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ANALYSIS OF REGENERATED SINGLE-SHAFT CERAMIC GAS-TURBINE ENGINES AND RESULTING FUEL ECONOMY IN A COMPACT CAR

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16. Abstract

Ranges in design and off-design operating conditions of an advanced gas turbine and their effects on fuel economy were analyzed. The assumed engine incorporated a single-stage radial-flow turbine and compressor with fixed geometry. Fuel economies were calculated over the Composite Driving Cycle on a 29° C (85° F) day with gasoline as the fuel in a 1588-kg (3500-lb) car. At a constant turbine-inlet temperature of 1644 K (2500° F), with a regenerator sized for a full-power effectiveness of 0.90, the best fuel economies ranged from 11.1 to 10.2 km/liter (26.2 to 22.5 mpg) for full-power turbine tip-speeds of 770 to 488 m/sec (2530 to 1600 ft/sec), respectively.

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SUMMARY

The purpose of this study was to take a first look at the fuel economy potentials of an advanced ceramic gas-turbine engine in a compact car. The assumed engine was a fixed-geometry, regenerated gas turbine operating with a maximum constant turbine-inlet temperature of 1644 K (2500° F). It incorporated a single shaft with a single-stage, radial-flow turbine and compressor. This gas-turbine, with several variations in design conditions, was analyzed for its design and off-design performance. For fuel-economy comparisons, a performance map for an existing spark-ignited piston engine was used. All engines were sized for 74.6 kW (100 hp), coupled to an advanced continuously variable speed-ratio transmission, and applied to a compact car. Vehicle test weight was 1588 kg (3500 lb) with a corresponding curb weight of 1452 kg (3200 lb). Fuel economies then were calculated over the Composite Driving Cycle using gasoline as the fuel on a 29° C (85° F) day at sea level.

At a constant turbine-inlet temperature of 1644 K, with a regenerator sized for 0.90 heat-transfer effectiveness and no other operational limits, the best fuel economy was 11.1 km/l (26.2 mpg), and required a design turbine-tip speed of 770 m/sec (2530 ft/sec) if a one-stage turbine is used. Limiting design turbine-tip speed to 488 m/sec (1600 ft/sec) while keeping the constant 1644 K operation resulted in 10.2 km/l (22.5 mpg); intermediate turbine-tip speeds produced intermediate fuel economies. The fuel economy calculated for the piston engine using the advanced transmission was 8.7 km/l (20.4 mpg). Hence, gas-turbine operation at 1644 K showed from an 11- to 28-percent advantage over the piston engine operation depending on attainable design turbine-tip speed.

Similar unlimited tip-speed operation at a constant 1311 K (1900° F) resulted in a fuel economy of 9.5 km/l (22.3 mpg). With a 488-m/sec limit and 1311 K operation, the gas turbine had the same fuel economy as the piston engine. Intermediate temperatures and/or tip speeds resulted in intermediate fuel economies. Operation of the gas turbine at its best design turbine-tip speed, but with limiting values of regenerator hot-inlet temperature also resulted in intermediate fuel economies. If a larger regenerator were practical, one with 0.94 design effectiveness, constant turbine-inlet temperature operation at 1644 K and no limit on tip speed would provide 12.2 km/l (28.6 mpg).

INTRODUCTION

Saving fuel and protecting our environment have become important national goals. In the automotive area, engines are being improved to conserve fuel and reduce pollution. For the near term, conventional engines can be made to meet national goals. In the long term, however, there are limits (ref. 1) to the improvements which can be obtained with conventional engines. And, reference 2 shows that a new engine is needed to dramatically lower fuel use and to substantially eliminate city air pollution from cars. Two alternate engines are recommended in reference 2, a Stirling engine and a Brayton (or gas-turbine) engine. Both engines, however, need technology advancements, particularly for dramatically better fuel economy. The recommended Advanced Engines of reference 2 use ceramic hot parts to allow high-temperature operation with an attendant high-level of engine efficiency.

This analytical study, as part of a cooperative program between NASA and the Energy Research and Development Administration (ERDA), was conducted to take an independent first-look at the fuel economy potentials of a high-temperature ceramic gas turbine in a conventional car. The scope here was limited to one specific gas-turbine/vehicle combination. However, effects of gas-turbine design conditions on fuel economy were examined in more detail than in reference 2.

A fixed-geometry, regenerated gas-turbine was assumed. It also incorporated a single-shaft arrangement with a single-stage, radial-flow turbine and compressor. The gas turbine was mated to an advanced, continuously variable speed-ratio transmission (CVT), and applied to a compact car. One digital computer program was used to analyze gas-turbine design and off-design performance. A second digital computer program was used to analyze vehicle fuel economy over the Composite Driving Cycle (ref. 1).

A turbine-inlet temperature of 1644 K (2500° F) was assumed as a development goal for a ceramic engine. However, since early versions may be designed for lower values, turbine-inlet temperatures of 1311, 1478, and 1644 K (1900°, 2200°, and 2500° F) were analyzed. Also, two full-power design values of regenerator heat-transfer effectiveness, 0.90 and 0.94, were studied. Design compressor pressure ratio was varied over a range of values at each temperature and effectiveness level. All gas turbines were sized for a maximum 74.6 kilowatts (100 hp). Total vehicle test weight was held at 1588 kilograms (3500 lb). This corresponds to a curb weight of 1452 kilograms (3200 lb). For comparisons, a spark-ignited piston engine performance map was used in the second computer program to obtain its fuel economy within the assumptions of the analysis. The piston-engine fuel economy was obtained for the same horsepower rating and with the same advanced CVT in the 1588-kilogram (3500-lb) test weight vehicle.

This approach, using equal horsepower and equal test weight between engine types, does not result in equal-performance vehicles (see Assumptions and Limitations). An equal-vehicle-performance approach such as in reference 2, would show a larger fuel economy advantage for the gas turbine relative to the piston engine than is shown in this analysis.

Results are presented first for the gas-turbine designs which would give the best fuel economy. These designs required constant turbine-inlet-temperature operation at all engine power levels, and also required high design turbine-tip speeds. Subsequently, results are presented for the effects of lower design values of turbine-tip speeds and of off-design temperature limits on fuel economy.

Some early results of this analysis were previously presented in reference 3. The early results included approximations for the effects of variable turbomachinery geomerty and vehicle improvements. Higher temperatures were also considered. And, higher fuel economies than are presented here were projected as possibilities for an advanced ceramic gas-turbine. Those higher potentials still exist (see CONCLUDING REMARKS), but require a more exact analysis for substantiation. The results here differ from reference 3 mainly in a more thorough evaluation of best-fuel economy designs and the effects of selected engine constraints on those designs.

ANALYSIS

The gas-turbine engine/vehicle model assumed for this analysis is described first. Then the major analytical assumptions are presented. The final part of this section describes the methods of analysis.

Engine/Vehicle Model

A schematic diagram of the engine and transmission arrangement is shown in figure 1. All hot parts of the engine are ceramic. The single-shaft gas turbine has a single-stage, radial-flow turbine and compressor; a rotary regenerator; and a combustor. A single-shaft gas turbine with heat recovery was chosen mainly because of the results of earlier studies (refs. 4, 5, and 6) at lower operating temperatures. All three of these references preferred the single-shaft engine. It is less complex than multiple-shaft versions, but does require an advanced CVT for driveability. All but reference 4 of the earlier studies preferred the gas turbine with heat recovery. It allows high engine efficiency without the need for very high operating pressures. Radial-flow turbomachinery was chosen because of its ability to operate at the required pressures in a single stage with high efficiency. The rotary regenerator was chosen over a recuperator because it is more compact, especially at high levels of heat-transfer effectiveness. Flow leakage losses, however, are larger in a regenerator than in a well-designed recuperator. No particular combustor was specified.

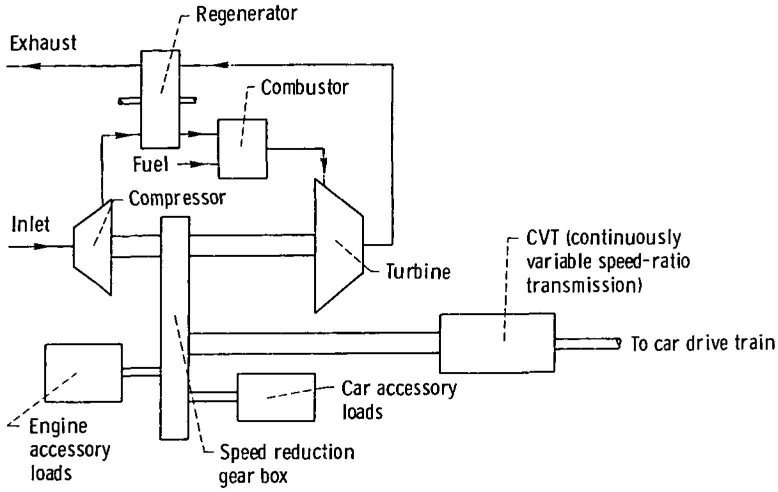

Figure 1. - Schematic diagram of engine/transmission arrangement.

The engine (fig. 1) was mated to the CVT through a speed-reduction gear box. A hydromechanical-type drive was chosen for the transmission. Engine and car accessory loads were separately driven from the gear box. Engine output net power was defined as that shaft power available to the transmission and car accessory loads.

The car itself was conventionally shaped and used radial tires. Car accessories did not include air conditioning.

Assumptions and Limitations

This analysis was limited to one specific gas-turbine/vehicle combination. And, most of the losses and characteristics of the combination were assigned values typical of current design practice (see Methods). Other analytical assumptions and general limitations are presented here categorically.

Engine/vehicle power rating. - Both gas-turbine and piston engines were rated for a full-power net output of 74.6 kilowatts (100 hp) and applied to a compact car with a 1588-kilogram (3500-lb) test weight.

Reference 2 shows that this approach does not result in equal-performance vehicles when comparing alternate engines. (Equal-performance vehicles have the same passenger room, range, acceleration capabilities, accessories, and aerodynamics. They also have equally acceptable driveability, safety, durability, and noise level.) To obtain equal-performance vehicles, reference 2 accounts for differences in engine specific power (hp/lb) and torque-speed characteristics and the propagated effects on vehicle weight. In a compact car with a 1542-kilogram (3400-lb) test weight, reference 2 uses a spark-ignited piston engine with a design maximum power of 93.3 kilowatt (125 hp).

The equal-performance vehicle with a single-shaft gas turbine required a test weight of 1343 kilograms (2960 lb) and a rating of 64 kilowatts (86 hp). Both of these trends, lower weight and power, would increase the fuel economy of the gas-turbine engine relative to the piston engine in this analysis.

Gas-turbine design features. - Variable turbomachinery geometry and power augmentation, such as supercharging or water injection, were not considered in this analysis.

Engine emissions. - Engine emissions were not analyzed. It was assumed that a combustor could be developed to meet or exceed emission standards.

Engine design point. - All engine components were assumed to be designed at the full-power rating of 74.6 kilowatts (100 hp).

Off-design engine operation. - Only one set of turbine and compressor performance maps was used. These maps were normalized to each design point condition. Therefore, it was implicitly assumed that the shape of these maps would not change over the range of investigated gas-turbine parameters.

Transient engine response. - Only steady-state gas-turbine performance was analyzed. It was assumed that the CVT could accommodate vehicle power demands without exceeding the design gas-turbine operating temperature and provide satisfactory engine and vehicle acceleration.

Vehicle-use pattern. - The Composite Driving Cycle (defined in ref. 1) was assumed to indicate average vehicle use. This cycle is a combination of city and highway driving such that the average vehicle speed is 53 kilometers per hour (33 mph) over its operating life.

Gas-turbine fuel. - Although the gas-turbine can operate with a broad range of hydrocarbon fuels, gasoline was assumed for the analysis. Use of diesel fuel in the gas-turbine would result in fuel economies on a miles-per-gallon basis 9 percent larger than those with gasoline. This is a direct result of a greater energy density (J/l or Btu/gal) for diesel fuel.

Regenerator size. - Based on current experimental gas-turbine/vehicle developments (ref. 7), it was assumed that a regenerator with a design heat-transfer effectiveness of 0.90 was practical.

Design-point regenerator seal leakage. - Design values of regenerator seal-leakage flow rates were assumed to be about one-half of those rates which have been achieved (ref. 6).

Ceramic turbine efficiency. - Successful fabrication of a ceramic-turbine rotor may require penalties in turbine efficiency due to thicker vanes and poor solidity. A 0.02 decrease in attainable laboratory turbine efficiencies was assumed as an allowance for ceramic fabrication.

Engine-idle conditions. - Idle conditions were assumed to be those which resulted in an engine output net power of 3.7 kilowatts (5.0 hp). For the gas-turbine, this approach resulted in idle speeds which ranged from about 45 to 53 percent of design speed.

Gas-turbine pressure losses. - Individual component pressure drops were not modeled. Rather, it was assumed that at each operating condition, a specific portion of the compressor pressure rise would be available to the turbine for expansion.

Gas-turbine thermal losses. - Heat losses were neglected.

Transmission efficiency. - It was assumed that the variation of CVT efficiency with vehicle speed and engine output could be linearly approximated in discrete vehicle speed ranges.

Combustion efficiency. - Combustion efficiency was assumed to be 100 percent. Furthermore, the enthalpy required to vaporize the fuel was neglected in the combustor energy balance.

Air and gas properties. - All calculations used enthalpy-entropy variations based on the ideal-gas relations (specific heats as functions of temperature) for the constituent gases. A hydrogen-to-carbon mass ratio of 0.16 was assumed for the fuel.

Methods

Two separate digital computer programs were used: one to evaluate design and off-design performance of gas turbines, and, the other to evaluate vehicle fuel economy over the Composite Driving Cycle. Engine design-point compressor pressure ratio was treated as the main independent parameter in the overall evaluation process. Methods of analysis are presented for the design point, off-design operation, and for engine/vehicle driving cycle fuel economies.

Gas-turbine design point. - Each first calculation was made for the design-point conditions. Major independent input parameters were turbine-inlet temperature, compressor pressure ratio, regenerator effectiveness, and limiting turbine tip-speed, if any.

Certain engine losses were assigned fixed values at the design point. Those are shown in table I. Regenerator and compressor seal-leakage flow rates were assumed to

TABLE I. - ENGINE LOSSES ASSIGNED

AT THE DESIGN POINT

Loss	Assigned design point values	Typical idle-speed values
Total pressure loss (percent of compressor pressure rise)	15	4
Bearing loss, kW (hp)	4.3 (5.8)	1.0 (1.3)
Engine accessory load, kW (hp)	3.7 (5.0)	1.4 (1.9)
Speed reduction gear loss, kW (hp)	2.5 (3.3)	.1 (0.2)

be dependent on design-point compressor pressure ratio. The variations are shown in figure 2. All of these losses, other than the regenerator leakage rates, are representative of current design practice.

Turbine and compressor efficiency were treated as dependent variables. Compressor efficiency was a function of its specific speed (see ref. 8 for compressor terminology), total pressure ratio, and mass-flow rate. In general, a compressor specific speed of 0.775 was assigned. This assignment results in the highest obtainable levels of compressor efficiency. Figure 3 then shows the corresponding variations of compressor efficiency with pressure ratio and corrected mass-flow ratio. Below a corrected mass-flow rate of 3.6 kilograms per second (8.0 lb/sec), efficiency was reduced for size ef-

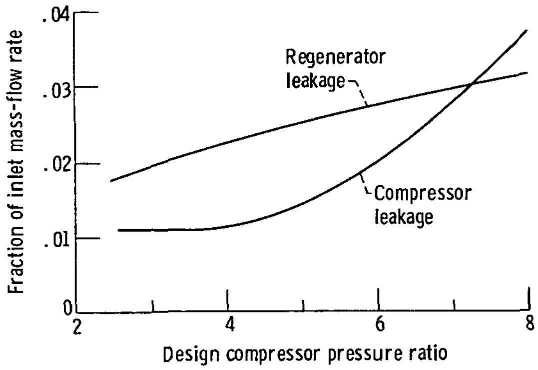

Figure 2. - Design-point variations in seal leakage flow rates.

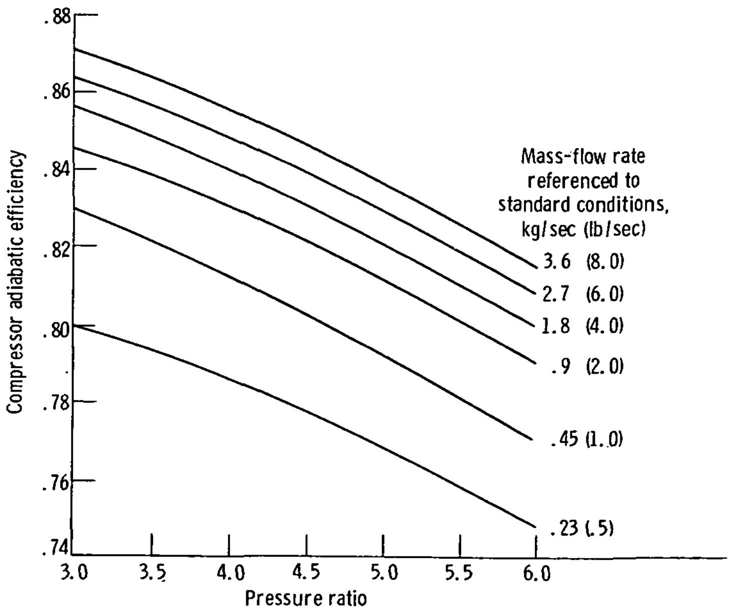

Figure 3. - Variation of compressor efficiency with design pressure ratio and mass-flow rate. Compressor specific speed, 0.775.

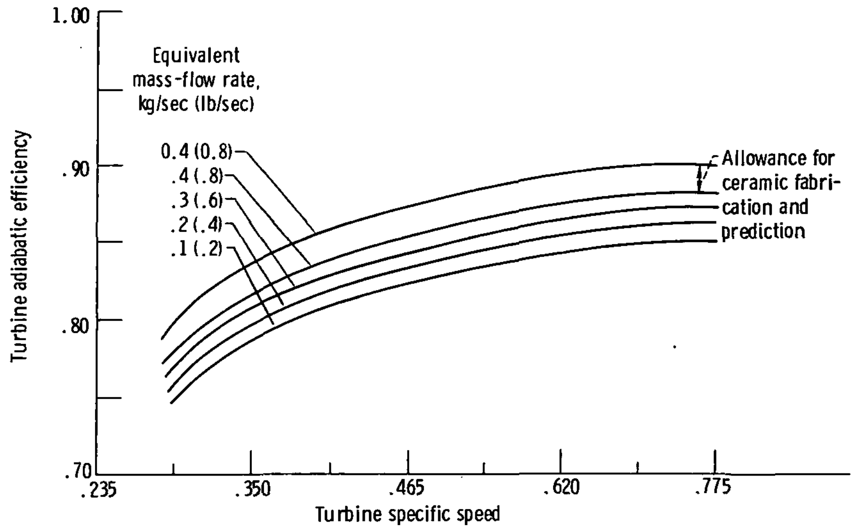

Figure 4. - Variation of turbine efficiency with design specific speed and mass-flow rate.

fects. Turbine efficiency was a function of its specific speed, equivalent mass-flow rate, and blade-to-jet speed ratio (see ref. 9 for turbine terminology). Figure 4 shows the variations of turbine efficiency (inlet to diffuser exit) for optimum blade-to-jet speed ratios. The assumed 0.02 decrease from attainable laboratory turbine efficiencies is shown in the figure. A smaller size effect than for the compressor was imposed for equivalent mass-flow rates below 0.4 kilogram per second (0.8 lb/sec). All turbine and compressor efficiencies in this analysis were influenced by the size effects. When turbine-tip speed limits were imposed, the efficiencies from figure 4 were further decreased for nonoptimum blade-to-jet speed ratios. Normalized variations in efficiency with blade-to-jet speed ratio from the data in reference 10 were used.

An iterative calculation determined the mass-flow rates for the assigned independent parameters and fixed-level of engine output. Shaft rotational speed resulted from the specified compressor specific speed. Compressor-tip diameter was estimated by assuming an inlet critical velocity ratio of 0.3 and using the head coefficients of reference 8. Without a limiting tip speed, turbine diameter was calculated using the optimum blade-to-jet speed ratios and exit-loss factors of reference 11. With a limiting tip speed, the exit-loss factors were increased from the data of reference 10.

Gas-turbine off design. - Off-design gas turbine calculations were made for 5 percent step decreases from the design engine speed. Engine turbomachinery matching was such that either the design turbine-inlet temperature or an assumed limiting regenerator hot-inlet temperature was met within ±1 percent. Analytical turbine and compressor performance maps, generated by Lewis computer codes, were used. The compressor maps had a design pressure ratio of 4 and assumed 30^{o}-backswept blading. The turbine maps were generated for a 1644 K (2500^{o} F) inlet temperature and a design pressure ratio of 4.25. Scaling techniques were used to normalize these maps to the design conditions of each calculation.

Variations in other component performance and parasitic losses were accounted for in the off-design calculations. Regenerator effectiveness was varied inversely with one plus its cold-side mass flow rate raised to a power. System total pressure losses were lumped together and varied with compressor-inlet mass flow rate raised to a power. The exponents for both the effectiveness and pressure loss variations were based on the work of reference 7. Bearing losses were varied proportional to shaft-speed squared. Engine accessory load and reduction gear loss were varied linearly with output power. Table I also shows some typical idle-power values for these losses. All leakage flows were distributed throughout the engine. Flow areas were calculated for the design pressure ratios and conditions in each flow path. Off-design leakage flow rates were calculated based on the flow path area and prevailing temperature and pressure. At idle conditions, the regenerator and compressor seal leakages, expressed in percent of inlet mass-flow rate were about 50 percent larger than their design values (fig. 2).

Engine/vehicle driving cycle. - Output from the engine analysis program was used as input to the driving-cycle computer program. Specifically, the calculated variation of engine output net power and net specific fuel consumption (lb/hr/hp) with engine speed was required by the driving-cycle program. Then for a set of assigned characteristics, this program calculated Composite Driving Cycle fuel economies.

Efficiency characteristics for the assumed CVT were approximated from the results of reference 12. The approximation is shown in figure 5. Below a vehicle speed of

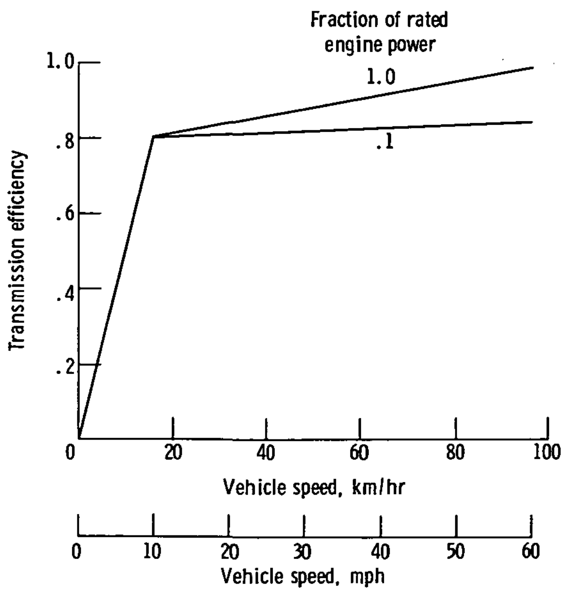

Figure 5. - Approximation of hydromechanical transmission efficiency.

16 kilometers per hour (10 mph) efficiency was independent of engine output power. Above 16 kilometers per hour (10 mph), the slope of the linearized approximation was varied directly with the fraction of rated engine power.

Various vehicle constants which were assumed in the analysis are listed in table II. The vehicle constants are typical of current compact cars. The car accessory load was varied slightly with engine speed. At 50 percent speed, the car accessory load was 1.3 kilowatts (1.7 hp).

TABLE II. - ENGINE/VEHICLE CHARACTERISTICS ASSIGNED FOR DRIVING-CYCLE ANALYSIS

Vehicle test weight, kg (lb)	1588 (3500)
Effective frontal area (drag coefficient x frontal area), m^2 (ft^2)	0.9 (10)
Wheel radius, m (ft)	0.3 (1)
Wheel moment of inertia, kg-m^2 (lbm-ft^2)	16.1 (383)
Engine moment of inertia, kg-cm^2 (lbm-in.^{2})	1.5 (0.5)
Rear-axle gear ratio	3.08
Vehicle accessory load (at design engine speed), kW (hp)	1.4 (1.8)

Fuel	Gasoline	Diesel oil
Density, kg/liter (lb/gal)	0.752 (6.28)	0.839 (7.00)
Lower heating value, J/g (Btu/lb)	43 200 (18 600)	42 370 (18 230)
Energy density, J/liter (Btu/gal)	3.250×10^7 (116 800)	3.5555×10^7 (127 600)

The driving cycle computer program included calculations for fuel consumed during both engine and vehicle accelerations. Idle fuel-flow rates were used during decelerations. At each 1-second increment in the driving cycles, the horsepower required at the car wheels was calculated. This power was the sum of that required to overcome air drag, rolling resistance, and car acceleration. Required engine power then was the sum of the wheel horsepower, power losses in the rear axle and transmission, and power for engine acceleration and car accessories.

For comparative purposes, a performance map for a midsize (130 kW or 180 hp) spark-ignition engine, supplied by the ERDA staff, was used in the analysis. The design-power condition on this map was normalized to 74.6 kilowatts (100 hp). And, because of the assumed CVT, the operating line on the piston-engine map was through the locus of least values of specific fuel consumption with engine speed.

RESULTS

Results are presented first for the gas-turbine designs which result in the best fuel economy. The effects of temperature and tip-speed constraints on fuel economy are presented following the best fuel-economy results. Constrainted solutions are presented only for the design regenerator effectiveness of 0.90.

All calculations were based on the use of gasoline on an 29° C (85° F) day at sea level. Fuel economy would improve, of course, if the denser diesel fuel were used.

Best Fuel Economy Designs

Most results are presented for a design regenerator effectiveness of 0.90. The resulting regenerator size is felt to be compatible with a standard compact engine compartment (see Assumptions and Limitations). Comparative results are also presented for the effects of a larger regenerator, 0.94 effectiveness, on fuel economy.

Regenerator effectiveness, 0.90. - Figure 6 shows the overall results. The variation of fuel economy with design compressor pressure ratio is plotted for the three turbine-inlet temperatures. Each curve was determined from calculations at four values of design compressor pressure ratio with a step size of 0.5. For example, the 1644 K (2500° F) curve used data for design compressor pressure ratios of 4.0, 4.5, 5.0, and 5.5. Over the calculated ranges in pressure ratio, the curves were fairly flat. How-

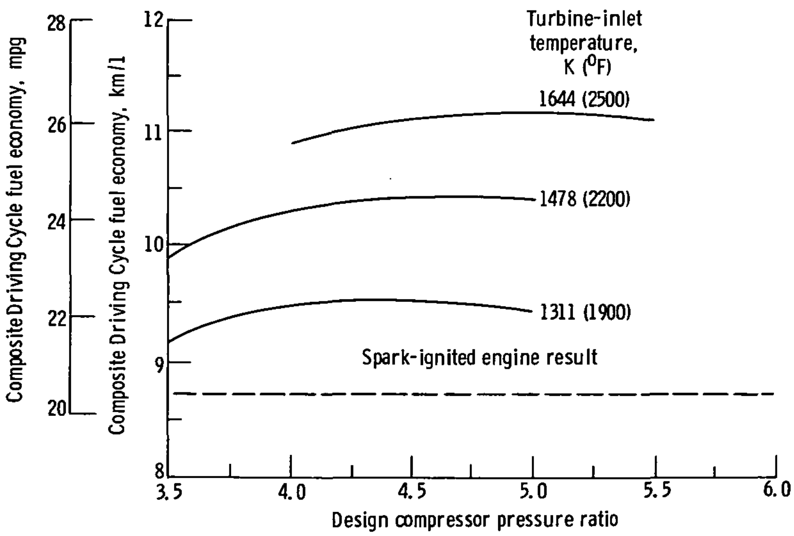

Figure 6. - Variation in fuel economy with design pressure ratio and for constant turbine-inlet temperature operation. Design regenerator effectiveness, 0.90.

ever, there was a value of design pressure ratio for best fuel economy at each temperature. The best design pressure ratio varied from about 4.4 at 1311 K (1900° F) to about 5 at 1644 K (2500° F). Highest fuel economies were: 9.5 kilometers per liter (22.3 mpg) at 1311 K (1900° F), 10.4 kilometers per liter (24.5 mpg) at 1478 K (2200° F), and 11.1 kilometers per liter (26.2 mpg) at 1644 K (2500° F).

The piston-engine analysis resulted in a fuel economy of 8.7 kilometers per liter (20.4 mpg). This level is shown in figure 6 for reference. All of the gas-turbine fuel economies in figure 6 exceeded the piston-engine level. The peak gas-turbine fuel economies were higher than the piston-engine economy by 9 percent at 1311 K (1900° F), by 20 percent at 1478 K (2200° F) and by 28 percent at 1644 K (2500° F).

Further gas-turbine design characteristics for peak fuel economy conditions are presented in figures 7 and 8. Figure 7 shows the calculated variations of specific fuel consumption with engine output power for the gas turbines. For reference, the piston engine curve, used in the driving cycle analysis, is also shown. The shape of the piston-engine curve is flattened somewhat because of the CVT. The gas-turbine curves have similar shapes and show a nonlinear decrease in specific fuel consumption with increasing temperature. Minima in specific fuel consumption occurred at about 45 kilowatts (60 hp) for the gas turbines, and varied from 0.25 gram per hour per watt (0.41 lb/hr/hp) at 1311 K (1900° F), to 0.22 gram per hour per watt (0.37 lb/hr/hp) at 1478 K (2200° F) to 0.21 gram per hour per watt (0.35 lb/hr/hp) at 1644 K (2500° F). In comparison the least specific fuel consumption for the piston engine occurred at about 23 kilowatts (30 hp) and was 0.29 gram per hour per watt (0.47 lb/hr/hp).

Figure 8 shows plots of the gas-turbine design parameters for best fuel economy

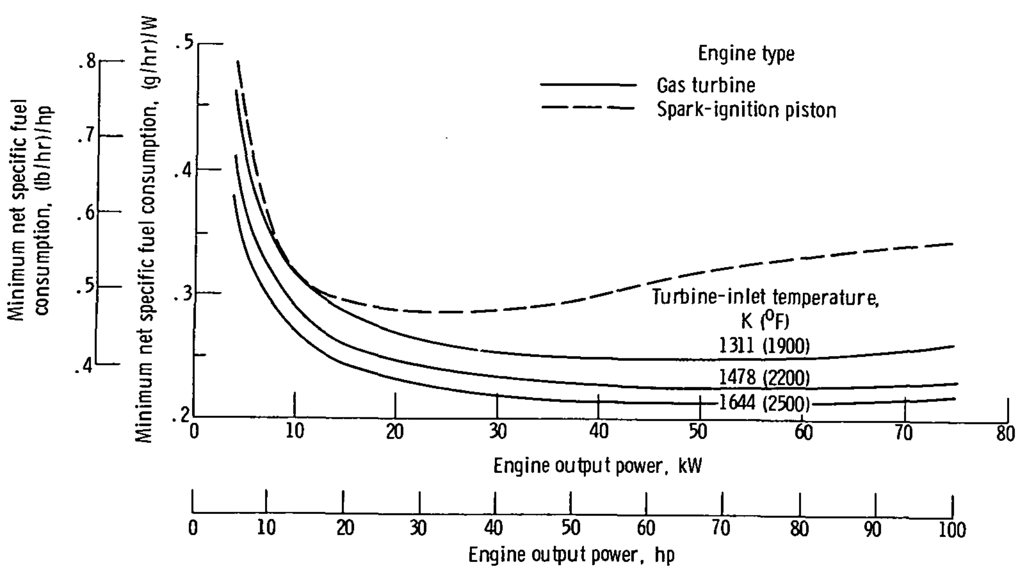

Figure 7. - Design and off-design engine fuel consumption for best fuel economy. Gas-turbine operation at constant turbine-inlet temperature; design regenerator effectiveness, 0.90.

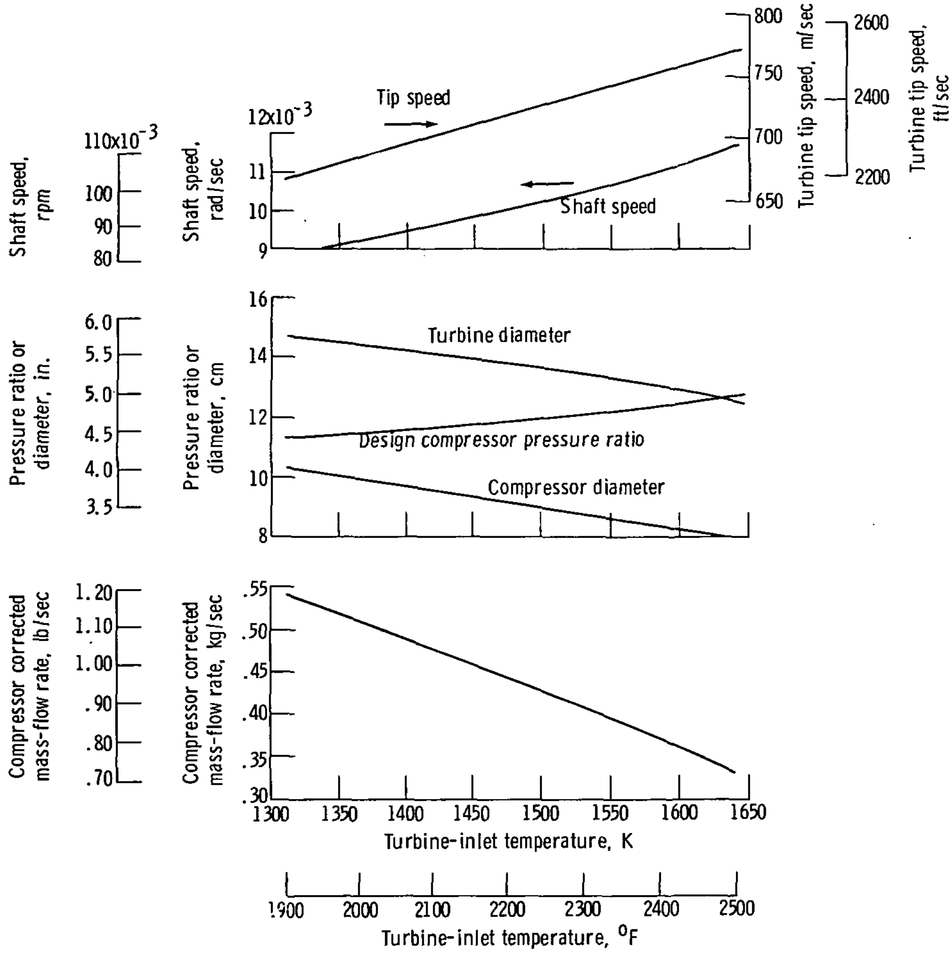

Figure 8. - Gas turbine design parameters for best fuel economy. Full-power regenerator effectiveness, 0.90.

against turbine-inlet temperature. Compressor-inlet mass-flow rate decreased with increasing operating temperatures mainly because of increasing specific power and the fixed output power level. Both turbine- and compressor-tip diameters decreased with higher temperatures, while required shaft speed increased. These trends were mainly due to decreasing volume flow rates. Resulting turbine-tip speeds increased from about 670 meters per second (2200 ft/sec) at 1311 K (1900° F), to 710 meters per second (2320 ft/sec) at 1478 K (2200° F), to 770 meters per second (2530 ft/sec) at 1644 K (2500° F). These high tip speeds imply high blade and disk working stresses. Whether or not ceramics can be adapted to such tip speeds cannot be fully answered at this time.

The ratio of stress to density in a turbine material is proportional to tip-speed squared. Current aircraft superalloys have a lower allowable stress-to-density ratio than those of small, well-formed ceramic samples. And, working turbine tip-speeds higher than those of superalloys might be expected with ceramics. Such a first-order comparison would allow ceramic tip-speeds on the order of 750 meters per second (2500 ft/sec). However, allowable ceramic stresses are also a function of fal cation flaw size and must be treated on a statistical basis. With development, cer: ic turbines may approach tip speeds indicated by allowable stress-to-density ratios; bu early ceramic turbines will probably be designed for more conservative tip speeds. Very similar fuel economies could also be obtained at lower tip speeds by use of a two-stage turbine although with some sacrifices in engine complexity and cost.

Values of design-point compressor pressure ratios are also cross-plotted with temperature in figure 8. These values gave peak fuel economy for vehicle use over the Composite Driving Cycle. Thermodynamic calculations showed that these pressure ratios were higher than those which would give the highest efficiency at design-point operation of the engine/vehicle. And, at idle-power operation, the resulting pressure ratios were lower than the optimum thermodynamic pressure ratios. Hence, the evaluation process resulted in the selection of a design compressor pressure ratio which gave thermodynamically optimum cycle efficiency at some off-design, midpower conditions.

Regenerator effectiveness, 0.94. - The 0.94 design-effectiveness-level was somewhat arbitrarily chosen. However, other internal Lewis studies have shown that the change from 0.90 to 0.94 with the same core geometry and the same relative pressure drop would almost double the volume and mass of a regenerator. Whether or not such a regenerator is practical was beyond the scope of this study.

Comparative effects of the larger regenerator are shown in figure 9. These re-

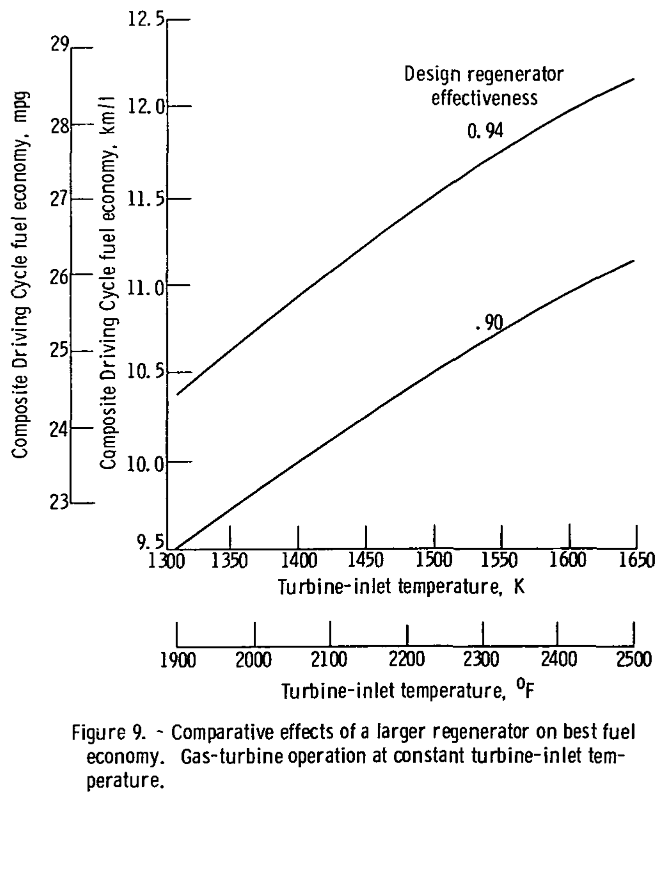

Figure 9. - Comparative effects of a larger regenerator on best fuel economy. Gas-turbine operation at constant turbine-inlet temperature.

sults are from the best fuel economy designs at each of the three turbine-inlet temperatures. Over the temperature range, the fuel economy improvement with the higher design effectiveness was nearby constant at about 1 kilometer per liter (2 mpg). At 1644 K (2500° F) with 0.94 effectiveness, the fuel economy was 12.2 kilometers per liter (28.6 mpg). Compared to the piston engine result, this would be a 40 percent improvement in fuel economy.

Constrained Gas-Turbine Designs

Results are presented for the separate effects of turbine-tip speed limits and two regenerator hot-inlet temperature limits on fuel economy. Tip-speed-limited results are presented first.

Lower design turbine-tip speeds. - Two methods of reducing the tip speed of the single-stage turbine were examined. The major method investigated here was to directly cut back on turbine-tip diameter while maintaining rotational speed. This method results in nonoptimum turbine blade-to-jet speed ratios and, hence, lower levels of design and off-design turbine efficiency. To maintain work while reducing rotor tip diameter, requires backsweeping the rotor entrance, increasing rotor-exit swirl, or some combination of these effects. Details of the best approach were not investigated.

Results of cutting back on tip diameter are shown in figure 10. Fuel economy is plotted gainst turbine-tip speed for operation at the three values of constant turbine-inlet temperature. The piston-engine fuel economy is also shown for comparisons. Most fuel

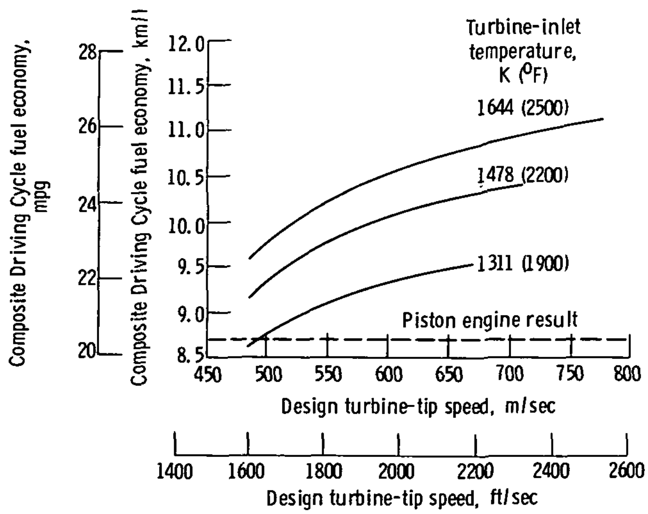

Figure 10. - Effect of reduced turbine diameter on design tip speed and fuel economy. Gas-turbine operation at constant turbine-inlet temperature; regenerator effectiveness, 0.90.

economies of the gas-turbine solutions with constrained turbine-tip speeds still exceeded that of the piston engine. However, the degree of improvement over the piston engine depends on both the maximum tip-speed and operating temperature which can be achieved If the design turbine-tip speed is limited to 488 miles per second (1600 ft/sec), operation at 1311 K (1900° F) would show no gas-turbine advantage; operation at 1478 K (2200° F) would result in 9.1 kilometers per liter (21.5 mpg) and a 5-percent improvement; and, at 1644 K (2500° F) the fuel economy would be 10.2 kilometers per liter (22.5 mpg) for an 11-percent improvement. At 1644 K (2500° F) with a 610-meter-per-second (2000-ft/sec) design tip speed, gas-turbine fuel economy would be 10.6 kilometers per liter (24.9 mpg) for a 22-percent improvement over the piston engine with the advanced CVT.

The other method examined for lower design tip speeds, was to accept design pressure ratios less than those which resulted in peak fuel economy. Reducing the design pressure ratio reduces the turbine jet speed (the velocity which would be obtained by isentropically expanding the flow to exit static pressure). Since the optimum blade-to-jet speed ratio is nearly constant, the turbine-tip speed is also reduced. Figure 11 shows the effects of design pressure ratio on turbine-tip speed for the three turbine-inlet temperatures. The locus of pressure ratios for peak-fuel-economy values (from fig. 6) is also shown. The rate of change in tip speed with design compressor pressure ratio is gradual; a decrease of 1.0 from the optimum pressure ratio resulted in a tip-speed decrease of about 46 meters per second (150 ft/sec). Effects of reduced pressure ratio on fuel economy can be found from figure 6. However, for identical tip-speed reductions, the fuel economy penalties with cutting back on diameter were about one-half those with a reduction from optimum pressure ratio.

An effective way of reducing turbine diameter and tip speed is to use more than a

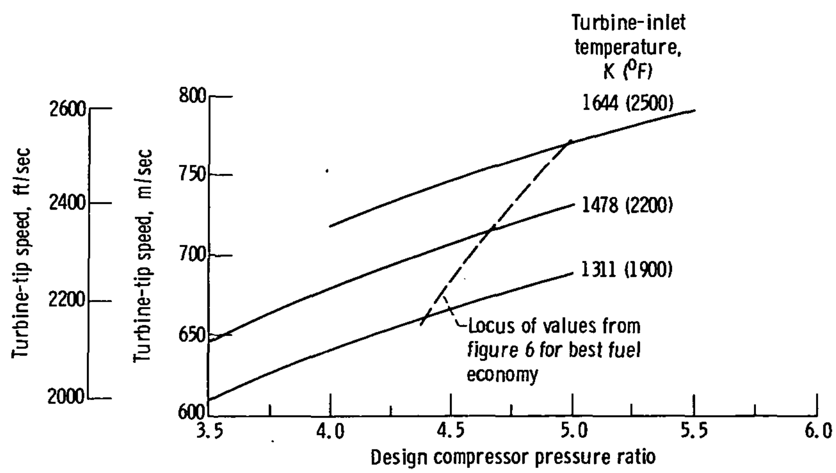

Figure 11. - Design-point variation of turbine-tip speed with temperature and compressor pressure ratio. Full-power regenerator effectiveness, 0.90.

one-stage turbine. Multistaging reduces specific work per stage. And, two or more turbine stages might allow a higher overall turbine efficiency. However, it also would add complexity, weight, and probably cost. Calculations for a 1644 K (2500° F) design with a compressor pressure ratio of 5 showed that a two-stage turbine, would lower the required turbine-tip speed from 770 meters per second (2530 ft/sec) to about 530 meters per second (1750 ft/sec) at the same shaft speed.

Off-design regenerator temperature limits. - Current experimental gas-turbine developments (ref. 7) are designed to operate with a constant turbine-outlet temperature (or a constant regenerator hot-inlet temperature). Such operation may be desirable with a conventional transmission to allow for engine acceleration without exceeding some allowable turbine-inlet temperature. It may also be desirable, however, to keep the regenerator operating temperature below a prescribed limit. For example, the 1644 K (2500° F), constant turbine-inlet temperature operation resulted in a regenerator hot-inlet temperature of 1250 K (1790° F) at full power. At idle power this temperature rose to about 1480 K (2200° F). Results for both constant-turbine-inlet and constant-turbine-outlet operation are shown in figure 12. Effects of an intermediate type of temperature limit are also shown by the dashed curve in figure 12. This operation is with the off-design, hot-inlet regenerator temperature, restricted to no more than 278 K (500° F) below the design turbine-inlet temperature.

At a design turbine-inlet temperature of 1644 K (2500° F), operation at a constant turbine-outlet temperature (1250 K or 1790° F) resulted in a fuel economy of 10.6 kilometers per liter (25.0 mpg). With a 1366 K (2000° F), off-design regenerator

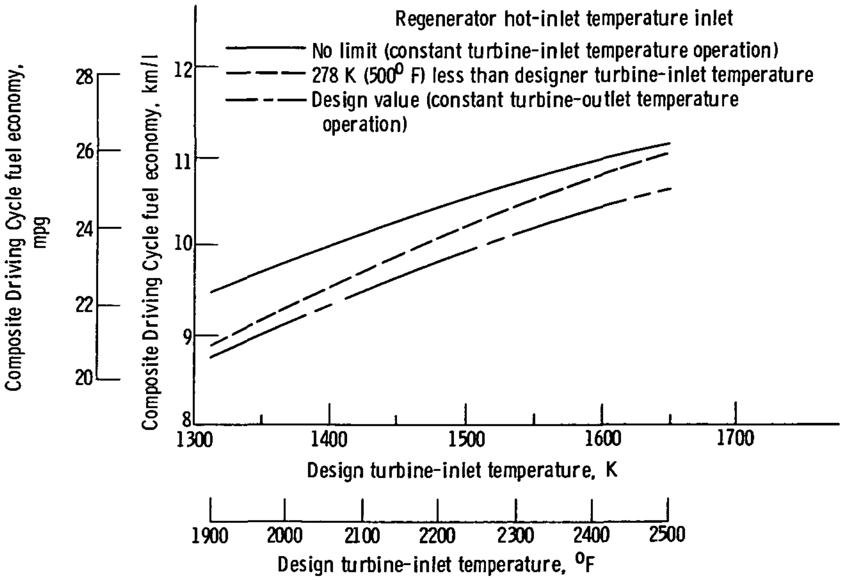

Figure 12. - Effect of off-design regenerator temperature limits on fuel economy. Full-power regenerator effectiveness, 0.90.

temperature limit and a design turbine-inlet temperature of 1644 K (2500° F), fuel economy was 11.1 kilometers per liter (26.0 mpg). Effects of both types of off-design limiting temperatures on fuel economy increased with decreasing design turbine-inlet temperature.

CONCLUDING REMARKS

This study examined the fuel economies which might be obtained with a particular arrangement of an advanced gas-turbine engine and transmission in a compact car. Since performance depends on the operational temperatures and turbine-tip speeds which can be obtained with development, a spectrum of designs with and without operational limits were analyzed. And, even with some operational limits, an advanced ceramic engine appears to offer fuel economy advantages over a piston engine using an advanced transmission.

Obviously, this study did not consider all potential ceramic engine/vehicle design options. And, further gains in advanced gas-turbine/vehicle fuel economy may be attainable.

One important design option that was not exercised here would be the adjustment of design power and vehicle mass in the comparison of the gas-turbine with the piston engine. Such adjustments could have a first-order effect on the relative fuel economies. Estimations by the authors based on the methods of reference 2 showed that the approximate 30-percent improvement shown here for unlimited ceramic gas turbine would be more like a 40-percent improvement with mass and power-level adjustments.

Other gas-turbine design options needing further study include: (1) use of variable turbomachinery stators; (2) use of water injection for power augmentation; (3) use of multishaft engine types, perhaps with power transfer among them; and (4) designing for part-power operation. Each of these, taken one at a time, is believed by the authors to have second-order effects on attainable fuel economy. Taken together, however, they could become first-order effects and further increase fuel economy. Granted, most of these options would also add complexity and cost to the engine design. And, it should be noted that variable turbomachinery stators may be required to obtain adequate vehicle response.

One further option is that of maximum temperature. Operating temperatures higher than 1644 K (2500° F), which was taken here a study limit, may be attainable with ceramics. Silicon carbide is known to decompose before 1922 K (3000° F), but silicon nitride does not.

The overall point is that there are still several options for further fuel economy gains that will require a continuing evaluation.

SUMMARY OF RESULTS

The purpose of this analytical study was to take a first look at the fuel economy potentials of a ceramic gas-turbine engine in a compact car. The engine was a fixed-geometry, regenerated gas turbine operating with a maximum constant turbine-inlet temperature of 1644 K (2500° F). The gas-turbine incorporated a single-shaft with a single-stage radial-flow turbine and compressor, and a rotary regenerator. A design regenerator heat-transfer effectiveness of 0.90 was used. The engine was mated to an advanced continuously variable speed-ratio transmission. Most engine/vehicle losses were assigned values typical of current practice. And, fuel economies were calculated over the Composite Driving Cycle using gasoline as the fuel on an 29° C (85° F) day at sea level.

For comparisons, a spark-ignited piston engine was also evaluated over the driving cycle. The same advanced transmission was used. And, both the gas-turbines and the piston engine were sized for 74.6 kilowatts (100 hp) and evaluated for a vehicle test weight of 1488 kilograms (3500 lb).

The performance of advanced ceramic gas turbines will depend on the operational temperatures and stresses which can be attained with development. Hence, a spectrum of designs, both with and without operational limits, were analyzed. The significant results were:

1. With gas-turbine operation at a constant turbine-inlet temperature of 1644 K (2500° F), fuel economy was 10.2 kilometers per liter (22.5 mpg) when design turbine-tip speed was limited to 488 meters per second (1600 ft/sec). Raising the limiting tip speed to 610 meters per second (2000 ft/sec) resulted in 10.6 kilometers per liter (24.9 mpg). And, operation at the best tip speed of 770 meters per second (2530 ft/sec) resulted in 11.1 kilometers per liter (26.2 mpg). The fuel economy calculated for the piston engine using the advanced transmission was 8.7 kilometers per liter (20.4 mpg). Hence, gas-turbine operation at 1644 K (2500° F) showed an 11-percent advantage over the piston engine operation when tip speed was limited to 488 meters per second (1600 ft/sec), 22 percent with a limit of 610 meters per second (2000 ft/sec), and 28 percent with no operational limits. With a two-stage turbine, similar fuel economies could be achieved at lower tip speeds.

2. Design and operation of the gas-turbine at a constant turbine-inlet temperature of 1475 K (2200° F) and with its best design turbine-tip speed resulted in a fuel economy of 10.4 kilometers per liter (24.5 mpg). Similar operation at a constant 1311 K (1900° F) resulted in 9.5 kilometers per liter (22.3 mpg). If the design turbine-tip speed is limited to 488 meters per second (1600 ft/sec), fuel economy at 1311 K (1900° F) would equal that for the piston engine, and at 1475 K (2200° F) the fuel economy would rise to 9.1 kilometers per liter (21.5 mpg).

3. Operation of the gas turbine, designed for 1644 K (2500° F) and its best turbine-tip speed, at a constant turbine-outlet temperature of 1250 K (1790° F) resulted in a fuel

economy of 10.6 kilometers per liter (25.0 mpg). Operation of the same gas-turbine design with its off-design upper limit on turbine-outlet temperature raised to 1366 K (2000° F) resulted in 11.1 kilometers per liter (26.0 mpg).

4. If a larger regenerator, one with a design effectiveness of 0.94 were practical, the fuel economies without tip-speed limits would be increased by about 1 kilometer per liter (2 mpg) at each constant operating temperature. At 1644 K (2500° F) with a 0.94 regenerator effectiveness, fuel economy was 12.2 kilometers per liter (28.6 mpg).

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