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Upgrading Automotive Gas Turbine Technology

An Experimental Evaluation of Improvement Concepts

Peter R. Angell
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Chrysler Corporation

SOCIETY OF AUTOMOTIVE ENGINEERS

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INTRODUCTION

ON NOVEMBER 22, 1972, Chrysler Corporation was awarded a Government contract E(11-1)-2749, through the Advanced Automotive Power Systems Division of the Environmental Protection Agency. Management of the contract was transferred to the Energy Research and Development Administration Division of Transportation Energy Conservation upon formation of the agency in January 1975. The contract covers the development of an automotive gas turbine engine which will meet the current 1978 Federal emission standards of 0.41 HC, 3.4 CO and 0.40 NO_X grams per mile, and have performance, reliability, and potential manufacturing costs comparable to present production powerplants. Overall program duration is 4 years and technological developments from both this government sponsored effort and Chrysler's ongoing gas turbine activity are being used to demonstrate program goals.

The Chrysler Corporation Sixth Generation Gas Turbine has served both as a baseline reference and as a test bed for development of improvements. Baseline vehicles are intermediate size 4 door sedans modified to accept the baseline gas turbine engine. The design and development of this engine and vehicle system, prior to its use for this program, is described in Parts 1 and 2 of this report.

Since the start of the contract the baseline engine has been used to develop and evaluate upgraded engine design concepts such as:

- The free rotor arrangement
- A torque converter lock-up
- Low emissions burners
- An integrated electronic control system
- Ceramic regenerator systems
- Power augmentation
- Linerless insulation
- Low cost turbine wheels etc.

These concept evaluations are discussed in section 3.

A free rotor automotive gas turbine is about to be released under the Government contract which will include many of these concepts. This **Upgraded Engine Design** is discussed fully in SAE Report 760279 by Dr. G. A. Ball, J. I. Gumaer and T. M. Sebestyen. (1)

1. BASELINE ENGINE

The baseline engine which is Chrysler's sixth generation gas turbine powerplant, Figures 1 and 2, was released in January, 1965. The engine was originally targeted for a 3600 lb. vehicle exhibiting fuel economy and performance comparable to vehicles of that era. Principal design goals were: improved rotor response, cost reduction, and capability of driving production type vehicle accessories, specifically air conditioning and power steering from the power turbine. Exhaust emissions, particularly NO_X, were not a consideration in the design of the combustion system although CO and HC were controlled to achieve a burner efficiency approaching 100% with minimum odor, carbon formation, etc. Powerplant weight and geometric configuration were considered on a current basis, i.e., of reasonable mass and to fit an existing vehicle.

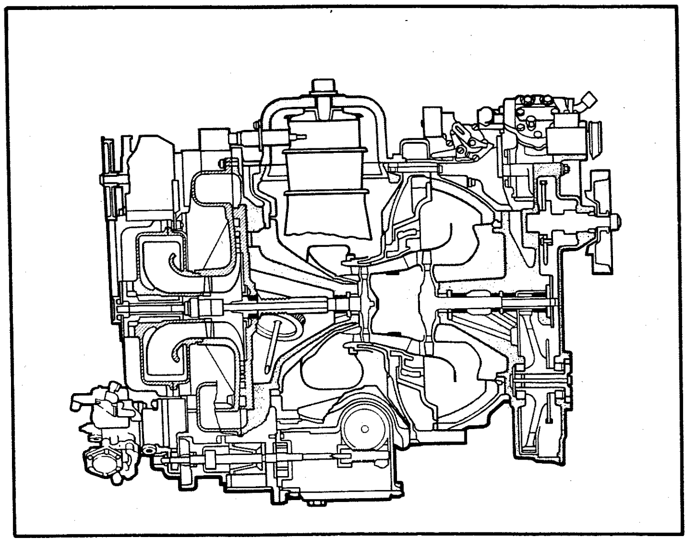


Figure 1

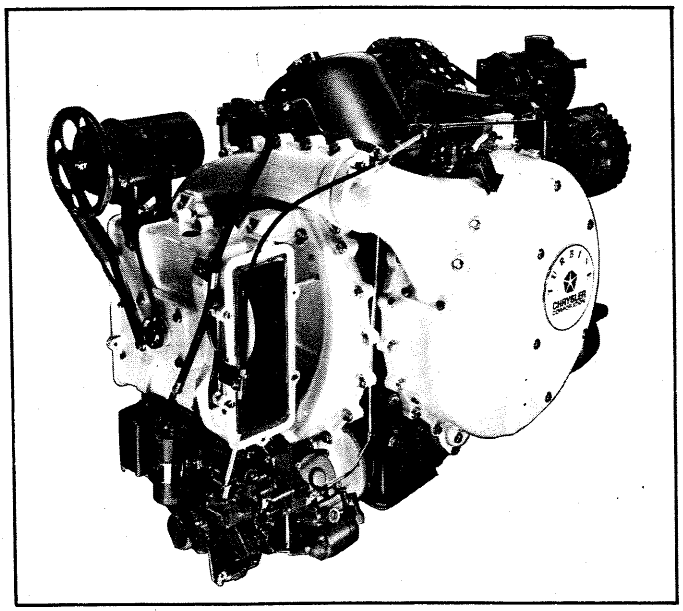


Figure 2 Sixth Generation Gas Turbine

Early development efforts extended into a large number of areas including:

Rotor Dynamics—oil film whirl, stability, noise control.
Gearing—high speed worm, helical reduction, regenerator drive gears, power turbine nozzle mechanism.
Housing—stress levels, deflections, insulation.
Bearings—journal, thrust.
Seals—high speed.
Starting System—belt drive, clutch.
Controls—fuel and variable power turbine nozzle, linkage, start and safety systems.
Engine Auxiliaries—air pump, oil pump, fuel pump.
Compressor—stall/surge margin, reduction of inertia, assembly techniques.
Lube System—oil heat rejection, distribution system, oil types.
Turbine Hardware—stress and inertia reduction, material, coatings, blade vibration, increased cycle temperature.
Regenerator—assembly techniques, seal development, matrix configuration, durability, fabrication techniques.
Combustor—material development, ignitor, ignition system, fuel nozzle, fuel types, distribution, cold starting (−20°F).
Material Development—Chrysler Research alloys, high temperature dry lubricants.
Intake and Exhaust Ducting—minimize losses, control noise.
Vehicle Accessories—air conditioning, heater, power steering and brakes.
Drive Train—automatic clutch, torque converters, axle ratios, part-throttle kick-down.
Durability—endurance test of *total* powerplant.

These efforts culminated in the successful demonstration of a 1966 2 Dr. Dodge Coronet, Figure 3, powered by a reliable automotive type gas turbine powerplant.

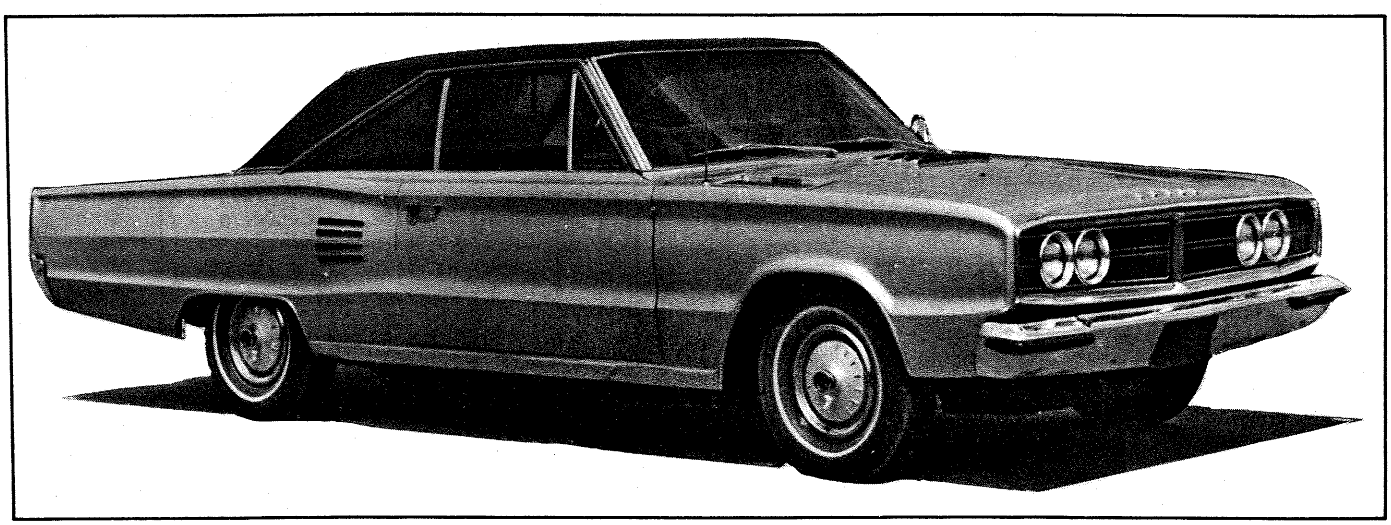


Figure 3 Turbine Powered Dodge Coronet

CHARACTERIZATION AND PERFORMANCE

The sixth generation Chrysler turbine engine is a low pressure ratio, regenerative, free power turbine design. It incorporates a single stage centrifugal compressor, single can type combustor, two axial turbine wheel stages and variable power turbine nozzle blading.

ENGINE CHARACTERIZATION

RPM Comp.	22,877	27,452	32,028	36,603	41,178	43,466	44,610
Press. Ratio Comp., P2/P1	1.545	1.865	2.290	2.825	3.500	3.865	4.080
Eff. Comp. (1-2, Total)	.765	.776	.792	.797	.784	.769	.766
Eff. C.T. (5-6, Total)	.76	.78	.80	.82	.84	.85	.86
Eff. P.T. (6-8, Total)	.67	.67	.68	.685	.69	.695	.70
Eff. Burner	.984	.986	.989	.993	.997	.998	.998
Effectiveness Reg.	.901	.898	.892	.885	.877	.874	.873
△ H Comp., BTU/LB	22.64	32.87	44.12	56.70	71.79	80.17	84.40
△ H C.T., BTU/LB	24.57	35.05	46.59	59.53	75.02	83.68	88.01
△ H P.T., BTU/LB	8.70	15.42	23.77	32.11	40.02	42.84	48.34
HP, Acc. & Loss, C.T.	1.85	2.47	3.33	4.46	5.86	6.70	7.12
HP, Acc. & Loss, P.T.	1.80	2.40	3.50	4.90	5.40	5.90	5.90
HP Net Output, P.T.	9.5	21.4	42.1	71.3	109.9	128.2	150.9
Fuel Flow, LB/HR	12.5	19.2	29.7	44.1	63.1	73.4	81.5
BSFC, LB/HP-HR	1.32	.90	.71	.62	.57	.57	.54
Gas Flow, LB/SEC							
W1	.790	1.010	1.293	1.620	2.001	2.181	2.274
W3	.750	.957	1.224	1.532	1.889	2.059	2.146
W4	.772	.986	1.259	1.575	1.942	2.114	2.204
W5	.781	.997	1.275	1.596	1.970	2.146	2.238
W6	.769	.979	1.253	1.570	1.941	2.116	2.207
W8	.787	1.006	1.288	1.613	1.993	2.172	2.266
W9	.791	1.012	1.296	1.626	2.010	2.192	2.287
Pressures (Total), PSIA							
P1	14.67	14.65	14.63	14.58	14.53	14.49	14.48
P2 (=P3)	22.66	27.33	33.49	41.20	50.84	56.02	59.06
P4	22.48	27.12	33.27	40.97	50.60	55.78	58.81
P5	22.08	26.52	32.49	39.98	49.42	54.42	57.42
P6	17.07	18.69	20.88	23.38	25.98	27.01	28.42
P8	15.03	15.15	15.32	15.55	15.87	16.04	16.16
P9	14.74	14.77	14.82	14.90	15.03	15.10	15.15
Temperatures (Total), °F							
T1	85.	85.	85.	85.	85.	85.	85.
T2	178.0	220.3	266.8	318.6	380.6	414.8	432.0
T4	1145.5	1148.4	1150.0	1152.1	1155.1	1158.0	1198.0
T5	1431.4	1494.6	1568.3	1647.7	1730.8	1772.3	1850.0
T6	1338.0	1362.9	1394.4	1425.6	1453.5	1463.7	1527.5
T8	1290.6	1291.1	1292.2	1293.4	1294.4	1294.6	1339.5
T9	343.3	383.2	429.5	480.6	540.1	571.2	594.9
Flow Leaks LB/LB Into Station							
20	.00204	.00214	.00218	.00224	.00230	.00235	.00238
25	.00689	.00636	.00585	.00539	.00507	.00498	.00496
34	.035						.035
38	.00235	.00290	.00340	.00385	.00425	.00450	.00460
39	.00470	.00580	.00680	.00770	.00850	.00900	.0092
46	.00384	.00408	.00432	.00460	.00493	.00517	.00522
48	.00235	.00290	.00340	.00385	.00425	.00450	.00460
68	.0196	.0214	.0213	.0209	.0197	.0188	.0185
80	.00151	.00156	.00164	.00167	.00173	.00178	.00186
Heat Leaks BTU/LB Flow into Station							
20	.354	.393	.416	.423	.435	.444	.445
40	1.870	1.762	1.553	1.336	1.138	1.055	1.069
50	.631	.980	1.742	2.480	3.085	3.311	3.441
60	.768	.899	1.119	1.663	1.734	1.900	1.882
80	1.989	1.855	1.626	1.394	1.185	1.098	1.110

Figure 4

Pertinent 85°F full power characteristics along with other descriptive data are itemized below:

Power	150 HP @ 3500 rpm
Pressure Ratio	4.1
Airflow	2.29 Lb/Sec
Compressor Speed	44,610 rpm
Turbine Inlet Temperature	
Steady State	1850°F
Acceleration	2000°F
Power Turbine	
Maximum Speed	45,500 rpm
Reduction Gear Ratio	9.6875
Regenerator	
Speed	21 rpm
Matrix Diameter	15.5 in.
Matrix Height	3.5 in.
Stock Thickness	.002 in.
Overall Dimensions	
Length to Transmission Mounting Flange	35.5 in.
Width	27.6 in.
Height	29.9 in.
Weight	
Complete Engine—Dry	600 Lb.
Gas Generator Acceleration, Idle-to-Maximum Speed	1.2 Sec.

Ten of these engines as well as three 1973 intermediate size vehicles were built as baseline hardware for this program. Engine characterization is shown in Figures 4 and 5 and station notation reference on Figure 6. Typical engine performance is shown in Figures 7, 8, 9, 10 and engine exhaust emissions on Figure 11. Vehicle exhaust emission levels, CO and NO_X, were lower than those required to meet the 1975 standards (0.4 gm/mi HC, 3.4 gm/mi CO and 3.1 gm/mi NOx).

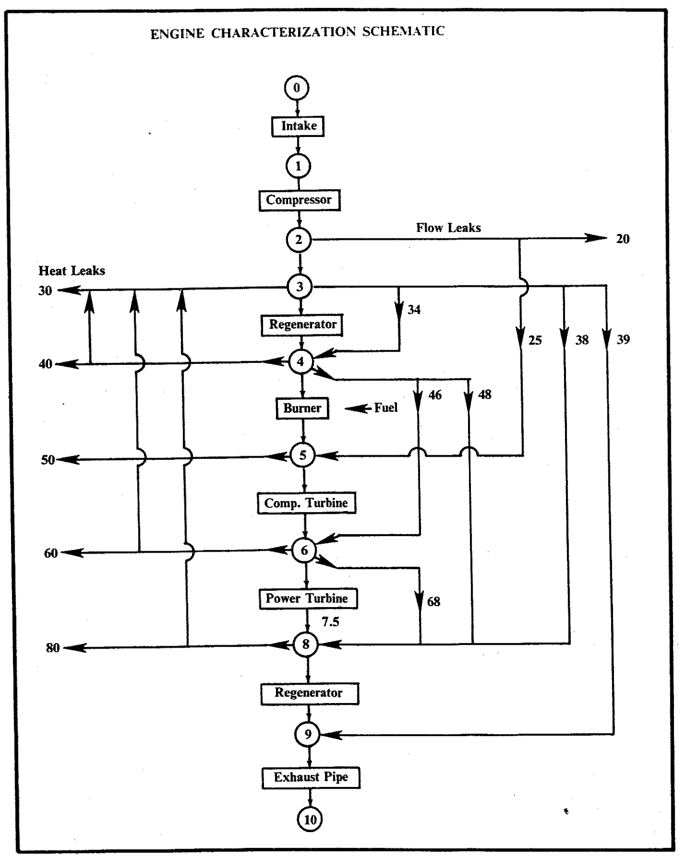


Figure 5

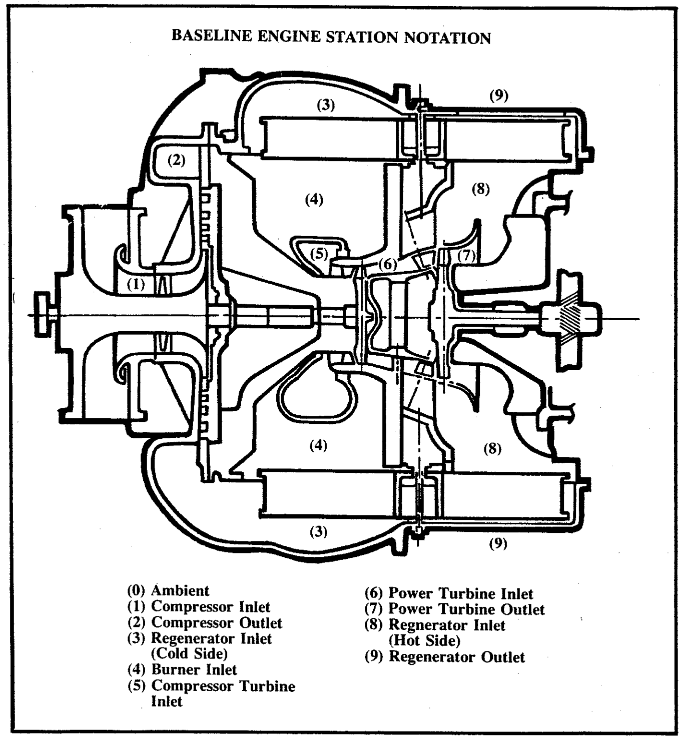


Figure 6

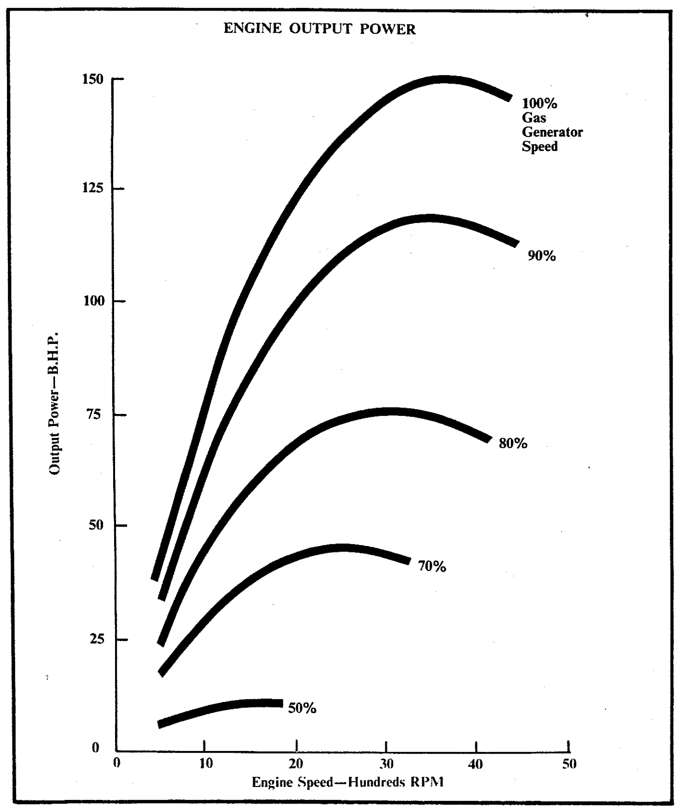


Figure 7

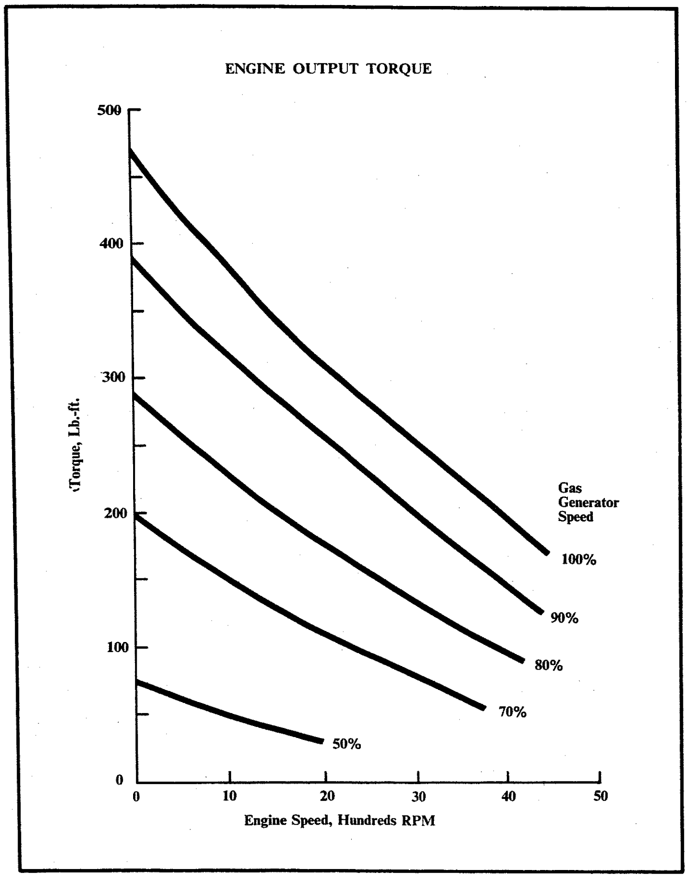


Figure 8

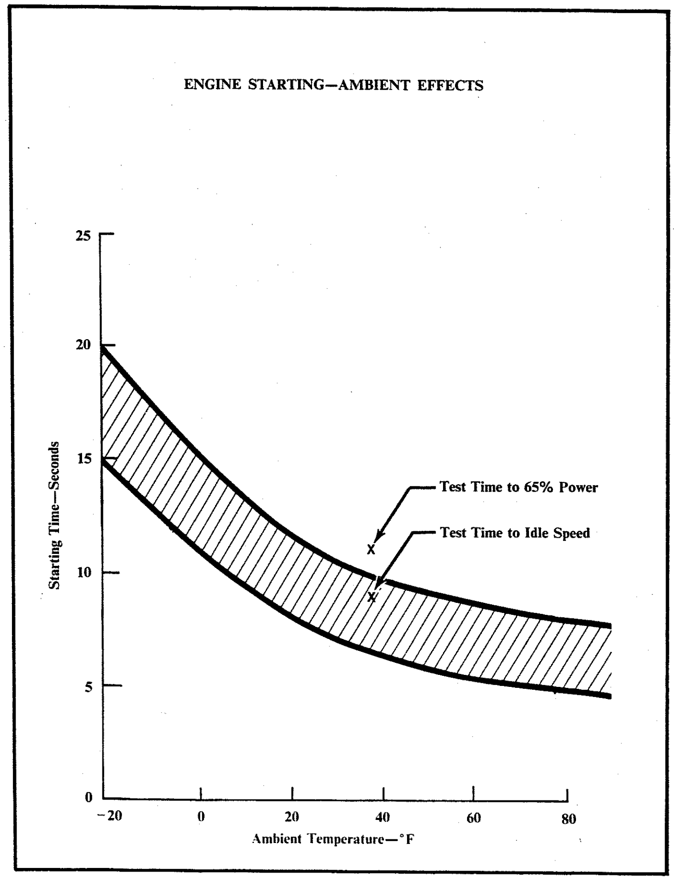


Figure 10

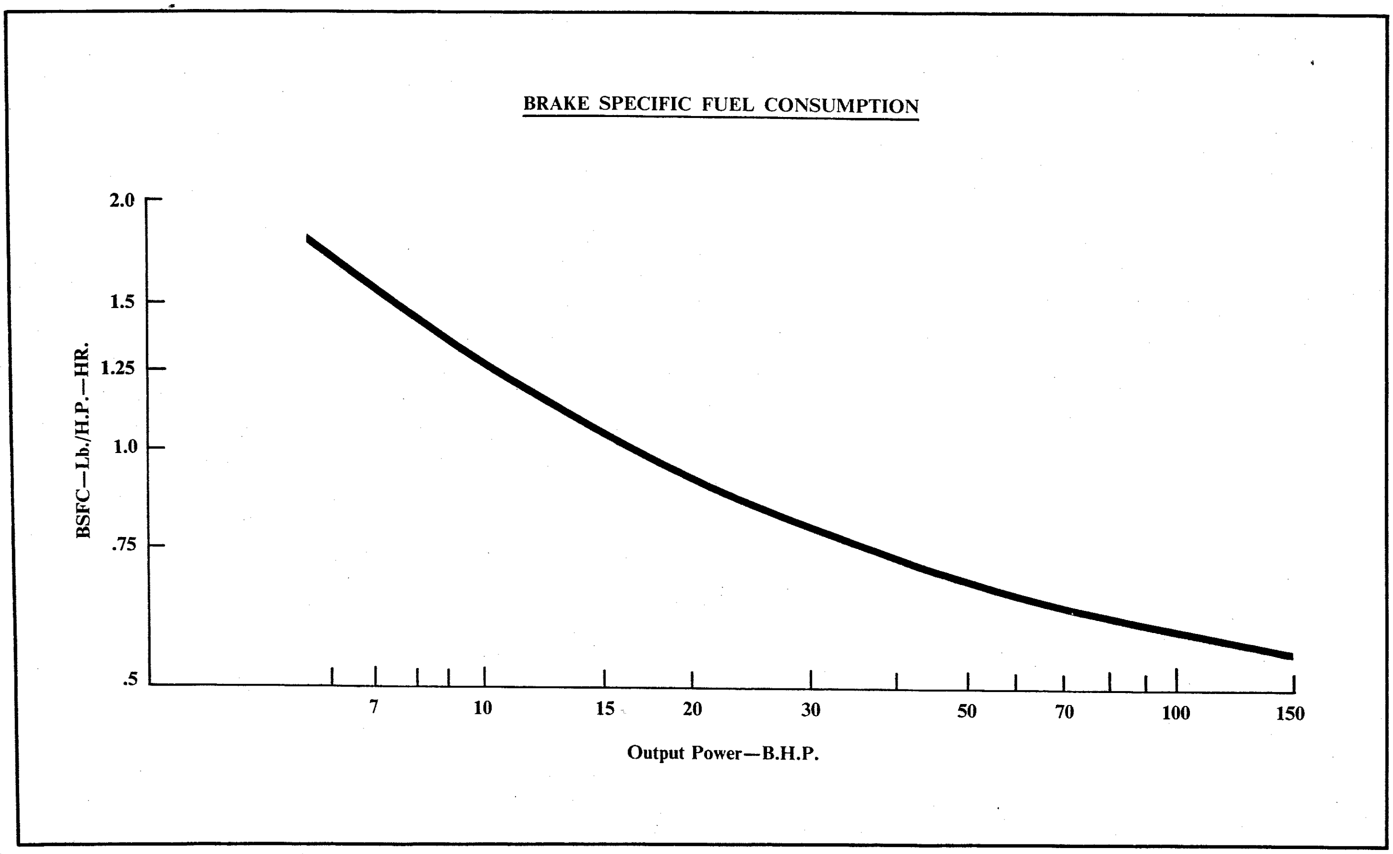


Figure 9

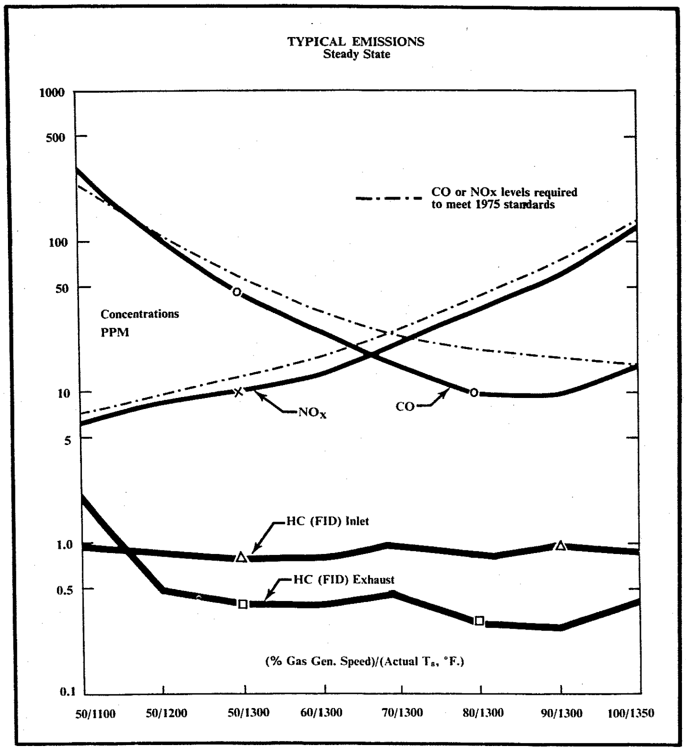


Figure 11

ENGINE DURABILITY

Evaluation of baseline engine durability relied principally on dynamometer cell testing. An endurance cycle was used which was designed to subject turbine components to an accelerated life test through the use of a transient type cycle rather than the steady state mode.

This cycle evolved from years of test experience from the proving grounds, the highways of this country and accumulated knowledge of various test cell endurance schedules. Since steady state operation in a vehicle is the exception rather than the rule, an all-encompassing test cycle including several starts, part and full throttle accelerations and a shutdown period to expose rotor bearings to soak back temperatures was chosen.

The cycle for the baseline engine was of one hour duration and included:

- 9 starts
- 15 wide open throttle accelerations
- 4 part throttle accelerations
- 14.5 minutes total shutdown time (includes 10.5 minute soak period).

This cycle is equivalent to an average vehicle speed of 49 MPH (assuming typical axle ratio, tire size, etc.). An automatic programmer controlled the engine over the one hour cycle. Safeties are provided for overtemperature, overspeed, low oil pressure, no start condition, etc. Strip chart recorders provide a continuous record of events. Typical speed recordings which characterize the cycle are shown in Figure 12. Failed or malfunctioning parts were repaired or replaced as necessary to continue testing.

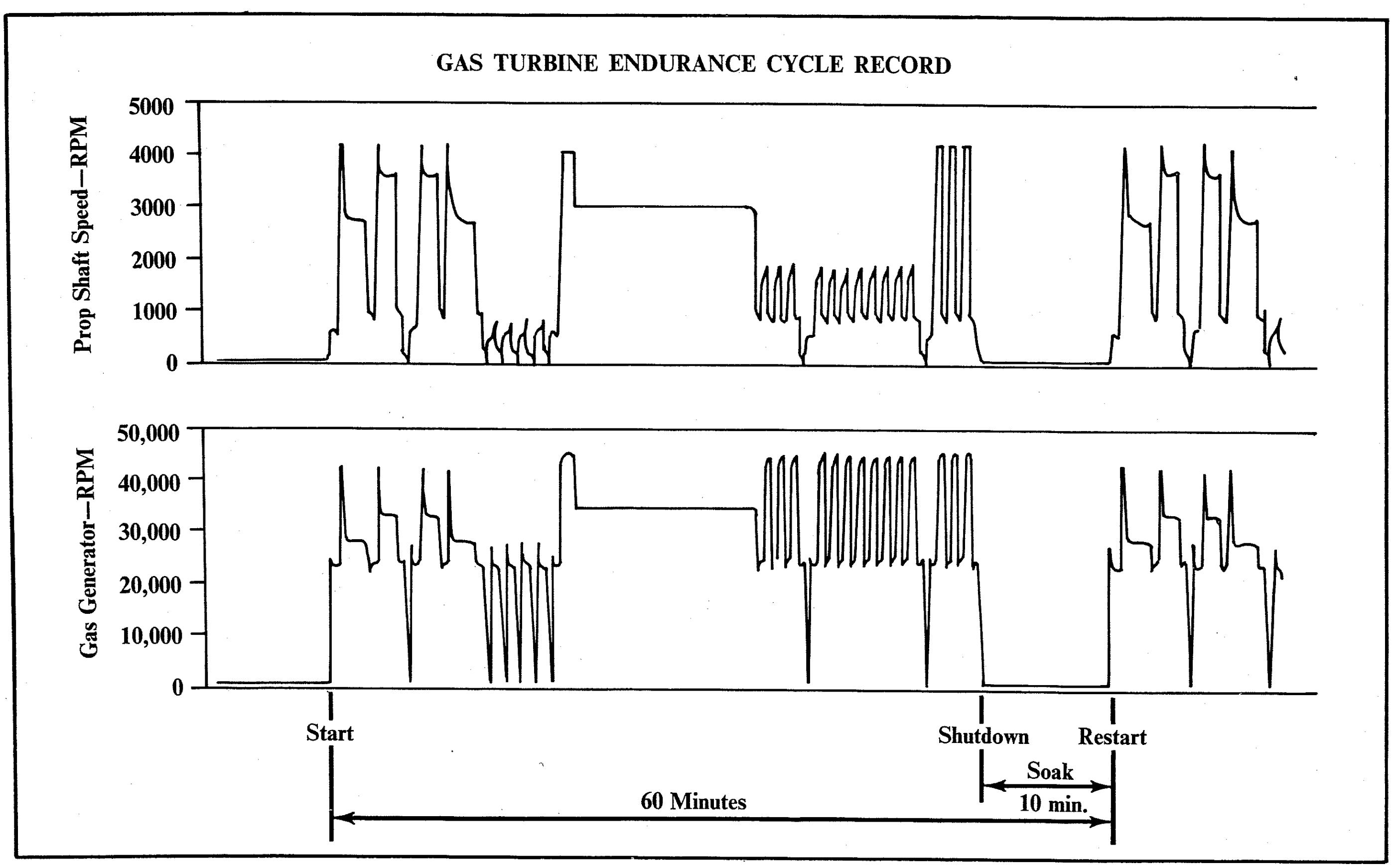


Figure 12

A baseline engine running this cycle was tested for a total of 4653.7 hrs.

Total test time on some of the significant components at final teardown is shown below. Where applicable, also shown are total test hours on components replaced at earlier teardown inspections because of part failure or update as indicated.

	Final Teardown Component		Earlier Components	
	Hours	**Condition**	**Hours**	**Reason for Removal**
Engine Housing	**4653.7**	**Good**		**Not Removed**
Engine Insulation (Linerless)	**820.7**	**Good**	**3828.0**	**Update**
Burner Cap Insulation (Linerless)	**1218.4**	**Good**	**2256.0**	**Update**
Impeller	**4654.1**	**Good**		**Not Removed**
First Stage Nozzle	**4277.0**	**Good**		**Not Removed**
First Stage Wheel	**3970.1**	**Good**		**Not Removed**
Second Stage Wheel	**573.5**	**Good**	**3831.8**	**Disc Cracks**
Total No. of Starts	**40861**			

Final teardown revealed that the most significantly distressed components were located in the burner section although the only non-useable part was the burner tube.

2. BASELINE VEHICLE ENGINE INSTALLATION

Three 1973 intermediate size, 4 door sedans were modified to accept the baseline engine. The modifications included:

- Chassis—Widen front track.
- Modify K member.
- Relocate torsion bar rear anchors, revise rear cross member, modify underbody to provide clearance for exhaust ducts.
- Relocate steering gear and modify linkage, steering column.
- Modify radiator yoke to receive A/C condensor, electric fan, engine and transmission oil coolers.
- Modify front fender sheet metal to accept engine inlet ducting. Install in-tank fuel pump and return line.
- Install *hydraulic* brake booster and accumulator.
- Modify transmission linkage.
- Revise car comfort system—install hot gas/water heat exchanger and air conditioning compressor (axial type).
- Revise wiring harness as required.

Figure 13 shows a cross section and Figure 14 a fully assembled vehicle.

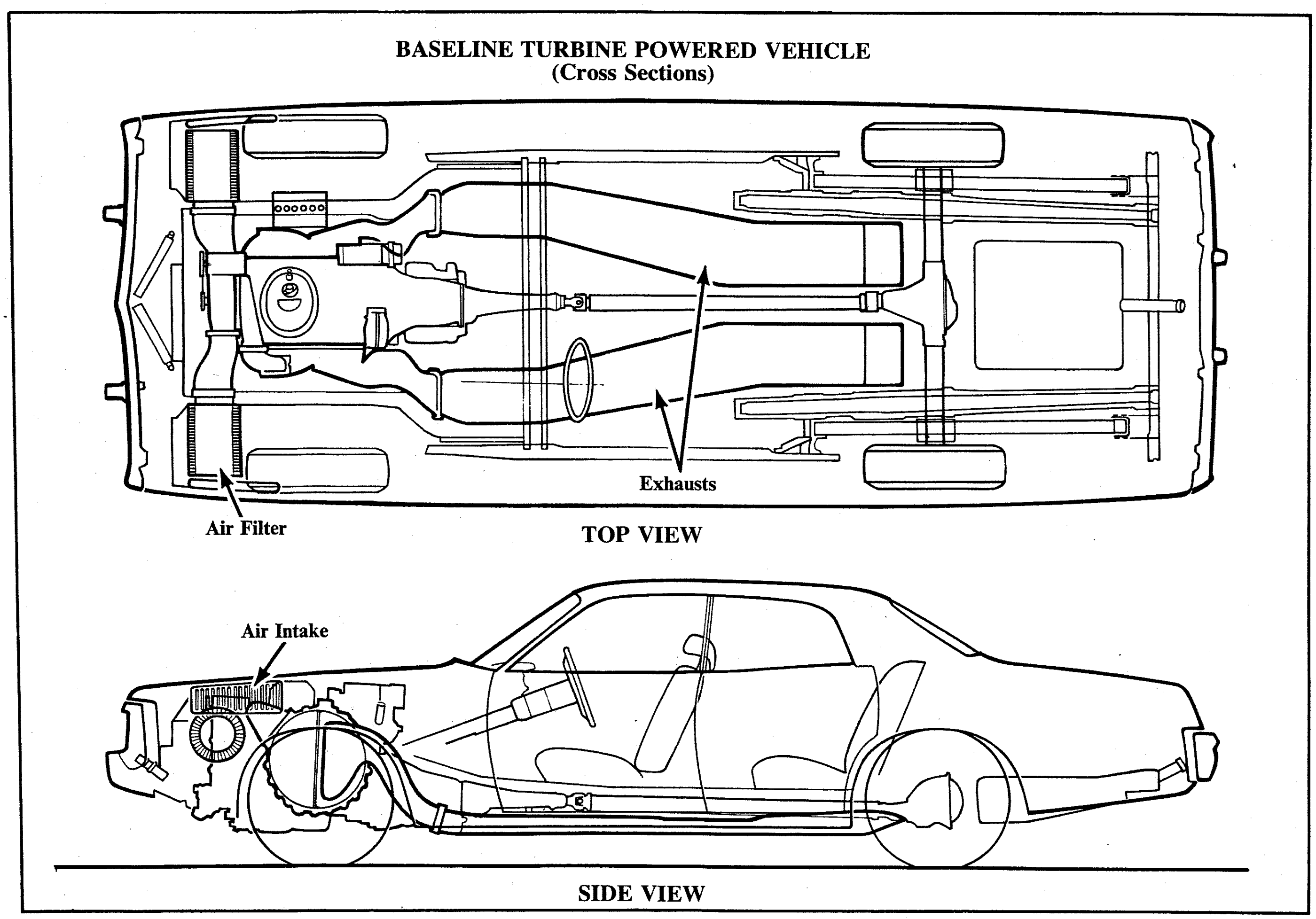


Figure 13

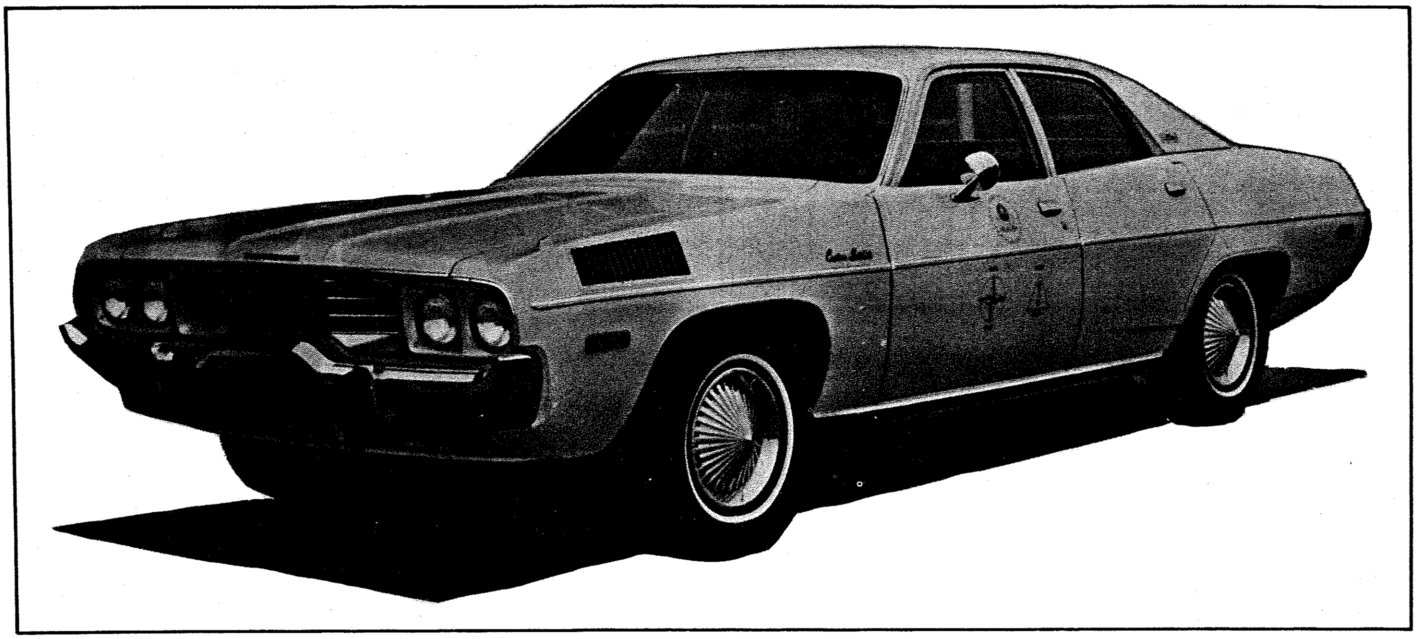


Figure 14 Baseline Turbine Powered Vehicle

VEHICLE PERFORMANCE

Baseline engine/vehicle performance was documented for several configurations at the proving grounds using established test procedures. Driveability was evaluated during several road trips and ride/drive demonstrations.

Figure 15 shows the uncorrected speed-time and distance trace for two vehicles. Their weights and ambient temperature test conditions are noted. The speed-time values are also shown with and without air conditioning corrected to an 85°F day. Performance was judged satisfactory. Figure 16 shows the engine/vehicle braking capability which compared favorably to the current conventional vehicles. The braking position was fixed by the baseline control system and the full engine braking potential

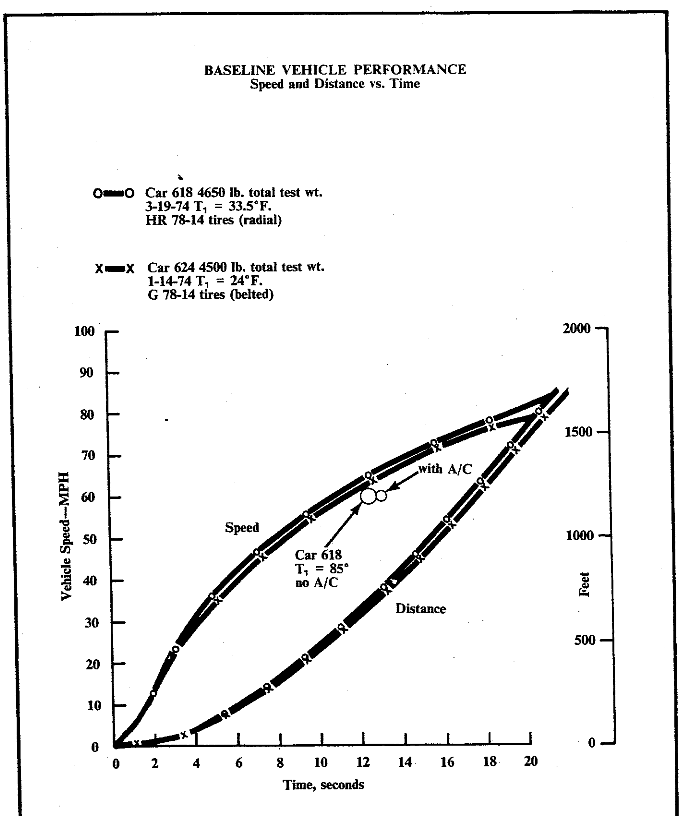


Figure 15

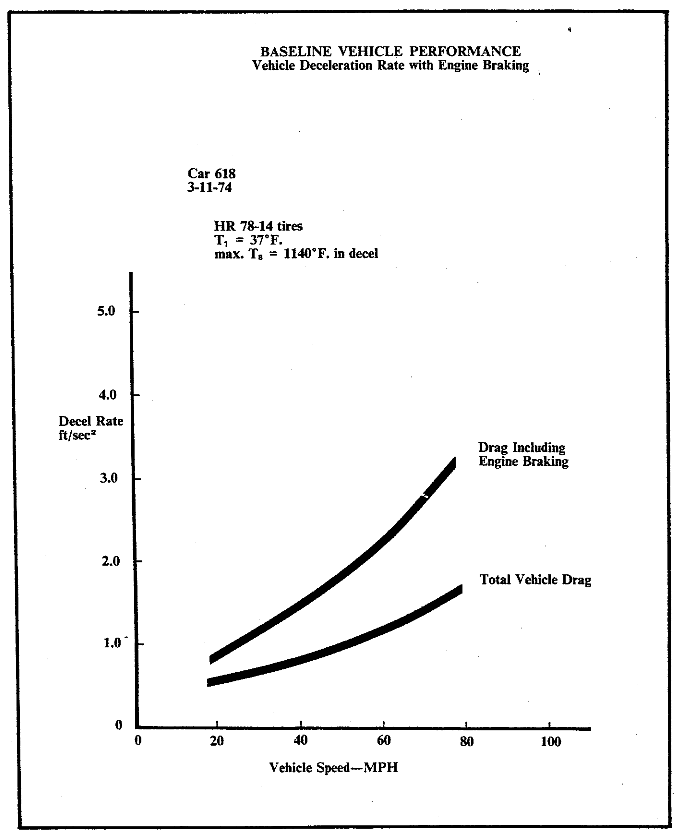


Figure 16

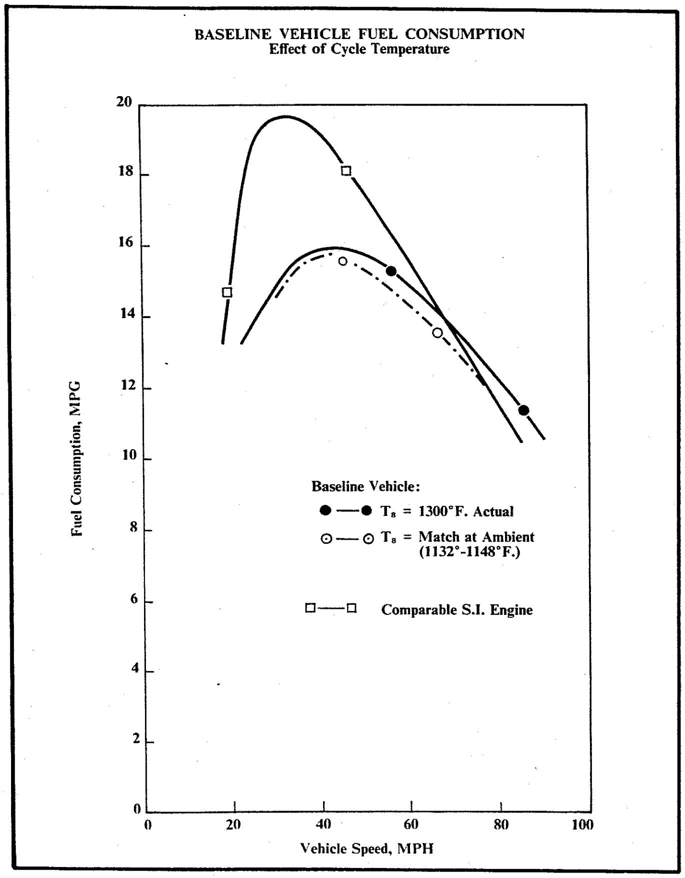


Figure 17

could not be realized due to linkage limitations. Figure 17 shows the fuel economy for varying road load speeds to 90 MPH, at match cycle temperatures of T_8/O = 1300°F and T8 = actual 1300°F for T_1 = 33-38°F. Note that these data were documented using the open loop hydromechanical baseline controls and as a result operating conditions were not optimized. Baseline metal regenerator cores were utilized for these tests. As indicated, fuel economy—particularly at low speed—would be less than with a comparable SI reciprocating engine.

These tests were supplemented by vehicle evaluations at the E.P.A. facility in Ann Arbor, Michigan, and are referenced in Technical Report (2).

The introduction of the integrated electronic control system (closed loop on power turbine exhaust temperature) will optimize vehicle operation in the drive and braking modes and will result in improved fuel economy and reduced emissions as described further in this report.

In summation, the majority of the evaluators who drove the vehicles judged the rotor response as acceptable. Most evaluators were totally satisfied with vehicle noise levels and driveability. Overall comments by technically knowledgeable personnel were favorable.

NOISE CONTROL

Principal noise sources emanating from gas turbine powerplants are attributed to:

1. airborne intake high frequency noise.
2. airborne exhaust noise.
3. rotor noise.
4. high speed gear noise.

Airborne intake noise can be readily attenuated by directing the inlet air through a minimum of two duct bends of maximum allowable size lined with a suitable acoustic material. Exhaust noise is treated in a similar manner. Additionally, as a general rule minimization of noise at the exhaust terminus requires a diffusion section which reduces gas velocity to < 100 ft./sec. at rated power.

Rotor generated noise attributed to rotor imbalance and shaft bending can be negated by isolating the rotor sleeve bearings from the bearing carrier via oil film damping etc. This technique has been successful in the baseline engines. Noise control of high speed gearing required precise manufacture of the involute surfaces and modification of the involute i.e. crowning, surface finish and treatment, e.g. tuff-triding, oil film thickness control etc. Gear tooth generated noise under lightly loaded conditions in drive to braking mode of operation can be controlled by bonding aluminum dampers fabricated of dead soft .06 material to the gear disc with a suitable adhesive.

Baseline vehicle proving grounds tests for compliance with SAE Standard J986a (wide-open throttle acceleration from 30 mph at 100 ft.) indicated sound levels of 75.1 and 75.0 dBA for the right and left sides respectively. By comparison, the EPA Prototype Vehicle Standard is 77 dBA, the legal standard is currently 86 dBA and equivalent reciprocating engines were measured 81-82 dBA. Static noise evaluations of a baseline vehicle and a 1974 conventional vehicle powered by a V8 engine are compared in the following chart.

Comparative Vehicle Noise Levels
dBA

	Baseline Turbine (1973) (Vehicle)	Baseline Turbine (1966) (Vehicle)	1974 Production V-8
Idle—At Front of Car	71	70	66
Idle—At Rear of Car	62.5	64	68
30 MPH—Driver's Ear	60	61	59
60 MPH—Driver's Ear	70	68	72

Further improvements were made to the noise control of a second vehicle and were documented at an independent test facility by EPA personnel.

Discrete frequencies were taken and the required A-weighting applied to arrive at the results in Table 1. Thus, the noise level for the car is 73 decibels, the highest average value recorded.

The data in Table 1 were taken to obtain additional information on the vehicle. Turbine whine was noticeable but not objectionable inside the car between 35 and 55 mph.

Table 1

SOUND TEST RESULTS
SAE J986a Drive-By Test

	Vehicle's Left Side Decibels	Vehicle's Right Side Decibels
Run 1	74	71
Run 2	72	71
Average	73	71

SAE J986a Drive-By Test, Discrete Frequencies

Frequency (Hertz)		
125 Hz	78	76
	80	75
250 Hz	71	72
	71	71
500 Hz	72	71
	73	71
1000 Hz	74	70
	72	68
2000 Hz	64	62
	61	61

SAE J986a Drive-By Test, Discrete Frequencies*

125	82	76
	76	77
	78	
250	73	74
	75	74
500	72	72
	73	72
1000	68	68
	68	67
2000	60	60
	62	60

*Procedure modified: Vehicle accelerated wide open throttle from stop instead of wide open throttle from 30 mph.

INLET AND EXHAUST DUCTING

An important consideration of gas turbine—vehicle installation is to provide adequate engine inlet ducting for the air filter capacity as well as noise control. Inlet restriction as well as exhaust restriction have an adverse and appreciable effect on engine performance. a 1.0 in. H_2O exhaust pressure loss results in a loss of 1 HP, at design speed shown in Figure 18. Restrictions generated in the inlet ducting/filter system manifests itself in a similar loss of performance.

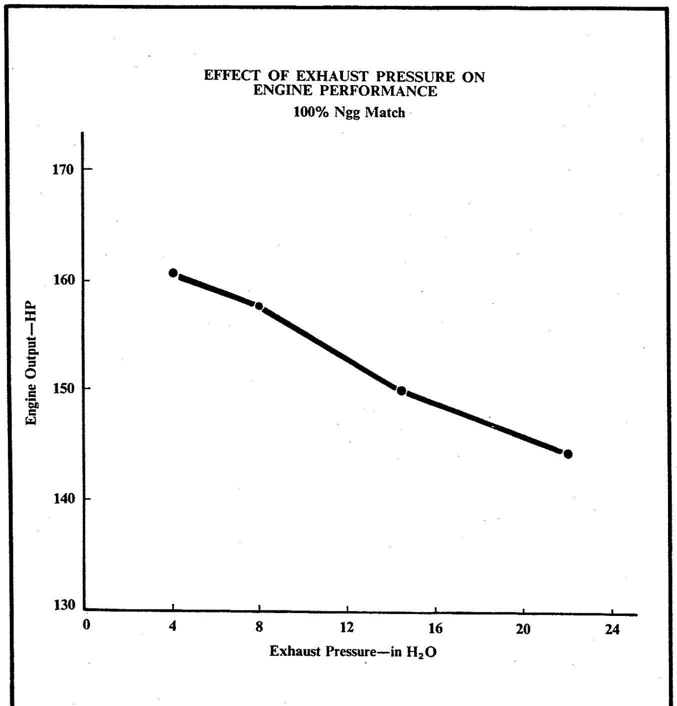


Figure 18

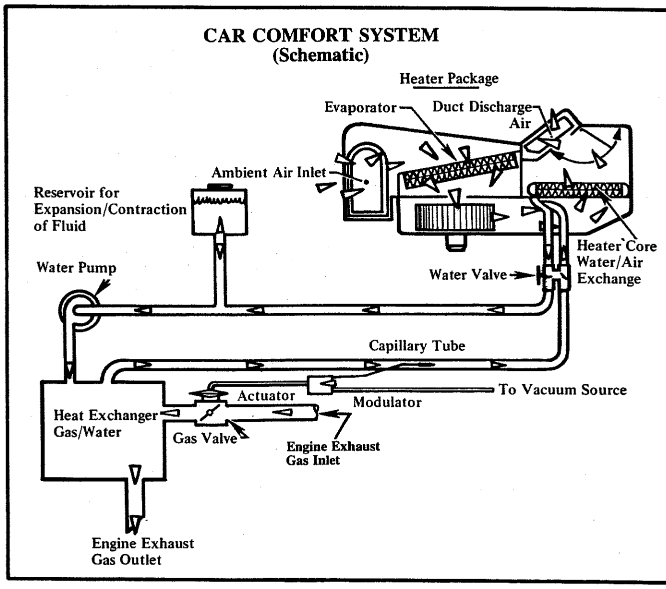


Figure 19

CAR COMFORT SYSTEMS

The use of an intermediate gas-to-water heat exchanger was predicated by two basic factors.

(a) The possible toxic gas effect of exposing Freon 12 to high temperature.

(b) The heat exchanger was readily adaptable to the production configuration of a passenger compartment heating and air conditioning system and associated controls.

In this system, power turbine exhaust gas is passed through a gas-to-liquid heat exchanger containing conventional coolant (50% water/ethylene glycol) which is circulated through the standard vehicle passenger compartment heater core. A 12V DC pump circulates the coolant through both the heat exchanger and the heater core. Thus passenger compartment configuration retains the desirable reheat feature for the air conditioning system and eliminates any possibility of the decomposition of Freon due to contact with a high temperature surface. Figure 19 is a schematic of this system. Production louver controls are actuated from a vacuum reservoir integral with a trunk mounted vacuum pump on current vehicles only.

Heater system evaluation at −10°F demonstrated performance superior to that installed in conventional vehicles. The system was not optimized by any means but will be developed in future vehicles.

VEHICLE OPERATION LOG

Vehicle running time/use cycle was automatically registered on a bank of pressure sensitive timers installed in the vehicle's trunk, Figure 20. The timers are actuated via calibrated pressure switches and sense engine compressor pressure. The engine pressure signal approximates engine speed.

Figure 21 summarizes the engine's use cycle. Note that 46% of the total engine run time is at idle. A majority of operating time is at speeds less than 80% Ngg. Current experience shows that the maximum speed attained during the Federal drive cycle is 80% Ngg.

The relationship between engine output power and accelerator pedal α, also shown on Figure 21, was designed to simulate that of a conventional vehicle e.g. 25% throttle equivalent to 40% power. This arrangement was implemented to improve the driver's performance "feel," a psychological consideration as opposed to technical.

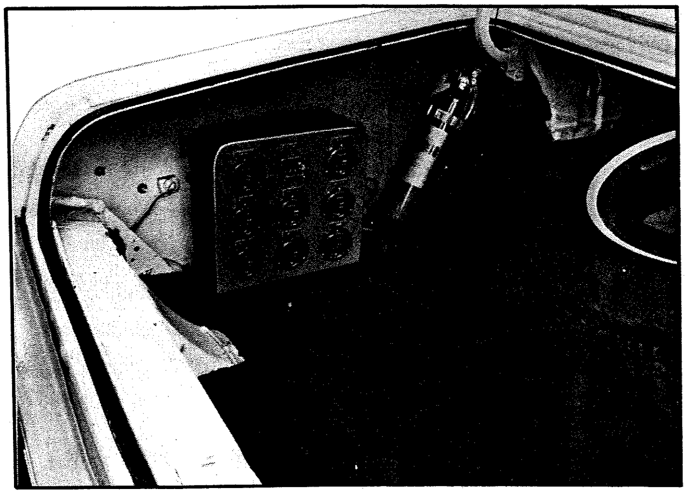


Figure 20 Vehicle Log Recorder

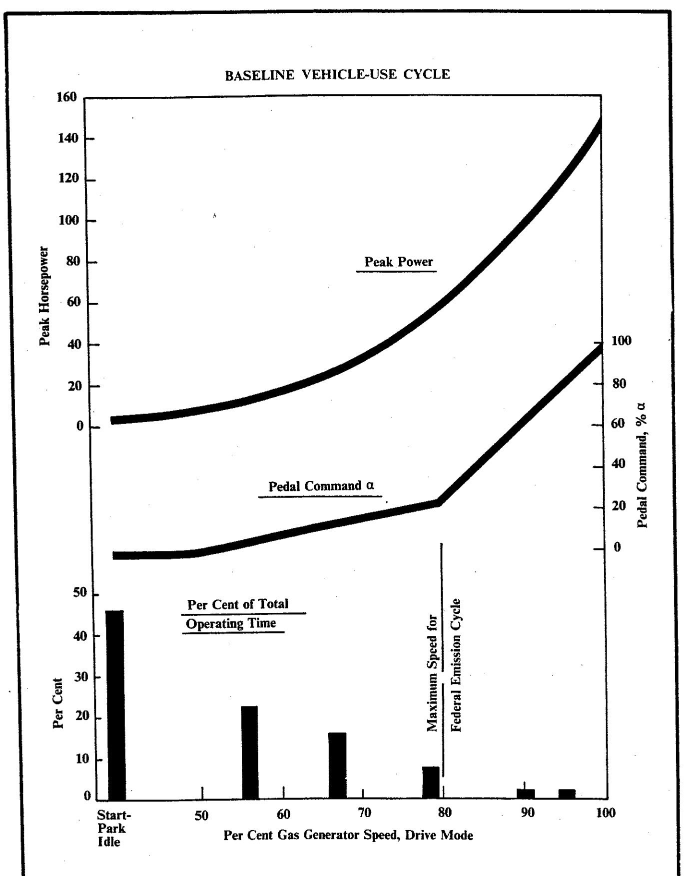


Figure 21

VEHICLE BRAKE SYSTEM

In several previous gas turbine installations conventional brake boosters were modified to operate off engine pressure as a vacuum source was unavailable. Space limitations of current and future installations require the use of the relatively small size hydraulic brake boosters—currently being introduced to the market. Oil pressure is tapped off the flow control circuit of the power steering pump. Figure 22 describes the hydraulic system and includes use of power steering back pressure to actuate engine controls i.e., the power turbine nozzle actuator, and the variable inlet guide vane actuator. An accumulator supplies pressure during an emergency situation.

This system was tested on two vehicles and judged acceptable for future installations.

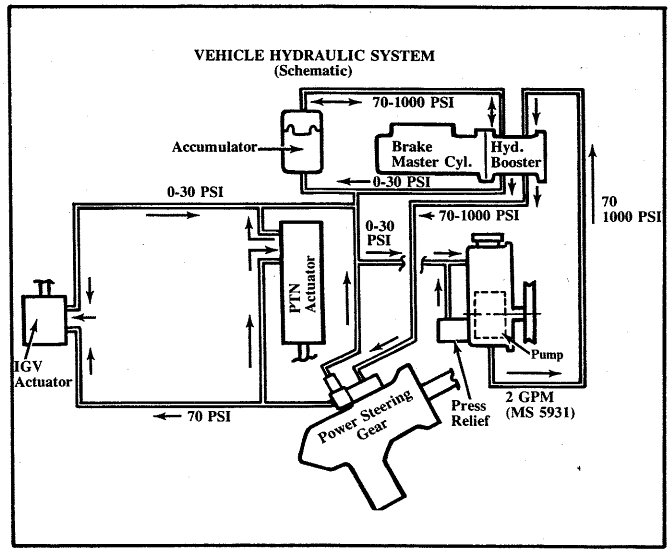


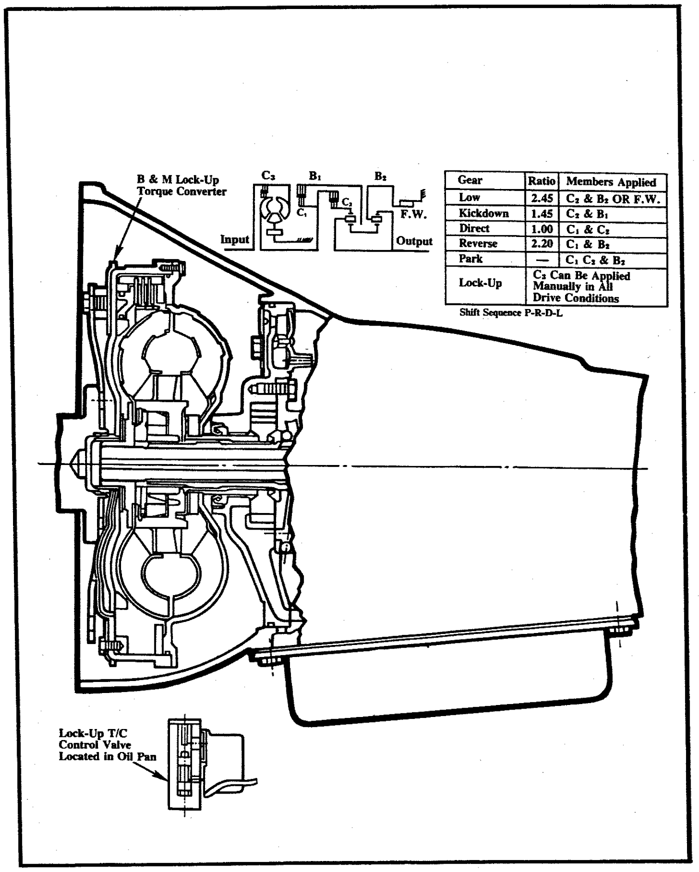
Figure 22

3. BASELINE SYSTEM IMPROVEMENT CONCEPTS

LOCK-UP TORQUE CONVERTER

The baseline engine uses a conventional three-speed automatic transmission. The torque converter is used to provide slippage so that power-turbine-driven vehicle accessories will function properly at zero and low vehicle speeds. It also allows the use of hydro-dynamic reduction gear pinion bearings. This results in an overall loss of drive-train effiency at normal road loads. An automatic slip clutch was considered as a more ideal solution, but major development would be required to obtain smooth engagement at all oil temperatures and throttle demand conditions.

To investigate the effects of torque converter slippage, an available unit was procured which incorporates a wet-disc clutch pack around the outside diameter of a 9.5 inch torque converter, Figure 23. With the clutch disengaged, normal torque converter characteristics as shown on Figure 24 are obtained. With the clutch engaged, the converter slip is eliminated and the unit rotates as a single piece.



Gear	Ratio	Members Applied
Low	2.45	C_2 & B_2 OR F.W.
Kickdown	1.45	C_2 & B_1
Direct	1.00	C_1 & C_2
Reverse	2.20	C_1 & B_2
Park	—	C_1 C_2 & B_2
Lock-Up	C_3 Can Be Applied Manually in All Drive Conditions	

Shift Sequence P-R-D-L

Figure 23 Lock-up Torque Converter

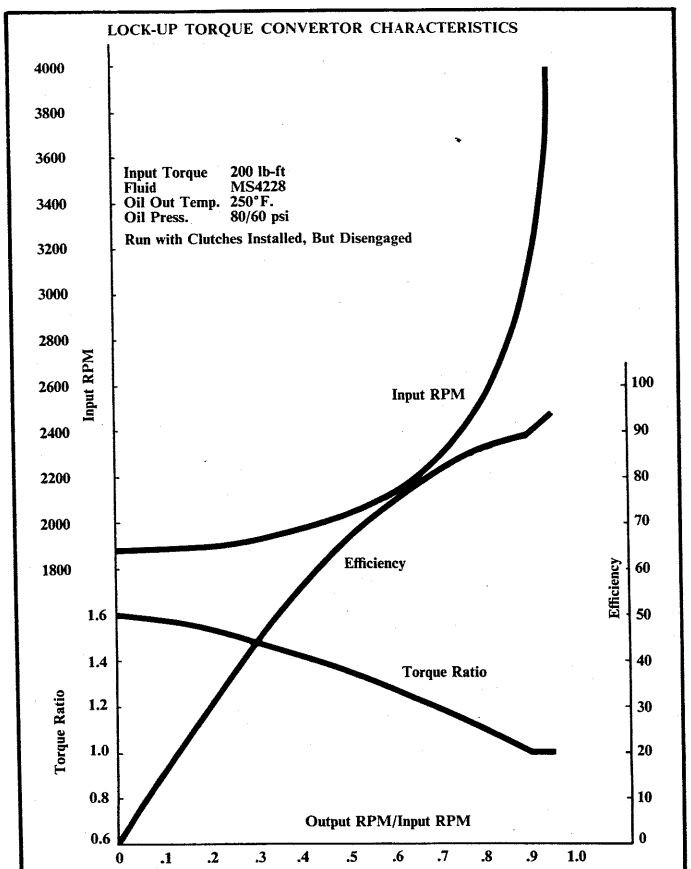


Figure 24

The transmission assembly was made such that pressure was available to the clutch only in second and direct gears. The closed-throttle 1-2 (and 2-1) shift point was raised to 10 mph to prevent dragging the power turbine speed too low which would result in loss of power assist for steering. A solenoid valve permitted driver choice to operate without lock-up.

Proving ground test results for vehicle acceleration, engine braking, and road load fuel economy are shown on Figures 25 and 26. Significant improvements are obvious in all cases. Figure 27 shows that for a given vehicle speed the engine will operate at a lower power level.

VEHICLE PERFORMANCE COMPARISON
LOCK-UP TORQUE CONVERTOR

Acceleration Performance

	W/O Lock-up T_1 = 31°F	W/Lock-up T_1 = 33°F
0-30 MPH	4.3 sec.	4.3 sec.
0-60 MPH	12.1 sec.	11.7 sec.
50-70 MPH	7.6 sec.	7.1 sec.
Distance in 5 sec.	117 ft.	118 ft.
Distance in 10 sec.	448 ft.	453 ft.
Distance in 15 sec.	894 ft.	907 ft.
Distance in 20 sec.	1422 ft.	1447 ft.

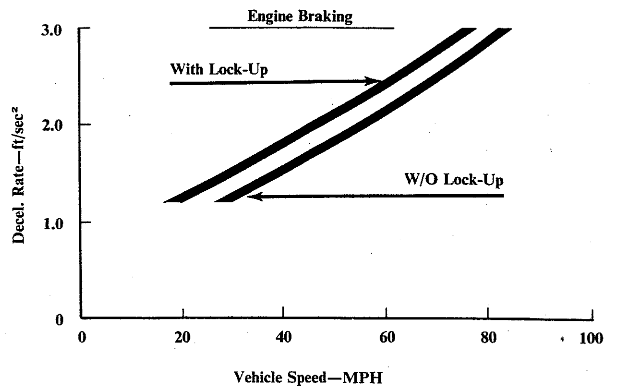


Figure 25

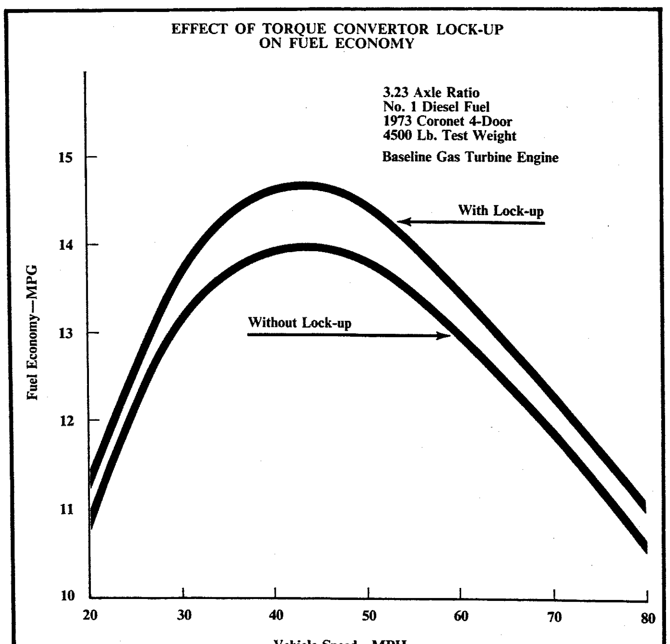


Figure 26

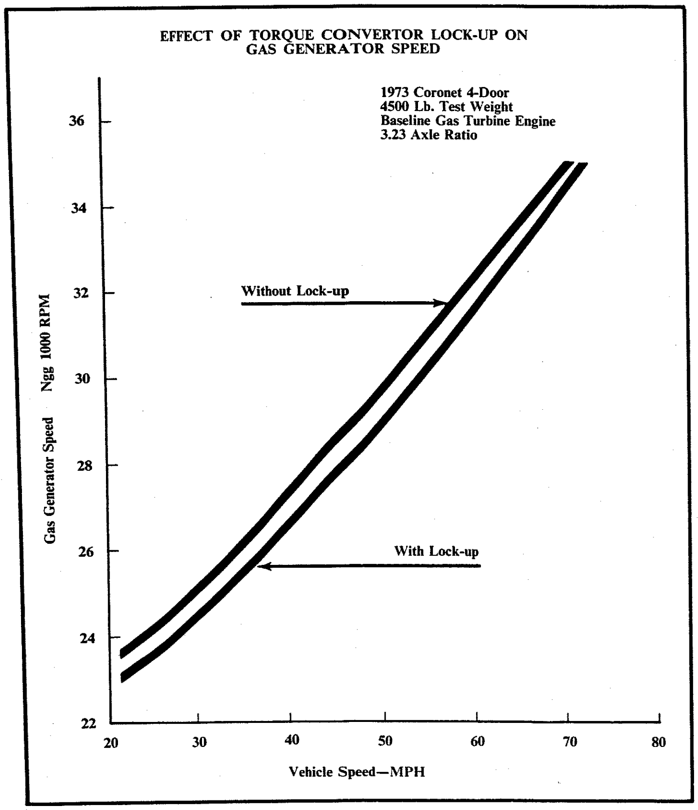


Figure 27

As discussed in the following section, the higher power turbine accessory load of the free rotor arrangement necessitates use of a lock-up during low speed engine braking to avoid accessory under-speeding.

FREE ROTOR

Free rotor is the identification given to the engine concept whereby all accessory drives (engine or vehicle) are removed from the gas generator. The baseline engine was designed with engine auxiliaries (air pump, oil pump, regenerators) driven through gearing from the gas generator, and vehicle accessories (alternator, power steering pump, air conditioning compressor) driven from the power turbine. Schematics of baseline and free rotor arrangements are shown in Figures 28 and 29. Potential advantages of a free rotor system would be:

1. Improved rotor response.
2. Reduced overall engine noise.
3. Simplified gas generator design.
4. Improved cold starting.
5. Use of gas bearings.
6. Improved idle fuel economy.
7. More efficient usage of the power turbine stage at idle.

Apparent disadvantages are:

1. The need for more sophisticated controls.
2. The need for a variable ratio power turbine accessory drive.

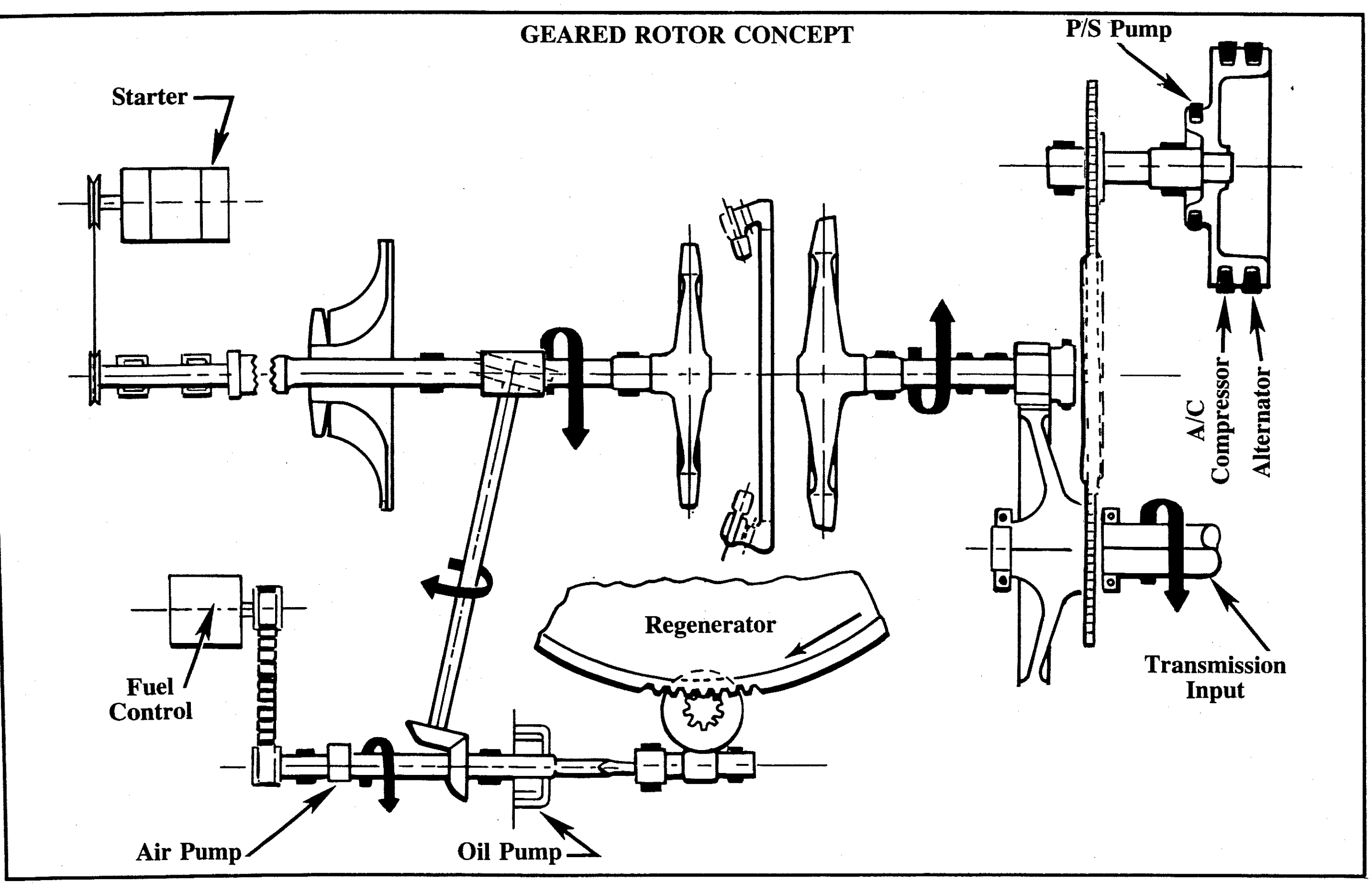


Figure 28

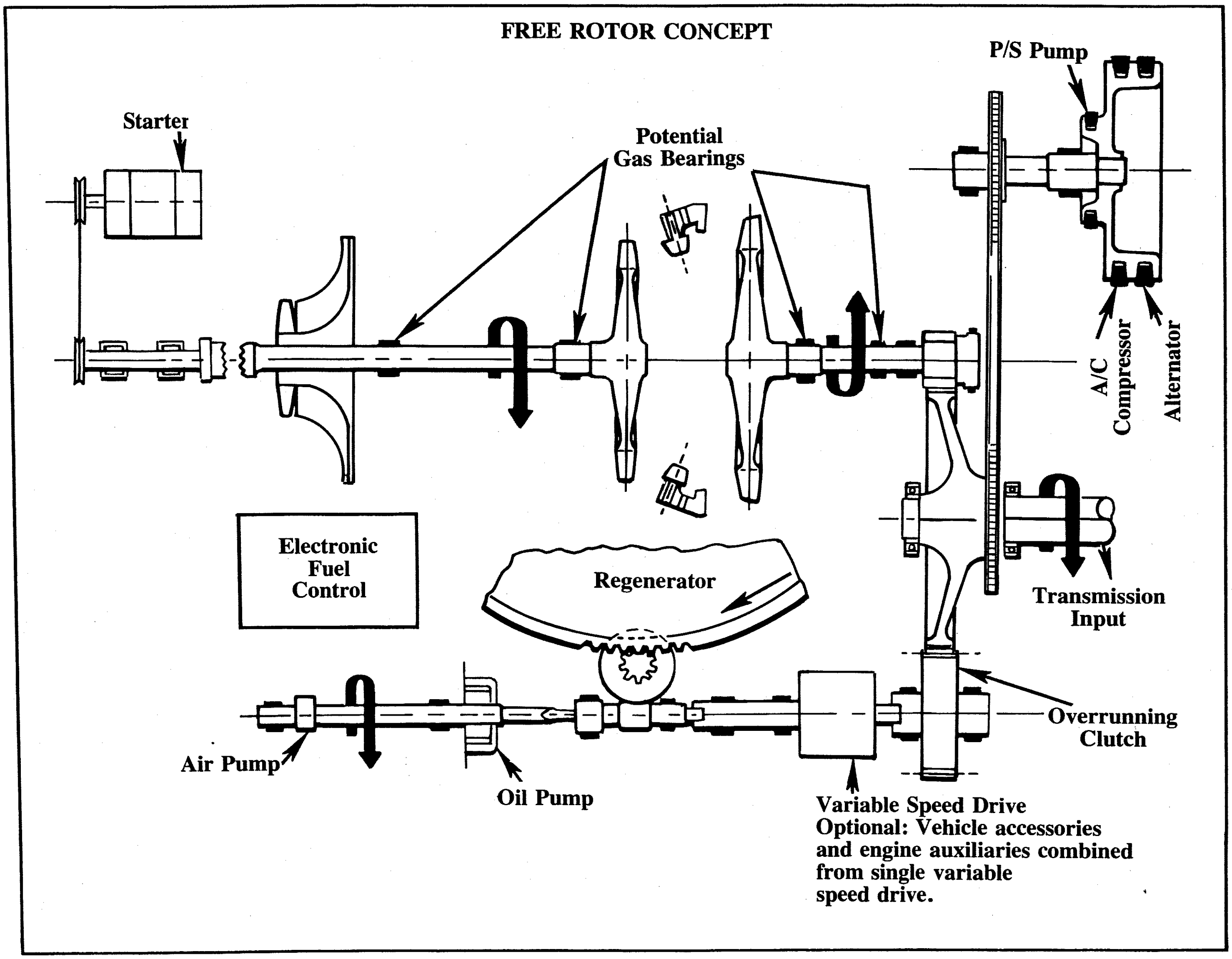


Figure 29

PRELIMINARY TESTING

A baseline engine was converted to a free rotor configuration for *test cell use* only as shown in Figure 30. A hydromechanical fuel control system with speed limiter was improvised as a suitable electronic control system was not available. In this arrangement engine auxiliaries were driven externally and an equivalent HP value was applied to the power turbine via an alternator/load bank.

Test results using identical fuel schedules and selected turbine exhaust temperatures, T_8, indicated response times of *approximately 1.2 sec.* for both the geared rotor and free rotor configurations using the slope intercept method. However, the free rotor had a slightly improved rate of rise during the initial part of the acceleration relative to the standard engine. Typical free-rotor test results are shown in Figure 31. Rotor response characteristics are shown in Figure 32.

In subsequent tests to "optimize" rotor and vehicle response, it was determined that opening of the power turbine nozzles just prior to initiation of the rotor acceleration resulted in an improvement in rotor response, a reduction in the integrated transient fuel flow and a substantial reduction of emissions, specifically NO_X. This latter effect is discussed later in detail in the dissertation on emissions.

Consider free rotor and geared rotor engines operating at idle speed and at equal engine output conditions. If the power turbine nozzles are opened prior to the rotor accel for both concepts and the acceleration temperatures, T_5, are equal, then the torque available for rotor acceleration will be greater for the free rotor concept since it is not burdened with driving the engine auxiliaries. Because geared rotor auxiliaries account for a minimum of 5% of the rotor power, such a scheme should result in a proportional improvement in response with a free rotor. Power turbine inertia is sufficient to maintain the speed of the auxiliaries during this short transient period. Application of nozzle opening should be of greater benefit to a free rotor arrangement.

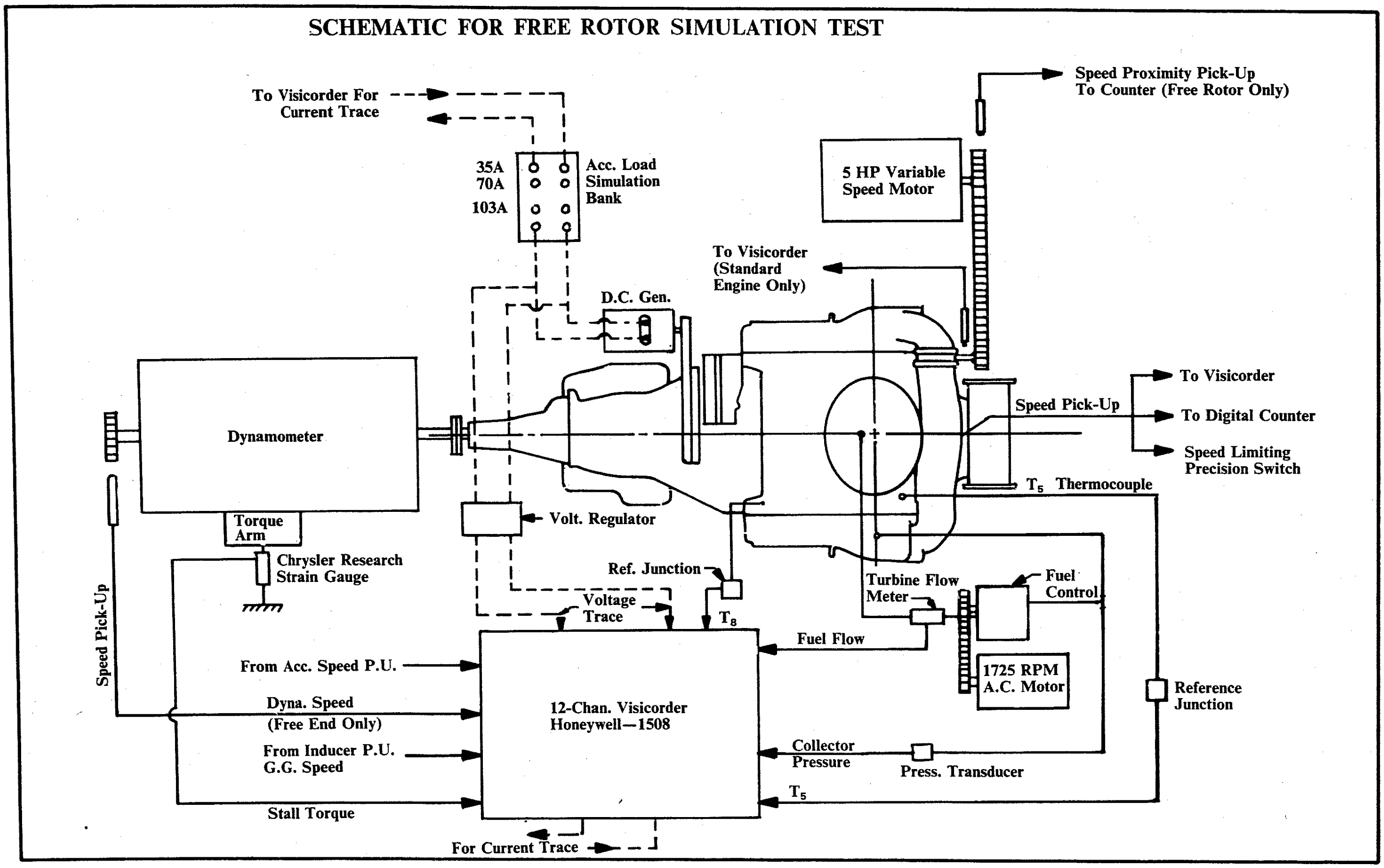


Figure 30

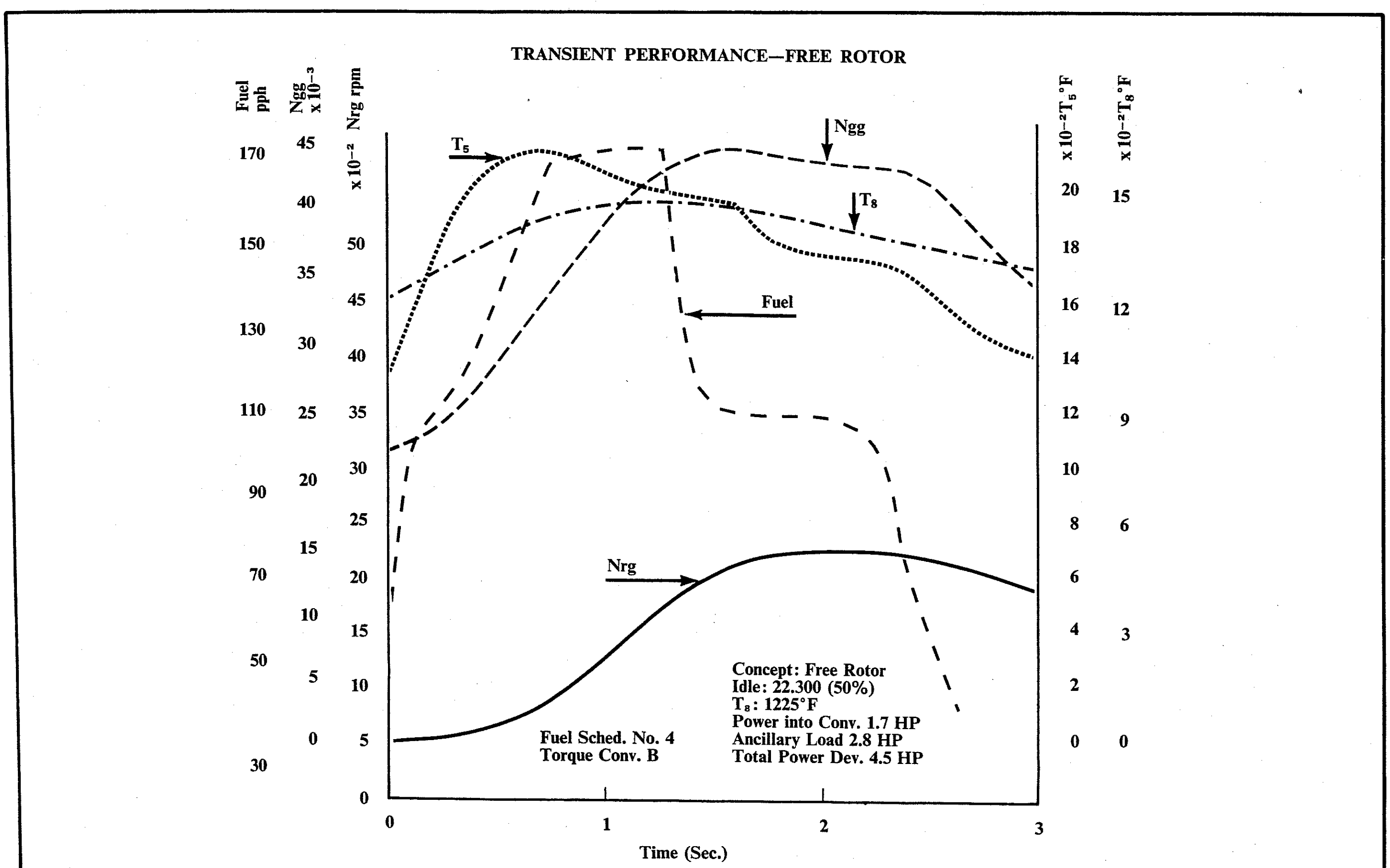


Figure 31

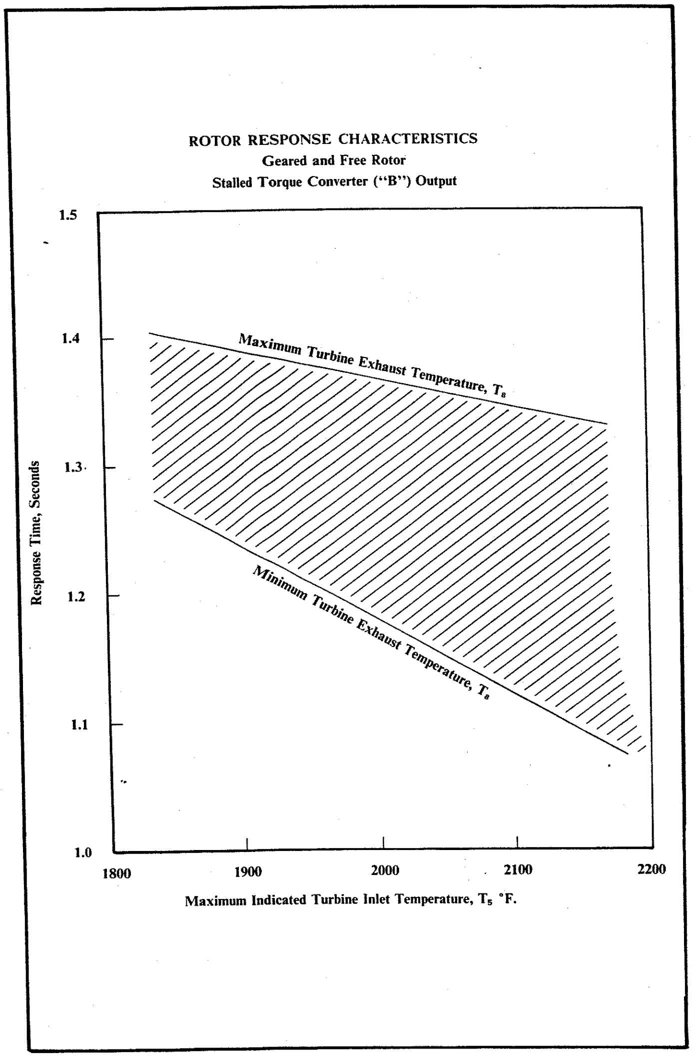


Figure 32

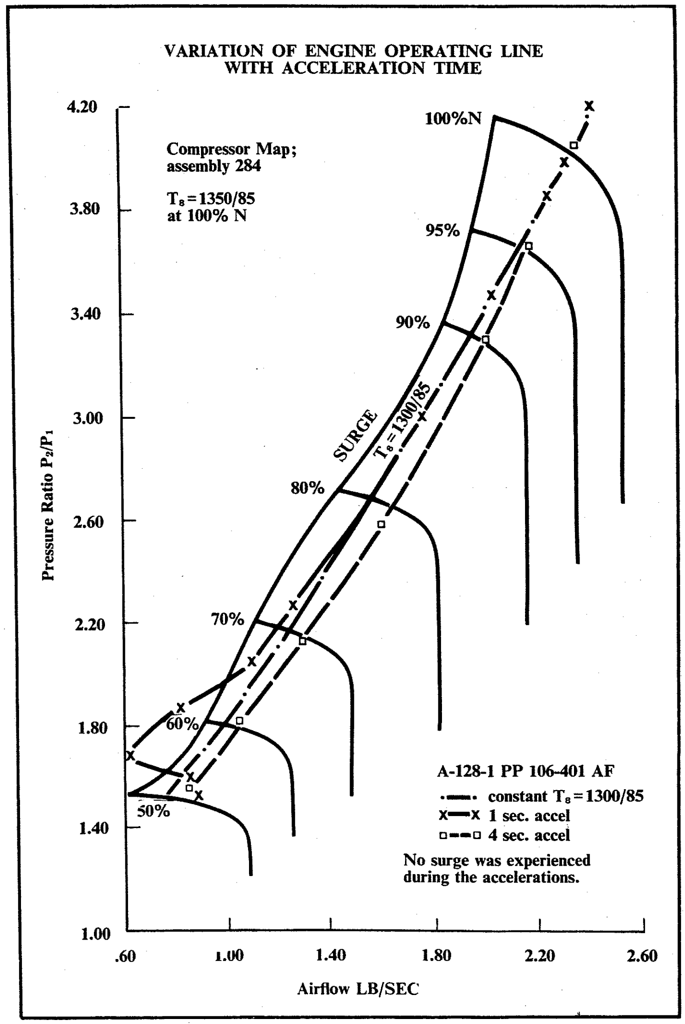


Figure 33

TRANSIENT AIR FLOW CHARACTERISTIC

An interesting phenomenon was noted during the rotor accel tests and is discussed briefly.

Figure 33 shows a typical surge line for the baseline engine. Superimposed are the transient and steady state characteristics of the same engine. This figure shows the compressor collector pressure during acceleration to be significantly higher than levels representative of steady state operation. The difference is attributable to cooling of the compression process by metal components as they absorb heat in approaching steady state temperatures.

LOW SPEED ENGINE CHARACTERISTICS

Theoretical analysis of the free rotor concept showed a potential for lower idle fuel flow. However, the tests indicated that fuel flow differences were minimal and within the experimental error limits. Test results are tabulated below. Note that the power turbine output for the free rotor was increased by 1 HP to compensate for the power required to drive the engine auxiliaries at idle.

Free Versus Geared Rotor Fuel Flow at 50 Percent Speed Road Load Points for Baseline Vehicle

			Minimum Accessory		Full Accessory (Including A/C)	
RPM	MPH		Power Turbine Horse-power	Fuel Flow Lb/Hr	Power Turbine Horse-power	Fuel Flow Lb/Hr
600	10	Geared Rotor	2.4	10.4	5.2	11.1
		Free Rotor	3.4**	10.4	6.2**	11.4
900	20	Geared Rotor	5.2	11.0	*	
		Free Rotor	6.2**	10.9		

*Requires higher gas generator speed.

**Includes 1 hp compensation for externally driven engine auxiliary drive (regenerator, lube pump, fuel control, air pump).

REGENERATOR UTILIZATION FACTOR

In the "Free Rotor" concept the regenerator cores are driven by the power turbine. This alters the proportionality of gas generator speed (airflow) to regenerator core speed which the geared rotor possesses.

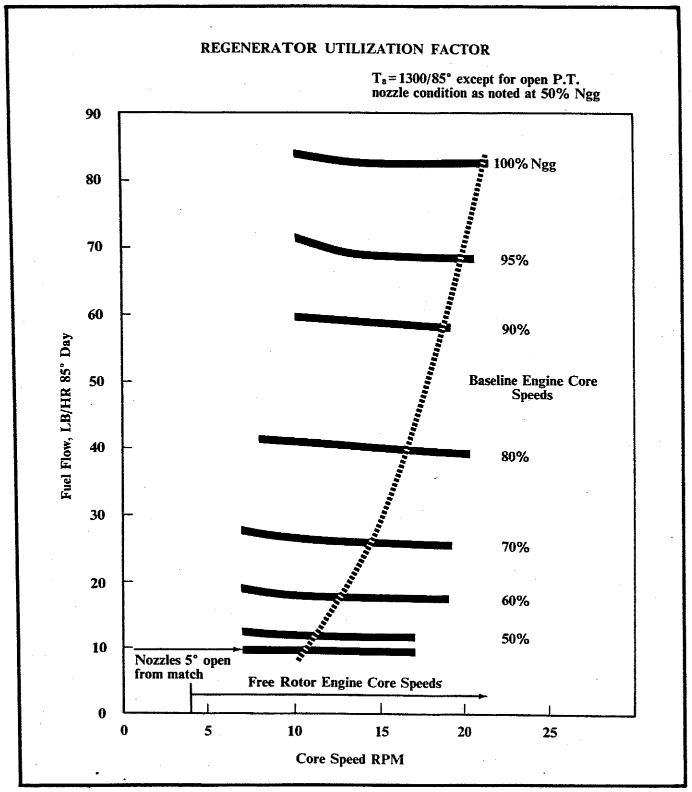


Figure 34

Under acceleration conditions the regenerator core rotation may not be optimum to maintain core utilization and economy of operation.

Sensitivity of metal regenerators to significant speed changes was tested and is shown on Figure 34. Relatively little effect of core speed on fuel consumption of the engine is indicated. Approximately 30% speed reduction is required to increase the fuel flow by 10% at idle conditions. At higher speeds the effect of core speed change becomes even less significant.

Increasing the regenerator speed considerably beyond the maximum utilization condition produced no significant changes.

ENGINE CONVERSION FOR VEHICLE APPLICATION

Conversion to a free rotor engine required innovative solutions in a number of critical areas

- fuel delivery
- starting
- power turbine speed governing
- oil and air pump, regenerator drive slip clutch.

A complete free rotor engine conversion was therefore undertaken to demonstrate feasibility of the system. This included:

- The auxiliaries (air pump, oil pump, and regenerators) normally driven by the gas generator were transferred to the power turbine by relatively simple modifications as shown in Figure 35.
- Addition of an electro magnetic face clutch mechanism to rotate the accessory shaft during cranking yet disengage at some predetermined time in the start sequence.
- A closed loop electronic fuel control system was incorporated replacing the hydromechanical fuel control.
- Removal of the high speed worm wheel and drive bevel gear. The engine auxiliaries would remain the same being driven either from the starter or power turbine.

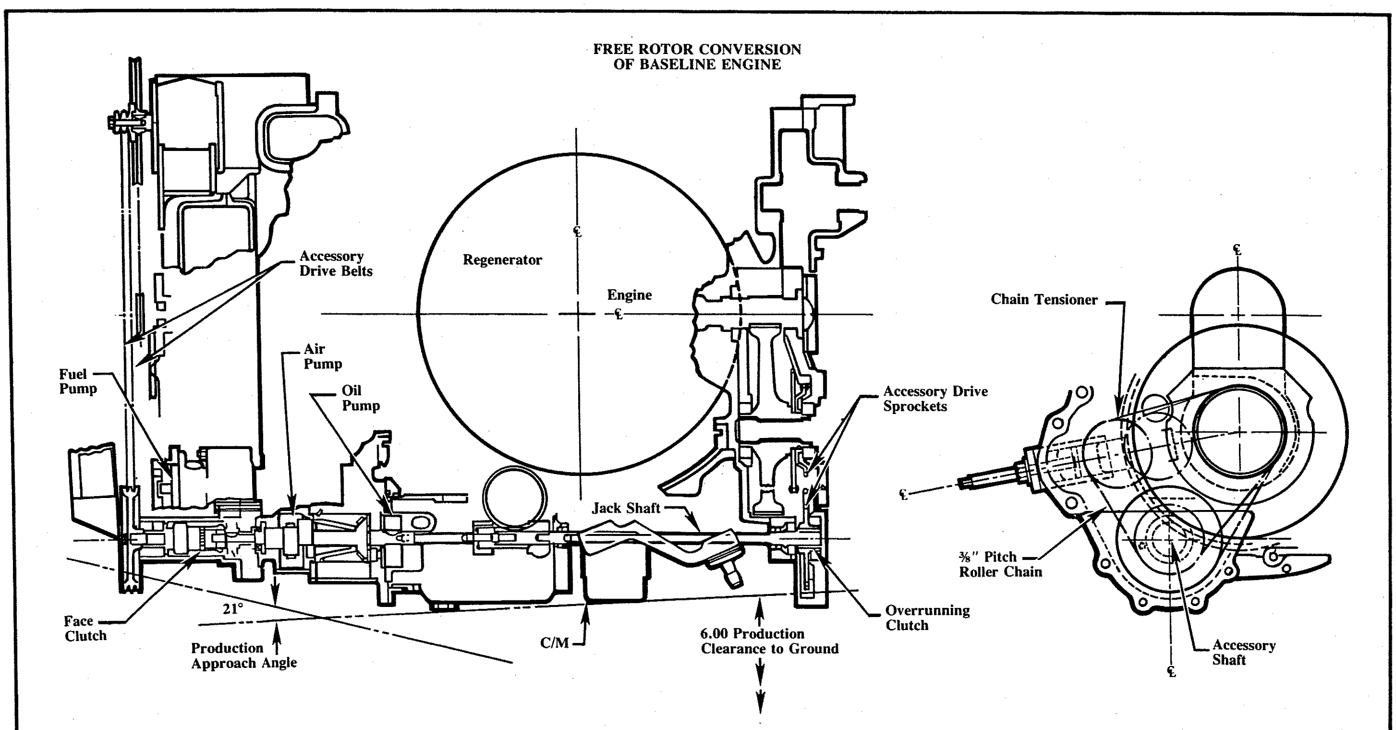


Figure 35

- A 6 to 1 ratio pulley system in tandem with the existing gas generator starter provided the correct accessory speeds during starting via dual polyflex belts.
- A drive member was installed on the rear face of the regenerator worm to accept the driving jack shaft from the power turbine. A side pocket was welded to a power turbine casting. This new addition houses the driven sprocket for the engine auxiliaries and the chain tensioner. A second sprocket was added to the existing one on the reduction gear to complete the auxiliary drive system. The driven sprocket incorporated an overrunning clutch to release during the start mode until power turbine speed increased and eventual lock-up occurred driving the engine auxiliaries by the power turbine.

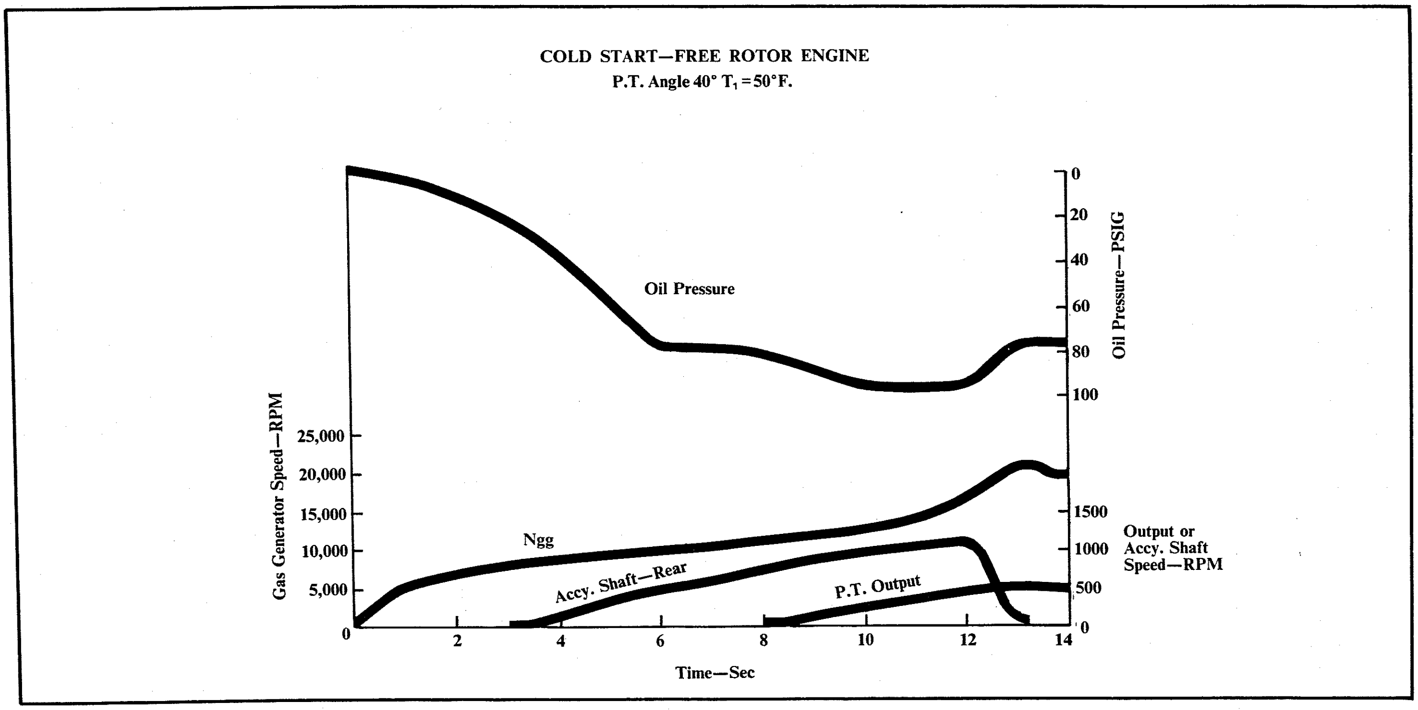


Figure 36

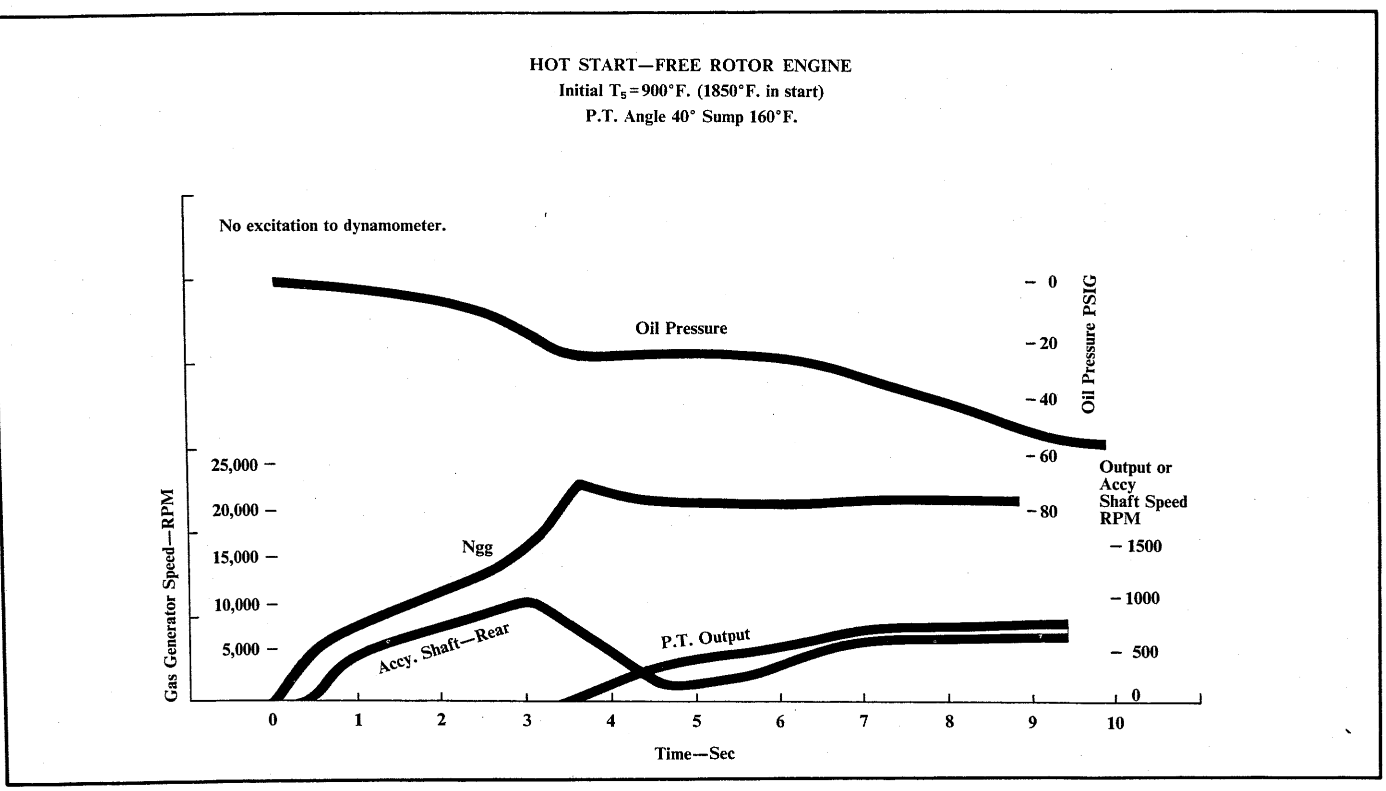


Figure 37

- A speed limiter installed in the above drive system prevented over speeding the engine auxiliaries at higher power turbine speeds. This was a simple friction slip clutch device with a centrifugal assist. The system was abandoned for simplicity and will be considered at a later date.

In the initial free rotor conversion, the power turbine would not accelerate fast enough to smoothly pick up the auxiliary shaft during starts. As a result, the auxiliary shaft speed would fall off sufficiently at starter dropout to prevent a smooth increase in oil pressure. Figures 36 and 37 show these trends for both a cold and hot start condition. The lack of smooth pickup of the auxiliary shaft during starting was reduced by closing down the variable nozzles resulting in greater torque to the power wheel.

A wide rotor oil pump (.825 in. nominal) is used as the optional narrow rotor pump (.50 in. nominal) resulted in unacceptably low oil pressure at the lower speeds.

Peak power of this engine was 145 horsepower. Peak power prop shaft speeds were generally slightly lower than the geared rotor counterpart. The basic peak power curve is shown in Figure 38.

The overall sound quality of the free rotor was greatly improved over the geared rotor. However, high frequency air flow noise previously masked by gearing noise is now dominant. This noise emanates from the scroll-elbow-regenerator cover regions.

In summation: limited experience has shown that a laboratory, free-rotor gas turbine is feasible with current electronic control technology. Additional experience is required to develop a reliable control system.

Methods of noise control must be investigated to reduce internal air flow noises.

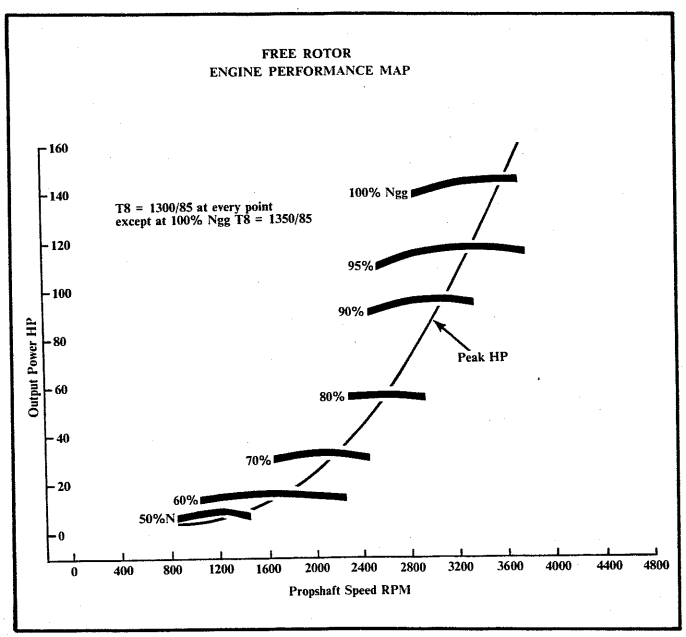


Figure 38

A speed limiting device has shown practical use in a gas turbine engine, however it will require further integration into the design of the auxiliary drive system.

VEHICLE TESTS

The use of the free rotor engine in a vehicle has required some special consideration but has resulted in a very acceptable system having low noise level, good starting, and good driveability.

The control system provides an output speed limit of 3000 rpm to prevent overspeeding the auxiliary air pump. This is done by throttling back fuel flow to reduce output power. However, this condition is seldom in evidence since 3000 rpm represents about 60 mph in second gear or 85 mph in high with the existing 2.76 rear axle ratio.

The major item requiring vehicle development was the low speed engine braking condition. Several items should be noted:

1. Compared to the baseline engine, this engine has an increased power turbine load ahead of the torque converter consisting of the engine oil and air pumps and the regenerator drive.
2. The engine oil pump was also used to provide actuator and trimmer power.
3. The engine oil pump has marginal capacity at 600 rpm output speed.
4. The power turbine nozzles are reversed during engine braking, thus the engine airflow is applying reverse torque on the power turbine wheel.
5. The torque converter is a poor coupling at low speeds, especially for the reversed torque of engine braking.

These conditions combine in such a manner that the power turbine could be *stalled* in engine braking at about 30 mph when operating with full vehicle accessory load i.e., air conditioning, alternator and power steering. When the power turbine stops rotating, the engine oil pump is also stopped so there is no oil pressure at the actuator or mechanical provisions to return it to the power position. The first stage is still running, without oil pressure or auxiliary air supply for the fuel nozzle, until the driver shuts the engine off. (Gasoline fuel is mandatory because of the ceramic regenerators in this engine, apparently vaporizing well enough to support combustion without the auxiliary air supply). Since the oil pump is also driven by the starter, a restart would return the actuator to the driving position. The upgaded power turbine nozzle actuator is designed for fail safe operation.

One way to avoid stalling the output is to come out of braking at higher speeds. On this car, braking would have to be deactivated below 40 mph. Engine braking goals have been to provide a maximum of braking down to a minimum vehicle speed. This has been accomplished as follows:

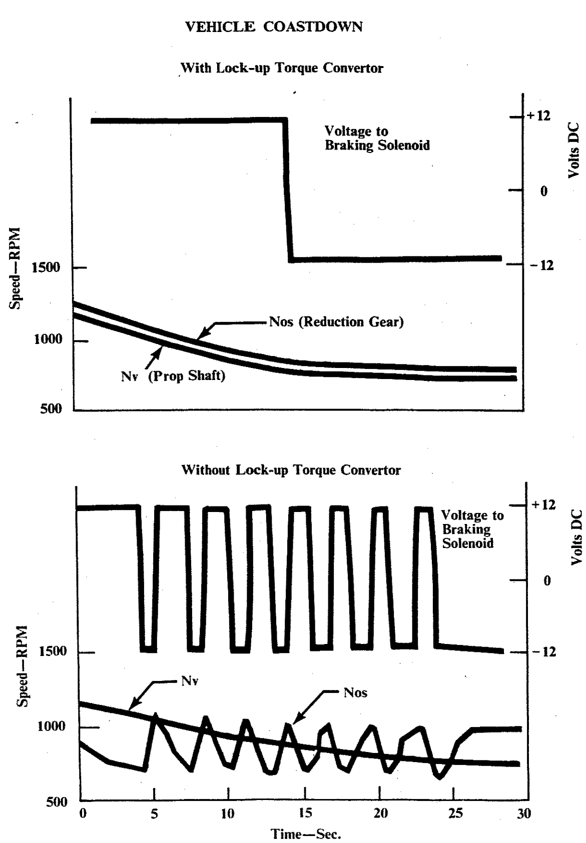


Figure 39

The power turbine nozzle actuator and trimmer were plumbed into the vehicle (power steering and brakes) oil supply which has greater capacity for maintaining pressure at low speeds. The lock-up torque converter was installed to eliminate converter slip. The controls are set to come out of engine braking at about 17 mph, before the lock-up clutch disengages at about 12 mph. Engine braking has been satisfactory with these changes, without problems of staying in braking and losing oil pressure.

Figure 39 illustrates the effects of torque converter lock-up during normal engine braking from 35 mph without air conditioning or wheel braking. Without lock-up the engine output speed quickly drops to 700 rpm requiring elimination of braking, then returns to over 1,000 rpm as the nozzles return to the power position. Since braking is being demanded in response to prop shaft speed, cycling results until the vehicle speed drops to 17 mph. The cycling could be eliminated by sacrificing engine braking in this speed range, but use of the lock-up provides the best of both. Note that for the first 5 seconds of each figure, engine braking is 50% better with lock-up than without (steeper decrease of Nv). A lock-up converter must be considered essential with the free-rotor concept if good engine braking is to be maintained.

Normal engine shut-downs were timed to evaluate the loss of oil pressure versus rotation of the free rotor first stage. Some oil pressure exists until the power turbine stops rotating, which takes at least eleven seconds even with maximum load applied (steering turned against stop). The first stage rotor normally coasts for 18 seconds after key-off, thus about seven seconds without oil pressure. No bearing problems have yet been encountered, even in combination with the many losses of oil pressure, while running, relative to the engine braking conditions.

The idle speed was reduced to 47½% in order to gain an improvement of about 5% in fuel economy over the Federal Emission test cycle. Fuel economy is still relatively poor with the early ceramic regenerators used. The reduced idle speed also helps to lower the idle noise level. In combination with the integrated control system which opens the power turbine nozzles during compressor accelerations, good response and driveability are maintained.

EMISSION CONTROL

It is generally believed that the gas turbine has continuous combustion, inherently low HC and CO emission levels, and especially with a regenerative engine, high NO_X levels due to high combustion temperatures. The *automotive* gas turbine is far from steady state and generally does *not* have continuous combustion. During a Federal Emissions Test Cycle ("normal driving") the fuel flow varies abruptly between zero and 140 pph while the engine air flow changes by only a factor of 2. See Figure 40. The high fuel spikes are necessary to overcome the gas generator inertia and provide quick response to driver demands for increased power. Additionally for good driveability it is necessary to shut off the fuel when power is decreased to minimize gas generator coast down time. This also minimizes fuel consumption. Though conventional combustor technology can provide very low steady state HC and CO emissions, these contaminants are likely to be real problems during transient operation. Emission tests also include exhaust sampling of a cold start which contributes substantially to HC and CO emissions.

The steady state range of the combustor for the baseline engine must extend from engine braking at .0018 overall F/A ratio to sustained wide-open-throttle operation at .0113 F/A. For acceleration transients, overall F/A increases to .024. The automotive combustion system is also required to be compact, quiet, durable, odor free, low cost, non-smoking and very reliable. It is highly desirable that it operate equally well on a range of fuels from non-leaded gasoline to Diesel 1, in any mixture, without adjustment.

The high burner inlet temperature of a regenerative engine does *not* make it a worse producer of NO_X than a non-regenerative engine. It *does* permit leaner combustion and lower peak temperatures so that low NO_X is obtained while retaining HC and CO control.

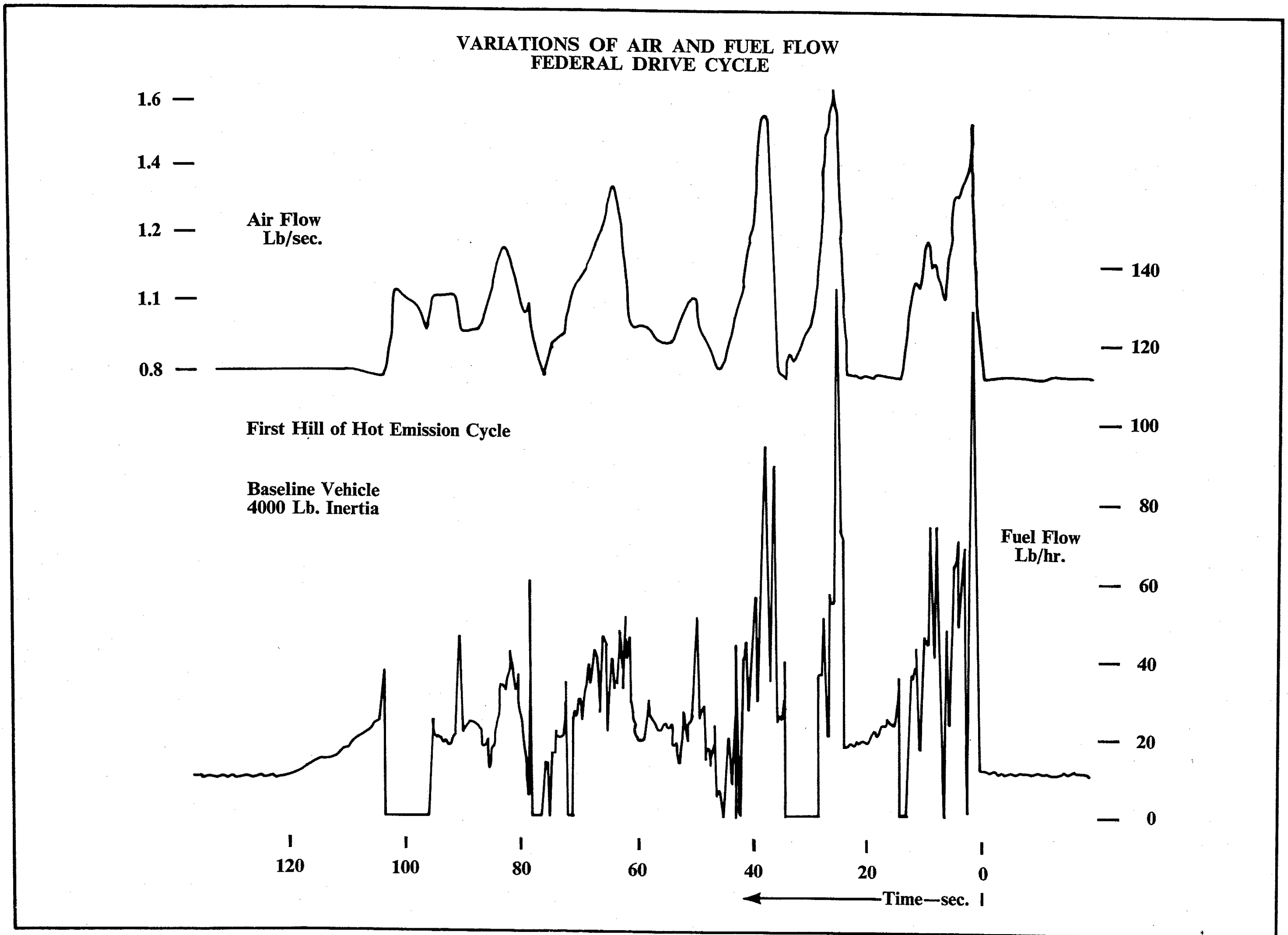


Figure 40

COMBUSTOR DEVELOPMENT HIGHLIGHTS

Ambient Effects

Figure 41 shows the typical trend of exhaust effects due to inlet air contamination. Any addition of NO_X or CO to the inlet tends to increase the exhaust concentration by an equal amount. In some cases there has been less change to exhaust CO levels than made to inlet levels, indicating some ability to oxidize incoming CO as would be expected, especially when operating with high burner or engine cycle temperatures. Inlet HC is partially converted to CO.

This raises the question of correcting gas turbine exhaust emissions for intake contamination. The most accepted procedure is to subtract inlet contaminant mass from the corresponding exhaust mass, thus charging the vehicle for only the net change to the atmosphere. However, it is recognized that this procedure may not yield the exact values that the vehicle would have produced with a perfectly clean inlet. It is advisable to conduct tests with minimum intake contamination.

It must be noted that the hot FID hydrocarbon analyzer is used, consistent with operation on Diesel fuel, and tends to read higher HC levels than a cold unit. Experience has

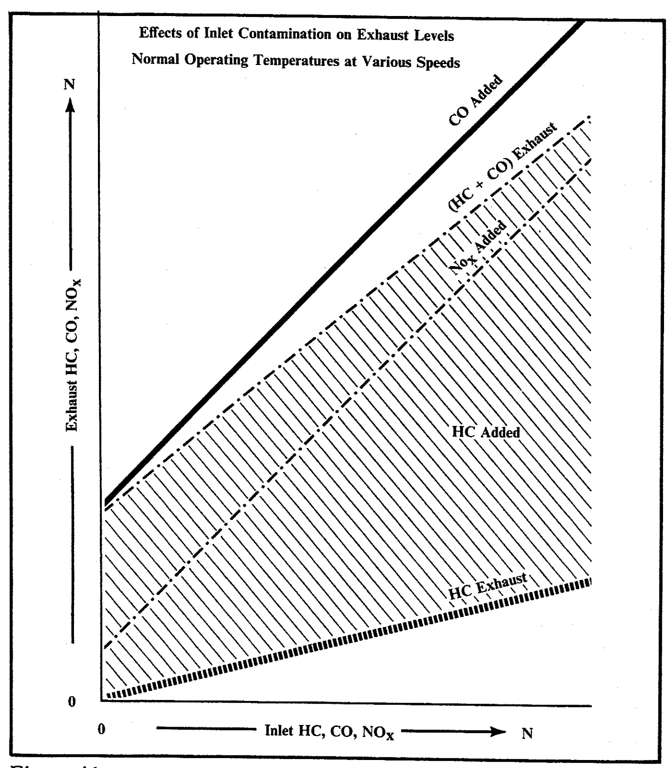


Figure 41

shown that ambient air HC levels are higher with hot analysis. However, our chassis rolls test data uses hot FID for the exhaust only. The inlet values based on cold FID analysis, are, therefore, low by comparison, resulting in conservative correction of HC data.

The effects of ambient humidity must also be considered. Tests with droplet combustion all show a tendency for lower NO_X with increasing humidity. The exact extent seems to vary with the details of the burner, but the correction to 75 grains humidity as specified in the Federal Register for gasoline piston engines is a reasonable approximation and is used by Chrysler. Limited experience with humidity effects on premixed burners indicates no effect on NO_X emissions if engine temperature levels are maintained. Humidity corrections are not used when testing premixed burners, though inlet humidity is maintained near 75 grains when possible.

COLD STARTING

In order to evaluate a burner in an engine or vehicle, it must first be able to cold start and warm-up properly. In order to meet emission standards it must light off quickly once fuel begins flowing. Automotive goals require reliable starting down to −20°F. The conventional burner used by Chrysler in the 1960's met this requirement consistently, even on Number 1 Diesel fuel.

These burners use air atomizing fuel nozzles which require an air pump at all times. Nozzle air pressure of 2-3 psi is required for light off, 3-6 psid for idle and up to 10 psid for WOT fuel flows. The engine driven air pump provided on the baseline engine is well developed and virtually trouble free. However, it does have cost, size, and weight and its use compromises the engine design to provide a suitable drive. It has been a long-term goal to eliminate this requirement.

All recent single stage premixed work has been conducted without the use of an air pump. Hundreds of room temperature starts have been made with gasoline fuel and hundreds of hours of operation conducted. Running and hot engine restarting have also been successful on Diesel fuel. Room temperature starting on Diesel fuel is borderline at this writing, though little effort has been applied toward development. The present concept runs well without an air pump, and, at the worst, will need a starter driven air pump or an air storage system for starting only.

This latest burner uses a pressure atomizing torch fuel nozzle requiring 20 psid fuel pressure. Fuel is supplied to the torch first when starting, so that torch light-off occurs before fuel reaches the premixer. The main fuel nozzle is an air blast type with good atomization when differential air pressure exceeds ten inches of water. This atomization must be sufficient during cold cranking to permit propagation of the torch flame to the *droplet* mixture. (Conversion to premixed burning occurs automatically as regenerated air reaches the premixer.) Most development and demonstration work has used total pressure from the compressor to provide maximum nozzle air ΔP for starting. Since this air by-passes the regenerator, some penalty in fuel consumption is encountered. Present effort is concentrating on using only burner ΔP for this nozzle. Both commercially available and in-house designed air blast nozzles have been used.

SHUT-DOWN

It is beneficial to use a 3-way solenoid (See Figure 50) at the fuel nozzle so that engine air pressure purges the fuel back to the tank. This prevents fuel from vaporizing into the hot engine, and forming deposits of soft soot. Purging therefore helps avoid plugging of nozzle passages and prevents the discharge of soft soot on restarting. It also provides the fill-time lag so that on start-up the torch can be lit before fuel reaches the premixer.

DETERMINATION OF VEHICLE EXHAUST EMISSIONS—FEDERAL DRIVE CYCLE

The Federal CVS emission testing system which is specified for gasoline light duty vehicles is not suitable for turbine emission testing. The turbine air flow is higher than most CVS system capacities. Large capacity CVS systems would further dilute the already diluted turbine exhaust and make accurate emissions determination difficult. The baseline engine/vehicle must average only 3.5 ppm NOx to meet the 0.4 gram/mile standard. Also, the use of Diesel fuel requires hot sample lines and continuous analysis to prevent loss of sample by condensation.

The test procedure in use involves the continuous analysis of all emission species. Fast response instruments are essential for accurate data. The vehicle is started and driven over the cycle as prescribed in the Federal Register. Emission sampling is begun a few seconds before the key is turned on and continues after key off until the gas generator has stopped rotating. The emission levels and suitable airflow signal are continuously recorded on a multichannel tape. See Figure 42.

Most testing has used gas generator speed as an airflow signal, with the engine airflow calibration in the computer program. Some use of an exhaust flow measuring system has shown good results. As more testing is done with different size engines, and especially with variable inlet guide vanes and variable nozzles, direct exhaust flow measurement is planned.

The CVS bag sampling system is used to collect average intake air contaminant levels during a test. These contaminant levels generally are nearly constant during a test period. Efforts are made to maintain ambient humidity at a nominal 75 grains.

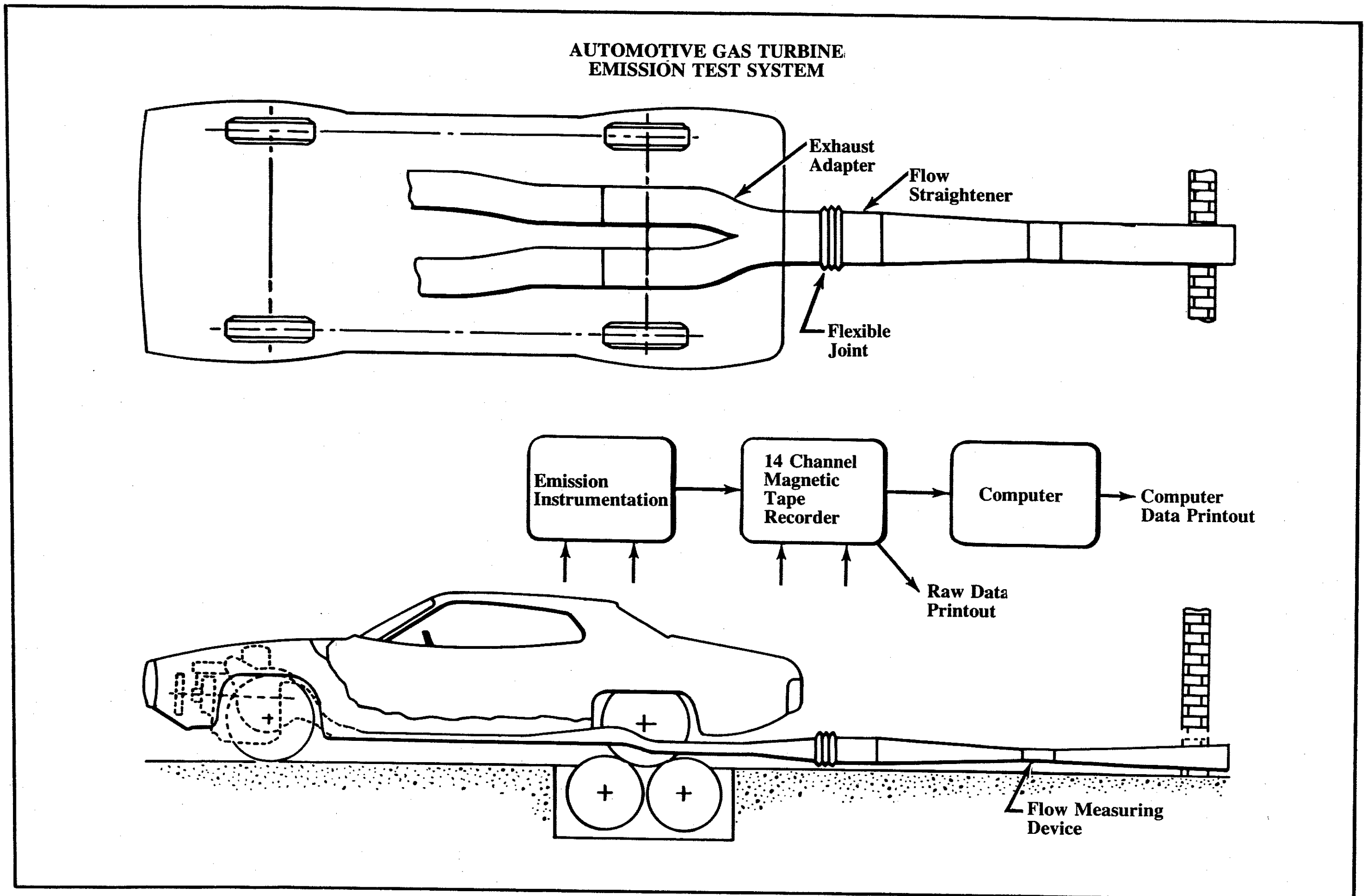


Figure 42

CALCULATIONS

The tape recorded data are processed by computer to yield raw exhaust mass emissions. The calculations are made at one-second intervals (grams per second) and summed over the entire cycle. For 1975 test procedures, the cycle is split at the 505 second point and the usual weighting formula applied.

The program also prints the total cycle air flow and uses it with the inlet sample bag concentrations to compute intake contaminant masses. The difference between intake and raw exhaust is the net emission mass.

Inputs of wet and dry bulb test temperatures permit computation of the inlet humidity. NO_X emissions are corrected to 75 grains as specified in the Federal Test Procedure. It is believed that this correction is valid for droplet diffusion flames, but not for homogeneous reactions. Complete reporting includes raw, net, and net corrected results.

The program also provides a breakdown of the emissions according to idle, acceleration, cruise, and deceleration modes of operation and for specified sections of the cycle time. Total carbon computations of fuel consumption are available through for accuracy, reliance has generally been on the weighed-can method.

ODOR—SMOKE

Odor and smoke problems are minor on the baseline vehicles. The only visible smoke problems have been when operating the baseline burner at low combustion temperatures on No. 2 Diesel fuel. These conditions are also conducive to odor formation, especially during and just after a cold start. Improvements to the engine control system to maintain temperature control have resulted in barely objectionable odors when starting or operating on No. 1 Diesel and virtually no problem on gasoline.

The advanced burner concept also has little or no odor problem with gasoline fuel; Diesel fuel has not yet been used for vehicle testing.

Experience so far has indicated that odor problems will not be severe if exhaust hydrocarbons are under control. A portable stand engine installation is being built for the purpose of ongoing odor evaluations, including cold ambient starting.

EVAPORATIVE AND VENT

The major effort has been in the development of a combustor which produces low emissions. However, complete control of HC emissions includes the lube system and fuel system venting. The production type carbon canister vapor saver system is not applicable to the gas turbine because there is no vacuum purge available and because most of the inlet airflow does not enter the combustion process. With Diesel fuel, no tank vapor saver is needed but one will be necessary if gasoline is used. Lubrication system venting can be a problem if the oil gets hot enough to begin vaporizing. The baseline cars vent the lubrication system into the engine exhaust so that it is included in emission tests.

GAS TURBINE COMBUSTOR DEVELOPMENT

Two basic paths of combustor development have been followed. One, using droplet diffusion flames to obtain low (less than 3.0 gr/mi) NO_X levels, yielded the combustor provided in the baseline engines. The second uses homogeneous reaction to get very low (less than 0.4 gr/mi) NO_X levels. All efforts have been toward combustion control rather than clean-up and low emissions in all normal operation, not just on the Federal Test Cycle.

Combustor testing begins on one of two "engine fixtures." These are complete baseline engines except that they have no power turbine wheel. Shop air is supplied at the power turbine wheel location to provide control of regenerator inlet temperature (emission sampling is ahead of this air injection). Thus, a burner concept starts out in its ultimate environment with the first test being its ability to start the engine from room temperature. The burner is tested in the quasi-steady state conditions of an engine and its control system. Because both variable power turbine nozzles and variable cooling air are used, turbine inlet and regenerator inlet temperatures can be set off the normal operating conditions. The one shortcoming of these fixtures is that by omitting the power turbine and output load, the burner does not operate under true conditions during all transients, specifically engine braking.

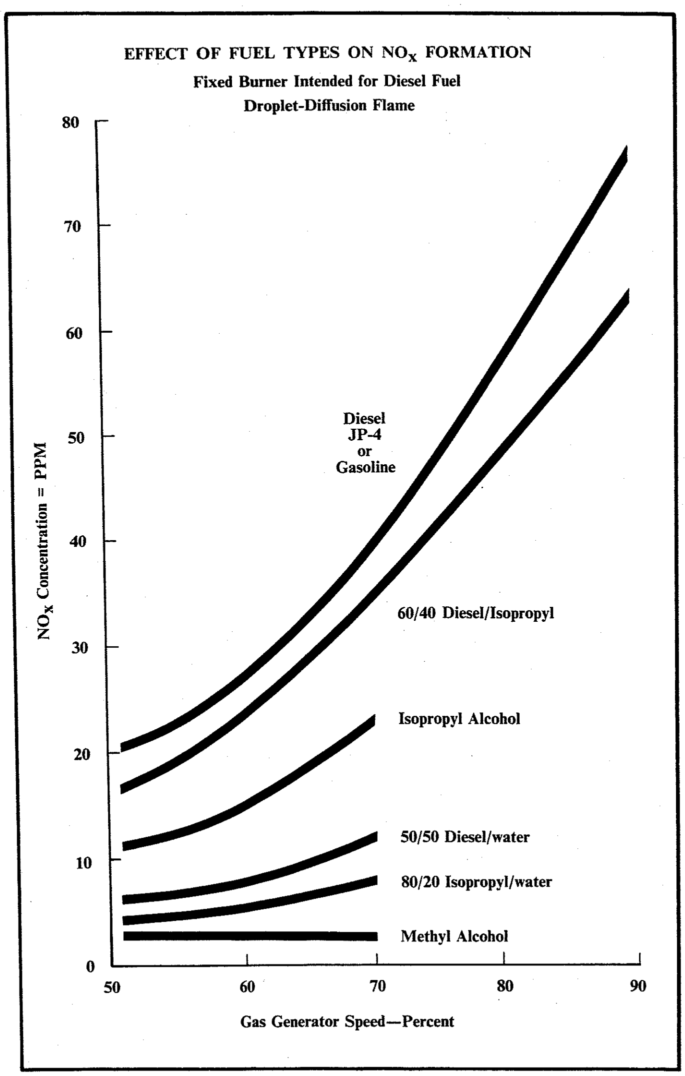


Figure 43

A burner which has satisfactory start and steady-state characteristics, must undergo final testing and development on a complete dynamometer engine. The dynamometer has driving capability to simulate engine braking and the engine includes the automatic power turbine nozzle actuator. This facility is used, along with actual vehicles, to develop transient emission control.

Early in the transition from conventional combustors to low-NO_X concepts, an investigation of alternate fuels was conducted. Figure 43 shows that gasoline, Diesel 1, and JP4 are all about equal while fuels with lower heating values generate less NO_X (but usually higher HC and CO). No efforts have been made to develop a burner for alcohol because of fuel cost and system requirements. It is believed that technology developed for gasoline and Diesel fuel will be suitable for use with alcohol as well. All droplet-diffusion development has been on Diesel fuel with occassional evaluation on gasoline.

DROPLET-DIFFUSION FLAMES

There are two things to be done to control NO_X in droplet-diffusion flames, minimize the peak temperature and minimize the time at that temperature.(3) For CO control, the air-fuel mixing must be well defined and sufficient reaction zone residence time at temperature provided. If CO is controlled HC will generally not be a problem, except at flame-out and light-off conditions. Since these conditions occur up to 200 times during the Federal Cycle, precise fuel control and ignition requirements must be met. The most developed burner along these lines is the baseline burner provided for the EPA/ERDA engines and vehicles and described in the contract Quarterly Reports (4).

This concept was first demonstrated on Chrysler Car 208 on 12/1/71 resulting in emission levels of 0.29, 3.41, and 2.68 grams/mile net corrected HC (Cold FID), CO and NO_X over a 1972 cold start cycle. The baseline burner is a direct evolution and recent testing on a baseline vehicle resulted in 0.85 HC (Hot FID), 2.58 CO, and 2.10 NO_X net corrected grams/mile on the 1975 cold-start cycle. On the highway-fuel-economy-cycle emissions were 0.09 HC, 0.66 CO, and 1.19 NO_X. Though no standard exists for this cycle, the data exhibits the proper spirit of emission control. It should be noted that driving the prescribed emission test cycle requires the driver to constantly switch between accelerator and brake, thus making transient emission characteristics far more dominant than steady-state

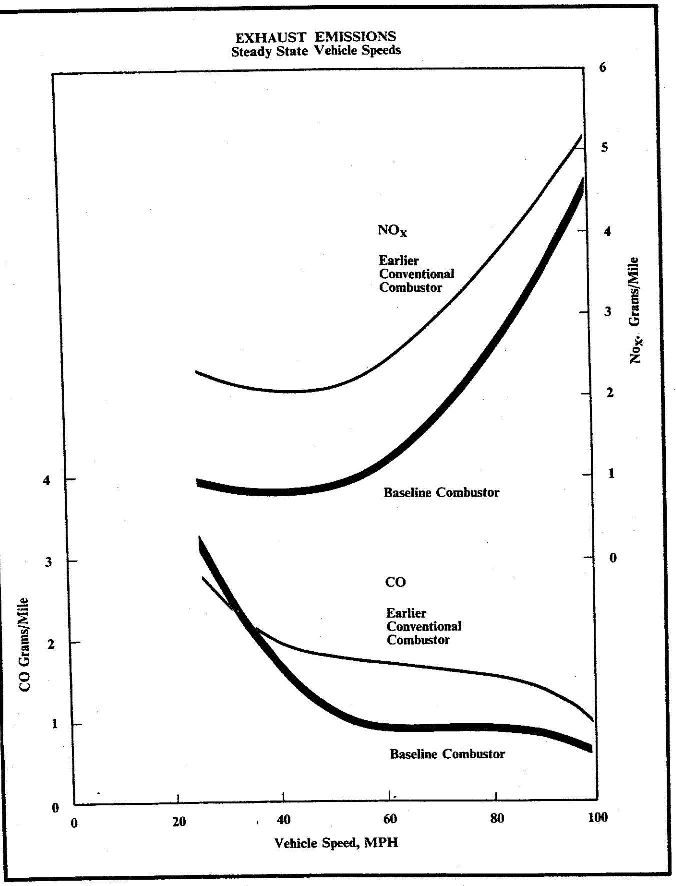


Figure 44

emissions. The fuel-economy cycle is more typical of the modulating throttle, plus occasional braking, characteristic of normal and prudent real-life vehicle operation.

Approximate steady state NO_X and CO levels are compared with an earlier conventional burner in Figure 44. Exhaust HC is typically lower than intake HC for these conditions. Extensive baseline vehicle emission tests were conducted by EPA with results reported on Pages 5-7 and Appendix A of the Ninth Quarterly Progress Report.(4) These results show that this concept is capable of meeting relatively low NO_X standards even in a full sized car with relatively poor fuel economy. This same technology applied to an advanced gas turbine vehicle having twice the fuel economy should produce half the emissions output.

Additional proprietary effort has shown promise of still lower emissions with this basic concept. It is likely that advanced gas turbines could meet NO_X standards of 1.0 gr/mi with this type of combustion system. Note that the baseline burner has not compromised size, weight, cost, vehicle performance, or fuel economy as compared to the conventional burner of the 1960's.

PREMIXED BURNING

It is commonly accepted that very low NO_X levels (along with low HC and CO) can only be achieved by avoiding droplet burning; that is, by burning a homogeneous, prevaporized and premixed charge. Since the gas turbine must be lean overall, it is most practical to premix lean rather than rich.

The first demonstration of this combustion process was made early in 1972. A stable lean reaction was produced with very low emissions and no visible flame. This concept was supplied with gasoline fuel with vaporization and mixing taking place simultaneously within the burner. It would cold start as a droplet-diffusion flame and convert automatically to premixed conditions as the engine warmed up. No flame holding devices were used. However, range of operation between lean flame-out and high NO_X levels was very narrow.

For the control of automotive emissions, range must be defined as the operating limits within which the emissions, levels are low enough to meet the emission standards involved. Experience has shown that flammability limits are wider than emission limits. For reasonable combustor size the lean limit, where CO becomes excessive, is when reaction zone temperature is about 2000-2100°F. The upper limit, high NO_X levels, is at about 3000-3200°F. These limits will vary somewhat relative to actual size of engine and vehicle, fuel type, control system, etc. Several things can be done toward providing the wide range of overall fuel/air ratios required by the gas turbine.

One of the most obvious, and widely proclaimed as necessary, is the use of variable geometry burners which provide increased combustion airflow to correspond with increases in fuel flow. Chrysler has never felt that this would really be practical because of increased cost, size and weight, questionable reliability, and precise, fast response control requirements. However, at one time there was no other apparent solution and many attempts were conducted, keeping within "reasonable" limits of size and complexity. These efforts produced moderate range, but never enough, and were plagued with flashback and instability problems.

A less restricted approach was taken by Solar Division, International Harvester, under EPA Funded Contract No.

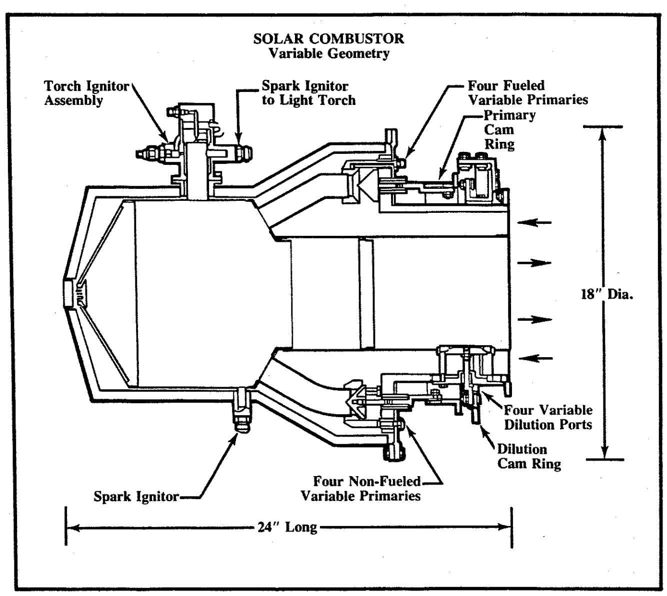


Figure 45

68-01-0464. The result was a large, complex burner, Figure 45, capable of very low emissions over a wide range of steady-speed conditions. It was tested on one of the Chrysler engine fixtures, with results shown on Figure 46. It had poor response to transient operation and mechanical problems with the 12 variable ports. No attempt was made

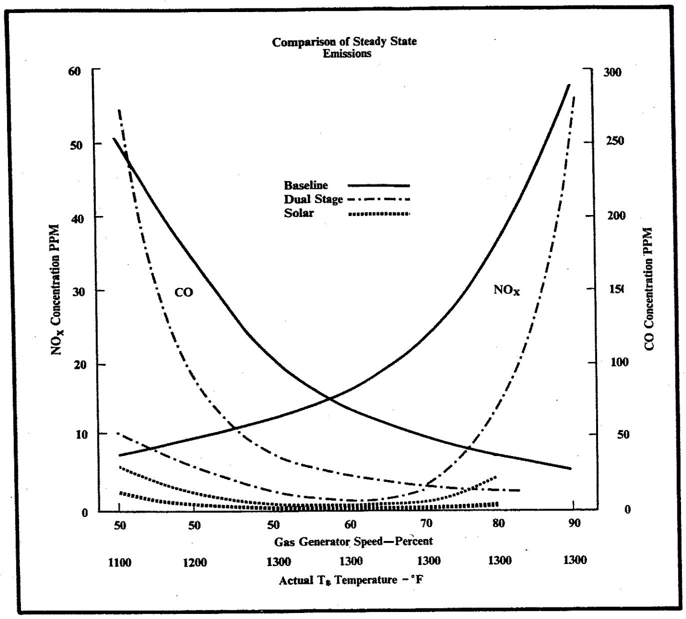


Figure 46

to provide an automatic control system for that burner which was developed principally for fixture testing.

An alternate approach was conceived in the fall of 1973, avoiding the use of variable geometry. A dual stage combustor as shown on Figure 47 and described in the Ninth Quarterly (4) was used with the fuel split between stages being varied to optimize the emissions. Pressure drop was limited to 3% though burner size and weight were somewhat greater than desired.

Considerable development effort was conducted on this concept including dynamometer engine and vehicle operation. A crude hydro-mechanical fuel splitting system enabled adjustment for the good range of steady-state emissions shown on Figure 46 in comparison with the Solar burner results. With a compromise setting it permitted street driving of the vehicle but not optimum emissions.

Further efforts were put forth to develop an electronic control system using two electronic metering valves for fuel splitting. A programmable analog computer was used for this activity. Problems of instability due to burner/control interaction during transient operation prevented the achievement of good transient cycle emission results. However, it is estimated that this combustor with a suitable fuel-splitting control could be developed to meet a NO_X level of 1.0 gr/mi in a baseline vehicle or 0.4 gr/mi in an upgraded vehicle. Another breakthrough caused major development activity to be redirected in early 1975.

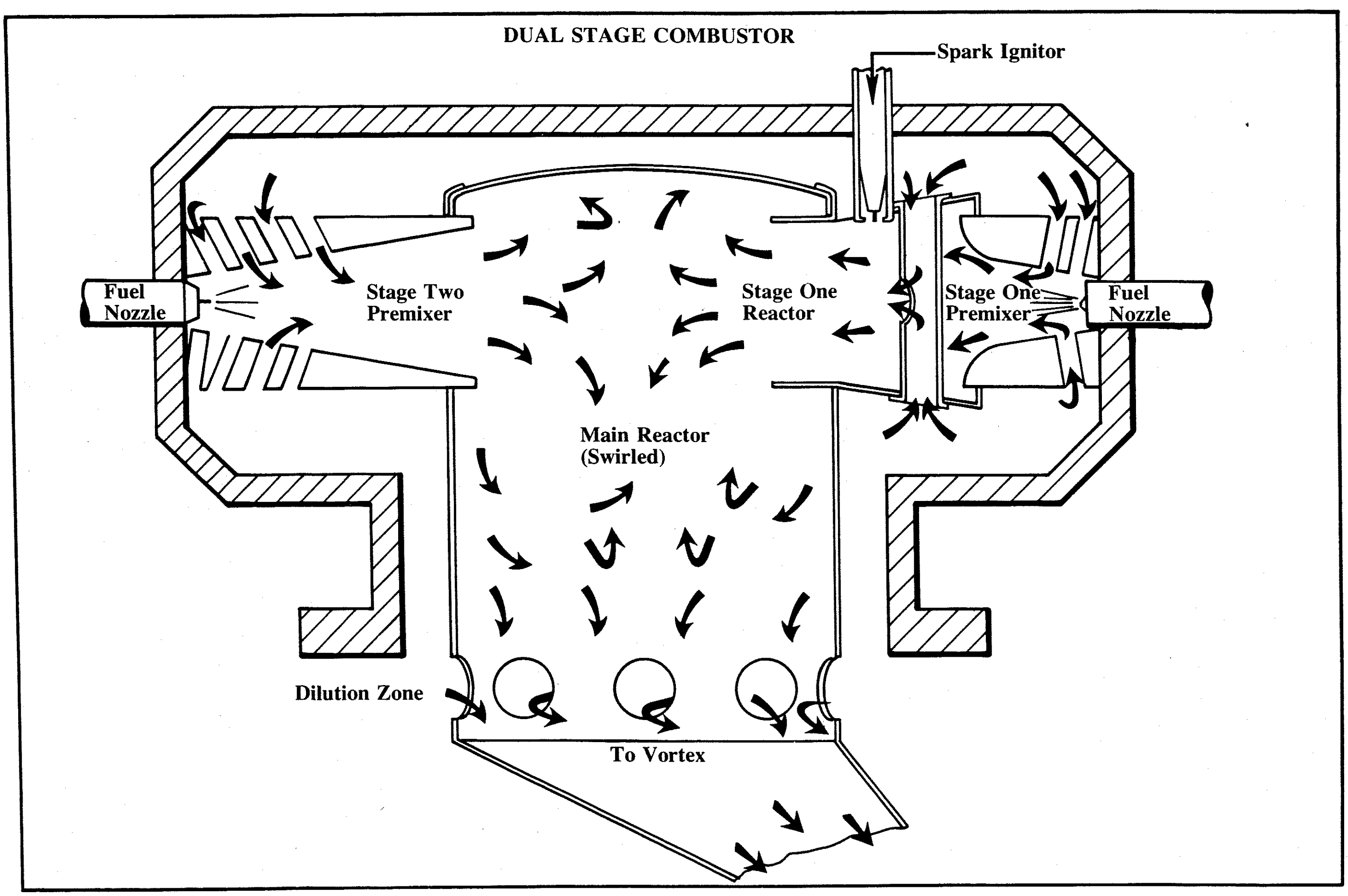


Figure 47

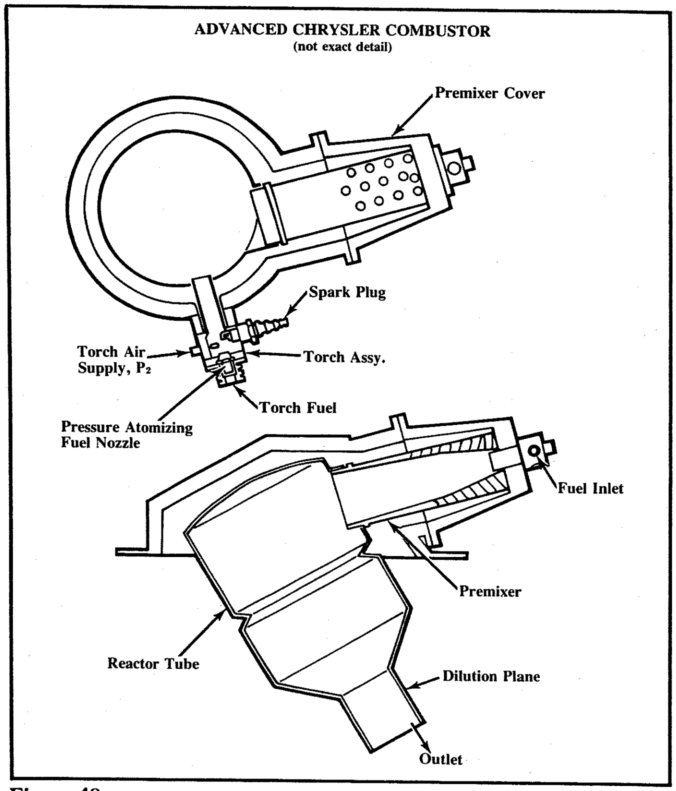


Figure 48

In this latest concept, Figure 48, use of a torch ignitor provides the required wide range of steady-speed emissions control from a single stage lean premixer. As compared to spark ignition, the torch is continuous, provides more energy, and contacts a larger volume of premixed charge. This has permitted stable, low emissions operation down to premixed primary zone fuel-air ratios of .015-.020. For the first time, a compact and reasonably simple combustor has demonstrated with potential for meeting the most stringent exhaust emission standards while maintaining "automotive practice." This concept was first run on a burner fixture in late February (1975) and by May 1 was being demonstrated on a vehicle with both chassis rolls and street driving. This concept has shown definite potential for development to meet the 1978 standards even on the baseline size vehicle. Best NO_X emissions demonstrated as of this writing over a hot 72 cycle are 0.47 gr/mi with HC and CO needing further deceleration control development. Only the torch flame is maintained during compressor decelerations. At steady vehicle speeds on the chassis rolls exhaust HC levels were lower than intake values and the following raw exhaust (not corrected for inlet contamination) emissions of CO and NO_X have been demonstrated:

	CO		NOx	
Condition	**PPM**	**GR/MI**	**PPM**	**GR/MI**
Curb Idle	**37**	**—**	**1.8**	**—**
15 MPH	**47**	**4.0**	**1.8**	**0.15**
30 MPH	**26**	**1.2**	**2.0**	**0.09**
40 MPH	**14**	**0.5**	**2.3**	**0.09**
50 MPH	**12**	**0.4**	**5.9**	**0.21**
60 MPH	**12**	**0.4**	**12.8**	**0.41**

Emissions over the highway fuel economy cycle were 1.57 HC (Hot FID), 2.50 CO, and 0.39 NO_X grams/mile.

Still another means of improving burner low-emissions-range is by raising the burner inlet temperature. (5) Though this would increase NO_X from a given burner, it permits the burner airflow to be reapportioned for leaner operation at the minimum fuel requirement (CO limit). Then the fuel flow can be increased a greater amount before reaching the NO_X formation limit. Because of present material limitations, little use has yet been made of this principle, but it is in the right direction for excellent emission control in advanced efficient, high cycle temperature regenerative gas turbines.

Another area for attention is the reaction zone. Strong recirculation and minimization of wall quenching help to obtain a good lean limit. While mixture distribution is very uniform in these reactors, wall cooling can result in high CO levels. By having the premixed charge enter at a tangential position strong recirculation results so that a faster and more complete reaction takes place. In addition, a fence around the reaction zone wall is used to mix in the cooler gasses flowing there.

TRANSIENT OPERATION

Acceleration NO_X Control

Having a burner which can operate with low emissions over a wide steady-speed range is only part of the battle. Making it comply with transient emission standards is still difficult. Both our droplet-diffusion baseline and advanced premixed burners operate lean steady-state but become rich on start-up and full acceleration schedules. Refer to Figure 49 and note that the various lines are characteristic of any fixed geometry burner, not specified for a particular one. The operating line a-d for speeds from idle to 80%, which is the range required to drive the Federal emission cycle, lies between the CO and NO_X limit lines indicating good emissions over this range.

If the throttle is suddenly moved from idle to 80% demand, the fuel flow will follow the schedule a-b-c-d and about 0.7 seconds will elapse for the speed change. NO_X is controlled both by the rich combustion process and the short duration of time. If the throttle is moved slowly enough to limit the fuel schedule to bounds defined by a-e-f-d, then the NO_X is controlled by the leanness of combustion, though such an acceleration will take 6-8 seconds.

The area between lines e-f and b-c is the problem, with operation just below line g-h being the hottest combustion and therefore having the highest NO_X formation *rate*, but still being of short duration. Premixed burners receive a more pronounced effect because all of the mixture is at peak conditions, whereas droplet burning has rich and lean regions, each with NO_X formation rates lower than the average mixture rate.

In driving the Federal Text Cycle, the actual throttle movements are dependent on many factors such as vehicle

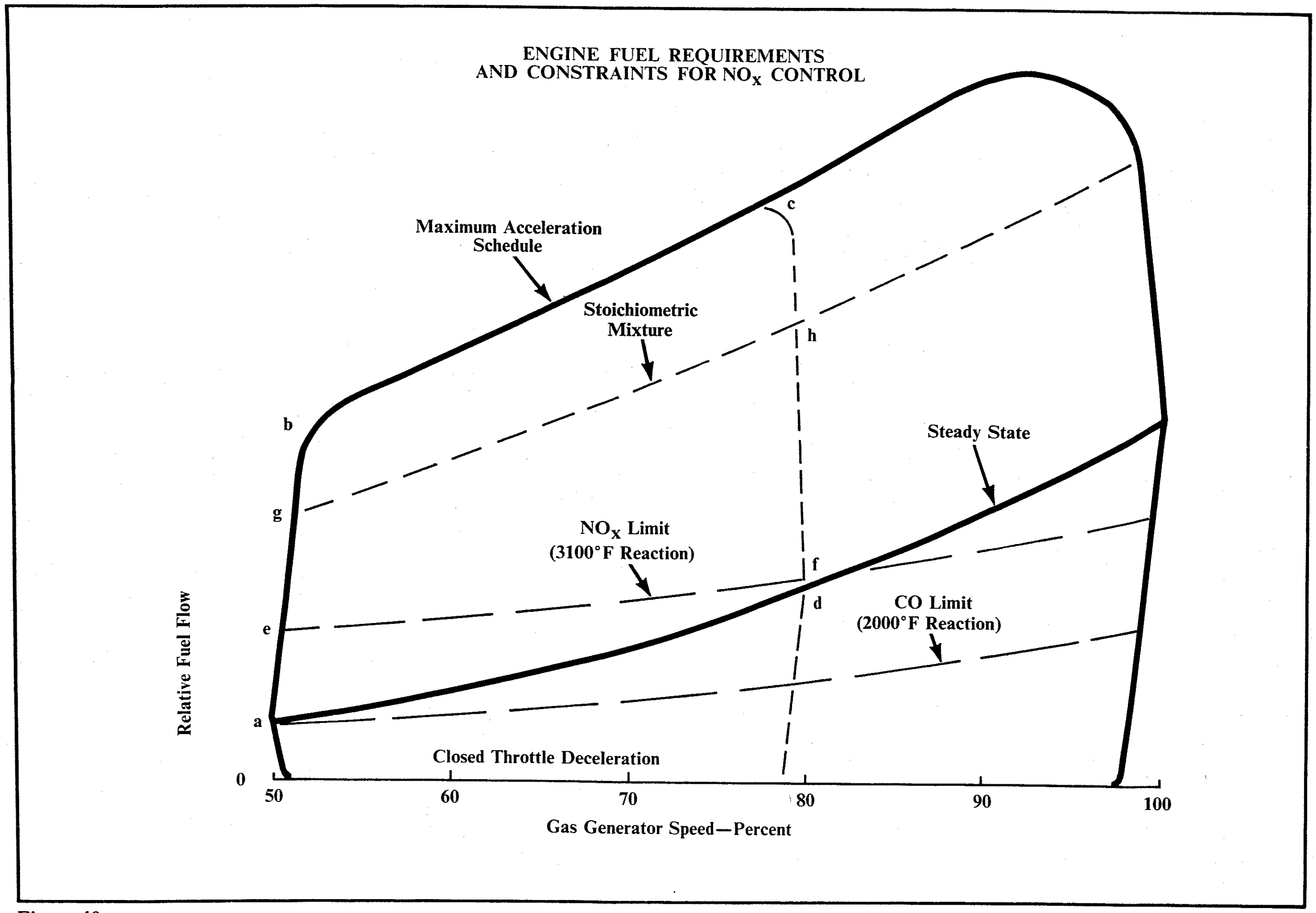


Figure 49

response, driver attitude and ability, throttle sensitivity and repeatability, and compressor response. An effective emission control system must account for *all* conditions. One way to accomplish acceleration NO_X control is to use a fuel control system which avoids this operating region, but can jump across it. Figure 50 is a schematic of such a system as used on early vehicle demonstrations of the single-stage-lean-premixed burner. For slower throttle movements acceleration fuel is limited by orifice "A" to levels below those producing high NO_X. Orifice "B" is opened only when speed demand is significantly greater than actual speed, thus A and B flow in parallel to jump to the maximum fuel schedule. For moderate rates of throttle movement, several jumps back and forth may occur as speed error is alternately high and low.

This schematic also shows the power turbine nozzle blip arrangement used to assist in controlling acceleration NO_X. By opening these nozzles slightly during the acceleration more energy can be extracted by the first stage turbine wheel. Figure 51 shows the effect of blipping on gas generator speed and acceleration fuel flow for a stepped throttle movement from idle to 70% Ngg. Note that with blip, the speed actually overshoots demand at less than 0.6 seconds, and less total fuel is required. Improved NO_X control is primarily the result of this shorter duration of acceleration fuel flow. This blip action is also triggered by a significant speed error signal and, in this case, the solenoid controlling orifice "B" is delayed until the blip actually takes place, thus accounting for the hydraulic lag time. The blip is accomplished by closing the nozzle actuator drain line and feeding it with oil supply to force the piston out until switch 1 is tripped. This leaves the drain closed, but stops feeding oil supply. Switch 2 prevents activation when the piston is in the extended, engine braking, position.

The improvement in compressor rotor response does not necessarily mean an improvement in vehicle response, however, because the second stage is not being fully accelerated until the blip is over and the nozzles return to the power position.

DECELERATION HC & CO CONTROL

The problem of controlling deceleration HC and CO is similar to controlling acceleration NO_X. Again refer to Figure 49. If the throttle is suddenly closed from 80% speed, fuel flow is abruptly stopped and the nozzle bled back to the fuel tank, resulting in only a very narrow HC spike as the lean flame limit is passed. A second narrow HC spike occurs as fuel flow returns and the burner relights. These narrow spikes do not prevent meeting the HC

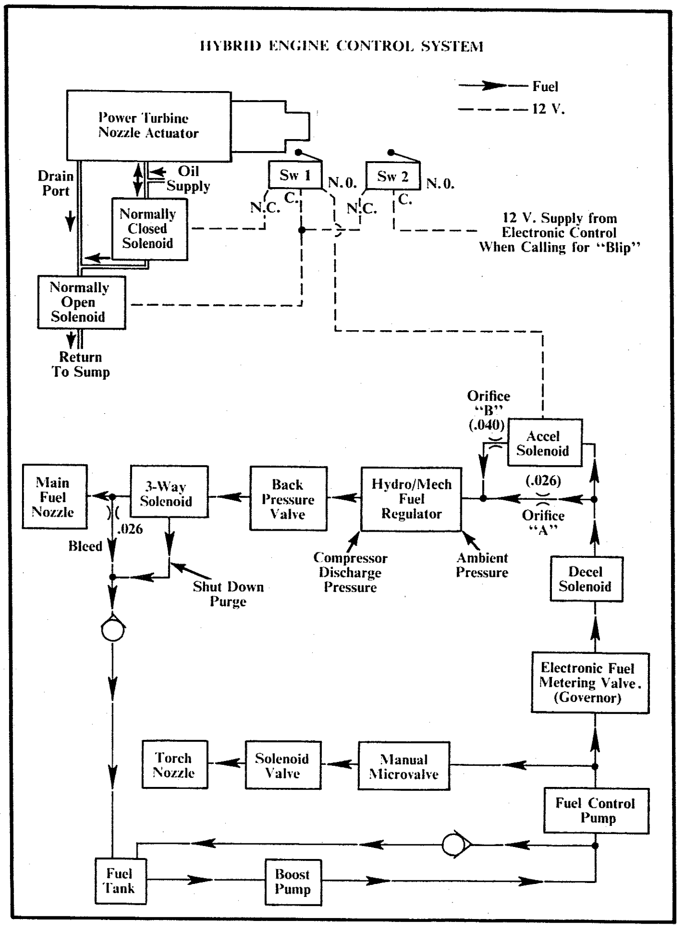


Figure 50

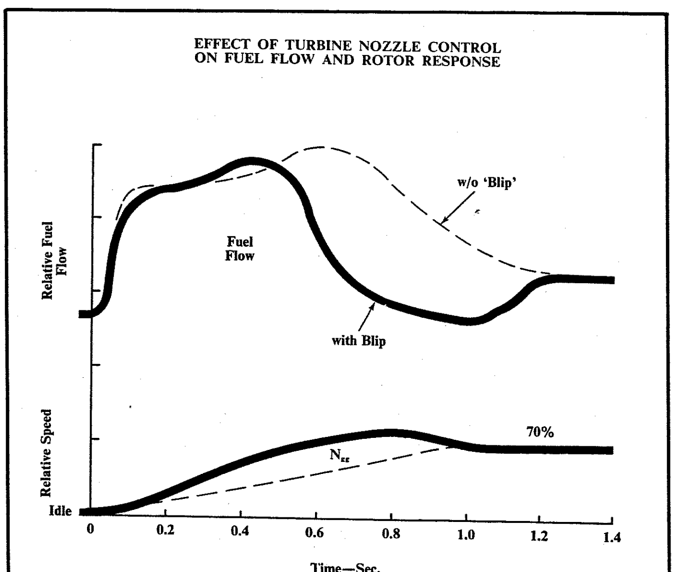


Figure 51

emission standards of 0.41 gr/mi. No CO spiking is evident under these conditions.

However, if the throttle is closed gradually in such a manner that the fuel flow drops just below the CO limit line, then CO and HC will be high throughout the deceleration and the standards may not be met. To insure HC and CO control, the fuel control system should shut off the fuel whenever demand drops below the CO limit line established for the particular burner.

An alternate approach is to maintain deceleration fuel flow at the CO limit line and thus maintain continuous combustion. This approach has shown the ability to maintain HC levels below intake values throughout the entire emission test cycle. However, driveability is compromised somewhat as gas generator decelerations are slower and fuel consumption is increased. No significant NOx penalty is encountered.

PRESSURE DROP EFFECTS

The air pressure loss across the combustor assembly is very important to low pressure, automotive gas turbines, especially at low power levels. Figure 52 shows that a given engine loses power and suffers an increase of specific fuel consumption with increasing pressure drop. The minimum pressure drop allows the smallest engine to provide given vehicle performance and the best BSFC so that maximum fuel economy can be obtained. Pressure drops greater than 3% should not be considered for automotive use. The baseline burner has approximately 2% pressure loss.

In order to use variable geometry burners and maintain very low pressure drops, both primary and dilution zones may need to be varied, such as on the Solar burner.

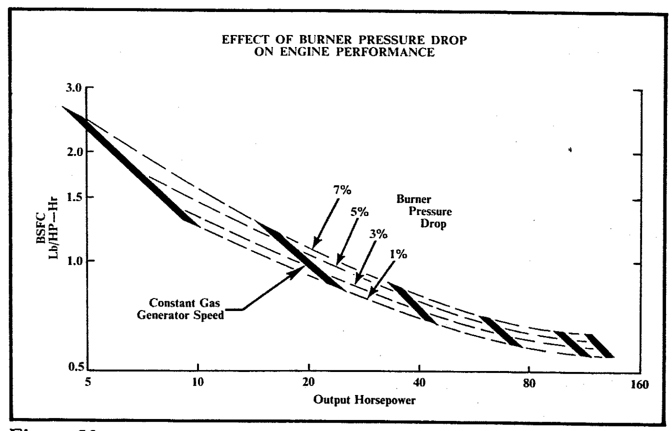


Figure 52

COMBUSTOR MATERIALS

Ceramics

A limited effort has been made to evaluate ceramic components in combustor systems. Two silicon nitride burner tubes were run with the baseline burner configuration. One silicon-carbide-coated graphite first stage reactor tube was evaluated with the dual stage concept. All developed cracks after short running times.

Such applications are desirable because of lower weight, thermal inertia, and potential low cost.

IN SUMMATION

1. Emission standards of 0.41 HC, 3.4 CO and 0.40 NOx can probably be met, on a laboratory basis, by a gas

turbine automobile of 4,000 pounds using present engine and fixed geometry burner technology.

2. Advanced gas turbines with improved fuel economy can certainly meet the above standards with present fixed geometry burner concepts.
3. The burner and fuel control system will operate satisfactorily, with equally low emissions, on fuels or mixtures of fuels ranging from unleaded gasoline to No. 1 Diesel without adjustment or modification.
4. A tank evaporative emission (HC) control system must be developed before gasoline or other volatile fuels can be permitted for general use. Additional effort may also be required to control HC emission from the lubrication system.
5. The combustion control techniques used should be equally valid with potential fuel alternatives such as alcohols or hydrogen, though probably requiring modification or adjustment from a system developed for conventional hydrocarbon fuels.
6. This level of emission control can be achieved with only small penalties in size, weight, or cost and without compromising fuel economy or performance characteristics.
7. Further development might be required for control of odor, sulfates or combustion noise if any becomes a problem.
8. A tremendous amount of development effort is required to translate these concepts into production designs with required durability, cold start ability, and margin for production tolerances.
9. Combustor pressure drop has substantial influence on engine performance and fuel consumption.

ENGINE CONTROLS

Baseline Hydromechanical Control

The baseline engine was equipped with hydromechanical controls for fuel metering and power turbine nozzle actuation. Additionally, a relay type start-safety protection package was utilized, which incorporated electronic over-temperature and light-off detection.

The fuel control provided the functions of steady state gas generator speed governing, start/acceleration fuel scheduling and fuel shut off on decelerations. As shown in Figure 53 the fuel control, driven by the gas generator, consists of a positive displacement pump, a pressure regulator, flyweight

BASELINE ENGINE CONTROL SYSTEM

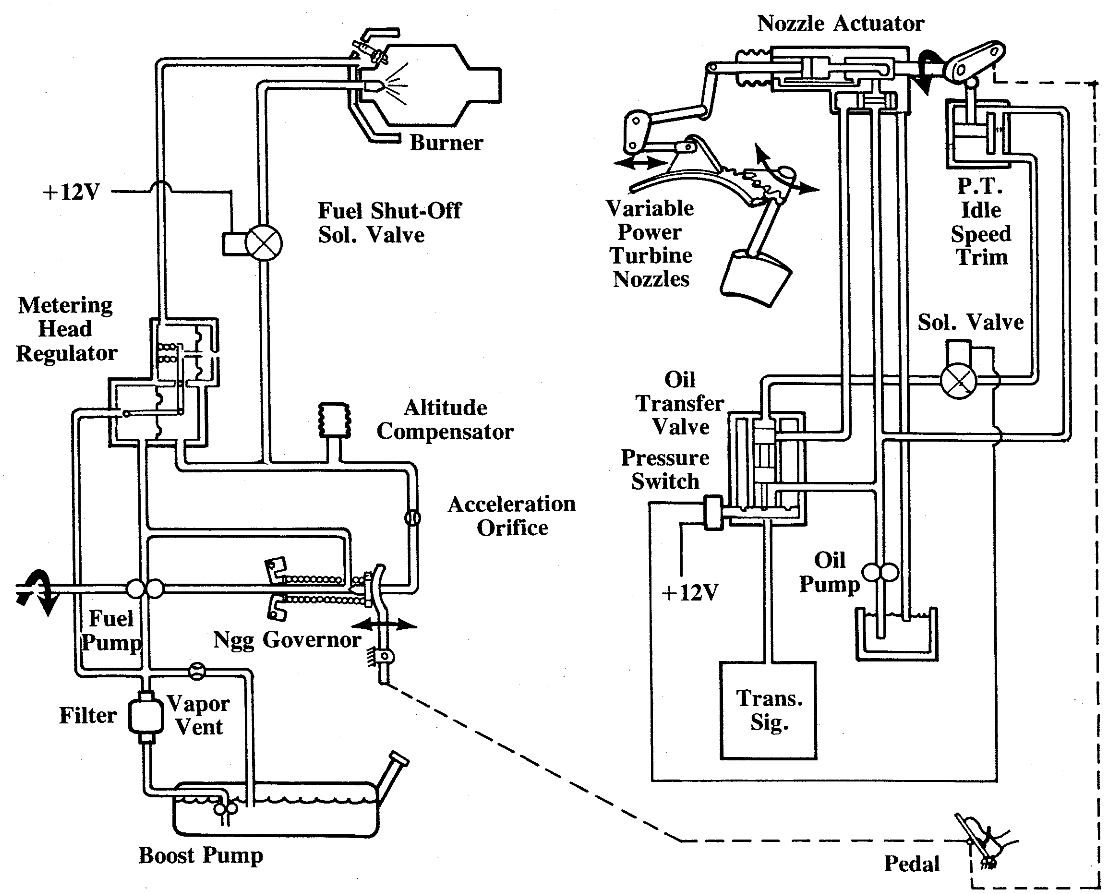


Figure 53

governor, an acceleration orifice and an altitude compensator. Start/acceleration fuel is scheduled as a function of compressor discharge pressure with altitude trim. Schedule changes were accomplished by orifice sizing.

The power turbine nozzle actuator, supplied with engine oil pressure, was a hydraulic positioner which provided modulation of the nozzles in accordance with a cam generated schedule based on turbine exhaust temperature (T8). This actuator also provided a fixed braking position of the power turbine nozzles for vehicle braking. Also shown in Figure 53 is the hydraulic circuit for the nozzle actuator. Transmission governor pressure is utilized (through a transfer valve) to provide a braking signal to the actuator. With vehicle speed above 15 MPH and the input arm of the actuator in the idle position, full line pressure is applied to the back of the piston to extend it to the braking position. A two-position idle is accomplished with an idle positioner to accommodate increased accessory loading, specifically the air conditioning compressor load.

In addition to the hydromechanical fuel control and power turbine nozzle actuator the gas turbine engine requires a safety system. This start-safety system provides automatic start sequencing upon initiation of the key switch. A no-start condition results if the transmission is not in the start/park position. There are four starting mode aborts as follows:

- Failure of the burner to light within 8 seconds. A light off constitutes a 100°F/Sec. rise on turbine inlet temperature (T_5).
- Failure to reach 25 PSI oil pressure within 5 seconds.
- An overtemperature (T_5>2000°F) for 2 seconds duration.
- Cranking time in excess of 20 seconds.

A restart can be made in the latter three cases, however, if the burner does not light off, a 30 second reset time is required to allow fuel to drain before restarting. The start safety system logic is shown in Figure 54.

These hydromechanical controls are simple, reliable and low cost, however, they are deficient in the following areas:

1. No allowance for ambient temperature compensation, power limiting or speed correction.
2. Turbine exhaust temperature (T_8) is based on a cam generated average temperature schedule instead of actual temperature. This does not allow optimum economy, emissions and performance due to variations in ambient temperature, engine leakage, engine conditions etc.

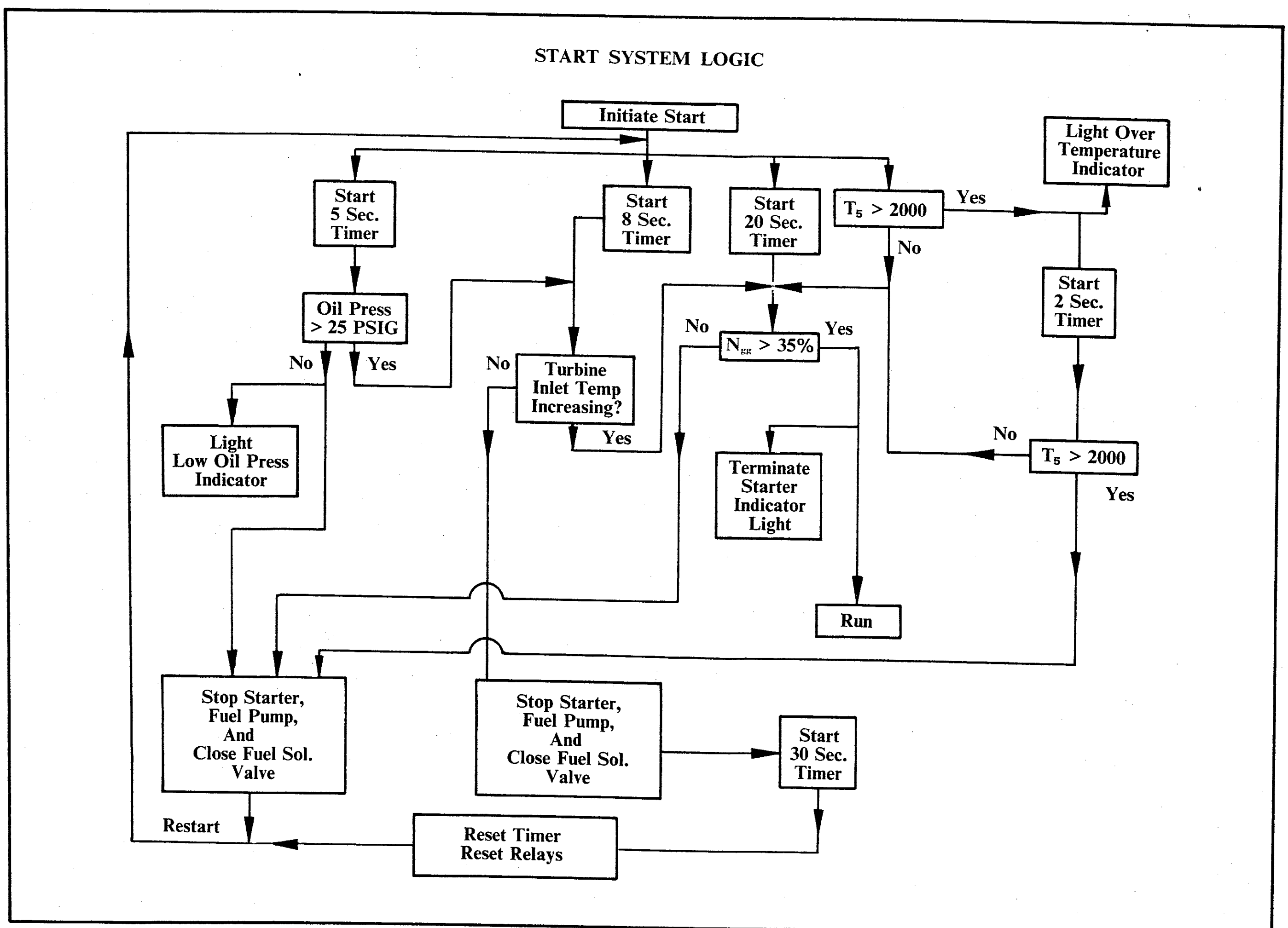


Figure 54

3. Adaptations of the hydromechanical controls for idle speed control and nozzle opening or "blipping" on gas generator acceleration requires complex additions such as linkage, solenoid valves, etc., and has resulted in performance less than optimum.

Additionally the control demands of the upgraded gas turbine engine, i.e., free rotor gas generator, variable inlet guide vanes, water injection and low emissions combustor require increased control complexity. Design and development of an integrated engine control system was therefore undertaken to investigate whether and to what extent these deficiencies can be overcome and new control requirements satisfied in a cost effective manner. This activity is reported in further detail in references (6) and (7).

This electronic approach was selected to provide flexibility in the development stage with final implementation dependent on the requirements.(8) Present electronic controllers, being of a developmental nature, are either hard wired or printed circuit board with discrete components. Considerations toward extending electronic control concepts and functions to high volume production include LSI (large scale integration) technology, microprocessors and digital implementation.

A listing of engine and control system parameters is shown in Figure 55 and a simplified block diagram of the electronic control system is represented by Figure 56.

The functions of the various control loops are listed below:

1. Fuel Control
   - Gas generator speed governing based on ambient correction.
   - Start/acceleration fuel scheduling as a function of $N_{gg}/\sqrt{\theta}$ with the start schedule altered by T_8.
   - Fuel shut off on decelerations with minimum decel flow at 10% speed error.
   - Turbine inlet temperature limiting (T_5) during starting and accelerations.
   - Maximum speed limiting for gas generator rotor and output shaft.
   - Output shaft idle speed governing in conjunction with either the power turbine nozzles or inlet guide vanes.
2. Nozzle Control
   - Temperature control (T_8) in power and braking mode.
   - Engine braking.
   - Output shaft idle speed governing.
   - Open nozzles for start and acceleration modes.
   - Open nozzles for loss of T_5 or T_8 sensors.
3. Inlet Guide Vane Control
   - Low speed power modulation.
   - Output shaft idle speed governing.
   - Power augmentation at maximum gas generator speed.
4. Water Injection
   - Power augmentation.
5. Start-Stop Logic

   The same functions of the baseline engine package were incorporated in MOS (metal oxide-semiconductor) digital logic.
6. Diagnostics

   The electronic control system incorporates a diagnostic connector with 55 test points to evaluate engine and control system performance. Test points include conditioned and unconditioned sensor inputs, computed schedules, logic commands and control system outputs.

ENGINE AND CONTROL SYSTEM PARAMETERS

T_5	Gas Generator Turbine Inlet Temperature
T_8	Turbine Exhaust Temperature
T_1	Ambient Temperature
P_1	Ambient Pressure
Ngg	Gas Generator Shaft Speed
$\frac{N_{gg}}{\sqrt{\theta}}$	Corrected Gas Generator Speed
δ	Ambient Pressure Correction = $\frac{P_1 \text{ PSIA}}{14.7 \text{ PSIA}}$
θ	Ambient Temperature Correction, = $\frac{T_1\ ^\circ R}{545^\circ R}$
N_{OS}	Output Shaft Speed
α	Throttle Pedal Position
β	Power Turbine Nozzle Angle Position
γ	Inlet Guide Vane Angle Position
W_f	Fuel Flow in Pounds Per Hour
V_V	Vehicle Velocity

Figure 55 Engine and Control System Parameters

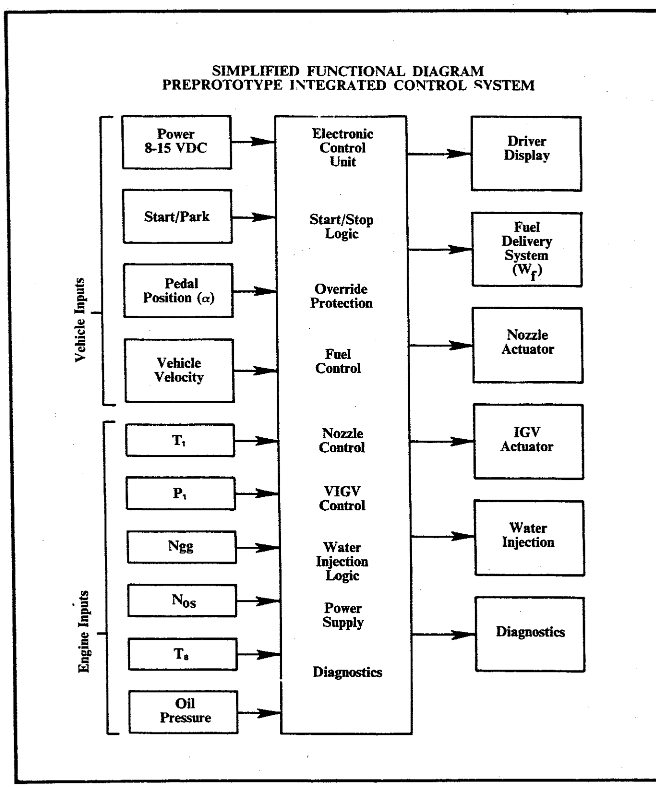


Figure 56 Simplified Functional Diagram Preprototype Integrated Control System

In conjunction with the development of the electronic control systems a computer simulation model of the baseline engine was constructed and utilized to assist in control concept definition and evaluation of steady state and transient characteristics. The engine model was effective in defining loop gains and stabilizing terms, however, some schedule changes were necessary during engine and vehicle testing to improve driveability and surge-free operation.

ELECTRONIC CONTROL IMPLEMENTATION

The electronic controller was configured with eight wired circuit boards using analog circuitry for control functions and digital logic for starting and engine protection. Pulsed output stages were utilized for the actuators and variable speed fuel pump motor. Various circuit modifications were required during early operation and involved such problems as power supply operation at low voltages, stable low speed operation of the fuel pump motor and noise interferences with the TTL (transistor-transistor logic) start logic. Other problems encountered were internal wiring connections as well as harness connections. Figure 57 shows the electronic controller in the test cell along with calibration equipment and instrumentation for diagnostic purposes. Figure 58 shows the controller as it was mounted in the trunk of the vehicle. The test cell installation of the programmable analog controller is shown in Figure 59. A self contained vehicle module is contained in the rack mounted configuration located on top of the pin programmable read only memory unit. This system utilizes analog sensor and output stages and performs computations sequentially in accordance with a clock controlled digital software program.

FUEL METERING COMPONENTS

The fuel metering was accomplished by a variable speed constant displacement pump. Fuel flow was metered as a closed loop function of pump speed. Fuel flows ranged from 5-180 PPH and the pump was turned off on gas generator decelerations. Later, with the combustor torch igniter requiring continuous fuel flow, a constant speed fuel pump was utilized with a proportional metering valve and volumetric flow sensor.

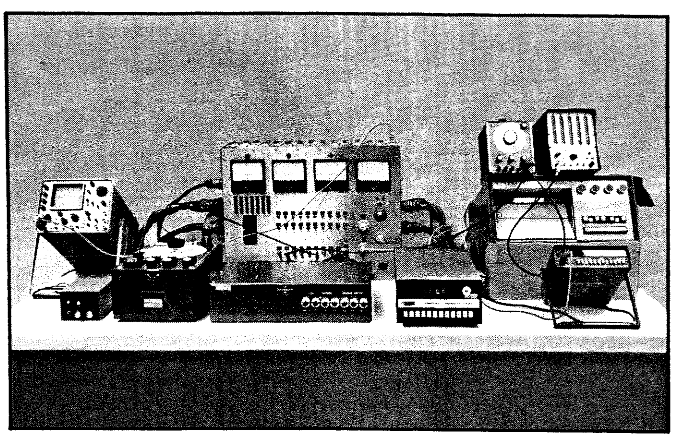


Figure 57 Integrated Control Diagnostic System

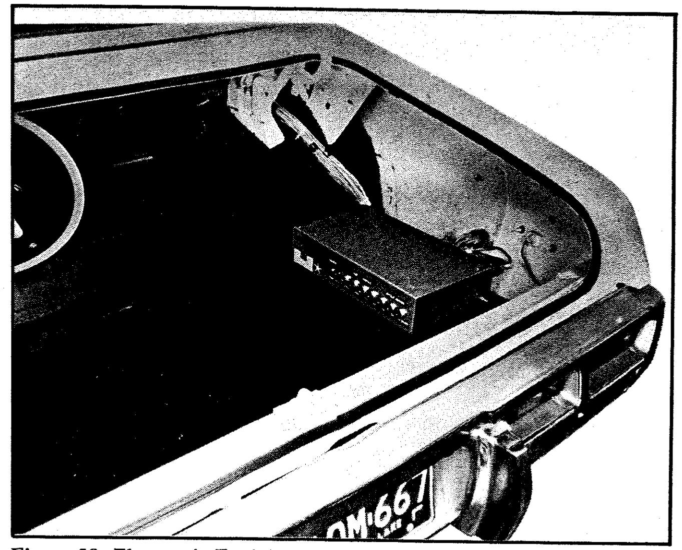


Figure 58 Electronic Fuel Control Vehicle Installation

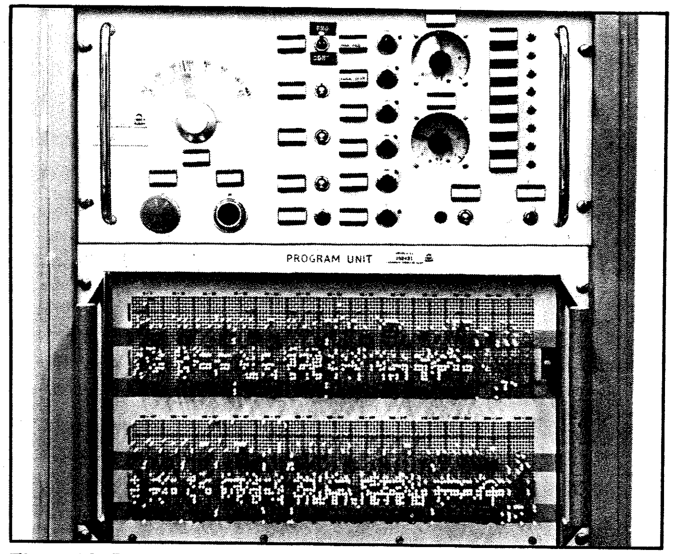


Figure 59 Programmable Analog Control System

SENSORS

Speed, position, temperature, flow and pressure sensors are utilized with the electronic control system. Speed signals (Ngg, Nos and Vv) were sensed with magnetic pickups, T_5 and T_8 were sensed with thermocouples and ambient temperature by means of a resistance sensor. Positions (α, $\Delta\beta$ and γ) were sensed with linear potentiometers and ambient pressure utilized a LVDT (linear variable differential transformer) diaphragm transducer. Fuel flow was sensed with a paddle wheel flow sensor. Some test activity was performed on fluidic and thermister high temperature sensors, however, response, accuracy and reliability problems have not been resolved, and continued development in this area is needed.

ACTUATORS

The electronic control systems utilized available actuators. For power turbine nozzle actuation an electrohydraulic actuator provided closed loop trim control

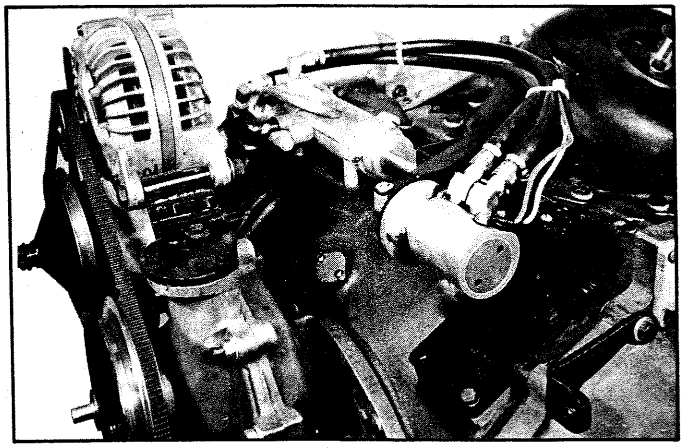


Figure 60 Power Turbine Nozzle Actuator Installation

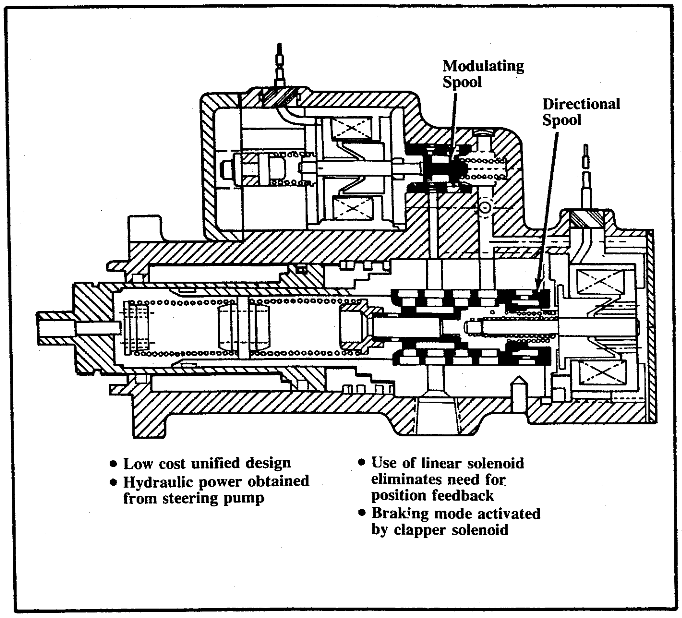


Figure 61 Upgraded Engine Power Turbine Nozzle Actuator

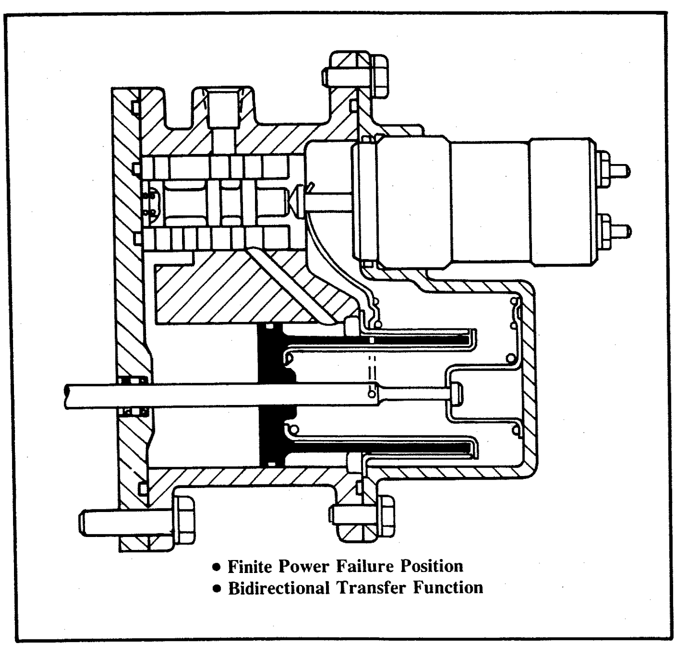


Figure 62 Upgraded Engine Inlet Guide Vane Actuator

in conjunction with the baseline hydromechanical actuator. A three-way solenoid valve selected the power or braking mode on the baseline unit. The inlet guide vane actuator was an extended stroke modification of the electrohydraulic trim actuator. The combined power turbine nozzle actuator assembly is shown in Figure 60. New actuators have been designed for the power turbine nozzles and inlet guide vanes on the upgraded engine and are shown in Figures 61 and 62. These actuators incorporate mechanical feedback to the main servo valve and eliminate the requirement for electrical position feedback.

ELECTRONIC ENGINE CONTROLS

In summation, electronic engine control was effective and was demonstrated on three vehicles. The programmable analog control with its ease of application and added flexibility proved to be a valuable development tool for control work as mentioned in reference (7). All control requirements of the gas turbine engine were integrated into a single electronic control unit and were optimized for performance, economy and emissions control. As mentioned in reference (6) the gas turbine engine requires greater control complexity when compared with a reciprocating engine. Further simplification of the control implementation is required along with added development of sensor interface elements. The higher acquisition cost of the electronic control package is only part of the more important total engine life cycle cost. It is in this area where cost effectiveness will be shown.

CERAMIC REGENERATORS

Background

Metallic regenerators have been developed and tested for several years prior to their use in the baseline engine. Maximum durability has exceeded 3000 hours in an endurance engine.

The concept of a ceramic regenerator however, offers potential advantages in a number of areas, and subsequent to the design of the Sixth Generation engine, substantial progress in the area of ceramic regenerator technology has been made. Specific areas of potential advantage are:

1. Improved engine efficiency by allowing increased engine operating temperature.
2. Lighter weight—approximately one-half that of a metal regenerator.
3. Higher effectiveness at low flow rates, due to lower axial heat conduction.
4. Simpler sealing because of low thermal distortion.
5. Potentially lower cost due to the use of non-critical materials.

Some original work was done in this field as early as 1960. These tests were confined to the regenerator fixture, and the matrices of that time showed deficiencies of a technological and material nature. Thinner, stronger stock and finer, more uniform passages were required. These preliminary studies indicated that ceramic regenerators would satisfy the automotive gas turbine requirement if satisfactory technological progress could be made. This, of course, would involve development not only of matrix material and shape, but also of suitable drive, suspension, and sealing systems.

During the late 1960's and early 1970's, ceramic regenerator work was deferred pending development of a reliable design by the ceramics industry. By 1973, such designs were readily available, and the task of converting the baseline engine was begun.

CONVERSION TO CERAMIC REGENERATORS

When the decision was made to convert the baseline Sixth Generation engine from metallic to ceramic regenerators, several basic guidelines were established:

1. The ceramic regenerators should fit in the existing envelope with a minimum of mechanical changes.
2. Seals, drive gear, and other related components should closely follow the established, trouble-free designs proven in the baseline engine.
3. The conversion should take place with a minimum of delay, necessitating the use of off-the-shelf core and seal materials.

MATRIX SELECTION

Comparison of theoretical performance between several versions of available ceramic matrices and the current metal regenerator indicated that, for the same size and pressure drop, the ceramic core should be superior to the metal core at idle and part throttle conditions. The best overall performance should be from a ceramic matrix with rectangular passages. Results of this study are shown on Figure 63.

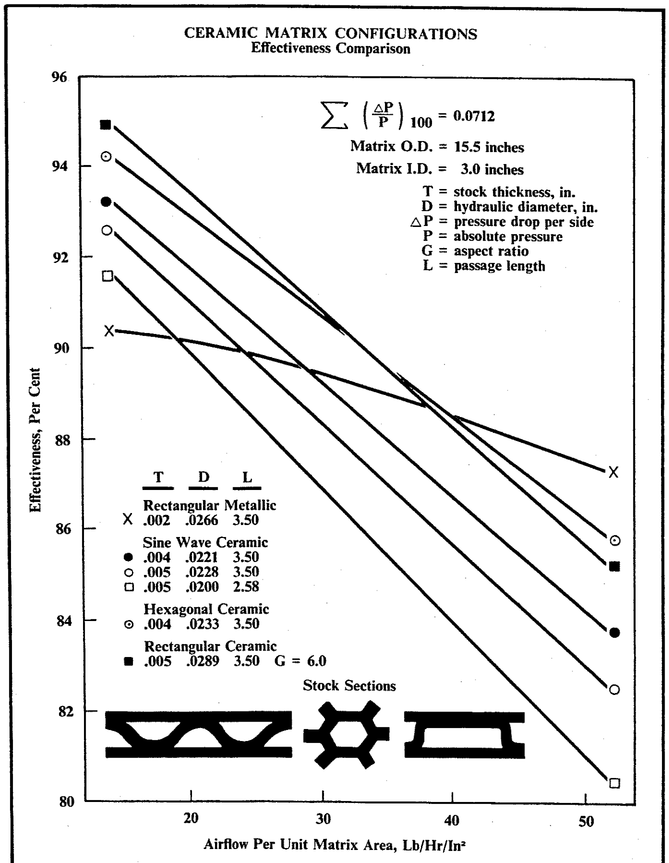


Figure 63

On the basis of the above comparison, specifications were drawn and discussed with prospective vendors. Ceramic technology was not then capable of producing a rectangular matrix, but two alternate types were proposed:

Type "A," a wound triangular or "sine wave" shape, and

Type "B," nested glass tubes forming a hexagonal shape.

Samples of currently-available matrices of both types were inspected for pressure drop, and the final selection of each type matrix was based on its closeness of match to the baseline engine's pressure drop. Since envelope constraints required use of the same overall diameter as the Sixth Generation metallic regenerator, matrix thickness was adjusted as necessary to correct the pressure drop. Final matrix designs selected are shown on Table 2. Baseline metallic matrix specifications are shown for comparison.

Table 2
BASELINE METALLIC AND CERAMIC MATRIX SPECIFICATIONS

Matrix Configuration	Hydraulic Diameter	Stock Thickness	Passage Length	% Open Area
Metallic (rectangular)	.028 in.	.0020 in.	3.50 in.	85
Type "A" (triangular)	.025 in.	.0045 in.	2.58 in.	67
Type "B" (hexagonal)	.027 in.	.0045 in.	3.00 in.	67

REGENERATOR PROCUREMENT AND INSPECTION

Several ceramic cores of each type were procured, inspected, and fixture-tested prior to installation in the engines. Inspection procedures were patterned after those developed for metal regenerators. Preliminary screening for acceptable pressure drop is done with the cold flow fixture shown in Figure 64. Metered air is introduced to a known area of the matrix, and allowed to exhaust to atmosphere. Using the pressure, temperature, and flow conditions in the flow cup, the pressure drop under engine operating conditions may be calculated as shown in Figure 65. These data are used not only for core acceptance, but also for ensuring that each engine is fitted with a matched set of cores.

Following flow check, a leak test fixture is used to check for porosity and internal voids. A cylindrical volume of matrix is sealed at each side of the core, and pressurized. A flow meter measures the make-up air required to maintain pressure.

Each regenerator is then dimensionally checked, with particular emphasis on surface flatness. Close-up photographs of the matrix are taken for computer-aided graphical analysis of passage geometry.

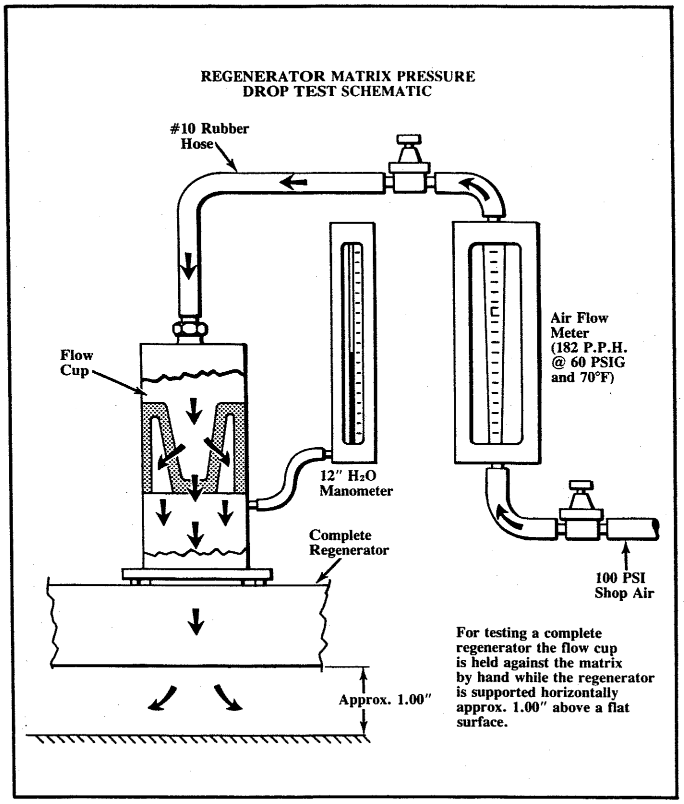


Figure 64

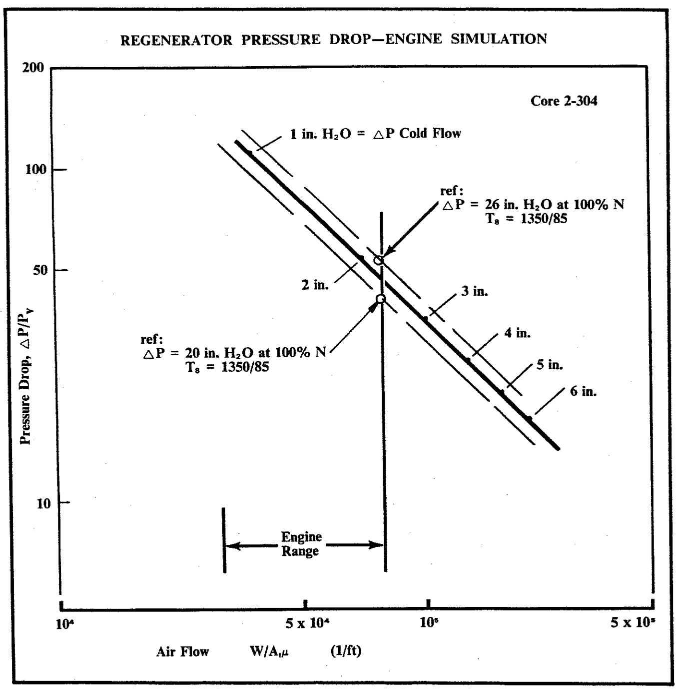


Figure 65

Finally, each regenerator is fitted with a ring gear and tested in the regenerator fixture. This fixture is designed to match, as closely as possible, actual engine conditions. Compressed air is first heated to simulate the required compressor discharge temperature, and then passed through the high-pressure side of the matrix, where it picks up additional heat. The air is then throttled to simulate the pressure drop across the turbine wheels, and heated to a fixture-limited T_8 of 1200°F. The hot air then passes through the low-pressure side of the matrix, where it gives up much of its heat before being exhausted to atmosphere. A variable-speed external drive system is used to rotate the regenerator, and drive speed and torque are monitored to evaluate seal coefficient of friction. Standard ASME metering orifices are used to check airflows throughout the fixture, permitting accurate measurement of seal leakage. Temperatures are recorded by means of thermocouple grids near both faces of the matrix, and effectiveness is calculated, corrected through a computer program for actual engine T_8 conditions of 1350°F. Typical effectiveness curves for baseline type "A" and "B" matrices are shown on Figure 66.

EFFECTIVENESS vs. AIRFLOW FOR TYPE "A", TYPE "B", AND BASELINE METALLIC MATRICES

	Hydraulic Diameter In.	Stock Thickness In.	Passage Length In.	% Open Area	ΔP "H_2O
× —— × Baseline Type "A" Wound Triangular (Sine Wave)	0.025	0.0045	2.58	67	24
⊙ —— ⊙ Baseline Type "B" Nested Tubes (Hexagonal)	0.027	0.0045	3.00	67	24
△ —— △ Thinwall Type "A" Wound Triangular (Sine Wave)	0.020	0.0030	3.00	76	32
— — — Baseline Metallic	0.027	0.0020	3.50	85	24

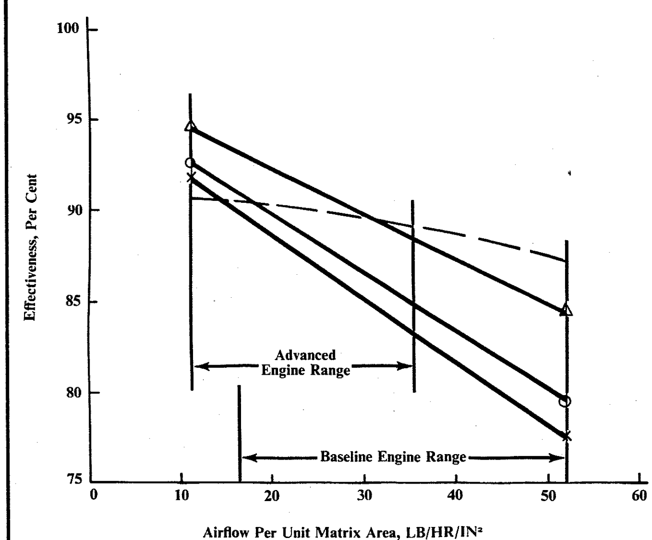


Figure 66

DRIVE SYSTEM

Drive and suspension methods are similar to metallic regenerator practice. Optimum seal performance requires freedom of matrix movement in all directions except radially. To achieve this freedom, the spherical graphite bearing of the metal core has been replaced by an elliptical graphite sleeve sliding in a cylindrical bore in the ceramic hub. The core is rim-driven using a drive pinion and ring gear identical to those used with metal cores. Past experience with a variety of suspension and drive systems has demonstrated the superiority of the center-support-rim-drive concept with metal cores, and this method has worked equally well with ceramics.

Two different methods of attaching the metal ring gear to the ceramic matrix rim have been tried:

1. Mechanical mounting was used successfully with early type "A" matrices. This method utilizes solid ceramic drive pins cemented in the matrix rim, which are engaged by spring-loaded metal drive shoes suspended from the gear and rim. This method has the best life potential at extreme temperatures, but the cost penalty is greater, and acceptable gear runouts are difficult to achieve.
2. Elastomeric mounting, now the preferred method, is accomplished by fixturing the gear and its rim concentric with the matrix, and injecting silicone rubber into the annular space between them. This method is both economical and accurate—ring gear runouts of less than .010" are typical—but requires careful design to keep the elastomer temperature below 550°F. More detailed information on elastomer potential is given under "Elastomeric Drive Development," below. Figure 67 illustrates a typical elastomeric mount.

SEAL SYSTEM

In keeping with the conversion guidelines, standard diaphragm-type baseplates were used for both inner and outer seals, as shown in Figure 68. Inner rim and outer "D" rubbing seals are high-temperature graphite from the baseline engine, and only the crossarm seal coating has been changed to achieve compatibility with the ceramic matrix. The coating selected was 85% nickel oxide and 15% calcium fluoride, which was the best available material at that time. This coating has the advantages of low coefficient of friction ($\simeq$0.2) and extremely low wear rate (theoretical coating life $\simeq$40,000 hours), but alleged health hazards attributed to nickel compounds dictate the need for the eventual replacement of the nickel oxide. Chrysler's program to perfect a suitable substitute is discussed under "Seal Coating Development," below.

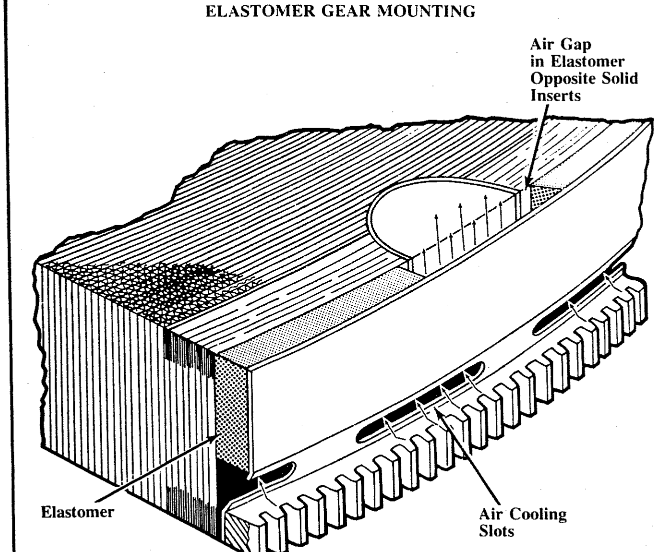


Figure 67

SYSTEM PERFORMANCE

Half of the baseline engines have been fitted with ceramic regenerators, and 1600 core hours have been logged to date. As shown on Figure 66, baseline regenerators of both types "A" and "B" show the expected fuel economy penalty at higher powers. As noted above, envelope constraints dictated that, for equal pressure drop, the ceramic cores would have shorter than optimum passage lengths, with corresponding loss of heat transfer ability. As discussed under "Advanced Regenerator Design," below, increased matrix area (reduced airflow per unit area) can produce idle and part-power fuel economy well in excess of that attainable with a comparable mass of metallic matrix. Regenerators and seals have performed well throughout the program, showing no signs of distress after 300 to 500 hours. An isolated low-time (6 hours) failure of a type "B" matrix occurred in the regenerator fixture, but this failure was attributed to internal stresses resulting from a hub change, and would not be characteristic of a typical production item.

One of the baseline vehicles, the Free Rotor engine, is currently running with ceramic regenerators. Over 1100 miles—equivalent to 50 engine hours—have been logged to date. Regenerator and seal systems have been trouble-free.

ACCELERATED LIFE TESTING

Several regenerators of both types have been endurance-tested for hundreds of hours under the severe accelerated wear and thermal shock test cycle. This cycle,

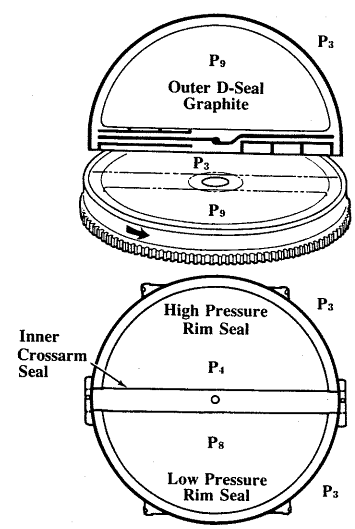


Figure 68

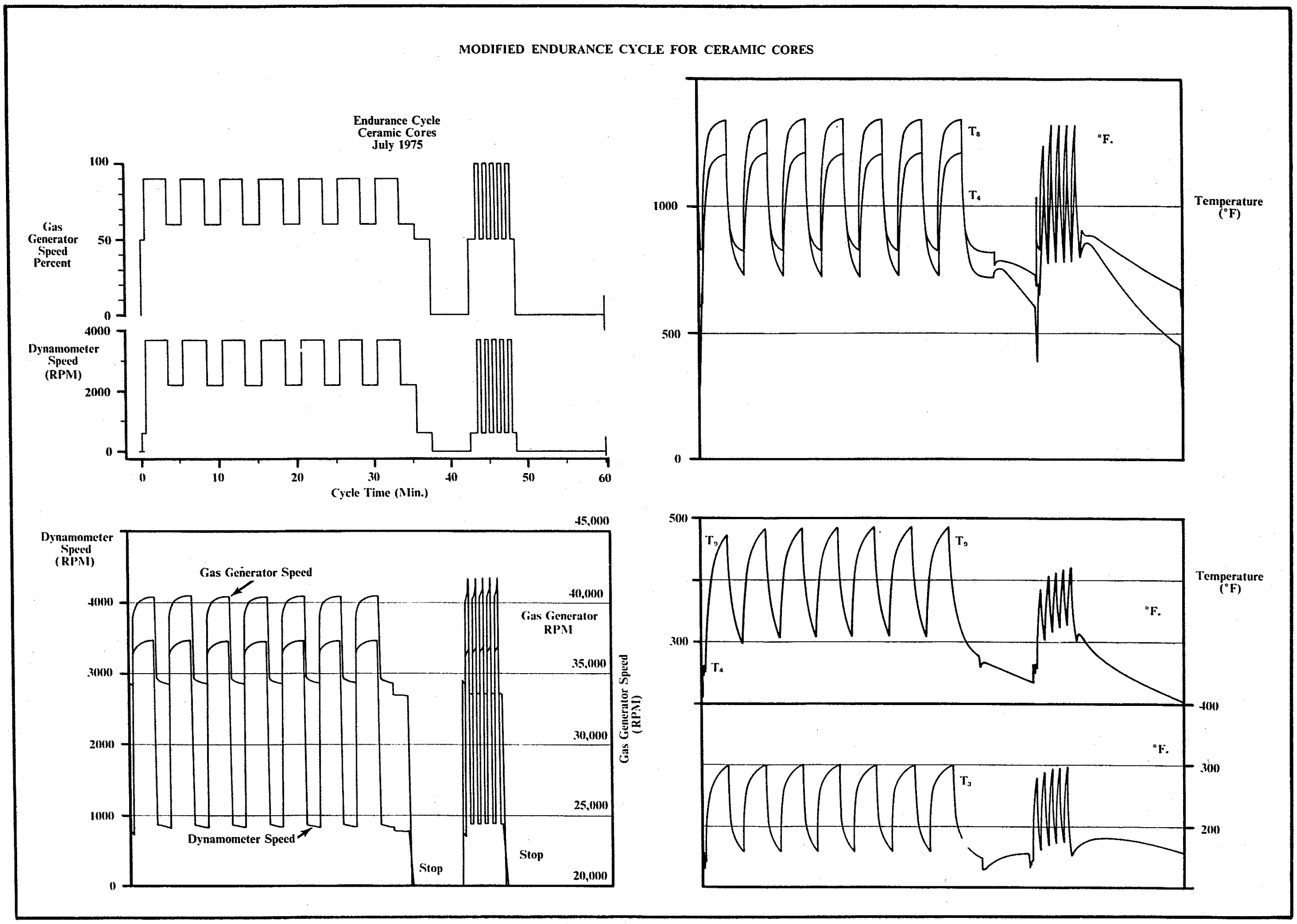


Figure 69

shown on Figure 69, alternately subjects the matrix to 1400°F at high power conditions, followed by rapid cool-down to 900°F idle. The thermal transients in this test sequence are far more extreme than any anticipated vehicular duty cycle, and are designed to screen various matrix configurations for susceptibility to thermal fatigue cracking, see Figure 70. A 60% to 100% speed acceleration was included to subject the elastomeric core ring gear attachment system to high torque conditions. The engine was subsequently modified to have the regenerator cores driven directly from the power turbine reduction gearing independent of the gas generator to simulate the free rotor concept utilized in the upgraded engine. This drive arrangement results in a more severe matrix temperature gradient under conditions of cold engine start-up followed by a rapid demand for power. Testing is continuing on upgraded regenerator assemblies. To date, 334 endurance cycles have been completed on a type "A" matrix, and 144 cycles on a type "B," without a failure.

A torque measuring system was designed for installation in the regenerator drive train. This permits determination of actual regenerator core torque requirements at engine operating conditions. The relative coefficient of friction of the crossarm seals will be observed with this arrangement.

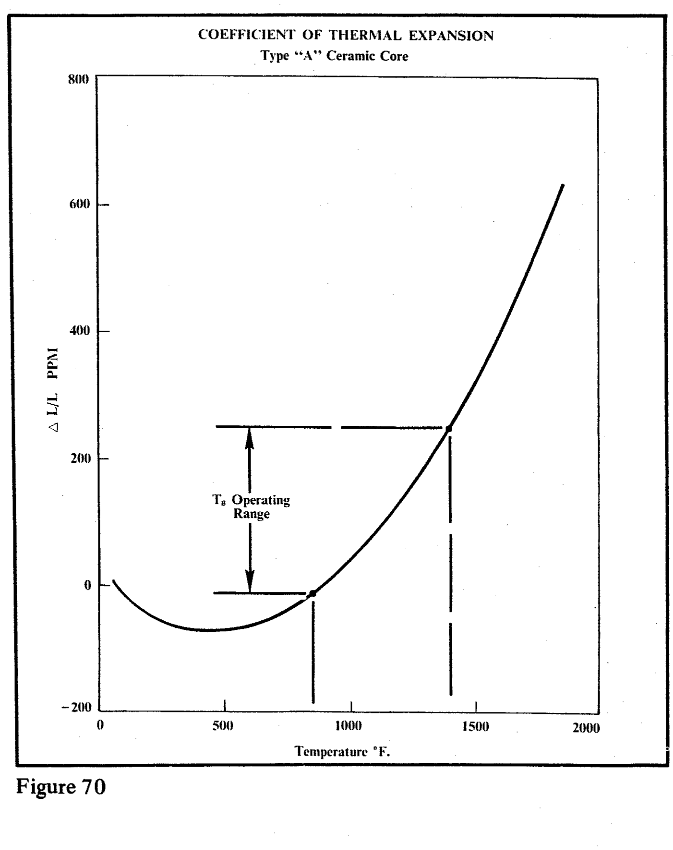


Figure 70

SEAL COATING DEVELOPMENT

As discussed above, concern has been expressed over the use of nickel oxide as a seal coating material because of the alleged health hazard of nickel compounds. While the amount of nickel oxide in the baseline engine's exhaust is extremely small—on the order of 1 $\mu g/m^3$—Chrysler has continued its basic research in the area of seal coatings, seeking an acceptable substitute. A number of potential materials have been tested, as listed in Table 3. Fixture testing consists of rotating a small matrix sample against a simulated seal in an electrically heated furnace. Rubbing speed, seal load, and operating temperature may be set to match any anticipated engine condition, and drive torque is continuously recorded. Figure 71 shows the results of several of these tests, and illustrates the lower coefficient of friction of the type "A" matrix as compared to the type "B", regardless of the coating used. This phenomenon has been verified on fixture test of full-sized regenerators, where typical drive torque of 70 lb-ft (100% Ngg, type "A" matrix) compare to 90 lb-ft for type "B". It is believed that this discrepancy results from the basic design of the matrix: The nested glass tubes are less porous than the wound ceramic, and the calcium fluoride, which acts as a dry lubricant, is less readily embedded in the glass.

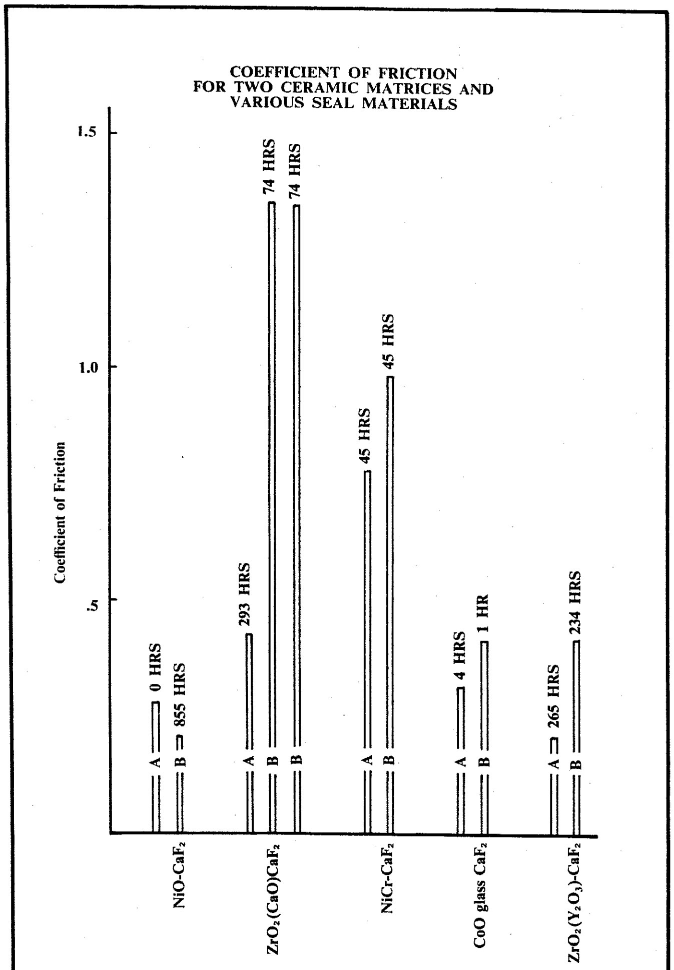


Figure 71

To date only one advanced coating—calcia-stabilized zirconia—has been engine tested. Wear rate is considerably higher than with nickel oxide, as shown on Table 4, but the projected life of 2300 hours is still quite adequate for a development program. However, the higher torque required (110 lb-ft) produced excessive gear wear, and the search for a coating material with better properties continues.

Table 3
POTENTIAL SEAL COATING MATERIALS

Seal Composition	Mode of Application
$NiO\text{-}CaF_2$	Plasma Spray
$ZrO_2(CaO)\text{-}CaF_2$	Plasma Spray
$CoO\text{-}BaO\text{-}B_2O_3\text{-}CaF_2$	Glass Glaze
$CoO\text{-}BaO\text{-}B_2O_3\text{-}CaF_2$	Glass Glaze
$NiCr\text{-}CaF_2$	Plasma Spray
$CoO\text{-}CaF_2$	Plasma Spray
$ZrO_2(Y_2O_3)\text{-}CaF_2$	Plasma Spray
$Li_2O(ZnO)SiO_2$	Glass/Ceramic

Table 4
Crossarm Seal Wear Rates
Engine Endurance Test

Test Conditions	Wear Rate (μm/hr)	Seal Life (hr)*
Type "A" Matrix $NiO\text{-}CaF_2$ Seal 250 hours	.022	40,000
Type "B" Matrix $NiO\text{-}CaF_2$ Seal 250 hours	.056	16,000
Type "A" Matrix ZrO_2 (CaO)-CaF_2 170 hours	.380	2,300

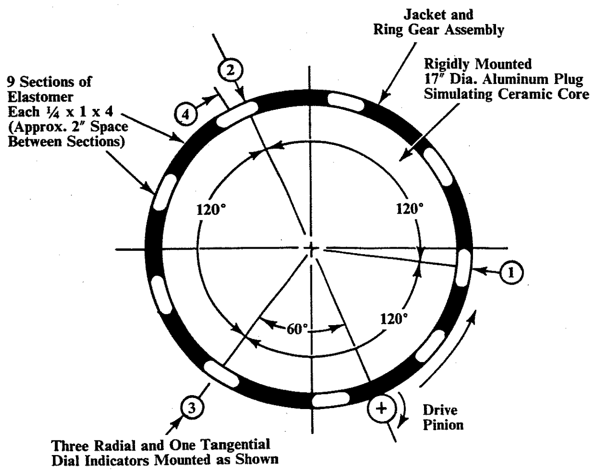
*Based on nominal coating thickness of .035 in. (890 μm).

ELASTOMERIC DRIVE DEVELOPMENT

All ceramic regenerators now on test use the elastomeric mounting concept. During development of this method, a number of regenerators exhibited small areas of cohesive failure of the elastomer. These tears originated at stress risers such as bubbles in the elastomer surface, and enlarged very slowly over a period of several hundred test hours, until the mount was deemed unsafe for further running, and replaced. These failures are believed to be caused by a gradual shrinkage of the material over an extended period at high temperature, together with a gradual loss of properties. A comprehensive program has been established to fully document changes of properties with heat aging, and a concurrent design effort will examine possible changes in elastomer thickness and gear rim arrangement. If necessary, fillers will be added to the elastomer to achieve the desired properties.

It should be noted at this point that the basic concept of this mount is a sound one. To verify the integrity of the design, a mount was static tested to failure using the arrangement shown on Figure 72. Gradually increasing torque was applied to the drive pinion, and ring gear deflections noted. The mount sustained a load of 720 lb-ft, as compared to a typical maximum torque of 70 lb-ft.

DEFLECTION TEST
ELASTOMERIC MOUNTING



Torque lb-ft		Deflection at Each Indicator, In.			
		1	2	3	4
At Pinion	At Core	Radial	Radial	Radial	Tangential
5	90	+.005	+.001	-.006	+.012
10	180	+.009	+.001	-.011	+.047
15	270	+.016	+.001	-.017	+.074
40	720	(BOND TO METAL FAILED)			+.175

Figure 72

IMPROVED EFFECTIVENESS MATRIX DEVELOPMENT

Once the feasibility of operating the baseline ceramic regenerators in the baseline engine was proven, the next step was to test matrices of increasingly higher effectiveness, and actually demonstrate the performance improvement shown possible by the theoretical studies

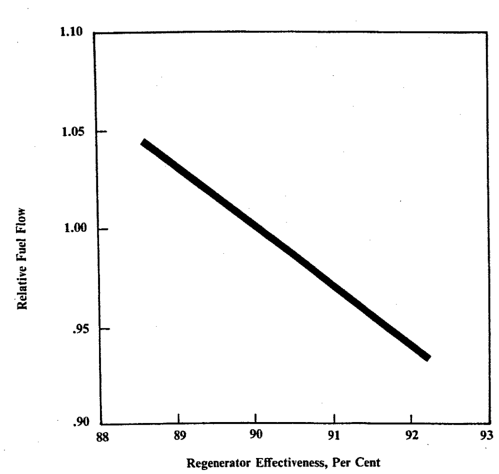


Figure 73

illustrated in Figure 63. As shown on Figure 73, 1% increase in regenerator effectiveness results in a 3% reduction in idle fuel consumption. For this part of the program, it was decided to work with type "A" matrix, in view of its proven lower torque requirements and potential for the greatest reduction in wall thickness. Several cores were tested with increasingly thinner walls and smaller hydraulic diameters, to the limit of the vendor's tooling capability. New tooling was made on the basis of the earlier test results, and the final set of cores achieved the program goals of .003 in. wall thickness, .020 in. hydraulic diameter, and 76% open area. The fixture test results are shown on Figure 66. The upgraded engine ΔP sized for the baseline engine was 32 in. H_2O, somewhat higher than the baseline engine's design point of 24 inches, resulting in a power loss of 8 HP. The gain in engine fuel economy is substantial, as shown on Figure 74. Referring back to Figure 66, it can be seen that the advanced engine, scheduled for test early in 1976, will have a lower airflow per unit matrix area, reducing the ΔP to the design value. At these low airflows the effectiveness of the ceramic matrix shows the greatest improvement over the metallic type. These advantages, combined with the benefits from higher cycle temperature made possible by the ceramic matrix, show the ceramic regenerator to be a strong contender among potential automotive turbine heat exchangers.

POWER AUGMENTATION

Variable Inlet Guide Vanes

Incorporating variable compressor inlet guide vanes into the engine system permits an increase in airflow at high speed and a reduction in flow at low speeds. This arrangement offers the possibility of a smaller basic engine design resulting in improved fuel economy. Such an improvement is obtained by (1) reducing engine air flow at idle and (2) requiring that the engine operate at a higher cycle pressure ratio at any given output load requirement.

Use of guide vanes at idle to further reduce flow results in higher operating temperatures, and depending on the level of change in compressor efficiency, might also result in a fuel savings. Higher idle temperature of itself is desirable for reasons of:

1. Fuel savings under normal cyclic engine operation by eliminating cooling and heating of the regenerator.
2. Reducing the amount of fuel required to achieve acceleration temperatures.
3. Reduction of CO and HC emissions.

In addition to a reduction in engine size, positive preswirl at low speeds tends to shift the compressor characteristic to lower flows (throttling), enabling higher turbine inlet temperatures with an attendant rise in thermal efficiency (fuel economy). A more detailed discussion on the ramifications of VIGV augmentation can be found in references (1) and (9).

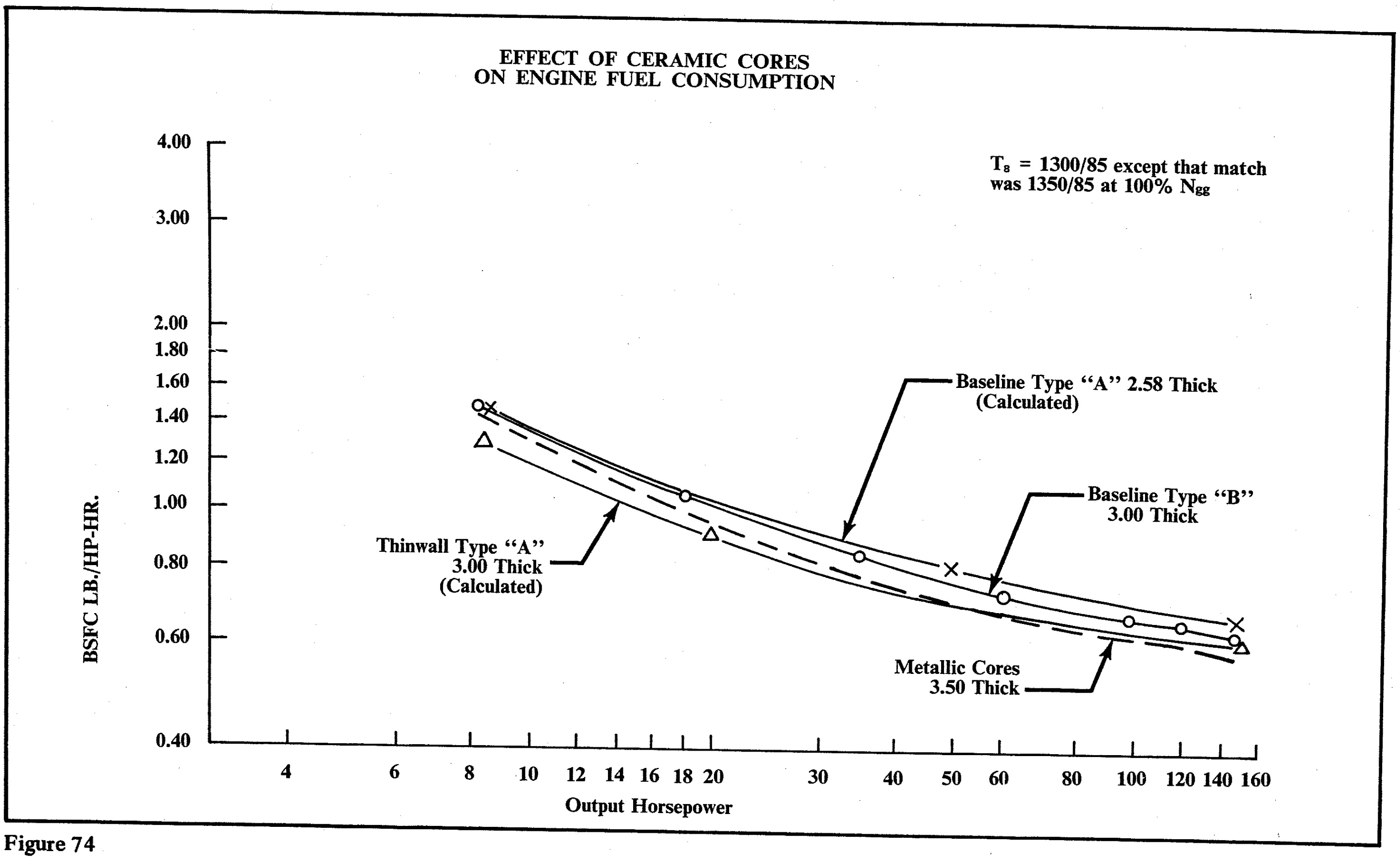


Figure 74

Guide varies augment compressor performance through their ability to alter the change-in-angular-momentum; i.e., the Euler work, of the impeller ($U_1 Vu_1—U_2 Vu_2$).

By deflecting the inlet flow either in the direction of (positive preswirl) or opposite to (negative preswirl) the rotational direction of the impeller, the Euler work is reduced or increased, respectively. Inlet guide vanes, therefore, simply create an inlet tangential component of velocity either positive or negative, depending on the direction of flow turning. Increased work results in increased flow and pressure.

The adaptation of guide vanes to the baseline engine is shown in Figures 75 and 76. A more detailed description is contained in Reference (9).

ENGINE PERFORMANCE

A 50% speed performance map for various guide vane positive deflection angles is shown in Figure 77. This shows that at any given idle power level a slight improvement in fuel economy occurred with increasing guide vane deflection. The ability to maintain high idle temperature without generating excessive idle power is also illustrated. The 50% speed compressor map over this range of operation is shown in Figure 78. This includes a 5 HP operating line to illustrate the extent of compressor efficiency decrease with increased deflection. As discussed in Reference (9), turbine efficiency shifts to a slightly more efficient map point with positive guide vane deflection.

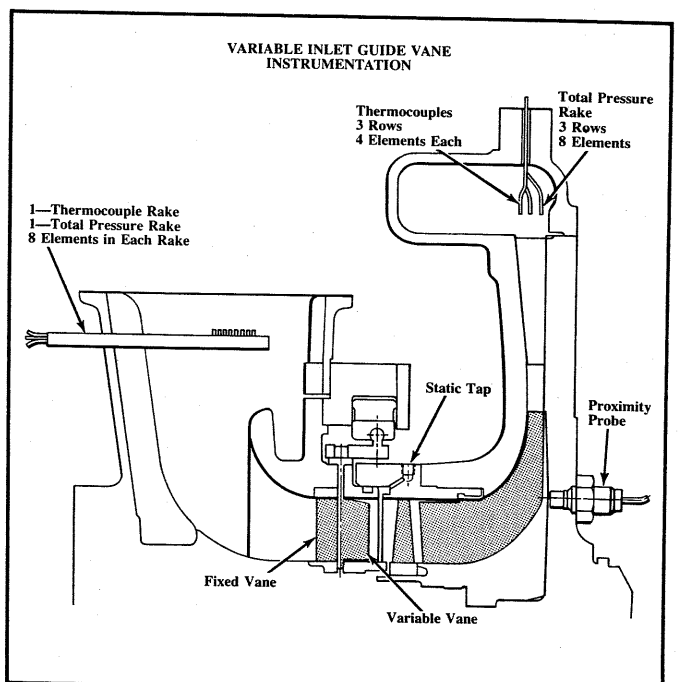


Figure 75

Power augmentation at 100% speed is shown in Figure 79, and the corresponding compressor map in Figure 80. As discussed in Reference (9), for this baseline engine with its high work speed parameter compressor turbine ($\Delta Cu_i/U$ = 2.4) the 4% power boost is about all that should be expected. The reason is that the discharge velocity from such a turbine is normally high, and when driving an augmented compressor becomes significantly higher. Since this velocity is input to the power turbine section, power turbine stage efficiency drops off rapidly as augmentation increases. However, as also discussed in the reference, with a slightly more conservative compressor turbine design ($\Delta Cu/U = 2.1$), as is being used in the upgraded engine of Reference (1), power turbine inlet velocity level is lowered sufficiently such that 10% augmentation should be obtained.

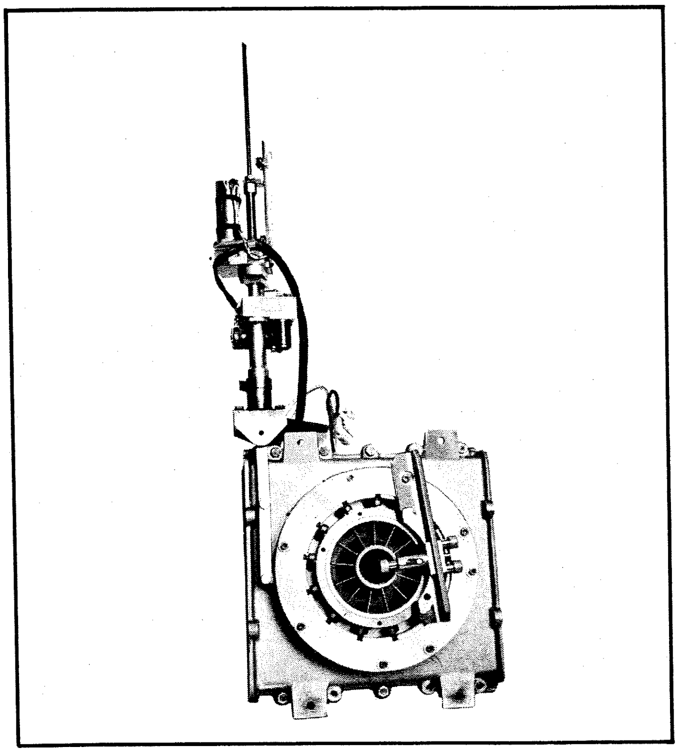


Figure 76 Calibration of Inlet Guide Vane Assembly

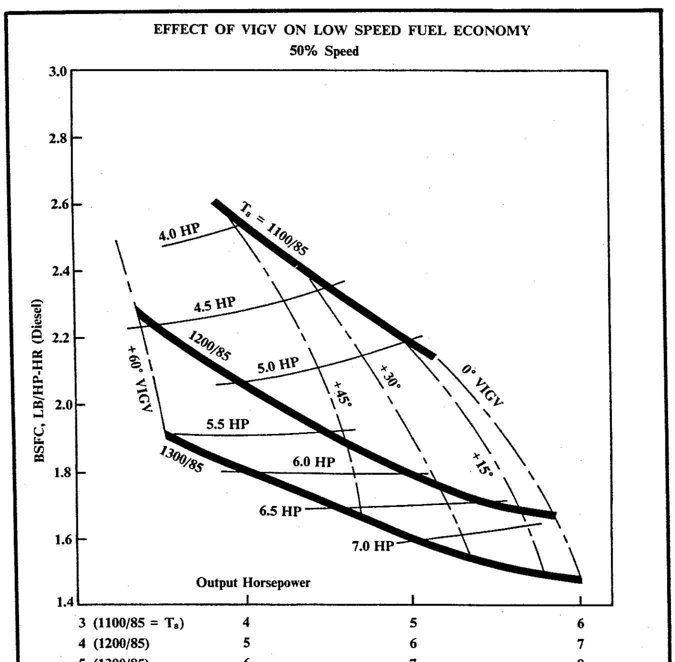


Figure 77

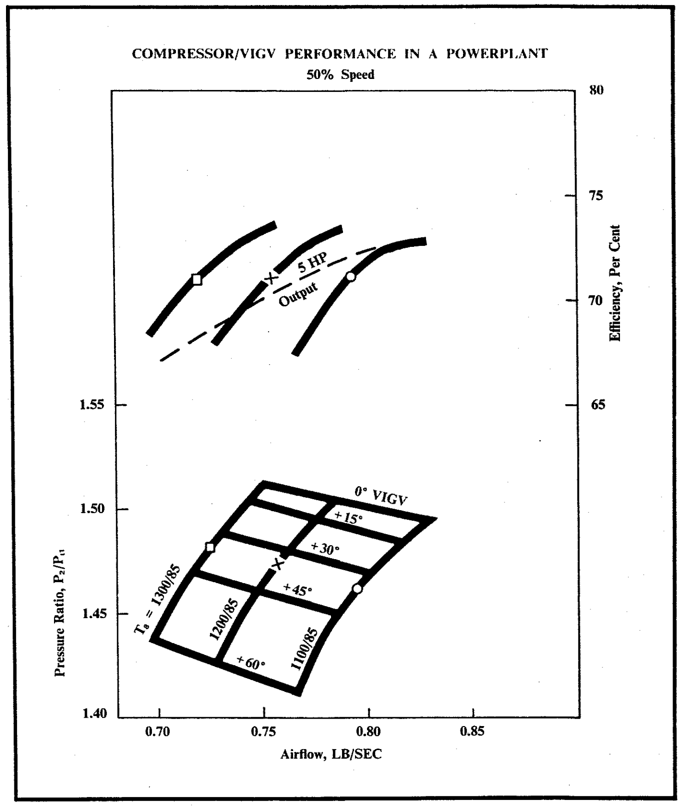


Figure 78

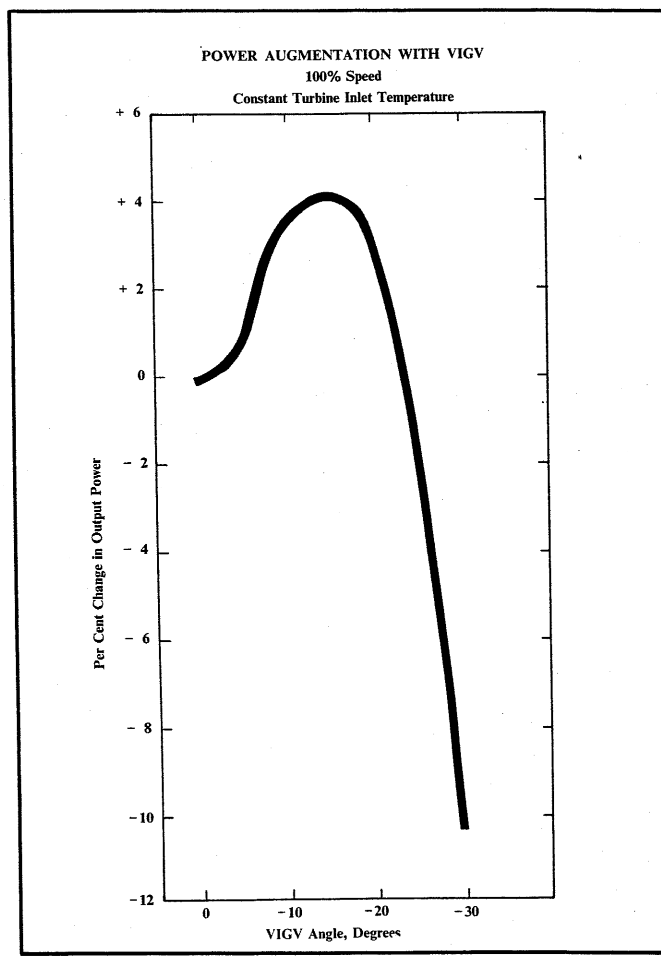


Figure 79

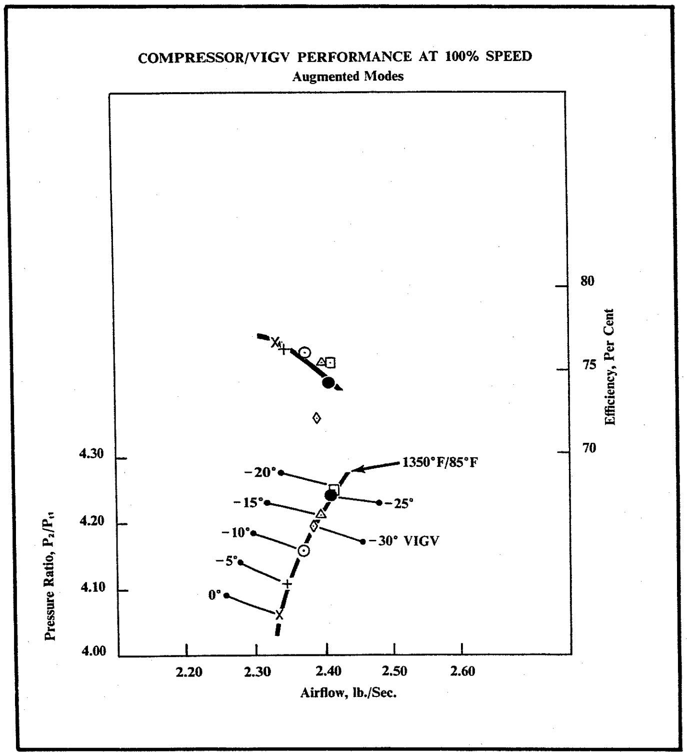


Figure 80

WATER INJECTION

Although water injection for aircraft gas turbines is commonly used, little data exists pertaining to regenerative automotive gas turbine applications.

Problems of hardware packaging, engine characteristics, vehicle response, and component integrity were to be investigated.

The addition of water is essentially an evaporative process to: 1. depress compressor inlet temperature to allow running over corrected speeds to increase engine mass flow, and 2. absorb heat during the compression process (intercool). If water injection was to have application to this engine, augmented power goals must be met at 100% inlet humidity conditions.

ENGINE TEST

Modified air atomizing fuel nozzles were used initially as these were readily available and allowed engine tests to commence. The resultant spray was exceptionally soft and fine. The nozzles were located approximately 2 ft. upstream of the impeller eye but downstream of the filters. This was arbitrarily selected to allow sufficient length for mixing and evaporation. The fiberglass intake liners were coated to prevent erosion of these parts. Distilled water was delivered and metered to the intake. Distilled water is used exclusively as prior feasibility tests with tap water resulted in contamination by mineral deposits.

Humidity to the engine inlet air plenum was controlled by an upstream steam coil. Additional compressor inlet air thermocouples were used upstream of the spray nozzles to set compressor speed during water addition.

Typical engine characteristic curves are shown in Figures 81a & 81b for a constant corrected turbine inlet temperature. Temperature was adjusted as required via the power turbine variable nozzles.

A water rate of 100 pph yields an optimum augmented power increase of 10%. The flatness of the power curve at higher injection rates is due to loss of power turbine efficiency as the nozzles must be opened to control temperature with increased water rates.

BSFC is nearly flat over the test range with pressure ratio increased 5%. The compressor discharge temperatures T_2 were proportionately reduced with added water flow and represent average values. These temperature levels were relatively stable to 150 pph water flow but become increasingly erratic at higher water flows. This would imply that the cooling process is moving further into the impeller with resultant non-uniformity entering the diffuser. Lower NOx emissions with the increased water rates are attributed to a reduction of peak combustion temperatures. CO levels were relatively unaffected, hydrocarbon levels were reduced at a rate similar to NOx. These results are shown in Figure 82.

Engine oil sump temperatures were depressed with increased water rate and are due to the compressor section and surrounding structure becoming the heat sink with added water rate. As a result, the heat path is directed towards these components and not into the oil cooled support housing.

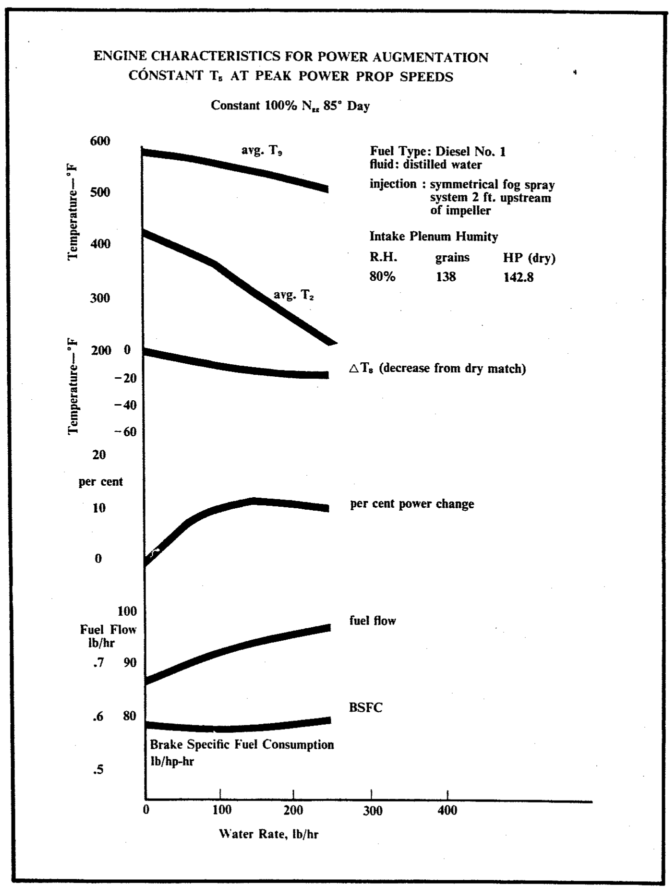


Figure 81a

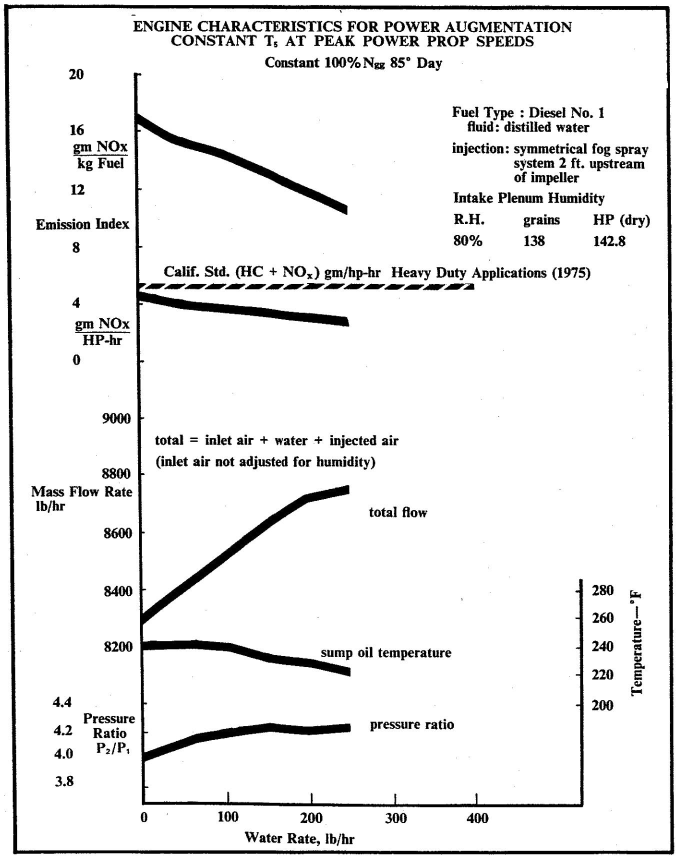


Figure 81b

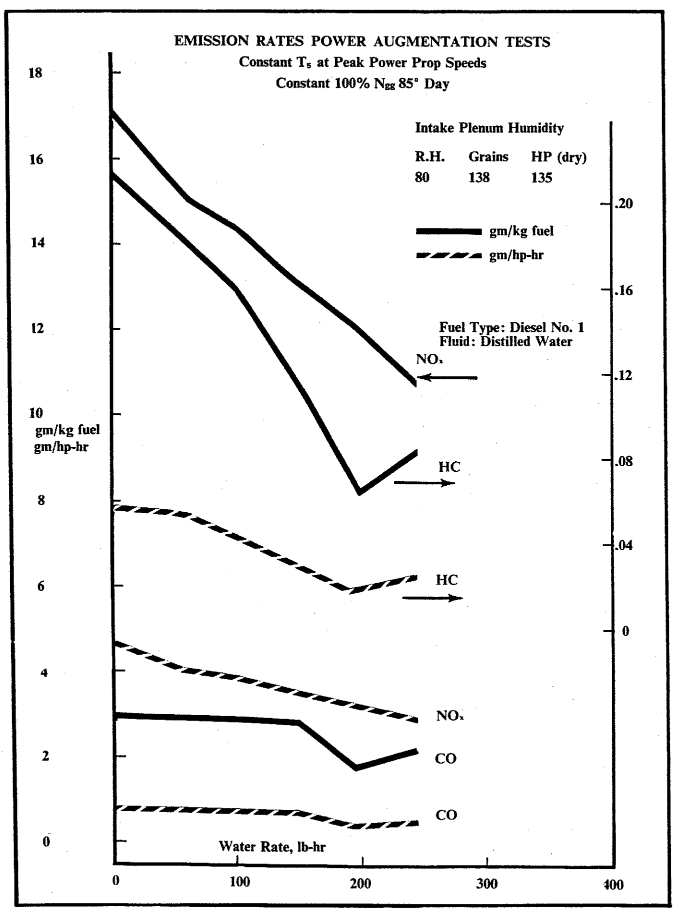


Figure 82

Inspection of the compressor section showed water injection related damage to the aluminum impeller, cascade and diffuser plate. The impeller damage varied from mild pitting of the leading edge to severe pitting on the pressure side of the O.D. blade edges. Heavy blade material breakout was evident at both edges of the steel shroud. Leading edge damage increased radially.

It is suggested that the probable mechanism(s) involved in the edge deterioration is related to mechanical destruction of the protective surface oxides by erosion, turbulence and/or cyclic cavitation forces leading to rapid deterioration of the surface macro-structure.

Figure 83 is a photograph showing the impeller damage.

Cascade and diffuser plate damage was confined to the leading edges and pressure side just aft the leading edge, and was of an erosive nature.

The variations in NOx with humidity were investigated at 50 through 80% speed using the steam coil to vary humidity. The speeds were selected to characterize the NOx pattern over the normal speed range encountered during vehicle chassis rolls emission tests. NOx values (ppm) were observed to decrease with increased humidity.

Transient emission tests for 50-100% acceleration conditions showed that NOx was depressed with water injection. Figure 84 shows the results for injecting water at 50, 80, 90, and 97.5% speeds. All the above emission evaluations were conducted with the baseline combustor (droplet diffusion flame).

Additional augmentation tests using two other types of injection resulted in similar engine parameter results. Compressor damage was identical to that previously described. The other methods tried were:

(a) Direct axial injection into the impeller eye using pressure atomizing nozzles. Two and four nozzles were tried, Figure 85.

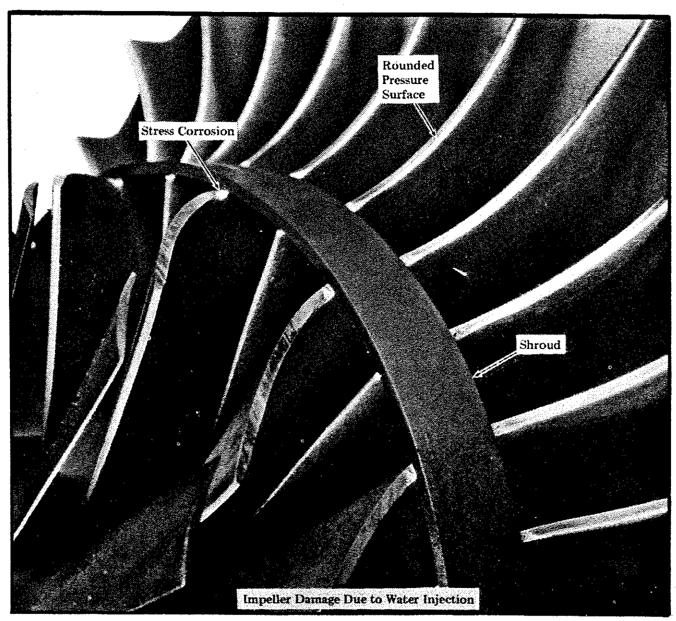


Figure 83

(b) Single modified air atomizing nozzle at the top of the air intake box, injection through the VIGV's.

No attempts have been made to determine erosion rate versus injection techniques as this was beyond the scope of work.

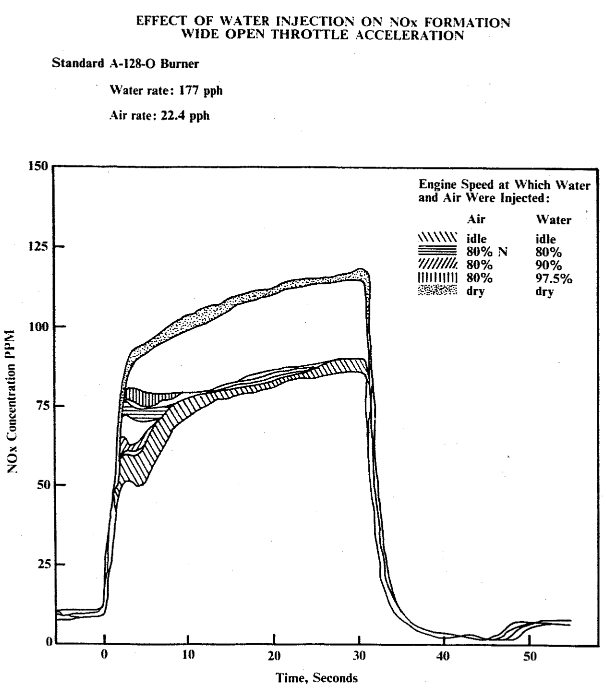


Figure 84

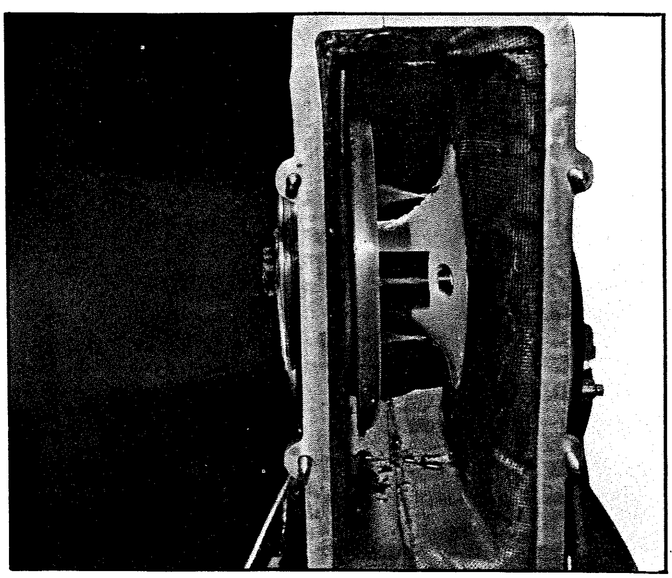


Figure 85 Axial Injection of Water into the Impeller

The use of water did not appear to cause degradation of other engine components nor contaminate the engine oil system for total injection quantities of 30 to 300 gallons of water.

Water injection through the VIGV's produced augmentation levels identical to those previously documented and showed that each system's contribution to augmenting power was directly additive, Figure 86.

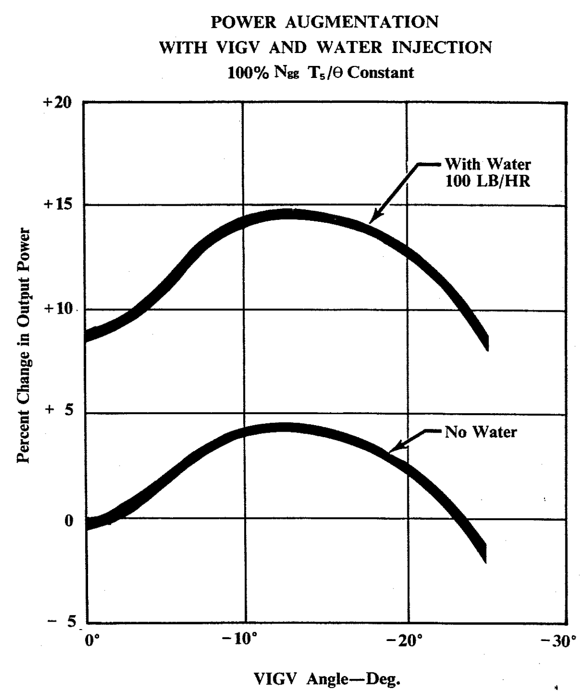


Figure 86

IN SUMMATION

1. Engine intake water injection will augment engine power.
2. Water management, spray quality and distribution must be optimized for each engine system.
3. A surface protection system is required to prolong the mechanical and aerodynamic integrity of aluminum compressor components.
4. Only *distilled* water is satisfactory.

VEHICLE TESTS

A practical demonstration of augmentation was conducted in a baseline vehicle. This was a combined test of VIGV and water injection to determine the feasibility of the upgraded engine design philosophy. Water injection was accomplished via a single air atomizing fuel nozzle in the top of the air intake, Figure 87. Compressor discharge air is used to pressurize the water tank and force water to the nozzle. Compressor discharge is also supplied to the water nozzle to assist in atomization. Both air and water flows were controlled by fixed orifices in the nozzle fittings. A 7 gallon tank is mounted in the trunk over the rear axle.

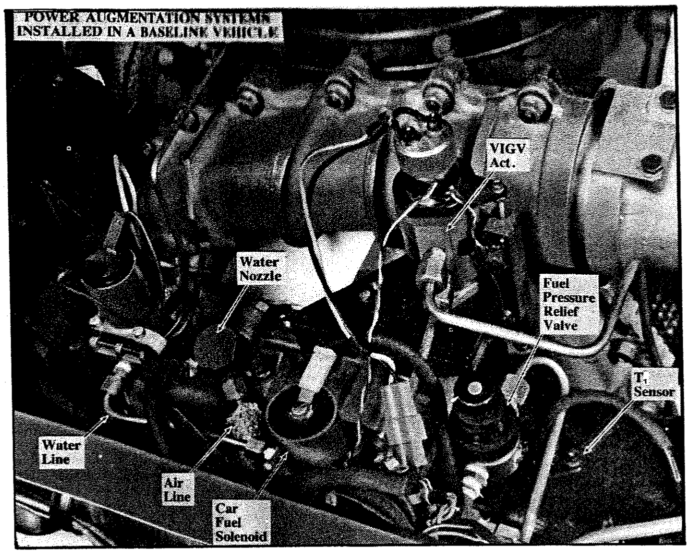


Figure 87 Power Augmentation Systems Installed in a Baseline Vehicle

A schematic showing the system is shown in Figure 88.

The engine system is controlled by the integrated electronic control system. Water injection command is initiated at 95% Ngg with WOT demand and energizes a relay to open the nozzle air and water solenoids.

The use of a solenoid in the pressurizing line to the tank is a feature to prevent bleeding off compressor discharge during starting.

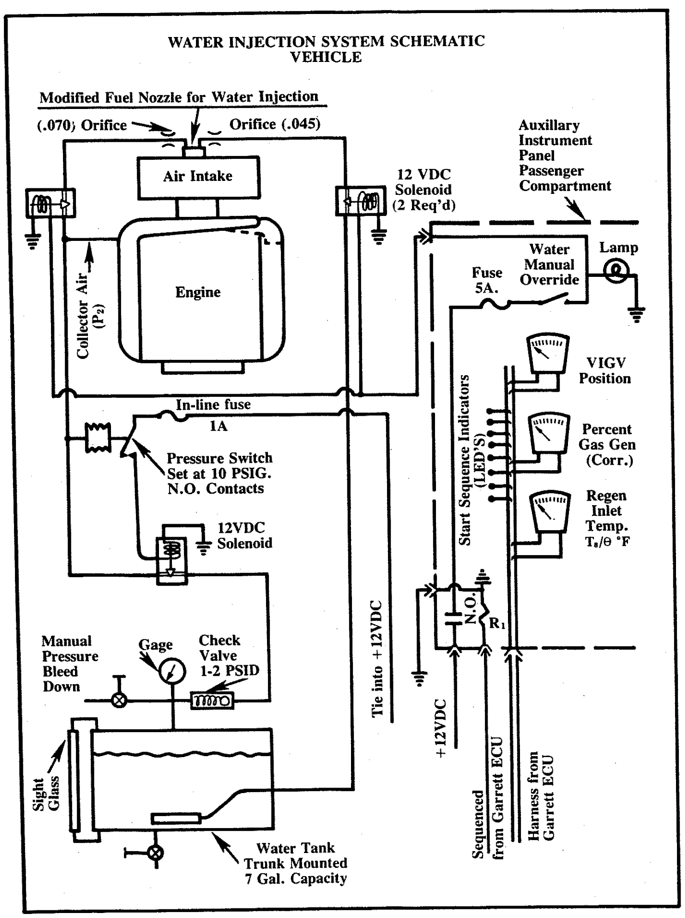


Figure 88

The control system had complete authority to govern speed, limit maximum turbine inlet temperature during accelerations, control T_8 during steady state, and momentarily open the power turbine nozzles during a rotor transient.

Performance tests were conducted at the proving grounds. Using established test procedures, three modes of operation were tested to determine the relative vehicle response to the augmentation modes. Test results showed improved vehicle performance but did not completely verify the steady state power increase due to water injection and/or VIGV's, Figure 89.

Vehicular water injection experience is limited and has not been optimized for performance, economy and emissions. Nevertheless, it does offer a novel approach to increase power output on demand.

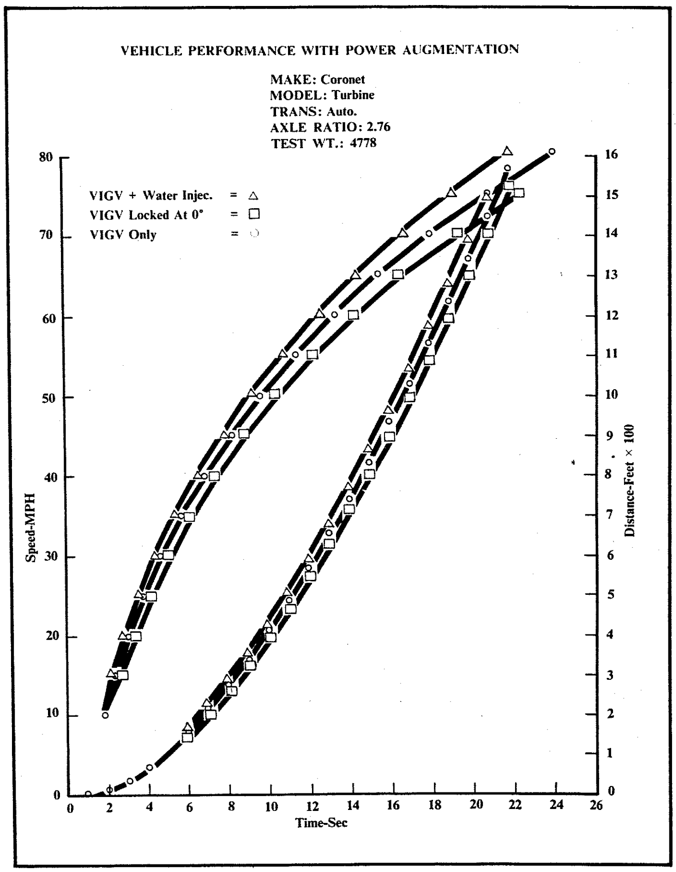


Figure 89

COST REDUCTION AND MANUFACTURING

Reduction of cost and development of production-oriented manufacturing processes will demand a maximum effort if the gas turbine engine is to compete seriously with conventional and/or alternate engines.

Good progress was achieved in development of new turbine wheel manufacturing, insulation fabrication and installation, and development of a unique regenerator seal. More subtle cost reductions were achieved with the ignitor/ignition system, free rotor concept, elimination of the auxiliary air pump, elastomeric drive development, and some of the control components. The former three items will be discussed briefly.

COMPRESSOR TURBINE WHEEL

Turbine wheel expense is a major consideration in the volume production of an automotive gas turbine engine. ERDA is currently sponsoring two different studies concerned with methods of reducing processing costs in manufacturing compressor turbine wheels while maintaining acceptable performance of the finished wheel.

The methods under study are:

- The AiRefrac Process—A reuseable pattern process developed by the AiResearch Casting Company.
- Gatorizing—A hot isothermal forging process developed by United Technologies.

Both processes will provide integrally bladed turbine wheels.

The AiRefrac Process utilizes reuseable patterns and proprietary molding techniques. To evaluate the process the AiResearch Casting Company was asked to cast sample wheels in IN 792+Hf material. Overall casting quality of the initial parts was relatively good and the metallurgical properties were to specifications. The parts were also dimensionally acceptable. One of these wheels is currently on test at the higher temperatures of the upgraded engine.

Gatorizing, a hot isothermal forging process developed by United Technologies, is the second turbine wheel production method under study. This program is also funded by ERDA.

LINERLESS INSULATION

Recognizing the high cost of sheet metal combined with production assembly problems, a limited in-house investigation was initiated during the mid 60's into the use of linerless insulation.

The burner cap insulation and the T_8 zone engine housing sheet metal were replaced with linerless insulation which was formed in place. The use of present day adhesives and sealants has greatly advanced linerless insulation technology.

To date one burner cap insulated in the late 60's has accumulated almost 3000 hours of combined endurance and steady state/transient tests. Surface erosion is almost non-existent and the cement requires only occasional

LINERLESS INSULATION EVALUATION SUMMARY

Table 5

	Chrysler			Foseco		
Description	Power Plant*	Hours	Condition	Power Plant	Hours	Condition
Burner Cap S/N 111A***	102	1,662	Minor spalling/repair	102*** Burner Cap S/N 113	53	Minor cement repair
S/N 118	102/106	2,908	Cement repair			
S/N 403	102/429	1,825	Minor repair			
High Pressure Housing	102	1,129	Minor cracks	427	15	
	406**			429	767	Minor cracks
	428	224	Minor cracks			
Gas Generator Support						
S/N 108	425	20	Good condition			
Front Bulkhead	102	1,129	Cracks and separation	427	15	
	406**			429	767	Serious cracks/repair
	428	224	Cracks/repair			
Low Pressure Housing	102	1,129	Minor cracks; erosion/repair	427	15	
	406**			429	767	Minor cracks and erosion
	428	224	Severe erosion/repair			
Power Turbine Support						
P.T. 110***	102	246	Minor cracks	(P.T. 109) 100	759	Good
P.T. 403	425	10	Good condition			
P.T. 406	429	767	Hairline cracks/repair			
P.T. 409	428	164	Good			

*Power Plant 102—4000-Hour Endurance Engine (Loaner)
Power Plant 106—General Performance Engine (Loaner)
Power Plant 406—Reinforced Bulkhead
Power Plant 427—Scaled Down Burner Fixture
Power Plant 428—Program Free Rotor Engine
Power Plant 429—Program Endurance Engine

**Parts Not Yet on Test
***Insulation Out of Service

replacement. Engine housings using Chrysler material and technology showed similar promising results.

As part of a cost reduction program requirement of the ERDA contract, dual insulation test and development programs were initiated. Both Foseco and Chrysler linerless insulations are being tested to determine life characteristics. A summary of the efforts to date with comments are given in Table 5, accompanied by an engine cross sectional view identifying these areas within the engine, Figure 90.

Aside from rare unexplained occurrence of erosion or material break-out, both insulations show comparable life.

Linerless insulated sub-assemblies yield heat losses similar to that of the sheet metal/insulation system of the baseline engine. A gas generator instrumented with skin thermocouples, and a power turbine, were fitted with linerless insulation as shown in Figure 91 and 92, tested, and then fitted with standard sheet metal over the linerless insulation. Material removal was minimal to fit the sheet metal. This approach allowed considerable flexibility and practicality to the test as removal of the insulation requires the biscuit to be destroyed and possible loss of instrumentation.

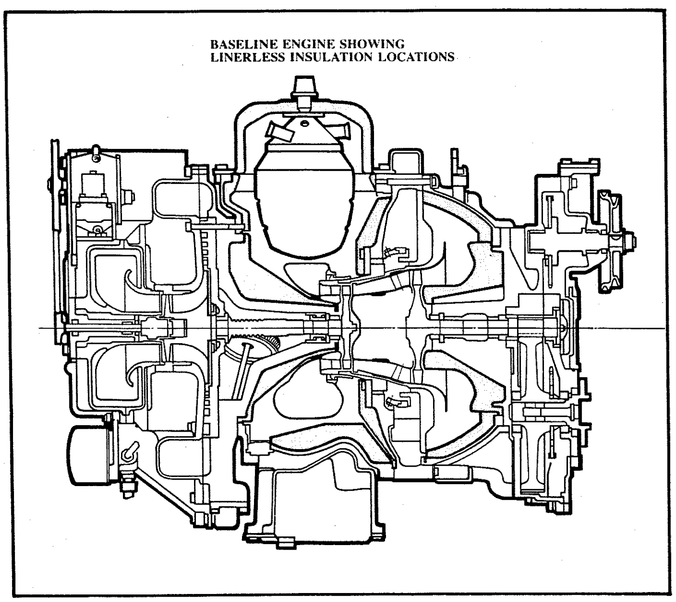


Figure 90

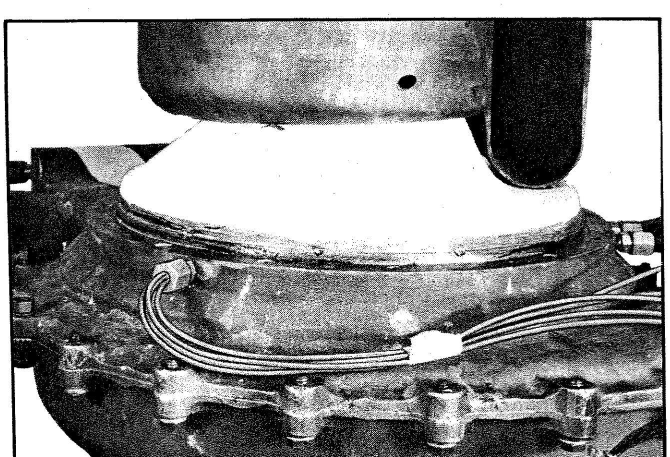


Figure 91 Linerless Insulation Gas Generator

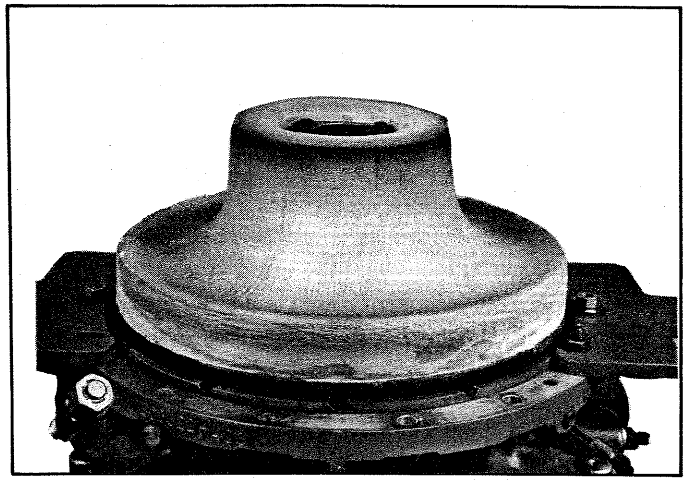


Figure 92 Linerless Insulation Power Turbine

Simultaneous performance and oil heat rejection tests for both sub-assemblies showed no significant change in heat rejection and BSFC for the two types of insulation techniques. Additional supporting data were provided by the gas generator support skin thermocouples in which temperatures repeated very well. The BSFC results are given in Figure 93. A complete linerless insulated engine housing is shown in Figure 94.

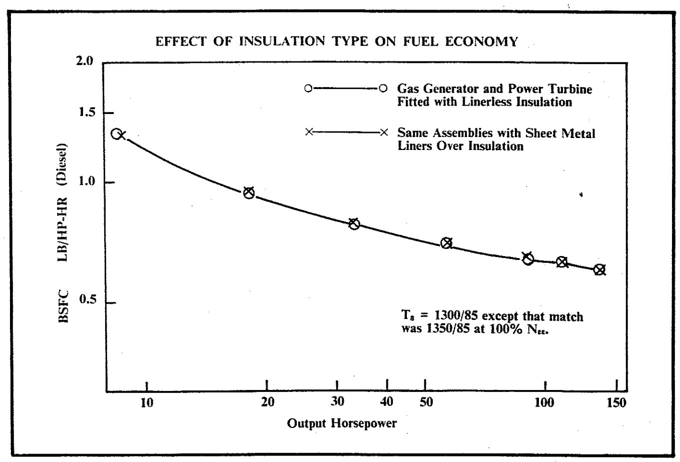


Figure 93

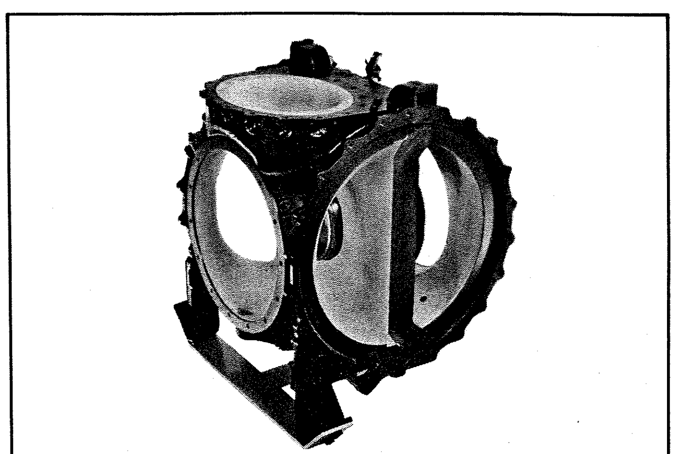


Figure 94 Linerless Insulation Engine Housing

IN SUMMATION

1. There is no measurable fuel penalty using linerless fitted sub-assemblies in the baseline engine. The engine housing system requires investigation before final judgement can be made.
2. Linerless insulation has a cost saving potential. High volume production tooling would produce dimensionally consistent parts, requiring little or no fitting.
3. Insulation damage in production would be negligible as design constraints and applicable tooling would exist. Disassembly would virtually not exist thus minimizing damage. Application of linerless insulation to the burner cap has demonstrated more than adequate life where a low velocity-moderate temperature environment exists.
4. Engine housing structural deflections must be minimal to prevent separation at structure-insulation interfaces.
5. Structural materials and adhesives must be compatible to provide adequate bonding.
6. Insulation thickness and minimum radius of curvature limits have been established for the upgraded engine.
7. Ceramic regenerator cores have not exhibited the predicted superior low speed effectiveness comparable to the fully developed metal cores. Additional effort will be required to reach this goal.
8. Nickel oxide seals have performed satisfactorily with extremely low wear rates in fixture, engine and vehicle applications but alleged health hazards preclude their wide spread use. Alternate materials are being screened. Limited tests in an engine using $ZrO_2(CaO)CaF_2$ yielded satisfactory results.
9. Engine controls must be more sophisticated than the current conventional practice i.e., closed loop on engine cycle temperature managed by electronic devices.
10. Prototype integrated electronic engine control systems have been successfully applied to automotive gas turbine vehicles. Long term reliability was not established. It appears that shielded wiring harnesses are not required. Temperature sensor performance remains a problem. A rigorous cost reduction program is warranted.
11. Conventional lubricants performed satisfactorily for extended periods of operation.
12. Several cost reduction programs such as linerless insulation, regenerator drive gear/core attachment, combustor/ignitor, turbine wheel, free rotor configuration etc. have indicated good progress. A major effort will be required to develop the numerous manufacturing processes which are virtually non-existent.
13. Noise control techniques have been successfully demonstrated for gas turbine vehicles. A 72 dBA, standard can be met using the SAE J-986a drive-by test for a 4500 lb. vehicle.
14. A lock-up torque converter, which directly couples the engine to the 3-speed transmission at the time of the 1-2 shift, is necessary for optimum free rotor vehicle performance. A slip clutch (no torque converter) may be the ultimate arrangement.
15. Air intake filter and exhaust ducting must be configured for minimum restriction as these losses have an appreciable effect on engine/vehicle performance.
16. The gas/liquid/air heater system coupled to a conventional air conditioning system with reheat capability is a satisfactory car comfort system.

GENERAL CONCLUSIONS

1. The regenerative, free power turbine gas turbine engine is a viable candidate as an alternate power plant for automotive application.
2. The free rotor concept (driving engine auxiliaries from the power turbine) appears to be a more practical and lower cost automotive power plant arrangement than the geared rotor configuration, (driving engine auxiliaries from the compressor turbine).
3. Engine performance, braking and starting are acceptable for automotive application. Low speed fuel consumption—principally at idle—is excessive.
4. Exhaust emissions can be controlled in the combustion process with no penalty to fuel economy or driveability. The 1975 statutory emission regulations (0.41 gm/mi HC, 3.4 gm/mi CO, and 3.1 gm/mi NOx) were met. The 1978 standards (.41 gm/mi HC, 3.4 gm/mi CO, and 0.4 gm/mi NOx) appear possible using a variety of fuels with fixed geometry burner concepts currently being developed.
5. Based on laboratory samples, satisfactory power plant endurance, durability, reliability and serviceability were achieved.
6. Low speed fuel consumption, BSFC, can readily be improved by reducing engine size and augmenting power to maintain vehicle performance. Water injection and variable inlet guide vanes are successful methods of power augmentation. The VIGV approach needs to be optimized while water injection requires corrosion/erosion protection of the compressor assembly and development of the water injection system. VIGV may be useful for reduction of idle fuel flow, rotor response and emission control.

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C. H. Mader	—	Aerodynamics
T. D. Nogle	—	Combustors
E. Z. Trieskey	—	Data Processing

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