program.

The tasks for the prototype vehicle program, while not delineated, are in essence the same as those shown for the test bed vehicle; the main difference is the use of more developed components and of design data obtained from the test bed program. Detailed definition of the prototype vehicle program should not be attempted until specifications have been released for the test bed vehicle and some data on component performance characteristics generated.

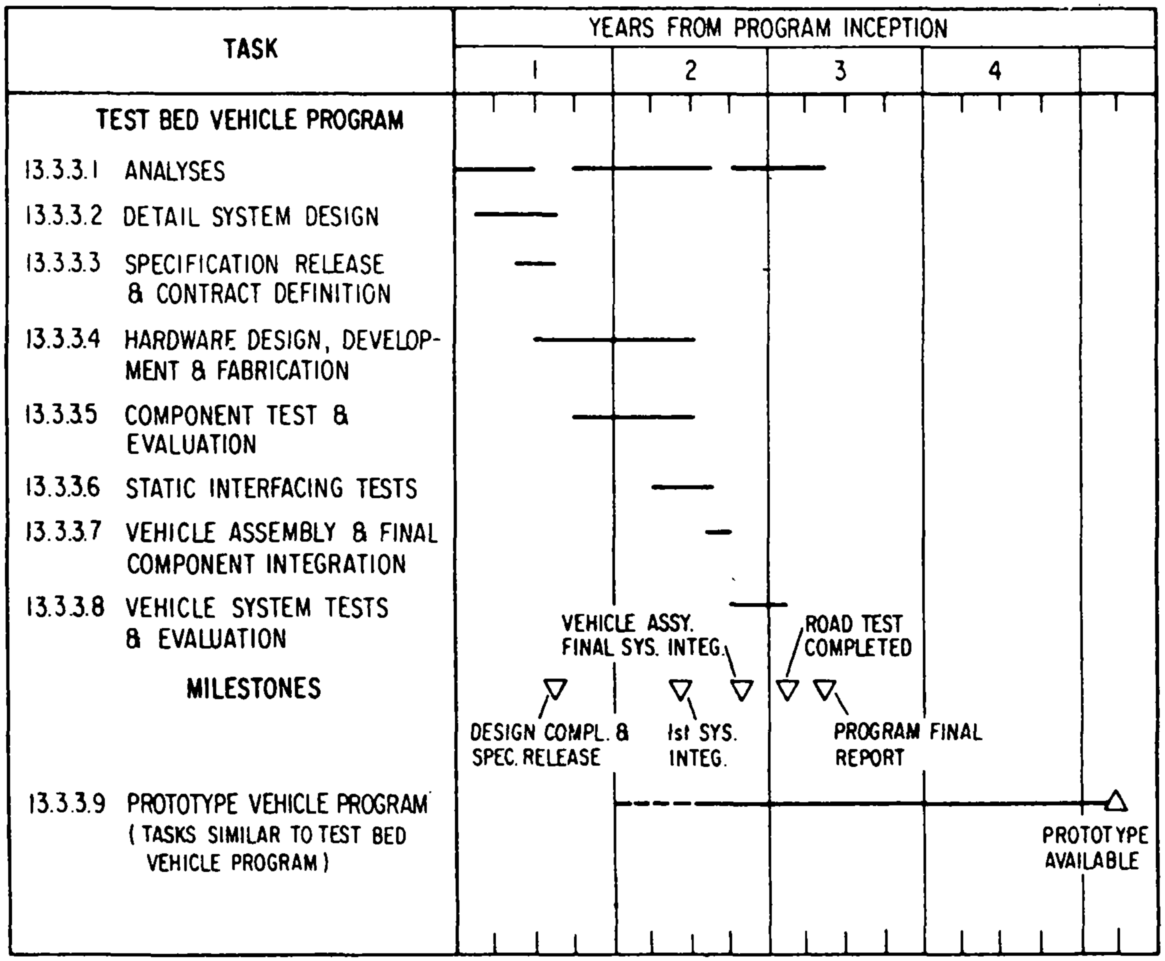

Figure 13-3. Test Bed Vehicle Development Program Schedule