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DOT/NASA COMPARATIVE ASSESSMENT OF BRAYTON ENGINES FOR GUIDEWAY VEHICLES AND BUSSES

VOLUME II—ANALYSIS AND RESULTS

NATIONAL AERONAUTICS AND SPACE ADMINISTRATION

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VOLUME II—ANALYSIS AND RESULTS

Prepared by Lewis Research Center

Scientific and Technical Information Office
NATIONAL AERONAUTICS AND SPACE ADMINISTRATIO

PREFACE

This is the second volume of a report describing a study of gas turbine engines for heavy duty transportation. The study was conducted by the NASA Lewis Research Center with support from the Marshall Space Flight Center at the request of the Department of Transportation (DOT). The objectives of this study were to provide a definition of the potential for turbine engines to minimize pollution, energy consumption, and noise; to provide a useful way of comparing these engine types based on consistent assumptions and a common analytical approach; and to provide a compendium of comparative performance data that could serve as a technical basis for future planning. Closed, semiclosed, and open cycle gas turbine engines were studied. Emphasis was placed on making a consistent comparison among these engine types. The background and reason for this comparative assessment of Brayton engines for the heavy duty transportation application is given in the INTRODUCTION to volume I.

As stated in the PREFACE to volume I, contributions to the study were made by numerous individuals, many of whom prepared sections of this volume. This second volume contains all of the results dealing with engine comparisons as well as supporting studies. In volume I much of the information from this volume is summarized and conclusions of the study are given.

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INTRODUCTION

This volume contains the technical assessment of several Brayton engine types for application to the guideway vehicles and buses described in volume I. Because of the volume of technical information contained herein, the arrangement of volume II is now explained.

Section 1 contains a summary of the assumptions, applications, and methods of analysis. Also included are a discussion of the approach taken, the technical program flow chart, and the weighting criteria used for performance evaluation.

The technical summary and discussion are included in section 2. The results of both the engine screening phase and the conceptual design phase are presented. The unique feature of this section is the engine comparison discussion which is made on the bases of weight, performance, emissions and noise, technology status, and growth potential. Although most of the information and results in section 2 were taken from the sections following it, the different types of engines are compared only in section 2.

Sections 1 and 2 give the reader a general understanding of how this study was conducted and an indepth summary of the results and conclusions.

The remainder of volume II contains the detailed study results, engine compartment layouts, component analysis, and methods of analysis for the reader interested in greater detail. Sections 3 and 4 constitute the engine screening phase. In addition to the results of the preliminary cycle screening, section 3 presents a description of the cycles considered with the group designations used throughout this report. Section 5 presents the results of the conceptual design phase; the results cover both design-point and off-design performance.

Following section 5 are the technical appendixes which contain information that supports the analysis. Appendixes A to F include descriptions of the heat exchangers and combustors, a discussion of the design-point and off-design analysis techniques, and considerations related to emissions and noise, engine compartment layouts, and engine-mission integration. Appendixes G to I contain supporting studies.

1. STUDY METHODS

The engines analyzed in this study included many variations of open, semiclosed, and closed Brayton cycles. For convenience the engine types were grouped by thermodynamic similarities into five categories. Groups I and II are closed Brayton cycles which differ in the type of combustion loop used. Group I engines have a combustion loop using a conventional combustor with either excess air (Ia) or recirculated combustion gas (Ib) as the diluent. Group II engines have an integrated combustor heat source heat exchanger. The semiclosed Brayton cycle engines considered were designated as group III. Group III engines use combustion gas as the working fluid, and most of it is recirculated back to the combustor as the diluent. The group IV engine is a special case of the semiclosed cycle engine which uses hydrogen as the fuel and oxygen as the oxidizer. In this case the recirculated combustion gas is condensed and returned to the combustor as liquid water. The open-cycle Brayton engine, with and without recuperation, was designated as

TABLE 1-1. - ENGINES CONSIDERED IN STUDY

Engine designation	Engine description
Group I:	Closed Brayton cycle; diluent controlled combustor exit temperature
Ia	Excess air diluent
Ib	Recirculated combustion products diluent
Group II	Closed Brayton cycle - combustion temperature controlled by heat transfer in combined surface combustor and heat source heat exchanger; near stoichiometric fuel/air ratio
Group III	Semiclosed Brayton cycle; recirculated combustion products as working fluid and pressurized by turbocharged combustion air; near stoichiometric fuel/air ratio
Group IV	Similar to group III except hydrogen and oxygen are reactants with steam as working fluid
Group V	Open Brayton cycle

group V. This engine was considered in the conceptual design phase. The engine group descriptions and designations are summarized in table 1-1. The engine groups are fully described in section 3.

Closed and semiclosed engines (groups I to IV) were studied over a wide range of conditions in the engine screening phase in order to select an engine type for a more detailed study. The selected engine was then compared with open recuperated and unrecuperated engines (group V) in a conceptual design phase. The fuels considered were cryogenically tanked hydrogen and methane and a hydrocarbon fuel typified by kerosene.

General Approach

A flow diagram of the study procedures is presented in figure 1-1. The study was divided into two major phases, an engine screening phase and a conceptual design phase.

ENGINE SCREENING PHASE

The first step in the engine screening phase was a preliminary thermodynamic comparison (see section 3) of many of the possible variations of open, semiclosed, and closed Brayton cycles including reheat and intercooling. This cycle screening involved both quantitative comparisons based on cycle efficiency and flow rates and qualitative comparisons based on factors such as complexity, estimated relative size, and expected off-design power level performance. The quantitative comparisons were based on preliminary, simplified cycle calculations that neglected fan power parasitics. This thermodynamic comparison was to reduce the number of engines to be considered in the more detailed engine screening analysis.

The engine screening analysis (see section 4) was performed to further reduce the number of engines and fuels being carried through the evaluation process. In this task of the engine screening phase consideration was given to two applications: These were a 400-horsepower urban bus and a 300-mph tracked air-cushion vehicle (TACV). A number of engine variations were considered for the TACV. These variations included single- and two-shaft engines, use of turbine cooling, intercooling during the compression process,

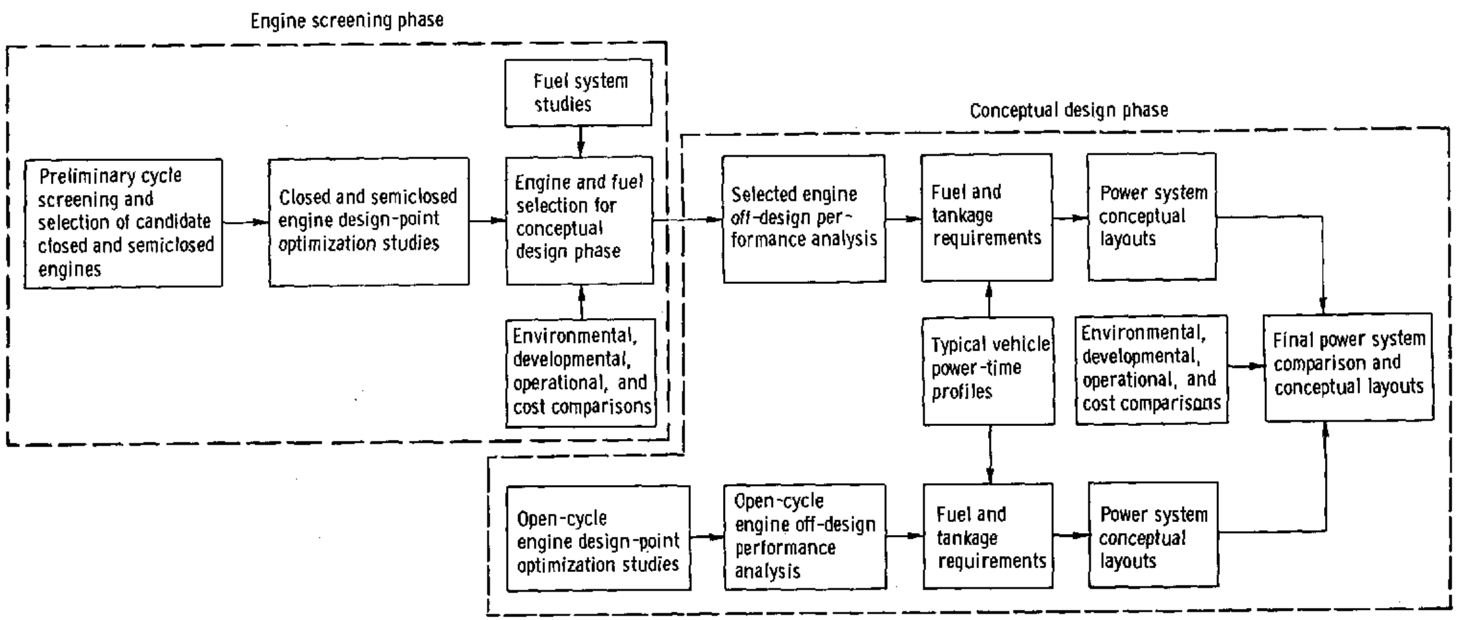


Figure 1-1. - Study flow diagram.

and several working fluid molecular weights in the closed-cycle engines. Methane was the reference fuel used for specific fuel consumption (SFC) and weight comparisons except for the hydrogen-oxygen engine (group IV). In order to provide good performance over a range of power, the use of an infinitely variable transmission was assumed for all engines. For the bus application simplicity and compactness were considered to be important attributes. For this reason bus configurations were limited to single-shaft engines. No intercooling was used, and turbine inlet temperatures were limited to values attainable without the use of turbine cooling.

Engine performance was based on net engine power including fan power parasitics. Comparisons of design-point SFC and engine specific weight, together with qualitative consideration of performance at off-design power levels, complexity, cost, emissions, noise, growth potential, and technology status, were used to select the engine type and fuel to be used for the more detailed study and comparison with open-cycle engines (group V) in the conceptual design phase.

The evaluation of the fuels was made independently of the engine performance analysis and is presented in appendix G. The fuel selection was based on availability, safety, and handling and also on required fuel-tankage system weight and volume for specified applications.

CONCEPTUAL DESIGN PHASE

The conceptual design phase (see section 5) was devoted to a more detailed comparison of the performance and characteristics of two types of engines. The best engines of the closed and semiclosed engines were compared to the open-cycle engines on the basis of design-point performance, off-design power level performance, and engine-vehicle integration (see appendix E). Performance was based on reference fuel used. Off-design power-level performance was quantitatively evaluated in this phase. A detailed analysis is necessary because of the variations in the off-design SFC between the open and closed or semiclosed engines. The off-design performance of an engine is reflected in system weight by determining the fuel and tankage requirements for a particular mission profile as described in appendix F. To better assess the effects of off-design power-level performance four applications were considered in the conceptual design phase. In addition to the urban bus and the 300-mph TACV, a 150-mph urban TACV and a present-day locomotive application, both powered by single 5000-horsepower engines, were considered. In this phase, the configurations considered for each application were limited to those now discussed.

In the conceptual design the bus engine configuration and operating limits were identical to those used in the engine screening phase. In this design phase both cooled and uncooled turbines were considered for the TACV and locomotive applications. The effect of intercooling was investigated. Two versions of the 300-mph TACV were considered. The first used two 7500-horsepower single-shaft engines coupled to infinitely variable speed transmissions to produce the required 15 000-horsepower net electric power. The transmission for each engine had two output shafts operating at independently variable speed ratios. One output shaft operated at constant speed to drive the alternator supplying the levitation fan, train auxiliary, and cooling fan loads. The other shaft drove the alternator supplying the LIM power at speeds determined by the vehicle speed requirements. Engine speed could be varied to provide good part-load performance. The second version used two engines, each driving a 5000-horsepower LIM drive alternator through an infinitely variable speed transmission. A third engine operating through a fixed ratio gear box was used to drive the 5000-horsepower alternator which supplied power for the levitation fans and train auxiliaries.

The 150-mph urban TACV used a single engine to provide the 5000-horsepower electric power for both motive and auxiliary power. A single-shaft engine coupled to an infinitely variable speed transmission with two output shafts similar to that used for the 300-mph TACV was assumed for this application. A single 5000-horsepower engine was also assumed for the locomotive application. In this case, a two-shaft configuration was assumed to supply mechanical power through an infinitely variable speed transmission.

Fuel consumption over the various assumed vehicle mission power-time profiles was used as a basis for comparing the engine groups in the conceptual design phase rather than the design-point fuel consumption. The vehicle mission power-time profiles used were simplified profiles based on a series of steady-state operating power levels over the mission or trip time. Periods of acceleration were approximated by a number of steady-state operating power levels. The power-time profiles assumed for each vehicle are described in appendix F. The comparative transient response, such as acceleration, was limited to qualitative considerations. Other engine attributes that were considered were complexity, cost, emissions, noise, growth potential, and technology status. The emissions and noise characteristics are discussed in appendix D.

Consideration was given to the ease of integrating the engine types into an assumed engine compartment for each application. As discussed in section 1 (volume 1), no attempt was made in this study to design a vehicle around an engine. Instead existing vehicles or concepts were used. A present-day urban bus and locomotive were used for those applications. The TACV configuration was obtained from a survey of high-speed ground transportation concepts that resulted from previous Department of Transportation work. The vehicles are described in detail in appendix F.

Technical Approach

Although this study was conducted in two phases, an engine screening phase and a conceptual design phase, the technical approach can best be presented as a design-point evaluation and a mission-performance evaluation. The screening phase was based only on design-point analysis while the conceptual design phase used both design-point and mission-performance evaluation. The total calculational procedure flow diagram is shown in figure 1-2.

DESIGN -POINT EVALUATION

For engine design-point evaluation, cycle calculation subroutines were written for each engine type. These subroutines included options for a number of possible variations. These variations included single- and two-shaft engines, use of reheat within the expansion process and/or intercooling during the compression process, use of turbine cooling, use of alternate fuels, and provisions for considering different molecular weights of the cycle working fluid in the closed Brayton cycle engines. Design subroutines (see appen-

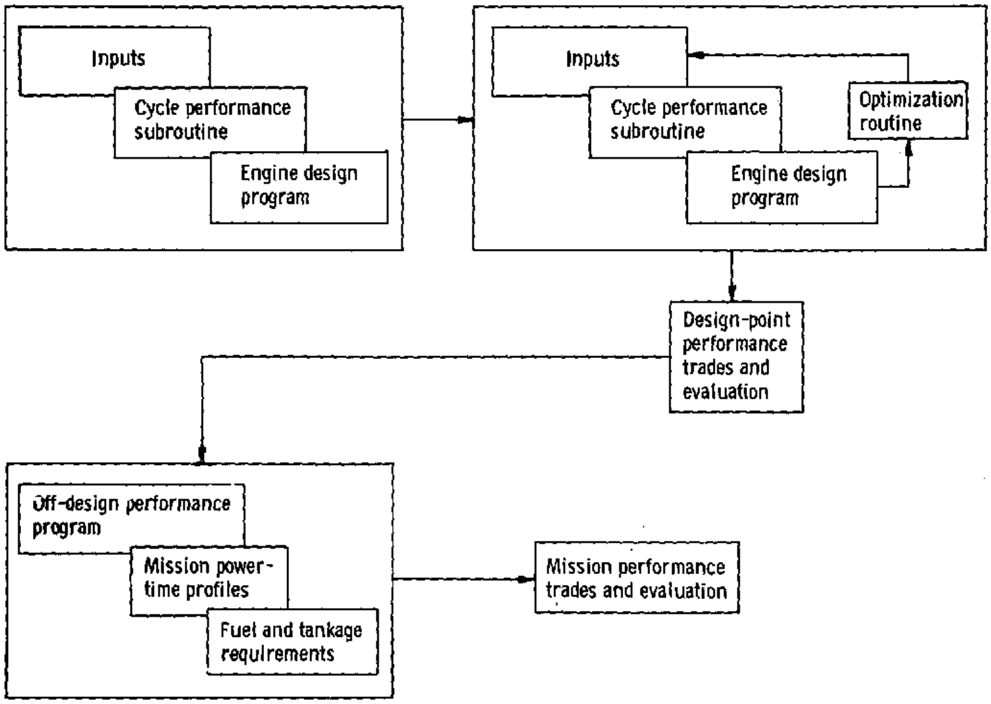


Figure 1-2. - Calculation procedure flow diagram.

dixes A to F) were written for each major component of the engines to form an engine design program. In writing the component design subroutines, emphasis was placed on those which were regarded as critical from the standpoint of weight and those which were most influential in determining optimum engine design parameters. These components were primarily the heat exchangers. Less emphasis was placed on the other components. It was found in the initial sizing calculations of the waste heat exchangers for groups I to IV that the fan-cooled, air-flow frontal area needs were large. Also, if dimensional constraints were not used, the resulting heat exchangers could not be conveniently packaged in a vehicle (see appendix A). Therefore airflow frontal area constraints of 2 by 4 feet in the bus and 9 by 18 feet in the train were assumed based on intended vehicle envelopes (see appendixes E and F). Although these area constraints could not be exceeded, dimensions less than these limits were allowed.

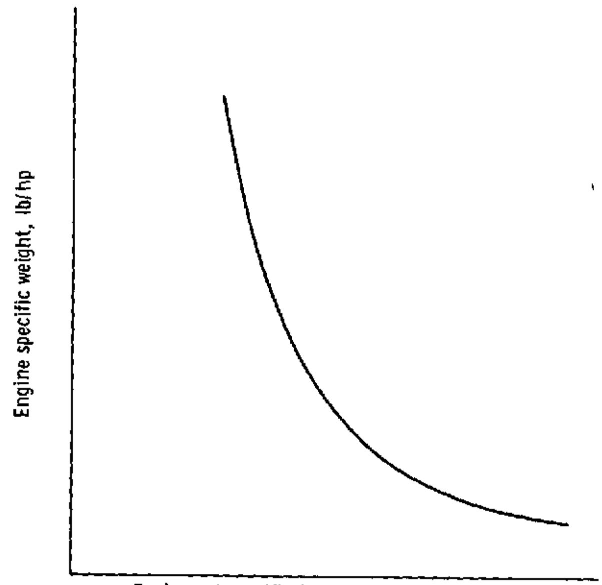


Figure 1-3. - Engine specific weight and design-point performance.

The cycle subroutine and engine design program combined with an input array to form a system design-point program. A design-point power system optimization program (PSOP) was obtained by closing the loop between the system design program output and the input array through an optimization routine. This PSOP permitted as many as 40 engine parameters to be varied so as to optimize design-point performance for any desired figure of merit. The figure of merit could be any combination of performance parameters or operational characteristics with appropriate weighting given to each. Performance curves typical of the example shown in figure 1-3 were generated by using SFC and engine specific weight in the figure of merit. Trade-offs in design-point performance were obtained for each engine type and variation by varying the relative weight given to SFC as compared to specific weight. Each point on such a curve represents the weight of an engine with operating parameters optimized for minimum weight for that particular value of design-point SFC. A number of engine designs covering the range from low to high design point SFC was selected from the design-point performance curves, and the corresponding design-point parameters were put into the off-design performance program.

MISSION - PERFORMANCE EVALUATION

As discussed in the previous section, a number of engine designs were selected from the design-point trades for input into a mission-performance program in order to permit an evaluation of the engines over representative vehicle mission power-time profiles. In the mission-performance evaluation variations of SFC as a function of power over a range of engine speeds were obtained from the off-design performance program. An example is shown in figure 1-4. The off-design performance characteristics were used in conjunc-

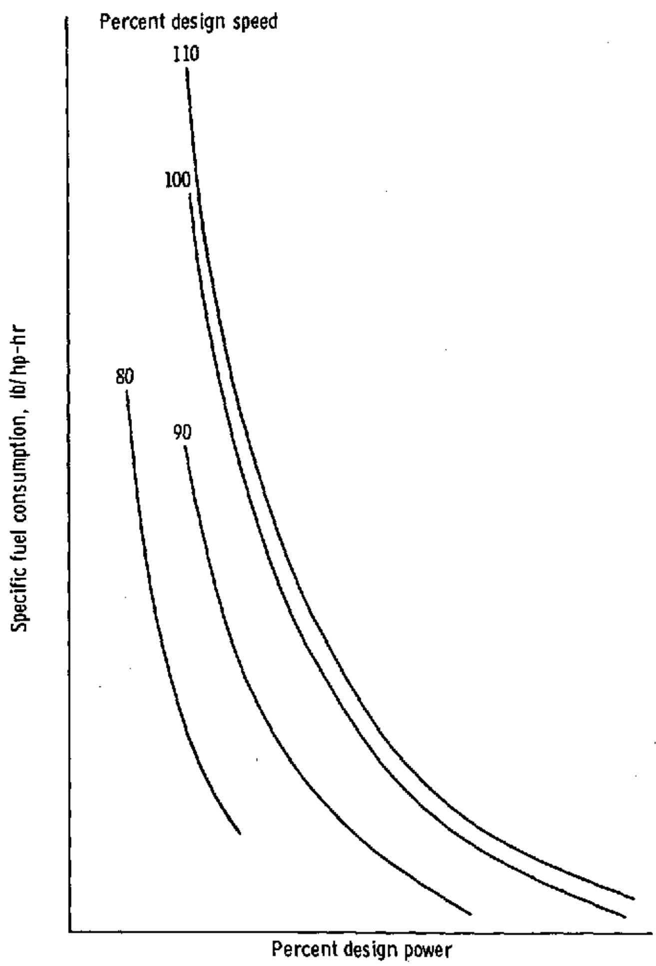


Figure 1-4. - Engine off-design performance.

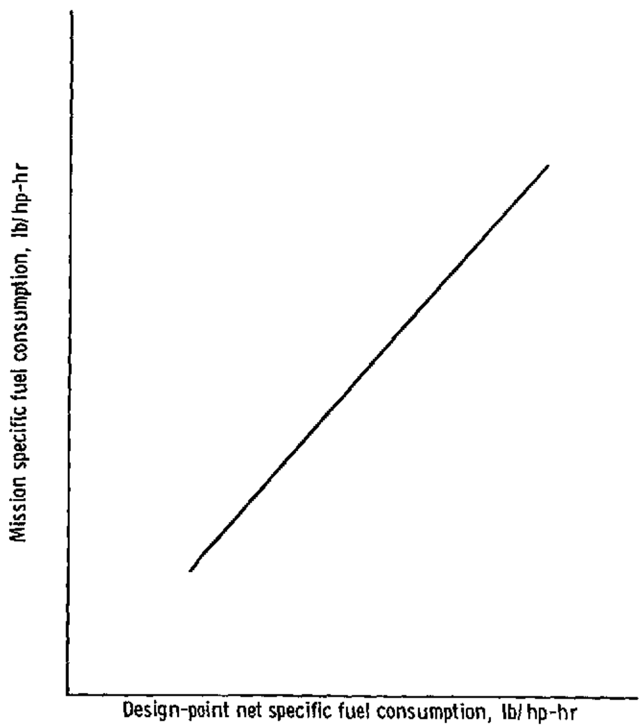


Figure 1-5. - Variation of mission specific fuel consumption with engine design-point specific fuel consumption.

tion with the appropriate mission power-time profile and the vehicle transmission efficiency (function of engine speed and load) to show the variation of mission SFC (total fuel used in mission divided by total mission energy requirements in horsepower-hours) as a function of design-point SFC for each engine and mission. An example of a resulting curve of mission SFC as a function of engine design-point SFC is shown in figure 1-5. Finally, total fuel requirements were calculated from the mission SFC and mission energy requirements. The fuel requirements were combined with the required tank weight and engine weight to show the variation of total power system specific weight as a function of fuel load for each engine and mission. An example is shown in figure 1-6. This curve illustrates the trade-off between the total power system specific weights (engine plus fuel and tankage) and the fuel load for the specified mission. Each point on this curve represents an engine with design-point parameters optimized to result in minimum total power system

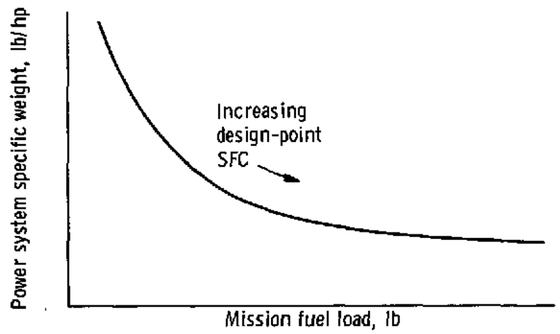


Figure 1-6. - Variation of power system specific weight with mission fuel load.

weight for the mission for any given fuel load. The curve serves to demonstrate that, although minimum total power system weight occurs at a rather high fuel load, it is possible to reduce substantially the quantity of fuel required for the mission by selecting engine designs with lower design-point SFC's but higher weights. The effect is a trade-off between reduced fuel consumption and an increase in total power system weight (engine plus fuel and tankage) required to perform the mission.

Final comparisons between the engines for each application were made on the basis of performance including power system weight, volume, and specific fuel consumption, estimates of environmental effects including emissions and noise, and qualitative evaluations of such factors as growth potential, technology status, multifuel and multipurpose capabilities, and engine compartment integration.

2. TECHNICAL SUMMARY AND DISCUSSION

In this section the different engine types are compared on the bases of weight, performance, emissions and noise, technology status, and growth potential. In addition to making the engine comparisons, since most of the information used was taken from following sections, this section also serves as a summary of the results of the study. For the reader interested in greater depth, sections 3 to 5 present the complete, detailed results of this study.

This section is divided into two parts, the first is concerned with the engine screening phase and the second with the conceptual design phase. In the engine screening phase the off-design performance comparisons between closed and semiclosed cycles were made qualitatively, while in the second phase the off-design performance of the semiclosed and open cycles was analyzed quantitatively. Therefore, in the conceptual design comparisons between semiclosed and open cycles, off-design performance is quantitatively combined with the engine weights from section 4 and the mission requirements from appendix F to show the trade-off between total power system weight (engine, transmission, fuel, and tanks) against the total fuel consumed for a variety of mission applications.

The conclusions of each phase of the study are summarized at the end of the part of this section dealing with each phase. The conclusions or discussion dealing with the study as a whole are summarized in the final parts of this section. The last section addresses the general logic for selecting and applying the gas turbine type systems.

Engine Screening Phase Comparison

HYDROGEN - OXYGEN ENGINE AND BRAYTON ENGINES

The stoichiometric hydrogen-oxygen engine (group IV) was examined because of its long-term future potential as an engine with zero emissions and high efficiency. This concept could also provide a basis for adapting the heavy duty transportation system to a possible future hydrogen economy. The technology basis for this engine is largely at hand with the extensive hydrogen-oxygen rocket engine experience of the space program; but, such powerplants have not yet been carried beyond the prototype stage. Several studies of this type of powerplant for both mobile and stationary application have been made. Furthermore, NASA is developing the technology for a hydrogen-oxygen auxiliary power unit (APU) which incorporates several features of this system but which controls combustion temperature by hydrogen dilution rather than water injection.

As shown by the results of section 4, the group IV engine weight is lower than that of any of the closed or semiclosed Brayton engines considered (group I, II, or III). However, group IV, unlike any of the others, requires that the oxidant be carried aboard the vehicle. As a result, for the same mission, the weight of consumables (hydrogen and oxygen) and the weight of the tanks required are significantly greater for the group IV engine. Consequently, even though the engine weight is less, the total power system weight (including consumables and tanks) is greater for the group IV engine than for the others.

It is also shown by the engine design-point analysis that the dimensions of the air-cooled condenser for the group IV engine would be a problem in packaging the engine and integrating it into the vehicle. The engine optimization can be constrained in order to control the maximum dimension of the condenser but at considerable expense in reduced efficiency, which in turn increases the weight of the required consumables (fuel and oxygen).

Because of the scope of this study, the potential of this engine was not fully investigated; for example, the potential performance with high turbine inlet temperature (above 1800° F) was not analyzed. Also, for other applications, where weight of the consumables is not as significant a factor or where air cooling is not required, the comparison of the group IV engine with the others would have to be reexamined. However, on the basis of the results already discussed, the group IV engine was not considered further in this study.

CLOSED AND SEMICLOSED BRAYTON CYCLE ENGINES

Weight and Performance

As previously described the engine performance comparisons between the closed and semiclosed engines were made on the basis of design-point SFC without quantitative consideration of off-design performance. This was done on the assumption that both engine types have the potential for a rather small variation in SFC with power level so that the design-point SFC is a good indication of engine performance over a wide range of power. The off-design operation of these engines is discussed further in this section after the comparisons between engine weight and design-point SFC are discussed.

The engine design-point analysis (section 4) shows that the net efficiencies of groups Ia and Ib are comparable but that the group Ia engine is heavier. The air-flow rate for the group Ia air preheater is much greater than it is for the group Ib engine. As a result the group Ia preheater weight is greater, and this accounts for most of the difference in engine weights.

The group II engine is lighter and more efficient than either variation of the group I engine. The total flow rate in the group II combustion loop is much lower than that of group I. As a result the combustion loop fan power requirements are lower, and the resulting net efficiency is higher. Also, because of the lower flow rate the combustion loop weight for the group II engine is lower, and the resulting total engine weight is lower.

Variations in the closed Brayton cycle such as the reheat of the gas during expansion and/or the cooling of the gas during compression increase the gross cycle efficiency by narrowing the range of temperatures at which heat is added to or rejected from the cycle. However, this is at the expense of adding a reheat and/or intercooler heat exchanger to the system. The analysis showed that even though adding reheat or intercooling increases the gross cycle efficiency this increase is not fully reflected in the net efficiency. The cycle analysis of section 3 shows that the addition of reheat significantly increases the combustion loop flow rates, and in the case of group Ia the associated increased thermal losses of the combustion loop result in a decrease, rather than increase, in net overall efficiency. The effect of including intercooling is examined in section 4; here it is shown that the net efficiency was decreased by adding intercooling for groups I and II due to the increased fan power required for coolant air. It should be noted that this conclusion is influenced by the fact that in order to meet the guideline engine compartment dimensions the engine optimization was constrained so that the combined maximum dimensions of the intercooler and waste heat exchangers were the same as those for the waste heat exchanger alone for the design without intercooling.

A comparison of the optimized engine design points for the closed- and semiclosed-cycle engines in section 4 shows that for the train application groups II and III are comparable, with group III being slightly lighter. In the case of the bus engine group III is noticeably lighter and more efficient. As explained in section 4 these comparisons are a function of the constraints concerning maximum cycle pressure and waste heat exchanger size placed on the optimization of the engine design. For group III intercooling was shown to result in an increase in net system efficiency.

When comparing engine weight and performance for the closed- and semiclosed-cycle engines an important factor to consider is their corresponding limits on turbine inlet temperature. The turbine inlet temperature for closed-cycle engines is limited by the temperature capabilities of the heat-source heat exchanger. In the designs considered in section 4 the turbine inlet temperature for the closed-cycle engines was taken as 1500° F, which corresponds to a maximum heat-source heat exchanger metal temperature in the range of 1700° F. Since the group III engine uses the combustion gas as working fluid, eliminating the heat-source heat exchanger, the turbine inlet temperature can be correspondingly higher. For the comparisons made in section 4 the turbine inlet temperature for group III was taken as 1700° F so that the maximum turbine surface temperatures would roughly correspond to the peak temperature in the closed-cycle heat-source heat exchanger. Furthermore, since the turbine inlet temperature of the group III engine is not limited by a heat-source heat exchanger, the group III engine can use turbine cooling to permit still higher turbine inlet temperatures and hence higher efficiency. However, as the turbine inlet temperature is increased the turbine exhaust temperature, which is the inlet to the recuperator, increases until the recuperator temperature reaches the same limitations assumed for the closed-cycle heat-source heat exchanger. This limit on the turbine inlet temperature is about 2100° F for the group III optimized engine design points considered.

One of the most commonly cited attributes of the closed-cycle Brayton is its ability to maintain high efficiency over a wide range of power level. This is accomplished by adjusting the system inventory, and hence mass flow, with essentially constant engine speed and temperatures. The semiclosed-cycle Brayton was shown in section 3 to be mainly closed from the thermodynamic standpoint. Therefore, with an inventory adjustment for the power level control, the semiclosed cycle can be expected to share this attribute of the closed cycle (this was shown to be true in the second phase of the study in section 5).

The closed-cycle engine requires a separate inventory adjustment system which would rapidly extract gas from or pressurize and inject gas into the otherwise closed power system gas loop. The weight of such an inventory control system and the power required for it were not included in the closed-cycle engine results presented in section 4. Although conceptually simple, a practical scheme for accomplishing inventory adjustment to operate the engine by this method over rapid power transients was not found during the study. The alternative is to use inventory adjustment for long-term changes in power level but not for rapid transients or short-term power changes. This would result in a reduction in efficiency during those periods in which inventory adjustment is not used. This effect on performance was not used. This effect on performance was not evaluated.

Since the semiclosed cycle is selfpressurizing, it does not require a separate inventory adjustment system. Control of the turbocharger can be used to control system inventory. A preliminary examination of a control scheme to accomplish this is included in appendix C. Further analysis is required to predict the time response of this power level control. The weight of the turbocharger and its effect on engine performance was included in the engine design-point results presented in section 4. From this standpoint the results presented for the group III engine are more complete than those for the closed-cycle engines.

Exhaust Emissions and Noise

A commonly recognized advantage of a closed-cycle Brayton engine is that of external near-ambient-pressure combustion with a tolerance of a wide variety of fuels. This feature provides both flexibility and low emissions. However, as shown by the engine performance analysis, it is necessary to use combustion inlet air preheat for good efficiency. The effect of combustor primary air inlet temperature as well as pressure level and combustor type must be considered when comparing the engine types on the basis of exhaust emissions.

Groups Ia and Ib differ only in the source of combustor diluent. This does not substantially alter the primary combustion zone, and there is no first-order difference between the two from the standpoint of exhaust emissions. For emission comparison purposes these two can be considered as a single class.

In the group II combustor a surface combustor is integrated with the heat-source heat exchanger, and heat transfer from the combustion zone to the heat exchanger is used to control temperature. The peak temperatures can be limited in this way to low enough levels to reduce considerably the production of NO_x while still providing sufficiently complete combustion to limit the generation of carbon monoxide and hydrocarbons. From the discussion in appendix D it can be seen that the low emissions potential of the surface combustor is substantially better than that of the diluent class of combustor. As a result the emissions control potential of the group II engine is superior to that of the group I engines.

The group III engine as analyzed uses a combustor similar to that of group Ib which uses recirculated combustion products as the diluent. The group III combustor, since it is within the power conversion gas loop, operates at peak cycle pressure compared to the near-atmospheric-pressure operation of the group I combustor. As discussed in appendix D this would result in higher NO_x emissions for the group III engine. The emissions from the group II engine, which uses a surface combustor, would be substantially lower than those from the group III engine. However, a surface combustor with this semiclosed cycle is also possible. In this way the emissions might be reduced to a level near that which is possible with the group II engine. However, such a combustor would probably be larger than the diluent-controlled type of combustor considered in the analysis. Also a surface combustor - heat exchanger could be used with a semiclosed cycle such as that described in section 3 figure 3-1 (p. 44) in which case it would operate at the lower pressure level of the cycle. This variation in the semiclosed cycle was not studied. Furthermore, a catalytic combustor (see appendix D) which has the potential for very low emission levels might be used with either the closed or semiclosed cycle. In the case of the semiclosed cycle, an arrangement such as that in figure 2-7 (p. 32) of section 3 could be used with a catalytic combustor.

It was therefore concluded that, although the emissions potential of the group II cycle is superior to that of the group III cycle, the group III (or some other variation of the semiclosed cycle) could be improved from the emissions standpoint so that this is not a strong factor in selecting between the two groups.

In considering the use of a surface (or catalytic) combustor another factor which must be included is that the fuel and primary air are premixed and then introduced to the combustion zone. This will affect techniques of shutdown and startup and may in addition affect performance by limiting the amount of air preheat in order to avoid autoignition of the fuel-air mixture. A limitation on preheat would result in a reduction in performance of the group II engine. If a surface combustor (or catalytic combustor) were used with a semiclosed cycle a similar limitation on primary air temperature would result in a limitation on the pressure ratio of the air supply compressor pressure ratio and hence the cycle pressure level. This would result in some weight penalty and a performance penalty. The resulting performance penalty of the semiclosed engine would not be expected to be as severe as that of the closed cycle.

For the closed Brayton systems the dominant noise sources are those contributions external to the power conversion loop, i. e., coolant and combustion air fans, transmission, and alternator. Experience indicates that the transmission is a strong noise source but should lend itself to a state-of-the-art external acoustic treatment. Since the transmission is common to all the power systems considered, however, it is not a strong factor when selecting between the systems. The other noise sources were examined in appendix D, and it was found that these sources could be limited below 75 dBA for all closed systems.

In the semiclosed cycles, the turbocharger and the rejection of gas from the pressurized loop present noise sources different than those of the closed engines. The turbocharger is similar to an open-cycle gas turbine and as such is amenable to acoustic treatment on the same basis as current technology efforts related to quieting aircraft engines. The application of these engines to ground systems also permits additional noise attenuation to be incorporated into the engine compartment. As concluded in appendix D, for comparable levels of treatment the group III engine noise may be 5 dBA higher than that for the closed systems. This difference was not considered to be large enough to make noise levels a strong influence when selecting between these groups of engines.

Technology Status and Growth Potential

The closed Brayton system is neither new nor untried. A number of such plants are in operation in Europe (stationary applications) combined with waste heat utilization. Also, NASA has had a substantial program in the development of the closed Brayton for space power. To this time, however, the closed Brayton has not been adapted to mobile terrestrial applications.

The thermal input to the closed cycle is through a heat-source heat exchanger. As discussed previously the materials technology of this heat exchanger limits the turbine inlet temperature and hence the potential performance of the engine. This heat exchanger must be able to withstand not only the temperature level but thermal cycle fatigue due to engine startup and shutdown, and the substantial pressure difference between the combustion gas side and the working gas side. On the basis of current and forecast materials technology it appears that this heat exchanger will limit the turbine inlet temperature to about 1500° F.

The group II variation of the closed Brayton cycle uses a surface combustor and heat-source heat exchanger integrated into a single unit. Such a combustor - heat exchanger has been demonstrated as being feasible but the concept still requires development.

The semiclosed-cycle Brayton engine (group III) combines some of the advantageous characteristics of the closed- and open-cycle Brayton engines, but it apparently has not been reduced to practice. The cycle configuration analyzed in this study employs a conventional type combustor and uses the combustion gas as working fluid. As previously discussed this eliminates the heat-source heat exchanger and its limitation on turbine inlet temperature. Use of the state-of-the-art turbine cooling would therefore allow higher turbine inlet temperatures for this cycle.

A disadvantage of the use of the combustion gas as the working fluid, however, is that condensation from the combustion products in the waste heat exchanger will result in the need for special attention to materials compatibility and for water extraction in the heat exchanger. Although this does not appear to detract from feasibility, the materials selection may adversely affect cost and weight in comparison to the closed-cycle waste heat exchangers.

As discussed in the Weight and Performance section, the best off-design performance of a closed-cycle Brayton is attained when the power level is controlled by inventory adjustment. However, the complexity and attendant development problems for inventory control for the closed cycle were not assessed since a practical scheme for accomplishing inventory adjustment for rapid power transients was not found. To the extent that rapid transient response using inventory adjustment is required, control of the closed Brayton is an unresolved issue. For the semiclosed cycle the control of the power level by inventory adjustment can be accomplished by controlling the turbocharger (appendix C). The control of this turbocharger is, however, a developmental problem.

In order to operate either the closed- or the semiclosed-cycle engines with inventory adjustment to control power level, the engine speed remains constant, independent of power level or load speed. As a result, an infinitely variable transmission would be required. These have been developed and are commercially available in sizes up to several hundreds of horsepower capacity. Further development is required in the power level range required for the train application. The use of an infinitely variable transmission also requires a sophisticated control system. Since the requirement is assumed common to all the engines and therefore is not a factor in selection between them, the control system was not studied.

FUELS AND TANKAGE

During the engine screening phase of the study the closed and semiclosed engines were considered using several fuels. A fuels evaluation was made independent of the engine performance analysis. All of the Brayton engines considered are tolerant of a wide range of fuels without the restraints of cetane or octane numbers, for example. Thus the selection of fuel was decoupled from engine selection and was analyzed separately by considering such factors as economics, availability, and hazards qualitatively and such factors as weight and volume quantitatively.

It is clear from the results shown in section 4 and appendix B that the use of methane or hydrogen fuel is a disadvantage (in weight and predominantly in volume) compared to the use of a liquid hydrocarbon fuel such as kerosene for the applications investigated. This is particularly true for the TACV application. On this basis alone the further consideration of methane (or hydrogen) in this study might be dropped. However, it should be recalled that an objective of this study was an assessment of the potential, including flexibility, of the engine types considered. Any of the types of Brayton engines considered would be more easily accommodated by the present ground transportation system if it were designed for using a more conventional fuel such as kerosene or diesel fuel. If in the future a change in the transportation fuel economy resulted in the introduction of hydrogen or methane the Brayton engine could accommodate the change while the current diesel engine might become obsolete.

The heavy-duty ground transportation system (principally rail) presently accounts for approximately 4 percent of the total energy consumed in the transportation sector of the United States. It does not appear therefore that at this time a selection of a new fuel for the heavy-duty ground transportation system will be a strong factor in dictating the future fuel economy for the nation. On the other hand, because of the relatively small demand and depot type fueling of the heavy-duty system, new fuels could be introduced into the heavy-duty ground transportation sector as an evolutionary step if future changes do occur, or are required, in the total transportation fuel economy of the nation. Using an engine capable of using both present and future fuels would be the first evolutionary step.

As discussed in appendix A, the cost for the alternate fuels is substantially higher and likely to remain so for the foreseeable future, even on the basis of synthetic hydrocarbon fuel liquids from coal. If synthetic fuel liquids from coal do become a viable and significant factor, it will then be important to assess the additional cost factors involved to obtain acceptable cetane values for diesel applications against the lower process cost and higher yields for external or turbine combustor quality fuels. It is anticipated that this factor will also favor the evolutionary introduction of the alternate engines studied here.

The previous discussion leads to the conclusion that the engine selected to be studied in the conceptual design phase of this study must be compatible with a hydrocarbon liquid fuel as well as alternate fuels (as are the Brayton engines studied, i. e., groups I, II, and III). Also, since it was concluded that in the conceptual design phase it would be more instructive to base the engine design trade-offs on the use of a fuel requiring conventional tankage, kerosene was chosen as representative.

ENGINE SELECTION OF ENGINE SCREENING PHASE

On the basis of the engine comparisons already discussed one of the engines analyzed in the engine screening phase was selected by the DOT/NASA Steering Committee for further consideration in the conceptual design phase. The basis for this selection is summarized in this section. As explained in the previous section, kerosene was selected as the fuel to be considered in the conceptual design phase.

The group IV engine was not selected for further consideration. The reasons, in addition to the fact that kerosene was selected as the fuel for the second study phase, were the large weight and volume of the required hydrogen, oxygen, and tankage and the large size of the condenser required for efficient operation.

Of the closed cycles (groups I and II) considered, the group II engine is clearly superior on the basis of weight, performance, and exhaust emissions. On the basis of noise level the two groups should be comparable. Cost would be comparable except that the combustor - heat exchanger required for group II requires more development. Considering these factors, it was con-

TABLE 2-1. - ENGINE COMPARISON SUMMARY -

ENGINE SCREENING PHASE

[X indicates engine with best performance. When X appears in both columns, no discernible difference was observed. When ? appears, further study is needed to assess engine performance for the criterion.]

Criterion	Group II	Group III
Low specific fuel consumption and weight		X
Low volume		X
Good part power performance	X	X
Growth potential		X
Multifuel capability	?	X
Low emissions potential	X	?
Noise	X	
Minimum technology issues	X	X

cluded that the selection to be made between closed- and semiclosed-cycle engines should be a selection between groups II and III with no further consideration of group I.

The criteria considered in comparing groups II and III are listed in table 2-1 and the two groups are comparatively rated. An X is placed in the column judged to be the best on the basis of each factor listed. As noted in the previous discussions, some of the criteria listed were not individually strong selection factors between the two engine types. Also, for some of the criteria there was judged to be little discernible difference between the engines; in those cases an X was placed in both columns.

On the basis of multifuel capability group II was given a question mark since this aspect of the operation of surface combustors (which group II requires) needs further investigation. From the standpoint of low emissions potential group II (using a surface combustor with near atmospheric combustion) was rated above group III. Group III could also employ a surface combustor to reduce significantly the emission levels compared to what is expected from the diluent-controlled-type combustor used in the particular cycle configuration analyzed. But the applicability of the surface combustor to the higher pressure level of this semiclosed cycle needs further study and therefore group III was given a question mark for this criteria. The group II engine tends to have higher pressures than the group III engine. The consequences of this from the standpoint of hazards have not been assessed, but in general the lower the pressure level the better. In the event of leakage or vandalism the group III engine should be the more forgiving. Because of the full-time inventory control feature of the group III engine (the turbocharger continuously pressurizes the cycle) it should be more capable of maintaining acceptable performance for longer periods of time than the group II engine for any given leak rate. This is especially true if the group II engine is designed to use a working fluid other than air.

The technology issues concerning groups II and III, meaning those areas which must be addressed prior to commitment to development, are summarized as follows:

Group II:

Heat-source heat exchanger limits maximum cycle temperature
Inventory control limits power setting response
Heat exchanger fabrication and life
Requires surface combustor

Group III:

Recuperator limits maximum cycle temperature
Can use variety of combustors but -

Conventional	Emissions are problem
Catalytic	Life must be demonstrated
Surface	Requires development

Heat exchanger fabrication and life
Heat exchanger fouling
Turbocharger performance needs verification

As has been previously discussed the heat-source heat exchanger limits the maximum turbine inlet temperature in group II but not in the case of group III. Substantial advancements would therefore be required to raise the turbine inlet temperature and hence performance of the group II engine. In addition, this combustor - heat-source heat exchanger which is used by group II requires further development. This would also be true for group III if, as discussed previously, a surface combustor or catalytic combustor is used on that type of engine to obtain a potential for low emissions similar to that predicted for group II.

A fundamental problem with any closed or semiclosed Brayton engine will be heat exchanger packaging, fabrication, and life. This is particularly true of the waste heat exchanger. The problem is predominately one of scale which does not appear to be a barrier to development, but rather a question of demonstration and verification. The problem with group III is somewhat more complex; the group III waste heat exchanger must be compatible with the combustion gases and condensed water vapor. Although this does not detract from feasibility, it is important to assess with respect to materials selection. The material selected may adversely affect weight and cost, particularly if a stainless steel is required. Fouling should not be a serious problem but verification is required.

On the basis of the considerations summarized in table 2-1 and those listed previously, the Joint Steering Committee for the study selected the group III engine for further analysis.

Conceptual Design Phase Comparison

SEMICLOSED AND OPEN BRAYTON CYCLE ENGINES

Weight and Performance

As explained at the beginning of this section, the performance comparisons of the various closed- and semiclosed-cycle engines considered in the engine screening phase were made on the basis of design-point SFC. This was justified by the fact that for these engines the variation in SFC with power from the design point is not large. But in the conceptual design phase the open-cycle Brayton was also considered, and the off-design variation in the SFC of this engine was expected to be greater. Therefore, in order to com-

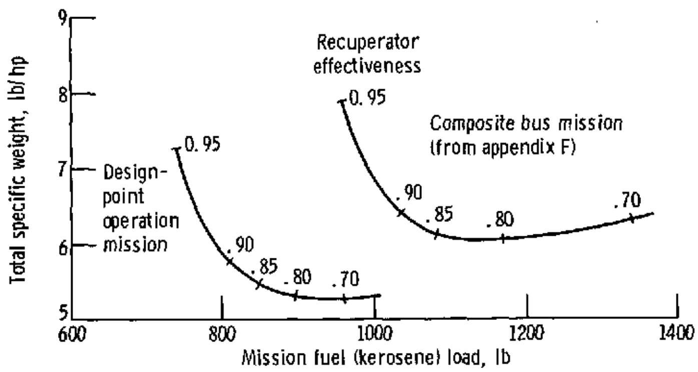


Figure 2-1. - Total system weights for group V. 400-Horsepower bus engine for two missions with equal work done. Total system weight includes engine, transmission, fuel, and tanks.

pare this engine's performance to the others, the off-design performance characteristics must be included.

In this section, therefore, the performance comparison of the semiclosed (group III) engine and the open (group V) engine will be made by comparing the total fuel expenditure for a variety of missions. Also, in this way the effect of off-design performance is brought into the trade-off between total system weight and performance in selecting the design point for each type of engine. As an example, consider figure 2-1 where the group V total system weight (engine, transmission, fuel, and tanks) is given as a function of mission fuel load for two different missions. The composite bus mission referred to is that given in appendix F. Each point on the curve represents a different engine design point taken from figure 5-4(a) (p. 98) of section 5. The design point is identified by the recuperator effectiveness. As the recuperator effectiveness is increased the performance (both design point and off-design) is improved and thus less fuel is required. At the high recuperator effectiveness end of the curve the total weight increases as a result of increasing recuperator weight and at the other end of the curve the total weight increases as a result of increasing fuel and tank weight. The design-point operation mission in figure 2-1 assumes that the engine operates at the design-point SFC throughout the mission, and the mission length is sufficient to produce the same amount of engine work as the composite bus mission. A comparison of the curves shows the effects of the off-design performance of the engine; for the composite bus mission the engine is operated at off-design power levels (and hence with SFC higher than that at design point), which results in comparatively higher fuel loads. As the recuperator effectiveness decreases, the off-design performance of the engine differs more from the design point, and the difference between the required fuel loads for the two missions in figure 2-1 increases. For the design-point operation mission the curve in figure 2-1 shows that a recuperator effectiveness of slightly more than 0. 70 results in the minimum total weight; for a longer mission (more work done) the minimum would shift to higher effectiveness. For the composite bus mission, however, because higher recuperator effectiveness yields better off-design performance, the minimum system weight is achieved with a higher effectiveness (above 0. 8). The optimum choice of the engine design point depends on the power profile of the application. The mission power profile and engine off-design performance are obviously also factors which must be considered when comparing the performance of different engine types. One situation which could arise when comparing closed- (or semiclosed-) and open-cycle Brayton engines is indicated by the sketch in figure 2-2. A closed- (or semiclosed-) cycle Brayton might be represented by curve A and an open-cycle Brayton by curve B. Engine A has a higher SFC at full power than engine B but it also has a flatter variation in SFC with power so that at low power the performance of engine A is better then engine B. A mission which includes a lot of operation at low powers could re-

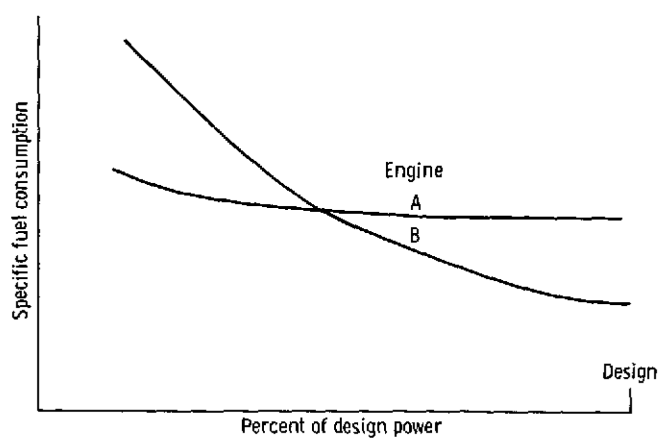


Figure 2-2. - Engine off-design performance.

sult in engine A requiring less fuel than engine B. In the case of the closed (or semiclosed) and open Brayton engines the open-cycle engine is lighter; however, depending on the mission and the relative weight of fuel and engine, the total weight of either system could be lighter. And for each type of engine a range of designs are possible, the trade-offs between which are mission dependent as was shown by figure 2-6. Therefore, in order to make a comparison on an equal basis the semiclosed- and open-cycle engines are compared here by using the curves of total system weight as a function of the total fuel load for each type of engine for a variety of missions. The mission power profiles used are those described in appendix F. The comparison between groups III and V for these missions is given in figures 2-3 to 2-6.

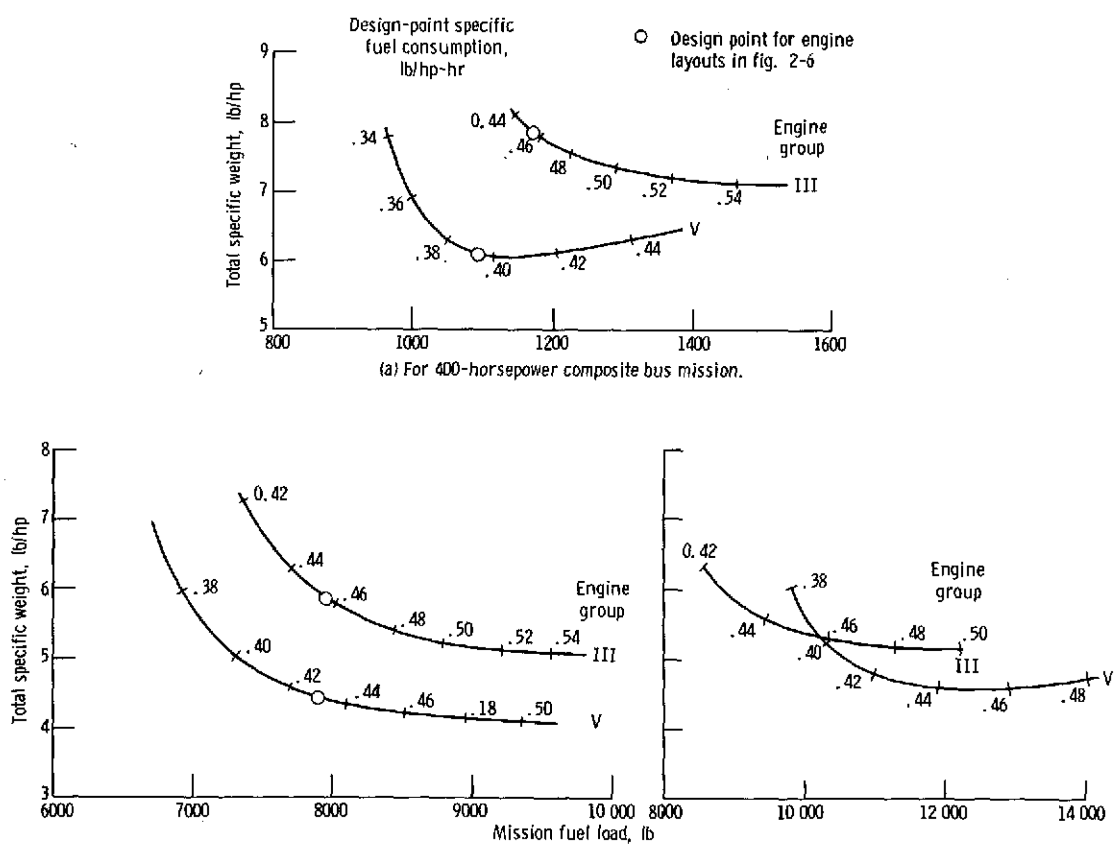


Figure 2-3. - Total system weights (includes engine, transmission, fuel, and tanks).

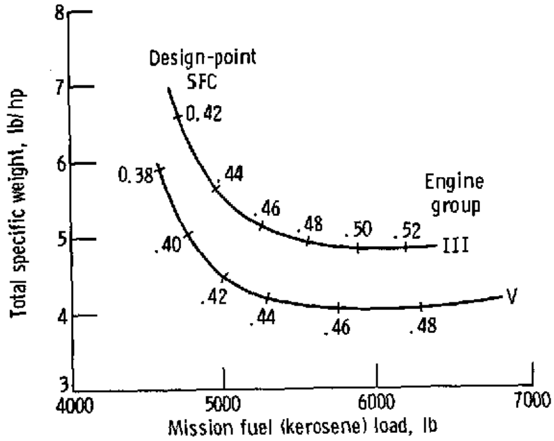


Figure 2-4. - Total system weights for 5000-horsepower two-shaft engine. Vehicle drive for TACV mission from appendix F. Total weight includes engine, alternator, transmission, fuel, and tanks.

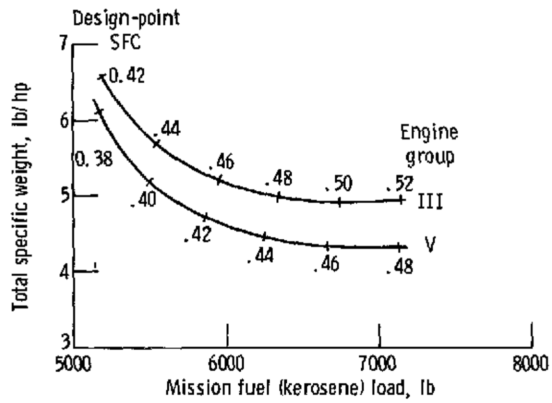


Figure 2-5. - Total system weights for 5000-horsepower two-shaft engine. Urban TACV mission from appendix F. Total weight includes engine, alternator, transmission, fuel, and tanks.

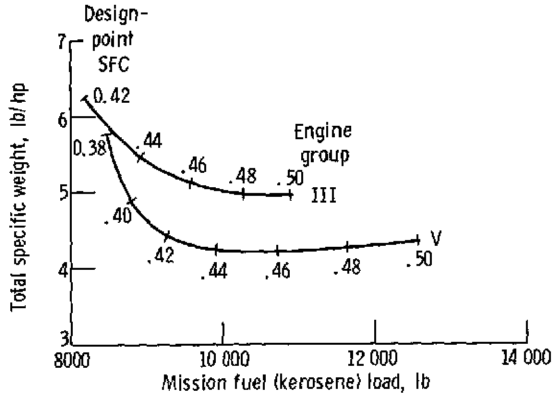


Figure 2-6. - Total system weights for 5000-horsepower locomotive. Mission (without idle) from appendix F. Total weight includes engine, transmission, fuel, and tanks.

Except for the locomotive application considered in figures 2-3(c) and 2-6 the total system weight and/or mission fuel load for the group V engines are less than for the group III engines for the range of designs considered. When comparing figures 2-3(c) and 2-6 it is seen that idle is more of a penalty on the group V engine than on the group III engine; for this case engine shutdown rather than idling would be a consideration for the group V engine. As explained in the OFF-DESIGN PERFORMANCE section the single-shaft group V engine can be operated at off-design power levels while holding the turbine inlet temperature at the design-point value by using an infinitely variable transmission to allow engine speed to be reduced independent of vehicle speed. This results in a much improved off-design performance over what can be obtained when turbine inlet temperature is reduced with power reduction. The improvement, in fact, results in an SFC variation with power which is almost as flat as that achieved with the closed- or semiclosed-cycle engines. As a result the trade-offs and comparisons shown in figures 2-1 and 2-3(a) and (b) do not show as much dependence on mission as would be expected if the turbine inlet temperature of the group V engine had been reduced at off-design power levels. Since using the infinitely variable transmission is assumed for the group III engine (see OFF-DESIGN PERFORMANCE section), its use with the group V engine also provides comparability.

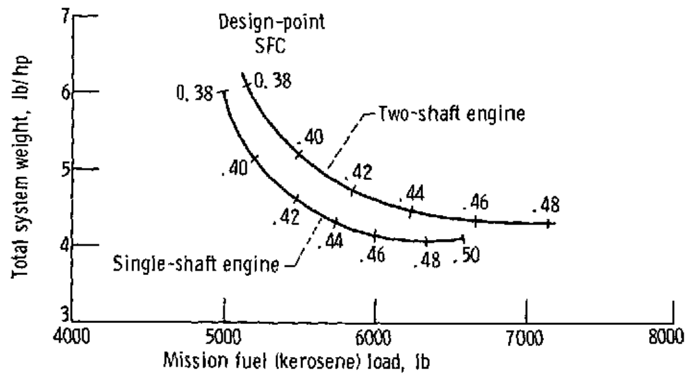


Figure 2-7. - Total system weights for 5000-horsepower single-shaft and two-shaft group V engines. Urban TACV mission from appendix F. Total weight includes engine, alternator, transmission, fuel, and tanks.

The group V engine considered in figures 2-3(c) and 2-4 to 2-6 is the two-shaft engine which does require reduction in turbine inlet temperature with power reduction (see OFF-DESIGN PERFORMANCE section). In these cases the group III engines compare a little more favorably. To show the effect of the off-design performance on the group V engine tradeoff, both the double-shaft and single-shaft engines are considered in figure 2-7 for the urban TACV mission. As the design-point SFC is increased the difference in mission fuel load required for the two engines is increased.

Exhaust Emissions and Noise

The combustors used by the group III and group V engines studied are the same type and both operate at the maximum cycle pressure. The major differences which affect emissions are the pressure level and temperature of the input primary air for equal turbine inlet temperatures. The effect is primarily on NO_x emissions.

The group III engine tends to optimize at a high combustor pressure (25 atmospheres for the train application and 12 atmospheres for the bus application). The group V engine optimizes at a lower combustor pressure level with a variation from about 3.5 atmospheres at a high recuperator effectiveness to about 7 atmospheres at a low recuperator effectiveness (see section 7).

The temperature of the primary air input to the group III engine combustor is about 1100° F for the engines for the train application and 800° F for the bus application. For each of these applications, this temperature is relatively constant over the range of optimized designs considered in section 4. In the case of the group V engine the temperature of the combustor primary air varies considerably over the range of optimized engine designs considered in section 4. At high recuperator effectiveness (about 0.95) this temperature is about 1150° F for engines at the power level for the train application and about 1200° F for the bus engine. At lower recuperator effectiveness the primary air temperature is 750° to 800° F.

If one considers both the pressure level and the temperature of the primary air and the fact that the results shown in the previous section indicate that the group V engine gives minimum total weight with recuperator effectiveness below 0.9, the information discussed in section 12 leads to the conclusion that the group V engine would yield lower NO_x emissions than the group III engine. In the case of hydrocarbon and carbon monoxide groups III and V are comparable.

Either group III or group V could use a surface or catalytic combustor, and emission levels would be substantially reduced and would in that case be comparable. These cases were not analyzed in this study. Since the surface combustor requires premixing of the fuel and primary air at near stoichiometric ratio, both engine designs might have to be restrained to limit the primary air temperature to avoid autoignition. In the case of the catalytic combustor operation is so lean that this is not a consideration. It is not expected, however, that these considerations would affect the conclusions about the relative performance of these two engine types.

With respect to noise there does not appear to be a strong difference between the two engine types. In the case of group III either the rejection of gas or the air supply compressor inlet dominates as a noise source, and in the case of group V the dominant source is the compressor inlet. In both cases the untreated engines would be unacceptable according to the guideline values (see appendix D). On the basis of the discussion in appendix D, however, it appears that both engine types can be quieted to acceptable levels with about the same level of acoustic treatment of the engine and the engine compartment. Noise does not, therefore, constitute a strong selection factor.

Technology Status and Growth Potential

The technology status of the semiclosed engine has been discussed in a previous section (p. 20). The open-cycle Brayton engine represents a well developed technology. The application of this type of engine is dominated by the use of the simple (unrecuperated) open cycle. However, the regenerated or recuperated engine has also seen substantial development and limited application. As previously stated, a fixed heat exchanger used to recover part of the waste heat of the turbine exhaust by transferring it to the compressor exit air is referred to here as a recuperator. A rotary heat exchanger used for the same purpose is referred to as a regenerator.

The growth of the open-cycle Brayton has closely followed the development of high temperature alloys and turbine cooling techniques. Advancements in these two areas have now permitted practical applications with turbine inlet temperatures at 2200° F and higher.

The automotive industry, both in this country and abroad, is trying to reduce the cost and undesirable emissions of the regenerated open-cycle Brayton engine. The EPA is also supporting an effort aimed at solving these problems and accelerating the large-scale introduction of the engine.

The major attributes of this engine which have spurred its development are light weight and low volume. The major impediments to widespread use in the heavy-duty ground transportation system have been cost and part-load SFC when compared to the diesel engine. As a result of the efforts just noted, costs have been coming down and recent developments in power transmission (i. e., the infinitely variable transmission) offer new potential for improved part-load fuel economy.

Using an infinitely variable transmission permits a single-shaft engine speed to be varied on a schedule independent of the output speed required for vehicle operation. The engine can operate at constant turbine inlet temperature, maintaining high efficiency over a wide range of power levels by reducing engine speed with decreasing power level. As pointed out in the OFF-DESIGN PERFORMANCE section (p. 102) the temperature of the combustion gas at recuperator inlet increases with decreasing power level under these conditions. Therefore, depending on the design turbine inlet temperature and the turbomachinery characteristics, the recuperator temperature capability might place a low power level limit on this mode of operation.

As discussed in previous sections (pp. 15 and 20), the ability to operate with constant turbine inlet temperature and hence high efficiency over a power level range is an advantage of closed and semiclosed cycles. The single-shaft group V engine with its use of an infinitely variable transmission also has this capability. In addition, the group V engine does not need a waste heat exchanger, and it is this component that dominates engine-vehicle integration problems (see appendix E) because of its large size. (The waste heat exchanger, since it is dominated by ambient air side heat-transfer considerations, cannot be substantially reduced in size by increasing the power system pressure level.) Therefore the group V engine offers comparable fuel economy at lower weight, size, and cost and with increased flexibility of application. Low weight and volume may permit new options for vehicle design and optimization.

ENGINE SELECTION OF CONCEPTUAL DESIGN PHASE

On the basis of the analysis presented and the previous comparisons the conclusions are summarized in table 2-2 where the criteria considered are listed and group III and V engines are comparatively rated. Group II, although not considered in this phase of the study, is included in this summary table for completeness.

With the exception of those applications which would be biased toward operation at very low fractions of design power (<20 percent) or idle for large fractions of their mission profile, the group V engine (for the configuration studied) is either competitive or superior on the basis of weight and fuel economy. In any event, the elimination of the waste heat exchanger makes the group V engine more attractive from the standpoint of integration flexibility.

The importance of engine weight and size on vehicle-engine integration is illustrated by considering the TACV. The TACV design was based on wayside power pickup. To integrate the group III waste heat exchanger would require modifications of the vehicle envelope. In addition, the group III engine and fuel (for mission profile considered here) would require a minimum of 60 percent of vehicle gross weight (as per study guidelines, appendix F)

TABLE 2-2. - ENGINE COMPARISON SUMMARY-

CONCEPTIONAL DESIGN PHASE

[X indicates engine with best performance. When X appears in more than one column, no discernible difference was observed. When ? appears, further study is needed to assess engine performance for the criterion.]

Criteria	Group II	Group III	Group V
Low specific fuel consumption and weight			X
Low volume			X
Good transient response			X
Growth potential		X	X
Good part power performance	X	X	X
Multifuel capability	?	X	X
Low emission potential	X	?	?
Flexibility			X
Noise	X		
Minimum technology issues			X

TABLE 2-3. - EMISSIONS POTENTIAL SUMMARY

Group	Combustor	Emission index, g/kg of fuel		
		HC	CO	NO_x
II	Surface	<1	<10	<1
III	Conventional	↓	<10	<10
	Catalytic	↓	<1	<.5
V	Conventional	↓	<10	<5

while the group V engine would require about 50 percent. These engines replace the power conditioning unit currently contemplated which accounts for about 25 percent of vehicle gross weight. However, these engines could provide mechanical shaft power for direct drive of the lift fans (a possibility not considered in the analysis). If these engines were to exploit this possibility, the total engine and fuel weight excluding the LIM would be about 50 and 37 percent of gross vehicle weight for groups III and V, respectively. It is reasonable to assume that if these vehicles were designed for onboard power the fraction of gross vehicle weight allocated to propulsion and life power would be further reduced.

The emissions potential of the closed-, semiclosed-, and open-cycle engines studied are summarized in table 2-3. The group II engine appears to have the lowest emissions; however, the surface-type combustor used in group II may also be applicable to the other engines. The catalytic combustor appears to ultimately offer the lowest emissions, and it could be used for both groups III and V. As noted both the surface and catalytic combustors, however, require development.

The technology issues for each of the Brayton engines are summarized as follows:

Group II:

- Heat-source heat exchanger limits maximum cycle temperature
- Inventory control limits power setting response
- Heat exchanger fabrication and life
- Requires surface combustor

Group III:

- Recuperator limits maximum cycle temperature
- Can use variety of combustors but -
  - Conventional Emissions are problem
  - Catalytic Life must be demonstrated
  - Surface Requires development
- Heat exchanger fabrication and life
- Heat exchanger fouling
- Turbocharger performance needs verification

Group V:

- Recuperator limits maximum cycle temperature
- Can use variety of combustors (same as group III)
- Low power already available to meet California 1975 standards
- Higher power needs development of recuperator and transmission

These have been previously discussed for groups II and III but are repeated here for comparison with group V. The items listed for group V reflect its advanced state of development and application. The major difference between the group V engine as studied here and current practice is the use of the infinitely variable transmission and requisite controls and the use of a fixed recuperator, particularly at higher powers.

Concluding Remarks

STUDY METHODS

In any study such as this there are inevitably considerations, options, and arrangements that consciously or unconsciously are omitted. This fact must be considered when conclusions are drawn. Some of these factors may have first-order effects which might bias selection unnecessarily or more importantly, unknowingly. This was recognized at the initiation of the study and to a large extent influenced the conduct of the study and the way in which data have been displayed. As a result, primary emphasis was given to valid comparisons and hence to consistency of treatment among the various engine groups rather than to the absolute values themselves. A more detailed treatment was given to the critical components which had a substantial effect on engine selection. For example, it becomes readily apparent from the preceding sections that heat exchangers dominate the size of these systems and have, in fact, influenced the conclusions. Hence, the heat exchanger designs were more detailed than say those of the gear box or ducts. The latter were not given as much detailed consideration since they were relatively small contributers to size, weight, and performance, but the same models were used in each of the engines. It was also recognized, however, that study results would be used not only for comparison purposes to select the "best" of the engine types considered for a given application, but also they would be used to assess the attractiveness of that engine for the application considered. Therefore, it was considered important that all the component models should be representative. An attempt has been made to present sufficient information to allow the reader to identify the contribution of each component and to make an independent judgment of the validity of these results.

It is felt that the comparisons made between the engine types considered are valid; a more detailed analysis would be required to provide a high confidence in the absolute values of performance. Many of the considerations or options which were omitted were mentioned in the previous sections of this report. Several will be discussed in the following paragraphs from the standpoint of their effect on the engine comparisons made in this section.

When the combustor - heat exchanger of the group II engine was studied it was not pressurized with a separate turbocharger. This reduced its size. However, even if this approach is adopted, it is not expected to influence the overall engine comparisons of this study.

The closed and semiclosed engines for best performance must be operated at constant turbine inlet temperature and at constant speed independent of load speed requirements. Therefore, an infinitely variable transmission was assumed to exploit this advantage in performance. It might be contended that the use of a multispeed automatic transmission should also be considered. However, it is felt that such a consideration would not change the conclusions of the study.

Both the group III and V engines could have included turbine liquid cooling and/or variable geometry. The comparison would not have changed, however, since both engines would display comparable improvements in performance and increase the disparity with group II.

The group III cycle as considered is but one example of the semiclosed type of Brayton cycle. Many other variations are possible, some of which were described in section 3. Many of these cycles which were not considered in detail share characteristics with those which were studied. The results of analyzing the cycle variations which were studied provide insight into the potential advantages and disadvantages of those variations which were not studied. Although it cannot be concluded that the group III cycle as studied is the best of the semiclosed cycles, it is, however, clear from the results that the conclusions concerning the comparison of the open and semiclosed cycles for the applications of interest here would not have been substantially changed by further analysis of other semiclosed-cycle variations.

Finally, several points are concerned with the fact that in all cases the engines were optimized with the design point at maximum power. In the case of group V, it would seem more desirable to have optimized the engine size at a lower power level and trade improved low end SFC against some penalty at full power for those applications where the mission profile is dominated by part power operation. Another related possibility not exploited is the ability of both groups III and V to operate for short periods of time (short with respect to total life) at higher than design turbine inlet temperature; for example, maximum power at turbine inlet temperature above 1900° F and sustained operation at lower power limited to 1700° F. One way to include these effects would be to include the engine-off design analysis and mission profile within the engine design optimization computer program. (As discussed in section 1 this was not done.) Although these options should be considered when developing these engines for a particular application, if they had been included in this study the conclusions would not have been qualitatively changed. Optimizing at a power level less than maximum would tend to make group V comparatively better, and considering using higher turbine inlet temperatures for short time peak power would benefit both groups III and V at the expense of group II.

In the mode of off-design operation considered for the single-shaft group V engine, the turbine inlet temperature is held constant and the engine speed is reduced with decreasing power. As discussed in the OFF-DESIGN PERFORMANCE section (p. 102) this results in an increase in the temperature of the combustion gases at the inlet to the recuperator as the power level is decreased. For a particular engine design this could result in a lower limit on the power level for this mode of operation because of the temperature limitations on the recuperator. In such a case a reduction in turbine inlet temperature (and hence in efficiency) to reach lower power levels would be one possibility. Alternatively, the design-point conditions could be compromised by increasing the design-point pressure ratio and hence reducing the design-point recuperator inlet temperature to keep this temperature within safe operating limits at lower power levels. A reduction in design-point performance would be traded off against an improvement in the lower power level performance. Again this should be considered when developing a specific engine design, but including it would not have changed the outcome of this study.

ENGINE CHARACTERISTICS AND APPLICATIONS

All the Brayton engines considered are thermodynamically similar. The differences are in the way the inventory is controlled (groups I and II against group III), in the way heat is added to the working gas (directly with the combustor in the gas loop or indirectly with a separate combustor loop), and in the way waste heat is rejected (direct exhaust to the atmosphere or by means of an air-cooled waste heat exchanger). Each of these characteristics has distinct advantages for different types of applications. And, therefore, for a variety of applications the most desirable type of Brayton engine could differ from one to the other.

For some applications the working fluid must be conserved, and for these applications the closed cycle must be used, there is no trade-off. Such a situation could arise from using a working fluid other than air in a space application and/or where the fluid may become contaminated and cannot be released (as, for example, when a Brayton is used with a gas-cooled nuclear reactor heat source).

When waste heat must be rejected directly to the atmosphere then the use of a waste heat exchanger (closed or semiclosed cycle) will present a penalty in size, weight, and parasitic fan power compared to an open cycle. Although the pressure level of a closed or semiclosed cycle can be raised above that of an open cycle and therefore the size of some of the components can be reduced, the waste heat exchanger is little affected since heat-transfer characteristics on the ambient air side is limiting. Furthermore, the waste heat exchanger is coupled to the recuperator in size and performance. The higher the amount of recuperation, the lower the load on the waste heat exchanger. Therefore, the closed and semiclosed cycles tend to optimize with high recuperator effectiveness (as shown in section 4). The open-cycle Brayton, on the other hand, can use lower recuperation without having a significant deleterious effect on another component. As a result, for a given performance level the open-cycle recuperator may be lighter than that of the closed cycle due to its possibly lower effectiveness even though the gas pressure in the open-cycle recuperator may be much lower than in the closed cycle. This is shown by the results in section 4 and is an example of an effect not likely to be clearly defined by qualitative insight alone.

For the applications considered in this study, power system size and weight are significant factors, and waste heat is rejected to ambient air. In light of the preceding discussion it is not surprising that group V appears to be the best of the types studied. However, if for some other application waste heat rejection to water were possible the conclusions might be changed. In such a case the waste heat exchanger size (water cooled) would be a stronger function of the heat transfer on the Brayton gas side (in contrast to an ambient air-cooled waste heat exchanger) and the waste heat exchanger could therefore be significantly reduced in size by increasing pressure level. In this situation the overall size and/or weight of a high pressure closed or semiclosed Brayton engine might be lower than that for an open Brayton cycle engine.

If a substantial fraction of the power system waste heat is to be used in some application, the previous inferences also are changed. In such a case, a waste heat exchanger becomes, by definition, a part of the system, and it may be advantageous to close the loop and raise the pressure level to allow reduced size and weight of the other components.

The effect of mission application is also an important factor in other ways. If rapid power transients are required and a closed-cycle inventory adjustment system with rapid response is not practical, the choice becomes one between the semiclosed and open cycles. Furthermore, the previous selection logic is only valid if the cycles are compared on an equal basis with respect to turbine inlet temperature. To compare an open cycle in which the turbine inlet temperature is reduced to reduce power output to a closed or semiclosed cycle in which the turbine inlet temperature might be held constant while pressure level is reduced to reduce power requires more than a qualitative assessment. In such a case quantitative analysis would be required in order to compare the fuel economy of two such engines over a mission profile in which power is significantly varied. The application of the engine therefore not only influences the choice of engine type but affects the amount of analysis necessary to make a choice between engine types.

3. THERMODYNAMIC CYCLE DESCRIPTIONS AND SELECTION

As indicated earlier, this section presents a detailed description of the closed, semiclosed, and open Brayton cycles considered in this study as well as a discussion of the preliminary cycle screening comparisons.

The purpose of the preliminary thermodynamic cycle screening, which was part of the engine screening phase, was to select a limited number of engines from each of the three types for further analysis. This cycle screening involved both quantitative comparisons based on cycle efficiency and flow rates and qualitative comparisons based on such factors as complexity, estimated relative size, and expected off-design power-level performance. The quantitative comparisons were based on preliminary simplified cycle calculations. The results of these calculations are given in appendix I.

In the next section the closed, semiclosed, and open Brayton cycles with some of their variations are described. Included with the descriptions is a discussion of the cycle screening comparisons.

The cycles which were selected for a more detailed analysis in the engine screening phase (section 4) fit conveniently into five general thermodynamic groups. These group assignments and designations are explained later in the Group Designations for Selected Cycles section (p. 60). Throughout this report the cycles and engines studies are referred to by these group designations.

Cycle Descriptions

CLOSED BRAYTON CYCLES

The thermal input to the closed Brayton system comes from a separate

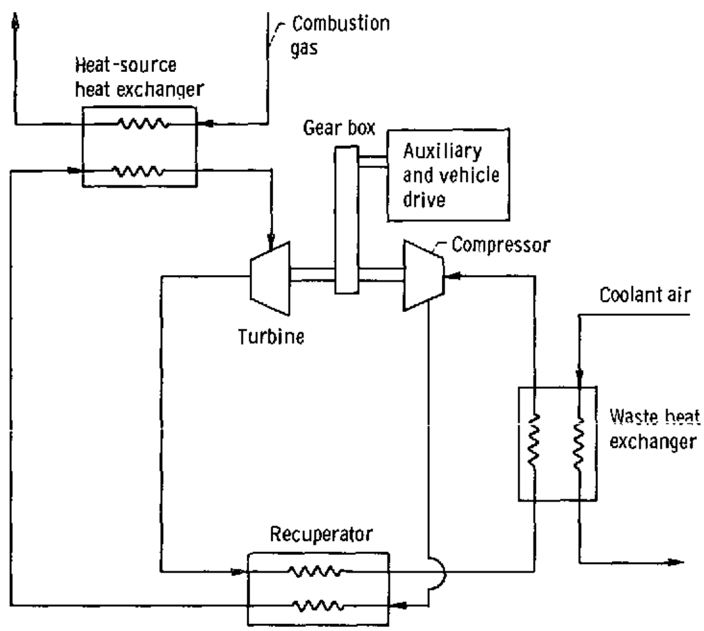


Figure 3-1. - Schematic of basic closed Brayton cycle.

combustion loop. The Brayton working fluid is heated by the combustion gas as it flows through a heat-source heat exchanger. The Brayton working fluid pressure and composition are independent of the combustion gas. The closed Brayton cycle is shown in figure 3-1. After being heated in the heat-source heat exchanger, the Brayton cycle fluid is expanded through a turbine to produce power to drive the compressor and auxiliaries and for vehicle propulsion. The fluid then flows through the recuperator, where part of its thermal energy is transferred to the compressor exit flow, and then through the waste heat exchanger where heat is rejected to the coolant air. After being compressed, the gas flows through the high pressure side of the recuperator and then back to the heat-source heat exchanger.

Several variations in this basic cycle were considered in the study. Since the cycle is closed, the working fluid can be independently chosen. Air and monatomic inert gases were considered. Both the single-shaft arrangement (fig. 3-1) and a two-shaft arrangement, in which on turbine drives the compressor and provides the auxiliary power required and a second turbine

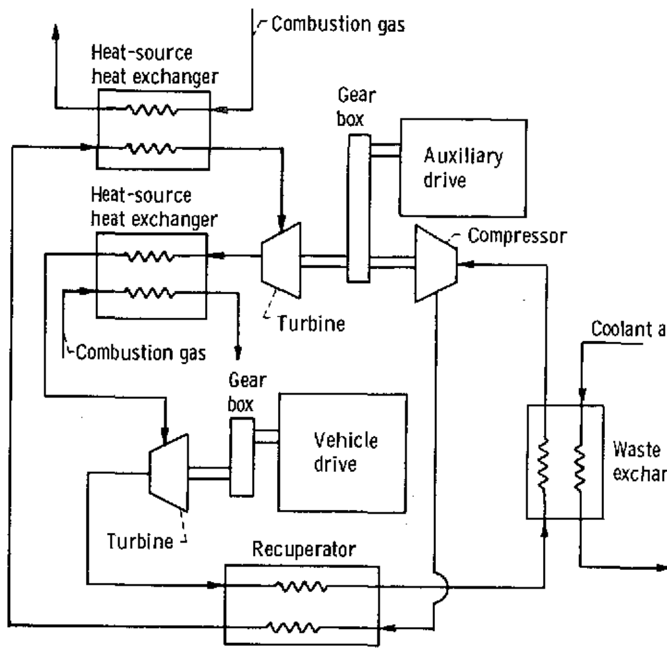


Figure 3-2. - Schematic of closed Brayton cycle with reheat.

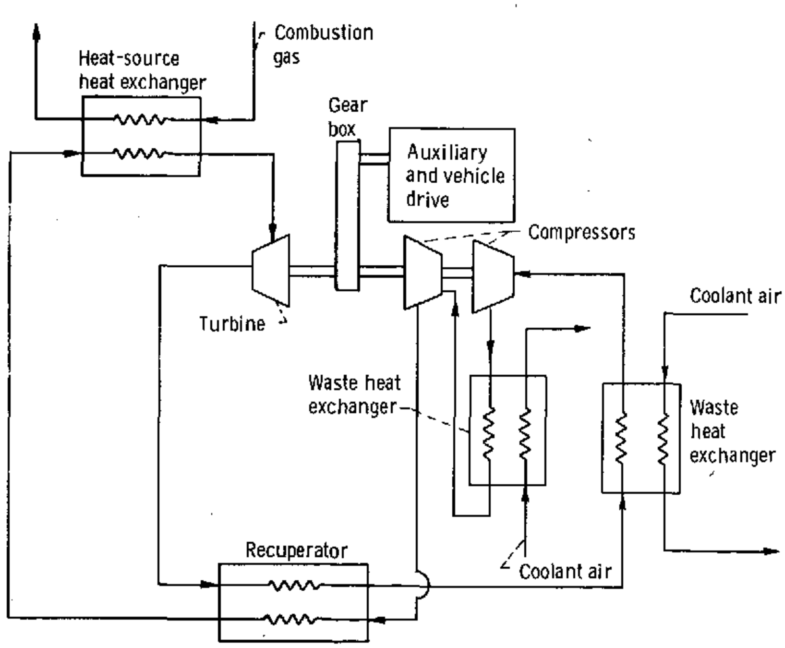


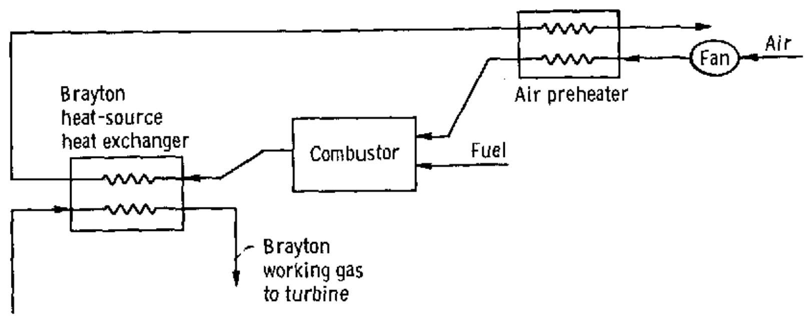
Figure 3-3. - Schematic of closed Brayton cycle with intercooling.

provides the vehicle propulsion power, were considered. For the two-shaft system the option of reheating the working gas in a second heat-source heat exchanger between the turbines as shown in figure 3-2 was included. Finally, the option of intercooling or cooling the working gas between two stages of compression as shown in figure 3-3 was included.

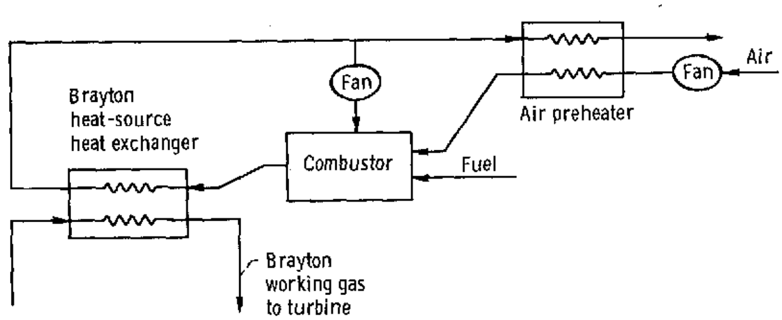
Reheat and intercooling increase the efficiency of the closed cycle by narrowing the temperature range over which heat is added to or rejected from the cycle. However, each cycle requires the addition of a heat exchanger. The effect of the added weight and pressure loss on the total system performance cannot of course be fully evaluated until the engine design-point analysis phase. The effect of reheat on the combustion loop is discussed later in this section.

The three types of combustion loops considered are shown schematically in figure 3-4. In the combustion loop of figure 3-4 gas from the combustor flows through the heat-source heat exchanger transferring part of its thermal energy to the Brayton working gas. The enthalpy of the combustion gas at the exit of the heat-source heat exchanger is appreciable; the energy removed from the combustion gas flow in the heat-source heat exchanger is limited by the effectiveness of the heat exchanger and by the temperature of the incoming Brayton gas. An air preheater is therefore used to conserve part of this energy; the exhaust gas flow is cooled before rejection to the atmosphere by transferring energy to the incoming air. A conventional combustor is used. Both the combustor primary and secondary air are preheated. The overall air-fuel ratio is determined by the required combustor exit gas temperature. This temperature is, in turn, determined by the specified Brayton turbine inlet temperature and the specified temperature difference between the combustor exit gas and the turbine inlet.

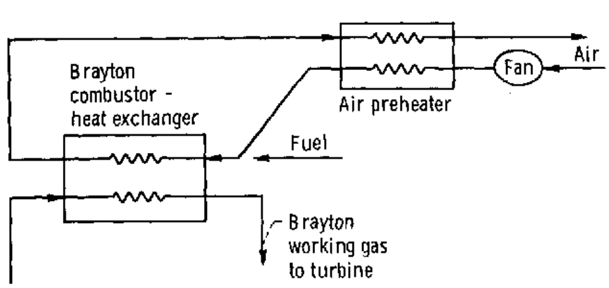
As the effectiveness of the preheater is increased, the temperature of the exhaust gas rejected to the atmosphere approaches the ambient air temperature, the wasted thermal energy is reduced, and the combustion loop efficiency and consequently the overall efficiency are increased. As shown by the cycle results in appendix I, the efficiency of this combustion loop is quite sensitive to the preheater effectiveness. Unfortunately, as the preheater effectiveness increases, and the temperature of the combustor diluent (secondary air) increases, the diluent flow rate required to maintain a given combustor exit



(a) Excess air as combustor diluent.



(b) Recirculated combustion products as combustor diluent.



(c) Combined combustor and heat-source heat exchanger.

Figure 3-4. - Combustion loop schematics for closed Brayton.

temperature increases. An increase in preheater effectiveness with the intention of improving efficiency would then result in an increase in preheater size not only because of increased effectiveness but also because of the increased flow rate through it. Also, as the flow rate is increased the fan power requirement for the loop also increases, tending to reduce the net efficiency, which indicates that a trade-off between exhaust gas thermal recovery and fan power requirements would be necessary.

An improvement of this combustion loop configuration, in that the performance is less sensitive to the air preheater size and performance, is shown in figure 3-4(b). In this case, part of the exhaust gas is recirculated to the combustor to serve as the diluent, and the air input through the preheater is only the amount required to ensure complete combustion of the fuel. The recirculated flow rate, however, is comparable to the secondary air flow rate in the previously discussed combustion loop with high preheater effectiveness. This high flow requirement and the fact that the temperature of the flow through the exhaust recirculation fan is much higher than through the air supply fan are disadvantages of this combustion loop arrangement.

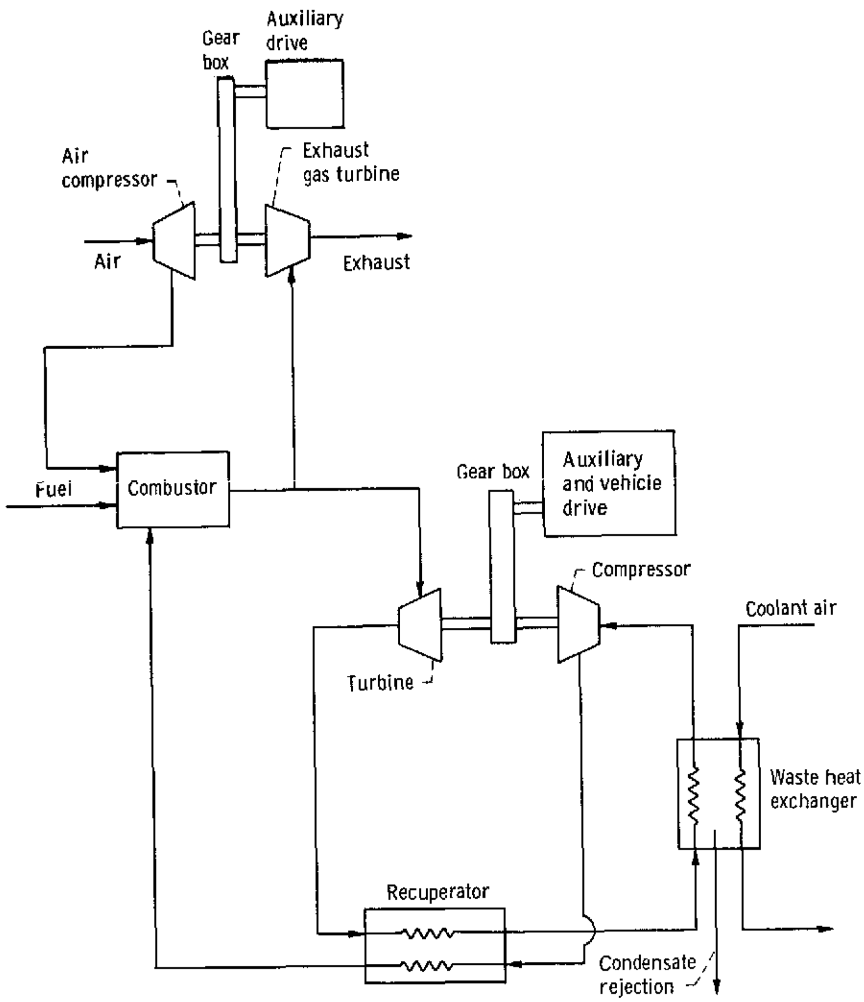
The schematics in figures 3-4(a) and (b) both show a conventional combustor with a gaseous diluent for temperature control. In figure 3-4(c) the combustor and heat source heat exchanger are combined into one unit so that heat transfer from the combustion zone to the Brayton working gas in the heat exchanger can be used to control combustion temperature. Potentially, with effective design, a gaseous diluent would not be required and the flow rate in this loop would equal the fuel flow rate plus slightly more than stoichiometric air (i.e., no diluent air required). This combustion loop (fig. 3-4(c)) is potentially the most efficient of the three for it has the lowest flow rates and smallest fan powers.

If the closed Brayton cycle includes reheat the combustion loops in figure 3-4 must be modified to include two heat-source heat exchangers. As shown in appendix I the combustion loop for the reheated Brayton cycle is less efficient than that for the basic cycle. The addition of reheat increases the efficiency of the closed Brayton by narrowing the temperature range over which heat is added to the cycle. This, however, also increases the temperature of the combustion gas leaving the heat source heat exchanger. With the same preheater effectiveness the exhaust temperature (and therefore the thermal loss) of the combustion loop for the reheat cycle will be higher than that for the cycle without reheat. For the combustion loop of figure 3-4(a) it is shown in appendix I that even though reheat increases the closed Brayton cycle efficiency the reduction in combustion loop efficiency can be sufficient to result in a reduction in overall efficiency. Therefore, the use of reheat with the combustion loop configuration was not considered further in the study. The reduction in combustion loop efficiency with the addition of reheat for the other two combustion loops in figure 3-4 is not as pronounced. But because of the added complexity, the use of reheat with any of these cases was not further considered.

SEMICLOSED BRAYTON CYCLES

In the semiclosed cycles which were selected for analysis in section 4 the combustor is included within the Brayton cycle; the combustion gas is used as the Brayton working fluid and is recirculated to the combustor. One of the many possible variations of such cycles is shown in figure 3-5(a). This cycle is similar to that proposed in reference 1. As shown, the flow path of the recirculated working gas is similar to that of the closed-cycle working fluid shown in figure 3-1. However, rather than being heated in a heat source heat exchanger, the gases in this case are heated directly by combustion of the fuel. The cycle is partly open in that the fuel and combustion air are supplied to the combustor and an equal mass flow of combustion products must be rejected from some point in the system in order to maintain constant inventory. Part of this mass rejection takes place in the waste heat exchanger where water formed by the combustion of fuel is condensed from the recirculated combustion products as they are cooled prior to compression. The remainder of the rejected flow can be expanded through an exhaust turbine to produce useful power.

As discussed in appendix I the semiclosed cycle can be analyzed as a combination of an open and a closed Brayton cycle. The gas which is recirculated is the working fluid of the closed portion, and the fuel and air supplied to the combustor and the combustion products not recirculated are the working fluid of the open portion. The open portion determines the system pressure

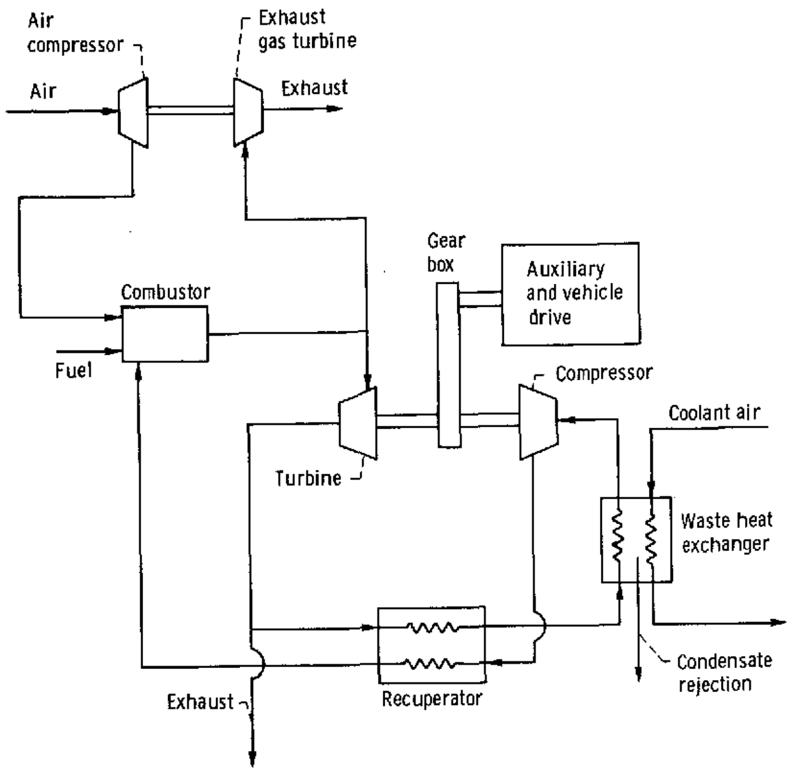


(a) Example with exhaust at single point.

Figure 3-5. - Schematic of semiclosed Brayton cycles with combustion air input at cycle high pressure level

level and provides the thermal input to the closed portion by heating the recirculated gas to the turbine inlet temperature. The closed portion provides the combustor diluent for the open portion; the air supplied to the combustor is near stoichiometric with only enough excess to ensure good combustion efficiency.

The results in appendix I show that thermally the cycle is more than

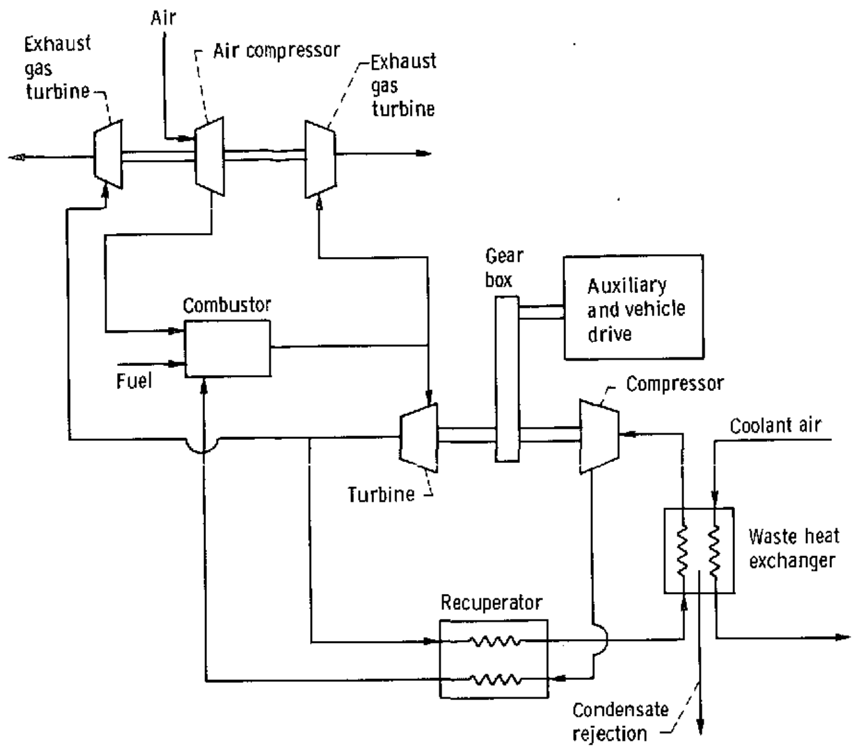


(b) Example with exhaust at multiple points.

Figure 3-5. - Continued.

three-quarters closed (i. e., more than three-quarters of the fuel heat release is transferred to the working fluid of the closed portion) over a wide range of pressure levels. Potentially, then, such a cycle could have the desirable closed Brayton characteristics of compact heat exchangers and good efficiency at off-design power levels. But since the open-cycle portion (with appropriate controls) determines system pressure level, the semiclosed cycle does not need the separate inventory control system required by a closed Brayton to achieve its best off-design performance characteristics. Also, this semiclosed cycle has the advantage that its turbine inlet temperature is not limited by the temperature capabilities of the heat source heat exchanger as is the case for closed Brayton cycles.

Most of the cycle variations considered for the closed Brayton cycle are



(c) Example with exhaust at multiple points using two exhaust gas turbines.

Figure 3-5. - Concluded.

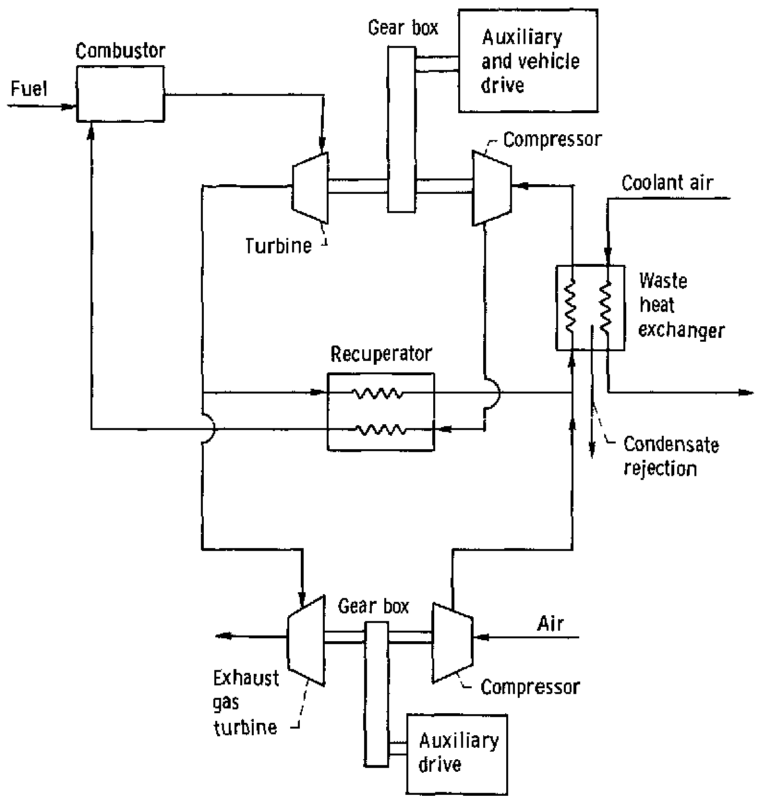
also possible for the closed portion of the semiclosed cycle. Single-shaft and two-shaft arrangements and compressor intercooling were considered. Reheat was not considered because of the apparent complexity.

Many variations in the open portion of the semiclosed cycle or in the way the open portion is integrated with the closed portion are possible; that is, many variations in the point of exhaust gas rejection and air input to the cycle are possible. For most conditions of interest the power produced in the exhaust turbine for the particular cycle arrangement in figure 3-5(a) would exceed the power required by the air compressor. If this power cannot be effectively utilized, other points of rejection of the exhaust gas (open-cycle gas) can be considered. For example, rather than exhausting gas at combustor exit as shown in figure 3-5(a), the exhaust gas can first be expanded through the main turbine (or through one or both of the main turbines in the two-shaft arrangement) to produce useful power before being separated from the recirculated gas and expanded through the exhaust turbine. Or the gas could also be passed through the recuperator before being rejected. Obviously, as the inlet temperature and pressure of the exhaust turbine are reduced the power produced is reduced until a point is reached where it is less than that required to power the air compressor and an additional source has to be provided.

If more flexibility to match the exhaust turbine and air compressor powers is desired, two or more exhaust points could be considered. An example is shown in figure 3-5(b). The arrangement has been slightly changed from figure 3-5(a) such that the flow through the exhaust turbine is only the amount sufficient to produce a power equal to the air compressor requirement. The rest of the exhaust gas is expanded through the main turbine to produce useful power and then rejected from the system. Rather than waste the available energy of the gas rejected at the second point, the exhaust gas could also be expanded through an additional turbine. The sum of the powers produced by both exhaust turbines could then be matched to the requirement of the air compressor by controlling the relative flow rates in the two exhaust turbines. This is indicated in figure 3-5(c). The excess power produced by the single exhaust turbine in figure 3-5(a) has in effect been transferred to the main turbine by the arrangement in figure 3-5(c).

Although many variations are possible, only a few were considered in this study, and most of the effort discussed later in section 4 considered the combination of rejection points represented by figure 3-5(b). Up to this point the variations in the open portion of the cycle discussed have been concerned only with the point of exhaust gas rejection. The semiclosed-cycle configuration can also be changed by changing the point where the air is supplied. Although these variations were not considered in this study, they could be of interest and some possibilities are described briefly.

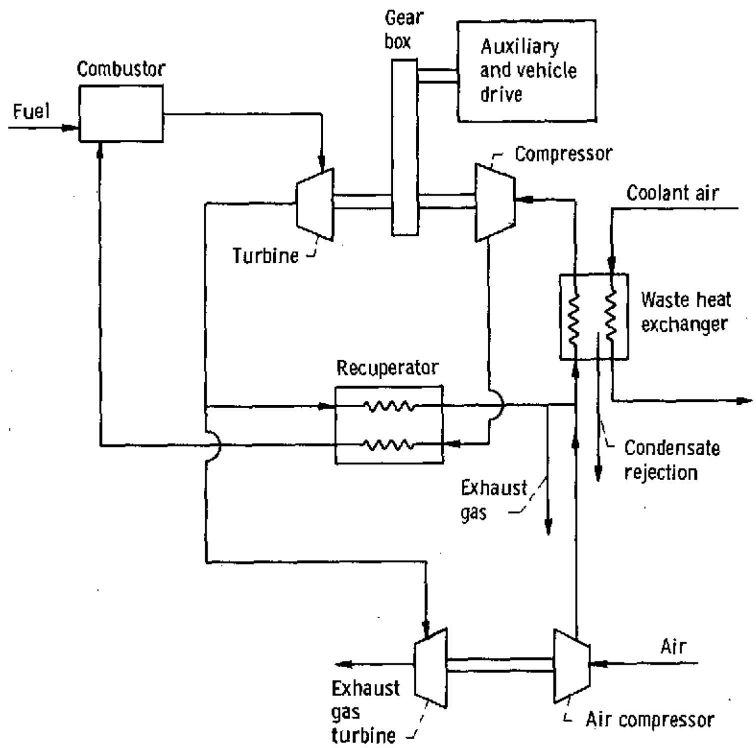
In the cycles considered in figure 3-5 the supplied air must be compressed to the combustor pressure, the peak pressure in the cycle. If the air could be supplied to the low pressure side of the cycle as shown in figure 3-6, the pressure ratio of the air compressor and the power involved could be considerably reduced. The gas reaching the combustor in figure 3-6 would be a mixture of air and combustion products, and the mixture of it and the fuel



(a) Example with exhaust at single point.

Figure 3-6. - Schematic of a semiclosed cycle with combustion air input at cycle low pressure level.

would not likely be flammable. Such a cycle would require a catalytic combustor. Or conversely, if the use of a catalytic combustor were to be considered from the standpoint of reducing emissions, the cycles in figure 3-6 would be of interest as an improvement over those of figure 3-5. As in the case of figure 3-5, the cycle in figure 3-6 can be varied in many ways by changing the point at which the exhaust gas is separated from the recirculated gas. In the case of figure 3-6(a) the exhaust gas is taken at the exit of the main turbine, and, as indicated, for most conditions of interest the power produced in the exhaust turbine would exceed that required by the air compressor. One variation which could be used is shown in figure 3-6(b) where



(b) Example with exhaust at multiple points.

Figure 3-6. - Concluded.

the exhaust turbine gas flow is enough to match the turbine and air compressor powers and the rest of the exhaust gas is rejected after flowing through the recuperator.

Another modification of the semiclosed cycle would allow introducing the air supply to the low pressure side of the cycle without the requirement of a catalytic combustor as was the case in figure 3-6. This is shown in figure 3-7. Instead of using the combustion products as the working fluid for the closed portion of the cycle, air is used. A combustor - heat exchanger similar to that of the combustion loop shown in figure 3-4(c) (p. 47) is used as a heat-source heat exchanger for the closed portion of the cycle. The combustion air is bled from the main gas loop at exit of the main turbine and the combustion products at exit of the combustor heat exchanger are expanded through an exhaust turbine before being rejected to the atmosphere. An air

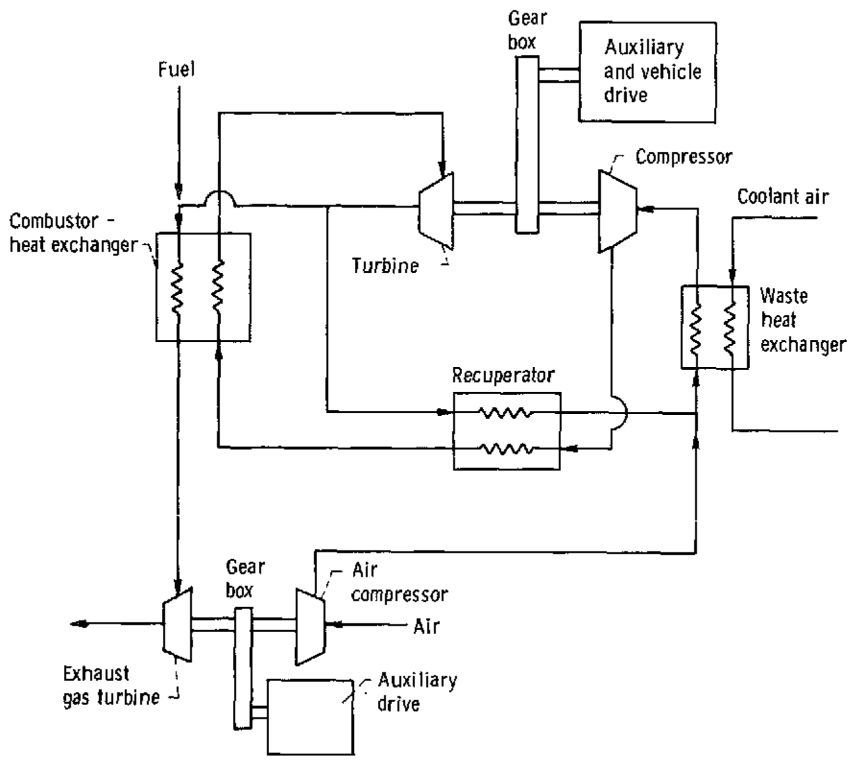


Figure 3-7. - Schematic of semiclosed cycle with air working fluid.

compressor supplies air to the main loop at the waste heat exchanger inlet to replace the inventory bled off for the combustion process. The exhaust turbine powers the air compressor.

If an integrated combustor - heat exchanger (heat transfer is used to control the combustion temperature) is used, the combustion air flow rate required could be near the stoichiometric rate. Hence, the flow rate in the air supply compressor would be near the stoichiometric rate. This would result in the cycle being (as was that in fig. 3-5) largely closed from the thermodynamic standpoint. Except for the attempt to keep the combustion air near the stoichiometric rate, the cycle in figure 3-7 is essentially the same as that proposed in reference 2.

This semiclosed cycle has much more in common with the closed cycles considered in the study than do the other semiclosed cycles discussed. It is seen that the flow path of the closed portion of the cycle (the recirculated air) is the same as that in figure 3-1. The combustor heat exchanger differs from that in figure 3-4(c) in that the pressure level on the combustion side would be higher in figure 3-7. Like the closed cycle, it is limited in turbine inlet temperature by the temperature capability of the heat-source heat exchanger. The cycle in figure 3-7 like the other semiclosed cycles includes the means for inventory control of the closed portion to change output power. However, it has the advantages over the semiclosed cycles of figure 3-5 that it does not use combustion gases as the working fluid in the main loop, it does not require condensate removal in the waste heat exchanger, and the pressure ratio and power required for the air supply compressor would be much smaller. The combustor - heat exchanger unit of figure 3-7 should, however, be larger than the combustors in figure 3-5.

The semiclosed cycles of the type shown in figures 3-6 and 3-7 were not analyzed in this study but were qualitatively discussed in section 2.

Another variation of the semiclosed cycle in figure 3-5 is that in which hydrogen and oxygen, in stoichiometric proportions, are used as combustor reactants. Recirculated water is used as the combustor diluent so that the turbine inlet working fluid is pure, superheated steam. The waste heat exchanger becomes an air-cooled condenser, and the compressor is replaced by a pump. The portion of the working fluid which is not recirculated to the combustor can be rejected at condenser inlet, if the pressure is above atmospheric, or at exit of the pump.

The temperature which can be achieved at the turbine inlet with this cycle is much higher than that of conventional Rankine cycles because of the temperature limits of conventional boilers. The expansion through the turbine, because of the high superheat for cases of interest here, occurs entirely within the dry region. A recuperator is therefore used to recover part of the energy of the superheated steam at the turbine exit by transferring it to the water recirculated to the combustor. In general, the steam at recuperator exit is still superheated so that waste heat rejection does not take place at constant temperature. The heat which must be rejected to desuperheat the steam might be used to preheat the hydrogen and oxygen before they are introduced into the combustor. This was not, however, considered and would not substantially affect the conclusions of the analysis.

If just the thermodynamic cycles are considered, this hydrogen-oxygen cycle is inherently more efficient than the Brayton cycles considered for the same source and sink temperatures in that for the hydrogen-oxygen cycle the waste heat rejection takes place at a much more uniform temperature. However, because the engine is air cooled and the fan power required decreases the net cycle efficiency, this apparent advantage cannot be fully realized in terms of net efficiency. If the condensing temperature was the same as the compressor inlet temperature of the Brayton cycles the coolant air flow rate per unit of heat rejected would be much higher for the hydrogen-oxygen cycle because of the narrower range of heat rejection temperature and therefore the lower potential temperature rise of the coolant air. An increase in condensing temperature (pressure) would decrease the gross cycle efficiency but it would also decrease the coolant air flow (and hence fan power) per unit of heat rejected. The result of this is that the peak net cycle efficiency for the hydrogen-oxygen cycle occurs at a condensing temperature which is in general higher than the compressor inlet temperature of the Brayton cycles.

OPEN BRAYTON CYCLES

The basic cycle schematic for the recuperated version of the open cycles analyzed is shown in figure 3-8. The input air is compressed, heated in the recuperator, and then supplied to the combustor. The amount of combustor excess air, or secondary air, is the amount required to maintain the specified turbine inlet temperature. After expanding through the turbine the combustion gas passes through the recuperator to recover part of its thermal energy and is then exhausted. The recuperator considered in this study was a fixed boundary heat exchanger rather than a rotating regenerator commonly used for open-cycle Brayton engines. The cycle is affected by this assumption in that seal leakage for rotating regenerators could have a significant effect on cycle performance.

Both the single-shaft arrangement shown and a two-shaft arrangement with a first expansion turbine driving the compressor and providing auxiliary power and with a second expansion turbine providing vehicle drive were con-

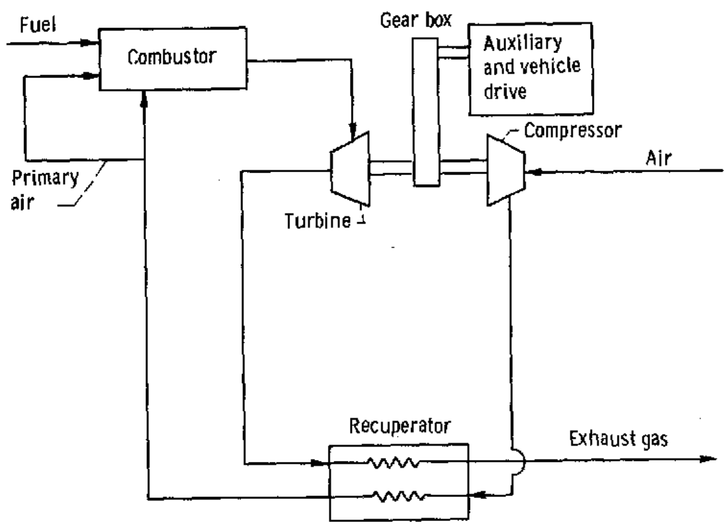


Figure 3-8. - Schematic of basic open Brayton cycle.

sidered. Reheat between the two turbines was not considered, which is consistent with the analysis of the semiclosed cycles.

Another cycle variation considered was that of recuperator bypass of the combustor primary air. One way in which NO_x emissions can be reduced is to reduce the temperature of the primary air. This could be done by bypassing the primary air around the recuperator so that it enters the combustor at compressor exit temperature. For the sake of thermal efficiency, especially at off-design power levels, the recuperator would still be used for the secondary air. The penalty of this bypass on the cycle efficiency will be shown.

Because the open-cycle engine does not have a waste heat exchanger, heat-source heat exchanger, or air preheater, it would be expected to have a weight advantage over closed- and semiclosed-cycle engines. Also, because it does not have a waste heat exchanger, its thermodynamic performance is unaffected by the penalties associated with the waste heat exchanger, that is, temperature difference between ambient air and compressor inlet, pressure losses, and air coolant fan power. Like the semiclosed-cycle studies, the open cycle is not limited in its turbine inlet temperature by the temperature capabilities of the heat-source heat exchanger as is the closed cycle.

These advantages have to be compared with the major disadvantage of the open cycle; that is, the decrease in efficiency at off-design power levels is expected to be greater than that of the closed and semiclosed cycles. The final comparison must therefore be made for specific missions so that the mission fuel expenditure and total weight of engine and fuel can be compared for open- and closed- or semiclosed-cycle engines. This comparison was discussed in section 5.

Group Designations for Selected Cycles

For convenience of identification the cycles which were studied were divided into five groups. Group I includes the closed Brayton cycles with combustion loops using conventional combustors and gaseous diluents (i. e., those shown in figs. 3-4(a) and (b)). The cycles which use excess air in the combustion loop are referred to as group Ia, and those which use recirculated combustion gas are referred to as group Ib. Group II cycles are also closed Brayton but with the combustion loop of figure 3-4(c) which uses a combustor heat exchanger unit with heat-transfer control of combustion temperatures. The semiclosed Brayton cycles studied are designated as group III. A special case of the semiclosed cycle is that which uses hydrogen fuel and pure oxygen as the oxidizer. In this case the working fluid is condensed and recirculated to the combustor as liquid water. The waste heat exchanger and compressor of the Brayton cycle are replaced by a condenser and pump in this cycle. Because this hydrogen-oxygen cycle differs so much from the other semiclosed cycles it was treated separately and designated as group IV. Finally, the open-cycle Brayton was designed as group V. For reference, the cycles studied are shown according to groups in figures 3-9 to 3-13.

References

1. New, W. R.: Gas Turbine Power Plant and Method. U. S. Patent 2,303,381, United States, Dec. 1942.

2. Traupel, W.: Gas Turbine Plant. U. S. Patent 2,268,270, United States, Dec. 1941.

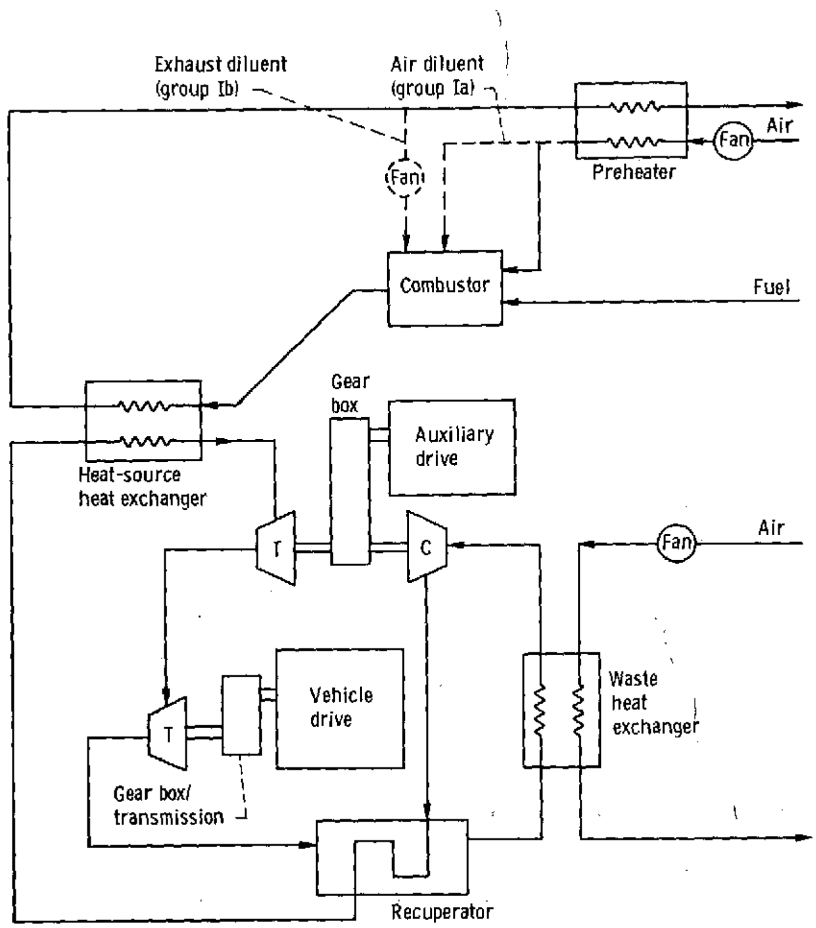


Figure 3-9. - Group I schematic. Description: closed Brayton; atmospheric pressure combustion; inert gas or air working fluid.

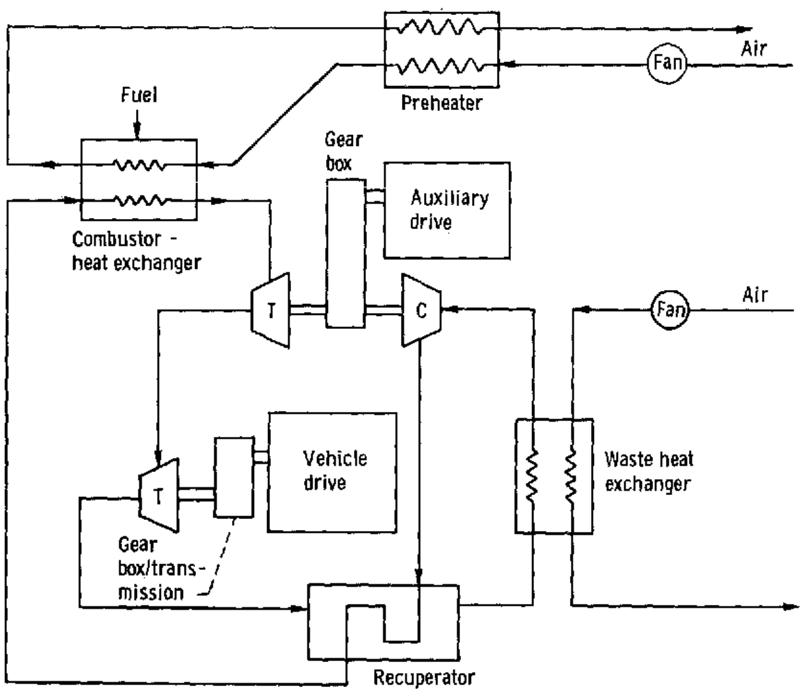


Figure 3-10. - Group II schematic. Description: closed Brayton; heat-transfer combustion temperature control stoichiometric; atmospheric pressure combustion; inert gas or air working fluid.

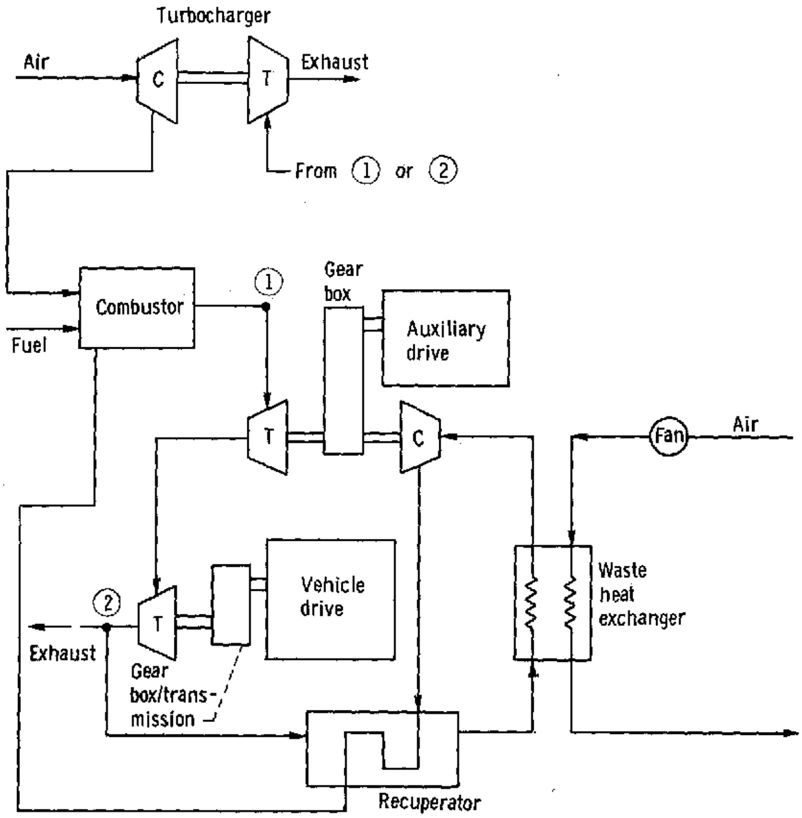


Figure 3-11. - Group III schematic. Description: semiclosed Brayton; combustion gas working fluid; turbocharged high pressure combustion.

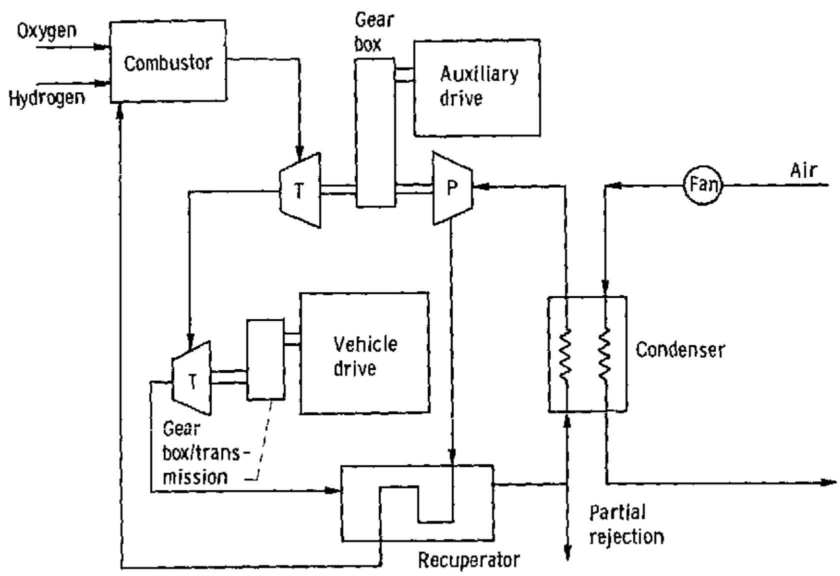


Figure 3-12. - Group IV schematic. Description: semiclosed cycle; hydrogen-oxygen reactants, steam working fluid; high pressure combustion.

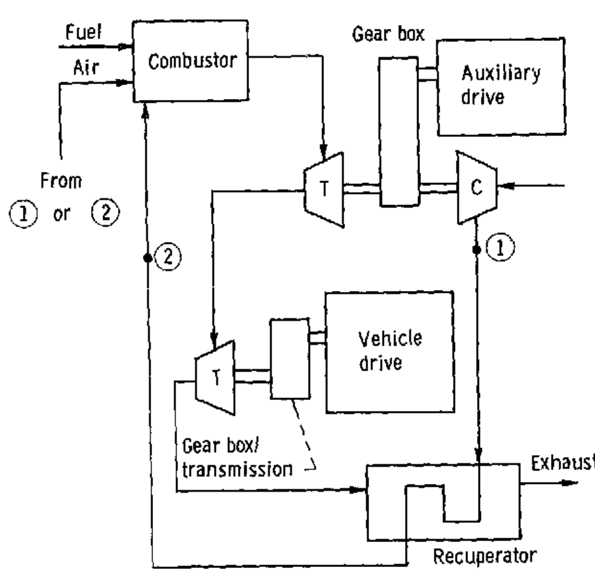


Figure 3-13. - Group V schematic. Recuperated open cycle.

4. ENGINE SCREENING ANALYSIS

The engine analysis of the engine screening phase was based on design-point analysis as discussed in section 1. Engine groups I, II, III, and IV with some thermodynamic variations were compared. Minimum engine weights were generated as a function of engine design-point SFC. General methods of the design-point studies are presented in appendix B.

Assumptions and Constraints

Table 4-1 lists the assumptions and constraints that were common to all of the engines studies in the screening analysis. Table 4-1(a) lists those assumptions and constraints which effect the power system configuration. Net power output for the bus was fixed at 400 horsepower. This power would be available to a transmission for both motive and auxiliary loads. Net electric-power output for the train engine was fixed at 7500 horsepower. In this case, 5000 horsepower was assumed for motive loads and 2500 for auxiliary loads.

Radial-flow turbomachinery was assumed for the bus application and axial-flow turbomachinery for the train application. Preliminary calculations showed no advantage of either a single- or two-shaft machinery arrangement. For bus engines a single-shaft arrangement was assumed. Since the train engine power output was split between a constant auxiliary load and a variable motive load, a two-shaft arrangement was assumed.

For preliminary sizing of the fuel system, it was assumed that the fuel tank would provide 1500 horsepower-hours of operation at the design-point SFC for the bus application and 18 000 horsepower-hours for the train.

As previously discussed in section 1 (p. 1) air-flow frontal area constraints of 2 by 4 feet for the bus and 9 by 18 feet for the train were assumed based on vehicle envelopes. For both vehicles the ambient air temperature was assumed to be 80° F.

TABLE 4-1. - DESIGN-POINT ASSUMPTIONS AND CONSTRAINTS -

ENGINE SCREENING PHASE

(a) Power system configuration

	Vehicle	
	Bus	Train
Power output, hp	[a]400	[b]7500
Type of turbomachinery	Radial flow	Axial flow
Turbomachinery shaft arrangement	Single	Double
Fuel tank energy capacity, engine output in hp-hr	1500	18 000
Waste heat exchanger air flow frontal area, ft	2 by 4	9 by 18
Ambient temperature, ^{o}F	80	80

(b) Engine losses and efficiencies

	Vehicle	
	Bus	Train
Turbine polytropic efficiency	0.87	0.89
Compressor polytropic efficiency	0.86	0.87
Alternator electromagnetic efficiency	[c]0.93 to 0.945	[c]0.93 to 0.945
Thermal losses, percent	5	5
Mechanical shaft losses, percent	5	5
Fan efficiency	0.85	0.85
Fan drive efficiency	0.98	0.90

[a]Shaft power.
[b]Electric power.
[c]Section 10.

Table 4-1(b) lists those engine losses and efficiencies which were assumed constant in the studies. These were considered to be representative of the current state of art for the bus and train applications. Initial calculations for the bus engines showed that to get reasonable rotor diameters and speeds it was necessary to assign less than optimum turbomachinery specific speeds (appendix B). The reduced polytropic efficiencies assumed for the bus engines accounted for the nonoptimum specific speeds. Thermal losses were accounted for by an increase in the ideal fuel mass flow rate. The mechanical shaft losses included windage, bearing, and gear box losses. Fan power needs were the main power system parasitic losses. The gross power output of each engine was increased to supply the fan-power needs. For bus engines, it was assumed that the fans were belt driven, while for train engines a motor drive was assumed.

Brayton Cycle Engine Trade-offs

CLOSED ENGINES

Although in the study of the closed Brayton cycle engines (engine groups I and II) the maximum cycle temperatures and compressor outlet pressure were not optimized, they were assigned fixed values. A temperature of 1700° F was considered to be the upper limit for superalloys in these applications where multiple stop-start cycles result in thermal fatigue damage. Hence, in the group I engines the combustor exit temperature was set at 1700° F with a corresponding turbine inlet temperature of 1500° F. In the group II engines the turbine inlet temperature was also set at 1500° F, and the temperature difference between the combustion products leaving the combustor and the Brayton working gas entering the combustor was set at 50° F. Compressor outlet maximum pressure for TACV engines was set at 800 psia. No trade-off was made on structure thickness within the heat exchanger and turbomachinery component models (appendix B). However, 800 psia was within the capabilities of the assumed material thicknesses. Since the bus engine would be used in more densely populated areas, compressor outlet pressure was set lower at 300 psia for this application.

Combustion-Loop Variations

The combustion-loop variations for closed Brayton cycle engines are discussed and ranked on the basis of thermodynamic considerations in the previous section and in appendix I. Potentially, engine group II was shown to be superior because of the reduced combustion air-flow requirements. And, group Ib was better than Ia because of its reduced combustion air-flow requirements in the preheater. Results of the design-point studies did not alter this ranking.

Figure 4-1 introduces the format of design-point study results and shows the performance of group II bus and train engines. The fuel was methane, while the Brayton working gas was air. Specific weight is plotted against net SFC. Specific weight curves are shown for the engine and some of the engine components, and the sum of engine and initial estimates are shown for the fuel and tanks. Train engine weight includes all of the heat exchangers, ducting, turbomachinery, reduction gear boxes, and alternators. Bus engine weights include the same components except for the alternators. The combustion-loop component weights for group II include those of the preheater, the combined surface combustor - heat exchanger, and the interconnecting ducts.

The curves of engine specific weight are the loci of several optimized solutions. Each point on the engine specific weight curve provides a potential set of minimum specific weight design conditions for a given SFC. These solutions also satisfy the waste heat exchanger surface area constraints. Other solutions, or possible sets of design conditions, would either fall above the engine specific weight curve or exceed the surface area constraints.

Since the fuel and tank weights are estimated at this point in the study, the more important curves are those of engine specific weight. The fuel and tank weights are included to show the variation in relative weights of the engine and fuel system. Engine weight decreased with increasing net SFC (lower conversion efficiency). As net SFC increased, compressor pressure ratio and component gas pressure drops increased, and heat exchanger heat-transfer effectiveness decreased. The main effect on engine weight was the decrease in recuperator size as its effectiveness decreased. Values of recuperator effectiveness are shown in figure 4-1 by marks on the recuperator weight curves. Comparing the engine and recuperator curves shows that the change in recuperator weight was greater than the change in weight of all the other engine components combined. At the maximum net SFC shown in figure 4-1, engine weight was still decreasing. Continued increases in net SFC would further reduce recuperator weight and conversion efficiency. However, because of the decrease in conversion efficiency the other system heat exchangers must increase in weight. And at some higher value of net SFC a minimum engine weight would occur.

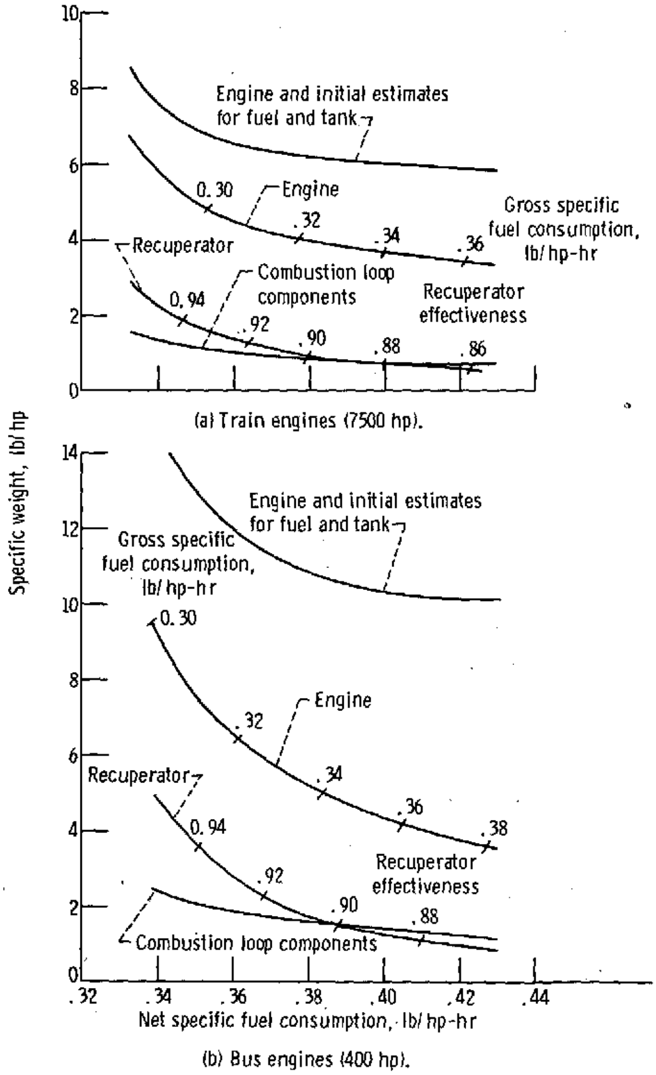


Figure 4-1. - Group II engine performance. Fuel, methane; Brayton gas, air.

Values of gross SFC are indicated on the engine weight curves. Gross SFC is based on the difference between the generated turbine power and the required compressor power. Hence, the gross values account for no engine losses other than the assumed thermal loss. And the difference between net and gross specific fuel consumption is a measure of parasitic losses (shaft losses and fan power needs).

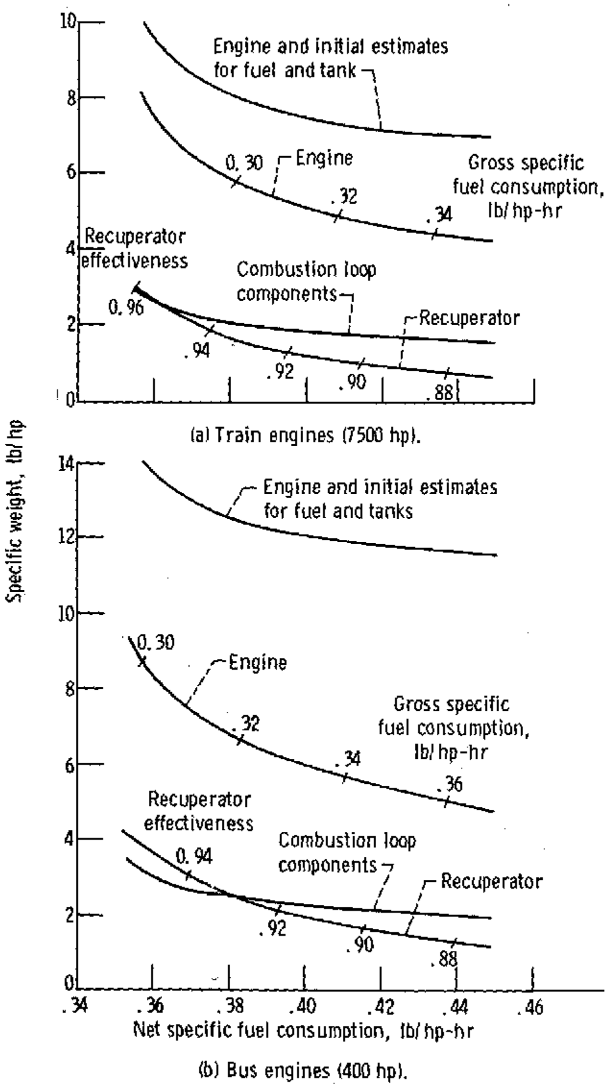
Figure 4-2 shows the similar results for group Ib engines. All trends were like those for the group II engines. However, the group Ib engines were heavier than group II engines at the same design-point net SFC. The group II combustion-loop components were smaller than those of group Ib because of lower mass-flow rates. The same levels of gross SFC were reached with the group Ib engines as those with group II. However, the total parasitic losses for the group Ib engines were higher.

A comparison was also made between the group Ia and Ib train engines. Fan power needs were about the same for both of these combustor-loop arrangements. However, the group Ia train engine specific weight was about 0.6 percent per horsepower heavier than that for the group Ib engine. The weight difference was mostly due to a larger preheater for the group Ia engine as was predicted from thermodynamic considerations.

Working Gas

One potential advantage for closed Brayton cycle engines is the freedom to choose the working gas. Air, single inert gases, or mixtures of inert gases are among the possible choices. The gas properties of specific heat, thermal conductivity, and viscosity strongly affect the design of Brayton cycle components. Turbomachinery components favor a high molecular weight gas mainly because they have lower specific heats. Lower specific heat reduces the required machinery specific work and number of stages for a given application. Conversely, the heat-transfer components favor a low molecular weight gas because they generally have a higher heat-transfer coefficient.

Binary mixtures of a high molecular weight inert gas (such as xenon or krypton) and a low molecular weight inert gas (such as helium or neon) are advantageous. Such mixtures can provide a choice in the average gas molecular weight and do provide an improved heat-transfer coefficient compared



(b) Bus engines (400 hp).

Figure 4-2. - Group Ib engine performance. Fuel, methane; Brayton gas, air.

to the single intermediate molecular weight inert gases. For example, if a Brayton cycle system were to be designed at the molecular weight of neon, argon, or krypton, the use of a helium-xenon mixture at the same molecular weight would allow a reduction in heat exchanger weight.

The cost of working gases in ground power systems is important. Xenon and krypton were not considered to be practical for these applications because of their rarity and hence high costs. However, gas mixtures of helium and argon were studied and compared to air. At the molecular weight of air, engine weight was the same with both air and the mixture. Comparison with air properties showed that the mixture did have a larger coefficient of thermal conductivity by about 25 percent, but that the mixture also had a larger viscosity by about 27 percent. These effects were off-setting and resulted in no weight savings. Small engine weight savings were calculated with the helium-argon mixtures at molecular weights less than air. At a mixture molecular weight of 5, there was a 13 percent decrease in engine weight. However, the number of turbomachinery stages more than doubled.

There is an additional complication associated with the choice and use of the closed Brayton cycle working gas which has not yet been discussed. To obtain good part-power engine performance (high off-design efficiency) with a closed Brayton cycle, gas inventory control would be required. Simply reducing combustor fuel and air mass flow rates reduce turbine inlet temperature and spoils the system efficiency. Therefore, economy dictates that for long operating periods at less than full power gas inventory should be reduced. This allows the design-point turbine inlet temperature and turbomachinery speed to be maintained. With air as the working gas, the inventory control system would need a moisture and particle removing device, a charging compressor, and injection and vent values. With inert gas mixtures and their costs, a closed inventory control system with a storage tank would be needed. Weights for such an inventory control system were not included in the engine design-point results.

Air was selected as the closed Brayton engine working gas for the bus and train application because it is naturally available and its use is less complicated compared to inert gases.

TABLE 4-2. - EFFECTS OF INTERCOOLING ON

GROUP II TRAIN ENGINE

[Fuel, methane; working gas, air.]

	Intercooling	
	Without	With
Total waste heat exchanger length, ft	18	18
Compressor pressure ratio	3.9	6.2
Gross specific fuel consumption, lb/hp-hr	0.357	0.312
Net specific fuel consumption, lb/hp-hr	0.389	0.403
Gross shaft power, hp	8 800	9 700
Power losses, hp:		
Total	1 300	2 200
Shaft	950	1 050
Fan	350	1 150
Initial weight estimates, lb:		
Total	45 200	45 600
Tank and fuel	17 050	17 900
Engine	28 150	27 700

Intercooling

Table 4-2 lists the main effects of intercooling between compressors on group II train engines. The comparison in table 4-2 is made at equal values of the engine figure of merit (appendix B). Both solutions were constrained so that the dimensions of the total waste heat exchanger air-flow frontal area did not exceed 9 by 18 feet; that is, the intercooler was included within these dimensions. Gross SFC was lower but the fan power needs tripled with intercooling. And, the resulting net SFC was higher with intercooling than without. No advantage was found for intercooling the group II engines.

SEMICLOSED ENGINES

Compared to engine groups I and II, the semiclosed Brayton cycle engines (group III) are characterized by higher pressure combustion and higher turbine inlet temperature capabilities. High-pressure combustion occurs because the combustor is within the cycle. Higher turbine inlet temperature is possible since the combustion products are the working gas, and they directly enter the turbine without need for an intermediate heat exchanger.

For basic group III engine performance, the combustor exit temperature was fixed at 1700° F. This was the same limit used for the closed Brayton cycle engines. Because of the absence of a heat source heat exchanger in group III engines, higher temperature operation with the use of turbine blade cooling was also considered. For design-point performance with turbine cooling the combustor exit temperature was set at 2100° F. This temperature was chosen to be representative of the current state of art for turbine cooling. For all of the group III engines, the turbocharger compressor pressure ratio was limited to no more than 25 to preclude excessively large numbers of machinery stages.

For the basic group III performance (1700° F turbine inlet temperature) enough of the working gases was bled from the combustor exit through the turbocharger turbine to match the required turbocharger compressor work (see section 3). The remaining excess cycle flow was bled from the engine as gases at the waste heat exchanger inlet and as water at the waste heat exchanger outlet. For the cooled turbine solutions (2100° F turbine inlet temperature), all the excess gas flow was bled from the drive-turbine exit through the turbocharger turbine. This was done to avoid the need for cooling the turbocharger turbine. In this case, however, there was not enough energy in the bleed flow to satisfy the turbocharger compressor work. The required additional power was assumed to be taken from the auxiliary drive shaft. Physically this situation would require coupling the auxiliary shaft and the turbocharger shaft.

Basic Performance

The basic group III engine performance is presented in figure 4-3. The previous format is repeated. However, the specific weight of the combustion-loop components is not shown. For group III the combustion-loop components are the combustor, the turbocharger, and the associated ducts. The

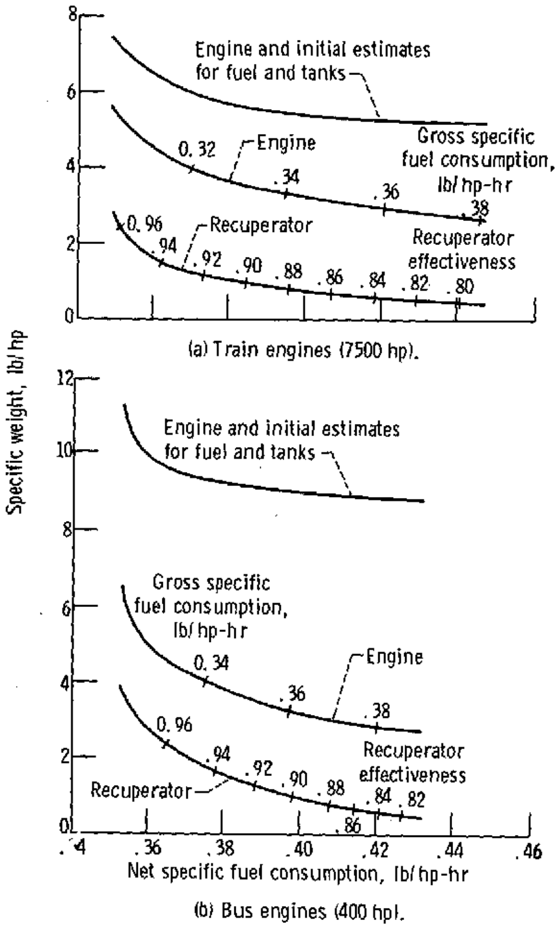


Figure 4-3. - Basic group III engine performance. Fuel, methane.

sum of these specific weights were relatively small and almost constant with SFC.

All of the trends for the group III engines are similar to those for groups I and II. In general, group III had lower engine specific weights than group II because of the lighter combustion-loop component weights. Also, group III engines had low parasitic losses. These losses increased SFC for the train engines by about 15 percent and for the bus engines by about 10 percent.

In spite of a 200° F turbine inlet temperature increase, group III engines operated over about the same range of gross SFC as did groups I and II. A comparison of the optimized solutions showed two conditions which tend to increase gross SFC and offset the expected improvement due to the 200° F temperature rise. The group III weight-optimized engine solutions had ratios, higher compressor inlet temperatures, and larger pressure losses within the closed-loop portion of its cycle.

Turbine Cooling

Effects of the higher turbine inlet temperature are shown in figure 4-4. Engine specific weight is plotted against net SFC. The scale of the ordinate in figure 4-4 was doubled compared to the previous figures. Both cooled and uncooled engine weight curves are shown. A substantial improvement in net design-point SFC resulted with the higher turbine inlet temperature. Net SFC was lowered by about 20 percent for the same engine specific weight.

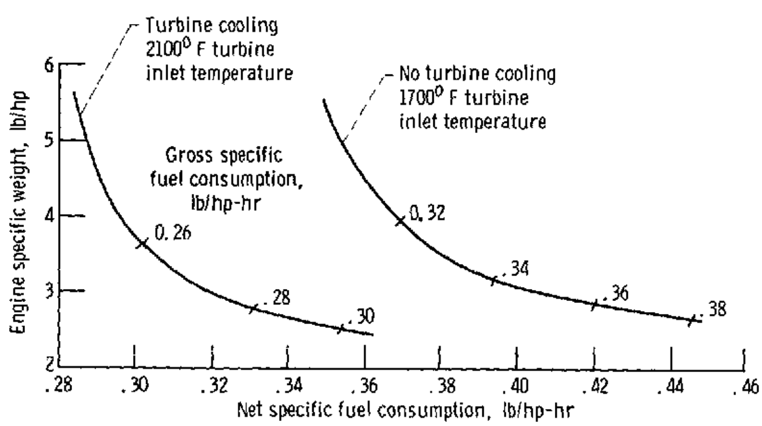


Figure 4-4. - Effects of turbine cooling on group III train engine performance. Fuel, methane.

Intercooling

Effects of intercooling on group III train engines are listed in table 4-3. The comparison is made at equal values of the engine figure of merit (see appendix B). Both with and without intercooling the waste heat exchanger air-cooled frontal area was constrained to 9 by 18 feet. Gross SFC with intercooling was about 6 percent lower than without. Fan power needs with intercooling increased by 100 horsepower while the shaft losses remained about the same. The net SFC with intercooling was also 6 percent better than without. Both engine and total system weight showed some savings with intercooling.

Although intercooling complicates engine arrangement within the vehicle, there were potential benefits for its use with the semiclosed Brayton cycle engines.

TABLE 4-3. - EFFECTS OF INTERCOOLING ON BASIC GROUP III TRAIN ENGINES

[Fuel, methane.]

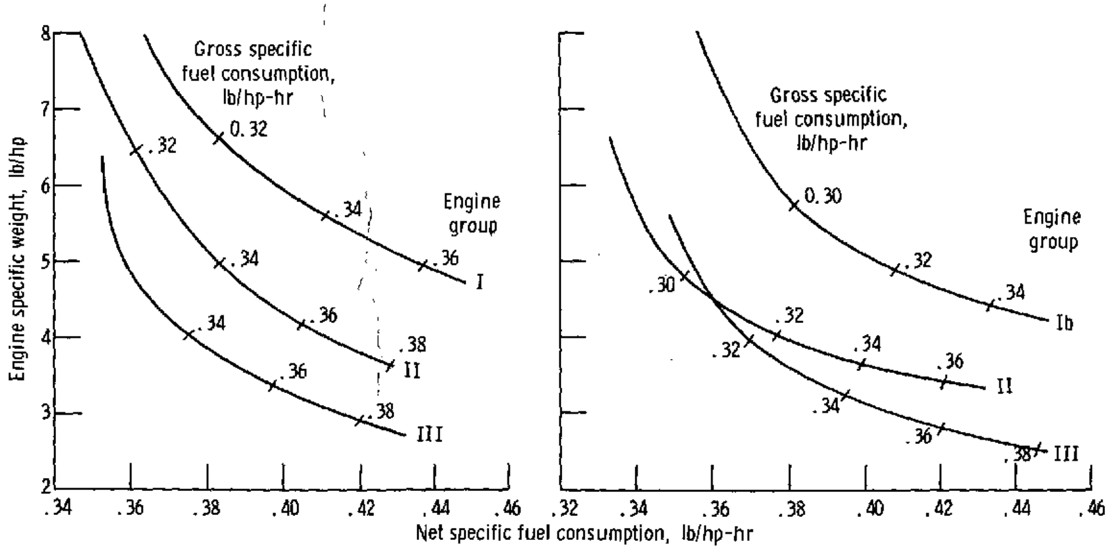
	Intercooling	
	Without	With
Total waste heat exchanger length, ft	18	18
Compressor pressure ratio	4.3	6.8
Gross specific fuel consumption, lb/hp-hr	0.324	0.303
Net specific fuel consumption, lb/hp-hr	0.514	0.354
Gross shaft power, hp	8 660	8 770
Power losses, hp:		
Total	1 160	1 270
Shaft	890	900
Fan	270	370
Initial weight estimates, lb		
Total	43 400	41 700
Tank and fuel	15 200	14 300
Engine	28 200	27 400

RELATIVE ENGINE PERFORMANCE

Basic engine performance for groups Ib, II and III are compared in figure 4-5. These engine specific weight curves are repeated from previous figures. However, the scale of the ordinate in figure 4-5 was doubled. No intercooling or turbine cooling was used for the comparisons, and methane was the fuel.

For the train engines (fig. 4-5(b)) group III had the lightest engine specific weight above the design-point net SFC of 0.36 pound per horsepower-hour. For the bus engines (fig. 4-5(a)) the group III engine was lightest over its range in fuel consumption. Reasons for these relative performances are complex and are now discussed.

As pointed out previously, all three engine groups optimized with about the same potential operating range in gross SFC but group III had the least parasitic losses. Also, group III had the least weight in combustion-loop components. Conflicting factors, mainly in the relative size of the recuperator and waste heat exchanger, led to a large reduction in group III bus engine weight compared to group II and a more modest reduction between train engine weights.



(a) For 400-horsepower bus engines.

(b) For 7500-horsepower train engines.

Figure 4-5. - Comparison of basic Brayton cycle engine group performance. Fuel, methane.

Heat exchanger weight is a function of the required heat-transfer effectiveness, capacity rate ratio (the ratio of products of mass-flow rate and specific heat at constant pressure), pressure level, and pressure drops. The recuperators in groups I and II have equal mass-flow rates in the hot-side and cold-side passages and have a capacity rate ratio near unity. However, the group III recuperator has about 5 percent more mass flow in its hot side than in its cold side. This effect, holding all other factors constant, would result in a lower weight for the group III recuperator. For the bus engines, recuperator pressure levels and pressure drops were about the same at equal values of effectiveness for all three engine groups. Group III recuperators for the bus engine were about one-half the weight of those for group II. Pressure level had a conflicting effect among the train engines. The group III engines optimized at the turbocharger compressor pressure ratio limit of 25. Group III engine's main compressor optimized at higher pressure ratios than those for groups I or II. The resulting recuperator pressure levels for group III train engines were about 370 psia on the cold side and about 90 psia on the hot side. In comparison, the group II recuperator pressure levels were higher, about 800 psia on the cold side and about 220 psia on the hot side. The net effect of differences in capacity rate ratio and pressure level on the train engine recuperators was that at an effectiveness of 0.86 the recuperator specific weights were equal and above 0.86 the group III recuperator became increasingly lighter. At an effectiveness of 0.95 the group III recuperator was about 25 percent lighter. Although group III recuperators were lighter at high effectiveness, this engine group did require higher levels of effectiveness to achieve the same gross SFC.

Similar effects caused a reversal of the relative waste heat exchanger weight between the bus and train applications. For bus engines, the group III waste heat exchangers were lighter than those of groups I and II. For train engines, the group III waste heat exchangers were heavier. Group III bus heat exchangers were lighter mainly because they optimized at lower capacity rate ratios, but for train applications they were heavier mainly because of the lower pressure level.

Although the group III engines optimized at the turbocharger compressor pressure limit, the rate of change of engine weight at the limit was small. Raising the limit would allow little weight reduction at the expense of more turbocharger machinery stages.

FUEL COMPARISON

Fuel comparisons are presented in table 4-4. They are made at equal values of the engine figure of merit for group III. It should be remembered that the fuel and tank weights here are preliminary estimates based on the arbitrary number of assumed horsepower-hours (table 4-1). Also, the cryogenic tank weight calculations were made using the most conservative design practices described in appendix H.

Only methane and kerosene were considered for the bus application. For the train, hydrogen was considered along with methane and kerosene. Differences in fuel consumptions and fuel weight were mainly due to the changes in heating values among the fuels. Engine weights were nearly independent of the fuel. The lightest initial estimates of total weight were obtained with kerosene. Its conventional tankage weight was less than the cryogenic tankage weight with methane or hydrogen. Furthermore, kerosene tank volumes were the smallest as shown by the equivalent tank sizes in table 4-4.

TABLE 4-4. - FUEL COMPARISONS FOR BASIC GROUP III ENGINES

	Vehicle				
	Bus		Train		
	Fuel				
	Methane	Kerosene	Methane	Kerosene	Hydrogen (air)
Net specific fuel consumption, lb/hp-hr	0.35	0.41	0.37	0.42	0.14
Engine specific weight, lb/hp	3.4	3.4	3.8	3.8	4.0
Initial weight estimates, lb:					
Total	3380	2020	43 400	36 000	52 700
Tank	1480	40	8 500	220	20 200
Fuel	530	620	6 700	7 800	2 500
Engine	1370	1360	28 200	28 000	30 000
Equivalent tank size, diameter times length, ft	2 by 16	2 by 4	4 by 32	4 by 13	4 by 70

Hydrogen-Oxygen Engine

As stated previously, the semiclosed cycle using hydrogen and oxygen combustor reactants differed from the other semiclosed cycles to such an extent that it was treated separately as group IV (see section 3). Group IV is also unique among all the cycles studied in that the total system weight must include the oxidant and its tank. In the other groups the specific fuel consumption serves as a measure of efficiency and as an indication of the rate of use of consumables. In the case of group IV the specific reactant consumption (SRC) serves the latter purpose. In the results to be presented the engine weight is given as a function of both SFC and SRC.

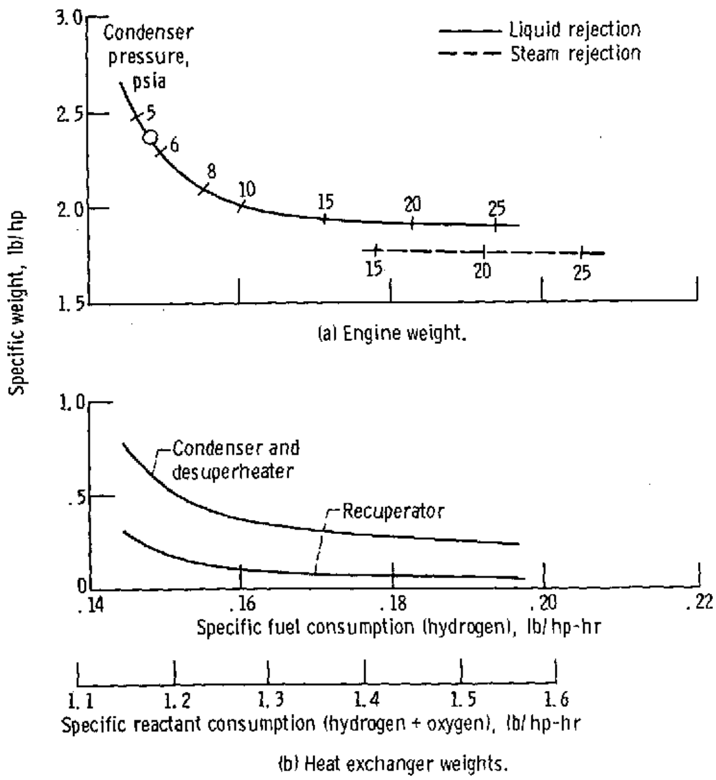


Figure 4-6. - Group IV weight and design-point performance. Net power output, 7500 horsepower; turbine inlet temperature, 1800° F; combustion chamber pressure, 600 psia; ambient temperature, 80° F.

In figure 4-6(a) the optimum engine weight is given as a function of design-point net SFC for a combustion chamber pressure of 600 psia and turbine inlet temperature of 1800° F. As explained previously the curve defines the envelope above which all nonoptimum engine designs fall. At a particular design-point SFC, parameters including all heat exchanger effectivenesses and pressure drops, condensing pressure, and duct pressure losses are adjusted to obtain the minimum engine weight.

For reference, a hydrogen SFC of 0.14 pound per horsepower-hour efficiency of about 35 percent and an SFC of 0.18 pound per horsepower-hour to an efficiency of 27.5 percent. The SFC values in figure 4-6 include losses and parasitics such as air-coolant fan power. For the design point indicated by the circle in figure 4-6, the gross SFC without any losses or parasitics included is 0.12 pound per horsepower-hour (41 percent) compared to the net value 0.15 pound per horsepower-hour shown. Indicated on the curve in figure 4-6(a) is the variation in condensing pressure; as the condensing pressure (temperature) is increased the SFC increases while the engine weight decreases. The variation in engine weight is due to the variation in heat exchanger weights which is shown in figure 4-6(b) (the other components are relatively constant in weight). As the SFC is increased, the heat rejection from the air-cooled condenser and desuperheater increases. In spite of this, the curves in figure 4-6(b) show these heat exchangers decreasing in weight as SFC is increased. This is because the temperature difference between the ambient air and condensing steam increases as the design-point SFC (and hence condensing pressure) is increased. A further increase in condensing pressure beyond the range shown in the figure would, however, result in an increase in engine weight.

The solid curves in figure 4-6 assume that the portion of the working fluid which is not recirculated to the combustor is condensed and rejected from the system as water at the pump exit. If the condensing pressure were to be above atmospheric pressure, there obviously would be an advantage in reducing the desuperheater and condenser flow rates and hence size by rejecting this portion of the working fluid as steam before the desuperheater inlet. The dashed curve in figure 4-5(a) is the optimum engine weight curve assuming this is the case.

The specific reactant consumptions shown in figure 4-6 are much higher than the specific fuel consumptions for the other cycles considered. Even though the group IV engine is lighter the weight of the consumables obviously will be heavier than for the other groups. Also, because both of the reactants would either be carried as cryogenic liquids or high pressure gases, the tanks would be much heavier than those required for the other cycles, which could operate on a fuel such as kerosene.

In figure 4-7 the total system weight which includes engine, consumables, and cryogenic tanks is given for the case corresponding to figure 4-6.

The weights of consumables and tanks were determined with the assumption that the engine is operated at design-point efficiency for 18 000 horsepower-

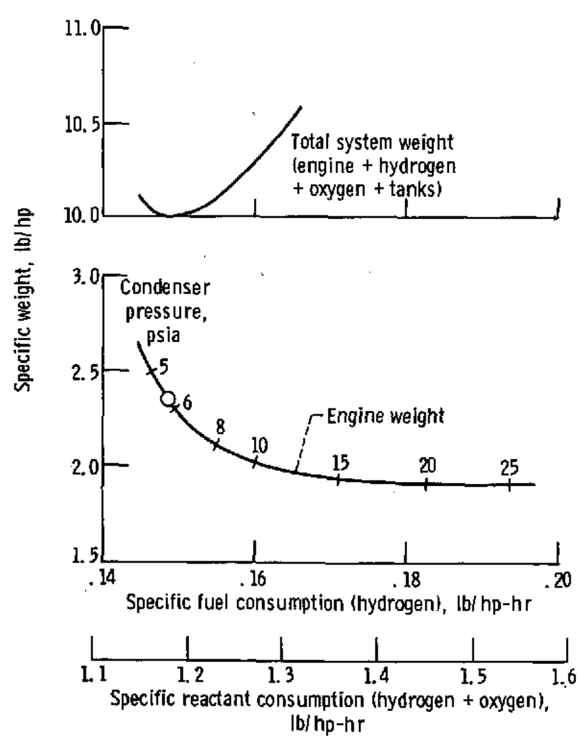


Figure 4-7. - Group IV engine, reactant, and tank weights. Mission, 18 000 horsepower-hours at full power; tanks, cryogenic, 4 foot diameter, and cylindrical; net power output, 7500 horsepower; turbine inlet temperature, 1800° F; combustion chamber pressure, 600 psia; ambient temperature, 80° F.

TABLE 4-5. - EXAMPLE TOTAL SYSTEM WEIGHT BREAKDOWN FOR GROUP IV ENGINE FOR 18 000 HORSEPOWER-HOUR ENGINE OUTPUT AT DESIGN-POINT EFFICIENCY

[Specific fuel consumption, 0.15 lb/hp-hr; net power output, 7500 hp; turbine inlet temperature, 1800° F; combustion chamber pressure, 600 psia; ambient air temperature, 80° F.]

	Specific weight, lb/hp
Engine	2.34
Hydrogen	.36
Oxygen	2.83
Hydrogen tank (cylindrical, 4 ft diam)	2.87
Oxygen tank (cylindrical, 4 ft diam)	1.60
Total	10.00

hour engine output. As shown, this results in a minimum total weight of about 10.0 pounds per horsepower which occurs at a condenser pressure well below atmospheric pressure. The weak breakdown for the minimum total weight design point indicated by the circle in figure 4-7 is given in table 4-5. As shown, both the weights of the consumables and of the two tanks exceed the engine weight. The tanks were assumed to be cylindrical, 4 feet in diameter, and to correspond to the most conservative designs considered in appendix H.

For a shorter mission than that assumed for figure 4-7, the consumables and tanks would be correspondingly reduced in weight so that the minimum in total system weight would be reduced and shifted to higher design-point SFC.

As discussed previously (in section 3), the air-coolant flow rates for the group IV engine tend to be higher than for the Brayton engines. To keep the air fan power requirements from becoming excessive, the condenser tends to optimize with small air-side pressure loss by keeping the air-flow length small and the air-flow frontal area high. The condenser dimensions for the case of figure 4-6 significantly exceed the engine compartment dimensional guideline constraints for the 7500-horsepower engine as will be shown shortly. As in the case of the other groups, the optimization must therefore be constrained to bring the heat exchangers within allowable dimensions. As an illustration of the effects of constraints, in figure 4-8 the air-side relative pressure loss was not allowed to be less than 2 percent in order to increase the air-flow length and decrease the air-flow frontal area. Shown for comparison are the unconstrained results of figure 4-6 which have less than 2 percent pressure loss. Indicated on the condenser desuperheater weight curves is the air-side frontal area of the condenser for both cases.

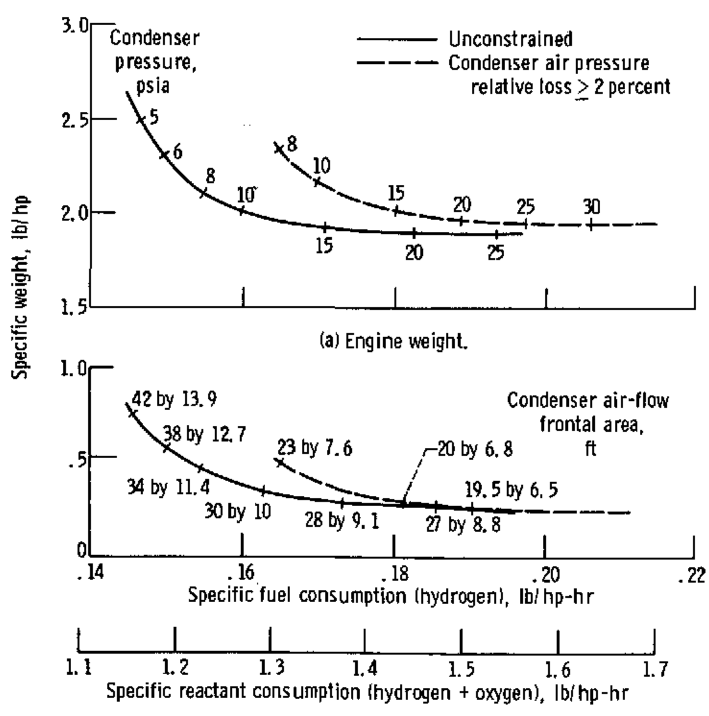


Figure 4-8. - Effect of size constraint on group IV weight and design-point performance. Net power output, 7500 horsepower; turbine inlet temperature, 1800° F; and combustion chamber pressure, 600 psia.

As the condensing pressure is decreased, the heat rejection temperature approaches the ambient air temperature, the air-flow rate increases, and the condenser air-flow area rapidly increases. It is seen by comparing the two sets of results in figure 4-8 that at the same condenser pressure the curve for the constrained case is shifted to higher SFC and higher weight. This is due to a significant increase in air fan power. For the same net output power, the gross power for the constrained case is higher, resulting in the increased weight. This effect would actually be increased if the weight of the air fan motors were included in the curves. With the higher air-side pressure losses of the constrained case, the air-side frontal area of the condenser is reduced as shown in figure 4-8 and the smallest dimensions shown approach the study guidelines for the engine compartment size. However, as shown, the penalty in SRC is substantial.

Concluding Remarks

The engine screening analysis was part of the total evaluation process for bus and train power systems. These studies were used mainly to narrow the number of engine types and potential fuels in the conceptual design phase. In the engine screening phase, closed and semiclosed Brayton cycle engines (groups I, II, and III) and a hydrogen-oxygen fueled engine (group IV) were analyzed. Methane, hydrogen, and kerosene were considered as potential fuels. Optimized system weights were generated as a function of design-point SFC for each engine type with some thermodynamic variations. Engine geometry was constrained to fit within typical bus and train envelopes.

Among three possible combustion-loop arrangements for the closed Brayton cycle engines one was shown to have the best performance. The use of a combined surface combustor Z-Z heat exchanger resulted in the lowest weight and specific fuel consumption for the closed-cycle engines. Both air and helium-argon mixtures were considered for the closed-cycle working gas in train engines. The helium-argon mixture at the molecular weight of air showed no advantage. And air was selected as the closed Brayton engine working gas. No advantage was found for intercooling between compressor with the closed Brayton cycle engines.

The semiclosed Brayton cycle engine which was studied was shown to have better performance than the best of the closed engines. For bus engines the semiclosed engines were lighter over the whole range of design-point SFC. The semiclosed train engines were lighter over all but the lowest part of the range in design-point fuel consumption. The semiclosed cycle considered also can employ turbine cooling to allow use of higher turbine inlet temperatures. Furthermore, intercooling between compressor stages showed performance gains for the semiclosed engines.

An analysis of the hydrogen-oxygen semiclosed cycle engine showed that although it would have the lightest engine weight of all those studied it would require a very heavy fuel system. Effects of the vehicle geometry constraint on waste heat exchanger area reduce performance and consequently add to the fuel system weight.

The semiclosed Brayton cycle engine (group III) and kerosene as the fuel were selected for more detailed analysis in the conceptual design phase.

5. CONCEPTUAL DESIGN PHASE

Based on the results of the engine screening phase, the semiclosed Brayton cycle engine (group III) and kerosene as the fuel were selected for the conceptual design phase of the study. The group III engine was reexamined in more detail and compared to the open Brayton engine (group V). The conceptual design phase consisted of a design-point analysis (see appendix B) and a quantitative off-design power level analysis (see appendix C). The results of the design-point analysis were used as input for the off-design and overall engine-mission analyses.

Design-Point Performance

As discussed in section 2 the initial comparisons in the conceptual design phase were based on design-point performance. The general methods of design-point studies are presented in appendix B.

ASSUMPTIONS

Table 5-1 lists the assumptions for the conceptual design phase. Footnote a indicates the assumptions which were changed from those in the engine screening phase (see table 4-1). Two net power output levels were studied for the train engines. The 5000-horsepower level was added for the possibility of using two such engines for motive power in the TACV application with a third engine or alternate source for levitation and housekeeping power. A double-shaft turbomachinery arrangement was assumed where the turbine compressor shaft acted as a gas generator. The assumed fuel tank energy capacity was scaled down directly with the output power level, while the waste heat exchanger surface area constraint was maintained. For the 7500-horsepower output level a single-shaft turbomachinery arrangement was specified. In this case it was assumed that a

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variable-speed transmission could be used with one of the two alternators for motive power needs.

Examination of potential operating conditions from the results of the engine screening phase led to some changes in the turbomachinery assumptions. For the bus application the turbines were changed from radial-flow to axial-flow components mainly because of the high tip speeds. Also, for both engine applications the assumed turbine polytropic efficiency was reduced from

TABLE 5-1. - DESIGN-POINT ASSUMPTIONS - CONCEPTIONAL

DESIGN PHASE

(a) Power system configuration

	Vehicle		
	Bus	Train	
Power output, hp	400	7 500	[a]5 000
Turbomachinery type			
Compressors	Radial	Axial	Axial
Turbines	Axial[a]	Axial	Axial
Turbomachinery shaft arrangement	Single	Single[a]	Double
Fuel tank energy capacity, engine output in hp-hr	1500	18 000	12 000
Waste heat exchanger air flow frontal area, ft	2 by 4	9 by 18	9 by 18
Ambient temperature, ^{o}F	80	80	80

(b) Engine losses and efficiencies

	Vehicle	
	Bus	Train
Turbine polytropic efficiency	0.87	[a]0.87
Compressor polytropic efficiency	[a]Modelled	0.87
Alternator electromagnetic efficiency	[b]0.93 to 0.945	[b]0.93 to 0.945
Thermal losses, percent	5	5
Mechanical shaft losses, percent	5	5
Fan efficiency	0.85	0.85
Fan drive efficiency	0.98	0.90

[a]Change from table II-5.
[b]Section 10.

0.89 to 0.87. Resulting turbine adiabatic efficiencies, with the higher value and relatively high pressure ratios, were deemed to be too optimistic. The radial-flow compressors which were used for the bus application operated over a range of design pressure ratios and at relatively low equivalent mass flow rates. For the turbocharger compressor of group III nonoptimum values of specific speed were required. To account for more than the effects of pressure ratio a radial-flow compressor efficiency model was added to the analysis. With this model (appendix B) efficiency was determined as a function of pressure ratio, equivalent mass flow, and specific speed.

SEMICLOSED BRAYTON CYCLE ENGINES

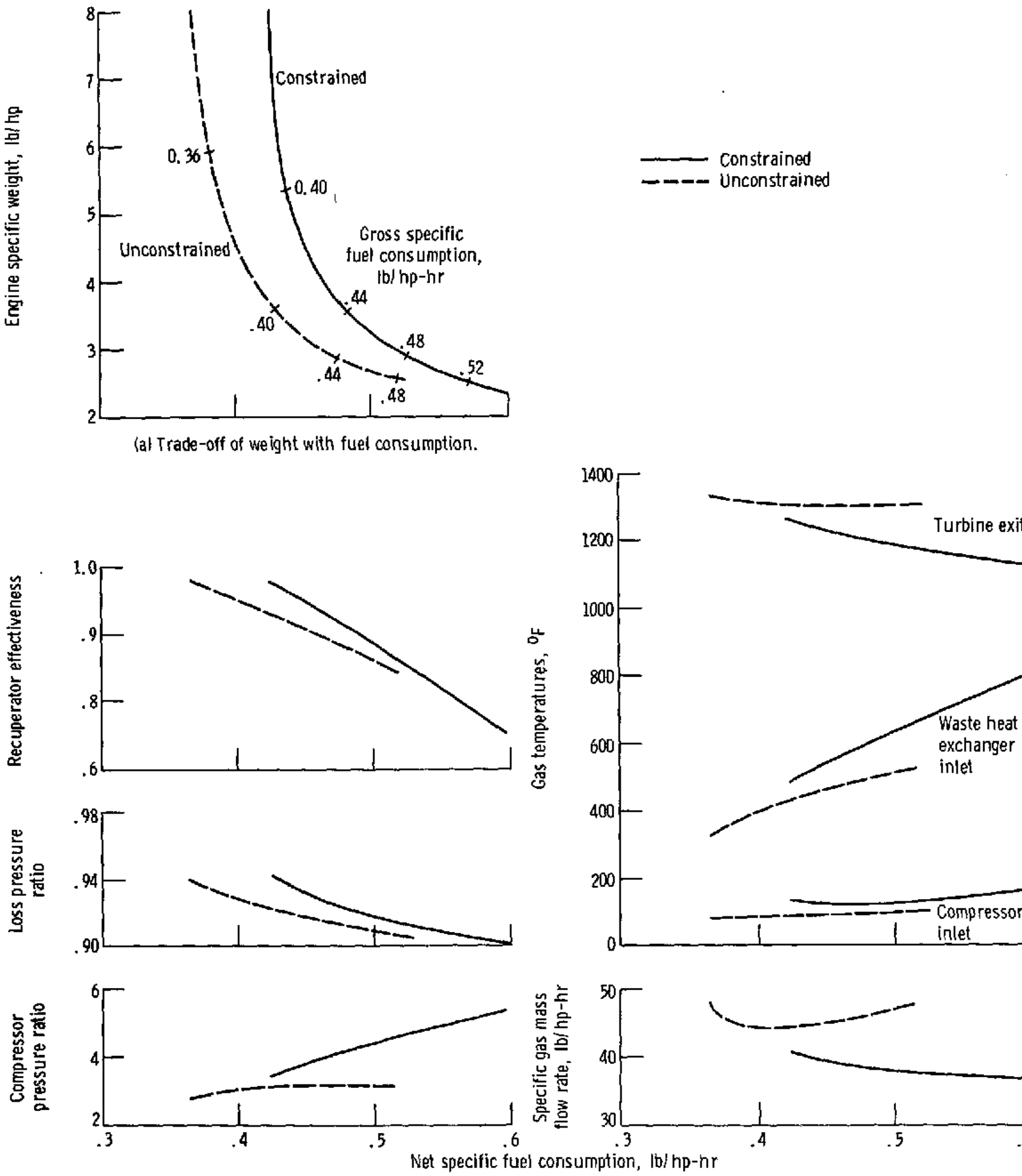
Bus Application

Details of the revised group III bus engine performance are presented in the three parts of figure 5-1. Figure 5-1(a) shows optimized engine weights, figure 5-1(b) shows the required cycle parameters, and figure 5-1(c) shows the resulting gas temperatures and flow rates. In each part the abscissa is net specific fuel consumption. The levels of consumption are larger than those of the previous figures mainly because of the switch in fuel from methane to kerosene. Two sets of optimized results are shown. The solid lines show results with the waste heat exchanger surface area constraint. The dashed lines show unconstrained results.

Figure 5-1(a) shows that at the same engine specific weight the penalty of the constraint ranged from about 10 to 15 percent in net specific fuel consumption. The waste heat exchanger air frontal area for the unconstrained case was 3.6 times quieter than the constrained area at 0.52 pound per horsepower-hour and 17 times greater at 0.36 pound per horsepower-hour. A comparison of the solutions shows the effects of the vehicle dimensional constraint on engine design characteristics.

The main differences between the solutions were that the constrained engines had higher fan power losses, higher compressor pressure ratios (fig. 5-1(b)), and higher values of compressor inlet temperature (fig. 5-1(c)).

Fan power losses are reflected in figure 5-1(a) as the difference between net and gross fuel consumption. Each of the effects causes a penalty in fuel



(b) Cycle parameters.

(c) Gas temperatures and mass flow rate.

Figure 5-1. - Group III bus engine performance at 400 horsepower. Fuel, methane.

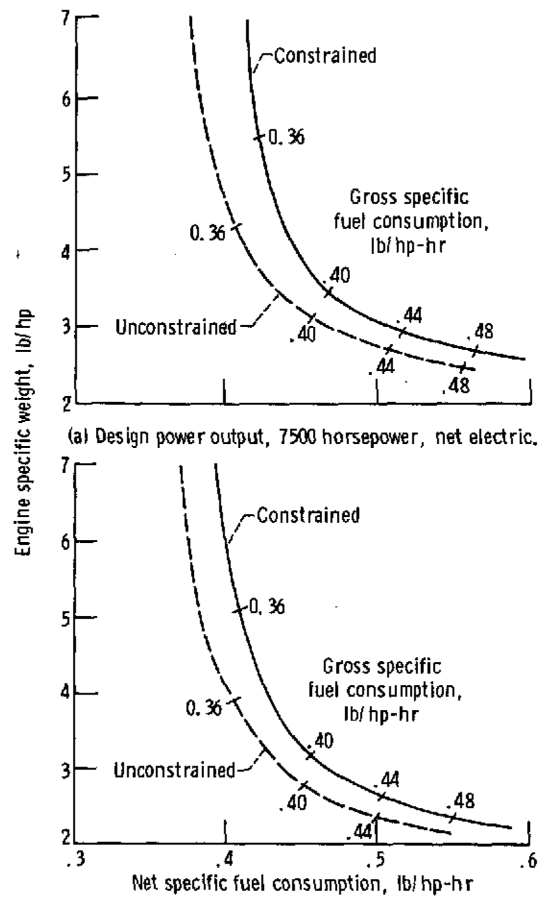
consumption. Fan power is a direct parasitic loss. High values of compressor pressure ratio indicate a shift away from the optimum efficiency values based on thermodynamic performance alone (appendix I). And, higher compressor inlet temperatures also result in poorer thermodynamic efficiency.

Reasons for these effects are interrelated. The unconstrained waste heat exchangers had short coolant air-flow lengths, relatively long working gas-flow lengths, and very long nonflow dimensions (see appendix A for explanation of heat exchanger geometry). The constrained heat exchangers had longer air-flow lengths, shorter gas-flow lengths, and shorter nonflow dimensions. Also, the constrained heat exchangers had lower air mass flow rates. Fan power increased in spite of the lower air mass flow rates because of higher pressure drops with the longer air-flow length. Higher values of compressor inlet temperature occurred for the constrained case because the geometry limits restricted the waste heat exchanger heat-transfer effectivenesses to lower values. Furthermore, when optimizing the constrained case, in order to compact the waste heat exchanger geometry without large penalties in engine efficiency, low working gas-side pressure drops resulted. The low drops were achieved with a shorter gas-flow length and a lower gas-flow velocity (or inlet pressure). Since both constrained and unconstrained solutions optimized at the same values of turbocharger pressure ratio, lower waste heat exchanger inlet pressures result in higher compressor pressure ratios. The higher pressure ratios in turn cause the lower turbine exit temperature (fig. 5-1(c)), working gas mass-flow rate, and the higher waste heat exchanger inlet temperature.

Further conceptual design details are presented in appendix B.

Train Application

Basic performance for the group III train engines is presented in figure 5-2. Trade-offs of engine weight and fuel consumption are shown for the 7500-horsepower output level in figure 5-2(a) and for the 5000-horsepower output level in figure 5-2(b). Cycle parameters and conditions varied in a manner similar to those of the bus and are not shown.



(a) Design power output, 7500 horsepower, net electric.

(b) Design power output, 5000 horsepower, net electric.

Figure 5-2. - Basic group III train engine performance. Fuel, kerosene.

Figure 5-2(a) for the 7500-horsepower engines shows that at the same specific weight the penalty for the waste heat exchanger constraint was about 10 percent in net specific fuel consumption. The unconstrained waste heat exchanger air frontal areas varied from 2.5 times the constrained area at 0.57 pound per horsepower-hour to 8.5 times at 0.38 pound per horsepower-hour.

Engine design-point performance at the 5000-horsepower output power level (fig. 5-2(b)) was better than that at 7500 horsepower. Lower fuel consumptions occurred mainly because the same waste heat exchanger air frontal area constraint was applied to the lower power level engine and was therefore more easily satisfied. Lower engine specific weights were due almost entirely to lower waste heat exchanger specific weights. The penalty of the waste heat exchanger constraint was 5 to 6 percent in SFC for the 5000-horsepower engines. And the unconstrained heat exchanger air frontal areas varied from 1.7 times the constrained area at 0.55 pound per horsepower-hour to 6.5 times at 0.37 pound per horsepower-hour.

Thermodynamic Variations

It was shown in the engine screening phase that both the use of higher inlet temperatures with turbine cooling and intercooling improved group III engine performance but added mechanical complexity. These two operating options were reassessed in the conceptual design phase. Results are shown in figure 5-3(a) for the 7500-horsepower train engine and in figure 5-3(b) for the 5000-horsepower version. All solutions were constrained. The reference case (1700° F turbine inlet temperature) is repeated from figures 5-2(a) and (b).

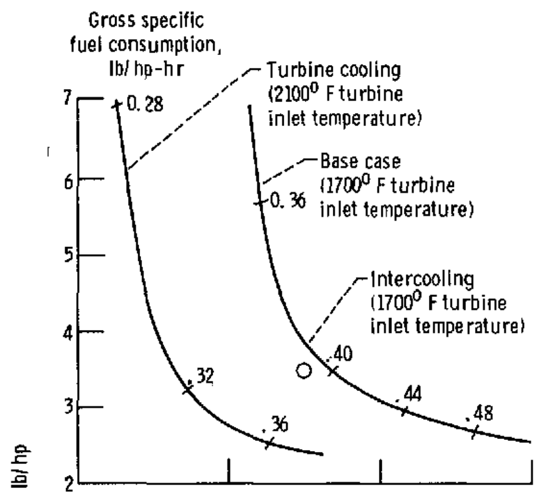
Turbine cooling resulted in about a 25-percent reduction in net specific fuel consumption at both power levels. The single solutions for intercooling showed about a 3-percent improvement in SFC at the 7500-horsepower level and about 9-percent improvement at the 5000-horsepower level.

OPEN BRAYTON CYCLE ENGINES

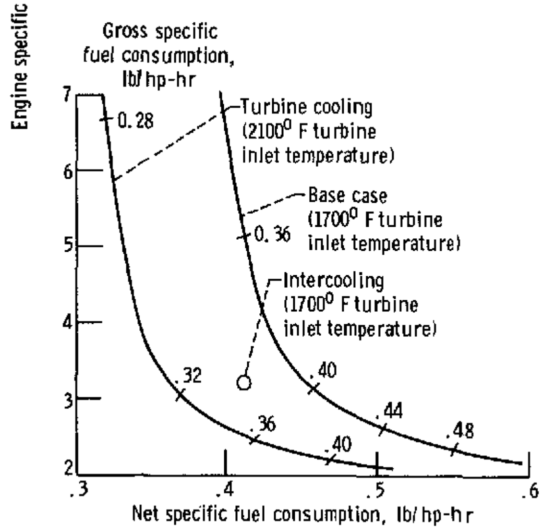
Bus Application

The engine specific weight as a function of the design-point SFC for the 400-horsepower single-shaft bus engine is given in figure 5-4(a). The variation in engine weight is primarily due to the recuperator weight variation which is also shown. The weight changes of the other components are small in comparison.

Over the range of design-point SFC's shown, all the engine parameters vary simultaneously except the turbine inlet temperature, which is held constant at 1700° F. At low design-point SFC's the engine optimizes at



(a) Design power output, 7500 horsepower, net electric.



(b) Design power output, 5000 horsepower, net electric.

Figure 5-3. - Effects of turbine cooling and intercooling on group III train engine performance. Fuel, kerosene.

high recuperator effectiveness and low compressor pressure ratio. As the design-point SFC is increased the recuperator effectiveness is decreased and compressor pressure ratio is increased. The variation in recuperator effectiveness is also shown on the recuperator weight curve. The compressor pressure ratio variation is shown in figure 5-4(b). The pressure ratio for the unrecuperated engine (not shown in the figure) is about 14. The gas temperatures at the inlet and exit on both sides of the recuperator are shown in figure 5-4(c) for the range of design points considered in figure 5-4(a). The effect of the variations in compressor pressure ratio and recuperator effectiveness is evident. Finally, in figure 5-4(d) the combustion gas flow rate for the range of designs is shown.

Because of the absence of the waste heat exchanger, the open-cycle engine is lighter than the semiclosed engine and the design-point SFC is slightly lower. The SFC is lower for several reasons. First, the optimization did not have to be constrained as was the case with the semiclosed and closed cycles where the unconstrained optimized engines resulted in waste heat exchanger dimensions which exceeded the guideline engine compartment dimensions. Second, unlike the closed and semiclosed engines, the open-cycle engine has no temperature difference between ambient air and compressor inlet temperature. Therefore, the compressor inlet temperature is lower in the case of the open-cycle engine. Finally, the open-cycle engine does not have to generate power for waste heat rejection coolant air fans as in the other cases.

These weight and design-point performance advantages of the open-cycle engine, of course, have to be balanced against the expected superiority of the closed or semiclosed engines in off-design power level operation. These comparisons will be made in later sections of this report. Also, the closed and semiclosed cycles have the advantage in that the lowest pressure within the cycle can be above atmospheric pressure, so that the size of the recuperator can be smaller than a recuperator with equivalent performance for the open cycle. However, a valid comparison of the recuperator weights for the two optimized engines is not that straightforward. As shown by the group III results, the system tends to optimize at relatively high values of recuperator effectiveness over the range of SFC's considered. For the closed and semiclosed cycles, as the recuperation is decreased

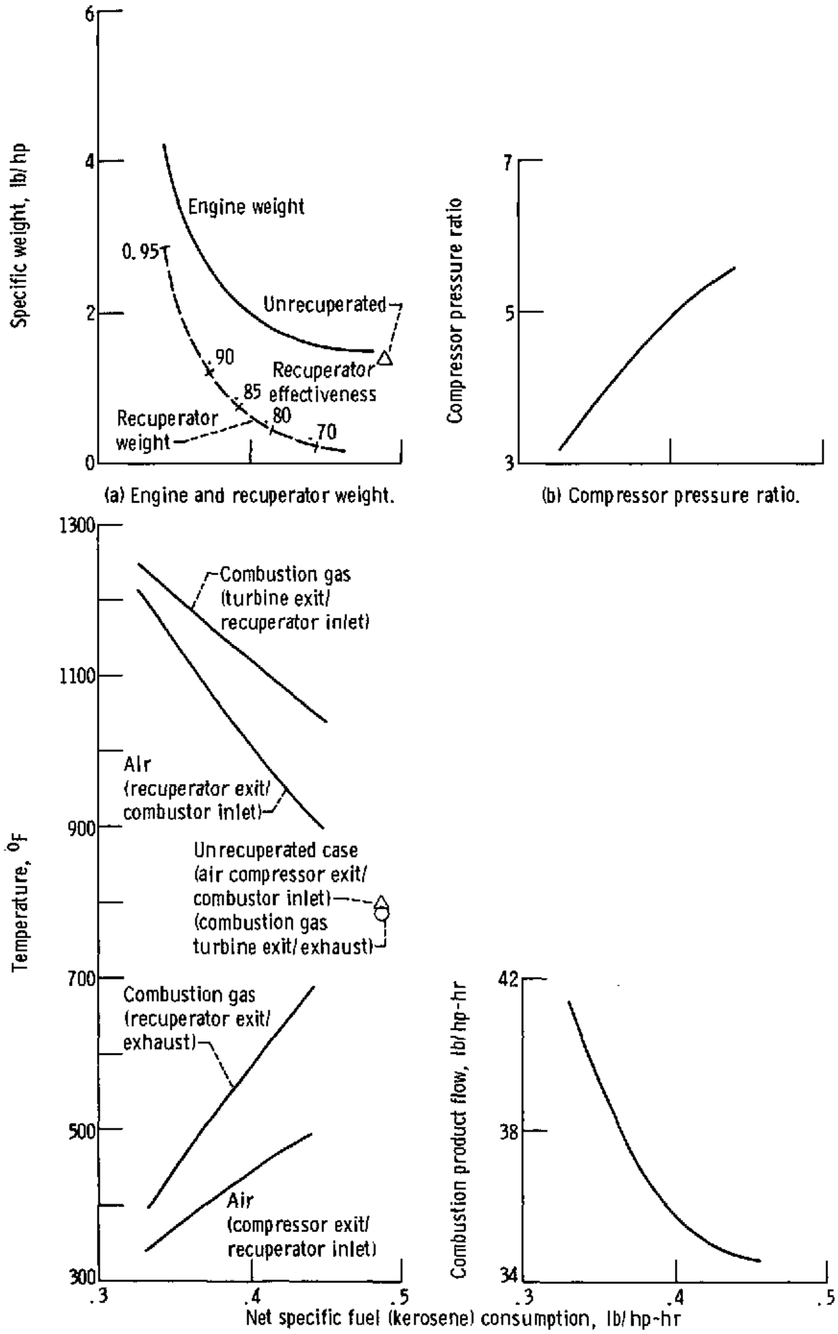


Figure 5-4. - Weight and design-point performance for group V 400-horsepower single-shaft bus engine. Turbine inlet temperature, 1700° F.

and the cycle becomes less efficient, the amount of heat rejected in the waste heat exchanger is increased and thus the weight is increased. Eventually a point is reached where the sum of recuperator and waste heat exchanger weight increases and produces a minimum in the curve of engine weight against design-point SFC. In contrast, the open-cycle engine weight decreases monotonically as the recuperator effectiveness is decreased to zero, and the minimum engine weight corresponds to an unrecuperated engine. When the recuperators of open to closed and semiclosed engines are compared, the effects of this interaction between the recuperator and the waste heat exchanger in optimizing the closed and semiclosed engines has to be considered.

Train Application

In figures 5-5 and 5-6 the engine specific weights for the 5000-horsepower two-shaft and the 7500-horsepower single-shaft train engines are given. Again, as in the case of the bus engine, the optimization of these cases did not have to be constrained in order to fit the guideline engine compartment dimensions. Also, the variation in engine weight shown in these

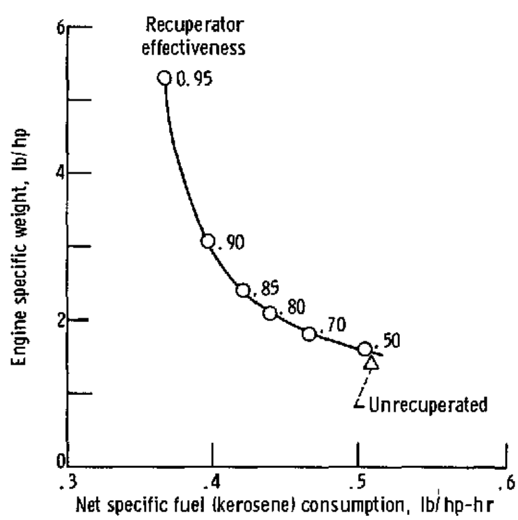


Figure 5-5. - Weight and design-point performance for group V 5000-horsepower two-shaft train engine. Turbine inlet temperature, 1700° F.

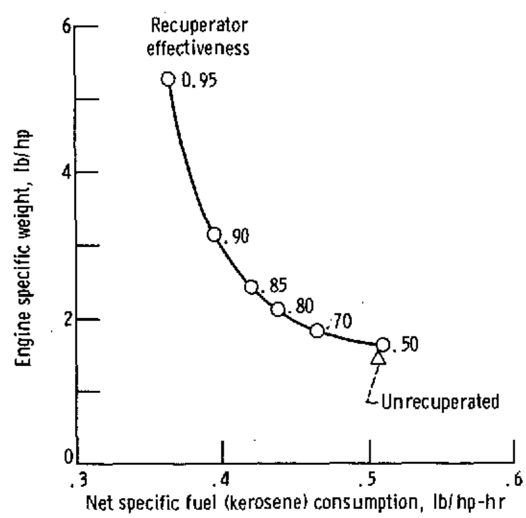


Figure 5-6. - Weight and design-point performance for group V 7500-horsepower single-shaft train engine. Turbine inlet temperature, 1700° F.

figures is again due primarily to the variation in recuperator weight. In these cases the recuperator effectiveness is indicated on the engine weight curves for reference. These engine weights include the weight of the alternator and the SFC values are based on net electric power (the effect of the alternator efficiency is included).

In figures 5-4 to 5-6 the engine weight curves for the recuperated engines are plotted only up to the SFC corresponding to that for the unrecuperated engine. With the recuperator effectiveness lower than the range plotted, the SFC for the recuperated engine is worse than that for the unrecuperated engine. This is because the amount of energy being recuperated has decreased to the point where the adverse effect of the pressure loss in the recuperator outweighs the beneficial effect of the recuperated energy. It must be remembered that when comparing the unrecuperated and recuperated engines, in addition to the design-point performance advantage shown in these figures, the recuperated engine has more favorable variation in SFC with power level changes from the design-point power. The effect of recuperator design-point effectiveness on the off-design performance of the open-cycle engine will be considered in the OFF-DESIGN PERFORMANCE section.

Thermodynamic Variations

The 5000-horsepower optimized train engine performance with a 2100° F turbine inlet temperature is compared to that of figure 5-5 (with the 1700° F turbine inlet temperature) in figure 5-7. The effect of using compressor exit air for turbine cooling is included for the 2100° F turbine inlet temperature design. The higher turbine inlet temperature results not only in lower SFC but in lower engine weight. For equal effectiveness, the recuperator for the higher turbine inlet temperature is lighter since the gas flow rate to produce the same power is lower for higher turbine inlet temperature. This is the primary reason for the lighter weight shown in figure 5-7 for the higher temperature engine.

As the recuperator effectiveness for the open-cycle engine increases the temperature of the primary air entering the combustor increases (as shown in fig. 5-4(c) for the bus engine). If a conventional combustor is used this results in more difficulty in controlling NO_x emissions. If a

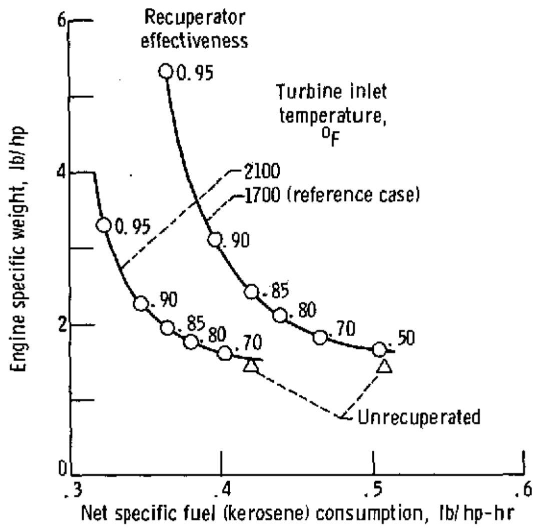


Figure 5-7. - Effect of increased turbine inlet temperature for group V 5000-horsepower two-shaft train engine.

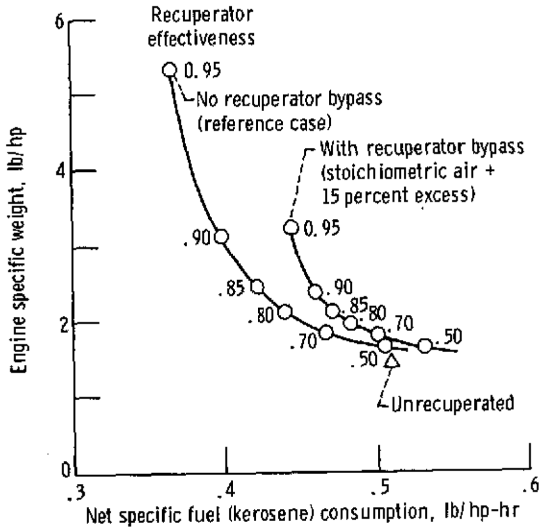


Figure 5-8. - Effect of recuperator bypass on group V 5000-horsepower two-shaft train engine. Turbine inlet temperature, 1700° F.

surface combustor in which the primary air and fuel are premixed prior to entering the combustion zone is used, the air temperature must be kept below the preignition limit. One way of keeping the combustor primary air inlet temperature at a lower level when a high effectiveness recuperator is used is to have some or all of the primary air bypass the recuperator; the secondary air (a large part of the total air flow) would still flow through the recuperator to remove heat from the turbine exhaust flow. In figure 5-8 the effect of this recuperator bypass is shown for the 5000-horsepower engine. The engine weight curve, optimized assuming the primary air bypasses the recuperator, is compared to the reference case (from fig. 5-5). Since only part of the flow is recuperated, the SFC is higher, and since the flow through the recuperator is smaller, the recuperator and hence the engine is lighter for the bypass case. Figure 5-8 shows only the performance at design-point power level, which is assumed here to be maximum power. At lower power levels, depending on how the engine is operated off design, it may not be necessary to bypass the recuperator to keep the temperature of the primary air at acceptable levels, the full recuperation might then be used. In such a case the off-design performance would differ somewhat from the case where bypass is used at all power levels.

Off-Design Performance

In order to make a valid comparison of semiclosed- (group III) and open-cycle (group V) engines it was necessary to analyze the off-design performance for several assumed missions. This off-design performance information was used in conjunction with several mission profiles to determine mission fuel requirements for the performance comparisons in section 2.

SEMICLOSED BRAYTON CYCLE ENGINES

The group III engine, a semiclosed Brayton power system, should exhibit off-design characteristics similar to the closed Brayton (see appendix I). The closed and semiclosed Brayton power systems can match the power level demand by adjustment of the inventory (pressure level) in the system while maintaining the design-point temperatures within the system and the design-point turbomachinery speeds. When this is done the system efficiency for closed Brayton engines is almost unaffected as the power level is reduced. A closed Brayton engine requires a separate inventory adjustment (or pressure level control) system. The group III engine, however, with appropriate control of the turbocharger inherently includes the means for system pressure level control (see appendix C for a discussion of group III engine controls).

The off-design performance of the group III single- and two-shaft engines was analyzed. The two-shaft system supplies power to two alternators. The auxiliary alternator drives the fixed loads on the vehicle and the required fans on the engine. The vehicle drive alternator supplies the motive power for the vehicle. The vehicle drive alternator, therefore, must either be able to operate at various speeds and power levels, or be driven through a variable speed transmission. The two-shaft engine was considered for use with the train vehicle. For the single-shaft engine both auxiliary and vehicle motive power are derived from the same shaft. A variable speed transmission would be used to control the speed of the separate alternators. The single-shaft engines were considered for the train and bus applications.

The off-design performance of the group III systems was calculated using a model based on IBM's modeling program CSMP. A description is included in reference 1, and the system model is discussed in appendix C. As discussed in section 3 many variations of the semiclosed cycle are possible. But in order to keep the number analyzed to a reasonable limit, the off-design computer code was set up to analyze only the variations in figure 3-5 (p. 50) which has a single exhaust point. For most ranges of operating conditions, the work of the exhaust gas turbine in this case would exceed the requirements of the air supply compressor so that there would be a net output power from the turbocharger shaft. In order to avoid the necessity of providing a means of making use of this turbocharger output it was decided to consider another cycle variation (see fig. 3-5(b)) in the engine design-point analysis of section 5. The cycle arrangement considered in section 5 included multiple points of rejection of gas from the system to allow matching the turbine and compressor power on the turbocharger shaft independent of pressure level. The off-design characteristics calculated for the cycle of figure 3-5(a) were assumed to apply to the variation considered in the engine design-point analysis. This was considered to be sufficiently accurate for the purposes of this study. For each engine the off-design performance at reduced power was calculated for three design points. The resulting curves of SFC as a function of shaft speed and power are presented in the following section.

For this analysis the turbine inlet temperature and compressor inlet temperatures are assumed constant at 1700° and 80° F, respectively. Turbine inlet temperature is controlled by adjusting the diluent fuel ratio. Compressor inlet temperature is a function of ambient air temperature and waste heat exchanger effectiveness.

Two of the major parameters that affect system efficiency are recuperator effectiveness and the relative pressure loss ($\Delta P/P$). For fixed machinery speed, as the pressure (power) decreases these parameters increase. System efficiency increases as the recuperator effectiveness increases, but it decreases as the relative pressure loss increases. At very low pressures the relative pressure loss becomes so great that the system is not self-sustaining. The overall effect on efficiency depends on the rate of change of each of these parameters with changing power level.

System efficiency is also affected by the turbomachinery efficiencies. This information is included in the turbomachinery maps that are used in this simulation. The efficiencies of the turbocharger components were assumed constant because of the complexity required to include them in the simulation.

Bus Application

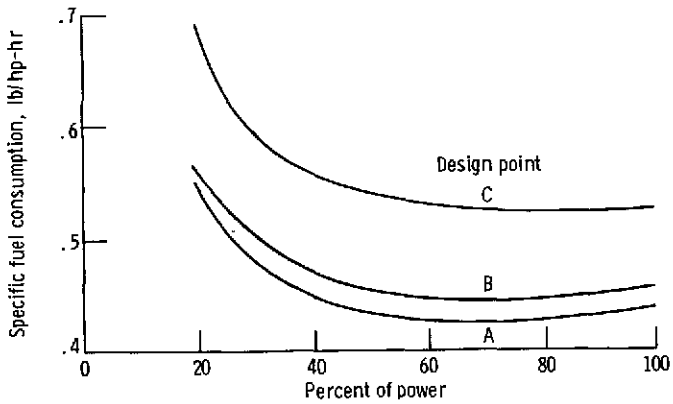
The off-design performance for the bus engine is shown in figure 5-9(a) for three design points. The design points correspond to three points on the curve of optimum design shown in figure 5-1. This curve is reproduced in figure 5-9(b) showing the design points used.

The SFC decreases slightly as power decreases. This reflects the effect increasing recuperator effectiveness has on system efficiency. At 50 percent of design power, the SFC begins to increase sharply. This is caused by the increase in relative pressure loss. The curves for different design points are all similar in shape.

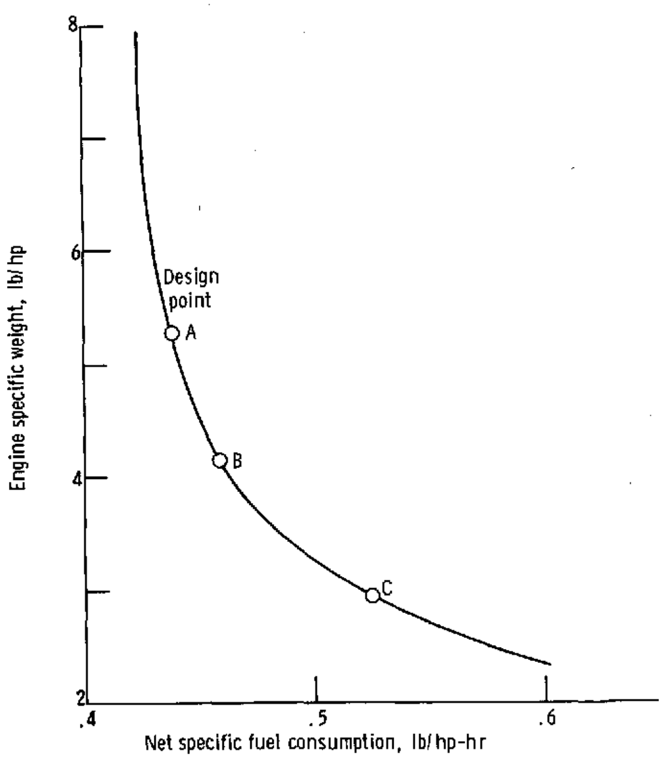
Train Application

The performance for the single-shaft train engine is shown in figure 5-10(a) for the design points shown in figure 5-10(b). This figure is reproduced from figure 5-2. Also indicated in this figure is the effect of turbomachinery speed on SFC. These curves are similar to the curves for the bus engine. The effects of the design point are greater in this system because of the higher pressure level. Specific fuel consumption is a minimum at about 50 percent of design power and increases as power decreases further. The detrimental effect of speed on SFC shown here illustrates the penalty for changing engine speed and indicates why constant engine speed is desired.

The two-shaft engine has a constant-speed shaft and a variable-speed shaft. The results of the off-design analysis for this system are shown in figure 5-11(a) for the design points indicated in figure 5-11(b). This figure is reproduced from figure 5-2(b). Also shown in this figure is the effect of varying vehicle drive shaft speed. As was shown previously, the recuper-

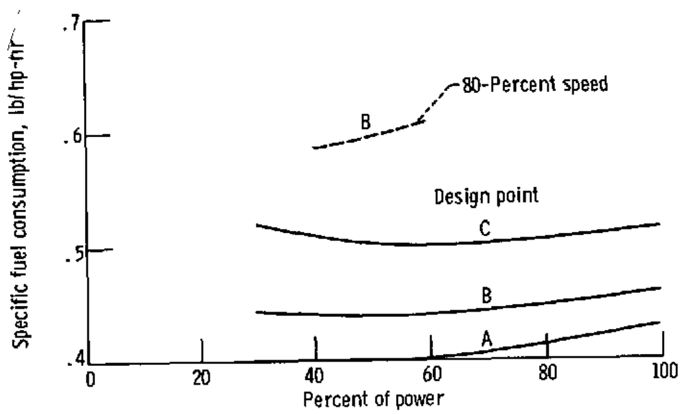


(a) Specific fuel consumption against percent of power for various design points.

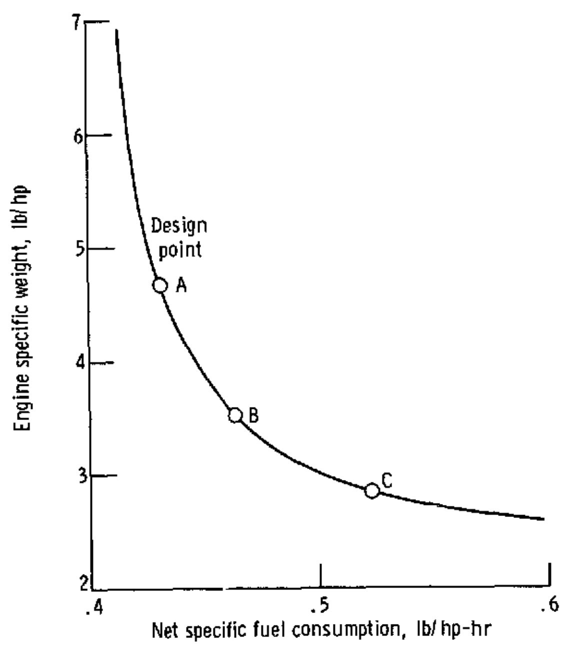


(b) Engine design points used for off-design analysis.

Figure 5-9. - Group III single-shaft bus engine.

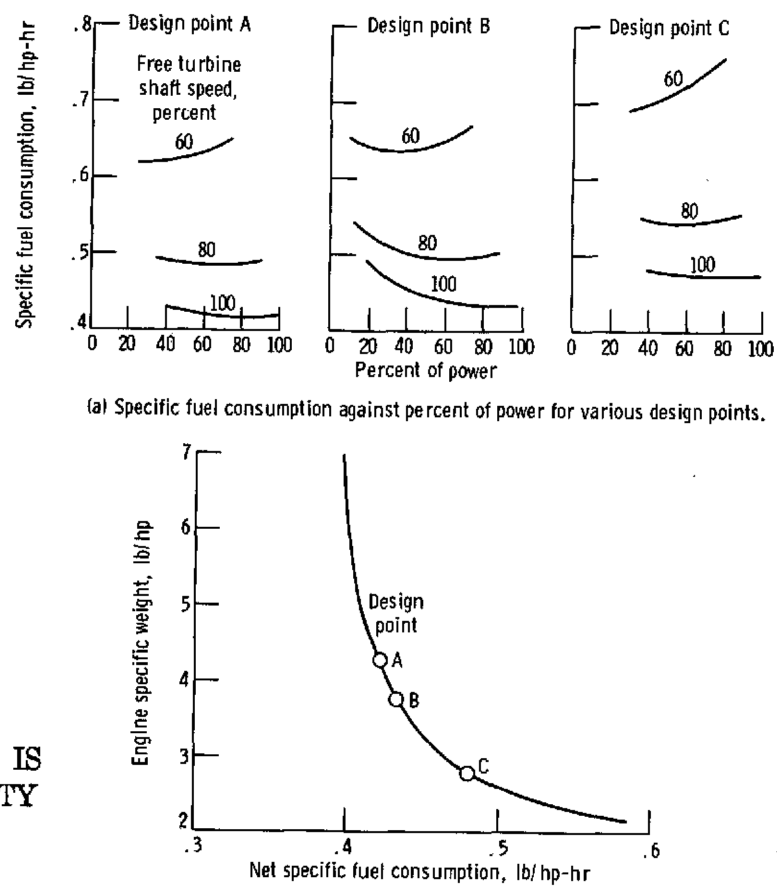


(a) Specific fuel consumption against percent of power for various design points.



(b) Design points used in off-design analysis.

Figure 5-10. - Group III single-shaft train engine.



(a) Specific fuel consumption against percent of power for various design points.

(b) Design points used in off-design analysis.

Figure 5-11. - Group III two-shaft train engine.

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ator keeps SFC relatively flat to about 50 percent of design power. The sensitivity of SFC to speed is less for the two-shaft engine.

OPEN BRAYTON CYCLE ENGINE

Since the compressor inlet pressure for the open-cycle (group V) engine is ambient pressure, reducing the power requires either reducing the shaft speed or the turbine inlet temperature. The turbine inlet temperature was a dependent variable in this cycle for a given shaft speed and power level. The maximum temperature considered without turbine cooling was 1700° F. The compressor inlet temperature was held constant at 80° F.

The off-design performance of the group V engines was calculated using GENENG, a computer code for calculating one- or two-shaft turbojet and turbofan cycles. A detailed description of GENENG can be found in reference 2. A description of the modifications made to the code to simulate the group V engines and the techniques the code uses to match performance of the components for off-design are presented in appendix C. For each engine (bus, single-shaft train, and two-shaft train) off-design performance was calculated for three design points. The resulting curves of the SFC as a function of shaft speed and horsepower are presented in the following section.

Bus Application

The off-design performance characteristics of the single-shaft bus engine are shown in figure 5-12(a) for design points shown in figure 5-12(b) which are reproduced from figure 5-4(a). The design turbine inlet temperature limit of 1700° F is also shown. If the engine is operated at a constant turbine inlet temperature of 1700° F, the SFC remains close to the design value to about 50 percent power and then begins to increase rapidly. The initial flatness of this curve can be attributed to the improvement in recuperator effectiveness with decreasing engine air flow. To obtain this performance requires that shaft speed decrease with power level. This requires an infinitely variable speed transmission. A decrease in shaft speed results in a decrease in pressure ratio and, therefore, an increase in recuperator hot side inlet temperature. This temperature increase must be considered when operating at low power levels to avoid materials problems in the recuperator. For design point A in figure 5-12 the recuperator inlet temperature increases from 1140° F at design power to 1540° F at 10 percent of design power.

Train Application

The off-design characteristics of the single-shaft train engine shown in figure 5-13 differ only slightly from the bus case. The three design points chosen are shown in figure 5-13(b) which is reproduced from figure 5-6. The differences between the single-shaft bus and train cases can

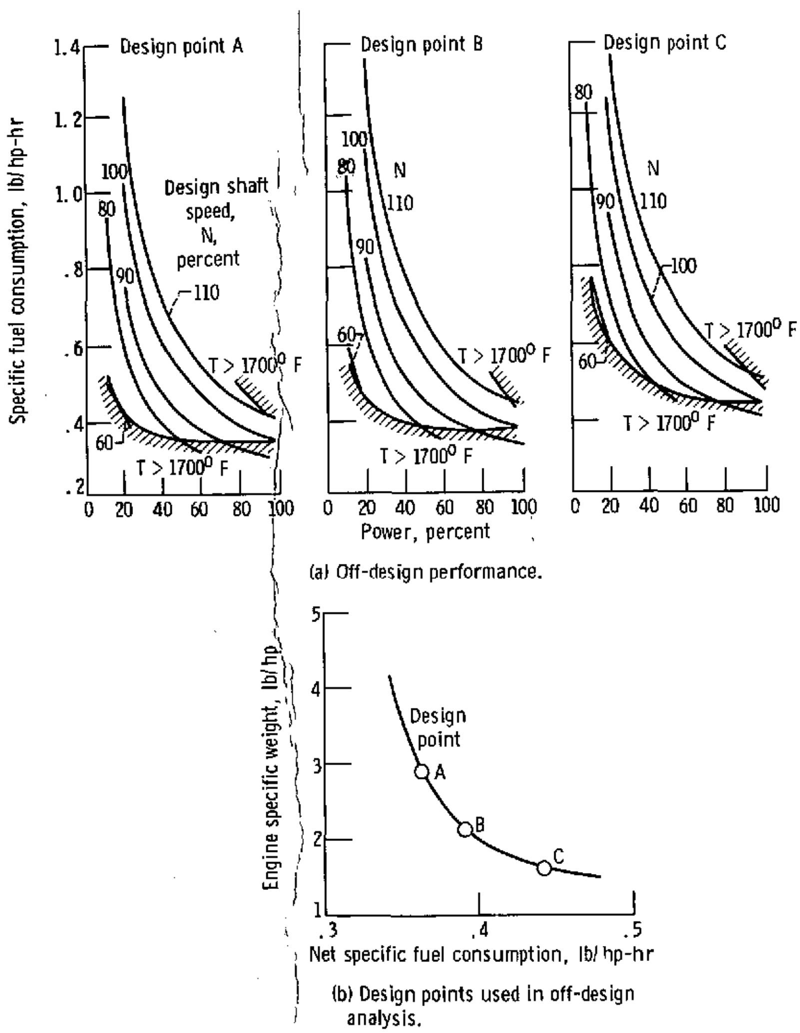
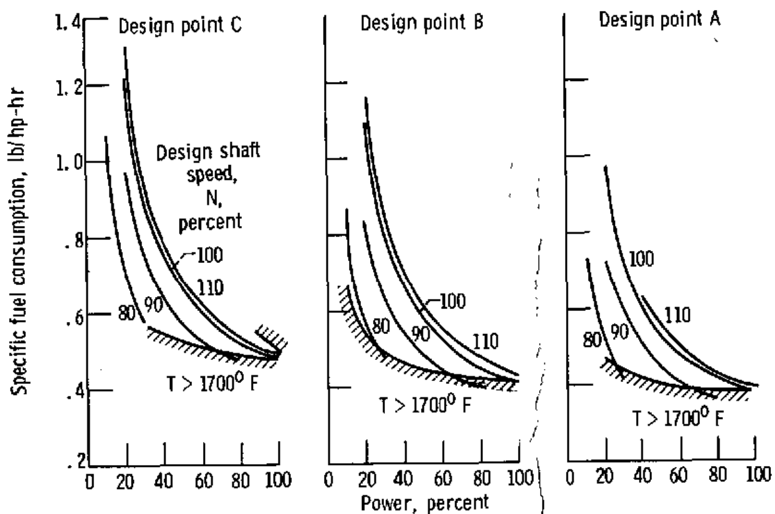
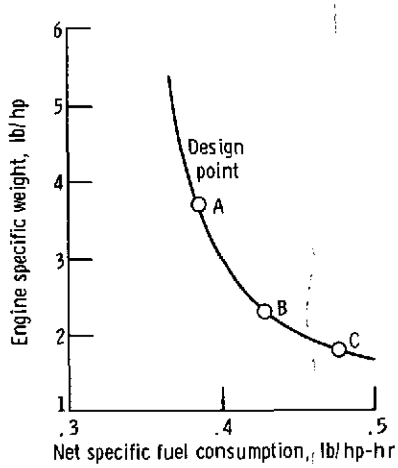


Figure 5-12. - Single-shaft group V bus engine.



(a) Off-design performance.



(b) Design points used in off-design analysis.

Figure 5-13. - Single-shaft group V engine.

be attributed not only to their being different design points but also to the compressor performance maps used. A radial compressor design was chosen for the bus engine and an axial compressor for the train. If the engine is operated at a constant turbine inlet temperature of 1700° F, the SFC remains close to the design value to about 50 percent power and then begins to increase rapidly. Again, to obtain this performance requires adjusting shaft speed as power level decreases which is permitted by a variable speed transmission. The effect of a decrease in speed on recuperator inlet temperature must also be considered in this case.

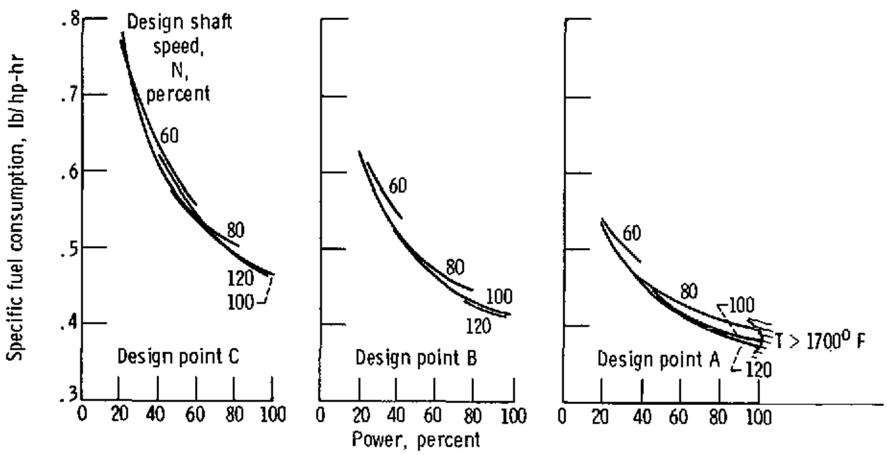
The performance of the two-shaft engine is shown in figure 5-14. Off-design characteristics of this engine are considerably different from those of the preceding cases. The constant speed lines for the power turbine are less steep and there is not the temperature limitation for the same speed ranges. The only way to decrease the power level along the minimum SFC path for this engine is to decrease turbine inlet temperature. For this reason SFC increases more rapidly as power decreases.

Concluding Remarks

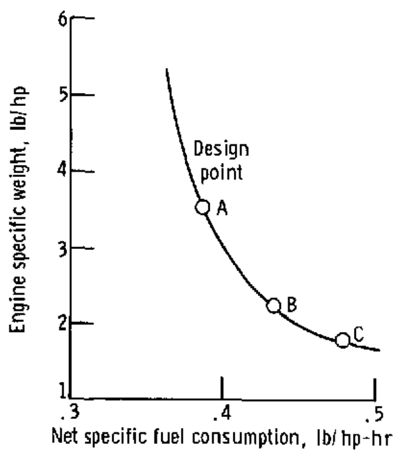
In the conceptual design phase the semiclosed Brayton cycle was reassessed in more detail and compared with the open Brayton cycle engines. Kerosene was used as the fuel. Final system selection was dependent not only on the design-point engine performance but also on the engine off-design performance and engine operation over trip profiles (see appendix F).

The design-point results show that the open-cycle engine performance was better and the engine weight was lighter than the semiclosed cycle engine. The results of the design-point performance analysis were used as input to the off-design analysis.

The off-design results illustrate that the best performance for the group III engines is attained when power level is changed by changing system pressure level while maintaining turbine inlet temperature and engine speed at their design-point values. In order to hold the engine speed constant, independent of load-speed requirements, the use of an infinitely variable transmission was assumed.



(a) Off-design performance.



(b) Design points used for off-design analysis.

Figure 5-14. - Two-shaft group V engine.

The group V off-design results show that if the infinitely variable transmission is used for the single-shaft engine, so that engine speed can be varied with power level (independent of load speed), the turbine inlet temperature can be maintained at the design-point value. In this case, the group V off-design performance becomes comparable to that of the group III engine. However, the recuperator inlet temperatures would increase with decreasing power level. Depending on the turbine inlet temperature and turbomachinery characteristics this could place a lower limit on the power level for this mode of operation due to temperature capabilities of the recuperator.

Efficiency of an infinitely variable speed transmission is a function of the difference between the input- and output-shaft speeds. Therefore, for low load-speed operation the group V engine could result in a better overall efficiency since it could run at lower engine speed without as much of a penalty in SFC.

The results of this portion of the study are used along with mission profile data (from appendix F) to calculate an overall mission fuel requirement. The final engine performance comparisons are made in section 2.

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APPENDIX A

HEAT EXCHANGERS

The heat exchangers required in gas turbine systems and their functions are presented in appendix I. Table A-1 presents the heat exchangers required for each of the engine groups studied. The combustors and combustor - heat exchanger are described in appendix D.

The heat exchangers in a gas turbine system occupy the major portion of the engine volume and also represent a large fraction of total engine weight. In addition, routing the ducting and heat exchanger placement constitute major tasks in engine layout and component integration. To limit the volume of the heat exchangers, mobile gas turbine applications require heat-transfer surfaces with a small passage dimension (low hydraulic radius) and closely spaced fins. These compact heat-transfer surfaces, representative of those described by Kays and London (ref. 1),

TABLE A-1. - ENGINE GROUP HEAT

EXCHANGER REQUIREMENTS

Engine group	Preheater	Heat-source heat exchanger	Recuperator	Waste heat exchanger	Condenser
I	Yes	Yes	Yes	Yes	No
II	Yes	No[a]	Yes	Yes	No
III	No	No[a]	Yes	Yes[b]	No
IV	No	No[a]	Yes	No	Yes
V	No	No[a]	Yes	No	No

[a]These groups use a combustor or combustor - heat exchanger.

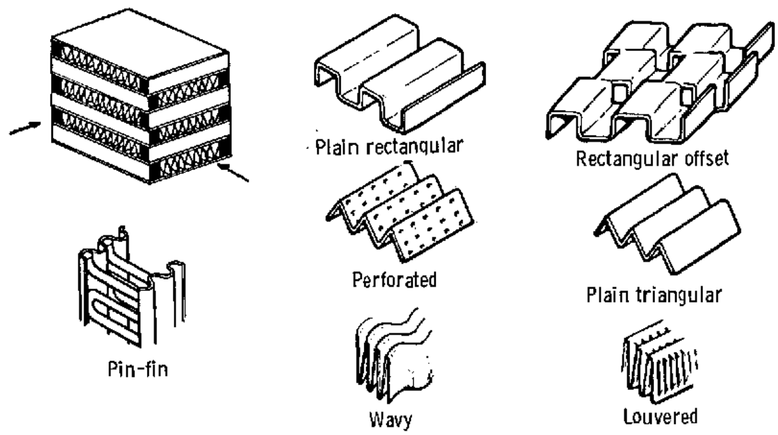
[b]Group III waste heat exchanger does have condensation of water vapor from combustion products but it is not specifically a condenser.

have heat-transfer area densities of the order of 300 to 500 square feet per cubic foot of volume.

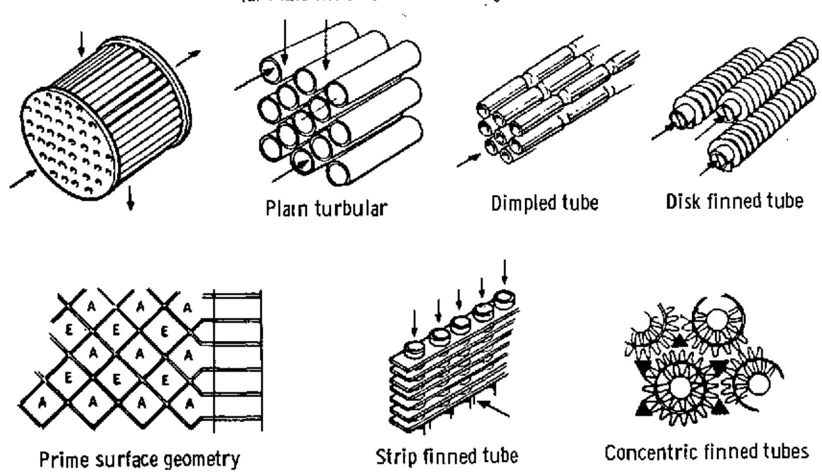
The heat exchangers considered in this study are of the fixed boundary type rather than the rotary type. Rotary regenerators were not considered because of the uncertainty of size effects and the lack of definitive data on seal life, seal leakage, and drive power requirements over the engine power range. It is doubtful that a rotary regenerator could effectively function at the 7500-horsepower limit. The mass of a single unit would be prohibitive, while paralleling a large number of small ones would be almost impossible within a reasonable engine envelope. However, a number of regenerated engines are operating in the range of 400 horsepower. Although a more comprehensive engine development program would certainly consider regenerators in greater depth, it was felt that the use of a fixed or rotary heat exchanger would not affect system selection.

Surface Geometries

Various types of heat-transfer surfaces are shown in figure A-1. There are basically two types of surfaces, tubular and plate fin. In general, plate-fin heat exchangers are brazed and pressure limited, while tubular units are welded and capable of high pressure containment. Brayton cycle space power systems design studies (refs. 2 and 3) have shown that the rectangular offset, plate-fin geometries yield minimum weight and volume gas to gas recuperators for a wide range of design conditions and working fluids. Since all the heat exchangers transfer heat from gas to gas (low pressure vapor on one side of group IV condenser), plate-fin geometries were used. Off-set rectangular surfaces (as described in ref. 3) were used for the recuperator and gas side of the heat-source heat exchanger and waste heat exchanger. A plain perforated-fin matrix, shown in reference 4 to offer improved performance over offset rectangular and louvered-fin surfaces in automotive radiator applications, was selected for the preheater, the combustion gas side of the heat-source heat exchanger, and the air side of the waste heat exchanger. This should be an optimum



(a) Plate fin extended surface geometries.



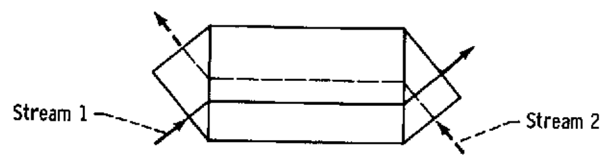
(b) Tubular, prime, and extended surface tubular geometries.

Figure A-1. - Heat exchanger surface geometries.

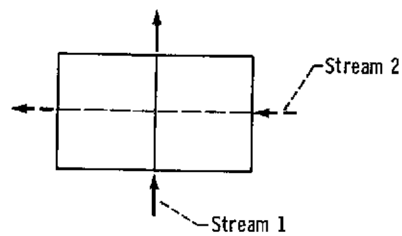
surface since the limiting heat transfer fluid is air at approximately atmospheric pressure in all cases.

Flow Configurations

Counterflow, simple crossflow, and cross-counterflow heat exchanger configurations were considered in this study.



(a) Counterflow core with triangular ends.



(b) Crossflow.

Figure A-2. - Flow configurations.

COUNTERFLOW

Figure A-2(a) illustrates the flow arrangement of a counterflow heat exchanger. Counterflow is the most efficient configuration; it requires the least heat-transfer area for a given effectiveness. However, since both fluid streams share inlet and exit faces, manifolding and headering are complicated. The counterflow core is the central rectangular section. It can be seen that the core flow length is identical for both fluid streams. Even allowing for geometry variations on opposite sides of the core, pure counterflow is best applied to heat exchangers with fairly well matched streams such as the preheater or recuperator. To prevent the mixing of the streams at the opposite faces, triangular end sections are included. The end sections separate and turn the fluids by extending the plates beyond the end of the core section. In low pressure systems where the heat exchangers tend to have large frontal areas and short flow lengths, end sections can become a major part of total heat exchanger weight. Therefore, the end section weight and pressure loss are included in system optimization and heat exchanger design. No credit is taken for heat transfer in the end sections since the fluids are in crossflow and low performance plain fins are used.

CROSSFLOW

Single-pass crossflow, as shown in figure A-2(b), offers simple manifolding since the fluid streams are at right angles and enter and exit through adjacent faces. It is a useful arrangement when the fluids are not well matched (i.e., different heat-transfer rates, densities, or flow rates) since the flow faces and lengths are more independent than in counterflow. The heat-source heat exchanger and waste heat exchanger have crossflow construction to better accommodate the large differences in pressure level, heat-transfer area required, and pressure loss between the streams.

CROSS- COUNTERFLOW

When a high effectiveness is required, the single-pass crossflow arrangement, although simple and convenient for compact core design, suffers a severe penalty in required area. The efficiency of the crossflow arrangement can be improved by adding additional crossflow passes in series in an overall counterflow configuration. A three-pass, cross-counterflow heat exchanger is shown schematically in figure A-3. For a large number of passes, cross-counterflow approaches counterflow in required heat-transfer area. In general, the passes are not physically separated as shown but are achieved by baffling or turning the flow internally (fig. A-4). The additional pressure losses required to turn the flow must be accounted for in the design. To reduce volume in the turning sections the higher density side is multipassed because it is less sensitive to pressure loss. The low pressure or air side flows through in a straight pass.

Packaging

Early in this study it was found that the crossflow heat exchangers sized for the minimum system weight conditions did not conveniently fit

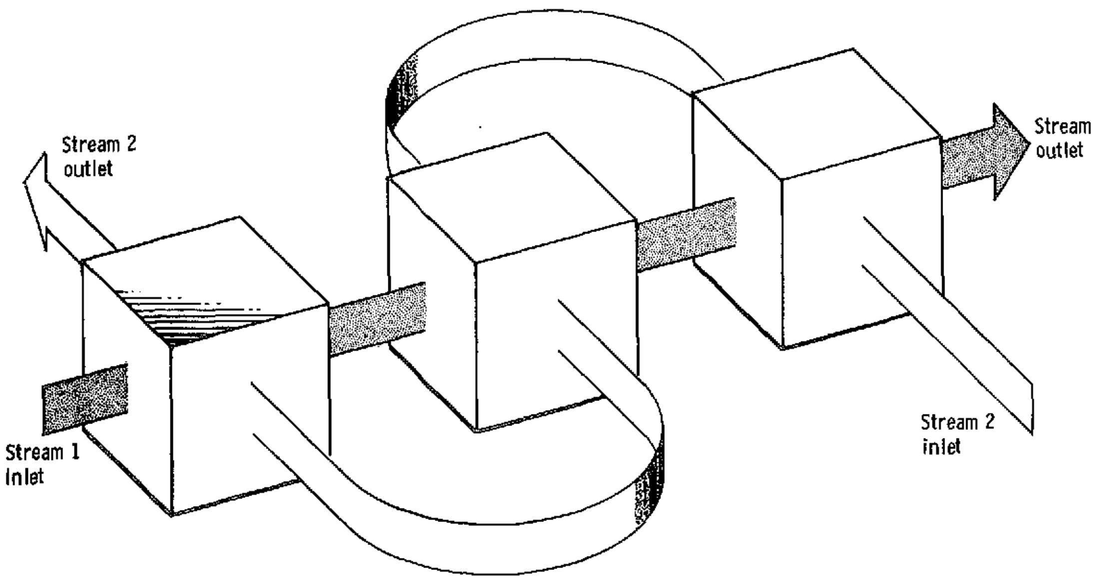


Figure A-3. - Three-pass cross-counterflow heat exchanger.

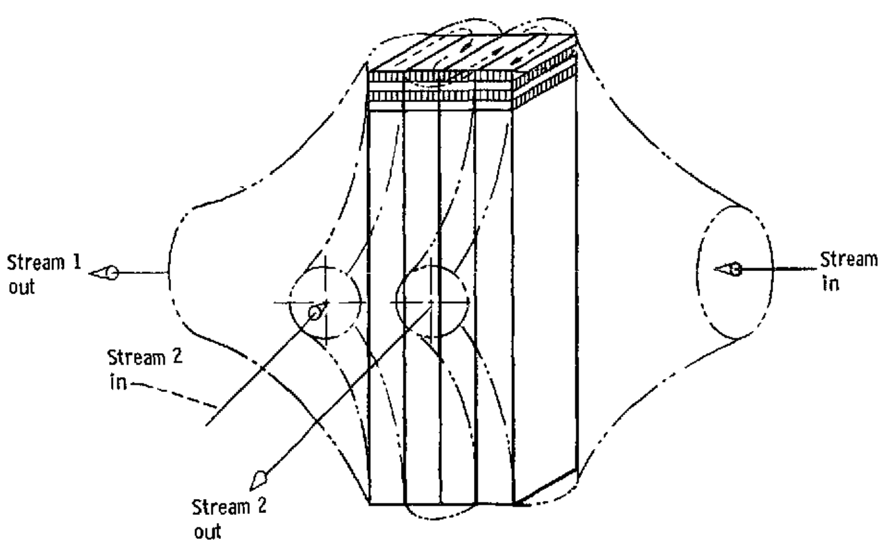


Figure A-4. - Multipass cross-counterflow heat exchanger.

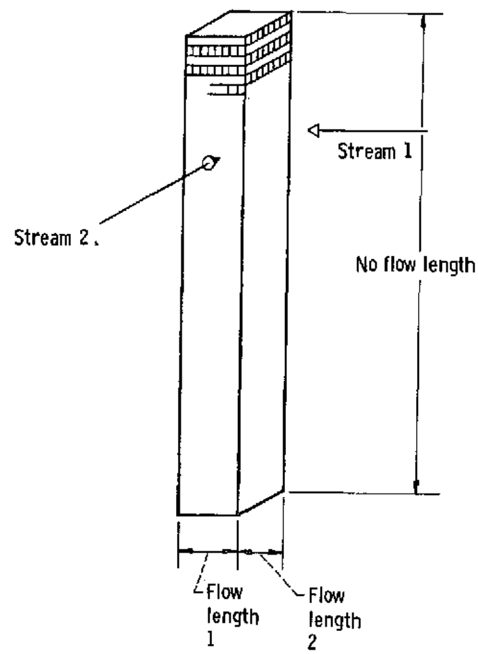


Figure A-5. - Single module single pass crossflow.

into a reasonable engine envelope. This was true for both the heat-source heat exchanger and the waste heat exchanger. Figure A-5 illustrates the problem in aspect ratio (no-flow length divided by minimum-flow length) for a single-pass crossflow unit. A crossflow heat exchanger is characterized by a hot flow length, a cold flow length, and a no-flow length or stack height. The no-flow length is the dimension along which no-flow occurs; in a plate-fin heat exchanger it is the stack height of the passages. As illustrated in the figure, the stack height tends to be very much larger than either flow length especially at high values of effectiveness and low allowable pressure drops.

In order to bring the heat exchangers within the selected engine envelope, a packaging concept with minimal effect on performance was developed. A crossflow heat exchanger can be split along the no-flow length to yield a number of parallel modules as shown schematically in figure A-6. The low pressure stream flows through in a single, straight pass as in the unpackaged unit while the high pressure stream does require additional entrance and exit turn losses. Turning the high pressure side is favored

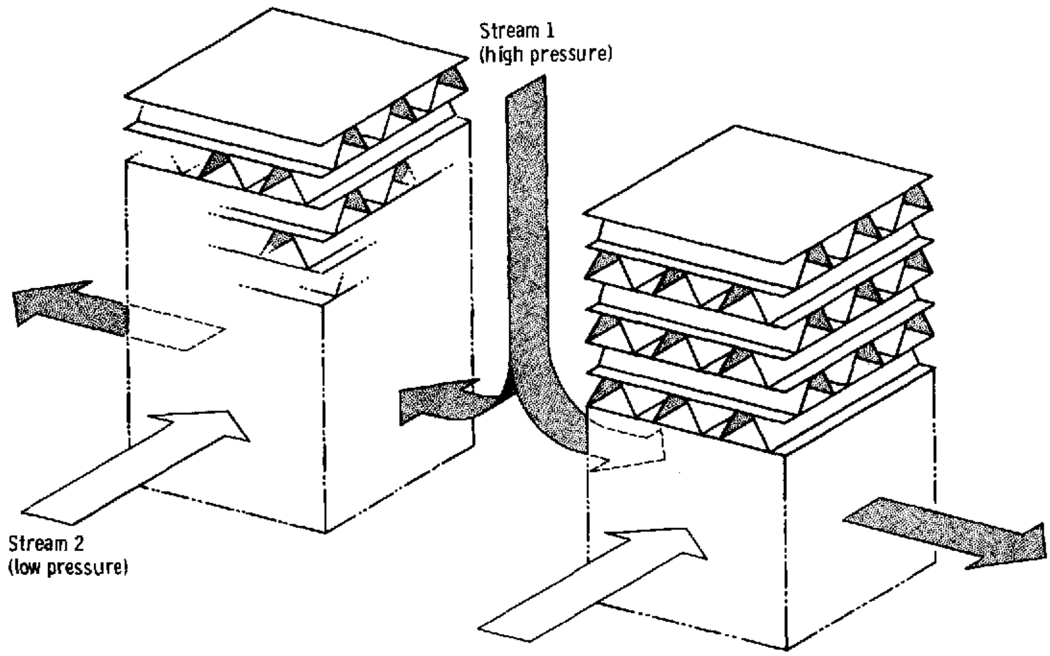


Figure A-6. - Crossflow heat exchanger packaging.

because overall engine performance and heat exchanger size are less sensitive to the pressure drop on the high pressure side. For packaging, an even number of core modules is assumed to make efficient use of manifolding as illustrated in figure A-7. The major problem is in ducting the high pressure gas flow in and out of the core with the required distribution and collection manifolds. Ducting for the air (low pressure) side is not a problem.

For the counterflow arrangement, the height to width aspect ratio of the core face is picked to establish a reasonable end section configuration. The counterflow core dimensions are defined in figure A-8.

Packaging the counterflow recuperator into an engine was not a problem because the relatively high working fluid pressures reduce the volume. A recuperator aspect ratio of two was assumed. A typical recuperator configuration with triangular end sections is shown in figure A-8. The flow paths for the high and low pressure stream are shown in figure A-9. Offset rectangular fins are used in the core section. To reduce friction

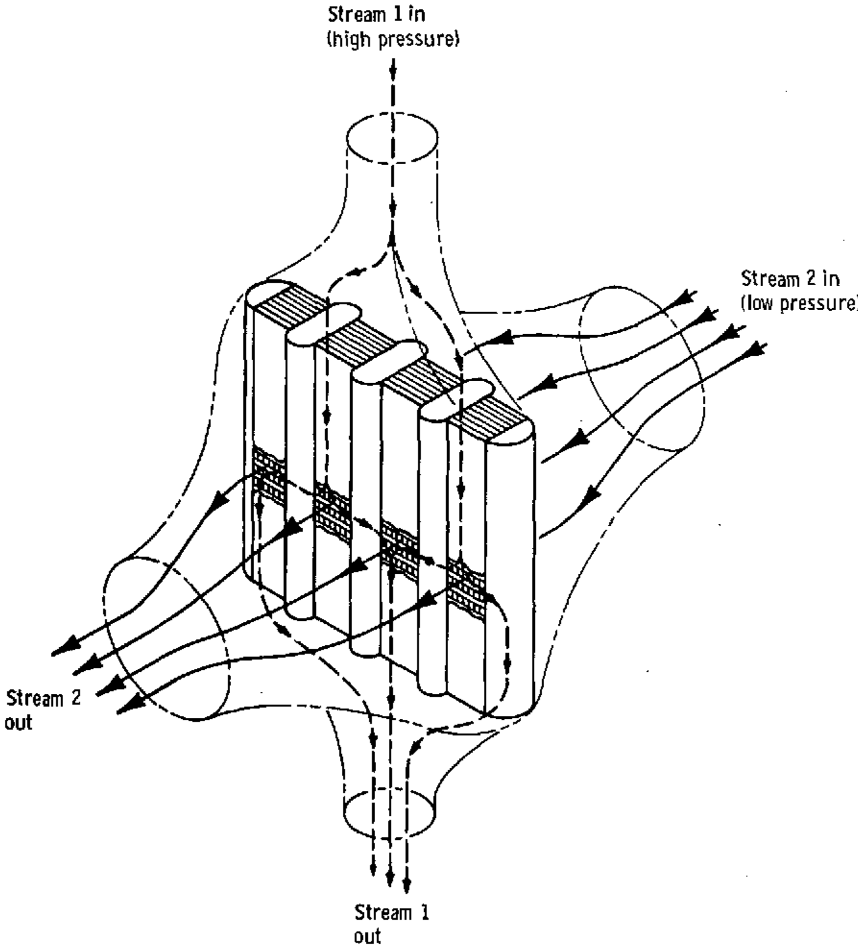


Figure A-7. - Modular crossflow heat exchanger.

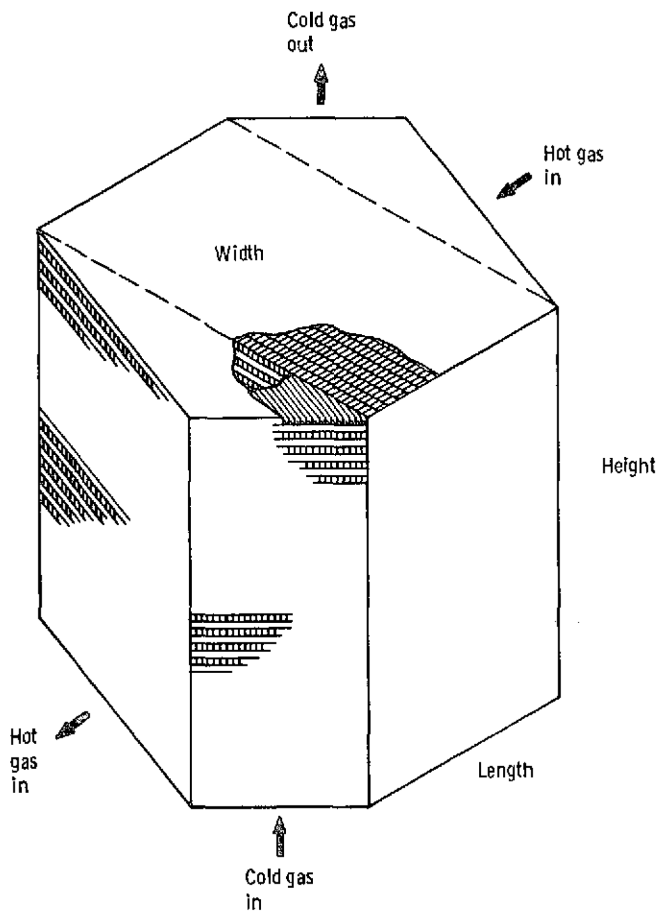
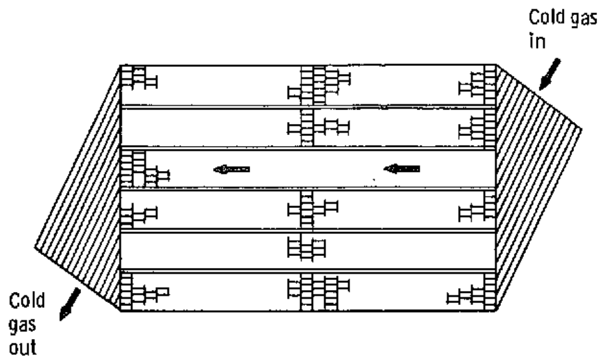
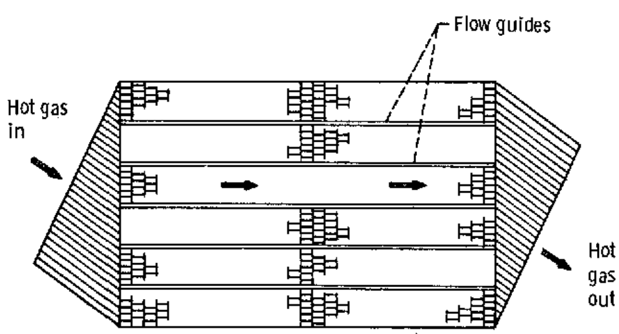


Figure A-8. - Plate-fin counterflow recuperator.



(a) Cold side flows (high pressure).



(b) Hot side flows (low pressure).

Figure A-9. - Recuperator core and end section flow paths.

pressure drop in the end sections, plain rectangular fins (no offset) are used. Guide strips running the length of the core are used to prevent any crossflow which would be detrimental to performance.

The preheater, because of the relatively low gas pressure level and allowable pressure drop, tends to have a large frontal area and short flow length. To reduce the end section weight relative to the core, an aspect ratio of six was assumed and two triangular end sections were used at each end as shown in figure A-10. This arrangement complicates manifolding and collecting the streams but does yield a better weight and pressure drop distribution between the end sections and core. The flow paths for the two sides of the preheater are shown in figure A-11. Plain perforated fins are

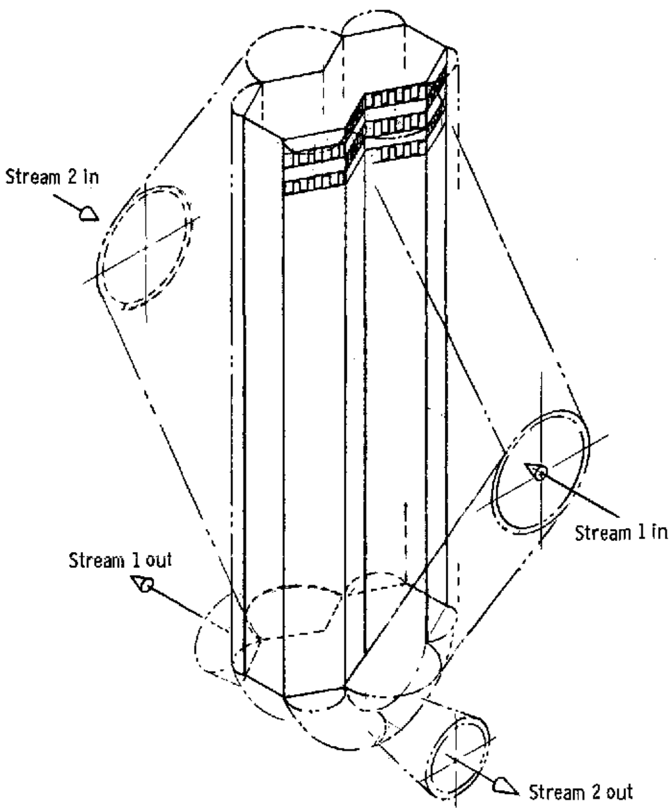


Figure A-10. - Counterflow air preheater.

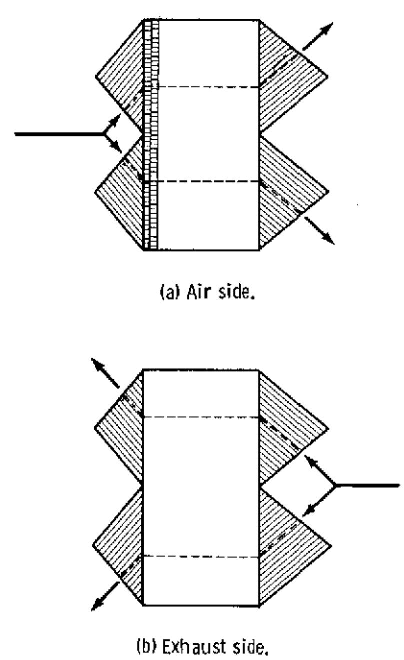


Figure A-11. - Preheater core and end section flow paths.

used in the core and end sections. Flow guides are also required in the core section.

Although the group IV hydrogen-oxygen engine is attractive as an advanced nonpolluting system, it is severely hampered by condenser (waste heat exchanger) packaging. The waste heat exchanger in the group IV engine is actually an air-cooled steam condenser. Because the steam side condensation coefficient is so much larger than the air-side heat-transfer coefficient, the required heat-transfer area is independent of flow configuration and is determined by air-side performance. A plate-fin core similar to configurations presented in reference 4 was assumed since the stream conditions are quite similar to those in an automotive steam engine.

It was found that the low condensing temperature on the steam side limited the driving temperature between the streams. Therefore, a relatively large heat transfer area was required. A typical condenser core layout is shown in figure A-12. Characteristically, the core has a large air-side face area and a short air-side flow length. The no-flow length is again the largest dimension. While in the other engines packaging was accomplished by taking advantage of the density of the high pressure stream, in the group IV engine this was not possible. In this engine, efficiency is a function of condenser temperature which is in turn dependent on pressure level. Increasing the steam-side pressure to increase density for better packaging also increases the temperature which reduces efficiency. It was found that the optimized group IV engine required a condenser that could not be packaged into a reasonable engine envelope. Dividing the core into a number of parallel modules as was previously discussed did not result in a better overall package. For reasonable pressure losses, the low steam-side density required large spacing between modules.

Condenser size and the problems it introduces in engine-vehicle integration were identified as one problem in the practical application of the group IV engine. However, a more novel approach to waste heat rejection might improve the possibilities. For example, a rotary condenser or augmenting the air-side heat transfer by the injection of a condensate might result in a smaller, more convenient condenser package.

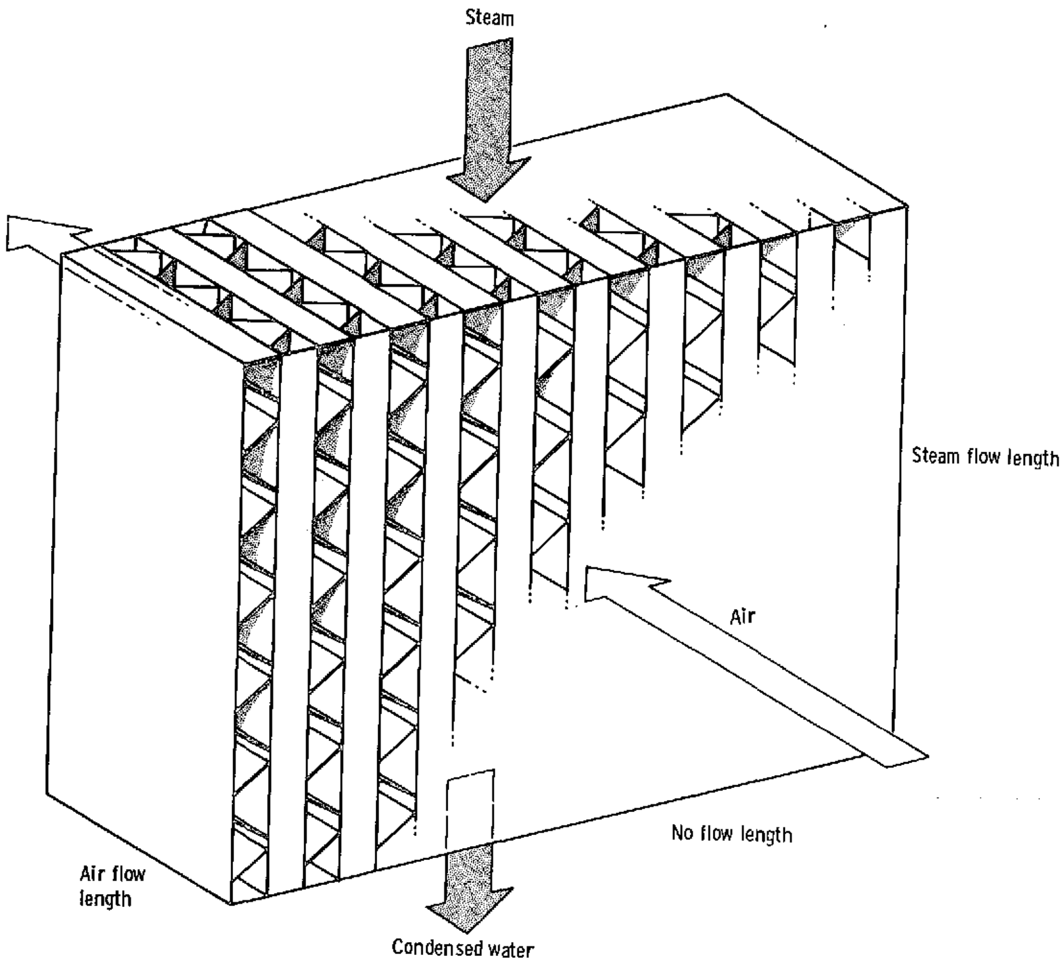


Figure A-12. - Group IV condenser.

Materials

Conventional aerospace heat exchanger materials were considered in this study. The high temperature heat exchanger materials were assumed to be superalloys or stainless steel. Aluminum was assumed for the low temperature waste heat exchanger. The long-term effects of the combustion products in the group III engine on aluminum were not evaluated. These materials were selected as best meeting the severe engine requirements, but a much more complete and comprehensive structural analysis would be necessary for final materials selection.

Ceramic materials offer the necessary high temperature oxidation resistance and strength, but little has been done to demonstrate the practicality of a ceramic heat exchanger. However, there is some experience with ceramic rotary regenerators in automotive and truck gas turbine engines.

Several companies have demonstrated the feasibility of fabricating crossflow and counterflow ceramic core sections of complex geometries. The data required for sizing such cores are also available (ref. 5). Little is known of the mechanical properties of such complex matrices, how to integrate the ceramic units into the metal ductwork of an engine, and the techniques necessary for fabricating large heat exchangers.

Fabrication

In this study, all weight calculations have assumed a conventional header-bar construction as illustrated in figure A-13. The header bars which separate the plates serve as passage seals and support the core during brazing. Because of the large mass difference between header bars and plates, this type of construction tends to impose severe thermal stresses on the core joints during startup and rapid temperature variations. It is possible to reduce weight and thermal stresses by using hollow header bars. The reduced thermal stresses will improve cycle life.

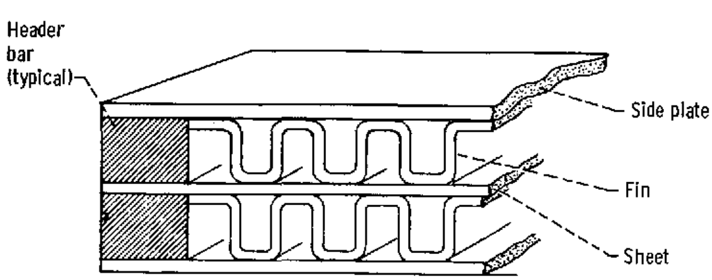


Figure A-13. - Plate-fin construction.

A fabrication approach, similar to that used in present automotive radiators, where the adjacent plates are crimped and bonded together to form passage seals and structural joints appears promising because it eliminates header bars completely. This approach has an apparent weight and cycle life advantage over present header-bar construction. However, this formed plate approach has only been used in low pressure applications for relatively small engines. A technology program would be required to qualify this approach at high pressures and to determine fabricability in large sizes.

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APPENDIX B

GENERAL METHODS OF DESIGN-POINT STUDIES

The general procedures used in the engine design-point studies are presented to illustrate the main assumptions of the analysis. The general logic of the computer codes developed to optimize each of the engine types is outlined and the models used to simulate the engine components are described. The mathematical details are not included.

As previously discussed, the main purpose of this study was to make valid comparisons between the various engine types studied. Therefore, the assumptions and procedures were applied consistently for all engine groups. Furthermore, emphasis was placed on those components (specifically heat exchangers) which are the largest contributors to the total engine weight and size or are primarily responsible for the variation in engine weight with design-point conditions. Other components (such as ducting or gear boxes) were included, but not treated in as much detail. Finally, some components were not included in the computer simulation. Those which were not included were considered either to be small contributors to engine weight or nearly identical in weight for all engines studied. The number and types of components varied among the engine groups and depended on the application and operating options being studied. Typical but specific geometries were assumed for the main engine components.

System Synthesis and Optimization

To conduct the design-point studies of sections 3 and 4, digital computer programs called PSOPs (power system optimization programs) were developed. A separate PSOP was used for each of the six engine groups or types of power system. Each PSOP included a main program, a thermodynamic-cycle subroutine, component subroutines, and a common optimization subroutine. The subroutines were written by members of the staff at Lewis Research Center. Many of the component subroutines were common among the PSOPs. The main programs provide the logic for subroutine calling sequences depending on power system operational options. They also handle problem input-output, any necessary iterations, and system calculations. Engine operating options varied among the PSOPs. A general list of problem options is as follows:

Bus or TACV engine
Choice of fuel (kerosene, methane, hydrogen)
Choice of working gas (for closed cycles)
One- or two-shaft turbomachinery arrangements
Radial- or axial-flow turbomachines
One stage of intercooling between compressors
Turbine cooling
Locations for bleeding working fluid (for semiclosed cycles)

The general flow diagram for the PSOPs is shown in figure B-1. The fixed inputs are those quantities which define the option under consideration (such as the power level, engine configuration, etc.), while the variable inputs are those which are changed in order to optimize the engine design point. In general, the types of variable parameters include compressor pressure ratios, heat exchanger effectivenesses, individual component and ducting relative pressure drops, waste heat exchanger capacity-rate ratios, and cooling-fan pressure rises.

The cycle subroutine is the first called. Thermodynamic cycle calculations are performed to obtain the temperatures, pressure ratios, gas compositions and properties, and the specific mass flow rates in the gas loops. Using these, the absolute flow rates and pressures are then calculated in the main program. The appropriate component subroutines are then entered to determine individual component sizes and weights. The component weights are then summed, the net system efficiency is calculated, and the figure of merit for this "nonoptimum" solution is determined. (The figure of merit is the one-dimensional quantity which is optimized; it will be discussed subsequently.) The optimization subroutine then determines whether or not the solution is optimum. If not, one of the

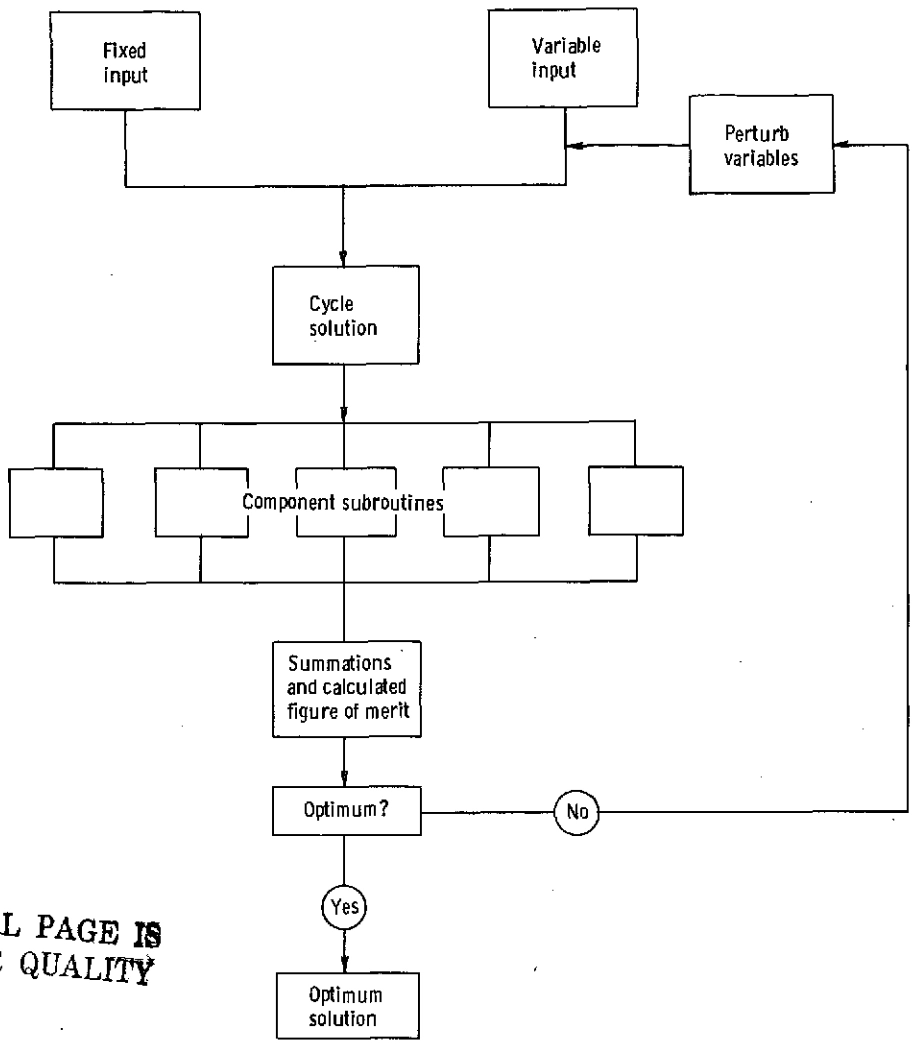

Figure B-1. - PSOP flow diagram.

variable parameters is perturbed and the entire procedure is repeated. This is continued until all the parameters have been adjusted to yield the solution with the optimum value of the figure of merit.

Different forms of the figure of merit were used during the study. All had the common objective to produce the curves of minimum engine weight as a function of engine design-point SFC which were presented in sections 4 and 5. The particular form is not important since they produce the same results. But, as an example, one of the forms of the figure of merit (FOM) which was used is

$$\text{FOM} = (\text{engine specific weight})^{x} (\text{SFC})^{(1-x)}$$

For a particular value of x, minimization of this figure of merit yields a solution corresponding to one point on the envelops of engine weight as a function of design-point SFC. Variations of the exponent x yields the entire envelope curve.

Geometry constraints on the power system waste heat exchangers were satisfied in the optimization process. If either of the two air-flow frontal area dimensions exceeded the imposed limits, the figure of merit was increased or penalized. The increase was directly proportional to the amount that each dimension was exceeded. Hence, in minimizing the figure of merit, solutions were obtained that either satisfied the constraints or could be made arbitrarily close to satisfying the constraints.

A separate cycle subroutine was used for each PSOP. In each case, calculations of the thermodynamic conditions in the combustor and coolant loops are included. The air-fuel ratio in the case of groups Ia and V and the ratio of recirculated combustion gas to fuel in the case of groups Ib, III, and IV are calculated by using an energy balance on the combustor. In all cases the composition of the combustion gas is calculated and the thermodynamic properties are calculated as functions of temperature and composition. For group III the effects of water condensation from the combustion gases in the waste heat exchanger on the flow rates and gas composition are included. Except for group IV, perfect gas relationships were used in calculating expansion and compression processes, with the gas properties evaluated at inlet temperature and composition. In the case of group IV, correlations of the enthalpy and entropy as functions of temperature and pressure were used.

Component Models and Assumptions

Packaging and configuration details for the power system heat exchangers and fuel tanks are described in appendixes A and H. Here the general procedures and assumptions in the computer component subroutines, other than the fuel tank, are presented.

The input to each component subroutine was its performance requirements. In most cases the requirements were inlet and outlet temperatures and pressures, mass flow rates, and efficiency or effectiveness. The subroutines then sized the component to its requirements. Output from the subroutines included weight, dimensions, and characteristic parameters.

HEAT EXCHANGERS AND COMBUSTORS

All of the heat exchanger models assumed specific internal geometries (described in appendix A). Within the computer subroutines, these models were scaled to satisfy the heat-transfer requirements of the problem. For all but the combustors the models were also scaled for pressure drop requirements. The combustor models were assigned constant values of relative pressure drop (pressure drop divided by inlet pressure).

Fuel temperature entering the combustors was assumed to be 80° F. Also, for all combustors more than stoichiometric air mass flow rates were assumed for the primary zone to account for typical real performance. With methane or kerosene as the fuel, the excess air was 15 percent. With hydrogen, the excess air was 10 percent.

Heat Exchangers

Each of the heat exchangers (preheater, recuperator, heat source, and waste heat) used plate-fin heat-transfer surfaces (appendix A). Constant values of metal thicknesses were used. The high pressure, gas-side fins were 0.006 inch thick, while the low pressure (near atmospheric) fins were 0.004 inch thick. All plates between fins were 0.008 inch thick. Total heat exchanger weight was increased by 10 percent to allow for brazing material.

Heat exchanger core relative pressure drop was assumed to be 80 percent of the total allotted to the heat exchanger. Iterations on core frontal area were used to satisfy the assigned relative pressure drop within 0.1 percent.

Both the heat-source and waste heat exchangers were cross-counterflow and used multipasses on the high pressure gas side. Within these component subroutines, the number of passes was incremented up from one until the required heat-transfer effectiveness could be satisfied. The waste heat exchanger was also modularized (appendix A) to satisfy dimensional constraints. The number of modules was also incremented up from one until the constraints could be satisfied. Spacing between modules was determined by requiring that the cooling air velocity entering the module be 100 feet per second. Each gas turn between passes or into modules was assumed to lose 20 percent of the entering velocity head.

Surface Combustor

A sketch of the assumed surface combustor - heat exchanger core geometry is shown in figure B-2. Premixed fuel and air flow through the porous plates and burn near the downstream surface. Heat is transferred from the combustion zone to the plate and then radiated to the two rows of closely space, internally finned tubes. The Brayton cycle working gas flows once through the length of these tubes. In an actual design, for the sake of efficiency, the combustion gases would be cooled before exiting the unit by the incoming Brayton gas. Although this was assumed in the performance calculations, the weight of such a gas-to-gas heat exchange section was not included in this component model.

The tube inside diameter was 0.375 inch with 0.0275-inch wall thickness for the high pressure train applications and 0.0123 inch for the lower pressure bus applications. Rosette fins, 0.006 inch thick, occupied three-fourths of the tube diameter. The porous plate burner total thickness was 0.625 inch. Outer wall plates were 0.0625 inch thick. Total heat exchanger weight was increased by 10 percent to allow for external structure.

Relative pressure drop in the Brayton gas core was assumed to be 80 percent of the total. And iterations on core frontal area were used to

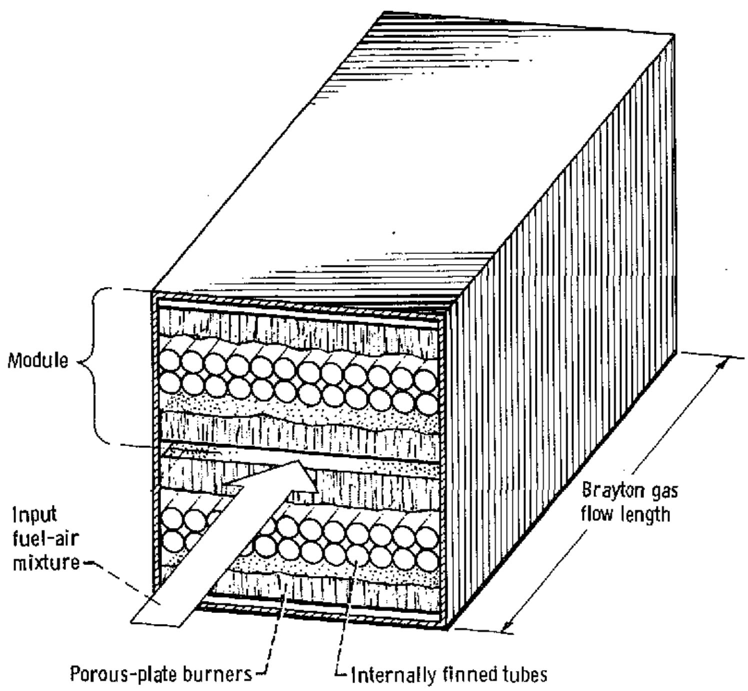

Figure B-2. - Surface combustor core geometry.

satisfy the assigned relative pressure drop within 0.1 percent. The number of core modules was incremently raised from one until the core face aspect ratio was equal to or less than 2. The relative pressure drop on the combustion side of the heat exchanger was fixed at 3 percent.

Atmospheric Combustor

This type of combustor was used in the closed Brayton cycle engine (groups Ia and Ib). A double can arrangement (appendix D) with insulation between the walls was assumed. Can walls were made of 0.0625 inch thick sheet steel. The combustor cross-sectional area was calculated based on assumed values of entering air velocity. About 100 feet per second was used with methane or kerosene, while 150 feet per second was used with hydrogen. Combustor mixing length was calculated based on the area and a space heating rate of 3×10^6 Btu/(hr)(ft^3). The relative pressure drop in these combustors was fixed at 3 percent.

High-Pressure Combustors

Engine groups III, IV, and V used high pressure combustors. Arrangements are described in appendix D. External pressure shells were cylindrical with hemispherical ends. The pressure shells were made of steel and cooled by the working fluids. Allowable stresses were based on 1-percent creep in 50 000 hours. The relative pressure drop in these combustors was fixed at 4 percent.

The hydrogen-oxygen combustor of group IV had a 0.15-inch-thick inner copper liner that was assumed cooled to 600° F by recirculated water from the waste heat exchanger. The thickness of the outer pressure shell was calculated for an allowable stress of 25 000 psi. The combustor diameter was calculated by assuming a combustion zone Mach number of 0.05. Combustor length was assumed to be 3.9 times the diameter.

The combustor pressure shell for engine groups III and V was assumed to be cooled to 1300° F by the recirculated combustion products. Thickness of the pressure shell was calculated for an allowable stress of 13 000 psi. The internal swirl can diameter was calculated by assuming a primary air velocity of 100 feet per second. The mixing zone diameter calculation assumed a velocity of 200 feet per second.

TURBOMACHINERY

Since machinery weight was known to be small compared to that of the heat exchangers, simplified machinery models were assumed. Machinery weights and overall dimensions were scaled. Emphasis here is placed on the criteria used to determine turbomachinery rotor diameter, number of stages, and rotational speeds. In general, the compressor sizing criteria were used to set turbocompressor shaft speeds. In the two-shaft engines the turbine sizing criteria for the free turbine set its shaft speed. Since the use of reduction gear boxes was assumed, alternator or transmission speeds were set independently. Alternator speeds were a function of the power output level. For bus engines the design-point shaft speed to the transmission was set at 6000 rpm. Constant levels of polytropic efficiency were used for the turbines and compressors. An exception was for radial-flow compressors during the conceptual design phase of the studies where an efficiency model was used. The efficiency model was based on a compilation of air compressor performance. Efficiency was a function of specific speed, equivalent weight flow rate, and pressure ratio. Maximum compressor polytropic efficiency was 0.878. This efficiency occurred at a specific speed of 0.775, a pressure ratio of 2, and an equivalent weight flow rate of 8 pounds per second or more. Polytropic efficiency decreased at other values of specific speed and pressure ratio and at lower equivalent weight flow rates. Within the main PSOP computer program the compressor polytropic efficiency was iterated on until the assumed value agreed with the calculated value within 0.001.

Axial-Flow Turbines

The axial-flow turbines were assumed to have the same mean blade diameter for each stage. Using turbine cooling was an option. For uncooled turbines, the maximum mean blade speed was assumed to be a linear function of the turbine inlet temperature in the range from 1600° to 1800° F. At 1800° F the maximum allowable mean blade speed was 1200 feet per second. At 1600° F the maximum was 1900 feet per second. At lower temperature, the maximum mean blade speed was 2200 feet per second.

Turbine rotor diameters were calculated by assuming an exit axial Mach number of 0.6 and a last stage hub to tip diameter ratio of 0.7. The number of required turbine stages was assumed to be 0.8 divided by the overall turbine speed-work parameter[1] and then rounded upward to the next integer. Reference 1 shows that this assumption should result in a high efficiency turbine.

If turbine rotational speed was not specified by a compressor, it was set by the combination of maximum allowable mean blade speed and the mean diameter calculated at the last stage. If the speed was specified

[1]The overall turbine speed-work parameter is defined as the mean blade speed squared divided by the product of the acceleration due to gravity and the actual head to be produced by the turbine.

by a compressor, mean blade speed was calculated and compared to the allowable value. If the allowable blade speed was exceeded, the procedure was reversed. The turbine criteria were then used to set the rotational speed, and the compressor was recalculated.

In the turbine cooling option all stages whose inlet temperature exceeded 1700° F were cooled. Bleed flow from the compressor outlet was used for the cooling. The cooling flow was ducted into both stator and rotor blades and it exited through the blade trailing edges. The amount of cooling flow was determined from the correlation on page 37 of reference 2. Both rotor and stator walls were assumed to be cooled to 1700° F. It was also assumed that one-fourth of the coolant flow introduced into each stage produced useful work in that stage. And all of that coolant flow produced useful work in the next stage. The thermodynamic-cycle subroutine corrected for the ratio of cooled to uncooled turbine work in its calculations.

Radial-Flow Turbines

Single-stage radial inflow turbines were assumed. Rotor tip diameters were calculated by assuming an exit loss term[2] of 1.03 and a blade to jet speed ratio[3] of 0.70. Such values are typical of high efficiency turbines that have a design-point specific speed in the range from about 0.45 to 0.7.

If rotational speed was not set by a compressor, it was set by assigning a turbine specific speed of 0.7. If the resulting tip speed exceeded 1600 feet per second the solution was flagged for rejection. When the turbine rotational speed was specified by a compressor, turbine specific speed was calculated to check if it was in the 0.45 to 0.7 range. If the rotational speed resulted in a turbine tip speed greater than 1600 feet per second, the procedure was reversed. Then the turbine was allowed to set the speed and the compressor was recalculated.

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[2]The exit loss term is defined as the ratio of turbine total efficiency to turbine static efficiency.

[3]The blade to jet speed ratio is defined as the rotor tip speed divided by the square root of twice the acceleration due to gravity times the turbine ideal static head.

Axial-Flow Compressors

The axial-flow compressors were assumed to have the same blade tip diameter for each stage. Compressor tip speed was limited to no more than 1800 feet per second. Rotor diameters were calculated subject to some assumed velocity diagram and geometric limits. The entrance flow angle relative to the rotor was assumed to be 60°, and the maximum relative Mach number entering the first-stage rotor blades was 1.25. The first-stage hub to tip diameter ratio was limited to 0.5 or more, while the similar last-stage ratio was limited to 0.9 or less. The number of required compressor stages was assumed to be the overall compressor work-speed parameter[4] divided by 0.3 and then rounded upward to the next integer.

In the compressor sizing procedure, the first-stage hub to tip diameter ratio and the rotor relative Mach number are initially set at their limits. If the calculated rotor tip speed exceeds its limit it is set at the limit, 1800 feet per second. If rotational speed is not specified by a turbine, it is calculated from the tip speed subject to an assumed tip diameter limit of 4.8 inches or more. If the initial diameter ratio and relative Mach number values result in a too small tip diameter, one or the other is adjusted to satisfy the geometric constraints. If the rotational speed was specified by a turbine, the compressor tip diameter and last-stage hub to tip diameter ratio are calculated. And if the last-stage diameter ratio exceeds its 0.9 limit, the relative Mach number assumption is reduced until the geometry constraints are satisfied.

Radial-Flow Compressors

Single-stage radial-outflow compressors were assumed at all locations other than for the bus turbocharger in group III engines. In that case, two stages were assumed. Rotor tip diameters were calculated assuming an inlet critical velocity ratio of 0.30 and a head coefficient[5] of 0.74. Reference 3 shows that these assumptions are typical for high efficiency compressors with a design-point specific speed in the range from about 0.65

[4]The overall compressor work-speed parameter is defined as the product of the acceleration due to gravity and the actual head produced by the compressor divided by the blade-tip speed squared.

to 0.95. If rotational speed was not set by a turbine or another compressor, it was set by assigning a compressor specific speed of 0.77. When speed was previously determined the resulting compressor specific speed was calculated.

Alternators and Gear Boxes

Alternators were assumed to be salient, wound-pole machines. This type, rather than a solid rotor alternator, was used because it is inherently smaller and lighter. Generally the winding current densities and magnetic flux densities in this type of machine result in low loss, high efficiency alternators. The alternator size and weight were scaled from similar machines and based on the net shaft power available to the alternator. Three base designs, depending on power level, were used. The assumed characteristics are shown in table B-1.

The gear boxes were conceptually a single, inline helical gear set. Shaft diameters were sized to handle the required torque. Gear diameters and face width were proportional to their shaft diameters. The gear and enclosing box weight calculations assumed the density of steel.

TABLE B-1. - ALTERNATOR CHARACTERISTICS

	Shaft power, hp		
	5000 or less	6000 to 7500	7500 or more
Electromagnetic efficiency	0.93	0.945	0.945
Rotational speed, rpm	8000	6000	1500
Number of poles	6	6	8
Frequency, Hz	400	300	300
Specific weight, lb/shaft hp	1.1	0.93	1.0

[5]The head coefficient is defined as the product of the acceleration due to gravity and the ideal head to be produced divided by the square of the rotor tip speed.

DUCTING

Each duct was assumed to have a fixed length and two 90° turns. In general, bus duct lengths were one-half those for the train. The duct diameter was iterated on to satisfy the required pressure drop. High strength alloys were assumed. And wall thicknesses were calculated as functions of allowable stress for the given temperature and pressure. Two inches of insulation were assumed on all high temperature ducts.

References

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3. Galvas, Michael R.: Analytical Correlation of Centrifugal Compressor Design Geometry for Maximum Efficiency with Specific Speed. NASA TN D-6729, 1972.

APPENDIX C

GENERAL METHODS OF OFF-DESIGN ANALYSIS

The off-design performance was predicted for several design points for the groups III and V engines. The design points were selected from the results of section 5. The results of the off-design analyses performed are summarized and discussed in section 5.

The group III engines were simulated with a program using CSMP, an IBM dynamic modeling program described in reference 1. The group V engines were simulated using GENENG (generalized engine), a NASA program for calculating design and off-design performance for turbofan and turbojet engines (ref. 2). GENENG was modified to include recuperation. The purpose of this section is to outline the procedures used in the off-design analyses.

Semiclosed Brayton Cycle Engines

The group III system is a turbocharged, semiclosed Brayton system. The two configurations that were studied are shown in figure C-1. These configurations are representative of the systems discussed in section 3.

The group III mode of operation is to reduce system pressure level and hence mass flow rate in order to reduce power output. When this is done, the turbomachinery speed and turbine inlet temperatures are maintained at a constant value. As the flow rate is reduced from the design-point values, the relative pressure losses, the heat exchanger effectivenesses, and the turbomachinery efficiencies change from their design values. As a result, the SFC changes with power level. This simulation was used to predict that variation. The off-design performance was calcu-

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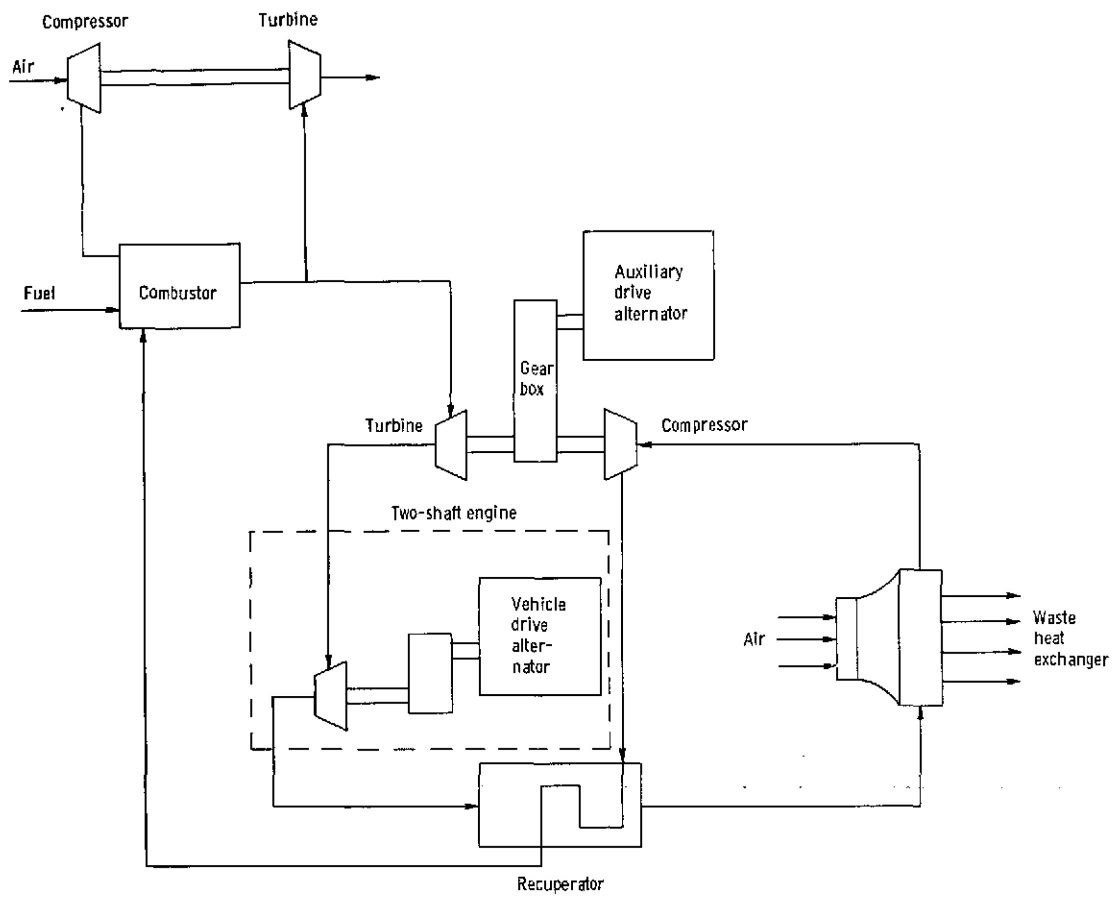


Figure C-1. - Schematic diagram of one- and two-shaft group III engines.

lated for several design points. These design points were selected from section 5.

To insure that this method of part-power operation is feasible, a brief assessment of engine control was made and is described in this section.

METHOD OF SIMULATION

The group III engines were simulated using a digital computer model originally developed to investigate dynamic performance of closed Brayton systems. This model was programmed using CSMP. The model was modified to include a turbocharger model and appropriate inventory controls.

An operating point of this system is described by the pressure distribution, the temperature distribution, and the turbomachinery speeds. These variables are continuously calculated by the computer and can be used to predict the response of the system for a wide variety of inputs. The simulation contains all of the equations necessary to define system operation.

The turbomachinery performance is simulated using operating maps. These maps are taken from representative machinery and scaled to each design point. Shaft speed is supplied to the program, and the power is calculated by summing the torques acting on the shaft. The turbine and compressor torques are derived from the machinery operating maps. The representative shaft losses are windage, bearing, thermal, and electromagnetic. The windage losses and bearing losses are assumed to vary as speed cubed, and the thermal and electromagnetic losses are assumed constant.

For these studies the combustor exit temperature and compressor inlet temperature were assumed to be constant. The turbine and compressor outlet temperatures are functions of the inlet temperatures and the power transferred. The effectiveness of the recuperator is calculated from

$$E = \frac{1}{1 + \left(\frac{1 - E_{DES}}{E_{DES}}\right)\left(\frac{W_{DES}}{W}\right)^{-0.4}}$$

where E_{DES} and W_{DES} are the design-point effectiveness and flow rate, respectively, and W is actual flow rate. This equation is used with the recuperator inlet temperatures to calculate the recuperator outlet temperatures. The variation of recuperator relative pressure loss with flow rate in terms of design pressure loss varies according to

$$\left(\frac{\Delta P}{P}\right) = \left(\frac{W}{W_{DES}}\right)^{-0.62}\left(\frac{\Delta P}{P}\right)_{DES}$$

CONTROL OF SINGLE-SHAFT SEMICLOSED 7500-HORSEPOWER BRAYTON CYCLE SYSTEM

A representation of the system showing possible control loops is given in figure C-2. Of primary importance would be the control of the speed of the main rotor shaft as the power extracted by the alternators is varied. To maintain efficiency the inlet temperature to the main turbine should be regulated to a fixed value. The inlet temperature could be inferred from turbine

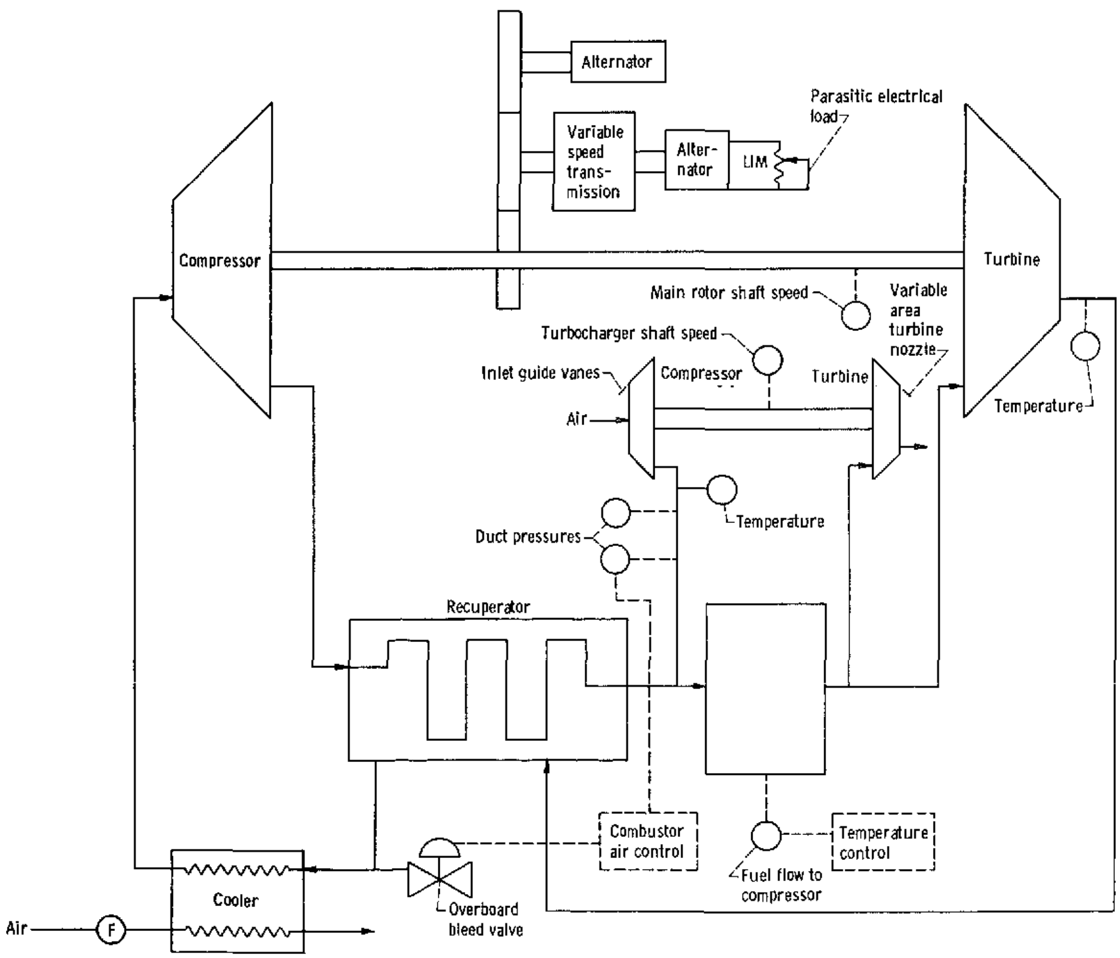


Figure C-2. - Conceptual control design for 7500-horsepower single-shaft group III engine.

outlet temperature using present-day high reliability components. It would be important to maintain control of the turbocharger shaft speed and to introduce sufficient airflow for complete combustion of the fuel flow to the combustor. Control could be implemented by controlling fuel flow to the combustor, using inlet guide vanes and a variable area turbine nozzle for the turbocompressor, using an overboard bleed valve, and using a parasitic electrical load.

Control Modes

Steady-state operation at full and partial power. - For temperature control the fuel flow would be used to control turbine discharge temperature. This would insure high efficiency system operation at full and partial power conditions.

For main shaft speed control the loop inventory would be used to control the main shaft speed. In steady state this would be controlled by the turbocompressor inlet guide vanes and variable area turbine nozzle. Thus, if the main rotor shaft speed decreases slightly, the inlet guide vanes and variable area turbine nozzle will be manipulated to increase the inventory. The resulting increased density in the loop would increase the speed of the main rotor shaft.

For combustion airflow control, to insure an atmospheric air inflow rate great enough for complete combustion of the fuel flow, turbocompressor duct pressures and temperature would be used to determine the air inflow rate. This would be compared to the fuel flow rate and if too low the bleed valve would be opened. The resulting outflow from the loop would depress the main rotor shaft speed slightly and cause the speed control to manipulate turbocompressor variable geometry to increase air inflow rate to maintain loop inventory.

Decreasing power level. - A decrease in power extracted from the loop would cause the speed of the main shaft to increase above the design value. This would be sensed by the main shaft rotor speed sensor, and the speed control would manipulate the turbocompressor inlet guide vanes and turbine nozzle to decrease loop inventory. In addition, the combustion airflow control could be overridden to the open bleed valve. To accommodate more rapid decreases in power the turbine temperature control could be overridden to depress the fuel flow rate. A separate fuel shutoff valve could be used to protect both rotors from overspeed. This separate emergency system would receive signals from both the main rotor shaft and the turbocharger shaft.

Increasing power level. - Increasing the power extraction from the loop would cause the main shaft speed to decrease below the design value. This would be sensed by the speed control and the turbocompressor inlet guide vanes and variable area turbine nozzle manipulated to increase loop inventory. The resulting increase in turbocompressor duct airflow rate would be sensed by the combustion airflow control and would reduce the opening of the dump valve.

Other Controls, Coupling Signals, and Implementation

Separate controls, not described herein, would be required for the cooler fan, the alternators, the levitation fans, and the linear induction motors. Coupling signals between these controls would be required. During power changes a speed error signal from the engine controls could be used to limit the rate of change of power supplied to the LIM alternator. The parasitic load could be manipulated by the alternator control to limit the rate of power change during the decreasing load conditions. Also, anticipatory signals from the alternator controls could give the engine control more time to react to changes in the power load initiated by the operator.

Further Controls Definition

During a preliminary system design, a computerized system dynamic model should be implemented to further define the control system requirements. Compressor and turbine maps including variable geometry features would be included. Results of such a study would disclose additional control requirements or possible simplifications. Important results would be the control requirements during system startup, control features required to prevent surge of either compressor or blowout of the combustor. Another important result would be the prediction of the time history of power available during increasing power conditions. This could be important when selecting the capacity of the turbocompressor.

During the simulation study the advantages of alternate control modes could be examined. For example, an alternate way of improving the rate of power increase would be to temporarily set the temperature control loop for a lower turbine operating temperature. This would cause the speed control to increase loop inventory. Then when the power load was applied, the turbine temperature could be quickly increased by manipulating fuel flow to provide the necessary power output.

Other alternate control modes could also be examined during the simulation study. It might be possible to control speed with fuel flow and temperature with inventory. This would give a tighter, more responsive, control of speed. Another possible improvement would be to measure the mass flow leaving the bleed valve and compare it with the inflow rate giving a quicker indication of increasing or decreasing loop inventory.

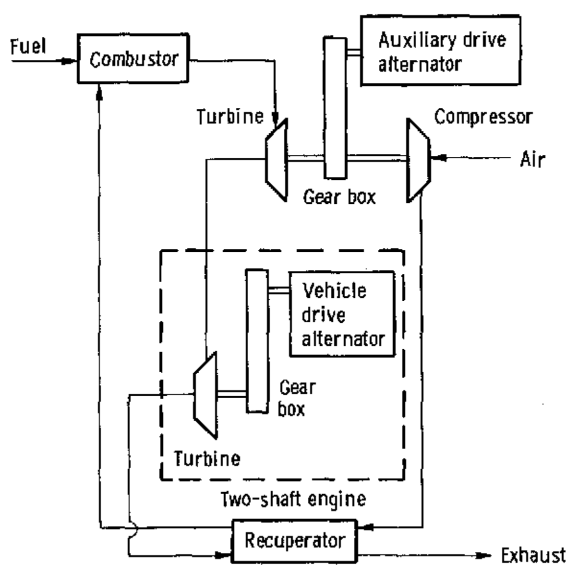


Figure C-3. - Schematic diagram of group V open-cycle engine.

Open Brayton Cycle Engine

The group V open cycle is shown schematically in figure C-3. The mode of operation for group V is to reduce turbomachinery speed or turbine inlet temperature to reduce power output. When these parameters are changed, the relative pressure losses, the recuperator effectiveness, and the turbomachinery efficiencies also change. This results in a change in SFC. This section discusses the methods required to model these changes.

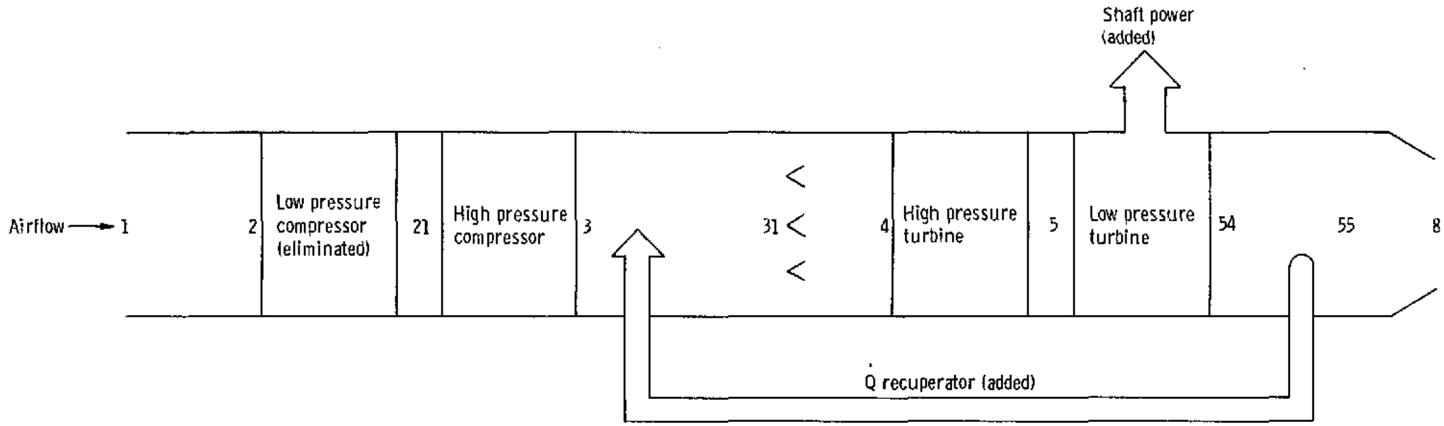
The group V engines were simulated using GENENG, a program for calculating design and off-design performance for turbofan and one- or two-spool turbojet engines. Off-design calculations use input component performance maps which are scaled to the desired design-point values of mass flow, pressure ratio, efficiency, and corrected speed.

The two-spool turbojet cycle considered by GENENG (shown in fig. C-4(a)) was reconfigured to simulate the two-spool turboshaft engines. The outer or low-pressure compressor was eliminated, shaft power extraction was added to the LP spool, and a recuperator loop was added between the compressor and turbine exits. To simulate the one-spool turboshaft engines, the one-spool turbojet configuration of GENENG shown in figure C-4(b) required only the addition of the recuperator loop. The various bleeds and afterburners shown in these figures are optional in GENENG and are controlled by input as desired. They were all set to zero for the present study.

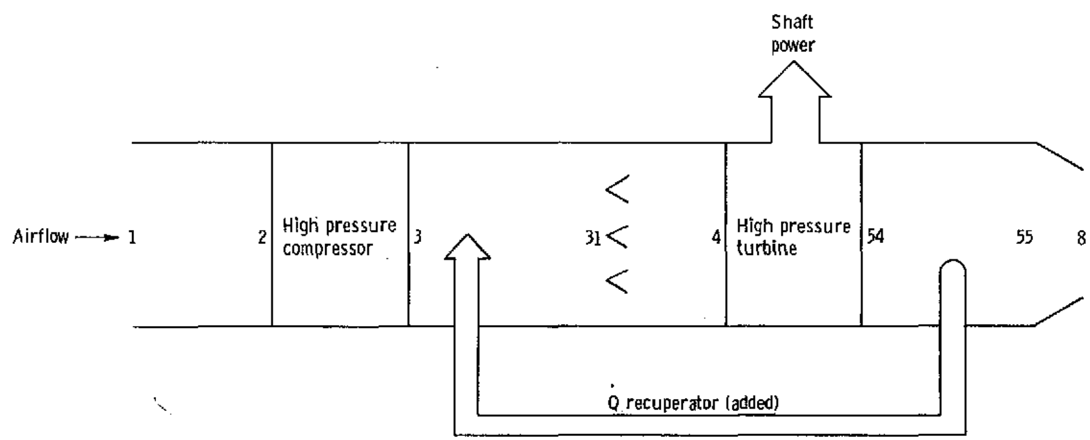
Conditions at station 2 are input as $T_2 = 540^{\circ}$ R and $P_2 = 0.9935$ atmosphere (14.6 psia). Since there is no low pressure compressor the conditions at station 21 are the same as those at station 2.

The air is compressed to conditions at station 3 and passes through the recuperator where it is heated and where it suffers a pressure loss.

Combustor airflow is heated and is then expanded through the high pressure turbine to conditions at station 5. The enthalpy at station 5 is calculated by making a power balance. The combustion products then pass through the recuperator before being exhausted to the atmosphere.



(a) Two-shaft engine.



(b) One-shaft engine.

Figure C-4. - Modifications made to GENENG turbojet cycles to simulate group V engines.

In addition, the physical speeds of the compressor and the high pressure turbine must match. There is no corresponding constraint on the low pressure turbine shaft speed.

When T_{54} has been determined, the temperature and pressure out of the low pressure side of the recuperator are calculated.

The assumed variation of recuperator effectiveness and recuperator relative pressure loss is the same as for the group III engine.

The turbomachinery component performance maps required for off-design calculations are supplied as block data. The entire map is scaled by GENENG (for mass flow, pressure ratio, efficiency, and, in some cases of turbines, corrected speed) according to the ratio of map values at that point and what the values must necessarily be for the input design point. The maps were chosen to be representative of their flow types.

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APPENDIX D

COMBUSTOR EMISSIONS AND ENGINE NOISE

In evaluating new engines for application to the transportation industry, it is essential to consider their environmental effects. The bus application considered in this study includes considerable operation in urban areas that will require low levels of noise and exhaust emissions. The TACV application considered operation in the Northeast Corridor. Although full-power operation in densely populated areas would be minimized, noise and exhaust emissions during station approach and departure or cruise between stations in urban areas must still be considered.

In this section the engine groups are compared on the basis of noise and exhaust emissions. Nominal design conditions were used to estimate noise and exhaust emissions typical of an engine group. Exhaust emissions are not only determined by engine operating conditions but are also a function of the type of combustor and fuel. Therefore, to estimate emissions, a particular type of combustor and a set of design parameters were selected for each engine group. Discussion of the controlling parameters in combustor design and operation and their effects on emissions and on combustor concept selection has been included.

Emissions estimates are presented for three closed-cycle engines (groups Ia, Ib, and II), a semiclosed engine (group III), and an open-cycle engine (group V) over the range from idle to full power. For the closed-cycle engines, groups Ia and Ib use the same combustor concept but different diluents; the group II engine uses a surface combustor concept. The group IV engine is a nonpolluting variation of the semiclosed engines. Since the group IV engine burns cryogenic hydrogen and oxygen, the only combustion product is water vapor, which is condensed for recirculation. Therefore, the group IV engine is not included in the emissions summary.

Only the noise emissions from the power generation system were considered. Although the transmission might be a significant source of noise, it was not specifically treated since it would be essentially the same for all engines within an application and therefore would not serve as a discriminator between groups. A brief discussion of the sources of noise, methods of noise suppression, and noise estimates is included.

Emissions

In order to compare the engines on the basis of exhaust emissions, the operating conditions required of the combustor by the cycle must be considered. For example, the combustor for the closed cycles operates at near ambient pressure, and those of the semiclosed and open cycles operate at the cycle peak pressure. In addition to the pressure, the temperature and composition of the gases at inlet affect the emissions and are different for each of the engine types. As discussed in the description of the engine group (sec. 3), the semiclosed cycle engine (group III) and one version of the closed-cycle engine (group Ib) use recirculation combustion gases as combustor diluent, while the open-cycle engine (group V) and another version of the closed-cycle engine (group Ia) use excess air as the diluent.

In addition to the operating conditions, the combustor concept must be considered in some detail before emissions estimates can be made. The group II engine, for example, is a closed-cycle engine that differs only from the group I engines in the type of combustion used. The group II engine used an integrated surface combustor and heat-source heat exchanger, and the group I engines use a diluent controlled combustor and a separate heat exchanger.

As a consequence, before comparing the engines on the basis of emissions, their combustor operating conditions and the types of combustors considered for each engine type are described. The combustor parameters or operating conditions that are a function of the engine type, and their effect on emission trends are discussed in this section.

The specific combustor concept assumed for each engine type for purposes of emissions comparisons are also described.

COMBUSTOR PARAMETERS

The objective of this appendix is to compare the closed, semiclosed, and open Brayton cycle engines on the basis of exhaust emissions. From that perspective, the combustor parameters that are of primary importance here are those that are imposed on the combustor by the engine, that is, pressure, temperature and composition of the inlet gases. The effects of these engine dependent parameters on combustor emissions are shown.

Other parameters, such as combustor geometry, reference velocity, volumetric heat release rates, type of fuel preparation, and injection, would be chosen for each engine type to meet design requirements or constraints on size, combustion efficiency, pressure drop, exit temperature profile, stability limits, maintainability, reliability, etc. These parameters were considered only so far as necessary to obtain estimated emission ranges for each engine and to insure that the engines are compared on an equal basis. The combustor configurations assumed are described in the following section. Those parameters that were specified to determine combustor size and weight for the engine weight and performance analysis are briefly discussed in appendix B.

The pressure and temperature of the combustor inlet air have significant effects on the emissions of hydrocarbons (HC), carbon monoxide (CO), and oxides of nitrogen (NO_x). For a specific type of combustor, as pressure and temperature are increased, the HC and CO emissions are reduced and the NO_x emissions are increased. The combustor inlet pressure and temperature differ significantly among the engine types considered and for each type they vary with the engine design point. For the designs considered in the engine analysis of sections 4 and 5 these conditions are given in table D-1 for full power operation. For the closed cycle engine, the combustor loop operates near the atmospheric pressure. The inlet temperature ranges shown correspond to a range of air preheater and heat-source heat exchanger effectivenesses.

TABLE D-1. - COMBUSTOR CONDITIONS FOR RANGE OF

DESIGN POINTS CONSIDERED IN ENGINE ANALYSIS

Engine group	Type of diluent	Temperature range, $^{\circ}F$		Pressure range, atm
		Primary air	Diluent	
Ia	Air	400 to 800	400 to 800	1
Ib	Recirculated combustion gas	400 to 800	400 to 800	1
II	(a)	400 to 800	(a)	1
III	Recirculated combustion gas	800 to 1100	1000 to 1200	12 to 25
V	Air	750 to 1200	750 to 1200	3.5 to 7

[a]Not applicable. Surface combustor integrated with heat-source heat exchanger, so near stoichiometric air-fuel ratios are assumed.

The group III and V engine combustors operate at the cycle peak pressure. As was explained in section 5 the group III combustor pressure was limited to 12 atmospheres for the bus application but was allowed to optimize near 25 atmospheres for the train application. The primary air for the group III engine is heated as a result of the work of compression and the 800° F value corresponds to the 12-atmosphere pressure, while the 1100° F corresponds to the 25-atmosphere case. The pressure level variation shown for the group V engine corresponds to optimum values over a range of recuperator effectiveness. The 3.5-atmosphere pressure and 1200° F inlet temperature correspond to a highly recuperated engine; the 7-atmosphere pressure and 750° F inlet temperature conditions correspond to engines with the lowest recuperations considered. The unrecuperated engine would have an inlet temperature of about the same as this and a pressure in the range of 10 to 14 atmospheres.

ENGINE COMBUSTOR CONCEPTS

Combustors were studied in the context of emissions levels and system requirements. Emissions considerations affected selection of candidate combustors and required a review of current state-of-the-art and advanced combustor concepts. The review of combustor concepts indicated that current combustors have no potential for low emissions and therefore, were, not considered in this study. However, a number of advanced and developmental combustor concepts are applicable to this study. A description of these concepts and their application to the engine groups is presented here.

Diluent Swirl-Can Combustors

A swirl-can combustor (fig. D-1) consists of a diffuser, combustion zone, and diluent zone. Unlike a conventional combustor, the primary zone has a number of small, individual swirl-can modules making up the burner. Each swirl-can module consists of a carburetor, swirler, and flame stabilizer (fig. D-2). Approximately 15 percent of the airflow is ported into the swirl-cans for combustion; the remaining air flows between the swirl cans providing cooling, completing combustion, and quenching the exhaust gases. This concept provides more efficient control of fuel-air preparation before combustion. Also, overall swirl-can combustor length is considerably less than that of a conventional annular combustor resulting in shorter residence times at high temperature and therefore reduced NO_x emissions. Experimental data show lower NO_x, CO, and HC emissions compared to conventional annular combustors. Operating experience indicates that high combustion efficiency can be maintained

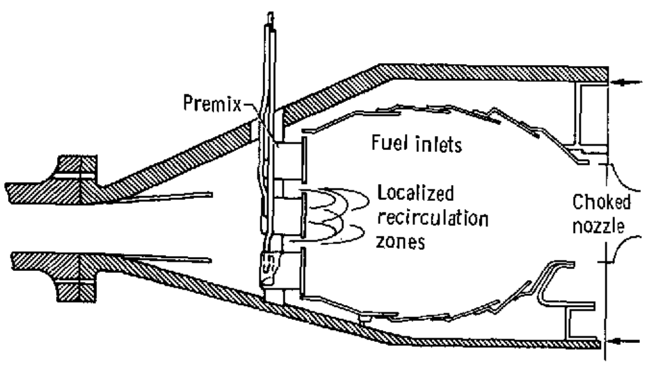

Figure D-1. - Modular swirl can combustor.

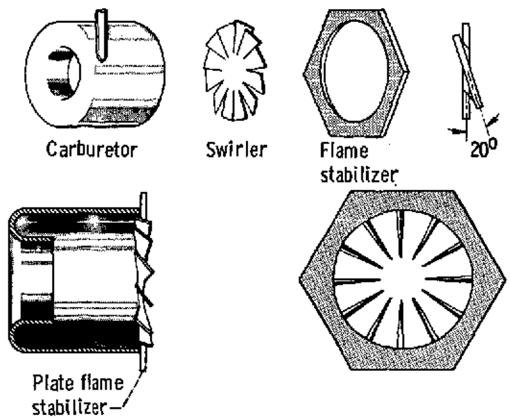

Figure D-2. - Swirl can module details.

over a range of engine power by varying the number of swirl-can modules in operation (fuel staging). In this way each swirl can module operates near its design point and therefore at high efficiency.

The swirl-can combustor concept was assumed for the group I, III, and V engines. The group Ia combustor operates near atmospheric pressure with preheated air diluent. Combustion air is also preheated. Because of low air density, the combustor has a large cross-sectional area. Figure D-3 illustrates conceptually the group Ia swirl-can com-

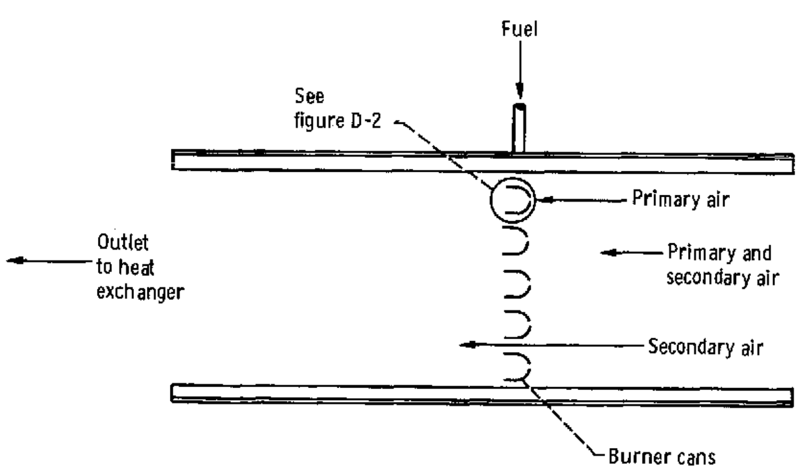

Figure D-3. - Group Ia combustor.

bustor. Fuel staging (varying number of burners in operation) can be used to maintain high combustion efficiency over a range of engine power.

The assumed group Ib combustor illustrated in figure D-4 uses a modular swirl-can concept with cooled exhaust product diluent. Primary air is preheated. This combustor concept, unlike the group Ia combustor, has separate primary and secondary air ducting. The primary air is ducted directly to the inlet of each swirl-can module while the diluent flows between modules for cooling. This arrangement permits efficient use of exhaust gas waste-heat and also reduces the amount of oxygen for NO_x production. Although swirl-can combustors have demonstrated low emissions with air diluent, there is no operating experience using exhaust product diluent.

The combustors assumed for groups III and V would also use the swirl-can concept. Since the combustors for both these groups are pressurized, their concept is conceptually similar to an aircraft engine swirl-can combustor. A representative combustor is illustrated in figure D-1. In both groups, the primary air is preheated. The group V diluent is preheated air, and the group III diluent is exhaust gas recirculated by the power system.

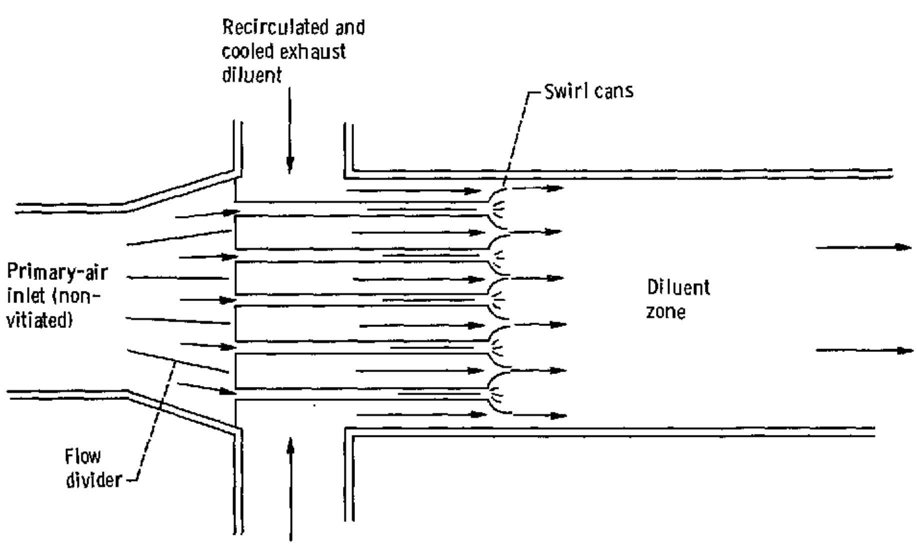

Figure D-4. - Group Ib swirl-can combustor concept.

Surface Combustors

Although swirl-can combustors have lower emissions than conventional burners, still lower emissions can be realized by use of a burner with a nonadiabatic combustion zone. A porous-plate surface combustor is such a device. Porous-plate surface combustors using natural gas as fuel are now used in space heating and industrial applications. The surface combustor is presently being developed for use with liquid fuels in advanced automotive engines. Burner surfaces can be fabricated in flat-plate, cylindrical, or polygon configurations using ceramics or sintered metals.

An objective of this study was the evaluation of a closed Brayton cycle system using a surface combustor (group II). Application of this concept to a group II system is shown in figure D-5.

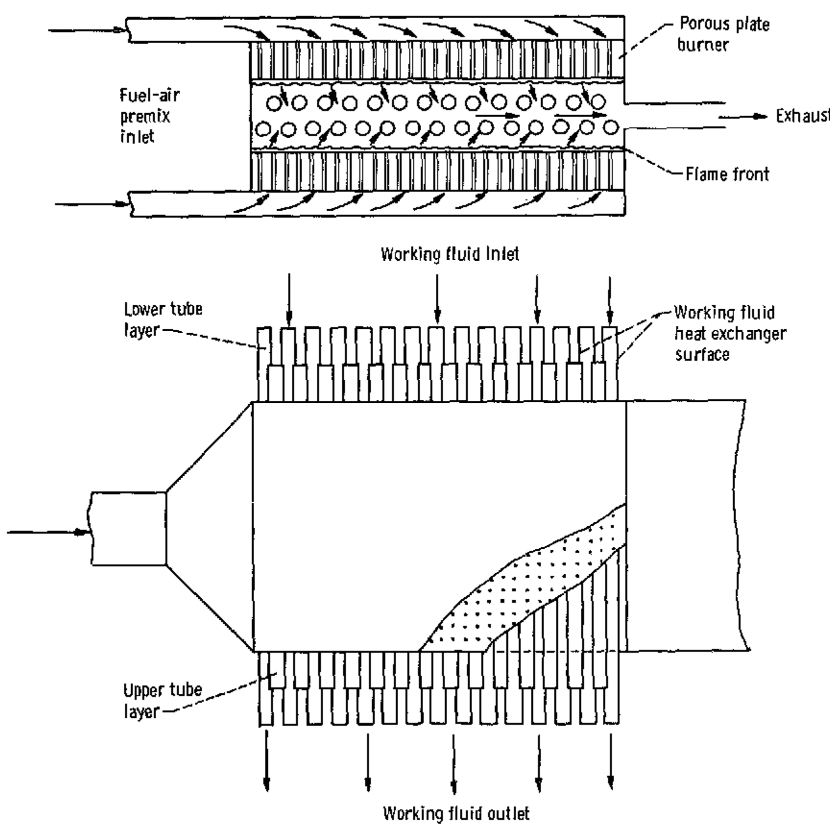

Figure D-5. - Group II surface combustor concept.

In a surface combustor, combustion occurs at or near the surface of a porous matrix (generally ceramic). Premixing of fuel and air is required before it enters the porous matrix. This premixing tends to insure uniform fuel concentration and to provide intimate fuel-air contact, both of which are required for low emissions. The flame front is uniform across the surface of the matrix and is generally less than 4 millimeters high. Combustion occurs near the stoichiometric fuel-air ratio. Combustion-zone temperatures are heat-transfer controlled below 3000° F by conductive heat transfer to the burner surface which in turn radiates heat to the working fluid heat exchanger surface. Unlike conventional and swirl-can combustors, where cooling occurs after combustion, cooling of the flame occurs during the combustion process in surface combustion. The low-flame temperature and short residence time results in low NO_x emissions.

Although emissions estimates were made only for the group II surface combustor, which operates near atmospheric pressure, consideration was given to a pressurized surface combustor concept. Figure D-6 illustrates a possible surface combustor configuration for engine groups III and V. The burner geometry is cylindrical and consists of a pressure shell and cooled heat receiving surface around the porous burner. The pressurized fuel-air mixture enters through the cool side of the porous matrix. The sec-

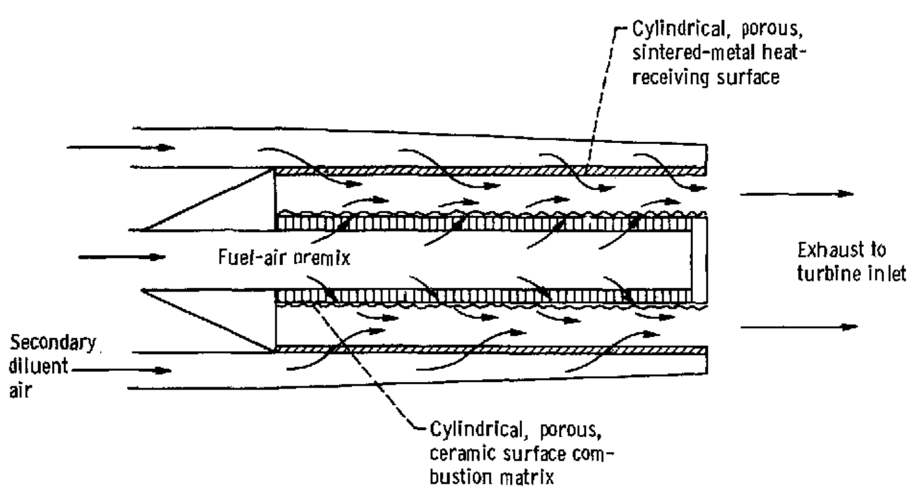

Figure D-6. - Group V surface combustor concept.

ondary air enters the combustion chamber through the porous heat receiving surface, thus providing cooling. The burner surface is cooled by radiation to the surrounding heat receiving surface which is in turn cooled by convection to the secondary air. In the group V burner, the secondary cooling fluid is air; in the group III burner it is recirculated exhaust gas. Good emissions performance over a range of power can be obtained by a combination of fuel staging and fuel-air ratio control. Fuel staging can be achieved by using a number of small burner modules to make up the combustor.

Catalytic Combustors

A recent innovation in combustor technology has potentially lower emissions than even the surface combustor. The catalytic combustor uses a catalyst to promote the oxidation of fuel at temperatures below 3000° F. Demonstration catalytic combustors have been built, but they have not been used for mobile applications. To indicate the possible reductions in emissions to be realized by developing this concept, emissions estimates were also made for this type of combustor.

The catalytic combustor has a porous or honeycomb matrix which contains the catalyst. In appearance and operation, it can be quite similar to the surface combustor. However, unlike other combustors, it can operate at fuel-air ratios well below the stoichiometric mixture. This is due to the action of the catalyst which promotes complete combustion even at low temperatures and fuel-air ratios. The efficiency or action of a catalyst varies with temperature. Therefore, for best performance a catalytic combustor should be operated in a specified temperature range. To operate over a different range of turbine-inlet temperatures, part of the secondary air can be bypassed. Startup is initiated by preheating the matrix to a minimum reaction temperature and flowing the preheated fuel-air mixture into the combustor. Premixing the fuel and air is also required. Experimental data indicate that exhaust product temperatures can be maintained between 1800° and 2300° F without bypass diluent.

EMISSIONS ESTIMATES

Emissions estimates were made for the engine groups Ia, Ib, II, III, and V using the assumed combustor concepts and operating parameters. Three fuels were considered: kerosene-air, methane-air, and hydrogen-air. The results are summarized in figure D-7. Consideration was also given to the catalytic combustor concept in the following discussion.

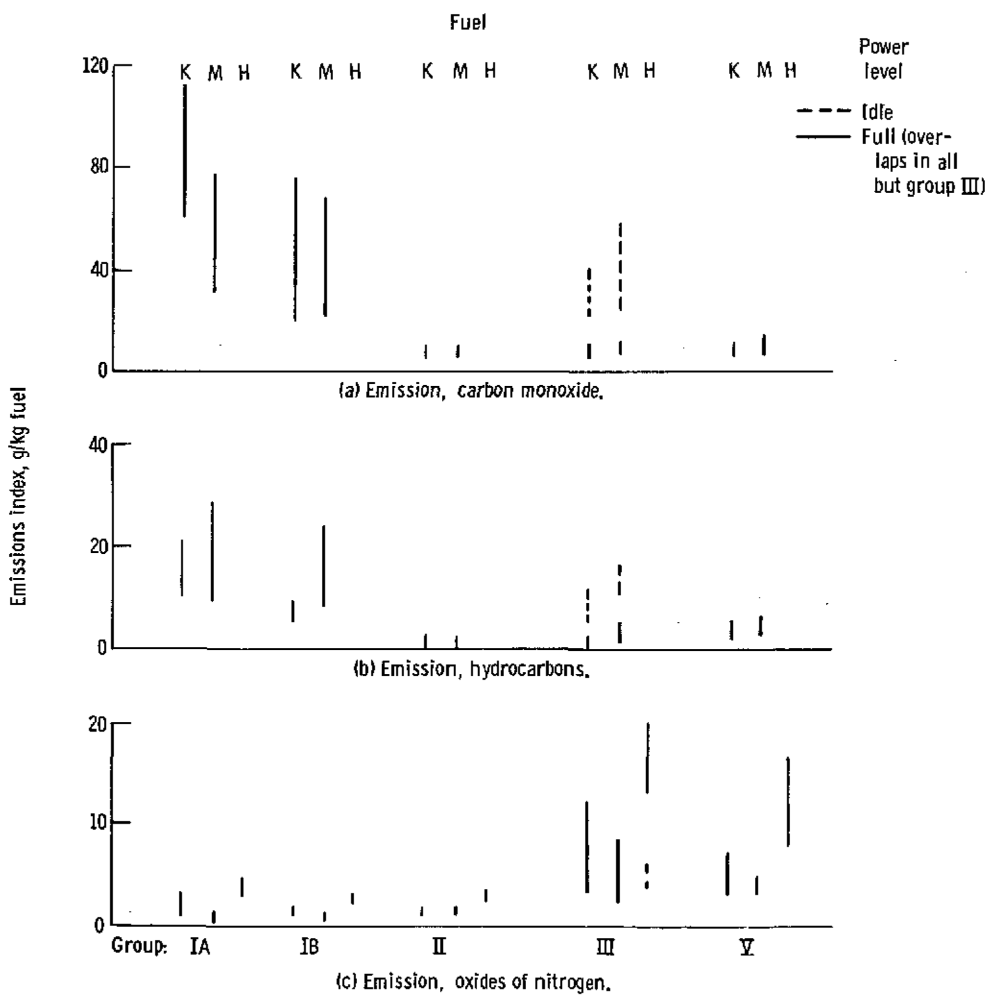

Figure D-7. - Emissions estimates. (Note: K denotes kerosene; M, methane; and H, hydrogen.)

The engines are grouped by type of combustion system. The closed systems (Ia, Ib, and II) all have external combustors operating near atmospheric pressure. Groups Ia and Ib have swirl-can combustors while group II assumes a surface combustor concept. Group Ia has air diluent while Ib uses exhaust gas recirculation. The semiclosed (group III) and open (group V) systems use internal, pressurized, swirl-can combustors. The group III engines use exhaust gas recirculation, and group V uses air diluent.

The emissions in figure D-7 are in the units of emissions index which is defined as grams of pollutant per kilogram of fuel used. The emissions index is shown as a range of values partly because of the uncertainty in the calculation but also to include a range of operating conditions as discussed previously. Idle and full-power conditions are included. In general (except group III) the emissions at idle and full power are expected to be approximately constant. The group III estimates indicate the deleterious effects of reduced temperature on CO and HC emissions. At idle, the primary-air inlet temperature drops by several hundred degrees, which increases CO and HC emissions while reducing NO_x formation.

Emissions estimates were made for kerosene, methane, and hydrogen. Kerosene and methane were considered as primary fuels while hydrogen was considered as a possible alternate fuel. (See appendix G.) The only combustion product of hydrogen is water vapor so there are no HC and CO emissions. However, NO_x is formed through combination of oxygen and nitrogen in the air. The emissions characteristics for each of the fuels tend to hold for all groups so that a general fuels comparison can be made.

Hydrogen shows higher NO_x emissions than both kerosene and methane. This is primarily due to the high combustion zone temperature. The emissions levels of kerosene and methane are nearly comparable except for the CO emissions for group Ia.

Figure D-7, in a comparison of engine groups, shows that group Ib has lower emissions than Ia, the result of the use of exhaust-gas recirculation. Of the closed engines, group II with its surface combustor has the lowest emissions. The open and semiclosed engines have comparable emissions levels at full power. However, at idle group III has much higher CO and HC levels but lower NO_x than group V. Of the five engine-combustor systems shown in figure D-7, group II has the potential for lowest emissions. This is due to the surface combustor. If surface combustors were considered for the open and semiclosed engines, they would have a comparably low emissions potential.

As a final comparison, emissions were also estimated for a catalytic combustor. The catalytic combustor emissions index is estimated to be less than 0.2 for NO_x and very nearly zero for HC. This concept has the potential for bettering the emissions from the other types of combustors considered. Because it is at this time primarily a laboratory device and has only been demonstrated for gaseous fuels, considerable development is required.

Engine Noise Estimates

Operating experience with Brayton cycle power systems and aircraft jet engines has shown the compressor to be the major source of objectionable noise in such systems. The characteristic jet engine whine and high level broadband noise occurs at the air inlet in a fan or compressor stage. Another source of noise is the high velocity exhaust stream. This is broadband noise from the dissipating jet.

It has been found that closed Brayton systems for space electric power are inherently quiet and free of objectionable noise. This is partly due to the sealed working fluid loop so that the turbomachinery inlet and exhaust streams are not open to the atmosphere. Therefore, it is felt that closed-cycle engines (groups I and II) would have no objectionable noise and could easily meet urban noise standards. This would also be true for the group IV engine since no compressor is used and the turbine exhaust is condensed for recirculation to the combustor. However, the potential for objectionable noise, from at least one source, does exist in both the group III and group V engines.

The noise sources in the engine groups III and V have been evaluated and the estimated noise levels of these sources have been determined for each application. The noise levels are given in the units of decibels (A), a measurement that is weighted for the varying sensitivity of the human ear with frequency. The noise levels are specifically for maximum power at a distance of 50 feet from the source. The exposure of humans to these conditions would generally be short and infrequent. A bus would normally develop the maximum 400 horsepower only when accelerating to high speeds, such as on entering a freeway or passing at high speed. A train would likely not use full power for acceleration until clear of the terminal area. A conservative estimate of the noise level that is objectionable for prolonged exposure is 90 dB(A).

Table D-2 shows estimated noise levels obtainable by acoustical treatment for those sources that are near or exceed the 90-dB(A) level. Two values for acoustical treatment are shown. The term "treated engine" refers to treatment that is an integral part of the engine. In the case of compressor noise, splitter rings are inserted into the inlet duct, and all internal surfaces are lined with acoustically absorbing material. Figure D-8 illustrates this type of treatment in a turbofan aircraft engine. Engine treatment for exhaust noise also consists of acoustic lining of the

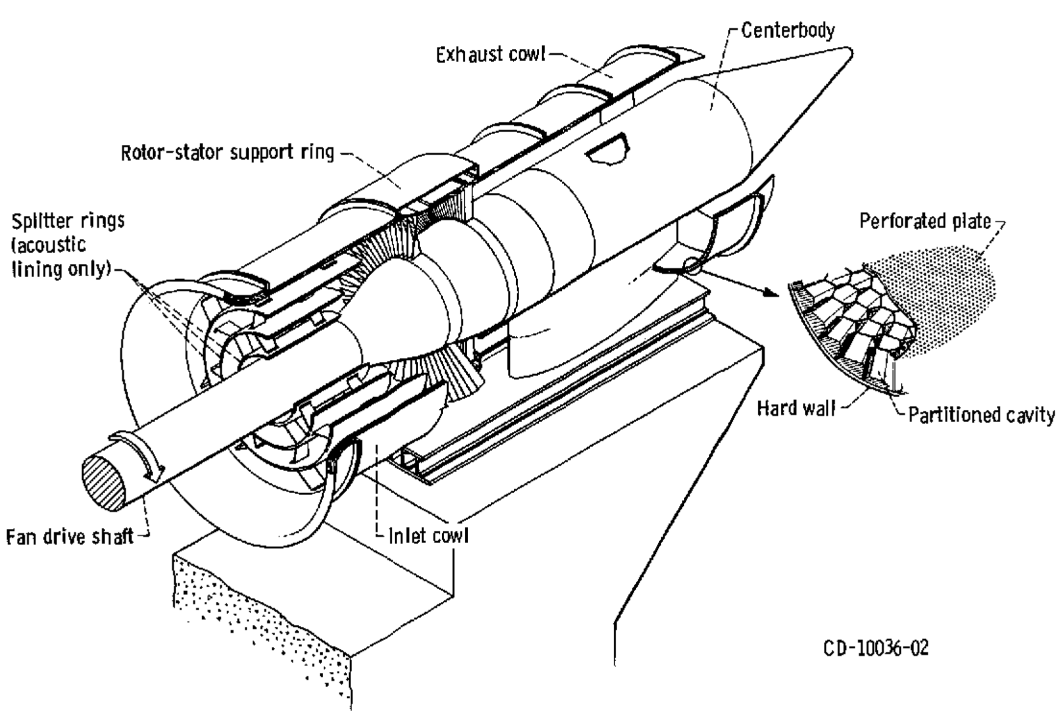

Figure D-8. - Cutaway view of fan and suppressor assembly.

TABLE D-2. - MAXIMUM POWER NOISE ESTIMATES

(a) Group III train

	Turbocharger		Engine exhaust	Cooling fan
	Inlet	Exhaust		
	Noise estimated, dB(A)±2 at 5 ft			
Engine	106	46	105	85
Treated engine	91	--	77	70
Engine in treated compartment	<75	--	<75	<70

(b) Group III bus

	Compressor inlet	Turbine exhaust	Exhaust	Cooling fan
	Noise estimate, dB(A)±2 at 5 ft			
Engine	95	48	103	77
Treated engine	80	--	77	---
Engine in treated compartment	<75	--	<75	<75

(c) Group V train and bus

	Application	Compressor inlet	Recuperator exhaust
		Noise estimates, dB(A)±2 at 50 ft	
Engine	Train	113	43
Treated engine		98	---
Engine in treated compartment		<80	---
Engine	Bus	101	35
Treated engine		86	---
Engine in treated compartment		<75	---

duct walls; however, additional air must be mixed with the primary exhaust jet to reduce the exhaust velocity and noise level. An air ejector is one method of inducing additional airflow and mixing.

The reduced noise levels for "engine in treated compartment" (table D-2) refer to an alternate or additional method of reducing the noise levels. In this case, the entire propulsive unit is contained well within the walls of the engine compartment. The compartment walls, as well as some additional noise diverting bottles, are lined with acoustically absorbing material. Extremely high levels of noise reduction are obtainable with this method. The penalty for such reductions, however, is an increase in the size of the engine compartment.

The noise estimates that are tabulated are based on the experience and noise correlations developed under aircraft noise reduction programs being conducted at the Lewis Research Center. The predicted noise levels are based on performance and design parameters specified in sufficient detail for the use of existing prediction formulas. For the untreated noise levels, conventional design practice for turbomachinery is assumed. The level of compressor noise can be reduced by incorporating some of the known low-noise design features developed for aircraft engines. These features have not been considered in the present estimates.

Remarks

Combustors considered in this study were advanced state of the art and developmental concepts. Swirl-can combustors were considered for groups I, III, and V. Group II assumed the use of a surface combustor. In addition, emissions estimates were made for a catalytic combustor.

From an emissions standpoint, no significant difference was found between kerosene and methane fuels. Hydrogen emissions are limited to NO_x, which makes it attractive as a future fuel. The group II system with a surface combustor had the lowest emissions potential of all groups. Groups III and V had comparable emissions at full power but were higher than group II. At idle group III had substantially higher emissions than group V. Emissions from the group III and V engines could be reduced to approximately the group II levels with a surface combustor. The catalytic combustor concept has the potential for even lower emissions than a surface combustor but considerable development is required.

In both the open and semiclosed (groups III and V) Brayton engines, the potential for objectional noise exists from at least one source. It can be concluded that this noise is controllable to below 75 dBA with existing technology. Generally, engine treatment will reduce the noise to acceptable levels. Only for the group V train application is engine compartment treatment clearly indicated to reduce compressor noise. The location of the compressor within the compartment is very important and tends to influence vehicle design. Therefore, consideration should be given to noise suppression in the initial stages of design and component packaging. Noise suppression may be particularly difficult when suppression devices are additions to a prepackaged propulsion unit. For the semiclosed systems (group III) noise could probably be reduced to be within 5 dB of the closed systems.

APPENDIX E
ENGINE COMPARTMENT LAYOUTS

Generalized design constraints that affected engine performance were dictated by considerations of power system integration into the various vehicles evaluated. Each of the power systems could have been separately optimized for each vehicle, but the objective of this study was a broad assessment that sought, not so much to optimize the design for each, as to display the potential for diverse application of the conceptual systems studied.

The objective of this section is to display how the assumed vehicles constrained engine design as part of the conceptual design phase. Engine design configurations were made similar to allow comparison between each of the conceptual engine designs. The design layouts were a necessary step in assessing the ability and characteristics of each engine group to meet vehicle envelope requirements in a way that tends to minimize the impact on vehicle design while at the same time providing flexibility, accessibility, maintainability, and reasonable efficiency in utilizing available vehicle space. The layouts presented in this section are of the group III and V engines only since these were the engines selected for the conceptual design comparison.

No attempt was made to redesign the various vehicles to adapt these conceptual engines in an optimum way. This is a serious shortcoming when assessing the benefits that might accrue to the introduction of new powerplants, but it does not negate the validity of the selection process which was the objective of this study. For example, the TACV configuration used was evolved on the basis of wayside power. The design would most certainly have been different if on-board power had been considered from the start.

Vehicle Constraints

GUIDEWAY VEHICLES

For the high power applications, the 300-mph TACV application represented the most stringent packaging requirements from the standpoint of volume, size, and weight. The constraints for this vehicle were then used as the guideline for the over-the-road locomotive and 150-mph urban TACV applications and the calculated component designs were then configured within the acceptable envelope of each vehicle. As will be shown, these configurations still provided design flexibility not only for this application, but for the other vehicles considered as well.

The maximum available length of the engine compartments was assumed on the basis of 100 passengers, baggage, and crew space. The maximum compartment volume was then determined on the basis of the outside envelope with consideration of the air levitation plenums (approximately 2 ft high) beneath the compartment and allowance for thermal insulation, noise attenuation, and structure. These considerations led to the engine compartment envelope limitations shown in figure E-1. Within this envelope two approaches were examined for engine integration. The first was to place a separate engine compartment at each end of the vehicle and the second approach was to combine the two engines in a single compartment at either end of the vehicle. In all cases it was assumed the vehicle must be able to be propelled in either direction. The passenger compartment space of 50 feet was derived on the basis of six abreast seating with a central aisle, with allowance for bulkheads and separation space. This appears adequate, but minimal for comfort. If both engines were integrated into a single compartment, passenger space is increased to nearly 70 feet which appears more than adequate by current aircraft standards. Both approaches will be shown.

On the basis of these compartment envelopes, it was necessary to constrain the maximum dimensions of the waste heat exchanger to a 9 by 18 foot airflow frontal area - the maximum available. This assumed that the waste heat exchanger could be placed across the top of the engine compartment. This derived waste heat exchanger constraint was then used during both

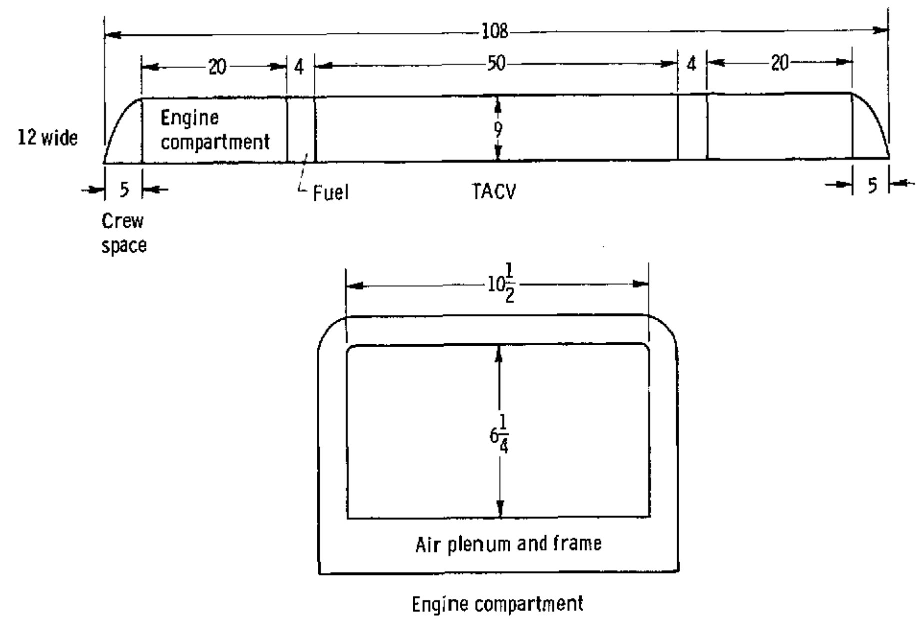

Figure E-1. - TACV envelope. (Dimensions are in ft.)

screening and conceptual design phases of the study and was a major influence on closed or semiclosed system performance. Since one of the guidelines was minimum or no vehicle redesign, it was assumed for all cases that waste heat exchanger cooling air was to be supplied by fans without benefit of ram air cooling. This allowed cooling air to be drawn in and exhausted through flush grills. However, air intakes could be provided at the front of the vehicle and ducted to the engine compartment. This approach required that cooling air be an engine parasitic power requirement which amounted to several hundred horsepower, depending on the design point selected. A benefit of this assumption on engine performance, was the ability to go off-design at low speeds where high powers were required while still maintaining design values of compressor inlet temperature. However, this assumption did penalize system performance at design since the power system had to provide cooling power. Any further evaluation of closed or semiclosed systems should include vehicle design interactions with the provision of ram air for both levitation and cooling.

The waste heat exchanger airflow length was not constrained by the vehicle, but was a parameter in engine optimization to limit pressure drop and hence parasitic fan power requirements. In the case of intercooling the 9- by 18-foot constraint air side frontal area was applied, which forced the optimization to treat both the waste heat exchanger and intercooler within this same frontal area constraint.

In the case of the urban TACV and locomotive, both of which were assessed at the 5000-horsepower level, the 9- by 18-foot frontal area limitation was compatible and no new constraint was imposed. It is, however, implicit that for these engines the size of the waste heat exchanger could have been smaller. The implication of this common constraint for the 5000-horsepower applications is to permit more waste heat exchanger area per horsepower which resulted in somewhat better performance at the lower horsepower. This is evident from the results shown in sections 5 and 7.

URBAN BUS

The primary guideline used in the conceptual layouts of the bus engine configuration was to maintain the "conventional" engine compartment of existing urban buses. Typically, the engine is in the rear, below the rear window and under the rear bench passenger seat. As shown in figure E-2 the engine compartment utilizes the full width of the bus extending upward to the rear window, approximately 66 inches, and half way to the rear axle, approximately 56 inches. The total volume is about 157 cubic feet. Fuel is stored in tanks located between the bus axles.

The major component constraint imposed was on the waste heat exchanger. On the basis of preliminary calculations, it appeared that the frontal area limitation on this heat exchanger would be 2 by 4 feet. However, as conceptual design proceeded, it appeared that this limitation could have been relaxed to two 4- by 4-foot heat exchangers, and this option was also laid out. In any event, even at the 2- by 4-foot constraint the available area per horsepower was greater than for the higher power applications, and it does not appear to be as significant a limitation.

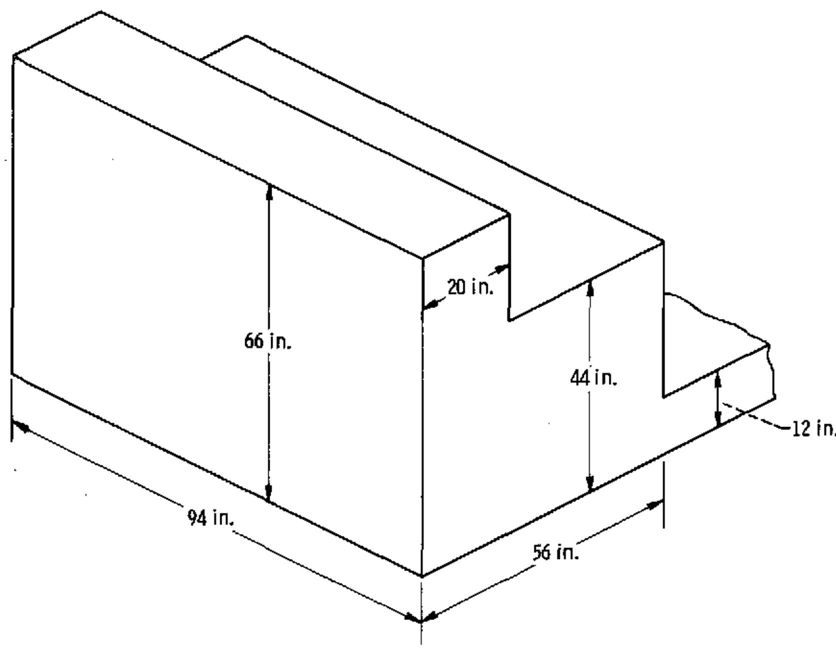

Figure E-2. - Bus engine compartment. Bus length, 40 feet; width, 8.5 feet; height, 10.25 feet.

TACV Layouts

SEMICLOSED BRAYTON SYSTEM (GROUP III)

The group III engine layout for the TACV is shown in figure E-3. This engine is a single-shaft unit producing 7500 horsepower electrical output from an alternator coupled through an infinitely variable transmission as shown. This configuration is for integration as two units in separate compartments one at each end of the vehicle. In this concept the waste heat exchanger fixes the compartment length. However, there appears to be no real constraint on volume utilization, which is more than adequate. This is illustrated conceptually in figure E-4. As shown, there is considerable freedom for arrangement of components, and no real attempt was necessary to compact the arrangement since packaging is not a serious problem. Figure E-5 illustrates this engine compartment integrated with a TACV. Based on component accessibility, maintenance of the engine should be straight-

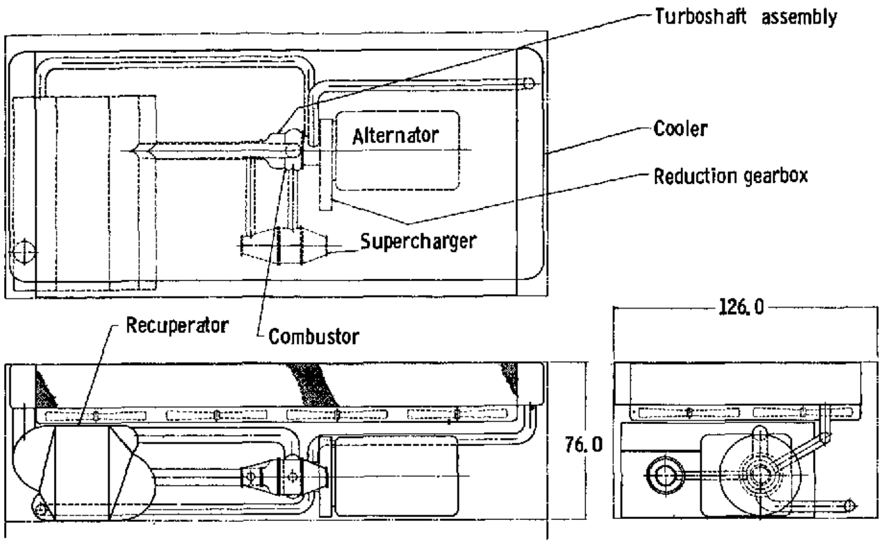

Figure E-3. - Group III 7500-horsepower, single-shaft TACV component layout.

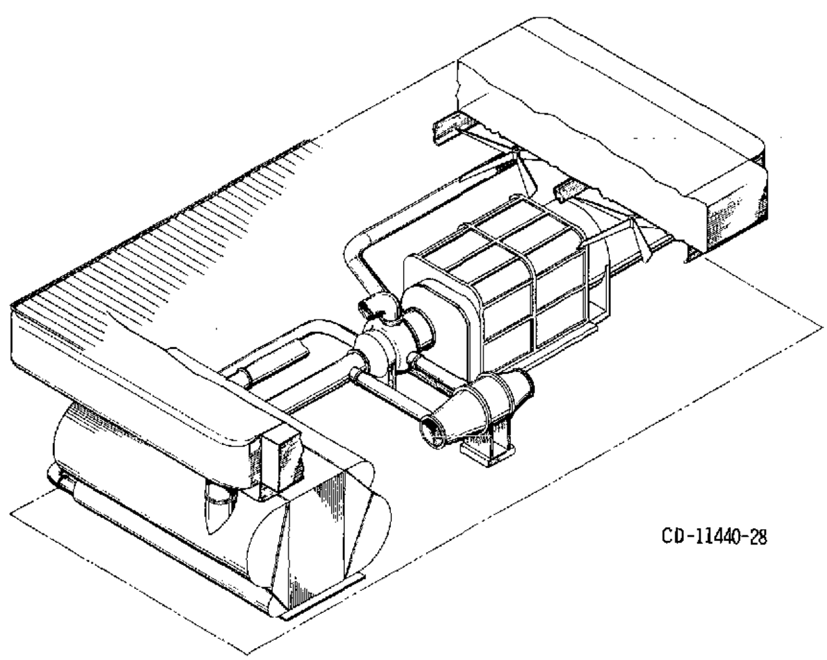

Figure E-4. - Group III engine compartment schematic layout.

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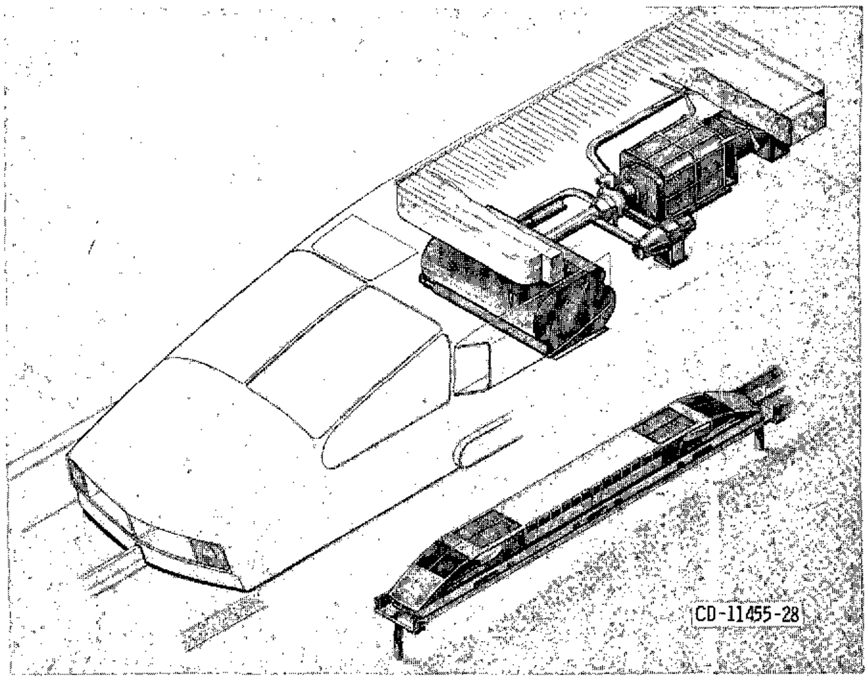


Figure E-5. - Group III 7500-horsepower TACV.

forward. The TACV has been shown with louvered sides and top for inlet and exhaust of cooling air. Scoops could also be provided, as shown, to take advantage of ram air cooling.

The other approach, integrating both engines in a single compartment, is illustrated in figure E-6. In this approach the compartment requires more attention to packaging. The packaging arrangement shown minimizes the length of hot ducts and still permits easy access for maintenance and/or replacement. The waste heat exchangers would be serviced as a unit from the top of the compartment. In this case the height of the vehicle had to be raised to accommodate the heat exchanger compartment. Cooling air would have to be ducted through scoops at the front and beneath the crew compartment. An isometric view of this engine arrangement is shown in figure E-7. Thus it appears that there is considerable flexibility in engine arrangement, and the waste heat exchanger dominates integration in either configuration.

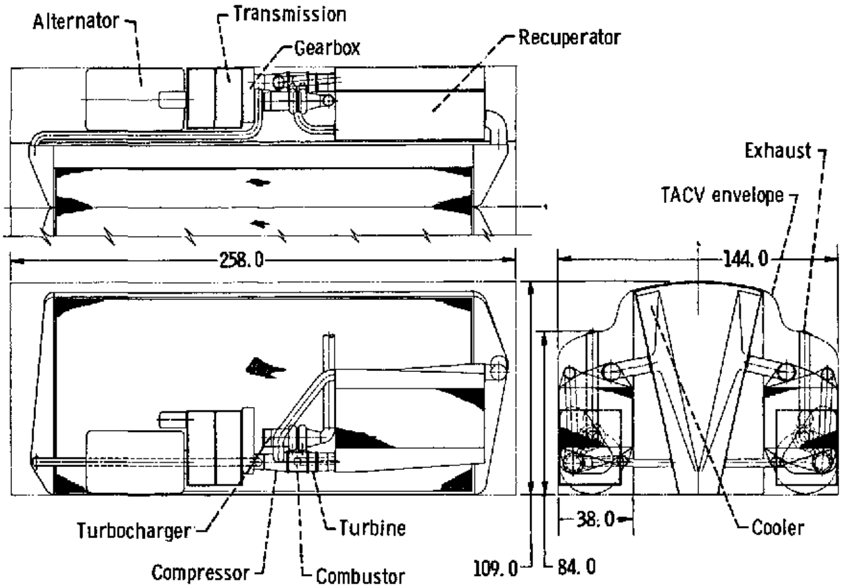


Figure E-6. - Conceptual design group III TACV (two engines in compartment).

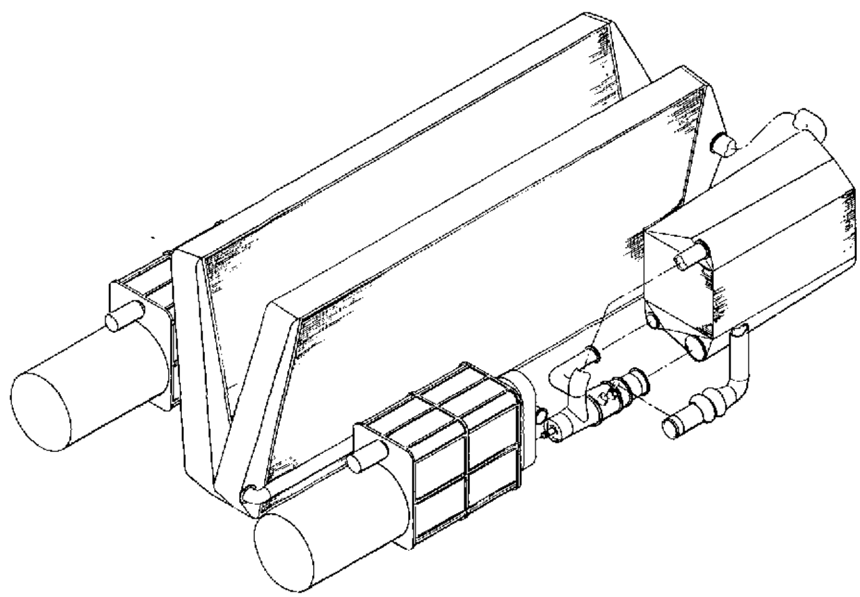


Figure E-7. - Conceptual design group III (two 7500-horsepower, single-shaft engines in compartment).

TABLE E-1. - GROUP III 7500-HORSEPOWER TACV

Component	Size	Weight, lb	Description
Rotating assembly:	------------------	1500	--------------------------------
Turbocompressor -			Single shaft; 30 000 rpm
Turbine	15 in. tip diam	----	Two stage axial; adiabatic efficiency, 89%; pressure ratio, 4.0
Compressor	9 in. tip diam	----	Five stage axial; adiabatic efficiency, 84%; pressure ratio, 4.4
Turbocharger -			Single shaft; 28 000 rpm
Turbine	13 in. tip diam	----	Four stage axial; adiabatic efficiency, 90%; pressure ratio, 24
Compressor	10 in. tip diam	----	Five stage axial plus one stage radial; adiabatic efficiency, 80%; pressure ratio, 25
Combustor	------------------	----	Annular can, diluent cooled
Recuperator	92 by 46 by 25 in.	8800	Plate fin with triangular and manifold; effectiveness, 0.916
Waste heat exchanger	110 by 208 by 9.4 in.	5700	Plate fin; two pass, four module; effectiveness, 0.94
Gearbox and transmission	------------------	9000	Infinitely variable; 5:1 reduction into transmission
Alternator	23-in.-diam rotor; 58 in. long; 37-in.-diam casing	8400	Eight pole wound; efficiency, 94.5%

The components for each of these arrangements are the same and are summarized in table E-1 for the calculated design point shown in figure 2-5 of volume I. A description of the heat exchangers appears in appendix A.

OPEN, RECUPERATED BRAYTON SYSTEMS (GROUP V)

The group V engine integration is substantially simplified by the elimination of the requirement for a waste heat exchanger. These engines can easily be integrated either in a single compartment or two compartments, one at each end of the vehicle as in the case of the group III engine. Figure E-8 is a layout of the major group V components, and their arrangement is depicted

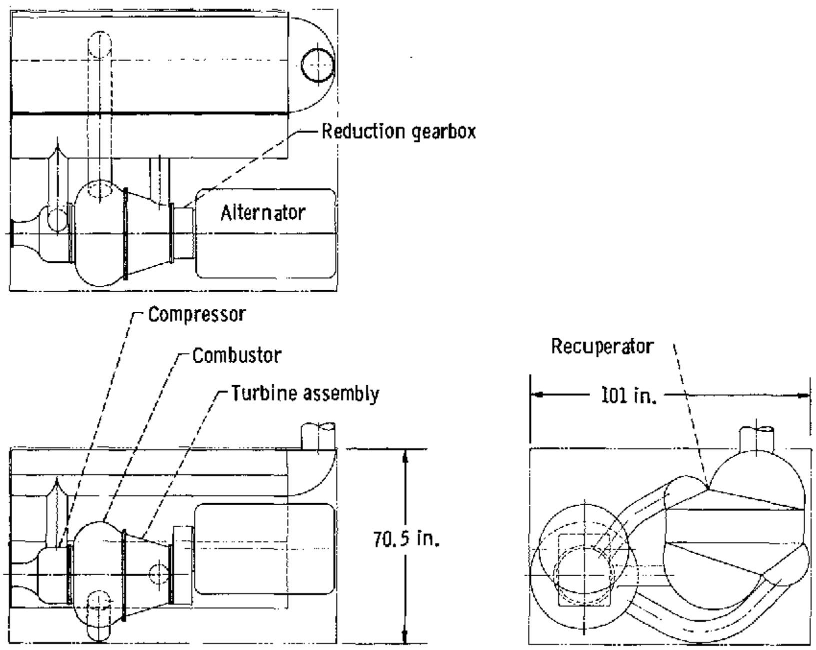


Figure E-8. - Group V (open cycle) 7500-horsepower, single-shaft TACV component layout.

in the isometric view shown in figure E-9 as they might appear in a single compartment.

This engine is compact enough that is might be palletized as a module for repair or replacement as a complete engine or as separate components. The exhaust of this unit is at the recuperator outlet and is ducted upward through the top of vehicle. Air inlets would be either from the sides or from scoops at the front of the TACV. Table E-2 summarizes the group V engine components for this application. In general the turbomachinery sizes and weights are higher than those for the group III engine since the open recuperated cycle tends to optimize at lower speeds and is at a lower system pressure level.

TABLE E-2. - GROUP V 7500-HORSEPOWER TACV

Component	Size	Weight, lb	Description
Rotating assembly:	--------------------	1500	--------------------------------
Turbocompressor -	--------------------	----	Single shaft; 13 000 rpm
Turbine	33-in. tip diam	----	Two stage axial; adiabatic efficiency, 89%; pressure ratio, 4.8
Compressor	21-in. tip diam	----	Six stage axial; adiabatic efficiency, 84%; pressure ratio, 5.5
Combustor	--------------------	----	Can; diluent cooled
Recuperator	120 by 60 by 12 in.	6000	Plate fin with triangular and manifolds; effectiveness, 0.916
Gearbox and transmission	--------------------	8500	Infinitely variable; 2.2:1 reduction into transmission
Alternator	23-in.-diam rotor, 57 in. long; 37-in.-diam casing	8000	Eight-pole ac wound; efficiency, 95%

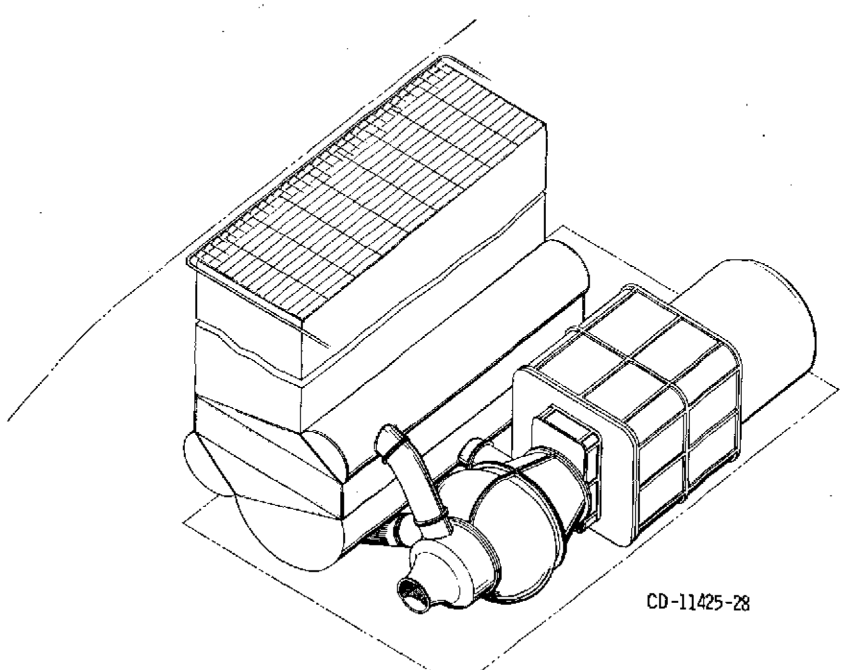


Figure E-9. - Assembled group V TACV engine.

Bus Layouts

SEMICLOSED BRAYTON SYSTEMS (GROUP III)

A group III engine was configured for installation in the 400-horsepower bus application shown in figure E-10. The engine shown corresponds to the design-point selection shown in figure 2-5(a) of volume I. This view and design represents the case where the waste heat exchanger frontal area is limited to 2 by 4 feet. Substantial volume is available for ancillary equipment such as air conditioning. If the entire volume were used, the waste heat exchanger could be enlarged to incorporate two units, each with a 4- by 4-foot frontal area. The performance improvement for this engine would approximate the unconstrained case for the 400-horsepower bus as shown in section 5. The reference case, however, assumes the 2- by 4-foot frontal area constraint for the waste heat exchanger. An exploded view of this engine is shown in figure E-11 broken down into the major assemblies. The

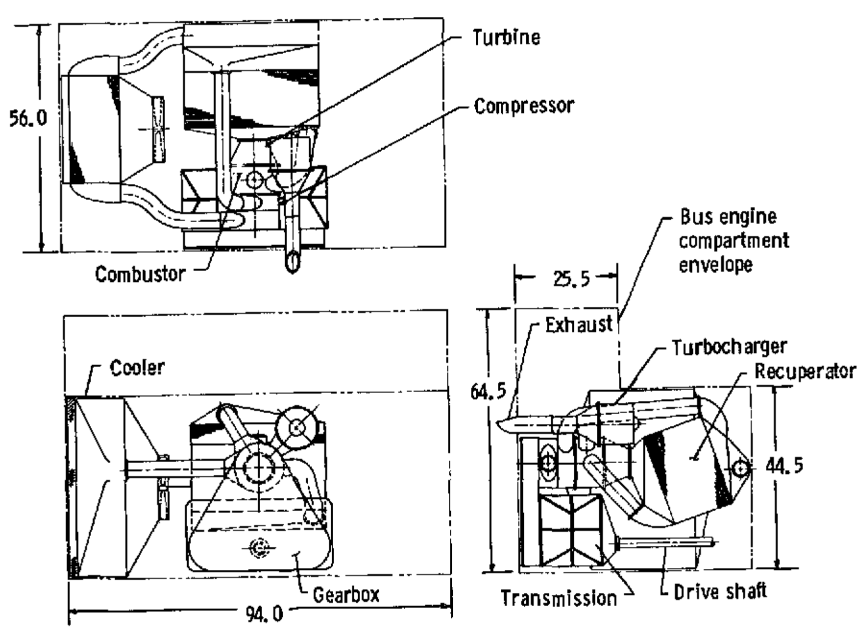


Figure E-10. -Group III 400-horsepower, single-shaft bus component layout.

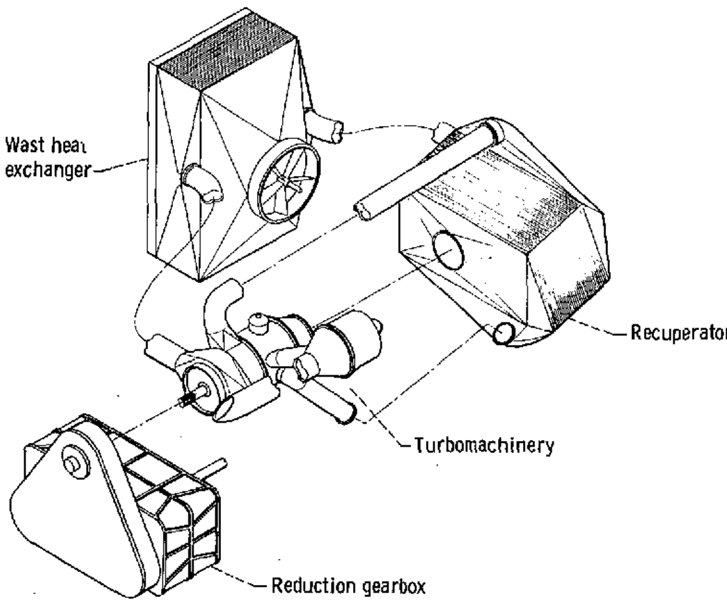


Figure E-11. - 400-Horsepower, single-shaft bus engine components.

reduction gear transfer case is directly coupled to the output shaft at the compressor (low temperature) end of the turbodrive system. The infinitely variable transmission is located on centerline beneath the turbine. This permits ready access for replacement or repair without disturbing the other components. Service can thus be either from the rear or below the engine compartment. The turbocharger is close-coupled to the turbocompressor but displaced to the side. This permits direct access to the turbocompressor unit itself or the combustor, which could be serviced from the top without interference. The turbine exhaust duct is kept short and dumps directly to the recuperator. Turbocharger exhaust is to the rear and upward. The waste heat exchanger is located on the left (traffic) side of the vehicle and exhausts downward. Temperatures and flow rates at idle should not pose a pedestrian problem. The waste heat exchanger can also be serviced directly and independent of the other major components. The recuperator is situated below the rear passenger bench seat, but thermally isolated by its own insulation. An objective of this layout was to minimize hot duct length and hence thermal losses and to minimize turning losses.

TABLE E-3. - GROUP III 400-HORSEPOWER BUS

Component	Size	Weight, lb	Description
Rotating assembly:	--------------------	200	--------------------------------
Turbocompressor -	--------------------	---	Single shaft; 72 000 rpm
Turbine	4.7-in. tip diam	---	Two-stage axial; adiabatic efficiency, 88%; pressure ratio, 3.4
Compressor	4.6-in. tip diam	---	One-stage radial; adiabatic efficiency, 82%; pressure ratio, 3.7
Turbocharger -			Single shaft; 100 000 rpm
Turbine	3.1-in. tip diam	---	Four-stage axial; adiabatic efficiency, 84%; pressure ratio, 11.5
Compressor	[a]3.2, 3.7 in. tip diam	---	Two-stage radial; adiabatic efficiency, 76%; pressure ratio, 12
Combustor	--------------------	---	Can; diluent cooled
Recuperator	34 by 17 by 11 in.	800	Plate fin with triangular end manifolds; effectiveness, 0.937
Waste heat exchanger	26 by 48 by 11 in.	200	Plate fin with triangular end manifolds; effectiveness, 0.90
Gearbox and transmission	--------------------	750	Infinitely variable; 12:1 reduction into transmission

[a]First stage, second stage.

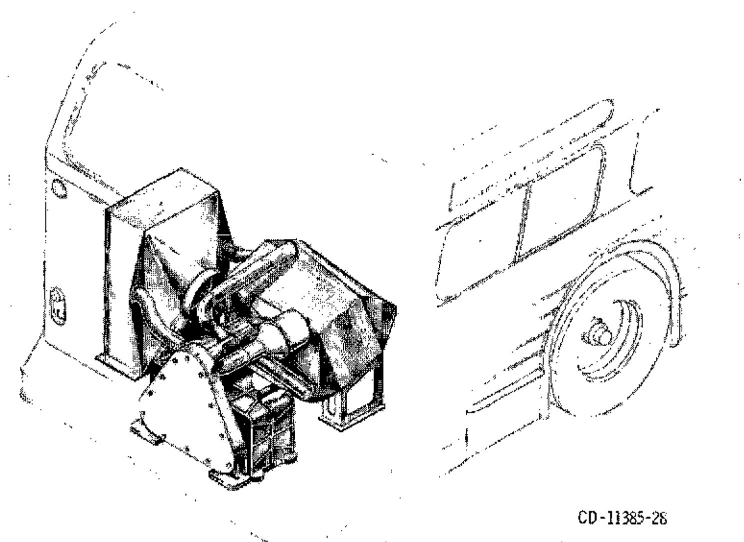


Figure E-12. - Conceptual group III bus engine configuration.

The performance characteristics and description of components are shown in table E-3. An artist's concept of this engine installation is shown in figure E-12.

OPEN, RECUPERATED BRAYTON SYSTEMS (GROUP V)

The open-recuperated, group V, 400-horsepower bus engine layout of components is shown in figure E-13. Again, as was the case for the TACV, the engine integration is far simpler because of the absence of the requirement for a waste heat exchanger. The general arrangement of the remaining components is the same as that for the group III engine. The gear reduction-transfer case is on the compressor end of the engine to the rear of the bus. The infinitely variable transmission is on the centerline below the turbomachinery package. Access to the turbomachinery, is, however, simplified by the absence of the turbocharger. The combustor is top mounted (same as group III) for ready access and replacement without disturbing other com-

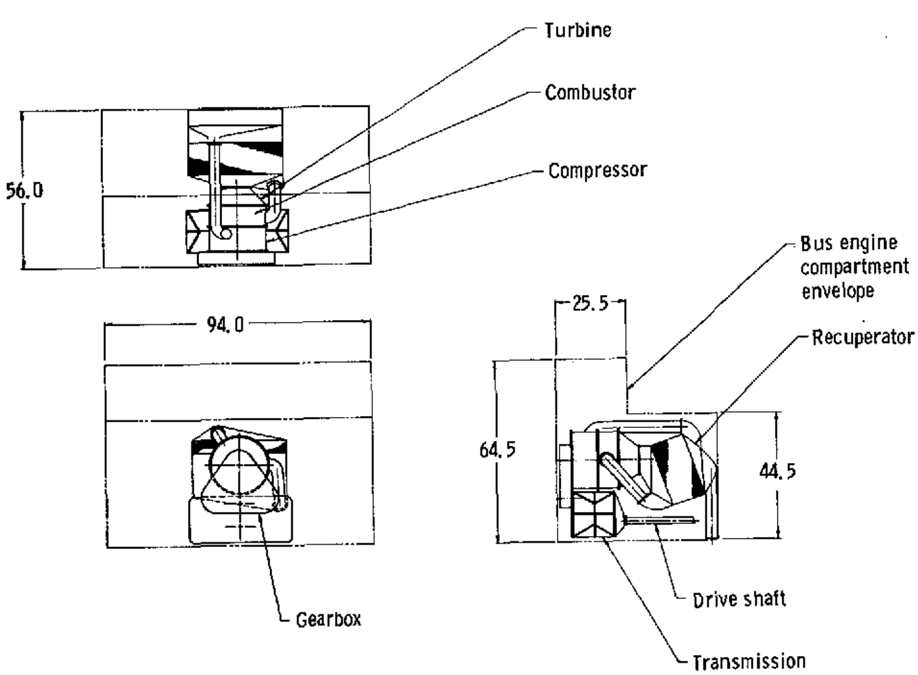


Figure E-13. - Group V 400-horsepower bus engine component layout.

ponents. An exploded view is shown in figure E-14. The recuperator exhaust can be more clearly seen in this view and is directed down beneath the bus. Alternatively this exhaust could be ducted as a single duct or divided into two symmetrical ducts exhausting up and to the rear.

The performance characteristics and description of components are shown in table E-4.

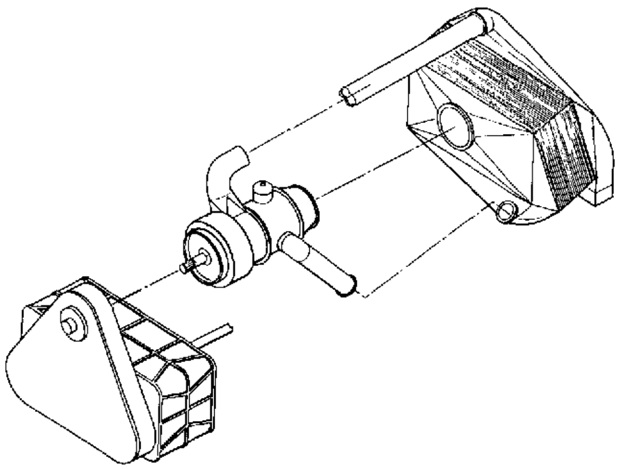


Figure E-14. - Group V 400-horsepower bus engine assembly.

TABLE E-4. - GROUP V 400-HORSEPOWER BUS

Component	Size	Weight, lb	Description
Rotating assembly:	---------------	250	------------------------------
Turbocompressor -	---------------	---	Single shaft; 51 000 rpm
Turbine	8-in. tip diam	---	Two-stage axial; adiabatic efficiency, 89%; pressure ratio, 4.3
Compressor	7-in. tip diam	---	One-stage radial; adiabatic efficiency, 83%; pressure ratio, 4.9
Combustor	---------------	---	Annular can; diluent cooled
Recuperator	34 by 17 by 11 in.	300	Plate fin with triangular end manifolds; effectiveness, 0.853
Gearbox and transmission	---------------	650	Infinitely variable; 8.5:1 reduction into transmission

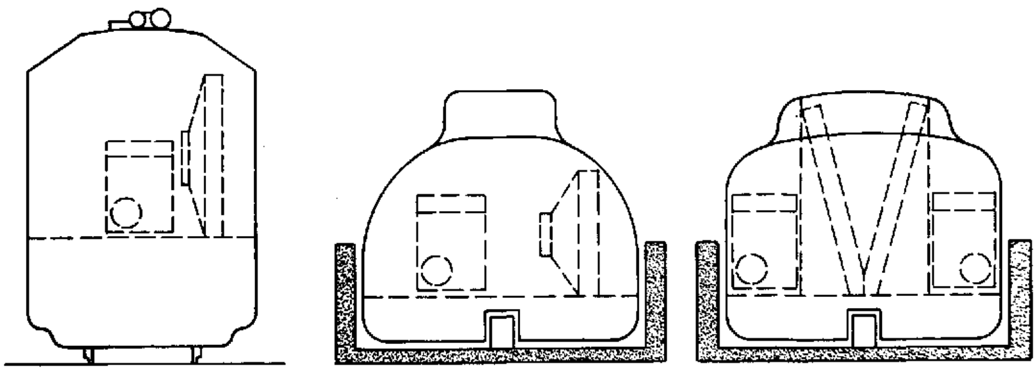


Figure E-15. - Engine installations for different applications of group III engines.

Urban TACV and Locomotive Application

A cursory look was taken at the integration of the 5000-horsepower two shaft, group III engine for an urban TACV and locomotive. It appears that there is no significant problem integrating the group V engine for these applications. In integrating the group III engine, the primary constraint was in cross-section geometry of the vehicles and its effect on waste heat exchanger dimensions rather than length. Figure E-15 is a cross section view of a typical diesel locomotive and urban TACV compared with the interurban TACV. This illustration assumes the same 9- by 18-foot flow cross-section limitation for the waste heat exchanger and indicates that no serious problem should be encountered integrating the group III engine for these applications on the basis of the concepts developed in this study.

Remarks

The layouts clearly show the impact on design of the waste heat exchanger for the semiclosed group III engine. However, there appears to be sufficient volume and dimensional flexibility in each of the vehicles evaluated for integration of the group III engine. The only case where vehicle geometry was substantially affected was in the case of two 7500-horsepower units in a single compartment. The turbocharger presented no integration problem. It would appear that the closed Brayton would pose the same constraints, but further aggravated by the requirement for the surface combustor and preheater.

The group V engine posed no problem of integration and, due to its simplicity and small volume, enjoys a substantial advantage on flexibility for adaptation. It is concluded that the vehicle geometry would be nearly independent of engine integration for the group V engine.

APPENDIX F

ENGINE-MISSION CONSIDERATIONS

This section describes the mission and vehicle input used for the conceptual design comparisons (groups III and V). These mission and vehicle inputs are required for two reasons: (1) the power demand on the engine as a function of time during a mission is required to determine the total fuel consumption, and (2) the vehicle imposes constraints in both weight and size on the engine that affect the choice of engine design point.

The calculation of the overall efficiency of an engine for a particular mission application can be divided into five steps. The first two, establishing engine design-point SFC and determining the variation in SFC with engine load, were the subjects of sections 4 and 5. The last three, choosing representative mission load demand as a function of time, determining transmission efficiency as a function of load and speed, and using all this information to determine the total fuel consumption are described in this section.

As explained in section 4, the engine design points were optimized under the constraint that the engine fit a compartment of the appropriate vehicle. In appendix E the conceptual layout of these engines within the vehicle engine compartments was described. The vehicle models used to determine the appropriate engine compartment sizes are discussed in this appendix. A vehicle total weight is used to provide a reference for evaluating engine weights. The effect of variations in engine and/or fuel weight on vehicle total weight and hence vehicle performance (acceleration capabilities or power requirements) was not considered in this section. However, because of the importance of power system total weight, its tradeoff with the mission fuel consumption is discussed in section 2.

The vehicle operational information needed in this part of the study may be divided into two parts: the vehicle model and the mission profile model.

It should be emphasized here that these models served as a framework for comparing types of Brayton engine (not for comparing on-board with wayside power in the TACV example). Hence, the models were adapted from existing design information and not redefined for this study.

For the vehicle model the main considerations were to establish the volume restrictions and the drag and performance characteristics. For the mission model (a profile of engine power versus time), many variations are, of course, possible. However, for the purposes of this study, comparison of the Brayton engines for only one representative mission or driving cycle was evaluated for each vehicle application.

Vehicle Models

Four vehicles, each with a representative mission profile, were considered: a typical urban bus, an interurban tracked air-cushion vehicle (TACV) that would serve an area such as the Northeast Corridor and cruise at 300 mph, an urban TACV that would cruise at 150 mph, and a typical railroad locomotive.

For the urban bus the vehicle envelope, specifications, and performance were taken directly from DOT specifications (ref. 1), which are essentially those of existing buses (see fig. E-2). The bus would use a 400-horsepower engine, larger than present buses to provide extra acceleration for, as an example, the entrances to high-speed highways, and for sustaining 70-mph speeds on steeper grades.

The TACV is a train type of vehicle supported by a cushion of air, propelled in this case by a linear induction motor and constrained to a guideway. Information for the TACV model was obtained from a survey of high-speed ground-transportation concepts (using wayside power collection for vehicle power) that resulted from previous DOT work (in particular, ref. 2). From the conceptual designs of reference 2, the total power needed by a 300-mph TACV is about 15 000 horsepower.

A representative vehicle was selected from reference 2. The vehicle envelope is shown in figure F-1; it is 108 feet long, 12 feet wide, and 9 feet high. The 12-foot width would allow a nominal 100 passengers to be seated

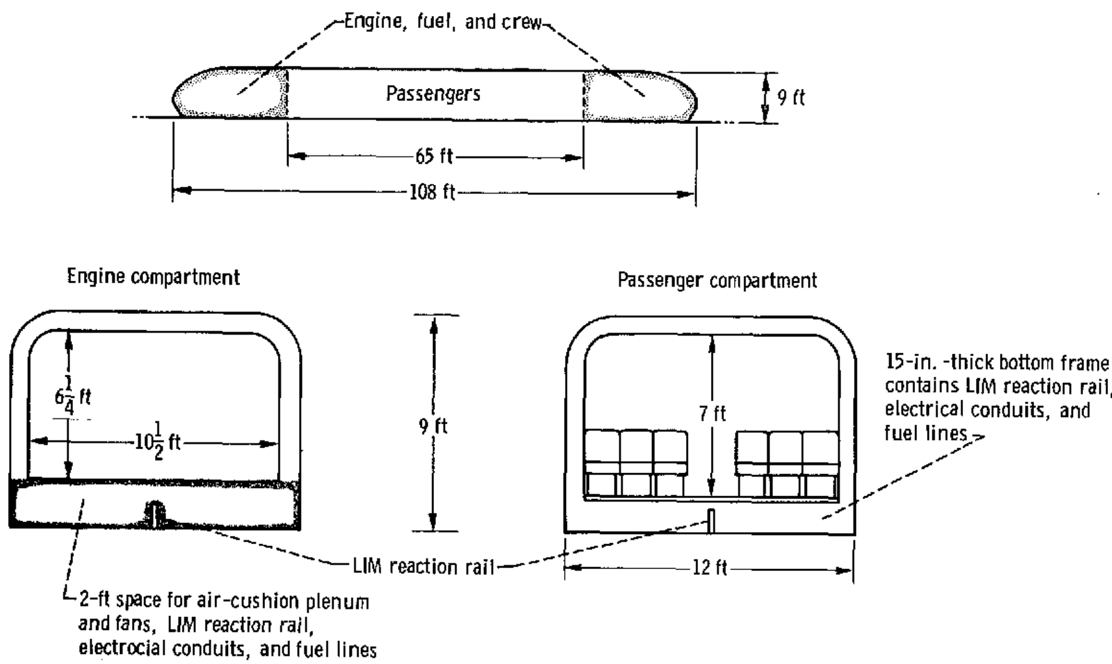


Figure F-1. - TACV train envelope.

six abreast, thus shortening the passenger compartment to 65 feet and allowing more room for the engines and fuel tanks. The 9-foot height allows ample passenger head room plus room for the air-cushion lift fans and plenum.

Because the TACV model is essentially that of reference 2, the drag characteristics (needed to establish the mission profile) in reference 2 were also used. The performance specifications for the TACV are a cruise speed of 300 mph, a 1-percent grade capability at all stages of the mission, and an acceleration and deceleration of 0.1 g (2.2 mph/sec). To determine the power needed to provide this performance, a guideline of 120 000 pounds gross weight was set, corresponding to that of a TACV with guideway power.

The urban TACV, designed for shorter commuter runs, would have a cruising speed of 150 mph, carry 60 passengers, weigh 60 000 pounds, and require only about 5000 horsepower.

A locomotive was considered because it is potentially a more immediate application of high-power Brayton engines by retrofitting an existing vehicle.

Furthermore, the weight restriction is not so severe for this application. The 5000-horsepower urban TACV engine variation was used for the locomotive, although this is somewhat higher power than current locomotives.

Mission Models

Although in actual practice power is a continuous variable, in this study no transient engine performance was considered. The increasing power during acceleration was calculated at 50-mph increments and then further grouped into steps of constant power. Braking was approximated as a single step and was considered to be dynamic - the engine providing only enough power in reverse so that, when added to the drag, a deceleration of 0.1 g is maintained.

URBAN BUS

The urban bus driving cycle (taken from ref. 2) is as follows: The bus range is 400 miles. For verification, the operating profile for this range is (a) 10 trips, 10 miles each at 50 mph, accelerating at maximum acceleration to 50 mph on each trip and (b) 600 trips, 0.5 mile each at 20 mph, accelerating at 0.1 g to 20 mph on each trip. For the 50-mph portion of this composite mission, a station stop time of 60 seconds is assumed. For the 20-mph portion, a station stop time of 20 seconds is assumed. The road load associated with various bus speeds and a selected acceleration profile is given in table F-1. The corresponding engine loads can be determined using the efficiencies of an infinitely variable transmission (table F-2). The engine load must also include a constant 60-horsepower for vehicle accessories that was assumed as representative.

The driving cycle thus translates into the composite mission profile shown in figure F-2(a). Note that the 20-mph portion of the mission requires less than 100 horsepower except for a brief peak during the 0.1-g acceleration. Similarly for the 50-mph portion the power needed stays under 200 horsepower except for a brief peak during acceleration. Hence, for this urban mission, 400 horsepower is needed only for brief periods when accelerating to 50 mph.

APPENDIX F - ENGINE-MISSION CONSIDERATIONS

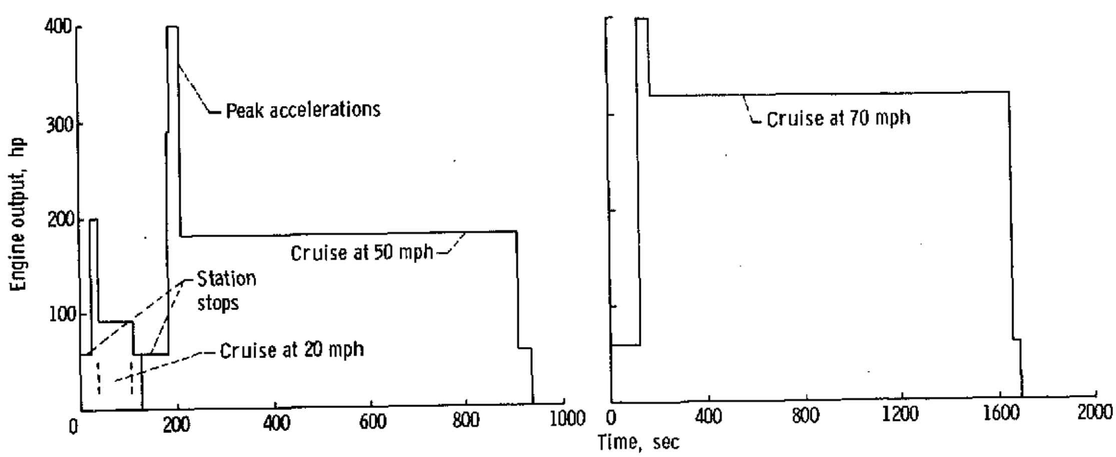
TABLE F-1. - BUS ROAD LOADS[a]

Speed, mph	Cruise road load, hp	Accelera-tion, g's	Accelera-tion load, hp
5	3	0.20	94
10	6	.18	168
20	13	.14	262
30	25	.09	258
40	50	.06	224
50	85	.04	173
60	130	.015	79
65	165	.007	42
70	210	-----	0

[a]Maximum power delivered to wheels is about 290 hp at about 85 percent engine efficiency.

TABLE F-2. - EFFICIENCIES

Device	Power, percent	Engine speed, percent	Effi-ciency, percent
Electric motors (TACV)	---	---	85
Rear axle (bus)	---	---	95
Transmission (infinitely variable)	10	60	50
	25	60	70
	40	80	75
	45	80	75
	75	90	80
	90	100	85
	100	100	90



(a) With two stops, after 0.5 mile (20-mph cruise) and after 10 miles (50 mph cruise).

(b) 30-mile, 70-mph-cruise trip.

Figure F-2. - Bus mission.

If the same bus is considered for an interurban mission, traveling on interstate highways at 70 mph (fig. F-2(b)), 400 horsepower is much more appropriate. Assumptions for this comparative bus mission are a station stop time of 2 minutes and trip segments of 30 miles. This comparison of the urban and interurban missions emphasizes their different power needs: most of the time the urban bus will need either 100 or 200 horsepower; the interurban bus will need over 300 horsepower.

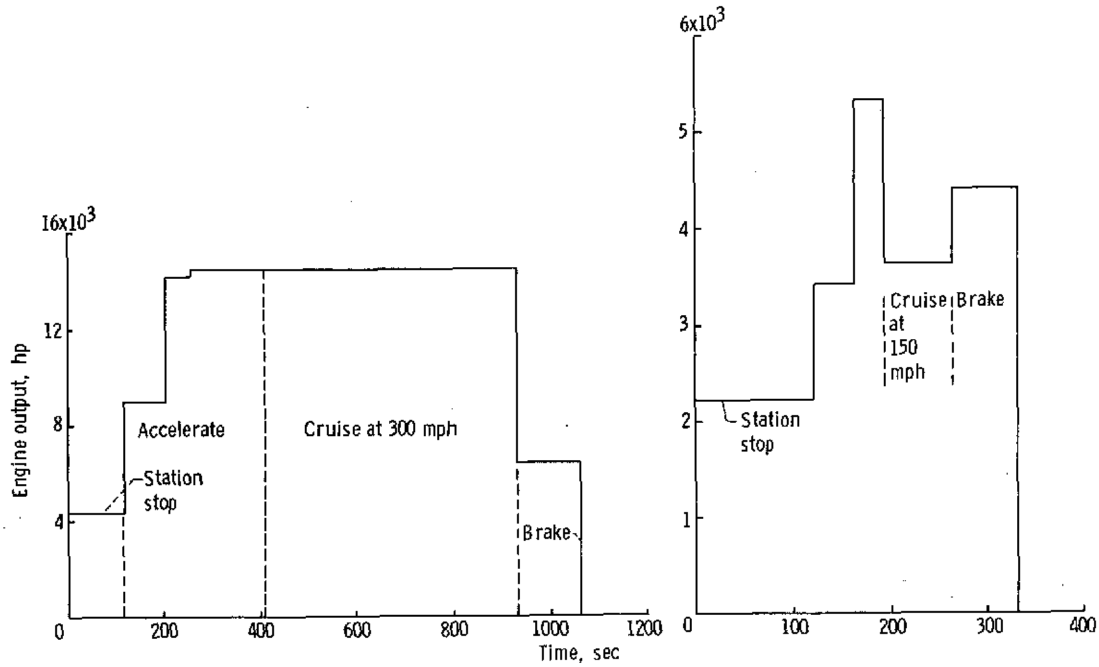
INTERURBAN TACV

The TACV mission profile was chosen to be a 2-minute station stop, constant acceleration at 0.1 g (2.2 mph/sec) until cruise speed is reached or a power limitation forces the acceleration to be reduced, cruise at 300 mph, and constant dynamic braking at 0.1 g (reverse power). During the station stop the TACV is assumed to operate on the air cushion. For simplicity the aggregate effect of several factors, supplemental lift from dynamic and ram airflow, transmission and motor efficiencies, variations in cushion air needed for guidance, was considered constant for all vehicle velocities.

Three types of drag were considered - aerodynamic, skin friction, and ram air. Based on equations and coefficients given in reference 2, the total drag and the propulsive power needed to overcome it were determined for various velocities. In 50-mph increments the propulsive power needed for drag, acceleration, and lift (including guidance) was determined. Once this road load was determined, an 85 percent electric motor and LIM efficiency and the transmission efficiency (dependent on engine speed) (table F-2) were used to obtain the engine output power.

A station-to-station distance (trip length) of 55 miles was selected; eight of these trips would equal the length of the northeast corridor from Boston to Washington (about 440 miles).

The profile of the engine power needed versus time into the mission is shown in figure F-3(a). The power for acceleration, while computed at 50 mph increments, fell conveniently into three steps. Based on the assumptions and approximations of this study, a power limitation forces the acceleration to drop below 0.1 g before reaching 300 mph. The assumption of



(a) Interurban TACV; one 55-mile 300-mph-cruise trip.

(b) Urban TACV mission; one 5-mile, 150-mph-cruise trip.

Figure F-3. - Interurban TACV missions.

1 percent grade capability during this acceleration and during cruise would boost the power requirement to more than 15 000 horsepower. The total time for one trip of 55 miles including station stop is about 17.6 minutes for an average speed of 190 mph.

URBAN TACV

The urban TACV mission profile is shown in figure F-3(b). Again, a 2-minute station stop is assumed; acceleration is constant at 0.1 g until 150 mph is reached, cruise is at 150 mph followed by constant dynamic braking at 0.1 g. The trip length is 5 miles. Note that the initial power for braking is greater than the cruise power, but that in figure F-3(a) for the 300-mph TACV this was not the case. (At 300 mph the drag provides enough decelerating force to achieve the 0.1 g initially.) Based on the assumptions and level of detail in these calculations, the 0.1-g acceleration just before reaching cruise speed would require slightly more than 5000 horsepower. The time between stops is 5.5 minutes for an average speed of 55 mph. Although it is unlikely that a fast urban commuter train would make such frequent stops (distance between stops, 5 miles), this choice of mission provides a comparison with Brayton engine performance when the engine operates off-design much of the time.

TABLE F-3. - GENERAL MOTORS ROAD LOCOMOTIVE LOAD-SPEED CYCLE[a]

Throttle position[b]	Percent of time
[c]8	30
7	3
6	
5	
4	
3	
2	
1	
Idle	40
Dynamic brake	8

[a]Ref. 3.
[b]Nonidle time is at an average of 65% power.
[c]For this study assumed to be full engine power.

TABLE F-4. - MISSION PROFILE SUMMARY

Vehicle	Total power, hp	Percent of time at power	Percent of power
TACV	15 000	15	25
		15	40
		5	70
		50	85
		15	95
Urban TACV	5 000	35	35
		15	55
		22	60
		21	70
		7	100
Locomotive	5 000	40	Idle
		30	35
		30	100
Bus composite mission	400	23	15
		61	22
		8.7	45
		6.9	50
		.3	95
Bus (70 mph) mission	400	8.5	15
		89	80
		2.5	90

LOCOMOTIVE

The power needs of a present locomotive were studied for two reasons: (1) to provide a contrast with the 300-mph TACV, which spends much of its operating time at or near design point (presently a large fraction of the locomotive engine operating time is at an idea condition) and (2) to examine a derivative application in which retrofitting existing vehicles might be both desirable and easy to accomplish.

The locomotive mission, the General Motors Load-Speed Cycle (table F-3), is taken from an unpublished report by Battelle Columbus Laboratories. In this cycle the engine idles 40 percent of the time and is at an average of 65 percent power during the nonidle time. The engine is operated at full power about 30 percent of the time. Hence, accelerating and braking is at an average of 30 percent power about 30 percent of the time.

Mission Specific Fuel Consumption

To compare various design points for an engine, it proved useful to define an aggregate parameter, mission SFC, as the total fuel used in a mission divided by the total number of horsepower-hours the engine was operational during that mission. The total fuel needed per mission was determined from the mission profiles (engine power versus time) of figures F-2(a) and F-3, table F-3, and the engine SFC versus power level curves of section 5. A summary of the mission profiles reduced to a form useful for calculational purposes is given in table F-4. Integration of each mission profile provided the number of horsepower-hours. The fuel weights for each vehicle were determined as follows:

For the 300-mph TACV fuel is provided for ten 55-mile trips; stop time between trips is 2 minutes (eight trips constitute a representative northeast corridor). For the urban bus one tank of fuel must provide for six hundred 0.5-mile trips (assuming 20-sec stops) and ten 10-mile trips (assuming 60-sec stops). (See ref. 1.) The urban TACV fuel weight arbitrarily provides a 200-mile range of forty 5-mile trips (2-min stops). The locomotive fuel tank arbitrarily provides 10 hours of operation.

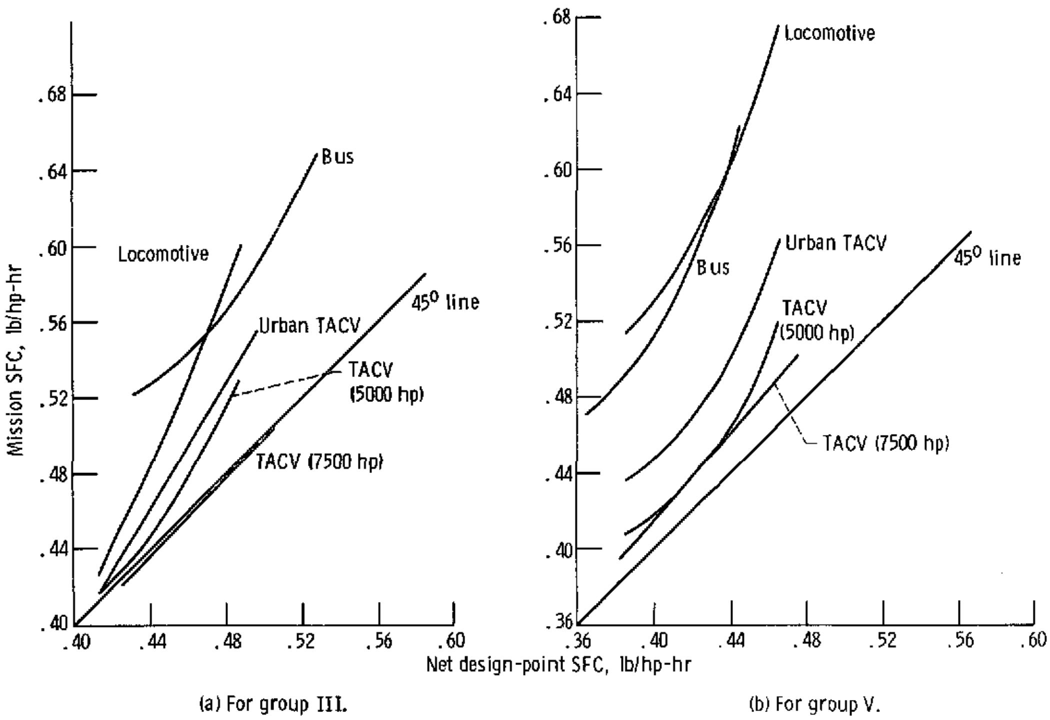


Figure F-4. - For group III.

In figure F-4 the mission SFC is plotted against design-point SFC for engine groups III and V. Note the 45° line; if the engine operated at design point, all the time or if the SFC did not vary with engine load, the mission SFC line would coincide with the 45° line. However, the engine efficiency does decrease as the engine load decreases. (See figs. in sec. 8.) Thus, vehicles that spend more time at low power will generally have a greater difference between mission SFC and design-point SFC. This assumption is supported by figure F-4. However, an exception in figure F-4(a) shows the 300-mph TACV curve lying slightly below the 45° line, reflecting a better off-design than design-point performance at power levels where the engine would spend much of its operating time (fig. 5-10).

Remarks

Approximate vehicle and mission models and their assumed and derived characteristics (weights and powers) were used to allow an assessment and comparison of the potential performances of these Brayton engines. The mission models described previously were used with the results of section 5 to determine the mission SFC's in figure F-4. The mission SFC for the group III engine is less sensitive to the mission than group V because of its flatter variation of SFC with off-design power levels.

Except for the bus mission, the mission SFC for the group III engine is less than the group V engine for the same design-point SFC. However, as shown in section 5 the engine weights for groups III and V differ and in each case vary considerably with design-point SFC. The optimum choice of the design-point SFC is, therefore, likely to be different for the two engines, and the comparison of the performance for any mission must be made using the optimum design point for each. This is done in section 2 using the information of this section and the engine weight from section 5.

An urban bus would need about 200 horsepower during most of its mission and about 400 horsepower for only brief periods during acceleration. It would be desirable to design its engine for lower power with the provision for overpowering to provide peak power needs.

References

1. Specification for Design and Performance of 40 Ft. Transit Buses. Booz, Allen Applied Research, Inc., 1971.

2. HSGT Systems Engineering Study, Tracked Air Cushion Vehicle. TRW Systems Group (NECTP-219), 1969.

APPENDIX G
FUELS STUDY

Recognizing national concern over increasing pollution and the developing shortage of petroleum and natural gas, the Department of Transportation requested that this engine study include an evaluation of methane and hydrogen as potential fuels for future transportation systems. This section of the study treats the production and distribution of these fuels including the practical considerations of cost and safety.

Hydrogen and methane are evaluated as fuels for powering advanced high-speed ground transportation systems at levels of energy use appropriate to (1) a single regional system and (2) a nationwide system. For the regional system the northeast corridor was selected and a model established based on DOT studies. The model assumes a high-speed, tracked, air-cushion vehicle (TACV) system operating between five major cities - Washington, D.C.; Philadelphia, Pennsylvania; New York, New York; Milford, Connecticut; and Boston, Massachusetts. Production, distribution, and fueling problems are evaluated specifically against this model. For the nationwide system it was not practical to establish a model and evaluate detailed production and distribution problems. Instead, the cost evaluations generated for the northeast corridor were extrapolated to the energy consumption levels estimated to be appropriate to the nationwide system level (assumed equivalent to six regional systems). In addition, the cost of a gas pipeline system potentially applicable to a nationwide transportation system was evaluated.

Three fuel-oxidizer systems, hydrogen-air, methane-air, and hydrogen-oxygen, were evaluated and compared with the conventional systems, gasoline-air and kerosene-air. The use of hydrogen as a fuel with air eliminates all combustion pollution problems except for the generation of

NO_x and, when burned with pure oxygen, eliminates the NO_x also since the only combustion product is water vapor. Methane, although it does not have the pollution free potential of hydrogen, is cleaner burning than gasoline and kerosene. It is also available today in the form of natural gas (90 percent methane) and is being considered for large-scale production from coal.

Hydrogen is expensive and of low density but is a candidate for the fuel of the future for land transportation. Hydrogen is readily produced from water by electrolysis. In the future it is anticipated that nuclear or solar power could be used to produce hydrogen by electrolysis or advanced thermal or chemical processes. In the interim, hydrogen could be produced from coal or methane. During the interim, government policy will influence the production and consumption of oil and natural gas, possibly for industries other than transportation. This would lower the supplies of liquid petroleum gases (propane, butane) and liquid and compressed natural gas, which usage is currently increasing in trucks, autos, and buses. Government agencies and various automotive and transportation companies are converting increasingly more automotive vehicles to the use of cleaner burning fuels in the effort to clean up the atmosphere.

Two significant areas must be considered if liquid hydrogen or liquid methane should be adopted for a nationwide transportation system. The first is the complexity of the nationwide logistics system that would be required to manufacture, distribute, and store these cryogenic fuels. Next is the nationwide education and training program that would be required to assure the safe use of these fuels, not because they would necessarily be any less safe than conventional fuels but rather because the safety and handling procedures would be new and different from those presently used for conventional fuels.

The study provides information and data on the physical characteristics of the fuels; the production processes for hydrogen and methane; the distribution, storage, and loading for hydrogen, methane, and oxygen; facility costs for hydrogen; product costs for hydrogen, methane, oxygen, gasoline, and kerosene; and some safety aspects of hydrogen, methane, and oxygen.

Summary

FUELS

Hydrogen may be produced from various sources and processes. These include steam reforming of natural gas, partial oxidation of petroleum, coal gasification, electrolysis of water, and thermochemical separation of water. Hydrogen produced from coal starting in the 1980's and from nuclear powered electrolytic plants starting in the 1990's should receive strong consideration as a transportation fuel of the future. Methane may be produced from natural gas, petroleum, coal, and organic wastes by a variety of processes. For most applications natural gas would, of course, be used "as is" rather than separating it to obtain pure methane. Because of volume limitations in most transportation applications, hydrogen and methane would likely be stored in liquid rather than gaseous form. A review of the physical properties of liquid hydrogen and liquid methane reveals some of the key considerations of the use of these fuels compared with more conventional fuels such as kerosene. Considering the differences in densities and heating values, a liquid hydrogen tank would require more than four times the volume of a kerosene or gasoline tank to have the same energy content; and a liquid methane tank, nearly twice the volume. In addition, liquid hydrogen and liquid methane must be stored at cryogenic temperatures: -423° F for hydrogen and -258° F for methane. This generally will require the use of vacuum-insulated tanks for both fuels, although more conventional insulation may be adequate for some methane applications. Storage of hydrogen as a hydride is a possible alternative to storage as a liquid. Hydrides allow hydrogen storage at ambient temperatures at approximately the same density as liquid hydrogen. However, the hydrides themselves are heavy, so that, although the volume is equivalent, the weight of the storage system is substantially greater than with liquid hydrogen. The heat that must be added to release the hydrogen could be obtained from the engine exhaust. Additional research and development are still required to provide a lightweight, safe, and inexpensive hydride storage system applicable to transportation vehicles.

NORTHEAST CORRIDOR TACV

Hydrogen and methane were evaluated as fuels for powering a regional, advanced, high-speed ground transportation system. The northeast corridor was selected for this evaluation, and a model established based on existing DOT studies (ref. 1). The model assumes a high-speed, tracked, air-cushion vehicle (TACV) system serving five major cities - Washington, D.C.; Philadelphia, Pennsylvania; New York, New York; Milford, Connecticut; and Boston, Massachusetts. The model assumes 60 four-car trains per day, each way, between Washington and Boston; and 43 three-car trains per day, each way, between Philadelphia and Milford. Each car is assumed to be self-powered, to have an average power consumption of 12 000 horsepower, and an average speed of 212 miles per hour. Based on these assumptions and the assumed fuel consumption rates of 0.40 and 0.15 pound per horsepower-hour for methane and hydrogen, respectively, and allowing for cryogenic handling and distribution losses of 10 and 25 percent for methane and hydrogen, a production rate requirements of 2.734×10^{6} pounds per day of hydrogen and 6.145×10^{6} pounds per day of methane results. Fuel costs based on present production techniques for hydrogen and methane were estimated and compared with gasoline and kerosene. No credit was taken for recovery and secondary use of the cryogenic fuel losses. In summary, the cost (in cents) per passenger mile was estimated as follows (based on 1972 costs):

Liquid hydrogen -	
From natural gas or petroleum	1.8
From coal	2.1
From electrolysis	3.0
Liquid oxygen from air separation	1.3
Liquid methane -	
From pipeline natural gas	1.2
From shipped liquid natural gas	2.1
Kerosene	0.7
Gasoline	0.8

Hydrogen costs do not appear to be competitive today. However, it is anticipated that limited petroleum supplies will force gasoline and kerosene prices up while electrolytic production of hydrogen with nuclear-generated power will reduce hydrogen costs. Thus, hydrogen may become competitive in the 1980's with coal gasification and in the 1990's with electrolysis.

NATIONWIDE TRANSPORTATION SYSTEMS

The results of the estimates for the northeast corridor were extrapolated directly to provide some insight into the magnitude of production requirements and costs for a nationwide advanced transportation system. The estimated energy requirements for all-passenger rail and intercity bus needs for 1985 is approximately 0.3×10^{15} Btu per year or about six times the northeast corridor demand. The yearly fuel requirements and cost are as follows:

System operation area	Hydrogen			
	Yearly requirement, lb	Cost of electrolysis	Yearly requirement, lb	Cost
Northeast Regional	1.00×10^9	$\$180\times10^6$	2.40×10^9	40×10^6
Nationwide	5.82	1048	13.97	233

A gas pipeline system for hydrogen or methane was laid out for a nationwide advanced transportation system by superimposing the potential pipeline system over the existing U.S. natural gas pipeline system and terminating it at strategic distribution locations. This system includes 7447 miles of 30-inch pipe and 196 booster stations at a total cost of $\$4184\times10^6$. It would be capable of delivering all the fuel (hydrogen or methane) required for a nationwide system (0.3×10^{15} Btu/yr). Production facilities at supply points and liquification facilities for final distribution are not included in the cost estimates.

SAFETY AND REGULATIONS

Hydrogen and methane, both as gases and cryogenic liquids, have been handled on an industrial basis for the past 20 years in large quantities. Satisfactory safety procedures have been established and shipment of liquid hydrogen and methane by truck and railroad tank car is common place.

In adapting the established procedures for these fuels to use in a transportation system, additional consideration must be given to protecting passengers and the public in the event of a leak, spill, or accident. Specific factors associated with hydrogen are that it is odorless, burns with a colorless flame, and has wide flammability limits. Additives are being considered to color the flame and to give it any odor. This is feasible for the gaseous state, but finding additives compatible with liquid-hydrogen temperatures seems unlikely. Fire and leak detection devices must be used. Methane burns with a visible flame and has relatively narrow flammability limits at low pressures (1 to 5 atm). While also odorless, the use of additives is standard for the gaseous state and are being developed for use with the liquid state. Both hydrogen and methane storage tanks should include special design considerations to limit pressure buildup, safely dispose of any gas that must be vented, and provide adequate fire protection.

An argument can be made that hydrogen is safer than gasoline because it would dissipate quickly and because its ignition temperature is higher than gasoline's. However, appropriate safety procedures for hydrogen are substantially new and different. Personnel involved with the use and handling of liquid hydrogen or liquid methane must, therefore, be appropriately trained in the required precautions and procedures for these fuels. New regulations are needed to cover the use of methane or hydrogen as a transportation fuel. These must be understandable, enforceable, and protective to personnel and property.

Remarks

Both methane and hydrogen are potential fuels for future transportation systems. The technology exists today that would allow liquid methane to be selected as the fuel for a given transportation system. (This has essentially been done in a limited way in certain fleet vehicle operations which have converted to liquified and pressurized natural gas.) It offers the advantage of ready transition from the natural fuel form (natural gas) to a synthetic fuel (similar composition) produced from coal, waste material, or some other source. This same easy transition would, of course, be available with kerosene, which may also be produced synthetically. A final selection for a given application would have to consider a variety of factors including the initial availability and cost of the natural fuel, the eventual availability and cost of the synthetic fuel, the criticality of emissions, and the complexity and cost of using cryogenic fuels.

Although the basic technology exists that would allow a transportation system to be operated on hydrogen, significant advances are required to make it practical. The two most critical needs are for a low-cost, high-density onboard fuel storage system and for competitive hydrogen production costs. It is estimated that electrolytic hydrogen may become competitive with fossil fuels in the 1990's and it seems likely that the storage problem may also be solved by that time. If the air pollution problems become sufficiently severe, the advantages of hydrogen in minimizing emissions could substantially hasten the advent of a hydrogen-fueled transportation system. Exhaust emissions may be reduced to zero through the use of the hydrogen-oxygen system.

Because of the magnitude of the fuel distribution problem, the need for safety education and training and regulatory requirements, transportation vehicles would have to be phases gradually into the use of hydrogen or methane. The following lists present sequences for such a transition:

Methane	Hydrogen
(1) Trains and buses	(1) Aircraft
(2) Trucks	(2) Trains and buses
(3) Automobiles	(3) Trucks
(4) Aircraft	(4) Automobiles

For methane, trains and buses are shown first, where operations can be initiated in a limited and controlled fashion in small fleets or restricted operations. For hydrogen, aircraft are shown first because the low fuel weight (lb/Btu) is particularly advantageous and operations can again be initiated in a limited and controlled manner.

The following additional studies are recommended to provide a more detailed evaluation than was possible in this study and to explore new areas that have been defined by this study:

(1) Design requirements, technology needs, and economics of a nationwide hydrogen logistics system.

(2) Safety and regulatory requirements for hydrogen and liquid methane.

(3) Potential approaches to low-cost, high-density on-board hydrogen storage, including consideration of slush hydrogen for aircraft.

Discussion

FUELS

This section describes production processes of hydrogen, methane, and the hydrides. Their physical properties as well as a comparison of thermal values (hydrogen and methane) are listed. A brief comparison of various hydrogen storage systems is also included. This discussion lists several major areas for study in order to evaluate a vehicle design that utilizes a hydrogen or hydride fuel system. The composition of kerosene and gasoline, the physical properties of both, and the typical composition of gasoline obtained from West Texas crude oil is included. The starting materials in most instances for the present means of producing hydrogen and methane are the fossiliferous materials - natural gas, petroleum, and coal. General terms will be used in describing the methods of production.

Hydrogen Production Process

(1) Steam reform process of natural gas - More hydrogen is probably produced by this process than any other. In the steam reform process, methane reacts with steam in the presence of a catalyst to form carbon monoxide (CO) and hydrogen. The CO in turn reacts with steam to form carbon dioxide (CO_2) and hydrogen. The thermal efficiency (lb H_2 out) × (H_2 heating value)/ (lb NG in) (NG heating value) in plants using about 100 standard cubic feet of

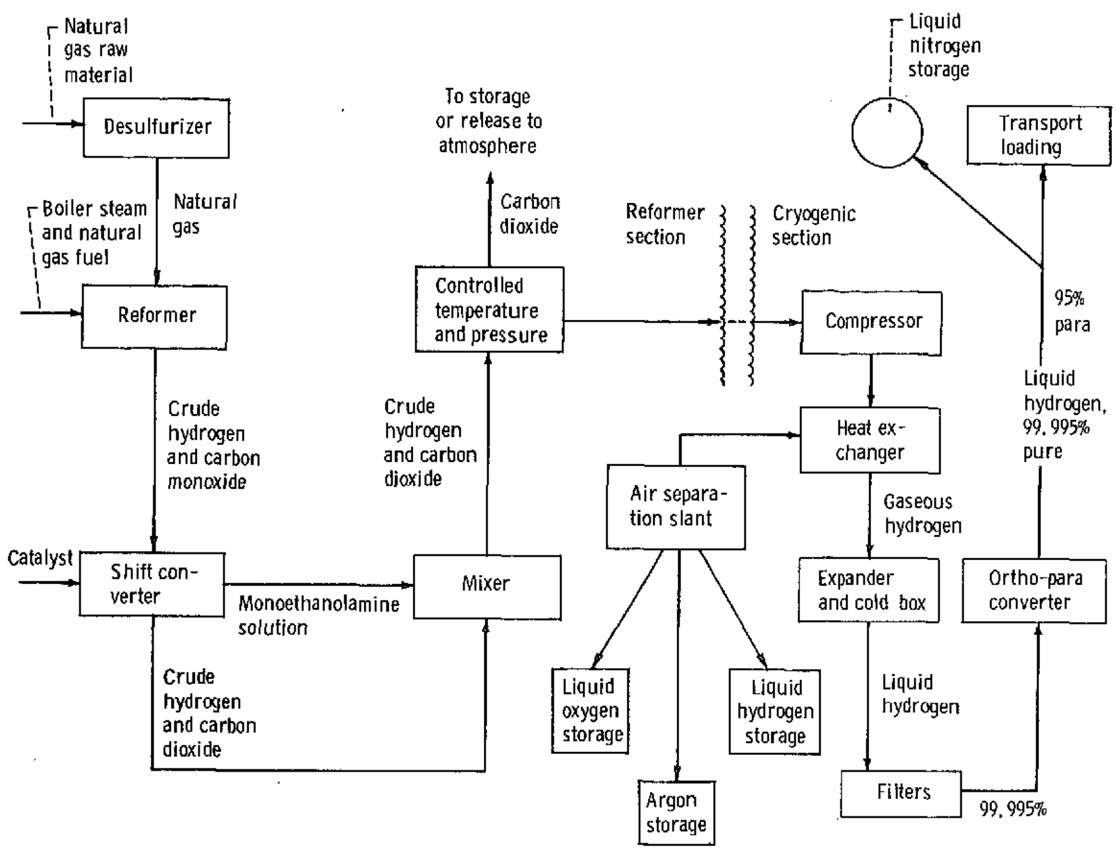


Figure G-1. - Liquid-hydrogen processing flow chart (steam reforming of natural gas).

natural gas to produce 1 pound of hydrogen (192 std. ft^3) gas is approximately 52 percent. Figure G-1 is a flow chart of this process (ref. 2).

(2) Cracking of petroleum - Much hydrogen is presently being produced in petroleum refineries, but a large portion is used in hydrogenating unsaturated hydrocarbons and is not available (ref. 3).

(3) Steam reform process of synthetic natural gas - This process will work with synthetic natural gas (SNG) from either petroleum or coal as long as the SNG contains light-weight hydrocarbons and CO (ref. 4).

(4) Water - gas process for coal gasification - Coal is reacted with steam at an elevated temperature to form hydrogen and CO. The CO in turn

reacts with steam to form CO_2 and H_2. After the removal of CO_2 by absorption, the hydrogen is essentially pure.

(5) Partial oxidation of hydrocarbons - In a typical partial oxidation process, a crude oil is mixed with steam and fed in carefully controlled quantities to a generator where the fuel reacts with preheated oxygen. This produces a synthesis gas consisting primarily of CO and hydrogen. The shift reaction is used in converting the CO to CO_2 and hydrogen. The CO_2 produced is removed by absorption leaving essentially pure H_2 (ref. 2).

(6) Electrolysis of water - By Faraday's Law 76.2 kilowatt-hours of electrical energy will ideally liberate 1000 cubic feet of hydrogen (60° F and 1 atm) by electrolysis. This is the theoretical ideal case where the production facility operates at 100 percent thermal efficiency. Under present state of the art an efficiency of only about 55 percent is expected. At 55 percent efficiency 145 kilowatt-hours is required to liberate 1000 cubic feet of hydrogen.

(7) Other - Other sources and methods of production include thermochemical (EURATOM and the Institute of Gas Technology) and radiolytic processes. These processes, though not yet developed, offer the potential for the direct production of hydrogen from water using thermal and/or nuclear energy. In addition, there are biological processes for the generation of hydrogen which may become viable sources in the future. Such methods could prove significantly more efficient than generating electricity and then producing hydrogen by electrolysis.

Methane Production Process

(1) Natural gas - The U.S. natural gas supply contains, on the average, about 85 percent methane. Therefore, in using the natural gas supply no chemical production would be required for obtaining methane; only a chemical extraction is required (ref. 3).

(2) Petroleum - (a) Catalytic conversion by the catalytic rich gas (CRG) process. This process used naptha as feedstock (ref. 4). (b) The British Gas Council's fluidized bed hydrogenation process. This process handles petroleum feedstock from heavy naptha to whole crude (ref. 4). (c) The method rich gas (MRG) process by the Japan Gasoline Co. uses light-liquid

hydrocarbons as feedstock (ref. 4). (d) Gassynthan process by Lurgi - Note: Most processes for converting petroleums to gases use light-liquid hydrocarbons as the feedstock and the processes are similar, varying primarily in the formulation of the nickel catalysts used (ref. 4).

(3) Coal - (a) Pyrolysis or destructive distillation process: This process is being used by Food Machinery Corp. (FMC) for Project COED (Char-Oil-Energy-Development). An oil fraction is produced by pyrolysis for further refining. Generally, with the correct cracking procedures, an oil fraction can be cracked to yield large concentrations of methane (ref. 4). (b) Solvation process: This process yields an extract enriched in hydrogen, either by a solvent or dispersing action alone, or by using a hydrogen donor solvent such as tetralin. The Consolidated Coal Company is using this process for obtaining an extraction which is hydrogenated to produce a feedstock for further refining (ref. 4). (c) Direct hydrogenation process: Coal is reacted directly with hydrogen at high temperatures in the presence of a catalyst to form methane and other hydrocarbons. The Hygas Process being installed by Institute of Gas Technology is a direct hydrogenation process (ref. 4). (d) Indirect hydrogenation process: A synthesis gas is first produced by means of the following reaction

$$C + H_2O \rightarrow CO + H_2$$

The synthesis gas is then methanated by a catalyzed Fischer-Tropsch reaction:

$$CO + 3H_2 \rightarrow CH_4 + H_2O \tag{1}$$

$$2CO + 2H_2 \rightarrow CH_4 + CO_2 \tag{2}$$

The prime economic consideration in most of the gasification processes for producing methane from coal is the source and cost of the hydrogen required. For large amounts the hydrogen would be produced using coal and water as described in the water-gas process for producing hydrogen.

(4) Other sources and methods of production - (a) methane produced from organic waste such as rubber and plastic by nickel catalyst methanation in

steam reaction (ref. 1). (b) Biological fermentation of waste such as cellulose and sewage sludge. The average anerobic sewage treatment plant has the capability of producing 1 standard cubic foot of methane per day per capita. (c) Synthesis such as heating and pressurization of calcium carbonate with water, ferrous oxide, and silicon dioxide. This process has been successfully carried out in the laboratory (ref. 5).

Physical Properties

Methane - Some physical properties of the colorless, odorless liquid methane follow:

Melting point, ^{o}F	-296.7
Boiling point, ^{o}F	-258.5
Critical temperature, ^{o}F	-116.1
Critical pressure, atm	45.8
Density, lb/gal	3.537
Heat of combustion (net), Btu/lb; Btu/gal	21 052; 76 052
Flammable limits (lower to upper), vol. % methane gas in air	5.3 to 13.9

Hydrogen - Some physical properties of the colorless, odorless liquid hydrogen follow:

Melting point, ^{o}F	-434.6
Boiling point, ^{o}F	-423.0
Critical temperature, ^{o}F	-400.0
Critical pressure, atm	12.98
Density, lb/gal	0.591
Heat of combustion (net), Btu/lb; Btu/gal	51 571; 30 487
Flammable limits (lower-upper), vol. % H in air	4.1 to 74.2

Kerosene - Kerosene is a mixture of petroleum hydrocarbons, chiefly of the methane series having from 10 to 16 carbon atoms per molecule. Specifications are established for specific grades of kerosene by the government and by refiners. The specifications are developed based on performance observations. For example, fuels used for lighting require a high paraffinic content (less smoke) and low viscosity for wick feeding; kerosene, with a high aromatic and napthenic content, gives a flame too smoky for lamps, but it may be used for vehicles such as tractors. The following table lists some of the physical properties of this pale yellow (or colorless), oily liquid:

Density, g/cm^3; lb/gal 0.81; 6.75
Boiling range, ^oF . . . . 175 to 325
Flash point, ^oF . . . . 150 to 185
Heat of combustion, Btu/lb 19 000
Flammable limits (lower-upper), vol.% kerosene in air . . . . 0.7 to 5.0

Gasoline - Gasoline is a mixture of hydrocarbons ranging from C_3 to C_{12}. The actual composition depends on the source of the crude processing techniques used for refining and on the additives used to prevent knocking, gum formation, oxidation, corrosion, etc. Some properties of this highly flammable, volatile liquid follow:

Density, g/cm^3; lb/gal 0.74; 6.16
Boiling range, ^oF . . . . 80 to 430
Flash point, ^oF . . . . -50
Heat of combustion, Btu/lb 19 200
Flammable limits (lower-upper), vol.% gasoline in air . . . . . 1.3 to 6.0

A typical composition of gasoline from West Texas crude oil is given in table G-1.

TABLE G-1. - TYPICAL COMPOSITION OF GASOLINE FROM WEST TEXAS CRUDE OIL

Refinery process	Component content, vol. %					Octane number (F-1)	Sulfur content, wt. %	Nitrogen content, wt. %	ASTM boiling range, ^oF
	Paraffins	Naphthenes	Olefins	Diolefins	Aromatics				
Straight run	58	34	0	0	8	71	0.07	(0.02)	130 - 229
Thermally cracked	25	19	43	3	11	72	.24	(.05)	162 - 357
Catalytically cracked	26	12	27	1	35	90	.1	(.01)	90 - 438
Catalytically reformed	29	4	0	0	67	97	(.01)	------	114 - 443
Hydrocracked	10	30	0	0	60	95	(.01)	(.001)	150 - 410

Heating Values

The following table presents a comparison of the thermal values for methane, hydrogen, kerosene, and gasoline on a weight and liquid volume bases:

Fuel	Lower heating value	
	Btu/lb	Btu/gal
Hydrogen	51 600	30 500
Methane	21 500	76 000
Kerosene	19 000	128 000
Gasoline	19 200	118 000

As can be seen hydrogen has a substantial advantage over the other fuels on a weight basis but a substantial disadvantage on a volume basis.

Hydrides and Hydrogen Storage

A hydride is a binary compound composed of hydrogen and another element. Some hydrides are formed by reacting hydrogen gas directly with the other element, and some can only be formed by indirect means. Most hydrides break down when heated. Also, many of the hydrides will hydrolyze when brought in contact with water. Hydrides are divided into three classes (based on the differences in their chemical structures, physical properties, and chemical behavior):

Ionic hydrides: These are salt-like compounds in which the hydrogen is present as the negatively charged hydride ion.

Covalent hydrides: These compounds are volatile gases, liquids, and solids in which the chemical bonding is primarily nonpolar, electron-pair sharing. This class of hydrides comprises a major portion of the known hydrides.

Transitional metal hydrides: Hydrides of the transitional elements have a wide variation in their nature and properties. Some appear to contain hydrogen as molecular hydrogen physically absorbed in rifts or defects in the metal. Some appear to contain hydrogen as if it, too, were a metal; that is, hydrogen assumes definite positions in the metal lattice structure, and the properties of the compound remain metallic.

Metal hydrides may be used for vehicular propulsion. Investigations (unpublished) of the equilibrium relationships and kinetics of the reversible reaction of hydrogen with magnesium-nickel and magnesium-copper alloys reveal that these alloys may be used in a system as a convenient and inexpensive method of storing hydrogen. The magnesium and copper catalyst would merely act as a carrier and would not be consumed. The heat energy needed to dissociate the hydride could be obtained from the waste heat of hot exhaust gases. In addition, the fuel storage bed could be regenerated by supplying hydrogen at pressures higher than the dissociation pressure while the bed is maintained at a temperature where the reaction kinetics are satisfactory. The depleted fuel container would most likely be refilled at a service station. Various hydrogen storage systems are compared in the following table:

Storage and system	Weight of container and fuel, lb	Contained volume, ft^3
Gas at 200 psi	2250	66
Liquid (cryogenic)	353	10.2
Magnesium hydride[a]	[b]692	10.8

[a]40 Percent voids.
[b]Hydride, 592 lb; container, 100 lb.

The table indicates that the major disadvantage of compressed hydrogen gas storage is the weight of the container and that the major disadvantage of hydride storage is the weight of the binary element. The lowest weight system is liquid hydrogen storage.

It appears that the major areas of study required to evaluate the full potential of an optimized, hydrogen-engine - hydride-fuel-system vehicle design include investigations of hydride bed characteristics when coupled to an exhaust-gas heat source, engine design integration with the hydride system, and safety of the hydride system.

Hydrogen and Oxygen Production

Table G-2 lists the largest existing liquid hydrogen plants and their production capacities. Figure G-2 shows the possible production phasing for hydrogen, and figure G-3 for oxygen. It is estimated that coal gasification may become a significant source of hydrogen in the 1980's and that hydrogen and oxygen from electrolysis of water would become competitive in the 1990's.

TABLE G-2. - EXISTING LIQUID-HYDROGEN PLANTS

Company	Location	Capacity, ton/day
Air Products	Long Beach, Calif.	30
	New Orleans, La.	30
Linde	Ontario, Calif.	30
	Sacramento, Calif.	[a]60
	Ashtabula, Ohio	[b]7
AiReduction	Pedricktown, N.J.	6

[a]Being scrapped.

[b]Initial maximum capacity to be 8.5 tons/day (7 tons/day design minimum) can be incrementally expanded as required with 7 ton/day modules.

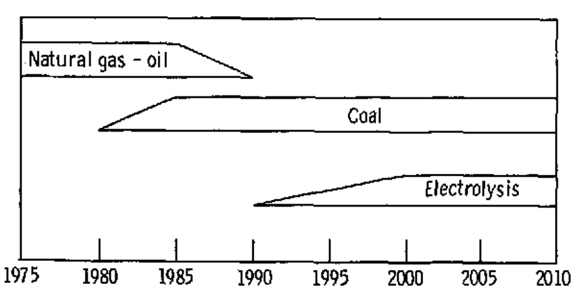


Figure G-2. - Liquid-hydrogen production process phasing.

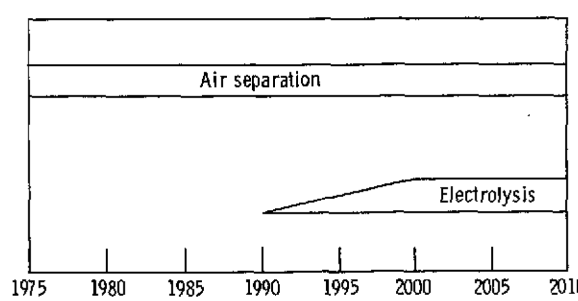


Figure G-3. - Liquid-oxygen production process phasing.

NORTHEAST CORRIDOR TRANSPORTATION SYSTEM

This section provides basic production, storage, and distribution quantity requirements and costs for liquid hydrogen, liquid oxygen, and liquid methane for a northeast corridor TACV. To evaluate fuel needs for an advanced high-speed transportation system, a model was established based on existing DOT studies of the northeast corridor. A reserve or contingency quantity of 20 percent is added to the actual consumption requirement of the three liquids to provide on-board requirements. In addition, a factor has been applied to allow for all cryogenic losses through distribution and, in turn, provide the total quantities for sizing production plant capacity. No credit is taken for possible secondary use of the fuel losses which could be collected and reprocessed or used for local heating. Methane losses would have to be collected to prevent pollution. Figure G-4 shows the northeast corridor and the principal terminals.

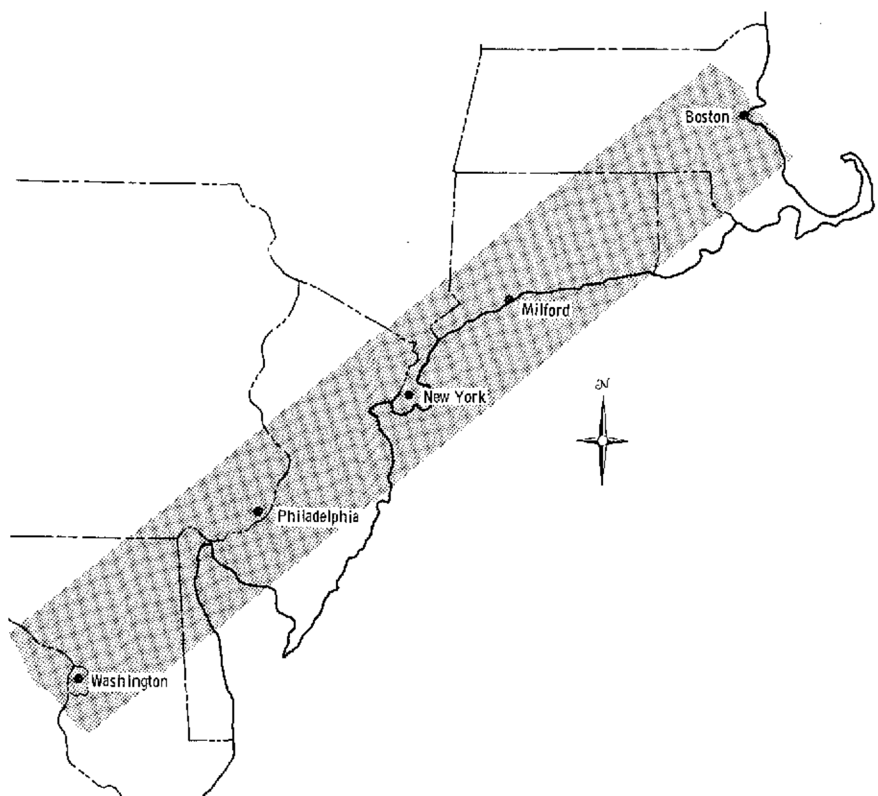


Figure G-4. - Northeast corridor.

TABLE G-3. - NORTHEAST CORRIDOR MODEL AND ANALYSIS

(a) Base for mission calculations

Average vehicle power level, hp	12 000
Number of engines	2
Distance, mile, from -	
Washington to Boston	450
Philadelphia to Milford	161
Passenger miles per year	6045
Fuel consumption rates, lb/hp-hr, for -	
Liquid methane	0.40
Liquid hydrogen[a]	0.15
Liquid oxygen[a]	1.31
Kerosene	0.45

(b) Daily train usage

Train origin-destination	Number of trains per day	Number of cars[b] per train
Washington-Boston	60	4
Philadelphia-Milford	43	3

(c) Loss factor for distribution

Liquid methane	1.1
Liquid hydrogen	1.25
Liquid oxygen	1.25
Kerosene	1.0
Average yearly production plant operating time, day	350
Daily TACV operating time, hr	24
Distance between fueling sites, mile	150

(d) Daily total mileage

Train origin-destination	Number of trains daily		Number of cars per train		Distance one way, mile		Total mileage
Washington-Boston and reverse	120	×	4	×	450	=	216 000
Philadelphia-Milford and reverse	86	×	3	×	161	=	41 538
						Total:	257 538

(e) Daily fuel requirements without losses

Average vehicle speed, mph	212
Total vehicle use per day, hr	1215
Daily energy output[c], hp-hr	14 580 000
Daily fuel consumption[d], lb, for -	
Liquid methane	5.832×10^{6}
Liquid hydrogen	2.187×10^{6}
Liquid oxygen	17.496×10^{6}
Kerosene	6.561×10^{6}

(f) Total daily fuel requirements with losses[e]

Consummable	Daily fuel requirements, ton (lb), based on yearly plant operation of - 365 days	350 days[f]
Liquid methane	3 208 (6.415×10^{6})	3 345
Liquid hydrogen	1 367 (2.734×10^{6})	1 425
Liquid oxygen	10 935 (21.870×10^{9})	11 403
Kerosene	3 281 (6.561×10^{6})	3 421

(g) Consumption per car (vehicle)[g]

Consummable	Assumed fuel consumption lb/hp-hr	lb/12 000 hp-hr	lb/total range[h]	ft^3/total range	gal/total range
Range, 450 miles					
Liquid methane	0.40	4 800	10 176	387	2877
Liquid hydrogen	.15	1 800	3 816	863	6457
Liquid oxygen	1.20	14 400	30 528	428	3204
Kerosene	.45	5 400	11 448	229	1696
Range, 600 miles					
Liquid methane	0.40	4 800	13 584	517	3841
Liquid hydrogen	.15	1 800	5 094	1152	8619
Liquid oxygen	1.20	14 400	40 752	571	4278
Kerosene	.45	5 400	15 282	306	2264

[a]Stoichiometric mixture of hydrogen and oxygen.
[b]Car capacity, 100 passengers.
[c]1215 hr times 12 000 hp.
[d]Daily output times fuel consumption rate.
[e]Fuel burned times buy-to-use ratio.
[f]Accounts for plant down time. Ratio 365/350 = 1.0428
[g]For sizing on-board storage tanks, assume a 20% reserve.
[h]Average speed, 212 mph; therefore, $t_{450} = 2.12$ hr and $t_{600} = 2.83$ hr.

Model and Analysis

The model used and the analysis of total fuel requirements is given in table G-3. The engine specific fuel consumption (SFC) values used were assumed on the basis of preliminary engine analysis results from section 4. For four refueling sites at Washington, Philadelphia, Milford, and Boston and train fuel ranges of 450 to 600 miles, the number of fuelings required at each site per day and the average time available per fueling is listed in table G-4.

Fuel Cost Analysis

Table G-5 tabulates estimated fuel costs in 1972 dollars for the northeast corridor TACV. Daily and annual costs are shown for liquid hydrogen, liquid methane, liquid oxygen, gasoline, and kerosene. The table footnotes explain the methods of arriving at the fuel costs. Costs are based on f. o. b. liquid storage tank prices at liquefying plants in the New York City area. Costs of liquid fuel distribution are not included. In 1972 dollars, cost per million

TABLE G-4. - REFUELING SITE REQUIREMENTS FOR NORTHEAST CORRIDOR TACV TRANSPORTATION SYSTEM

[Refueling sites at Washington (W), Philadelphia (P), Milford (M), and Boston (B).]

Train range, mile	Average fueling time[a], min				Train origin-destination	Number of fuelings[b] at each site per day				
	W	P	M	B		W	P	M	B	Total
450	24	100	100	24	W-B	30	---	---	30	60
					B-W	30	---	---	30	60
					P-M	--	$14\frac{1}{3}$	---	--	$14\frac{1}{3}$
					M-P	--	---	$14\frac{1}{3}$	--	$14\frac{1}{3}$
						60	$14\frac{1}{3}$	$14\frac{1}{3}$	60	$148\frac{2}{3}$
600	96	35	35	96	W-B	15	---	30	--	45
					B-W	--	30	---	15	45
					P-M	--	$10\frac{3}{4}$	---	--	$10\frac{3}{4}$
					M-P	--	---	$10\frac{3}{4}$	--	$10\frac{3}{4}$
						15	$40\frac{3}{4}$	$40\frac{3}{4}$	15	$111\frac{1}{2}$

[a]Assumes equal time and distance spacing of trains, 24 hours per day, with one fueling at each site. Fueling time is inversely proportional to number of fueling booths.

[b]Applies to entire train. W-B and B-W trains each have four cars; P-M and M-P trains have three. Each car must be refueled. Number of cars to be fueled is four and three times the fuelings shown at each site for W-B - B-W and P-M - M-P trains, respectively.

TABLE G-5. - COST OF REQUIRED CONSUMMABLES FOR NORTHEAST CORRIDOR

TACV TRANSPORTATION SYSTEM - 1972 DOLLARS

Consummable	Daily requirement, lb (a)	Source	Unit	Cost of fuel requirements		Cost per million Btu, $	Cost[b] per passenger mile, ¢
				Daily	Yearly		
Liquid hydrogen	2.734×10^6	Steam reform of natural gas	[c]10.60	290 000	106.0×10^6	2.06	1.8
		Partial oxidation of petroleum	[c]10.60	290 000	106.0	2.06	1.8
		Electrolysis of water	[c,d]18.00	493 000	180.0	3.49	3.0
		Coal gasification	[e]12.50	342 000	125.0	2.43	2.1
Liquid methane	6.415	Purification of pipeline natural gas	[f]3.00	193 000	70.4	1.40	1.2
		Purification of shipped liquid natural gas	[g]5.30	340 000	124.1	2.47	2.1
Liquid oxygen	[h]21.870	Air separation	[i]1.00	219 000	80.0	----	1.3
Gasoline	[j]6.488	Petroleum	[k]1.94	126 000	46.0	1.01	.8
Kerosene[l]	[j]6.561	Petroelum	[m]1.67	110 000	40.2	.88	.7

[a]See table III-3(f).

[b]Based on 6.045×10^9 passenger miles per year.

[c]From ref. 8, with the 1967 dollars escalated to 1972 at 5.5% compounded per annum. Cost is f.o.b. liquefaction plant in the New York City vicinity.

[d]Electrolytic hydrogen in 1972 cost more than fuels from other sources. But by 1985 fossil fuels will become more scarce and their price may rise enough to make electrolytically produced hydrogen more practical. Electric power from nuclear plants (fast breeder plants especially) for electrolytic hydrogen after 1985 might tend to hold down the cost.

[e]Extrapolated for ref. 8 data. Assumes cost of hydrogen from coal (lignite at \$7/ton) is same as cost of hydrogen from natural gas at $\$0.75/10^6$ Btu ($\$0.72/10^3$ ft^3).

[f]1972 Dollars based on the following: natural gas to New York City at $\$0.50/10^3$ ft^3 (natural gas is 90% methane), purification to 99% methane, and liquefaction at $\$0.60/10^3$ ft^3 of natural gas. (From Oil and Gas Journal.) Cost is f.o.b. liquefaction plant in the New York City vicinity.

[g]1972 Dollars based on the following: liquid natural gas received in New York City at $\$1.37/10^3$ ft^3, purification to 99% methane, and reliquefaction at $\$0.60/10^3$ ft^3 (liquid natural gas gassified). Cost is f.o.b. liquefaction plant in the New York City vicinity.

[h]Based on hydrogen requirements only (stoichiometric 8 to 1).

[i]Value is 20% less than current 1972 Air Force Stock Fund list price because of large quantities. Cost is f.o.b. liquefaction plant in the New York City vicinity.

[j]Based on heat of combustion equivalents of liquid methane requirements; 1 lb of liquid methane is equivalent to 1.120 lb of gasoline or 1.132 lb of kerosene.

[k]Average pump price (in 52 U.S. cities) of regular gasoline for 1972 to date (May 30, 1972), 34.62¢/gal. (From Oil and Gas Journal, June 5, 1972.) Cost used here is average refinery price or 12.12 ¢/gal. (Note: The average wholesale price of regular gasoline, excluding Texas, is approx. 32% less than the average pump price and the average refinery price is approx. 65% less than the average pump price.)

[l]This grade of kerosene would be similar to diesel fuel used for automotive engines.

[m]A good average pump price for kerosene is unavailable, but the average refinery price (10.42 ¢/gal) is 14% less than the average gasoline refinery price.

TABLE G-6. - PRODUCTION AND STORAGE FACILITY COSTS[a] FOR

NORTHEAST CORRIDOR TACV TRANSPORTATION SYSTEM

[Extrapolated from ref. 8; 1972 dollars.]

(a) Liquid hydrogen; daily requirement, 1425 tons

Assume 1500 ton/day liquid hydrogen plant:	
Plant capital investment, $	159.1×10^6
7-Day flywheel[b] storage capacity, lb	19.8×10^6
Cost for storage capacity, $	21.6×10^6
Assume 1500 ton/day gaseous hydrogen plant:	
Plant capital investment, $	26.8×10^6
Capital cost of liquefaction of gaseous hydrogen[c], $ -	
At each of four fueling sites	57.5×10^6
Total	230×10^6
7-Day flywheel[a] storage capacity at each of four sites, lb	5125×10^6
Cost of storage tank, $ -	
At each site	5.42×10^6
Total	21.68×10^6
Conclusion - Based on these figures, one centrally located 1500-ton per day plant is more attractive. But costs and losses involved in distributing liquid hydrogen in large quantities as well as the cost of gaseous hydrogen pipelines must be considered.	

(b) Liquid oxygen; daily requirement, 11 463 tons

Assume 12 000-ton/day liquid oxygen plant:	
Plant capital investment, $	120×10^6
7-Day flywheel storage capacity, lb	168×10^6
Assume four 300 ton/day liquid oxygen plants located near main TACV terminals:	
Plant capital investment (each), $	33×10^6
7-Day flywheel capacity at each site, lb	42×10^6
Total cost for storage[c]	22×10^6
Conclusion - Separate liquid oxygen plants appear more feasible. Logistics of distributing these large quantities from a single plant would present problems.	

[a]All costs are estimated averages and are intended to indicate magnitude of cost. Actual costs would vary with location.

[b]That storage volume capable of holding 7 days of plant production at maximum production capability.

[c]Transported by pipeline.

[d]Using vacuum jacketed tanks.

Btu's of liquid methane from domestic natural gas is lower than liquid nitrogen by 32 percent, higher than the gasoline refinery price (excluding taxes) by 40 percent, and higher than estimated kerosene refinery price (excluding taxes) by 60 percent. Likewise, costs per passenger mile for these fuels will show approximately the same percentage differences.

Table G-6 provides cost estimates for liquid hydrogen and liquid oxygen manufacturing and storage facilities.

Fuel Distribution

Potential approaches to the distribution and handling of fuels and to vehicle refueling are presented in figures G-5 to G-11. Figure G-5 suggests use of a pipeline for the transmission of gaseous fuels and oxygen through the corridor to liquefying plants and then to liquid storage at or near the re-

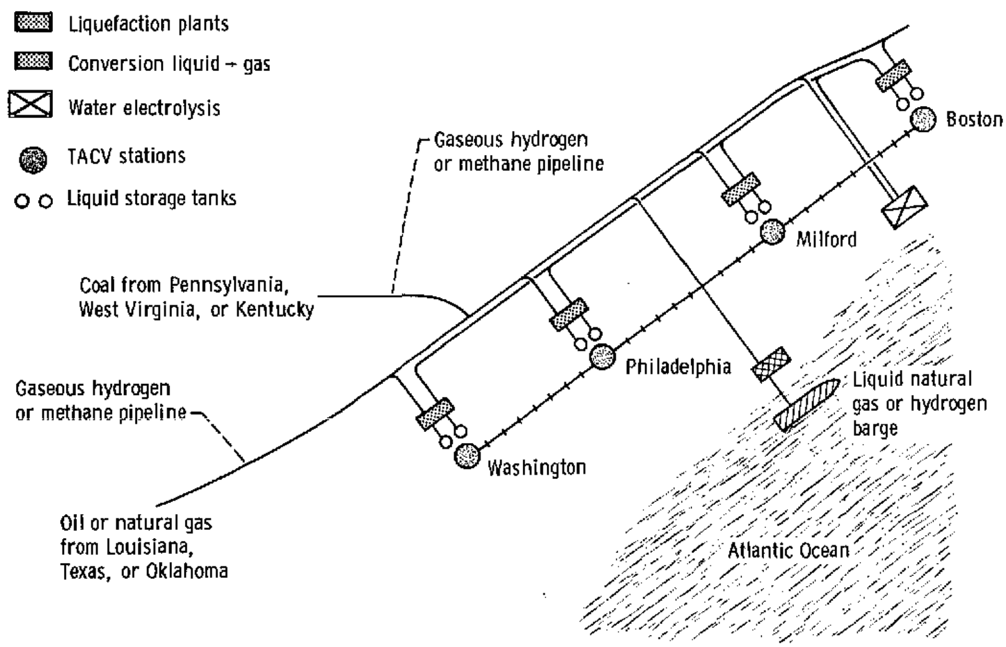


Figure G-5. - Transmission, liquefaction, and storage of hydrogen, methane, and oxygen.

fueling sites. Figure G-6 presents a method for delivering cryogenic liquids to station storage when distance or safety preclude the use of a pipeline. Figure G-7 shows train module concepts, which are various approaches to modular fuel and engine integration with the passenger part of the vehicle. Figures G-8 to G-10 provide switching and turntable arrangements for train refueling. And figure G-11 offers several methods for train refueling by the direct transfer of fuel and by exchange of fuel containers.

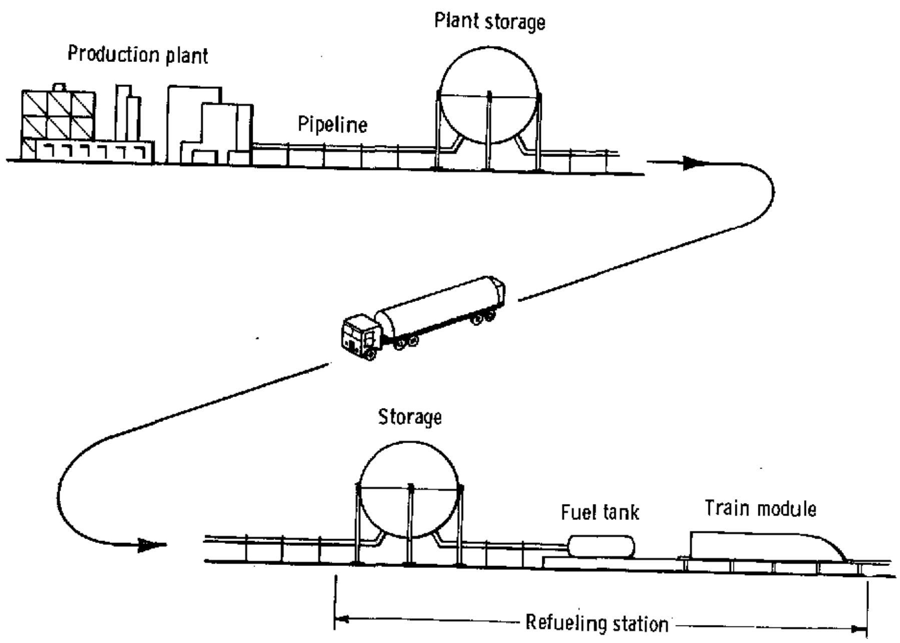
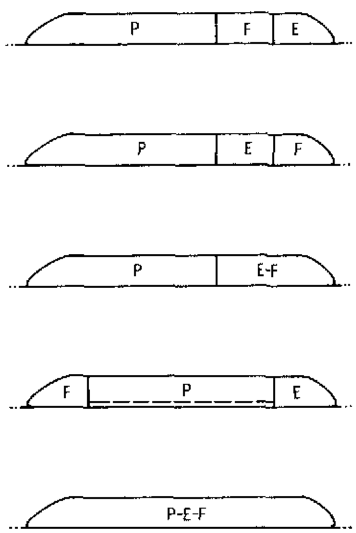


Figure G-6. - Cryogenic delivery and storage system.

P Passenger module
E Engine module
F Fuel module
E-F Combined engine and fuel module
P-E-F One inseparable unit



	Advantages	Disadvantages
(a)	Ease of module exchange	Refueling must be done either at the passenger station or E has to pull F to a remote fueling area; fuel line couplings between F and E are a potential problem
(b)	See (a)	F can be pulled to refueling station by conventional "Tugloc," while E can rest (not using aircushions); fuel line couplings between E and F are a potential problem
(c)	See (a). Also, eliminates fuel line couplings; E-F is shorter	E-F will have to be exchanged for a fueled unit
(d)	See (a)	F must be refueled at passenger station or must be pulled to fueling station by a utility engine; fuel lines must run through P; two fuel-line coupling points
(e)	Maximum fuel capacity; no couplings	In case of breakdown, the entire unit is idle; removal from track is complicated; requires large turntables

Figure G-7. - Train car assembly.

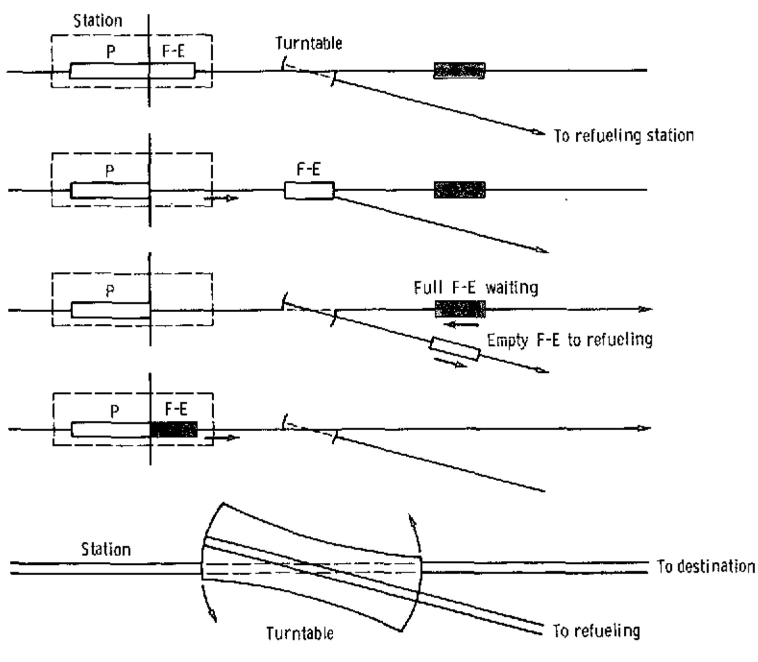


Figure G-8. - Exchange of fuel-engine modules at intermediate stations.

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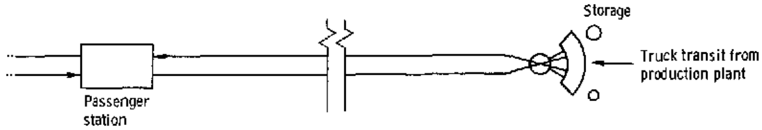
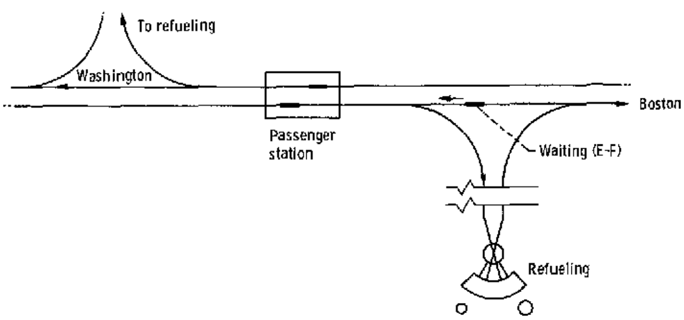
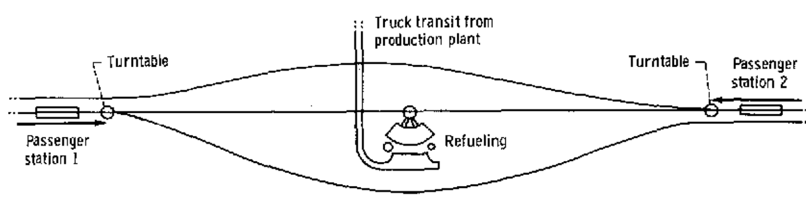


(a) At terminal (passengers leave train).



(b) At intermediate stations (passengers remain in train).



(c) At intermediate stations (passengers remain in train; F-E module separates from passenger module).

Figure G-9. - Passenger station and refueling station.

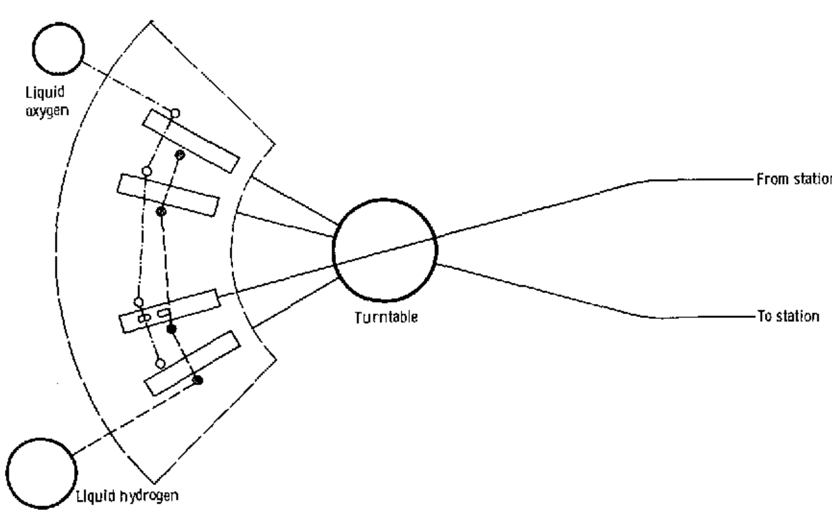
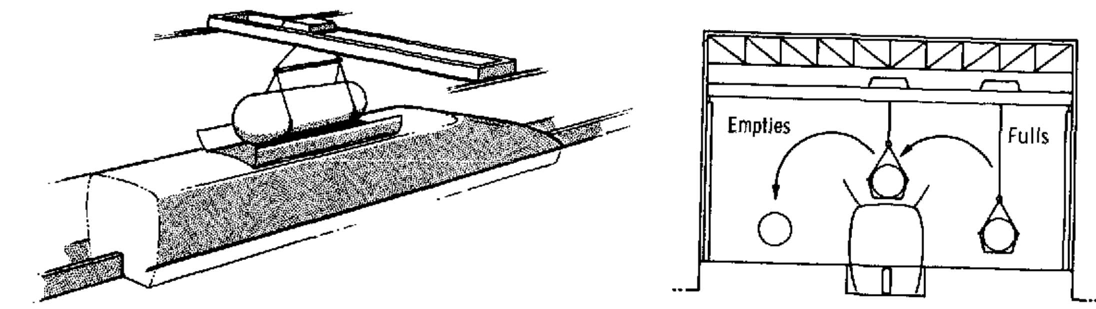
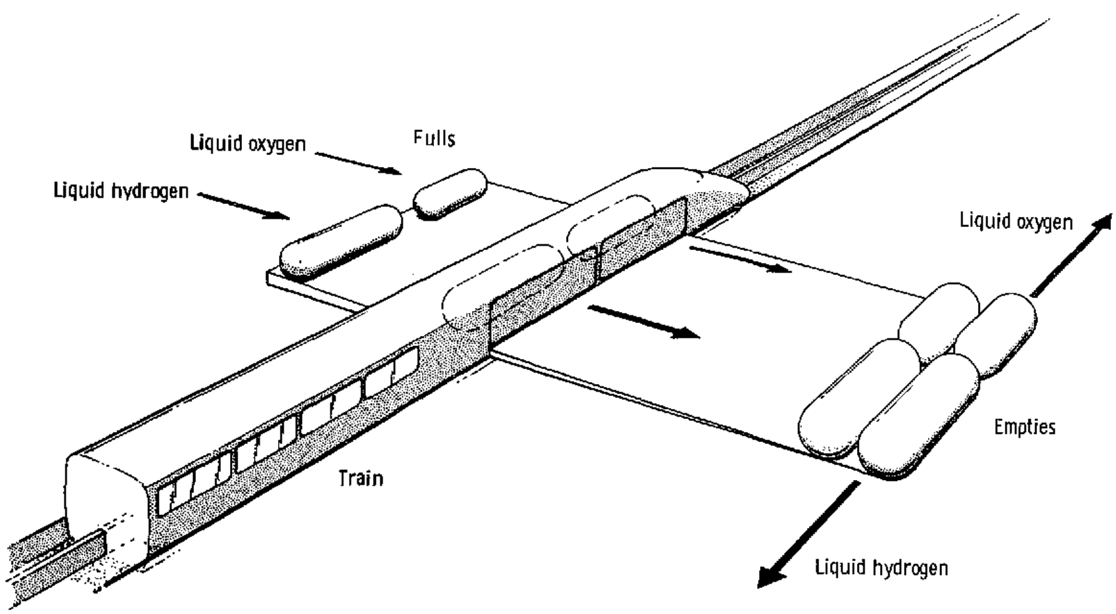


Figure G-10. - Direct refueling of train (or train module) without exchange of tanks.

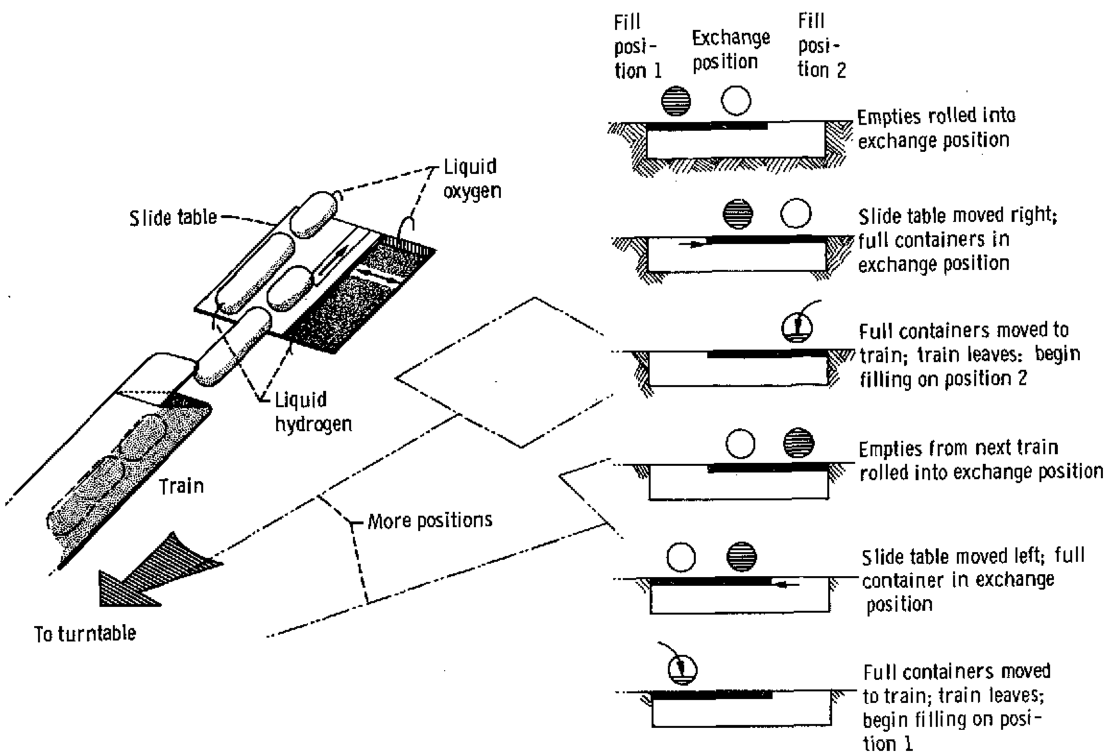


(a) Through roof. Disadvantage: Slow, hazardous, requires hangar, and limits structural integrity.



(b) Sideways. Advantage: Fairly quick operation can be performed at passenger station. Disadvantage: Transport to and from fueling station by truck, monorail conveyors, elevators, forklifts, etc. is time consuming and source of potential problems; fuel consuming; limits structural integrity.

Figure G-11. - Exchange of fuel containers.



(c) Through front end.

Figure G-11. - Concluded.

NATIONWIDE SYSTEM

Since a nationwide advanced transportation system model was not available, it was not practical to evaluate detailed production and distribution problems. Instead, the cost evaluations generated for the northeast corridor evaluation were extrapolated to the nationwide energy needs. The energy level selected, approximately, 0.3×10^{15} Btu per year, corresponds to estimates of the energy needs for all passenger rail and intercity bus service for the year 1985. It also is about six times the energy needs of the northeast corridor model that was used.

Cost of Fuels for Nationwide Advanced Transportation

Table G-7 compares annual costs of the various fuels to support total energy requirements of approximately 0.3×10^{15} Btu per year for a limited nationwide advanced transportation system.

TABLE G-7. - COST OF REQUIRED CONSUMMABLES FOR NATIONWIDE ADVANCED TRANSPORTATION SYSTEM - 1972 DOLLARS

Consummable	Yearly requirement[a], lb	Source	Unit cost, ¢/lb (b)	Yearly cost, $	Cost per million Btu, $
Liquid hydrogen	5.82×10^9	Steam reform of natural gas	10.60	617×10^6	2.06
		Partial oxidation of petroleum	10.60	617	2.06
		Electrolysis of water	18.00	1048	3.49
		Coal gasification	12.50	728	2.43
Liquid methane	13.66	Purification of pipeline natural gas	3.00	410	1.40
		Purification of shipped liquid natural gas	5.30	724	2.47
Liquid oxygen	[c]46.56	Air separation	1.00	466	----
Gasoline	[d]13.82	Petroleum	[e]1.94	268	1.01
Kerosene[f]	[d]13.97	Petroleum	[g]1.67	233	.88

[a]Converted from energy requirements as follows: hydrogen yields 51 571 Btu/lb and methane 21 502 Btu/lb.

[b]Same as unit cost used in table III-5.

[c]Based on hydrogen requirements only (stoichiometric 8 to 1).

[d]Based on heat of combustion equivalents as follows: gasoline yields 19 200 Btu/lb and kerosene 19 000 Btu/lb.

[e]Average pump price (in 52 U.S. cities) of regular gasoline for 1972 to date (May 30, 1972), 34.62 ¢/gal (Oil and Gas Journal). Cost used here is average refinery price or 12.12 ¢/gal. (Note: The average price of regular gasoline, excluding Texas, is approx. 32% less than the average pump price, and the average refinery price is approx. 65% less than the average pump price.)

[f]This grade of kerosene would be similar to diesel fuel used for automotive engines.

[g]A good average pump price for kerosene is unavailable, but the average refinery price (10.42 ¢/gal.) is 14% less than the average gasoline refinery price.

Nationwide Gas Pipeline System for Hydrogen and Methane

Figure G-12 is a map showing a potential pipeline system superimposed over the existing U.S. natural gas pipeline system. The potential pipeline terminates at strategic locations for possible distribution of hydrogen or methane for a limited nationwide advanced transportation system. The eight lettered squares A (New York) to H (Southern California) are assumed sites for gas plants to extract hydrogen or methane from natural gas. The 34 circled numbers indicate the various legs of the superimposed hydrogen or methane pipeline distribution system. Table G-8 provides the calculations for estimating the costs of the pipeline and booster stations. Estimated total cost of the system is $4 183 600 000 including land acquisition, 7447 miles of 30-inch (1200 psig) pipeline, and 196 booster stations. The cost of the eight conversion plants is not included. (Pipeline cost information was furnished by Southern Natural Gas Co., Birmingham, Alabama.) In figure G-12 it can be further assumed that liquid conversion plants can be strategically located to provide liquid hydrogen or methane to support the transportation system at major refueling terminals.

SAFETY CONSIDERATIONS

General

The gases and cryogenic fluids considered (hydrogen, methane, and oxygen) have been handled (produced, stored, and transferred) in large quantities for the past 20 years. Use of these fluids for future transportation systems will require safety emphasis on the protection of passengers and the local population in the event of propellant leaks, spills, or other accidents. A detailed comparison of these combinations is required as related to the primary safety requirements. Safety problems of concern with the hydrogen-oxygen combination are possibly more severe and would require more detailed studies. Detection and safety equipment will have to be developed to reduce, not only the probability of any accidents occurring, but also the consequences of any accident. Use of the separate tender concept, with propellants physically separated from passenger car, is a consideration. Fail-safe concepts

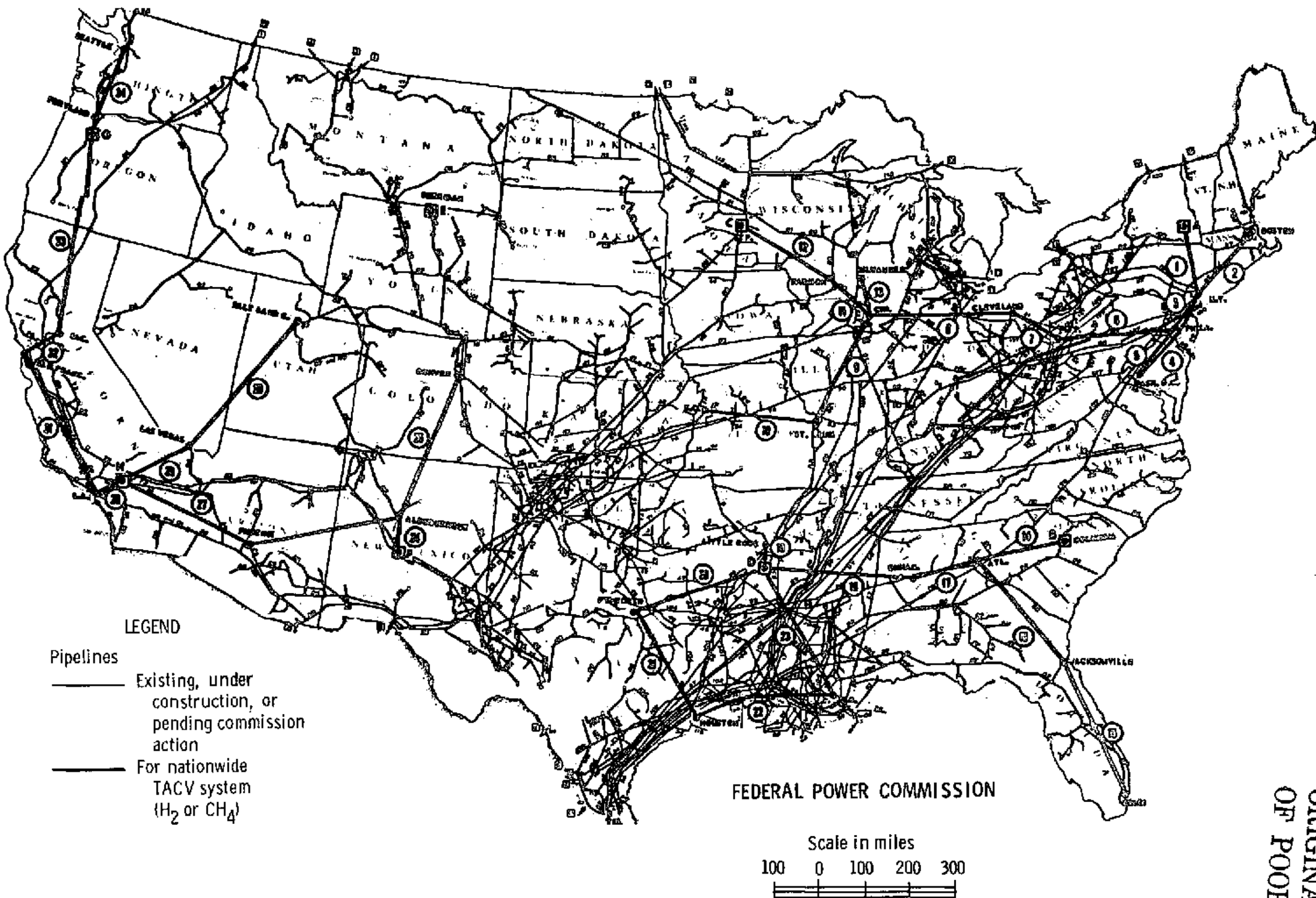


Figure G-12. - Nationwide gas pipeline system for hydrogen or methane.

TABLE G-8. - COST OF PIPELINES FOR GAS TRANSMISSION

Run	Terminal	Distance, mile	Cost per mile[a], $	Cost of pipeline[b], $	Number of boosters	Cost of booster[c], $	Total cost, $	Remarks
1	Plant A - New York	173	350×10^3	62.6×10^6	4	96×10^6	158.6×10^6	1 River crossing, $\$2.0\times10^6$
2	New York - Boston	186	500	95.0	5	120	215	1 River crossing, $\$2\times10^6$
3	New York - Philadelphia	96	550	52.8	2	48	100.8	----------------
4	New York - Washington, D.C.	208	550	118.4	5	120	238.4	2 River crossings, $\$2\times10^6$
5	Washington, D.C. - Philadelphia	125	550	68.8	3	72	140.8	----------------
6	Philadelphia - Pittsburgh	250	350	87.5	7	91	178.5	----------------
7	Pittsburgh - Cleveland	114	350	39.9	3	39	78.9	----------------
8	Cleveland - Chicago	313	350	109.6	8	104	213.6	----------------
9	Chicago - St. Louis	261	300	78.3	7	91	169.3	----------------
10	St. Louis - Kansas City	238	300	72.9	6	62.4	135.3	1 River crossing, $\$1.5\times10^6$
11	Chicago - Madison	113	300	33.9	3	31.2	65.1	----------------
12	Madison - Plant C	148	250	37.0	4	41.6	78.6	----------------
13	Chicago - Milwaukee	79	350	27.6	2	20.8	48.4	----------------
14	Plant B - Atlanta	179	275	50.7	5	37.5	88.2	1 River crossing, $\$1.5\times10^6$
15	Atlanta - Jacksonville	287	250	71.8	8	60	131.8	----------------
16	Jacksonville - Miami	322	300	96.6	9	67.5	164.1	----------------
17	Atlanta - Birmingham	144	275	39.6	4	30	69.6	----------------

[a]Cost per mile varies with the cost of the land. These values are based on $250 000 to $300 000 per mile in central Alabama.

[b]Pipeline costs including land acquisition are based on 30-in. class 3 lines. Maximum working pressure, 1200 psig.

[c]Booster stations located every 35 miles. Size of stations varies from 10 000 to 16 000 hp; cost varies from $750 to $1500 per hp.

TABLE G-8. - Concluded. COST OF PIPELINES FOR GAS TRANSMISSION

Run	Terminal	Distance, mile	Cost per mile[a] $	Cost of pipeline[b], $	Number of boosters	Cost of booster[c] $	Total cost, $	Remarks
18	Birmingham - Plant D	358	200	73.6	10	65	138.6	Mississippi River, $\$2\times10^6$
19	Plant D - Little Rock	79	200	15.8	2	13	28.8	----------------
20	Plant D - Ft. Worth	252	200	50.4	7	45.5	95.9	----------------
21	Ft. Worth - Houston	250	200	50.0	7	45.5	95.5	----------------
22	Houston - New Orleans	308	300	92.4	8	83.2	175.6	----------------
23	New Orleans - Plant D	300	200	60.0	8	60	120.0	----------------
24	Plant F - Albuquerque	68	175	11.9	1	6.5	18.4	----------------
25	Albuquerque - Denver	343	250	85.7	9	67.5	153.2	----------------
26	Albuquerque - Phoenix	343	200	68.6	9	58.5	127.1	----------------
27	Phoenix - Plant H	252	175	44.1	7	45.5	89.6	----------------
28	Plant H - Los Angeles	88	350	30.8	2	26	56.8	----------------
29	Plant H - Las Vegas	150	200	30.0	4	30	60	----------------
30	Las Vegas - Salt Lake City	362	200	72.4	10	65	137.4	----------------
31	Los Angeles - San Francisco	354	300	106.2	9	117	223.2	----------------
32	San Francisco - Sacramento	71	350	27.4	1	13	40.4	San Francisco Bay, $\$2.5\times10^6$
33	Sacramento - Plant G	450	250	112.5	12	124.8	237.3	----------------
34	Plant G - Seattle	183	250	45.8	5	65	110.8	----------------
	Totals	7447	---	2120.6×10^6	196	2063.0×10^6	4183.6×10^6	----------------

[a]Cost per mile varies with the cost of the land. These values are based on \$250 000 to \$300 000 per mile in central Alabama.

[b]Pipeline costs including land acquisition are based on 30-in. class 3 lines. Maximum working pressure, 1200 psig.

[c]Booster stations located every 35 miles. Size of stations varies from 10 000 to 16 000 hp; cost varies from \$750 to \$1500 per hp.

including failure mode studies should be used for system design guidelines to limit possible malfunctions. A safety plan must be developed encompassing materials, design, system safety analysis, inspection requirements, operational history, detailed check and operating procedures, and personnel certification.

Production

Hydrogen - Any gaseous hydrogen leaks must be monitored, and gaseous hydrogen and air mixtures must be kept below the critical level. (Flammable limits of hydrogen in dry air at atmospheric pressure are 4.1 to 74.2 volume percent hydrogen. In dry oxygen at atmospheric pressure, the flammable limits are 4.7 to 93.9 volume percent hydrogen.)

Visual fire detection is not practical. Hydrogen burns with a colorless flame. Additives can be used to add color to the flame. Class 1, group B, electrical equipment is required.

Methane - Gaseous methane production and handling is standard industrial practice. Class 1, group D, electrical equipment is required.

Oxygen - Liquid oxygen production both for industry and aerospace applications, has been produced in this country for many years. Material compatibility and cleanliness are keys to successful liquid oxygen production.

Storage and Piping System

Design and construction must consider the following:

(1) All applicable codes such as ASME pressure vessel, National Electrical Code, ASTM Standards, and NFPA Regulations

(2) Applications of quantity-distance requirements

(3) Proper marking of vessels and piping

(4) Fire protection-water deluge system

(5) Conversion of vented gases

(6) Burn stacks for disposal of hydrogen and methane

(7) Adequate ventilation

(8) Fail-safe valving systems

(9) Satisfactory materials for long-term use

(10) Leak detection

(11) Electrical grounding of system equipment

Operations considerations include the following:

(1) Detailed procedures must stress safety.

(2) Personnel protective equipment will be required.

(3) Inspection and maintenance procedures are required to insure cleanliness, to retest (periodic) vessels, to provide for calibration of gages and relief devices, and to cover emergencies (spills and fires).

(4) Thorough training of operational and maintenance personnel is required.

TACV On-Board Propellant Tanks

Tanks must conform to all applicable specifications and codes, such as Title 49 of the Federal Regulations and ASME, NFPA, ASTM, and NEC Documents.

The g-forces generated by train acceleration, deceleration, turning, and loads due to high winds must be considered.

Methane venting to the atmosphere should be prohibited to prevent hydrocarbon pollution. Hydrogen venting should be avoided if possible. Any venting of hydrogen should result in a nonflammable exhaust gas mixture.

In the event of leakage or spillage, adequate means must be provided to prevent propellant mixing. This also applies to mixing of vented gases. In the event of an accident, provision must be made for automatic purging to prevent an inflammable mixture from developing. It would be advantageous to have the dewars (if, for instance, liquid H_2 and O_2 are used) separated by some distance to prevent mixing. Devices such as sensing systems and automated purge systems to compensate for the unexpected must be provided. The cleanliness of dewars, especially for liquid oxygen service, must be strictly enforced. The same applies to all operational and maintenance procedures. Operating personnel must have instant knowledge regarding the condition of the propellant systems. They must be trained to react properly when hazardous conditions are approached or exceeded. Area warning systems concerning hazardous conditions must be developed.

TACV Refueling

Five concepts are considered: Removal and replacement of fuel containers, service station concept, replacement of fuel tender, replacement of entire engine-tender unit, and moving the engine to a refueled tender unit.

(1) Removal and Replacement of Fuel Containers:

Advantages - This is within the present state of the art and has good potential for safe operation. Containers would be reasonably small and could be installed and removed with automated equipment to allow rapid refueling. Passengers would not be exposed to large volumes of cryogenics since containers would be refilled outside of the terminal.

Disadvantages - The development of quick disconnects would be required to provide rapid, leak-proof operation. Extra containers and remote support equipment would be required at the terminal. Special emergency vent valves would be required if hold time at the terminal is extended.

(2) Service Station Concept:

Advantage - Procedures for storage and operation are developed and are standard practice in industry.

Disadvantages - Safe procedures must be developed for rapidly transferring large quantities of cryogenic fuel. Precautions must be taken to have satisfactory distances between the large quantities of stored propellants and the terminal to minimize the possible occurrence of accidents and their consequences. The need for large stores of cryogens close to the terminal might be avoided by hauling propellants to the fueling area in a prearranged schedule. However, losses due to transfer and venting make this method unattractive.

(3) Replace Fuel Tender:

Advantages - The complete fuel tender could be replaced rapidly with maximum safety (i.e., minimum exposure of passengers to large propellant quantities). The tender would be transported from the propellant plant (outside populated areas) to a terminal by conventional train where unit replacement would be accomplished by an automated ram or overhead crane.

Disadvantages - Additional tender cars would have to be purchased. Probably, this concept would be more expensive and no safer than concept (1).

(4) Replace Entire Engine-Tender Unit:

Advantages - This is similar to the previous concept and offers the same safety advantages. Safety is enhanced because there are no requirements to uncouple and couple propellant lines in the terminal. In addition, the engine-tender unit can be interchanged rapidly.

Disadvantages - The principal disadvantage is the high cost of additional engine-tender units.

(5) Move Engine to Refueled Tender Unit:

Advantage - The engine is disconnected from a depleted tender and is reconnected to a fueled tender. This is the same as concept (4), except some cryogenic connections will be required. In addition, the cost would be less than concept (4), in that fewer engines will be required.

Disadvantages - Operational costs would be more than concepts (1) and (2).

Concepts (1) and (2) appear to be the most cost effective. Safety requires highly reliable, leak-proof, quick disconnects and automatic purge systems in the engine compartment. Passengers would be suitably separated from engine and tender while in the terminal using perhaps a purged shroud. Concept (1) is best safety-wise, but has highest initial investment potential.

Regulations

This area will require much study and development. Some considerations are as follows:

(a) Transport of existing fuels and cryogens is regulated by the DOT.

(b) Rail movement must be in tank cars (DOT SPEC 113 A60-W-2).

(c) Transport by tank truck requires a special permit from DOT.

(d) Tank trucks are prohibited in tunnels.

(e) New regulations must be developed for all methods of transporting each cryogenic gas and liquid on a nationwide scale as well as for local areas, design of systems and components, refueling operations, and governing travel over bridges and through tunnels.

KEY TECHNOLOGY AREAS FOR FURTHER STUDY

The following is a list of fuel related key areas that will require in-depth study for the purposes of improving state-of-the-art, developing new materials and components, increasing quality, reliability, and system safety, and evaluating and optimizing costs:

Production processes - Hydrogen and methane from coal, hydrogen and oxygen from electrolysis of water, and synthetic methane from waste materials, cellulose products, carbonaceous materials, etc.

Transportation and distribution - Existing gas pipelines, new gas pipeline systems; liquid pipeline transfer systems including valves, pumps, instrumentation, etc.; rail tank cars (cryogenic); truck trailers (cryogenic); barges (cryogenic); liquid transfer methods; power transmission through cold hydrogen pipelines.

Storage - Configuration and design of large storage facilities, configuration and design of liquid fuel tanks for vehicles, and hydride storage systems.

Components and subsystems - Quick disconnect-connect apparatus, leak detection systems, insulation types and methods, venting systems, automatic purge systems, hydro-pneumatic loading systems, automated control systems, electrical hardware, and instrumentation.

Cost analysis and optimization - Production processes and methods, transportation equipment and methods, storage and distribution equipment and methods, and refueling equipment and methods.

Safety - Evaluate chronological history of incidents involving fire, detonation, personnel injury, property damage; the development of safety rules and practices; the development of training methods; and the development of fire extinguishment techniques.

Regulations - Development of Federal regulations.

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APPENDIX H
FUEL AND OXIDIZER STORAGE SYSTEM COMPARISON

In this section the weight and volume of the cryogenic on-board fuel storage systems required for the alternative fuels methane and hydrogen are compared with that of the state-of-the-art tanks of conventional liquid-hydrocarbon fuels. Because weight and volume are important considerations for mobile applications, only fuel storage in the cryogenic liquid (rather than gaseous) form was considered. This requires well insulated spherical or cylindrical pressure vessels. Vacuum jacketed multifoil insulated and foam insulated tanks were considered for methane, which has a normal boiling temperature of 201° R. For hydrogen, which has a boiling temperature of 37° R, only vacuum jacketed multifoil insulation was considered. A vacuum jacketed cryogenic oxygen tank for the hydrogen-oxygen semiclosed engine described in section 4 was also considered.

The cryogenic fuel storage systems were compared with conventional, kerosene tanks. Kerosene tanks were assumed to be rectangular shaped, conforming to the available volume. The tank weight was set at 7 percent of the fuel weight based on data from a bus manufacturer's standard design. It was felt that kerosene tanks protected from road hazards and supported by other structure could meet this weight guideline.

Fuel storage system comparisons were made for two of the mission applications considered in the engine comparisons. These were the 300-mph interurban tracked air-cushion vehicle (TACV) and the 400 horsepower bus, which were described in appendix F. A total of 15 000 horsepower would be required for the TACV propulsion and air levitation. The TACV configuration was adapted from the conceptual designs considered in reference 1, where propulsion was by a linear induction motor with wayside power pickup and onboard power conditioning. For onboard power generation the engine, the fuel, and the fuel-storage system would replace the power conditioning unit. For reference in comparing the fuel storage systems, the engine and fuel storage were allotted about 20 feet of vehicle length at each end of the vehicle. Because of aerodynamic drag, vehicle size and shape are important considerations; therefore, the fuel storage would have to fit within the 9-foot-high by 12-foot-wide vehicle cross section used in reference 1. A total vehicle weight of 120 000 pounds was taken as a guideline, and fuel storage system weights were referenced to the total weight for comparison. For the bus application the total vehicle weight and space available for engine and fuel in a standard bus were assumed for this study.

In general, the cryogenic fuel storage tanks were sized to handle a nominal trip time allowing 10 percent ullage volume at the start. All insulation systems were designed to restrict heat flux into the fuel to the amount that would cause only 2 percent of the total fuel capacity to vaporize in 24 hours. It was assumed that tanks would be filled with saturated liquid fuel, and the design pressures were selected to provide a significant margin above saturation pressure. When the design pressure is reached, the tanks would be vented. This covers the case where the vehicle must stop and wait at a time when fuel tank pressures have built up to the vent point. Normally one might consider burning vented fuels for propulsion up to a rate of 100 percent boiloff in the mission time.

An initial comparison of cryogenic fuel-system weights was made for tanks designed using different assumptions as discussed in the next section. A comparison of cylindrical storage and spherical storage was also made for cryogenic fuels. Within this framework the influence of the number and diameter of cylindrical storage vessels is shown. A breakdown of each fuel system is included so the weight differences can be compared on the component level. The influence of specific fuel consumption and mission time or range are shown for both the bus and the TACV.

Design Philosophy

The Interstate Commerce Commission (ICC) specifications for liquified hydrogen tank car tank design were used at first. For the inner pressure vessel, the material specified was fully annealed 304 stainless steel. In the equation for cylindrical wall thickness, a welded joint efficiency of 90 percent was specified. The design pressure was 240 psig, and the design stress was the ASTM specified minimum ultimate tensile strength (75 000 psi at 70° F). The minimum gage was 0.1875 inch. For the vacuum jacket, carbon steel with less than 0.31 percent carbon was specified. A critical collapsing pressure of 37.5 psig was given along with a minimum gage of 0.4375 inch for both the cylindrical and head sections.

The ICC specifications for railway tank cars caused a serious weight penalty for the type of propulsion system studied here. These specifications were intended to prevent fire damage even during relatively long exposures and to prevent a major disaster if several tank cars on a train were to crash at the same time. Accordingly, the ICC specifications were modified for this study. For the pressure vessel a working pressure of 40 psig was combined with an allowable stress of 15 000 psi to allow a wall thickness about 25 percent less than the ICC specification. In the vacuum-jacket cases 15 psig was added to the 40 psig to account for removal of atmospheric pressure. The minimum gage was assumed to be 0.100 inch, which conforms to the Code of Federal Regulations 49-178.36-10 for cylinders. For the vacuum jacket an external pressure of 40 psig was used. The minimum gage of the vacuum jacket was reduced from 0.4375 to 0.100 inch. A bake-out temperature of 500° F was assumed to be required during vacuum-jacket evacuation. Criteria for buckling of the vacuum jacket were adjusted to account for the higher temperature. All materials were assumed to be 304 stainless steel, and the 15 000-psi allowable stress gave a safety factor (SF) of 2 on the yield value. This approach will be termed "modified ICC" in the rest of this discussion.

A second design approach was intended to more nearly approximate aerospace practice for the design of fuel storage. Stainless steel (304) remained the pressure vessel material but, to decrease weight, 5083 aluminum was selected for the vacuum-jacket material. The 5083 aluminum was chosen because of its good properties in the as-welded condition. No heat-treatment would be required after vacuum-jacket fabrication. The allowable stress for the stainless was kept at 15 000 psi (SF = 2.0), and the value for aluminum was set at 11 000 psi (SF = 1.5). The design internal and design external pressures were set at 30 psig with 15 psig added to vacuum-jacketed internal pressures to account for removal of atmospheric pressure. The actual internal pressure would be maintained at 17.5 psia by relief valves. The vacuum-jacket equations were also modified to assume room-temperature vacuum application with no bake-out as used in the modified ICC case. This approach will be termed "conservative aerospace" in the rest of this discussion.

The ASME boiler code was reivewed for comparison with the conservative aerospace assumptions. For a given internal pressure the stainless-steel tank designs would be the same as those used if one assumes fully radiograph inspected weld joints. The ASME vacuum jacket criteria allow use of 15 psid from external pressure to vacuum. The designs under conservative aerospace practice would support 10 psid pressure using ASME boiler code formulas. Under the conservative aerospace design it was calculated that the vacuum jackets would support 30 psid pressure.

The general relationships used in the parametric calculations are given in the section Description of Calculations. The fuel system weight (including fuel) was multiplied by a contingency factor of 1.1 to allow for fittings, support hardware, weld lands, etc. This was an extra degree of conservatism in all the calculations since this factor is normally applied to hardware weight only.

The equations were arranged so that the cylindrical tank walls were twice the spherical end-cap thickness. For example, if the end-cap thickness was minimum gage, the cylindrical walls were assumed to be twice the minimum gage instead of separately calculating the values. This was a significant weight addition in cases where the end cap minimum gage was significantly larger than that required by pressure (e.g., modified ICC criteria).

The assumption of limiting the heat leak into the cryogenic fuels such that no more than 2 percent vaporization could occur in 24 hours was somewhat arbitrary. It was anticipated this would cover all possible enroute delays. This required use of vacuum jacketing for liquid hydrogen, which was a significant weight factor. If the allowable heat leak could be increased somewhat, alternative insulation systems could be considered to achieve significant weight savings.

Train Fuel Storage Results

The first comparison for cryogenic fuel storage was between the modified ICC and the conservative aerospace design assumptions. Both spherical and cylindrical storage shapes were considered. Nominal values of specific fuel consumption and operation time (sec. 4) were chosen to make the comparison. Table H-1 summarizes the design input conditions as well as the results.

The required fuel weights were calculated by assuming that each engine operates at full rated power at the specific fuel consumption listed and for the time specified. Although actual engine operation would be partially off design, the comparison of fuel systems would still be valid. When a nominal vehicle weight of 120 000 pounds is chosen, all the fuel system weights except two range from 9 to 16 percent of the vehicle total weight. The two exceptions were both modified ICC design assumptions applied to a 4-foot-diameter cylindrical tank with vacuum-jacketed construction (hydrogen and methane fuels). The major weight of these structures was in the vacuum jacketing. The length of these cylinders (32 ft for methane and 68 ft hydrogen) and the design criteria chosen (40 psi external pressure and 500° F maximum temperature) required a very thick wall to resist buckling of the vacuum jacket. The selection of 304 stainless steel made the vacuum jacket quite heavy - 4700 pounds for methane and 12 400 pounds for hydrogen. Tank calculations and results given in section 7 were based on modified ICC design criteria and included the heavy vacuum jackets associated with long cylinders.

A state of the art kerosene tank is included in table H-1 for comparison. This was based on tank weight being 7 percent of the fuel weight. Cryogenic fuel storage in vacuum-jacketed pressure vessels was weight competitive with state-of-the-art kerosene storage systems.

TABLE H-1. - FUEL STORAGE FOR 15 000-HORSEPOWER TRAIN

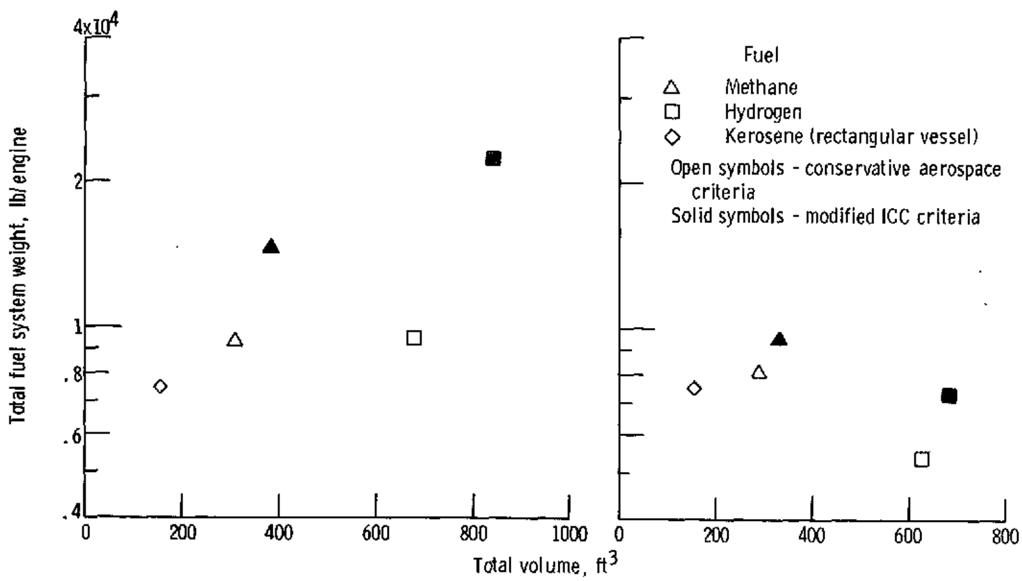
	Methane		Hydrogen		Kerosene
	Modified ICC	Conservative aerospace	Modified ICC	Conservative aerospace	
Net specific fuel consumption, lb/hp-hr	0.34	0.34	0.13	0.13	0.39
Operation time, hr	2.4	2.4	2.4	2.4	2.4
Fuel weight per engine, lb/engine	6100	6100	2300	2300	7000
Fuel volume per engine, ft^3/engine	230	230	510	510	140
Tank:					
Material	304SS	304SS	304SS	304SS	--------
Minimum gage, in.	0.100	0.030	0.100	0.030	--------
Internal design pressure, psia	55	45	55	45	--------
Allowable stress, psi	15 000	15 000	15 000	15 000	--------
Tank density, lb/in.^{3}	0.28	0.28	0.28	0.28	--------
Vacuum space, in.	4	2	4	2	--------
Vacuum jacket:					
Material	304SS	5083 Al	304SS	5083 Al	--------
Minimum gage, in.	0.100	0.030	0.100	0.030	--------
External design pressure, psi	40	30	40	30	--------
Allowable stress, psi	15 000	11 000	15 000	11 000	--------
Vacuum jacket density, lb/in.^{3}	0.28	0.096	0.28	0.096	--------
Maximum temperature of vacuum jacket, ^{o}F	500	70	500	70	--------
Sphere weight, lb/engine	3500	1944	5054	3149	--------
Sphere diameter (2 spheres), ft	8.56	8.23	10.95	10.62	--------
Cylinder weight, lb/engine	8461	3249	19 929	7172	--------
Cylinder diameter and length (2 required), ft	4 by 32	4 by 26	4 by 68	4 by 55	--------
Spherical totals:					
Total fuel system weight, lb	19 240	16 128	14 608	10 798	[a]15 000
Total vehicle weight, lb	120 000	120 000	120 000	120 000	[a]120 000
Fuel system weight as percent of total vehicle weight	16.0	13.4	12.2	9.0	[a]12.5
Cylindrical totals:					
Total fuel system weight, lb	29 162	18 738	44 354	18 844	--------
Total vehicle weight, lb	120 000	120 000	120 000	120 000	--------
Fuel system as percent of total vehicle weight	24.3	15.6	36.3	15.7	--------

[a]State of the art rectangular tank.

Figure H-1(a) shows the relation of the cryogenic fuels and design criteria for cylindrical pressure vessels. Figure H-1(b) shows the same relations for spherical pressure vessels. The spheres are lighter, particularly where the long cylindrical vacuum jackets are required.

It was decided that the modified ICC designs were not representative of state-of-the-art aerospace cryogenic storage technology (ref. 2), especially where long thin cylinders were concerned, because the thicknesses of all parts except the vacuum jackets were defined by minimum gage rather than pressure requirements.

The design criteria chosen for further discussion in this section were the conservative aerospace column of table H-1. These were selected as representative of state-of-the-art design but still relatively conservative by aerospace standards. Improvements in the conservative aerospace design will be discussed later. Data of section 7 were based on the modified ICC design criteria applied to long cylinders, however.



(a) Four-foot-diameter cylindrical vessel. (b) Spherical vessel.

Figure H-1. - Fuel system weight for two fuel vessel shapes. Number of engines per vehicle, two; engine output, 18 000 horsepower-hours.

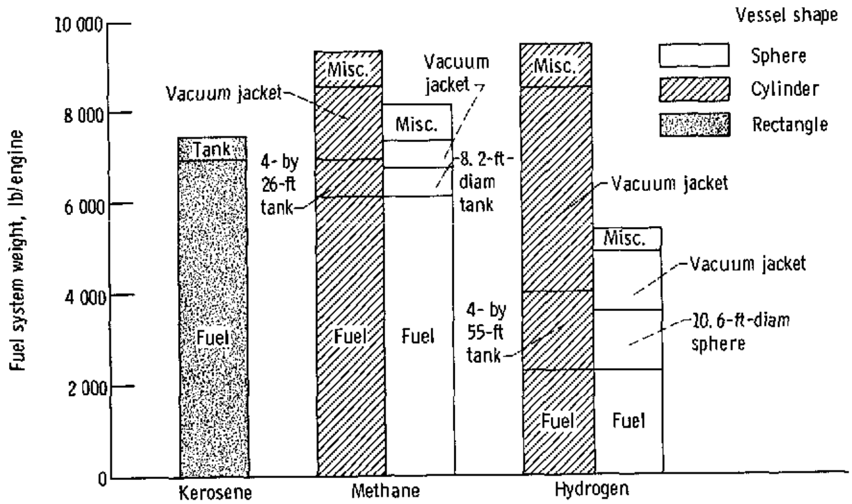


Figure H-2. - Fuel system weight estimates using conservative aerospace criteria. Number of engines per vehicle, one; engine output, 18 000 horsepower-hours.

Figure H-2 shows a weight breakdown by fuel for the conservative aerospace design. The state-of-the-art kerosene system is mostly fuel weight, so tank design parameters do not significantly influence system weight. Therefore, kerosene tanks can be fitted to existing space without weight penalties. Cryogenic fuels requiring vacuum jacketed insulation devote a greater fraction of the total weight to tankage. Therefore, it is more important for cryogenic fuel tanks to be carefully designed and efficiently packaged (e.g., spheres in fig. H-2).

The difference in vacuum-jacket weight between the spherical and cylindrical storage cases was largely due to the extra thickness required to prevent buckling for the 26- and 55-foot cylinders. Several methods were considered to avoid excessive vacuum-jacket length. Among these were ribs to support the jacket and effectively decrease its length to diameter ratio, diameter increases, and multiple cylinder applications. Diameter increases and multiple cylinder applications will be discussed later.

Figures H-1 and H-2 show that spherical fuel storage for methane is close to the weight of kerosene but with about 1.6 times the volume of kerosene. Hydrogen in a spherical tank has a 3200-pound per engine weight advantage over kerosene and methane but hydrogen requires four times the volume of kerosene for the same mission.

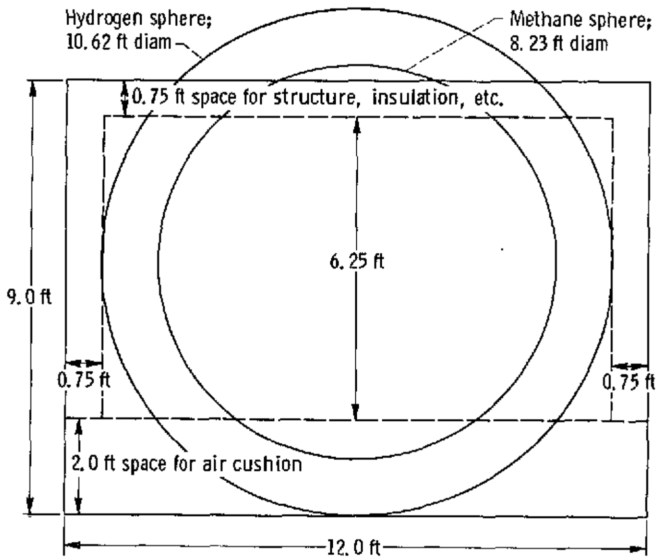


Figure H-3. - Fuel storage in spherical tanks for TACV.

Train fuel system packaging requirements are illustrated in figure H-3. The basic outside train dimensions are 12 feet wide by 9 feet high. A two-foot space under the propulsion compartments was assumed for the air cushion plenum. A 9-inch space was left around the outer cross section for car structure including skins, stiffeners, and insulation. The size of the spheres illustrates the scale of cryogenic fuel storage required for 2.4 hours of full power operation. The methane sphere intrudes somewhat on the reserved spaces but could probably be accommodated. The hydrogen sphere is definitely outside the vehicle limits and would require special structure or resizing the train cross section for accommodation. The smallest kerosene tank of rectangular cross section would be a 5.4-foot cube.

Alternative storage possibilities for cryogenic fuels would include multiple spheres or cylinders. The length of these combinations becomes important (as was noted on fig. H-2) where a 4-foot-diameter cylinder required 26 feet of length for methane and 55 feet of length for hydrogen. This length is not available unless one considers storing fuel under the passenger compartment.

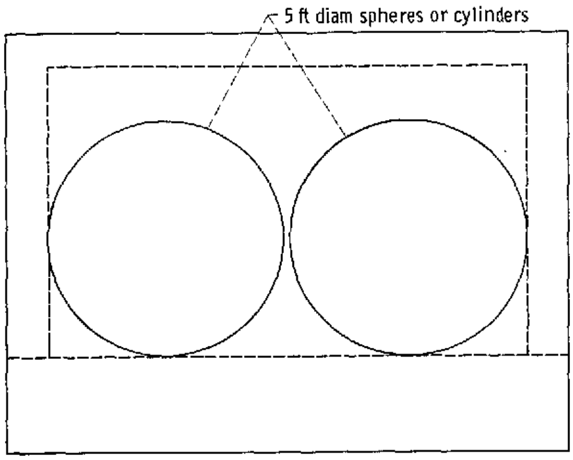


Figure H-4. - Five-foot-diameter fuel storage for TACV.

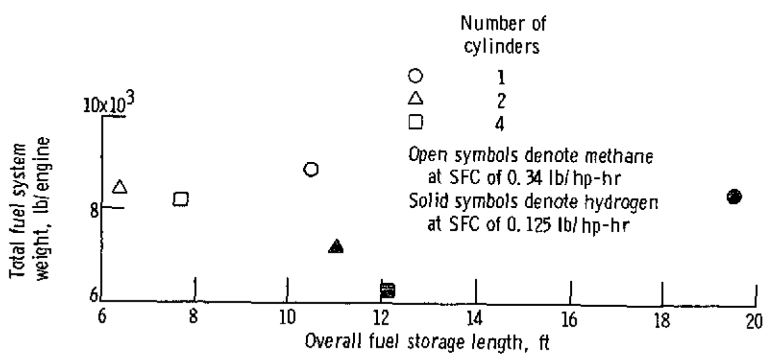
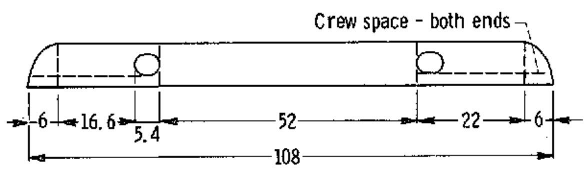


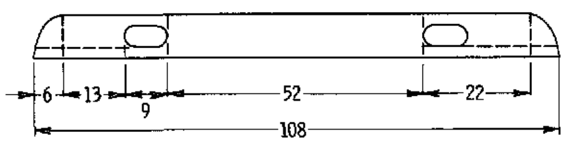
Figure H-5. - Influence of multiple cylinders on fuel system weight and length. Cylinder outside diameter, 5 feet; number of engines required, two; engine output, 18 000 horsepower-hours.

Figure H-4 gives a possible arrangement of side-by-side fuel storage using either 5-foot-diameter spheres or cylinders. The length relationship for cylinders is given in figure H-5. Only pairs of cylinders are considered for side-by-side storage. The length of one cylinder is given for reference. The weight saving for hydrogen is largely a decrease in vacuum-jacket thickness. Four 10-foot-long cylinders weigh about 1000 pounds less than two 18-foot-long cylinders. The weight saving requires an increase of about 2 feet in fuel storage length. A part of this weight saving could probably be effected by applying stiffeners to the 18-foot-long cylinders which would allow a thinner vacuum jacket.

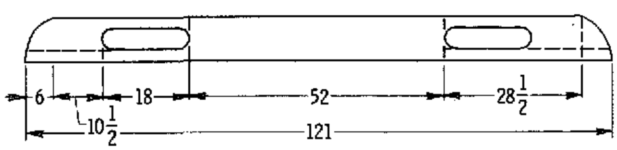
If the engine is about 11.5 feet long, as is the open-cycle Brayton of appendix E, the kerosene would fit within the remaining 10.5 feet as a 5.4-foot cube (or any convenient rectangular shape). Cylindrical storage for kerosene would require two 5-foot-diameter cylinders, each 5.7 feet long, essentially slightly elongated spheres. None of these arrangements would intrude on the reserved area significantly. Further space economy for kerosene could be made by specially shaped tanks designed to fit the reserved area and supported by normal structural members. The methane storage length of 9 feet for two 5-foot-diameter cylinders would use most of the alloted 10.5 feet. For hydrogen storage in two 5-foot-diameter cylinders, a length of 18 feet is required. This would necessitate lengthening the vehicle. Figure H-6 illustrates the length requirements for the various fuel systems with varying engine compartment lengths.



(a) Kerosene.



(b) Methane.



(c) Hydrogen.

Figure H-6. - Cylindrical fuel storage for TACV. Four cylinders per vehicle; cylinder diameter, 5 feet.

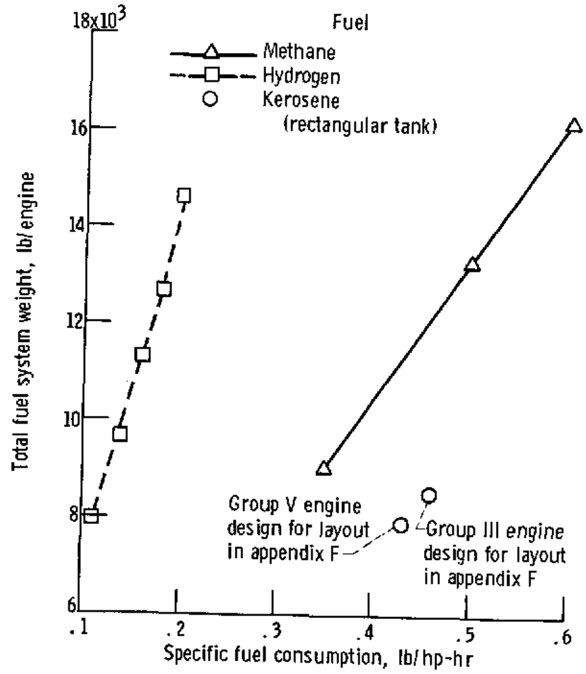


Figure H-7. - Influence of specific fuel consumption on fuel storage system weight. Two engines per vehicle; engine output, 18 000 horsepower-hours per engine; one cylinder per engine; cylinder diameter, 5 feet.

The sensitivity of fuel system weight to varying specific fuel consumption (SFC) was calculated for figure H-7. Mission profiles, duty cycles, and engine weight influenced the selection of design-point SFC. The SFC values used in table H-1 and in figures H-1 to H-5 were nominal values chosen for comparison of fuel storage parameters. The curves of figure H-7 are intended to cover the entire range of interest for each fuel. If SFC values other than those nominal values used previously are of interest, one may estimate the fuel system weight from figure H-7. A discussion of mission SFC for a variety of missions and engine designs is given in section 7. The values used in engine layouts of appendix E for kerosene are indicated on the curve. The hydrogen curve seems to be more sensitive to SFC than the others, but the relationships are similar - doubling the SFC approximately doubles the fuel storage weight.

The point is more clearly illustrated in figure H-8 where nominal values of SFC were assumed and the time of engine operation was varied. The time was translated into range at an average speed of 250 miles per hour. At that speed 10 55-mile segments could be completed within 2.4 hours of operating time. If fuel system weights are critical, one could shorten the range. The data of figure H-8 are for a single 5-foot-diameter cylindrical tank so that

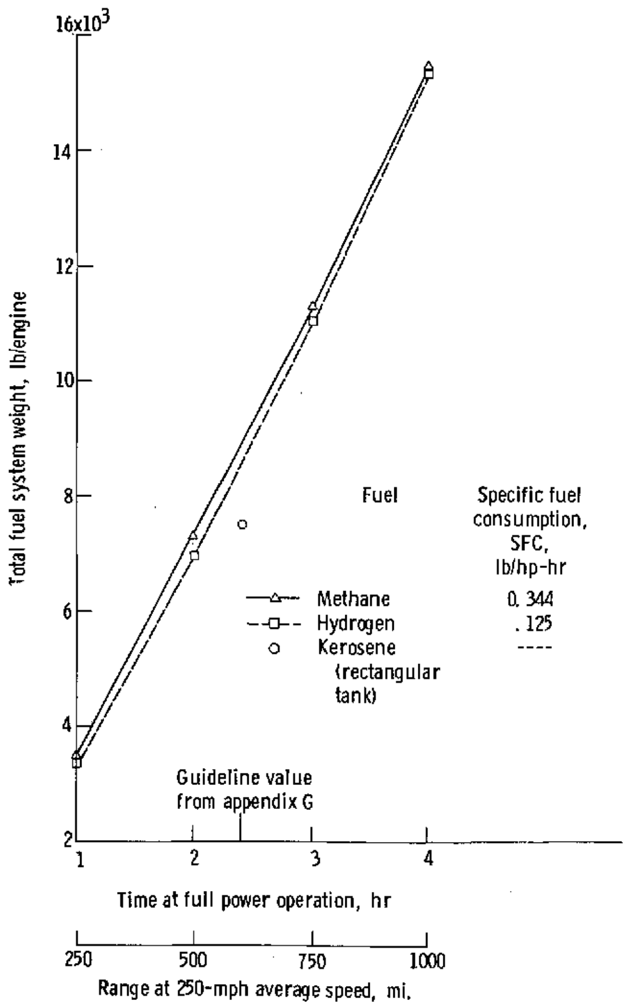


Figure H-8. - Influence of mission time on fuel system weight. Two engines per vehicle; engine power output, 7500 horsepower. One vacuum jacketed, cylindrical tank; cylinder diameter, 5 feet.

weight savings similar to those shown in figure H-5 could be expected if multiple tanks were used. The sensitivity of the kerosene fuel to operation time would be similar to that shown for cryogenic fuels.

The heat flux and insulation parameters were not varied from the nominal assumed values. The multilayer insulation weight was not a significant part of the total, so the only requirement was to make sure the number of layers needed to restrict the heat flux would fit in the vacuum space allowed. A more significant variation would be elimination of the vacuum jacket. One case of a polyurethane foam insulated methane tank was calculated to compare with vacuum jacketed fuel-storage weights. An effective conductivity of 0.15 Btu-inch per hour per square foot per $^{\circ}$R was used for the foam insulation, and a temperature drop of 320° R was assumed. The density of the foam was input at 2.0 pounds per cubic foot for the weight calculation. A 0.030-inch-thick aluminum skin was used over the foam insulation to provide mechanical protection and to seal against air and moisture. The total weight for a foam insulated sphere at the baseline condition (table H-1) would be 7800 pounds per engine or 15 600 pounds for the vehicle. The last value compares with a total of 16 128 pounds for the vacuum-jacketed methane sphere. The total weights are similar because the foam insulation and skin weigh 354 pounds per tank compared with 600 pounds per tank for the vacuum jacket. The thickness of the foam insulation required to limit boiloff to 2 percent in 24 hours was 8.24 inches for the spherical case. This results in an outside diameter of 9.31 feet for each sphere compared with the 8.23 feet for each vacuum jacketed sphere whose vacuum space was 2 inches. For storage of methane in cylindrical tanks, the foam insulation thickness would be 10 to 12 inches, depending on the tank diameter. This would necessitate enlarging the storage space either in diameter or length to account for the greater thickness. On the other hand, the foam insulated methane tanks could be tailored to the available area and perhaps save space even with the 12-inch thicknesses. Foam insulation for methane presents some packaging problems at essentially the same weight as vacuum jacketing. Vacuum-jacket cost and complexity would be significantly higher, however; so the economic and maintenance requirements might make the foam insulated system more attractive. If the boiloff criteria could be relaxed from

2 percent in 24 hours, a significant decrease in foam thickness could be realized.

The storage of liquid oxygen on board the train was studied briefly. For the same conditions as listed under the conservative aerospace column of table H-1, vacuum jacketed tanks were considered for two specific oxidant consumption values, 1.03 and 1.29 pounds per horsepower-hour (see sec. 7). Fuel system weight for each 7500-horsepower engine at the 2.4-hour operating time was 21 862 pounds for the lower consumption rate and 27 319 pounds for the higher rate. Sphere outside diameters would range from 8.56 to 9.19 feet. Considering that two of these spheres would be required per vehicle, the oxidant storage weight approaches 50 percent of a 120 000-pound vehicle weight. Most of the weight per engine (18 000 to 23 000 pounds) is in oxidant weight so that the basic decision involves whether the benefits of complete fuel-oxidant storage on board outweigh the volume and weight penalties associated with its storage. These values would undoubtedly require a larger, heavier vehicle than is now considered under the baseline case of table H-1.

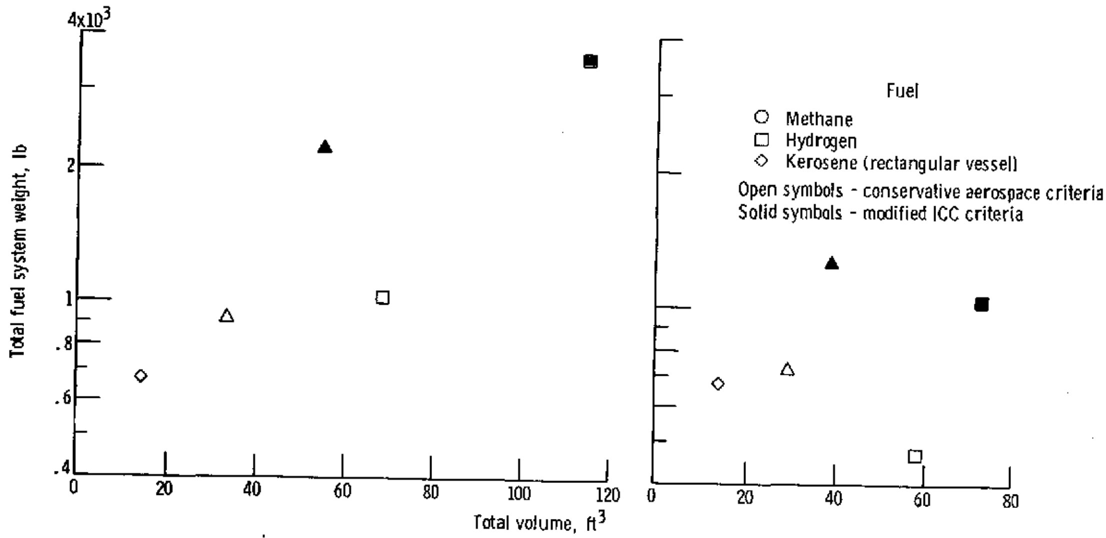
Bus Fuel Storage Results

Nominal values for SFC and operation time were assumed for bus applications (see table H-2). The combination gives a 1500 horsepower-hour engine output capability. A comparison of fuel storage system weight and size is included. As in the case of the train, the modified ICC designs were controlled by minimum gage specifications of 0.100 for spherical walls and 0.200 for cylindrical walls. In addition, the assumption of a 2-foot outside diameter with the 4-inch vacuum jacket spacing resulted in very long cylinders for methane and hydrogen storage. Steel vacuum jackets the minimum gage result in vacuum-jacket weights of 914 pounds for methane and 1855 pounds for hydrogen. For the purposes of this discussion, the modified ICC criteria were deemed unrealistic, so that the conservative aerospace case was selected as a baseline for discussion. Even the conservative aerospace vacuum jacketed cylindrical tanks were 11 and 23 feet long with jacket walls on the order of 0.200 inch thick. Aluminum jacket construction would save significant weight overall as may be seen in table H-2. The total fuel storage

TABLE H-2. - FUEL STORAGE FOR 400-HORSEPOWER BUS

	Methane		Hydrogen		Kerosene
	Modified ICC	Conservative aerospace	Modified ICC	Conservative aerospace	
Net specific fuel consumption, lb/hp-hr	0.36	0.36	0.13	0.13	0.42
Operation time, hr	3.75	3.75	3.75	3.75	3.75
Fuel weight per engine, lb/engine	540	540	188	188	630
Fuel volume per engine, ft^3/engine	22	22	49	49	13
Tank:					
Material	304SS				
Minimum gage, in.	0.100	0.030	0.100	0.030	--------
Internal design pressure, psia	55	45	55	45	--------
Allowable stress, psi	15 000	15 000	15 000	15 000	--------
Tank density, $lb/in.^3$	0.28	0.28	0.28	8.28	--------
Vacuum space, in.	4	2	4	2	--------
Vacuum jacket					
Material	304SS	5083 Al	304SS	5083 Al	--------
Minimum gage, in.	0.100	0.030	0.100	0.030	--------
External design pressure, psid	40	30	40	30	--------
Allowable stress, psi	15 000	11 000	15 000	11 000	--------
Vacuum jacket density, $lb/in.^3$	0.28	0.096	0.28	0.096	--------
Maximum temperature of vacuum jacket, ^{o}F	500	70	500	70	--------
Sphere weight, lb/engine	711	181	1044	282	--------
Sphere diameter (2 spheres), ft	4.21	3.85	5.19	4.83	--------
Cylinder diameter and length (2 required), ft	2 by 18.4	2 by 11.3	2 by 37.3	2 by 22.7	--------
Cylinder weight, lb/engine	1671	376	3379	838	--------
Spherical totals:					
Total fuel system weight, lb	1251	721	1231	470	[a]674
Total vehicle weight, lb	35 000	35 000	35 000	35 000	[a]35 000
Fuel system as percent of total vehicle weight	3.6	2.1	3.5	1.3	[a]1.9
Cylindrical totals:					
Total fuel system weight, lb	2211	916	3566	1025	--------
Total vehicle weight, lb	35 000	35 000	35 000	35 000	--------
Fuel system as percent of total vehicle weight	6.3	2.6	10.2	2.9	--------

[a]State-of-the-art rectangular tank.



(a) 4-foot-outside diameter cylindrical vessel.

(b) Spherical vessel.

Figure H-9. - Fuel system weight for two fuel vessel shapes. Engine output, 1500 horsepower-hours.

system weight for the conservative aerospace cases is only 2 to 4 percent of the total vehicle weight if a 35 000-pound vehicle gross weight is assumed. Again, the kerosene tank weight was set at 7 percent of the fuel weight.

Figure H-9 illustrates the weight and volume variations for spherical and cylindrical configurations. The influence of the modified ICC design criteria is quite obvious. Hydrogen fuel in a spherical tank is the lightest weight system for the conservative aerospace design, but it also has the largest volume.

The weight breakdowns for each of the conservative aerospace design systems is given in figure H-10. The kerosene system weight is mostly fuel, and the hydrogen system is mostly structure. The vacuum jacket for the hydrogen is quite heavy because of the wall thickness required to withstand buckling with a 22.1-foot-long cylinder.

The bus engine compartment with a semiclosed cycle (group III) engine installed is shown in figures E-12 and H-11. The only areas available for fuel storage would be along one side where the 24-inch spheres are shown in figure H-11. There would be room for four 24-inch spheres (or 4.19 ft^3 of fuel). For the open-cycle engine (group V), because the waste-heat exchanger is not present, there would be space for an additional four fuel tank spheres,

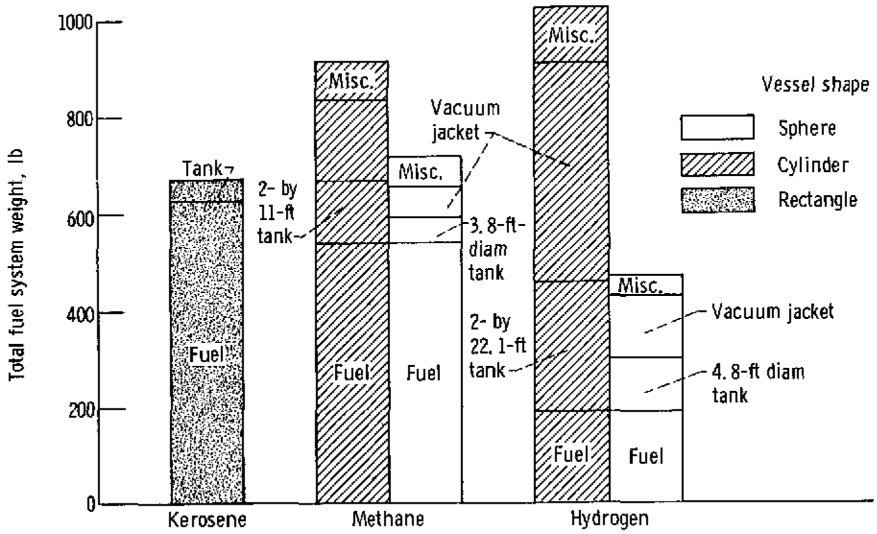


Figure H-10. - 1500-Horsepower-hour bus fuel system weight breakdown; conservative aerospace design.

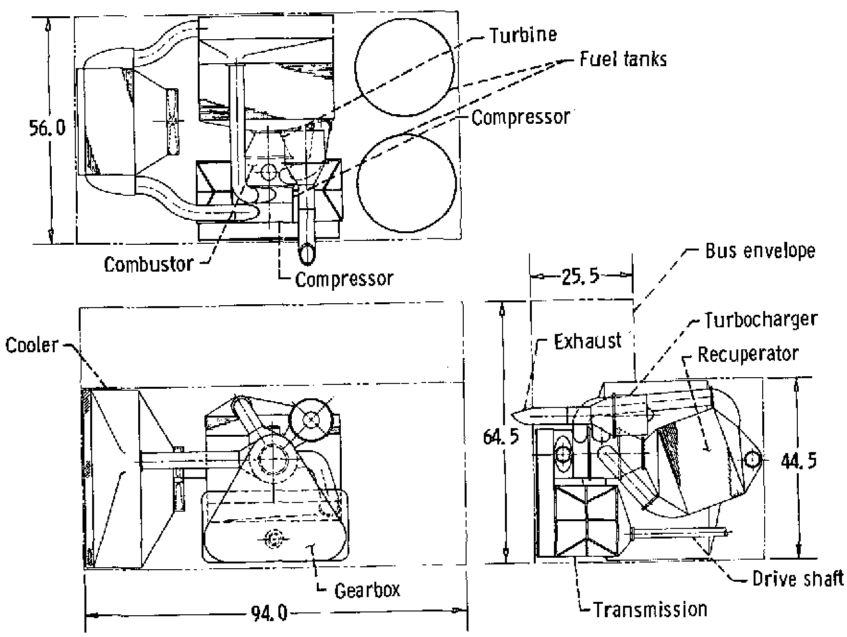


Figure H-11. - Engine layout for group III bus engine. Engine output, 400 horsepower (single shaft).

for a total volume of 8.38 cubic feet. This still would not provide the volume requirements given in table H-2 for any of the fuels. Storage over the engine compartment could be used, except that the rear window would be blocked. Fuel storage all around the engine would cause servicing, safety, and overheating problems.

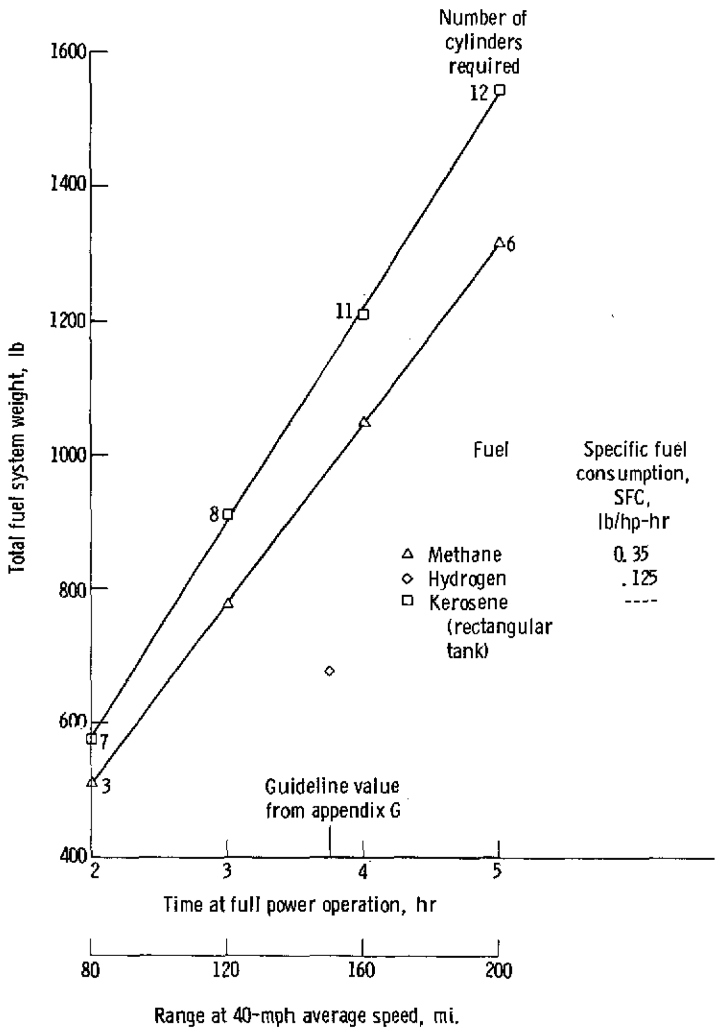


Figure H-12. - Bus fuel storage parameters. Vacuum jacketed cylindrical pressure vessel; jacket material, stainless steel with aluminum; tank length, 7 feet; outside diameter, 14 inches; engine output, 400 horsepower.

Fuel storage in contemporary buses is under the floor and ahead of the rear wheels. This compartment is approximately 7 feet wide by 7 feet long by 14 inches high. The use of this space for cylindrical tanks of cryogenic fuels was examined. All cylinders were assumed to be 14 inches in diameter to take advantage of the available space. Vacuum jacket space was also decreased from 2 inches for the nominal study to 1 inch. The results are plotted in figure H-12. Calculations were made for fuel system weight with tanks nominally 7 feet long. The numbers on the curves indicate the number of tanks used at each point calculations. The range of the bus at a 40-mph average speed is superimposed on the plot. Since the weight curves are approximately straight lines, extrapolations could be made to other ranges, such as to 400 miles. Kerosene fuel weight sensitivity would approximate the slope of the cryogenic fuels. The curves indicate that methane (five tanks) would fit in the present compartment space. Hydrogen, however, requires 11 tanks and an overall storage length of 13 feet. The present compartment could be enlarged to accommodate the hydrogen under the bus floor, or the engine compartment or exterior storage could be used.

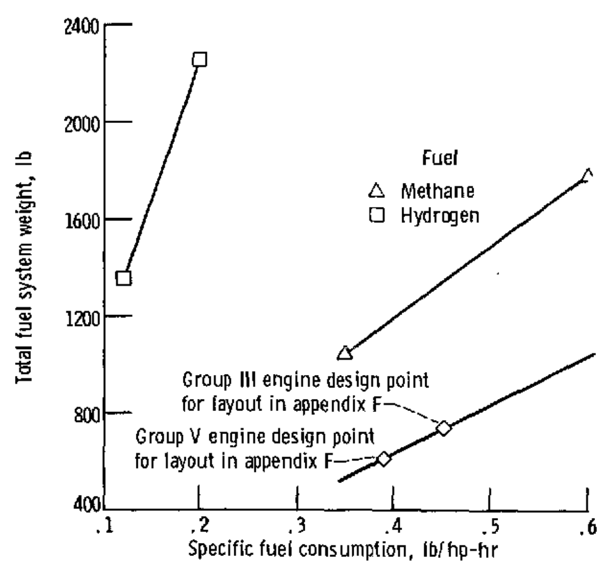


Figure H-13. - Influence of specific fuel consumption on fuel system weight. Multiple cylinders; cylinder length, 7 feet; cylinder outside diameter, 14 inches. engine output, 1500 horsepower-hours.

The sensitivity of fuel system weight to SFC variations in the assumed design point SFC is given in figure H-13. All curves are for single, 1-foot-outside-diameter, cylindrical tanks. The values used for kerosene in the engine layouts of appendix E are indicated on the kerosene curve.

Remarks

Parametric studies were made of possible fuel storage systems for a TACV and an urban bus. The fuels considered were kerosene, liquid methane, and liquid hydrogen. One oxygen storage design was considered for the train. The weight for cryogenic fuel storage was competitive with kerosene when conservative aerospace designs were used. Cryogenic designs using modified ICC criteria were also weight competitive unless long cylindrical tanks were used. The cryogenic fuels require rather heavy insulation systems (vacuum jackets). Volume requirements for kerosene were about 1/2 that for methane and 1/4 that for hydrogen. Cryogenic fuel storage could easily be provided for in the design of a TACV. Use of hydrogen fuel in buses would require extra space now reserved for other purposes. Several insulating schemes using foams showed promise, especially for cases where the boiloff criteria could be relaxed. More detailed designs would be required to establish weights. The designs used in this study were felt to be representative of conservative aerospace practice. Advanced design techniques and material selection could allow significant weight savings.

Description of Calculations

All fuel storage system calculations began with fuel weight and volume. The required weight was derived by multiplying mission specific fuel consumption and mission energy requirements in horsepower-hours. Tank volume was derived by dividing the propellant weight by the density and multiplying by 1.1, to allow a 10-percent ullage space.

Spherical tank wall thickness was calculated by the membrane stress equation of reference 3. Tank weights were added to fuel weight, and the result multiplied by 1.1 to allow additional weight for fittings, weld lands, support hardware, etc. Application of this factor to the fuel as well as the tank should allow some weight reduction when an actual tank is designed.

Cylindrical tank wall thickness was calculated by the membrane hoop stress equation of reference 3. End closures for cylindrical tanks were assumed to be hemispherical. Use of these equations results in cylindrical walls twice as thick as the ends. When the hemispherical ends were determined by minimum gage specifications, the cylindrical walls were made twice the minimum gage. No thickness was added for shear and bending stresses at the junction between hemispheres and cylinders - this allowance being in the 10 percent value discussed previously.

Vacuum jacketed tanks were designed for the same working pressure with allowance made to account for removal of the atmosphere from the pressure vessel. Vacuum jacket spacing was selected arbitrarily for each case as was the outside diameter of the vacuum jacketed assembly. Spherical vacuum-jacket required thickness was dependent on the material used, the external pressure assumed, and the maximum temperature of the vacuum jacket during its lifetime. Values were calculated using the curves of reference 2 (p. 108). They were derived from the classical Zoelly equation of reference 3 increased by a factor of 4.7 to account for experimental data of reference 2. Cylindrical vacuum jacket thickness was also a function of cylindrical length to diameter ratio. The data of reference 2 (p. 112) were used to derive thicknesses based on a known diameter. The curves were based on classical equations of reference 3 with the thickness multiplied by 1.3 to account for the experimental data of reference 2.

The heat-leak allowed was assumed to be the amount required to vaporize 2 percent of the stored fuel (full tank) in a 24-hours. Insulation used in the vacuum space was designed to meet this requirement. Double aluminized Mylar insulation was selected. The weight of the insulation was based on a density of 0.0042 pound per square foot per layer. The calculated thickness for the foam insulated case assumed the same heat leak but used a conductivity of 0.15 Btu-inch per hour per square foot per $^{\circ}$R. A temperature difference of 320° R was assumed across the insulation. The weight of the foam insulation was calculated using a nominal density of 2 pounds per cubic foot. An aluminum skin, 0.030-inch thick, was assumed to be required to protect the foam from damage and prevent water-vapor penetration of the foam.

In addition to the general equations, a minimum gage material thickness was assigned for all configurations. The minimum gage value was based on general handling considerations and was set at two levels: a 0.100-inch thickness to represent the Code of Federal Regulations 49-178.36-20 and a 0.030-inch thickness to represent conservative aerospace practice.

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2. Cryogenic Storage Systems Design, Fabrication and Evaluation. Instruments and Life Support Division, Bendix Corporation, March 27, 1968.
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APPENDIX I
THERMODYNAMIC CYCLE COMPARISONS OF CLOSED, SEMICLOSED, AND OPEN BRAYTON

During the preliminary cycle screening at the beginning of the study (see sec. 4), simplified cycle calculations were made to compare variations of closed, semiclosed, and open cycles. The results of these calculations are presented in this section.

The overall efficiency of the externally fired closed Brayton is the product of the closed Brayton cycle efficiency and the combustion loop efficiency. The closed Brayton cycle efficiency is the gross shaft power divided by the thermal input to the closed cycle; the combustion loop efficiency is the heat input to the closed cycle divided by the fuel heat release rate. Three variations of externally fired closed Braytons were considered in the study. They differ in the type of combustion loop used and will therefore be compared by comparing their combustion-loop efficiencies while holding the closed-cycle efficiency constant.

To compare the semiclosed Brayton cycle with the closed and open cycles, the semiclosed cycle is thermodynamically described as an open cycle superimposed on a closed cycle. Using this approach, the overall cycle efficiency of the semiclosed cycle, defined as gross shaft output power divided by fuel heat release rate, can be expressed as a function of the efficiencies of the closed and open portions. The overall efficiency of the semiclosed cycle can then be conveniently and consistently, compared with the

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overall efficiency of the externally fired closed Brayton cycles by holding the efficiencies of the closed portions equal. Open Brayton cycle efficiencies were also compared with the efficiency of the open portion of the semiclosed cycle. It will be shown that, in addition to allowing convenient comparison with completely closed and open cycles, this approach to analyzing semiclosed cycles contributes to the understanding of the changes in semiclosed cycle performance with variations in cycle parameters.

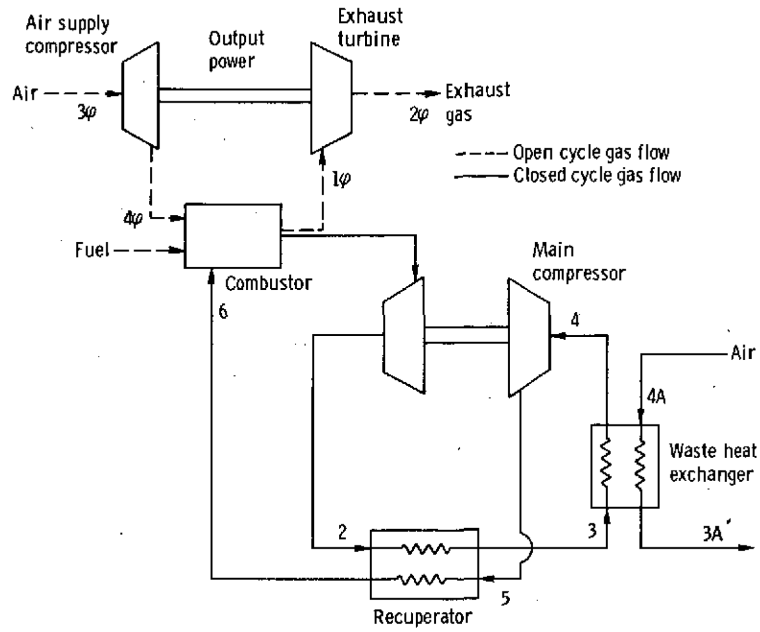
Approach

Some simplifying assumptions were made only for these preliminary calculations and were not used for the cycle calculations made for the analysis of sections 6 to 8. Perfect gases with uniform and constant properties were assumed. Pressure losses in all gas loops were neglected. As a result of neglecting pressure losses, the effects of fans (power required and temperature increases across them) that are required in the closed Brayton combustion loops and for the waste heat exchanger coolant air are not included.

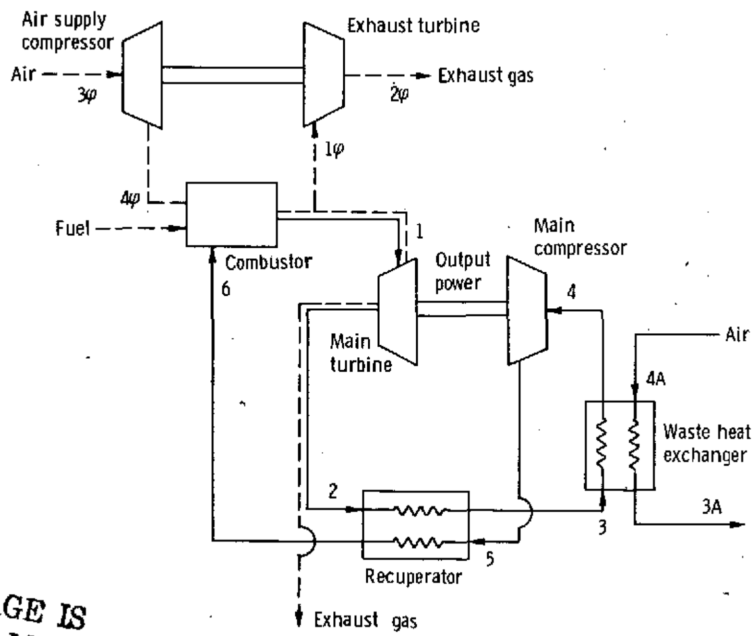
In the semiclosed cycles considered, combustion products are used as the working fluid of the closed portion. For the heat rejection temperatures of interest there would be condensation of water from the combustion products in the waste-heat exchanger. For purposes of these cycle calculations this condensation was neglected, although this assumption is not required in order to analyze the semiclosed cycle as a combination of closed and open cycles.

SEMICLOSED BRAYTON

In the type of semiclosed cycle considered here, the thermal energy released by the fuel is transferred directly to the cycle working gas, as in an open Brayton cycle. The combustor is included in the Brayton gas loop; the combustion gases being the Brayton working gas. Unlike an open Brayton cycle, however, excess air is not used as a combustor diluent. Most of the combustor exit gas is cooled in a waste-heat exchanger after expansion, compressed, and recirculated to the combustor as a diluent. The flow path



(a) Configuration with single exhaust point.



(b) Configuration with multiple exhaust points.

Figure I-1. - Semiclosed Brayton cycle schematics.

of the recirculated gas is similar to that of the working gas of a closed Brayton cycle. The combustion air and fuel, in near stoichiometric proportions, are raised to the combustor pressure and introduced into the combustor where they react and mix with the recirculated gases. To maintain constant inventory, part of the combustion gas (a mass flow equal to the incoming fuel and air) is exhausted to the atmosphere from some point in the cycle. Depending on the point of rejection, this exhausted gas could be expanded through a turbine to provide enough power to drive the air compressor. The flow path of the fuel and air and of their combustion products is therefore similar to that of an open Brayton cycle. The open portion of the cycle provides the thermal input to the closed portion, and the closed portion provides the combustor diluent for the open portion of the cycle.

By changing the manner of integrating the closed and open portions, the overall schematic of the semiclosed cycle can be rearranged in many ways. Two of the examples considered are shown in figure I-1, which indicates the closed and open working gases. In these configurations the working gases of the open and closed portions of the cycle enter the combustor separately and leave the combustor mixed. In part (a) a mass flow rate of gas approximately equal to that of the sum of air and fuel flow rates (the open-cycle working gas) is then separated from the closed-cycle working gas at combustor exit and expanded in a turbine to atmospheric pressure. This turbine (the open-cycle turbine) drives the air compressor (the open-cycle compressor) and produces some output shaft power. In figure I-1(b) only enough of the open-cycle working fluid is separated at combustor exit to provide turbine work sufficient to drive the open-cycle compressor. The remainder of the open-cycle gas is expanded through the main turbine, providing useful output power, and then is rejected from the system between the turbine and recuperator. After being heated in the combustor by direct mixing with the open-cycle combustion products, the closed-cycle working gas is expanded in the main turbine, cooled in the recuperator and waste-heat exchanger, recompressed in the main compressor, heated in the recuperator, and then recirculated to the combustor.

As previously mentioned, there will be condensation of water from the combustion gases in the waste heat exchanger. Thus the gas that is recirculated to the combustor will have slightly different composition than the total gas flow leaving the combustor. Also, the mass of the condensate removed in the waste heat exchanger should be considered a part of the open-cycle working gas. Then for the configuration of figure I-1(a), for example, the mass flow of gas in the open-cycle turbine would be slightly less than the sum of the air and fuel flow rates, the difference being equal to the amount of condensate removed in the waste-heat exchanger.

In terms of the efficiency of the closed portion η_{cB} and the open portion η_{oB}, the overall efficiency of the semiclosed cycle is

$$\eta_t = \eta_{cB}\left(\frac{\dot{Q}_{cB}}{\dot{Q}_{hv}}\right) + \eta_{oB}\left(1 - \frac{\dot{Q}_{cB}}{\dot{Q}_{hv}}\right) \tag{3}$$

where the quantity $\dot{Q}_{cB}/\dot{Q}_{hv}$ is the fraction of the fuel heat release rate that is transferred to the closed cycle (i.e., the thermal split between the closed and open portions) and the quantity $\dot{Q}_{hv}$ is the product of the fuel flow rate and the fuel heating value. Note that for the special case where the open cycle produces no useful output power (i.e., $\eta_{oB} = 0$) the sole function of the open portion is to act as a combustion loop for the closed portion. In this case equation (3) reduces to the product of η_{cB} and $\dot{Q}_{cB}/\dot{Q}_{hv}$, which is analogous to the overall efficiency expression for the closed Brayton; in this case $\dot{Q}_{cB}/\dot{Q}_{hv}$ is analogous to a combustion-loop efficiency.

For specified component efficiencies, heat-exchanger effectiveness, ambient air temperature, turbine-inlet temperature, pressure ratios across the two compressors, and pressure-loss distributions, the thermodynamic state of the gas at each point in the cycle can be determined by conventional cycle calculations. An energy balance on the combustor yields the ratio of the gas flow rate for the closed portion (the combustor diluent) to the gas flow rate for the open portion. For the open portion the air-fuel ratio is a function only of the type of fuel; it is specified to exceed the stoichiometric ratio by an amount sufficient to insure complete combustion. With the thermodynamic state points and these flow rate ratios, the terms in equation (3) can be evaluated.

CLOSED BRAYTON

The thermal input to the closed Brayton cycle is supplied by a separate combustion loop and is transferred from the combustion loop to the Brayton working gas in the Brayton heat-source heat exchanger. Three types of combustion loops considered here are shown in figure I-2. The combustion loops in figure 3-4(a) and (b) use conventional combustors operating near the ambient pressure. In the case of figure 3-4(a) sufficient excess air is used as a combustor diluent to obtain the required combustor-exit gas temperature. In place of the excess air, the combustion loop in figure 3-4(b) uses recirculated combustion gases as a combustor diluent. The amount of air input to this combustion loop is only enough to insure complete combustion of the fuel.

In the combustion loop of figure 3-4(c) the combustor and Brayton heat source heat exchanger are combined. The combustion temperatures are controlled by direct heat transfer to the Brayton working gas. Potentially, if the integration of the heat exchanger and combustor is good enough to make this heat transfer effective in controlling the combustion temperature, a gaseous diluent would not be required. The air input for this combustion loop could then also be just slightly above the stoichiometric value.

If the closed Brayton cycle considered uses reheat, that is, reheat of the Brayton working gas between two stages of turbine expansion, two Brayton heat-source heat exchangers would be required, and the combustion loops would be modified accordingly. In the case of figure 3-4(a) and (b) it was assumed that the combustion gases would be divided at the combustor exit, with part going to each of the two Brayton heat-source heat exchangers. In the case of figure I-2(c) the preheated air would be divided between two combustion-heat-exchanger units.

The overall efficiency of the externally fired closed Brayton cycles is the product of the efficiencies of the closed Brayton cycle and the combustion loop:

$$\eta_t = \eta_{cl}\eta_{cB} \tag{4}$$

The closed Brayton cycle efficiency is conventionally defined as the output shaft power divided by the heat input to the Brayton gas in the heat-source heat ex-

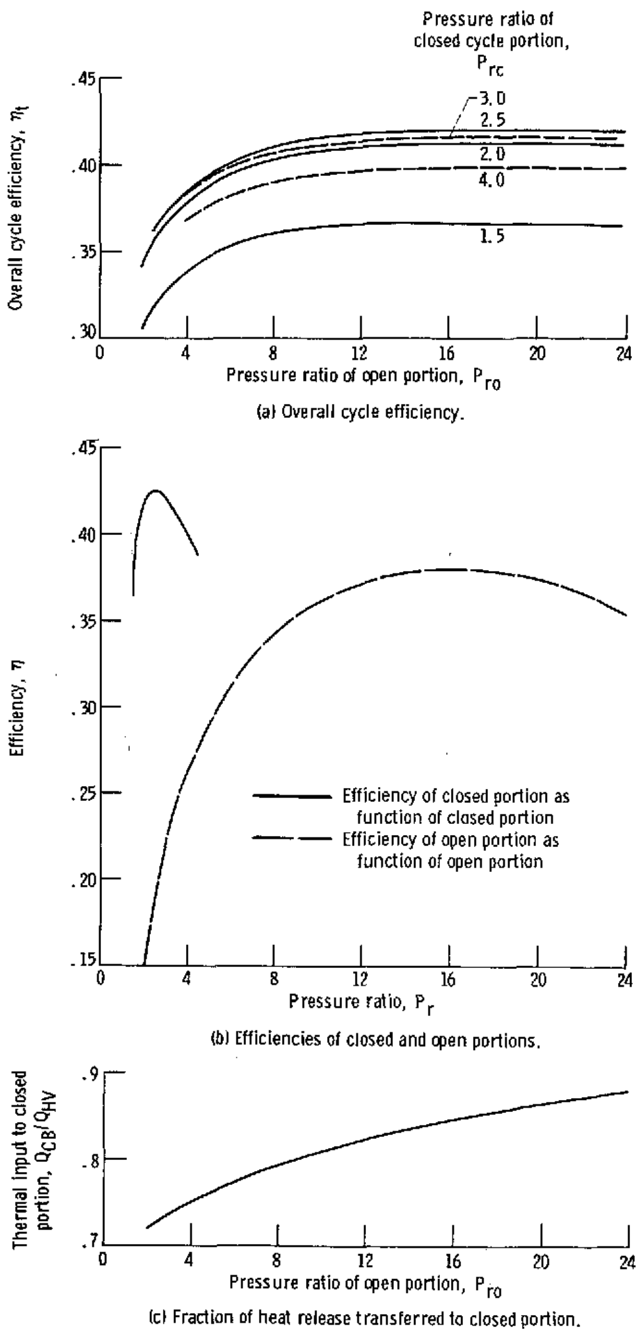


Figure I-2. - Semiclosed cycle configuration of figure I-1(a). Main turbine inlet temperature, 1600° F; main compressor inlet temperature, 140° F; ambient air temperature, 80° F; recuperator effectiveness, 0.9; pressure losses, 0; polytropic efficiency of turbines, 0.89; polytropic efficiency of compressors, 0.87.

changer. The combustor loop efficiency is defined as the heat transferred to the Brayton working gas divided by the product of the fuel flow rate and the fuel heating value.

To calculate the combustion-loop efficiency, the thermodynamic state points and relative flow rates must first be calculated. For the combustion loops in figures 3-4(a) and (b) the temperature difference between the combustor exit gas and the Brayton turbine-inlet temperature is assumed. Then for known Brayton gas temperatures and for specified values of heat-source heat exchanger and air preheater effectiveness, and pressure losses, the temperature at each location in the combustion loop can be calculated. A heat balance on the combustor yields the ratio of the combustor-exit flow rate to the fuel flow rate. This flow rate ratio and the temperature change of the combustion gas flow in the heat source heat exchanger are then used to calculate the combustion-loop efficiency, that is, the heat transferred to the Brayton cycle divided by the product of the fuel flow rate and heating value.

For the combustion loop of figure 3-4(c), the temperature difference between the combustion gases at the exit of the combustion-heat exchanger and the Brayton gas at inlet to the combustion-heat exchanger is assumed. With known Brayton gas temperatures and specified fuel-air ratio, air preheater effectiveness, and pressure losses, the temperatures at each point in the combustion loop can be calculated. This information is sufficient to calculate the combustion-loop efficiency.

By presenting the efficiencies as suggested by equations (3) and (4), the three types of closed Brayton cycles are compared with each other and with the semiclosed cycles, all with the same value for the closed Brayton cycle efficiency, η_{cB}.

Results

SEMICLOSED CYCLES

The overall efficiency and the efficiencies of the closed and open portions were calculated for the two configurations of figure I-1. The efficiencies and the thermal split between the closed and open portions are summarized here as a function of the pressure ratio of the closed-cycle portion and the system pressure level. The system pressure level is determined by the combustor air supply compressor pressure ratio, which is referred to as the open-cycle pressure ratio.

The overall cycle efficiency for the semiclosed cycle configuration of figure I-1(a) is given as a function of the pressure ratios of the closed and open portions in figure I-2(a). At a fixed value of the closed cycle pressure ratio, the overall cycle efficiency shown is constant over a broad range of cycle pressure level. In figure I-2(b) the efficiencies of the closed and open portions are given as functions of their respective pressure ratios. The efficiency of the closed portion is independent of the system pressure level. The efficiency of the open portion depends only on the system pressure level and is independent of the pressure ratio of the closed portion.

The fraction of the total fuel heat release, which is transferred to the closed portion, that is, the thermal split between the closed and open portions, is given in figure I-2(c) as a function of the open portion pressure ratio. For this particular cycle configuration it is independent of the closed-cycle parameters other than the turbine-inlet conditions. As shown, the semiclosed cycle becomes increasingly closed from the thermal standpoint as the system pressure level is increased. Simply, the heat input to the closed portion is the fuel heat release remaining after the open-cycle gases have been heated to the combustor-exit temperature. As the pressure level P_{ro} is increased, the temperature of the air input to the combustor increases so that the fraction of the fuel heating value required to further heat it to the combustor-exit temperature is decreased.

The overall cycle efficiency in figure I-2(a) is insensitive to pressure level because of the large weighting of the closed-cycle efficiency contribution to the overall efficiency. Also, because of the large weighting of the closed cycle efficiency, the total cycle efficiency peaks at values only slightly lower than the closed-cycle efficiency. The importance of the open-cycle efficiency should not, however, be minimized. For example, if the pressure ratio of the exhaust turbine is restricted so that the turbine work equals the air compressor work, the open-cycle efficiency would become zero and the total-cycle efficiency would be the product of the closed-cycle efficiency and $\dot{Q}_{cB}/\dot{Q}_{hv}$. The curves of figures I-2(b) and (c) show that the product of these two quantities would be significantly less than the curve of figure I-2(a).

In the cycle configuration of figure I-1(b), the power of the exhaust-gas turbine is matched to the power required by the air supply compressor, but without reducing the efficiency of the open-cycle portion to zero as just dis-

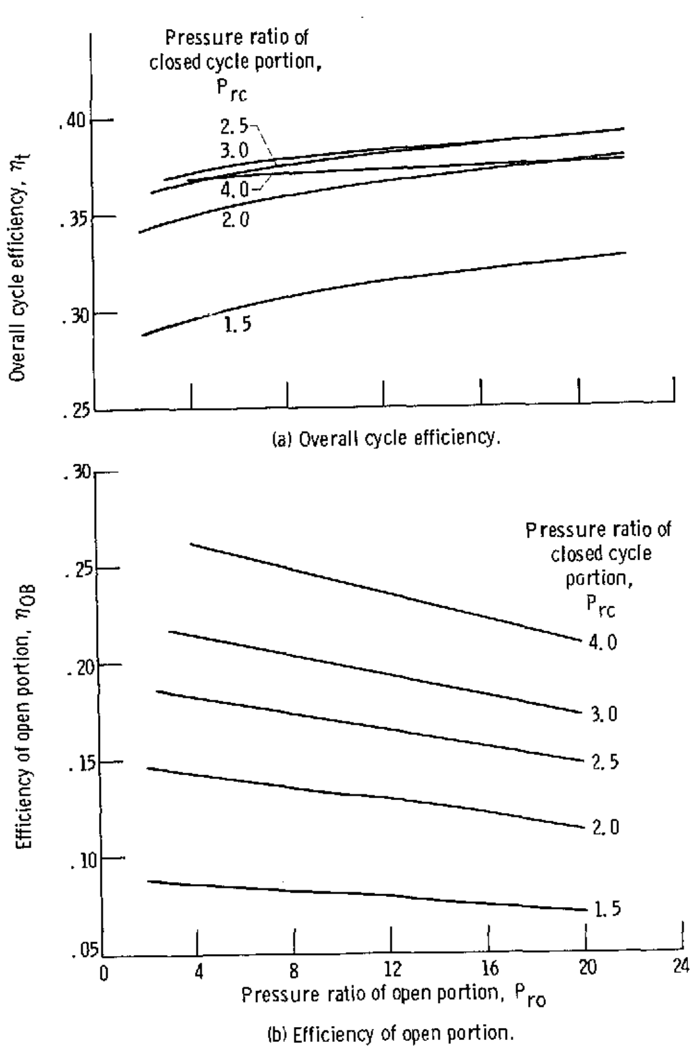


Figure I-3. - Semiclosed cycle configuration of figure I-1(b). Main turbine inlet temperature, 1600° F; main compressor inlet temperature, 140° F; ambient air temperature, 80° F; recuperator effectiveness, 0.9; pressure losses, 0; polytropic efficiency of turbine, 0.89; polytropic efficiency of compressors, 0.87.

cussed. Only part of the open-cycle working fluid flows through the exhaust turbine in this case. The remainder flows through the main turbine producing useful power (which is credited to the open portion of the cycle) and is then exhausted to the atmosphere.

The overall cycle efficiency for this configuration is given in figure I-3. For the assumptions made in this analysis, the efficiency of the closed portion of the cycle and the thermal split between closed and open portions are the same for this cycle configuration as for the one in figure I-1(a). In this case, however, the efficiency of the open portion of the cycle is dependent on both the open-cycle pressure ratio P_{ro} (system pressure level) and the closed-cycle pressure ratio P_{rc} as shown in figure I-3(b). Recall that part of the open-cycle gases are rejected from the system at a pressure at or above atmospheric. For a fixed value of P_{rc} and increasing P_{ro} or for a fixed value of P_{ro} and a decreasing P_{rc}, the pressure at which this portion of open cycle gas is rejected (and hence the power wasted) is increased. This is the reason for the behavior of the efficiency of the open-cycle portion shown in figure I-3(b).

A comparison of the overall cycle efficiencies for the two semiclosed configurations in figure I-1 shows that neither is a strong function of system pressure level over a broad range of pressures. Because some of the open-cycle gas in the configuration of figure I-1(b) is rejected at pressures greater than atmospheric, the efficiency of the open-cycle portion is lower for this case. This is reflected in slightly lower overall efficiency and by the slightly greater dependence on pressure level. The configuration of figure I-1 might, however, be preferred if the excess power produced by the exhaust turbine in configuration I-1(a) cannot be conveniently or effectively utilized.

Consider for a moment the effect on overall efficiency if the open-cycle gases, which are rejected at the exit of the main turbine in the configuration in figure I-1(b), had been passed through the recuperator and then rejected (at lower temperature) at the waste-heat exchanger inlet. In such a case, part of the thermal energy of that portion of the open-cycle gas is transferred to the closed-cycle gases on the high-pressure side of the recuperator. However, for the same recuperator effectiveness, the amount of energy transferred between the closed-cycle gases on the two sides of the recuperator is reduced because of the energy exchange between the open- and closed-cycle gases. As a result, the waste-heat-exchanger inlet temperature and the waste-heat rejection are higher. The net result is that the overall cycle efficiency is unchanged, the amount of energy recovered from the open-cycle gases as they pass through the recuperator being balanced by increased energy rejection in the waste heat exchanger.

As previously discussed the open-cycle portion of the semiclosed cycle differs from an independent open-cycle Brayton in that it depends on the closed portion of the cycle to provide the combustor diluent. The air-fuel ratio for the open portion of the semiclosed cycle is near stoichiometric, and only a portion of the fuel heating value is used to heat the open-cycle gases to turbine-inlet temperature. In the case of the semiclosed cycle, most of the fuel heating value is input to the closed portion as shown by figure I-2(c). In contrast, the independent open cycle provides its own combustor diluent (excess air) and consequently is charged with the entire fuel heating value. For the semiclosed cycle considered in figure I-1(a), the efficiency of the open-cycle portion is higher than that of a comparable independent open cycle as shown in figure I-4. This comparison is made only to illustrate the uniqueness of the open portion of the semiclosed cycle.

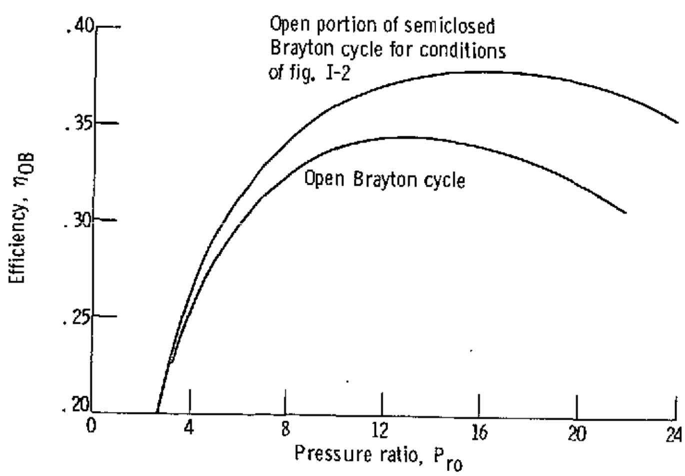
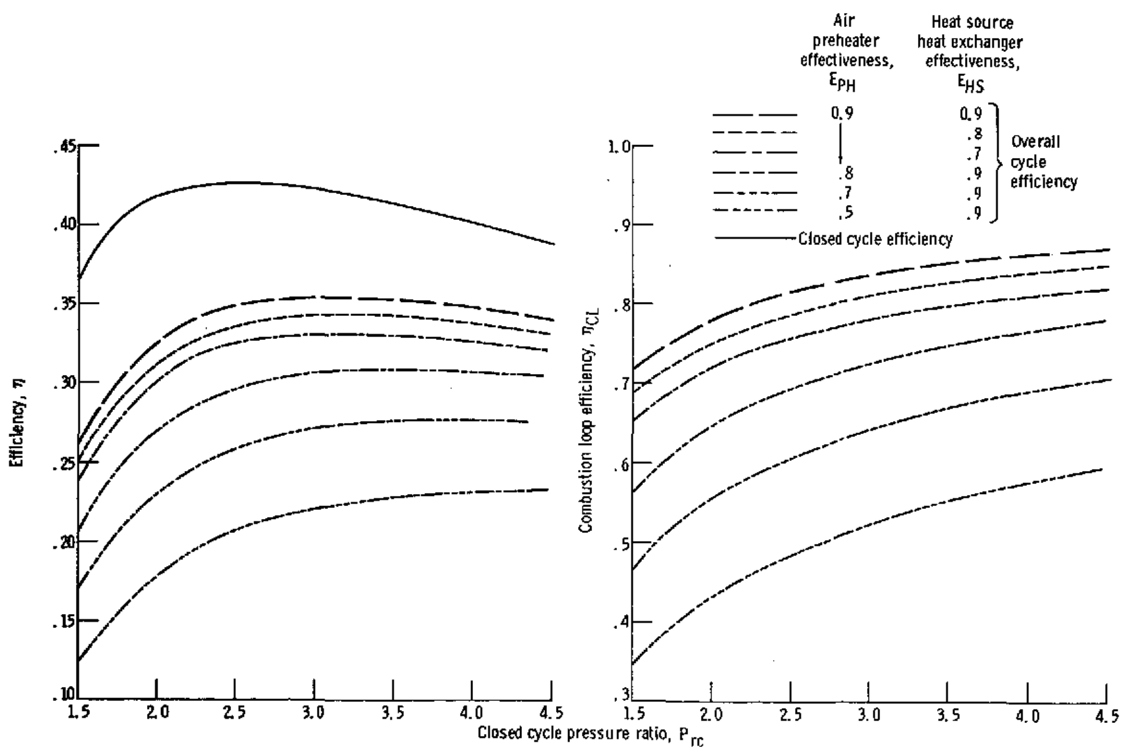


Figure I-4. - Comparison of open cycle efficiencies.

CLOSED CYCLES

The overall efficiencies for closed Brayton cycles using the combustion loops shown in figure 3-4 were calculated. A closed cycle, which is thermodynamically identical to the closed portion of the semiclosed cycle in the results just presented (fig. I-2(b)) was used. The three types of combustion loops are compared and each, in turn, can be compared with the semiclosed cycle results.

For the combustion loop of figure 3-4(a), the conventional combustor with air diluent, the overall cycle efficiency is shown in figure I-5(a) as a function of closed-cycle pressure ratio and the heat-transfer effectiveness of the heat-source heat exchanger and air preheater. Also shown is the efficiency of just the closed Brayton cycle. The difference between the closed Brayton



(a) Overall efficiency and closed cycle efficiency.

(b) Combustion loop efficiency.

Figure I-5. - Closed Brayton with combustion loop of figure 3-4(a). Turbine inlet temperature, 1600° F; compressor inlet temperature, 140° F; ambient air temperature, 80° F; recuperator effectiveness, 0.9; pressure losses, 0; polytropic efficiency of turbine, 0.89; polytropic efficiency of compressor, 0.87; temperature difference across heat source heat exchanger inlet, 50 F deg.

efficiency and the overall efficiency is the loss due to the combustion loop. The combustion-loop efficiency (fig. I-5(b)) is moderately affected by the heat-source heat exchanger performance and greatly affected by the air preheater effectiveness. The lower the air preheater effectiveness the higher the exhaust temperature and hence the energy loss of the combustion loop. Note that the overall cycle efficiency, even with both heat-source heat exchanger and the air preheater effectiveness at 0.9, is lower than the overall cycle efficiency for the semiclosed cycles considered. As the preheater effectiveness is increased, the combustor air-inlet temperature is increased and consequently the air-diluent flow rate must be increased to maintain the specified combustor-exit temperature. Had the fan power been included, the effect on efficiency would have been greater at the higher preheater effectiveness.

The flow rate in the air preheater in the combustion loop in figure 3-4(b) is less than the flow rate for the air-diluent combustor loop. As a result,

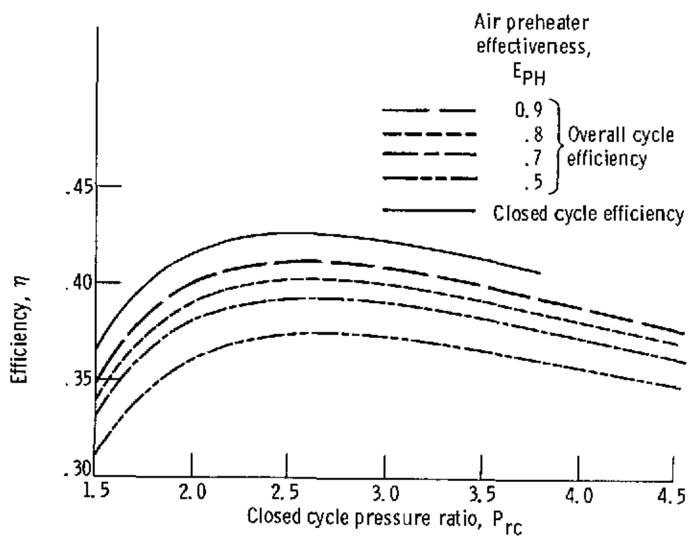
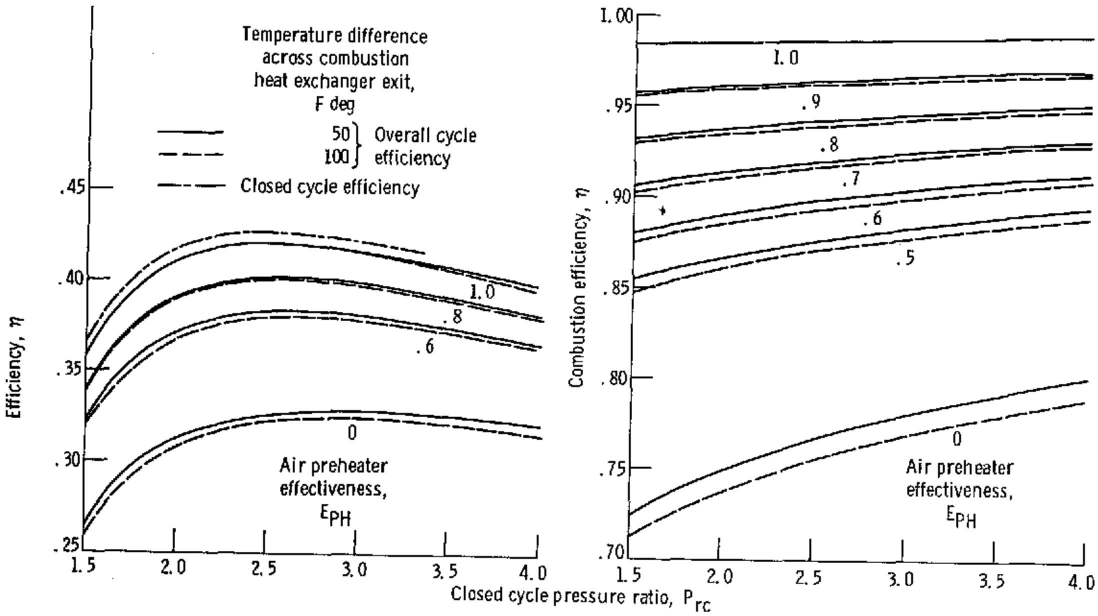


Figure I-6. - Efficiency of closed Brayton with combustion loop of figure 3-4(b). Turbine inlet temperature, 1600° F; compressor inlet temperature, 140° F; ambient air temperature, 80° F; recuperator effectiveness, 0.9; pressure losses, 0; polytropic efficiency of turbine, 0.89; polytropic efficiency of compressor, 0.87; heat source heat exchanger effectiveness, 0.9; excess air, 10 percent; temperature difference across heat source heat exchanger inlet, 50 F deg.



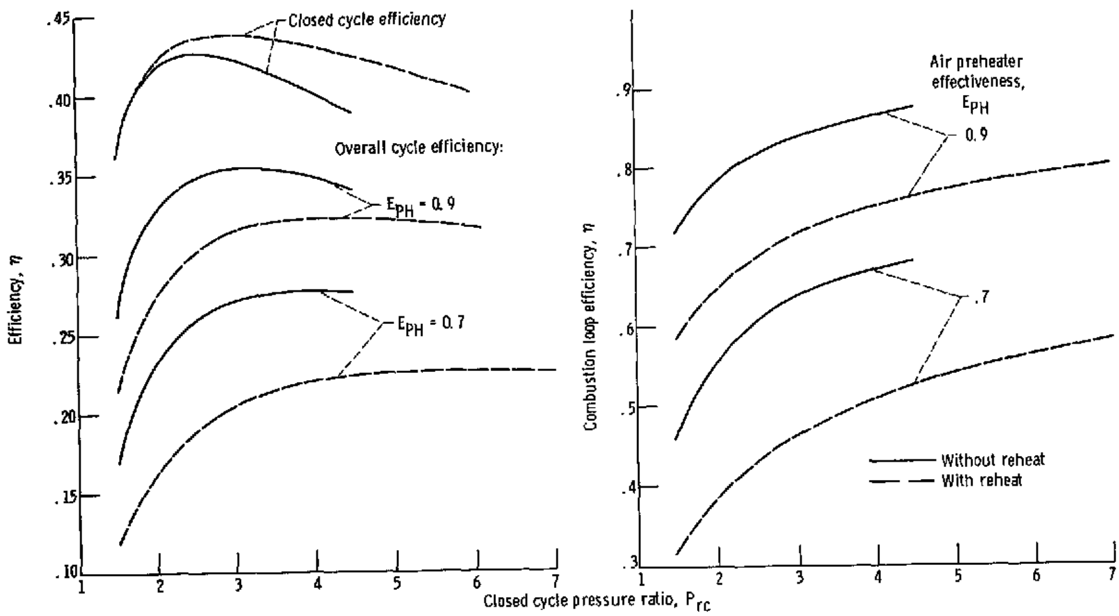
(a) Overall efficiency and closed cycle efficiency.

(b) Combustion loop efficiency.

Figure I-7. - Closed Brayton with combustion loop of figure 3-4(c). Turbine inlet temperature, 1600° F; compressor inlet temperature, 140° F; ambient air temperature, 80° F; recuperator effectiveness, 0.9; pressure losses, 0; polytropic efficiency of turbine, 0.89; polytropic efficiency of compressor, 0.87; excess air, 10 percent.

the overall cycle efficiency is much less sensitive to the preheater effectiveness and is higher than that for the air diluent case as shown by a comparison of figures I-6 and I-5(a). A disadvantage of the exhaust-diluent case over the air-diluent case, which is not reflected in these results, is that the fan for the recirculated exhaust diluent is located in a hot gas flow. With a preheater effectiveness of 0.9 the overall cycle efficiency shown in figure I-7 is just slightly less than that for the semiclosed cycle in figure I-2(a).

The flow rates in the preheater for the combustion loop in figure I-2(c) are similar to that in the preheater in the combustion loop in figure I-2(b). The efficiencies for this case (shown in figure I-7) are also close to those shown in figure I-6. The combustion loop efficiencies are shown in figure I-7(b). Had the fan powers been included, the efficiency for the case that uses the combustion-heat exchanger unit (fig. 3-4(c)) would be slightly higher than the case that uses the conventional combustor with exhaust gas diluent.



(a) Overall efficiency and closed cycle efficiency.

(b) Combustion loop efficiency.

Figure I-8. - Effect of reheat for conditions of figure I-5. Heat source heat exchanger effectiveness, 0.9.

The closed Brayton cycle using the near stoichiometric combustion-heat exchanger and that using the conventional combustor with exhaust gas diluent have comparable efficiencies. Both are superior in efficiency to the closed Brayton using the conventional combustor with air diluent, and both have overall efficiencies comparable to those for the semiclosed cycles considered.

The use of reheat increases the efficiency of the Brayton cycle (as shown in fig. I-8(a)) but decreases the combustion-loop efficiency (as shown in figure I-8(b)) for the combustion loop that uses air diluent. The reason is that when reheat is used, the temperature of the combustion gas at the exit of the heat-source heat exchanger is higher than when reheat is not used. As a result, for the same preheater effectiveness, the exhaust temperature and, therefore, the thermal loss is higher. As shown in figure I-8(a), the overall efficiency is acutally lower, for this case, when reheat is used. For the combustion loop that uses the combustor with exhaust-diluent, the situation is not as bad (as shown in fig. I-9). However, for both cases the use of reheat

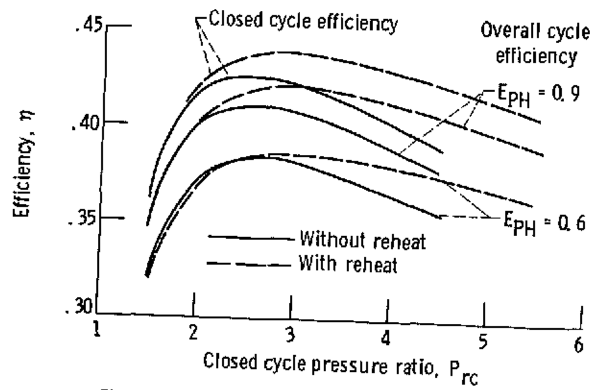


Figure I-9. - Effect of reheat on efficiency for conditions of figure I-6.

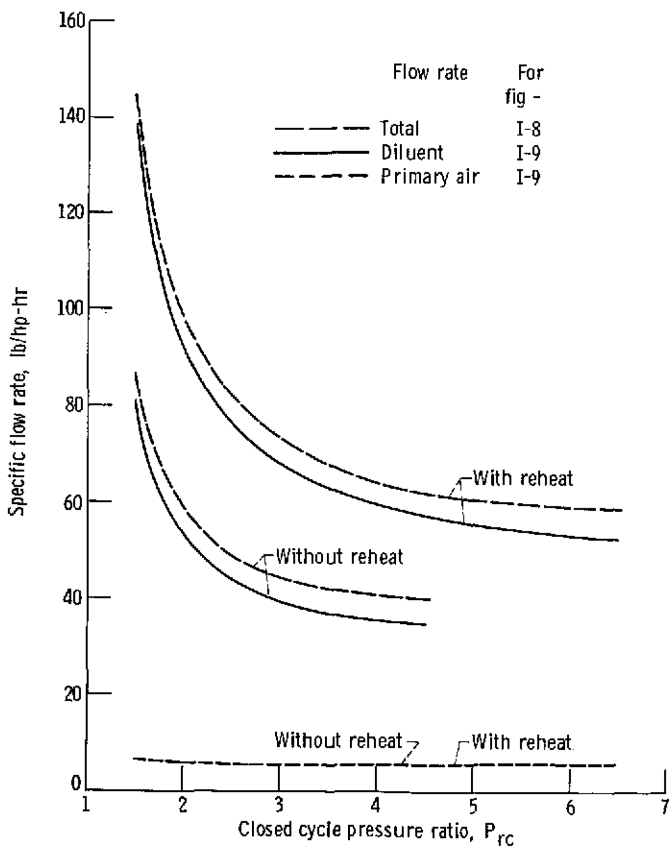


Figure I-10. - Combustion loop specific flow rates for results of figures I-8 and I-9 Heat source heat exchanger effectiveness, 0.9; air preheater effectiveness, 0.9.

results in higher flow rates in the combustion loop as shown in figure I-10. Again, this follows from the fact that with reheat the temperature of the combustion gas at the exit of the heat-source heat exchanger is higher. The temperature of the combustor diluent is then higher, which results in the requirement of higher diluent flow rate.

Remarks

A simplified cycle analysis was performed for several variations in semiclosed and externally fired closed Brayton cycles to obtain a first-order comparison of their cycle efficiencies. The semiclosed cycle efficiency was presented as a combination of a closed portion and an open portion efficiency to facilitate the comparisons.

The semiclosed cycle results show that the overall efficiency is heavily influenced by the efficiency of the closed portion; thermally the semiclosed cycles considered are more than three quarters closed, over a wide range of system pressure levels. As a result the overall efficiency of the semiclosed cycle is close to that achieved by the closed portion and only slightly dependent on system pressure level. If the power level is controlled by changing pressure level, it should therefore be expected that the off-design performance of the semiclosed cycle would also be similar to that of a closed cycle, which is known to maintain near design point efficiencies at partial power. The efficiency of the open portion is not, however, unimportant. This is shown by comparison of the results for the two different cycle configurations. For the second configuration, the efficiency of the open portion is reduced over that of the first configuration considered, and the effect on the overall efficiency is noticeable.

Of the three combustion-loop configurations considered for the externally fired closed Brayton, the one using a conventional combustor with air diluent is least efficient. Use of recirculated exhaust gas as a diluent results in better combustion-loop efficiency, and also reduces the sensitivity of the efficiency to air preheater performance. Combining the combustor and heat source heat exchanger into one unit so that heat transfer to the Brayton working gas controls combustion temperatures results in potentially the most efficient combustion loop for the closed Brayton.

In comparing the semiclosed cycles with the closed Brayton, it was found that the overall cycle efficiencies are comparable for comparable component performance, with only the externally fired closed Brayton using an air diluent controlled combustor having clearly poorer performance.

The open portion of the semiclosed cycle was basically different from an independent open Brayton cycle. The open portion of the semiclosed cycle is capable of producing useful power while serving as the pressure level control and heat source for the closed portion of the cycle. The closed portion in turn serves as combustor diluent for the open portion of the cycle.

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