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CONCEPTUAL DESIGN STUDY OF AN IMPROVED GAS TURBINE (IGT) POWERTRAIN

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FOREWORD

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I. SUMMARY

A study was conducted to establish the "Conceptual Design of an Improved Gas Turbine (IGT) Powertrain" which best meets the following goals:

1. At least a 20% improvement in fuel economy (as a result of powertrain thermal efficiency improvement) relative to a conventionally powered 1976 compact size passenger car, based on the EPA composite driving cycle
2. 1983 production engineering readiness
3. Reliability and life comparable to those of currently marketed powertrains
4. Driveability acceptable to the consumer
5. Meet or exceed Federal Emission Standards of 0.4, 3.4, and 0.4 gm/mi for HC, CO, and NO_x
6. Meet currently legislated noise and safety levels
7. Have competitive initial cost and a life-cycle cost no greater than those of conventionally powered vehicles

Several candidate configurations were initially reviewed in terms of their inherent physical and performance characteristics and their ability to meet the project objectives. From this group three powertrain concepts were selected as candidates worthy of additional investigation. These powertrains consisted of (1) a two-shaft engine with a conventional automatic transmission, (2) a single-shaft engine with a continuously variable transmission (CVT), and (3) a differentially geared arrangement wherein the engine and transmission functions are integrated into a single unit. These candidate powertrain systems were selected to incorporate high-effectiveness rotating regenerators as well as wide-range, variable-geometry aerodynamic components.

The three candidate configurations were evaluated in terms of performance potential, design/installation, and production cost. It was determined that powertrains based on the use of single- and two-shaft engines produced similar fuel economy improvements. Because of its relatively high content of high-speed gearing and high level of transmitted power, the differential powertrain had significantly poorer fuel economy. All three powertrain concepts lent themselves to acceptable engine arrangements, and all could be acceptably installed in the front-wheel-drive (FWD) vehicle which was selected for study because it represented the most difficult installation challenge which a gas turbine powertrain could be expected to meet.

Because of its poor fuel economy, the differential powertrain was dropped from the production cost study. On the other hand, the production cost for the powertrain consisting of a single-shaft engine and a CVT was found to be similar to that for the powertrain consisting of a two-shaft engine and a conventional automatic transmission. Although the single-shaft engine is less expensive, its lower cost was offset by the higher cost of its continuously variable transmission. The powertrain utilizing the two-shaft engine and the conventional transmission had the lowest development risk and cost because it did not require a new transmission. Whereas the powertrain with single-shaft engine and CVT was equivalent (to that using the two-shaft engine) in several categories, it failed to excel in any one category and was

inferior in others. Consequently, the two-shaft engine paired with a conventional automatic transmission was the powertrain recommended for development.

Analysis revealed that an engine with ceramic static gas path parts and metal turbine rotors failed to meet the goal of 20% increase in thermal efficiency and fuel economy. This finding led to the decision that the selected two-shaft engine should include a ceramic gasifier turbine rotor. Additional optimization of the selected two-shaft engine indicated a single-disk regenerator to be best and that the design pressure ratio should be about 4.5:1. The selected engine concept is as follows:

- o Two-shaft configuration
- o Centrifugal compressor/4.5 R_c/variable inlet guide vanes/variable diffuser vanes
- o Variable-geometry radial turbines (gasifier and power)
- o Ceramic gasifier turbine rotor plus associated ceramic static parts
- o 1288°C (2350°F) turbine inlet temperature (maximum power)
- o Ceramic rotating regenerator
- o Idle fuel flow of 1.4 kg/hr (3.0 lb/hr)

A powertrain consisting of a 63 kW (29°C day) (85 hp, 85°F day) size of this engine concept and a three-speed automatic transmission is estimated to yield a 20% improvement—10.0 km/l (23.5 mpg)—in combined cycle fuel economy over a 1976 base-line spark ignition vehicle of 1406 kg (3100 lb) curb weight.

An optimization of transmission modifications (three vs four speed, lockup clutch), gear ratios, and shift patterns was investigated. A powertrain based on the selected two-shaft engine and an optimized four-speed automatic transmission is estimated to improve the base-line vehicle fuel economy by 30% to 10.8 km/l (25.5 mpg).

The "sticker price" differential between the recommended turbine-powered vehicle and a conventional vehicle was evaluated. High-volume production levels were assumed. The estimated turbine vehicle selling price was higher than that of a conventionally powered vehicle; however, the premium was of the same order as for currently marketed fuel-efficient powered vehicles. It was estimated that this price premium could be narrowed after an initial period of production as the result of learning-curve improvements. Life cycle cost was evaluated, and that of the turbine-powered vehicle was found to be slightly better (lower) than that of the conventionally powered vehicle.

The improved gas turbine powertrain was analyzed from an installation standpoint for a transverse-mounted, front-wheel-drive vehicle. The 1980 Pontiac Phoenix, a compact size vehicle, was used as the base line. This vehicle size, 1225 kg (2700 lb) curb weight, and configuration are estimated to be representative of the future; therefore, a powertrain design and development based on that of the 1980 Phoenix is recommended. By 1983, the contract-specified timetable, a vehicle demonstration of significant fuel economy improvement and emissions compliance is feasible. This demonstration would be with an engine containing high-efficiency components, a ceramic regenerator, and ceramic static gas path parts. A 1983 engine dynamometer demonstration with a ceramic turbine rotor was recommended.

The initial portion of the overall development plan should include component development and ceramic turbine rotor development. These activities, along with system and design analysis, should be undertaken as long-lead research and development items.

II. INTRODUCTION

Detroit Diesel Allison (DDA) and Pontiac Motor Division (PMD) of General Motors Corporation have recently completed this "Conceptual Design Study of an Improved Gas Turbine Powertrain" under NASA Contract DEN3-28. This study is a part of the Department of Energy's Highway Vehicle Systems Energy Conservation Program. The objectives of the contract were to conduct a conceptual design study of a gas turbine powertrain for a compact size vehicle (1976 base line) which met the following objectives:

1. A 20% improvement in powertrain thermal efficiency over that of a conventional 1976 powertrain. This shall result in at least a 20% fuel economy improvement over a 1976 model year automotive vehicle over the EPA composite driving cycle. For the study, a compact class 1406 kg (3100 lb) curb weight* vehicle shall serve as the basis for comparison.
2. The powertrains considered should be ready to enter the production engineering phase of development in 1983.
3. Reliability and life features comparable with those of powertrains currently in the market.
4. Driveability acceptable to the consumer.
5. Meet or exceed the Federal Emissions Standards of 0.4 gm/mi, 3.4 gm/mi, 0.40 gm/mi for HC/CO/NO_x.
6. Meet noise and safety levels that are currently legislated.
7. Have competitive initial cost and a life cycle cost no greater than those of conventionally-powered automotive vehicles.

Several current and recently contracted programs directed toward gas turbine-powered passenger cars have indicated, within a range of opinions, that such powertrains offer potential for fuel economy improvement. The Jet Propulsion Laboratory's "Should We Have a New Engine?" is probably the most widely published theoretical analysis recommending, among others, that turbine engine development for passenger cars be initiated. Several private organizations, including General Motors, have conducted gas turbine engine research and development for passenger car application. This background, when supplemented by the emerging technology in ceramic materials, offers the possibility to add a higher operating temperature capability to turbine engines and thereby obtain higher thermal efficiency. It was thus appropriate to conduct a study to determine if a powerplant which offers the higher fuel efficiency so vitally needed by society is feasible.

The conceptual design of such an advanced powertrain requires the expertise of several different disciplines. DDA was the prime contractor, and through its Gas Turbine Engineering organization provided program management and mechanical design and performance, structural, and life cycle cost analyses. The DDA Transmission Engineering organization conducted the transmission design and analysis, and Pontiac Motor Division performed vehicle design and cost studies. Harrison Radiator Division supplied regenerator design data, and several ceramic suppliers provided material property and fabrication data.

*English units of measurements were used for the principal measurements and calculations in this report.

The basic logic-path followed is diagrammed in Figure 1. The engine concepts considered included simple and regenerated cycles, intercooled and augmented versions, fixed- and variable-geometry components, various degrees of ceramic materials utilization, and one-shaft, two-shaft, and differentially geared configurations. Transmission concepts included conventional automatic types (three and four speeds, optional lockup clutch) and continuously variable designs (four-speed countershaft, single-range hydromechanical, Toric drive, variable belt).

It was assumed that, by 1983, ceramic materials development will advance to the point that the strength of silicon carbide, from low-cost processes, will match that currently exhibited by hot-pressed silicon nitride. Levels of turbomachinery efficiency were based on modest improvements to the current state of the art. This assumption matches the program objective of 1983 production engineering readiness.

The concept study consisted of five technical tasks, summarized as follows:

Task I - Powertrain Identification and Description
Identify powertrain concepts, their inherent physical and performance characteristics, and select those worthy of more detailed evaluation.

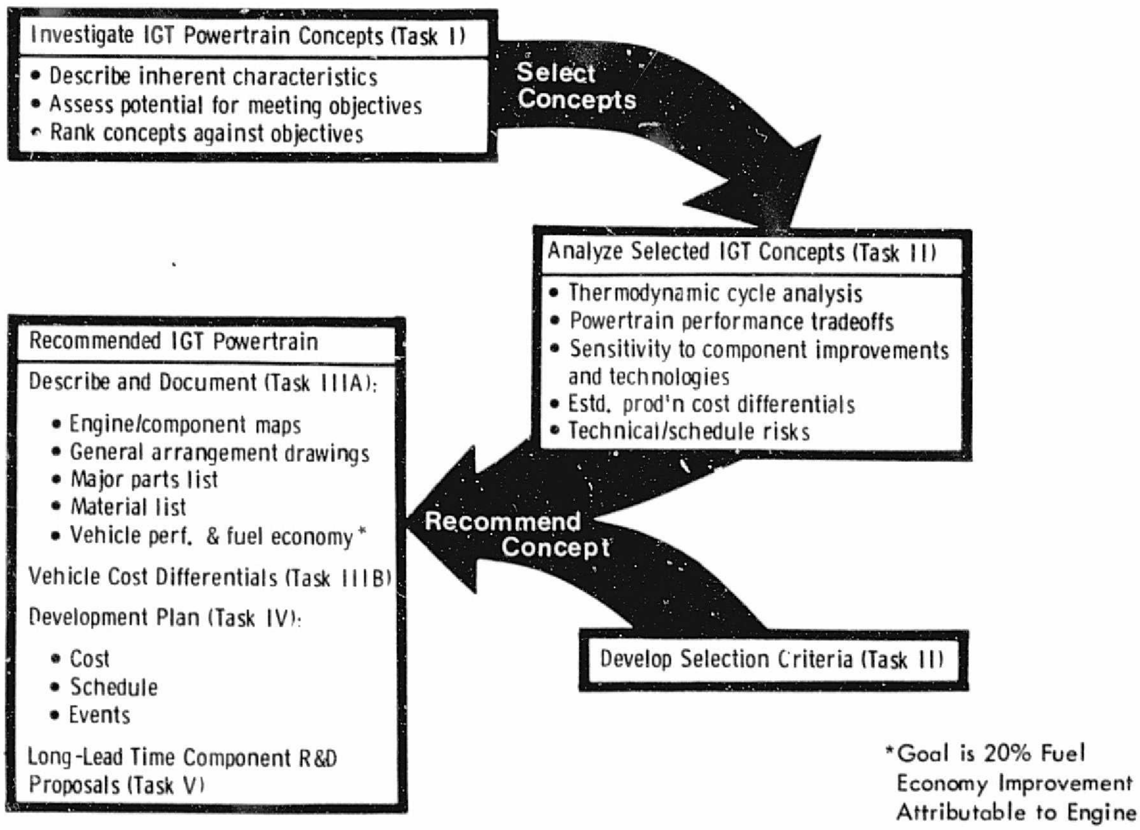

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Figure 1. Improved gas turbine powertrain conceptual design study work flow.

Task II - Powertrain Analysis
: Analyze those concepts chosen in Task I from standpoints of thermodynamic analysis, vehicle performance, engine design, risk, and production cost differences. Recommend a concept for development.

Task IIIA - Powertrain Optimization & Performance
: Optimize and define the components and engine performance of the selected powertrain and conceptually design the turbine powertrain and vehicle.

Task IIIB - Vehicle Cost Analysis
: Prepare an engineering estimate of the sticker price and life cycle cost differences between the turbine-powered and Otto-powered vehicles.

Task IV - Development Plan
: Prepare a plan for development of the recommended turbine powertrain including events, schedule, and cost.

Task V - Long-Lead Research & Development Requirements
: Identify the areas of component, material, and/or subsystem long-lead R&D required to support the development plan.

Sections I and IX of this report contain the SUMMARY and CONCLUSIONS. Sections III through VIII constitute the main text; the following is a brief summary of the contents of each of these sections.

III. BACKGROUND AND ASSUMPTIONS: Relevant computer models are described, base-line vehicles are discussed, temperature limits and component characteristics are presented.

IV. POWERTRAIN IDENTIFICATION AND DESCRIPTION: This section presents the various concepts considered and the rationale for selecting those which are subsequently studied in greater depth. The work reported in this section corresponds to Task I of the contract.

V. POWERTRAIN ANALYSIS: This section, corresponding to Task II, covers the in-depth analyses and evaluations of the candidate powertrain systems. After comparisons from several standpoints, a two-shaft engine with conventional automatic transmission is selected.

VI. POWERTRAIN OPTIMIZATION AND PERFORMANCE: The selected concept is further optimized in this section, which corresponds to Task IIIA. The concept is described in terms of components as well as overall system. Powertrain installation in the vehicle is included.

VII. VEHICLE COST ANALYSIS: The results of production cost (sticker price) and life cycle cost studies are presented in this section. The sticker price study was conducted for the year of initial production by Pontiac Motor Division. Cost reduction potential (resulting from design simplification, material substitution, and learning-curve effects) is also addressed.

VIII. DEVELOPMENT PLAN: A development plan was prepared (and submitted separately, as required by the contract Task IV). A summary of the plan, including long-lead research and development activity (contract Task V), is presented in this section.

III. BACKGROUND AND ASSUMPTIONS

This section is a discussion of the various assumptions and background data used for the study efforts of this report. Included are the primary computer models used for the engine, transmission, and vehicle characteristics and details of the base-line vehicles, ambient conditions, fuel characteristics, vehicle response and engine sizing criteria, engine thermal limits, and mechanical and aerodynamic component characteristics.

PRIMARY COMPUTER MODELS

The computer models used to calculate engine transmission and power train performance have been in use by DDA for several years and are continually updated to meet present-day needs.

Engine Performance Models

DDA computer performance programs are designed on a building-block system concept and consist of a calculation-controlling routine which links a system of generalized component subroutines into any desired type of engine arrangement (Figure 2). Inputs to the program include component arrangement in terms of compressor(s), combustor, turbine(s), and heat exchanger(s), pressure loss, reheat, transmissions, accessories, cycle losses, and other characteristics. Additional specialized subroutines can be easily and efficiently included. This approach allows maximum flexibility in studying a variety of cycles and uses the same program with the cycle defined by input data. The system also features rapid cycle matching procedures and direct transfer from design point to off-design calculation mode.

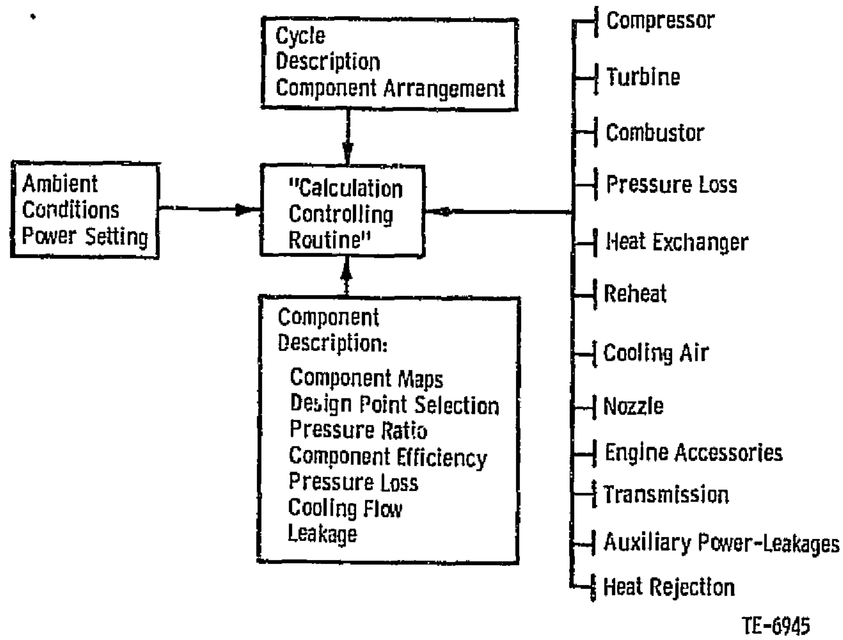

Figure 2. Partial list of generalized subroutines.

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Upon selection of the design point for each cycle to be examined, the necessary component data are defined. Appropriate information is developed for use in the computer to relate how the individual component performance is affected by variation in flow, pressure, and temperature. In the case of the compressor, these data take the form of conventional maps of corrected mass flow versus pressure ratio with lines of speed and efficiency. Expansion ratio versus flow and efficiency with lines of speed is used for the turbine. In particular cases where a component operates over a narrow range and/or where limiting the range of operation is desirable, a single-line map can be used.

The performance analysis program has the capability to scale component maps in terms of either constant or variable scale factors on any of the map parameters. For example, a given compressor map could be scaled in terms of flow, pressure ratio, or efficiency to match some design change for that compressor or to simulate the effect of variable geometry. This degree of flexibility provides for the rapid examination of variations to components and was particularly useful in applying one set of component characteristics to meet a range of horsepower sizes.

Heat exchanger performance is calculated by using the fundamental heat transfer characteristics of the surface being considered and solving the heat transfer and pressure loss equations in the performance program.

Combustor pressure drop and efficiency at off-design power are calculated as a function of both the combustor design and the flow conditions in and out of the combustor.

Outputs from the computer program include the two basic performance parameters of shaft horsepower and specific fuel consumption along with all flows, temperatures, and pressures at each station within the cycle.

A time history of engine performance during fuel rate, variable geometry, and gasifier rotor transients is rapidly accomplished by interfacing additional dynamic routines with the steady-state simulation. These additional routines perform the control of time functions, rotor dynamics, sensor lags, and heat storage effects. Thus, the dynamic simulation is achieved with little change to the steady-state model and makes maximum use of component characteristics prepared for the steady-state analysis.

Vehicle Models

The GPSIM (General Purpose Simulation) method is the most thorough and accurate analytic tool available within General Motors for evaluating the performance and fuel economy of vehicle systems. However, it requires extensive powertrain data and is relatively expensive to operate. Consequently, it is not practical to employ GPSIM each time that an estimate of driving cycle fuel economy is required. Hence, other modeling techniques were developed which traded accuracy for reduced preparation and operating times. These other methods are valid and are appropriate for use in many situations. In fact, constant use of GPSIM would be inappropriate.

The general areas of use for each method are as follows. GPSIM was used when accurate, absolute levels of driving cycle fuel economy were required. Examples are the base-line spark ignition vehicle and the final recommended gas turbine-powered vehicle. The simplified driving cycle, while less accurate on an absolute MPG basis, is felt to be accurate for quantifying differences. Thus, it was frequently used in the process of concept selection, especially when the road-load fuel economy of competing concepts was relatively close. Road-load fuel economy is not only quickly calculated but also lends itself to a graphical comparison of two systems. When clear and unmistakable trends are present in road-load fuel economy, selection decisions can be confidently made without the need for further analysis.

GPSIM (General Purpose Simulation)

GPSIM is a computer program developed and used by General Motors for simulating automotive vehicle performance and fuel economy. The program computes the operating conditions for the engine and driveline and the performance and fuel economy of the vehicle as it is operated in a prescribed manner.

The program can simulate a wide variety of vehicles which are described with combinations of steady-state test results and vehicle specifications. Piston engines, gas turbines, and electric powerplants can be used. Their power, fuel, and speed characteristics are input as data tables. A general purpose transmission description allows an arbitrary number of gear ratios, each using a torque converter, clutch, or other special device. Operating routes and driving cycles and schedules to be simulated are described with data tables. Cold starts cannot be simulated with GPSIM for spark-ignition or gas turbine engines. However, good agreement is achieved between GPSIM predictions and EPA driving cycle test results for spark-ignition vehicles. Therefore, neither engine type was corrected for cold starts.

A wide variety of engine accessory loads and/or driveline losses can also be specified. The simulator can make predictions from limited data or make more accurate computations by using more extensive and accurate data.

Besides standard output, additional variables can be selected from a large list and can also be plotted. Distribution statistics can also be generated for such variables.

Simplified Driving Cycle

The quantity of input data required to simulate a given powertrain/chassis combination is significant and prohibits a full simulation of each candidate powertrain. Therefore, a means to simplify the simulation without a significant loss in accuracy was investigated. Part of the output data from GPSIM consists of accumulations of time spent within various velocity-power cells during the federal urban and highway driving cycles. The separate urban and highway driving cycles were then combined. The urban driving cycle was scaled on the basis of the ratio of the distances traveled between the urban cycle and the highway cycle and added to the highway cycle to obtain the combined driving cycle shown in Table I. Analysis of the time spent at different vehicle speed intervals and engine power intervals indicated that the combined driving cycle could be approximated by four categories: idle, road load, acceleration, and deceleration. By time weighting these categories and subdividing the road load category into three further categories, the combined urban and highway cycle can be simulated by six combinations of horsepower required and vehicle speed. The fuel economy calculated by this simplified method was compared with the more accurate GPSIM results. The results of several checks that included gas turbine engines and conventional automatic transmissions indicated that both methods agreed within 2%.

TABLE I. PERCENT TIME DISTRIBUTION IN VELOCITY AND POWER CELLS ON COMBINED DRIVING CYCLE

Engine power, kW (shp)	Vehicle velocity, km/hr (mph) Idle	0	16.1 (10)	32.2 (20)	48.3 (30)	64.4 (40)	80.5 (50)	96.6 (60)
-3.8(-5)								
			0.1	0.14	0.92	0.66		
0								
		4.36	6.22	6.31	0.62	0.07	0.05	
3.8 (5)								
	13.85	2.28	0.33	11.25	1.38	0.29	0.42	
7.5(10)								
		1.39	0.91	4.56	3.52	0.41	0.47	
11.5(15)								
		0.84	1.13	1.73	0.20	1.20	0.93	
15.0(20)								
		0.39	2.11	2.82	0.62	5.58	7.52	
22.5(30)								
			0.96	0.37	0.68	1.87	6.39	
30.0(40)								
			0.26	0.35	0.01	0.81	1.61	
37.5(50)								
				0.26	0.20	0.02	0.22	
45.0(60)								
				0.07	0.24	0.03		
52.5(70)								

The six points were then adjusted from engine output horsepower to the basis of transmission output power. This model adjustment allowed the application of the simplified method to other transmission types. The adjusted six points comprising the simplified driving cycle are presented in Table II.

Road-Load Fuel Economy

Use of the GPSIM-modeled base-line vehicle allowed definition of road-load power requirements—i.e., the power required at steady-state velocities. Transmission losses must be added to the vehicle requirement to obtain the transmission input power requirement. The engine power required is then determined by adding the accessory load to the transmission input power. Road-load fuel economy is the result of the correspondence of engine fuel flow to the total power required for the given vehicle velocity. A road-load fuel economy curve (mpg vs mph) characterizes the vehicle at constant-velocity operation. Although such a curve does not indicate driving cycle fuel economy (because accelerations, decelerations, and idle are excluded), it is a useful measure of a given vehicle system. Vehicle systems which show a significant and/or consistent trend in terms of road-load mpg can be expected to show similar trends over the complete driving cycle.

TABLE II. SIMPLIFIED DRIVING CYCLE USED FOR FUEL ECONOMY PREDICTIONS

Cell	1	2	3	4	5	6
Condition	Idle	Road load	Raod load	Road load	Accel	Decel
Velocity, km/h (mph)	0	16.1 (10)	43.4 (27)	82.1 (51)	38.1 (24)	72.4 (45)
Transmission output power, kw (hp)	0 (0)	0.97 (1.3)	4.2 (5.6)	17.3 (23.2)	15.0 (20.1)	2.5 (3.3)
% time	14	13	27	23	18	5

Transmission Models

DDA has developed a series of computer programs to predict the performance of conventional automatic torque converter transmissions and continuously variable transmissions (CVT).

Hydromechanical System--CVT

This program model is used to describe the performance characteristics of hydrostatic or hydromechanical CVTs. Performance of the hydrostatic pump and motor variable drive mechanism as a function of speed, pressure, and displacement is formulated from test data of this type of drive used in military and commercial transmission development programs since 1955 and stored in a data library. For each new CVT arrangement, input data to the program include the hydrostatic size (displacement), input and output gearing and clutch losses as a function of speed, gearing efficiency, and modes of range operation (i.e., displacement relationship of pump and motor through the mechanical speed range). The output data from this program describe the transmission output performance in terms of system pressure, output torque, speed and horsepower, and overall efficiency at each pump/motor displacement of varying input power increments.

Variable-Ratio Steel Belt System--CVT

Performance of this type of drive was obtained from available test data.

Predicted Equipment Performance (PREP)

This is a DDA-developed computer program used to predict the performance and efficiency of conventional automatic torque converter-type transmission applications. The program has an extensive library of engine, torque converter, transmission, and vehicle data but new data (i.e., gas turbine engine performance data) can be introduced as an input in terms of output torque and speed characteristics. This program was used to develop the final performance characteristics of the selected powertrain.

Vehicle Acceleration

This computer program is used to develop vehicle acceleration characteristics of engine/transmission/vehicle combinations. The typical input data include:

- Engine power characteristics
  - Engine speed vs engine torque
- Vehicle weight
- Differential ratio
- Final drive ratio
- Tire rolling radius
- Rolling resistance
- Frontal area
- Driveline efficiency
- Traction coefficient
- Driveline inertia
- Wind resistance coefficient
- Transmission characteristics
  - Complete performance map (output torque vs output speed developed from one of the transmission performance programs) from idle to full throttle at various power levels

Standing start as well as rolling accelerations are calculated. Each acceleration is made in a series of discrete time steps which can be adjusted from a nominal value of 0.1 second to a shorter or longer time interval. At the beginning of each time interval, the transmission speed/torque characteristics are matched to those of the engine. That power which cannot be absorbed by the transmission serves to accelerate the engine. (In the case of the variable-drive transmission, the power distribution for engine acceleration and vehicle acceleration is programmed to meet a constant initial acceleration rate of engine rpm per second from idle to full throttle.) The output data at each time increment include:

- Vehicle speed (mph)
- Cumulative distance travelled
- Engine speed
- Transmission torques in and out
- Transmission horsepower in and out
- Vehicle and engine acceleration rates

BASE-LINE VEHICLES

The contract for this concept study specifices a 1976 vehicle to serve as the base line for the various elements of the study--namely, vehicle fuel economy and performance, engine design and installation (into vehicle), and cost (both acquisition and life cycle). As the concept study was executed, it became apparent that the engine design and installation studies should be structured around a vehicle that would represent a good candidate for production with a gas turbine powertrain in the 1980 to 1990 time frame. It was determined that the overall contract objectives of this study could be satisfied by utilizing two base-line vehicles. A 1976 General Motors compact vehicle was used for all performance and fuel economy studies. The mechanical design, installation, and cost studies used the 1980 General Motors X-body compact vehicle. Each of these is explained in the following paragraphs.

Performance and Fuel Economy Studies

Although this concept study has several important objectives, all of which must be met by a viable turbine-engine powertrain, fuel economy or thermal efficiency is clearly the most dominant. In addition, the base line of comparison is significant in absolute terms because factors can reasonably be expected to change with time, including the Otto cycle vehicle itself. Consequently, the contract vehicle specification was used as the fuel economy base line: 1406 kg (3100 lb) curb weight, 1545 kg (3400 lb) test weight, 8.3 km/l (19.6 mpg) using gasoline, three-speed automatic transmission, and a wide-open-throttle (WOT) acceleration capability of 0-96.6 km/hr (0-60 mph) in 15 seconds.

Table III is a listing of the contractual base-line vehicle specifications and characteristics. Because the contract specification represents an "average" 1976 compact vehicle, no manufacturer's offering would exactly match the specification. In General Motors' case, the X-body car (Citation, Phoenix, Omega, Skylark) closely approximates the specification and thus served as the starting point in arriving at an analytical vehicle matching the specification. Following minor iterations, a GPSIM computer model of the base-line vehicle was defined which acceptably matched the contract fuel economy and performance specifications. This model served as the economy and performance base line for all subsequent evaluations of candidate turbine engine powertrains.

The GPSIM data bank has data available for a number of vehicle, engine, and driveline characteristics which have been verified by various test procedures. The 1976 model year Chevrolet Nova class compact vehicle powered by an inline six-cylinder engine was chosen from the GPSIM data bank as most nearly characteristic of the nominal vehicle specified in the contract. This vehicle and drive train were then modified to correspond to the base-line vehicle. The base-line piston engine used is a 4.1 litre (250 CID), 1976 production Chevrolet inline six-cylinder engine which was then scaled to 3.6 litres (220 CID) to provide the specified 8.3 km/l (19.6 mpg) and 0-96.6 km/hr (0-60 mph) acceleration time of approximately 15.0 sec. Table IV lists the vehicle specifications used for GPSIM performance and economy modeling. C1, C2, and C3 are coefficients in the tractive effort equation,

$$\text{Tractive effort } (lb_f) = C1 + C3\ (mph) \times (\text{veh mass}) + C2\ (FA)\ (mph)^2$$

which GPSIM uses in the calculation of power requirement. Performance of the base-line car, with the engine sized for 8.3 km/l (19.6 mpg), was calculated:

TABLE III. CONTRACTUAL BASE-LINE VEHICLE SPECIFICATIONS	
Vehicle class	Compact
Curb weight	1406 kg (3100 lb)
Test weight	1542 kg (3400 lb)
Frontal area	9.75 m^2 (21.5 ft^2)
Wheel base	2.78 m (109.5 in.)
Axle Ratio	2.70:1
Tires	ER78-14
HP/WT ratio (reference)	0.03
Transmission	3-speed automatic
1st ratio	2.5 to 1
2nd ratio	1.5 to 1
3rd ratio	1.0 to 1
0-96.6 km/hr (0-60 mph) accel time	15.0 sec
Minimum range	483 km (300 mi)
1976 fuel economy (combined Federal Driving Cycle)	8.33 km/l (19.6 mpg)
Roominess index	6.93 m (273 in.)
Hip room	1.4 m (57 in.)
Shoulder room	1.4 m (55 in.)
Trunk space	0.45 m^3 (16.0 ft^3)

TABLE IV. BASE-LINE VEHICLE SPECIFICATIONS USED FOR GPSIM MODELING	
Vehicle class	Compact
Test weight	1542 kg (3400 lb)
Frontal area (FA)	2.09 m^2 (22.5 ft^2)
Wheel base	2.78 m (109.5 in.)
Axle ratio	2.70:1
Tires	ER78-14
Transmission	3-speed automatic THM 350
1st ratio	2.52:1
2nd ratio	1.52:1
3rd ratio	1.00:1
Tire efficiency	98%
C1, tractive effort coefficient	0.01306
C2, tractive effort coefficient	0.00134
C3, tractive effort coefficient	0.0

0-96.6 km/hr (0-60 mph) acceleration	15.5 sec
Distance traveled at 1.0 sec	1.8 m (5.9 ft)
Distance traveled at 96.6 km/hr (60 mph) point	259 m (850 ft)
48-114 km/hr (30-70 mph) pass--time	16.9 sec
—distance	402 m (1320 ft)

Design and Installation Studies

A very significant portion of this study was devoted to defining a logical engine general arrangement which was consistent with the vehicle design. The contract specification base-line vehicle was a 1976 compact type. Because such a vehicle design will be obsolete by the time that even a prototype improved turbine powertrain exists, basing the engine design on such constraints would have been counterproductive.

The likely configurations of compact vehicles for the 1980's were reviewed to determine the real constraints that would be imposed on an improved turbine engine powertrain. This review revealed that while conventional rear-wheel-drive (RWD) vehicles may exist (especially in larger body sizes), an equal if not greater probability exists that front-wheel-drive (FWD) vehicles will be produced. Accordingly, the RWD and FWD vehicle types were compared with a view to designing and installating turbine engines. The transverse-mounted FWD vehicle proved to be the greater challenge. Furthermore, an FWD engine arrangement was judged to be usable in an RWD vehicle, although the reverse was not necessarily true. Consequently, it was decided that the improved turbine engine concept design should be made to be consistent with an advanced FWD compact vehicle. Such an engine design could obviously be utilized in an FWD vehicle. It could also be easily adapted to RWD vehicles if these should become future candidates.

Of necessity, the transmission for the improved turbine vehicle was also selected to be of the FWD configuration. In the case of RWD vehicles, it was assumed that current automatic transmissions could be used with two-shaft engines. Other engine types (such as single-shaft) in RWD vehicles would require a new transmission.

Cost Studies

The cost estimation phase of this study was based on an FWD compact vehicle. It was assumed that the most likely gas turbine-powered compact vehicle would be an FWD configuration, not the 1976 RWD type. Therefore, the cost base line was logically and necessarily an FWD spark-ignition-powered vehicle. To evaluate a 1976 Otto-powered RWD vehicle with a turbine-powered FWD vehicle would have been a meaningless comparison because the vehicle differences would not have represented the changes required by the use of a gas turbine powertrain. The acquisition as well as the life-cycle cost studies of spark-ignition and gas turbine-powered vehicles were conducted for FWD compact vehicles. In both cases, the dollar differences were determined in terms of 1977 economics.

AMBIENT CONDITIONS AND FUEL USED

The contract required that all fuel economy and vehicle performance calculations and comparisons be made at 29°C (85°F), 152 m (500 ft) ambient conditions, using gasoline. The gas turbine-powered vehicle was thus designed to match the spark-ignition-powered vehicle performance at these conditions. Although fuel economy comparisons are made with gasoline, the life cycle cost and emissions studies assume the use of Diesel fuel No. 2 since it is the most likely fuel that would be used in a gas turbine-powered vehicle. (Refer to Appendix B.)

VEHICLE RESPONSE CRITERIA

An inverse relationship exists between vehicle performance and fuel economy; thus, a given value of fuel economy can be judged only if the vehicle's performance is also known. The contract base-line vehicle specification recognized this by requiring a 0-96.6 km/hr (0-60 mph) wide-open throttle acceleration time of 15 seconds. This represented the minimum vehicle response; a vehicle manufacturer could add other performance requirements if needed to achieve "acceptable driveability."

Driveability really means driver acceptance and, therefore, embraces the wide range of individual preferences in relation to many factors (maximum speeds, hill climbing, trailer pulling, etc). Public acceptance will never be known with certainty until large numbers of vehicles have been evaluated by a cross section of drivers. This fact represents risk—risk that engineering analysis might fail to consider performance factors important to the general public. To minimize this risk, the following additional requirements were placed upon the gas turbine-powered vehicle studied under this contract.

- Initial response. Otto-powered vehicles traditionally offer rapid vehicle response to throttle depression. This presents a potential problem to the turbine engine by virtue of its acceleration "lag" (the time required to accelerate the engine from idle to 100% rpm). The typical driver will probably be insensitive to engine acceleration, but he will be sensitive to the vehicle response. Examination of several acceptable Otto-powered vehicles revealed two facts: (1) significant motion occurs within the first second of a wide-open throttle acceleration and (2) considerable variation exists in the distance traveled in the first second. An engineering judgment was made that the gas turbine-powered vehicle must move a perceptible distance in the first second. However, a specific distance could not be named as a minimum acceptable target for the gas turbine-powered vehicle.

- Relative distance. The initial response constraint ensures that the turbine vehicle "starts" an acceleration maneuver; the 0-96.6 km/hr (0-60 mph) time constraint that it completes the maneuver. But what about the "getting there"? Conceivably, a turbine vehicle could accelerate slowly during the first few seconds (and fall well behind) but catch up in the last half of an acceleration maneuver. Such a characteristic would be annoying and uncomfortable (varying acceleration levels) and open a gap behind the front vehicle into which a rear vehicle could move (i.e., the turbine vehicle would be vulnerable to being passed). All of these concerns can be eliminated by requiring a turbine car to maintain a specified relative distance to adjacent Otto vehicles during the acceleration. To demand foot-by-foot distance matching is excessive. However, the turbine vehicle can reasonably be required to lose no more than a partial car length of its relative position throughout the overall acceleration.

- High-speed passing. In addition to acceleration from a standing start, a turbine vehicle must acceptably accelerate from a cruising position. This case typically involves only two vehicles, one passing the other. Therefore, the overall time (exposure) required to complete the pass is a sufficient constraint. The gas turbine-powered vehicle should require no greater passing time for a 48-114 km/hr (30-70 mph) pass than the base-line spark-ignition-powered vehicle.

ENGINE SIZING CRITERIA

Previous studies have produced a variety of results concerning the required maximum power of a turbine engine. These have ranged from considerably smaller than an Otto-cycle engine, according to JPL (Ref 1), to one of the same power as indicated by General Motors from their two-shaft engine experience (Ref 2), to a larger size, as estimated by General Motors for a single-shaft engine (Ref 2). Detailed engine and transmission designs, followed by extensive computer modeling and systems analysis, are required to determine the necessary engine size.

Such depth of analysis was beyond the contract scope during the concept selection phases (Tasks I and II). For these phases it was assumed that a gas turbine engine sized for 63 kW (85 hp) at 29°C (85°F), 152 m (500 ft) ambient conditions would provide adequate power for the vehicle. This assumption, which is consistent with the results of previous DDA studies, was judged not to introduce bias into the engine selection process. Indeed, to assume that engine size varied (with powertrain type) without benefit of supporting analysis could introduce bias. In order to be consistent with DDA analysis techniques, the actual "design point" of the engines is 75 kW (100 hp) for 15°C (59°F), sea level conditions. This results in 63 kW (85 hp) at the required ambient conditions.

During optimization of the selected powertrain concept (Task IIIA), computer model studies were conducted to determine the exact engine power rating required to match the previously discussed vehicle performance criteria. The results of this detailed analysis verified that the Task I and II assumption were very reasonable.

THERMAL LIMITS AND MECHANICAL CHARACTERISTICS

Several engine limits and characteristics were required to permit the evaluation of engine, powertrain, and vehicle performance for the variety of engine and powertrain types included in the study. These involved gas temperature structural limits for the turbine stages, determination of cooling air requirements for the turbine wheels, and the evaluation of rotor polar mass-moments of inertia. The results of these evaluations were incorporated in the computer performance models of the gas turbine engines as limits and characteristics.

Turbine Stage Temperature Limits

In order to perform a responsible assessment and comparison of engine/vehicle performance characteristics, thermal/structural turbine limits were required for each of the engine types to ensure that all of the study engines were configured with the highest possible cycle temperature which still provided adequate life and reasonable structural risk. Ceramic, air-cooled metal blades, and solid metal rotors were initially considered as candidates for these applications.

Absolute temperature limits for turbines with ceramic components were not established. Available data indicate that several ceramic materials are feasible at turbine temperatures to 1288°C (2350°F). At temperatures greater than 1371°C (2500°F), only one material—sintered alpha SiC—

appears feasible. Feasibility demonstrations and design studies from other programs indicate that current and evolving ceramics technology will permit the use of ceramic static parts with confidence in the time frame of the development program. However, the availability of a cost-effective ceramic rotor is much less certain and must, therefore, be viewed as a development effort to be conducted concurrently with the engine development. (The ceramic rotor feasibility is discussed in detail in another section of this report.)

Also, the small mass flow and physical size of the turbomachinery necessarily eliminates the possibility of using air-cooled metal turbine rotors as a cost-effective approach to higher engine cycle temperature.

Therefore, it was necessary to evaluate appropriate technology limits for metal turbine rotor systems with uncooled blading. To cover the variety of engine turbine types, this was done for axial, radial inflow, and reentry (axial) gasifier turbines. Limits were also evaluated for axial and radial metal power turbines for engines which might incorporate a ceramic gasifier turbine rotor.

To avoid an extensive mission cumulative damage analysis, several simplifying assumptions were made:

1. A 100-hour life is required at the engine maximum rating conditions, using minus-three-sigma material strengths. (All engine cruise conditions will be at 37.8°C (100°F) lower turbine inlet temperatures than the engine rating conditions.)
2. Turbine wheel and blade material will be Mar-M246. (Mar-M246 was chosen for this study evaluation because it is an available high-strength alloy with good castability and good coated oxidation resistance and has well defined strength/life characteristics.)
3. DDA engine experience will be used in determining the criteria limits.

The interrelationship of AN^2 (blade annulus area times the square of rotor speed as defined in Figure 3) and T_{Trel} (blade total relative gas temperature) has long been used at DDA as a parameter to reliably indicate the degree of difficulty and life sensitivity for axial turbine mechanical design. Therefore, this approach was adopted for the determination of axial turbine technology limits. A diagram showing the limiting relationships for these parameters as selected for the axial turbines is shown in Figure 3. Line A on the diagram imposes a structural limit on blade hub stress which provides a margin for vibratory stress. Line B represents the limiting blade stress rupture life envelope. Line C is an oxidation limit. Similar limits were applied to define limits for the radial inflow metal turbines. In the case of the radial turbines, the same T_{Trel} vs AN^2 criterion defined for the metal axial turbines was applied at the radial turbine rotor exit location (ref Figure 3). In addition, a limit criterion relating radial turbine inlet tip speed (V_T) and inlet gas total relative temperature was imposed. This relationship is presented in Figure 4.

For reentry turbines, the criterion of Figure 3 was used wherein the T_{Trel} was evaluated on time-temperature average basis for one revolution. That is, the residence time in each of the temperature zones is so brief that the blades will experience only an average gas temperature.

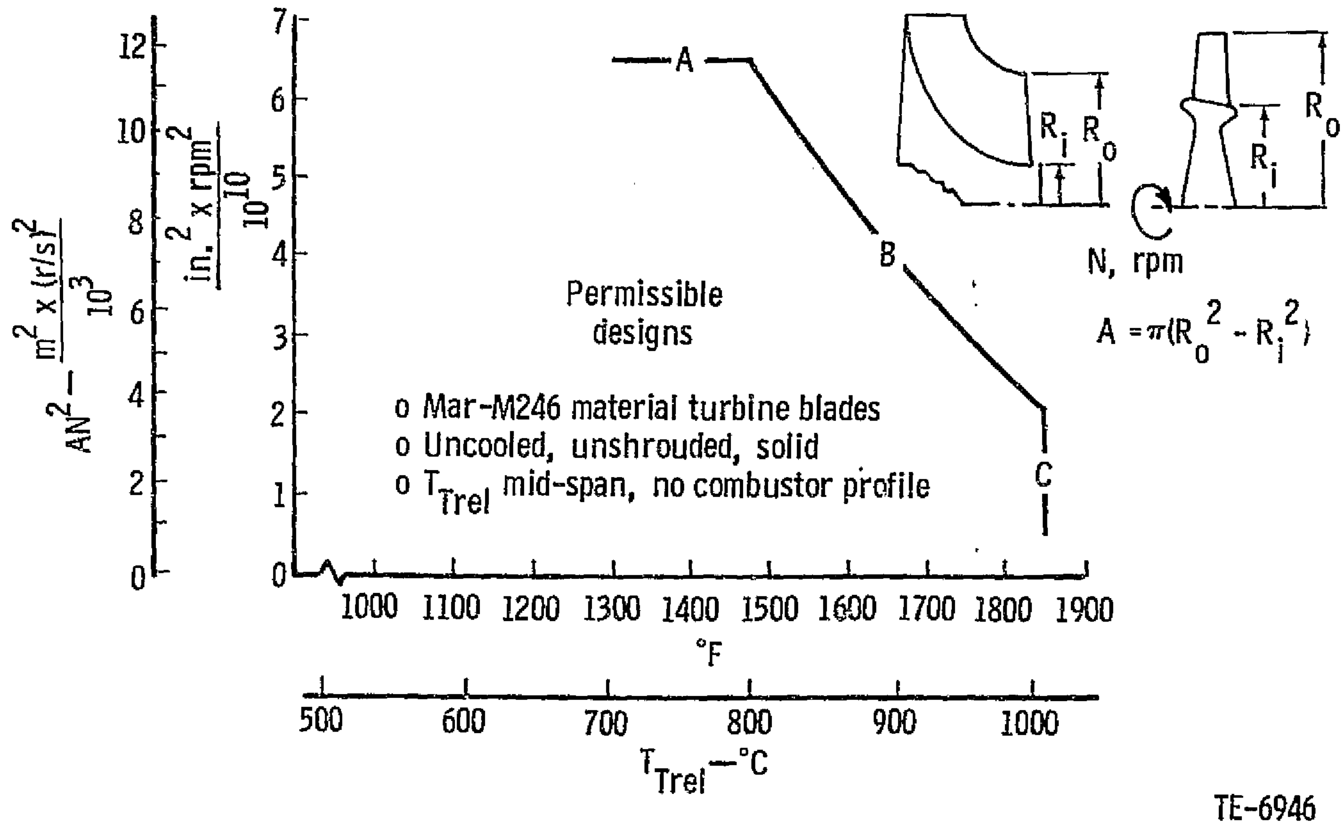

Figure 3. Design limits for axial and radial turbines.

The criteria of Figures 3 and 4 were used in conjunction with typical small turbine AN^2, T_{Trel}, and TIT (turbine inlet average gas temperature) to define limiting temperature levels for metal turbine rotors with uncooled blades for each of the study engine types as applicable. These limits, although different for the various engines, will result in the same turbine rotor life and are presented in Table V.

Turbine Wheel Cooling

The engine cycle analysis activities for the study engines required a determination of cooling flows for the turbine wheels and seals.

To facilitate this determination, without detailed analysis of the variety of engines and turbines in the study matrix, a characteristic was defined as a function of turbine interstage total temperature, using data from several DDA small gas turbine engines. The resulting characteristic for use in the study engines is shown in Figure 5. This characteristic is useful for both axial and radial inflow turbines except that to determine cooling values for the reentry turbine cavities the values obtained for "first-pass" conditions were doubled.

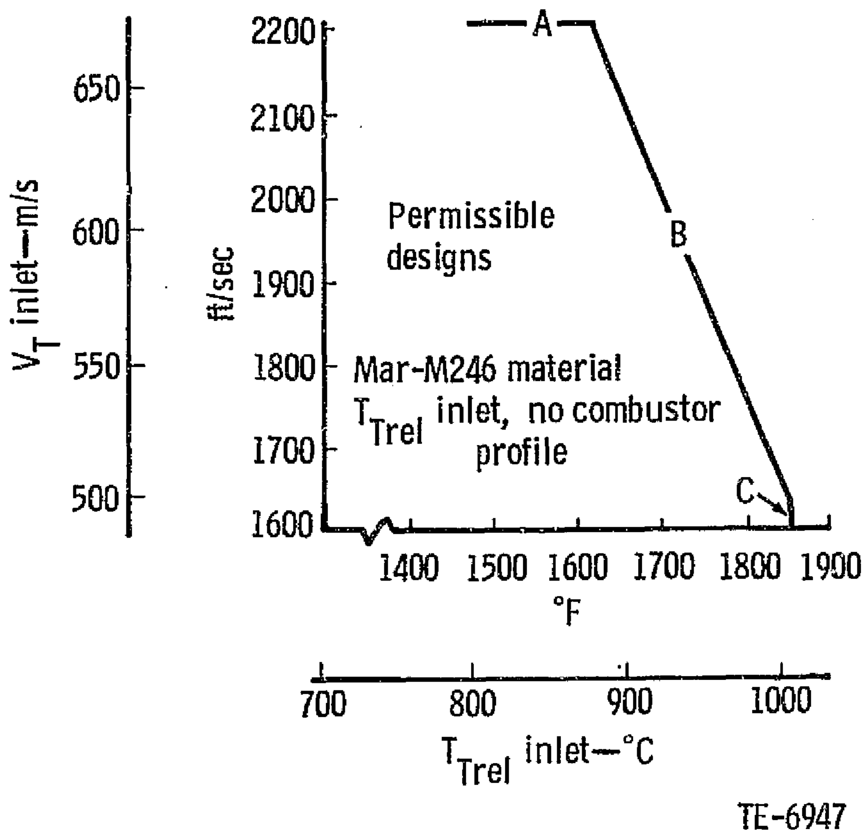

Figure 4. Design limits for radial turbines.

TABLE V. IGT ENGINE TURBINE INLET TEMPERATURE LIMITS WITH METAL ROTORS

	R_c*	TIT** °C	TIT** °F
<u>Two-Shaft Engine</u>			
Axial Gasifier Turbine	4.5	1068	1955
Radial Gasifier Turbine	4.5	1080	1976
Axial Power Turbine	4.5	1054	1930
Radial Power Turbine	4.5	1066	1951
<u>Differential or Single-Shaft Engine</u>			
Reentry Turbine	4.5	1195	2183
<u>Single-Shaft Engine</u>			
Axial Turbine	4.5	1049	1920
Radial Turbine	4.5	1049	1920
Radial Turbine	4.0	1063	1945

*Engine Cycle Pressure Ratio
**Average Turbine Inlet Temperature at Maximum Power Conditions

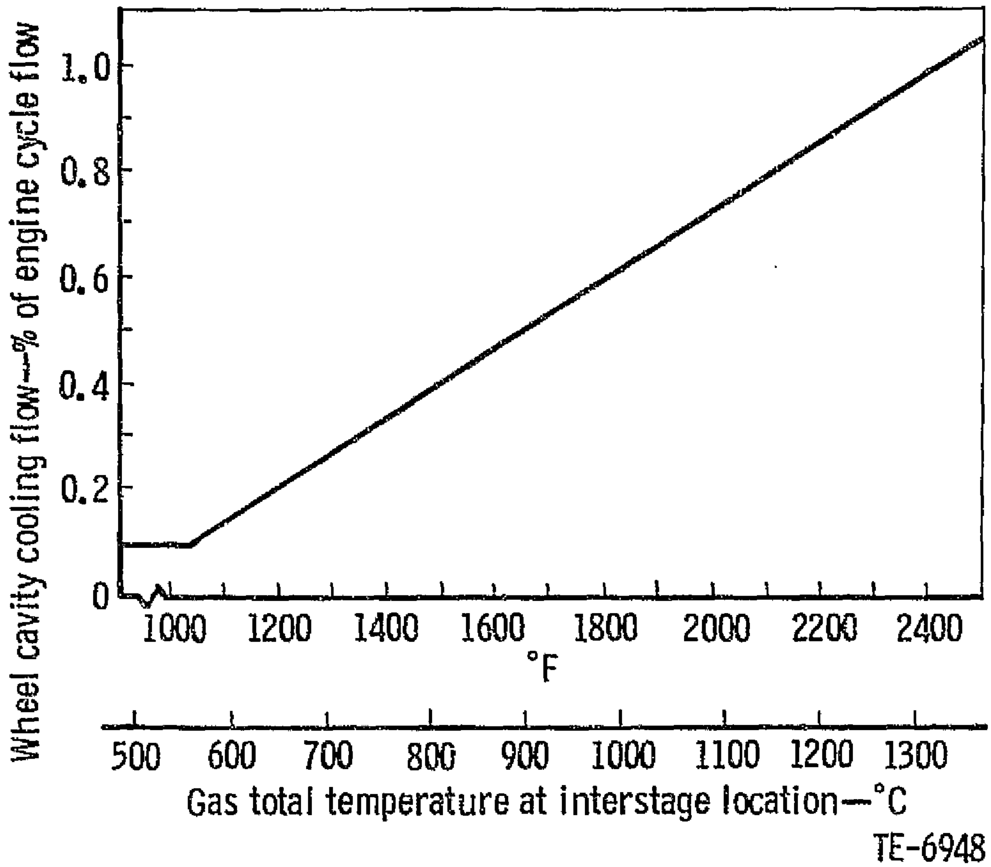

Figure 5. Turbine wheel cavity flow for cooling and seal leakage.

Rotor System Inertia

Inertia characteristics of engine rotor system components were evaluated from previous DDA small components to provide values for use in the study engine performance evaluations. All of the components are presented in Figure 6 as a function of engine inlet corrected airflow to simplify the determination of rotor total inertias. The compressor characteristic shown is for aluminum rotors with pressure ratios of 4.0 to 4.5. The turbine rotor relationships presented are for Mar-M246 material. Ceramic turbine rotors were assumed to have inertias equal to 1/3 of the Mar-M246 rotors. The scaling of rotor inertia versus engine airflow size was performed by assuming photographic scaling of the rotors. That is, axial and radial dimensions were proportional to the square root of design airflow.

AERODYNAMIC COMPONENT CHARACTERISTICS

Compressor Performance Characteristics

The compressor characteristics of the fixed- and variable-geometry compressors utilized in the study engine performance modeling have DDA compressor rig testing as their basis. The fixed-geometry compressor characteristics were scaled from the DDA 505-IIIA compressor. This compressor, which at design point delivers 1.92 kg/s (4.23 lb/sec) airflow at a pressure

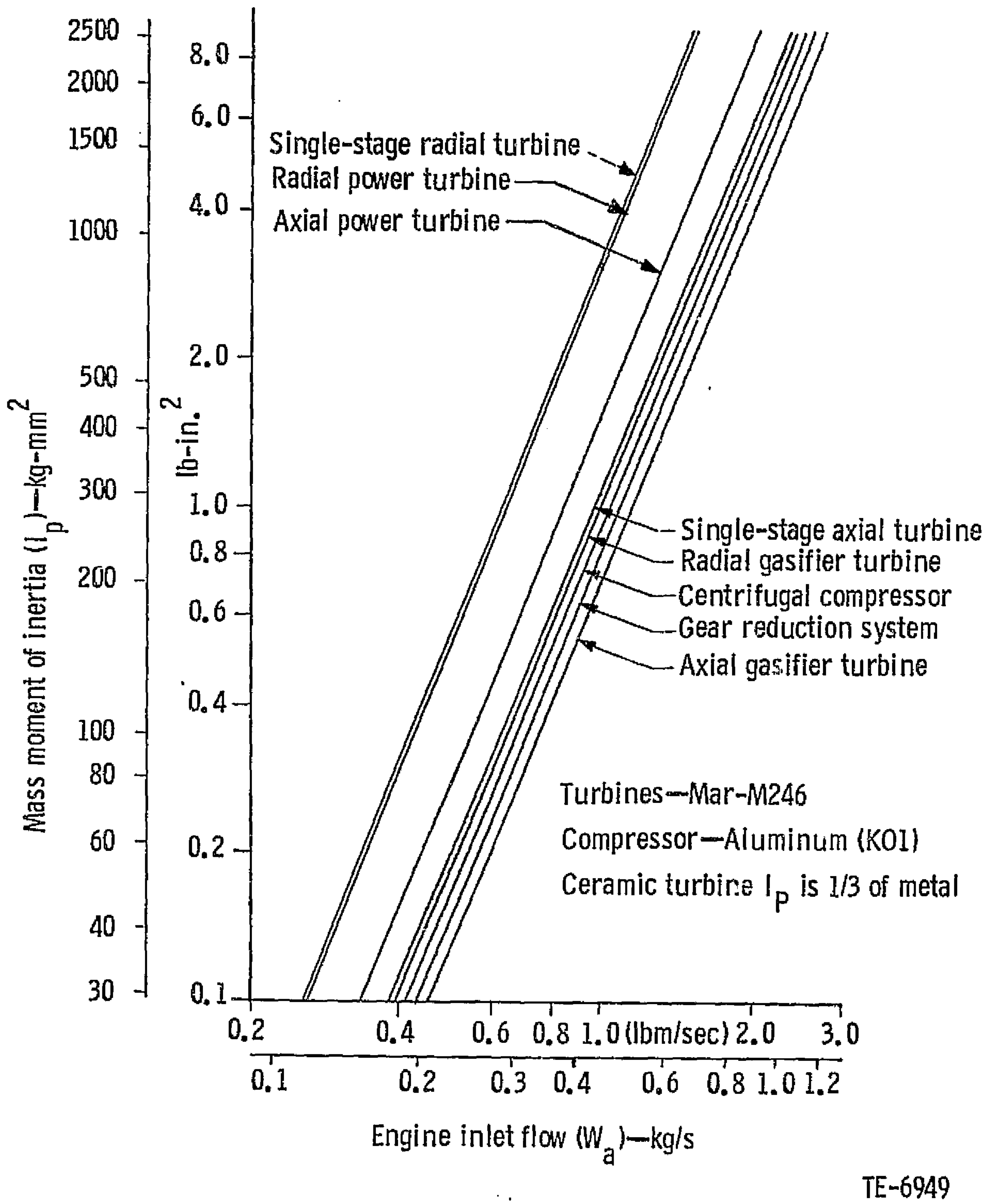

Figure 6. Engine rotor system component inertia scaling characteristics.

ratio of 4.5:1, represents recent DDA technology. It has demonstrated an extremely flat efficiency profile and a favorable surge characteristic. Along a constant turbine inlet temperature operating line which has an 8% minimum surge margin, the 505-IIIA shows no efficiency degradation from 100% $N/\sqrt{\theta}$ to 80% $N/\sqrt{\theta}$ and only a 2% efficiency loss when the speed is further reduced to 60% $N/\sqrt{\theta}$.

Variable-geometry design point performance levels were also estimated by adjusting the 505-IIIA match efficiency to account for pertinent differences while the off-design characteristics were obtained by appropriate scaling of the DDA VG225 composite map. The VG225 compressor has a design point of 1.36 kg/s (3.0 lb/sec) at a 4.9:1 pressure ratio and employs variable inlet guide vanes and variable setting angle diffuser vanes to obtain broad range. Use of the variable geometry allowed this compressor to demonstrate over 70% flow range, $[(W_{choke} - W_{surge})/W_{choke}] \times 100\%$, at all speeds from 60 to 100%.

Design point efficiencies were derived by adjusting the 505-IIIA performance for size and variable-geometry effects as applicable. The preliminary study engine designs maintained the 505-IIIA specific speed and closely bracketed its pressure ratio so that no adjustments were necessary for these factors. Size effect penalty assessments were determined, using Figure 7, which demonstrates the anticipated range of efficiency loss attendant with flow reduction.

For purposes of this study, the average was used as the applicable penalty. Variable-geometry inlet and exit vanes reduce the design point efficiency relative to a fixed-geometry design because of the losses associated with the addition of the inlet guide vane and of the clearance gaps of the movable

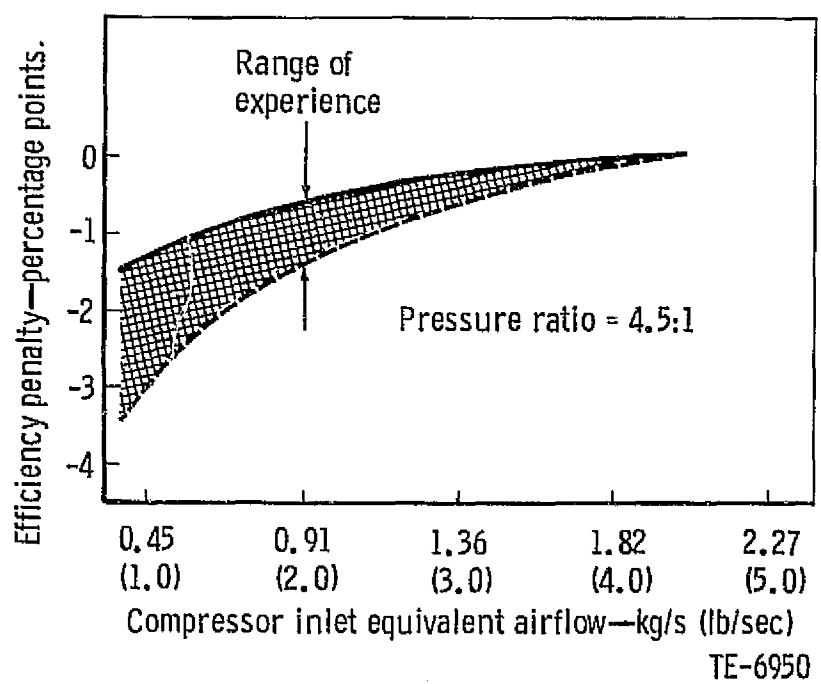

Figure 7. Compressor size-efficiency penalty.

diffuser vanes. These effects were assessed, using DDA rig test experience. A comparison of the maps from the VG225 and a similar technology fixed-geometry compressor having the same design conditions indicated a 1.7 percentage point efficiency decrement attributable to the variable geometry.

Table VI enumerates the efficiency adjustments applied to the fixed- and variable-geometry designs.

TABLE VI. COMPRESSOR EFFICIENCY PREDICTION

	Fixed geometry	Variable geometry
505-IIIA design point efficiency (η_{T-S} adiabatic)	82.0%	82.0%
Speed/R_c adjustment	0/0	0/0
Size adjustment (for $W\sqrt{\theta}/\delta$ = 0.45 kg/s) (1.0 lb/sec)	-3.05	-3.05
Variable geometry adjustment	---	-1.7
Predicted efficiency (η_{T-S} adiabatic)	78.95%	77.25%

These efficiencies represent the design point levels which were utilized in the study engine performance modeling for 0.45 kg/s (1.0 lb/sec) airflow and 4.5:1 R_c compressors. For other flow sizes, the compressor efficiency was adjusted, using Figure 7.

Turbine Performance Characteristics

Turbine concept selection studies have been based on first order analysis techniques of candidate turbine configurations sized to satisfy conceptual engine design requirements. These conceptual design requirements were based on preliminary cycle analyses of a 74.6 kW (100 hp) engine with a pressure ratio of 4.5 and turbine inlet temperature of 1149°C (2100°F). Turbine design requirements for candidate engines are listed in Table VII. For conceptual design, the single-shaft and differentially geared engines were considered, using the same cycle.

The basic turbine types considered were axial, radial, and reentry. Preliminary flow paths and velocity diagrams for each of the various turbines were generated to provide the basis for performance estimates and first order structural design criteria, such as relative blade temperature and blade stress levels. This information provided the foundation upon which to update cycle calculations with reference to component efficiency and structural limitations on metallic turbine inlet temperatures. Turbine efficiencies were continually updated through succeeding cycle calculations to reflect the size effect based on turbine inlet corrected airflow. Table VIII lists the pertinent loss elements which were considered in estimating efficiencies for each of the candidate turbines.

TABLE VII. TURBINE DESIGN REQUIREMENTS FOR CONCEPT ANALYSIS			
	Single-shaft and differential engine	Two-shaft engine	
		Gasifier	Power
Turbine inlet temperature, °C (°F)	1149 (2100)	1149 (2100)	958 (1742)
Turbine inlet pressure, kPa (psia)	429 (62.22)	429 (62.22)	210 (30.5)
Mass flow, kg/s (lb/sec)	0.428 (0.942)	0.421 (0.928)	0.430 (0.948)
Turbine power, kW (hp)	173.6 (232.8)	94.8 (127)	78 (105)
Equivalent flow ($W\sqrt{\theta_{cr}}\epsilon/\delta$), kg/s (lb/sec)	0.230 (0.507)	0.226 (0.449)	0.437 (0.936)
Equivalent work ($\Delta h/\theta_{cr}$), kJ/kg (Btu/lb)	84.7 (36.4)	47.0 (20.2)	43.5 (18.7)

TABLE VIII. LOSS PARAMETERS AND CRITERIA FOR EVALUATION OF CANDIDATE TURBINES		
Axial turbines	Radial turbines	Reentry turbines
Load chart efficiency Efficiency decrement attributable to: Size Reynolds number Clearance Variable geometry	Scroll loss Vane loss (loading-size-N_{Re}-clearance) Vaneless space Impeller (loading-size-N_{Re}-clearance) Exit loss Variable geometry	Conventional axial turbine loss Partial admission effect Effect of mass flow deficit attributable to rotor pumping and seal leakage

Turbine Configurations

The selection of turbine type must be considered with respect to engine arrangement. For the single-shaft engine, either a single-stage radial or a two-stage axial turbine are potential contenders. The single-stage radial turbine flow path, wheel speeds, and relative blade temperatures are illustrated in Figure 8. The 16.5 m (6.5 in.) radial wheel, with its tip speed of 655 m/s (2150 ft/sec), will exhibit relatively high blade and disk stress levels. The relative inlet blade temperature of 996°C (1825°F) would be too high for uncooled metallic construction.

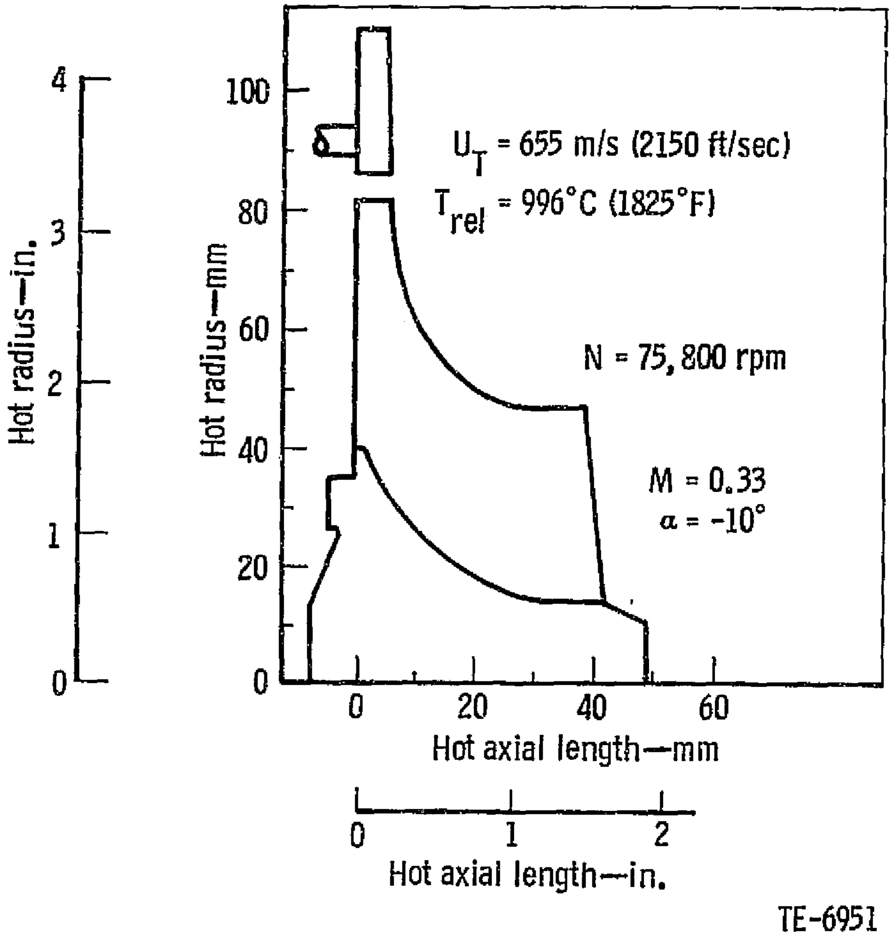

Figure 8. Radial turbine flow path—single-shaft engine.

To ease the problem of exhaust diffuser design to achieve low total pressure loss and uniform flow, the turbine exit Mach number and exit swirl should be as low as possible. The exit swirl of -10° and the exhaust Mach number of 0.33 are favorable for minimizing exhaust loss. The two-stage axial turbine flow path for the single-shaft engine is illustrated in Figure 9. Relative temperatures are too high for uncooled metallic construction.

Exhaust conditions of the two-stage axial turbine are more severe than those of the single-stage radial configuration. The major problem of the two-stage axial design is that of obtaining satisfactory efficiency levels because of the small blade height.

Turbine types for the differential engine include those previously discussed for the single-shaft engine. In addition, the added flexibility for design with different rotative speeds between the compressor and the turbine offers the possibility of utilizing a reentry turbine. The reentry turbine offers increased blade height and reduced relative blade temperatures as compared with the two-stage axial configuration. The flow path of the two-stage reentry turbine sized for the differential engine is illustrated in Figure 10. The turbine is geared to operate at one-half the compressor speed. The

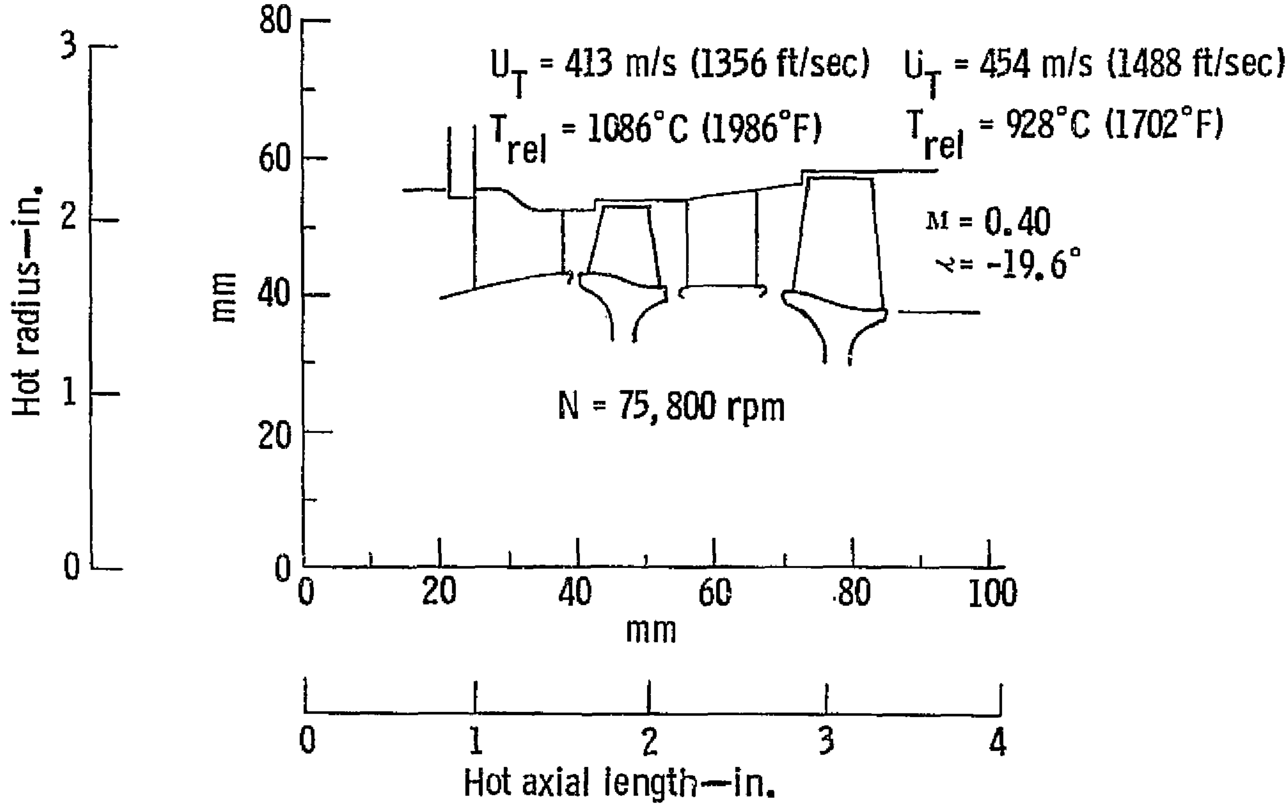

Figure 9. Two-stage axial turbine flow path--single-shaft engine.

use of 30% admission on the first stage results in a much larger turbine than the two-stage axial, thus reducing the adverse aerodynamic effects associated with small blade height. Because the cooler exhaust gas from the first stage passes through the second stage of the same wheel, the effective blade temperatures are reduced considerably. The second stage comprises almost twice the arc of the first stage, thus giving a 2:1 weight factor toward the inlet temperature of the second stage. Unfortunately, the reentry turbine poses additional losses which must be traded against its benefits. Partial admission, leakage, stage flow interaction, and crossover duct losses are common to reentry turbines. Additional reduction gearbox losses must also be assessed against the engine which employe the reentry turbine.

Turbine contenders for the two-shaft engine are the axial gasifier/axial power, radial gasifier/axial power, or radial gasifier/radial power. The axial gasifier/axial power turbine arrangement is illustrated in Figure 11. The small blade height of the gasifier turbine adversely affects its performance. Tip speeds and relative blade temperatures are similar to those of the two-stage axial for the single-shaft engine. The reduced rotative speed of the power turbine structurally allows a larger turbine exhaust area for exit Mach number and swirl angle reductions. The radial-axial flow path is illustrated in Figure 12. The radial gasifier exhibits a tip speed of 503 m/s (1650 ft/sec), which is very attractive for structural design. The

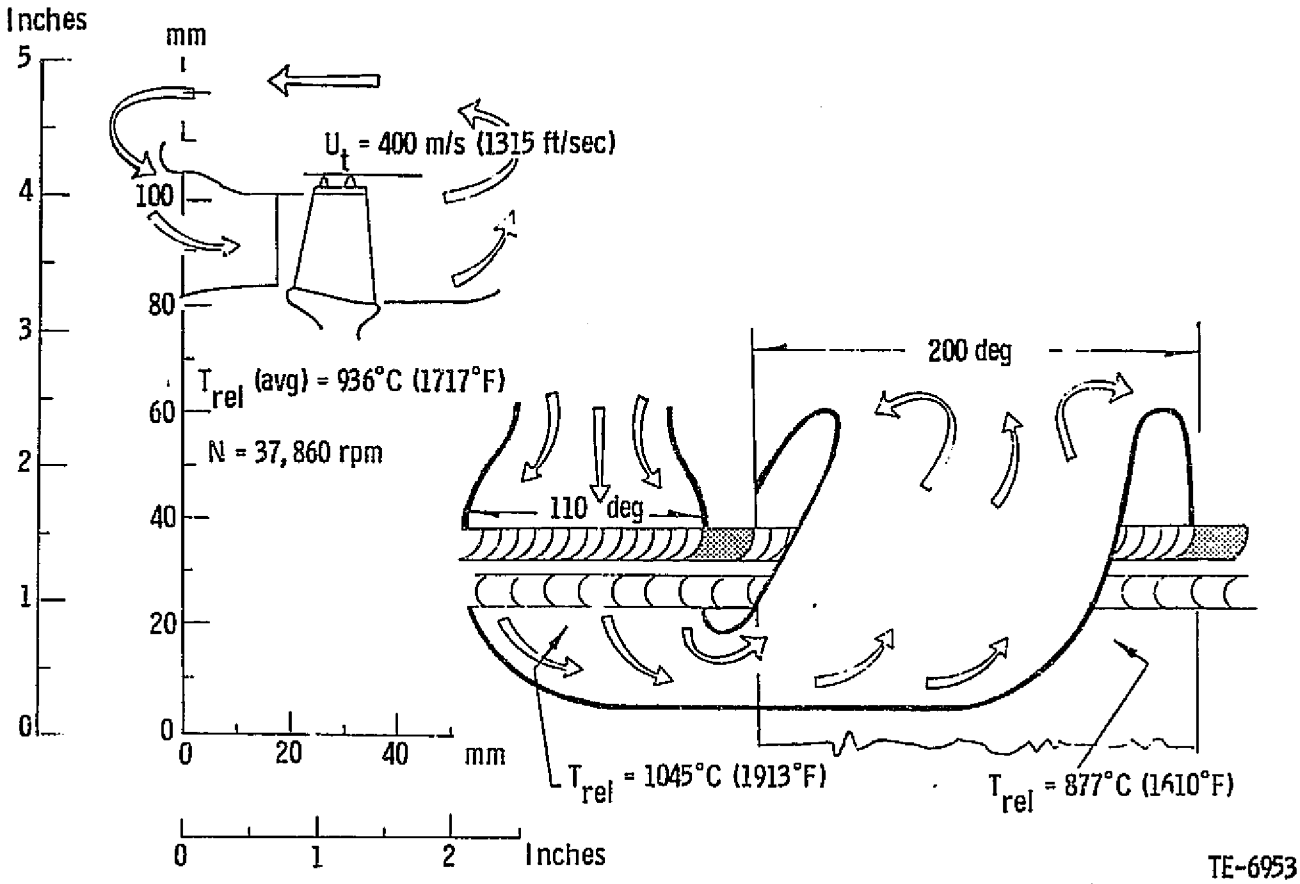

Figure 10. Reentry turbine flow path—differential engine.

axial power turbine, although on the small side, has sufficient blade height to achieve satisfactory efficiency. Typical of radial/axial configurations, a relatively long transition duct between turbine stages is required. To minimize the transition duct loss, the flow path was selected to maintain essentially constant velocity through the duct. The radial/axial configuration appears to offer minimum risk from both aerodynamic and structural considerations.

An additional turbine concept for the two-shaft engine is the radial gasifier/radial power turbine arrangement. The radial power turbine has the definite advantage of more efficient variable-geometry operation over a wide flow range over the axial power turbine. Furthermore, it offers less performance sensitivity to machining tolerances and size than its axial counterpart. These advantages, however, must be weighed against the disadvantage of flow path complexity in the interstage transition duct. The flow paths for the radial gasifier and radial power turbine are presented in Figure 13. The power turbine tip speed of 452 m/s (1482 ft/sec) is well within structural bounds.

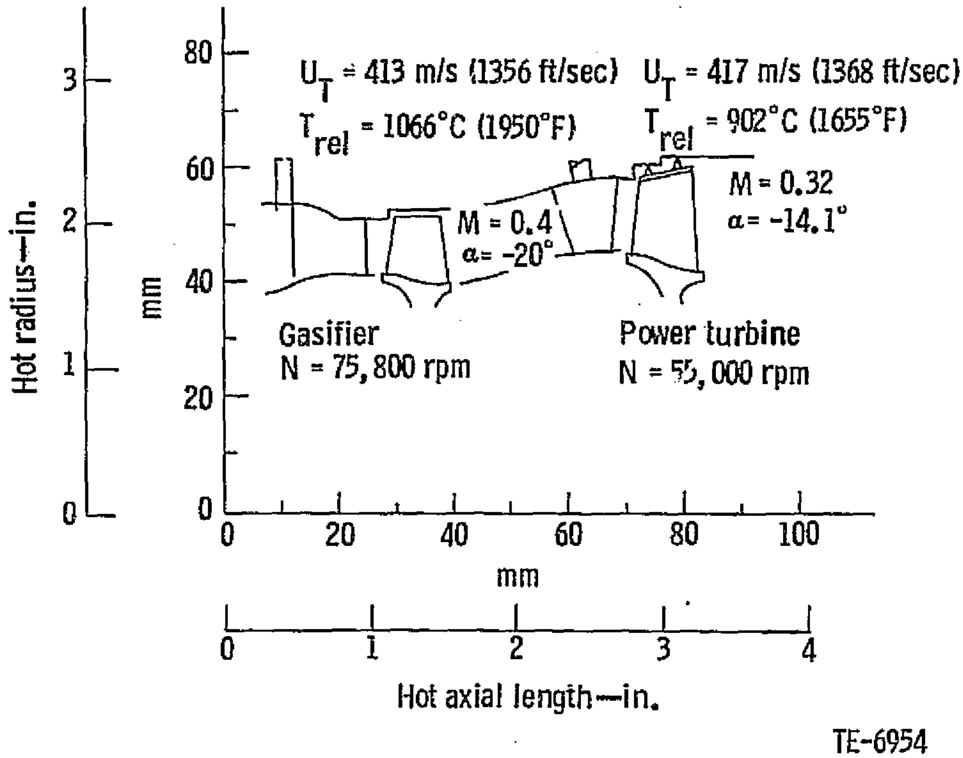

Figure 11. Axial-axial turbine flow path--two-shaft engine.

Size and Variable Geometry Effects

Although the attainment of good component performance is made difficult by its small physical size, the radial turbine exhibits less sensitivity to size effects than the axial turbine, as illustrated in Figure 14. This chart has been used for updating cycle calculations with regard to turbine efficiency as engine cycle analyses have been refined.

A compilation of in-house and open literature data has been used to establish the effect of variable geometry on turbine efficiency, as illustrated in Figure 15. The 100% flow corresponds to the engine full power setting. With pivotal vanes, the vane angle has been selected to allow maximum efficiency to occur at a reduced flow setting, therefore improving part-power engine performance. Partial admission techniques for flow reduction inherently display peak efficiency at the maximum flow setting. The wider flow range of the pivotal vane radial turbine over the axial version is clearly visible. This chart has been used in cycle calculations to reflect turbine efficiency changes with respect to variable geometry position.

Turbine Design Point Efficiency Values

Based on the preceding studies, full-power design point efficiency estimates were provided for cycle analyses. These values are listed in Table IX. The reentry turbine has the least background information on aerodynamic design. Further study to refine the quoted reentry turbine efficiency was not warranted because of the inferior calculated cycle performance of the dif-

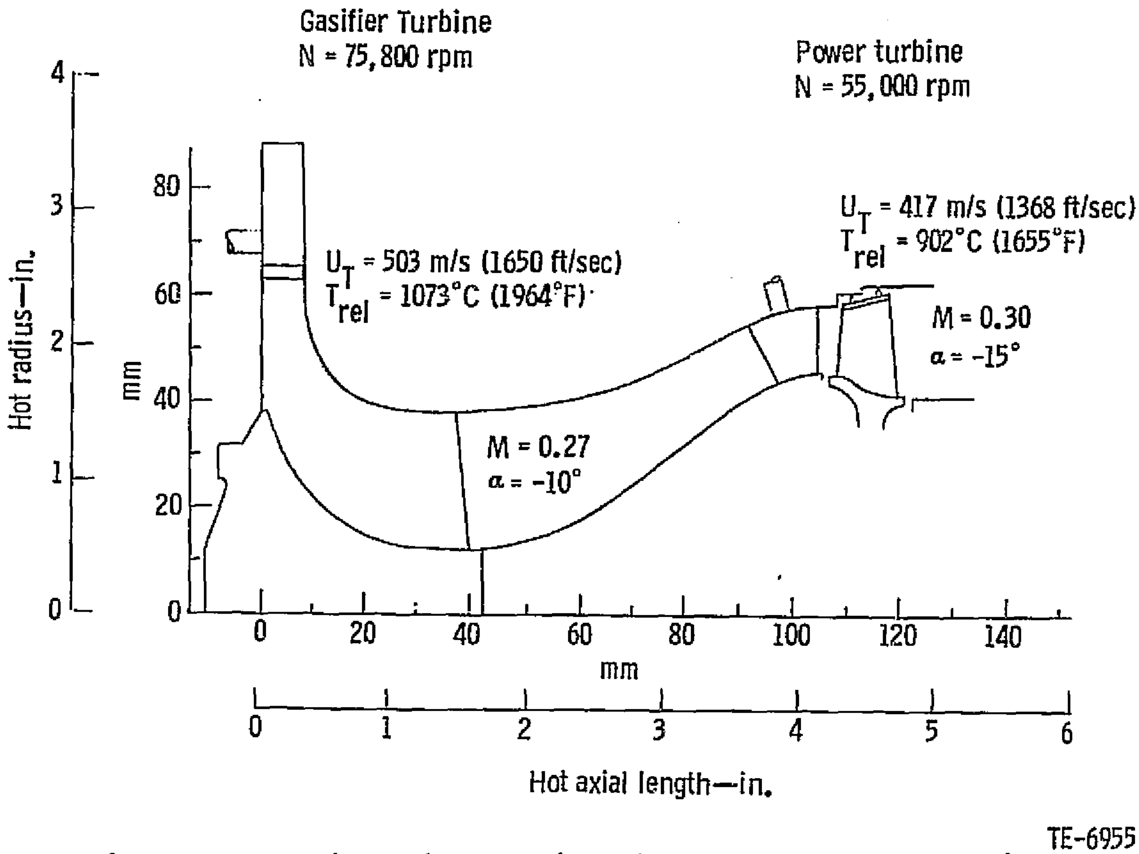

Figure 12. Radial-axial turbine flow path--two-shaft engine.

ferential engine. It should be recognized that turbine designs have been tailored to accent part-power performance. Radial turbines with pivotal vanes, for instance, exhibit peak efficiencies which are approximately 2% higher than those quoted for full power. These peak efficiencies occur at the 70% vane flow setting angle as reflecred in Figure 15.

Heat Exchanger Performance Characteristics

Rotating Regenerator

The regenerator performance model is based on the calculation procedure as presented by Kays and London (Ref 3). This procedure is the effectiveness-NTU method of analysis and is a well established and accepted procedure throughout the heat exchanger industry.

Input to the regenerator performance model, which is a part of the larger engine performance model, consists of parameters specifying the regenerator size, inlet flow conditions, and regenerator matrix characteristics. The resulting performance of the regenerator is expressed in terms of effectiveness and pressure loss.

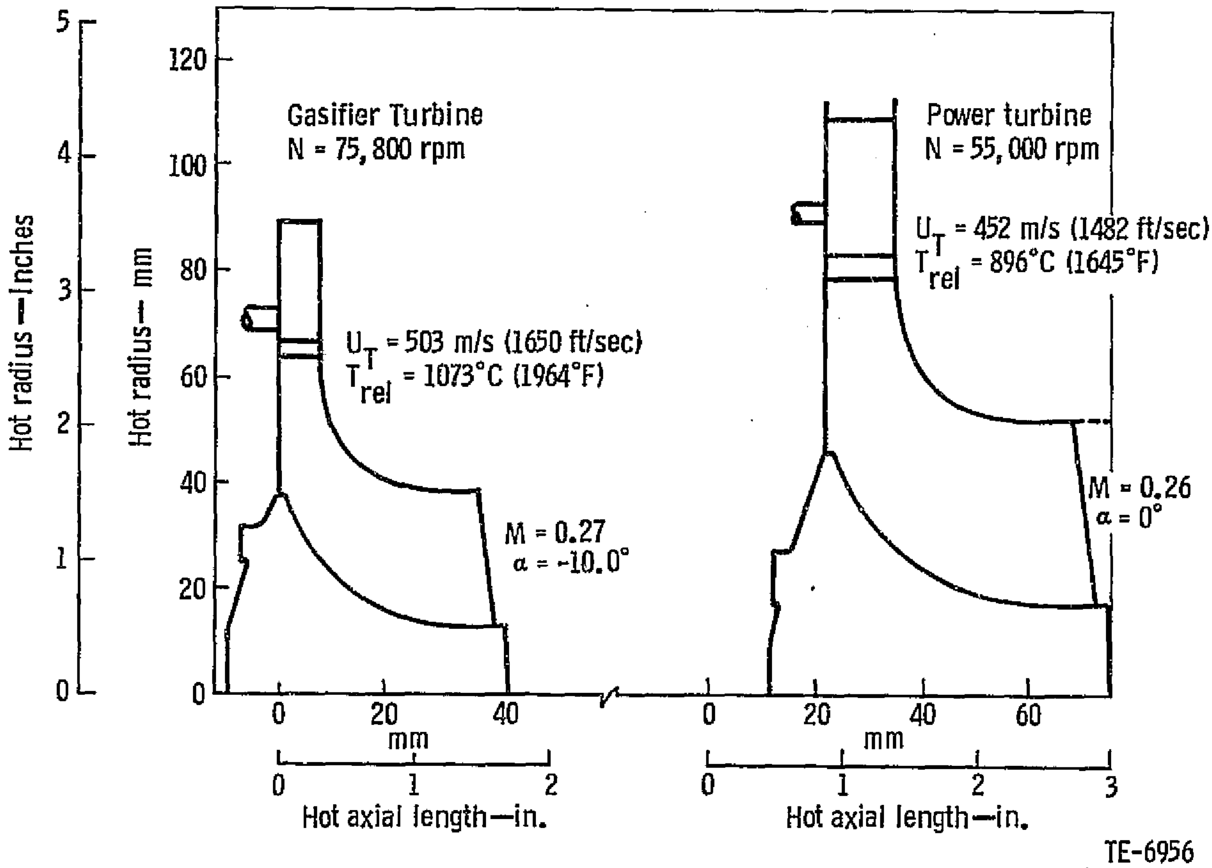

Figure 13. Radial-radial turbine flow paths—two-shaft engine

The regenerator disk selected for the studies is a matrix made of Corning 9460 aluminum silicate ceramic material. The matrix has a high surface-to-volume ratio and is made up of thin-wall triangular passages. The detailed matrix characteristics are listed in Table X. As the disk rotates, compressor discharge air flows through a section of it, and exhaust gases flow through the remainder. The seals which define the flow path consist of a graphite face seal against the moving disk combined with axially compliant leaves against a stationary disk housing. The matrix passages of the disk have a slight axial curvature which cause the entrance and exit of a passage near the disk outside diameter to be offset radially from each other by as much as 6.3 mm (0.25 in.). Because the inboard and outboard seals have the same OD, this passage curvature would adversely affect sealing action if left open. Therefore, the periphery of the disk face is filled underneath the seal faces. The pressure drop and effectiveness of the regenerator disk and the leakage of the seals all affect engine performance.

The selection of a regenerator system for the IGT engine included studies to determine the number and size of disks to be used. These studies consisted of performance evaluation and general arrangement (space available, plumbing, and mechanical drive) considerations.

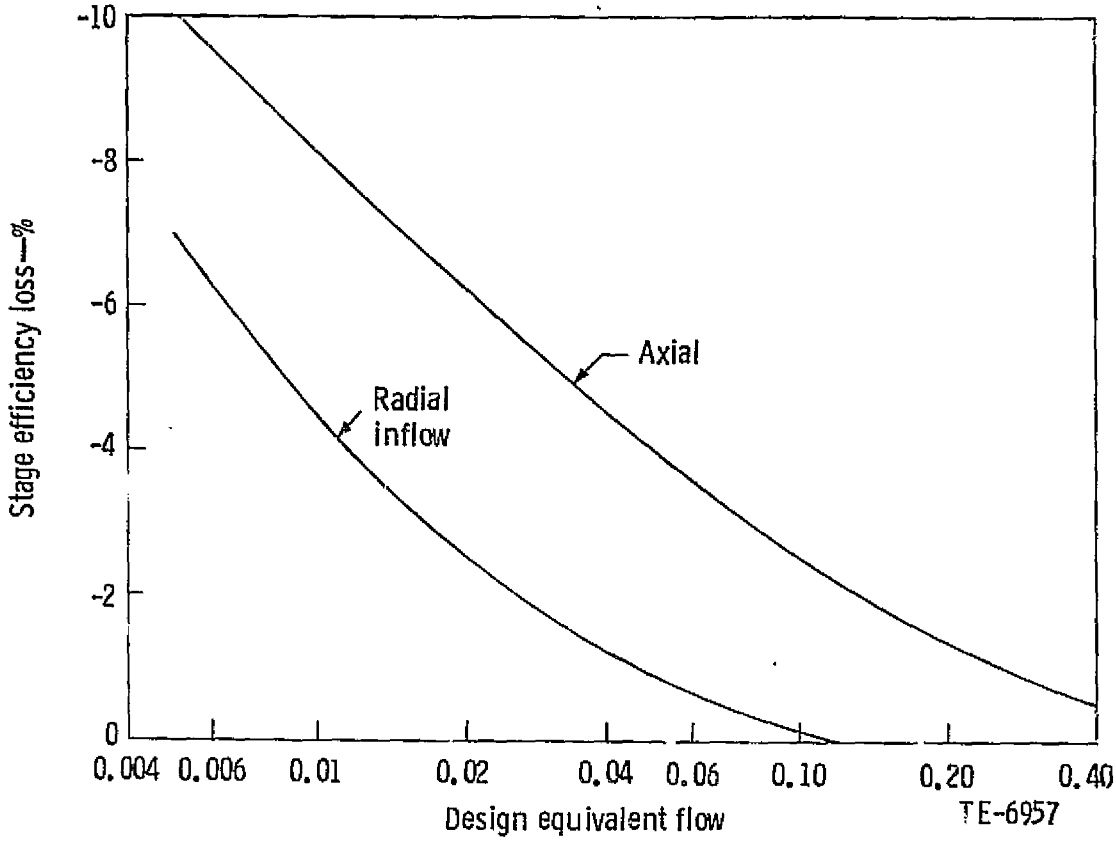

Figure 14. Turbine size effects.

TABLE IX. DESIGN POINT (FULL POWER) TURBINE EFFICIENCIES*

Single-shaft engine	
Radial with pivotal vanes	82.9%
Radial with partial admission	83.9%
Two-stage axial with partial admission	83.7%
Differential engine	
Radial with pivotal vanes	82.9%
Radial with partial admission	83.9%
Two-stage axial with partial admission	83.7%
Reentry with partial admission	83.9%
Two-shaft engine	
Axial gasifier with pivotal vanes	79%
Axial gasifier with partial admission	80%
Radial gasifier with pivotal vanes	84%
Radial gasifier with partial admission	85%
Axial power turbine with pivotal vanes	83%
Axial power turbine with partial admission	84%
Radial power turbine with pivotal vanes	84.5%
Radial power turbine with partial admission	85.5%

*Gasifier and single-stage turbine efficiencies are based on scroll inlet to turbine exit; power turbine efficiencies are based on vane inlet to turbine exit.

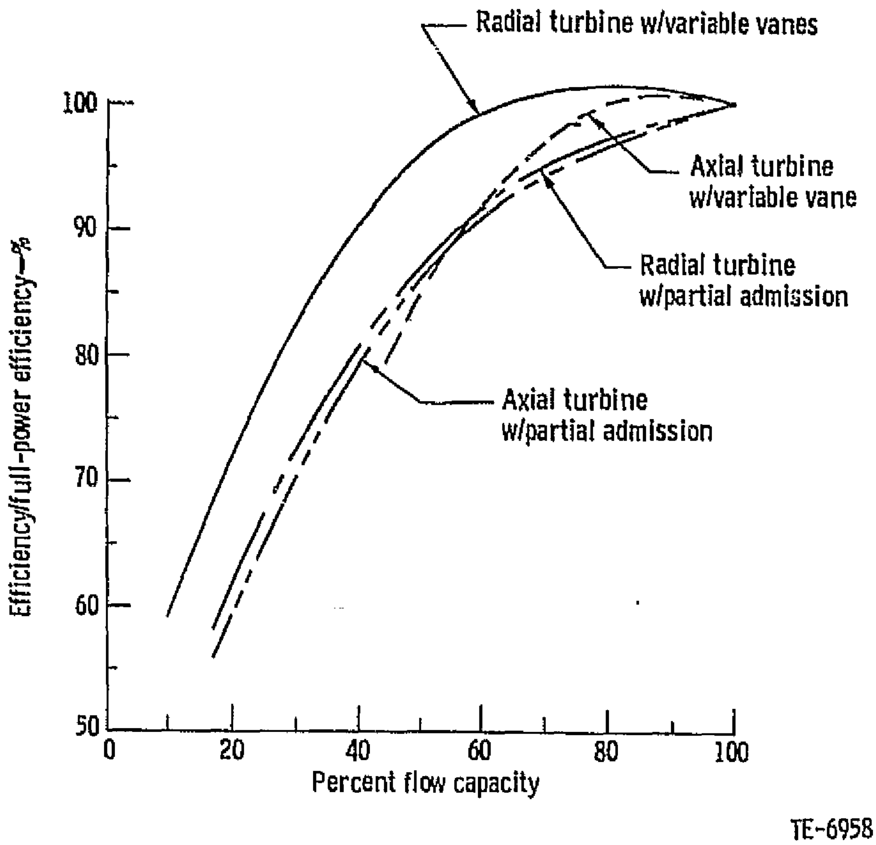

Figure 15. Turbine variable geometry effects.

TABLE X. CERAMIC REGENERATOR MATRIX CHARACTERISTICS	
Porosity (free flow area per unit frontal area)	0.71
Compactness (heat transfer area per unit volume)	6463 m^2/m^3 (1970 ft^2/ft^3)
Density (bulk)	554.2 kg/m^3 (34.6 lb/ft^3)
Hydraulic radius	0.110 mm (0.0003604 ft)
Specific heat	837.4 J/kg K (0.20 Btu/lb-°F)
Thermal conductivity	0.727 W/m-K (0.42 Btu-ft/hr-ft^2-°F)
Colburn surface factor, j	3.95/N_{Re}
Fanning friction factor, f	11.0/N_{Re}
where N_{Re} = Reynolds number	

Figure 16 shows regenerator performance over range of disk diameters and thicknesses for a two-disk system at engine design point conditions. As disk diameter increases, seal length increases with a corresponding leakage increase. Also shown is a performance point with a 46.7 cm (18.4 in.) diameter, 7.6 cm (3.0 in.) thick single-disk system. With allowance for seal blockage, this single-disk system has the same flow and heat transfer areas as a 35.6 cm (14.0 in.) diameter, 7.6 cm (3.0 in.) thick two-disk system and therefore has the same performance, 93.6% effectiveness and 5.9% pressure loss. The single-disk system, however, has lower leakage (4.3% versus 6.3%) because of the shorter seal length.

Recuperator

The recuperator performance model, like the regenerator, is based on the effectiveness-NTU method by Kays and London. Input to the model consists of parameters specifying the size, inlet flow conditions, and the heat transfer surface characteristics. The resulting performance is expressed in terms of effectiveness and pressure loss. In order to operate at the high cycle temperatures required in the IGT engine, a recuperator would have to be made of ceramic materials.

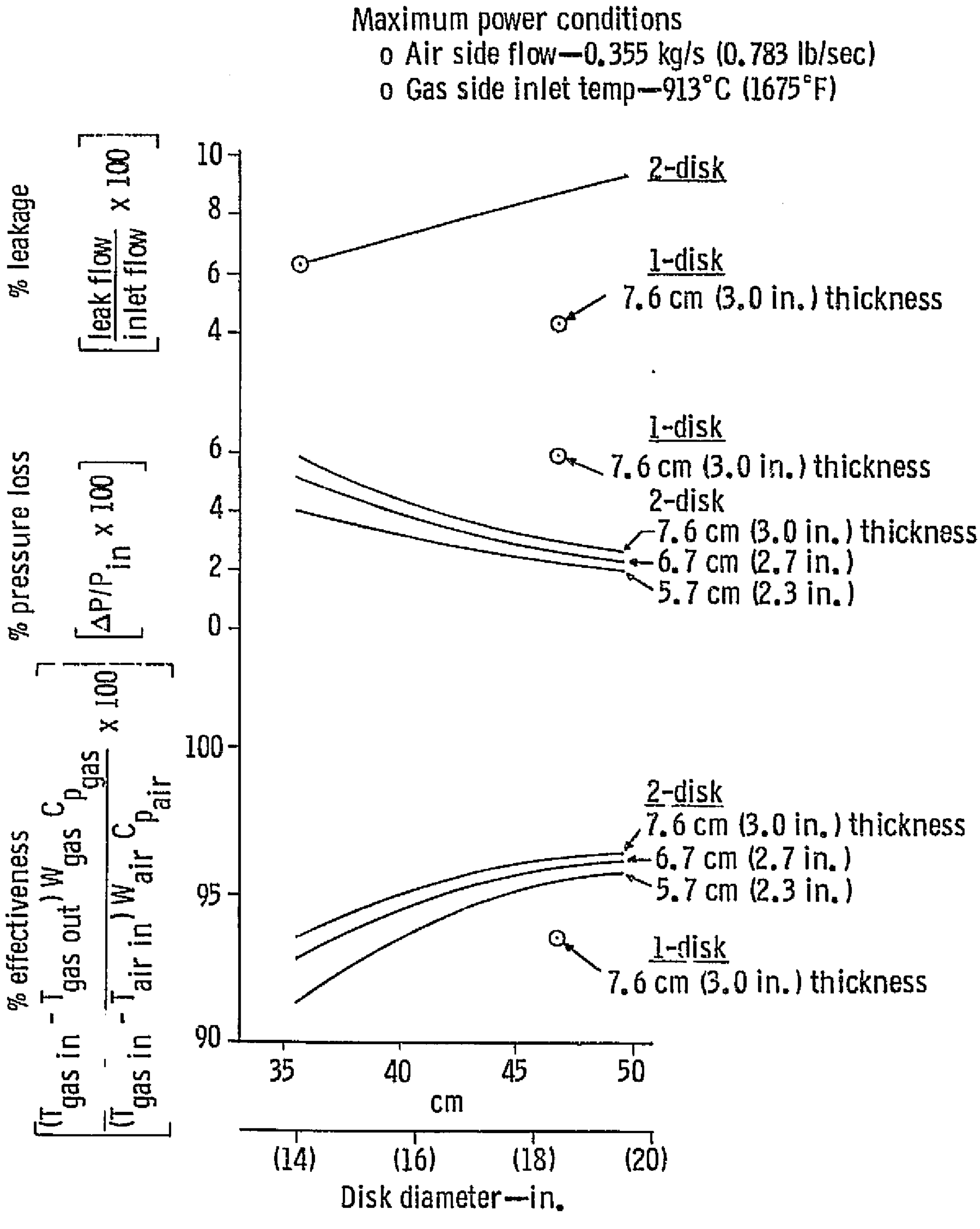

Figure 16. IGT regenerator performance versus disk size.

IV. POWERTRAIN IDENTIFICATION AND DESCRIPTION (TASK I)

During Task I, various improved gas turbine (IGT) powertrain concepts were considered toward the objective of describing the inherent physical and performance characteristics of the concepts and assessing their potential for meeting the objectives of the program. Existing data and study results were used as the basis for Task I. The end result of this task was the selection of a group of powertrain concepts and features for evaluation in greater depth during the "Task II—Powertrain Analysis" study phase.

In the interest of low cost, the simple cycle concept was reviewed but was rejected for inferior thermal efficiency. The two most common types of heat exchangers were compared, and the rotating regenerator was recommended. Compressor intercooling and power augmentation were considered as means of improving thermal efficiency. Intercooling proved to be ineffective and impractical, and augmentation was not recommended at this time. Wide-range variable-geometry aerodynamic components were considered and recommended for further study. To achieve the high cycle temperature needed for high thermal efficiency, the use of ceramic flow path parts was recommended. However, the inclusion of the turbine rotor in this group required further study. Single- and two-shaft engines as well as a differential engine were selected. Transmission studies identified several candidate continuously variable transmissions in addition to the conventional automatic transmission.

SIMPLE CYCLE VERSUS REGENERATION

A fundamental consideration in gas turbine design is whether to include a waste-heat recovery system (i.e., heat exchanger). This recovered heat is fed back into the cycle to provide a portion of the combustor temperature rise and thus reduce that required by the combustion of air and fuel. Excluding the heat exchanger is termed a simple cycle and offers the advantages of lower cost, weight, and size. A regenerative cycle includes a heat exchanger and offers the advantage of reduced fuel consumption. For optimum performance, each concept requires a different compressor pressure ratio: 4 or 5 to 1 for a regenerative cycle and about 10 to 1 for a simple cycle. Thus, for the regenerative cycle a low-cost, low-inertia aluminum single-stage centrifugal compressor can be used. The simple cycle probably requires a multistage compressor and clearly could not use an aluminum rotor because of the high operating temperature (except for the early stages). When each is at optimum pressure ratio, a high-effectiveness regenerative cycle has approximately 50% higher thermal efficiency than the simple cycle (Figure 17). Because the primary goal of the IGT program was high fuel economy, the simple cycle was rejected in favor of a highly regenerated cycle. This choice also allowed the use of the preferred compressor system.

HEAT EXCHANGER TYPE

Two types of air-to-air heat exchangers are candidates for a regenerative engine: a fixed or stationary <u>recuperator</u> and a rotating <u>regenerator</u>. Each has areas of superiority. The recuperator is simple (no moving parts) and has zero leakage; however, its heat transfer surface (its core) is less dense

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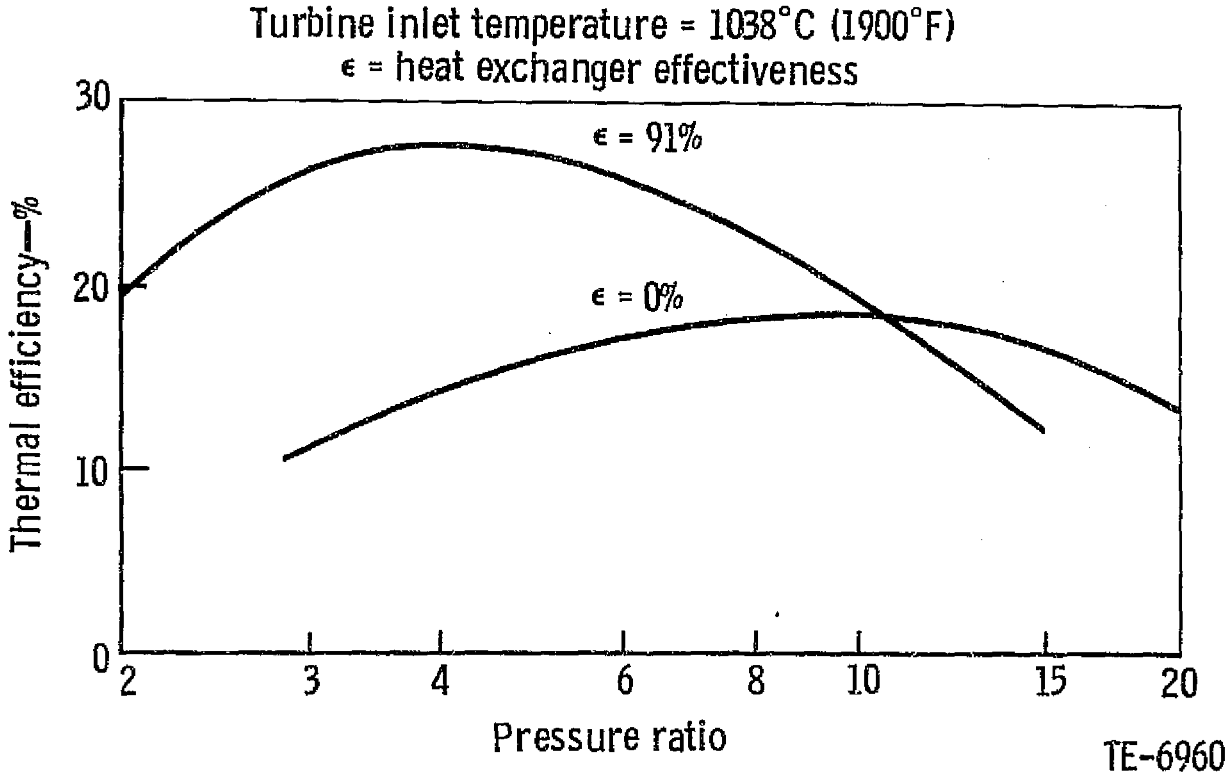


Figure 17. Thermal efficiency versus heat exchanger effectiveness and pressure ratio.

and, thus, the physical size required for a given effectiveness is greater. Regenerators achieve a higher effectiveness in a given envelope but have the disadvantage of seal leakage and seal wear. For passenger car application, regenerators are more highly developed from both performance and mechanical standpoints. For the IGT, the regenerator must be capable of accepting high-temperature exhaust gas. To meet this capability, its construction will require ceramic material. Ceramic regenerator development is already under way and showing good promise. Ceramic recuperator development is considerably further behind. Because of its superior performance (in a given volume) and its higher state of development with ceramic construction, a rotating ceramic regenerator was recommended for the IGT engine. However, the progress of ceramic recuperator development should be monitored for possible future consideration.

COMPRESSOR INTERCOOLING

Compressor intercooling involves a heat exchanger located approximately midway through the compression process; its purpose is to lower the inlet temperature to the second stage of compression. Atmospheric air is the only sink available; thus, the theoretical limit of intercooling is to provide ambient temperature to the second compressor. The "work of compression" for the cooled compressor is reduced, and in this manner intercooling has the potential for improving thermal efficiency. Figure 18 compares the thermal efficiency of an intercooled cycle with a noncooled cycle. At optimum pressure ratio, intercooling improves thermal efficiency by about 12%. (This

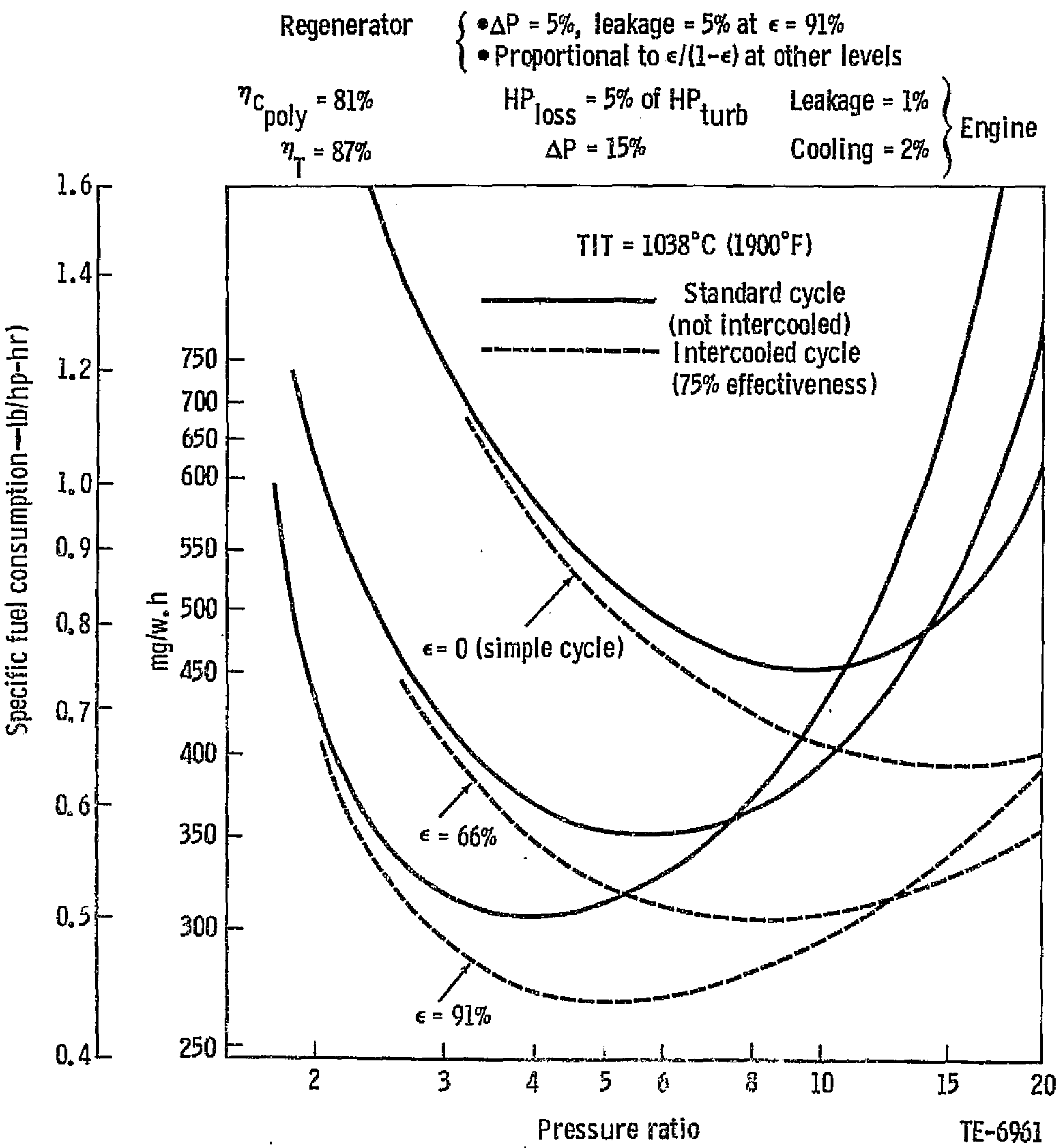


Figure 18. Thermal efficiency with compressor intercooling.

figure is somewhat optimistic in that no penalty for intercooler pressure drop or fan power has been included.) To achieve this improvement, the intercooled cycle requires a pressure ratio about 50% higher than the noncooled cycle. Operation at pressure ratios well below the optimum for either cycle results is very little improvement from intercooling. This point is significant because the very important low-power region of engine operation frequently occurs at pressure ratios below optimum. Intercooling does not improve the thermal efficiency of a simple cycle nearly as much as does regeneration; on an "either-or" basis, regeneration is much superior.

In addition to the fundamental intercooler components (heat exchanger and fan), the concept causes significant changes to other elements of the turbine engine:

- o The compression system must be "split" to allow installation of the intercooler. Thus, a single-stage centrifugal compressor could not be intercooled. Acceptable compressor configurations are axial-centrifugal or twin-centrifugal.
- o A higher-pressure-ratio compressor is required to utilize intercooling effectively. While this is compatible with a split compressor, secondary impacts occur in the regenerator and turbine systems.
- o Higher compressor pressure ratio is very likely to require additional turbine stages. This likelihood is independent of turbine type (axial vs radial) or engine arrangement (single- or two-shaft).
- o The high level of pressure ratio required by intercooling is estimated to result in excessive leakage through the seals of a rotating regenerator. Because this leakage would have a strong adverse effect on thermal efficiency, a rotating regenerator appears to be incompatible with an intercooled cycle. A fixed recuperator heat recovery system must therefore be used, and it is likely to have a lower effectiveness. Given the state of art in ceramic heat exchanger technology, the selection of a recuperator is judged to dictate metallic construction with an associated reduction in allowable turbine temperature (TIT). Future recuperators may be ceramic, however. This inherent lowering of TIT and effectiveness would probably prevent an intercooled cycle from exhibiting any net thermal efficiency improvement.

The cost of an intercooled engine is judged to be considerably higher than that of conventional turbine cycles because of the addition of a second heat exchanger system, a second compressor rotor, and a turbine stage or stages. As a result, intercooling is estimated to have a very low potential for meeting cost objectives—very likely an inability to do so.

Overall, intercooling is not recommended for further study because its potential for meeting the objectives of low cost and high thermal efficiency is very low.

AUGMENTATION

Passenger cars spend the majority of their operating time at very low levels of engine power. Typically, engine specific fuel consumption (sfc) rises rapidly at low power. The use of a small engine would allow most vehicle power requirements to be met at higher percentages of maximum power and would thus tend to avoid operation on the steep portion of the sfc curve (Figure 19). DDA calculations indicate that a 15% reduction in engine size would result in a 4 to 5% improvement in EPA driving cycle fuel economy.

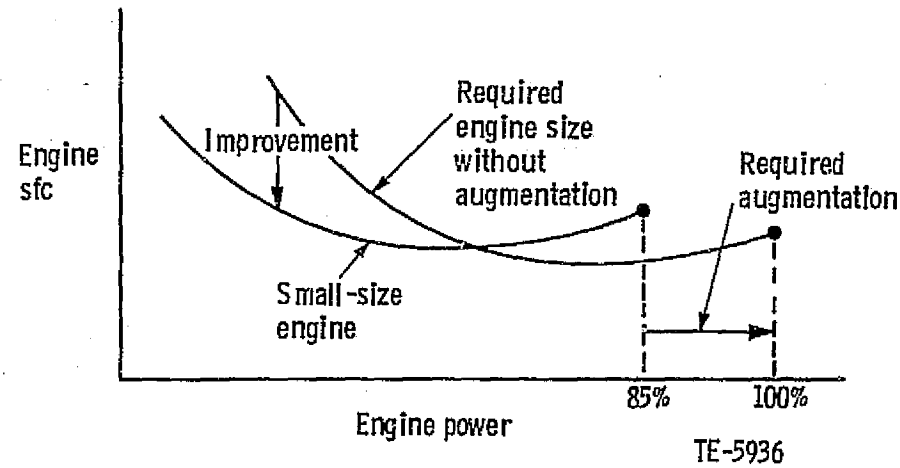


Figure 19. Effect of engine size on part-power sfc.

Good design practice will never allow oversizing the engine; its maximum power will be no greater than that required to meet the specified vehicle acceleration and driveability characteristics. Therefore, a small engine can be used only if its maximum power can be augmented.

Water injection at the compressor and/or combustor inlet is capable of providing about a 15% power augmentation. Its use would be "on demand" (e.g., wide-open throttle acceleration). Such an augmentation system has several drawbacks (complexity, cost, and nuisance--"another tank to keep filled") including safety (running out of water during a high-power maneuver). Some form of antifreeze additive might also be required.

The development of an augmentation system does not require technology advancement; it could always be added at a later stage, such as during prototype development, if then warranted. For the IGT powertrain system, DDA felt that the study should focus on those components and subsystems which require technology advancement and/or integrated development. Because power augmentation does not fit this category, its inclusion into the IGT concept is not recommended at this time.

WIDE-RANGE VARIABLE GEOMETRY

Typical passenger car operation requires very low power levels from its engine; the average is on the order of 15% of maximum power. Because conventional turbine engines exhibit sfc curves which rise rapidly at low power, overall fuel economy is actually determined by the part-power engine sfc characteristics. Turbine engines can have outstanding sfc at design point, and their fuel economy could be improved if a method could be found for keeping the engine at the design-point thermodynamic cycle while varying the power output. In other words, a variable-sized engine is needed. Power output is proportional to the flow rate of working fluid, and all engines reduce airflow in the process of lowering output. Typically, this is accomplished by lowering the speed (rpm). Usually, compressor pressure ratio and turbine inlet temperature also are reduced.

Wide-range variable geometry is a concept of varying the airflow of the components of a turbine engine. For example, a variable inlet guide vane plus a variable diffuser behind the impeller produces a variable-airflow compressor, and variable inlet vanes upstream of a turbine rotor yield a variable-flow turbine. Because of increased endwall clearances, the design point efficiency levels of variable-geometry aerodynamic components are lower than those of fixed-geometry components. The amount of decrement depends on the type of component and the specific variable geometry design; however, a typical range is a 1 to 2% efficiency change. An engine composed of such wide-range components can be made to vary its output power by simply changing the setting angle of the component variable-geometry elements. Previous designs, component rig tests, and engine tests have demonstrated the technical feasibility of this concept. Therefore, wide-range components are recommended as candidates for the design of an improved gas turbine engine. Variable-vane turbine designs are judged to be compatible with ceramic construction, especially for radial inflow types wherein the vanes are few in number and located between parallel walls.

ENGINE CONFIGURATION

A variety of basic mechanical configurations are available for incorporation into the design of an advanced turbine engine. If the engine were the only consideration, the choice would probably be the single-shaft configuration because of its simplicity. Its single turbine suggests low cost and potentially allows simpler implementation of the wide-range variable-geometry concept. However, the transmission required by a single-shaft engine must effectively possess continuous variability of speed ratio. This transmission requirement does not disqualify the single-shaft engine; however, it does dictate that other engine types also be considered.

To date, the two-shaft engine is the type most often picked for vehicular engine designs. This fact alone serves to qualify it as a candidate for an advanced engine. In addition to the large experience base with two-shaft engines, conventional transmissions, either manual or automatic shift, can be used to obtain the total powertrain. While the transmission which matches a two-shaft engine is simple, the additional turbine (and required shafting, etc) in the engine tends to be offsetting.

The single- and two-shaft engines are conventional concepts. Although not a new concept in principle, the differential engine has never been applied to the passenger car. As shown in Figure 20, this engine integrates the engine and transmission functions into a single unit. The compressor, turbine, and load (vehicle) are joined through differential gearing. Theoretically, such gearing permits stalling of the output shaft without requiring a slipping clutch. However, previous analysis has shown that simple differential gearing caused turbine inlet temperature to fall off at part power. To prevent this, a power transfer system is required. As indicated in Figure 21, the power transfer system assumes the appearance of an ordinary transmission (e.g., slipping clutch plus four-speed countershaft transmission). Although the differential engine concept quickly loses much of its theoretical simplicity, additional evaluation was deemed to be appropriate.

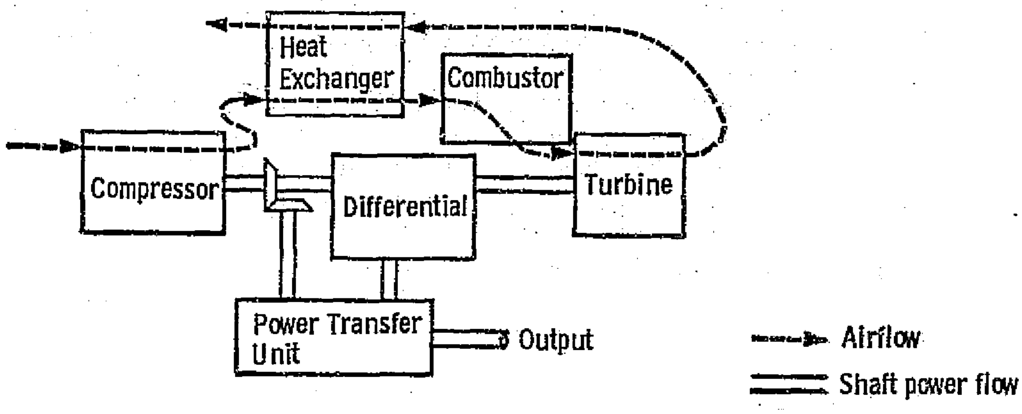


Figure 20. Differential engine powertrain schematic.

A three-shaft engine concept has recently been discussed in the literature. It consists basically of a gasifier section followed by two turbines which are linked to the load through differential gearing. It, too, combines the engine and transmission functions into a single package. This feature is judged to be the unique aspect of the three-shaft engine. Because the differential engine employs this same features without the additional turbines, further evaluation of the three-shaft engine was not undertaken.

CERAMIC COMPONENTS

To achieve the advanced turbine engine thermal efficiency goals, high cycle temperatures are needed. Toward this end, two approaches to engine design are recommended for further study. The first is an engine with the highest turbine inlet temperature possible with a metal turbine rotor(s). This engine concept is estimated to require ceramic turbine vanes, tip shroud, and regenerator. A ceramic burner and plenum may also be needed. This engine concept, if adequately efficient, avoids the more difficult design and development of a ceramic turbine rotor.

The maximum possible turbine inlet temperature would result from incorporating a ceramic turbine rotor into the engine design. In addition to the ceramic parts mentioned, this approach would also require the combustor, plenum, transition duct, and possibly the exhaust diffuser to be ceramic. In a two-shaft engine, the power turbine rotor could possibly be made of metal.

TRANSMISSION

A wide variety of transmission concepts was considered for the range of candidate gas turbine engines. They generally fall into the following categories:

- Conventional automatic transmissions readily adaptable to two-shaft gas turbines
- Infinitely-varying-ratio transmissions required by single-shaft or differential gas turbines

Figures 21 through 23 show the transmission concepts considered for each engine type. The two-shaft gas turbine engine, which has a rising torque characteristic as the power turbine speed decreases (similar to the internal combustion engine), is readily matched to a fully developed/high-efficiency conventional torque converter automatic transmission. The single-shaft gas turbine engine by nature of its torque/speed characteristic and a very narrow speed range of operation requires a continuously variable drive transmission to provide the wide range of operation required in vehicular propulsion. The differential gas turbine engine, incorporating a differential gear set linking the turbine, compressor, and output shaft can be adapted to mechanical gear transmissions or continuously variable transmissions as a means of increasing the overall coverage of this engine concept.

To achieve the primary objective of producing a gas turbine powertrain capable of a 20% fuel economy improvement over the 1976 automatic powertrain, the transmission candidates should exhibit a maximum efficiency performance over a vehicle speed range of 32 to 80 km/h (20 to 50 mph) to obtain maximum fuel economy on the federal driving cycles. Variable drives exhibit a very good efficiency characteristic when designed and optimized for the full-power engine capability (see Figures 24 through 26). Thus, in heavy-duty vehicle equipment, where power/weight ratios approach 6.6 to 8.2 mW/g (8 to 10 hp/ton) as compared with automotive applications at 49 mW/g (60 hp/ton),

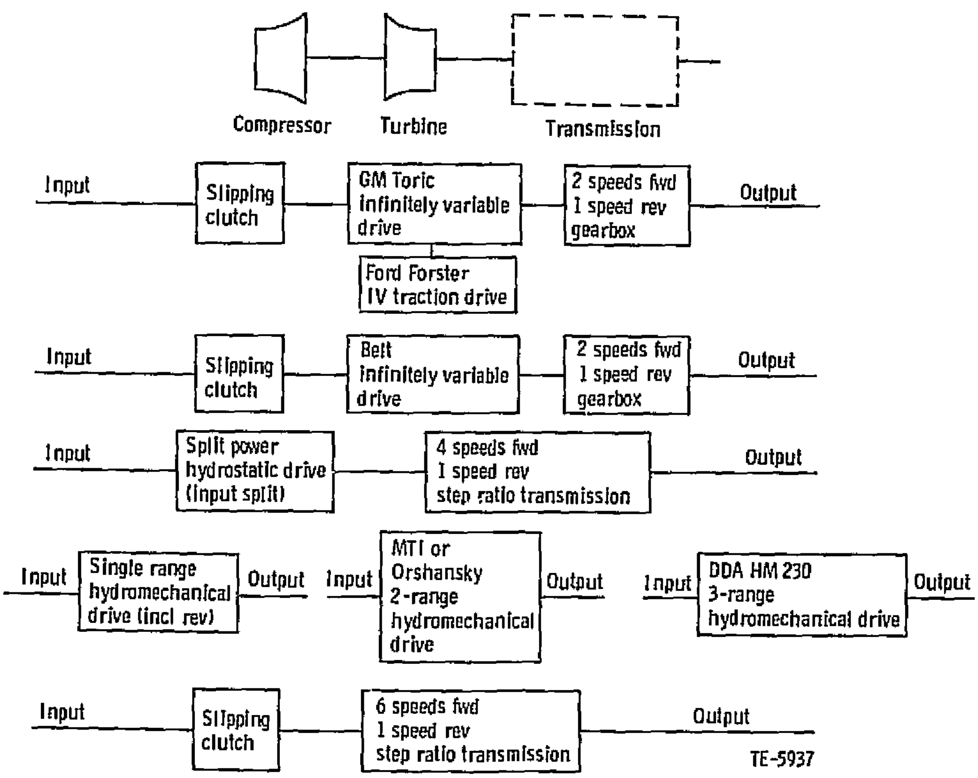


Figure 21. Transmission concepts for single-shaft gas turbine engines.

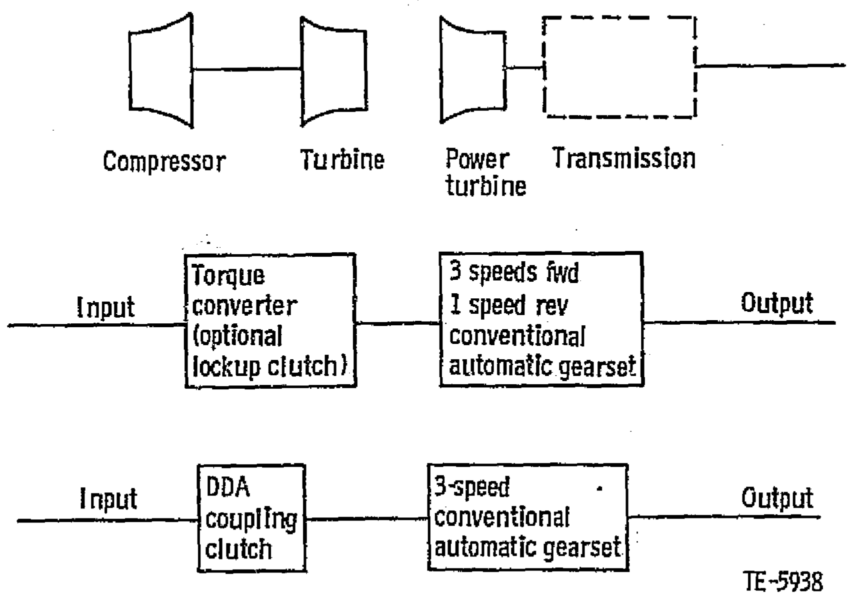


Figure 22. Transmission concepts for two-shaft gas turbine engines.

the variable-drive transmission is a good candidate for the near-maximum power vehicle operation because of its maximum power/efficiency characteristic. However, a significant transmission consideration for automotive applications is that typical driving conditions require a very low engine power output when compared with the total engine horsepower available. When operating over a broad speed range at very low power conditions (i.e., automotive), the efficiency of the subsystems (charging pumps, clutches, gears, etc) become the dominant factor in establishing the overall transmission efficiency. The trend for future automotive considerations is toward smaller, lighter cars and lower-power engines, which further emphasizes the importance of evaluating the transmission subsystem efficiencies because the efficiency of the variable drive itself will be optimized by proper ratio selection to match road load.

An overall view of the mechanical complexity of the transmissions considered for the three gas turbine engines is shown in Table XI. In previous studies and development testing of the Forster traction drive concept conducted outside of General Motors, the conclusions reached were that this type of variable drive was not practical for automotive use. General Motors has studied and tested the Orshansky two-range hydromechanical drive, and DDA Transmission Engineering hold patents on a three-range hydromechanical drive for truck applications. These multirange, variable-drive transmissions peak at 92% efficiency under full-power conditions and offer a broad range of ratio/speed coverage. However for a passenger car geared for 145 km/h (90 mph), a single-range hydromechanical transmission exhibits good full-power efficiency over a broad speed range and should exhibit equal or better overall transmission efficiency at low-power conditions because of fewer parasitic losses associated with the additional gears and clutches required in the multispeed, variable-drive transmissions.

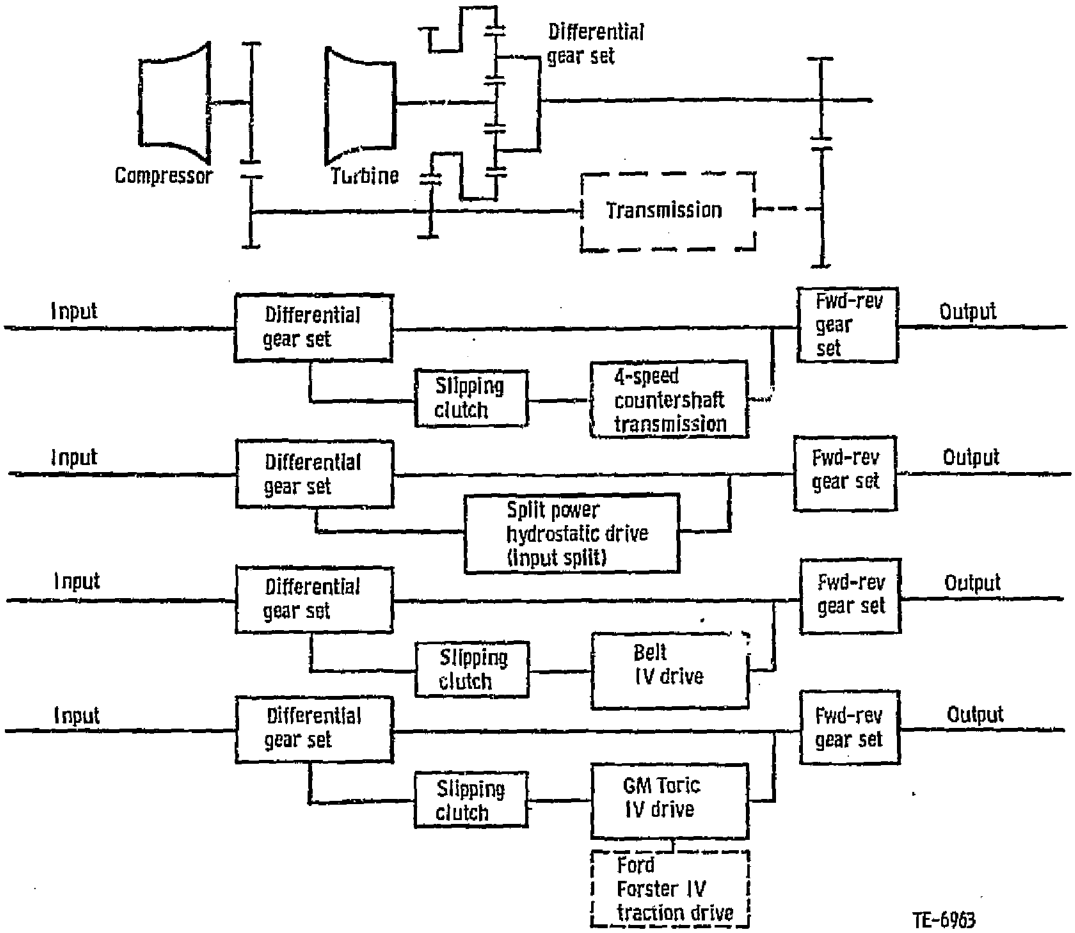


Figure 23. Transmission concepts for differential gas turbine engines.

TABLE XI. TRANSMISSION MECHANICAL COMPLEXITY COMPARISON

Engine	Transmission concept	Differential gear set	Spur transfer gear	Modulated slip clutch	Variable gearing: Spur gears	Variable gearing: Planetaries	Variable gearing: Clutches	Fwd-rev gearing: Planetaries	Fwd-rev gearing: Clutches
Differential gas turbine	4-speed mechanical spur gear train (4.5:1 coverage)	1	4 min	1	8 min	---	4	1	2
	6-speed mechanical planetary gear train (5.0:1 coverage)	1	6 min	1	---	3	5	---	2
	GM Toric IV drive (4.5:1 S.R.)	1	6 min	1	48 kW (64 hp) traction drive			1	2
	Transmatic IV drive (4.5:1 S.R.)	1	6 min	1	48 kW (64 hp) belt drive			1	2
	Hydrostatic IV drive	1	6 min	---	Hydraulic IV drive			---	2
Single-shaft gas turbine	GM Toric IV drive (4.5:1 S.R.)	---	2 min	1	75 kW (100 hp) traction drive			1	2
	Transmatic IV drive (4.5:1 S.R.)	---	2 min	1	75 kW (100 hp) traction drive			1	2
	Hydromechanical drive with 4-speed step ratio gearing	---	2 min	---	Hydraulic IV drive ---	3	5	---	---
	Single-range hydromechanical IV drive	---	2 min	---	Hydraulic IV drive ---	1	---	---	2
	3-range hydromechanical IV drive	---	2 min	---	Hydraulic IV drive 4	3	3	---	---
Two-shaft	Converter/LU clutch 3-speed automatic	---	2 min	---	Fluid coupling/LU clutch ---	3	5	---	---
	Coupling/LU clutch 6-speed automatic	---	2 min	---	Fluid coupling/LU clutch ---	3	5	---	---

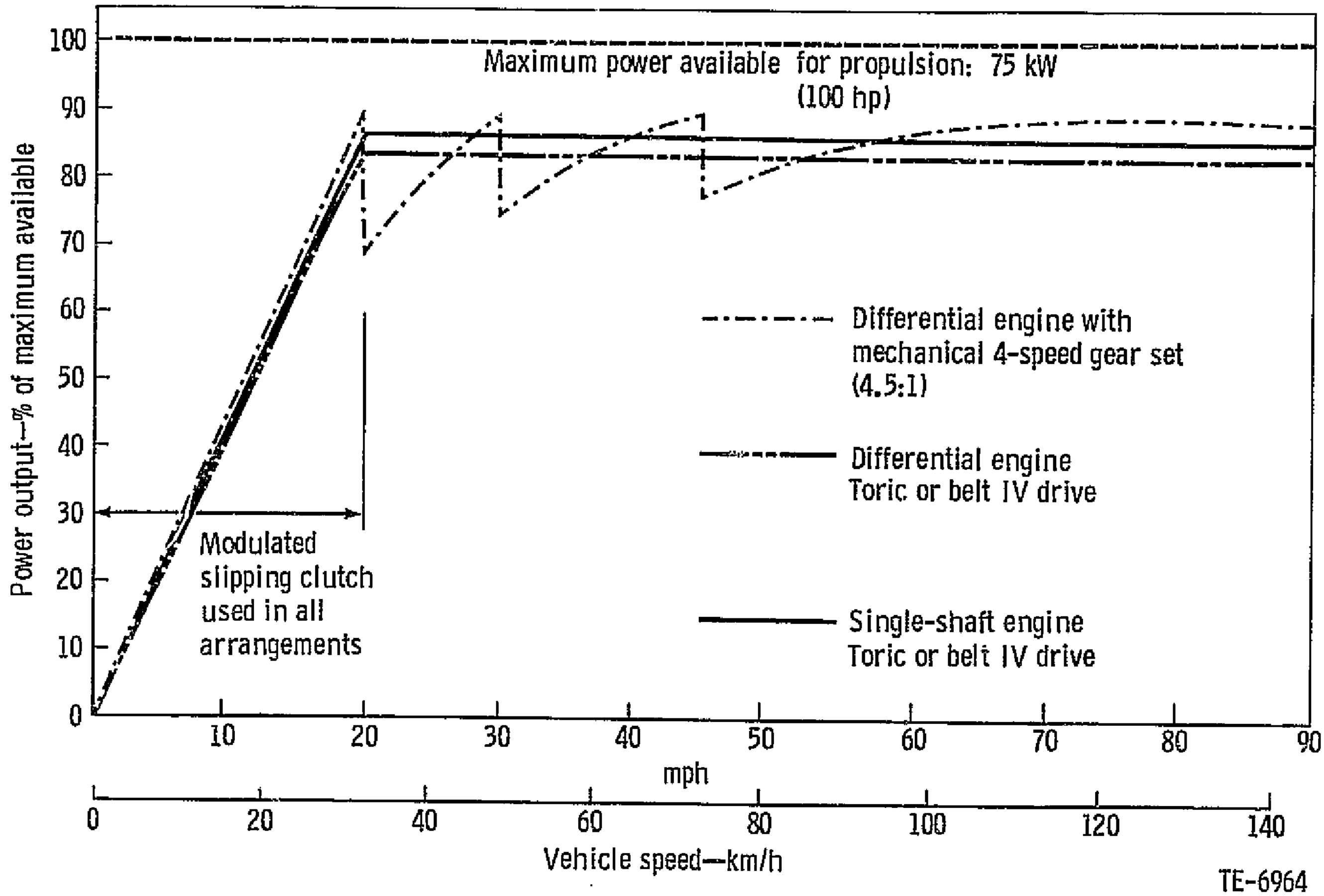


Figure 24. Full-power transmission performance comparison.

In the application of the DDA industrial two-shaft gas turbine engine with a DDA heavy-duty automatic transmission to the Greyhound bus, the existing powertrain compartment required a reduction in overall transmission length. This requirement was met by replacing the conventional torque converter with a patented DDA clutch/coupling arrangement. The coupling provided the required engine idle speed, and the coupling lockout clutch provided the initial torque reaction for vehicle acceleration. Road tests of this feature indicated that vehicle response was comparable to that with the torque converter automatic transmission. For automotive use, the torque will provide the engine torque multiplication required to meet the acceleration objective. The use of a clutch/coupling in place of the fully developed torque converter would add cost to the IGT development program and would offer no production cost, performance, or weight advantage. Therefore, the clutch/coupling concept for the two-shaft IGT engine will not be evaluated further.

Based on these preliminary considerations, the following transmission concepts were selected for further evaluation in Task II:

1. Single-range hydromechanical transmission
2. Automatic torque converter transmission
3. Four-speed mechanical countershaft transmission
4. Toric traction drive continuously-variable transmission
5. Belt drive continuously-variable transmission

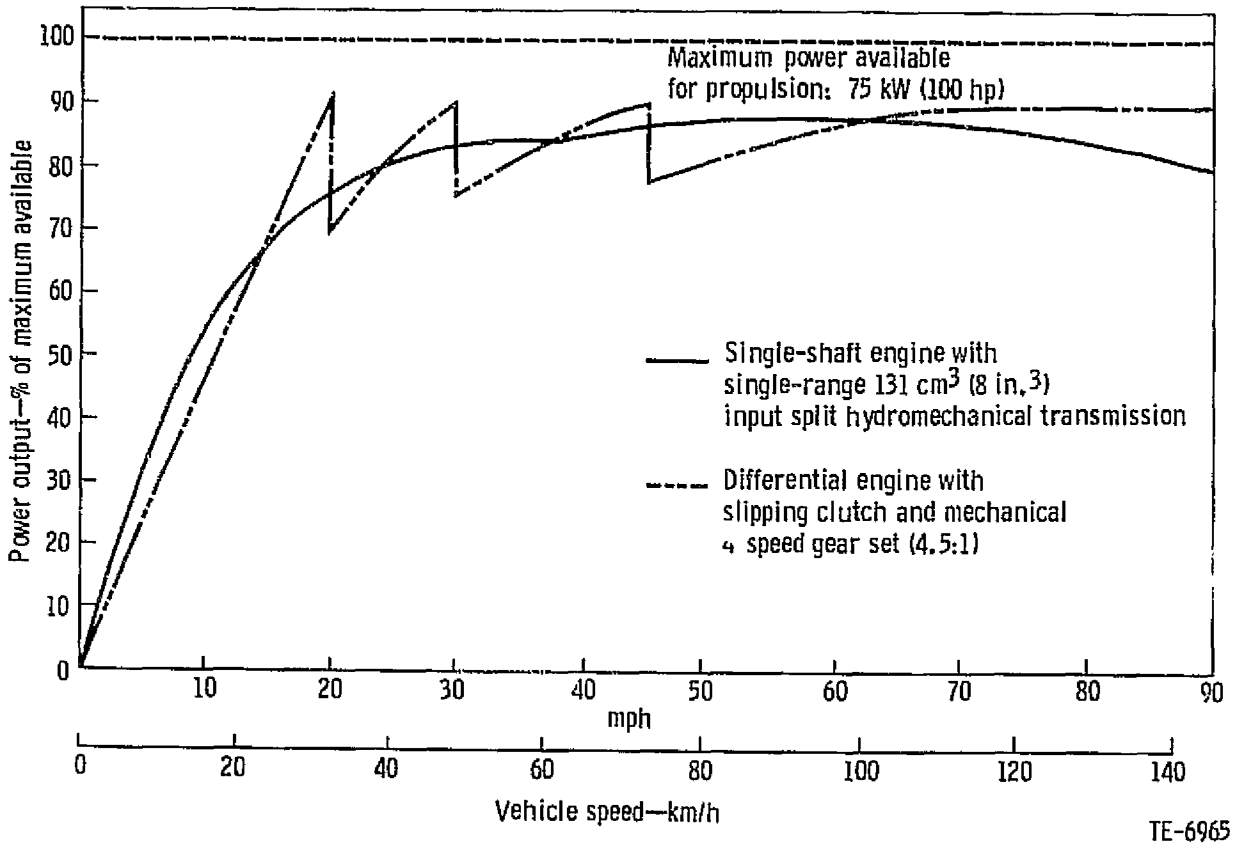


Figure 25. Full-power transmission performance comparison.

The advanced technology for the selected transmission concepts is considered in Task II.

The selected transmission concepts match with the following engine types:

- o Single-shaft engine: hydromechanical, Toric, belt
- o Two-shaft engine: automatic transmission plus convertor or coupling
- o Differential engine: mechanical countershaft, Toric, belt

CONCLUSIONS

An advanced gas turbine could include a variety of design features and configurations. To determine the optimum design, the following were recommended for further detailed study in the Task II powertrain analysis:

- o Rotating regenerator
- o Wide-range, variable-geometry components
- o Metal turbine rotor* or ceramic turbine rotor*

*Plus required static flow path ceramic parts.

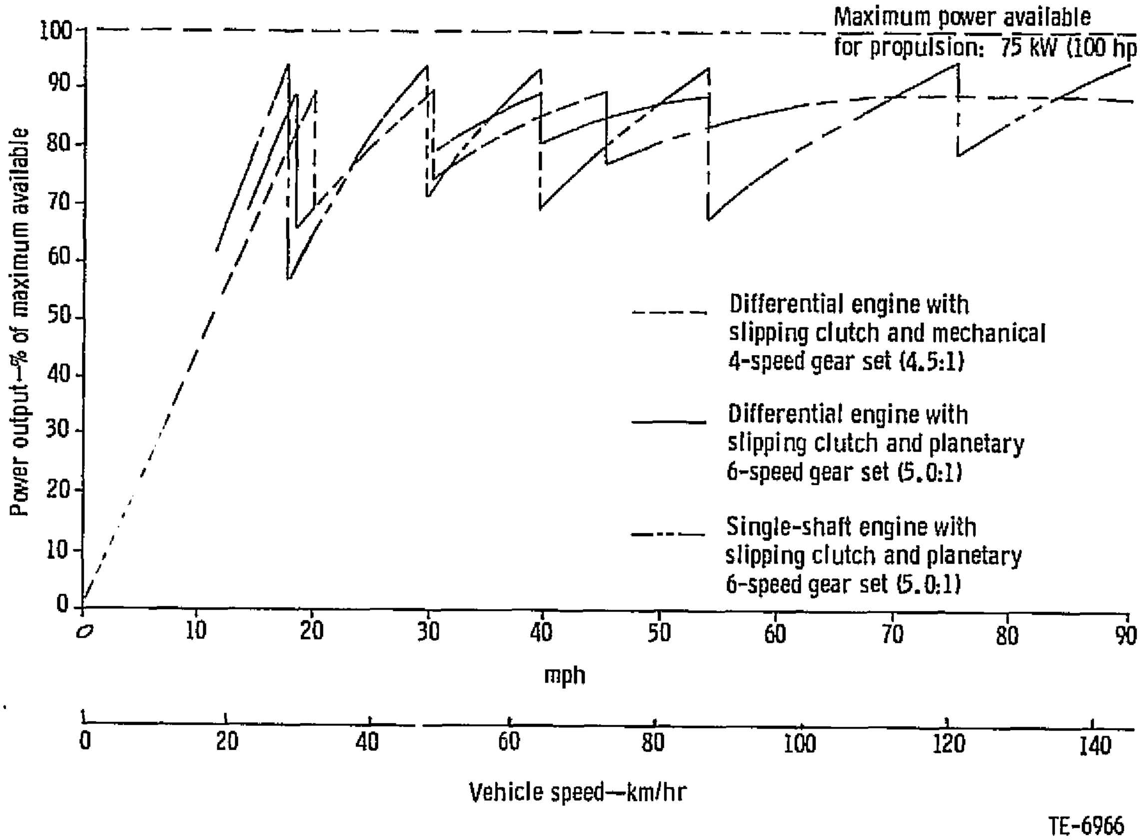


Figure 26. Full-power transmission performance comparison.

- o Single-shaft engine with single-range hydromechanical, Toric, or belt CV transmission
- o Two-shaft engine with conventional automatic transmission
- o Differential engine with mechanical countershaft, Toric, or belt transmission

V. POWERTRAIN ANALYSIS (TASK II)

This task is the middle step of a three-step process to select a recommended turbine engine powertrain for passenger car use. The first step (Task I) considered many concepts and identified those worthy of further analysis. In this step (Task II), the candidates are further analyzed and compared and a selection is made. The third step (Task IIIA) optimizes the selected concept.

CANDIDATE CONCEPTS

Three candidate gas turbine powertrain concepts were selected from those considered in Task I: single-shaft gas turbine engine with CVT, two-shaft gas turbine engine with conventional automatic transmission, and differential gas turbine engine wherein the engine and transmission functions are integrated into a single unit. All concepts incorporated high-effectiveness rotating regenerators to capture waste exhaust heat. This feature not only improves the engine thermal efficiency but also reduces the temperature level of the engine exhaust gas.

CONCEPT SELECTION ASSUMPTIONS

The designer of a gas turbine powertrain must make decisions concerning a great number of variables. Theoretically, a wide range of numerical values for each variable and all their combinations could be studied. The total number of cases to be considered under this approach quickly reaches impractical proportions and necessitates imposing limits and making simplifying assumptions.

Aerodynamic Components

Past DDA studies and rig tests have shown that compressor and turbine designs incorporating variable geometry possess wide-range airflow characteristics. This allows engine airflow at low power levels to be significantly lowered without corresponding large rpm or turbine inlet temperature reductions. The net result is lowered fuel consumption while good engine response is retained. One 1973 DDA study concluded that "variable geometry is required to make the turbine engine competitive with the fuel economy of an Otto-powered vehicle."

Each candidate powertrain was analyzed and compared under the assumption of the use of wide-range, variable-geometry components. This assumption allowed the maximum potential of each concept to be identified and did not introduce any unfair advantage (disadvantage) for any given candidate.

It is recognized that arguments exist against variable geometry (cost, lowered component efficiency, leakage). These effects are similar for all engine types and should not, therefore, influence the eventual outcome. The overall study plan included the analysis of the selected concept in the fixed- and variable-geometry configurations during the optimization phase (Task IIIA).

The physical configurations of variable geometry considered for this study comprised adaptations of systems previously demonstrated in the AGT-2 engine. A reduction in size is required to suit the smaller flow path, but the system would still be within the present state of the art and should attain the same high degree of reliability. Detail design of the variable geometry would consider the application of force balance to minimize actuation forces with the constraint of moving to the idle position with a loss of actuation force.

Turbine Rotor Temperature and Material

The turbine inlet temperatures were selected so that all engines would have equal life and were determined from the interrelationships of the generalized stress parameter of AN^2 (blade exit annulus area times the square of rotor speed), rotor tip speed, and T_{Trel} (blade total relative gas temperature). These criteria give different temperature levels for the single, two-shaft, and differential engines because of the difference in expansion ratio (hence, exit annulus area and tip speed). For a detailed analysis of the temperature selections, refer to Section III of this report.

At the program outset, DDA was hopeful that the thermal efficiency goal could be met with a metal turbine rotor but recognized that a temperature level might be required that would necessitate using a ceramic rotor. This possibility was analyzed by comparing all engines on a metal turbine basis, followed by an analysis of ceramic rotor potential. To maintain equal life with ceramic rotors, some differences in temperature levels for the various engines would be required for the reasons stated. Thus, the use metal temperature limits for comparing the engines is judged reasonable.

Pressure Ratio

A highly regenerated cycle optimizes at a low compressor pressure ratio. To achieve something close to this optimum value in the important low-power operation range, design point pressure ratio must be higher. Existing compressor aerodynamic technology would allow design values as high as 8-10:1. However, such high values are counter to the need for low-cost and low-inertia rotors for passenger car engines. As Table XII illustrates, only cast aluminum rotors meet the cost as well as the inertia requirements (low rotor iertia is required to minimize the rotor acceleration time from idle to rated rpm).

TABLE XII. COMPRESSOR ROTOR MATERIALS

		Rotor inertia	
		Low	High
Cost	Low	Cast aluminum	Cast steel
	High	Titanium	

The temperature-strength characteristic of aluminum limits maximum pressure ratio to about 5:1, certainly no higher than 6:1. This limit easily accommodates design pressure ratios of 4 to 5:1, which experience has shown to yield best vehicle fuel economy.

All of the candidate engine concepts were analyzed and compared with cast aluminum compressor impellers designed for 4.5:1 pressure ratio. This choice allows a practical, low-cost design and optimum engine performance. The selected concept was analyzed at other pressure ratios in the Task IIIA optimization phase.

Engine Power

The design point power for all the candidate concepts is 74.6 kW (100 hp) at 15°C (59°F), sea level ambient conditions. However, all fuel economy and performance analyses and comparisons are made at 29°C (85°F), 152 m (500 ft) ambient conditions. At these conditions, the maximum power of the candidate engine drops to approximately 63 kW (85 hp), which is judged to provide adequate power for the vehicle as discussed under the Engine Sizing Criteria heading in Section III of this report.

THERMODYNAMIC CYCLE ANALYSIS

The configurations that were selected from Task I for further study consisted of a two-shaft, a single-shaft, and a differentially geared engine. The following is a discussion of the design parameters, a survey of sample engine performance maps, and a discussion of control mode.

Design Point Parameters

Table XIII is a listing of the design point parameters. These are for all-metal rotors, each with a temperature rating that results in equal life for the three concepts. Past studies have indicated that a compressor pressure ratio of 4.5 to 1 is very close to optimum for highly regenerated cycles. Component efficiencies were selected on the basis of engine size 0.45 kg/s (1 lb/sec). The mechanical losses reflect the numbers of bearings and number of gears for each configuration. Engine accessory losses reflect the regenerator drive and the fuel and oil pumps. The single-shaft performance was calculated with two types of turbine configuration: a radial inflow and an axial reentry.

Component Maps

All engines use a variable-geometry centrifugal compressor. The compressor characteristics are based on the rig test results of an actual DDA compressor with variable inlet and exit vanes. The efficiency values have been adjusted for these engines to the level appropriate for nominally 0.45 kg/s (1 lb/sec) mass airflow as defined in Section III of this report. The actual value of airflow for the cycles is a result of each configuration being sized for 75 kW (100 hp) at 15°C (59°F), sea level ambient conditions. The efficiencies were not changed to reflect the resultant airflow of each configuration. The difference in airflow of the single-shaft radial turbine and the two-shaft configuration represents only 0.3% in compressor efficiency.

The actual compressor maps used for the cycle analysis are shown in Section VI (Figures 79 through 87) of this report. The efficiency, mass flow, and pressure ratio values were scaled as required to match the design point requirements of the different engines.

TABLE XIII. POWERTRAIN ANALYSIS DESIGN POINT PARAMETERS				
Engine configuration	Single-shaft		Differential	Two-shaft
Turbine configuration	Radial	Axial reentry	Radial	Radial
Turbine inlet temperature*, °C (°F)	1049 (1920)	1195 (2183)	1049 (1920)	1080 (1976)
Compressor inlet airflow, kg/s (lb/sec)	0.57 (1.25)	0.44 (0.96)	0.60 (1.33)	0.51 (1.13)
Compressor pressure ratio	4.5	4.5	4.5	4.5
Compressor efficiency, %	77	77	77	77
Gasifier turbine efficiency, %	82.9	83.7	83.2	84
Power turbine efficiency, %	---	---	---	83
Regenerator effectiveness, %	90	90	90	90
Total cycle pressure loss, %	13	13	13	14
Inlet, %	1.5	1.5	1.5	1.5
Regen cold side, %	0.5	0.5	0.5	0.5
Combustor, %	4.0	4.0	4.0	4.0
Transition duct, %	---	---	---	1.0
Turbine diffuser, %	2.0	2.0	2.0	2.0
Regen cold side, %	4.0	4.0	4.0	4.0
Exhaust to ambient, %	1.0	1.0	1.0	1.0
Regenerator leakage, %	5.0	5.0	5.0	5.0
Cooling air, %	1.8	1.8	1.8	1.8
Mechanical loss, kW (hp)	5.2 (7)	6.9 (9.3)	14.2**(19.1)	7.2 (9.6)
Engine accessory loss, kW (hp)	2.2 (3.0)	2.2 (3.0)	2.2 (3.0)	2.2 (3.0)
Heat loss	2.5% of engine fuel flow			
Compressor speed, rpm	67,800	77,400	65,700	71,300
Gasifier turbine speed, rpm	67,800	38,300	65,700	71,300
Power turbine speed, rpm	---	---	---	51,500

*Max continuous 56°C (100°F) less
**Includes transmission

Both the radial- and axial-flow turbines are used in the engine performance calculations. Each one uses some means of flow control--variable vanes on the radial turbine and partial admission on the axial turbine. Table XIV lists the various turbines and the source of the maps used to represent each.

TABLE XIV. SOURCE OF TURBINE MAPS			
Two-shaft engine			
	Type	Map	Figures
Gasifier turbine	Radial	AGT-2 rig test	88 and 89
Power turbine	Axial	505-4 predicted	69 and 70
Power turbine	Radial	AGT predicted	71 and 72
Single-shaft and differential engine			
	Type	Map	Figures
	Radial	AGT-2 rig test	Same as above
	Axial	250 rig test	

The axial turbine used in the single-shaft engine is a two-pass reentry turbine. This configuration was selected for study because it has two potential advantages over a two-stage design. First, because the gas makes two passes, the blades are longer. This result in better stage efficiency because of lower tip clearance effects. In addition, the turbine inlet temperature can be increased significantly because of temperature averaging by the rotor—i.e., the second-pass rotor inlet temperature is lower than the first-pass temperature and the rotor blading will see an average temperature because of high rotational speed.

The radial turbine map is based on the rig test results of the DDA AGT-2 variable-geometry radial-flow turbine. The 505-4 axial turbine map is based on a computer prediction which has shown good correlation with actual test data in the past.

Engine Performance Maps

The performance maps for the typical single-shaft engine are presented in Figures 27 and 28. The power and fuel flow maps are shown in terms of engine output speed, turbine rotor inlet temperature, and geometry setting. The compressor geometry and gasifier rotor geometry are coordinated so that as mass flow is reduced at a given speed and temperature, adequate surge margin is maintained. The curves are labeled in terms of compressor geometry position. When maximum travel of the geometry has been reached in the closed direction, a further reduction in power can be effected, if necessary, by a reduction in turbine rotor inlet temperature. The two-shaft engine map is plotted with lines of gasifier speed and turbine temperature

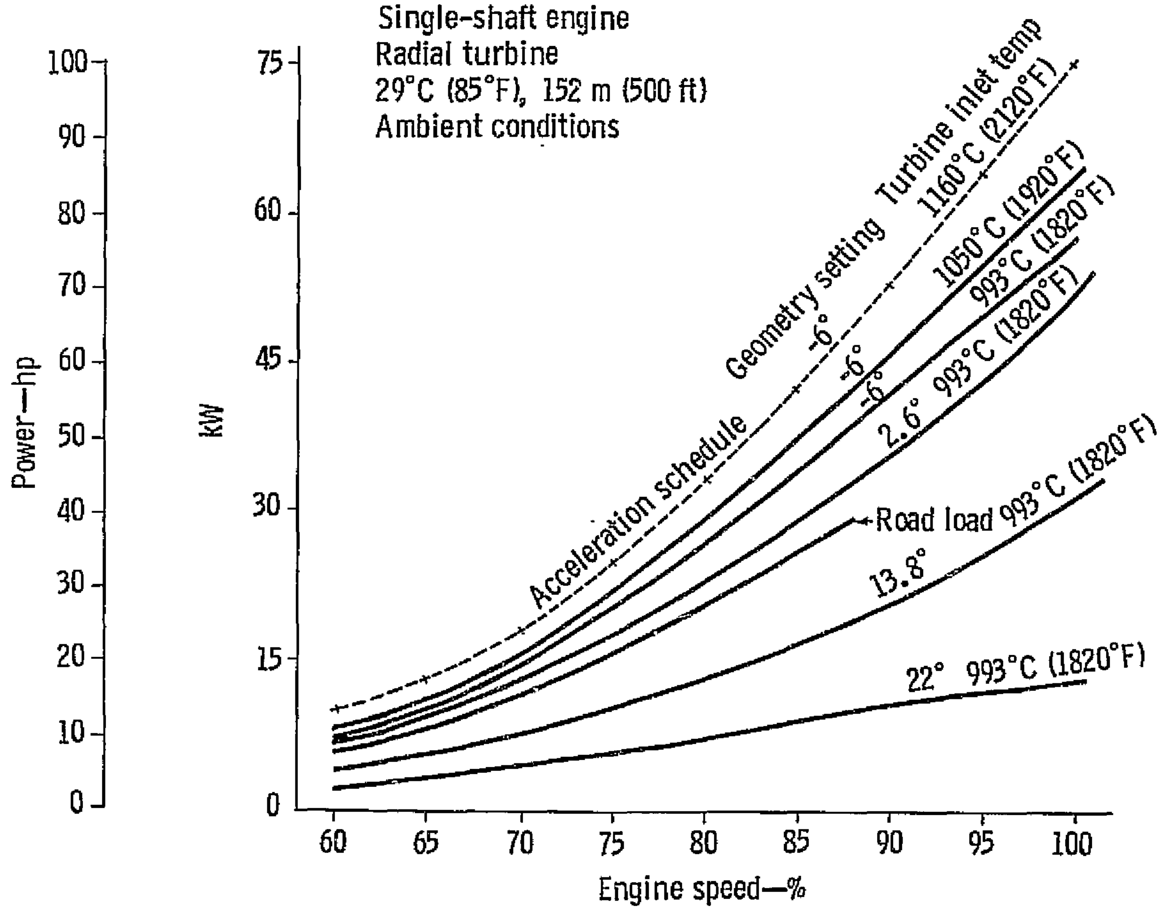


Figure 27. Single-shaft engine power map.

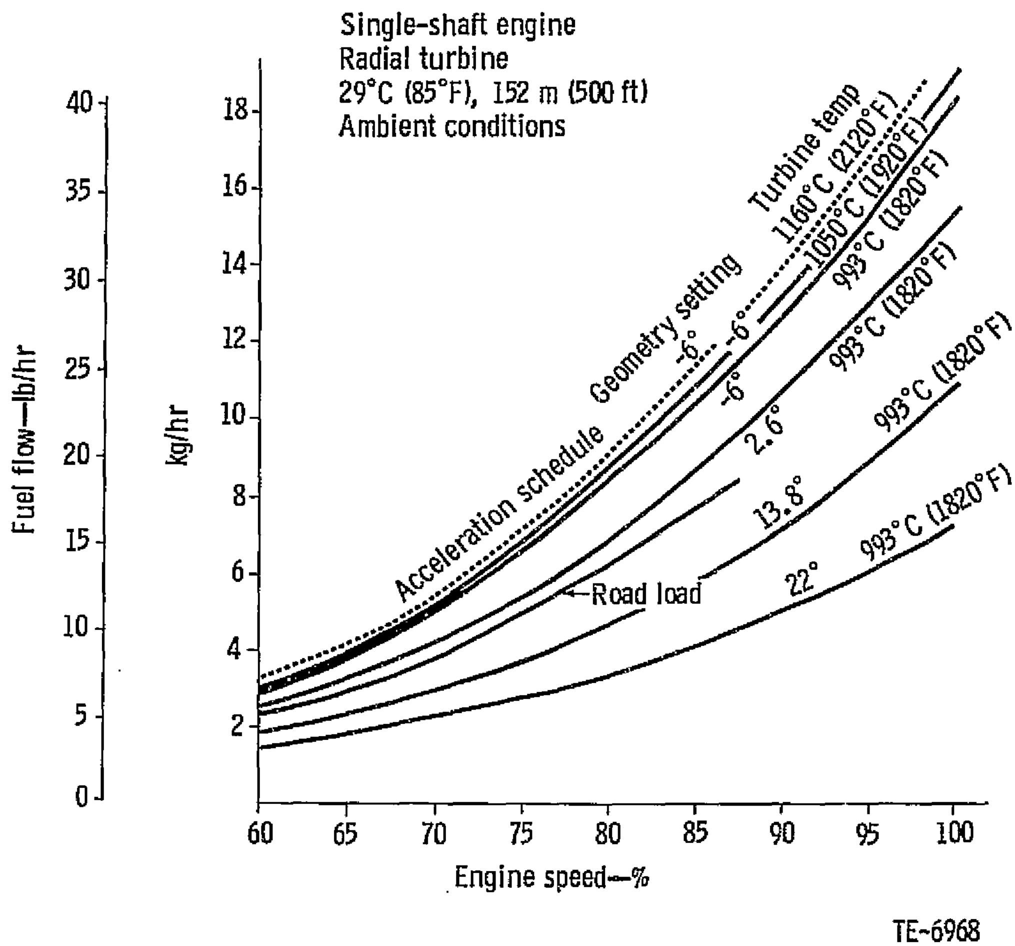


Figure 28. Single-shaft engine fuel map.

at the optimum geometry setting for the particular speed. When the minimum gasifier speed (rpm) is reached, power is further reduced by closing the variable geometry in the gasifier section.

The typical maps of power and fuel flow for a two-shaft engine are shown in Figures 29 and 30. The compressor variable geometry is used to reduce engine airflow at a particular speed; gasifier turbine geometry is varied so that the engine operating point on the compressor is matched at an optimum efficiency with adequate surge margin. The power turbine is varied to hold gasifier turbine rotor inlet temperature to the desired value.

On the performance maps, such as Figures 29 and 30, the geometry settings of -6°, 2.6°, 13.8°, and 22° refer to a compressor setting, where -6° is a high flow position and 22° a low flow setting.

The performance map of the differential engine is not available. Engineering judgment and discrete computer data points were used to estimate fuel economy. Figure 31 shows the variation of the engine accessory loads (regenerator drive and the fuel and oil pumps) as a function of gasifier speed. The vehicle accessories and the idle fuel flow values are discussed

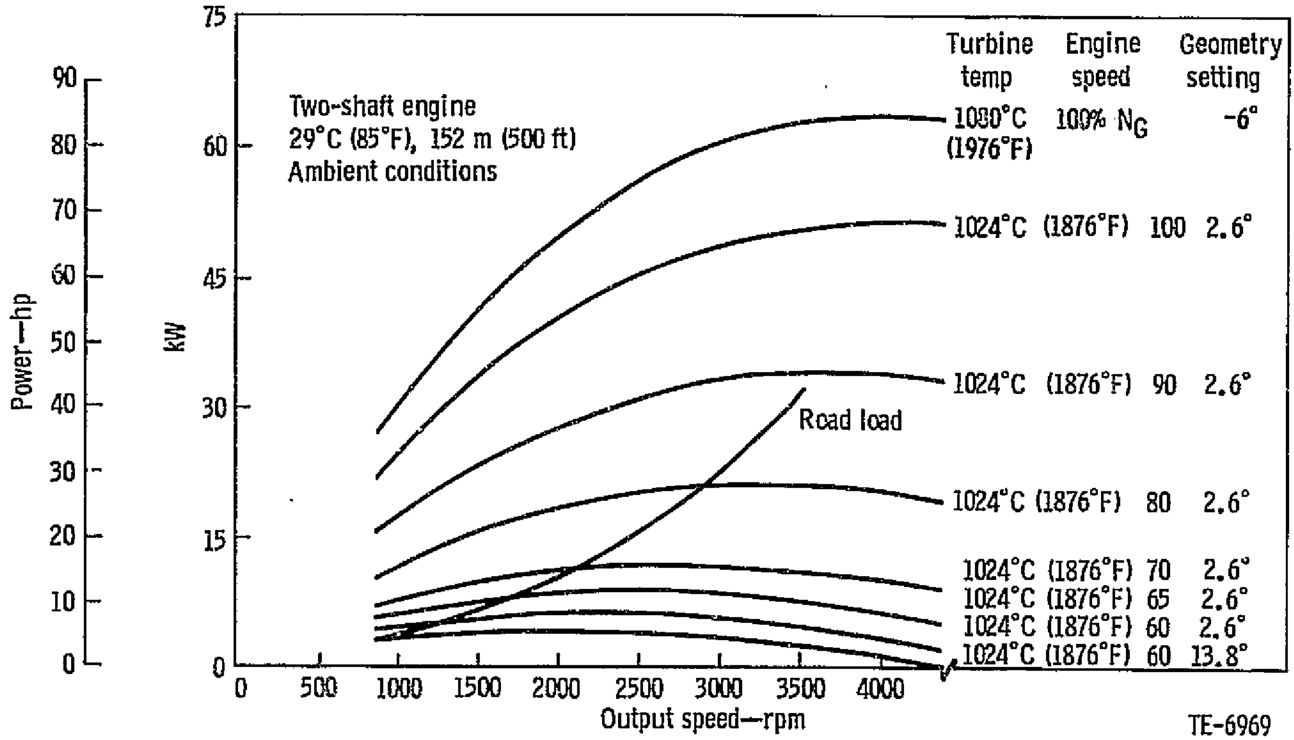


Figure 29. Two-shaft engine power map.

under the subsequent heading of Vehicle Fuel Economy because they relate to the engine and transmission match-up.

The control system for both the single- and two-shaft engines allows an increased turbine inlet temperature during compressor rotor system accelerations. This operating line is shown for the single-shaft engine only because it may last for 5 or more seconds—significantly longer than the typical 1-second period for the two-shaft engine. However, the vehicle fuel economy and performance calculations for the two-shaft engine accounted for the compressor rotor acceleration condition by use of a different map.

The output speed of the two-shaft engine was selected to be 4400 rpm at 100% power turbine speed in order to be within the speed and torque range of conventional automatic transmissions. For the single-shaft engine, the output speeds are given in percentages because the different CVT transmissions require different input speeds.

The road-load curve is shown in both the single-shaft and the two-shaft maps. The road load-curve is so positioned on the single-shaft engine that at any given speed at least 25% of the available power remains for engine and

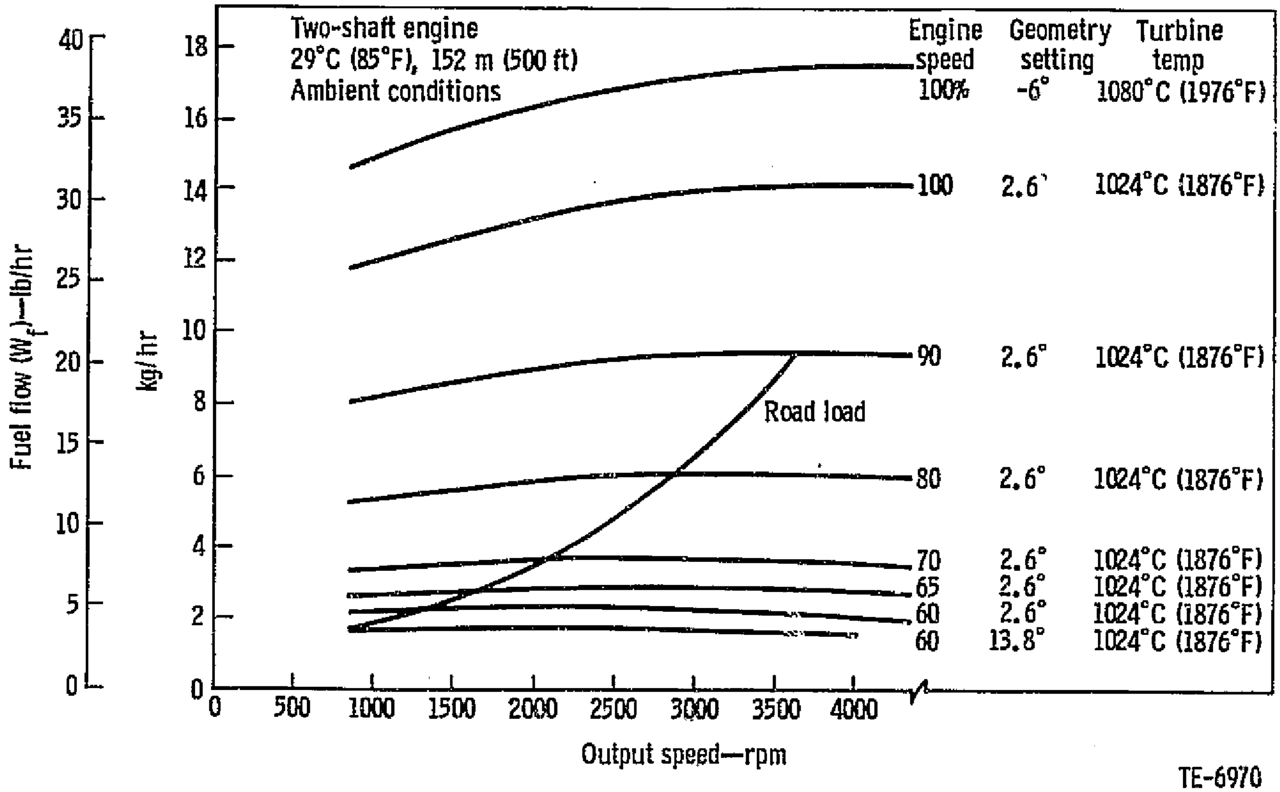


Figure 30. Two-shaft engine fuel map.

vehicle acceleration. This is necessary to provide the proper vehicle response in the first second and to reach cruising speed within a time that produces acceptable driveability. The minimum sfc plot (Figure 32) shows the best possible fuel consumption for typical single- and two-shaft engines. It is not always possible to operate the vehicle so that minimum sfc is attainable, and driving cycle fuel economy can be determined only by analysis of each driving cycle element.

A comparison of maximum torque characteristics is shown in Figure 33 for single-shaft and two-shaft engines. The single-shaft is so configured that a variable-ratio drive transmission is required. The two-shaft engine can use the conventional automatic with torque converter. One advantage of the two-shaft engine is that it would eliminate the necessity of a new transmission development program.

TRANSMISSION

A general search of data, reports, and engineering papers and a review of the GM Task Force effort expended in studying gas turbine powertrains leads to two general conclusions: (1) the conventional type of automotive auto-

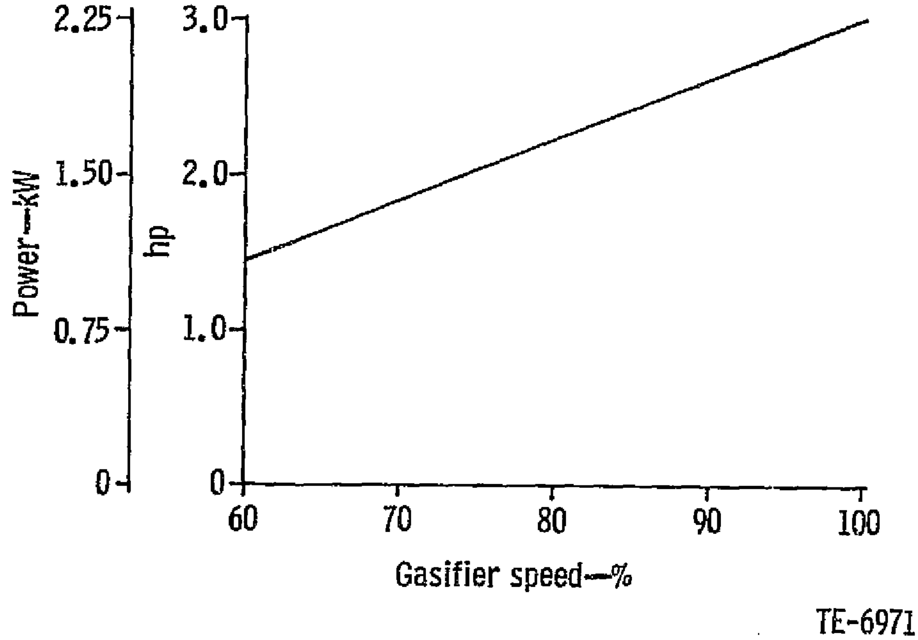


Figure 31. Engine accessory power.

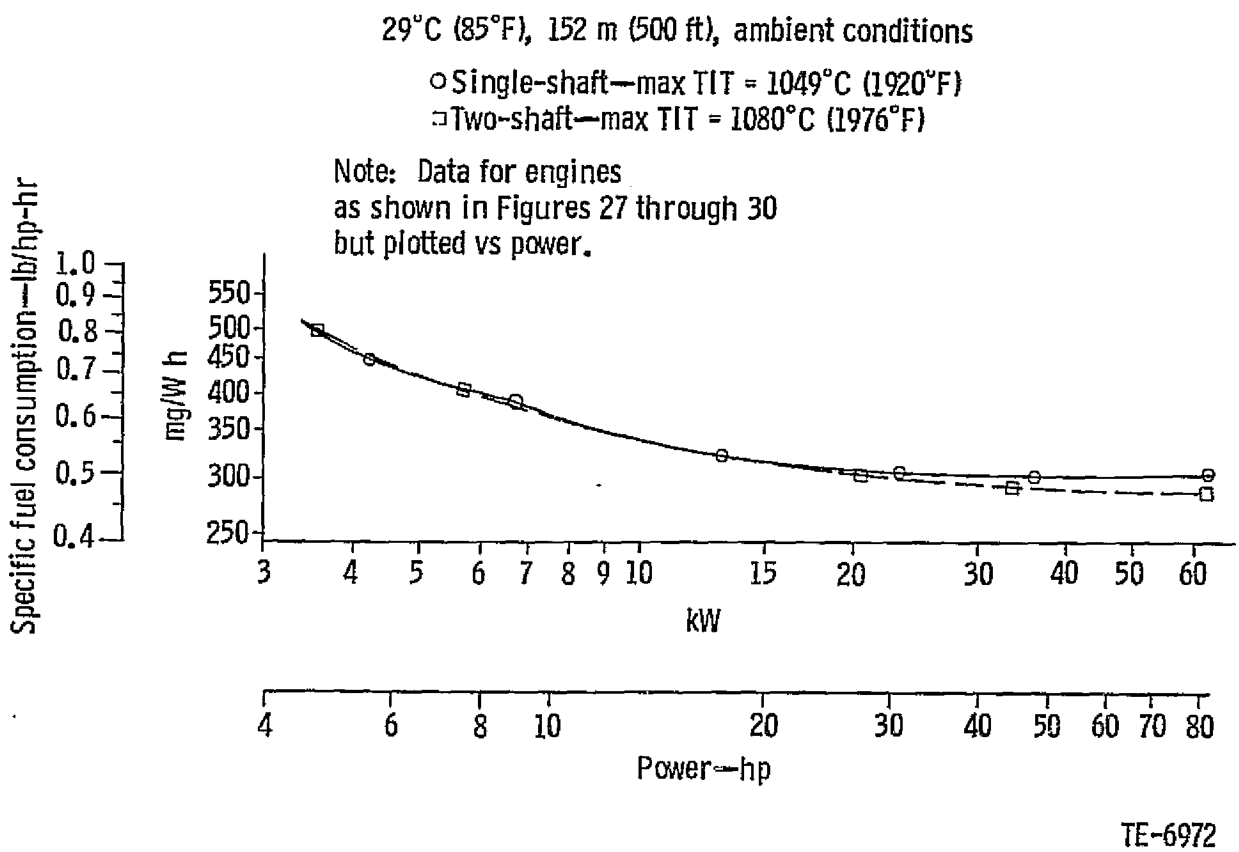


Figure 32. Comparison of one-shaft and two-shaft engines at minimum sfc.

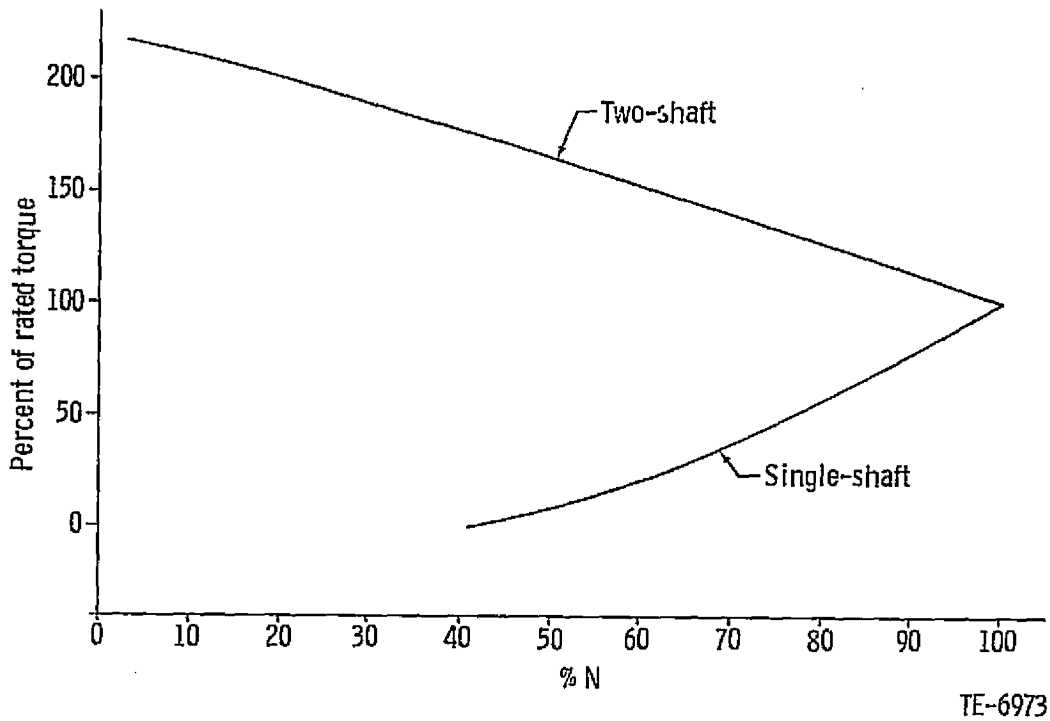


Figure 33. Torque characteristics of single- and two-shaft engines.

matic transmission is readily adaptable to the two-shaft gas turbine engine, and (2) the single-shaft or differential gas turbines require a transmission capable of continuously variable ratio over a wide range of vehicle speed operation.

A wide variety of continuously variable transmission (CVT) concepts were considered as described in Section IV of this report.

During this Task I analysis and search, the powertrains shown in Figure 34 were selected for further in-depth analysis and cross section development to establish fuel economy and performance characteristics, packaging arrangements, and detail configurations for cost analysis. The selection was sufficiently flexible to allow for configuration optimization--i.e., the single-shaft engine could also be matched to Toric or belt drive transmissions.

Arrangement Selections

A front-drive vehicle installation was selected for the gas turbine powertrain concept because the available compartment space presented the greatest challenge for both engine and transmission configuration arrangement. Concepts adapted to the front-drive installation can be readily rearranged from front to rear-drive installations. The differential and final drive reduction gearing is made integral with the transmission in the front-wheel-drive vehicle installation. The output drive shaft, in a transverse plane to the engine, is concentric with the transmission gearing and is coupled at each end of the transmission case to the drive wheels. The variable-drive mech-

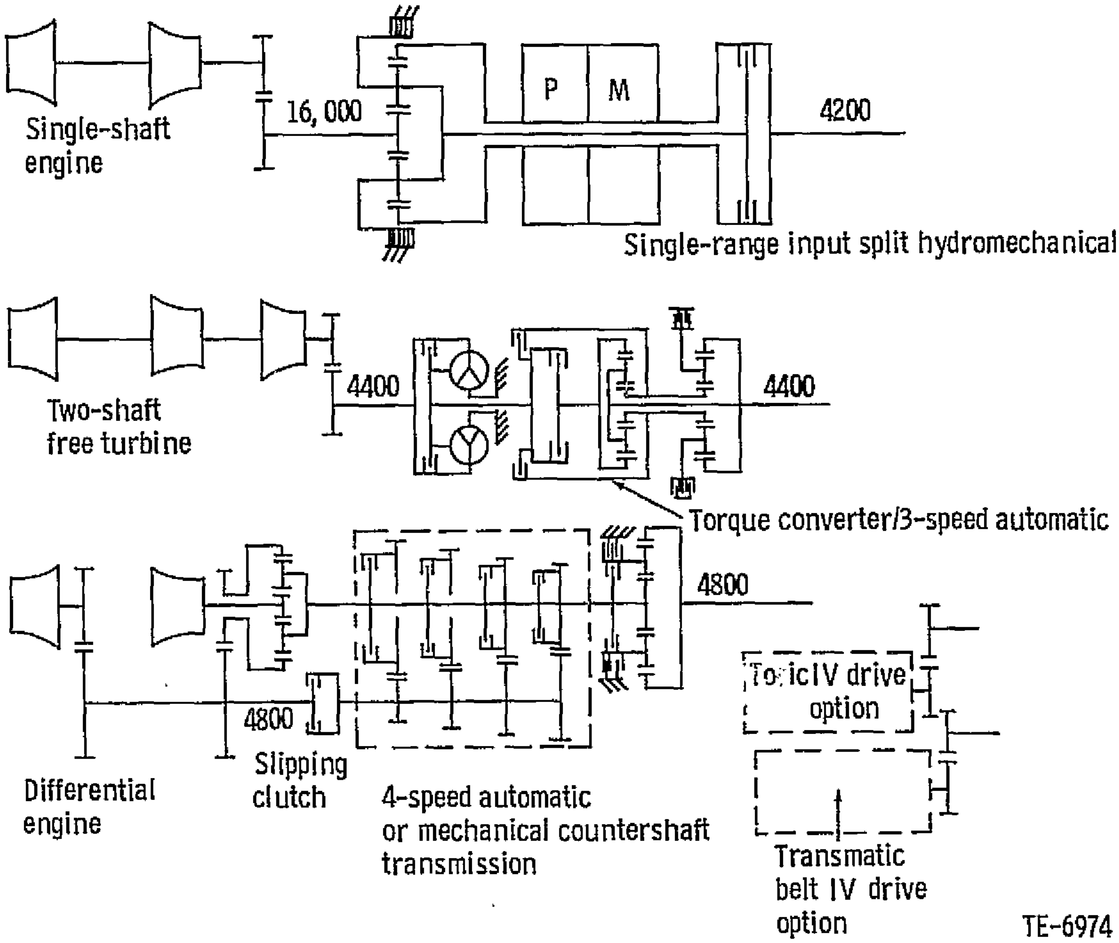


Figure 34. Transmission concepts for gas turbine engines.

anisms are concentric with the engine output, and a chain or belt drive transfers power to the transverse arrangement of the gear change mechanism.

Figure 35 shows the transverse transmission arrangement for a conventional torque converter automatic transmission. Figure 36 is a concept arrangement of a single-speed input split hydromechanical drive, where the IV hydrostatic drive is placed on the engine center line and, through chain drives, the mechanical and hydraulic power paths are combined at the cross-drive center line. Figure 37 is a transverse transmission arrangement of a variable-ratio steel belt drive with starting clutch, forward and reverse gearing, and clutch provision for extending the mechanical ratio coverage of the steel belt drive. Traction drives were studied but, within the space limitations of the front-wheel-drive installation, the housings to enclose the traction drive became too large when allowing for the cross-drive output shaft to pass through the center of the toroidal elements. Therefore, the traction drive for a front-wheel-drive installation was eliminated from consideration.

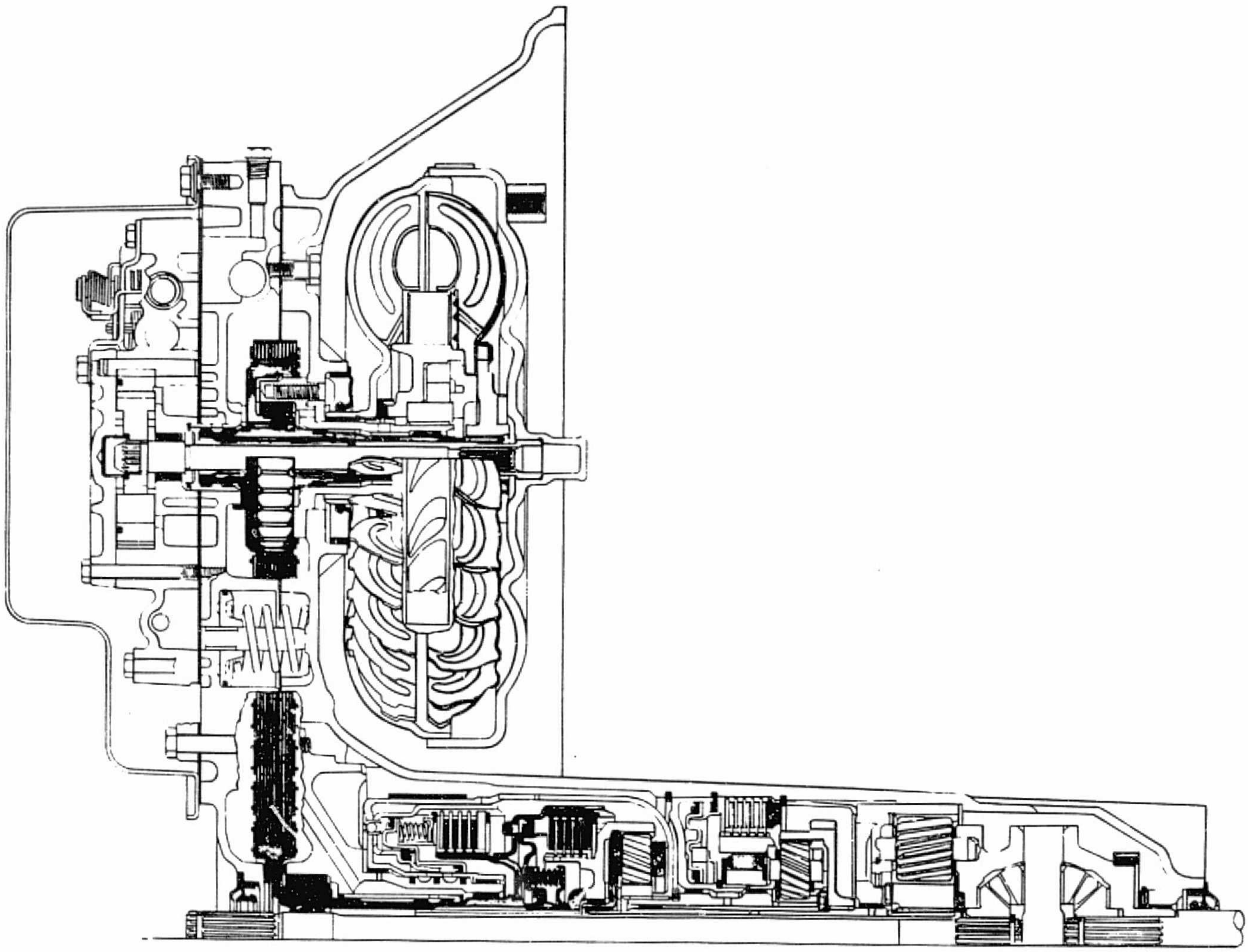


Figure 35. Three-speed automatic torque converter transmission.

Performance Studies

A conventional three-speed automotive planetary gear automatic transmission was selected for the two-shaft gas turbine engine. A steel belt variable drive with starting clutch and a single-range infinitely variable hydromechanical drive were selected for the single-shaft and differentially geared gas turbine engine performance studies. The overall efficiency of each of the transmissions selected is the combined efficiency of four component areas:

1. The charging pump (its flow and pressure) required to charge the converter and controls or position the variable drive mechanism
2. The operating efficiency of the torque converter or variable drive itself

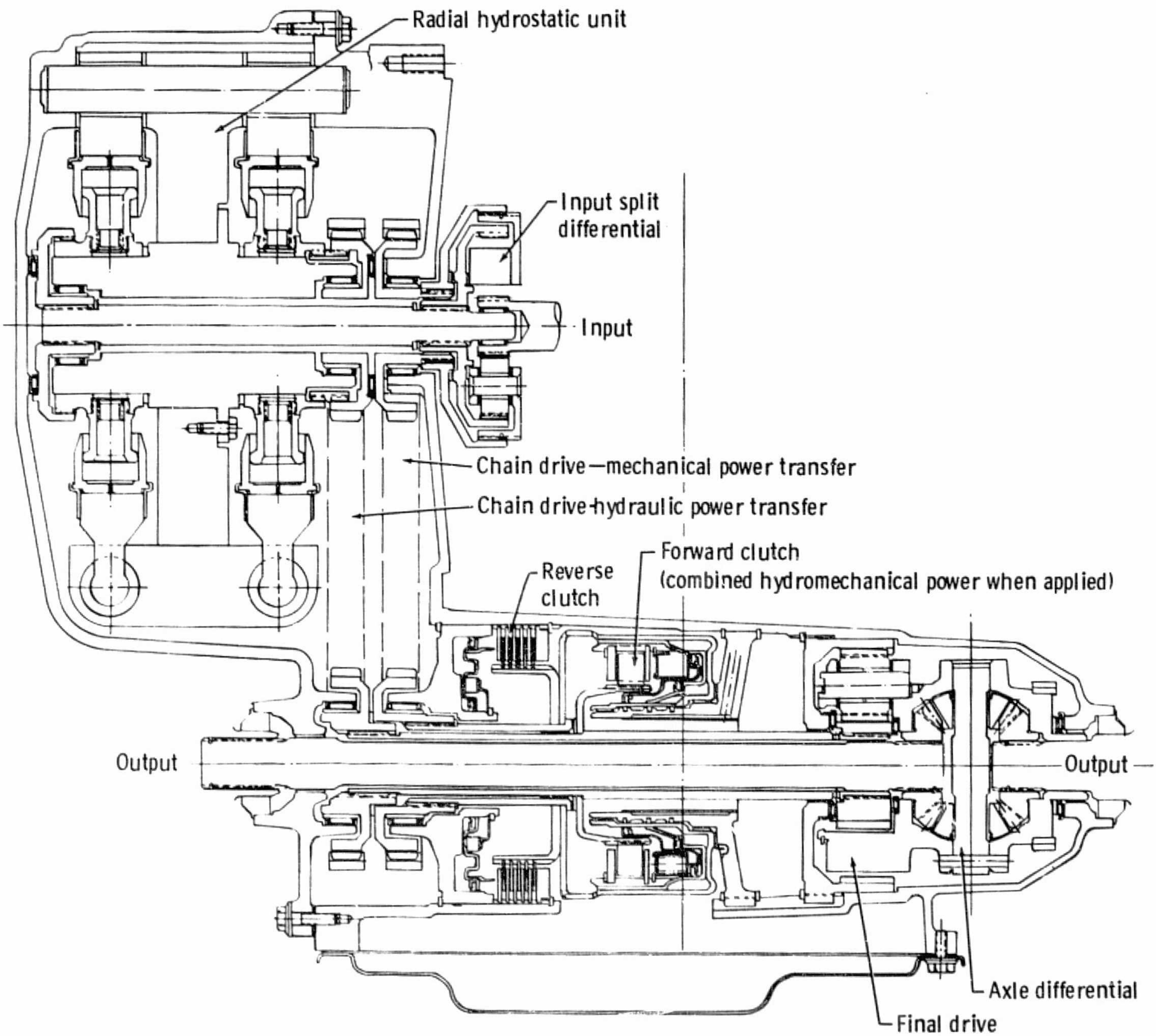


Figure 36. Single-range input split hydromechanical transmission.

3. Power flow loss through the gear change mechanism
4. An output efficiency accountable for the rotating clutches, shafts, and connecting drives

Typical performance of the two variable drives selected at 65% engine speed is shown in Figure 38. Performance data for the variable steel belt drive were obtained from the GM Technical Center, Warren, Michigan, where an actual automotive-type unit was recently tested. Performance data for the

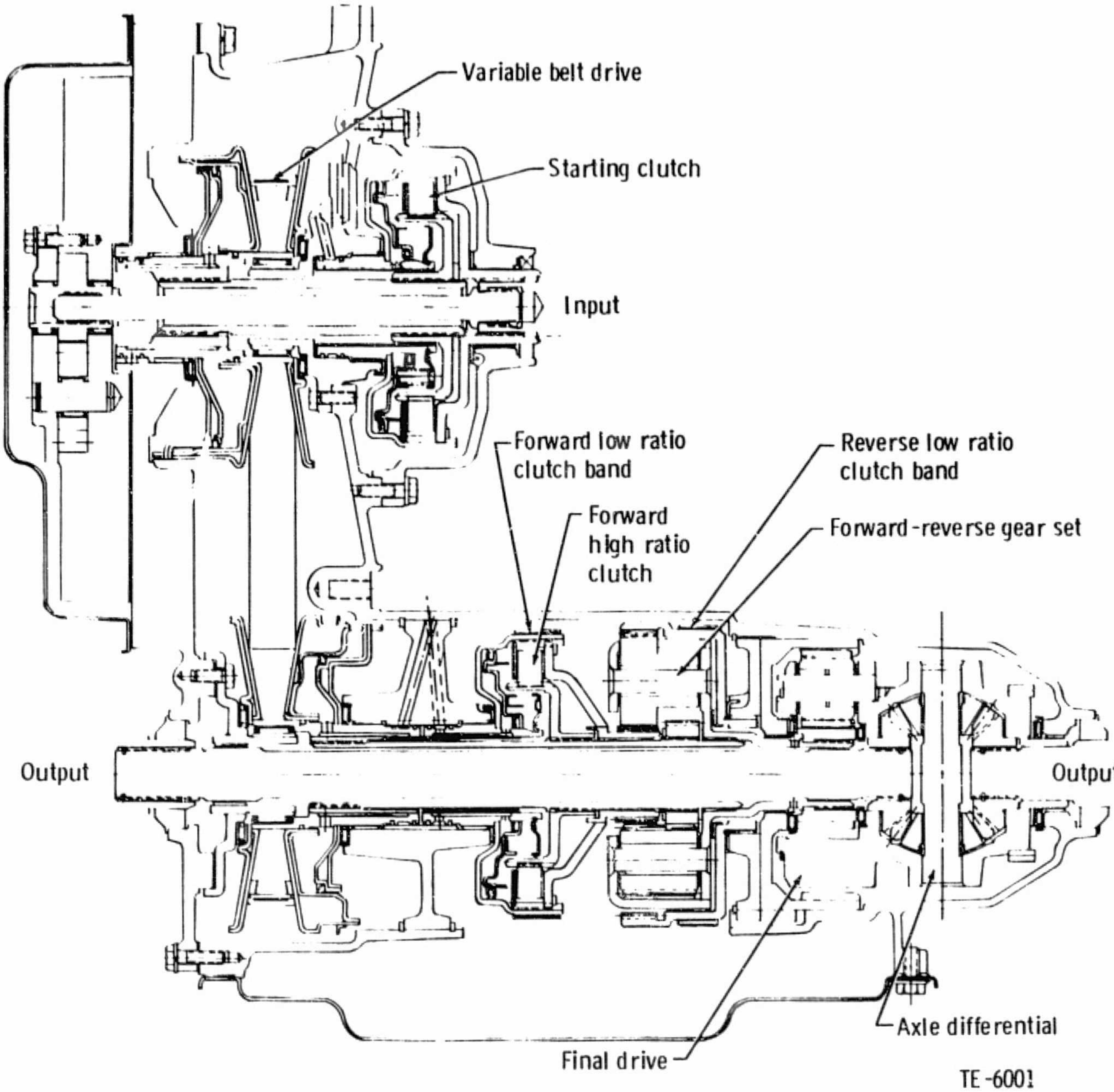


Figure 37. Variable-ratio steel belt transmission.

hydrostatic drive were also obtained from actual test data developed at DDA, Indianapolis, since 1955 in various military and commercial hydrostatic transmission development programs. Automotive and DDA heavy-duty transmission experience and test data were used in assessing the charging pump requirements and the pump, gear bearing, and spin losses. Figure 39 is a typical torque converter performance curve utilizing the factors K_T (ratio of turbine speed to the square root of turbine torque required for a given vehicle road load condition) and K_P (the ratio of the pump speed to the square root of the pump torque required at the same conditions) to establish the converter efficiency. A complete performance map of the trans-

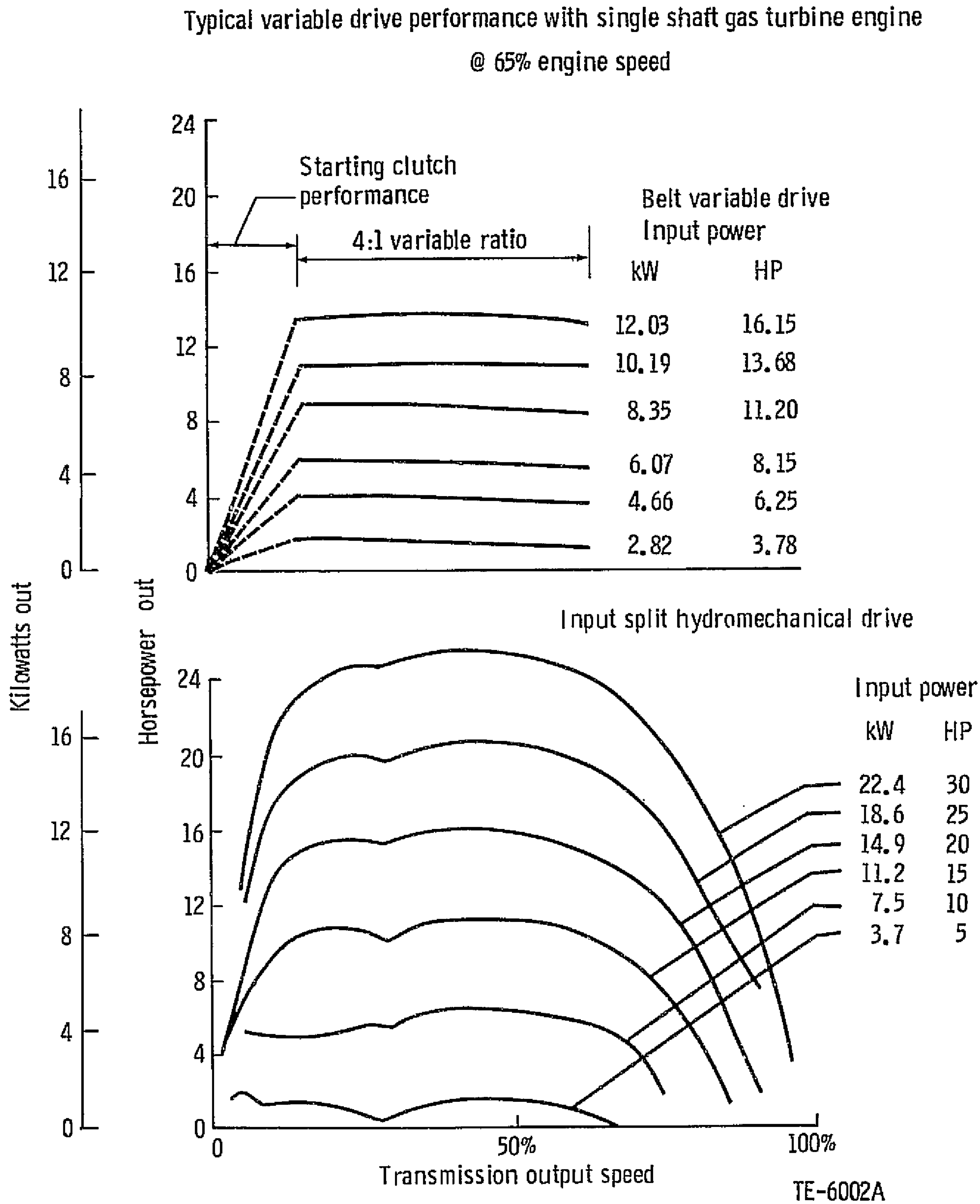


Figure 38. Typical variable-drive performance with single-shaft gas turbine engine at 65% engine speed.

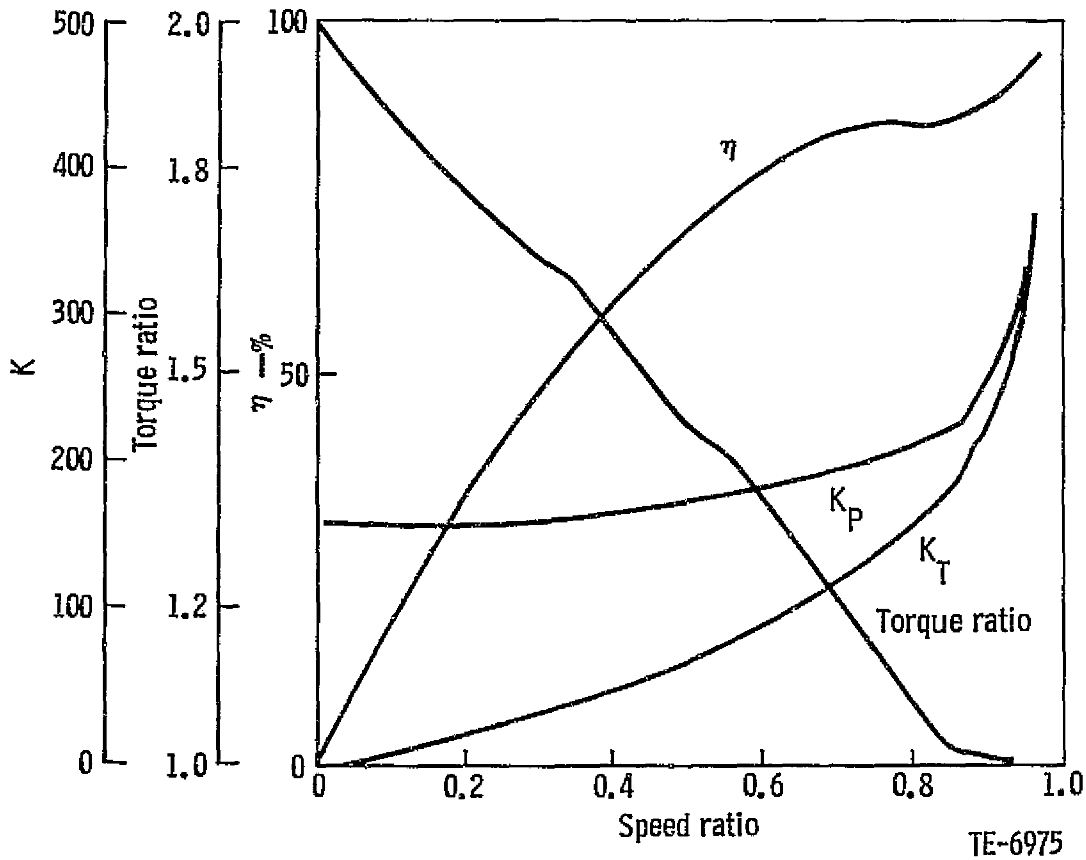


Figure 39. Typical torque converter performance.

missions at various engine speed and power settings (similar to Figure 38) was developed to describe the overall part-throttle to full-throttle transmission operation. These data were used in conjunction with the engine maps to calculate the vehicle fuel economy and performance data discussed later in this section.

The power required at the transmission input for the selected transmissions to meet typical output power requirements at road load conditions was developed and is summarized in Table XV. Figure 40 is a graphical representation of these data. Because of the high charging pump pressure required to position the hydrostatic reaction ring in the hydromechanical drive, the fixed pump losses at low road-load power requirements become a significant part of the total power required and, therefore, the efficiency falls drastically. Its superior efficiency led to the selection of the variable ratio belt drive over the single-range hydromechanical drive as the prime transmission candidate for use with the single-shaft gas turbine engine.

Transmission Losses Compared at Road Load

For comparison purposes, the transmission output power requirements at several road-load conditions were evaluated to compare the various detailed transmission losses. The results are summiarzed in Table XVI. The 32 km/h

TABLE XV. TRANSMISSION PERFORMANCE COMPARISON				
Vehicle speed, km/h (mph)	Transmission output power, kw (hp)	Transmission input power, kw (hp)		
		Single-range hydromechanical	Variable ratio steel belt	Three-speed conventional automatic
32 (20)	2.39 (3.20)	5.37 (7.20)	4.00 (5.36)	3.21 (4.31)
48 (30)	4.45 (5.97)	7.46 (10.00)	6.11 (8.20)	5.73 (7.68)
64 (40)	7.65 (10.26)	11.18 (15.00)	9.66 (12.96)	9.35 (12.54)
80 (50)	12.30 (16.50)	16.03 (21.50)	14.82 (19.88)	14.48 (19.42)
96 (60)	18.79 (25.20)	23.12 (31.00)	22.07 (29.60)	21.52 (29.86)

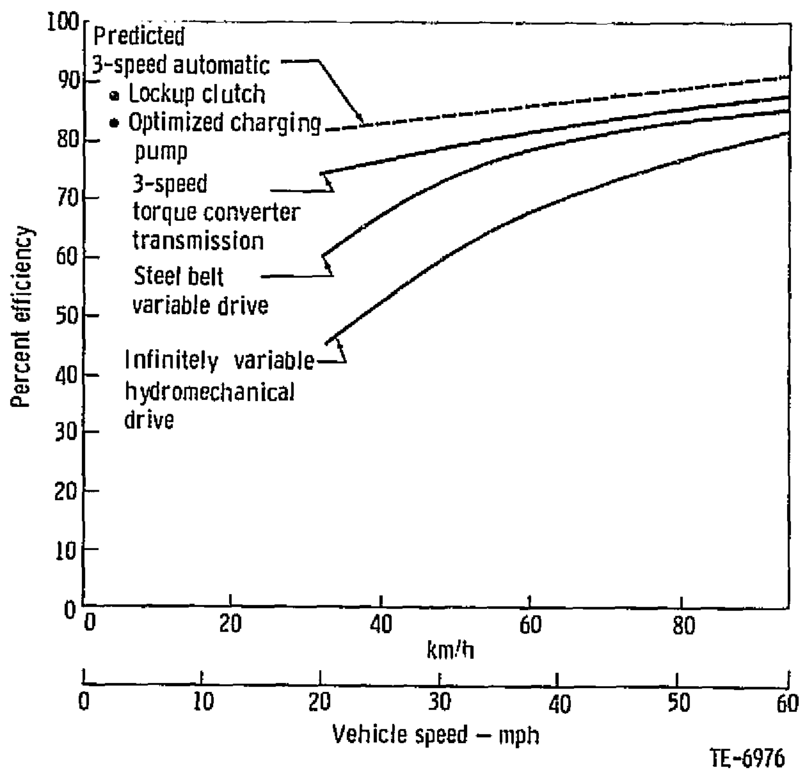


Figure 40. Transmission efficiency comparison at road load.

(20 mph) road-load requirement of 2.38 kW (3.2 hp) is examined in detail in the following paragraphs. This condition requires a transmission output speed (N_o) for the belt drive of 889 rpm (N_o = 4000/90 = 44.4) and 970 rpm for the torque converter drive (N_o/V = 4365/90 = 48.5). The conventional transmission is in third gear (1.00:1 drive) and, therefore, the above output speeds represent the speed at which the gear and clutch mechanism is rotating in each transmission. The transmission output power requirement is 2.4 kW (3.2 hp). For this speed condition, the output spin loss has been assessed: belt drive, 0.16 kW (0.22 hp); torque converter drive, 0.38 kW (0.51 hp). This loss is greater for the conventional

TABLE XVI. DETAILED TRANSMISSION LOSS COMPARISON STEEL BELT CVT VS 3-SPEED AUTOMATIC						
Vehicle speed, km/h (mph)		32.2 (20)	48.3 (30)	64.4 (40)	80.5 (50)	96.6 (60)
Transmission output power, kW (hp)		2.38 (3.20)	4.45 (5.97)	7.65 (10.26)	12.30 (16.50)	18.78 (25.20)
Transmission output speed, rpm	CVT	889	1333	1777	2222	2667
	3-speed	970	1455	1940	2425	2910
Output spin loss, kW (hp)	CVT	0.16 (0.22)	0.15 (0.33)	0.33 (0.44)	0.41 (0.55)	0.49 (0.66)
	3-speed	0.38 (0.51)	0.63 (0.85)	0.77 (1.03)	0.89 (1.20)	1.00 (1.34)
Gear mesh loss, kW (hp)	CVT	0.04 (0.05)	0.07 (0.10)	0.12 (0.16)	0.19 (0.26)	0.29 (0.39)
	3-speed	---	---	---	---	---
Power into gearing, kW (hp)	CVT	2.59 (3.47)	4.77 (6.40)	8.09 (10.86)	12.90 (17.31)	19.56 (26.25)
	3-speed	2.76 (3.71)	5.08 (6.82)	8.41 (11.29)	13.19 (17.70)	19.78 (26.54)
Variable drive efficiency, %	CVT	86.0	92.0	93.5	95.0	96.0
	3-speed	95.0	97.3	98.0	98.0	98.0
Power into variable drive, kW (hp)	CVT	3.01 (4.04)	5.19 (6.96)	8.66 (11.62)	13.58 (18.22)	20.37 (27.34)
	3-speed	2.91 (3.90)	5.22 (7.01)	8.58 (11.52)	13.46 (18.06)	20.18 (27.08)
Transmission input speed, rpm	CVT	1300	1300	1500	1600	1700
	3-speed	1012	1496	1980	2508	2992
Charging pump loss, kW (hp)	CVT	0.97 (1.30)	0.97 (1.30)	1.12 (1.50)	1.19 (1.60)	1.27 (1.70)
	3-speed	0.31 (0.42)	0.50 (0.67)	0.76 (1.02)	1.01 (1.36)	1.33 (1.78)
Transmission input power required, kW (hp)	CVT	3.98 (5.34)	6.23 (8.36)	9.78 (13.12)	14.77 (19.82)	21.64 (29.04)
	3-speed	3.22 (4.32)	5.72 (7.68)	9.34 (12.54)	14.47 (19.42)	21.51 (28.86)

transmission because of the increased number of planet gear sets and clutches required. Next, the power loss through the gear train is assessed (1 1/2% or 0.04 kW (0.05 hp) for the belt drive; no gear loss is assessed for the locked-up third gear torque converter drive). The resulting horsepower required into the gear change mechanism to develop the 2.4 kW (3.2 hp) output requirement is 2.59 kW (3.47 hp) for the belt drive and 2.76 kW (3.7 hp) for the torque converter drive.

Maximum engine input speed to the transmission is 4400 rpm for the conventional transmission and 20,000 rpm for the variable-ratio belt drive. The required reduction ratio for the variable drive unit is 1.46 to 1, assuming an engine speed of 65% or 1300 rpm coupled with the 889-rpm transmission output speed requirement. An efficiency of 86% for the variable drive was obtained from the steel belt drive data at the required output conditions of 889 rpm, 2.59 kW (3.47 hp), and 1.46 to 1 reduction ratio. The torque converter efficiency of 95% was derived from the torque converter absorption characteristics at the required output conditions of 970 rpm and 2.762 kW (3.706 hp). The torque converter factor, K_T, is 216 at these conditions. Therefore, the required power into the variable drive to develop the 2.4 kW (3.2 hp) transmission output requirement is 3.01 kW (3.90 hp) at 1021 rpm (970/0.95) for the converter drive. Reviewing the power into the variable drive required to develop the various output powers at road load as shown in Table XVI, it is apparent that the requirements are essentially the same as for the torque converter drive.

However, when the charging pump loss is included in the overall efficiency analysis, the variable belt drive as a complete transmission system is less efficient than a comparable torque converter drive system at these road-load conditions. Assessment of the mechanical power required to drive the charging pump for a CVT steel belt drive was based on THM 350 automatic transmission pump loss data. In the two-shaft gas turbine/conventional automatic transmission, the charging pump speed varies from approximately 400 rpm at idle to 4400 rpm maximum. The transmission mechanical component speeds (gears, bearings, and clutches) are a function of the transmission output speed ranging from 0 to 4400 rpm maximum. In the CVT/single shaft gas turbine application, the charging pump speed ranges from 1200 rpm (60% engine speed) to 2000 rpm (100% engine speed). With a 2:1 underdrive to 0.5:1 overdrive variable drive capability, the speed of the transmission components (driven belt sheave, forward/reverse gearing and clutches, additional ratio gearing if required) ranges from 0 to 4000 rpm maximum. The required input slipping clutch speed varies from 0 to 2000 rpm. The system pressure schedule to maintain position of the belt sheaves varies from 1379 to 2413 kPa (200 to 350 psi), depending on belt speed ratio. At the present state of the art, transmission flow requirements in the 1000-2500 rpm output speed range of 32-80 km/h (20-50 mph) should be approximately the same between the conventional automatic transmission and CVT belt drive because of leakage, lube, and cooling requirements. Over this output speed range in 1:1 gear drive, the conventional transmission charging pump is varying from 1000 to 2500 rpm at 414 kPa (60 psi) charging pressure. Over this same speed range, the charging pump for the CVT transmission is varying from 1200 to 1700 rpm at a varying pressure schedule from 1379 kPa (200 psi) minimum. THM 350 charging pump data indicate an approximate 5.8 N m (4.3 ft-lb) torque loss over a speed range of 1000 to 2000 rpm pump speed and 1034 kPa (1500 psi) charging pressure. This same pump flow at 1379 kPa (200 psi) charging pressure is estimated to require a 7.0 N m (5.2 ft-lb) input driving torque. For this study, a constant torque loss of 7.0 N m (5.2 ft-lb) was used to drive the CVT belt drive charging pump loss over the 1200 to 1700 rpm input speed range for road-load conditions. The resulting charging pump horsepower loss versus percent of maximum engine input speed to the transmission is shown plotted in Figure 41.

Thus, the total power required into the transmission system to develop the road-load output power requirement of 2.4 kW (3.2 hp) is 3.98 kW (5.34 hp) or 60% efficiency for the variable belt drive and 3.22 kW (4.32 hp) or 74% efficiency for the torque converter drive. As the power requirements are increased, the transmission parasitic losses become a smaller percentage of the total power, and two types of transmissions indicate nearly the same performance.

Projected Transmission Efficiency Improvements

It is feasible that a variable high-pressure-capacity pump can be developed for the steel belt variable drive that would vary the charging pressure as a function of speed ratio and/or applied load to minimize the pump parasitic loss at low road-load power requirements. To further optimize the conventional torque converter transmission, a lockup clutch and variable displacement pump is proposed. The resulting improvement in transmission efficiency when these changes are incorporated is shown in Figure 42 as an efficiency band for each of the two transmissions.

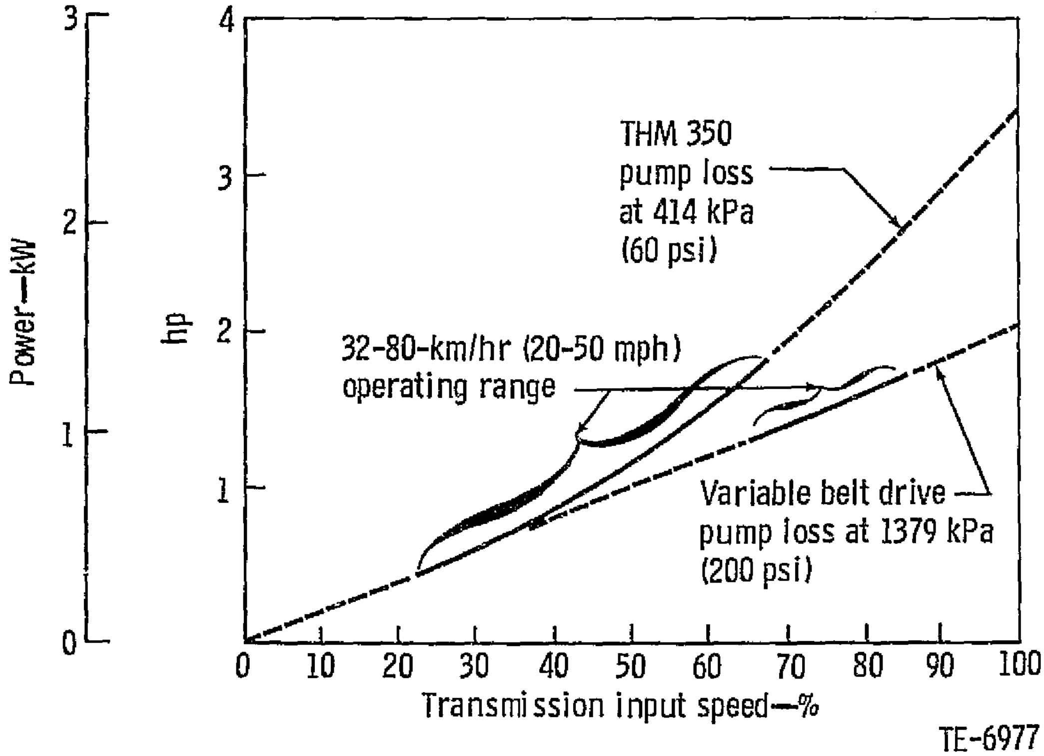


Figure 41. Transmission charging pump loss comparison.

Conclusions

As a result of evaluating four types of transmission drives applicable to gas turbine engines (namely, (1) conventional three-speed automotive type torque converter drive, (2) steel belt variable ratio drive, (3) input split hydromechanical drive, and (4) Toric traction drive), the following conclusions are offered:

- o Steel belt variable drive and hydromechanical drive are adaptable to a transverse mounted front-drive installation.
- o Toric traction drive is not suitable for front-drive installations.
- o Efficiency variation between transmission concepts is primarily a function of charging pump requirements.
- o Conventional three-speed automotive transmission has the highest efficiency for road-load conditions.
- o Further increase in conventional transmission efficiency is obtainable with a torque converter lockup clutch and optimized charging pump.
- o Further increase in variable ratio transmission efficiency is feasible with an optimized charging pump.

VEHICLE FUEL ECONOMY

Fuel economy over the EPA composite driving cycle was computed, using the GPSIM verified simplified method. Engine off-design horsepower and fuel maps as well as transmission efficiency characteristics and vehicle accessory loads were inputs for this procedure. The resulting overall vehicle

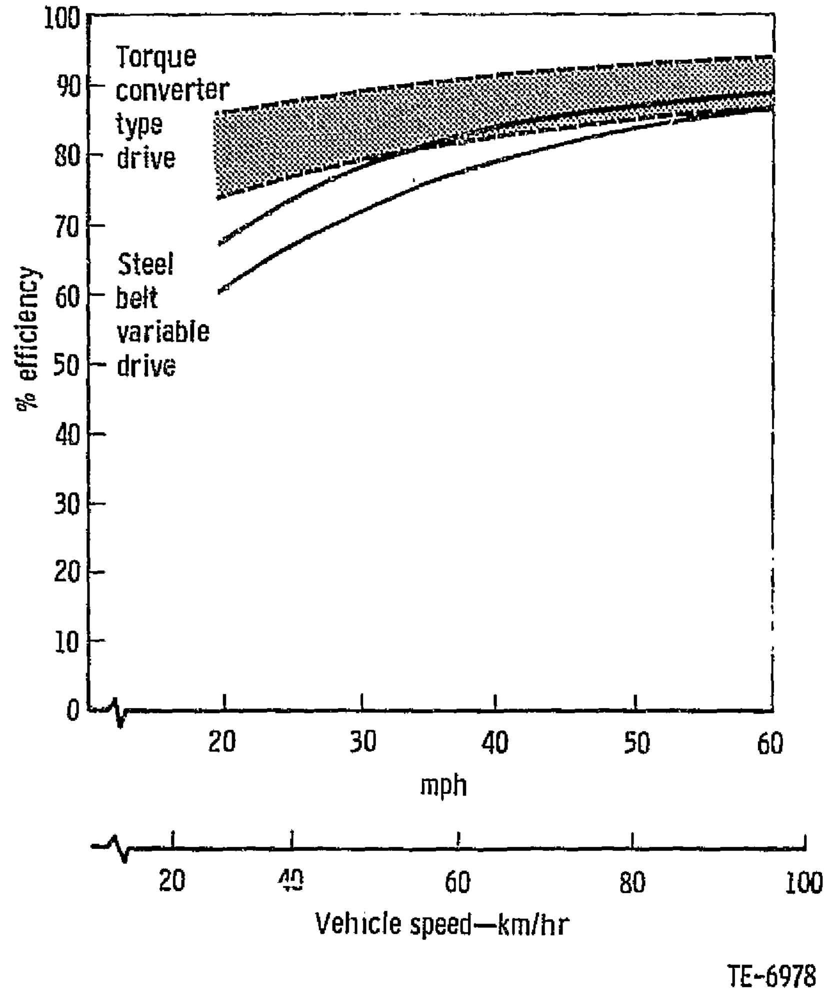


Figure 42. Transmission efficiency range comparison at road load.

fuel economy was the payoff parameter for comparing the various powertrains regarding their inherent ability to deliver maximum improvement potential. For all fuel economy analyses, gasoline equivalent fuel was used. Vehicle accessories include the power steering pump, alternator, and oil cooler fan. The power steering pump is unloaded (straight-ahead driving), the alternator providing sufficient output to balance the engine and vehicle electrical load with a fully charged battery. Total vehicle accessory power versus engine speed is shown in Figure 43. (Engine accessory power was shown in Figure 31.)

Note that the operating spoeed, in percent, for the single-shaft engine is considerably higher than that for the two-shaft engine as described earlier.

In addition, some of the powertrains were evaluated on a road-load fuel economy basis. This screening technique allowed certain of the concepts to be selected over others without the necessity of the more complex driving cycle analysis.

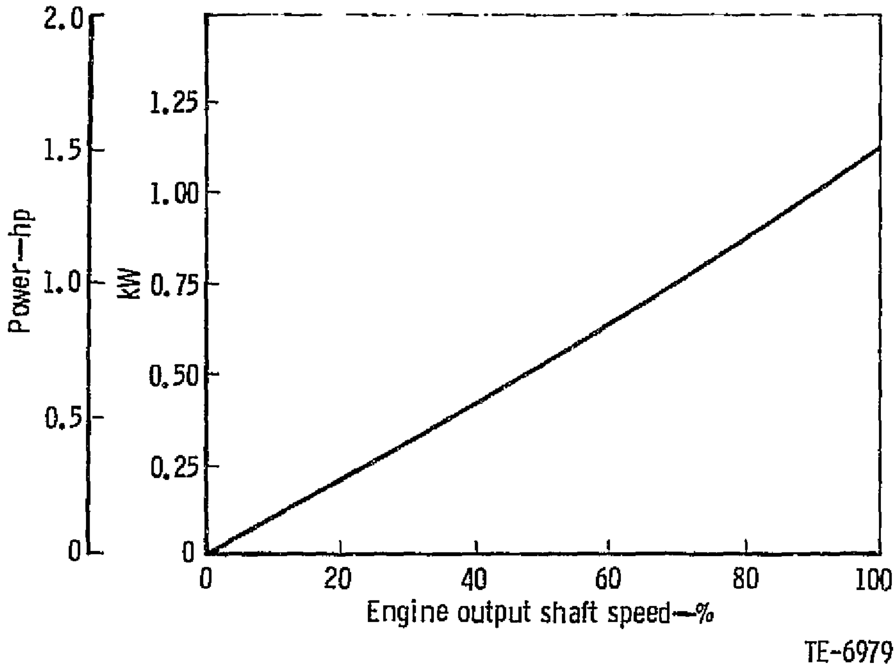


Figure 43. Vehicle accessory power requirements.

The idle fuel flow of the different powertrains is dependent on the transmission and accessory loads at the selected idle speed and the engine fuel flow versus power characteristics at the idle throttle setting. Table XVII is a listing of fuel flow, power, and speed at the idle condition of the various powertrains.

Single-Shaft Engine: Belt Versus Hydromechanical CVT

The single-shaft engine with radial turbine was studied with two continuously variable transmissions types. The steady-state road-load vehicle fuel economy for these two systems is shown in Figure 44. At 40 km/h (25 mph) the belt drive is 13% better on fuel economy. The 40-km/h (25 mph) point was selected because that is near the average road speed for the combined driving cycles. All-vehicle-speeds fuel economy with the belt CVT was consistently and significantly superior to the hydromechanical CVT. Based on this comparison, the hydromechanical CVT was no longer considered in the study.

Single-Shaft Engine: Radial Versus Reentry Turbine

The single-shaft engine was also studied with two types of turbines: a radial inflow and a reentry axial type. As explained in Section III of this report, the reentry turbine operates at one-half the compressor speed so that a reduction gear is required between the compressor and the turbine. This additional gear loss plus two additional bearing losses have a significant negative impact on engine sfc. Figure 45 is a comparison of the road-load fuel economy of the single-shaft engine with a radial inflow turbine versus an axial reentry type. Both were analyzed, using the belt transmission. The average speed for a vehicle on the combined driving cycle

TABLE XVII. FUEL FLOW, POWER, AND OUTPUT SPEED AT IDLE CONDITIONS.			
	Output speed, %	Power*, kW (hp)	Fuel flow, kg/h (lb/hr)
Single-shaft engine			
Hydromechanical transmission			
Radial turbine	60	3.7 (5.0)	2.0 (4.5)
Reentry turbine	60	3.7 (5.0)	1.9 (4.2)
Belt transmission			
Radial turbine	60	2.6 (3.5)	1.7 (3.8)
Reentry turbine	60	2.6 (3.5)	1.6 (3.5)
Two-shaft engine			
3-speed automatic transmission	10	1.6 (2.1)	1.4 (3.2)
Belt transmission	10	2.1 (2.8)	1.7 (3.7)
Differential engine	60**	NA***	2.0 (4.4)

*Power is the sum of the transmission input power plus the vehicle accessories.
**Speed for the differential engine is compressor speed because the output shaft is stalled.
***Engine powr has no meaning in this case because the output shaft is stalled.

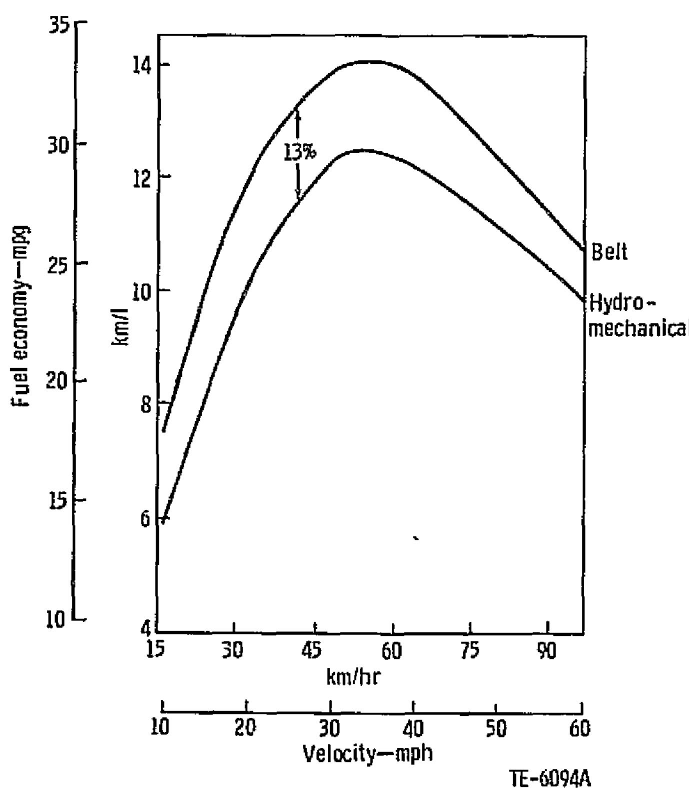


Figure 44. Road load fuel economy comparison, single-shaft engine--steel belt versus hydromechanical transmission.

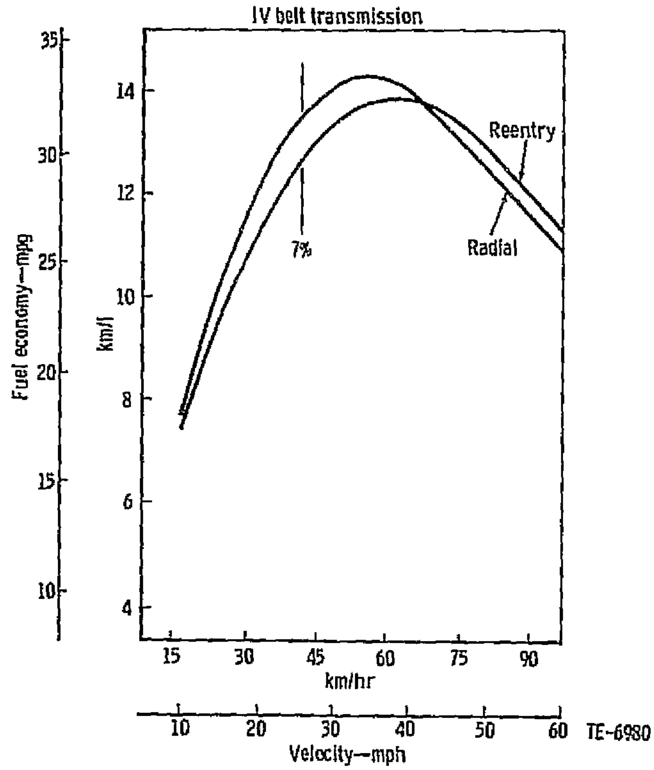


Figure 45. Road load fuel economy comparison, single-shaft engine--radial versus reentry turbine.

is approximately 40 km/h (25 mph). At this condition the radial turbine is 7% better. Because it has no performance advantage and introduces a higher risk, the reentry turbine was no longer considered as a candidate.

Two-Shaft Engine: Three-Speed Automatic Transmission Versus Belt CVT

A conventional three-speed transmission adapts easily to the torque characteristics of a two-shaft engine and positions the road-load curve on the engine performance map very close to the best sfc operating line. However, a continuously variable transmission offers total flexibility in matching the two-shaft engine to the vehicle power requirements. Consequently, a comparison of the three-speed automatic transmission and the belt drive CVT was made to determine which total system was best. Steady-state road load economy is shown in Figure 46. The conventional transmission is superior (at road-load conditions) by such a clear margin that its superiority over the combined driving cycle can be confidently predicted. Consequently, the two-shaft studies wre continued with a three-speed automatic transmission only.

Comparison of Optimum Two-Shaft, Single-Shaft, and Differential Powertrains

One of the candidates selected from Task I was the differential powertrain. The differentially geared engine requires a considerable amount of high-speed gearing, much of which transmits full-throttle output power and compressor load. It also requires power feedback between the output shaft

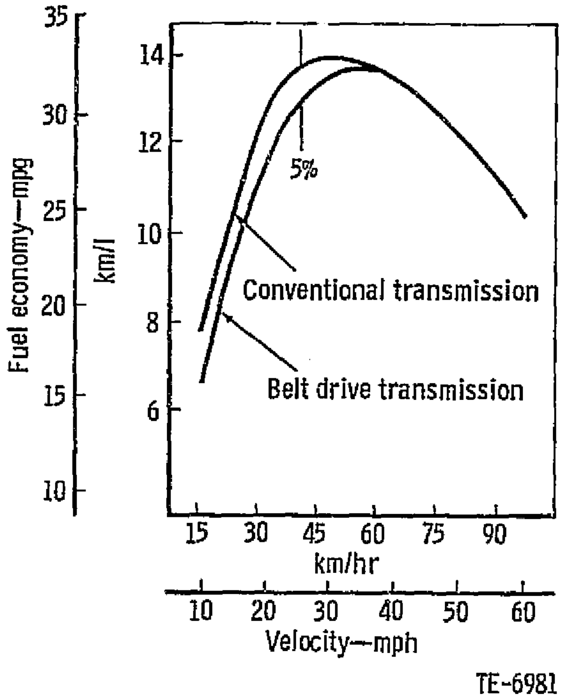


Figure 46. Road load fuel economy comparison, two-shaft engine--conventional three-speed automatic versus steel belt CVT transmission.

and the turbine shaft for turbine inlet temperature control. Because of its lower losses, a variable belt drive was selected to perform the temperature control function. Road-load fuel economy of the differential powertrain, shown in Figure 46, is significantly poorer than that of the other candidates because of the mechanical losses associated with the high-speed gearing which transmit high levels of horsepower. At the average velocity for the composite driving cycle, the differential powertrain had a fuel economy that was 28% below that of the other engine configurations. Because the differential engine is a poor fuel economy competitor, it was dropped from further study plans.

The remaining two powertrain candidates are (1) a single-shaft engine with a belt CVT transmission and (2) a two-shaft engine with conventional automatic transmission. On the basis of road-load fuel economy, these two are very close, as shown in Figure 47. In fact, they are too similar to permit an estimate of which would yield the best fuel economy over a driving cycle, as evidenced by the following fuel economy comparison based on a simplified composite driving cycle:

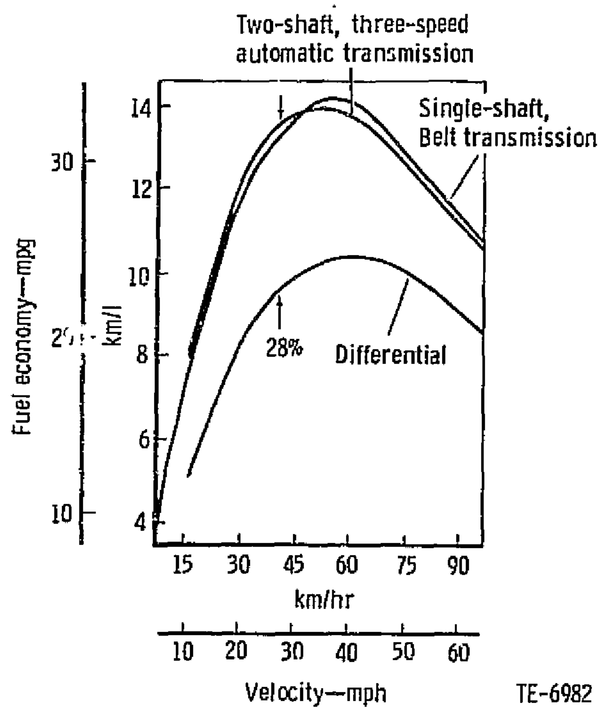


Figure 47. Road load fuel economy comparison of differential, two-shaft, and single-shaft powertrains.

	Fuel economy, km/l (mpg) (29°C (85°F), 152 m (500 ft) ambient conditions)
Base line	8.3 (19.6)
Goal	10.0 (23.5)
Single-shaft engine, CVT	8.6 (20.3)
Two-shaft engine, conventional transmission	8.8 (20.7)

Metal turbine rotors/twin ceramic regenerator disk assumed for both powertrains.

Although the two-shaft engine is slightly better, it does not exhibit a statistically significant advantage. Of greater importance is the fact that neither of these metal turbine rotor candidates meets the goal of 20% improvement over the base-line 1976 spark-ignition engine-powered vehicle of 10.0 km/l (23.5 mpg). Further, increases in component efficiencies, requiring significant improvements in technology, would be needed to meet the goal. As a result, the concept that is selected will require a ceramic rotor to operate at higher turbine temperature levels.

VEHICLE PERFORMANCE (ACCELERATION)

Vehicle performance parameters of initial response to throttle movement and time to accelerate from 0 to 97 km/h (0-60 mph) were evaluated. Comparisons of the single-shaft gas turbine engine with a variable steel belt drive transmission and a two-shaft gas turbine engine with a conventional three-speed automatic torque converter transmission were made relative to the 1976 conventional automotive power train consisting of a piston engine coupled to a three-speed automatic torque converter transmission. A 1542-kg (3400 lb) test weight vehicle was used in the acceleration studies. The axle ratio of 2.70 to 1 for the spark-ignition engine was chosen to optimize fuel economy. This results in a "geared" top vehicle speed of 192 km/h (119 mph) although the powertrain produces only enough power to drive the vehicle to approximately 153 km/h (95 mph). To minimize the transmission ratio coverage required, the gas turbine-powered vehicle was "geared" for a top speed of 145 km/h (90 mph). This decision was made in conjunction with input from Pontiac Motor Division. The axle ratio for the two-shaft engine with three-speed automatic transmission is 3.70 to 1, which corresponds to a maximum transmission output speed of 4400 rpm. For the single-shaft engine with the belt CVT transmission, the axle ratio is 3.36, which corresponds to a maximum transmission output speed of 4000 rpm.

The single-shaft engine vehicle accelerations from 0 to 97 km/h (0-60 mph) were studied. Idle speed was set at 60%, 65%, and 70% of maximum engine speed. The two-shaft engine idle was set at 60% gasifier speed. With the two-shaft gas turbine engine and torque converter transmission, the rate of vehicle acceleration is determined by the power turbine polar moment of inertia and the gasifier response time, which is approximately 1 sec. In the case of the single-shaft engine, the variable-ratio belt was programmed to accelerate the engine at a constant rate of 4.2% per second from idle to full speed. The results of studies to optimize the initial vehicle response by setting the acceleration rate at 3.1 and 2.1% per second are given in Table XVIII. The initial vehicle response characteristics of the single-shaft gas turbine engine and variable steel belt drive transmission are unacceptable with a variable drive ratio coverage of 4:1; therefore, an additional 1.8 to 1 reduction ratio was obtained by modifying the forward-reverse gearing with the addition of a lockup clutch. This configuration gave an overall variable drive ratio of 7.2:1. This additional torque ratio coverage approximately doubled the distance the vehicle traveled in the first second. However, the torque characteristics of the single-shaft engine combined with a high-mechanical-ratio variable-drive transmission do not provide sufficient torque to the wheels to indicate good response when compared with the torque multiplying characteristics of the two-shaft gas turbine engine and conventional three-speed torque converter transmission. With the two-shaft gas turbine powertrain, the initial response is such that this vehicle installation lags the piston engine vehicle by only 1.5 m (5 ft) in the first three seconds (Figure 48), and as both vehicles reach 97 km/h (60 mph) in the same time frame, the gas turbine powertrain has closed the gap to approximately 0.76 m (2.5 ft). Initial response can be further improved by increasing the available low gear mechanical ratio of the conventional transmission.

TABLE XVIII. WIDE-OPEN THROTTLE VEHICLE ACCELERATION COMPARISON: BASE-LINE WITH SINGLE- AND TWO-SHAFT ENGINES						
Configuration	Transmission	CVT acceleration rate, %/sec	Idle speed, %	Distance traveled in 1 sec m (ft)	Vehicle speed in 1st second, km/h (mph)	Acceleration time for 0-97 km/h (0-60 mph), sec
Base-line vehicle	3-speed automatic, 2.52 low gear ratio	---	---	1.8 (5.9)	14 (8.8)	15.6
Single-shaft engine	Variable belt CVT, 4:1 ratio range	4.2 4.2 4.2	60 65 70	0.18 (0.6) 0.30 (1.0) 0.52 (1.7)	1.6 (1.0) 2.6 (1.6) 4.2 (2.6)	15.54 15.01 14.53
	Variable belt CVT, 7.2:1 ratio range	4.2 4.2 4.2 3.1 2.1	60 65 70 60 60	0.37 (1.2) 0.58 (1.9) 0.91 (3.0) 0.43 (1.4) 0.52 (1.7)	3.1 (1.9) 4.8 (3.0) 2.4 (4.6) 3.4 (2.1) 3.9 (2.4)	14.56 14.02 13.51 15.29 16.76
Two-shaft engine	3-speed automatic, 2.52 low gear ratio	---	60	1.04 (3.4)	10.3(6.4)	15.40

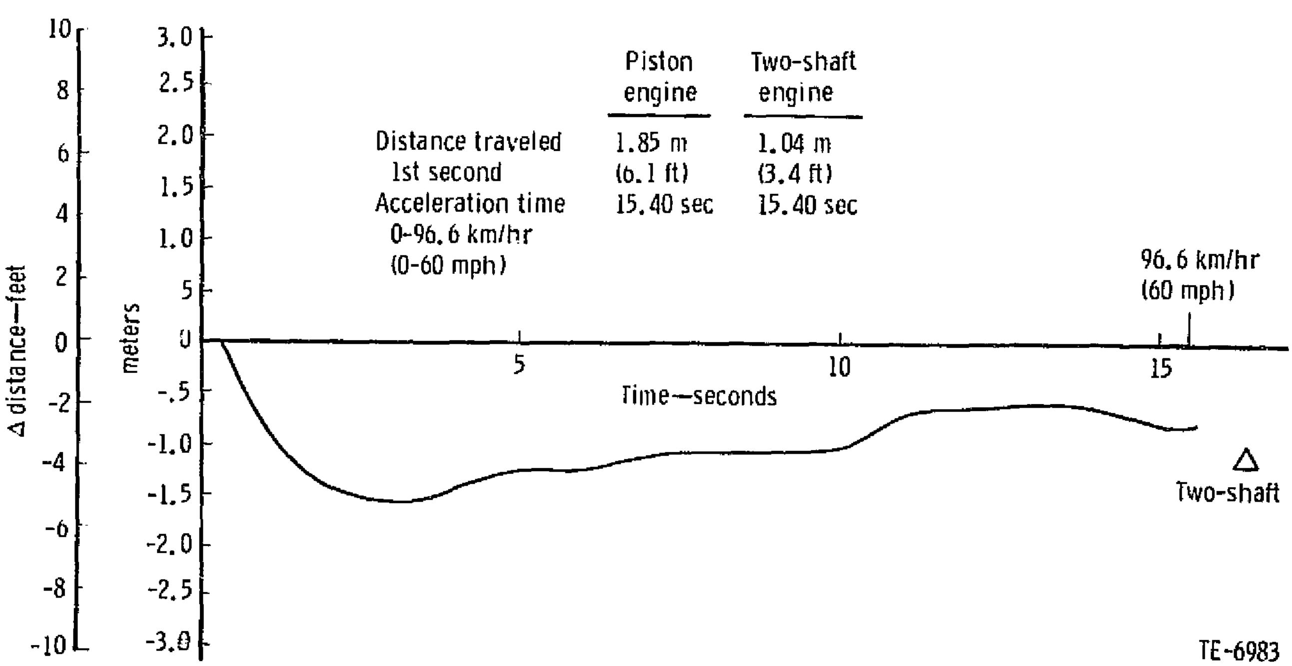


Figure 48. WOT acceleration comparison--base-line versus two-shaft turbine vehicles.

Based on these vehicle performance studies, where "driveability acceptance" is rated on the initial vehicle response to throttle as well as acceleration time to 97 km/h (60 mph), the two-shaft gas turbine engine with conventional automotive three-speed torque converter transmission best meets the objective.

DESIGN STUDIES

The physical size and configuration of the vehicle engine compartment imposed constraints on the optimum powertrain arrangement. These constraints were different locations of the driving wheels relative to engine location, such as the front engine driving the rear wheels or the front engine driving the front wheels. A front-wheel-drive (FWD) vehicle was judged to be the more difficult installation, and its design studies could be used to represent the problems that would be found in either vehicle. Therefore, all engine configurations examined were for FWD based on minimum change to vehicle. The hood line, front wheel suspension towers, transaxle location, engine cradle members, etc, were considered as remaining unchanged for the turbine engine installation studies.

Engine General Arrangements

The three basic engine configurations that were examined for suitability within the framework of the constraints imposed by the vehicle were the single-shaft engine, the two-shaft engine, and the differential engine. Within each of these, further variations were considered for the components of the three configurations. A matrix of the several engine installation schemes considered is shown in Table XIX. The regenerator was investigated as a single disk and as two disks except for the differential engine where, based on space availability, the use of two disks was not feasible.

TABLE XIX. IGT ENGINE/INSTALLATION SCHEMES

	Single-shaft engines					Two-shaft engines				Differential engines	
Regenerator disk											
One	X			X	X	X	X		X	X	X
Two		X	X					X			
Combustor											
Can	X		X		X	X	X	X	X	X	X
Annular		X		X							
Gasifier turbine											
Radial	X	X	X			X	X	X	X	X	
Axial				X							
Reentry					X						X
Power turbine											
Axial						X	X				
Radial								X	X		
Rotor coupling system											
Power transfer						X					
Belt										X	X
Vehicle transmission	(Envelope equivalent to conventional transmission)					Conventional manual conventional automatic				(Envelope equivalent to conventional transmission)	
Figure reference	49	50	51	52	53	54	55	56	57	59, 60	58

The combustor was considered as either a can-type or an annular combustor.

The three types of gasifier turbines included in the study were axial flow, radial inflow, and reentry. Radial and axial turbines were considered for the power turbine.

All engine/installation schemes used the same centrifugal compressor. For the purposes of these engine/installation studies, it was assumed that the transaxle stayed the same size and in the same location as for a conventionally powered car.

For the two-shaft engine, the transmission was the same as that for the conventionally powered car. For the single-shaft engine, the continuously variable transmission was designed to fit in the same envelope as the conventional transmission for the two-shaft engine.

Figure 49 shows the arrangement of a single-shaft engine with one regenerator disk, a can-type combustor, a radial inflow turbine, and a centrifugal compressor.

The arrangement shown in Figure 50 is for a single-shaft engine with two regenerator disks, an annular combustor, a radial inflow turbine, and a centrifugal compressor. Here the exhaust ducting from the regenerators to the vehicle exhaust system was severely cramped. Figure 51 shows the preceding arrangement with a can-type combustor in place of the annular combustor.

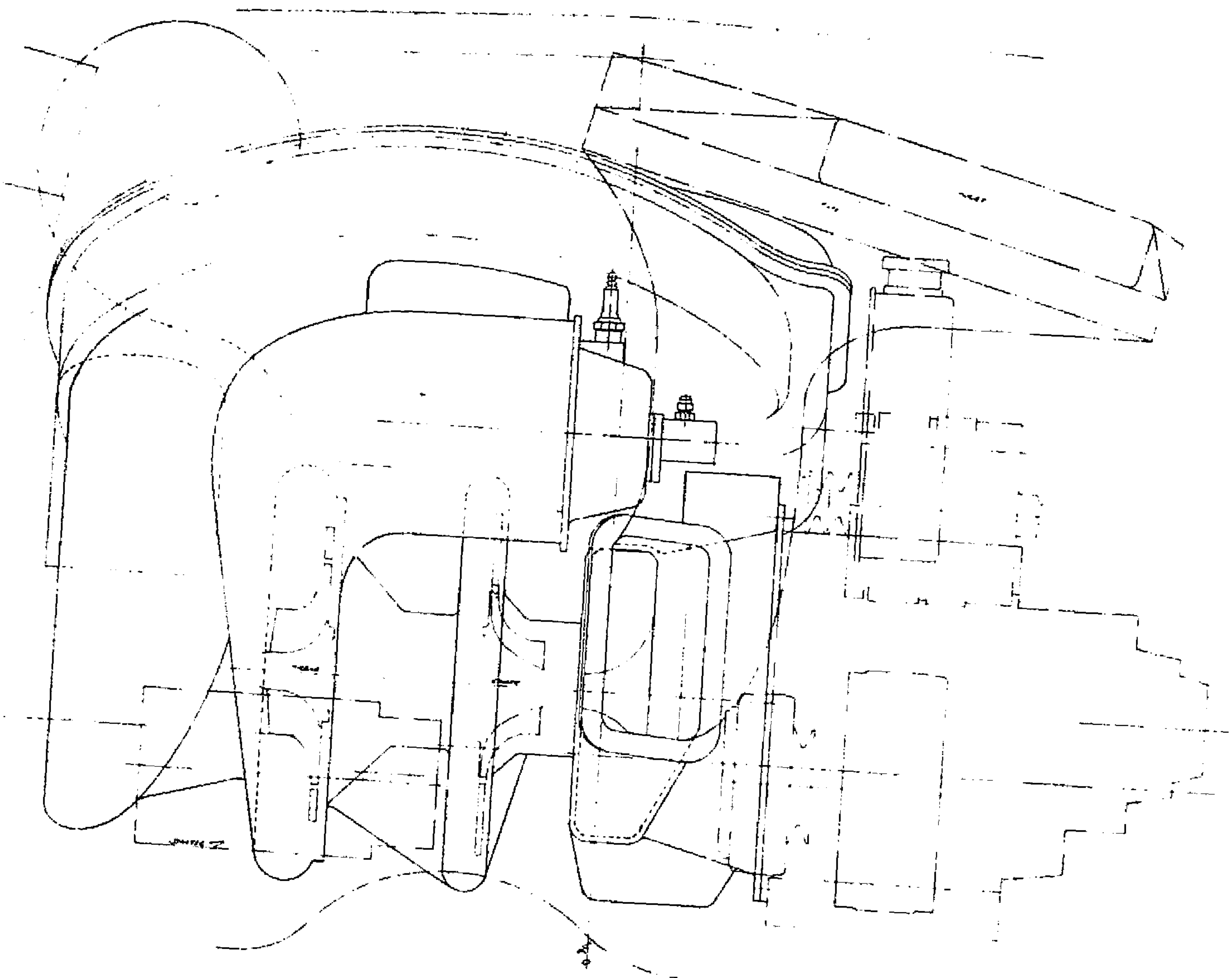
Figure 52 shows a single-shaft engine with one regenerator disk, an annular combustor, a two-stage axial-flow turbine, and a centrifugal compressor.

Figure 53 represents a single-shaft engine with an axial-flow, two-pass reentry turbine, a can combustor, one regenerator disk, and a centrifugal compressor.

Figures 54 and 55 each show a two-shaft engine with one regenerator disk, a can combustor, a radial-inflow gasifier turbine, a centrifugal compressor, and an axial-flow power turbine. Figure 54 indicates a power transfer device between the two turbine shafts with the starter driving through the reduction gearbox. Figure 55 does not have power transfer and the starter drive through a belt on the gasifier compressor shaft.

Figures 56 and 57 each show a two-shaft engine with centrifugal compressor, radial inflow power turbine, can combustor, and, respectively, two and one regenerator disks.

Figure 58 represents an engine installation of a differentially geared engine featuring a two-pass axial reentry turbine, a centrifugal compressor, a can combustor, a single regenerator disk, and a variable sheave belt drive torque transfer system between the output shaft and the turbine shaft. All differential engine studies incorporated this variable sheave belt drive. Further investigation determined inadequate belt life to make such a system feasible. Figures 59 and 60 show the external views and cross section of a differentially geared engine with a radial inflow turbine in lieu of the reentry turbine. All other features are as described previously.



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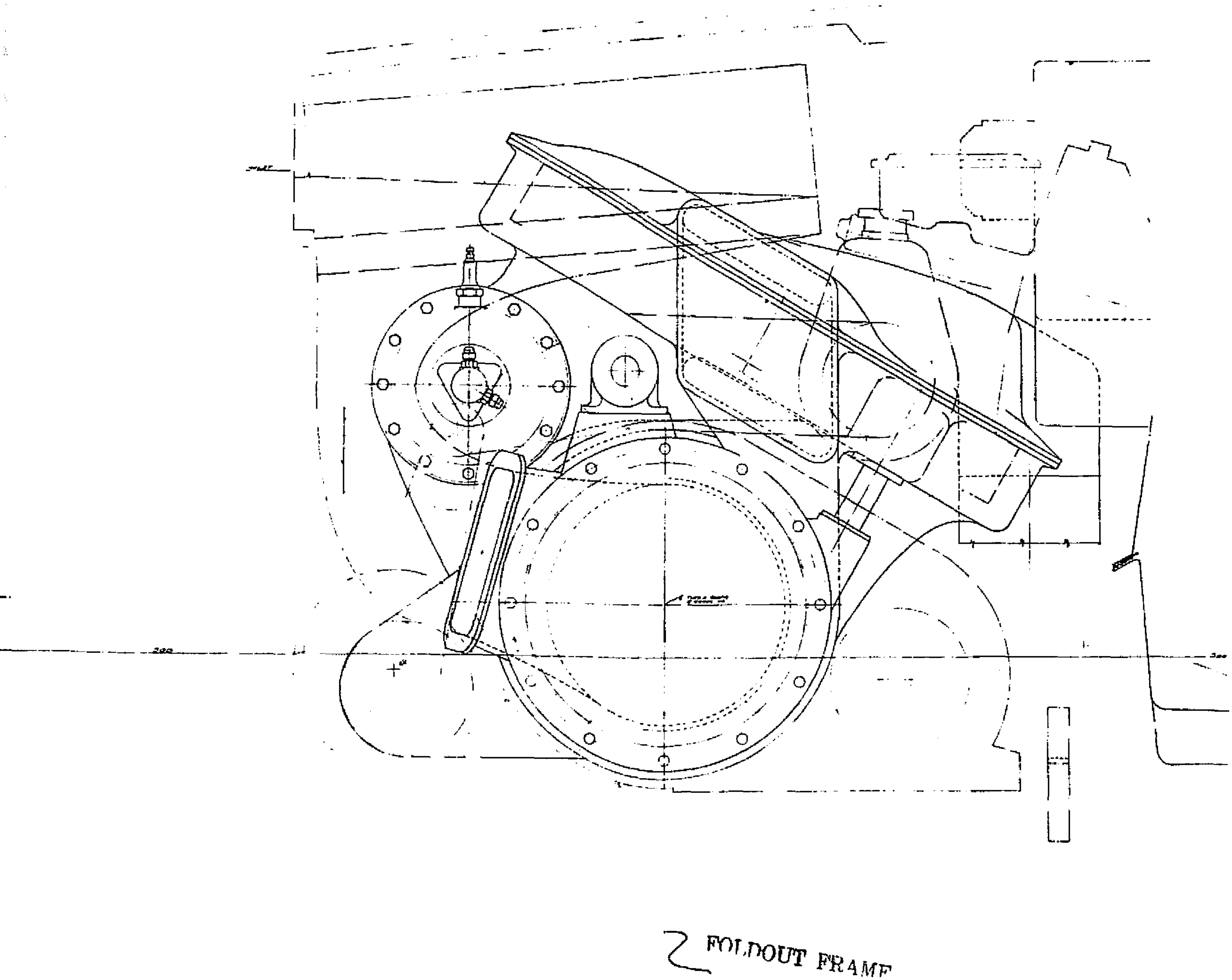

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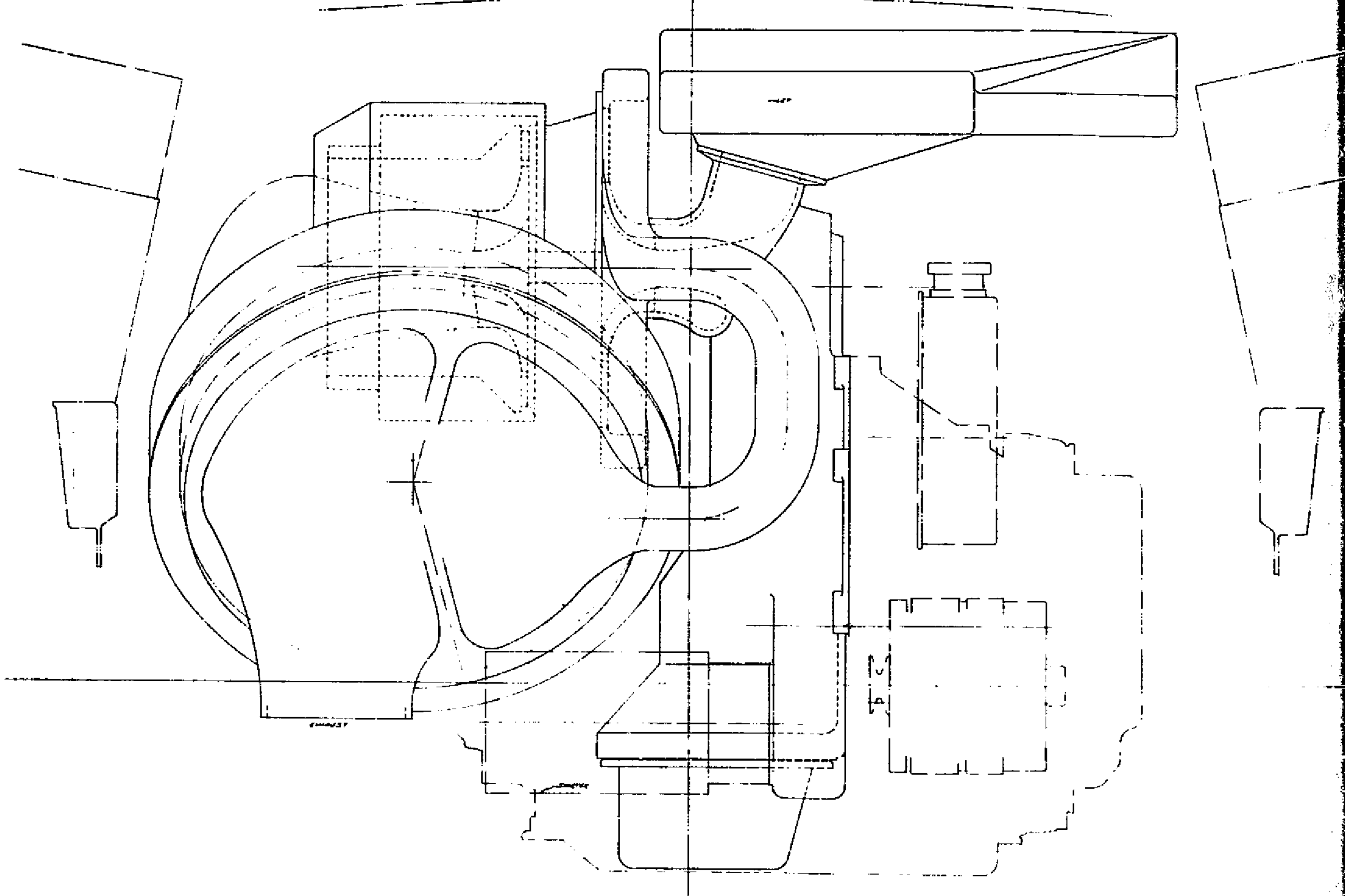
Figure 49. General arrangement--single-shaft engine, one regenerator, can combustor, radial-inflow turbine.



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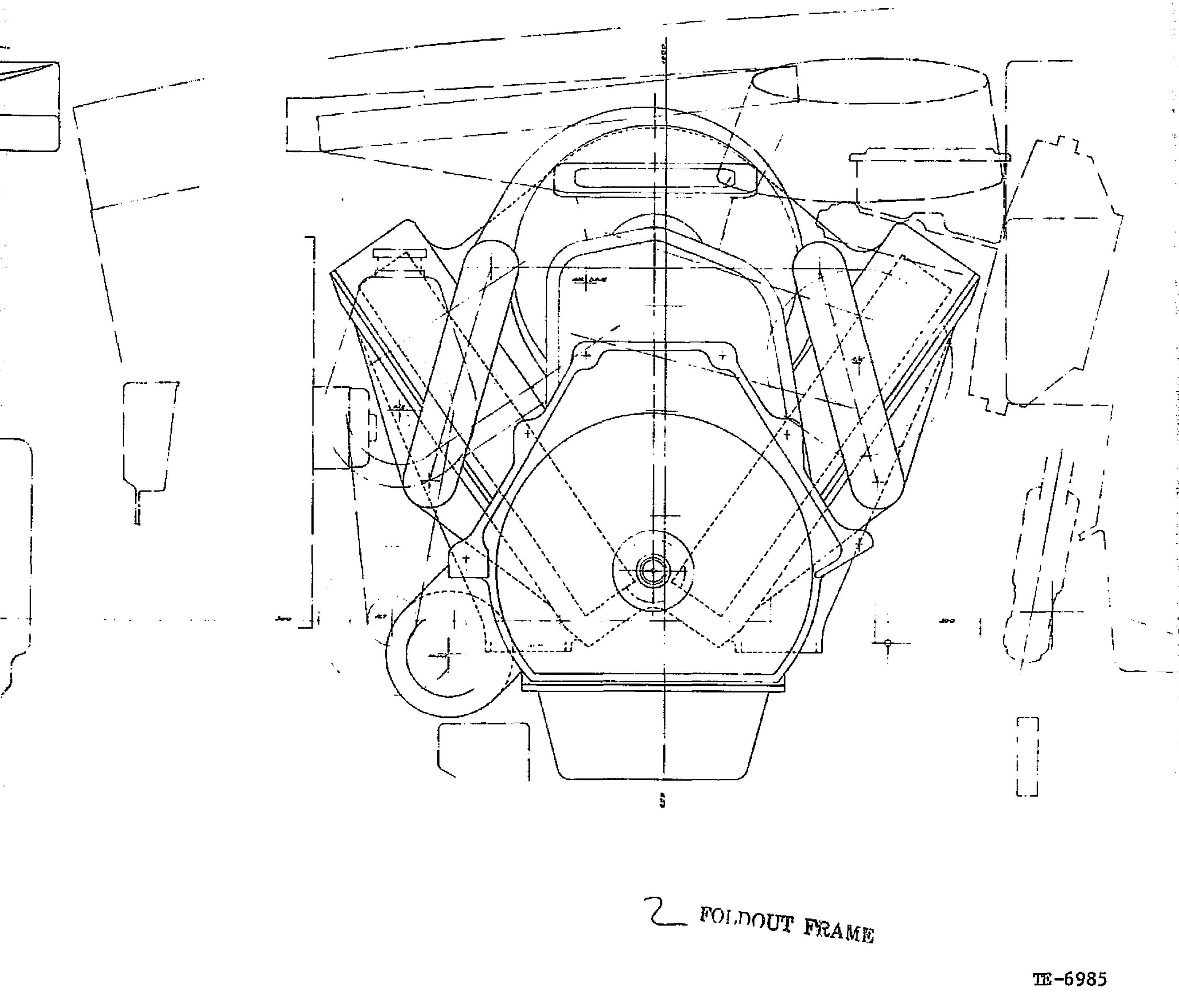

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Figure 50. General arrangement—single-shaft engine, two regenerators, annular combustor, radial-inflow turbine.

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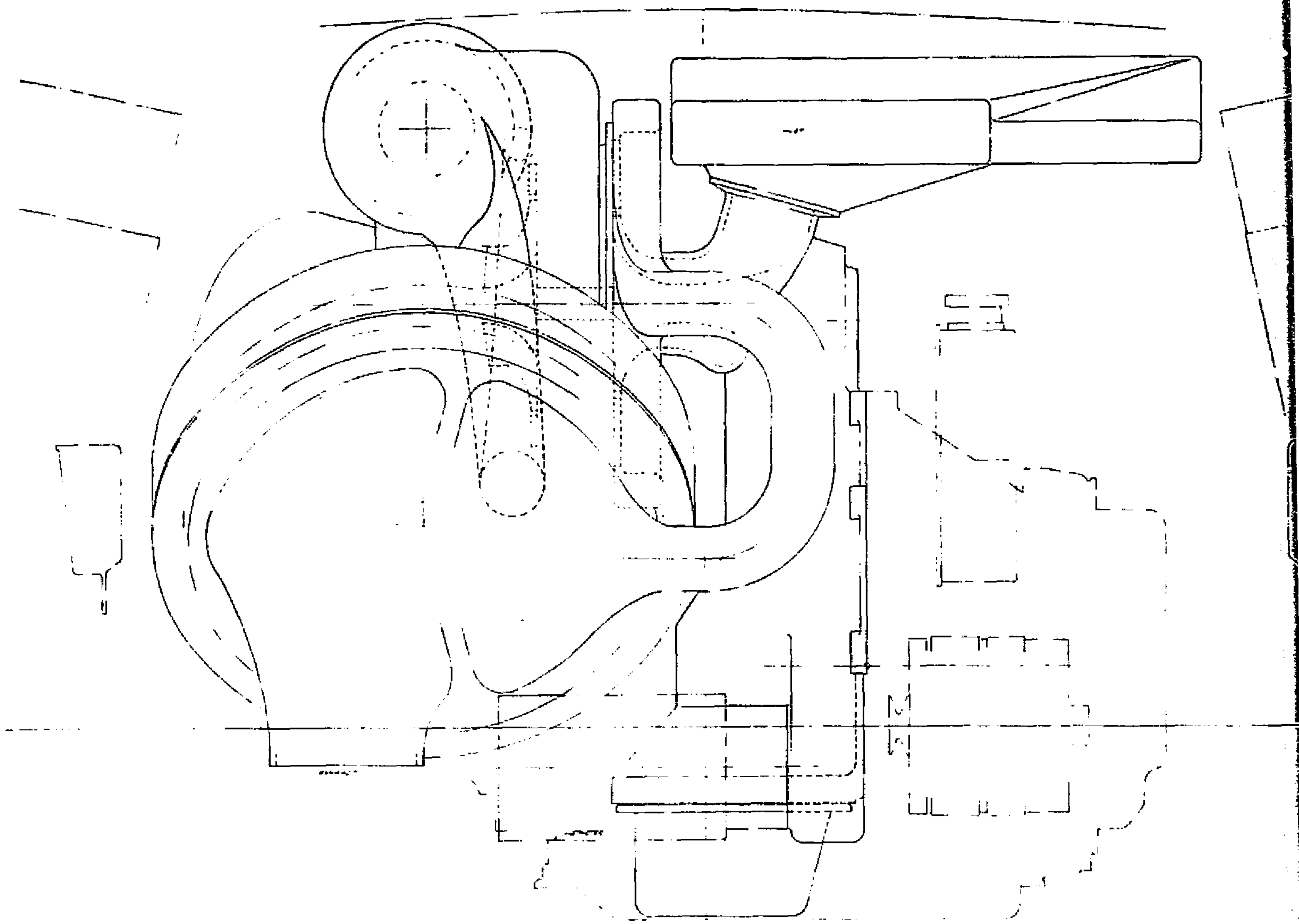

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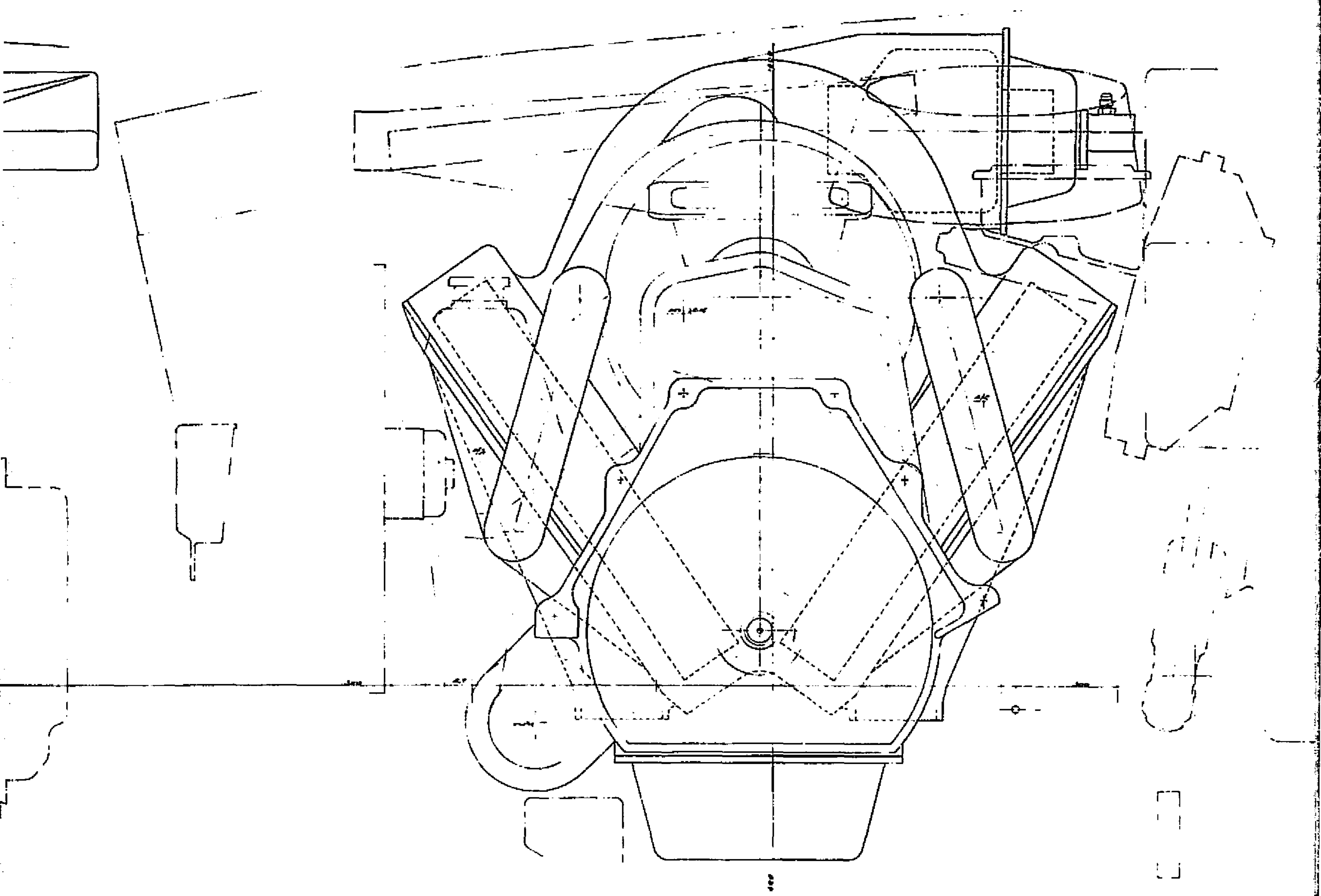

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Figure 51. General arrangement—single-shaft engine, two regenerators, can combustor, radial-inflow turbine.

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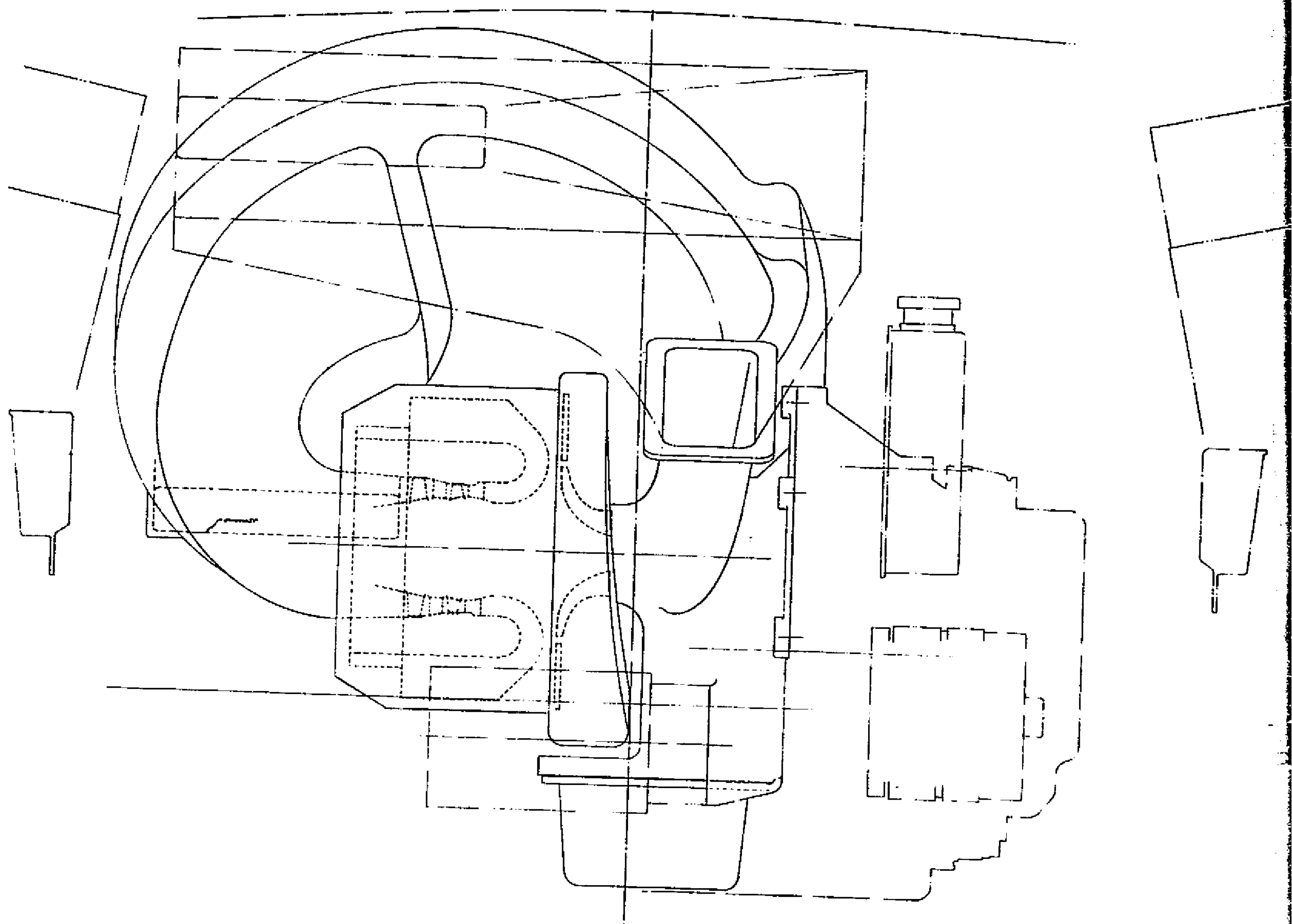

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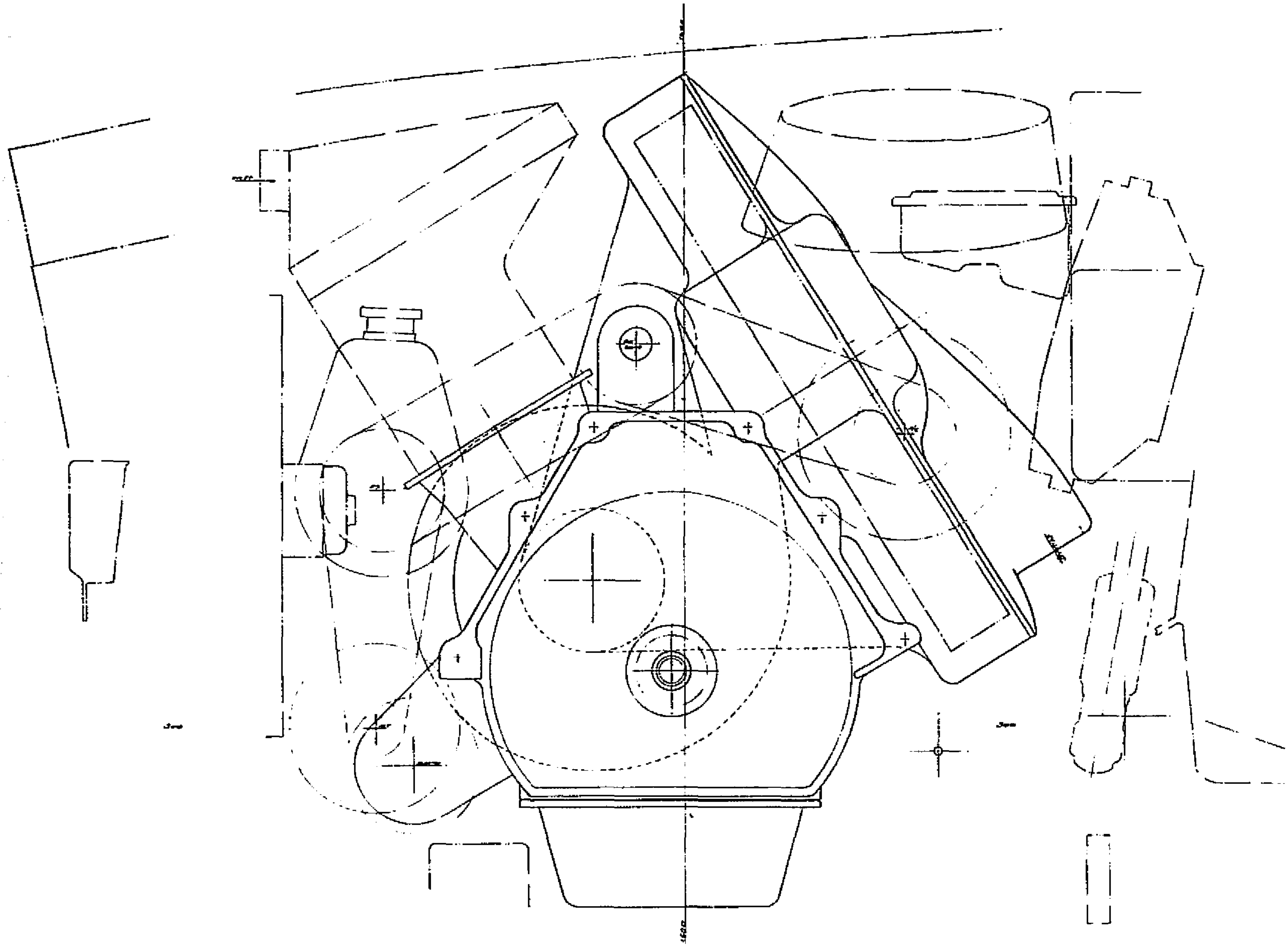

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Figure 52. General arrangement—single-shaft engine, one regenerator, annular combustor, two-stage axial-flow turbine.

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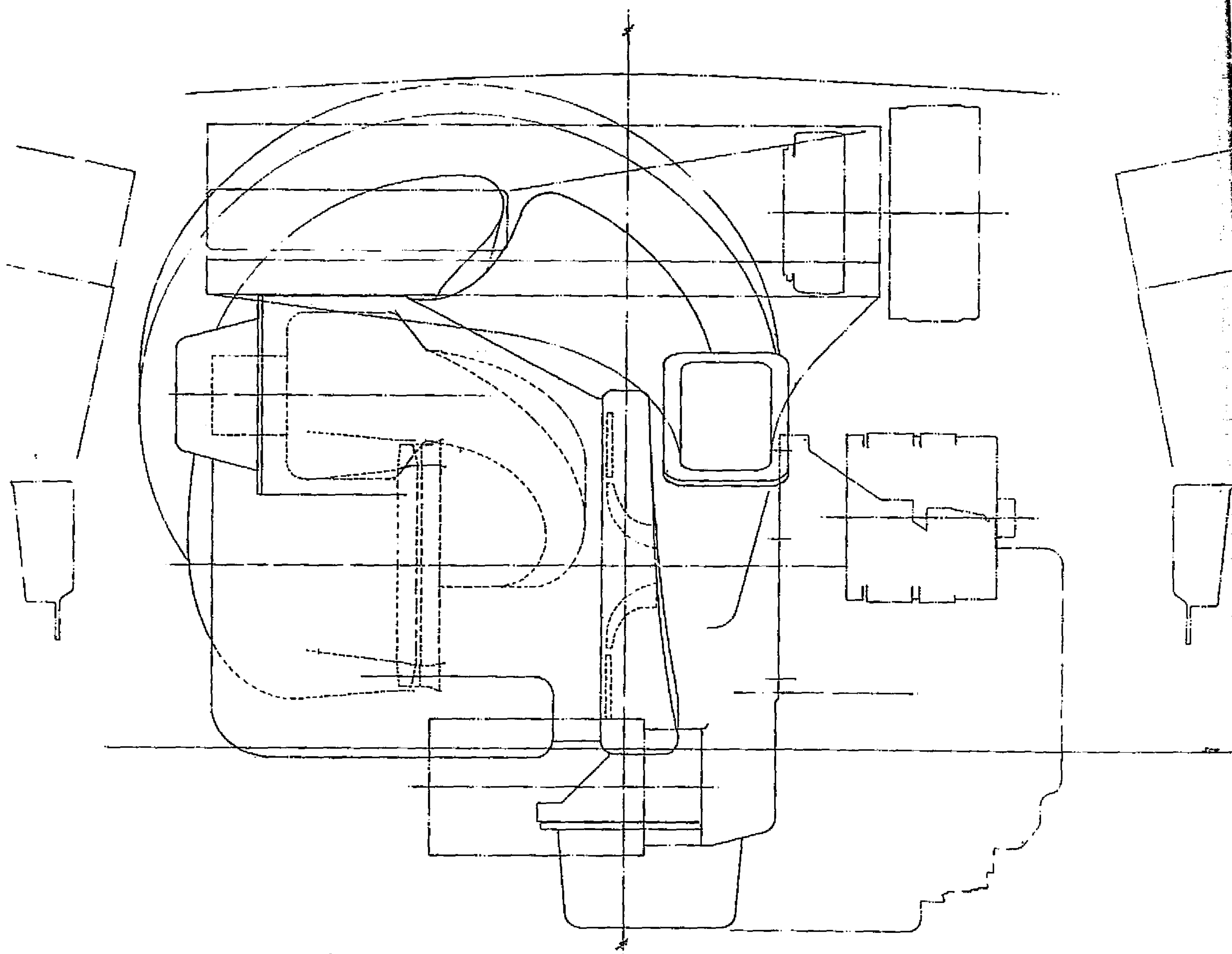

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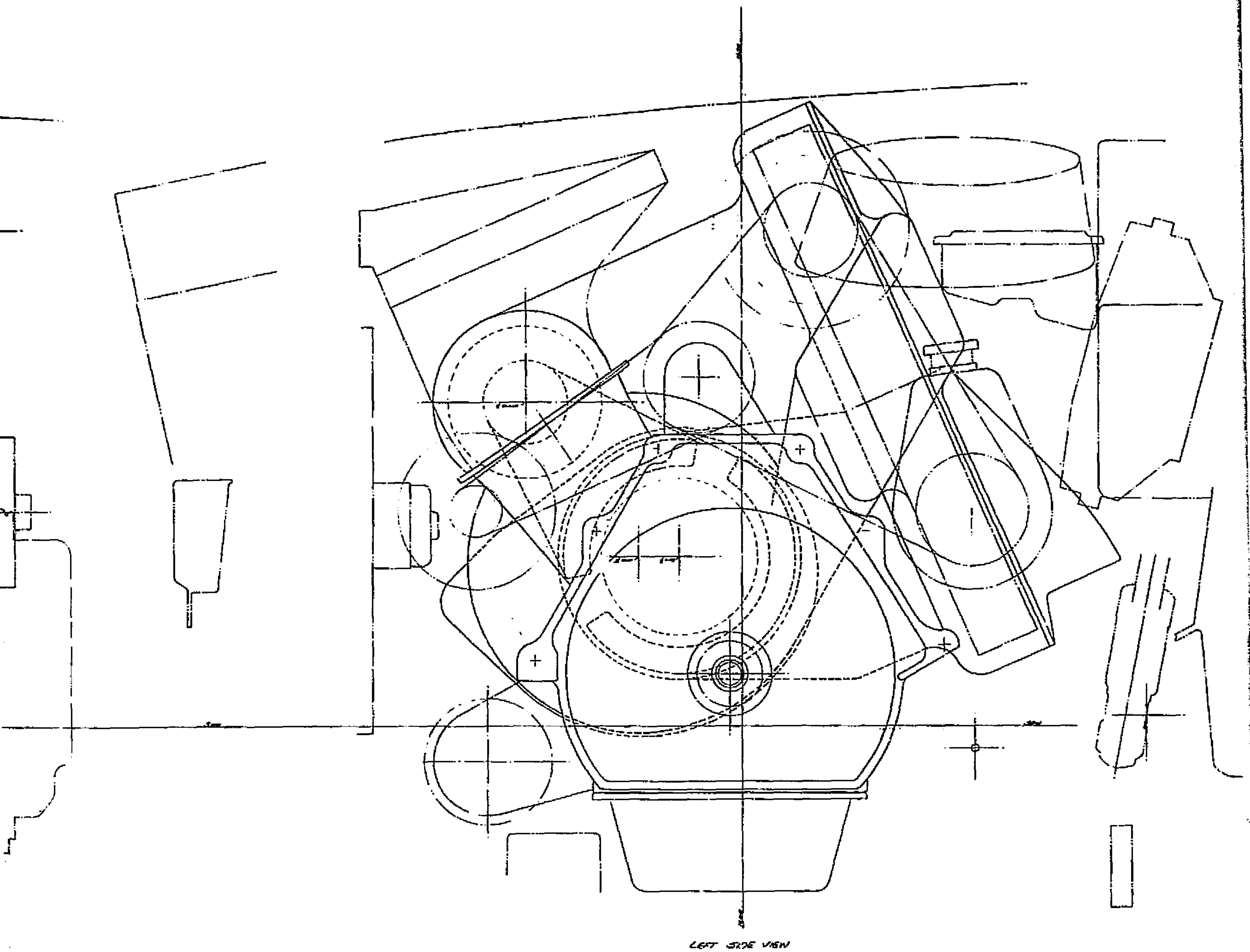

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Figure 53. General arrangement—single-shaft engine, one regenerator, can combustor, reentry turbine.

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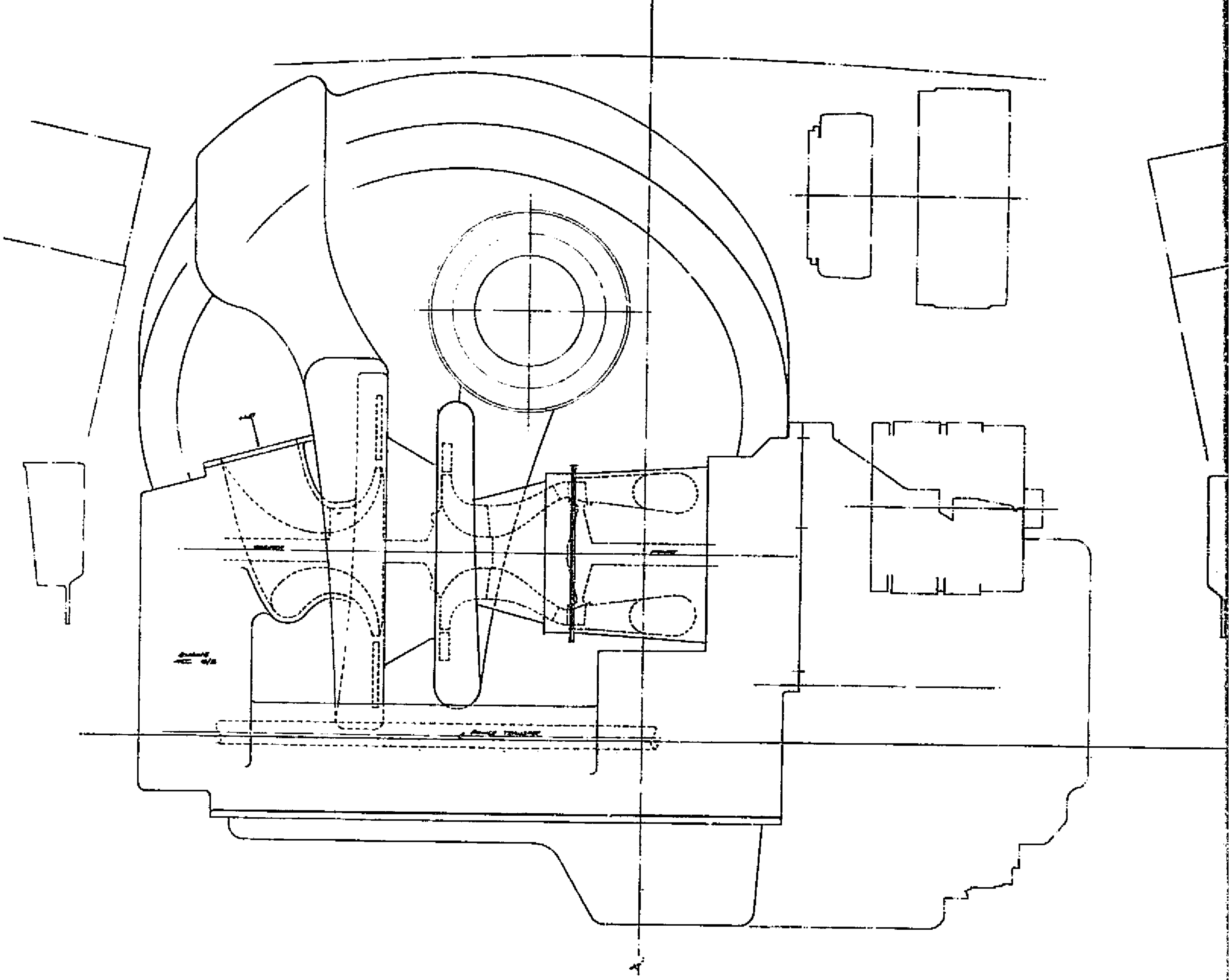

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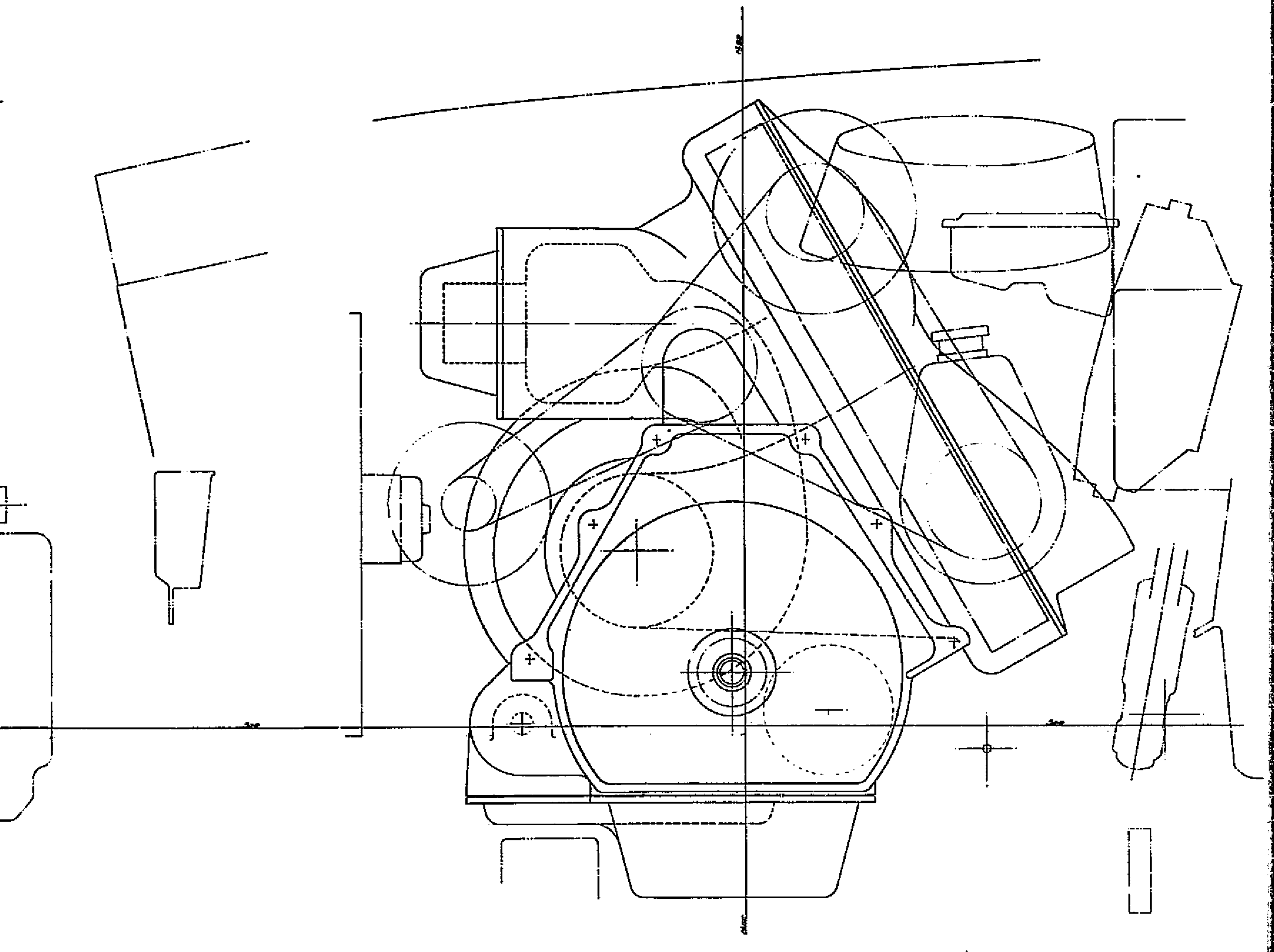

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Figure 54. General arrangement--two-shaft engine, one regenerator, can combustor, radial-inflow gasifier turbine, axial-flow power turbine, power transfer.

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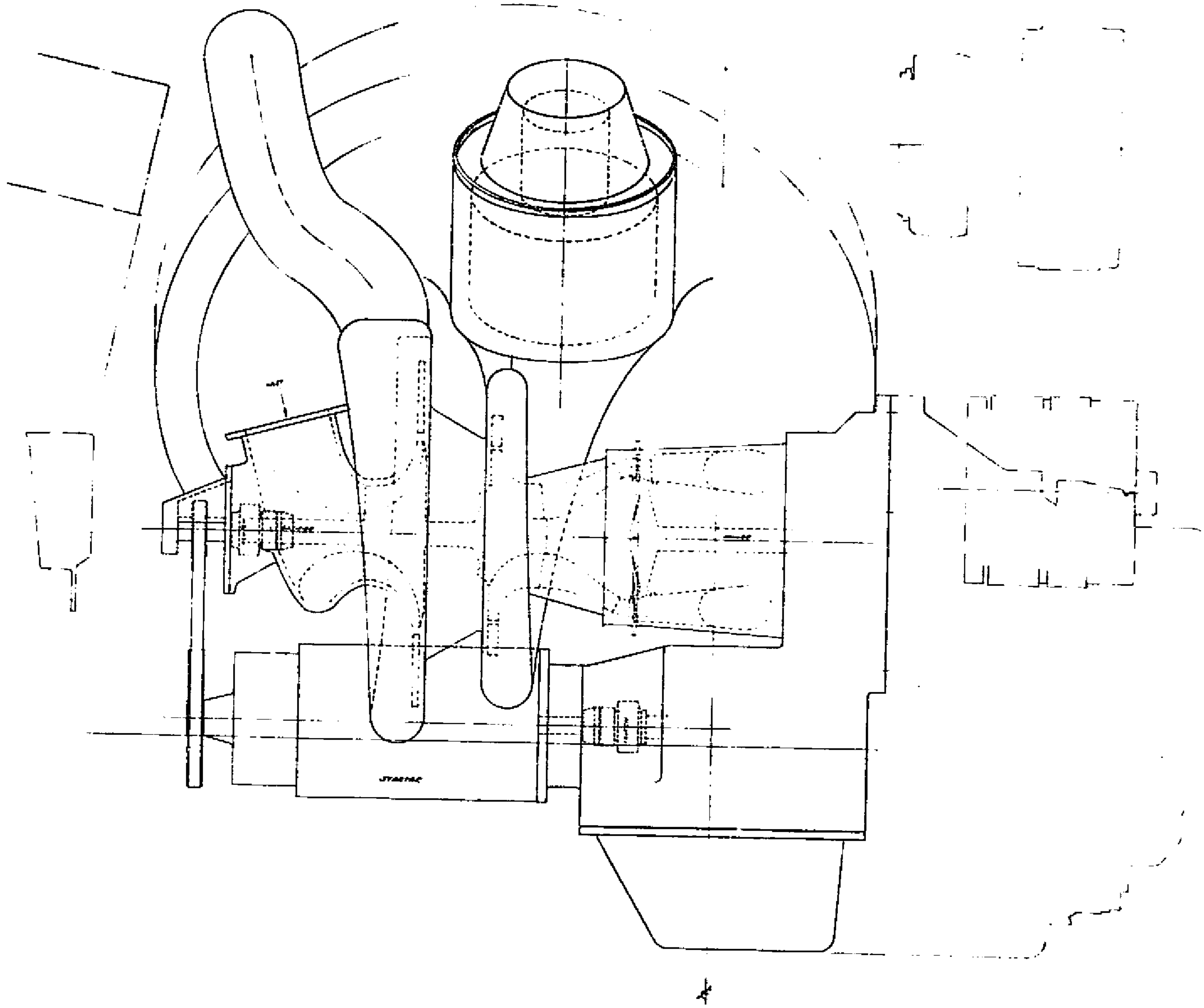

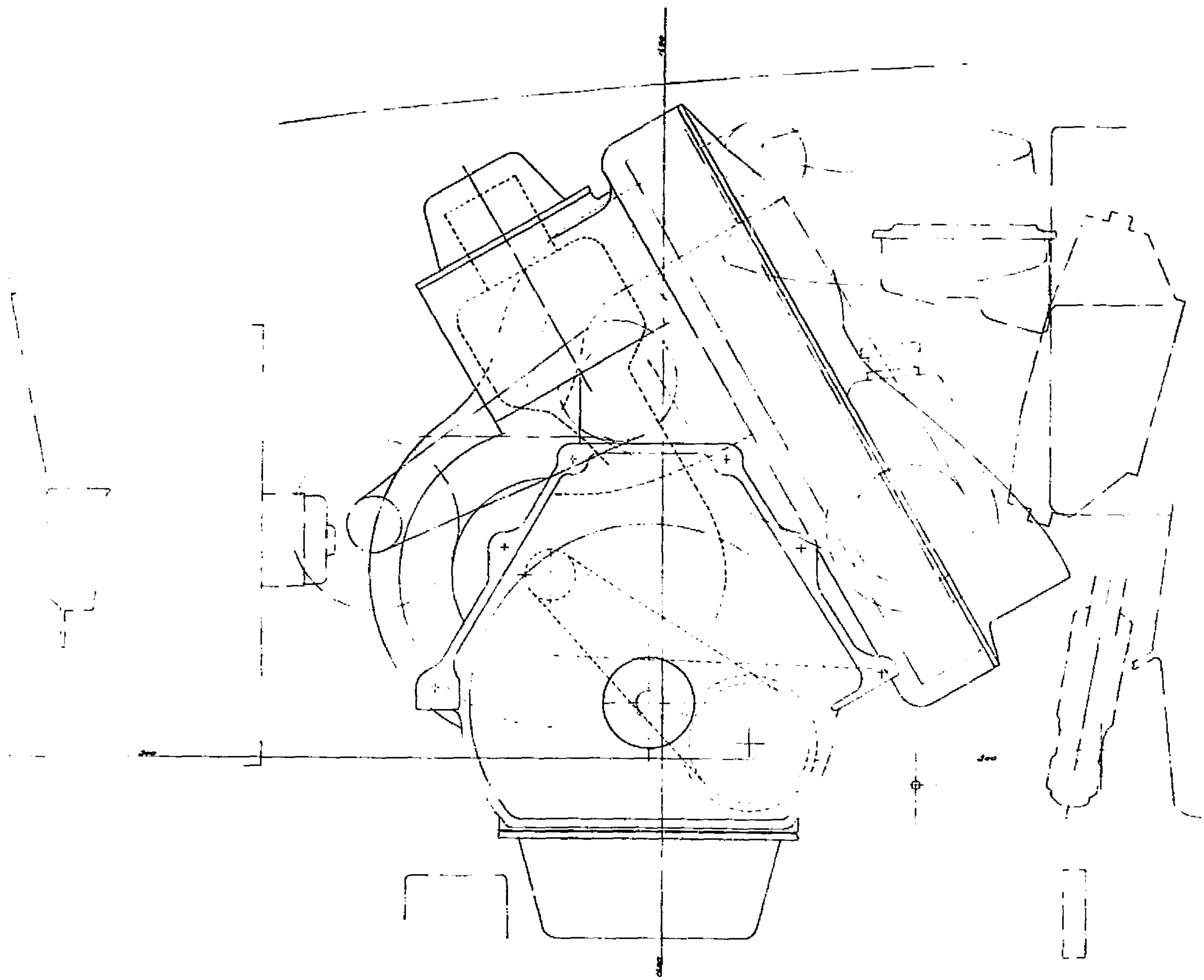

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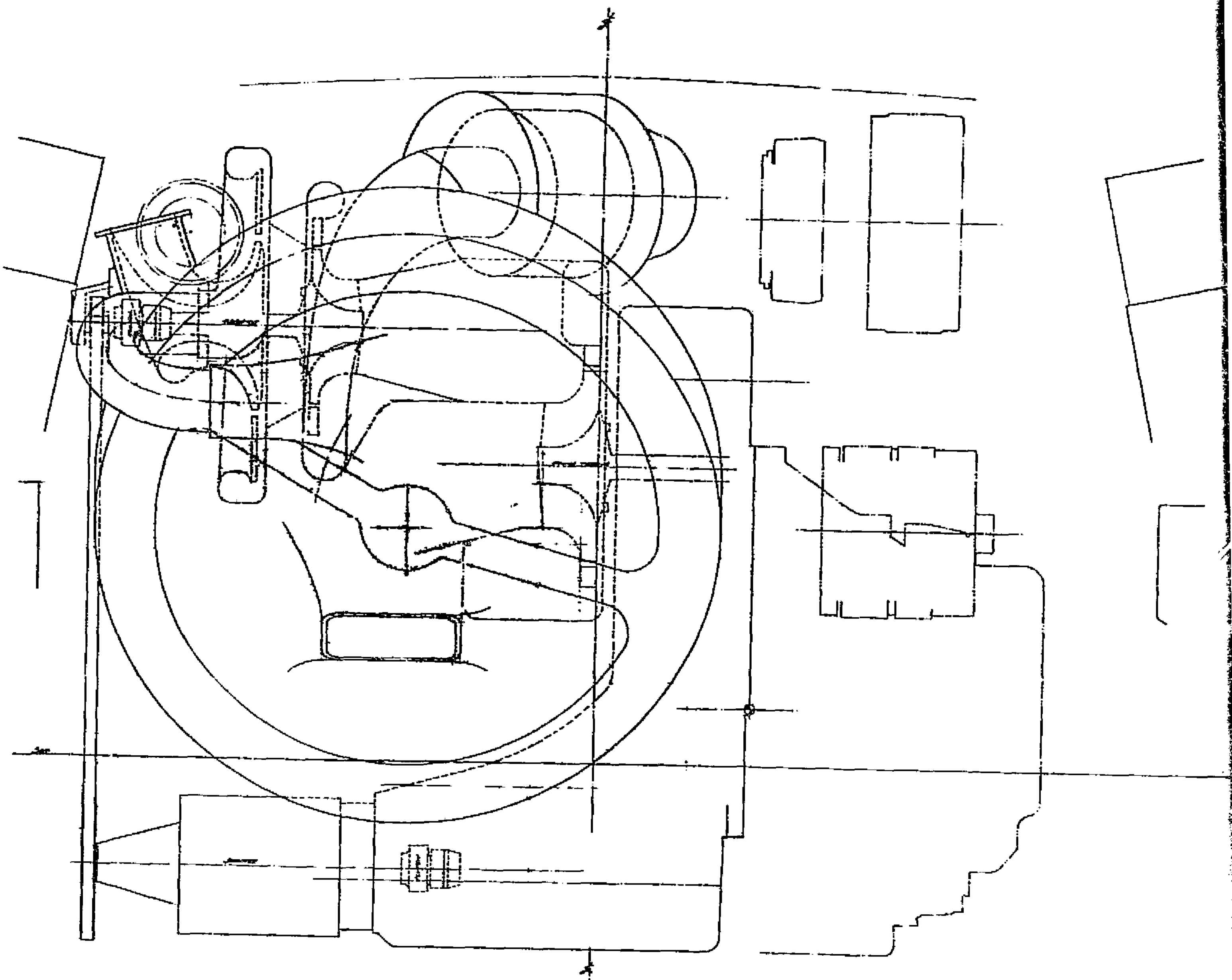
Figure 55. General arrangement—two-shaft engine, one regenerator, can combustor, radial-inflow gasifier turbine, axial-flow power turbine.



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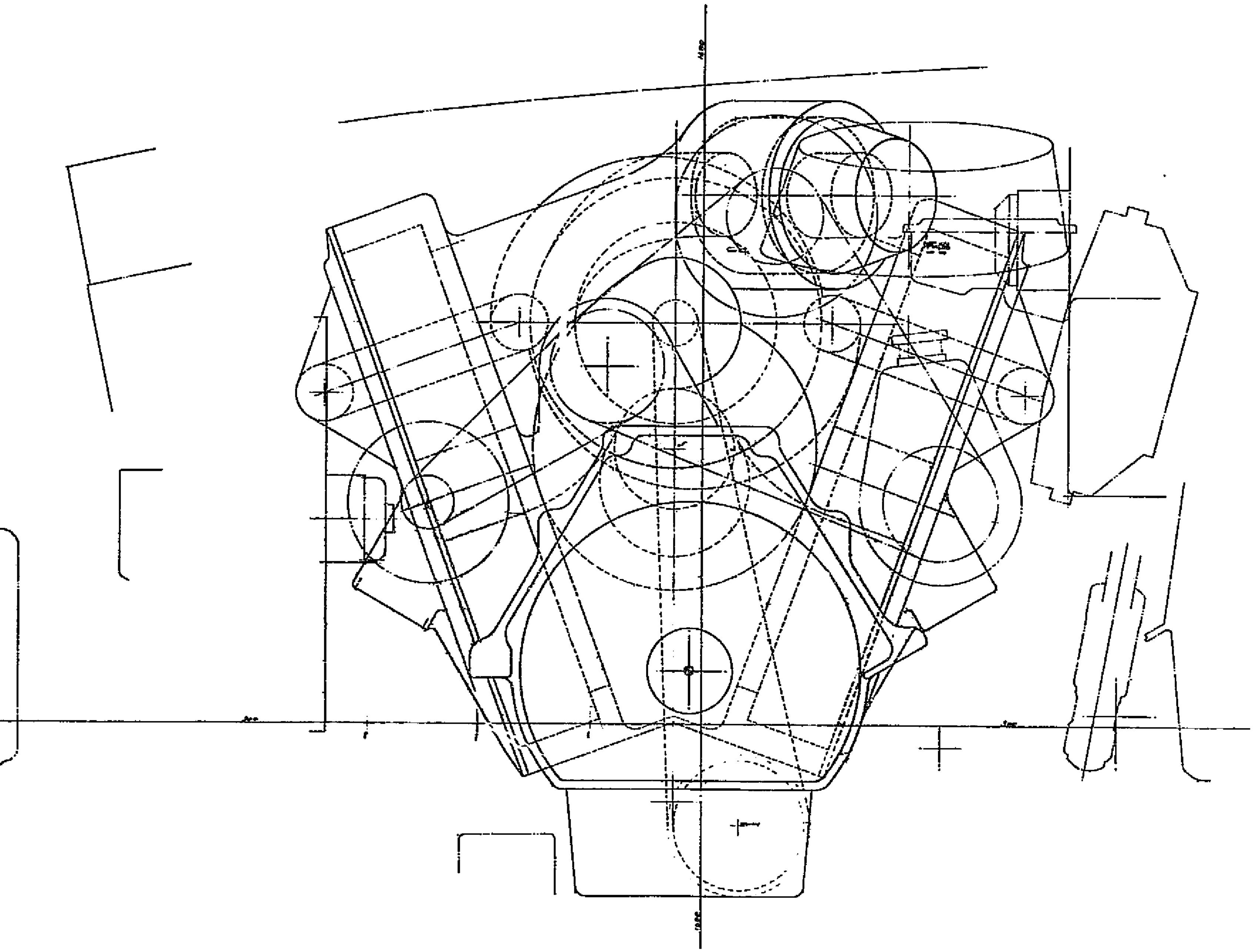

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Figure 56. General arrangement--two-shaft engine, two regenerators, can combustor, radial-inflow power turbine, radial-inflow gasifier turbine.

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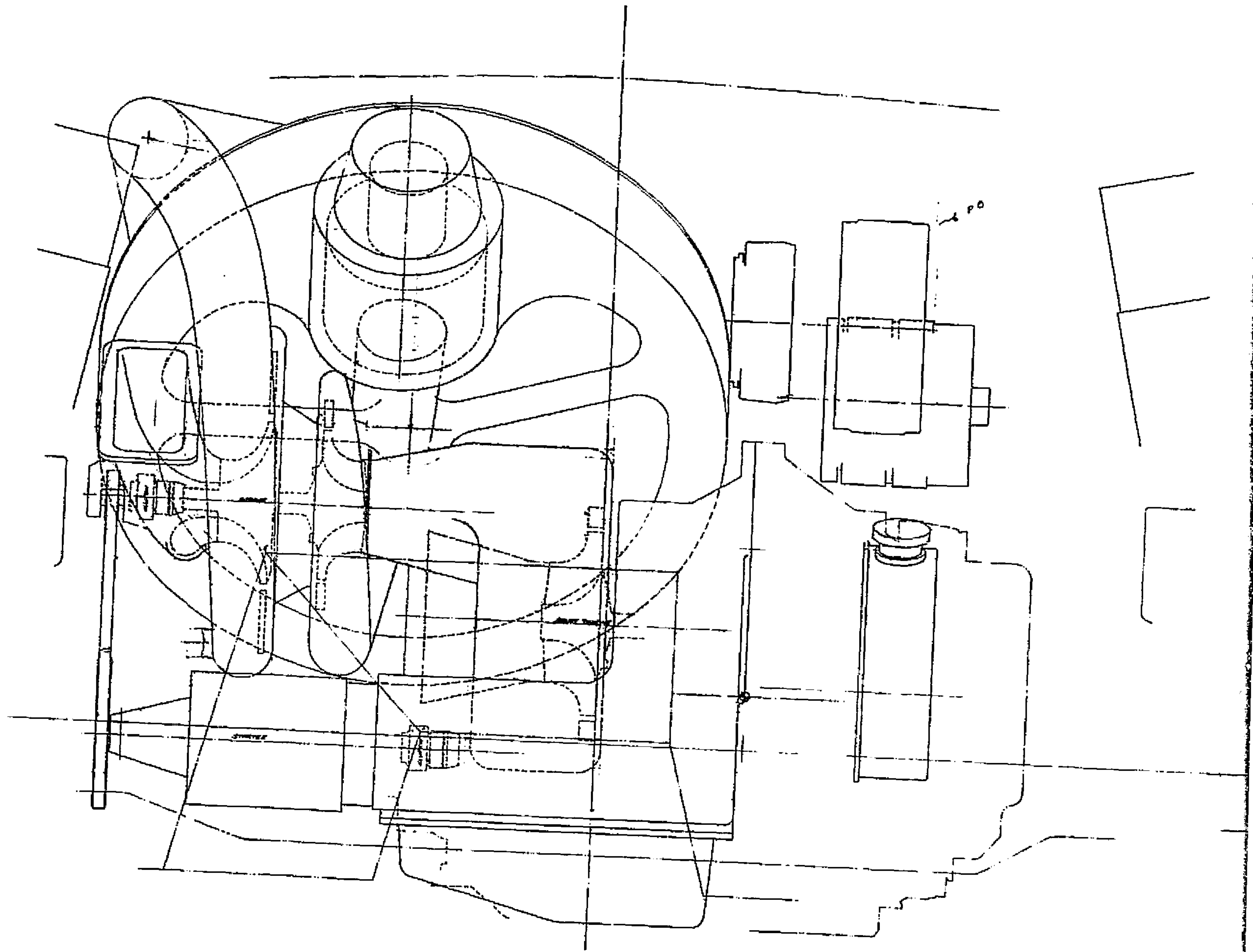

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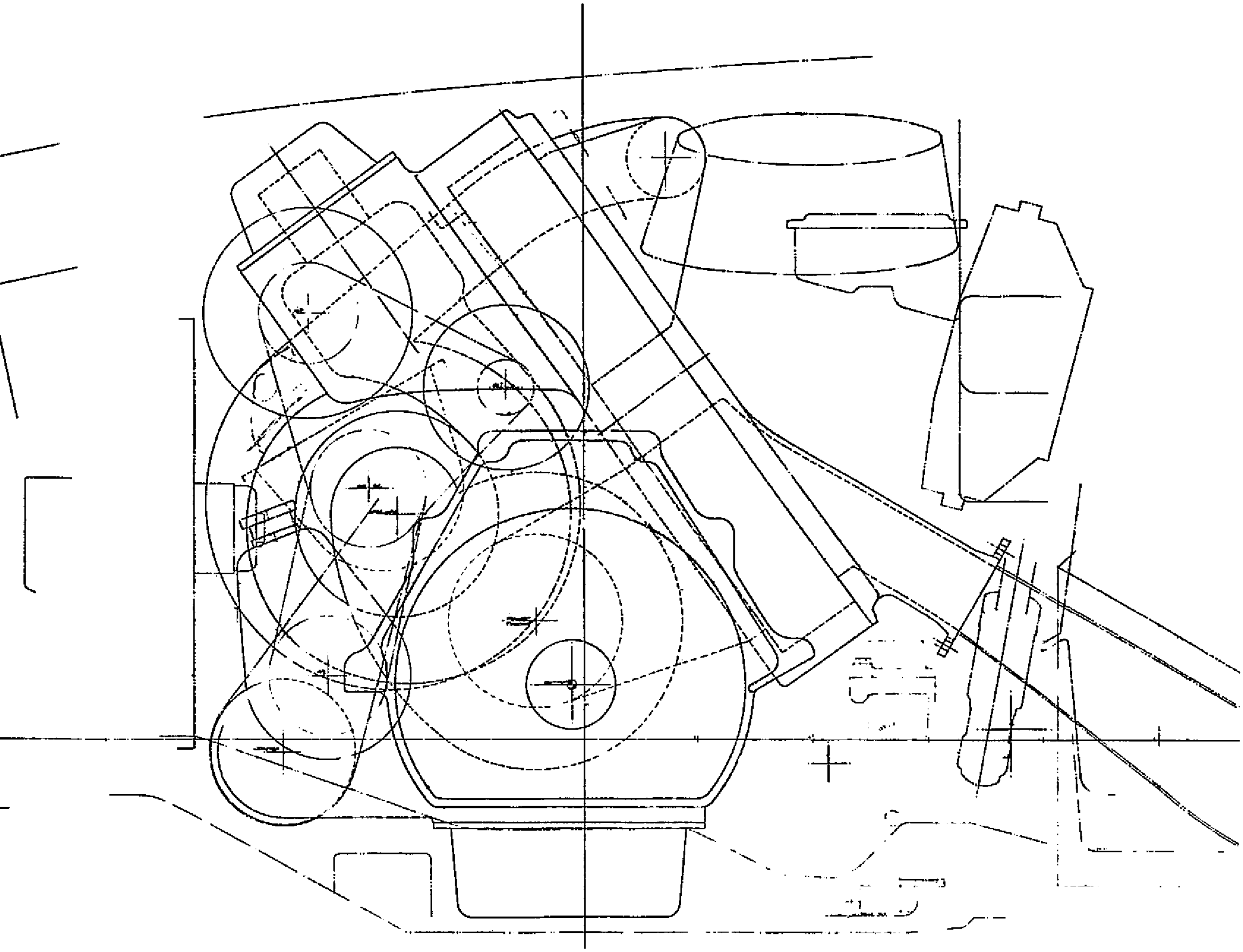

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Figure 57. General arrangment--two-shaft engine, one regenerator, can combustor, radial-inflow power turbine, radial-inflow gasifier turbine.

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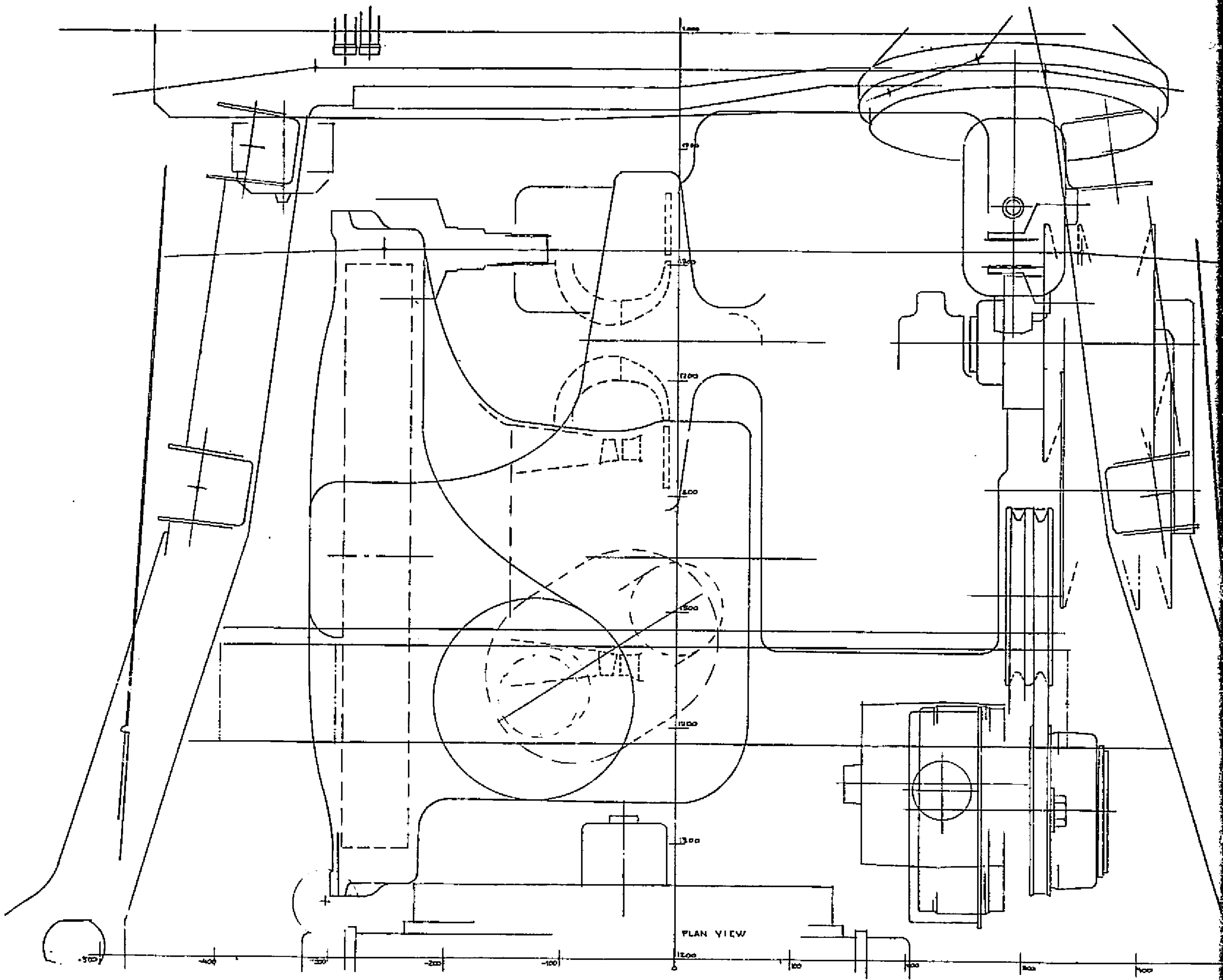

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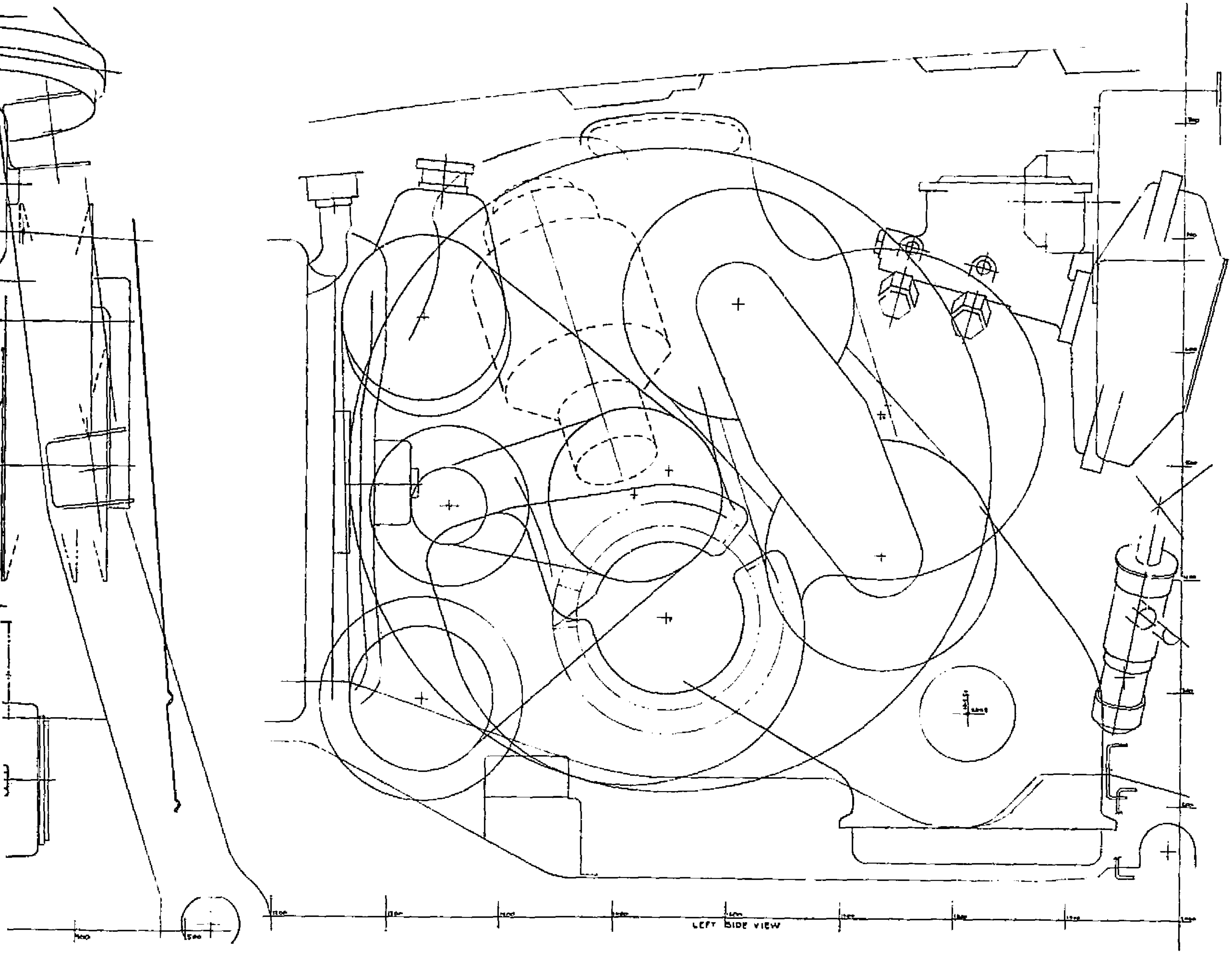

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Figure 58. General arrangement—differential engine, one regenerator, can combustor, reentry turbine.

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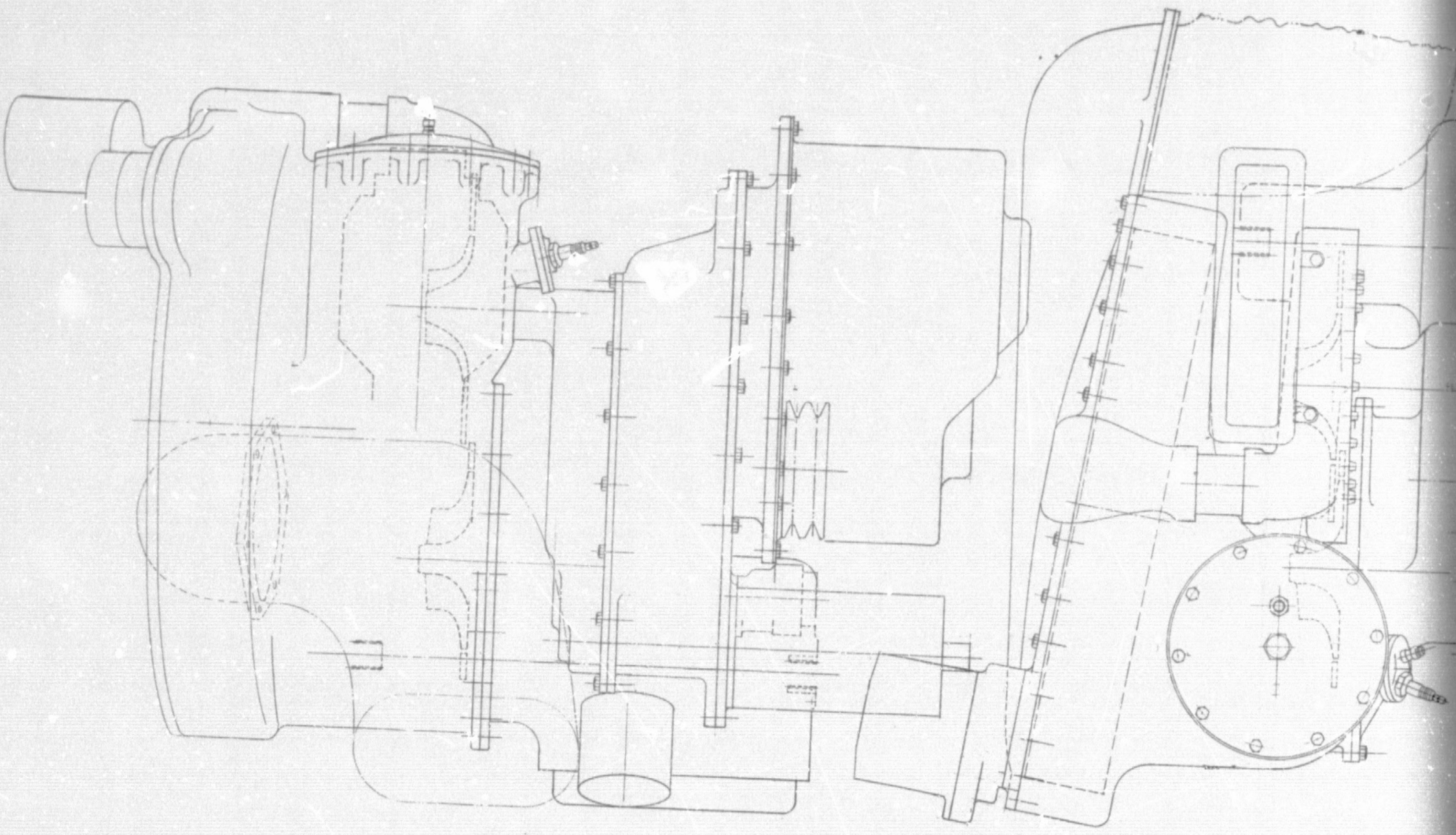

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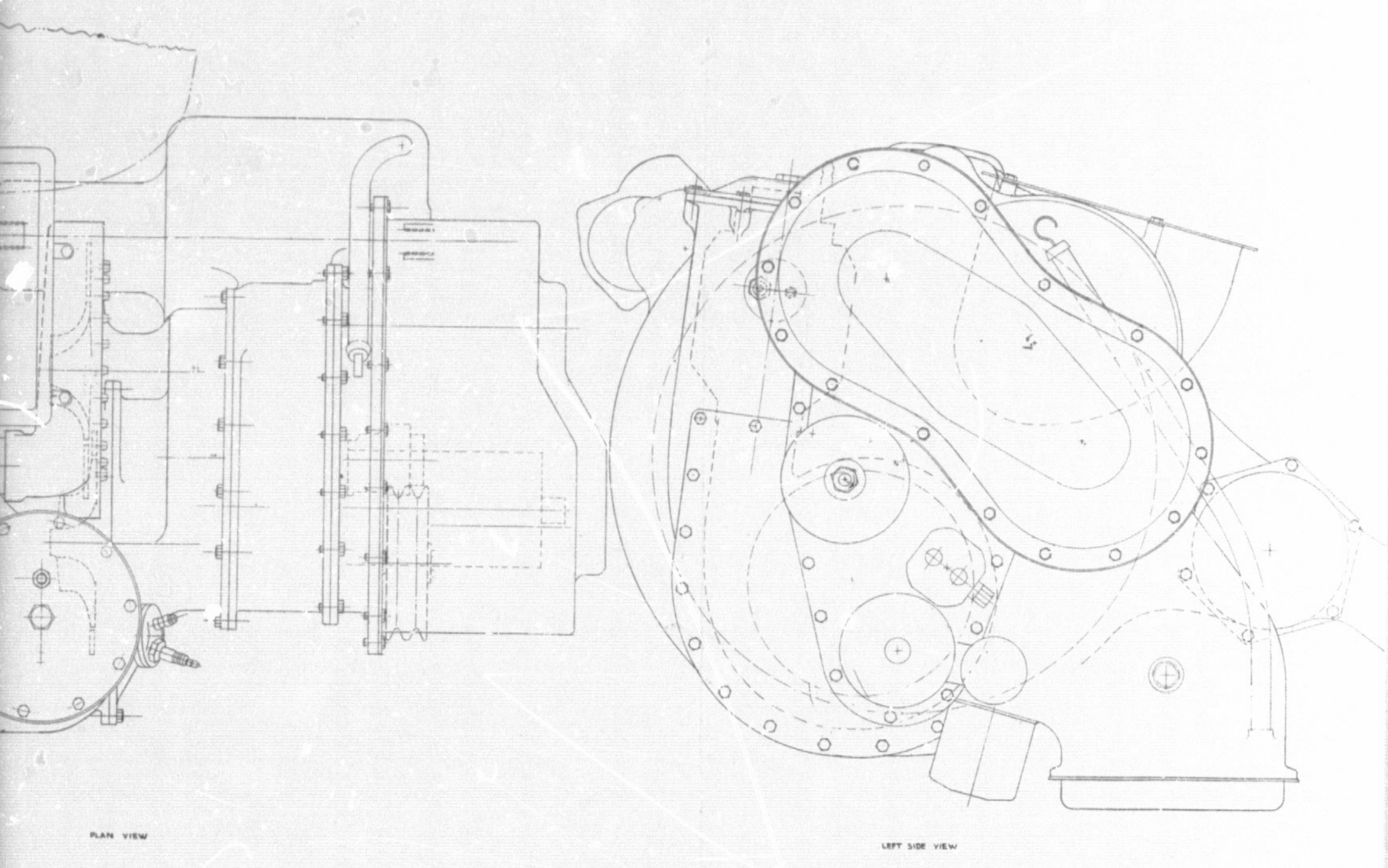

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Figure 59. External views--differential engine, one regenerator, can combustor, radial-inflow turbine.

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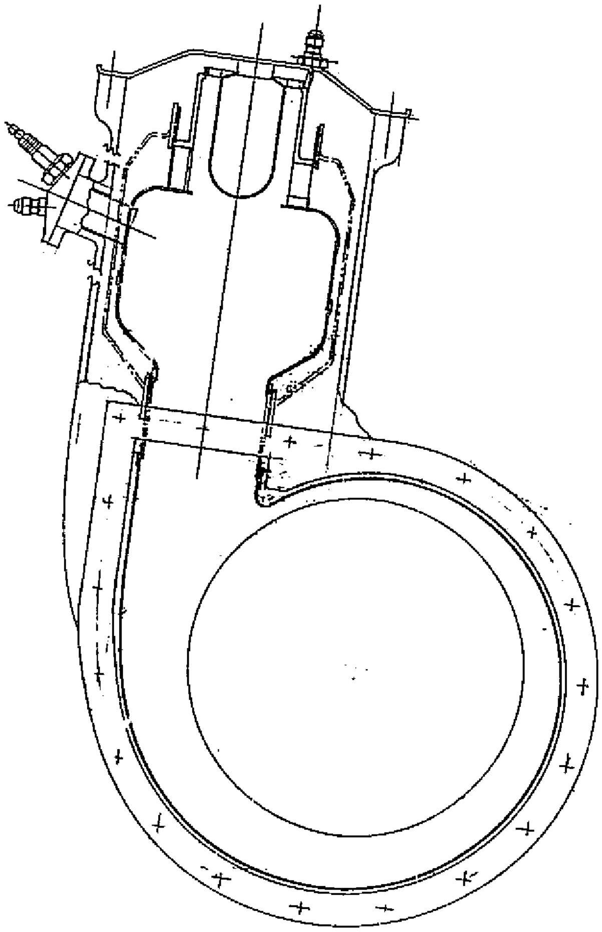

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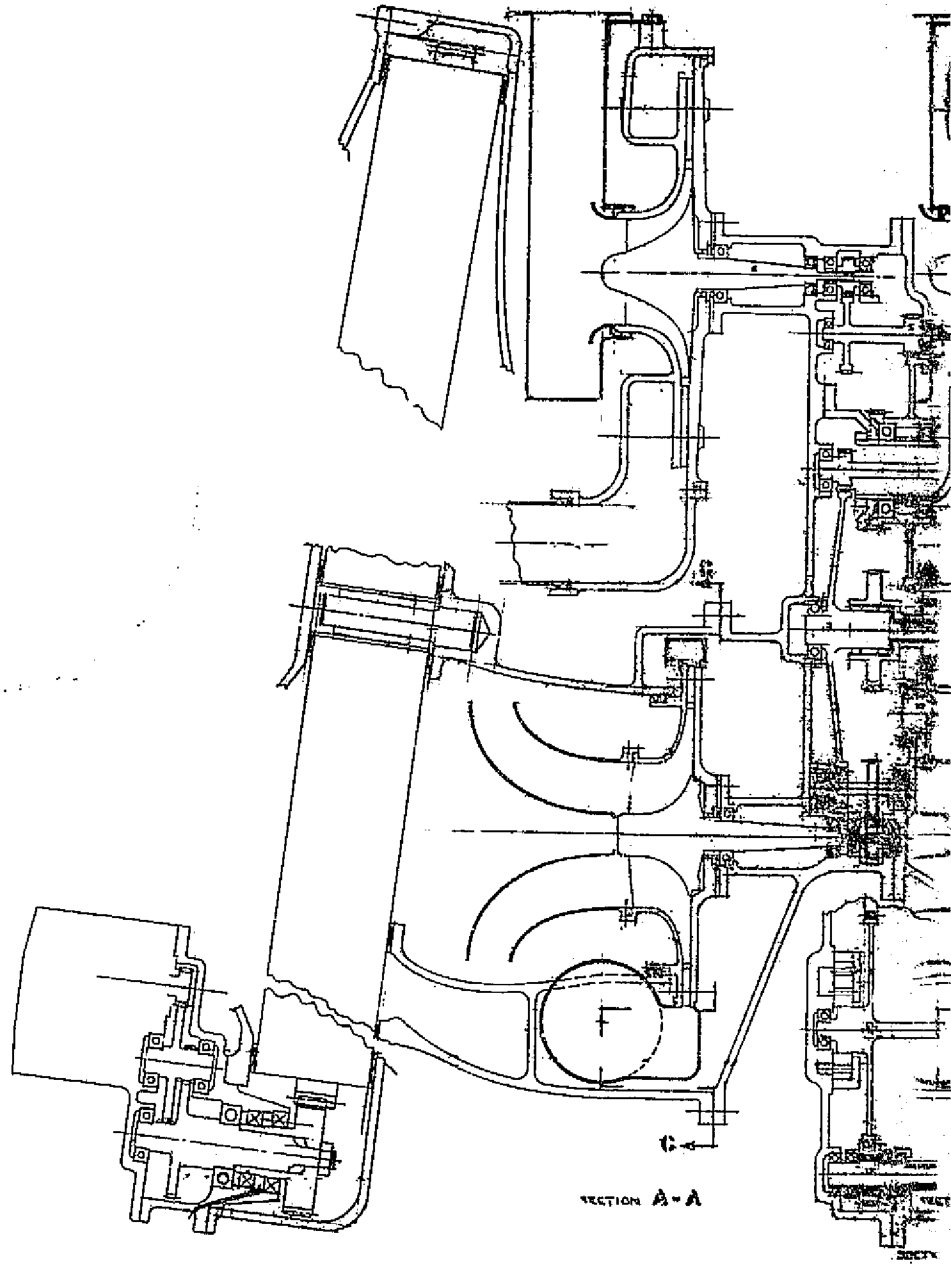

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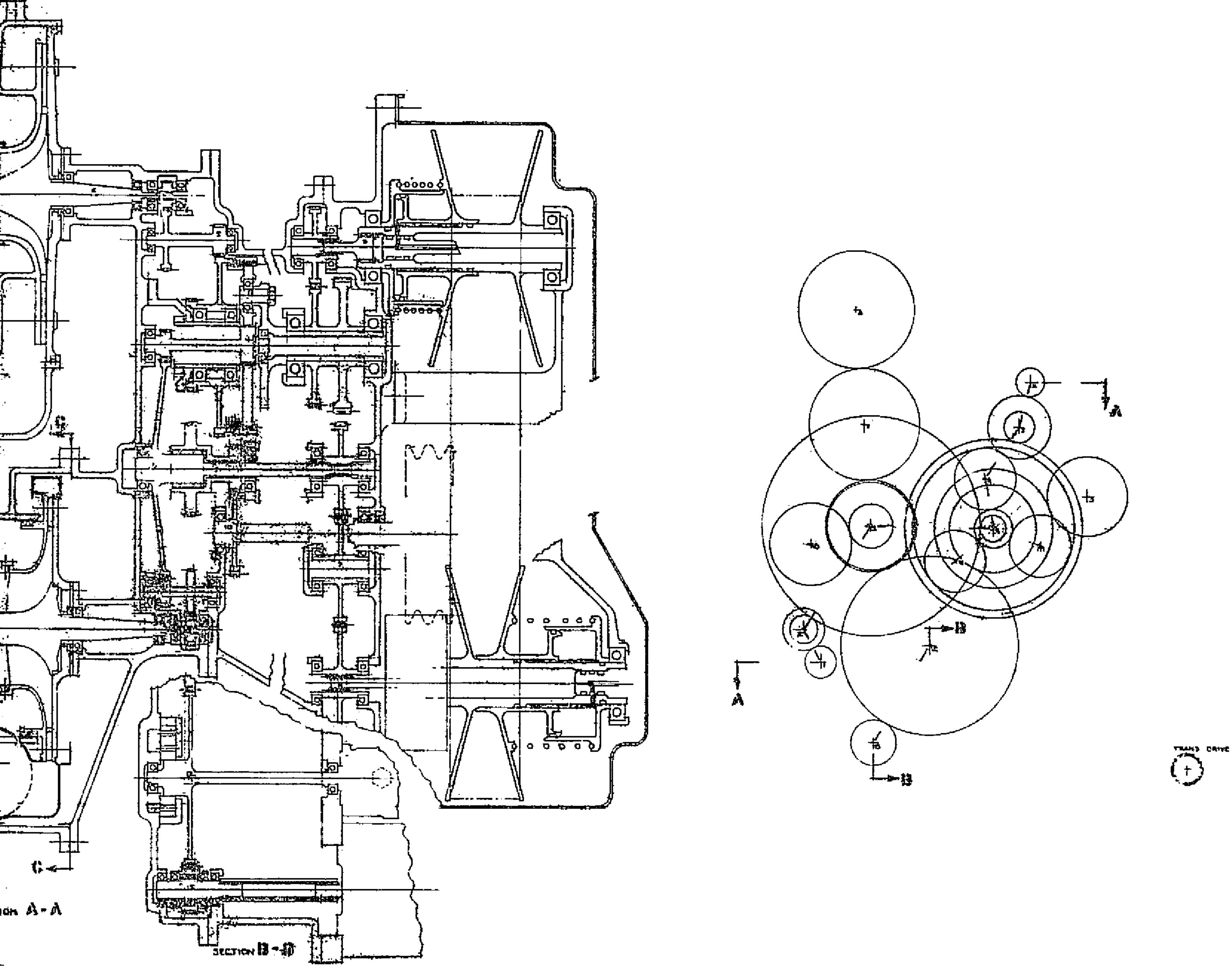

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Figure 60. General arrangement—differential engine, one regenerator, can combustor, radial-inflow turbine.

Each of the eleven engine arrangements shown in the attached sketches were packaged within the confines and constraints of the selected engine compartment with equal success, other than the exhaust problem mentioned for a two-disk installation. This problem could have been corrected by further design effort.

Installation

The installation criteria established for the IGT engine installation schemes were based on maximum commonality between a given conventionally powred vehicle and a turbine-powered vehicle.

All rigid vehicle features between the radiator and the firewall were retained in place. The radiator was assumed in place, even though a turbine engine does not require water cooling, as space provision for an engine oil cooler, transmission oil cooler, and air conditioning condenser.

The location and envelope of the transaxle, wheels, wheel suspension, and (except for the differentially geared engine) the transmission were retained. The front crossmember, side cradle members, and hood liner were not changed. The toe pan was modified as a result of the earlier studies to enhance the exhaust duct arrangement. Provision was made for vehicle accessories, such as alternator, power steering, and air conditioning.

Regenerator Size

One phase of the engine performance optimization was to determine the maximum size of regenerator disk within the confines of the vehicle engine compartment. A study was made to determine the maximum diameter disk that could be installed for 5.72 cm (2.25 in.) and for 7.62 cm (3.00 in.) thick (flow length) disks. These determinations were made separately for single-shaft and two-shaft engines, for both single-disk and twin-disk regenerators, and for the differentially geared engines with one disk. The results are listed in Table XX.

TABLE XX. MAXIMUM REGENERATOR SIZES (ESTIMATED)

	Engine				
	Single-shaft		Two-shaft		Differentially geared
Number of disks	One	Two	One	Two	One
Disk thickness, cm (in.)	5.72 (2.25)	5.72 (2.25)	7.62 (3.00)	7.62 (3.00)	7.62 (3.00)
Disk diameter, cm (in.)	58.09 (22.87)	49.53 (19.50)	56.69 (22.32)	46.99 (18.50)	53.85 (21.20)
Total area, cm^2 ($in.^2$)	1937.712 (300.346)	2651.969 (411.056)	1830.016 (361.650)	2333.221 (361.650)	1620.184 (251.129)
Total volume, cm^3 ($in.^3$)	11074.017 (675.778)	15156.002 (924.876)	13944.703 (850.958)	17779.116 (1084.948)	12345.817 (753.388)

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The regenerator limitations were found to be common for the single-shaft and the two-shaft engines. The maximum size for a single disk was found with the disk located transversely in the engine compartment and inclined approximately 45 degrees from the vertical, being limited by the hoodline and by vehicle features located at the engine compartment rear wall. The maximum size for two disks was obtained with the disks located transversely, front and back of the engine, and inclined approximately 20 degrees either way from vertical. Diameters for these were limited by the right cradle member and the transmission mounting flange.

For the differentially geared engine, the maximum single disk was located vertically along the right side of the engine compartment, being limited by the transaxle, the front crossmember, and the hoodline.

Exhaust

One of the important aspects of the early design studies was the coordination of engine design and vehicle design to provide a reasonable exhaust duct with minimum performance losses for the total engine/vehicle installation. The results of this effort are reported in the subsequent engine and vehicle design discussions.

Conclusions

The result of the design studies covering the matrix of engines considered was that all could be accommodated in a reasonable manner within the constraints of the engine compartment. The several schemes were evaluated based on consideration of the following items:

- Duct--Compressor to regenerator
- Duct--Regenerator to combustor
- Duct--Combustor to turbine
- Duct--Turbine to turbine if applicable
- Duct--Turbine to regenerator
- Duct--Regenerator to exhaust duct
- Structural continuity
- General envelope
- Assembly and major maintenance accessibility
- Accessory arrangement
- Impact on vehicle arrangement

No significant advantage was determined for any of the configurations; therefore, the candidate engine would have to be selected for other than mechanical and installation reasons.

CERAMICS POTENTIAL--TURBINE ROTOR

Structural design limits for uncooled metal radial turbines restrict the maximum rated turbine inlet temperature to 1049°C (1920°F) for a single-shaft engine and 1080°C (1976°F) for a two-shaft engine. To meet the engine performance goals, turbine inlet temperatures of the order of 1290°C (2350°F) are required. Therefore, the following study was made to determine the feasibility of ceramic radial turbines in the IGT engine with a maximum rated cycle temperature of 1290°C (2350°F). Radial turbines (Figure 61) were aerodynamically defined to meet the single-shaft and two-shaft engine work requirements. For the single-shaft engine, the required turbine tip speed is 655.3 m/s (2150 ft/sec). For the two-shaft engine, 506.9 m/s (1650 ft/sec) is required. To facilitate the evaluation of ceramic configurations of these rotors, an existing finite element computer model of the AGT-2 rotor was utilized with the material changed to hot-pressed silicon nitride (HPSN) with a characteristic strength of 410 MPa (59.4 ksi) and a Weibull exponent of 12. The AGT-2 radial-inflow turbine was designed to operate at a 594.4 m/s (1950 ft/sec) tip speed with an inlet temperature of 1038°C (1900°F). Steady-state and transient temperatures were then estimated for the AGT-2 ceramic rotor and were used to calculate maximum steady-state and transient stresses. The maximum principal stresses calculated in the blades and wheel are located as shown in Figure 62 and have values as shown in Table XXI.

To estimate maximum stresses in the ceramic turbines for the IGT, the analysis rotor speed was ratioed to 58,436 rpm for the single-shaft radial turbine at a 655 m/s (2150 ft/sec) tip speed and to 44,850 rpm for the two-shaft radial turbine at a 503 m/s (1650 ft/sec) tip speed. Maximum transient temperatures, along with the appropriate rotor speed, were used to calculate maximum transient principal stresses. The results are shown in Table XXII.

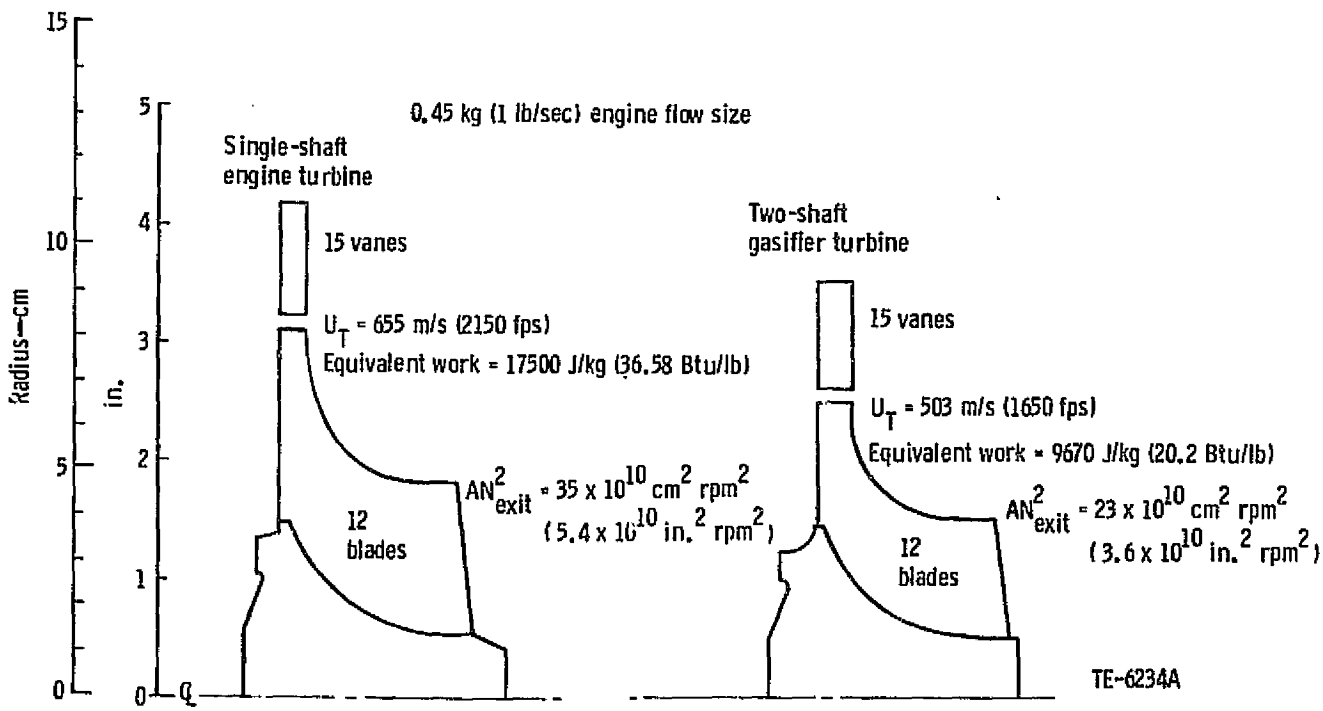

Figure 61. Radial turbine comparison.

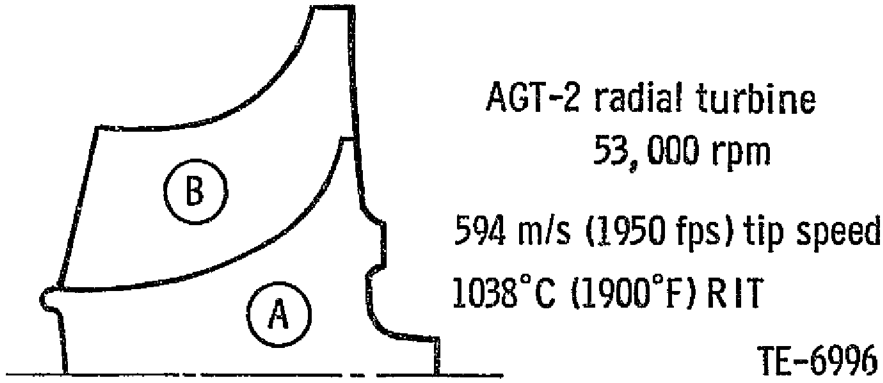

Figure 62. Maximum principal stress locations on ceramic radial turbine.

TABLE XXI. MAXIMUM PRINCIPAL STRESSES, MPa (ksi)

Location	A	B	CPS*
Mechanical	147.7 (21.4)	165.5 (24.0)	0.9999
Mechanical + thermal, steady state	190.4 (27.6)	170.4 (24.7)	0.9994
Mechanical + thermal, transient	236.0 (34.2)	230.5 (33.4)	0.9981
*CPS = Cumulative Probability of Survival			

TABLE XXII. MAXIMUM TRANSIENT STRESSES, MPa (ksi)

	Wheel	Blade	CPS*
Single-shaft	267.0 (38.7)	264.3 (38.3)	0.9901
Two-shaft	194.6 (28.2)	186.3 (27.0)	0.9998

*Cumulative Probability of Survival using:
- Material characteristic strength, σ_0 = 410 MPa (59.4 ksi)
- Material Weibull exponent, m = 12

These results indicate a failure rate of 1/100 for the single-shaft and 2/10,000 for the two-shaft ceramic turbine if they are made of HPSN.

HPSN is the strongest engineering ceramic material presently available. Components made of the material would be machined (diamond ground) from hot-pressed blocks. The part cost would appear expensive but the capability is technically available today.

A number of ceramic manufacturers have been requested for recommendations as to material, manufacturing method, costs (experimental and production), development funding, and achieveable strength level for the IGT ceramic radial turbine. Their response to date has indicated that a strength range of 518 to 690 MPa (75 to 100 ksi) (M.O.R. specimen) consistent with a Weibull exponent range of 15 to 20 could be achieved by 1983 for ceramic materials other than HPSN. These could be produced by injection molding or precision casting.

With these projected strength levels, a two-shaft ceramic radial turbine is much more feasible than a single-stage ceramic radial turbine for a single-shaft engine. Figure 63 shows the probability of failure for the IGT single-shaft and two-shaft radial turbines with current and projected ceramic material strengths.

Conclusions

1. A ceramic radial turbine with a tip speed of 502.9 m/s (1650 ft/sec) appears feasible for the two-shaft engine.
2. A ceramic radial turbine with a tip speed of 655.3 m/s (2150 ft/sec) does not appear to be feasible for the single-shaft engine without an accompanying high risk of failure.
3. Estimated maximum transient principal stresses (mechanical and thermal) for the two-shaft ceramic radial turbine are 186.3 MPa (27 ksi) in the airfoils and 194.6 MPa (28.2 ksi) in the wheel.
4. Estimated maximum stresses for the single-shaft ceramic turbine are 264.3 MPa (38.3 ksi) in the airfoils and 267 MPa (38.7 ksi) in the wheel.

Advertised properties:	HPSN Current	SiC Current	SiC 1983
Strength, MPa (ksi)	800 (116)	473 (68.6)	640-680 (93-98)
Weibull modulus (m)	12	12.3	18-20
Weibull strength, MPa (ksi) (So; k = 1.0, V = 1.0)	410 (59.4)	252 (36.6)	395-417 (57.3-60.4)

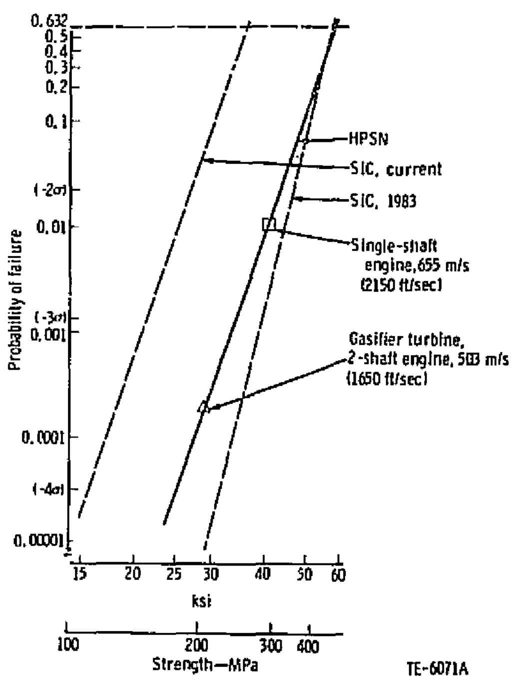

Figure 63. Ceramic radial turbine probability of failure.

COMPARATIVE RISK ASSESSMENT

High-volume production and marketing of passenger cars is a business venture containing many potential risks of great magnitude. It is thus clear that an alternative powertrain must attempt to minimize risk. This was accomplished on two fronts: realistic assessment of design and performance potential and relative risk of candidate concepts.

The first approach is implemented by designing low risk into the turbine powertrain. Since the IGT goals are beyond currently demonstrated turbine powertrains, some degree of risk is inherent. The real measure of risk is the probability of success (one minus the probability of failure). The DDA/PMD concept maximizes the probability of success by basing the design on adding realistic improvements to current state of the art and conservatively adjusting existing design experience to the different conditions unique to an advanced passenger car (especially small, low-airflow components). Ceramic technology currently being generated in parallel programs is being applied to the IGT design.

The second approach involves a comparative risk assessment of the candidate concepts to allow identifying the one with the lowest inherent risk level. The following discussions address various areas of relative risk. Because the differential engine concept was rejected for inferior fuel economy, the risk comparison is confined to two powertrains: (1) single-shaft engine and continuously variable transmission and (2) two-shaft engine and conventional automatic transmission. Only those areas where risks are different are addressed in this section; additional factors of risk for the selected concept are presented in Section VI.

Transmission Efficiency

The DDA studies indicated the conventional automatic transmission to have a higher efficiency level than the best CVT. This analysis was basedon careful consideration of the various loss mechanisms and various types of CVT designs and projections of the efficiencies attainable from advanced development. This conclusion also is intuitively correct; the added capability of continuous variability surely carries a "price tag" of lowered efficiency.

The higher efficiency of the conventional transmission is one of the reasons for projecting the two-shaft powertrain fuel economy to be slightly superior to that of the single-shaft powertrain. What if this transmission efficiency difference did not materialize? What if both transmissions were improved to their estimated maximum potentials? Table XXIII summarizes the results of this analysis.

The following conclusions are apparent:

- A single-shaft engine equipped with an improved CVT (same efficiency level as a conventional automatic) would have essentially the same fuel economy as the two-shaft engine equipped with a conventional automatic.
- The two-shaft fuel economy superiority (over single-shaft) becomes greater if their transmissions are developed to reach their estimated maximum efficiency potential.

TABLE XXIII. DRIVING TRANSMISSION EFFICIENCY COMPARISON		
		Relative fuel economy
Single-shaft engine	Best CVT transmission belt	97.5
	Same transmission efficiency as conventional automatic*	100.5
Two-shaft engine	Conventional automatic transmission	100 (Base)
	Improved automatic transmission	106
*Estimated maximum potential of CVT.		

Engine Configuration

From a system or parts-count standpoint, the two-shaft engine is more complex than the single-shaft engine. It has more shafts, seals, and bearings plus an additional turbine. However, these items are all known quantities supported by a wealth of background and experience. Simply using more of them does not increase the risk of program delay or failure, and two-shaft engine experience is much greater than single-shaft experience.

The single-shaft engine may have greater interface problems because the complete engine rotor with its relatively high inertia, the CVT and the remaining driveline are mechanically linked into one system. Such systems can cause high shaft loading from rapid transients and vehicle-induced shock loads. The two-shaft design isolates the power turbine and transmission system from the high-inertia compressor rotor system with a "soft" and forgiving aerodynamic coupling.

The single-shaft engine's turbine is more highly loaded aerodynamically because it provides the total work of expansion (i.e., a single-stage turbine). The two-shaft engine turbine system is more lightly loaded, having two stages of expansion, and is therefore less subject to the risk of falling below the goal efficiency level.

On balance, the single and two-shaft engines are judged to have similar risk levels.

Transmission Configuration

As the name implies, conventional automatic transmissions are virtually riskless. They exist and have an impressively large background of successful experience. This includes FWD transmissions, and even greater FWD experience will probably accrue during the next few years, whether or not turbine engines evolve. No significant development or start-up costs will be required if conventional automatic transmissions are employed in the IGT powertrain.

Although a CVT seems within the limits of technology, no design has been developed. Compared with the conventional automatic, the CVT thus possesses a very real risk of program delay. Further, failure to achieve efficiency goals could erode real improvements made in the turbine engine. Even if the CVT is successful technically, it would require a considerable amount of capital investment.

The CVT introduces risk unnecessarily. This added risk, both technical and financial, could be the item which leads to failure of turbine powertrain acceptance for passenger car use.

Ceramic Rotor Potential

The single-shaft engine turbine is more highly stressed (than the gasifier turbine of a two-shaft engine) because of its greater expansion ratio and the resulting higher tip speed.

Analysis indicates that a single-shaft ceramic turbine rotor is not feasible without an accompanying very high risk level. Solutions to this problem are to (1) initially design the single-- shaft engine with a two-stage turbine or (2) add a second turbine stage if technical problems arise. Neither of these alternatives is acceptable. The first eliminates the only advantage the single-shaft engine ever had over a two-shaft engine; the second approach introduces serious program delay and overrun probabilities.

Comparative Risk Summary and Conclusions

Table XXIV summarizes the areas where significant risk differences between the two candidate powertrains are judged to exist.

TABLE XXIV. RISK SUMMARY

	Lowest risk	Highest risk
1. Transmission efficiency	Conventional AT	CVT
2. Engine configuration	Equal	
3. Transmission configuration	Conventional AT	CVT
4. Ceramic rotor potential	Two-shaft	Single-shaft

The following conclusions have been reached:

- The powertrain consisting of a two-shaft engine and conventional automatic transmission has the lowest overall risk level.
- The single-shaft engine/CVT powertrain has the highest risk level in almost every individual category.
- Risk in the categories of transmission efficiency (1) and ceramic rotor potential (4) threatens the achievement of the thermal efficiency goal.
- Risk in the categories of engine configuration (2) and transmission configuration (3) threatens the program schedule and cost.

EMISSIONS

The emission goals of the IGT engine are listed in Table XXV.

TABLE XXV. EMISSION GOALS		
	g/mi	g/km
Hydrocarbon	0.41	0.26
Carbon monoxide	3.40	2.11
Oxides of nitrogen	0.40	0.25

These emission goals are in accord with the desired 90% pollutant reduction levels outlined by the Clean Air Act Amendments of 1970. In November 1973, a DDA low-emission combustor was used in an automotive gas turbine engine chassis dynamometer test and demonstrated that it would meet these very low automotive emission requirements. The results of this work were published in an SAE paper (ref 4) which was presented at the Automotive Engineering Congress and Exposition in February 1975.

An "emission principle" emerged from the GM-funded combustor development program which was essential for meeting the very low emission levels. As reported, the low-emission combustor used in this demonstration incorporated three basic features:

1. Prevaporized-premixed fuel and air
2. Lean F/A mixture within the reaction zone, having a maximum primary zone temperature of only about 1371°C (2500°F) for all combustor outlet temperatures
3. Provision for maintaining the low reaction zone temperature, as the overall engine fuel/air ratio and the combustor outlet temperature vary over the power range, by use of variable geometry or staged fuel concepts

The new challenge will be to maintain the emission performance at the higher combustor outlet temperatures and the smaller flow sizes required by the engine to meet the fuel economy goals.

The combustor design conditions were monitored during the powertrain performance optimization studies to determine effects on emission performance. The cycle trends were toward higher combustor inlet and outlet temperatures, reaching levels of 927°C (1700°F) and 1288°C (2350°F), respectively. The higher temperatures are favorable for decreased quantities of both hydrocarbons and carbon monoxide but could result in increased levels of oxides of nitrogen. Formation of oxides of nitrogen is an exponential function of reaction zone temperature and approximately a linear function of residence time. Reaction zone temperature may be controlled by variable geometry up to the point of equaling the turbine inlet temperature. Residence time is a variable of combustor dimensions.

The predicted emission performance for the IGT design conditions were studied, using a theoretical reaction rate kinetics model and sizes planned for the IGT. A typical plot of the Emission Index of CO and NO_x versus reaction zone temperature at a simulated engine operating condition is shown in Figure 64. For reference, the Emission Index that corresponds to the

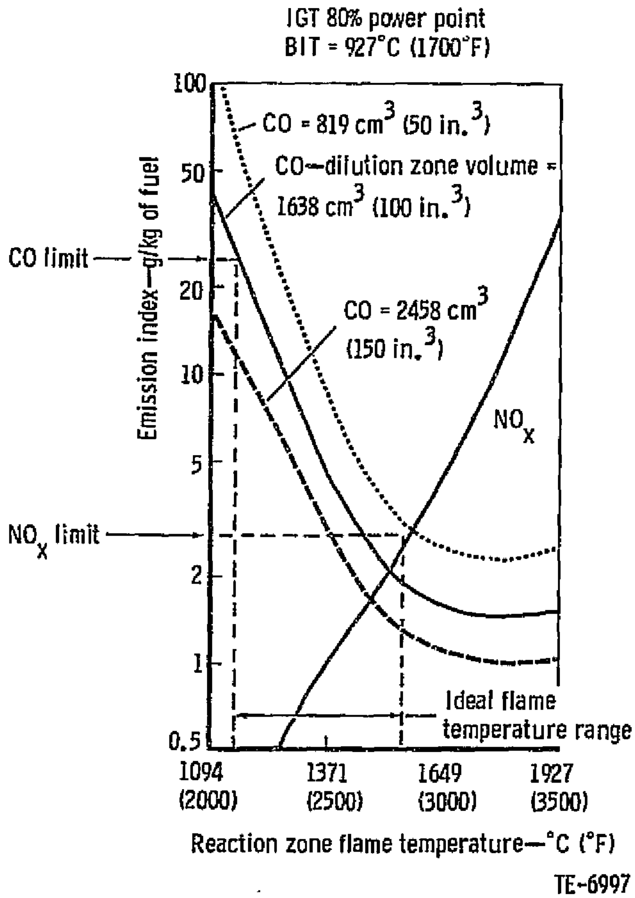

Figure 64. Predicted emissions using reaction kinetics model.

emission goals are shown. The g/mi units of the standards can be converted to EI by using the following relationship:

$$g/mi = \frac{0.454\ D_f\ (EI)}{MPG}$$

where D_f = fuel density, lb/gal
MPG = vehicle fuel economy, mi/gal
EI = emission index, g/kg fuel

The conversion from the emission standard to allowable Emission Index values is effected as follows:

Pollutant	Emission standard		Emission Index
	g/mi	(g/km)	g/kg of fuel
Hydrocarbon	0.41	(0.26)	2.93
Carbon monoxide	3.40	(2.11)	24.26
Oxides of nitrogen	0.40	(0.25)	2.85

The preceding values are based on the density of Diesel fuel No. 2 of 0.85 kg/l (7.1 lb/gal) and a fuel economy of 9.8 km/l (23 mpg) over the Federal Driving Cycle. Diesel fuel No. 2 was used because it is the most probable fuel that would be used in volume production.

Using the theoretical reaction kinetics emission predicted model, it was concluded that all of the engine types under review could meet the emission objectives. The studies showed that ideally the reaction zone flame temperature window might range from 1149°C (2100°F) to 1566°C (2850°F). This temperature range is slightly more than demonstrated with actual hardware and probably indicates that some incomplete mixing occurs in the real case. Because of the narrow flame temperature window, actual or ideal, it may not be possible to eliminate the requirement for variable combustor geometry of some form.

SAFETY AND NOISE

The safety of a turbine powertrain will equal the already high level enjoyed by the spark-ignition engine-powered vehicle. The high energy possessed by the rotors (compressor and turbine) of the turbine engine presents a potential hazard in the case of rotor failure. However, this has long been solved by designing for rotor containment. The inherently clean exhaust gas poses no problem even if leakage of exhaust gas were to occur. There is no known reason to expect the turbine powertrain to offer anything less than the safest of operating service.

Likewise, noise levels with a turbine powertrain will be attractively low. In shaft-power applications, this is one of the commonly accepted advantages of a turbine over piston-type powerplants. The air cleaner will shield high-frequency compressor noise from reaching the environment. Mufflers, per se, are not required to silence exhaust noise. The exhaust duct (larger than for spark-ignition engines) will provide adequate suppression of exhaust noise (steady-flow noise as opposed to the individual cylinder "explosions" fundamental to piston engines).

ESTIMATED PRODUCTION COST DIFFERENCES

A 1974 General Motors program involving Buick Motor Division, Hydra-matic, DDA, and other divisions encompassed the design, prototype fabrication, and high-volume production cost estimate of a variable-geometry automotive gas turbine engine (AGT-2) and Toric CV transmission. This data base served a starting point for comparing the two candidate powertrains of the current study: (1) single-shaft engine and belt CV transmission and (2) two-shaft engine and conventional automatic transmission. The differential engine powertrain was not costed because its poor fuel economy disqualified it from further consideration. Because of the proprietary nature of cost data and the realization that these estimates are not based on part-by-part costs of each powertrain, the results are presented in percentage format.

Table XXVI presents the relative costs of the candidate powertrains. Column one is the two-shaft engine with conventional automatic transmission, and column two is the single-shaft engine with belt CV transmission. The third column represents a single-shaft engine with two-stage turbine, a configuration which might be required to accommodate a ceramic turbine rotor design.

TABLE XXVI. RELATIVE COSTS OF CANDIDATE POWERTRAINS			
		Single-shaft	
	Two-shaft	1-stage turbine	2-stage turbine
Engine less turbine system	55.5	55.5	55.5
Turbine system	26.5	19	24
Engine	82	74.5	79.5
Transmission	18	22.5	22.5
Powertrain	100	97	102

Based on the referenced 1974 study, the turbine system of the conventional single-shaft engine accounts for approximately 25% of the engine cost (19 units out of 74.5 units). The remaining engine cost (55.5 units) is identical for both the two-shaft engine and the single-shaft engine with a two-stage turbine. However, both of these engines have turbine systems that differ from that of the conventional single-shaft engine. The two-shaft engine turbine system comprises two turbines, but its overall expansion ratio per turbine is lower, each turbine will be smaller and require less material. Therefore, the unit cost per turbine is estimated to be 30% lower for the two-shaft engine than for the single-shaft turbine. Thus, the two-shaft turbine system consists of two 70% turbines or a total cost factor of 140% (19 units x 140% = 26.5 units). The increased turbine cost causes the two-shaft engine cost to rise to 110% of the one-shaft engine (82 units vs 74.5 units). As mentioned elsewhere in this report, the feasibility of a ceramic turbine rotor for a conventional single-shaft engine is questionable because of the high tip speed and stress level. A two-shaft engine has lower work per turbine stage, resulting in considerably lower stress levels. The single-shaft engine may require similar stress levels; the only method to achieve this is by adding a second turbine stage. The cost of a two-stage turbine for the one-shaft engine is estimated to be 24 units, a value that falls between the two-shaft engine and the conventional single-shaft engine.

The total estimated engine cost is lowest for the conventional single-shaft engine and highest for a two-shaft engine. The cost of the two-stage turbine version of the single-shaft engine falls between the two.

The cost of a conventional automatic transmission was added to the two-shaft engine for a total powertrain cost of 100 units. The relative engine vs transmission costs were based on the same time frame of production. The variable belt CV transmission was estimated to have a 25% higher production cost than the conventional automatic transmission (18 units x 1.25 = 22.5 units).

Total powertrain costs are very close for all candidates. Within the accuracy of estimation, no significant difference in production cost is judged to exist.

Engine development costs are estimated to be similar for the two-shaft engine and the single-shaft engine with a two-stage turbine. Because of the higher risk associated with the ceramic turbine for a conventional single-shaft engine, its development cost will be higher.

The development cost for a conventional automatic transmission is very small, limited to its adaptation to a two-shaft engine. Significant costs are required to develop a continuously variable transmission.

The total development costs are thus lowest for the two-shaft powertrain and highest for the conventional single-shaft powertrain. The single-shaft engine with a two-stage turbine falls between these two.

POWERTRAIN SELECTION

Criteria

The contractual objectives of the IGT powertrain are to meet or exceed the following requirements:

1. A 20% improvement in powertrain thermal efficiency over that of a conventional 1976 powertrain. This shall result in at least a 20% fuel economy improvement over a 1976 model year automotive vehicle over the EPA composite driving cycle. For the study, a compact class, 1406 kg (3100 lb) curb weight vehicle shall serve as the basis of comparison.
2. The powertrains considered should be ready to enter the production engineering phase of development in 1983.
3. Reliability and life features comparable with those of powertrains currently in the market.
4. Drivability acceptable to the consumer.
5. Meet or exceed the Federal Emissions Standards of 0.4 g/mi, 3.4 g/mi, and 0.40 g/mi for $HC/CO/NO_x$.
6. Meet noise and safety levels that are currently legislated.
7. Have competitive initial cost and a life cycle cost no greater than that of conventionally powered automotive vehicles.

In addition to these stated contractual objectives, the implied contractual objective of minimum risk was added as the eighth IGT requirement. Although somewhat subjective and not quantifiable, risk minimization is an important element of any realistic decision-making process.

Requirement 1, fuel economy or thermal efficiency, is the most important payoff parameter of this study--the real reason for examining turbine powertrains.

Theoretically, the study would have generated several candidates from which the one which best exceeded the eight requirements would be selected. Tentative schemes for weighing each requirement and determining an overall "factor of merit" were defined. However, the thermal efficiency objective was sufficiently difficult to meet so that trade-off situations essentially did not exist. In practical terms, the process became one of defining powertrain conceptual designs that met the thermal efficiency goal and, simultaneously, did not violate the other requirements. Certain of the other requirements were easily met; some were more difficult.

The time frame (requirement 2) requires modification. DOE/NASA subsequently announced, at the semiannual Contractors' Coordination Meetings, revised plans which now call for an experimental IGT demonstration by 1983--the start of production engineering to follow later. The Task II study revealed that the TIT level dictated by a metal turbine rotor is too low to meet the thermal efficiency goal. This dictates the need for a ceramic rotor and thus does not allow the original 1983 Production Engineering timetable to be met with reasonable assurance.

Reliability and life (requirement 3) were staisfied by establishing stress levels and TIT limits which would allow the selected materials to deliver the necessary life and reliability levels. Drivability (requirement 4) can be achieved by sizing the engine and matching the transmission to deliver vehicle performance competitive with the 1976 base line. Emissions (requirement 5) compliance can confidently be predicted; however, the exact combustor configuration (variable vs fixed geometry, catalytic, etc) remains to be defined during combustor development. Noise and safety (requirement 6) do not pose a problem for turbine powertrains; neither do they offer trade-off possibilities for making the thermal efficiency goal easier to reach. Cost (requirement 7) was, for the selection process, confined to a comparison of the production cost differences between the candidate concepts. Overall vehicle selling price and life cycle cost analysis were a part of the vehicle characterization task (III). Risk analysis (requirement 8) was heavily weighed in making the final selection.

Selected Concept

The following summarizes the results of powertrain analysis (Task II) in terms of the criteria previously mentioned. Each of the summaries is based on material presented in Section V of the report.

- The differential engine was discarded because of inferior fuel consumption characteristics.
- Thermal Efficiency (fuel economy)--The two-shaft engine and conventional transmission is the best powertrain at 8.8 km/l (20.7 mpg); the single-shaft/CVT powertrain at 8.6 km/l (20.3 mpg) is very close behind.
- 1983 Production Engineering Readiness--Neither powertrain can meet the thermal efficiency goal with metal turbine rotors; the requirement for ceramic rotors raises doubts that this timetable can be met. The two-shaft engine can accommodate a ceramic rotor design; it is doubtful if a ceramic single-shaft engine with a single stage ceramic rotor would have an acceptable probability of success.
- Reliability and Life--The two-shaft powertrain is capable of meeting this requirement; the single-shaft engine with a single-stage ceramic rotor is judged unacceptable in this category.
- Drivability--Both the single-shaft/CVT and the two-shaft/conventional AT powertrains can meet the requirement; however, the single-shaft/CVT will be the most difficult.
- Emissions--Either powertrain concept can meet the emissions standard.
- Noise and Safety--Either powertrain concept can meet the noise and safety requirement.

- Production Cost--Production cost differences between the single-shaft/CVT and the two-shaft/conventional automatic transmission powertrains were, within estimating accuracy, nonexistent. If the single-shaft engine requires a two-stage ceramic turbine, its production cost would be marginally higher.
- Risk--Relative to the single-shaft/CVT powertrain, the two-shaft engine with conventional automatic transmission powertrain has lower or equal risk in every category, thus the lowest overall risk.

The two-shaft engine, conventional transmission powertrain is selected as being appropriate for development. It is superior to the single-shaft/CVT alternative in most requirements; in no instance has it been found to be inferior.

VI. IGT POWERTRAIN OPTIMIZATION AND PERFORMANCE (TASK IIIA)

The results of the Task II study with metal turbine rotors indicated that higher turbine rotor inlet temperatures would be required to meet the goal of a 20% thermal efficiency improvement over the base-line vehicle specified in the contract. The two-shaft engine with all-metal turbine rotors did not meet the goal. The optimization of various parameters to meet or exceed the goal is discussed in this section.

ENGINE PERFORMANCE OPTIMIZATION FOR FUEL ECONOMY IMPROVEMENT

A number of parameters that will result in fuel economy improvement can be investigated. The following is a listing of those that were studied at this time:

- o Gasifier turbine rotor inlet temperature
- o Compressor pressure ratio (R_c)
- o Regenerator disk size and the number of disks
- o Radial versus axial power turbine
- o Fixed versus variable geometry
- o Idle speed
- o Sensitivity study
- o Final selection

Each of these items is discussed in the following paragraphs.

Turbine Rotor Inlet Temperature

The following temperature ranges were investigated:

- o 1080°C (1976°F)--all-metal turbine rotors
- o 1288°C (2350°F)--ceramic gasifier turbine rotor
- o 1371°C (2500°F)--both rotors ceramic

These temperatures represent the maximum temperature rating for the engines. The governing temperature limits and the resulting driving cycle fuel economy at 29°C (85°F), 152 m (500 ft) ambient conditions are shown in the following tabulation:

Maximum TIT	Continuous TIT*	Maximum PTIT**	Maximum Regenerator	Fuel economy
1371°C (2500°F)	1315°C (2400°F)		1066°C (1950°F)	10.3 km/l (24.3 mpg)
1288°C (2350°F)	1232°C (2250°F)	1010°C (1850°F)		9.7 km/l (22.8 mpg)
1080°C (1976°F)	1024°C (1876°F)			9.2 km/l (21.6 mpg)

*TIT--gasifier turbine inlet temperature
**PTIT--power turbine inlet temperature

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All engines have a 4.5 compressor pressure ratio, single regenerator disk, and THM 350 transmission. A comparison of the part-load minimum sfc is shown in Figure 65. The goal of 20% requires 10.0 km/l (23.5 mpg) on the combined driving cycles. The all-metal engine listed here shows an improvement over the value achieved from Task II because of the effects of using one regenerator disk instead of two. However, it is still 8% below the goal. The 1288°C (2350°F) maximum temperature engine is 3% below the goal. However, further optimization of other parameters is expected to show that the 1288°C (2350°F) engine can meet the 20% improvement goal. This demonstrates that the selected engine design will require a ceramic gasifier turbine rotor. If only the gasifier rotor is to be ceramic, considerable risk is removed from the development program. Therefore, the 1371°C (2500°F) engine, which requires ceramic rotors at both locations, will be dropped from further consideration. However, eventual conversion of the power turbine from metal to ceramic could add some improvement potential.

Compressor Pressure Ratio

There is an optimum pressure ratio for each level of temperature and regenerator effectiveness. As the effectiveness increases, the minimum sfc will occur at a lower pressure ratio for a given level of temperature. Conversely, if the effectiveness is held constant and the temperature is increased, the optimum pressure ratio will increase for minimum sfc.

However, there is a rather wide range of pressure ratio in either case where minimum sfc will not vary significantly. Studies have shown that a 4.5 pressure ratio is near optimum for the cycle conditions under consideration. However, at 1371°C (2500°F), both 4.5 and 5.0 pressure ratio engines were evaluated. Performance maps were calculated for each cycle and by application of the simplified method to ascertain the driving cycle fuel economy, the results favored the 4.5 pressure ratio engine by 3.3%, as shown

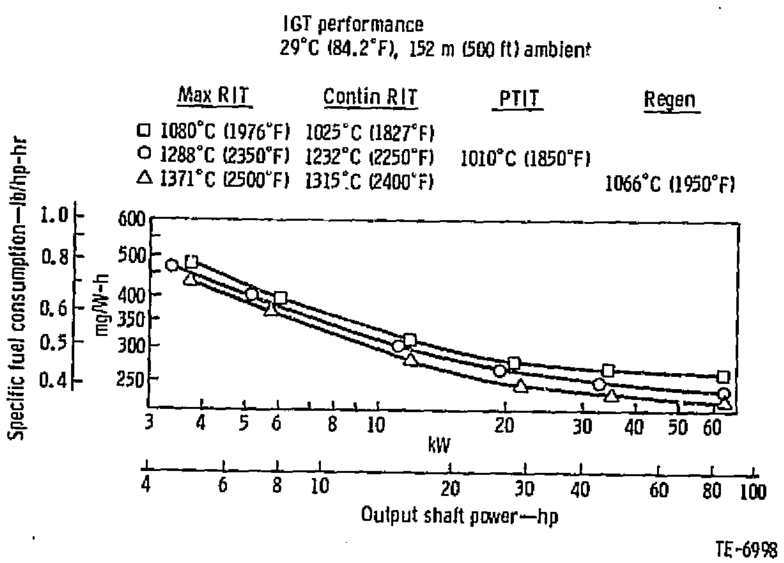


Figure 65. Effect of cycle temperature on sfc--two-shaft engine.

in Figure 66. Working gas leakage is directly proportional to compressor pressure ratio. An increase in compressor pressure ratio from 4.5 to 5.0 will increase leakage by approximately 10%. Also, compressor efficiency increases with the lower compressor pressure ratios. Improvement in leakage and compressor efficiency accounts for the improved fuel economy for the 4.5 compressor pressure ratio engine. The cycle was calculated at a 1371°C (2500°F) maximum temperature, using two 43 cm (16.9 in.) diameter regenerator disks, and a 1066°C (1950°F) regenerator hot-side inlet temperature limit, and a THM 350 transmission.

Regenerator Disk Diameter and Number of Disks

The rotating regenerator requires seals to prevent leakage of high-pressure air around the turbines. This amount of leakage is directly proportional to seal length or disk diameter and the number of disks. The pressure loss is a function of free flow area and flow length and will increase as diameter decreases. Regenerator effectiveness increases with an increase in heat transfer surface. Thus, there is a trade-off between leakage, pressure drop, and effectiveness relative to disk size and the use of one or two disks. Figure 67 shows the relationship of disk diameter, disk thickness,

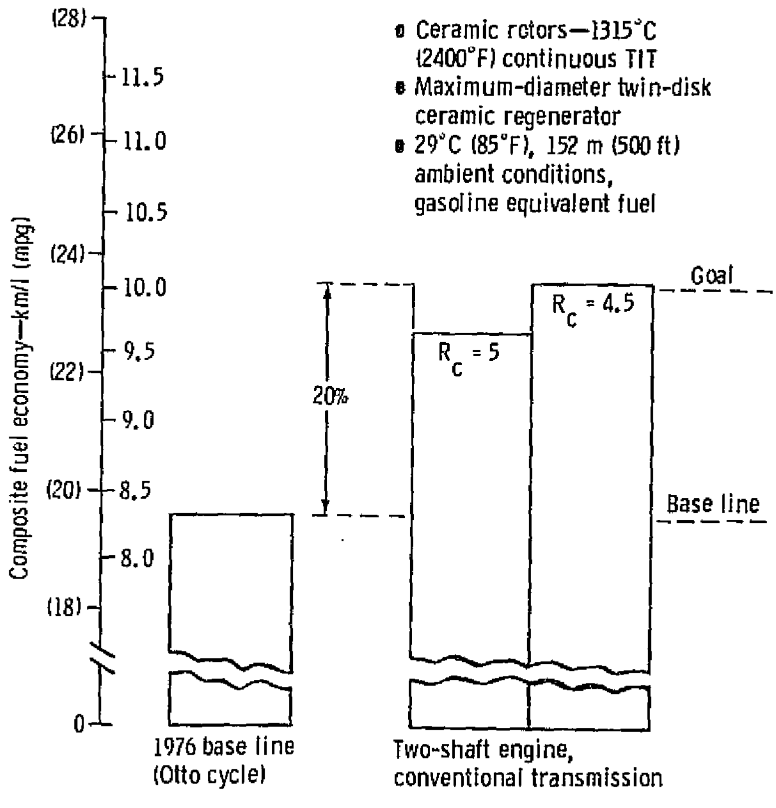


Figure 66. Effect of cycle pressure ratio on driving cycle fuel economy.

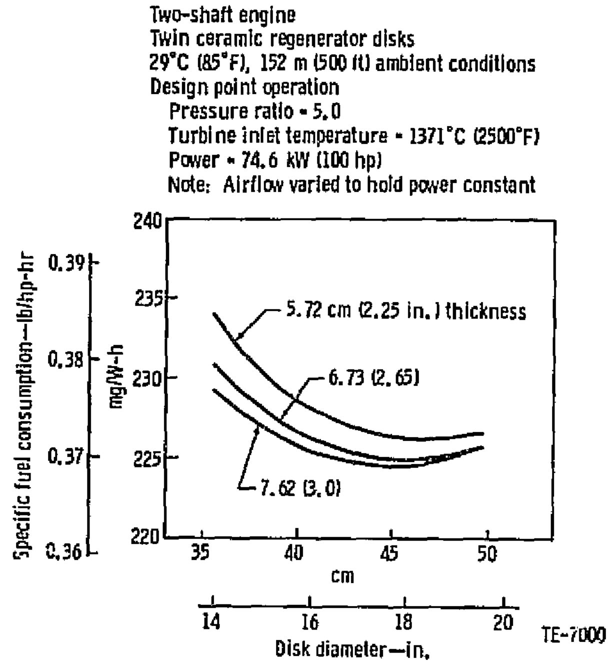


Figure 67. Effect of regenerator size on sfc.

and specific fuel consumption at the maximum power point and is indicative of trends at part load. Based on this analysis, two diameters of 35.6 cm (14.0 in.) and 43 cm (16.9 in.) at a 7.62-cm (3.09 in.) thickness were selected for comparison. Also, a single disk with the same heat transfer surface as the two 35.6-cm (14.0 in.) disks was calculated. This single disk had a diameter of 46.4 cm (18.3 in.) and was 7.62 cm (3.00 in.) thick.

Table XXVII shows how the three regenerators described affect leakage, pressure loss, and regenerator effectiveness at maximum power setting.

TABLE XXVII. EFFECT OF REGENERATOR SIZE AND NUMBER OF DISKS ON EFFECTIVENESS, PRESSURE LOSS, AND LEAKAGE AT MAXIMUM POWER

Number of disks	2	2	1
Disk diameter, cm (in.)	43 (16.9)	35.6 (14.0)	46.4 (18.3)
Regenerator effectiveness, %	96.6	94.3	94.3
Regenerator pressure loss, % $\Delta P/P$	3.66	6.00	5.83
Regenerator leakage, %	8.10	6.46	4.63

The effect of these three parameters on fuel economy for the combined driving cycle, as calculated by the simplified method, is shown in Figure 68. The 35.6-cm (14.0 in.) and 43-cm (16.9 in.) diameter disks result in similar fuel economy, but a single disk improved the fuel economy by 7.5% because of the reduced leakage.

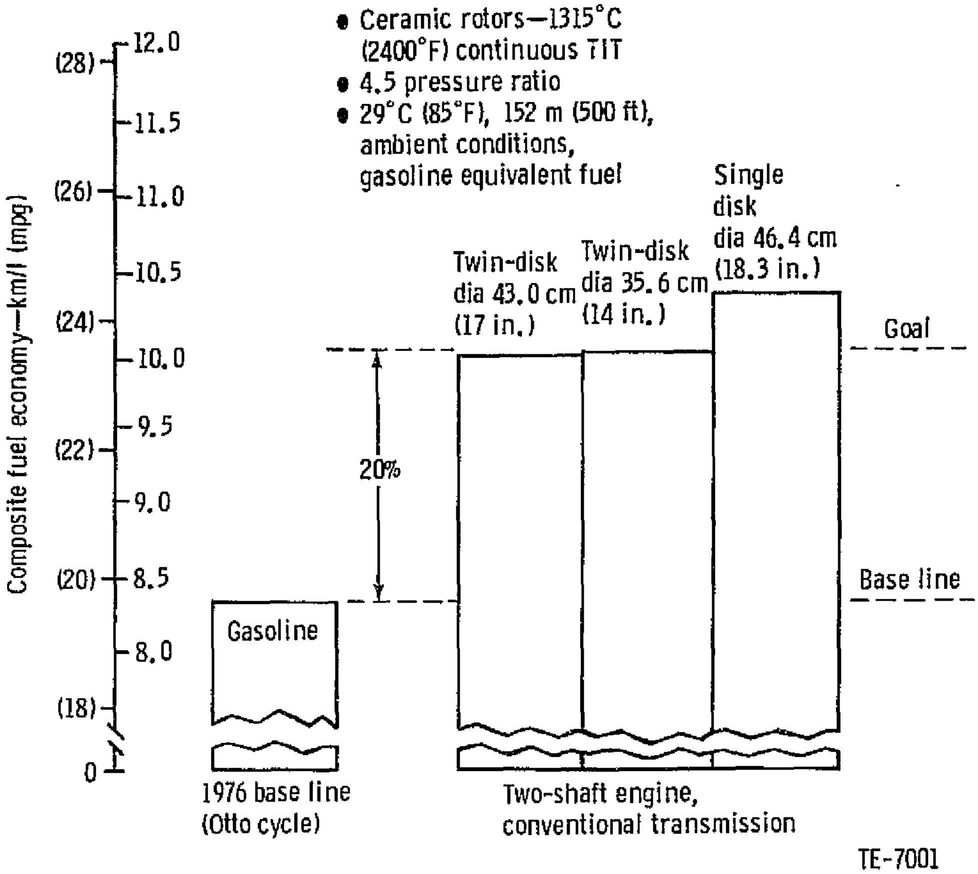


Figure 68. Effect of regenerator size on driving cycle fuel economy.

This study was completed for an engine cycle with a 4.5 pressure ratio, a 1371°C (2500°F) turbine rotor inlet temperature, a 1066°C (1950°F) regenerator hot-side inlet temperature limit, and using a THM 350 transmission.

Axial Versus Radial Power Turbine

A previous section of this report included a discussion of the radial and axial power turbine designs. Both turbines make use of variable geometry at the turbine inlet. The radial turbine performance was based on the use of variable vanes; that of the axial turbine was based on partial admission. The following is a discussion of the powertrain performance and includes engine performance maps of both types. The efficiency of the radial turbine does not drop as fast as that of the axial turbine when the flow capacity is reduced. Figures 69 through 72 are plots of the axial and radial turbine maps, respectively, in the design point geometry setting, and Figure 73 gives the lapse rate of efficiency with changes in flow setting. The minimum sfc is plotted in Figure 74, which shows that the radial turbine engine has a lower sfc at a given power output. However, the fuel economy calculated for combined driving cycles shows the axial turbine engine to be 0.7% better. This results from the shape of the power curves of the two configurations. Figure 75 shows power versus output speed for the radial turbine with the road-load curve overplotted. Note that the road-load curve does not pass through the minimum sfc points. However, the road-load curve on

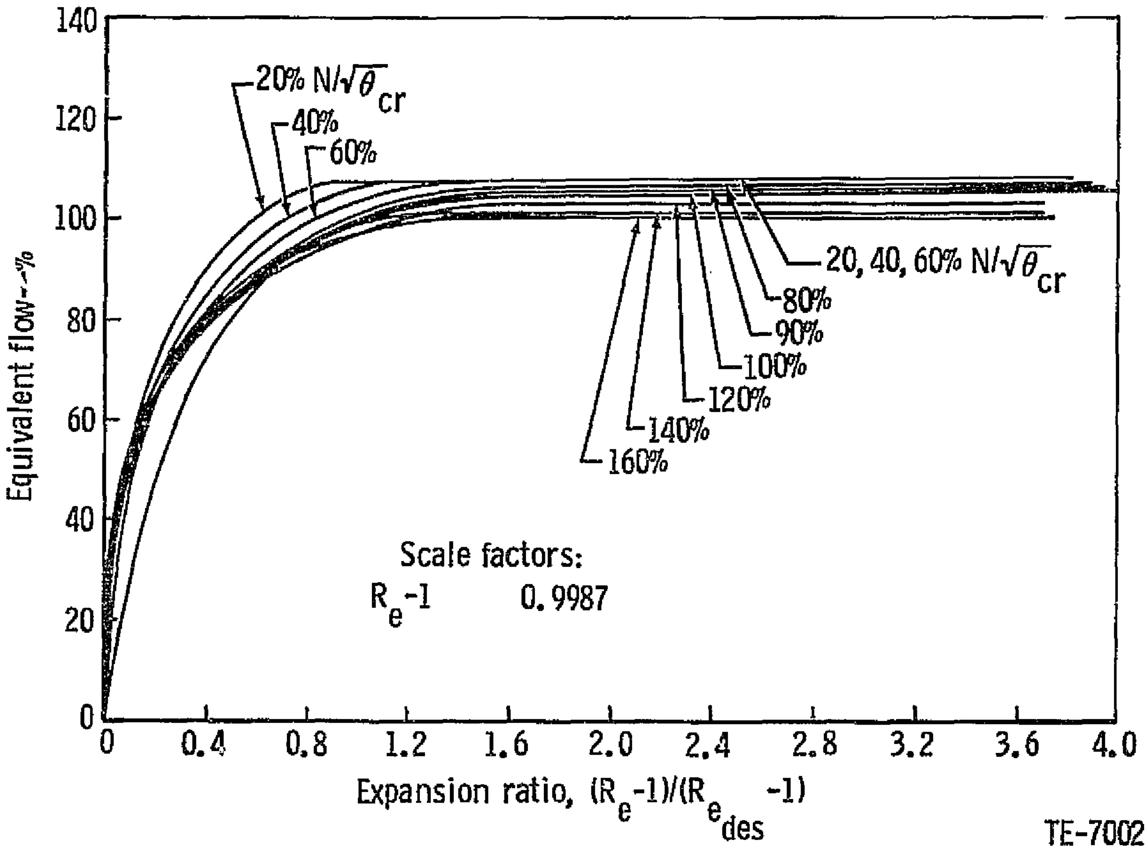


Figure 69. IGT axial power turbine map—flow versus expansion ratio at 100% flow setting.

the axial turbine performance map is much closer to minimum sfc, as shown in Figure 76. This explains why the minimum sfc for the radial power turbine engine can be better than that of the axial and still have a slightly worse fuel economy on the combined driving cycles.

There are several strategies for modifying the radial turbine engine characteristic to shift the relationship of the minimum sfc and road-load curves. A gearing change, such as axle ratio, can shift the road load curve, but this would complicate the transmission in order to maintain the same maximum road speed. Another approach is to shift or broaden the efficiency islands on the power turbine map through aerodynamic design modifications. This is the recommended approach for follow-on development programs. The radial power turbine engine configuration was selected because of its similarity to the gasifier turbine design. This should reduce the overall design and development effort, especially if the turbine inlet temperature is ever increased to a level where a ceramic power turbine is required. If, during the detailed aerodynamic design phase, the radial power turbine does not meet its performance requirement, an axial power turbine can be introduced.

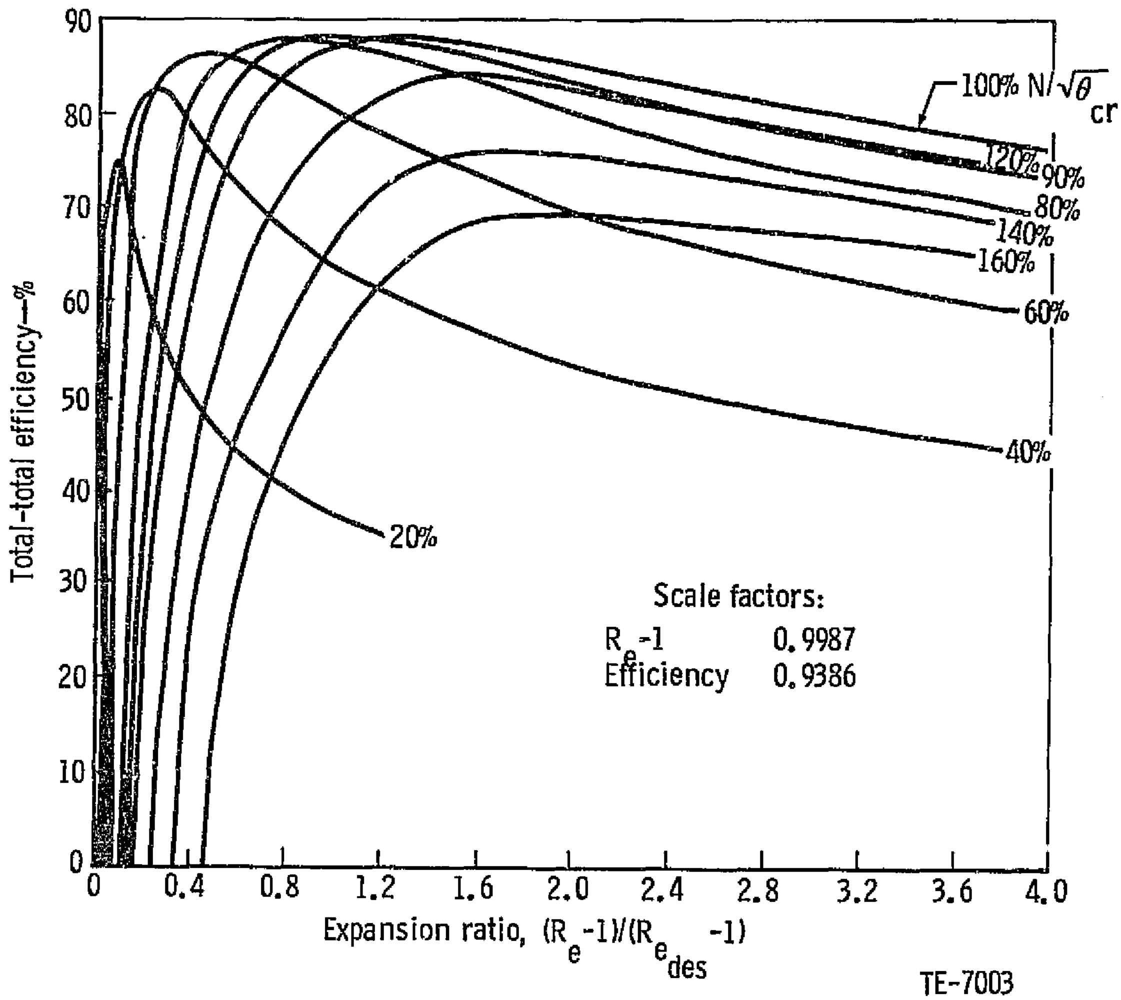


Figure 70. IGT axial power turbine map--efficiency versus expansion ratio at 100% flow setting.

Fixed and Variable Geometry

The variable-geometry engine has variable vanes at the compressor inlet and diffuser and at the gasifier turbine rotor inlet that are capable of changing the flow capacity of each component over a wide range. Component rig test results have shown that the flow can be varied over a range in excess of 3 to 1 at a given speed of rotation. A penalty on component efficiency is associated with the variable geometry. This penalty on compressor efficiency is 1.7%, and the gasifier turbine efficiency penalty is 1% at the maximum power condition. Both fixed- and variable-geometry configurations made use of variable geometry in the power turbine inlet vane to achieve part-load temperature control. Each cycle was calculated with a radial power turbine. The design point parameters at 15°C (59°F) are listed in Table XXVIII.

In this comparison, the cycle calculation was refined to better account for the leakage which would occur as the geometry in the gasifier is closed at constant speed. The quantity of flow leakage varies directly with compressor pressure. Hence, as engine airflow is reduced at constant speed and compressor pressure ratio, the percentage of leakage increases. Also,

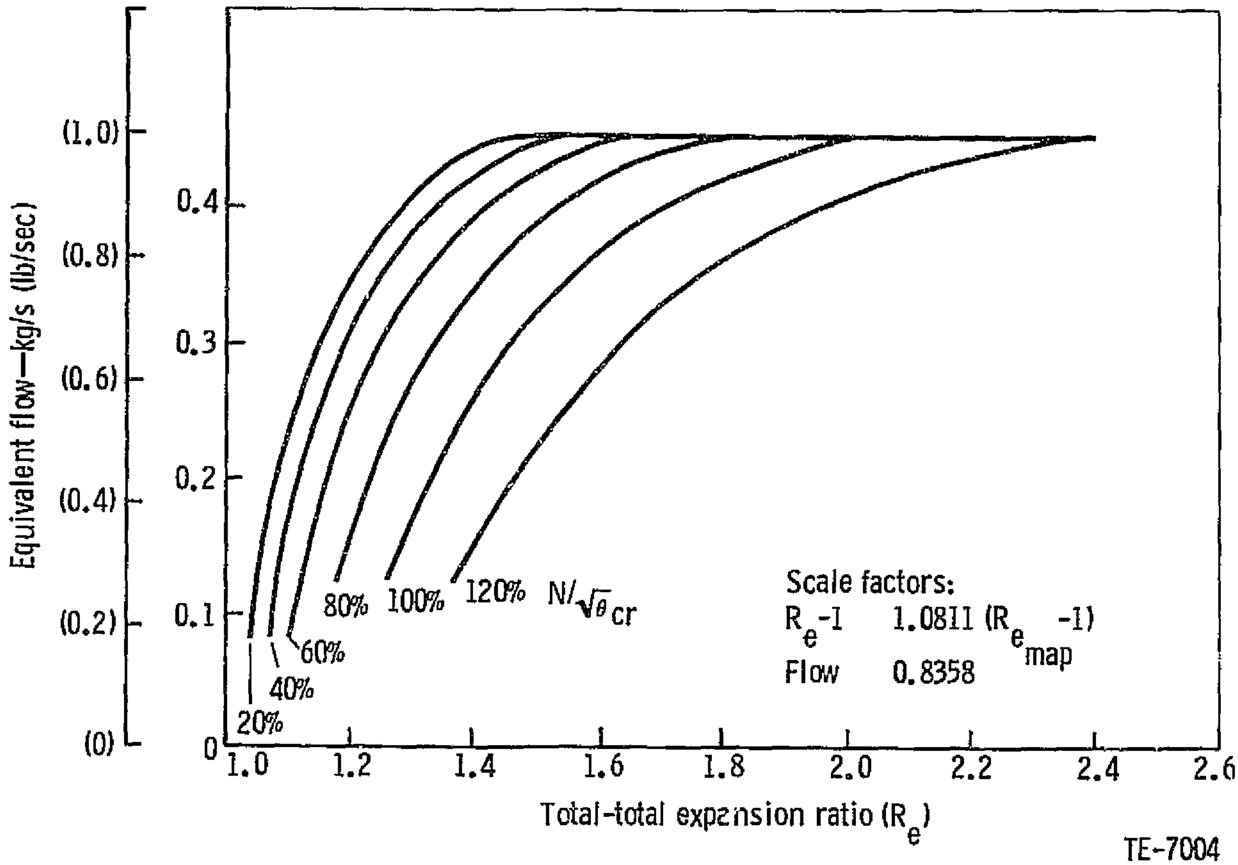


Figure 71. IGT radial power turbine map—flow versus expansion ratio at 100% flow setting.

TABLE XXVIII. COMPARISON OF FIXED- AND VARIABLE-GEOMETRY GASIFIER ENGINES AT MAXIMUM POWER, 15°C (59°F) AMBIENT

	Variable geometry	Fixed geometry
Compressor inlet pressure, kPa (psi)	99.8 (14.5)	99.8 (14.5)
Gasifier rotor inlet temperature, °C (°F)	1288 (2350)	1288 (2350)
Compressor inlet airflow, kg/s (lb/sec)	0.379 (0.836)	0.372 (0.820)
Compressor pressure ratio	4.5	4.5
Compressor efficiency, T-S	76.3	78.0
Gasifier turbine efficiency, T-T	83.7	84.7
Power turbine efficiency, T-T	84.5	84.5
Regenerator effectiveness, %	93.6	93.8
Total cycle pressure loss, %	15.4	15.4
Regenerator leakage, %	4.34	4.4
Outboard leakage, %	1.2	1.2
Turbine cooling, %	0.7	0.7
Mechanical loss + accessories, kW (hp)	7.5 (10.1)	7.5 (10.1)
Heat rejection from flow path, %*	2.5	2.5
Power, kW (hp)	75 (100.6)	75 (100.6)
Specific fuel consumption, mg/W h (lb/hp-hr)	225 (0.370)	219 (0.350)

*2.5 of Btu input is lost to heat rejection

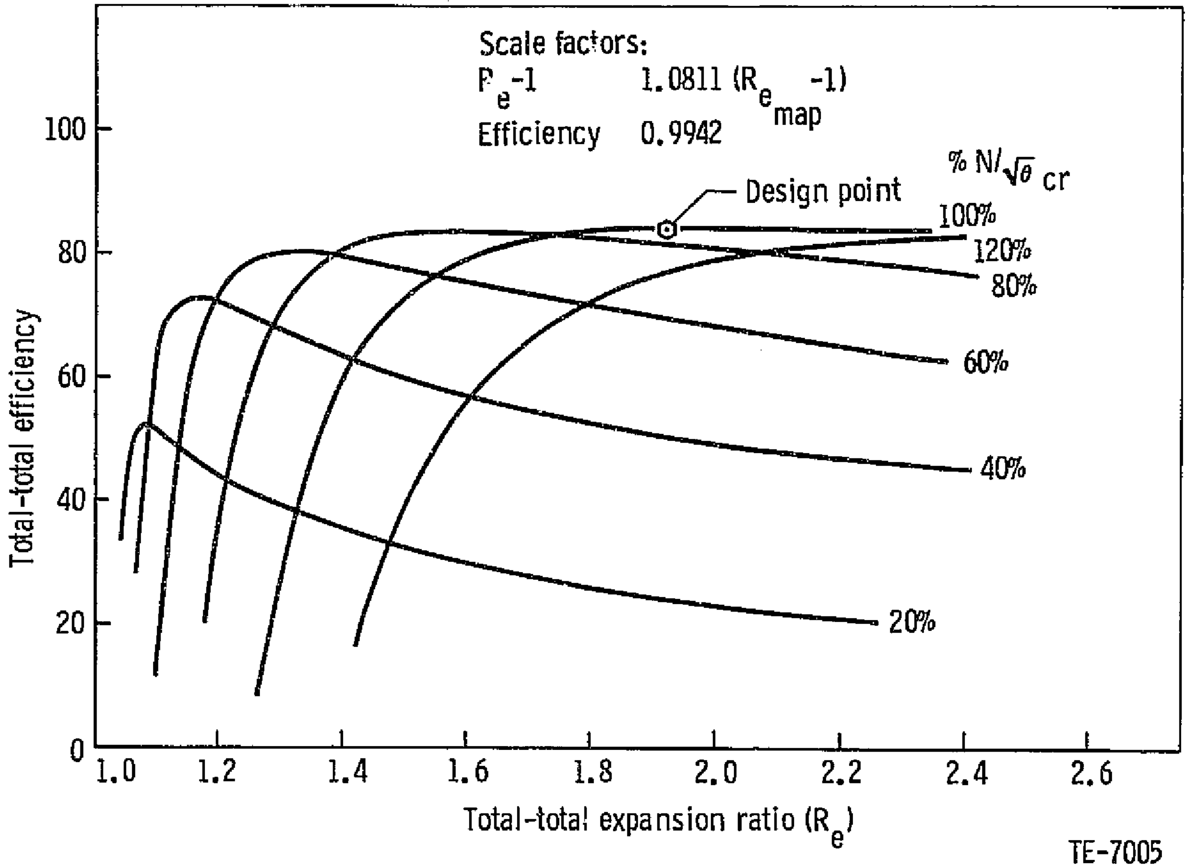


Figure 72. IGT radial power turbine map--efficiency versus expansion ratio at 100% flow setting.

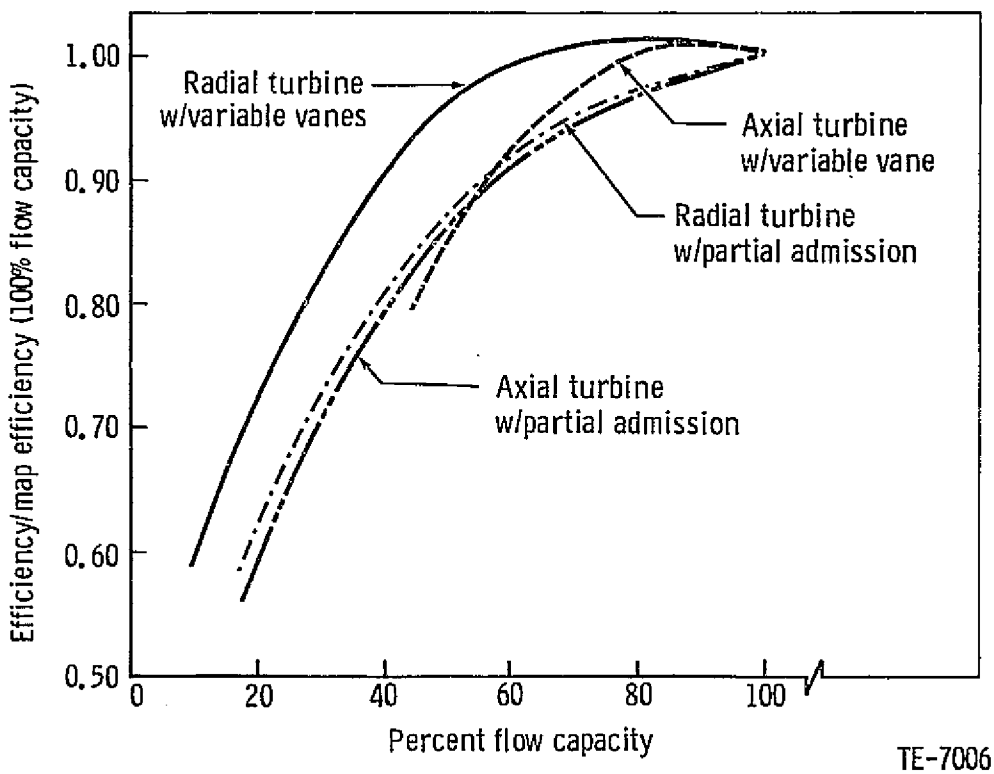


Figure 73. Effect of turbine flow capacity setting on efficiency.

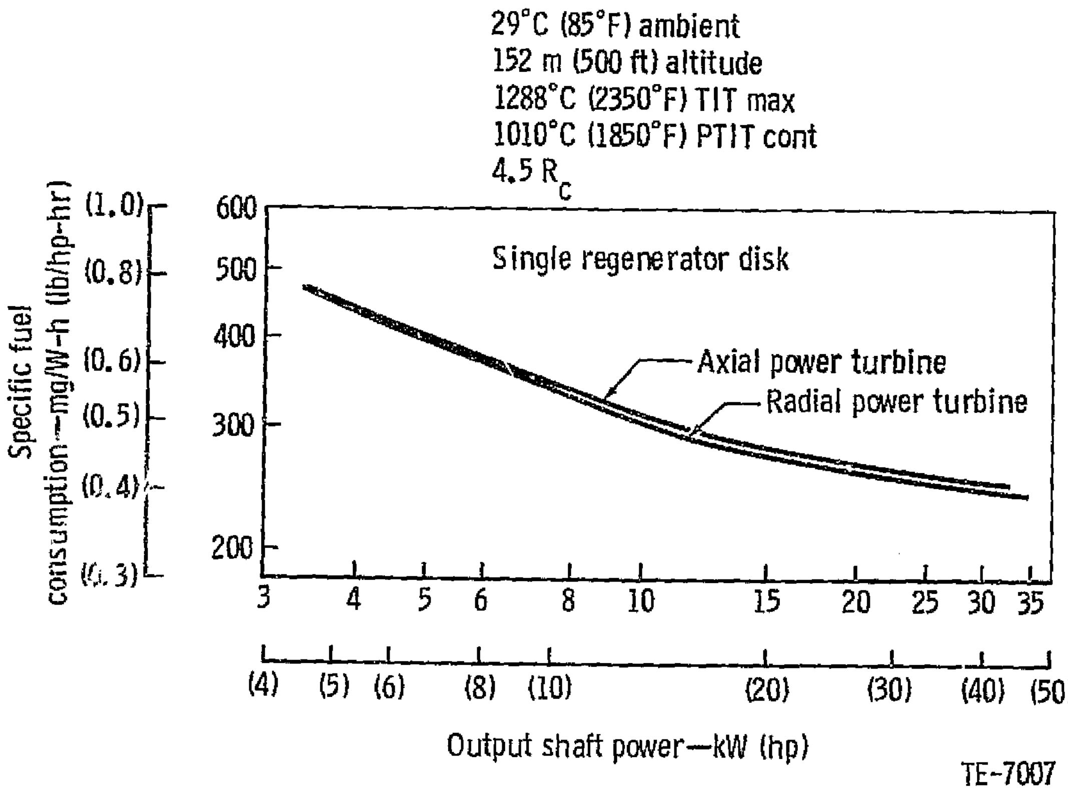


Figure 74. SFC versus power for radial and axial power turbines.

the compressor inlet pressure loss was scheduled as a function of inlet airflow. Prior to this time, the leakage and inlet pressure loss were assumed to be a constant percentage at all power settings. This change will not affect any comparisons made prior to the fixed- versus variable-geometry study.

One advantage of the variable-geometry configuration is the ability to idle at a higher gasifier speed without a significant loss in fuel economy. It is more efficient to reduce power at a given speed with variable geometry than to reduce temperature to achieve the same effect. Using the simplified method to calculate fuel economy on the combined driving cycle, the effects of idle speed for the fixed geometry versus variable geometry are shown in Figure 77. The higher idle speed improves vehicle response and minimizes the initial lag that is characteristic of gas turbine engines. Increasing the idle speed from 60% to 70% doubles the distance traveled in the first second of a wide-open throttle acceleration. Fuel economy with variable geometry is relatively insensitive to idle speed, however, the fixed-geometry engine is very idle sensitive. At 70% idle, variable geometry fuel economy is 20% superior to that of fixed geometry; at 60% it is 10% superior. A variable-geometry configuration is recommended for the engine design because of the increase in fuel economy and insensitivity to idle speed.

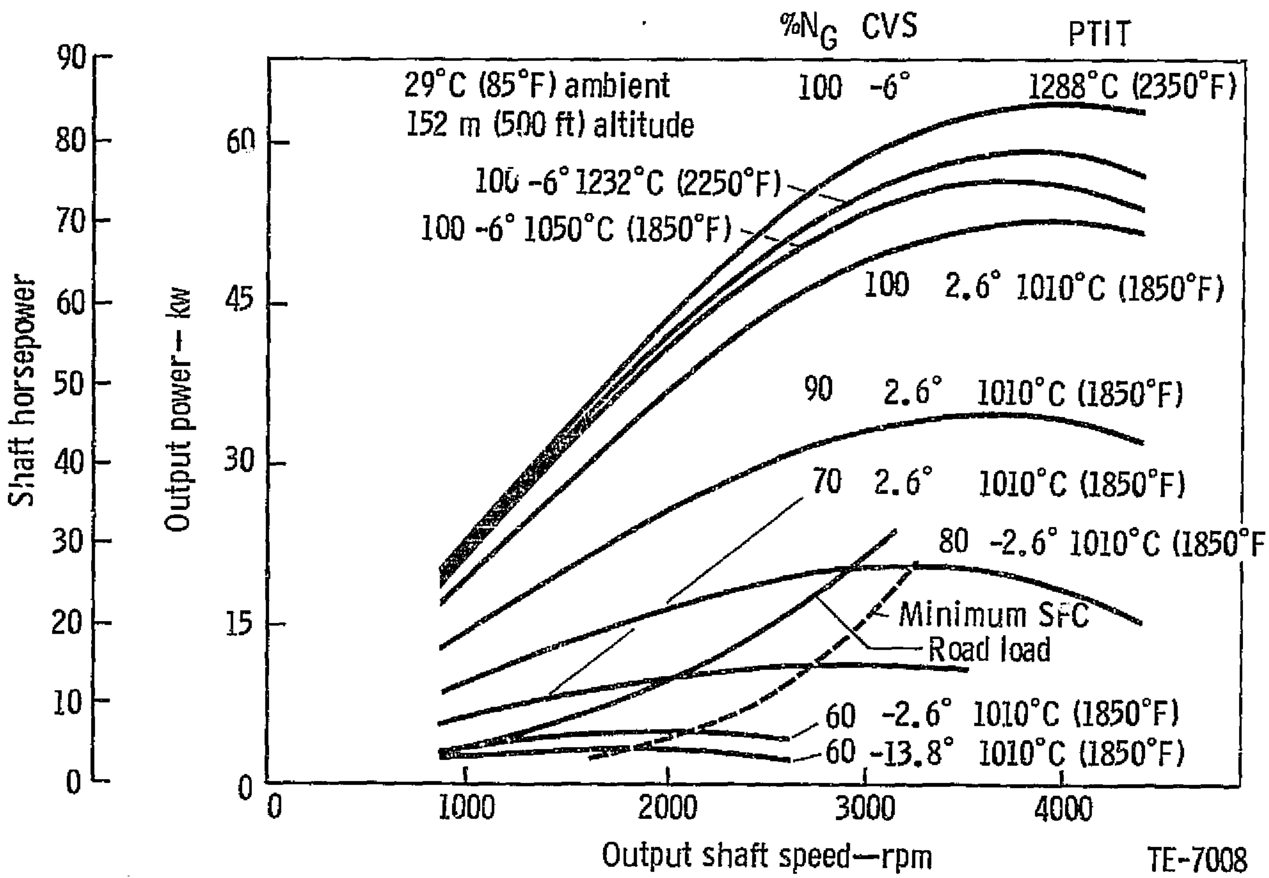


Figure 75. Power map for two-shaft engine with radial power turbine.

Sensitivity Study

This study involved varying each of the following engine cycle parameters individually and calculating the effect on fuel economy for the combined driving cycles:

- o Compressor efficiency
- o Gasifier turbine efficiency
- o Power turbine efficiency
- o Regenerator effectiveness
- o Cycle pressure loss
- o Mechanical loss
- o Regenerator leakage

A complete base-line set of off-design engine performance was first calculated. From this, a base-line fuel economy value was calculated, using the simplified method. A 1-point efficiency or loss change (1 hp in the case of the mechanical loss) was then assigned to each of the listed parameters on an individual basis. Engine performance and fuel economy were calculated in the same manner as for the base line for each parameter change. The results of the study are summarized in Figure 78.

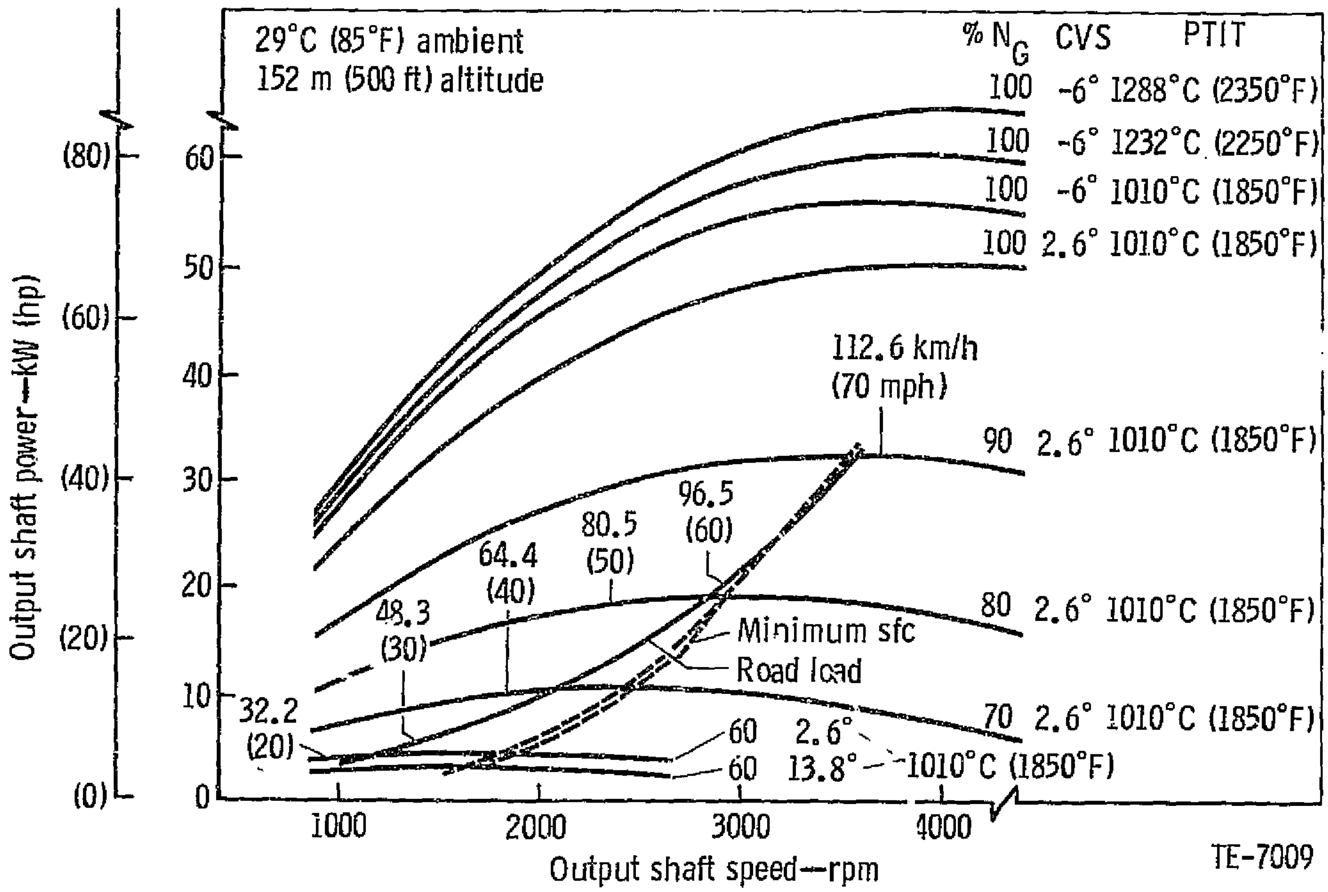


Figure 76. Power map for two-shaft engine with axial power turbine.

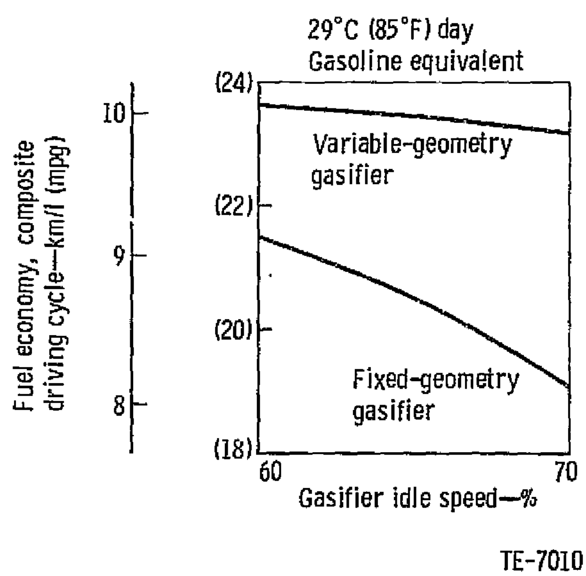


Figure 77. Effect of idle speed on fuel economy for fixed and variable-geometry gasifier two-shaft engines.

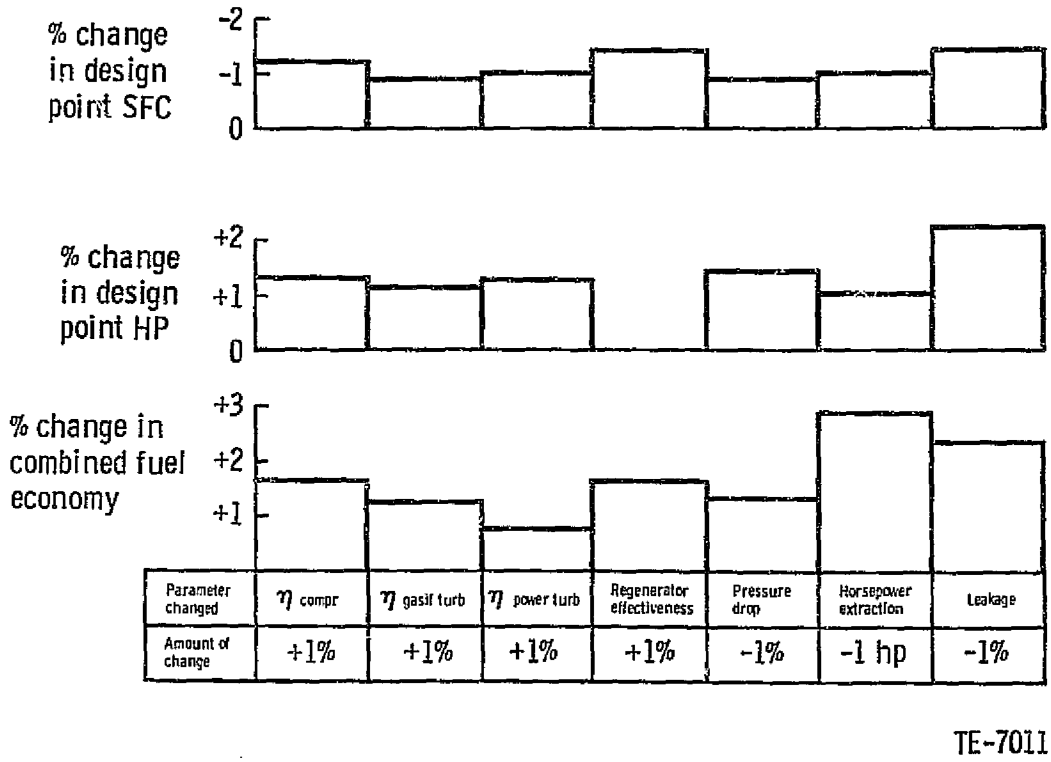


Figure 78. Sensitivity study.

Fuel economy is least sensitive to power turbine efficiency. Gasifier component efficiency, pressure drop, and regenerator effectiveness are of approximately equal significance. Flow leakage and mechanical loss have the strongest influence on fuel economy.

Final Selection

The final selection is a configuration of variable-geometry components, radial power turbines, a single regenerator disk, and a gasifier rotor inlet temperature of 1288°C (2350°F). Once the temperature level and single disk diameter were optimized, an engine with a 4.0 pressure ratio was compared with the initial selection of 4.5. Also, a disk diameter of 40.6 cm (16.0 in.) was calculated for comparison with the initial diameter of 46.4 cm (18.3 in.). The results, at 29°C (85°F), 152 m (500 ft) ambient conditions, are as follows:

TIT		Disk diameter			Fuel economy	
°C	(°F)	cm	(in.)	R_c	km/l	(mpg)
1288	(2350)	46.4	(18.3)	4.5	9.7	(22.8)
1288	(2350)	46.4	(18.3)	4.0	9.7	(22.8)
1288	(2350)	40.6	(16.0)	4.5	9.4	(22.1)

The 4.5 pressure ratio and 46.4-cm (18.3 in.) disk diameter yield the best fuel economy on the combined driving cycle.

There may be considerations during the final design phase that will require slight alteration of cycle parameters; however, the cycle defined during the concept study is shown in Table XXIX.

TABLE XXIX. IGT SELECTED TWO-SHAFT ENGINE MAXIMUM POWER CYCLE PARAMETERS

	CIT = 15°C (59°F)	CIT = 29°C (85°F)
Compressor inlet pressure, kPa (psi)	99.8 (14.5)	98.0 (14.2)
Gasifier rotor inlet temperature, °C (°F)	1288 (2350)	1288 (2350)
Compressor inlet airflow, kg/s (lb/sec)	0.379 (0.836)	0.352 (0.776)
Compressor pressure ratio	4.5	4.25
Compressor efficiency, T-S	76.3	76.4
Gasifier turbine efficiency, T-T	83.7	83.8
Power turbine efficiency, T-T	84.5	84.3
Regenerator effectiveness, %	93.6	94.1
Total cycle pressure loss, %	15.4	15.3
Regenerator leakage, %	4.34	4.34
Outboard leakage, %	1.2	1.2
Turbine cooling, %	0.7	0.7
Mechanical loss + accessories, kW (hp)	7.5 (10.1)	7.5 (10.1)
Heat rejection from flow path, %*	2.5	2.5
Power, kW (hp)	75 (100.6)	64.2 (86.1)
Specific fuel consumption, Mg/W h (lb/hp-hr)	225 (0.370)	235 (0.386)
Idle fuel flow, kg/h (lb/hr)	1.3 (2.9)	1.4 (3.0)

*2.5 of Btu input is lost to heat rejection

In compliance with the work statement of Task III, the engine performance is completely described including component maps. This section includes the complete performance of the recommended configuration:

- o Compressor maps (Figures 79 through 87)
- o Gasifier turbine maps (Figures 88 and 89)
- o Power turbine maps (Figures 71 and 72). Figures 69 through 72 represent a 100% flow capacity setting for the power turbine. Figure 73 shows how the turbine efficiency changes as the flow capacity of the power turbine is varied to hold TIT. Figure 73 is used to adjust the 100% setting to other geometry settings for off-design matching.
- o Power versus output speed (Figure 90)
- o Fuel flow versus output speed (Figure 91)

Operating limits and other features are described in Table XXX.

Vehicle Driveline Studies for Fuel Economy Improvement

With the engine already maximized in performance, the feasibility of improving fuel economy by driveline optimization was evaluated.

Compressor map scale factors:

Engine configuration	Axial power turbine	Radial power turbine
Compressor corrected flow	0.2905	0.2854
Compressor pressure ratio -1.0	0.895° ($R_{c_{map}}$ -1.0)	0.8951 ($R_{c_{map}}$ -1.0)
Compressor efficiency	1.0160	1.0167

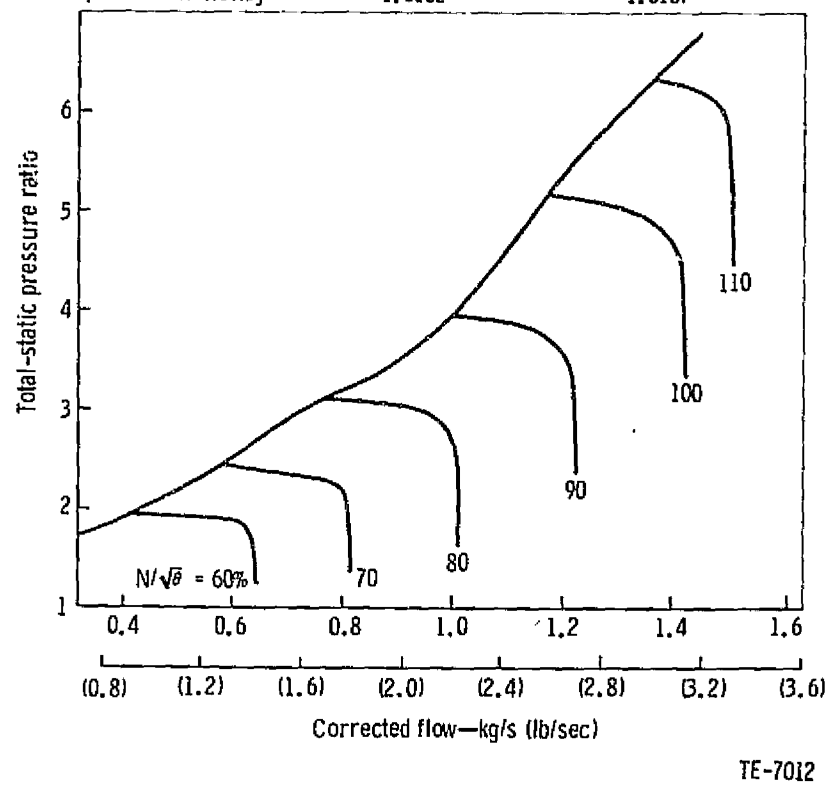


Figure 79. IGT compressor map (IGV setting = -6°).

TABLE XXX. IGT ENGINE OPERATING LIMITS AND FEATURES	
Maximum gasifier turbine inlet temperature	1288°C (2350°F)
Maximum continuous gasifier turbine inlet temperature	1232°C (2250°F)
Gasifier rotor acceleration temperature	1398°C (2550°F)
Maximum continuous power turbine inlet temperature	1010°C (1850°F)
Single regenerator disk	46.4 cm (18.3 in.)
Idle rpm	70%
Variable vanes at compressor inlet and exit	
Variable gasifier turbine inlet vanes	
Variable power turbine inlet vanes	

The GPSIM vehicle simulation computer model was used as an analytical means of studying the turbine engine in the vehicle for fuel economy evaluation. The selected concept engine was modeled by describing its output shaft torque and fuel requirement characteristics, over the full operating spectrum of the engine, as a function of the output shaft speed and a throttle function. The engine data were generated from the general off-design performance model described previously. Data were calculated for 29°C (85°F),

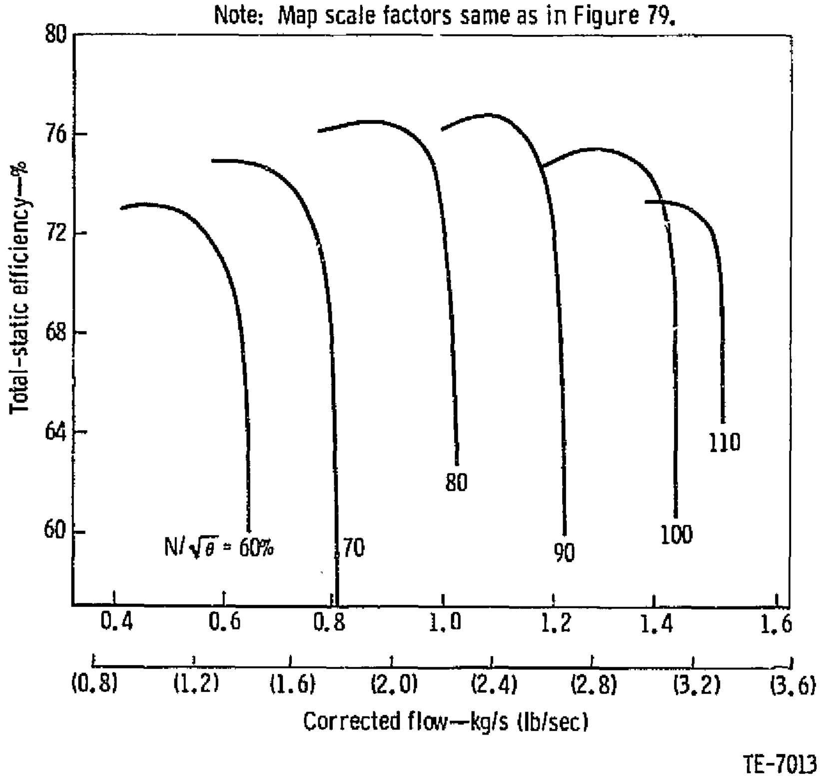


Figure 80. IGT compressor map (IGV setting = -6°).

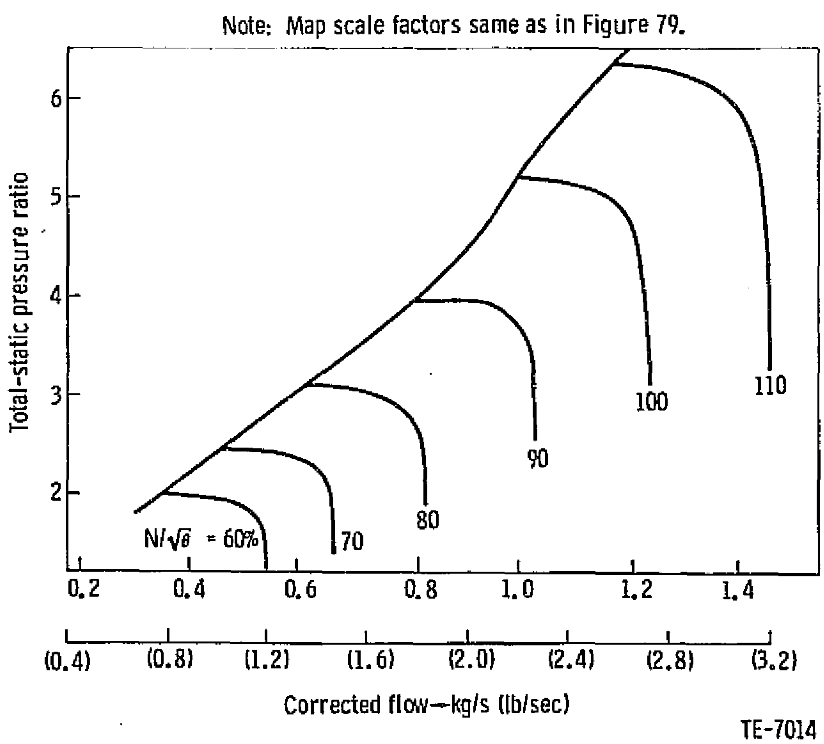


Figure 81. IGT compressor map (IGV setting = 2.6°).

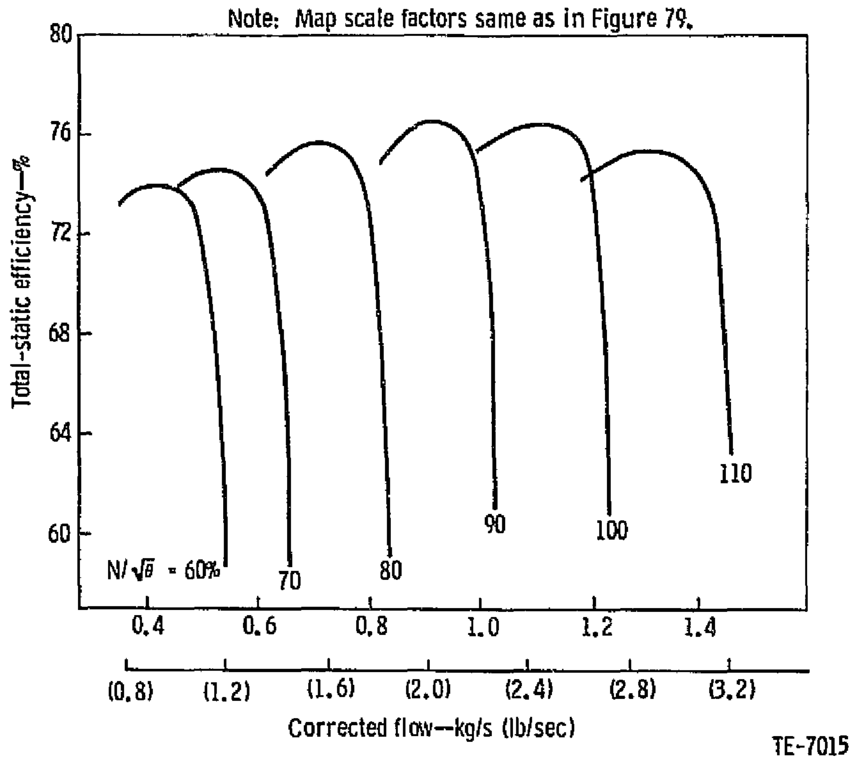


Figure 82. IGT compressor map (IGV setting = 2.6°).

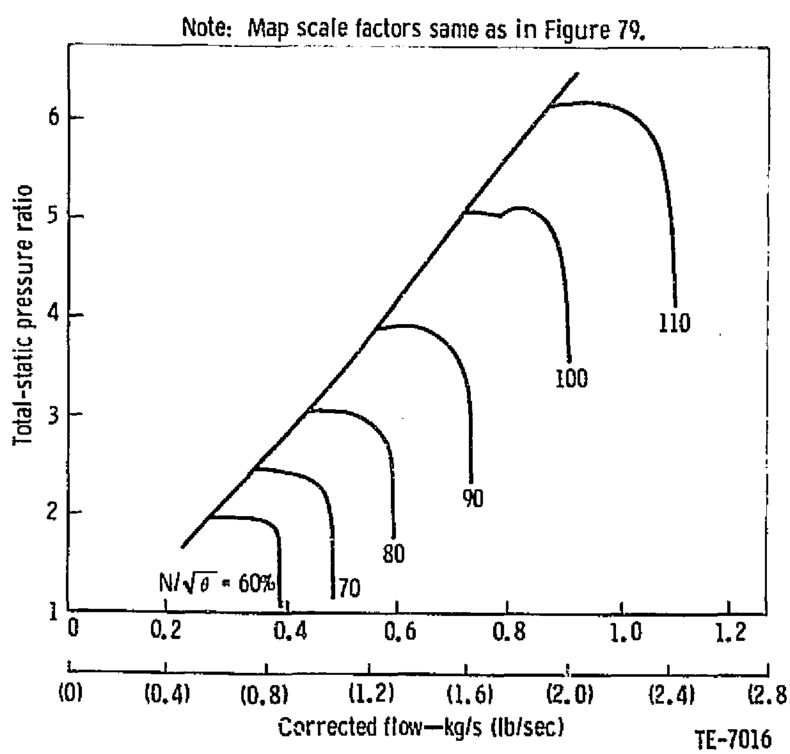


Figure 83. IGT compressor map (IGV setting = 13.8°).

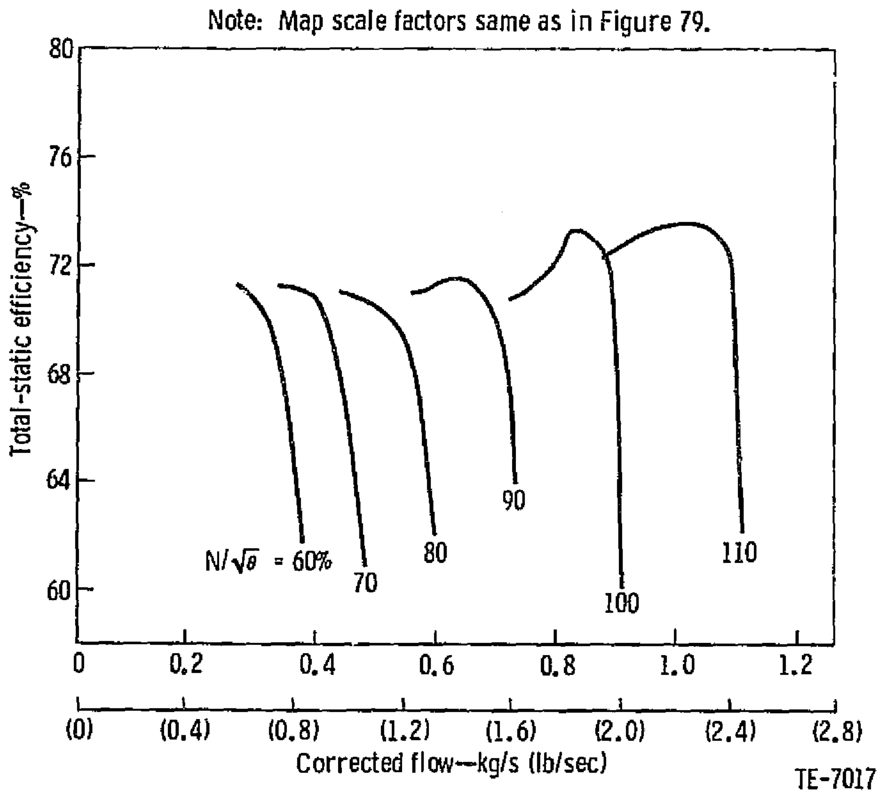


Figure 84. IGT compressor map (IGV setting = 13.8°).

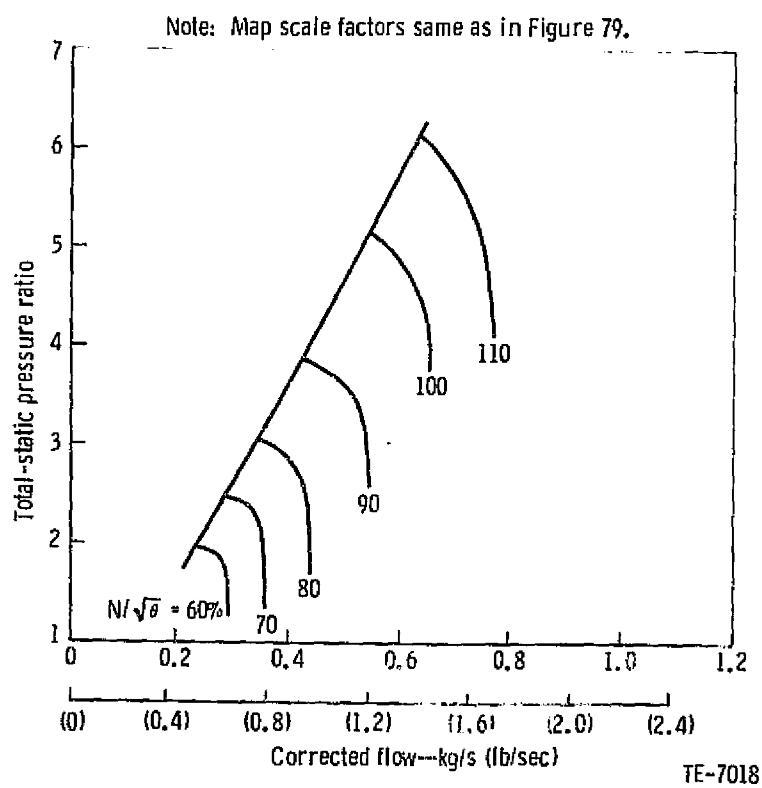


Figure 85. IGT compressor map (IGV setting = 22°).

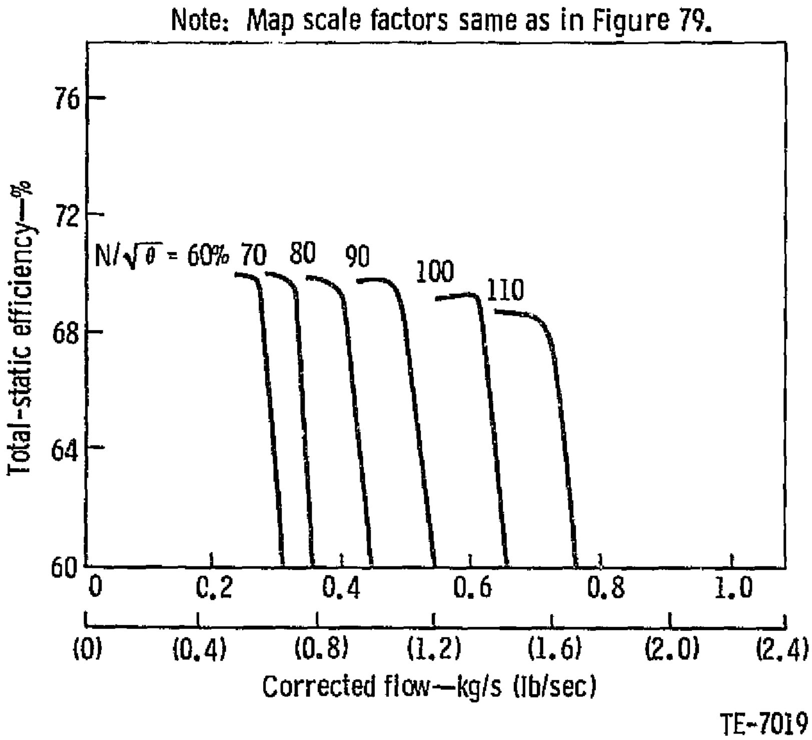


Figure 86. IGT compressor map (IGV setting = 22°).

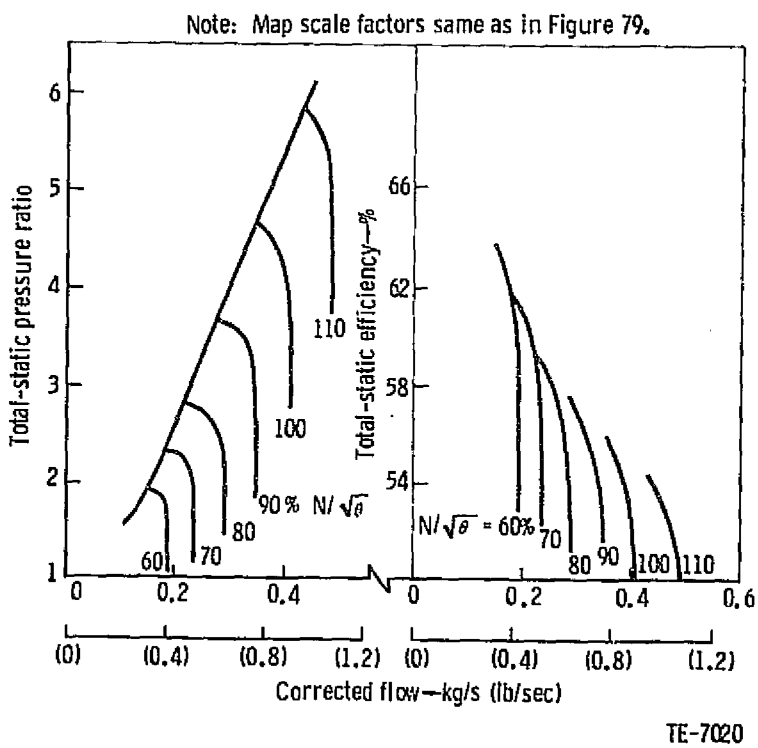


Figure 87. IGT compressor map (IGV setting = 31°).

Gasifier turbine map design point scale factors:

Engine configuration	Axial power turbine	Radial power turbine
Turbine equivalent flow	0.3355	0.3300
Turbine expansion ratio -1.0	0.9309 ($R_{e_{map}}$ -1.0)	0.9228 ($R_{e_{map}}$ -1.0)
Turbine efficiency	1.0470	1.0461

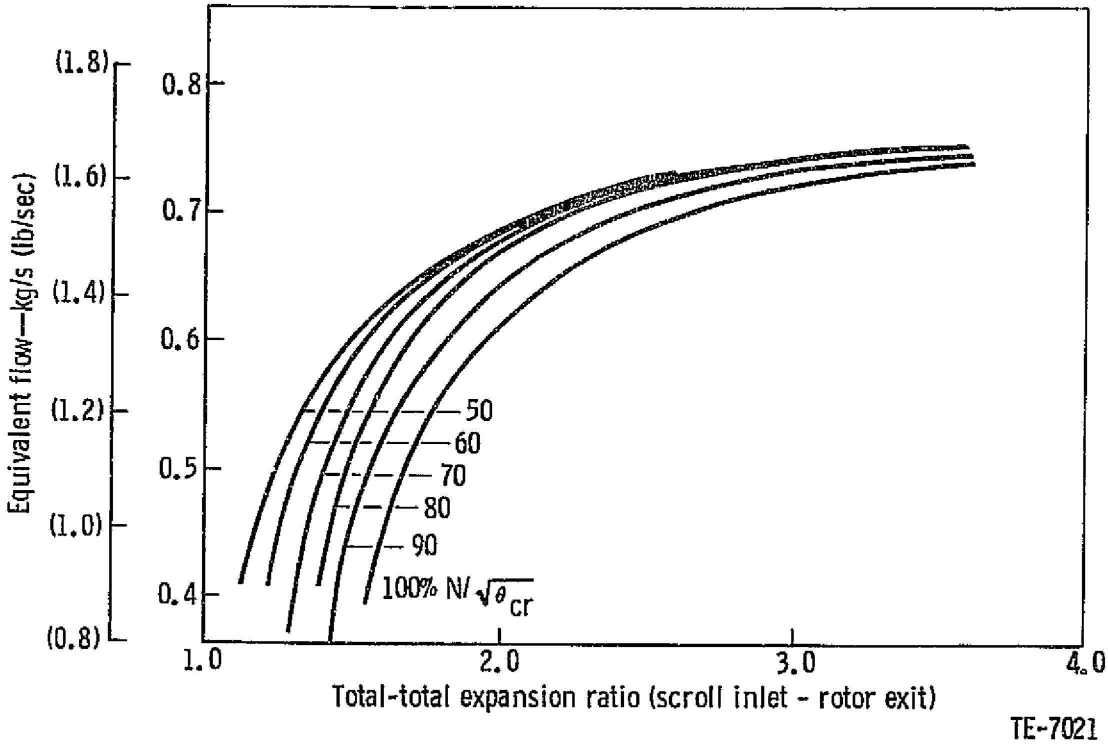


Figure 88. IGT gasifier turbine map (100% flow setting).

152 m (500 ft) ambient conditions. Separate tabluations were used to describe steady-state engine operation and transients (gasifier rotor accelerations). During steady state, the power turbine inlet temperature is held to 1010°C (1850°F) for most engine operation except at full throttle, where the engine is controlled to 1288°C (2350°F) gasifier turbine inlet temperature. During short periods of high power demand where the gasifier rotor is required to accelerate as quickly as possible, the gasifier turbine inlet temperature is controlled to a limit of 1400°C (2550°F) until 100% gasifier speed is reached. The program also considers the rotor inertia of the gasifier system and power turbine during transient maneuvers.

Analysis of the turbine engine was begun by replacing the piston engine model in the base-line vehicle simulation described previously. This provided a simulation with a conventional 1976 compact class vehicle. The transmission simulated is the General Motors Hydra-matic THM 350. The transmission model is quite detailed and includes torque converter losses on the pump and turbine sides, gear mesh efficiencies and spin losses, and gear shift torque reactions. The axle ratio was changed from 2.70:1 in the baseline vehicle to 3.70:1 to provide a vehicle capable of attaining 145 km/h

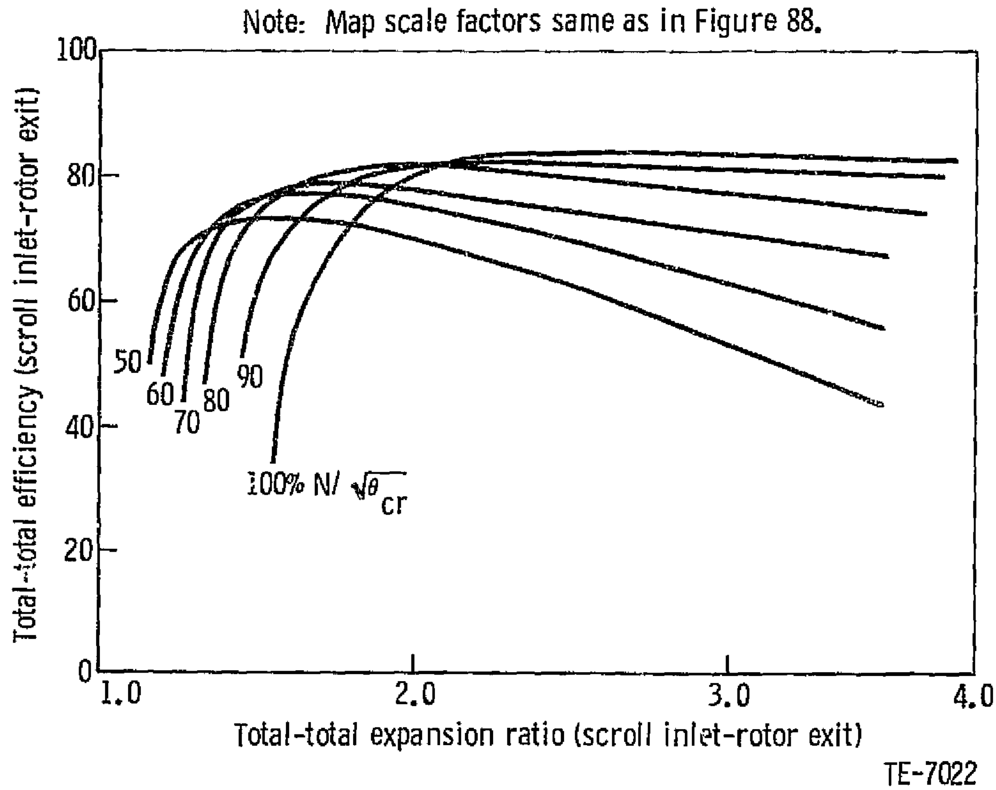


Figure 89. IGT gasifier turbine map (100% flow setting).

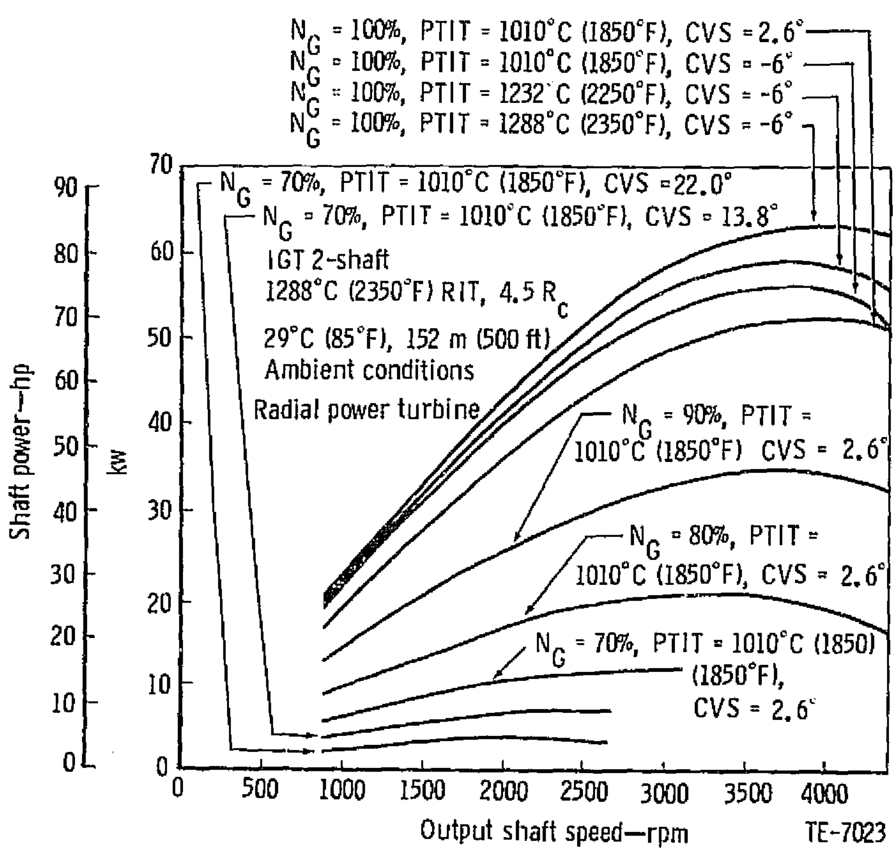


Figure 90. Selected IGT engine power map.

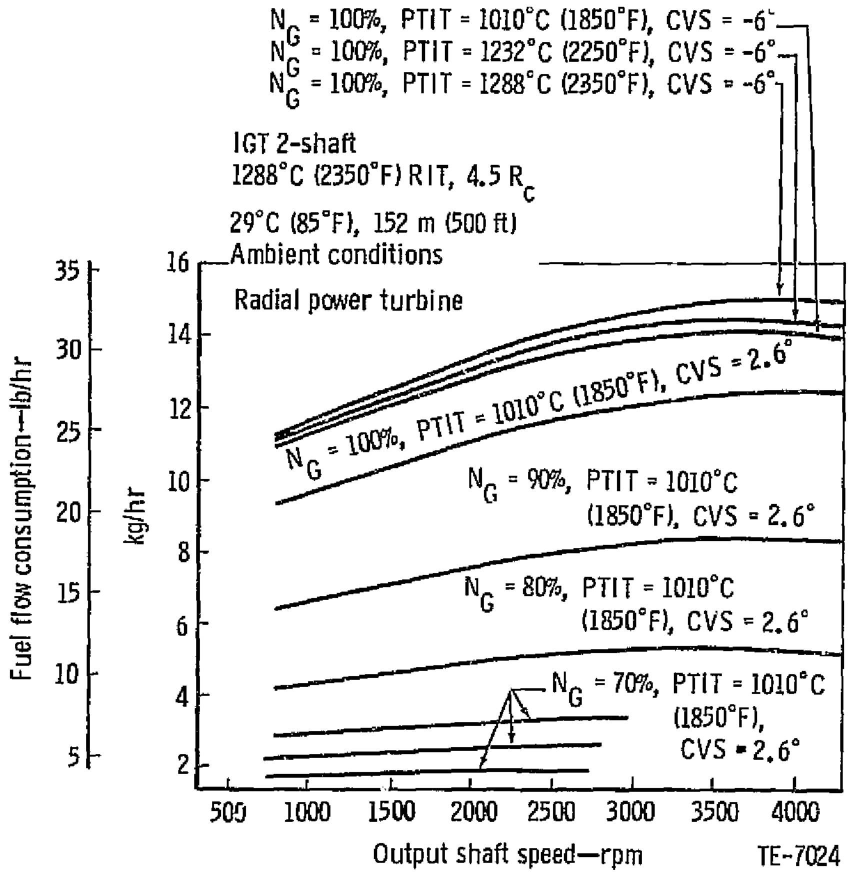


Figure 91. Selected IGT engine fuel map.

(90 mph) at 100% engine output shaft speed. The axle ratio was chosen to provide the desired maximum vehicle speed with the realization that the turbine engine exhibits the best specific fuel consumption at higher power turbine speeds.

Fuel economy is calculated in GPSIM by callout of a subroutine which calculates fuel usage over the Urban and Highway portions of the EPA Composite Driving Cycle, then combines the two by established EPA procedure to yield the combined fuel economy. The driving cycle is simulated by breakdown of the overall EPA Driving Cycle into specific components corresponding to the specific relationships between acceleration and deceleration rates, constant speed operation, and distance traveled. The fuel economy derived from simulation is then expressed in terms of a percentage fuel economy improvement over the base-line 8.3 km/l (19.6 mpg) achieved by the piston engine-powered vehicle. All fuel economy relations are expressed for the specified gasoline fuel having a heating value of 42.565 MJ/kg (18,300 Btu/lb) and a heat content of 31.623 MJ/l (113,460 Btu/gal).

Gasifier idle speed was studied to determine the trade-off between performance and economy by generating data at 60% and 70% idle speed. Figure 92 summarizes the results obtained and indicates that the turbine engine alone can achieve fuel economy improvements in excess of 20% over the base-line piston engine. The data indicate that the fuel economy gain is moderate when gasifier idle speed is decreased.

The turbine engine is operating at an inherent disadvantage when it is matched to an existing transmission. This is clearly shown in Figure 93, which compares the sfc characteristics of the two engine types. Existing drivelines have been optimized for the piston engine in the past few years to cause its operation to occur in the best sfc regions through transmission gear selection, low numerical axle ratios, and transmission part-throttle shift point scheduling. All of these bias the piston engine toward operating at the lowest engine speed consistent with good drivability of the vehicle.

The two-shaft turbine engine has an inherently lower sfc, but its fuel consumption characteristic suggests that it needs to operate at a higher output speed for a given engine power demand. This can be accomplished through gearing of the engine output shaft, transmission gearing, axle ratio selection, or a combination of the three. This trend could also be expected to enhance vehicle performance by allowing the turbine to operate in regions where more instantaneous torque is available.

The effect was studied by choosing a different engine output shaft reduction gear ratio, then coupling it with the standard THM 350 transmission torque converter and gear set but with the addition of a fourth overdrive gear range set to give the vehicle the desired top speed. The axle ratio was left

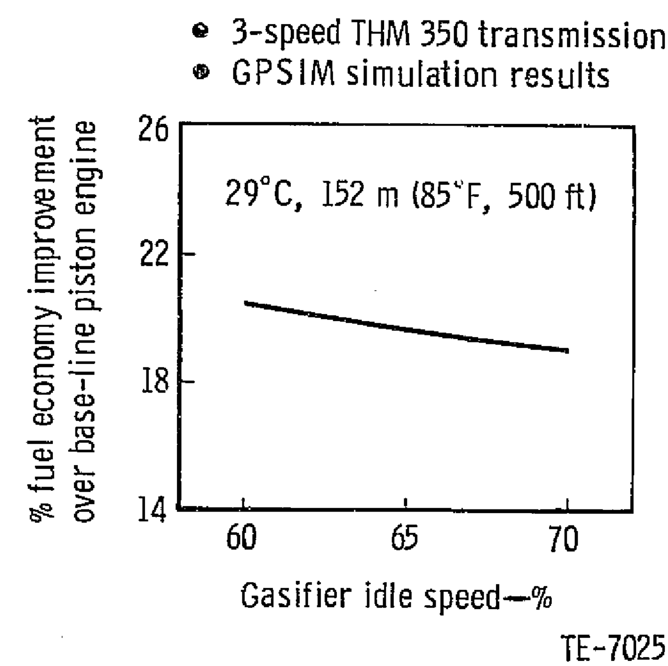


Figure 92. Effect of idle speed and ambient conditions on fuel economy.

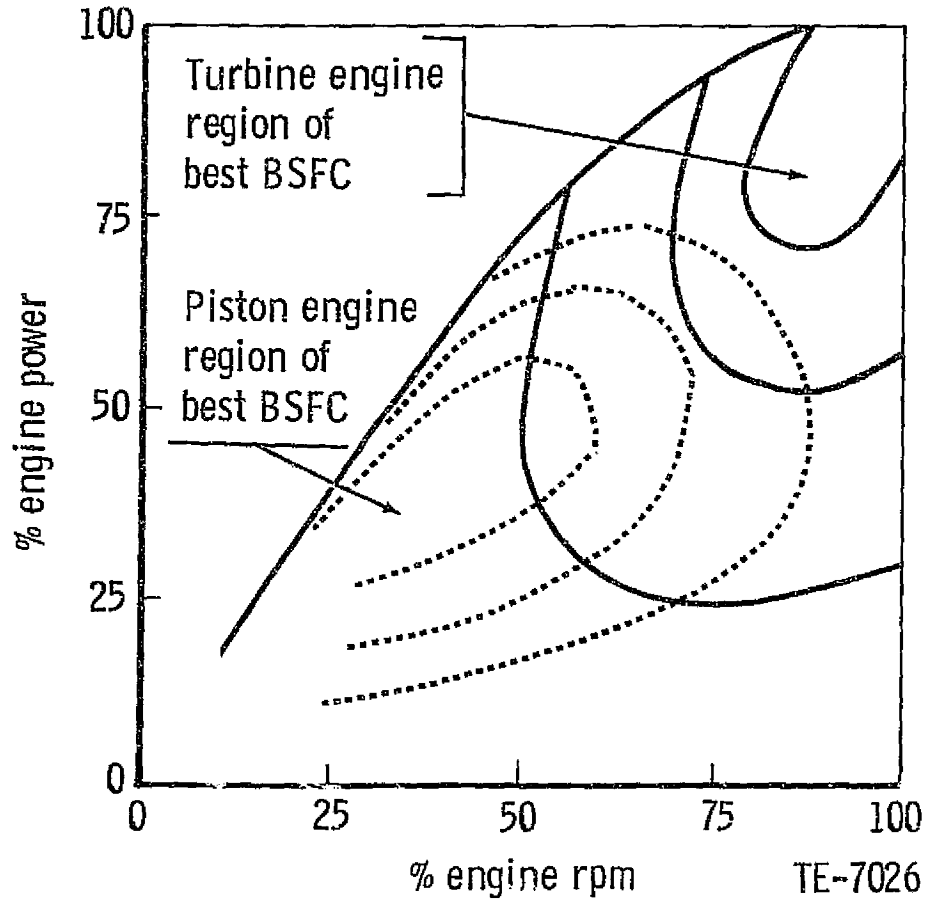


Figure 93. Comparison of sfc characteristics—piston engine versus two-shaft turbine engine.

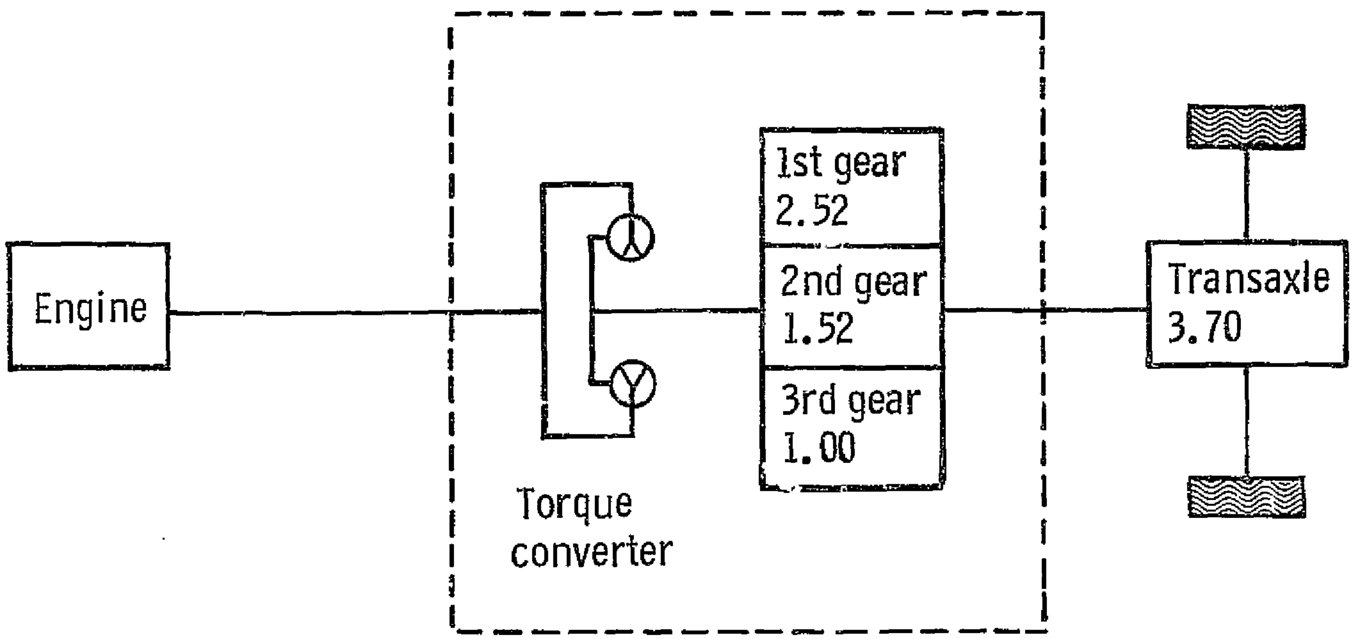
at its original 3.70:1 because this is near the maximum ratio which would fit the proposed vehicle installation. The engine reduction gear and the overdrive gear were chosen to keep the engine out of fourth gear during most normal vehicle operation since it appeared to penalize fuel economy.

Figure 94 describes the drivelines for the three-speed and four-speed automatic transmission configurations.

The results of the driveline alteration are summarized in Figure 95, which indicates that fuel economy improvements in the order of 1% to 1.5% can be seen for a four-speed gearbox. The optimization of gearing could provide some further improvements.

An important parameter studied is the effect of transmission shift schedule on vehicle fuel economy. During part-throttle operation the automatic transmission for a conventional piston engine is controlled by transmission output speed and engine manifold vacuum. For the turbine engine, a throttle function would be substituted for engine vacuum because it would perform the same control function--namely, providing the transmission with an indication of engine load demand. The shift schedule was set up to give approximately the same shift points as the base-line piston engine vehicle. The importance of shift schedule selection was then realized by utilizing an

1. Conventional three-speed THM 350 transmission



2. Four-speed automatic transmission with engine output reduction gear

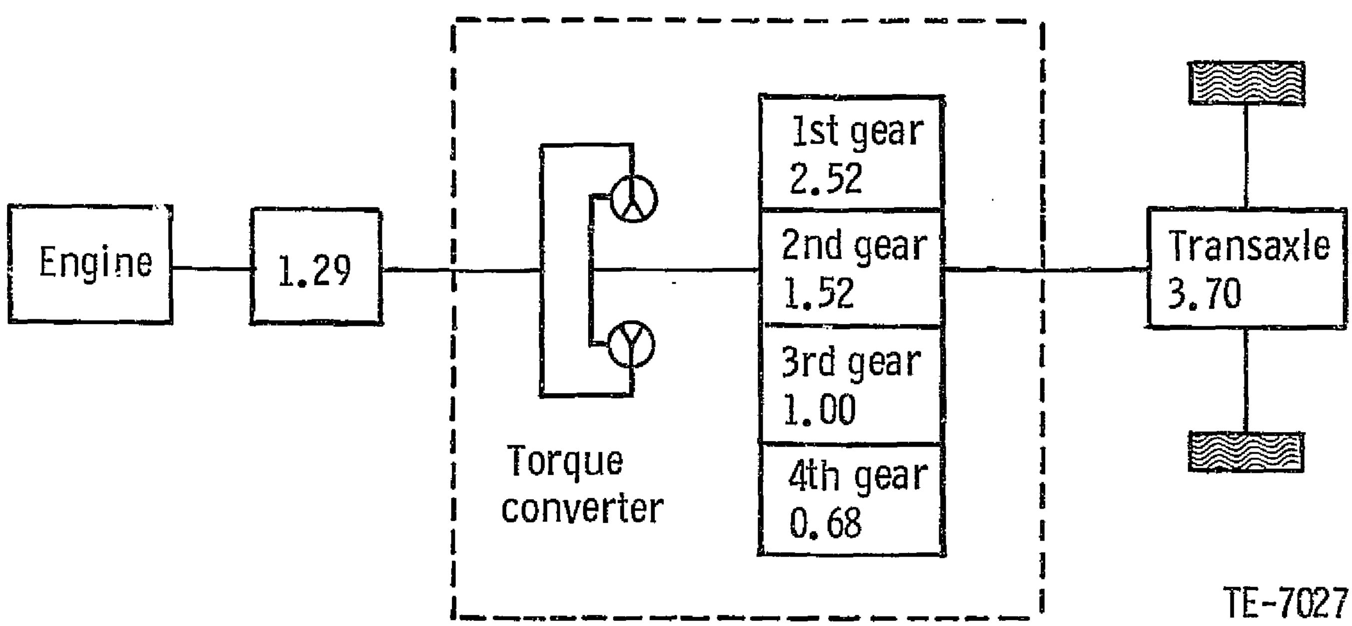


Figure 94. Comparison of driveline configurations studied.

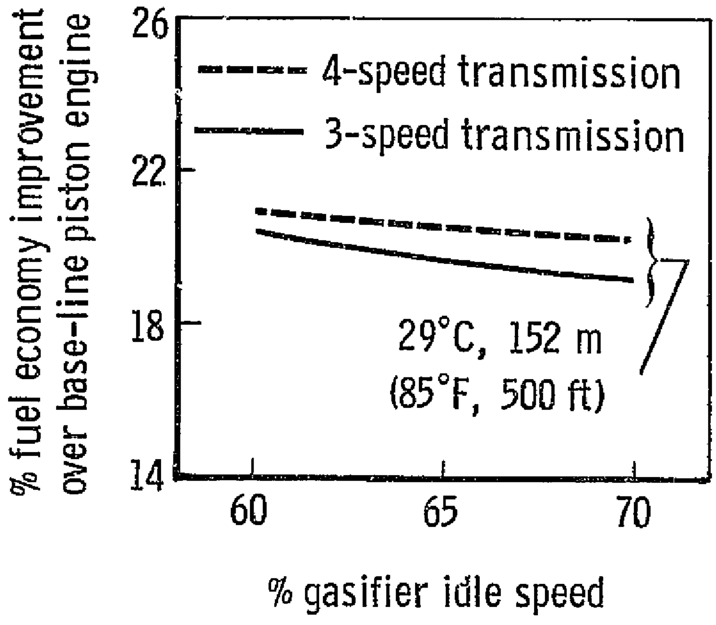


Figure 95. Effect of four- and three-speed transmissions on fuel economy.

option in GPSIM called "Best Economy Shifting," which automatically selects the transmission gear yielding the best fuel economy at each point in the driving cycle.

Figure 96 shows the effect of optimum shift point selection to be worth a 1.6% to 2.2% fuel economy improvement. The initial analysis of the shift pattern generated by "Best Economy Shifting" shows it to be reasonable. However, whether the full benefit shown can be gained will depend on further study of turbine engine transmission control systems.

The final parameter investigated in the driveline is the inclusion of a lockup-type torque converter and a reduced-loss state-of-the-art transmission. For this analysis the THM 350 transmission was replaced by the THM 125, which is a new transmission that has lower mechanical and hydraulic losses. In addition, transmission efficiency improvements for a lockup torque converter and optimized charging pump were included; both of these are being currently developed and evaluated within General Motors. Analysis was necessarily accomplished by hand calculation rather than the GPSIM program used for the other studies reported herein. Transmission power loss data were obtained for the THM 125 with lockup converter and low-loss charging pump. These data are shown in Figure 97, where they are compared with base-line THM 350 loss characteristics as applied to road-load conditions. The data yield an average fuel saving of 6% when applied along the road load from 32 km/h (20 mph) to 96 km/h (60 mph) vehicle speed.

It was then assumed that the 6% savings would apply uniformly over the EPA Composite Driving Cycle. These results indicate that significant improvements are available through optimization of existing concept transmissions. Figure 98 shows that the fuel economy improvement over the base-line piston

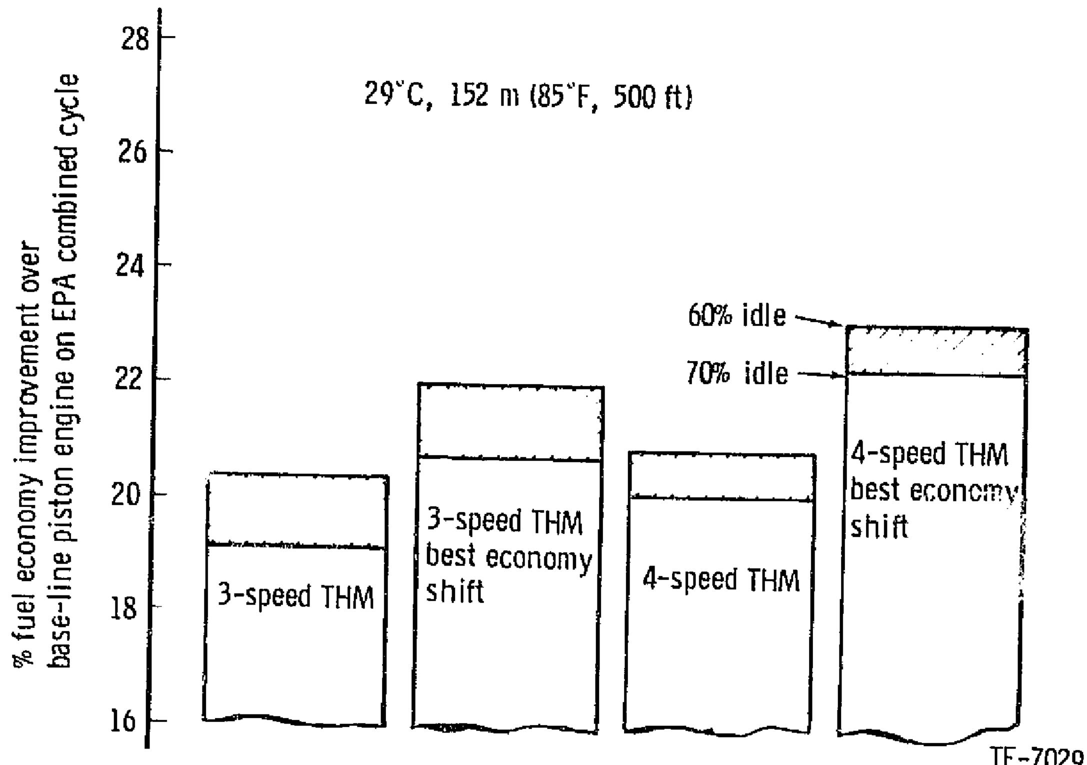


Figure 96. Effect of optimum shifting for best part-throttle economy.

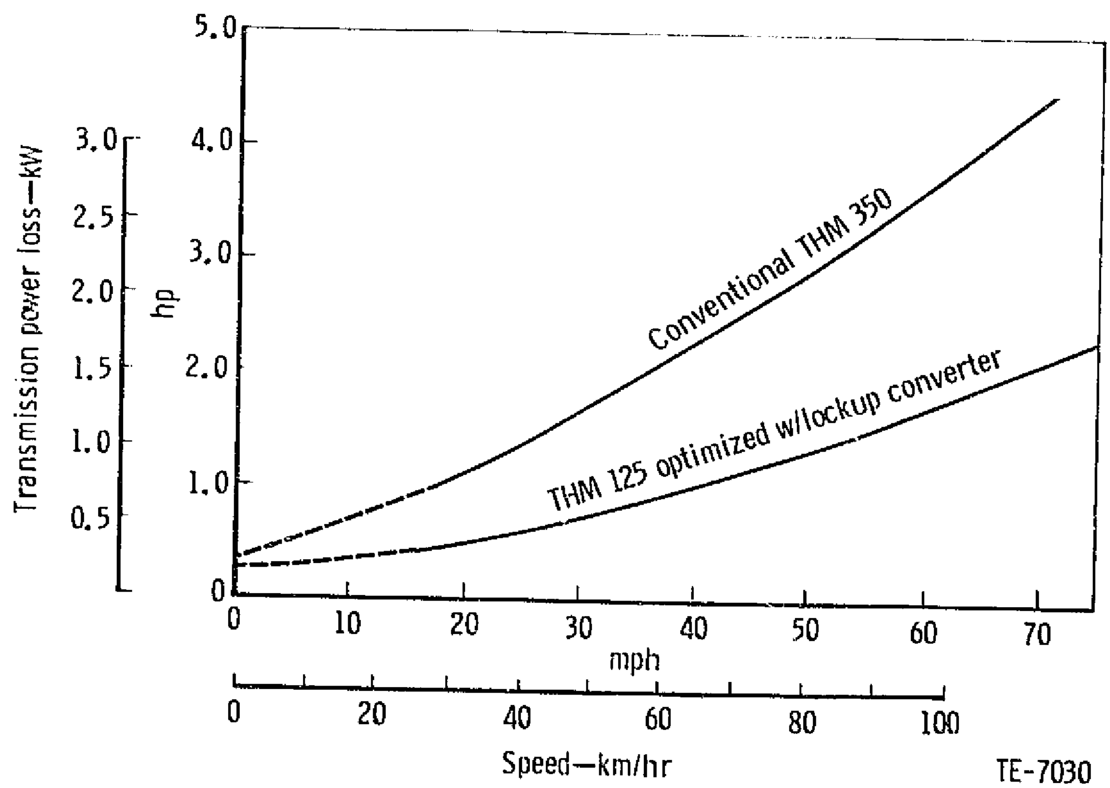


Figure 97. Transmission power loss comparison--THM 350 versus optimized THM 125.

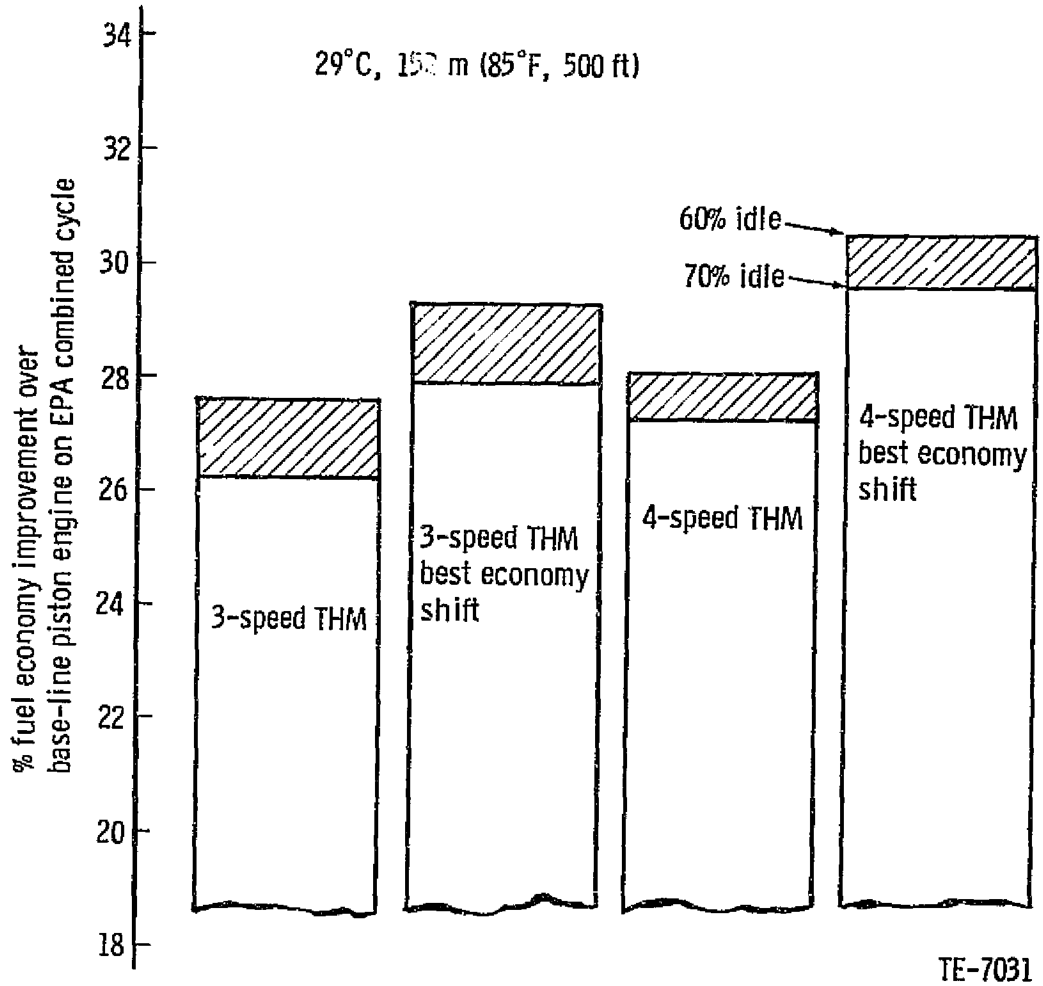


Figure 98. Effect of optimized transmission on fuel economy.

engine will be in the 30% range including transmission improvements. Future work would include developing the optimized transmission loss data in the format required to program GPSIM so that all comparisons would be on a consistent analytical basis.

VEHICLE PERFORMANCE

Vehicle performance characteristics were studied primarily by use of the GPSIM program in conjunction with the fuel economy studies reported previously. The effects of a three-speed versus four-speed transmission and of gasifier idle speed were found to be the primary factors affecting vehicle performance.

Vehicle driving characteristics were evaluated by using criteria applied within General Motors as indicative of relative acceptability between vehicles. The following performance calculations were made:

1. 0-97 km/h (0-60 mph) WOT acceleration--calculation of time and distance traveled in a wide-open throttle vehicle acceleration.
2. Distance traveled in first second at WOT—an indication of throttle response and driver feel to vehicle start-up from rest.
3. High-speed pass—calculation of the 48-113 km/h (30-70 mph) WOT vehicle acceleration in terms of time and distance required, which is indicative of the power capacity of the engine in a real-world highway maneuver.
4. Vehicle grade capability--calculation of the vehicle capability at steady-state road load conditions on various grades. The maximum gradeability criterion is that the vehicle must be able to start from rest on a 34% grade.

The results of GPSIM simulation of the 0-97 km/h (0-60 mph) WOT acceleration are shown in Figure 99. The turbine engine was sized to yield equivalent performance to the base-line piston engine with 70% gasifier idle and four-speed transmission on a 29°C (85°F) day. At this condition both vehicles would accelerate from 0 to 97 km/h (0-60 mph) in about 15.5 seconds. Figure 99 indicates several significant trends:

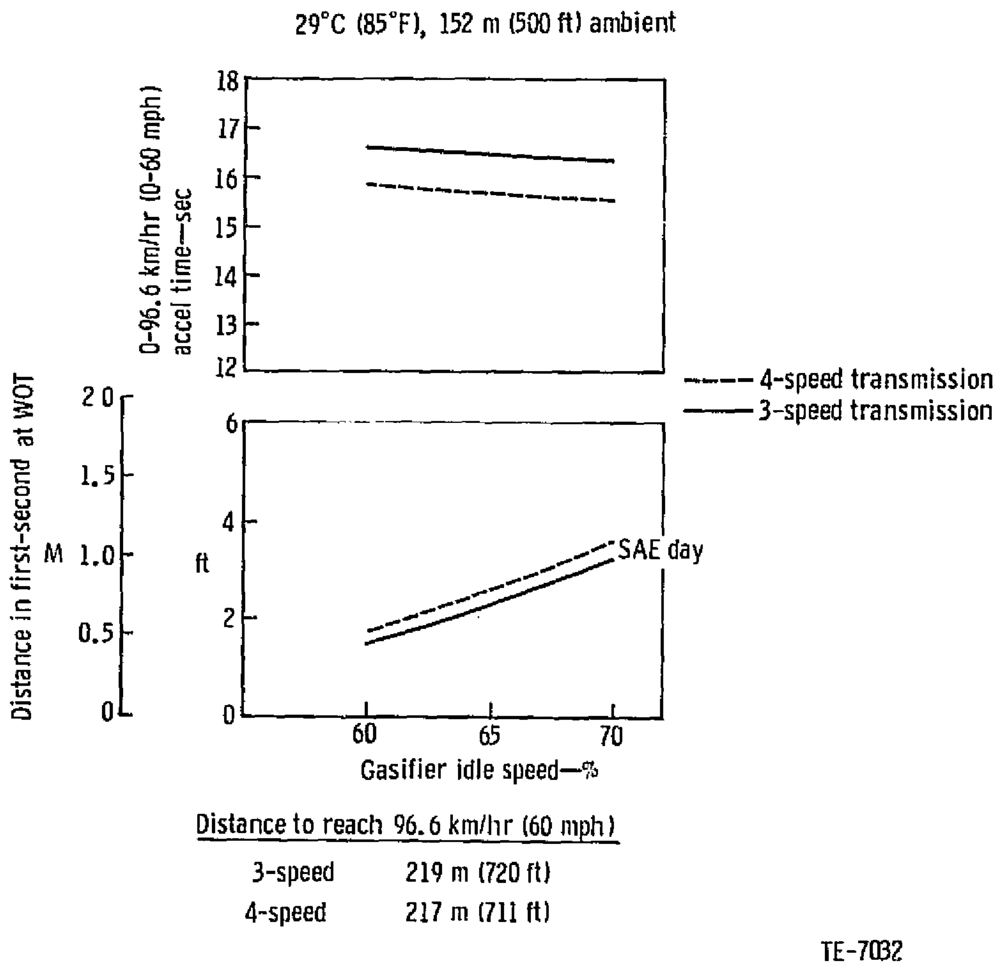


Figure 99. Turbine engine wide-open-throttle acceleration characteristics.

1. The effect of gasifier idle speed on overall vehicle acceleration performance is negligible for a given driveline configuration in the anticipated idle range.
2. The four-speed transmission enhances performance by providing additional torque multiplication to the drivewheels. Performance is improved approximately 5%, an 0.8-sec improvement in acceleration time, over the three-speed transmission.
3. The distance traveled in the first second of a wide-open throttle acceleration has a direct bearing on the selection of idle speed. It is strongly influenced by gasifier idle speed and is relatively insensitive to the difference in three-speed versus four-speed first gear ratio. This is not surprising, because the first second performance is primarily a function of instantaneous acceleration torque available and approximately twice as much power is available at 70% gasifier as at 60% idle at low output shaft speeds. The piston engine outperforms the turbine in this area, but distances of the 0.9-1.2 m (3-4 ft) magnitude are judged acceptable. A more detailed analysis of the engine-torque converter match at vehicle rest is expected to show improvement.

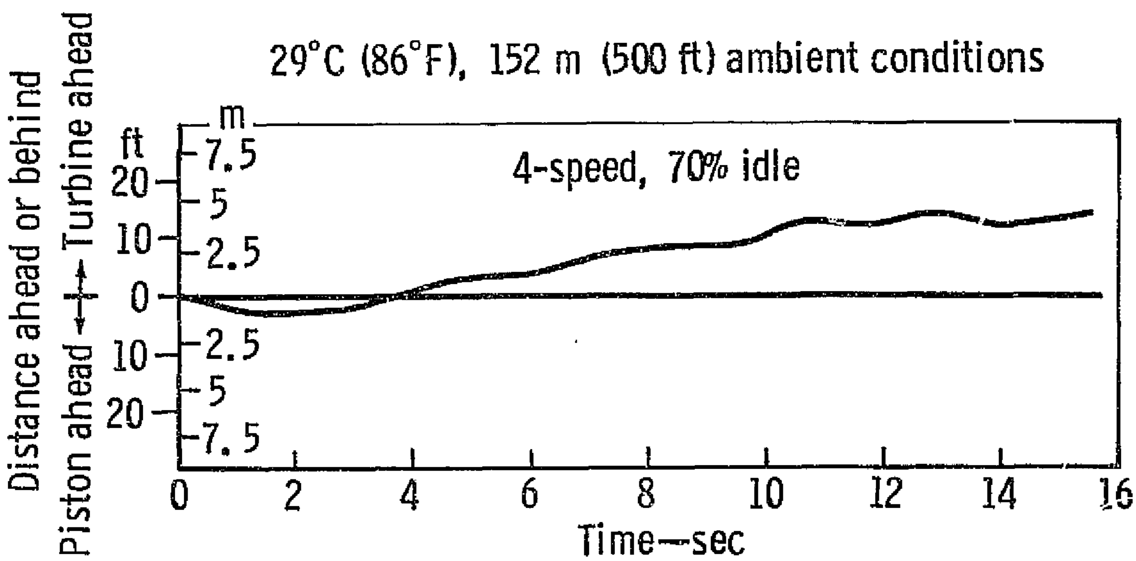
An alternative method of expressing relative performance between the piston engine vehicle and the turbine engine vehicle is to consider the distance ahead or behind during and at the end of an acceleration. Figure 100 shows by example the relative distance between vehicles throughout an acceleration. For all cases, the piston car maintains a small lead for the first several seconds but the final outcome is dependent on relative power available. The conclusion reached is that the driver would experience little difference in feel between the two types of powerplants.

The 48.3-112.6 km/h (30-70 mph) WOT high-speed pass is compared in Table XXXI. The times shown are acceptable and comparable with test results for piston engine vehicles currently marketed.

TABLE XXXI. TURBINE VEHICLE PERFORMANCE COMPARISON FOR 48-113 KM/H (30-70 MPH) PASSING MANEUVER

(29°C (85°F), 152 m (500 ft) ambient conditions)

3-speed THM 350	
Time to reach 113 km/h (70 mph)	18.9 sec
Distance	448 m (1470 ft)
4-speed automatic	
Time to reach 113 km/h (70 mph)	18.2 sec
Distance	435 m (1428 ft)
Base-line piston engine	
Time to reach 113 km/h (70 mph)	16.9 sec
Distance	402 m (1320 ft)



At end of 0-96.6 km/hr (0-60 mph)

70% gasifier idle	Which car ahead	How much
3-speed	Piston	10.7 m (35 ft)
4-speed	Turbine	4.3 m (14 ft)
60% gasifier idle		
3-speed	Piston	18.0 m (59 ft)
4-speed	Piston	2.7 m (9 ft)

Note: All calculations assume car that reaches 96.6 km/hr (60 mph) first maintains constant velocity until other car reaches 96.6 km/hr (60 mph).

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Figure 100. Performance comparison of base-line piston vehicle and turbine vehicle (showing distance ahead or behind during 0-96.6 km/h (0-60 mph) WOT acceleration).

Gradeability results are summarized in Figure 101. The data shown are for simulation of a three-speed THM 350 at the 29°C, 152 m (85°F, 500 ft) ambient condition. The choices of transmission and ambient condition represent the worst case with respect to the parameters studied. The data show the limiting power condition for 0%, 9%, and 34% grades with the transmission in the gear deemed appropriate for the condition. The data indicate that the engine is sized to provide adequate power for any grade encountered in real-world conditions. On a level road the vehicle is limited by maximum allowable engine rpm rather than by power available. On a 9% grade the engine rpm limit and power limit coincide at 91.7 km/h (57 mph). The vehicle is sufficiently powered to start from rest on a 34% grade and accelerate to 25 km/h (15.5 mph).

The studies and simulation data indicate that the selected turbine engine will provide adequate performance, comparable in most areas to that of a current piston engine vehicle, without sacrificing the established fuel economy goals. The greatest trade-off between fuel economy and performance appears to be in the area of initial driver feel of the vehicle starting from r... Acceptable performance tends to require a higher gasifier idle speed which adversely affects economy. Further simulation study is needed to better define the trade-offs in terms of matching torque converter stall characteristics, transmission gear ratio selection, and other associated parameters.

COMPONENTS

Compressor

The preliminary IGT compressor design addresses the broad range of surge-free operation required for acceptable engine performance and attempts to maximize operating efficiency levels. The range requirements will be met by the use of variable inlet guide vanes and variable setting angle diffuser vanes. This technique has been very effective in extending range but does result in an efficiency degradation. The variable geometry efficiency penalty coupled with the small flow size of 0.36 kg/s (0.8 lb/sec) of the compressor makes the achievement of high levels of efficiency a challenge. To achieve the full performance potential, the IGT design utilizes refinements to the technology from the DDA 505-IIIA. The 505-IIIA has demonstrated state-of-the-art performance and was used in the prediction of achievable IGT efficiencies. The IGT design parameters and requirements are listed in Table XXXII.

The preliminary compressor design has the variable guide vanes located in the radial inflow section of the axisymmetrical duct which turns the flow to axial at the inlet of the impeller. The impeller is of moderate specific speed design with a high exit backsweep angle. The radial vaneless space at the impeller exit has a minimum radius ratio of 1.08 (diffuser full open). The variable diffuser vanes are capable of being reset closed to give the desired flow range. The flow exits the diffuser into a single exit port collector which delivers the flow to the burner.

The preliminary design efforts were concentrated primarily on defining the most basic impeller design parameters. Both DDA testing and data from the published literature indicate an optimum specific speed for maximum compres-

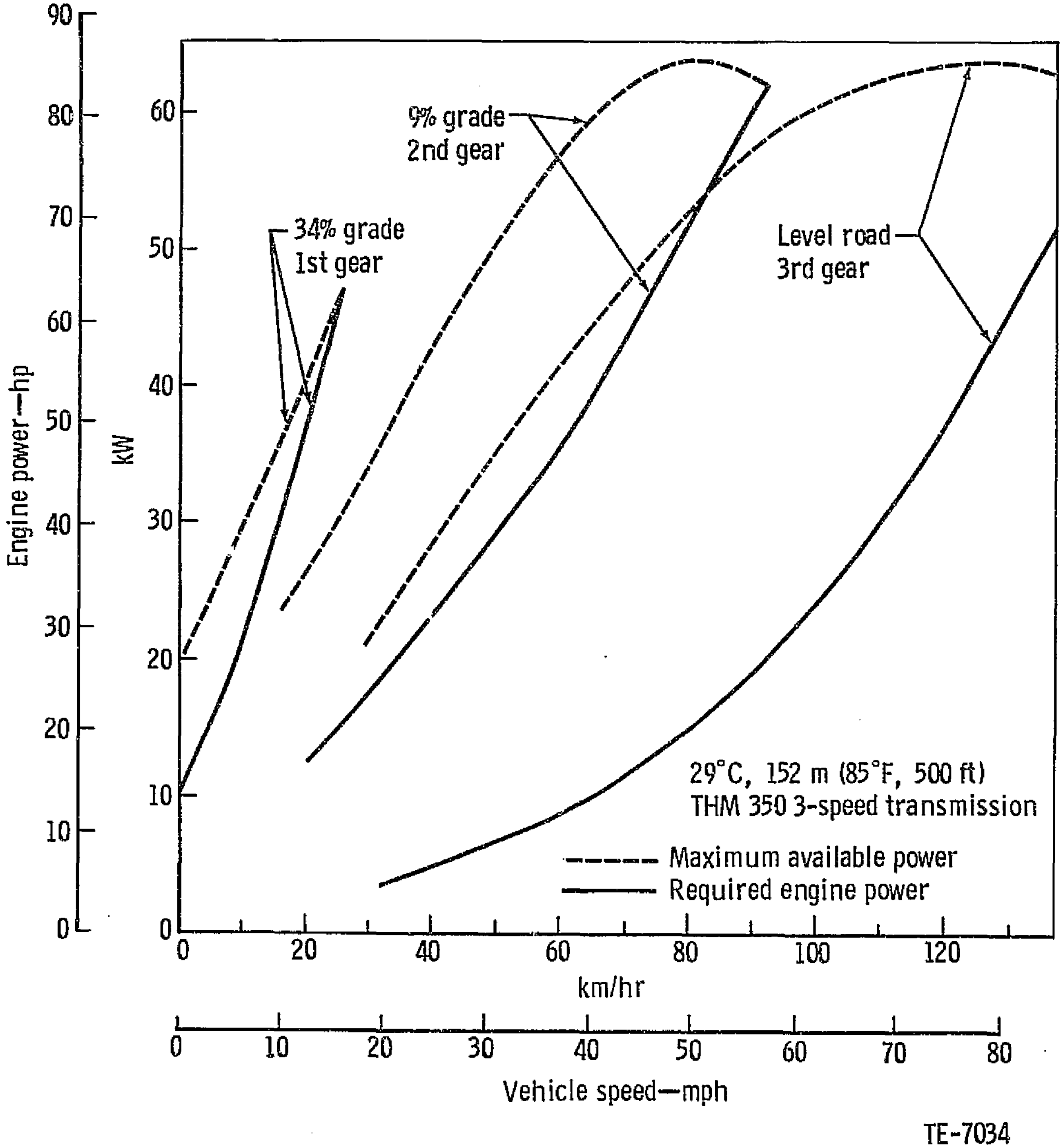


Figure 101. Turbine vehicle road load and grade capability.

TABLE XXXII. IGT VARIABLE-GEOMETRY COMPRESSOR DESIGN PARAMETERS		
Parameter	Units	Value
Compressor corrected flow	kg/s (lb/sec)	0.36 (0.80)
Compressor pressure ratio (total-to-static)	---	4.5:1
Compressor corrected speed	rpm	85,000
Compressor adiabatic efficiency goal (total-to-static)	%	76.25
Compressor specific speed	---	80
Range $\left[\left(\frac{W_{choke} - W_{surge}}{W_{choke}}\right) \times 100\right]$	%	72

sor efficiency. This background and some IGT parametric compressor designs led to the selection of a specific speed of 80, which should yield a maximum performance potential. The inducer hub/tip radius ratio was selected as 0.40 to be consistent with DDA experience for high-efficiency impellers. The inlet specific flow was set at 146.5 kg/m^2 (30 lb/ft^2) to maintain a moderate inducer tip relative Mach number and still yield inducer inlet angles which are quite tangential to minimize incidence changes as the flow is reduced.

DDA experience indicates that an increase in performance is achievable as backsweep is increased. Because a detailed impeller stress analysis was not completed, a 50-degree backsweep angle was chosen as an achievable goal on the basis of other DDA designs. The preliminary impeller included 15 full blades and 15 splitters which begin at 35% of the meridional distance through the impeller. An impeller flow analysis of the preliminary configuration was performed to check the blade loading levels. Moderately low loadings were achieved. On the basis of DDA empirical correlations, efficiency levels generally improve as impeller loading decreases. To attain the full potential, the final impeller design would be modified to achieve optimal loading and loading distributions.

The preliminary diffuser configuration consists of a constant-area vaneless space that has a 1.08 radius ratio when the vanes are at the full-open or design setting. Above the vaneless space, the diffuser has parallel sidewalls to maintain endwall clearances as the vanes are reset. The diffuser vanes were selected to be straight wedges with a 0.254 mm (0.010 in.) leading edge radius. The diffuser throat was sized to give an 8% minimum surge margin at design. The number of vanes (25) and the wedge thickness were selected to yield an eight-degree passage divergence angle at design setting. Vane reset studies indicate that the diffuser vane will require approximately an 11.5-degree reset to achieve the desired range. This means that the throat aspect ratio (axial dim./throat dim.) will vary from 0.87 to about 3.1 over the range of resets.

The off-design performance predicted for this IGT compressor was based on a DDA variable geometry "broad range" compressor test map. The IGT design point efficiency was derived by appropriate adjustments to the 505-IIIA demonstrated performance. All flow, pressure ratio, and efficiency values were then scaled from the original map.

The predicted IGT compressor map is shown in Figure 102. It demonstrates flat efficiency islands and over 70% range at all speeds from 60 to 100% $N/\sqrt{\theta}$.

Turbines

The two-shaft engine is structured around the use of two radial-inflow turbines--one for the gasifier and the other as the power turbine. Both turbines will incorporate variable flow capacity. The generalized characteristics of the turbine and associated goals are as follows:

	Gasifier	Power Turbine
Generalized characteristics		
Rotor inlet temperature, °C (°F)	1289 (2350)	1104 (2019)
Equivalent flow ($W\sqrt{\theta}/\delta$), kg/s (lb/sec)	0.253 (0.557)	0.451 (0.995)
Equivalent work ($\Delta h/\theta$), kJ/kg (Btu/lb)	43.3 (18.1)	46.1 (19.8)
Performance goals		
Efficiency (total-to-total), %	83.7	84.5
Flow range, %	50-100	60-113
Speed range, %	60-100	20-100

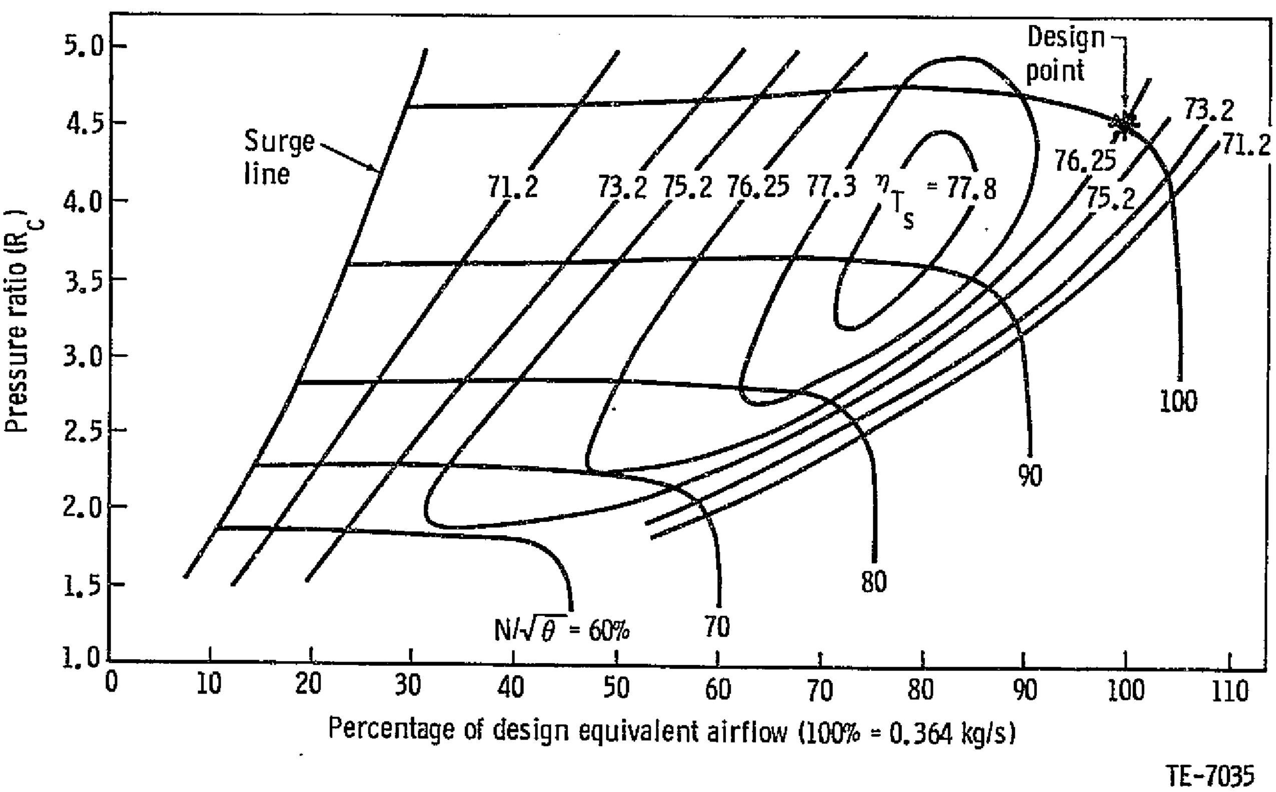


Figure 102. IGT variable-geometry composite compressor map.

The primary effort has been focused on the preliminary aerodynamic design of the gasifier turbine to expedite the development of the ceramic rotor. The power turbine activity has been directed toward defining turbine operating points along the engine operating lines in order to identify power turbine loss mechanisms. Through this identification, turbine design can be tailored toward accenting part-power performance.

Gasifier Turbine

The gasifier turbine flow path and design velocity triangle Mach numbers are illustrated in Figure 103. The turbine exhibits a tip speed of 500 m/s (1640 ft/sec). The specific speed of 78 and the aerodynamic load coefficient ($U_T/\sqrt{2gJ\,\Delta h_{is}}$) of 0.7 are nearly optimal values for radial turbine design. During the preliminary design phase, ceramic construction has posed no significant trade-off in aerodynamic performance. Consistent with previous DDA radial turbine design, radial blading has been used to minimize production cost.

Preliminary flow path calculations have been generated to provide velocity and temperature distributions for heat transfer and stress calculations. The circumferential x-y view of the turbine as seen from the inducer is shown in Figure 104. The predicted meanline blade-to-blade velocity distributions for this preliminary design are illustrated in Figure 105. These flow calculations will be refined following the structural updating of the blade thickness distribution.

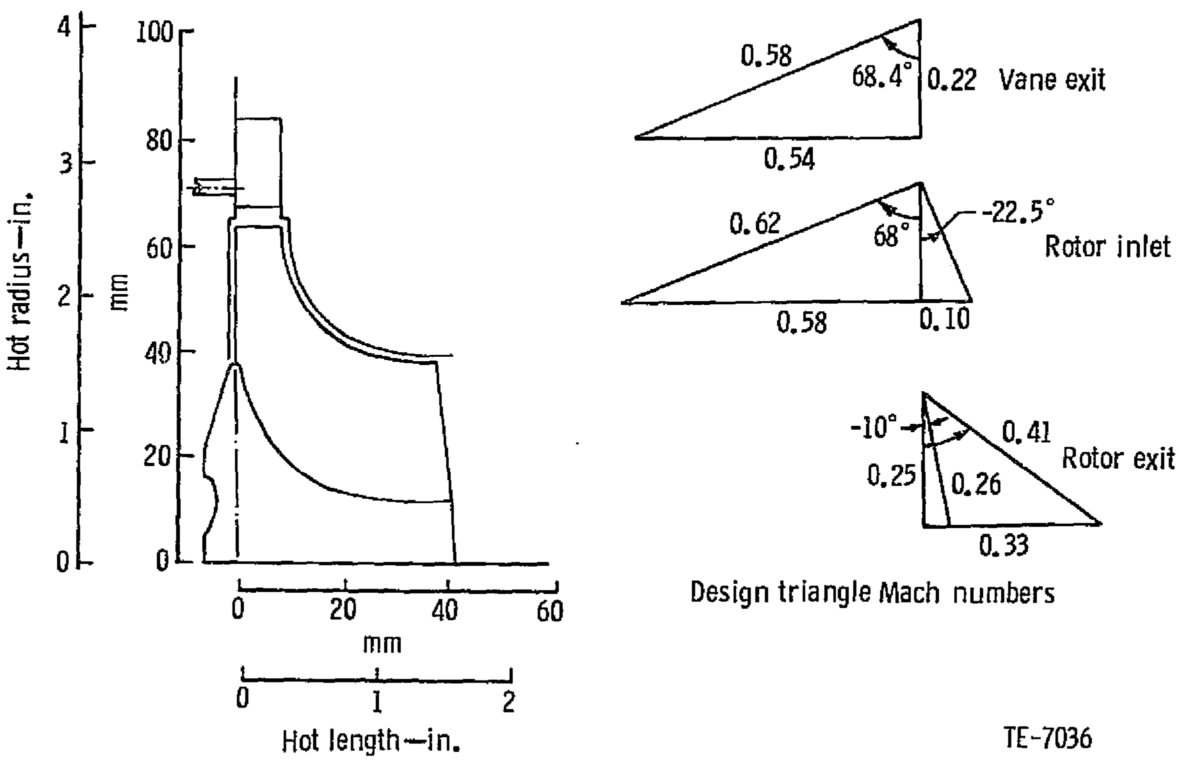


Figure 103. Flow path and velocity diagrams for IGT gasifier turbine.

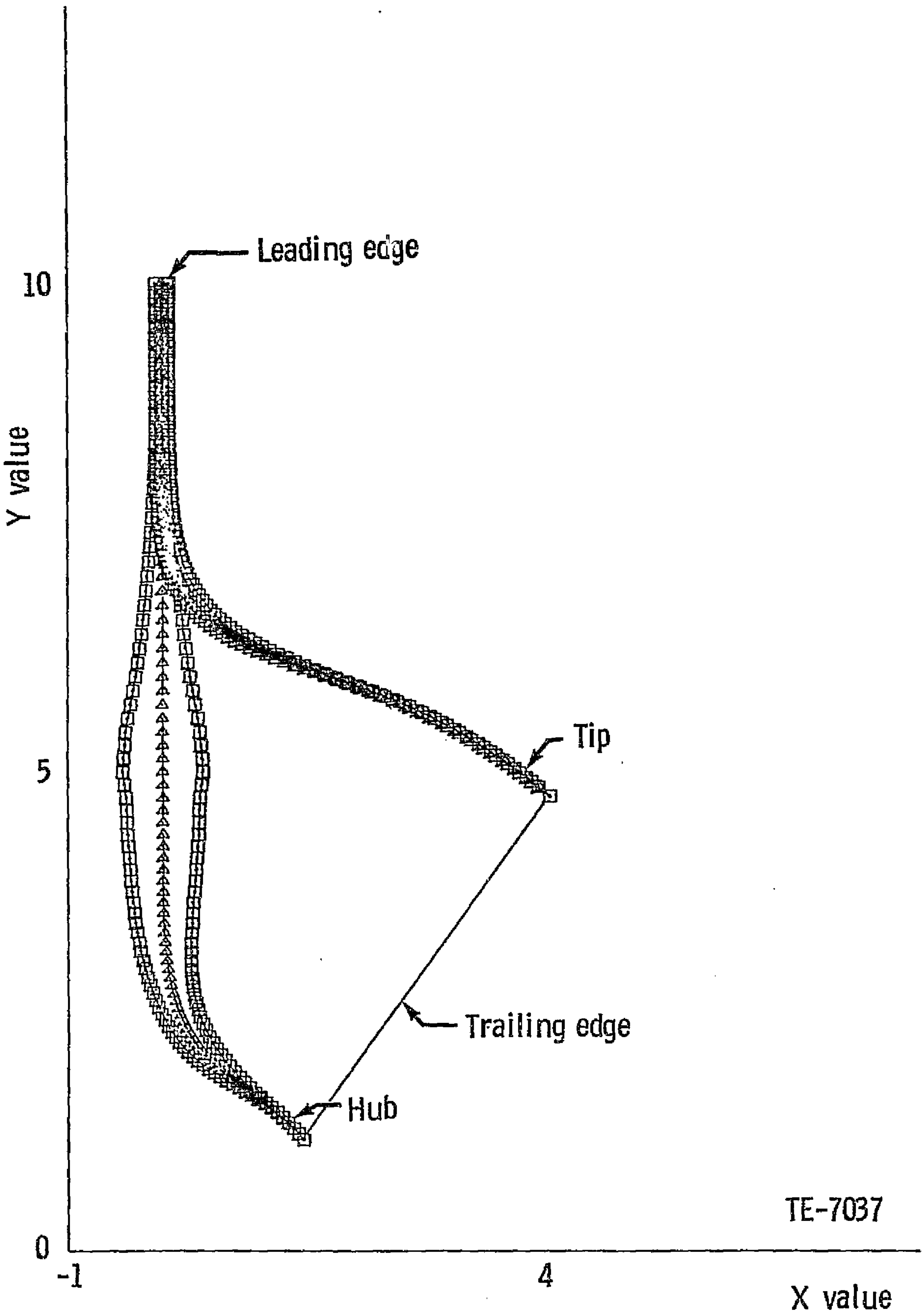


Figure 104. X-Y view—IGT gasifier rotor.

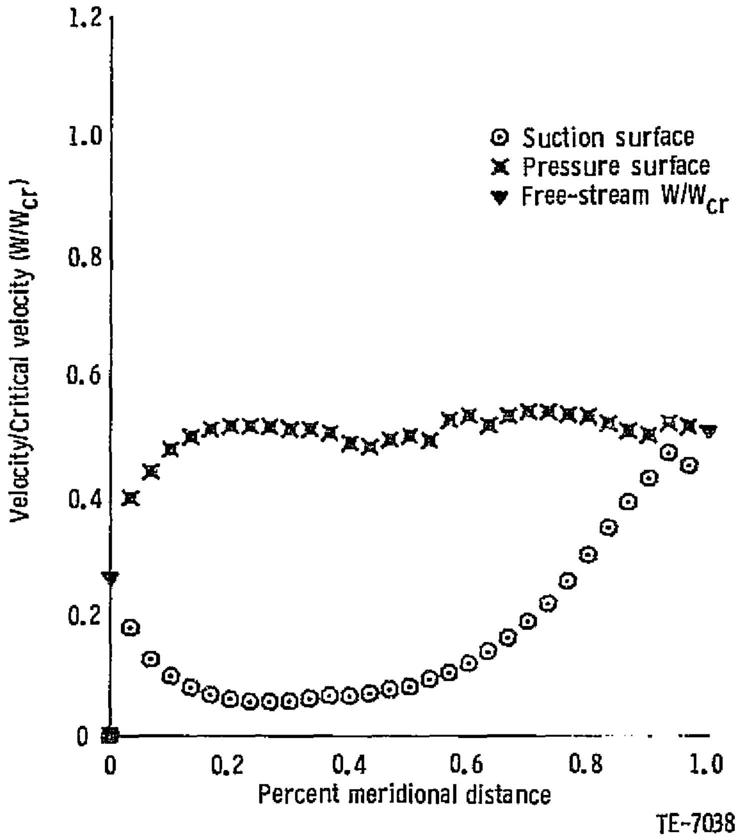


Figure 105. Meanline velocity distribution — IGT gasifier rotor.

The performance of the radial gasifier utilized for cycle calculations has been based on the test maps acquired on the AGT-2 turbine rig. Efficiencies and flow rates for the full-power vane setting angle as scaled to IGT engine requirements are shown in Figures 106 and 107. Efficiencies for reduced variable-geometry flow settings have been scaled on the basis of DDA experience.

Power Turbine

Because the power turbine is required to operate over such a broad range of speed and flow, attention was directed toward defining its operating characteristics before the start of the aerodynamic design. For this reason, a predicted preliminary design performance map was generated for the power turbine, using the DDA radial turbine performance code. This code not only predicts the overall performance but also subdivides losses among several loss mechanisms. Therefore, losses can be identified for any point of the turbine map. The predicted efficiency and flow characteristics for the full-power variable geometry setting are illustrated in Figure 108 and 109. Efficiencies at other variable geometry settings are obtained for cycle analyses through scaling similar to that of the gasifier turbine. The preliminary power turbine design exhibits a tip speed of 486 m/s (1593 ft/sec). The specific speed of 79.2 and aerodynamic load coefficient ($U_T/\sqrt{2gJ\Delta h_{is}}$) of 0.70 are nearly optimal values for radial turbine design.

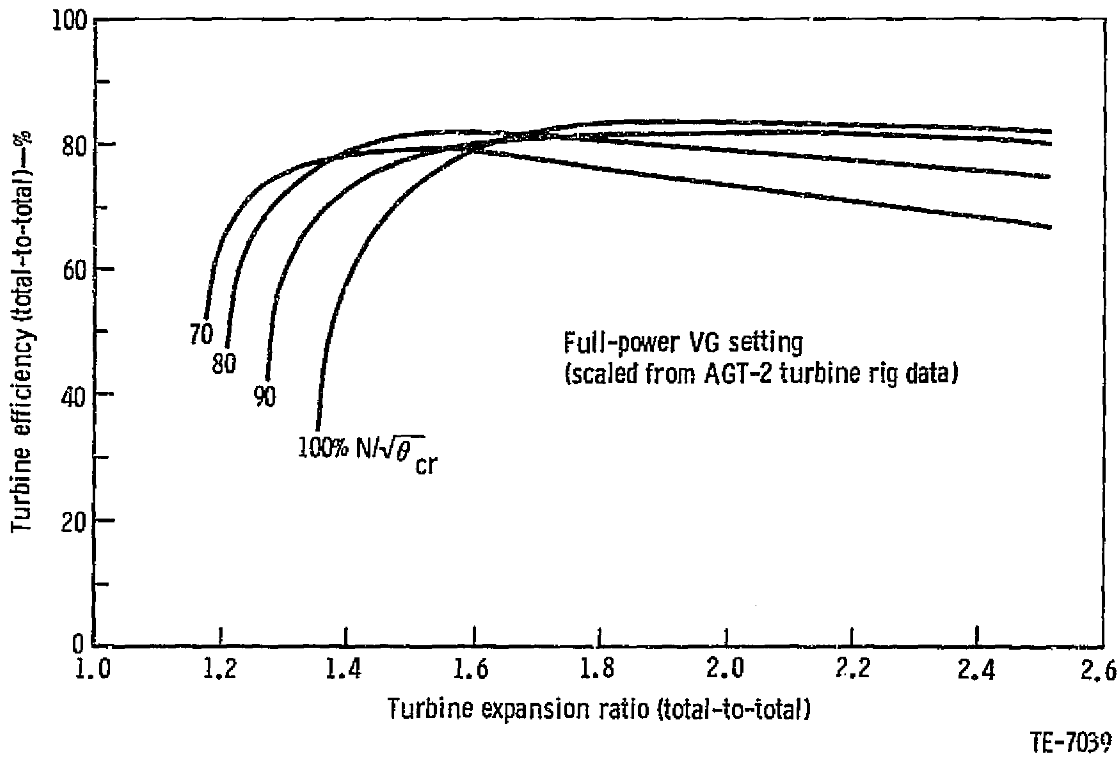


Figure 106. IGT gasifier turbine efficiency characteristics.

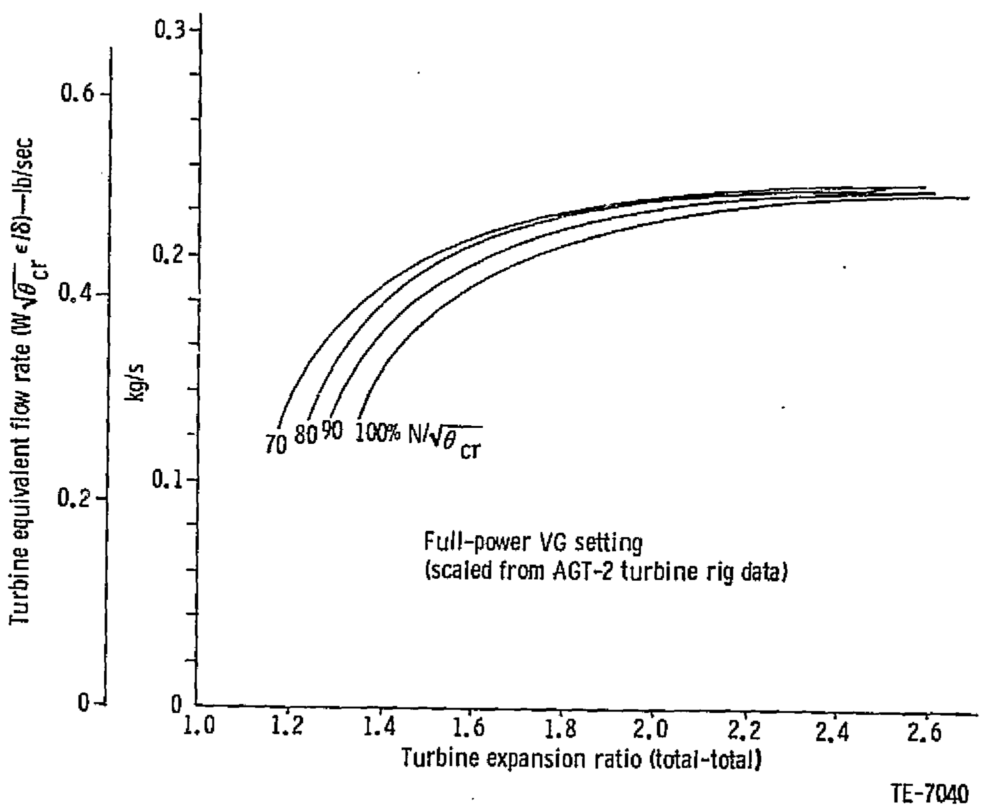


Figure 107. IGT gasifier turbine flow characteristics.

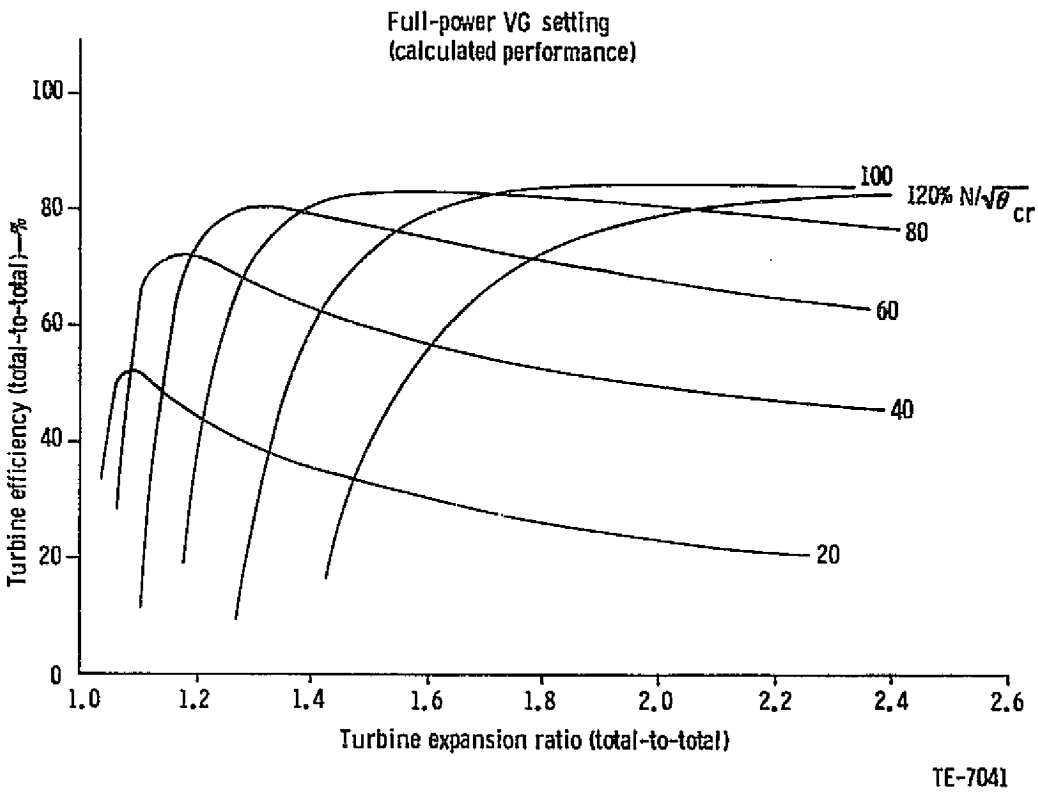


Figure 108. IGT power turbine efficiency characteristics.

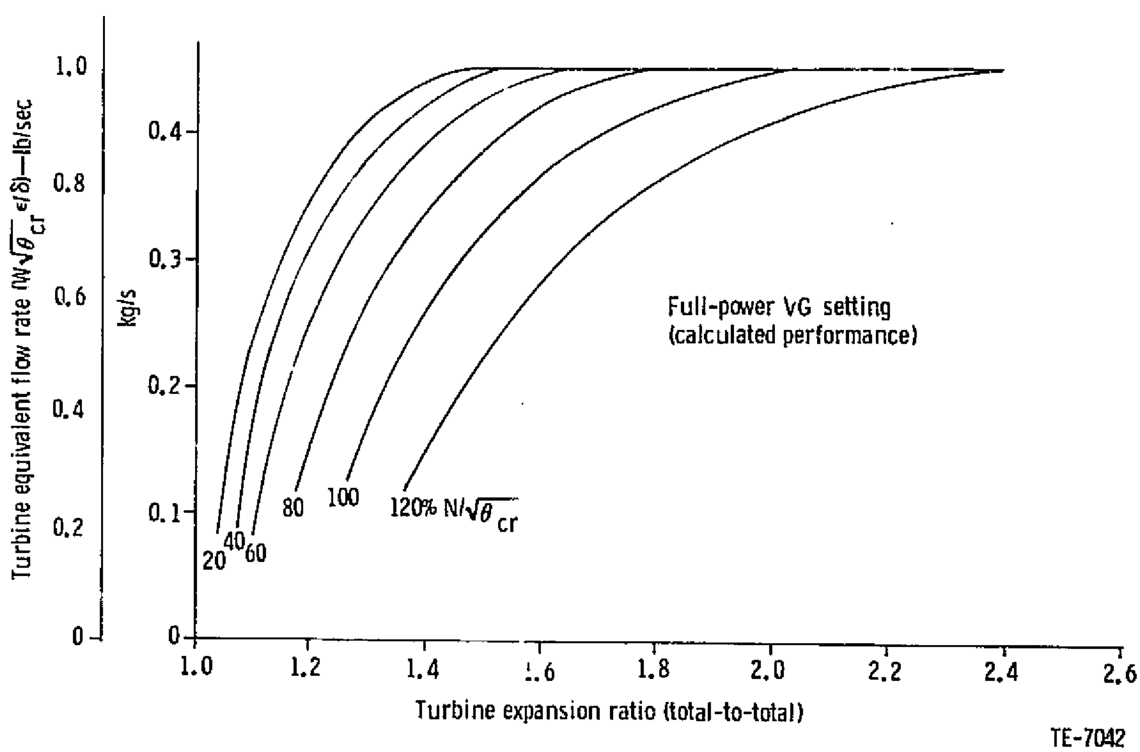


Figure 109. IGT power turbine flow characteristics.

The engine operating lines as seen by the power turbine are illustrated in Figure 110. The preliminary power turbine design results in low efficiency at the low engine power conditions. The calculated breakdown of the power turbine losses along the engine road-load line is illustrated in Figure 111. Volute/vane loss and rotor loss and rotor incidence loss are prime candidates for improvement. This information will guide the preliminary aerodynamic design iteration porcess to achieve a power turbine tailored to place performance emphasis on part engine power conditions.

Combustor

The combustor concept selected for the optimized IGT engine will be a prechamber configuration similar to the low-emission combustors of the GM-funded AGT programs as shown in Figure 112. The base-line design will feature a swirl prechamber followed by sudden expansion to a reaction zone which is closely coupled to a dilution zone. The design of the IGT combustor must include the basic features necessary for successful low emission operation. As discussed under "Emissions" (Section V), these include prevaporized-premixed fuel and air, low reaction zone temperatures, and either variable geometry or staged fuel. Accordingly, the prechamber combustor will have variable-geometry control of both primary zone and dilution zone air admission. The main fuel supply will be distributed on the prechamber wall in a thin film layer, where vaporization off the prechamber wall occurs, followed by premixing of the fuel vapor and swirler air.

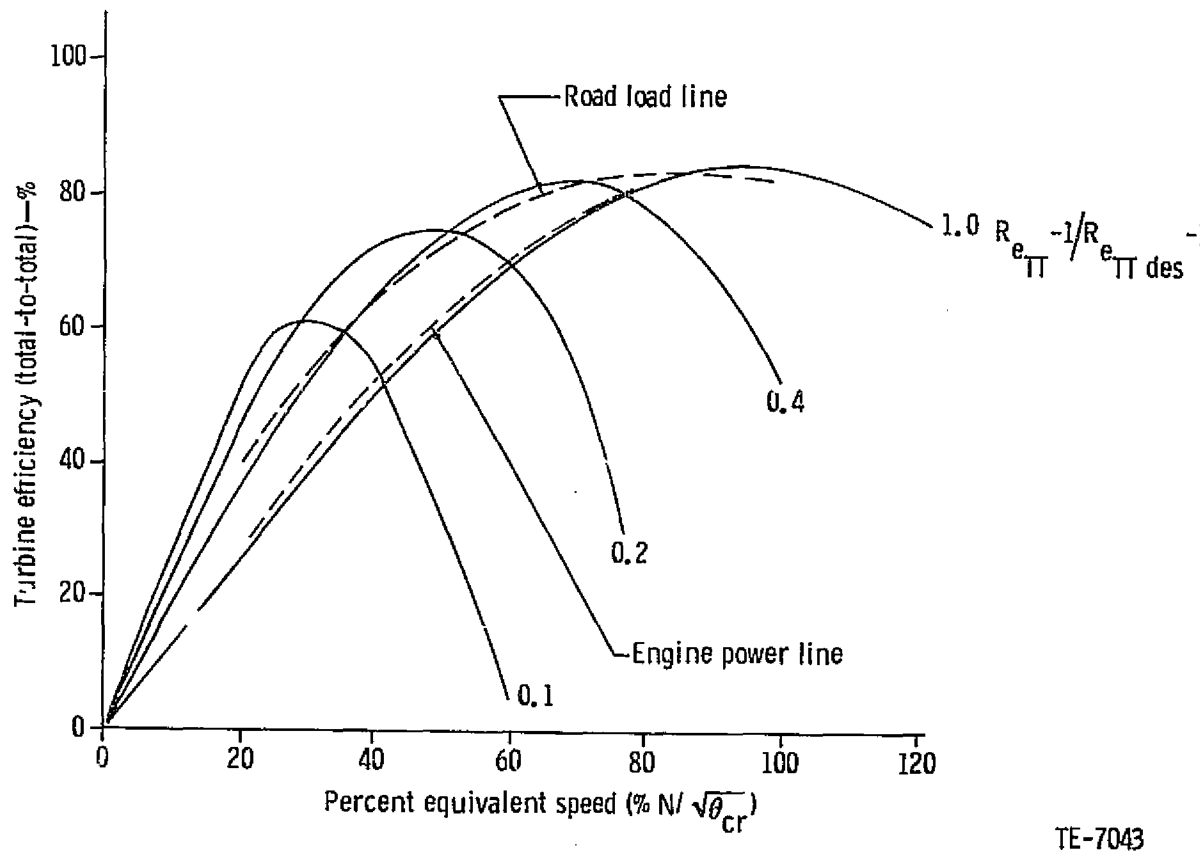


Figure 110. Engine operating lines superposed on preliminary power turbine efficiency map.

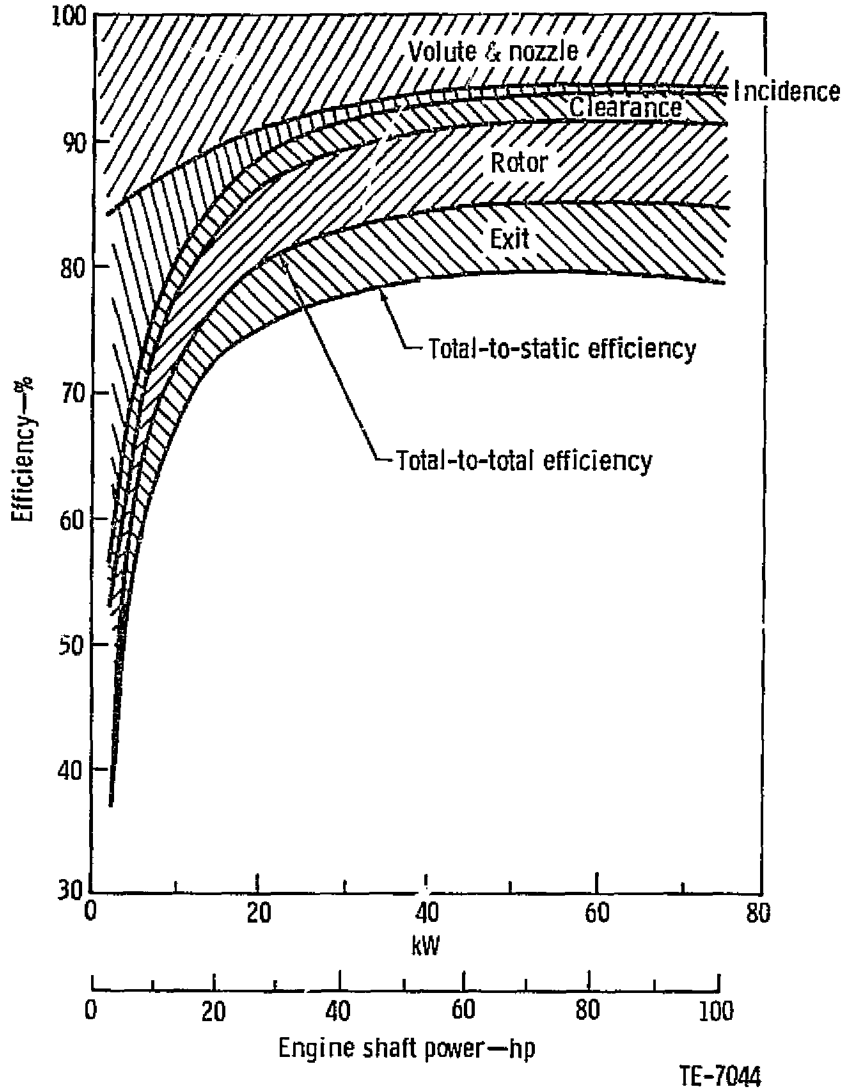


Figure 111. Predicted loss division of IGT power turbine along engine road load line—first iteration.

The detail design of the IGT combustor must also address new conditions of the optimized engine cycle. The major elements of these new design requirements are as follows:

1. Durability at increased combustor inlet and outlet temperatures
2. Pressure drop control over wide flow factor range
3. Preignition and fuel line vaporization and coking resulting from the high-temperature environment
4. Low, 0.27 kg/h (0.5 lb/hr) fuel rate pilot to provide continuous combustion with minimal emission contribution

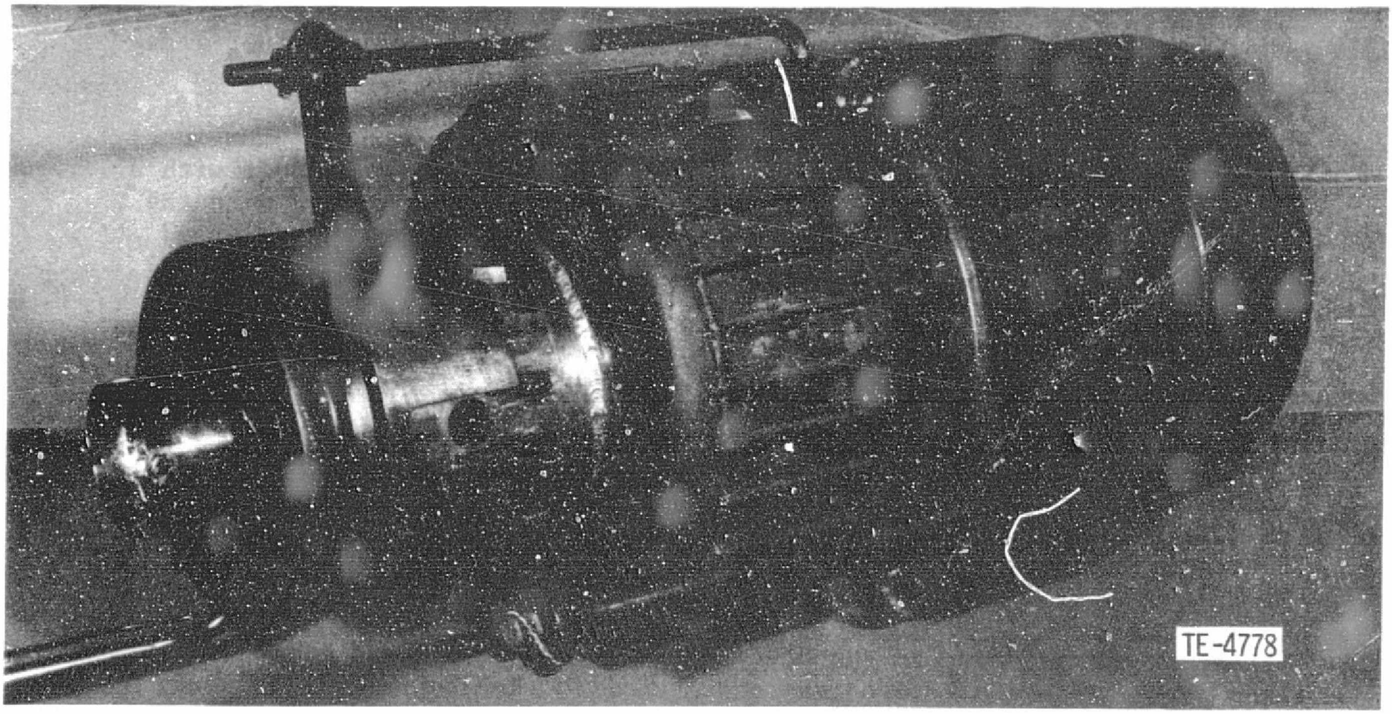


Figure 112. Low-emission combustor.

To achieve both durability and cost advantages, the combustor material must ultimately be a ceramic. The prime candidate material would be a form of silicon nitride. In its construction, the combustor will be segmented into shapes which reduce the stress in areas of complex shapes. During the development phase, a gradual transition will be made to ceramic sections, starting with simple shapes in the higher metal temperature zones. A transpiration-cooled material (Lamilloy®)* will be used to supplement the ceramic introduction. Lamilloy materials make very efficiency use of cooling air, which must be applied judiciously to minimize adverse effects on emissions.

Pressure drop control is an important consideration when dealing with wide flow range engines. The ratio of maximum to minimum flow factor ($W_a\sqrt{T}/P$) of the engines under review ranged from 2/1 to as high as 3/1. The ratio of maximum to minimum pressure drop for a fixed-geometry combustor which would be concomitant with this wide flow factor variation is proportional to the square of the flow factor. Thus, the maximum pressure drop could be as much as nine times the minimum pressure drop. By integrating the emission and pressure drop requirements into the variable geometry design, emissions can be controlled and pressure drop held nearly constant. This is done by balancing the proportions of primary zone or dilution zone variable geometry in combination with emissions and flow factor variables.

*Lamilloy is a registered trademark of the General Motors Corporation.

Prechamber preignition will be inhibited by offsetting the increased thermal inputs with higher prechamber velocities. Experience has shown that this thermal-aerodynamic balance must be maintained in the prechamber. Relative to fuel handling problems, it will be necessary to insulate the fuel lines from exposure to the high combustor inlet temperatures.

The low-fuel-flow pilot has been developed for the AGT engines. The function of the pilot is to facilitate main fuel ignition after shutoff with a negligible emission spike. Flameout protection is also provided with the pilot. The success of this operating mode has been clearly demonstrated with the General Motors AGT engines.

DDA has had wide experience with low-emission combustor design. The following concepts have been evaluated during GM-funded AGT programs:

- o External recirculation
- o Premix-prevaporization
- o Prechamber
- o Catalytic
- o Staged fuel, staged air
- o Fuel pyrolysis

The experience gained from the studies of these concepts and their variations places DDA in a knowledgeable position for developing an advanced-technology auotmotive gas turbine combustor. This experience will also aid and streamline the development task of converting new technology features to production acceptability.

Regenerator

The general arrangement of the proposed regenerator system is illustrated in Figure 113.

The system comprises the following basic components: a drive gear train from the power turbine rotor (only a portion of the train is shown); one rim-driven, hub-mounted ceramic regenerator disk; axially compliant face seals that define and control the gas flow path through the disk; a spherical hub bearing that facilitates slight "gimballing" of the disk as it reacts to varying face seal loads; and a hub spindle that is cantilever mounted on the regenerator cavity enclosure or housing. As previously stated, much of the CATE (Ceramic Applications for Turbine Engines, NASA Contract DEN3-17) regenerator technology will be directly applicable to the IGT engine.

The drive train must provide a speed reduction of at least 2500:1, based on estimated 100% design speeds in the range of 55,000-60,000 rpm for the power turbine rotor and 16-22 rpm for the regenerator disk. Because the regenerator gear and drive pinion must operate without the benefit of fluid lubrication and cooling, their teeth are expected to require very hard, heat-resistant coatings to provide adequate wear life.

The regenerator disk, in particular, will be essentially a replica of the unit now being developed in the CATE program (which is currently the Corning thin-wall T14-20 matrix), differing only in its outside diameter. It will be

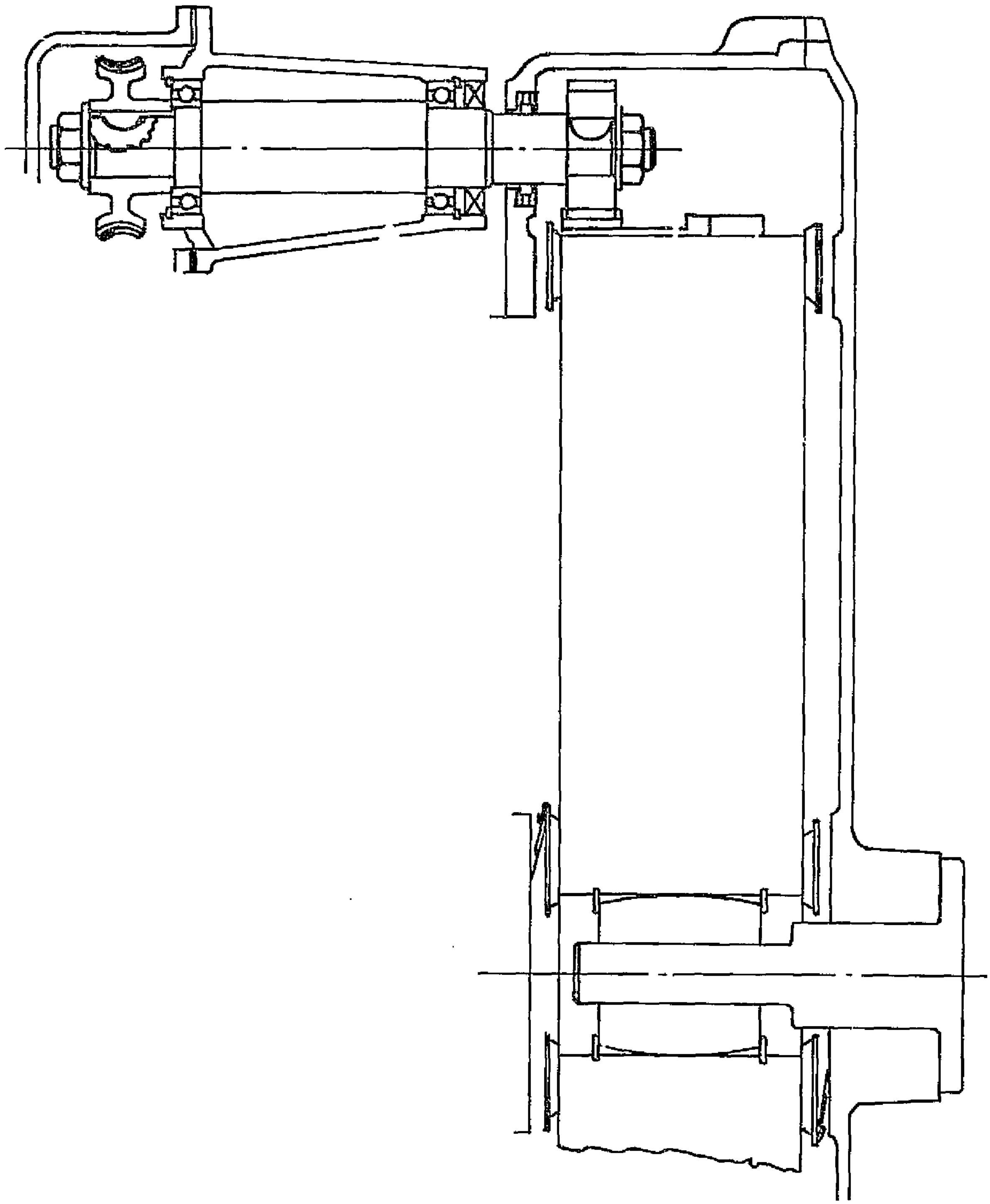


Figure 113. Partial cross section of regenerator system (preliminary).

76 mm (3.0 in.) thick and about 467 mm (18.4 in.) in diameter as opposed to a diameter of 640 mm (25.2 in.) for the CATE disk. Very high in surface-to-volume ratio, it will operate with a heat transfer effectiveness above 90% for all engine operational conditions. In addition to providing high performance in minimal space, the rotating heat exchanger offers the inherent advantage of self-cleaning as a result of the counterflow action of the two fluids. Elastomer is a prime material candidate for attaching the gear to the disk because of its favorable combination of properties: excellent molding and bonding characteristics that ensure uniform load distribution, plus adequate strength and resiliance over a broad temperature range.

Each face seal comprises three basic elements: a steel platform for structural rigidity and mounting; heat- and wear-resistant facings that interface with the disk; and axially compliant, pressure-actuated steel leaves that provide seals between the platforms and the regenerator housings. Because the basic function of the face seals is to prevent the high-pressure air on the forward side of the crossarms (and around the periphery of the disk) from bypassing the turbine by escaping directly into the low-pressure exhaust gas cavity on the aft side of the crossarms, their sealing efficiency is extremely important to engine performance. About 0.75 kW (1 hp) will be required to drive the regenerator, primarily to overcome the friction at the disk-to-seal interfaces and in the drive train gear meshes.

To ensure maximum contact between the disk and face seals over the full range of varying operational conditions that result from dimensional tolerances, thermal distortion, seal wear, and seal pressurization loads (a function of compressor impeller speed), a spherical hub bearing will be used to permit the disk to deviate from its normal plane of rotation. Carbon bearings developed for operation under moderate loads and temperatures up to 538°C (1000°F) are presently considered the most promising candidates.

The regenerator disk and bearing assembly will be supported by a central spindle mounted in the outer enclosure. In addition to providing the coolest environment for the spindle, outer wall mounting will facilitate the installation and removal of the regenerator disk and enclosure as a subassembly unit, thereby minimizing the possibility of disk handling damage. Spindle design considerations include the need for structural rigidity, a high-quality surface finish, and oxidation resistance to maintain satisfactory meshing of the regenerator gear with the pinion and minimum wear rates in the bearing bore.

A single regenerator disk system was selected for the IGT engine, based on the performance evaluation and the general arrangement evaluation. A single disk of 46.7 cm (18.4 in.) diameter and 7.6 cm (3.0 in.) thickness provides excellent performance and simple mechanical arrangement.

The following is a tabulation of input data to the regenerator performance model at the IGT engine design point. The resulting regenerator system performance is 93.6% effectiveness, 5.9% presure loss, and 4.34% leakage.

- Regenerator size

Diameter	467 mm (18.4 in.)
Thickness (flow length)	76 mm (3.0 in.)
Rim and crossarm seal blockage	566.5 cm^2 (87.8 $in.^2$)
Effective frontal area	1149.0 cm^2 (178.1 $in.^2$)
Gas-to-air-side area ratio	1.2:1
Rotational speed	17.6 rpm

- Flow conditions

Air side inlet flow	0.355 kg/s (0.783 lb/sec)
Air side inlet pressure	449.1 kPa (65.1 psia)
Air side inlet temperature	213°C (415°F)
Gas side inlet flow	0.367 kg/s (0.810 lb/sec)
Gas side inlet pressure	108.2 kPa (15.7 psia)
Gas side inlet temperature	913°C (1675°F)

Controls

Control Mode

The engine has four variables to be controlled:

1. Fuel flow
2. Power turbine nozzle position
3. Gasifier variable geometry position
4. Combustor variable geometry position

The fuel flow will be regulated to control rotor speed(s) and limit turbine gas temperatures as the engine is operated throughout start-up and power modulation.

The power turbine nozzles will be regulated to maintain the turbine gas temperature during steady state and positioned for dynamic braking when required.

The gasifier geometry will be operated in a high airflow position with gasifier speed at maximum for high power conditions. For idling and other low power conditions, the gasifier geometry will be varied to vary power while the gasifier speed is held constant at 70% of maximum. For intermediate or cruise power conditions, the gasifier geometry will be set to an optimum condition and the gasifier speed will be modulated to vary power.

The combustor geometry will be positioned to maintain the proper combustor reaction zone temperature required for low exhaust emissions. As the engine fuel/air ratio and combustor inlet temperatures varies, the position of the combustor geometry will be varied accordingly.

The power modulation will be selectable by a single input: the throttle. The fuel flow and variable geometries will be automatically positioned by the control system to maintain the necessary coordination of the variables. The basic engine parameters to be used will be rotor speeds and gas temperature(s).

Control System

DDA believes that an electronic control would best provide the closed-loop control required and maintain the appropriate coordination of the four variables in the many steady-state and dynamic modes of operation. The closed-loop, variable-geometry control will be on engine parameters and/or position feedback sensors to maintain the geometry position control required. For the development and demonstration phases of the engine program, a flexible, programmable design would be used to permit rapid changes in logic, schedules, settings, and dynamics. A production design could then follow to provide a more practical configuration.

The fuel system would include a fuel pump, a fuel metering valve, a flow divider, and a shutoff. An electric-motor-driven, positive-displacement pump appears to be the best candidate. The fuel metering valves in the development and demonstration program phases will vary fuel in response to the electrical input from the electronic control. A production design will possibly have a mechanical input from the throttle along with the electronic control signal. The flow divider valve is required to control the distribution of the metered flow between the main combustor manifold and the start-up fuel injection manifold. An electrically operated fuel shutoff valve will provide for fuel shutoff for engine shutdown.

All three variable-geometry actuation systems will be controlled by the electronic control on the initial engines, using engine oil to provide the actuating power. In a production design, it may be decided to power certain variables by compressor discharge air or even to operate a variable directly by mechanical means.

Transmission

DDA's commercial PREP computer program (PRedicted Equipment Performance) has been used to indicate the final performance of the selected three-speed conventional automatic torque converter transmission at varying engine throttle settings for a two-shaft gas turbine engine. The performance calculations include:

- o Charging pump loss
- o Torque converter efficiency
- o Gear mesh loss
- o Spin loss

The loss data were obtained from actual automotive transmission test data.

Figure 114 shows the transmission output power versus vehicle speed for various constant engine throttle settings. Figure 115 shows the transmission efficiency in third gear versus vehicle speed and throttle. Figure 116 shows the 100% throttle efficiency in all three gears versus vehicle speed.

ENGINE DESIGN FOR SELECTED ARRANGEMENT

Component sizes used for the selected arrangement design study were based on a 0.45 kg/s (1.0 lb/sec) mass flow at sea level static, standard day, 4.5

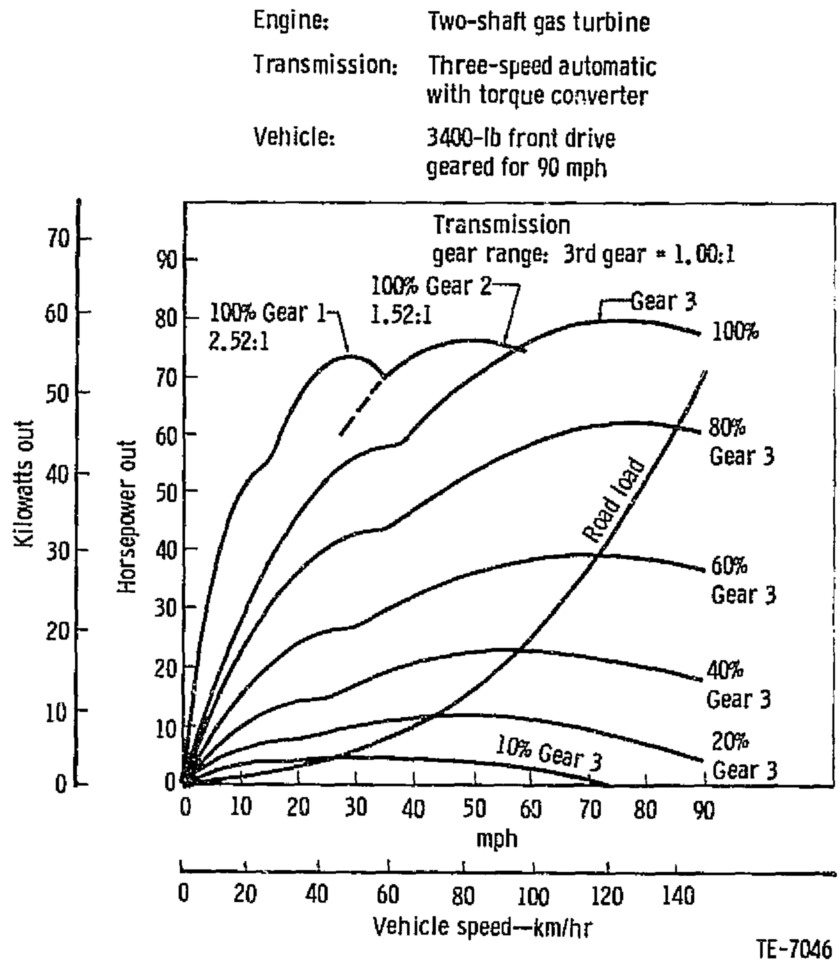


Figure 114. Transmission performance—HP_{out} versus vehicle speed at 10-100% throttle.

compression ratio, and 1149°C (2100°F) turbine inlet temperature. These parameters were selected to provide a base-line mechanical size that would adequately cover the ultimate parameters resulting from performance sensitivity studies. Thus, a meaningful design study was made with a minimum of perturbations from changing turbine temperature, mass flow, etc.

The arrangement selected for the design study is a two-shaft engine with a single regenerator disk, a can combustor, a centrifugal compressor, a radial-inflow gasifier turbine, and a radial-inflow power turbine. Figure 117 shows a left side view, a front view, and a plan view of the bare engine. Figure 118 shows a cross section through the major features of the selected engine along with the reduction gearbox cross section and the regenerator drive train.

The compressor is a centrifugal design that features variable inlet guide vanes and variable exit diffuser vanes to allow for engine operation over a wider flow range than permitted by a conventional fixed-geometry configuration. Air enters the compressor inlet annulus through a single cast passage and is given prewhirl by the radially mounted inlet guide vanes (IGV) before flowing into the impeller.

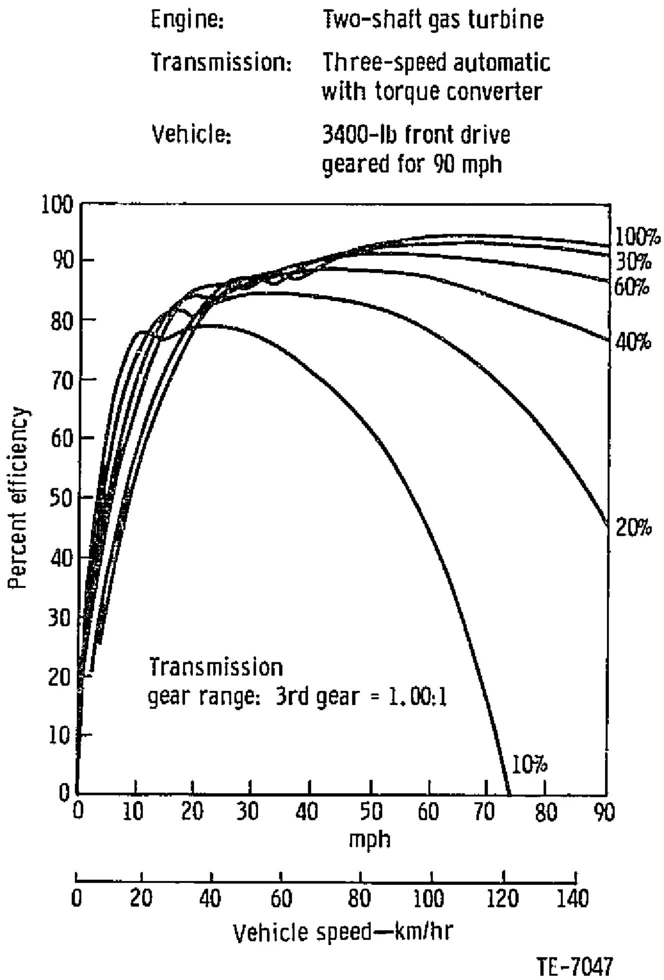


Figure 115. Transmission performance--efficiency versus vehicle speed at 10-100% throttle.

The variable-geometry vanes in each of the locations used are moved relative to each other to change the flow area (hence the airflow) through the gas path. This is accomplished by the motion of an engine oil-operated hydraulic actuator positioned by a function of the electronic control.

Air from the variable diffuser is discharged into a constant-velocity compressor scroll and is transferred to the regenerative heat exchanger through a transfer tube. The regenerator is a rotating counterflow heat exchanger with a sintered-ceramic, sinusoidal, spirally shaped, cellular-structure matrix. The heated air is collected from the regenerator and is fed to the combustion outer casing and thence into the combustor. The low-emission combustor features variable geometry in the primary zone. The heated air flows from the combustor through a transition section to a constant-velocity, spiral-annulus turbine scroll. The gas is fed to the radial-inflow turbine through variable turbine vanes and is discharged through a diffuser passage into a constant-velocity annulus and to the radial-inflow power turbine through variable turbine vanes. The ceramic gasifier rotor could be coupled to the metal shaft by bonded or mechanical joints. Either way will require a considerable effort in a development program. A bonding system re-

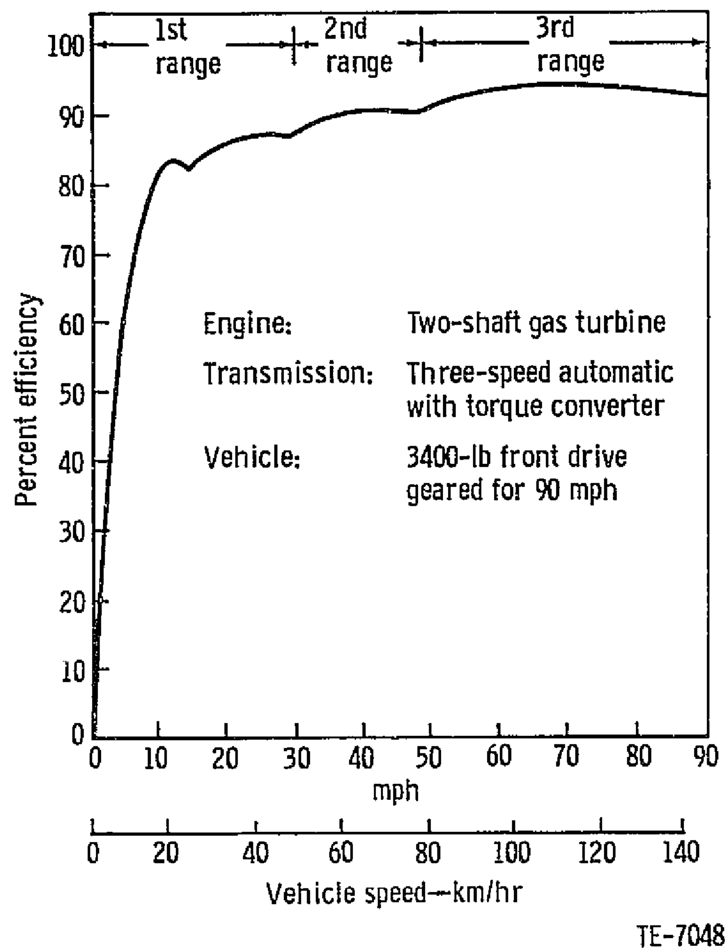


Figure 116. Transmission performance--efficiency versus vehicle speed at 100% throttle.

quires the development of materials as well as a system itself that will possess adequate strength at temperature.

Possible mechanical attachments would include but not be limited to Curvic couplings, pinned joints, and clamping arrangements. These must solve the problems of relative radial growth, dimensional accuracy, and compatibility of materials.

From the power turbine, the gas is discharged to an exit diffuser, turned, and distributed to the hot side of the regenerator. After flow through the regenerator core, the gas is discharged through a rear-directed exhaust port.

The engine gasifier rotor is so designed that the turbine shaft acts as a tie bolt to clamp the compressor impeller as well as provides the input drive from the clutch-coupled belt-transmitted starter drive. The rotor thrust is taken by a ball bearing located between the turbine wheel and the compressor impeller.

Rolling element bearings are used throughout the study engine, reflecting the years of experience at DDA with high-speed and high-temperature applications. The use of foil bearings has potential advantages, but such bearings do not have significantly higher temperature capabilities over the conventional bearing and would require an extensive development program.

The power turbine is mounted to the reduction gear case, driving through a two-stage reduction to the output shaft. Additionally, as shown in Figure 119, a worm and worm gear drives the regenerator disk and a spiral bevel gear set drives the oil pump. The engine oil pump not only supplies oil to the engine but also supplies the hydraulic power to operate the variable geometry actuators. Output torque is supplied to the transmission through a conventional automotive flex plate.

Materials for the candidate engine were selected for costing purposes as defined on specimen drawings of similar parts from other engine models. The final selection will be made in the design portion of Phase III.

A preliminary engine weight was estimated for the selected engine, using the materials assumed, scaling from the general arrangement drawing, and utilizing experience from other gas turbines. This engine weight of 156.5 kg (345 lb) assumed the full use of static ceramic parts.

The parts list used for costing purposes is presented in Table XXXIII.

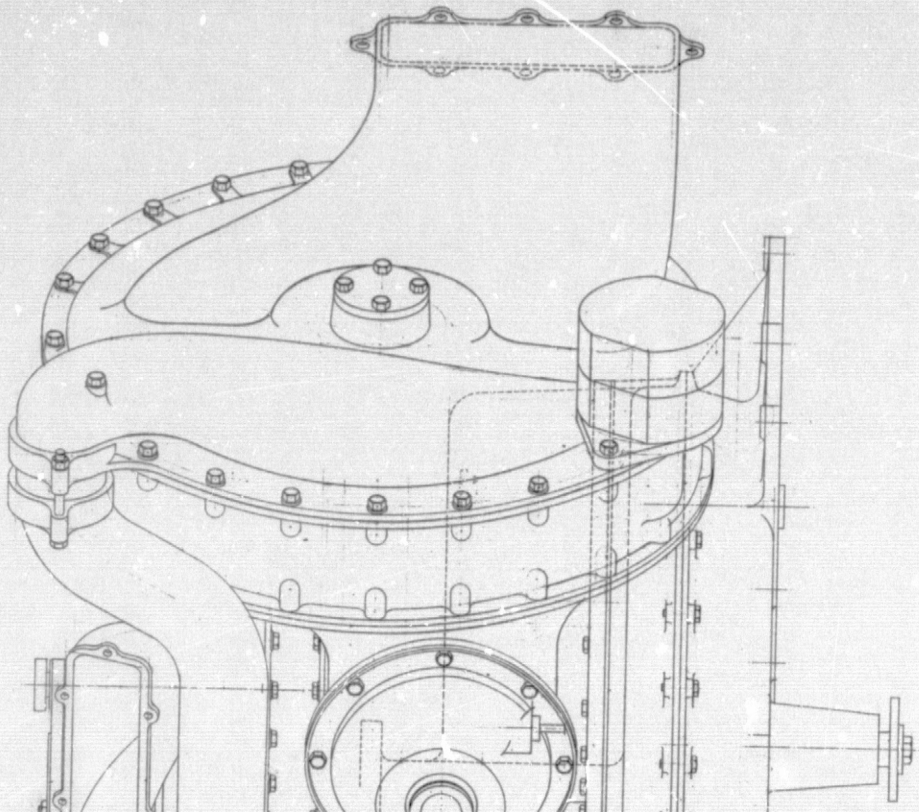
RISK ASSESSMENT

Many attempts have been made to analyze risk on a scientific and mathematical basis. For aerothermodynamic components, however, such methods are misleading and of little value. The risk assessment used for this analysis is subjective and based on the best judgments and abilities of knowledgeable technical experts at DDA. While the quantified assessments of risk presented in this section cannot be considered exact, the relative values have significant meaning.

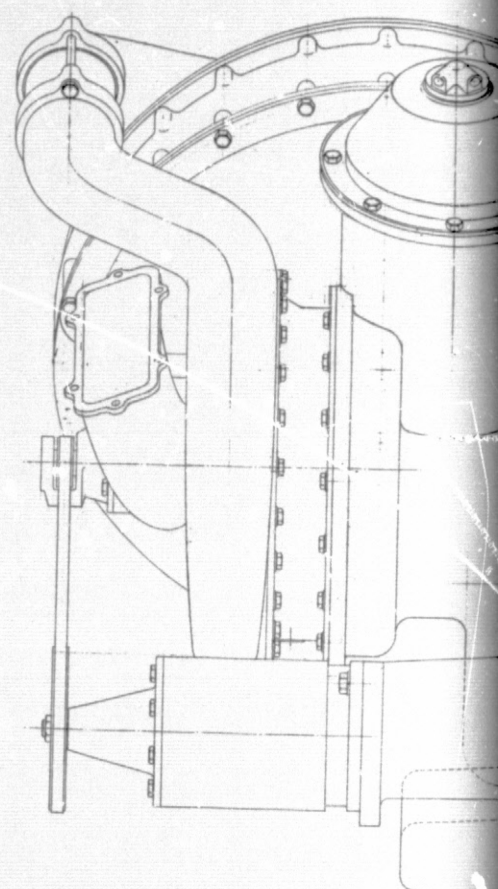
The objectives of this program require an advancement in the state of the art of small gas turbines and therefore involve some high-risk elements. These are justified by the large payback in energy conservation and environmental improvement which become possible as a result of a successful demonstration of the program goals. The resolution of these risks has been the deterrent to date in the commercial application of the gas turbire to the automobile.

Table XXXIV lists the various categories of risk against which the selected concept was assessed.

Tables XXXV and XXXVI present the assessments of risk, for the selected concept, in the areas of (1) performance and emissions and (2) mechanical, reliability, producibility, and safety. The two areas of high risk in the selected concept are the application of ceramics to the turbine rotors and the simultaneous achievement of high efficiency, wide flow range, and low cost in the small aerodynamic components. The approach that is recommended is to concentrate effort on the basic configuration, with variants for backup, and not to dilute effort on completely different alternates, such as



PLAN VIEW



FRONT VIEW

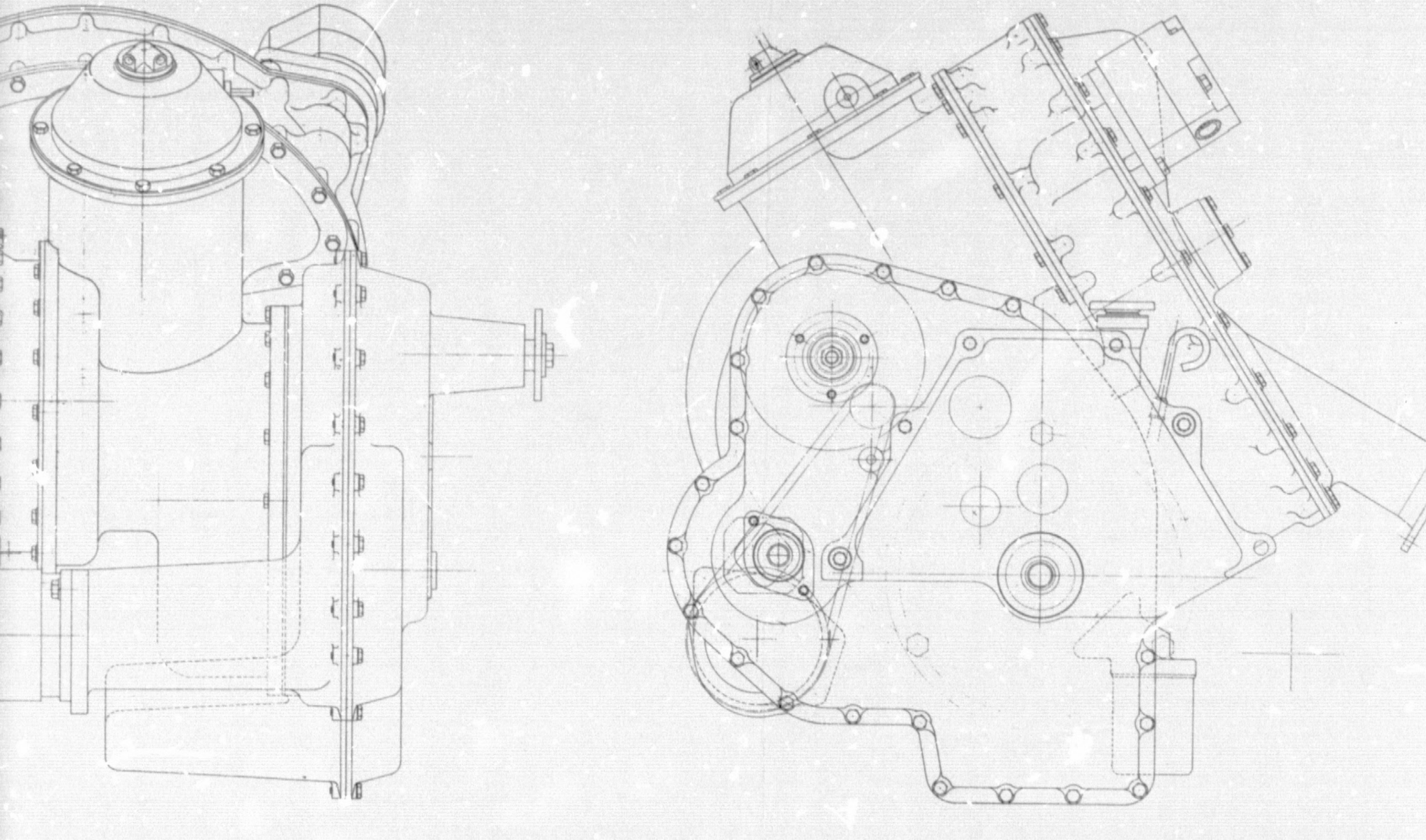


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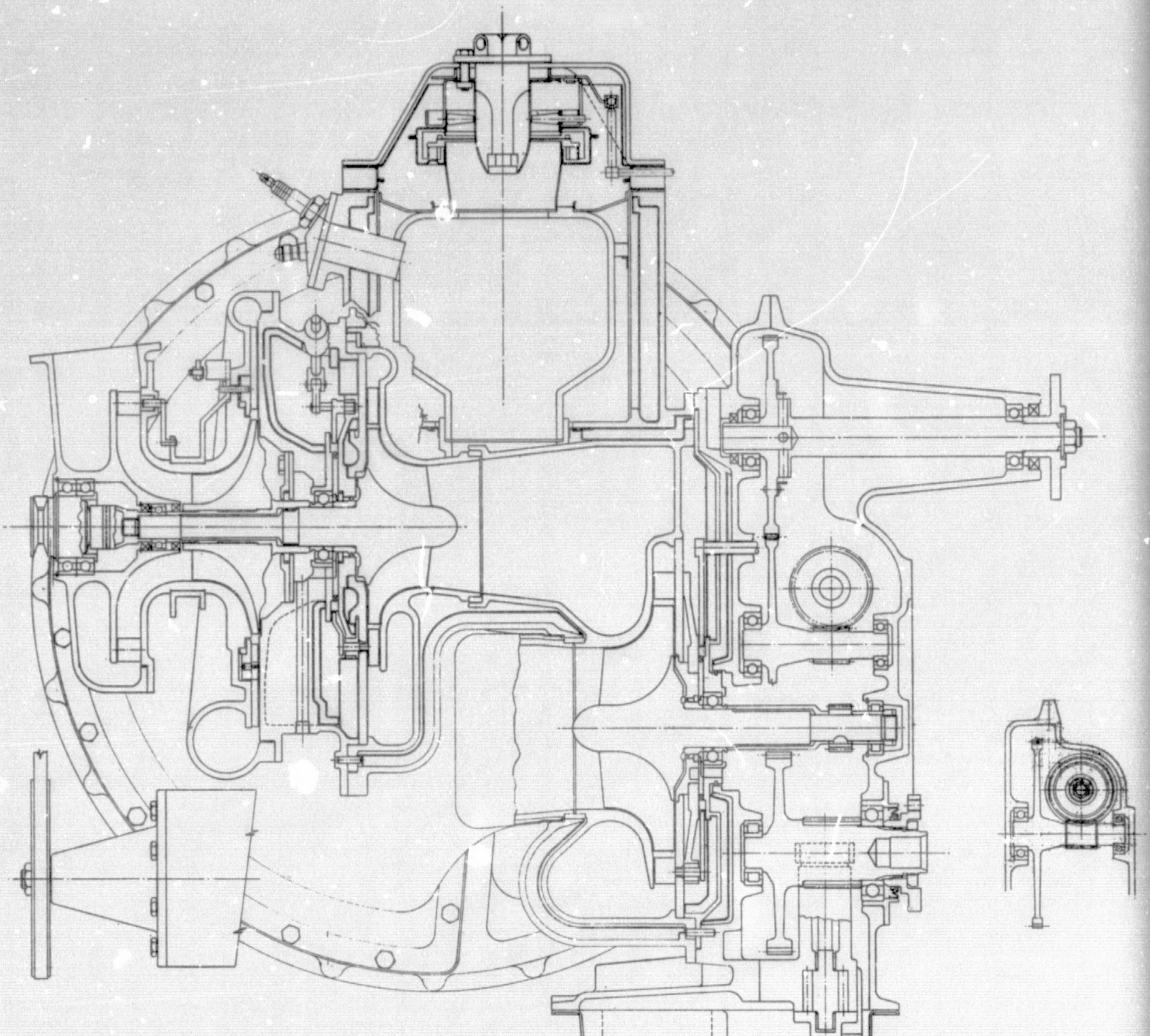
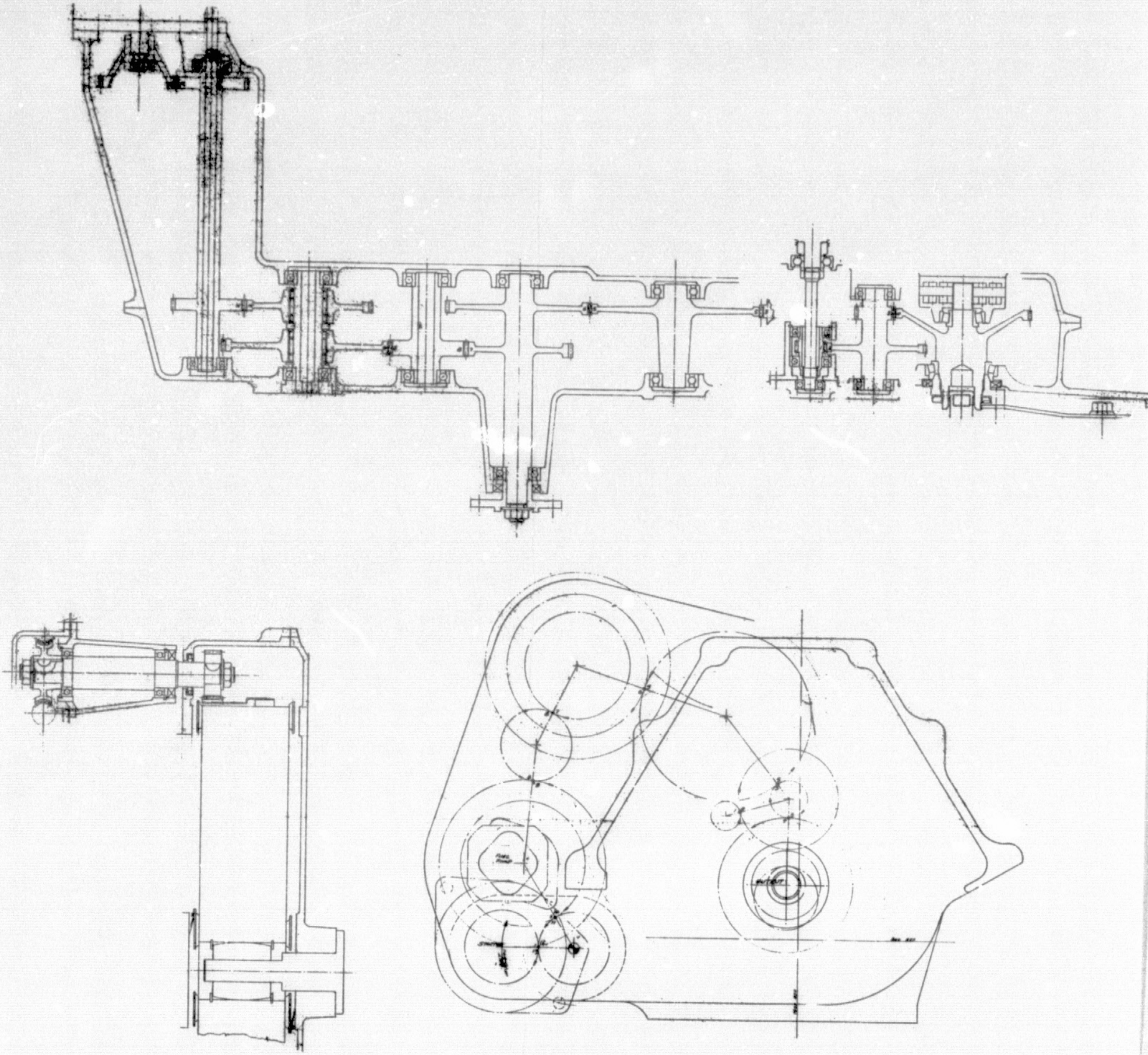
Figure 117. IGT two-shaft engine external arrangement.

2 FOLDOUT FRAME

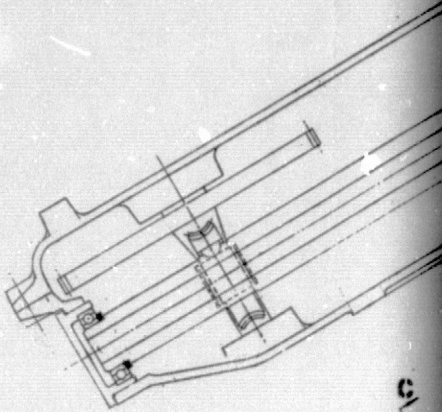


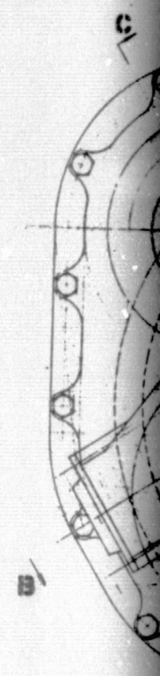
FOLDOUT FRAME 1

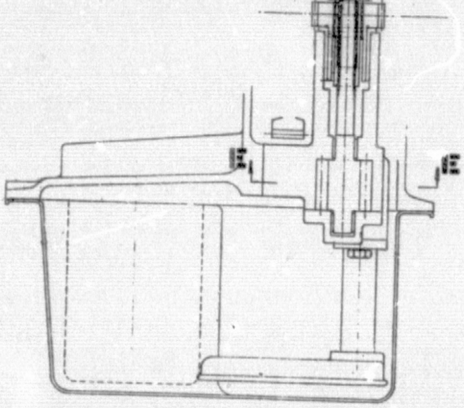


TE-7050

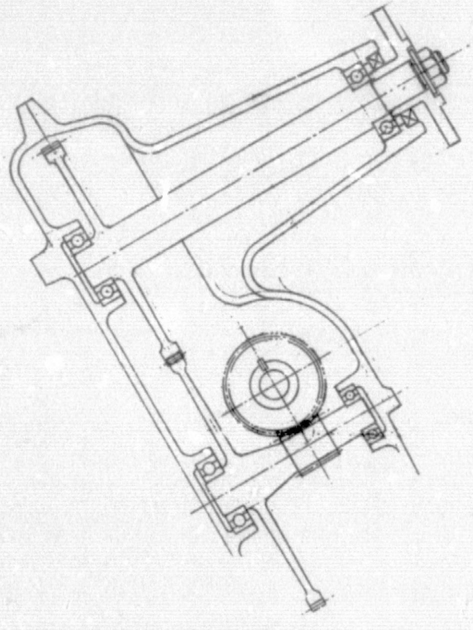
Figure 118. IGT two-shaft engine general arrangement.







SECTION D·D



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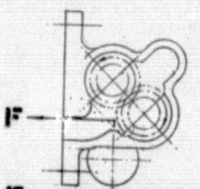
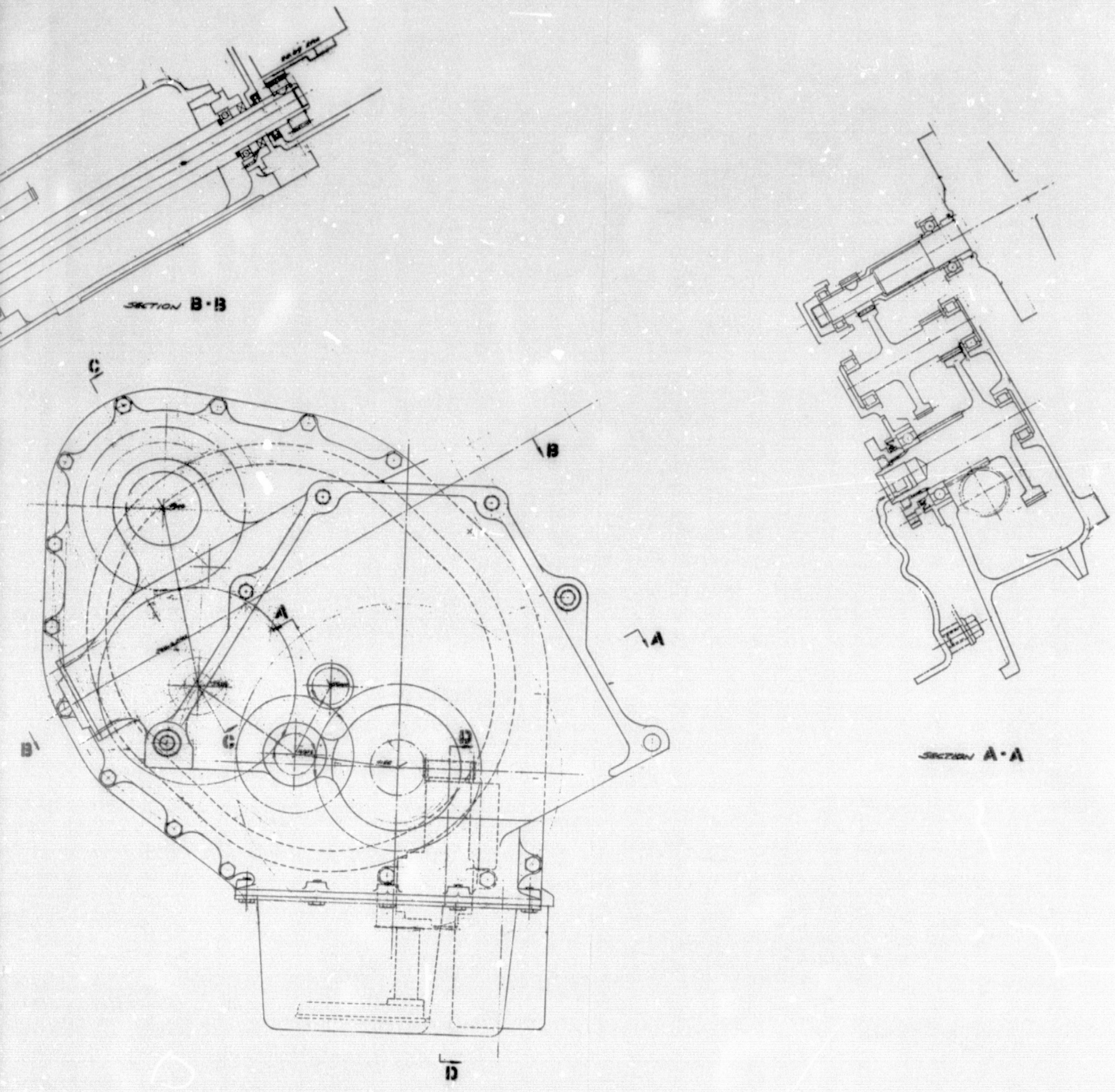


SECTION E·E

SECTION F·F

FOLDOUT FRAME 1



TE-7051

Figure 119. IGT two-shaft engine gear train arrangement.

TABLE XXXIII. IGT ENGINE PARTS LIST		
Group	Part Number	Description
6A1	ENGINE BLOCK	
	6A1-01	BLOCK-ENGINE
6A9	COMBUSTOR AND COVER	
	6A9-01	LINER ASSY, COMBUSTOR
	6A9-02	LINER, COMBUSTOR
	6A9-03	WASHER, RETAINING - INNER COMB.
	6A9-04	WASHER, RETAINING - OUTER COMB.
	6A9-05	TRANSITION, COMB. LINER
	6A9-06	COVER ASSY, COMBUSTOR
	6A9-07	COVER ASSY, COMBUSTOR
	6A9-08	COVER, COMBUSTOR
	6A9-09	HOUSING, DRAIN
	6A9-10	SHROUD, INNER - SWIRL VANE
	6A9-11	SEAL, SWIRL VANE
	6A9-12	SUPPORT, COMB. SWIRL VANE
	6A9-13	LEVER, COMB. SWIRL VANE
	6A9-14	VANE ASSY, SWIRL
	6A9-15	SHAFT, SWIRL VANE
	6A9-16	VANE, COMBUSTOR SWIRL
	6A9-17	RING, ACTUATOR COMBUSTOR
	6A9-18	NUT, RET - PILOT NOZ ASSY
	6A9-19	NUT PLATE, COMB. COVER ASSY
	6A9-20	BOLT, HEX HD NO. 10-24 x 19 mm (3/4 in.) LG
	6M6-23	NOZZLE ASSY, COMBUSTOR PILOT
	6M6-30	GASKET, PILOT NOZZLE
	6M6-31	BOLT, 1/4-20 x 29 mm (1-1/8 in.) LG
6A12	SCROLL ASSY & GASIFIER NOZZLE	
	6A12-01	SUPPORT ASSY, BEARING
	6A12-02	MOUNT ASSY, BEARING
	6A12-03	SEAL, PLATE
	6A12-04	RING ASSY, ACTUATOR VANE
	6A12-05	SUPPORT ASSY, TURBINE VANE
	6A12-06	VANE & SHAFT ASSY, TURB NOZ
	6A12-07	VANE, TURB
	6A12-08	SHAFT, TURB VANE
	6A12-09	ARM, TURBINE VANE
	6A12-10	SEAL, TURBINE OUTER
6B	REDUCTION GEAR HOUSING	
	6B-01	HOUSING ASSY, REDUCTION GEAR FRONT

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TABLE XXXIII. IGT ENGINE PARTS LIST (cont)		
Group	Part Number	Description
6C5	POWER TURBINE ROTOR	
	6C5-01	ROTOR ASSY, POWER TURBINE
	6C5-02	WHEEL, TURBINE
	6C5-03	SHAFT, TURBINE
	6C5-04	SEAL, TURBINE SHAFT
	6C5-05	BEARING, BALL - 15 x 32 x 10
	6C5-06	NUT, SELFLOCK 9/16-18
6D3	POWER TURBINE SCROLL & NOZZLE	
	6D3-01	VANE AND SHAFT ASSY, POWER TURB NOZ
	6D3-02	VANE, TURBINE NOZZLE
	6D3-03	SHAFT, TURBINE VANE
	6D3-04	ARM, TURBINE VANE
6F1	OIL PUMP ASSEMBLY	
	6F1-03	BODY, OIL PUMP-HIGH PRESSURE
	6F1-01	GEAR, ENGINE OIL PUMP HP DRIVE
	6F1-02	GEAR, ENGINE OIL PUMP HP DRIVEN
	6F1-04	TRANSFER PLATE, HP TO LP
	6F1-05	GEAR, ENGINE OIL PUMP LP DRIVE
	6F1-06	GEAR, ENGINE OIL PUMP LP DRIVEN
	6F1-07	BODY, OIL PUMP-LOW PRESSURE
	6F1-08	PUMP ASSY, OIL
	6F1-09	HOUSING ASSY, PUMP
	6F1-10	BUSHING, PUMP BEVEL PINION
	6F1-11	GEAR, PUMP DRIVE
	6F1-12	GEAR, DRIVEN - PUMP
6L2	REGENERATOR AND DRIVE	
	6L2-01	REGENERATOR DISK & RING GEAR ASSY
	6L2-02	SEAL ASSY, OUTBOARD
	6L2-03	SEAL ASSY, INBOARD CENTER SUPPORT MTG
	6L2-04	HOUSING, REGENERATOR
	6L2-05	COVER, REGENERATOR
6M6	FUEL NOZZLE	
	6M6-01	MANIFOLD & CARRIER ASSY, FUEL
	6M6-02	CARRIER RING ASSY, FUEL MANIFOLD
	6M6-03	CARRIER RING, FUEL MANIFOLD
	6M6-04	BUSHING, COMB. ACTUATOR LEVER
	6M6-05	LEVER, ACTUATOR - COMB.
	6M6-06	WASHER
	6M6-07	LEVER, COMB. ACTUATOR
	6M6-08	NUT
	6M6-09	MANIFOLD ASSY, FUEL

TABLE XXXIII. IGT ENGINE PARTS LIST (cont)		
Group	Part Number	Description
	6M6-10	MANIFOLD BODY & TUBE ASSY, FUEL
	6M6-11	BODY, FUEL MANIFOLD
	6M6-12	TUBE, MANIFOLD FUEL
	6M6-13	SHIM, FUEL METERING
	6M6-14	SPACER, FUEL MANIFOLD
	6M6-15	NUT, RETAINING - FUEL MAN.
	MS9202-085	PACKING, PREFORMED "O" RING
	141106	PIN, DOWEL 3 mm (0.125 in.) x 8 mm (0.312 in.) LG
	6M6-1[illegible]	COVER ASSY, FUEL MANIFOLD REAR
	6M[illegible]-17	COVER, FUEL MANIFOLD REAR
	6M6-18	HOLDER, COMBUSTOR FLAME
	6M6-19	COVER, FUEL MANIFOLD FRT
	6M6-20	SEAL, FUEL LINE
	137396	NUT, INVERTED FLARED TUBE
	6M6-21	NIPPLE, 1/16-27 x 32 mm (1-1/4 in.)
	6A9-06	COVER ASSY, COMBUSTOR
	6M6-23	NOZZLE ASSY, COMBUSTOR PILOT
	6M[illegible]-24	BODY, FUEL NOZZLE
	6M[illegible]	PINTLE ASSY, FUEL NOZZLE
	[illegible]	PINTLE, FUEL NOZZLE SPRAY TIP
	[illegible]27	TIP, SPRAY, FUEL NOZZLE
	6[illegible]-28	NUT ASSY, FUEL NOZZLE SPRAY TIP
	6M6-29	NUT, FUEL NOZZLE SPRAY TIP
	6M6-22	SHROUD, FUEL NOZZLE
	6M6-30	GASKET, PILOT NOZZLE
	6M6-31	BOLT, 1/4-20 x 29 mm (1-1/8 in.) LG
6M12	DIFFUSER AND COMPRESSOR SCROLL	
	6M12-01	SCROLL, COMPRESSOR DIFFUSER
	6M12-02	RING ASSY, ACTUATOR - DIFFUSER
	6M12-03	BUSHING, DIFFUSER DRIVE RING
	6M12-04	RING, ACTUATOR - DIFFUSER
	6M12-05	PIN, DRIVE - ACTUATOR RING
	6M12-0[illegible]	PIN, DRIVE DIFFUSER VANE
	6M12-07	SHIM, ACTUATOR RING
	6M12-08	VANE, DIFFUSER VARIABLE
	6M12-09	GEAR, DIFFUSER ACTUATOR
	6M12-10	STOP, ACTUATOR DRIVE RING
6M14	IMPELLER AND GASIFIER ROTOR	
	6M14-01	ROTOR ASSY, TURBINE
	6M14-02	WHEEL, TURBINE
	6M14-03	SHAFT, TURBINE
	6M14-04	SEAL, TURBINE SHAFT
	NO PART NUMBER	BEARING, BALL
	6M14-05	IMPELLER ASSY, COMPRESSOR
	6M14-06	IMPELLER, COMPRESSOR
	6M14-07	SLEEVE, COMPRESSOR IMPELLER

TABLE XXXIII. IGT ENGINE PARTS LIST (cont)		
Group	Part Number	Description
	NO PART NUMBER	BEARING, ROLLER
	NO PART NUMBER	CLUTCH ASSY, TURBINE STARTER
6X6	POWER TURBINE REDUCTION GEARS	
	6X6-01	GEAR, PINION TURBINE SHAFT
	6X6-03	BEARING, BALL
	6X6-02	GEAR, POWER TURBINE CLUSTER
	6X6-07	BEARING, BALL
	6X6-04	GEAR, OUTPUT DRIVE
6X9	POWER TURBINE ACCESSORY DRIVE	
	6X9-05	GEAR, ACCESSORY IDLER
	6X9-12	BEARING, BALL - 20 x 42 x 12
	6X9-06	GEAR, ACCESSORY DRIVE CLUSTER
	6X9-12	BEARING, BALL - 20 x 42 x 12
	6X9-08	GEAR, ACCESSORY AND FUEL PUMP
	6X9-12	BEARING, BALL - 20 x 42 x 12
	6X9-09	GEAR, ACCESSORY FUEL PUMP DRIVE
	6X9-10	GEAR, STARTER FUEL PUMP DRIVE
	6X9-15	CLUTCH, FUEL PUMP
	NO PART NUMBER	RING, SNAP - NARROW
	NO PART NUMBER	SNAP, WIDE
	6X9-12	BEARING, BALL - 20 x 42 x 12
	6X9-11	GEAR, STARTER FUEL PUMP CLUSTER
	6X9-14	BEARING, BALL - 17 x 35 x 10
	6X9-13	GEAR, STARTER FUEL PUMP
ALTERNATE TO ELIMINATE HYDRAULIC MOTOR REGENERATOR DRIVE AND TO ELIMINATE GEAR-DRIVEN FUEL PUMP		
	6X9-05	WORM, R.H.
	6X9-17	WORM GEAR, R. H.
	6X9-16	ACCESSORY DRIVE GEAR
	6X9-18	REGEN PINION
	6X9-19	CROSS HELICAL DRIVE GEAR
	6X9-20	CROSS HELICAL DRIVEN GEAR

TABLE XXXIV. RISK ASSESSMENT CATEGORIES

1. In production or qualified for release to production.
2. Demonstrated on a development engine, transmission, or vehicle.
3. Demonstrated during a component test. A small extension of conditions may be required.
4. Based on exploratory tests of similar components. Material changes and more severe environment may be required.
5. Based on research experiments, analysis, or theoretical investigations.

TABLE XXXV. RISK ASSESSMENT--PERFORMANCE AND EMISSIONS				
Item	Program goals	Program impact	Risk category*	Comments
Compressor	η_{TS} = 76.3% 8% surge margin	1% η gives 0.25 km/l (0.6 mpg)	3.5	Variable geometry flow range and efficiency are important to fuel economy
Combustor	0.26 g/km (0.41 g/mi) HC, 0.25 (0.4) NO_x, 2.11 (3.4) CO	Determines emissions	2.5	Emissions essential to marketability
Gasifier turbine & interstage duct	η = 83.7% at 70% rpm, P/P = 1.1%	1% η gives 0.13 km/l (0.3 mpg) 1% ΔP/P gives 0.21 km/l (0.5 mpg)	3.5	Flow range at good efficiency is important to fuel economy
Power turbine & exhaust diffuser	η = 84.5% at 70% rpm P/P = 2.1%	1% η gives 0.13 km/l (0.3 mpg) 1% P/P gives 0.21 km/l (0.5 mpg)	3.5	Wide rpm and flow range at good efficiency & uniform flow into regenerator aid fuel economy
Regenerator	ϵ = 93.6% ΔP/P = 5.5% Leak ~ 4.34%	1% ϵ gives 0.25 km/l (0.6 mpg) 1% leak gives 0.34 km/l (0.8 mpg)	3	Regenerator performance has an effect on fuel economy exceeded only by turbine temperature
Gears, bearings, accessories	Loss ≈ 7.5 kW (10 hp)	0.75 kW (1 hp) gives 0.42 km/l (1.0 mpg)	2	Minimum loss, side light load sfc
Transmission	η = 90% at 7.5 kW (10 hp)	1% η gives 0.17 km/l (0.4 mpg)	2	High efficiency at light load aids fuel economy on driving cycle
Controls	1-sec accel within emission limits	Fuel excursions increase emissions	2.5	Accurate tracking of variable geometry and fuel flow aids emissions and performance

*Refer to Table XXXIV.

air-cooled metal turbine rotors or fixed-geometry compressors. Within the time and probable funding available, the probability of success is greater with the effort properly concentrated.

In recognition of the risk, a high portion of program resources will be devoted to the development of ceramics. Good progress with ceramic stationary components has been made in the CATE program, and initial spin testing of ceramic rotor blade coupons has started. Ceramic vendors have also made turbocharger rotors which have been spun cold to reasonably high centrifugal stress levels. A concerted effort on ceramic rotors with more than one supplier and with alternate materials can be expected to advance the state of the art rapidly. The two-shaft engine design selected also minimizes the stresses imposed. The major concern, however, is that the rate of progress may be too slow for a five-year demonstration program. If the ceramic rotor development is slower than planned, the result would be an engine in which rates of temperature change and/or maximum rpm would be restricted to bring thermal and centrifugal stresses within the material limits. A steady-state road-load fuel economy test at rated turbine temperature would still be possible in these circumstances, so that the future potential of the engine would be evident.

TABLE XXXVI. RISK ASSESSMENT--MECHANICAL, RELIABILITY, PRODUCIBILITY, AND SAFETY				
Item	Program goals	Program impact	Risk category*	Comments
Compressor	R_c = [illegible].5	Low risk; allows maximum effort on ceramics	2	Producible, low-risk mechanical design
Variable-geometry ceramic combustor	T_3 = 940°C (1725°F) T_4 = 1288°C (2350°F)	Temperature pattern affects engine reliability	3	Ceramic needed for durability
Ceramic gasifier turbine & interstage duct	V_T = 500 m/s (1640 ft/sec) T_4 = 1288°C (2350°F)	Engine durability	4	Turbine temperature is needed for program fuel economy goal
Metal power turbine rotor, ceramic vanes	V_T = 526 m/s (1725 ft/sec) T_5 = 1010°C (1850°F)	Low risk		
Regenerator	T_6 = 982°C (1800°F)	Limits temperature on driving cycle	3	High-temperature seals & disks are needed for good part-load sfc.
Gears, bearings, accessories		Low risk for reliable engine running	2	
Transmission		Low risk for vehicle program	1	Very low risk for reliability
Controls	Transient T_4 = 1315°C (2400°F)		1	Minimum temperature excursions for reduced thermal stress for ceramics

*Refer to Table XXXIV.

A significant portion of the program resources should also be directed toward component development to achieve efficiency, flow range, and producibility. Precedence is given in the program plan to the flow range and producibility requirements; thus, a shortfall, if it occurs, would be in component efficiency. This is preferable to flow range restriction wherein one component could cause less-than-optimum operation of other components as well. The flexibility of the variable geometry and the electronic control in component matching in the engine would limit the effect of a reduced component efficiency on fuel economy to the factor shown in Table XXXV.

The regenerator is a moderately-high-risk component from a performance and a mechanical point of view. The performance risk is in the achievement of low leakage at temperatures in excess of present experience and is dependent on the development of hot seals with minimum thermal stress and distortion. The mechanical risks are in the increased temperature capability of the disk and of the hot seal wear face materials.

The combustor is included in the moderately-high-risk category because it is an untested combination of features which were previously demonstrated separately. The combination of ceramic construction of a variable-geometry premix/prevaporization combustor in a wide-flow-range engine under full electronic control is a task of considerable magnitude which must be recognized in the assignment of manpower and funds in a development program. It involves the coordinated effort of several disciplines and should be monitored frequently.

The remaining parts of the powertrain system development and demonstration are considered to be low risk and should respond to normal development effort in the five-year time frame. The risk assessments should be reviewed periodically, however, to reevaluate the level of risk in each area and to reallocate resources if necessary to minimize the overall risk of attaining the program goals.

VEHICLE DESIGN

Vehicle Specifications

Compact sedan	Wheelbase 2.654 m (104.5 in.)
Vehicle accessories	Power steering, power brakes, heater/air conditioning
Vehicle drive train	Transverse-mounted engine, torque converter with lockup clutch, three-speed automatic transmission, 3.7 rear axle ratio, 185/80R13 tires
Vehicle weight, curb	1235 kg (2723 lb)
test	1370 kg (3021 lb)
Engine oil cooler	Tube and fin type, 250 x 300 mm (9.8 x 11.8 in.)
Transmission oil cooler	Tube and fin type, 250 x 300 mm (9.8 x 11.8 in.)
Roominess index	7018 mm (276.3 in.)
Luggage capacity	0.37 m^3 (13.1 ft^3)

Vehicle Installation

The demonstration vehicle is pictured in Figure 120. This is basically the 1980 General Motors front-wheel-drive compact "X" car in which the powertrain is mounted transversely. The stock piston engine has been replaced with the gas turbine engine mounted in the same environment and mounted to the same frame mounting points. The cooling system components have been eliminated because the gas turbine engine is air cooled. Minor changes were required in the steering linkage, front stabilizer, and engine cradle rear crossmember to accommodate the exhaust system. A considerable amount of change was required in the underbody as discussed under the "Exhaust System" heading. Other related components were revised as described in Table XXXVII.

Transmission

The transmission is a modification of the 1980 production transmission, located in the production position so that no changes in the axle shafts and suspension are required.

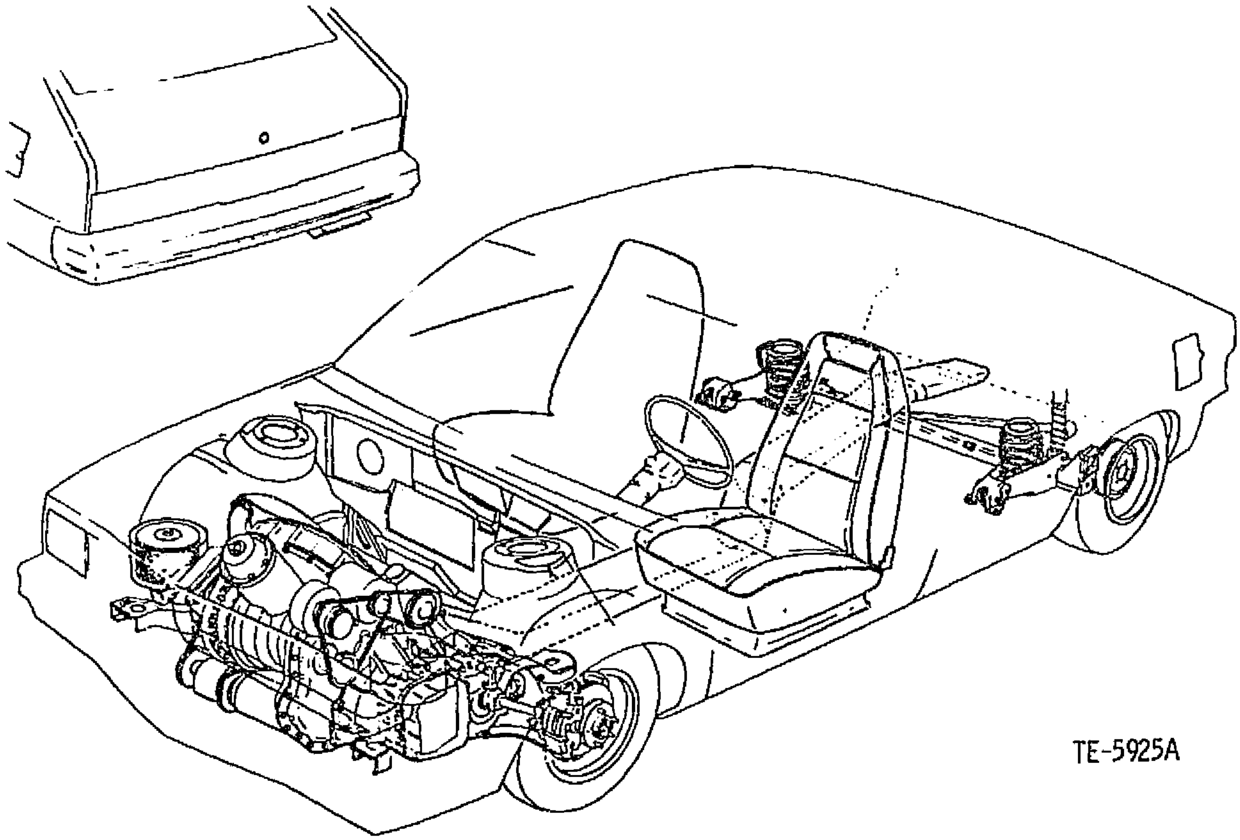

Figure 120. IGT engine vehicle installation.

TABLE XXXVII. IGT VEHICLE COMPONENT DEFINITION

UPC	Item	Status	Description
		Body	
1	Front Structure	Prod.	
	Seats	Prod.	
	Seat Belts	New	Front seat belt anchors relocated; new, longer belts.
	Floor Covering	New	Revised carpet and padding for new tunnel.
1	Windshield Opening	Prod.	
	Back Window Opening	Prod.	
	Center Pillar	Prod.	
	Roof Outer Panel	Prod.	
	Quarter Window Opening	Prod.	
	Rear Quarters	Prod.	
	Fuel Filler Door	Prod.	
	Wheelhouse Outer	Prod.	
	Rear End Panel	Prod.	
	Doors	Prod.	

TABLE XXXVII. IGT VEHICLE COMPONENT DEFINITION (cont)			
UPC	Item	Status	Description
	Body (cont)		
	Underbody	New	Revised toe pan, spare tire well and larger tunnel to accommodate turbine exhaust. New fuel tank hanger strap brackets & location. Brackets added to underbody for exhaust hangers.
1	Dash	New	Scalloped in center for exhaust clearance.
1	Deck Lid (Coupe)	Prod.	
1	Hatch Lid	Prod.	
1	Windshield Wipers	Prod.	
1A2	Windshield Washer	Prod.	
1A2	Cowl Vent Grille	Prod.	
1	Rear Side Market	Prod.	
1	Rear Side Marker Bezel	Prod.	
1	Combination Rear Lamp (base car)	Prod.	
1	Combination Rear Lamp (deluxe)	Prod.	
12C	Park and Signal	Prod.	
12C	Headlamp	Prod.	
12C	Front Side Market Lamp	Prod.	
12C	Front Side Market Bezel	Prod.	
	Front Sheet Metal		
11A	Front Fender Panels	Prod.	
	Fender Reinforcements	Prod.	
11D	Hood Outer	New	New hood design for engine air intake; inner and outer panels new.
	Hood Inner	New	
	Hood Latch Assy.	Prod.	
	Hood Hinge	Prod.	
	Hood Insulator Blanket	New	Revised for new hood.
	Hood Bumpers	Prod.	
11E	A/C Condenser Support & Baffle Assy.	Prod.	
	A/C Condenser Cross Brace	Prod.	
	Vertical Support	Prod.	
	Fender Skirt	Prod.	
11G	Name Plates & Emblems	New	Specific; stick on where possible. ABS chrome plated.
12B	Battery Tray	Prod.	
12C	Headlamps	Prod.	
	Headlamp Bezel	Prod.	
	Park & Signal Lamps	Prod.	
	Headlamp Mounting Panels	Prod.	

TABLE XXXVII. IGT VEHICLE COMPONENT DEFINITION (cont)			
UPC	Item	Status	Description
	Front Sheet Metal (cont)		
15A	Radiator Mounting	New	No radiator required. No hoses, thermostat, etc.
13C	Radiator Grille	Prod.	
14A1	Front Bumpber Facia	Prod.	
	Front Bumper Applique	Prod.	
	Front Bumper Air Slot Molding	Prod.	
	Front Bumper Reinforcements	Prod.	
	Front EA Units	Prod.	
	EA Unit Brackets	Prod.	
	Front Bumper Guard	Prod.	
	Front Rub Strip	Prod.	
	Front Bumper Closeouts	Prod.	
14A2	Rear Bumper Facia	Prod.	
	Rear Bumper Facia Applique	Prod.	
	Rear Bumper Reinforcements	Prod.	
	Rear EA Units	Prod.	
	Rear Bumper Guards	Prod.	
	Rear Rub Strip	Prod.	
	Rear Bumper Closeouts	Prod.	
14C	Front License Plate Mounting	Prod.	
	Rear License Plate Mounting	Prod.	
1A1	Body Requirements	Prod.	
1A2C	Body Ventilation	Prod.	
1A2G	A/C Compressor Parts	New	The radial 4 compressor used exclusively. All compressor mounting brackets are new.
1A2H	Heating & Air Conditioning Systems	Prod.	
	Heater	New	Heater core replaced by exhaust gas to air heat exchanger. Incorporates exchanger gas sensor & shutoff controls.
	Air Conditioning	Prod.	
	Heater Outlets	Prod.	
	Defroster Nozzles	Prod.	
	A/C Distribution	Prod.	
1A2J	HVAC Controls	Prod.	
1A2K	A/c Refrigerant System Parts	Prod.	
	Electrical and Instrument Panel		
1A2B	Instrument Panel	Prod.	
6Y1	Alternator	Prod.	
6Y4	Engine Electrical	New	New engine harnesses. New oil pressure and temperaure warning senders and gages

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TABLE XXXVII. IGT VEHICLE COMPONENT DEFINITION (cont)			
UPC	Item	Status	Description
	Electrical and Instrument Panel (cont)		
9A	Steering Columns	New	Same as 1980X except longer intermediate shaft: 25 mm (1.0 in.)
9B	Steering Wheels	Prod.	
12	Front End Electrical	Prod.	
12A	Battery	Prod.	
	Battery Cables	Prod.	
12H	Instrument Panel Wiring	Prod.	
	HVAC Electrical	Prod.	
	Chassis		
1B1	Cradle Mounts	New	New specific family of body mounts.
2B	Engine Cradle (Side & Crossmember)	New	Same as 1980 except new rear crossmember and mounts.
	Restraint Cable	Prod.	
3	Knuckle, Brake Assy	Prod.	
3A	Telescopic Strut & Shock Absorber Assy	Prod.	
	Lower Control Arm	Prod.	
	Steering Knuckle	Prod.	
	Lower Ball Joint	Prod.	
3B	Wheel Bearings	Prod.	
3C	Springs, Front Coil	Prod.	
3G	Front Stabilizer	New	Redesigned shape to provide exchanger system clearance.
4	Rear Suspension	New	Similar to 1980 except track bar lengthened and relocated new brackets.
4A5	Wheel Bearings	Prod.	
4B	Springs, Coil	Prod.	
	Coil Spring Insulator	Prod.	
4D	Trailing Arm and Beam Assy	New	Incorporates new track bar bracket.
	Track Bar	Prod.	
4E	Shock Absorber	Prod.	
4F	Stabilizer Bar	Prod.	
5A	Front Hub & Disk Assy	Prod.	
	Caliper Assy	Prod.	
5B	Rear Drum Brake Assy	Prod.	
	Rear Brake Wheel Cylinder	Prod.	
5C	Brake Pedal 6.5 ratio std sedan, 3.5:1 ratio power	Prod.	
	Pedal Bracket	Prod.	
5D	Parking Brake System	New	New cable length and attachment to underbody

TABLE XXXVII. IGT VEHICLE COMPONENT DEFINITION (cont)			
UPC	Item	Status	Description
	Chassis (cont)		
5E	Master Cylinder	New	New master cylinder for Hydroboost; like Olds diesel
	22.2 mm, Sedan PWR; 19 mm manual		
	Proportioning	New	Integral to master cylinder
5F	Brake Lines	New	Lines longer and rerouted across front of dash. 2 lines added from P/S pump.
5G	Power Booster & Hoses	New	Hydraulic supplied by P/S pump
6Q	Engine Mounting	New	3 engine-trans. mounts with torque reaction strut and lateral strut. Same location as 80X, but new struts and engine brackets.
9A			
	Steering Gear, Manual	Prod.	
	Steering Gear, Power	Prod.	
	Steering Gear Brackets	New	Mounting bracket to dash, new; gear lowered 25 mm (1.0 in.) from 80X.
	Steering Wheel	Prod.	
9C	Column Support	Prod.	
9D	Steering Linkage	New	80X except steering connecting rod length revised.
9E	Power Steering System	Prod.	
	Power Steering Pump	New	Modified to accommodate hydraulic brake booster for power brakes. Same as Olds diesel.
9E	Hoses	Prod.	
	Pipes	New	Revised length and rerouted across front of dash.
10B	Wheels & Wheel Trim	Prod.	
	Tires	Prod.	
	Spare Tire	Prod.	
	Hub Cap	Prod.	
	Wheel Cover (P02)	Prod.	
	Transmission-Axle		
3A4	Front Drive Axle Shafts	Prod.	
6C3	Flywheels & Ring Gear	Prod.	
	A/T Flexplate	New	Specific flexplate required with no ring gear.
6N2	Throttle Controls	Prod.	
	Engine Compartment	New	New throttle control systems.
	Passenger Compartment	Prod.	

TABLE XXXVII. IGT VEHICLE COMPONENT DEFINITION (cont)			
UPC	Item	Status	Description
	Transmission-Axle (cont)		
7	Automatic Transmission (three-speed M34)	New	1980X transverse mounted, LH side of vehicle. Single axis type (THM 125) with variable displacement pump. Recalibrated for turbine engine.
	Torque Converter dia = 254 mm (10.0 in.) Stall Torque Ratio = 2.00 Stall Speed at 135.6 N m (100 lb-ft) = 2100 rpm		
7A	Automatic Transmission Column Shift System	Prod.	
7A	At Modulation Controls M34 TV Control Cables	New	Automatic transmission TV cables for M34 are new length. New retainer bracket at engine.
7B3A	A/T Hardware Converter Housing Shield	New	To accommodate turbine engine.
	Transmission Oil Fill Tube	Prod.	
	Transmission Oil Level Dipstick	Prod.	
7B6	Automatic Transmission Cooling	New	Air-to-oil heat exchanger
	Transmission Oil Cooler Pipes (Upper and Lower)	New	Revised length.
	Transmission Oil Cooler Hoses	Prod.	
12J	Speedometer Gears		
	Speedo Drive Gear	Prod.	
	Speedo Driven Gears	Prod.	
	Speedo Sleeve Assy	prod.	
	Engine		
1A2G	A/C Compressor Mtg & Drive	New	New mounting brackets.
6	IGT Engine & Controls	New	
6C2	Accessory Drive Pulley	New	New pulley attached to accessory output shaft of turbine engine.
6G3	Engine Oil Cooler	New	Air-to-oil heat exchanger
6K1	Engine Fan & Drive	New	1980X electric drive fan with new mounting bracket.
6Y1	Alternator Mtg & Belt	New	New brackets.
9E	P/S Pump Mtg, Pulley and Belt	New	New brackets.

TABLE XXXVII. IGT VEHICLE COMPONENT DEFINITION (cont)			
UPC	Item	Status	Description
Induction-Emission-Exhaust-Fuel System			
6M3	Air Cleaners & Air Induction Air Cleaner	New	Dual air cleaners; air intake through hood, outlets ducted to engine air inlet. Polyfoam elements
6M4	Fuel Pump	New	Electric pump mounted on engine. Variable speed, 45.4 kg/h (100 lb/hr) at 1.38 MPa (200 psi), 41 W (0.5 hp) input.
8A	Fuel Tank, Attach	New	Reduced in size to 45.4 l (12 gal). Revised filler pipes. New hanger brackets and straps.
8B	Fuel Lines	New	Relocated to clear exhaust and to accommodate new tank.
8C & 8E	Exhaust System	New	0.02 m^2, 2mm (30 $in.^2$, 0.080 in.) thick aluminized steel duct from engine to right rear, exiting under bumper. No muffler or catalytic converter. Center section insulated for noise attenuation.

Engine Air Induction System

The air intake to the engine will be through a set of dual Polyfoam air cleaners located just behind the A/C condenser support. Figures 121 through 123 show that the air comes into a pair of ducts formed into the hood, entering the ducts from just behind the grille and before the A/C condenser. The air turns 90° before entering the hood vertically, providing good separation of water and foreign material.

The air then enters the air cleaners from the top into a cylindrical chamber, passes through the elements, and flows into a plenum mounted to the engine air inlet. The ducting from the left-hand air cleaner is amply sized to provide low-restriction balanced flow with the short duct right-hand air cleaner. The ducts are lined with acoustical insulation for attenuation of compressor noise.

Exhaust System

The exhaust leaves the engine through an aluminized steel exhaust duct which is illustrated in Figure 120. This duct requires an enlargement of the existing exhaust tunnel in the production vehicle. The duct passes over the axle at the right rear, with ample clearance of the 12-gallon fuel tank and

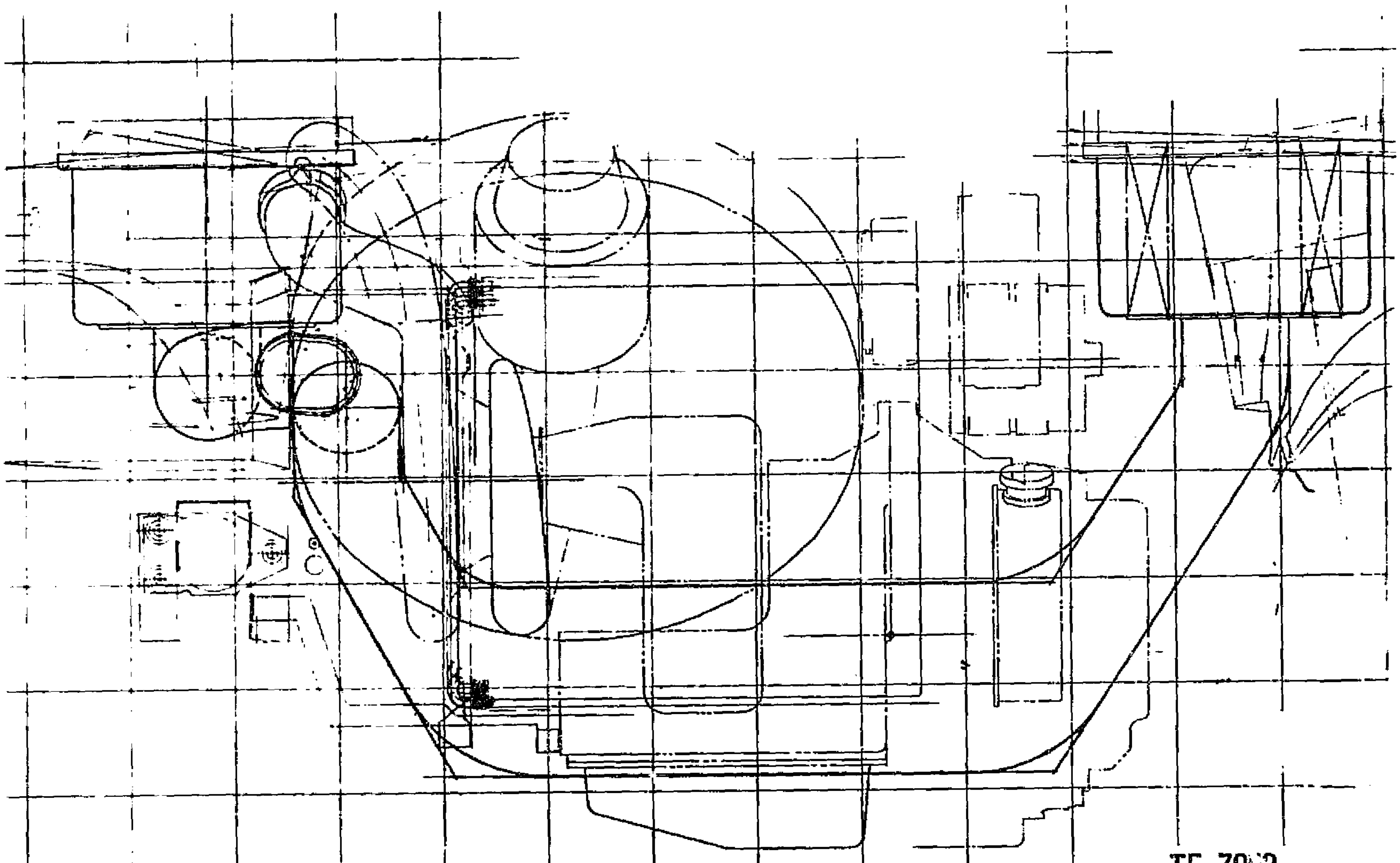

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Figure 121. IGT vehicle air induction system — front view.

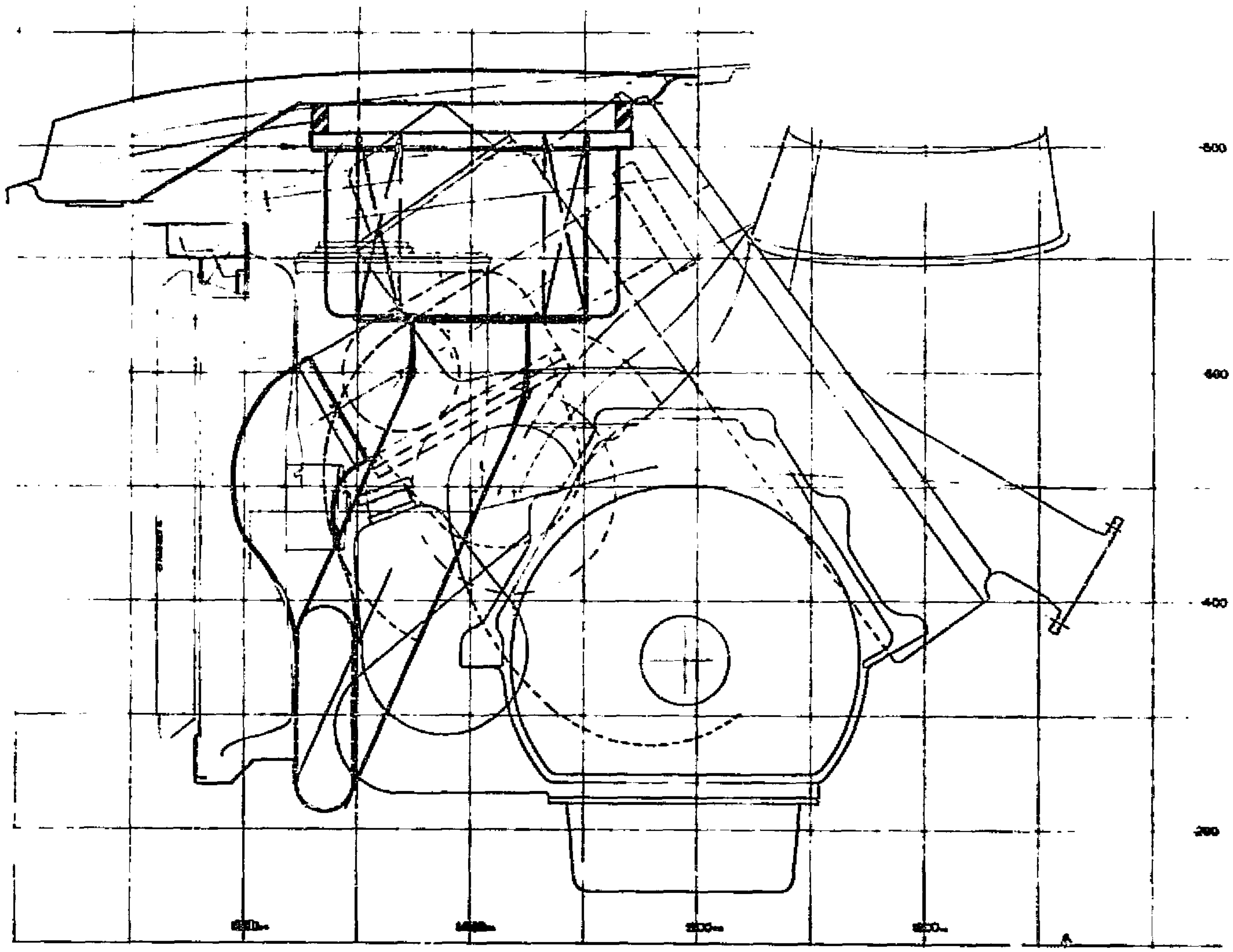

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Figure 122. IGT vehicle air induction system — left side view.

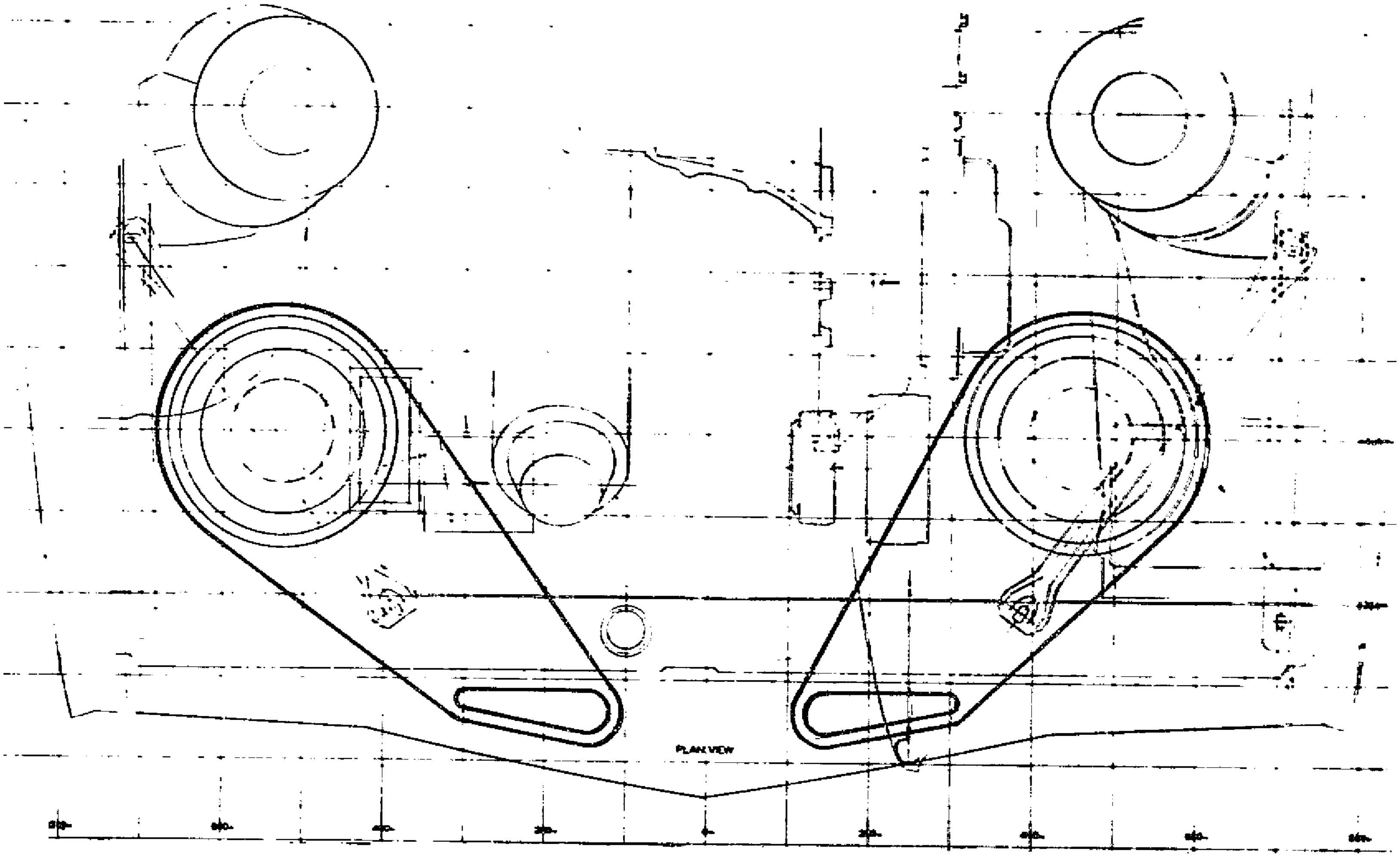

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Figure 123. IGT vehicle air induction system — plan view.

the spare tire well, and exits under the bumper as shown in the left upper corner of the illustration. The ducts will be manufactured in three sections for simplified production and assembly and to minimize the cost of service replacements. The intermediate section will be insulated for noise attenuation.

The pressure loss of the duct was analyzed at maximum power conditions. In a 0.02 m^2 (30 $in.^2$) duct area, which is consistent with the vehicle constraints, the pressure loss was estimated to be 1.5%. Duct bends and inlet/outlet conditions were included in the estimate. This loss is acceptable from an engine performance standpoint.

Vehicle Heating System

The vehicle heating system, as shown in Figure 124, is hot gas to air, eliminating the need for a cooling system with its attendant water pump and reservoir. The standard heater core is replaced with a stainless steel hot gas-to-air heat exchanger; the in-car system functions conventionally in other respects. The heat exchanger is heated by shunting a maximum of 10% of the exhaust gas from the hot side of the regenerator through the heat exchanger and back to the cold side of the regenerator. A regulator valve in the exhaust gas duct, with a pressure sensor located in the double-walled heat schanger, shuts off the exhaust flow in the event of a leak into the air space between the double-walled tubing. This feature eliminates any possibility of exhaust gases entering the vehicle through the heating system if the tubing should become perforated.

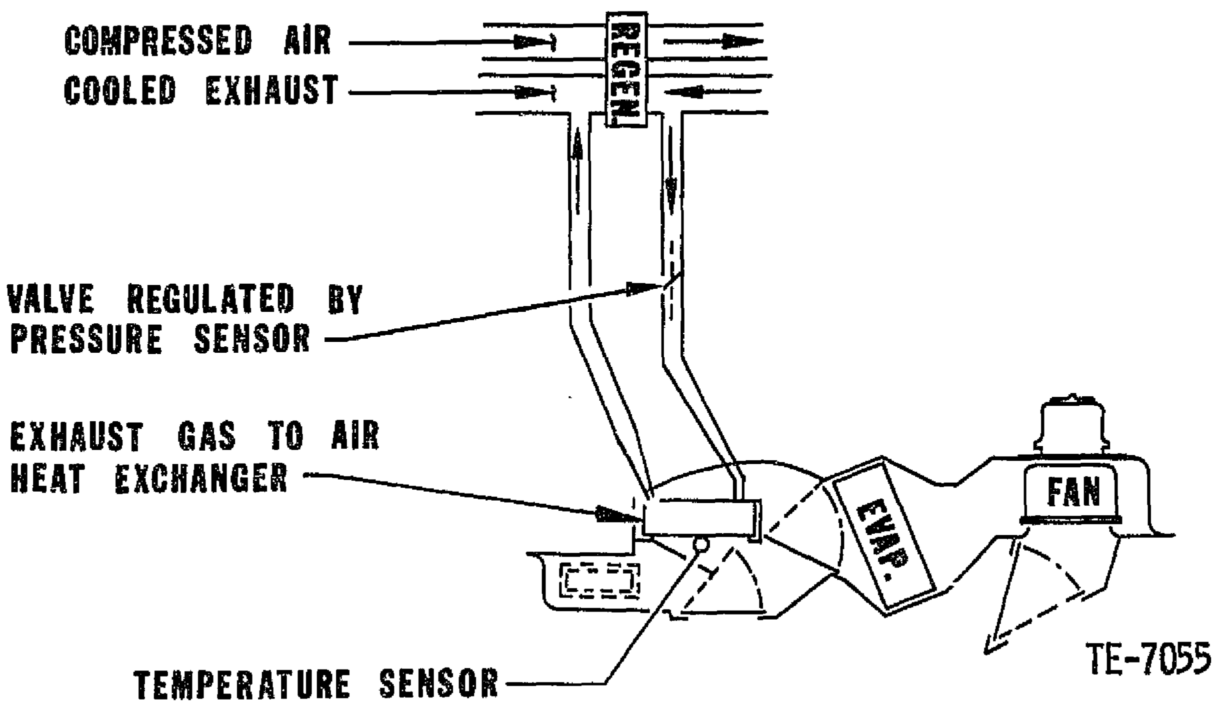

Figure 124. IGT vehicle heating and air conditioning system.

This proposal will be evaluated further during the vehicle design program to optimize the location of the heat exchanger (it may be more expeditious to mount the heat exchanger at the engine) and to reevaluate the desirability of a liquid-to-air heating system.

Vehicle Brake System

The vehicle braking system schematic is shown in Figure 125. This is a hydraulic pressure boost system already in production for cars equipped with a diesel engine and therefore requires no new technology.

Engine-Transmission Oil Coolers

A dual oil cooler assembly will be mounted behind the A/C condenser as shown in Figure 126. The engine oil and the transmission oil will each be cooled by a 250 mm x 300 mm (9.8 x 11.8 in.) tube-and-fin-type oil cooler.

SUMMARY

Based on a comparative analysis of powertrains incorporating differential, single-shaft, and two-shaft engines, the two-shaft configuration was selected. It is mated with a conventional automatic transmission. This selection was followed by additional optimization to maximize the fuel economy

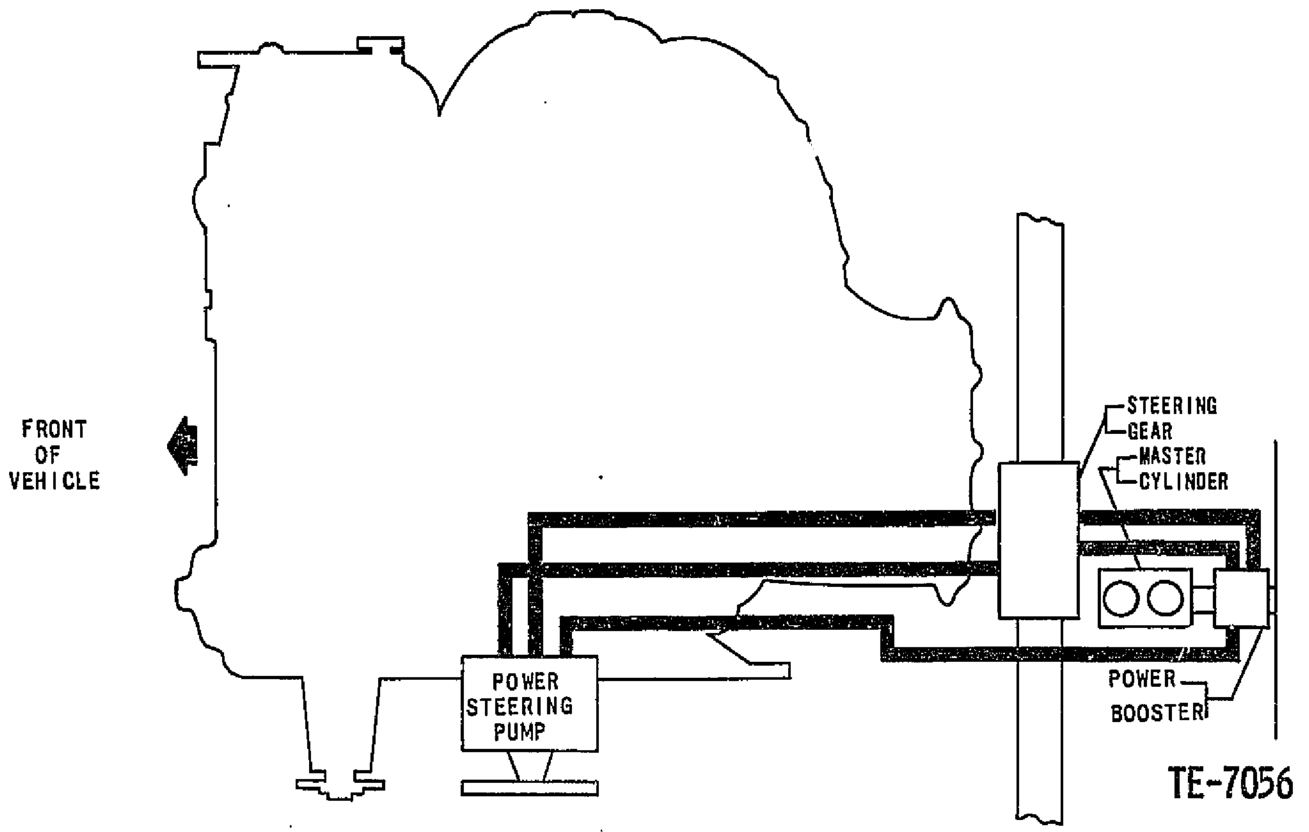

Figure 125. IGT vehicle hydraulic boost power brake system.

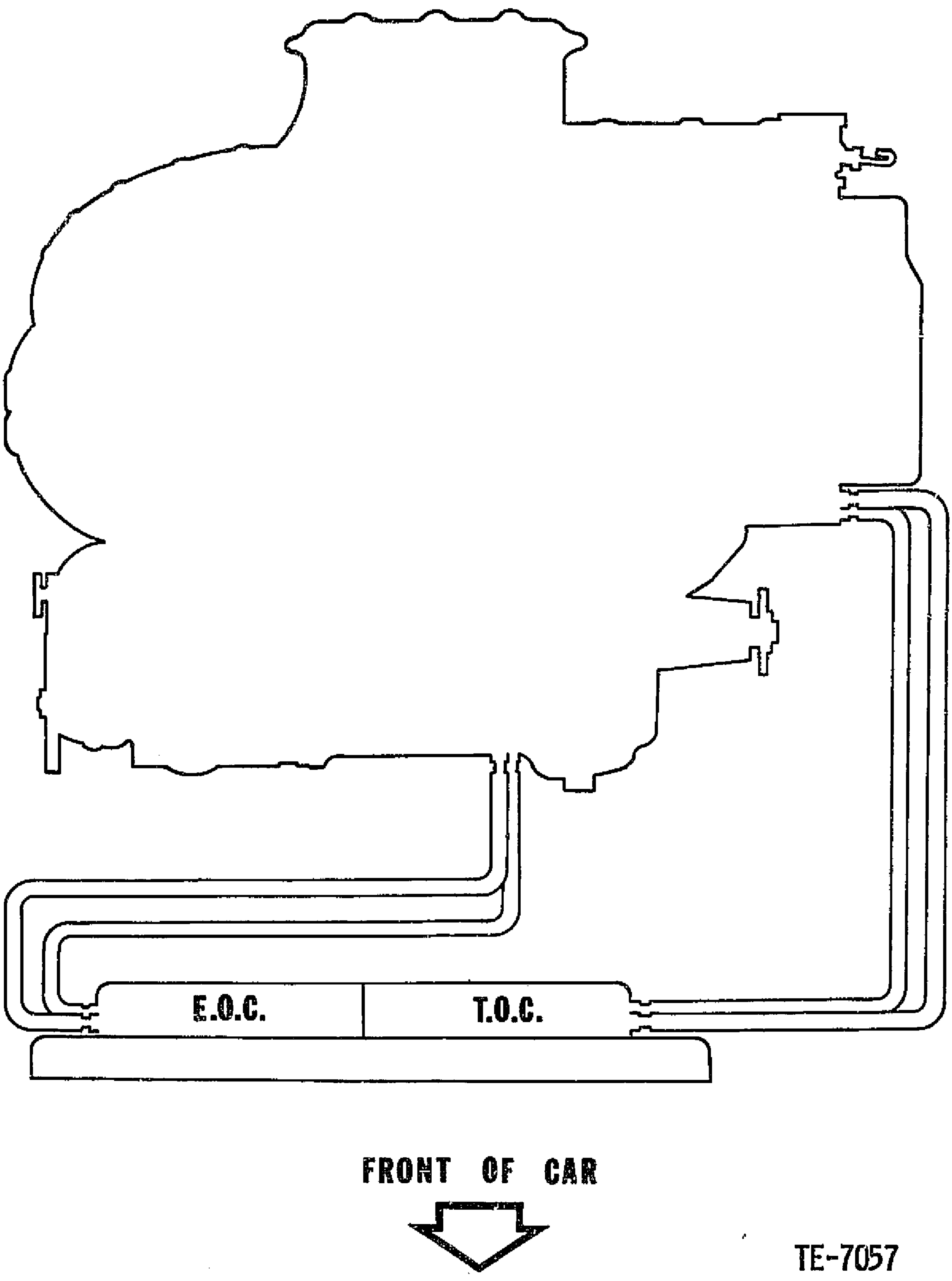

Figure 126. IGT vehicle engine and transmission oil coolers.

(thermal efficiency) improvement over a 1976 base-line Otto-powered vehicle. The incorporation of a ceramic gasifier turbine rotor allowed a higher turbine temperature, which contributed toward that goal. An optimized rotating regenerator design was also defined, as was improved low-power scheduling of the variable-geometry aerodynamic components. Both of these factors added further fuel economy gains. The optimization of transmission shift patterns and design improvements offered substantial improvement. In total, the potential fuel economy improvement is estimated to be 30.5% (29°C (85°F), 152 m (500 ft) ambient conditions, gasoline equivalent fuel).

The application of proved low-emission combustion technology will allow the advanced turbine powertrain to meet emission goals.

VII. VEHICLE COST ANALYSIS (TASK IIIB)

METHODOLOGY

The vehicle used to arrive at a "sticker price" comparison was the one studied under Task IV (Development Plan). This vehicle was developed around the new 1980 General Motors front-wheel-drive compact "X" car. It provided an excellent opportunity to cost a vehicle for which production cost data were available and which had the size, weight, and drivetrain configuration that are expected to be universal in the 1985-1990 time frame.

The vehicle cost analysis was relatively straightforward because no new methodology was involved. The engine cost analysis, however, involved considerable machining technology that is unconventional to normal high-volume production. The "new" component drawings were supplied to the Pontiac Industrial Engineering Department by Detroit Diesel Allison and were cost estimated along with the vehicle components by the procedure shown in Figure 127.

The cost analysis procedure involved the following steps:

1. Review each of the component and assembly drawings and determine which would be made in house and which would be purchased.

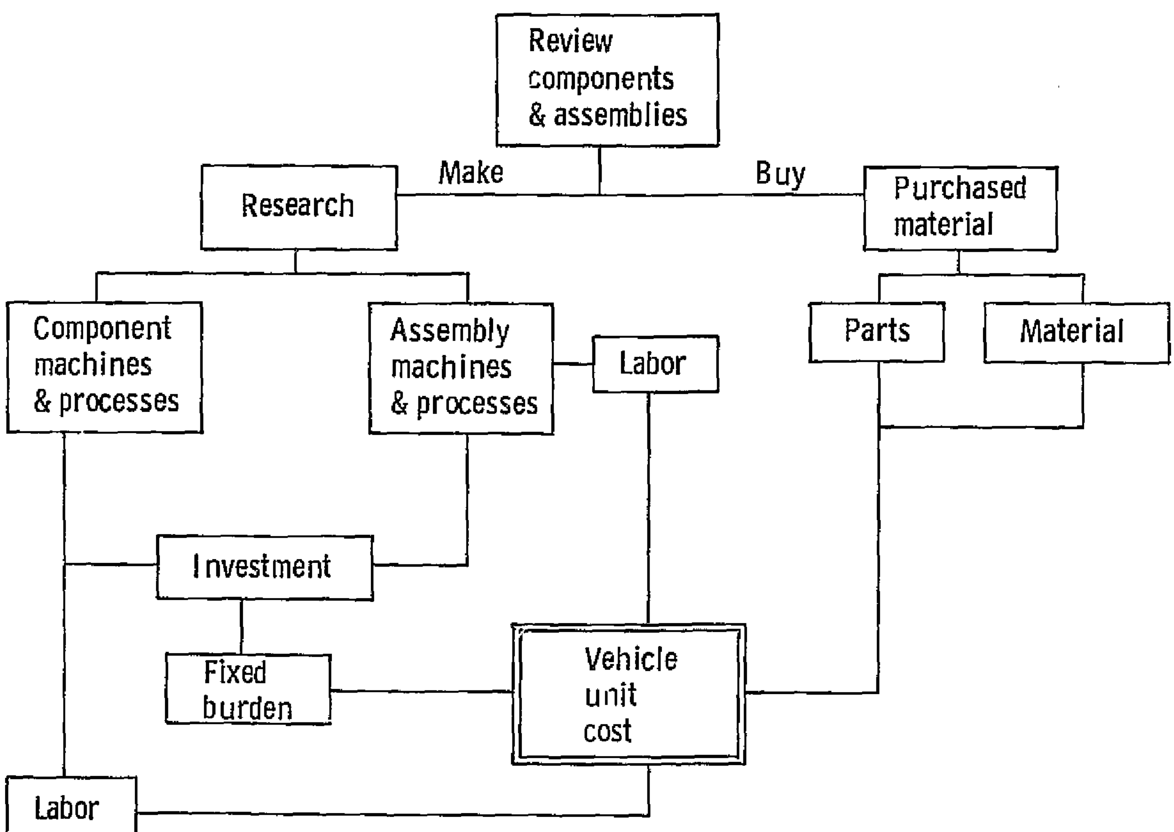

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Figure 127. Cost estimating procedure.

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2. Estimate the costs of purchased parts and material either through contact with suppliers or based on the cost of comparable current material.
3. Research "Make" components and assemblies for types of processes and machines required, using handbooks, experience, and consultations with machine and materials manufacturers.
4. Estimate and/or procure from suppliers the costs of machines, machine rates, and labor requirements.
5. Develop the total cost of investment and the resultant burden effect on unit cost for a single production module size of 400,000 vehicles and a minimum cost volume of 800,000 vehicles.

This procedure is normally applied to 100% of the engine and vehicle components and results in a complete cost analysis. The cost analysis for the engine under consideration varied from normal in that all parts were not available for estimating because a detailed design of the engine had not been made. The estimate of the total engine cost was developed by an extrapolation method based on a comparison of the number and cost of the parts analyzed relative to the cost impact of these same parts on the known Model 404 industrial gas turbine.

These part costs were progressively summed, beginning with the highest cost part, adding the next highest, etc. The similarity of the 404 engine, in number and type of parts (i.e., a two-shaft, regenerative turbine), to the candidate engine permitted these data to be used in a percent cost versus percent part number curve as shown in Figure 128. This curve shows that the first 20% of high-cost parts represents 80% of the engine cost. The reason for this is illustrated by the unit cost versus number of part numbers ranked by cost, as shown in Figure 129.

Thus, the number of drawings costed, including all high .ost drivers, permits the extrapolation of the total cost of the engine. As illustrated in Figure 129, the high cost driver of the control system was separated out and estimated to be 13% of the final cost, based on the industrial gas turbine experience. The costing effort for the engine, therefore, represented 72% of the final cost based on the 23% portion of drawings actually costed.

POWERTRAIN COST

The powertrain cost includes the engine, the engine accessories, and the transmission. The drive train beyond the transmission is a standard production unit and therefore does not affect the cost comparison.

The engine involved the greatest effort because it is all new. Many of the components and the machines required to process these components are "unconventional," resulting in cost estimates with considerable potential margin of error. The uncertainties and the error band involved in the costing are covered in more detail later in this section.

The engine accessories included were those required for the 33% option car, but these did not affect the comparative cost because no significant changes over production were required. A 33% option car is a car that includes all options expected to be sold on 33% or more of that car line as specified in

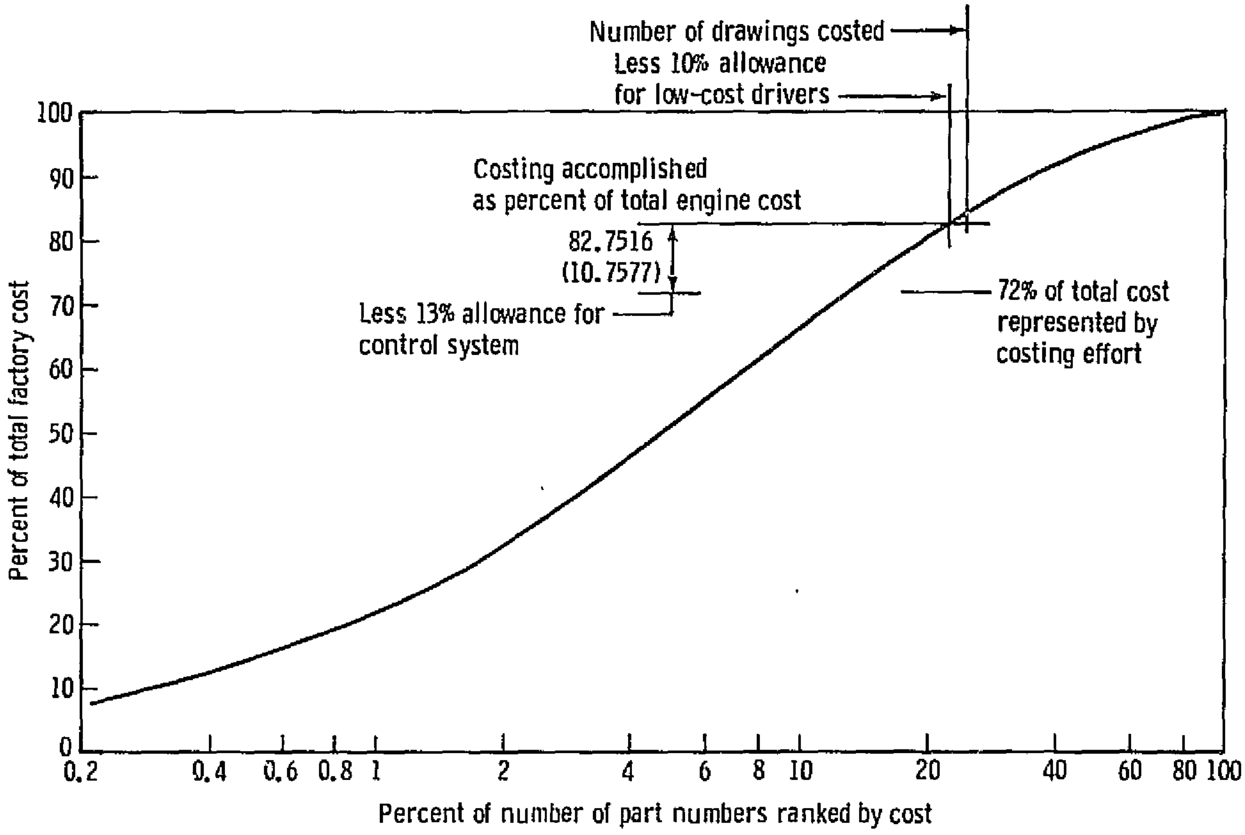

Figure 128. Cumulative engine cost versus number of parts.

the Federal Register, Part 86, Section 86.080-24, paragraph g. Specifically, these accessories would be the alternator, the air conditioning compressor, the power steering pump, and their respective mounting brackets.

The transmission also "washes out" of the comparison because the existing three-speed automatic transmission is used. The changes in gear ratios, torque converter, and calibration required to accommodate the turbine engine should not affect the unit cost.

The powertrain cost comparison therefore evolves into the basic gas turbine engine versus the six-cylinder engine of a comparable power range that is to be used in the 1980 compact Phoenix. A detailed cost estimate was made of the engine at volume levels of 400,000 (one full production module) and 800,000 (representative of minimum cost volume).

The net result of the powertrain cost analysis relative to that of the total vehicle was that approximately 99% of the price differential was assessed against the engine. The comparative data shown under the following Vehicle Cost heading, therefore, is approximately as representative of the engine price differential as it is of the vehicle. The cost impact of the two production volumes considered was not significant enough to affect the sale price of the vehicle (or powertrain).

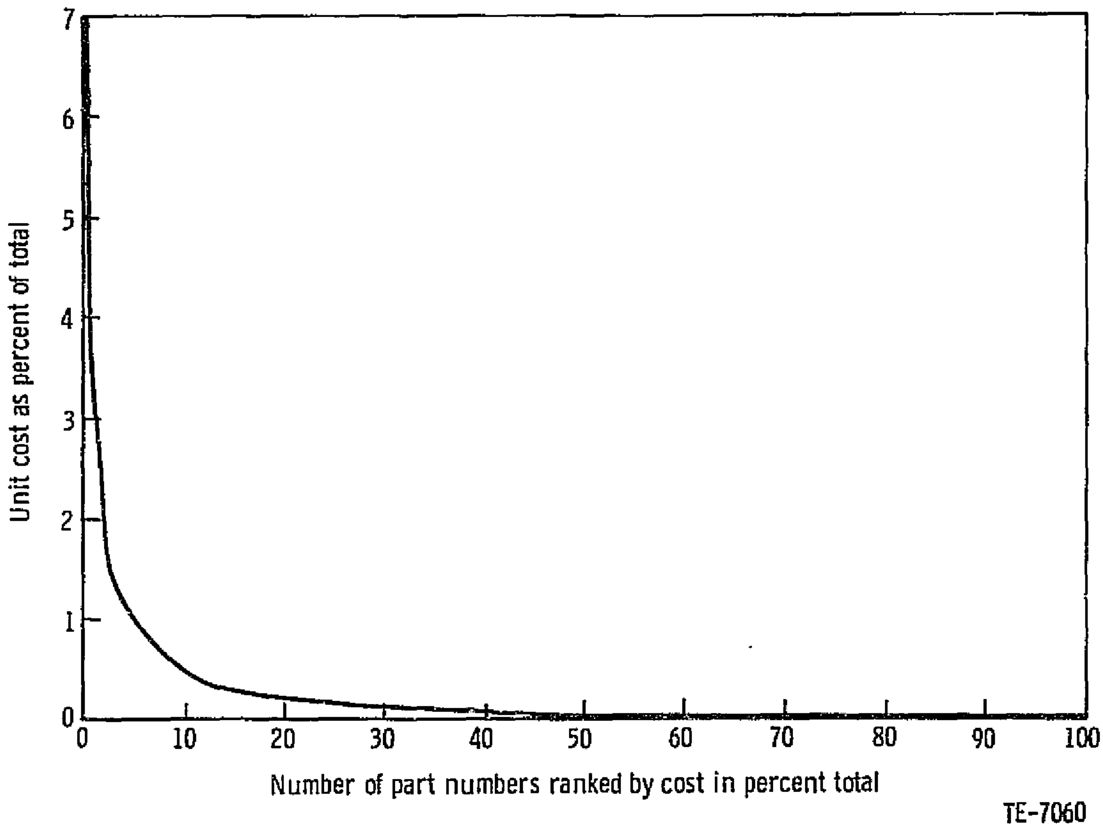

Figure 129. Distribution of cost drivers.

VEHICLE COST

The 1980 Pontiac Phoenix compact sedan with a six-cylinder engine was used for the cost analysis base line. The gas turbine powertrain has been designed into this car as described previously, providing a cost base for comparisons. All cost estimates were made at the same economic level as the 1980 vehicle and then adjusted to the 1977 economic level as specified by the contract. The analysis was made for the 33% car.

Detailed cost estimates of the complete vehicle, as modified to incorporate the gas turbine powertrain (along with the powertrain cost), were used to arrive at a sticker price comparison. The vehicle components requiring redesign to accommodate the gas turbine engine are listed beginning on page 208. The estimates were made for a volume of 400,000, which represents the minimum-volume module for full production of the engine, and for a volume of 800,000 vehicles, which represents the volume at which minimum cost can be realized. This was done primarily to ensure thoroughness in establishing the pricing because the cost is only one of many factors considered in establishing price. Final pricing is independent of the volume and, therefore, only one figure is shown. The following tabulation indicates the selling price difference and the "error band" price range resulting from the potential errors in the estimates as covered later:

Gas Turbine Vehicle Selling Price Differential

Nominal	Range
+$1000	+$780 to +$1130

The high initial cost of the gas turbine vehicle at this time is not surprising; it results from the unknowns in materials, tools, and processes required for high volume manufacturing of the gas turbine engine. This phenomenon is also accompanied by a proportionately high potential for cost reduction as the engine components, tools, and processes develop into "maturity." Figure 130 illustrates an estimate of the potential cost reduction (and timetable) as the engine evolves through the Experimental and Production Engineering phases into the "mature" engine.

The cost analysis ignores the effect that the turbine engine production might have on the spark-ignition engine volume (cost). This cause-effect is assumed to be only transitory as the replaced engine phases out of production.

The 1980 compact vehicle was fairly adaptable to the gas turbine powertrain, primarily because the front-wheel-drive configuration allows for routing of the exhaust system through the tunnel area of the floor pan. The following list describes the vehicle differences that are specific to the gas turbine engine:

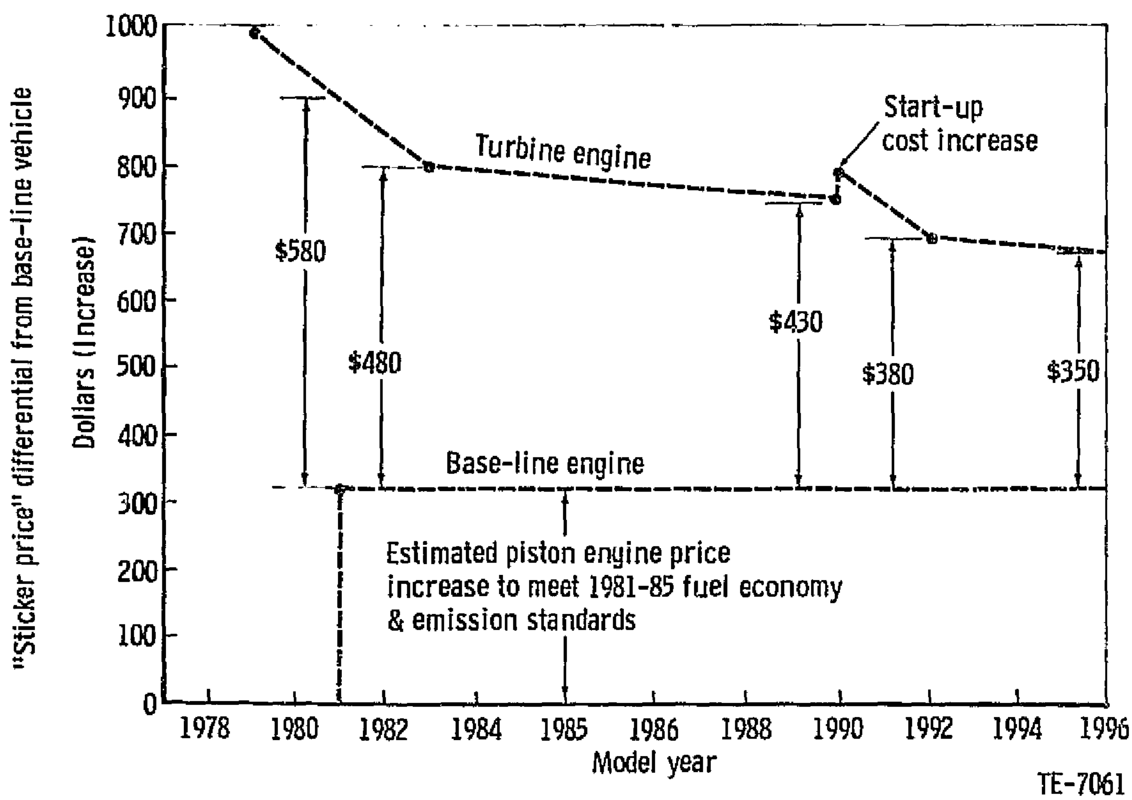

Figure 130. IGT vehicle "sticker price" comparison (all comparisons adjusted to 1977 economic level).

Underbody	Exhaust tunnel enlarged and toe pan scalloped to provide exhaust clearance. Spare tire well moved and reduced in size and fuel tank reduced in size to 12 gallons to provide exhaust clearance. New and relocated fuel tank and exhaust hangers and brackets.
Seat Belts	New, longer belts to accommodate relocation of anchors.
Floor Covering	New carpet and padding to accommodate underbody change.
Dash	Scalloped in center for exhaust clearance.
Hood	New hood inner and outer panel designs to incorporate engine air intake duct.
Hood Insulator	Revised for hood redesign.
Name Plates	Specific "Turbine" emblems.
Radiator & Mounting	No radiator, hoses, or thermostat.
Heater	Heater water core replaced by exhaust gas-to-air heat exchanger. Controls added for exhaust shutoff.
Engine Electrical	New engine wire harness; new oil pressure and temperature senders and gages.
Steering Column	Intermediate shaft, 25 mm (1 in.) longer.
Cradle Mounts	New mounts, tuning to accommodate new power-train.
Engine Cradle	New rear crossmember to accommodate exhaust.
Front Stabilizer	New stabilizer to accommodate exhaust.
Rear Suspension Trailing Arm & Beam Assy	Track bar bracket relocated for exhaust clearance.
Parking Brake Cable	New cable length and attachment location to underbody.
Parking Brake System	New cable length and attachment to underbody
Master Cylinder	New Master cylinder for Hydroboost
Proportioning	Integral to master cylinder.
Brake Lines	Lines longer and rerouted across front of dash. 2 lines added from P/S pump.
Power Booster & Hoses	Hydraulic supplied by P/S pump.
Engine Mounting	New struts and engine brackets to turbine engine.
Steering Gear Brackets	Mounting bracket to dash new. Gear lowered 25 mm (1 in.) from 80X.
Steering Linkage	Steering connecting rod length revised.
P/S Pump	Modified to accommodate hydraulic brake booster for power brakes.
Pipes	Revised length and rerouted across front of dash.
A/T Flexplate	Specific flexplate required with no ring gear.
Engine Compartment	New throttle control systems.

Automatic Transmission (3-speed M34)	New gear ratios and torque converter. Recalibrated for turbine engine.
M34 TV Control Cables	AT TV cables for M34 are new length. New retainer bracket at engine.
Converter Hsg Shield	To accommodate turbine engine
AT Cooling	New air-to-oil heat exchanger.
Transmission Oil Cooler Pipes (Upper & Lower)	New length.
A/C Compressor Mtg & Drive IGT Engine & Controls	New mounting brackets.
Accessory Drive Pulley	New pulley attached to accessory output shaft of turbine engine.
Engine Oil Cooler	New air-to-oil heat exchanger.
Engine Fan & Drive	New mounting bracket for electric fan.
Alt Mtg & Belt	New brackets.
P/S Pump Mtg, Pulley & Belt	New brackets.
Air Cleaner	Dual air cleaners, air intake through hood, outlets ducted to engine air inlet. Polyfoam elements.
Fuel Pump	New electric fuel pump mounted on engine.
Fuel Tank, Attach	Reduced in size to 12 gal. Revised filler pipes. New hanger brackets and straps.
Fuel Lines	Relocated to clear exhaust and to accommodate new tank.
Exhaust System	0.02 m^2, 2 mm (30 $in.^2$, 0.080 in.) thick aluminized steel duct from engine to right rear, exiting under bumper. No muffler or catalytic converter. Center section insulated for noise attenuation.

ERROR BAND ANALYSIS

The estimation of many of the engine components necessarily involved significant potential for error. Part detail drawings consisted primarily of very preliminary drawings based on similar parts from other DDA vehicular engines. License was taken to make revisions to improve machining and assembly without detailing the necessary interactions to determine the acceptability of these changes relative to engine performance. Assumptions had to be made concerning the types of machines (principally Electrochemical type) and machining rates needed to machine the special alloys as well as to the cost of the special alloys and materials required because such machines and materials are not used in high-volume manufacturing today. The availability of the exotic materials, in the quantities required, was not considered because of the anticipated change to ceramic components, nor was the cost effect of high demand on these alloys.

The materials and process engineers responsible for developing the cost data assigned error estimates to each component of the cost—i.e., equipment, tooling and its installation, purchased parts, and variable burden including labor. These factors were then ratioed relative to their specific portion of the total cost. This resulted in an estimated error band of +13% to -22% to be applied to the nominal cost. This error band was then assumed to have a linear relationship to the price and was therefore used to establish the price range specified in the price comparisons of the powertrain and the vehicle.

Although it was expedient and necessary to cost the engine with metallic aerodynamic components, it was assumed that the use of ceramic components, as anticipated for a production engine, should result in equivalent or lower cost.

LIFE CYCLE COST

Method

The life cycle cost (LCC) study was conducted in 1977 dollars but using a 1980 "X" body, 33% car. Several LCC categories were selected for analysis. Some relative nonsensitive categories were identified using a DOT report (ref 5). Other categories that were sensitive to the kind of power (turbine or Otto cycle) were separately analyzed in detail.

A computer simulation model was used to predict the maintenance and repair actions peculiar to the two powerplants for a reference life of 160 900 km (100,000 miles). Although originally designed for aircraft turbines, the simulation model logic was arranged to describe vehicle characteristics like those necessary to calculate automobile LCC. The current operating and support program, OS590, was used as a principal tool in the USAF-sponsored Reduced Cost Turbine Engine Concepts Program (Contract No. F33657-77-C-0425). A complete description of the program was delivered as a data item A0005 (ref 6). However, one particular logic flow chart (Figure 131) is the key to understanding the repair and maintenance portion of this study.

One point to keep in mind is that a "module" can be selected to be anything of interest. This study considered four items defined to be modules:

Engine, minor repair
Engine, major repair
Exhaust system
Water cooling system

In addition, unique elements of scheduled maintenance were defined in the computer input data. These unique elements were referenced to the 1978 Pontiac Maintenance Schedule (ref 7) expanded through the car life definition of 160 900 km (100,000 miles).

Consistent sets of assumptions were made for other LCC summary elements. For example, it was assumed that tire wear would not vary between kinds of engines; garaging, parking, and tolls were independent from engine type; vehicle scrap value at 160 900 km (100,000 miles) would be equal; mechanic labor rates at automobile dealer service centers would be independent from engine type; portions of insurance premium, such as public liability and property damage, would not vary with engine type; etc.

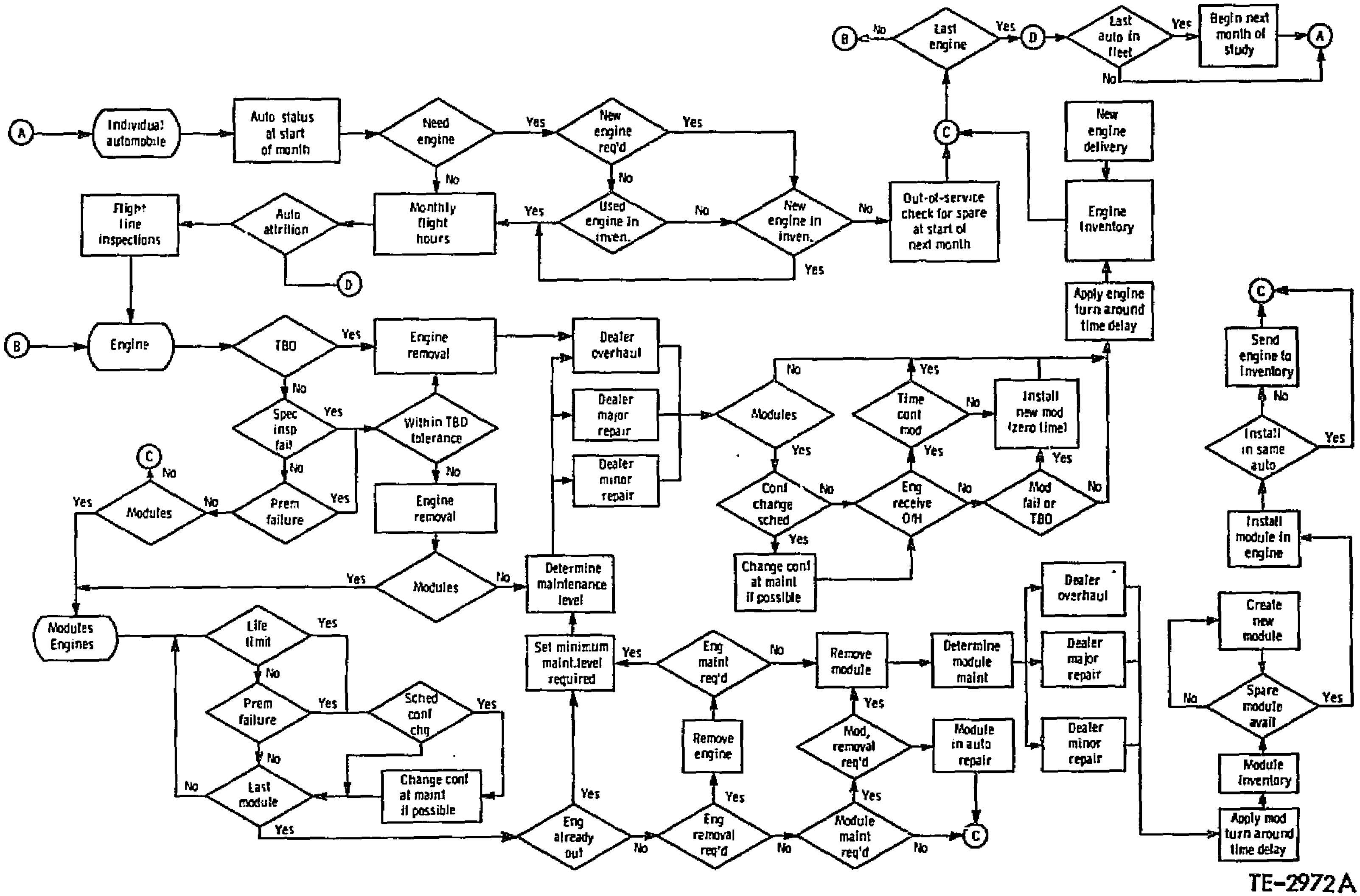

Figure 131. Life cycle cost computer model logic diagram.

The general approach to LCC analysis was well understood by DDA for commercial applications (ref 8). Published papers have dealt with LCC theory (ref 9) and application in design (ref 10) as well as specific customer analysis (ref 11). The proved logic contained in the OS590 program for operating costs became the basis for doing detail repair and maintenance for the Otto cycle car and the turbine car.

The results of the LCC study were shown as plus or minus dollars from the Otto cycle, 1980 "X" body, 33% car, adjusted to a 1977 economic base line. It was believed that GM proprietary interests could be protected while giving the reader an understandable dollar reference instead of a potentially misleading percentage loss or gain.

LCC Computer Model Inputs and Assumptions

No change in LCC was assigned to garage, parking, tolls, and fees.

Common logic elements used included:

- A fleet of new vehicles
- No loss of vehicle population (each vehicle attained 160 900 km (100,000 miles) of usage)
- Scheduled maintenance program (reference 1978 Pontiac schedule extended through 160 900 km (100,000 miles))
- Maintenance at a dealer service center where labor costs were assumed to be $20.00/hr
- Otto cycle vehicle at 8.3 km/l (19.6 mpg)
- Turbine engine vehicle at 11.5 km/l (27 mpg)--20% improvement with diesel fuel
- Otto cycle engine used no-lead gasoline at $0.184/l ($0.70/gal)
- Turbine engine used diesel fuel at $0.172/l ($0.65/gal)
- Distance to time conversion 48.27 km/h (30 mph). This conversion was necessary to convert certain turbine maintenance functions from time to distance units. The rate used closely approximates both the Advisors Circular, Federal Specification Test and the EPA urban/highway driving cycle.

Common elements excluded from the basic maintenance and repair calculations were:

- Tire rotation
- Chassis lubrication
- Vehicle inspection
- Wheel bearing repack
- Brake and power steering adjustment/replacement
- Body and bumper checks
- Air conditioning recharge and/or check

The maintenance schedules for the Otto cycle and the turbine engines are shown in Tables XXXVIII and XXXIX, respectively. It was observed that in addition to the longer maintenance intervals for the turbine, several maintenance items for the Otto cycle engine disappeared from the turbine schedule. Among these were:

- o Positive crankcase ventilation valve and filter
- o Evaporation control system filter
- o Catalytic converter
- o Muffler
- o Water cooling system (antifreeze)

TABLE XXXVIII. OTTO CYCLE SCHEDULED MAINTENANCE													
Action	Miles 7,500	15,000	22,500	30,000	37,500	45,000	52,500	60,000	67,500	75,000	82,500	90,000	97,500
Change oil	X	X	X	X	X	X	X	X	X	X	X	X	X
Oil filter	X		X		X		X		X		X		X
Carburetor choice check	X		X			X			X			X	
Carburetor mounting torque	X		X			X			X			X	
Idle speed adjust	X		X			X			X			X	
PCV valve & filter				X				X				X	
Spark plugs			X			X			X			X	
Engine timing			X			X			X			X	
ECS filter				X				X				X	
Air cleaner element				X				X				X	
Antifreeze				X				X				X	
Transmission fluid								X					
Fuel filter		X		X		X		X		X		X	

TABLE XXXIX. TURBINE SCHEDULED MAINTENANCE						
Action	Miles 15,000	30,000	45,000	60,000	75,000	90,000
Change oil			X			X
Oil filter			X			X
Fuel filter	X	X	X	X	X	X
Air cleaner element (2)		W		W		W
Igniter		X		X		X
Clean fuel nozzle				X		
Transmission fluid				X		
W = wash but not replace						

Unscheduled maintenance predictions were made, using the simulation model. A set of "flat rate" values for labor hours was estimated where GM flat-rate values were not available. Material costs for typical maintenance actions were estimated. A matrix of probable unscheduled maintenance actions was developed. This data set then bounded every data element needed to generate costs per occurrence. Rates for maintenance actions were reviewed, using in-house fleet maintenance data. Reliability values were developed from which frequency of each kind of maintenance could be developed. The resulting actions per car are shown in Table XL.

TABLE XL. ACTIONS PER CAR PER 160 900 KM (100,000 MILES)

Action	Otto cycle	Turbine
Minor engine	1.96	1.14
Turbine seal	---	0.11
Electronic control	---	0.16
Short block	0.02	---
Valves	0.26	---
Exhaust	1.37	1.20
Water pump & hoses	0.75	---
Water hoses	0.77	---

Fuel costs were calculated, based on quantity of the appropriate fuel used per unit distance multiplied by the 160 900 km (100,000 miles) and cost per unit for each car. The combined urban/highway cycle was used to determine the average rate of fuel usage.

Depreciation was calculated with the realization that a difference in selling price exists for each engine configuration. The same scrap value at 160 900 km (100,000 miles) was assumed for both engine configurations. No credit was taken for the probability of more residual turbine engine life.

Insurance was calculated, following the DOT study (ref 5). An adjustment was made to the collision portion of the cost. An Allstate insurance policy was used to determine the portion of a 1978-1979 premium allocated to collision. This amount was then adjusted to consider the difference in the manufacturer's suggested retail price of the turbine car.

Adding oil for the turbine is almost zero. Because no products of combustion mix with the oil and the oil cavity is not pressurized, oil life is anticipated to exceed 72 000 km (45,000 miles). The Otto cycle, on the other hand, begins with a small usage of oil and increases to about one quart in 1600 km (1000 miles) at 80 000 km (50,000 mi), which continues for the study life. Therefore, a total of 76 quarts of oil would be added over the car life.

Results of LCC Analysis

The results of the detail LCC analysis are summarized in Table XLI. The 160 900 km (100,000 mile) saving per turbine-powered vehicle was shown to be about $800. An additional saving of $180 could be realized through the use of the optimized four-speed transmission. This combination would yield a 30% improvement in fuel economy or 12.4 km/l (29.2 mpg) with diesel fuel.

TABLE XLI. TURBINE ENGINE CAR LCC SAVINGS OVER OTTO CYCLE ENGINE

		Saving/(Loss)
Depreciation		$(1,000)
Fuel		1,160
Repair and maintenance		630
Engine, minor	$70	
Engine, major	60	
Exhaust	(90)	
Water cooling system	180	
Scheduled maintenance	410	
Insurance		(70)
Add oil		82
		$ 802

LCC Sensitivity Analysis

Various tests were applied to the LCC results to help determine the sensitivity to error. The three largest drivers in the LCC study were:

- o Depreciation
- o Fuel
- o Repair and maintenance

Depreciation was a direct function of relative purchase price. An unfavorable 10% difference from the base line for the turbine would have affected the LCC result about 12%.

The fuel cost contribution was calculated by using a $0.012/l ($0.05/gal) differential. The most sensitive number in the analysis is the fuel differential. The net delta LCC varies more than a dollar for dollar basis. A 20% difference in fuel price saving results in nearly a 30% variance in LCC savings. Fortunately, fuel usage can be determined on the test stand. An error of 10% in SFC results in about 1% in LCC savings impact.

Repair and maintenance calculations showed the greatest weight in scheduled maintenance. A 10% variance in scheduled maintenance cost for the turbine resulted in about 5% variability in LCC savings. A substantially larger variance could be absorbed in each of the unshceduled turbine engine maintenance categories to match the 1% variance of scheduled maintenance.

It was clear that fuel price differential was the most sensitive variable in the study. Certain sources within the petroleum industry were contacted to confirm the price differential used. These sources were of the opinion that the \$0.012/l (\$0.05/gal) was the best estimate through 1980-1981. After that, no one was prepared to guess. These sources pointed out that in late 1977 and early 1978, the pump price differential between no-lead and diesel was 4+ cents per gallon attributable to a short-term oversupply caused by the severe winter. The average spread returned to about 5 cents per gallon in the late summer of 1978.

Conclusions

The LCC for the turbine-powered car showed a favorable cost difference over the life of the car. The purchase price differential appears competitive in that it was in the price range differential for diesel offerings for 1978 and 1979 GM cars. The volume fuel saving was in the national interest and showed a visible difference to the motorist when visiting the fuel pump.

The LCC analysis indicated an economic benefit for the turbine engine. The magnitude of LCC savings should be stable unless every significant LCC estimate for the turbine worsened by 12%. Because this set of circumstances is considered unlikely, the favorable LCC outlook was considered to be reasonably attainable.

VIII. DEVELOPMENT PLAN (TASK IV) AND LONG-LEAD RESEARCH AND DEVELOPMENT (TASK V)

The design, development, and demonstration of an experimental IGT powertrain is the second phase of a multiphased effort required to place a turbine-powered passenger car into volume production. The Conceptual Design Study of an Improved Gas Turbine (IGT) Powertrain (NASA Contract DEN3-28), which is the subject of this report, was the first phase. Additional follow-on phases (grouped under a collective term of "Production Evolution") will be required before volume production can be reached.

The second phase would be completed by 1983, the year in which the vehicle demonstration is required by the Automotive Propulsion Research and Development Act of 1978 (Title III, Public Law 95-238). This is the act that authorized the joint Government/Industry prototype development.

DDA has defined a development plan for the experimental phase. This plan itemizes the required tasks, schedule, and costs. A generalized estimate of the events, schedule, and costs of the follow-on Production Evolution phases has also been developed. In addition, DDA has delineated the requirements for long-lead research and development items. This section covers the events and schedules. The costs are considered to be proprietary information but were submitted to NASA in a separate document.

VEHICLE SELECTION

The 1983 Pontiac Phoenix vehicle was selected for the IGT development plan. This compact-class car, one of the General Motors "X" body family, was first marketed for the 1980 model year. It is an advanced-design vehicle that incorporates two features—transverse front-wheel drive and inertia weight—which are expected to be typical of passenger cars of the 1990's—the time period in which the gas turbine-powered vehicle is expected to be in full production. Front-wheel drive is rapidly becoming commonplace in automobiles so that the choice of such a vehicle for gas turbine development is consistent with the trend of the future. Similarly, the 1361-kg (3000 lb) inertia weight class of the Phoenix, although it is presently considered in the compact car range, is more representative of that of future full-size automobiles.

The 1980 Phoenix has reached production and is therefore already well defined. When the testing of the gas turbine powertrain begins in 1983, the test vehicle will be a carryover of the 1980 model with only minor modifications. Hence, the design and installation requirements for the IGT powertrain can be fully specified at the outset of the program. If, on the other hand, a car planned for a later market introduction were to be selected, the design and development of the IGT would inevitably have to respond to vehicle design iterations that could not have been recognized at the start of the program.

EXPERIMENTAL IGT DEVELOPMENT PLAN

The powertrain selected by DDA/PMD during this study is a two-shaft engine and a conventional automatic transmission. This powertrain is estimated to produce a 30% improvement in composite fuel economy over a 1976 compact

Otto-powered vehicle. Essential to achieving this goal are advanced, high-performance components plus ceramic components that will allow high operating cycle temperatures. Accordingly, this IGT development plan is an integration of component development, ceramic (materials, process, and part fabrication) development, and engine design and development. In addition, transmission and vehicle modifications are required to accommodate the turbine engine.

Strategy

Development of the ceramic turbine rotor is the highest-risk program element. The earliest anticipated availability of a ceramic rotor suitable for engine test is relatively late in the program; hence, initial engine testing is planned with a metal turbine rotor. This plan will permit the mechanical and performance development of the engine to be carried out prior to the availability of the ceramic rotor. Moreover, the metal rotor approach offers a backup to the possibility that the ceramic rotor development will not progress fast enough to allow for engine running with a ceramic rotor. DDA has devised a plan whereby the engine can initially be provided with a metal rotor and later be modified to accommodate a ceramic rotor so that both versions will allow meaningful vehicle/engine match-ups to be made. The following summarizes this strategy:

- Engine components will be sized for approximately 0.45 kg/s (1 lb/sec) airflow. Except for the gasifier turbine rotor, the engine will be designed for a 1286°C (2350°F) capability.
- The first engines will consist of components matched at 1080°C (1975°F) and will be rated at 60 kW (80 hp) at 29°C (85°F), 152 m (500 ft) ambient conditions. This turbine engine size in the 1983 compact vehicle will match the vehicle's performance with a spark-ignition base engine. Component matching for 1080°C (1975°F) is achieved by adjusting the variable-geometry elements to their appropriate positions. Gasoline-equivalent fuel economy (i.e., thermal efficiency) is estimated to be improved 20% over a 1976 same-weight vehicle.
- In parallel with the metal rotor engine development, a ceramic gasifier turbine rotor acceptable for engine testing will be developed. When ready, it can replace the metal rotor. This plus the rematching of the turbine nozzles for 1286°C (2350°F) will result in a larger and improved engine (rematching is accomplished by simply resetting the variable geometry). Because component airflow is unchanged, engine power will increase to 78 kW (105 hp). This power size is consistent with that of the optional engines expected to be offered with 1983 compact vehicles. A turbine of this size could also be applied to larger vehicles (intermediate or full-size) as a base engine. Fuel economy with the 1286°C (2350°F) engine is estimated to be improved 30% (gasoline equivalent) over a same-weight 1976 base line.

Development Plan Events and Schedule

The overall experimental IGT development plan requires five years to complete, as shown in Figure 132. Activity occurring "left" of the dashed line should be initiated as long-lead R&D tasks. Component development is started at the earliest possible date and focuses initially upon concept evalua-

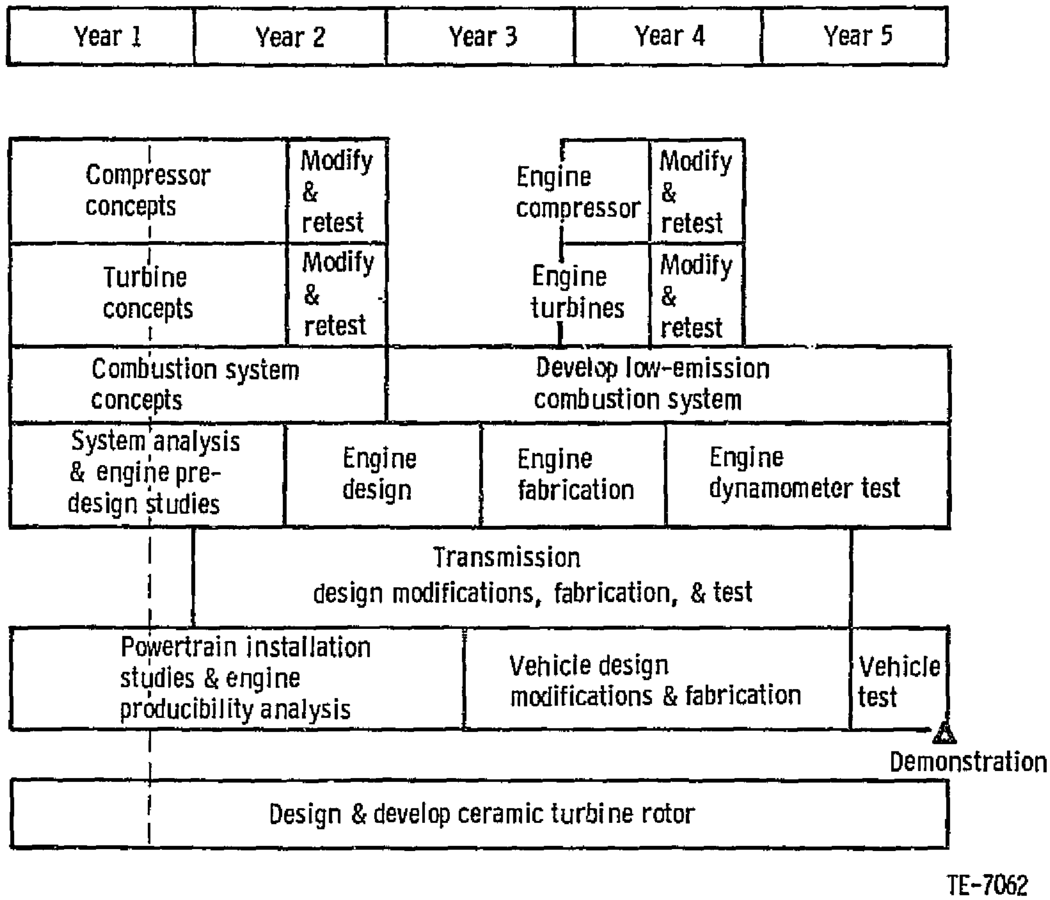

Figure 132. Experimental IGT powertrain development plan.

tion. Compressor and turbine concepts are directed toward solving the challenges associated with low airflow levels. This includes achieving high basic efficiency levels despite the adverse effects of low Reynolds number and nonscaling tolerances and clearances. Alternate means of achieving wide-range airflow characteristics will also be examined; the goals are to minimize the efficiency penalty and maximize producibility. Combustor concepts will concentrate on achieving low emissions with minimum use of variable geometry and with fuel delivery and management systems compatible with high 1286°C (2350°F) combustor outlet temperature. Because of the strong influence of component flow path on basic engine arrangement, rig test results of compressor and turbine concepts are judged necessary prior to starting engine hard-line design. This will allow the engine design to be based on the best component concepts and minimize the compromises which might result if the engine design were finalized earlier and made to accommodate a variety of possible component concepts. Subsequent component development improvements can be incorporated into the engine because the engine layout is based on the same component concepts.

Ceramic gasifier turbine rotor design and development should start at the earliest possible date. Although the turbine aerodynamic concept is being developed simultaneously, ceramic rotor development must be started early to solve the challenges of structural design and ceramic material/process/fabrication. A representative aerodynamic rotor can be selected for the initial ceramic rotor design; subsequent rotor changes required for aerodynamic reasons will not negate the progress made in ceramic development. Rotor development will continue throughout the program. If the anticipated progress is achieved, a ceramic rotor suitable for engine dynamometer test will be available during 1983; however, at this time a vehicle demonstration (emissions and fuel economy) of a ceramic rotor by 1983 is not deemed feasible.

Outstanding performance and fuel economy is a necessary but not sufficient condition for turbine engines to reach volume production. Producibility at low cost is also a requirement. The two-shaft engine selected during the Concept Design Study incorporated producible design features; however, the nature of a study comparing various powertrain concepts does not allow for adequate attention to producibility. Consequently, an in-depth producibility effort should be started in parallel with component development. The focus will be on the review and analysis of the engine concept which was "priced" during the Concept Study and on the analysis of the component concepts under consideration. Elements of the producibility study will be part function, tolerance, material, process, and assembly.

The component development, ceramic rotor development, and producibility study efforts must be coordinated to ensure the result of a maximum payoff in the engine design. This will be accomplished by a parallel task which includes system analysis and predesign studies. System analysis will include engine and vehicle computer modeling to ensure that component characteristics result in the desired system performance and fuel economy. Trade-offs will be examined to ensure that the overall vehicle is optimized. Engine predesign study is required to set dimensional constraints for component concepts and to evaluate components for compatibility with good engine arrangement potential.

Hard-line engine design is begun following the completion of the initial rig tests of the component concepts. Design is followed by parts fabrication/procurement and engine dynamometer testing. A total of five engines is required. Testing will be directed toward developing mechanical systems for function and reliability acceptable for vehicle emissions and fuel economy demonstration. Engine dynamometer testing is estimated at 2000 hours. Aerodynamic components of the engine design will be developed through rig testing. Rig improvements will be incorporated into the engine so that engine power and sfc can be developed.

Transmission and vehicle design modifications (minor) will be initiated as the engine design solidifies. The transmission will be tested as a component and, with the engine, as a system. The total system will be evaluated during the vehicle test; the ultimate goal is a vehicle demonstration of emissions and fuel economy.

Component & Parts Test Plan

Testing of components and parts starts the second year and continues to the end of the program, as shown in Figure 133. The initial test phases of the compressor, turbine, combustor, and ceramic rotor activities are designed to

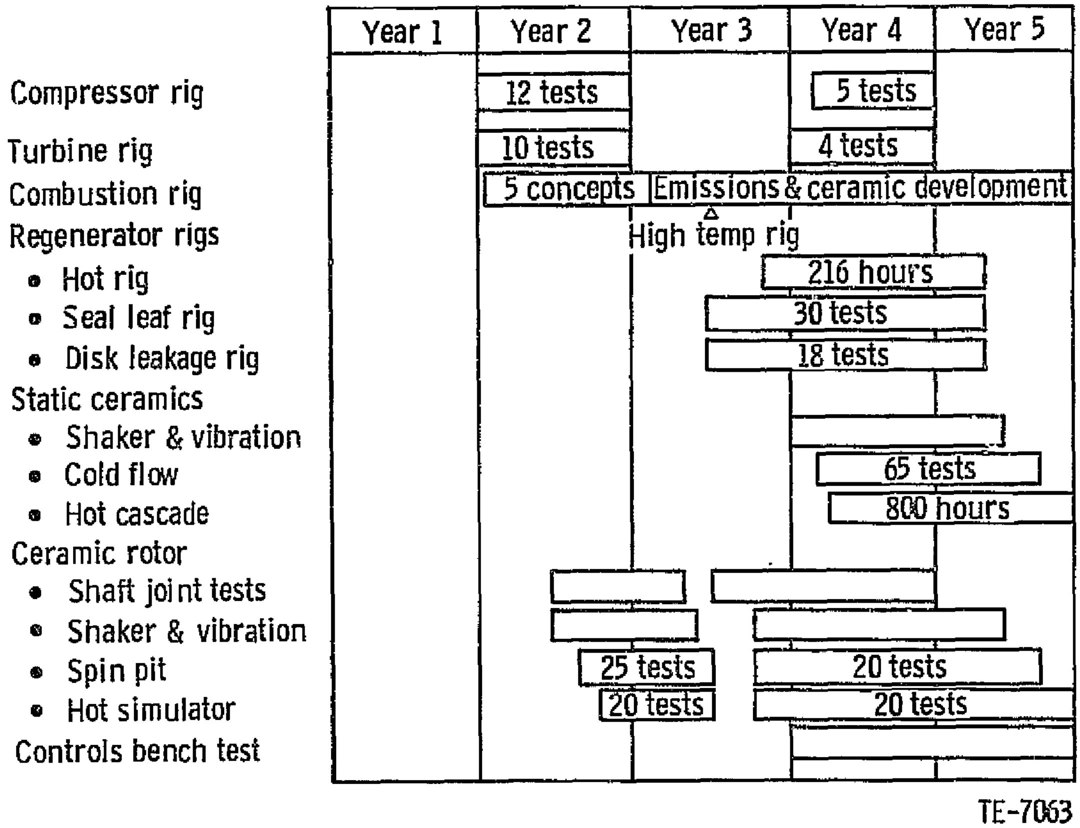

Figure 133. Component rig test plan.

evaluate various candidate concepts. Thereafter, the tests serve to evaluate and develop those concepts selected for incorporation into the engine design. The IGT engine ceramic regenerator is based on the technology being concurrently developed under the DOE/NASA-sponsored CATE program wherein DDA is developing ceramic components for the Model 404 industrial gas turbine engine. Consequently, the IGT program requires only rig testing to develop the specific regenerator design. Static ceramic parts are also based on the technology developed by the CATE program, and testing under the IGT program consists of evaluating and qualifying specific parts for engine test.

Engine Test Plan

The experimental IGT program requires five engines which will be tested for approximately 2000 hours (Figure 134). Engine testing starts mid-year in the fourth year. This schedule, the result of intentionally delaying the start of engine design until component concepts have been rig tested, is judged to be cost effective in meeting the overall program objectives. Initial testing includes two engine build configurations--one with an all-metal flow path (except for the ceramic regenerator) and the other with static ceramic parts (i.e., all of the ceramic parts except for the turbine rotor). This testing sequence allows the simultaneous development of performance, mechanical systems, and ceramic parts. During the fifth year,

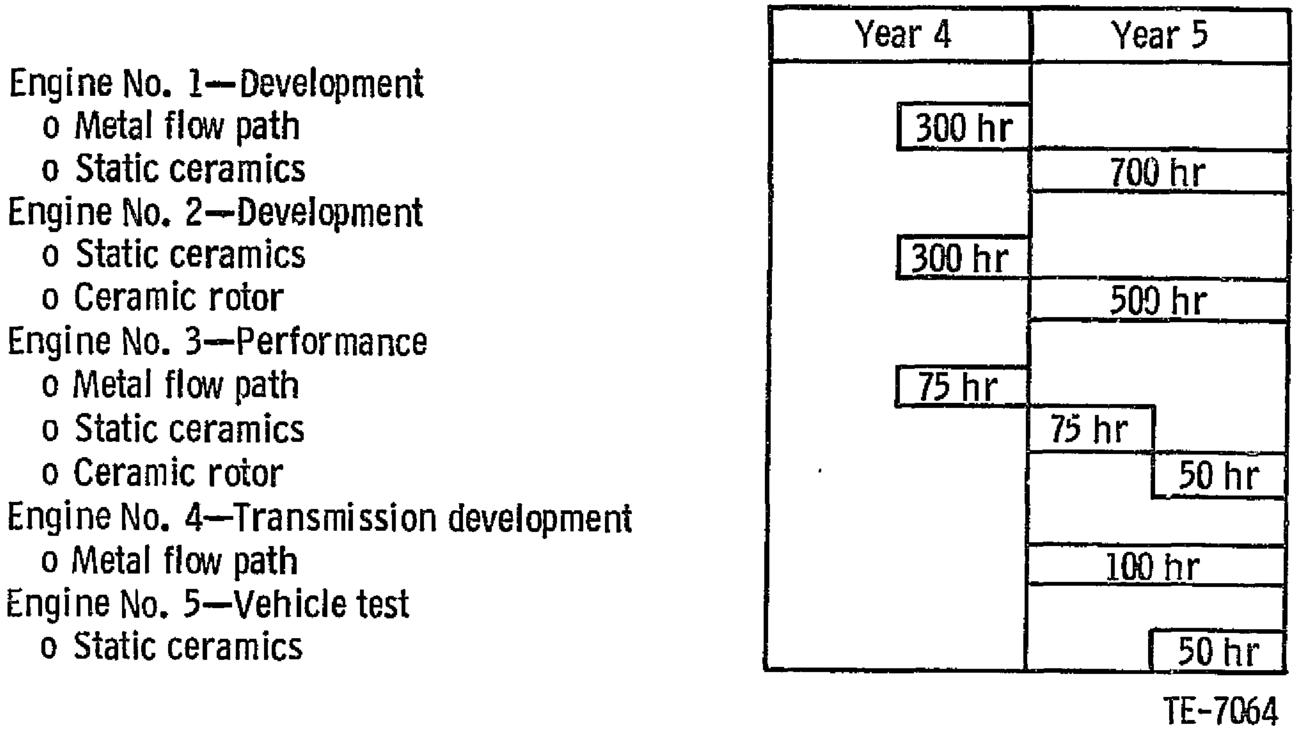

Figure 134. Engine test plan.

all engine development testing (engines 1, 2 and 3) is carried out with ceramic flow path parts including the ceramic rotor in engines 2 and 3. Engine 4 is used for engine/transmission systems testing and, consequently, is built with a metal flow path. The vehicle test engine (No. 5) will be built with static ceramic parts.

Basic Work Breakdown Structure

The experimental development program is made up of the following 11 basic tasks:

1.0 ENGINE DESIGN
Engine design includes all of the technical effort required to complete the engine design (except for the ceramic materials engineering effort of Task 7.0). Follow-up efforts required during parts fabrication and bench/rig tests are included; thus, Task 1.0 continues up to the point of the first engine test.

2.0 ENGINE PROCUREMENT
All engine hardware procurement is included in this task (purchased, in-house, metal, ceramic, etc). Engine hardware to be used for component or rig testing is a part of this task.

3.0 BENCH & RIG TESTS

This task includes all testing at the component or part level that does not fall into the aerodynamic performance or regenerator development categories. Typical tests include controls calibration, vibration, and spin and flow testing. Tests for the screening of ceramics parts and their qualification for engine running are also included.

4.0 ENGINE TEST

This task includes the design and fabrication of test equipment/tools, the preparation of the facilities to be used, and the assembly and dynamometer testing of five 1080°C (1975°F) engines (i.e., metal gasifier turbine rotors). The five engines are allocated to development (2), performance (1), vehicle (1), and transmission test (1). Dynamometer testing of the vehicle engine will be brief, consisting of check-out and calibration. Two 1080°C (1975°F) engines will be converted into 1286°C (2350°F) engines by replacement of the metal gasifier turbine rotor with one of ceramic. Dynamometer testing of the 1286°C (2350°F) engines will include development testing as well as performance and emission demonstrations.

5.0 ENGINE DEVELOPMENT

Engine design will be completed with the first engine run. All technical effort required to develop the engine will, from that point onward, fall under this Engine Development task. Such efforts include redesign and design modification, follow-up of hardware procurement and test, and data analysis.

A few Engine Design (Task 1.0) projects will be kept open past the initial engine test. An example is the ceramic gasifier turbine rotor, which will remain a design task until the ceramic rotor is engine tested.

6.0 COMPONENT DEVELOPMENT

This task accomplishes the concept technology and performance development of gas path components. To accomplish this goal, great emphasis will be placed on performance rig testing. The first part of this task includes the design, fabrication, and performance rig testing of component concepts. The second part is the performance development of the component concepts selected for the engine design. Engine hardware will be used for the second phase of rig tests (both their design and fabrication will be at no cost to this task). Analysis of test results and the design, fabrication, and test of component modifications provide the evolutionary process by which components are developed. Consequently, such efforts are included in this task.

7.0 CERAMIC MATERIALS DEVELOPMENT

This task includes the materials engineering and development required to characterize and qualify ceramic materials for engine design use. Support of the engine design, fabrication, and test phases will be provided for all materials.

8.0 TRANSMISSION

The two-shaft IGT engine requires a conventional automotive torque converter-type transmission in a transverse arrangement suitable for a front-drive vehicle installation. A basic Hydra-matic front drive transmission will be modified to incorporate (1) a torque converter optimized for the final IGT configuration, (2) a lockup clutch, (3) optimized gearing ratios for maximum engine performance, (4) transfer chain ratios and final drive gearing to achieve optimum performance and top-geared vehicle speed, and (5) control changes as required by gear ratio changes and lockup clutch arrangement.

Three pilot units will be built for a dynamometer development program that will include performance testing, power loss investigation and improvement, control functions, cooling requirements, and modification improvements. One unit will undergo durability testing.

Two final-design units will be built and functionally tested before engine/transmission powertrain testing and vehicle testing.

9.0 VEHICLE

A comprehensive program will be undertaken to result in the evolution of a turbine-powered vehicle from an existing base vehicle. This effort will require the design of new components, the revision of existing components, the procurement and/or fabrication of new and reworked components, the build of vehicles, and the characterization of the powertrain sufficient to provide a reasonably commercial turbine engine-powered vehicle for demonstration. The vehicle will be complete, including the normal high-volume options, such as automatic transmission, power steering, power brakes, heater, and air conditioning. In the interest of economics, low-volume options, such as manual transmission, cruise control, and automatic leveling systems, will not be included in the design and build activity. This preliminary development effort will not include any work related to meeting MVSS and barrier test requirements. Vehicle testing will culminate with a fuel economy and emissions demonstration.

10.0 ENGINE PRODUCIBILITY DEVELOPMENT

This activity involves the analysis of difficult-to-produce components of the engine to determine, evaluate, and optimize design materials and manufacturing processes relative to function and performance. Component changes will be monitored for compatibility with the vehicle to maintain the feasibility of obtaining the goal of a demonstration vehicle.

The producibility study and development work will be second in importance only to the engine development. The viability of this engine as a replacement for the piston engine will be dependent on the ability to develop components and assembly methods that will provide a cost-competitive alternate to the piston engine.

The materials, dimensional specifications, forming and machining, and assembly techniques will be analyzed in depth. The base materials and types of castings will be reviewed with the appropriate sources and correlated with the latest technology in casting, machining, and welding to optimize the material selection and forming techniques compatible with high-volume machine process capability. This study will eventually involve work with actual pieces as the program evolves to the point where hardware will be available.

11.0 PROGRAM MANGEMENT

This task consists of the Project Engineering and Administration required to manage, coordinate, and report the overall program. Meetings and exhibits, such as the semiannual Contractors' Coordination Meetings, are also included in this task.

LONG-LEAD RESEARCH AND DEVELOPMENT (TASK V)

An examination of the overall requirements for developing an IGT powertrain revealed that long-lead research and development was required in the fields of aerodynamic components (compressor, turbine, and combustor), ceramic rotor, and engine producibility. Simultaneous studies in the areas of engine/vehicle predesign and system analysis should also be undertaken to ensure a coordinated overall effort that will result in an integrated, optimized powertrain and vehicle.

The compressor and turbine R&D must be concentrated on high efficiency in low airflow size and wide-range, variable-geometry design concepts that will minimize both efficiency penalty and production cost. The ceramic rotor development must address the development of high-strength ceramic materials, process and fabrication techniques, and design optimization procedures. The producibility activity must address both design and manufacturing ingenuity so that an IGT powertrain, principally the turbine engine, can be developed within the framework of competitive selling price. Although the turbine has a favorable life-cycle cost relative to the base-line Otto engine, its adverse selling price should be lowered to better attract buyers.

GAS TURBINE VEHICLE PRODUCTION EVOLUTION

Figure 135 outlines the potential evolution of the gas turbine-powered vehicle through the various phases of development to the start of production. The realization of this evolution is dependent on the timely development of the ceramic components required to realize the full potential of the engine. The timing as shown would be possible only if no major problems occur to cause delays in any of the phases as scheduled.

Four-Vehicle Evaluation Supplement

As shown in Figure 135, a four-vehicle supplement to the DOE/NASA program has been included. This additional program is required to provide sufficient preliminary durability and development experience so that industry can make a prudent decision as to the feasibility of proceeding with an extensive experimental preproduction program.

Experimental Component Program

This phase would be very comprehensive, involving 40 or more vehicles and 100 or more engines. The engine at this time would be redesigned as predicted by the earlier durability testing and may be resized if vehicle trends so dictate. This program would provide industry management with the cost, performance, and durability data required to arrive at a decision with respect to production engineering and soft tooling (as required to build 1000 preproduction vehicles). This decision point, which would occur around the middle of 1986, would also involve the final selection of the vehicle for market introduction of the gas turbine powertrain.

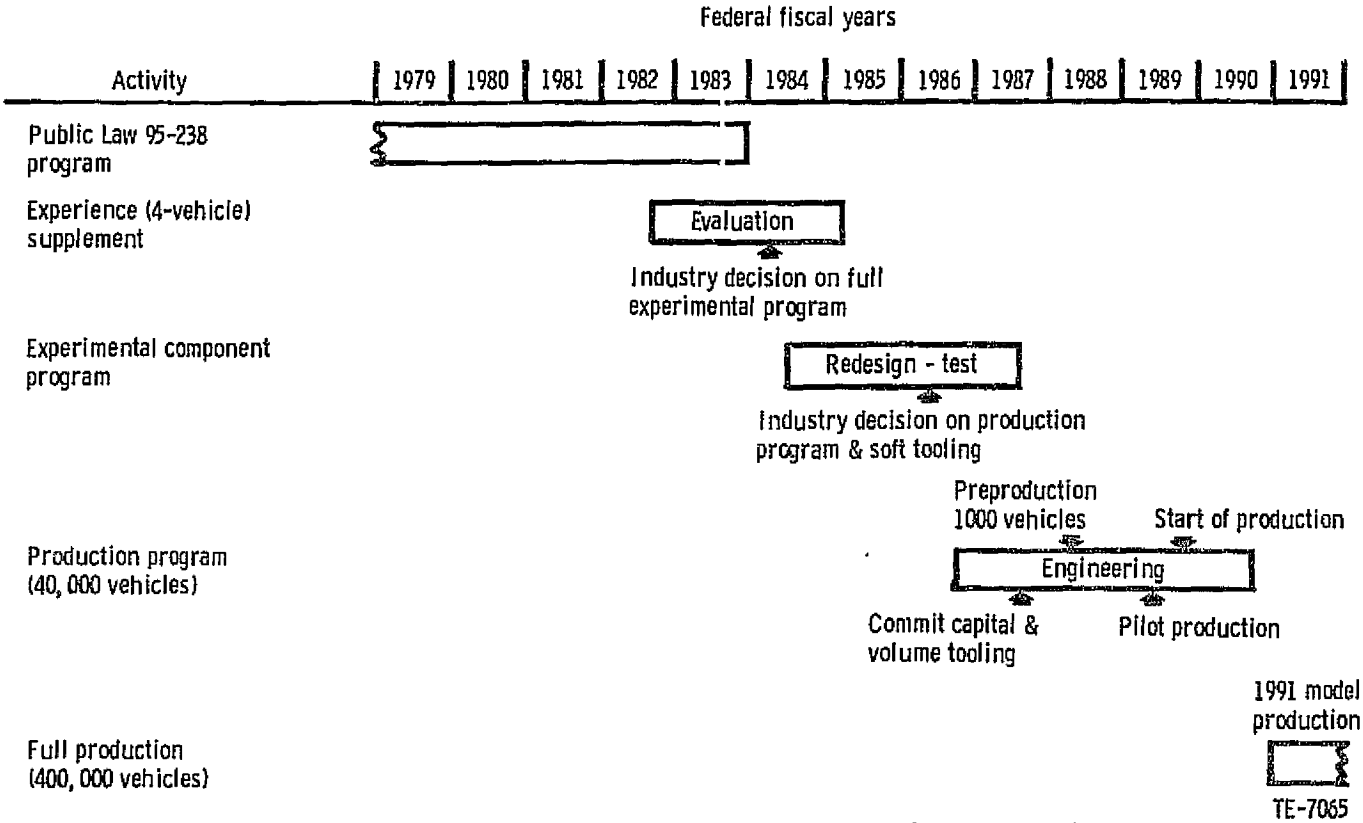

Figure 135. Gas turbine vehicle production evolution.

Production Program

The production phase would involve the final design iteration of the engine and the evolution of this design into production. This would involve the full powertrain and vehicle durability, performance, and development test programs necessary for production release, including certification to all required Federal Standards.

A 1000-vehicle preproduction field test would be conducted approximately 1.5 years ahead of the start of production. The engines would be manufactured with "soft tooling" that will have sufficient capability for low volume. These vehicles would receive customer evaluation in General Motors and suppliers' fleet service and would generate field data for "final tuning" of the engine.

The introduction of the new gas turbine powertrain would coincide with the introduction of the remainder of General Motors vehicles for the 1990 model year. The vehicle model used will have been selected in 1986 in the basis of such factors as the following:

1. 40,000 units the first year
2. Potential for a following-year expansion to 400,000 vehicles
3. Pricing

4. Market segment appeal
5. Performance and economy
6. Image
7. CAFE requirement

Full Production

The final event of this program evolution will be the acceleration to full production of one manufacturing module (400,000 units) with the start of the 1991 model year.

IX. CONCLUSIONS

This document reports the results of an Improved Gas Turbine (IGT) Conceptual Design Study. The objectives of the study were to define a turbine powertrain which offered at least a 20% improvement in driving cycle thermal efficiency, could enter production engineering by 1983, possessed comparable life, reliability, and driveability, met or exceeded Federal Emission Standards as well as legislated noise and safety levels, and had competitive initial cost and a life cycle cost no greater than that of conventionally powered vehicles.

The following conclusions were established as the result of this study.

1. An improvement of 20% in driving cycle thermal efficiency can be achieved with a low-sfc gas turbine engine. This engine includes wide-range aerodynamic components, ceramic regenerator and static gas path parts, and a ceramic gasifier turbine rotor.

   An improvement of 30% in thermal efficiency can be achieved with the low-sfc engine plus an optimized drive line. The drive line includes optimized gear ratios, shifting patterns, and charging pump size plus a lockup torque converter.

2. Powertrains consisting of (1) a two-shaft engine and automatic transmission and (2) a single-shaft engine and continuously variable transmission have equivalent thermal efficiencies and production costs. Both can meet noise, safety, and emission standards equally well. The two-shaft engine has a lower risk factor regarding the incorporation of a ceramic turbine rotor and, thereby, has a better life potential. The single-shaft powertrain would need a new continuously variable transmission and thus requires greater development cost and has a higher technical and schedular risk. A differential engine powertrain is unacceptable because of its inferior thermal efficiency.

3. Wide-range aerodynamic components result in significantly better driving cycle thermal efficiency than do fixed-geometry components, especially at higher engine idle speeds.

   The driving cycle thermal efficiency curve with a wide-range engine is relatively flat as idle speed is varied and thus offers the flexibility which may be required to provide good vehicle response.

4. The efficiency of the best continuously variable transmission (CVT) is lower than that of an automatic transmission. A variable-belt-type CVT has an efficiency that is superior to that of a hydromechanical ("hydrostatic") type CVT.

5. Emission requirements can be met with a variable-geometry, prechamber-type combustor.

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6. The "sticker price" of a turbine-powered vehicle is higher than that of a conventionally powered vehicle; however, the price premium is similar to that associated with currently marketed fuel-efficient engines.

   Life cycle cost is lower with a turbine-powered vehicle than with a conventionally powered vehicle.

7. A 1983 vehicle demonstration of emissions and fuel economy is possible with a turbine powertrain; the engine would include a metal gasifier turbine rotor.

   A dynamometer demonstration of a turbine engine with ceramic gasifier turbine rotor is feasible by 1983.

8. Long-lead research and development activity is required for small-size, wide-range aerodynamic components, a low-emission combustor, a ceramic turbine rotor, and engine producibility.

RECOMMENDATION

The development of a powertrain consisting of a two-shaft engine and automatic transmission should be initiated in a Government-sponsored follow-on program.

APPENDIX A

AMBIENT CONDITIONS

According to certain analyses, two standards of ambient conditions would be appropriate for the analysis of gas turbine-powered vehicles--one for vehicle performance (acceleration, response, etc) and another for fuel economy or thermal efficiency. The first case (performance) necessitates the consideration of the "worst case" while the second (economy) is appropriately simulated by average conditions.

VEHICLE PERFORMANCE

It is well documented that the slope of maximum power output versus ambient temperature is more severe for turbine engines than for Otto-cycle engines. For example, assume the two engine types had equal standard-day, 15°C (59°F) power ratings. The maximum power output of the turbine engine would then be greater than that of the Otto engine on a cold day and lower on a hot day. Acceleration and response are vehicle characteristics of which the driver is immediately aware. Furthermore, he will typically be more sensitive to inferior performance (of an alternate powerplant) than to its areas of superiority. The turbine vehicle must, therefore, be equal to the Otto vehicle under rather severe conditions. It is difficult to know where to draw the line; for example, should you search for the most extreme ambient temperature expected to be met? An engineering judgment porposed that 29.4°C (85°F) and 152.4 m (500 ft) altitude (SAE standard) was adequately severe. The turbine vehicle was thus designed to match the Otto vehicle performance at these conditions. Operation above 29.4°C (85°F) is infrequent enough to not warrant oversizing the turbine powerplant. Also, the performance of the Otto vehicle is not constant; it, too, falls off on hot days and at altitude. It is thus suggested that the driver's sensitivity to vehicle performance has some range of tolerance.

FUEL ECONOMY

Although the driver's sensitivity to vehicle performance is relatively keen, his awareness to fuel economy, if any, is an averaging process. Many drivers do not keep mileage records; their only awareness (again, if any) to fuel economy occurs when they view the window-sticker EPA rating of their new car. Those who do keep records record averages--wittingly or not. A given tank of fuel is consumed under varying driving and ambient conditions; thus, it averages several factors. Furthermore, the typical driver is judged to expect significant tank-to-tank variations (due, for example, to variations of "fullness" achieved at each refill). The point of this analysis is that, in general, the sensitivity of a given driver to fuel economy is not a critical mile-by-mile issue (even though vehicle performance may be).

Although individual drivers <u>may be</u> unaware of the fuel economy of their cars, the aggregate fuel economy of the nation's passenger car fleet is very important and is the most important objective of this study. Thus, it seems appropriate that fuel economy analyses be conducted at the "most typical" conditions. Several studies have already determined 15°C (59°F) to represent such conditions. As verification for the continental United States, DDA has examined public records (ref 12, 13, 14) concerning average temperatures as a function of location and time of day. These records confirm the validity of 15°C (59°F) as a representative figure. Sea level does not represent the average altitude but provides a conservative approach to estimating fuel economy. At 152.4 meters (500 feet) the engine would have to operate at slightly higher power settings; this would improve sfc, and thus fuel economy, very slightly.

For a given power level, the sfc of a gas turbine engine is somewhat better for a 15°C (59°F) ambient than for a 29.4°C (85°F) ambient. DDA's analysis of combined driving cycles has indicated the 15°C (59°F) ambient to give approximately 3% higher mileage per gallon than the 29.4°C (85°F) ambient.

APPENDIX B

FUEL HEAT CONTENT

One of the important assets of a turbine engine is its easy adaptability to a wide range of fuels. This characteristic is attributable to the fact that its combustion process is a continuous one. Turbine engine fuel carries no specification for octane limit, cetane limit, or volatility limit.

For the most part, vehicular turbine engines use Diesel Fuel No. 2, the same fuel which diesel-powered automobiles must use. This fuel has a higher heating value and a lower price than gasoline and thus offers several bookkeeping alternatives for comparing turbine-powered versus Otto-powered automobiles.

ENERGY EFFICIENCY

It is clear that the major goal of this concept study is to identify a powertrain whose consumption of energy is reduced as it performs its required task. The contract specifies that gasoline-equivalent fuel be used for calculation purposes (i.e., same heat value per unit of mass or volume as the Otto-cycle engine uses). This calculation requirement results in various fuel efficiency measurements that reflect thermal efficiency changes. Thermal efficiency, per se, is seldom used in the industry and never by the public. Specific fuel consumption (sfc) is a more common engine measure while the public's awareness of fuel efficiency rests with miles per gallon ("mileage"). Both sfc and mpg are truly sensitive to the particular fuel being used—sfc to Btu per mass and mpg to Btu per volume. The following table compares gasoline and diesel fuel:

	Gasoline	Diesel No. 2
Density, kg/l (lb/gak)	0.743 (6.2)	0.851 (7.1)
Heating value, weight, kJ/g (Btu/lb)	42.6 (18,300)	42.8 (18,400)
Heating value, volume, MJ/l (Btu/gal)	31.6 (113,460)	36.4 (130,000)

The NASA requirement of gasoline-equivalent fuel allows true thermal efficiency improvements to be quantified—an essential requirement when goals are expressed in terms of thermal efficiency. Accordingly, this report calculates fuel economy for a turbine vehicle based on gasoline. But the very same vehicle would actually attain a 15% higher mpg value when operating on Diesel Fuel No. 2. A dilemma is thus created when a commonly used term (miles per gallon) is used as a proxy for thermal efficiency. For the future, the following suggestion for definitions is recommended:

> Fuel Economy: A term matching what a vehicle operator actually does in checking his "mileage"—miles driven divided by gallons of fuel used. Percentage changes in predicted fuel economy should match what the operator would actually experience.

<u>Thermal Economy:</u> A term differing from fuel economy only in that an analytical substitution for gasoline-equivalent fuel is made. Percentage changes in thermal economy would be identical with percentage changes of thermal efficiency.

APPENDIX C

LIST OF ABBREVIATIONS AND SYMBOLS

A/C	Air conditioning
AN^2	Turbine exducer area x speed squared
AT	Automatic transmission
BSFC	Brake specific fuel consumption
CAFE	Corporate average fuel economy
CATE	Ceramic applications for turbine engines
CID	Cubic inch displacement
CIT	Compressor inlet temperature
CO	Carbon monoxide
CVS	Compressor vane setting
CVT	Continuously variable transmission
EI	Emission index
EOC	Engine oil cooler
EVAP	Evaporator
FWD	Front wheel drive
GPSIM	General purpose simulation
HC	Hydrocarbons
HPSN	Hot-pressed silicon nitride
HVAC	Heating, ventilation, and air conditioning
IGT	Improved gas turbine
IGV	Inlet guide vanes
IV	Infinitely variable
JPL	Jet Propulsion Laboratory, California Institute of Technology
LCC	Life cycle cost
m	Material Weibull exponent
MPG	Miles per gallon
N_G	Gasifier speed
NO_x	Oxides of nitrogen
N_{Re}	Reynolds number
P & E	Performance and economy
PREP	Predicted equipment performance
P/S	Power steering
PT	Power turbine
PTIT	Power turbine inlet average temperature
R_e	Expansion ratio
REGEN	Regenerator
RIT	Regenerator inlet temperature
RWD	Rear wheel drive
SiC	Silicon carbide
TIT	Turbine inlet average temperature
TOC	Transmission oil cooler
TV	Throttle valve
T_{Trel}	Total relative temperature
U_T or V_T	Radial turbine inlet tip speed
VG	Variable geometry
WOT	Wide open throttle
ϵ	Regenerator effectiveness
σ_o	Material characteristic strength

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