



**SOCIETY OF AUTOMOTIVE ENGINEERS, INC.**  
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# **The ERDA/Chrysler Upgraded Gas Turbine Engine Objectives and Design**

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ERDA

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## INTRODUCTION

THE CLEAN Air Act of 1970 and the Energy Crisis of 1973 have been instrumental in the United States Government becoming involved in automotive research for commercial purposes. This most envied "nation on wheels" has also become an unwitting producer of undesirable pollutants and a mammoth consumer of rapidly dwindling fuel supplies. Clearly, the assets and liabilities associated with the automobile had grown to national significance and coordinated measures had to be instituted on a national scale.

Responding to such challenges of modern society, the Energy Research and Development Administration (ERDA), through its Division of Transportation Energy Conservation (DTEC) is sponsoring several research and development projects with the goal of improving alternative automotive engines. One major project is with Chrysler Corporation: The Baseline Gas Turbine Development Program. Started in November of 1972 as a low-emission gas turbine demonstration effort, originally under the sponsorship of the U.S. Environmental Protection Agency (EPA), this project has been heavily modified first to include, then to emphasize the improvement of fuel economy.

The initial tasks of testing and evaluation of the baseline design engine and vehicle have been completed and so have those of modified hardware to verify planned improvements which have now been incorporated in the design of an all new Upgraded Engine.

This paper deals with the objectives, performance, and some of the design aspects of the Upgraded Engine. Details cover vehicle, engine, and component performance as well as mechanical design of the gas generator and power sections, the regenerator and its drive system, and the unique structure resulting from the application of a single rotary regenerator.

A series of papers that describe all major developments associated with the project through the design phase are being presented to professional societies in order to assure maximum availability of the information generated with public funds in a timely manner.

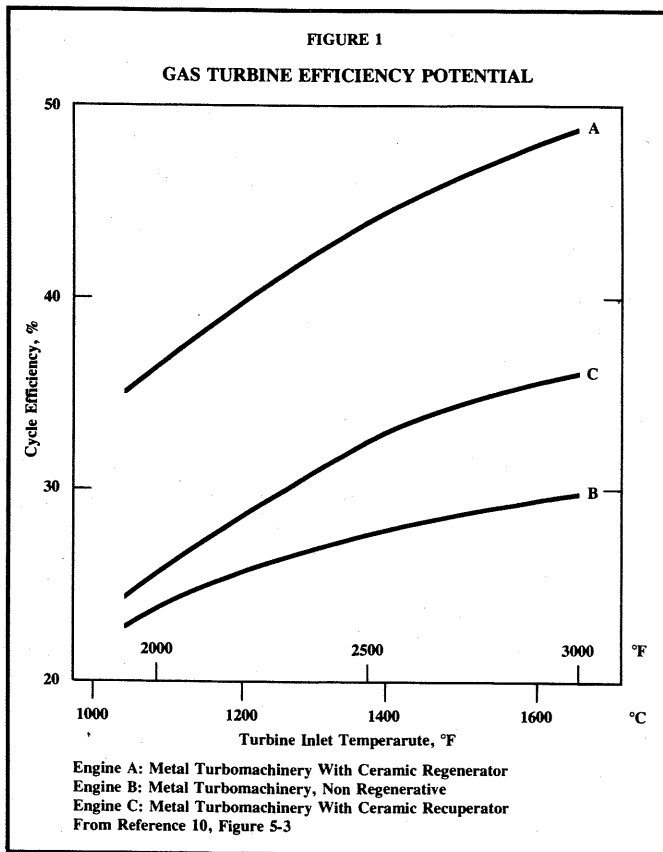
## OBJECTIVES

The ERDA has the goal of providing viable options for the nation's energy future. The gas turbine is one of the prime candidates as an alternate automotive power plant because of its excellent fuel economy potential (See Figure 1), inherent multi-fuel capability and low emissions. Even at the level of technology achieved with this relatively modest

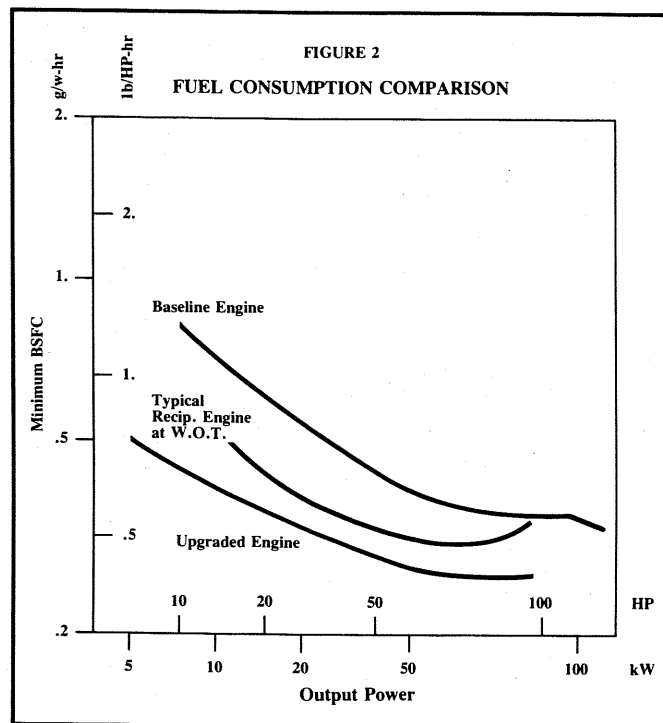
## ABSTRACT

Under a contract which began in November 1972, Chrysler Corporation has been conducting an automotive gas turbine improvement program for the Division of Transportation of the Energy Research and Development Administration. The final task of this program is to design, build, and demonstrate an Upgraded Engine. The design has been accomplished and is described in this paper. It utilizes a number of improvements developed and verified on the Chrysler Sixth Generation "Baseline" engines, e.g. variable inlet guide vanes, water injection, ceramic regenerators, an

integrated electronic control system, a free-rotor arrangement, a low emissions fixed geometry burner, and linerless insulation. Aerodynamic details to meet higher efficiency component specifications were provided by NASA Lewis. The design also incorporates a gas bearing on the rotor and improvements in arrangement and mechanical design. A vehicle system incorporating this engine should operate with fuel economy double that of the Baseline engine system and demonstrate emissions levels within the current 1978 Federal Standards.



program, the competitiveness of the gas turbine has become clearly evident (See Figure 2 and Table 1 for comparisons of fuel economy and emissions between the Upgraded Gas Turbine powered vehicle and its contemporary conventionally powered vehicle). However, this fuel economy, even with multi-fuel capability, will not be good



**TABLE 1**  
**EMISSIONS COMPARISON**

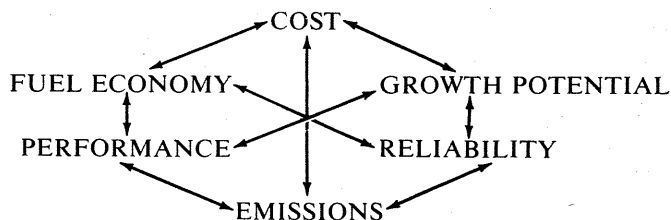
	EMISSIONS		
	HC	CO	NO <sub>x</sub>
1976 Production Compact	1.33	5.8	2.2
Upgraded Engine Vehicle (Projected)	.41	3.4	.40
1978 Statutory Standard	.41	3.4	.40

enough to meet ERDA objectives; the upcoming hardware demonstration will serve but a stepping stone, a firm base from where to launch a significant and coordinated effort aimed at realizing much more of the available potential.

By bringing these results to the professional societies, the ERDA wants an early introduction to the public of one of the possible future automobiles that will serve their transportation needs.

On the other hand, Chrysler Corporation, long a chief proponent of the gas turbine powered automobile, is vitally interested in the perfection for ultimate manufacture of the gas turbine. Manufacturing and sale of the gas turbine cannot commence until all requirements of the automobile industry have been met.

These are:



The gas turbine automobile must possess excellent qualifications in each of these categories and altogether as a package, it must be so much better than its competitors that the capital investment in new tooling, facilities, and the training of personnel are economically justifiable.

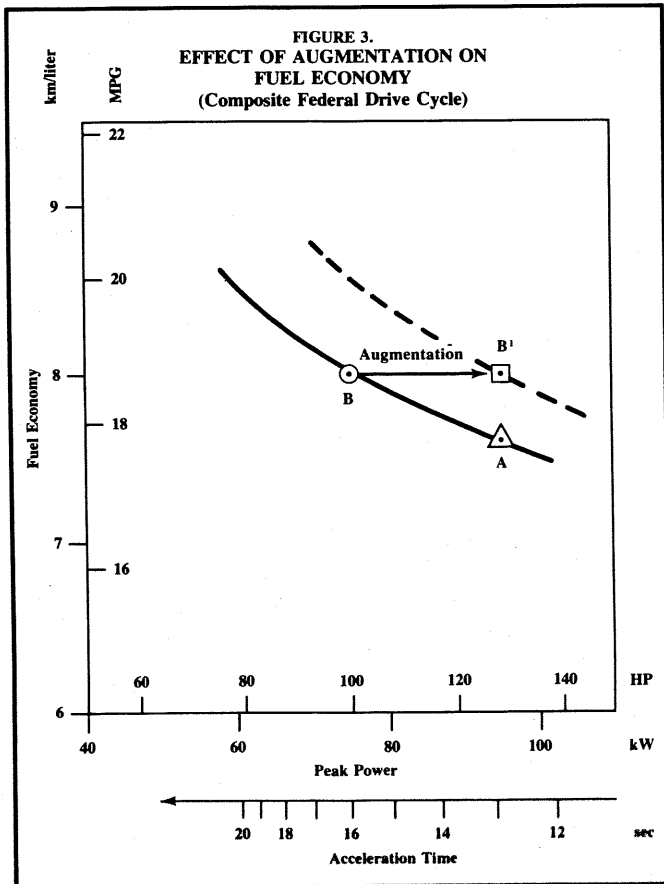
The Upgraded Engine will still need improvements particularly in the areas of cost and fuel economy before an automobile company can decide to manufacture. The answer to both may lie in the successful application of ceramics in more and more areas of the engine. With this design the first step has been taken by the incorporation of a ceramic regenerator. Related research efforts are being carefully watched and supported with much interest.

**ENGINE SIZING**

As mentioned earlier, our goal is to maximize fuel economy in an engine/vehicle system of "competitive" driveability. This was interpreted as being able to match the acceleration rate of an average 1973 compact vehicle (i.e., 0 to 96 km/hr (60 mph) in 13.5 sec). The vehicle acceleration

time goal determines the maximum engine power required for a given vehicle, driveline and engine torque characteristic. It also indirectly establishes the vehicle fuel economy level, since it specifies the vehicle power to weight ratio and thus the relative engine load factor level. A high maximum power engine in a given size vehicle, operating at a specified vehicle speed, consumes more fuel than a similar lower power engine. This is because the "smaller" engine would operate at a higher load factor (percent of maximum power), which is a more efficient operating condition.

These generalizations are shown for the subject compact vehicle in Figure 3. In order to meet the 13.5 second 0-96 km/hr acceleration performance, a peak output power of 92 kW (123 HP) is required. As shown by Point A in Figure 3 this results in a Federal Drive Cycle fuel economy of 7.6 km/liter (18 MPG). If 23% power augmentation is made available, the corresponding engine is represented by Points B and B' where the fuel economy is 8.0 km/liter (19 mpg). This engine gives the vehicle the fuel economy of a 75 kW (100HP) engine with the performance of a 92kW engine. In the Upgraded Engine 23% power augmentation is achieved by the use of water injection at the compressor inlet and the use of variable inlet guide vanes (VIGV). Note that the concept of augmentation is not new to the automotive scene. Actuation of the VIGV is almost directly analogous (albeit more complex) to opening the secondary throttles for high output with a four-barrel carburetor.



**TABLE 2**  
**UPGRADED ENGINE DESCRIPTION**

**GENERAL**  
 Type: Regenerative gas turbine  
 \*Rated Output: Power - 92kW (123 hp) @ 3280 rpm output shaft speed  
 Torque - 480 N.m (355 lb-ft) @ zero output shaft speed  
 Augmentation: Variable inlet guide vanes and water injection  
 Weight: 180 kg (400 lb)  
 Engine Configuration: Free rotor, designed for compact vehicle installation  
 Maximum gas generator acceleration, idle-to-full speed: 1.1 sec.  
 Fuels: Unleaded gasoline, diesel fuel, kerosene, JP-4, etc.

**COMPONENTS**  
 Compressor: Backward Curved Centrifugal with variable inlet guide vanes  
 Compressor Turbine: Axial  
 Power Turbine: Axial, variable geometry stator  
 Regenerator: One rotating ceramic disc  
 Burner: Fixed geometry

**DESIGN POINT CHARACTERISTICS**  
 Maximum gas generator shaft speed: 58,500 rpm  
 Maximum Power Turbine Speed: 70,000 rpm  
 Maximum output speed (after reduction gears): 4,700 rpm  
 Maximum Regenerator Speed: 31 rpm  
 Compressor Pressure Ratio: 4.2:1; 4.9:1 augmented  
 Compressor Air Flow: 0.61 kg/s (1.34 lb/s) design  
 0.69 kg/s (1.52 lb/s) augmented  
 Compressor Turbine Inlet Temperature: 1052°C (1925°F) design  
 Power Turbine Outlet Temperature: 748°C (1378°F)

\*Ambient Conditions: Temperature 29°C (85°F)  
 Pressure 1.013 bar (14.696 psia)

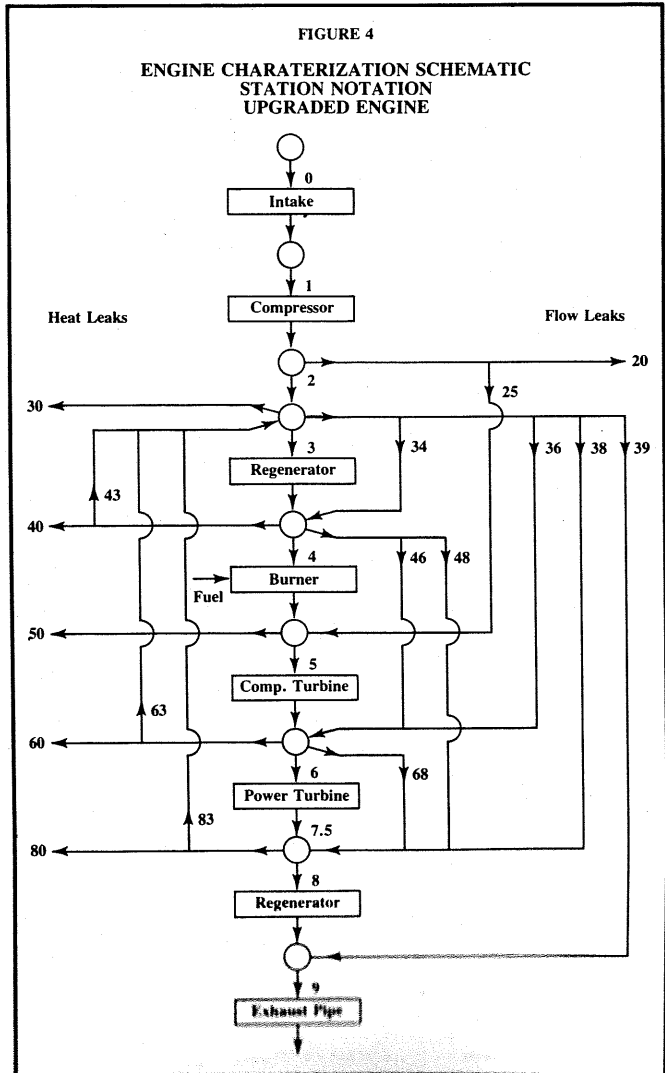


TABLE 3a  
 UPGRADED ENGINE CHARACTERIZATION  
 29.4°C, 1.013 bar  
 (SI Units)

Gas Generator Speed, RPM	29,250	35,100	40,950	46,800	52,650	55,575	58,500
FRACTION	0.5	0.6	0.7	0.8	0.9	0.95	1.0
Power Turbine Speed, RPM	26,000	34,000	40,600	44,500	50,000	52,000	50,000
Compressor Pressure Ratio	1.498	1.788	2.177	2.700	3.383	3.770	4.185
Component Efficiency							
Compressor, 1-2*	0.769	0.784	0.798	0.804	0.804	0.797	0.785
Compressor Turbine, 5-6	0.832	0.836	0.841	0.843	0.846	0.847	0.846
Power Turbine, 6-8	0.719	0.719	0.741	0.755	0.764	0.765	0.762
Burner	0.998	0.998	0.999	0.999	0.999	0.999	0.999
Regenerator	0.957	0.948	0.939	0.928	0.916	0.911	0.905
Parasitic Loss, kW							
Gas Generator	0.63	0.90	1.23	1.61	2.03	2.26	2.51
Power Turbine	1.46	2.15	3.12	3.90	5.15	5.73	5.57
Output Power, kW Net	4.86	10.69	20.71	36.25	56.71	67.49	77.83
Fuel Flow, kg/h Diesel	2.41	4.02	6.44	10.07	15.01	17.85	20.83
Specific Fuel Consumption, g/w-h	0.496	0.376	0.311	0.278	0.265	0.265	0.268
Gas Flow, kg/s							
Station 1	0.196	0.256	0.323	0.408	0.503	0.555	0.606
Station 3	0.191	0.249	0.315	0.397	0.489	0.538	0.587
Station 4	0.191	0.249	0.314	0.396	0.488	0.537	0.586
Station 5	0.192	0.250	0.315	0.397	0.489	0.539	0.588
Station 6	0.194	0.253	0.320	0.403	0.497	0.548	0.598
Station 8	0.196	0.255	0.322	0.407	0.502	0.554	0.604
Station 9	0.197	0.256	0.324	0.410	0.506	0.558	0.610
Pressure, kPa							
Station 1	101.17	101.05	100.89	100.64	100.28	100.05	99.80
Station 2=3	151.55	180.68	219.64	271.71	339.23	377.19	417.69
Station 4	150.40	179.38	218.23	270.20	337.66	375.60	416.08
Station 5	147.95	175.90	213.67	264.35	330.56	367.86	407.76
Station 6	120.41	132.10	147.31	165.41	186.19	195.61	203.37
Station 8	103.30	104.13	105.20	106.76	108.84	110.10	111.48
Station 9	101.59	101.79	102.12	102.68	103.54	104.11	104.77
Temperature, °C							
Station 1	29.4	29.4	29.4	29.4	29.4	29.4	29.4
Station 2	77.7	99.2	123.8	152.6	185.4	203.4	222.7
Station 3	86.2	105.4	128.2	155.8	187.6	205.2	224.3
Station 4	705.1	703.5	701.7	699.0	696.0	695.3	694.4
Station 5	829.4	864.6	907.3	954.6	1005.5	1029.5	1051.7
Station 6	776.1	792.1	812.6	834.1	855.5	863.4	868.5
Station 7.5	747.8	747.8	747.8	747.8	747.8	747.8	747.8
Station 8	741.3	742.3	743.0	743.6	744.1	744.3	744.5
Station 9	131.1	155.1	182.3	214.7	250.9	269.5	289.6
Flow Leaks, kg/kg Compressor Air Flow							
Station i to j							
i j = 20	.00242	.00251	.00257	.00266	.00280	.00285	.00286
25	.00431	.00360	.00308	.00270	.00242	.00230	.00218
34	.00251	.00220	.00199	.00170	.00152	.00143	.00135
36	.00910	.00942	.00971	.01011	.01068	.01093	.01104
38	.00345	.00360	.00371	.00384	.00399	.00404	.00402
39	.00478	.00546	.00616	.00696	.00785	.00826	.00852
46	.00539	.00577	.00616	.00667	.00735	.00768	.00792
48	.00133	.00186	.00245	.00312	.00386	.00422	.00450
68	.00236	.00245	.00247	.00245	.00239	.00232	.00218
Heat Leaks, J/kg Compressor Air Flow							
i j = 30	609.	712.	722.	795.	812.	812.	788.
40	2998.	2349.	1854.	1468.	1196.	1084.	996.
43	6178.	4654.	3503.	2624.	2010.	1761.	1582.
50	2468.	2058.	1737.	1463.	1263.	1175.	1096.
60	495.	402.	328.	267.	226.	207.	191.
63	788.	619.	488.	381.	305.	272.	249.
80	2821.	2231.	1777.	1416.	1163.	1056.	970.
83	2089.	1593.	1219.	926.	721.	635.	574.

\*Numerics Refer to Station Notation Figure 4

TABLE 3b  
**UPGRADED ENGINE CHARACTERIZATION**  
 85°F, 14.696 PSIA  
 (English Units)

Gas Generator Speed, RPM	29,250	35,100	40,950	46,800	52,650	55,575	58,500
FRACTION	0.5	0.6	0.7	0.8	0.9	0.95	1.0
Power Turbine Speed, RPM	26,000	34,000	40,600	44,500	50,000	52,000	50,000
Compressor Pressure Ratio	1.498	1.788	2.177	2.700	3.383	3.770	4.185
Component Efficiency							
Compressor, 1-2*	0.769	0.784	0.798	0.804	0.804	0.797	0.785
Compressor Turbine, 5-6	0.832	0.836	0.841	0.843	0.846	0.847	0.846
Power Turbine, 6-8	0.719	0.719	0.741	0.755	0.764	0.765	0.762
Burner	0.998	0.998	0.999	0.999	0.999	0.999	0.999
Regenerator	0.957	0.948	0.939	0.928	0.916	0.911	0.905
Parasitic Loss, HP							
Gas Generator	0.84	1.21	1.65	2.15	2.72	3.04	3.37
Power Turbine	1.95	2.89	4.19	5.22	6.90	7.68	7.46
Output Power, HP Net	6.52	14.33	27.76	48.59	76.02	90.47	104.32
Fuel Flow, lb/hr Diesel	5.32	8.86	14.19	22.20	33.09	39.36	45.93
Specific Fuel Consumption, lb/hp-hr	0.815	0.618	0.511	0.457	0.435	0.435	0.440
Gas Flow, lb/sec							
Station 1	0.433	0.564	0.713	0.900	1.110	1.223	1.335
Station 3	0.421	0.549	0.694	0.875	1.078	1.186	1.295
Station 4	0.421	0.548	0.693	0.874	1.076	1.185	1.293
Station 5	0.423	0.550	0.695	0.876	1.079	1.187	1.296
Station 6	0.428	0.558	0.704	0.889	1.096	1.207	1.318
Station 8	0.431	0.562	0.711	0.898	1.107	1.220	1.332
Station 9	0.433	0.565	0.715	0.904	1.116	1.230	1.344
Pressure, psia							
Station 1	14.673	14.657	14.633	14.596	14.544	14.511	14.476
Station 2=3	21.980	26.206	31.856	39.409	49.201	54.706	60.580
Station 4	21.814	26.017	31.651	39.190	48.974	54.476	60.348
Station 5	21.458	25.513	30.990	38.341	47.944	53.353	59.141
Station 6	17.463	19.159	21.365	23.990	27.004	28.370	29.495
Station 8	14.983	15.102	15.258	15.484	15.785	15.968	16.169
Station 9	14.734	14.764	14.811	14.893	15.018	15.100	15.196
Temperature, °F							
Station 1	85.0	85.0	85.0	85.0	85.0	85.0	85.0
Station 2	171.9	210.6	254.8	306.8	365.8	398.1	432.8
Station 3	187.1	221.6	262.8	312.4	369.8	401.4	435.7
Station 4	1301.2	1298.3	1295.0	1290.1	1284.8	1283.6	1281.9
Station 5	1524.9	1588.4	1665.1	1750.3	1841.9	1885.2	1925.0
Station 6	1429.0	1457.5	1494.6	1533.4	1571.9	1586.2	1595.2
Station 7.5	1378.0	1378.0	1378.0	1378.0	1378.0	1378.0	1378.0
Station 8	1366.4	1368.1	1369.4	1370.5	1371.3	1371.7	1372.1
Station 9	268.0	311.3	360.2	418.4	483.6	517.1	553.3
Flow Leaks, lb/lb Compressor Air Flow							
Station i to j							
i j = 20	.00242	.00251	.00257	.00266	.00280	.00285	.00286
25	.00431	.00360	.00308	.00270	.00242	.00230	.00218
34	.00251	.00220	.00199	.00170	.00152	.00143	.00135
36	.00910	.00942	.00971	.01011	.01068	.01093	.01104
38	.00345	.00360	.00371	.00384	.00399	.00404	.00402
39	.00478	.00546	.00616	.00696	.00785	.00826	.00852
46	.00539	.00577	.00616	.00667	.00735	.00768	.00792
48	.00133	.00186	.00245	.00312	.00386	.00422	.00450
68	.00236	.00245	.00247	.00245	.00239	.00232	.00218
Heat Leaks, BTU/lb. Compressor Air Flow							
i j = 30	.262	.306	.332	.342	.349	.349	.339
40	1.289	1.010	.797	.631	.514	.466	.428
43	2.656	2.001	1.506	1.128	.864	.757	.680
50	1.061	.885	.747	.629	.543	.505	.471
60	.213	.173	.141	.115	.097	.089	.082
63	.339	.266	.210	.164	.131	.117	.107
80	1.213	.959	.764	.609	.500	.454	.417
83	.898	.685	.524	.398	.310	.273	.247

\*Numerics Refer to Station Notation Figure 4

The engine is described briefly in Table 2 and more completely in the subsequent discussion. A schematic is given in Figure 4 to define the station notation and the corresponding state data are given in Tables 3A and 3B.

## ENGINE DESIGN

The two features of this engine which are new to the automotive gas turbine are the augmentation concept and the free rotor arrangement.

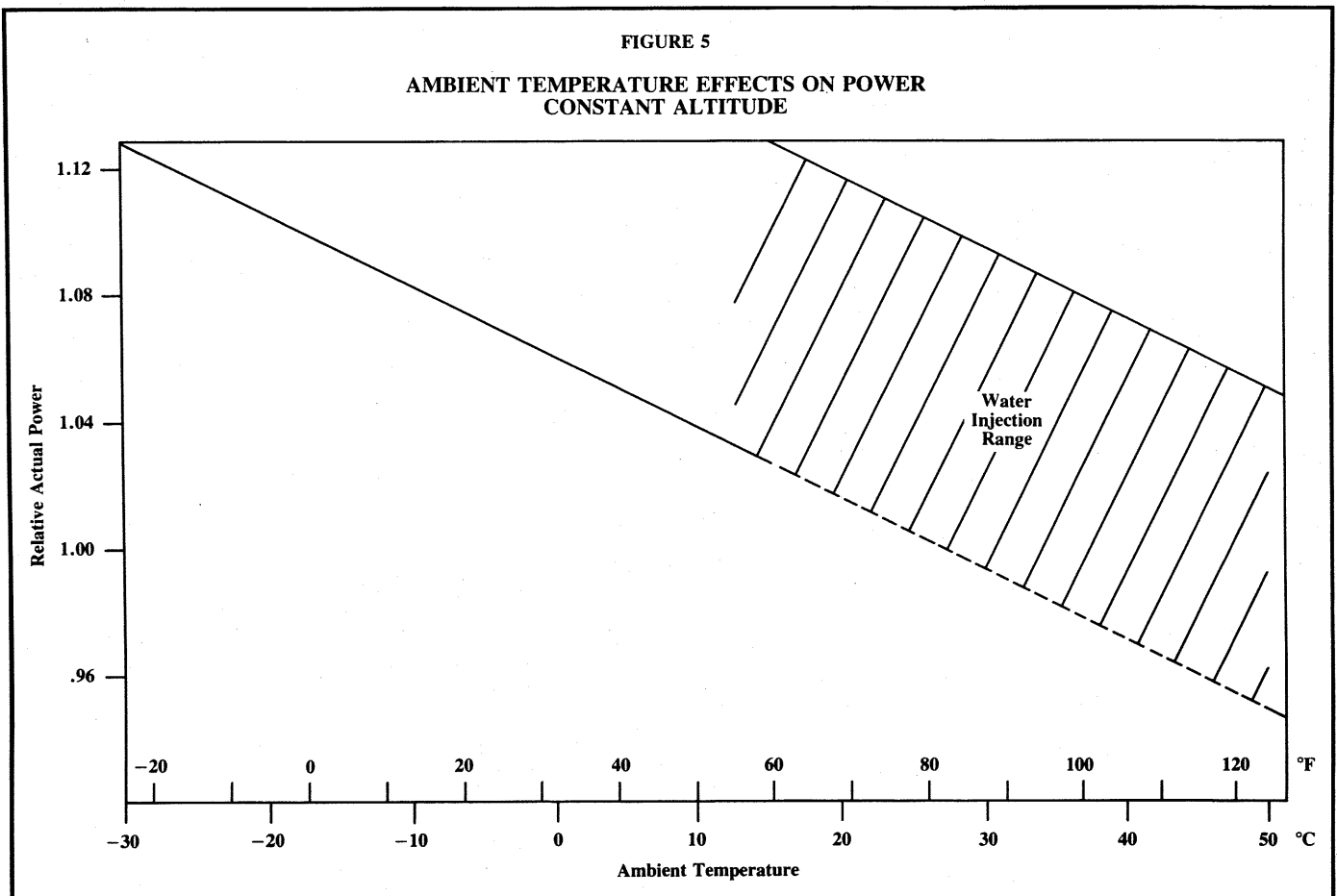
Power augmentation results from both water injection and variable inlet guide vanes (VIGV) at the compressor inlet. Water injection is simple to implement, requiring no more than a water fogging nozzle at the air intake bowl and a water tank pressurized from the compressor discharge. However, a water flow rate slightly larger than the fuel flow rate is needed in order to get a modest 10% boost in maximum engine power as shown in Reference 1. Therefore, to keep the water tank a reasonable size, the injection system is only activated at maximum power which is normally a small fraction of total operating time. Use of water injection only at ambients greater than 15°C (60°F) serves to conserve water, compensates for the gas turbine's characteristic temperature dependency as shown in Figure 5, and obviates freezing protection. Commercial grade distilled water is used to minimize compressor performance deterioration from scale deposits. In addition, the forward,

or inducer, section of the compressor is made from steel to minimize the effect of water droplet erosion.

Augmentation with VIGV is accomplished by imparting swirl to the air at the compressor inlet opposite to the direction of rotation. In this way, the change in angular momentum is increased for the fluid passing through the compressor. This results in increased pressure ratio and flow as though the compressor were running faster. Swirl in the plus or co-rotational direction is used at low vehicle speeds to regulate engine power while still maintaining engine temperature and efficiency. This works by reducing pressure rise and flow as though the compressor were running slower. A practical VIGV implementation consists of 13 uncambered airfoils (NACA 65-0010) with the front 45% non-moveable. This is capable of deflecting the incoming air through a range of angles from  $-30^\circ$  to  $+60^\circ$ .

The second novel feature of the Upgraded engine is the use of a free rotor. A "free rotor" configuration is a design in which all power takeoffs have been removed from the gas generator rotor. Thus all engine auxiliaries such as the oil pump, fuel pump, atomizing air pump and regenerator are driven from the power turbine instead of the gas generator. Such a free rotor design has a number of advantages.

- a) It is conducive to the use of a gas bearing. By removing a fraction of the bearing load due to power take-off, the bearing requirements are eased.



- b) It is quieter. Previous power take-off drives have been the source of identifiable gear noise.
- c) It is potentially less costly. The high speed gear system of a geared rotor is a significant cost item.
- d) Improved vehicle driveability. Although tests and calculations show small improvement in rotor response, experience with a prototype system suggests improved vehicle "feel". In particular, the combination of a free rotor and gas bearing will have superior cold starting performance.

At the same time there are a number of disadvantages and consequences for the Upgraded Engine as a result of the free rotor choice.

- a) The accessory drive is complex. Some means had to be provided to supply lubricating oil to the gas generator and atomizing air to the burner during the start sequence prior to the power turbine assuming the drive. It is presumed that technology advances will eventually eliminate the need for these services during the initial start period and that this disadvantage will not exist for future free rotor engines.
- b) The fuel control philosophy is changed. Previous fuel controls have been based on a mechanical governor driven from the gas generator. Removal of the accessory drive results in an electronic speed sensor as input to the fuel control. This is a consequence of the free rotor choice, but is not a disadvantage for the Upgraded Engine since other considerations resulted in the choice of an electronic fuel control. Under other circumstances, the lack of a convenient mechanical drive could be a disadvantage as a control limitation.
- c) The thermal loading distribution is altered. A gas bearing being cooled by compressor discharge air at 232°C (450°F) maximum as compared to the oil bearing running at a maximum oil temperature of 121°C (250°F) will cause higher temperatures in the shaft, bearing and support structure during both running and soakback conditions.

During the initial stages of the design layout, these factors were recognized and weighed as to relative importance. On balance the consensus was that the advantages outweighed the disadvantages for the purposes of the Upgraded Engine. Since the consequences of this choice are non-trivial, each of the above topics deserves further discussion.

## GAS GENERATOR BEARINGS

The development of the bearing system for the gas generator is described in some detail elsewhere (Ref. 2). For present purposes, it is sufficient to note that the gas bearings are physically larger than their oil counterparts because of their load-carrying characteristics. In fact, the front bearing is carried over as an oil bearing primarily because a gas thrust bearing would be so large as to pose an inertia problem. Moreover, the rear bearing in its present configuration contributes a small but measurable amount to

the rotor inertia. If the bearing had to carry the additional radial load associated with a power take-off, a further rotor inertial penalty would be incurred. The additional size would also pose an awkward design problem since space in the vicinity of the compressor turbine nozzle and vortex is very limited. It is also worth mentioning that interest in gas bearing technology is in part in anticipation of their potential use in high temperature non-metallic applications.

## NOISE

Generally speaking, high speed gearing tends to create noise that is an annoyance in a car. One high speed pinion on the power turbine shaft is necessary, but the use of the free rotor concept eliminates a second high speed gear drive. The effect of this arrangement on noise is most noticeable at idle where the power turbine may be idling as low as 10% speed. On a geared rotor engine, the gas generator never gets below 50% speed at idle and its contribution to engine noise is appreciable.

## COST

Engine designs that minimize the noise problems of high speed gearing require among other features, high quality precision gears and bearings that tend to be expensive. To the extent that the free rotor design eliminates one such gear set, it represents a cost saving. Of course, as noted above, the accessory drive of the Upgraded Engine has become complex and thus implies a cost penalty. This complication would be unnecessary if the burner and the engine bearings could function without the atomizing air and oil pumps during the portion of the starting sequence prior to the power turbine driving the engine accessories. As for the oil problem, a simple local oil storage system is one of several possible solutions that appears practical. Therefore, at the present time, the major obstacle to this simplification is the development of a low emission burner that does not require supplementary air for start, but such a design could well develop from the current state-of-the-art. Taking the long term view, then, it appears that the free rotor concept is consistent with an eventual simple, low cost design.

## ROTOR RESPONSE

The subject of rotor response can be made to sound deceptively simple by saying that the acceleration of a rotor mass is proportional to the excess torque available. This simple statement becomes complex when the term "excess torque" is considered in detail. The excess torque is produced by the increase in turbine inlet temperature over the steady state value during the acceleration and the magnitude of this torque is affected by the work split between the compressor turbine and the power turbine. In comparing the Upgraded Engine to the Baseline Engine, four significant changes have been made which affect the excess torque as well as the rotor polar moment of inertia.

1. The engine is smaller in size. This reduces the rotor inertia.

2. The compressor and turbine designs have been changed in order to improve efficiency at the expense of a modest increase in inertia.
3. The accessory load and its effective inertia have been removed from the compressor turbine and applied to the power turbine.
4. The rear bearing of the rotor has been changed from an oil bearing to a gas bearing with a resulting change in inertia and parasitic losses.

The effects of many of these factors are small and interrelated in a complex manner. When comparing the acceleration performance of the two engines, however, the major effects are that the reduced rotor inertia due to the smaller engine size is balanced against the increased inertia of the specific design of the compressor impeller, the turbine wheel and the gas bearing. The resulting comparison is given in Figure 6.

One consideration, however, should be non-controversial. A combination of a free rotor with a gas bearing and a low speed drive for accessories should have a decided cold start advantage over a high speed drive system with oil bearings.

As far as the known disadvantages of a free rotor design are concerned, the ones noted above are either self-explanatory or have been described elsewhere (Ref. 2). Suffice it to say that as the design has been worked out, no unexpected, insurmountable design problems have been uncovered. The ultimate test, of course, will be the performance of the system in the running engine.

In the ensuing discussion, frequent comparisons are made with the Baseline engine. This engine was used as the basis for development of the upgraded concepts and is thoroughly

discussed in the companion paper Reference 1. A brief description is provided in the Appendix.

## HOUSING GEOMETRY

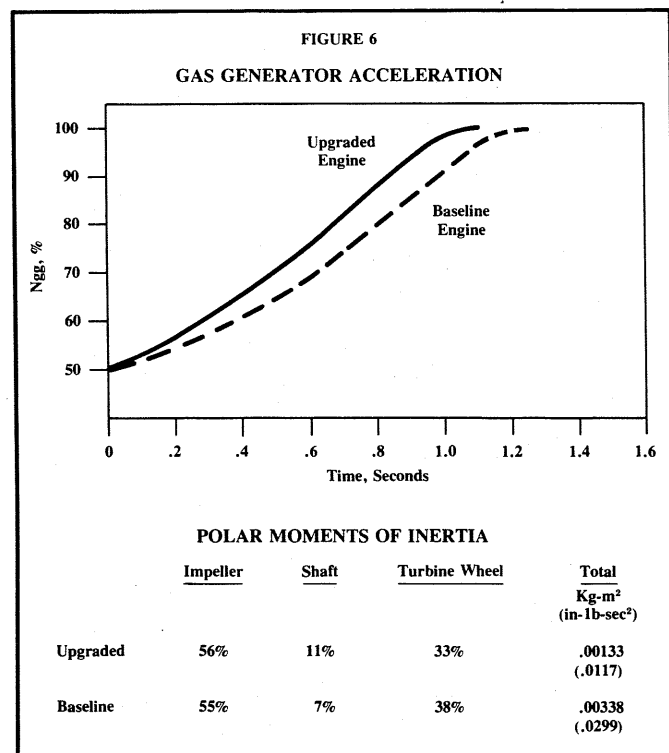
In the selection of the housing concept for the Upgraded engine there were numerous factors to consider, many of them conflicting. For example, a prime requirement for the engine, as previously discussed, is that the engine should possess outstanding fuel economy, particularly at ordinary road load conditions. This requires a large regenerator, in fact, a size that would have been termed "oversize" a few years ago. This tends to conflict with a parallel current approach to fuel economical personal transportation — the compact car. The regenerator (or equally for a recuperator as well) is a bulky component and "oversize" heat exchangers become increasingly difficult to package in a conventional vehicle engine compartment.

There is, however, a future potential for minimizing this situation through the use of high pressure, high temperature cycles. For the Upgraded engine this option was not open to us so we faced the problem of installing a large volume regenerator.

In selecting the housing geometry, a choice had to be made as to whether to provide for a single or a multi-regenerator configuration. The choice was not easy since in addition to the obvious differences in unit sizes required there were other differences such as leakage paths and overall costs to consider. Gas leakage paths, in spite of sophisticated regenerator seals, are a continual concern to the turbine engine designer. Two main considerations in determining the magnitude of these leaks are the length of sealing surface and the distortions of the housing as it is exposed to the various conditions of its operating range. The single regenerator has less sealing length but requires an unsymmetrical housing. In addition, the single regenerator requires fewer parts, seals, drive gears, exhaust ducts, etc. This general subject has been discussed in a paper by John S. Collman et. al. Reference 3.

Reference to Figures 7a thru 7d show that the single regenerator concept won out for the Upgraded engine. Probably the most significant factor in this selection was the installation of the regenerator in a non-vertical plane. Tilting the regenerator 20° to the vertical permitted the large single regenerator to fit between the side rails and suspension of the vehicle. Locating the regenerator on the right hand side of the compartment permits the exhaust duct to be free of the steering components. The general overall geometry of the engine appears to fit reasonably well into the available space as shown in Figure 7e.

Although the single regenerator solved the width problem, it created a length problem. The regenerator and compressor were now in conflict with one another. A radial compressor and an efficient radial diffuser take up considerable space in a radial direction. For "normal" size regenerators (or for dual regenerators) it is conventional to put the compressor forward of the regenerator and provide



simple ducting to get the air from the compressor discharge to the face of the regenerator. The space limitations for the engine did not permit this conventional approach, so the compressor was "tucked" in under the front of the regenerator with the result that the compressor diffuser must include a radial to axial bend prior to dumping into the collector duct. A somewhat unusual flow path for collecting the compressor discharge air was selected that collected the air by moving it radially inward in front of the diffuser cover and then periodically moving it radially outboard into ducts. Two of these ducts — or passages — lead directly to the regenerator face through openings in the forward portion of the regenerator platform. The third duct provides a passage to the high pressure bulkhead area for cooling both

the bulkhead and the variable power turbine nozzle mechanism. This subject is discussed in more detail in a subsequent section.

One final comment on the gas generator assembly as it fits into and affects the housing geometry concerns the provision for lubricating oil for the rotor bearing. It was decided that external oil lines are preferable to internal ones. Internal lines require appropriate seals between the gas generator assembly and the housing and this was considered to be an undesirable approach. In order to provide a lubricating oil path from the bearing to a fitting on the outside surface of the engine, a system of flanges and pilots was worked out. The solution is shown in Figure 8.

FIGURE 7a  
FRONT VIEW

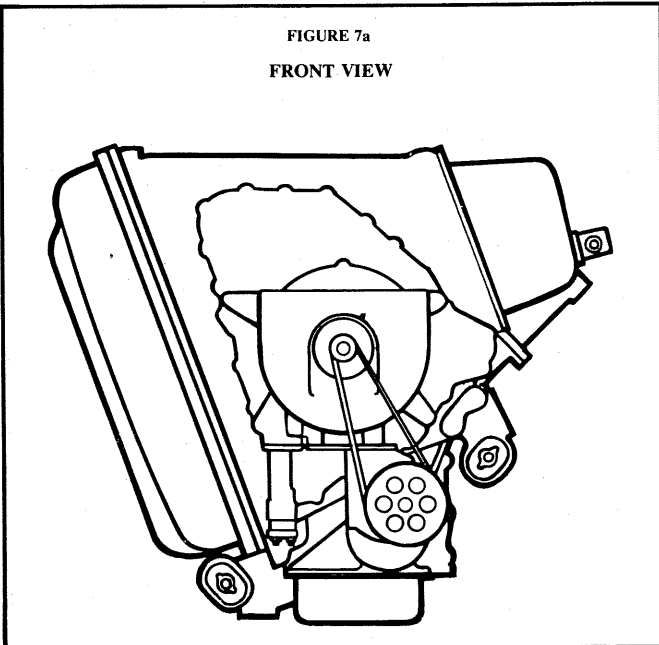


FIGURE 7b  
REAR VIEW

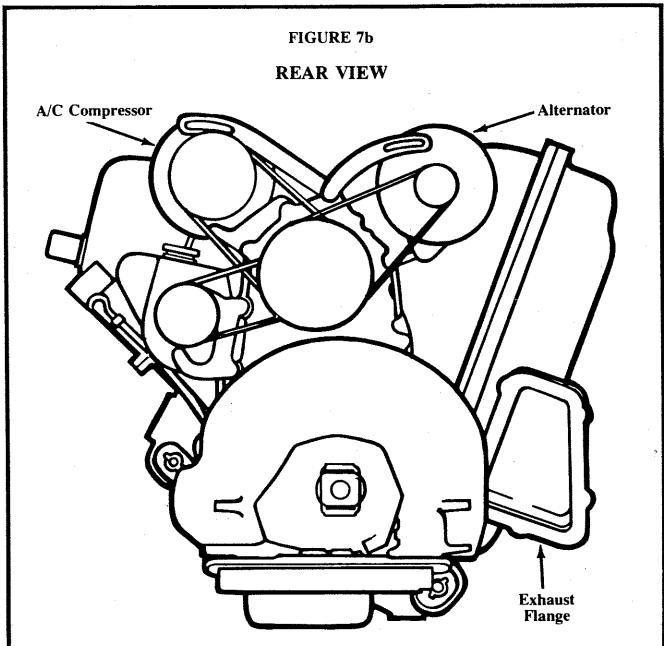
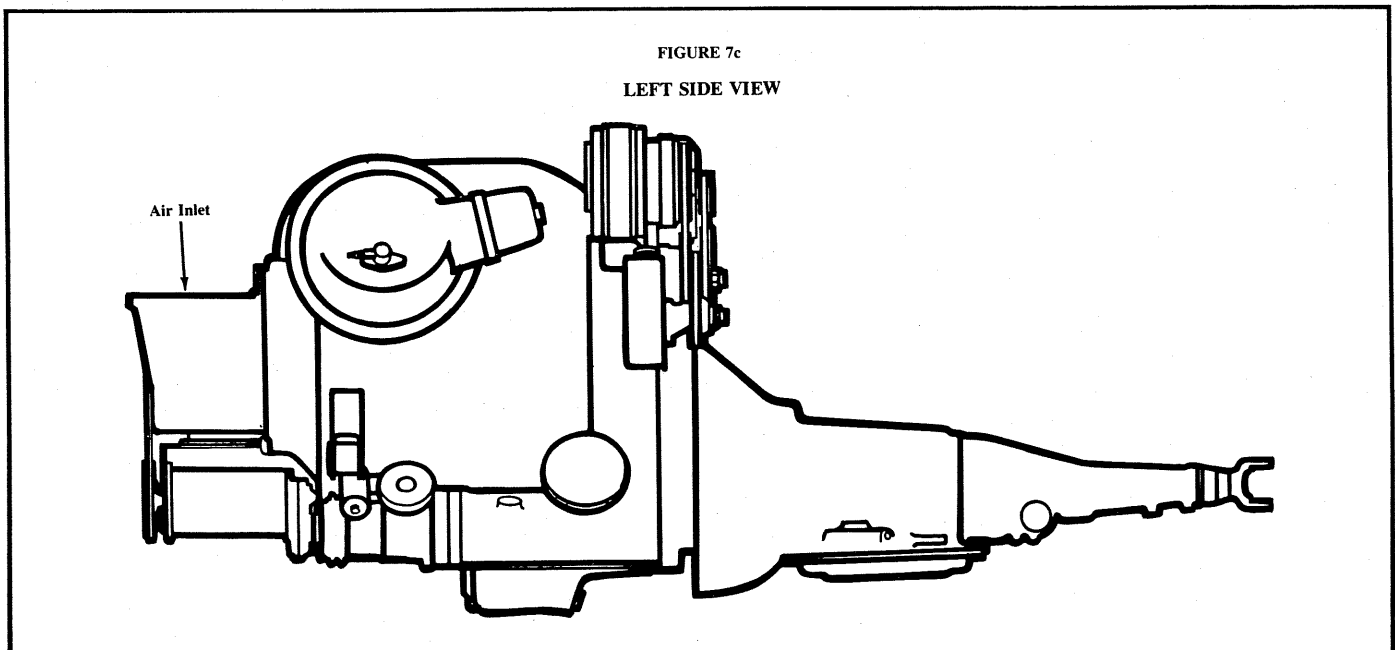


FIGURE 7c  
LEFT SIDE VIEW



We return now to the question of bulkhead cooling referred to previously. A choice can be made of using either the high pressure compressor discharge air or the low pressure regenerator discharge air for the cooling medium. These two media differ in temperature by the regenerator "ineffectiveness", of the order of 50°C. Once again, the choice was a debatable one involving many factors such as the direction and magnitude of the pressure difference across the variable power turbine nozzle stems and the

resulting leakage. Probably the one factor most significant in the choice of compressor discharge air as the cooling source relates to the development potential of the engine. An obviously desirable direction of development is toward higher cycle temperatures. Such increases generally cause the cooling requirements of various parts to become more critical and it was deemed that even the relatively small 50°C difference in favor of the compressor discharge air could be significant in the development potential.

FIGURE 7d  
TOP VIEW

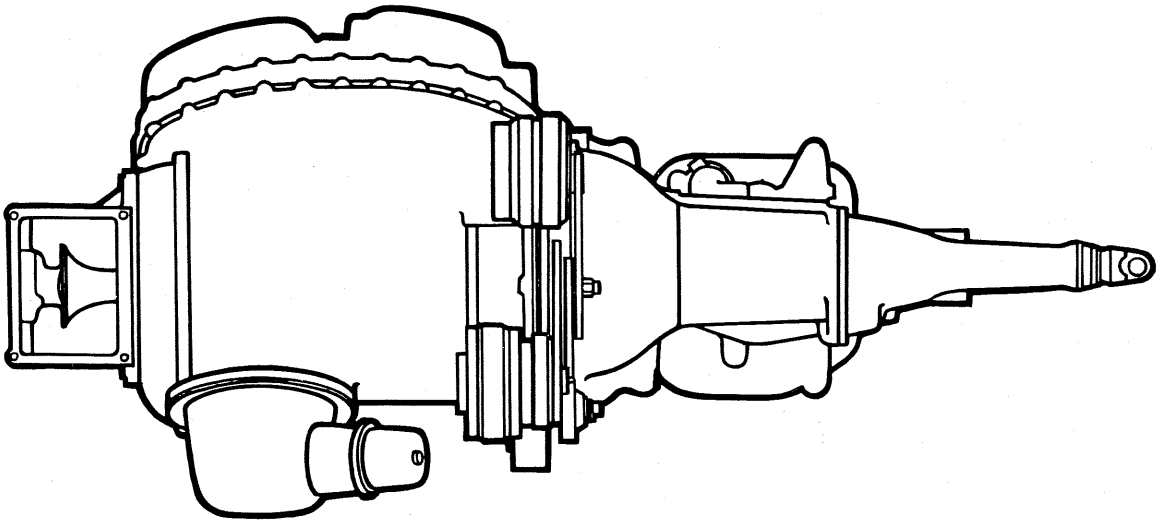
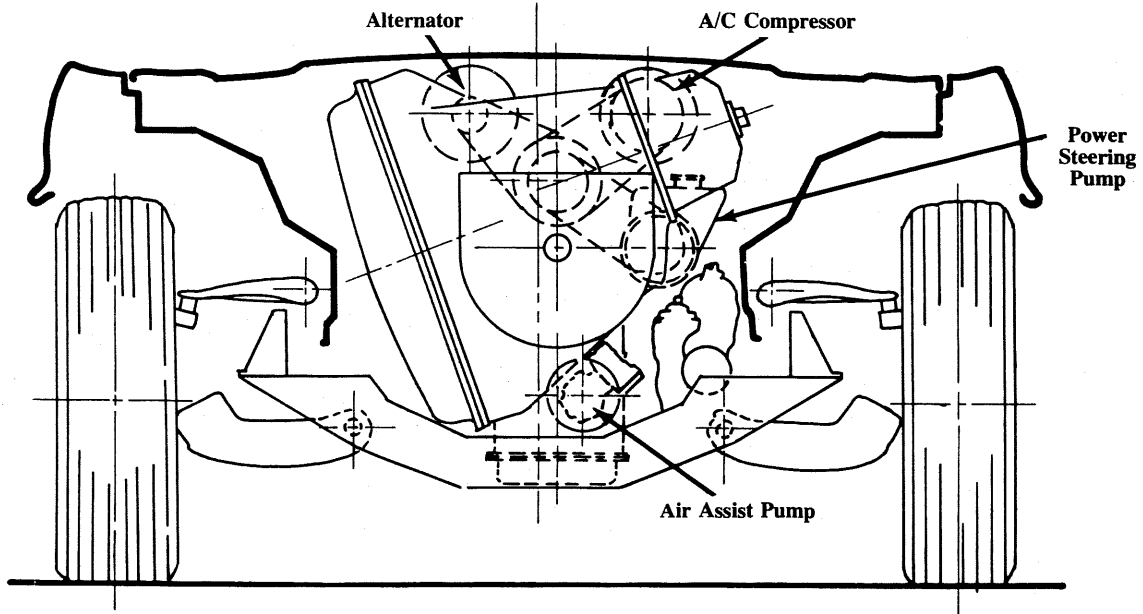
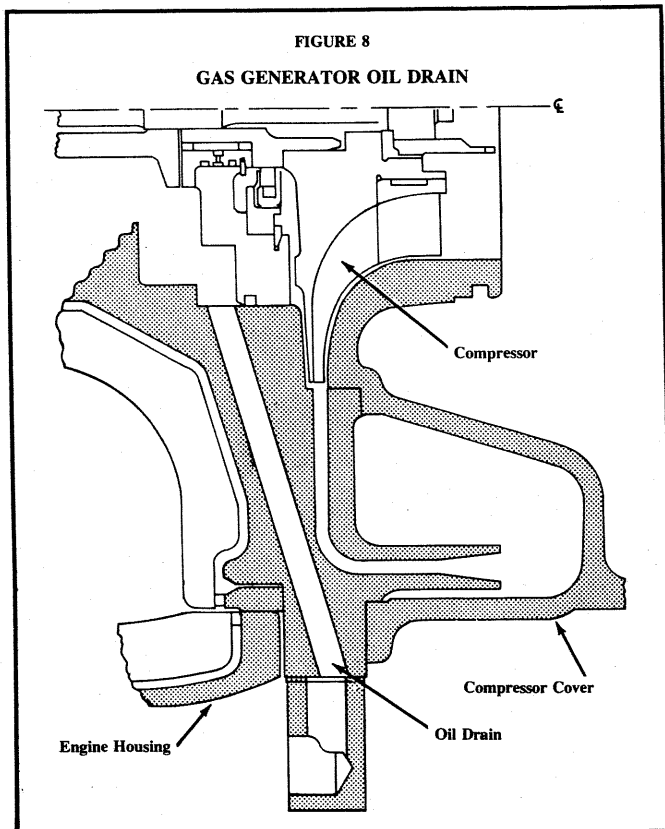


FIGURE 7e  
UPGRADED ENGINE INSTALLATION



The rear of the engine housing terminates in a flange to which the power turbine and reduction gear housing attaches. There are many details associated with this housing assembly including the provision for an adequate power turbine exhaust diffuser, the location of the engine oil sump and accessory drive shafting, etc. Considerations such as heat loss from the gas stream, heat input to sump oil, oil foaming, and ultimate low cost automotive design all are considerations brought to bear. There appear to be no particular problems nor unique features that deserve discussion at this particular time.

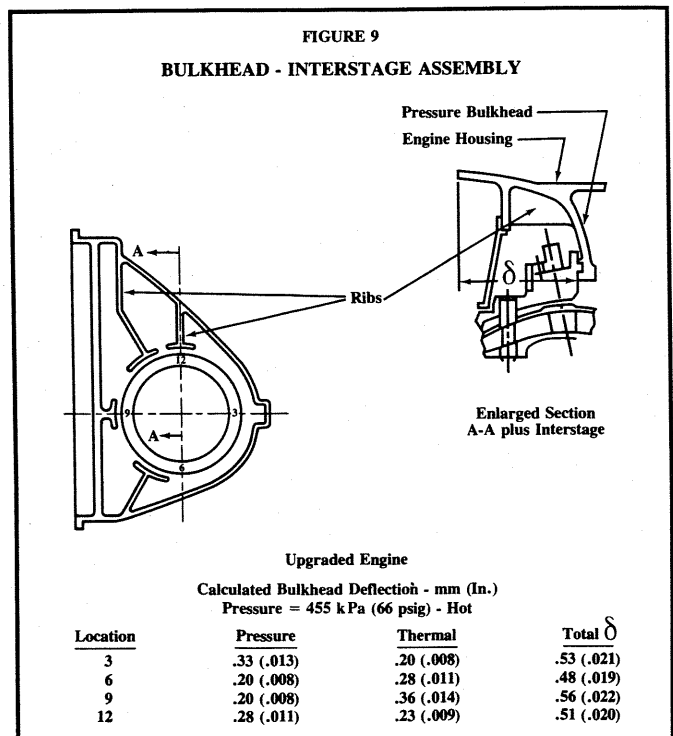
A critical feature of the housing that represents a change from the Baseline configuration is the housing thermal insulation. It is essential to keep the structure temperature in bounds; excessive heat loss and the resultant waste of fuel has disastrous effects on fuel economy and engine compartment temperatures. The Baseline engine has metal insulation liners to contain the insulating material. Not only are these liners relatively expensive and difficult to install, but the insulation can easily become loose and migrate into the engine interfering with seals and causing trouble. Considerable effort has been expended in developing a "linerless" insulation; that is, a molded insulation that can be cemented to the housing and in itself form the flow passages within the housing and not erode from scrubbing action of the gas flow. Significant success has been achieved through these efforts, but it is likely that additional development work will be required to meet the specific environment associated with the new engine housing at its higher temperature levels.



As a final item in the housing design, it seems appropriate to report the results of housing bulkhead deflection calculations as an indication of the level of effort that has gone into the design. A relatively stable housing is essential if minimum gaps are to be maintained in the leakage paths and rotating component clearances. Such control is vital to the overall efficiency of the engine, and thus to achieve our goal of good fuel economy, it is necessary to assure that, in spite of its lack of symmetry, the housing will not warp out of shape. One important indication of housing stability is the deflection of the bulkhead under load. Figure 9 illustrates a study of the bulkhead area. After checking the method of computation by comparing computed results with some measurements made on the Baseline housing, the method was applied to the Upgraded housing. The deflections shown in the figure are due to both pressure loads and thermal distortion. Such calculations are rarely precise, but at least it is possible to verify that these total deflections are less in magnitude and more uniform than the Baseline, where deflections due to pressure loads alone of more than .75 mm (0.030") were measured. In addition to these specific tests, running experience with the Baseline engine, including some rubbed turbine wheels, implied appreciable warpage and deflections of 1.3 mm (.050") or more. Compared to these deflection data, the calculated deflections of the Upgraded housing suggest strongly that we have achieved our goal of providing a stable environment for the flow paths.

## TURBO-MACHINERY AND PERFORMANCE MAPS

Prior to the design of the turbo-machinery components, specifications for these components were laid down. The requirements for an automotive turbine are stringent and the



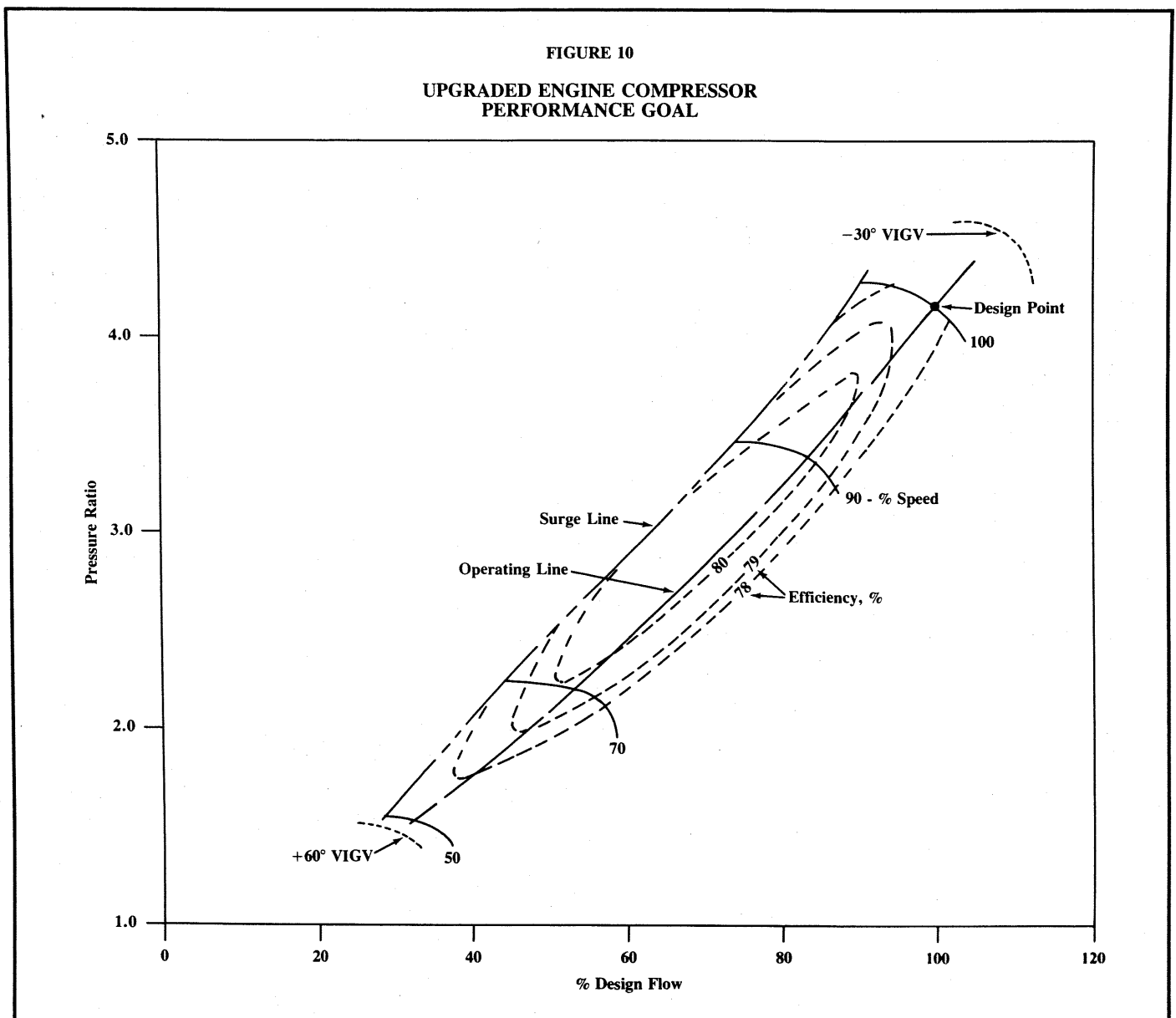
resulting specifications greatly limit the design freedom. On the other hand, it is important to assure that the requirements are valid and that the design is not unnecessarily restricted. The Upgraded Engine components emphasize fuel economy whereas the Baseline engine components were designed to have a minimum of rotational inertia, even at the expense of some aerodynamic efficiency. The augmentation concept being used for the Upgraded engine as well as the specification of a smaller vehicle has permitted a large reduction in the engine power and, hence, size. In addition, an increase in maximum turbine inlet temperature to 1052°C (1925°F) improves the specific power of the engine, also reducing its size.

It is well known that the responsiveness of an engine, as determined by the time required for the gas generator to reach full speed from idle, is directly related to its power level. However, improved responsiveness over that of the Baseline vehicles is not needed for adequate driveability.

This allowed the gas generator rotational inertia/efficiency compromises to favor efficiency. These compromises permitted employing 30° of backward curvature on the compressor impeller blades and setting the compressor turbine "design point" further from the limit loaded (and less efficient) condition. Backward curvature results in a more efficient basic compressor as well as increasing the useable flow range between surge and choke. The range improvement will permit keeping regenerator inlet temperature ( $T_8$ ) up to the full load value of 748°C (1378°F) for all part load conditions. This, too, will improve economy. References 4 through 6 describe the detailed aerodynamic design performed by NASA Lewis for the compressor, compressor turbine, and power turbine.

### COMPRESSOR

The resulting compressor uses a 24 bladed backward curved impeller. Backward curved impellers have good



efficiency and flow range, but at the expense of polar inertia. On the other hand, the broad flow range of this compressor will permit flexibility in the control system which, together with the high efficiency, is expected to make a significant contribution to the fuel economy of the engine. A map of predicted performance is given in Figure 10.

**COMPRESSOR TURBINE**

Two parameters that determine the efficiency potential of a turbine are the Mach number level at maximum speed and the work factor\* (or degree of reaction). In both of these

\*Work factor is the ratio of turbine specific work to the square of the tip speed and is commonly expressed in several forms, e.g.

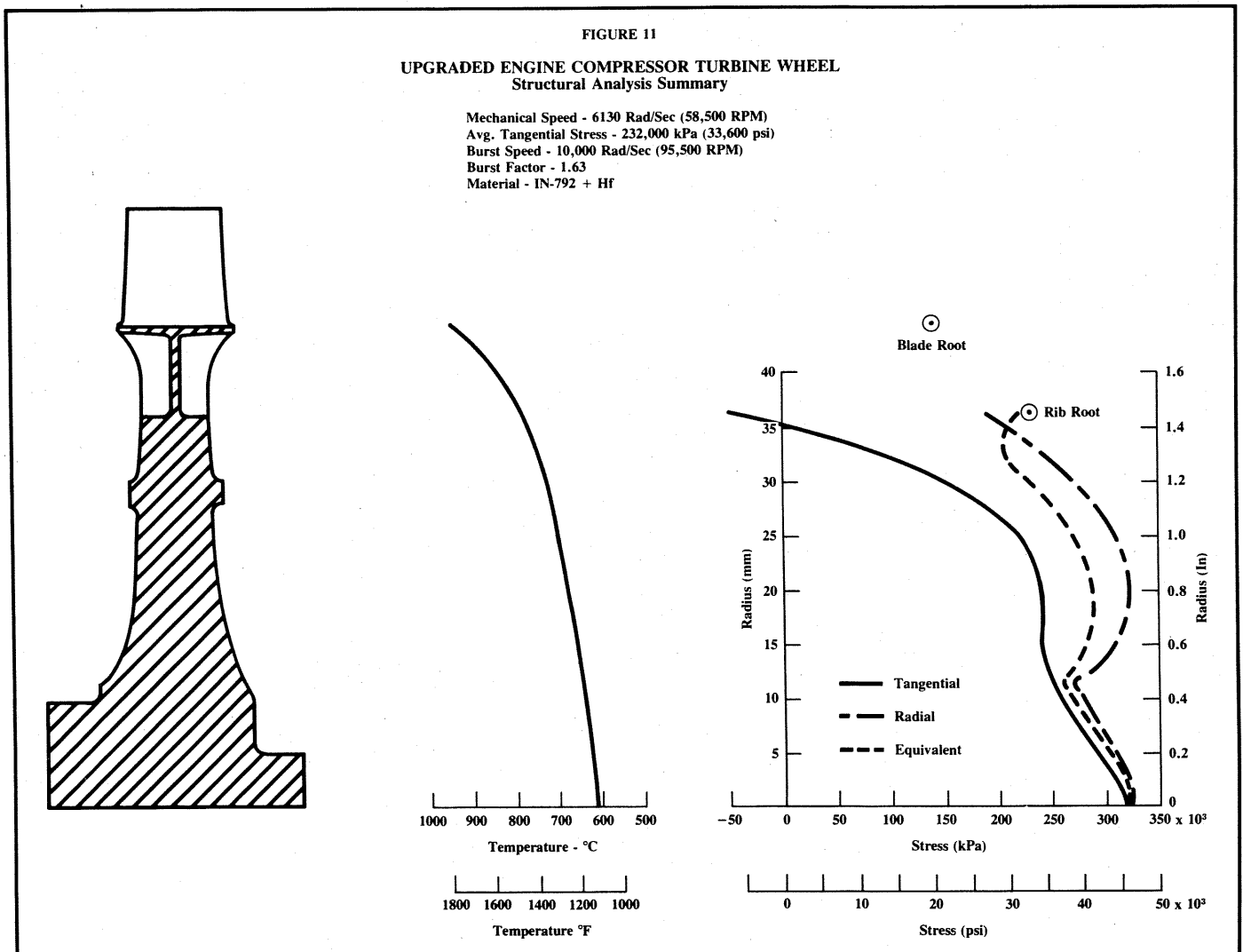
$$\text{Work Factor} = \frac{\Delta h}{U_m^2} = \frac{U_m \Delta C_u}{U_m^2} = \frac{\Delta C_u}{U_m}$$

The specific work is  $U_m \Delta C_u$ . Since  $U_m$  is tip speed and  $C_u$  measures the turning, the Work Factor expresses the relative importance of  $\Delta C_u$  in the production of work.

parameters, the Upgraded engine operates at levels that are potentially more efficient than the Baseline. The work factor is 2.1 as opposed to 2.4 which results in velocity triangles that are less extreme. The Upgraded design is essentially subsonic which should add about two points to its general efficiency level. As opposed to this, the Baseline engine compressor turbine (in order to minimize inertia) operates in the transonic regime near limit loading. Currently available materials have permitted an increase in turbine inlet temperature to 1052°C (1925°F). The resulting wheel temperature and stress distributions are shown in Figure 11. Figure 12 shows the performance map for the stage.

**POWER TURBINE**

In keeping with the intent of the Upgraded engine, a great deal of emphasis was given to the moderate and low speed road load efficiency. This led to a design where the peak power speed, or "design" speed, is at an RPM considerably below the value for level road maximum vehicle velocity. In fact, maximum vehicle speed occurs at about 50% over the "design" speed. As a result, the road



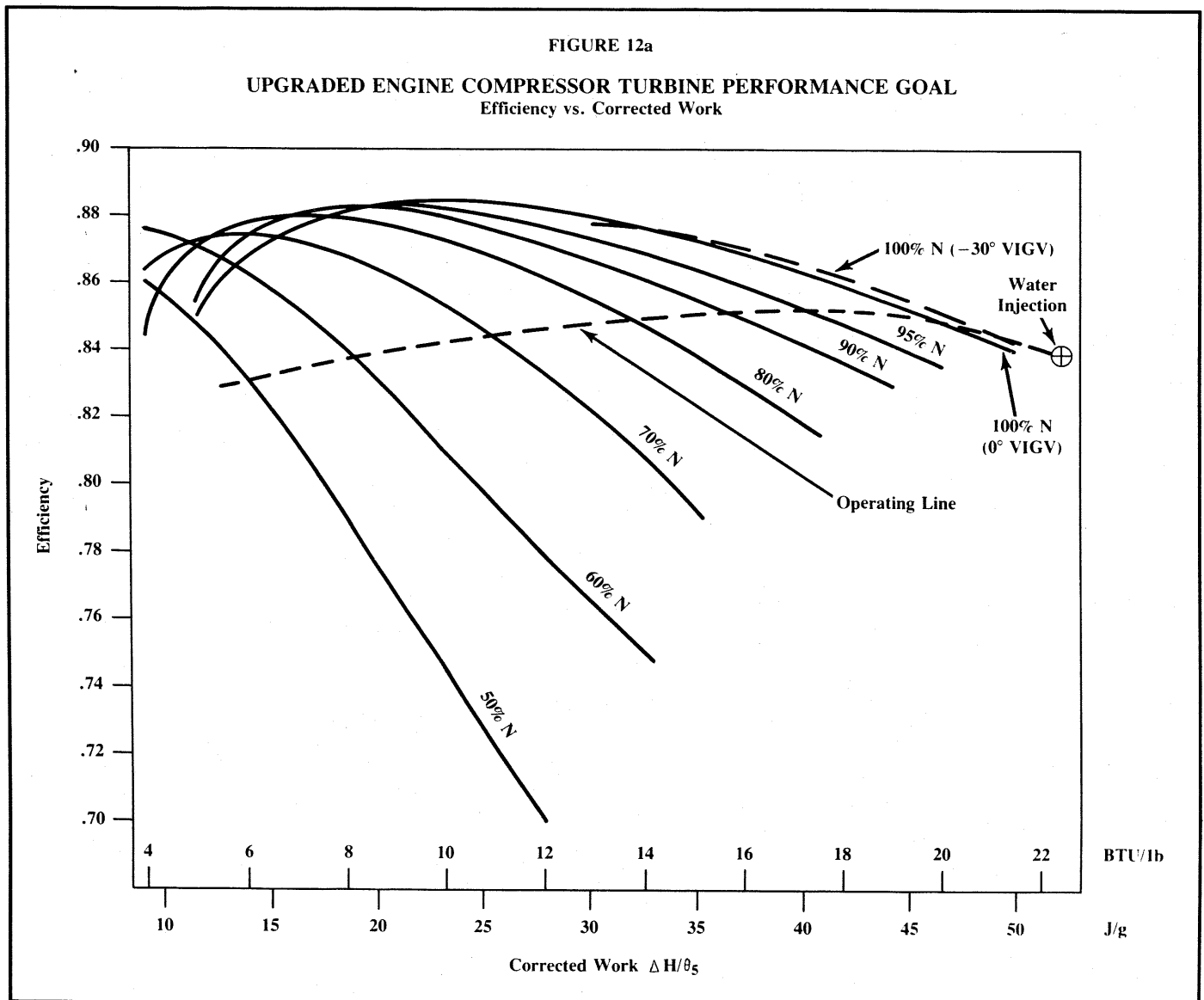
load line through the gears lies within the band of the turbine performance map that is within two points of the maximum efficiency at any speed. Thus, the design provides close to the best possible fuel economy. One undesirable result is that the maximum stress speed of the wheel is quite high and in general the power turbine operates at relatively high speeds. It has been difficult to keep the stress levels within the limits of available materials. Figure 13 illustrates the temperature and stress distribution. Moreover, the high speed journal bearings and the power turbine reduction gears present obvious additional problems as speeds go up. As experience with the Upgraded engine is accumulated and the power turbine wheel life has been evaluated, it will be possible to determine if the trade-off of efficiency gains as compared to design speed has met the goals of the program. The performance map is shown in Figure 14.

**CONTROL**

The control system for this engine, including the novel burner controls and the power turbine nozzle and VIGV control systems, is of sufficient scope to warrant a separate discussion. This is the subject of a joint paper with Chrysler and AiResearch to be presented soon.

**ACCESSORY DRIVES**

The adoption of the free rotor configuration required considerable change in the philosophy of the engine accessory drives. This is primarily a result of the much larger speed range through which the accessories are required to run. In the Baseline engine, with gas generator driven accessories, their speed range was 2:1 from maximum output to idle. In the Upgraded engine, with power turbine driven accessories, the normal speed range is 8:1 from maximum transmission input shaft speed to torque converter uncouple speed at idle. The result of this is that

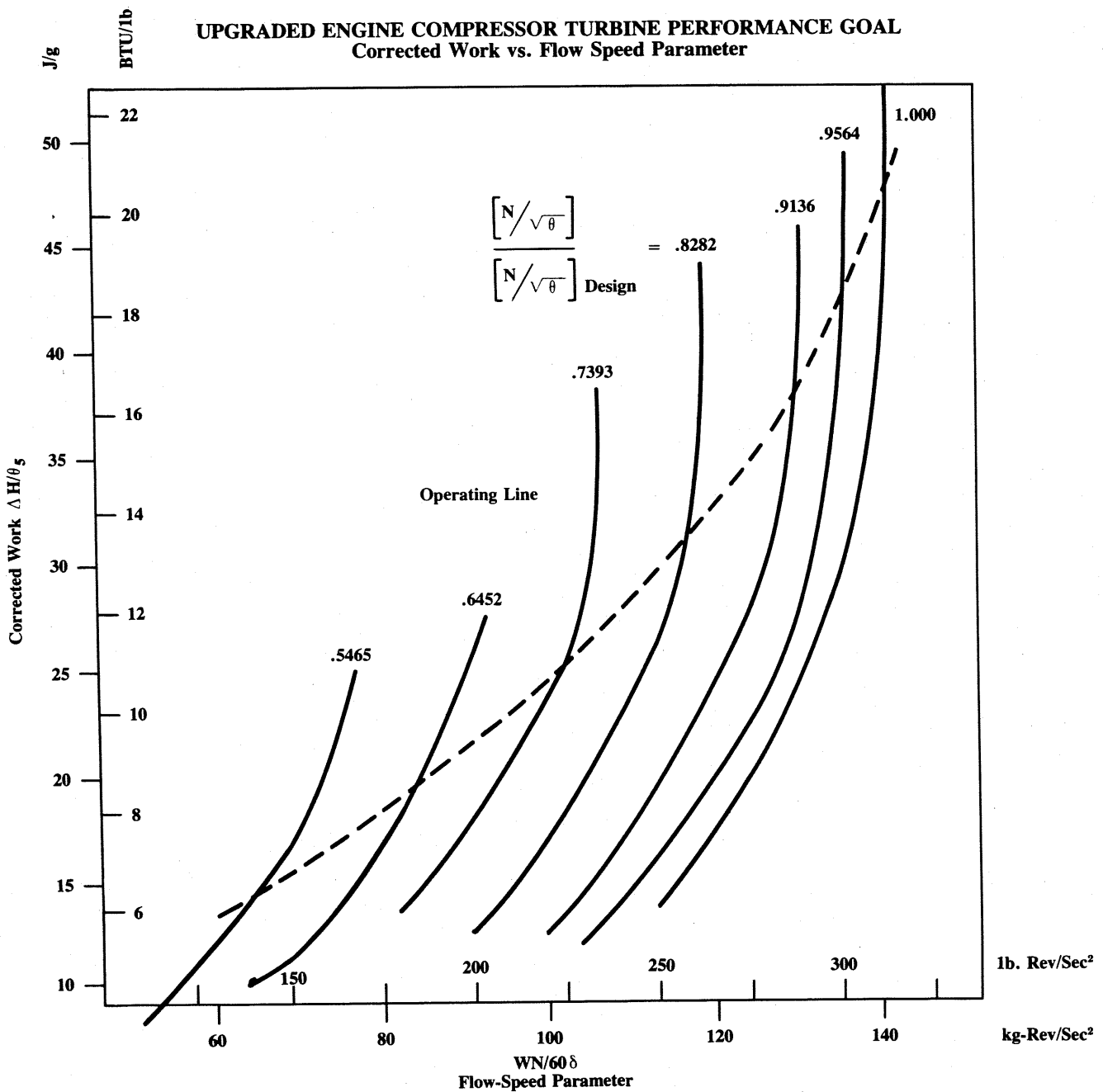


all of the engine accessories, the atomizing air pump, lubricating oil pump and regenerator, are arranged to run just fast enough at idle to satisfy the requirements. This minimizes the excess capacity at maximum speed and its attendant parasitic losses. In line with this, the hydraulic system was divided into both the lubrication and muscle systems.

Hydraulic power for both the VIGV and power turbine nozzle actuation is obtained from what was the power steering pump overflow. This permitted reduction of the basic engine oil pump size since it is required to supply only the lubrication needs of the engine.

In addition, the starting philosophy required revision for special provision to drive certain accessories in order to start the free rotor engine. It is essential that the air pump run to provide combustor atomizing air and that the oil pump run to provide lubrication for the gas generator thrust bearing and front journal bearing. This is accomplished by using an electrically actuated clutch that engages the fore and aft accessory drive shaft from the front during the start cycle using the starter motor to drive just the air pump and oil pump but not the regenerator. However, when the power turbine starts to rotate, it drives the regenerator and will also pick up the air and oil pumps via a rear

**FIGURE 12b**  
**UPGRADED ENGINE COMPRESSOR TURBINE PERFORMANCE GOAL**  
**Corrected Work vs. Flow Speed Parameter**



overrunning clutch when the starter motor shuts off.

The starter motor belt drives an inertia clutch for engaging the gas generator rotor.

**REGENERATOR AND SEAL**

Honeycomb ceramic is used for the regenerator. This permits maintaining regenerator inlet temperature, T8, at a maximum value of 748°C (1378°F) throughout the entire operating envelope of the engine. As well, in the interest of fuel economy, it permits using a relatively oversize single core without it being unreasonably heavy. The core has an effective diameter of 502 mm (19.8 in.) and a thickness of 83 mm (3.27 in.). The effectiveness is given in Table 3.

A self-aligning center bearing support is used with rim drive. The drive gear is attached to the core with a bonded elastomer.

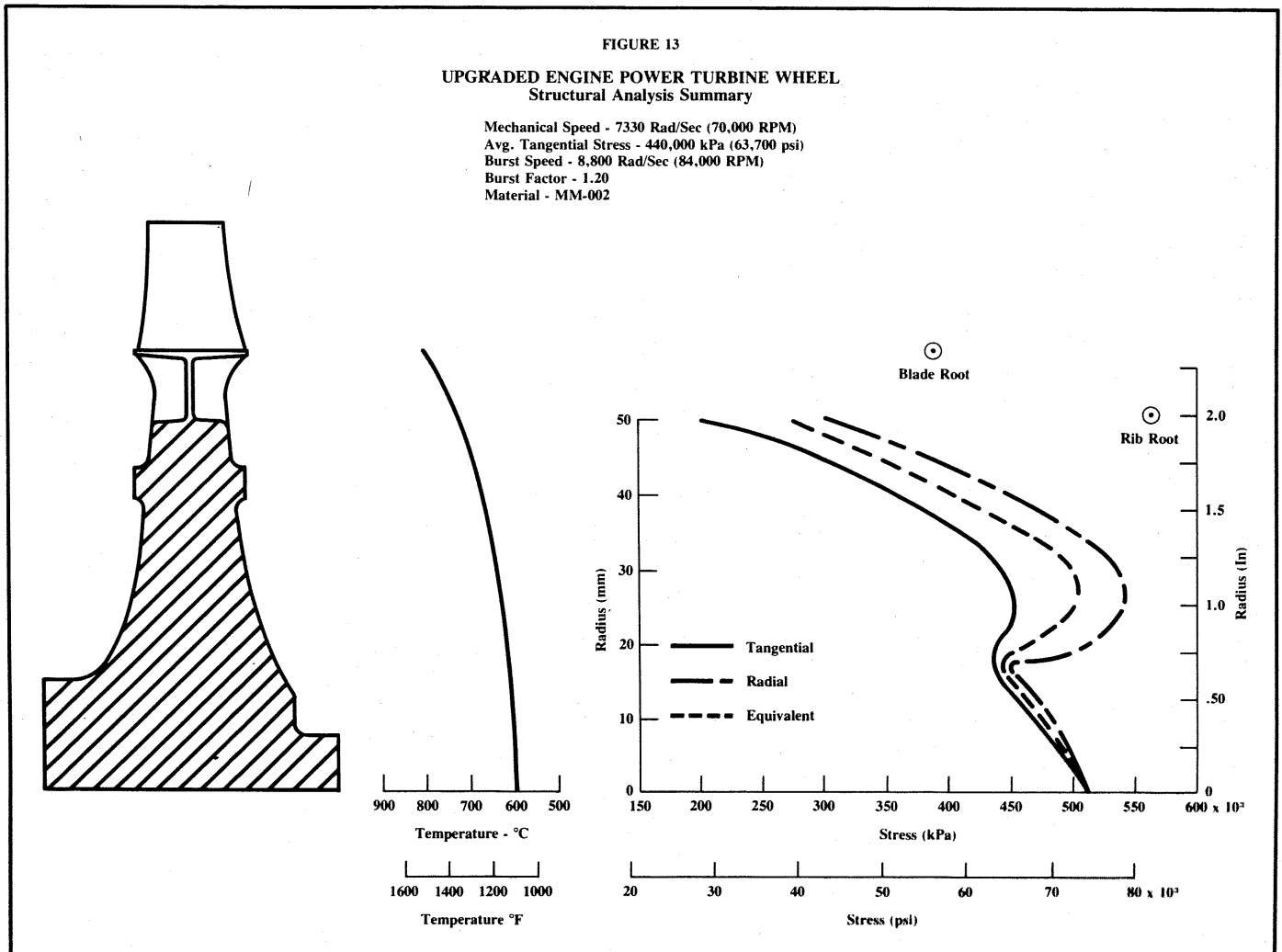
The seals for the regenerator are similar to the Baseline engine seals. The flexible members have been slightly revised to reduce the clamping force to that needed for conformability with the less distorted ceramic regenerator. This approach was used to minimize the investment in a less than optimum design. A proprietary seal system is

being developed that will eventually replace the relatively expensive Baseline type seal.

Until rubbing seal face coatings with acceptable friction characteristics are developed, NiO will be used. Although fine particles of this material are known to be carcinogenic, based on the established wear rate NiO emissions will be only 0.2% of the hazard threshold level.

**GEARS AND BEARINGS**

The high speed gears and bearings of a gas turbine engine are in some aspects more critical than conventional moderate speed automotive drives. The criteria by which we judge their performance, however, remain the same, namely, quietness, long life, low parasitic power, etc. Recent comprehensive compilations of data (References 7 through 9) have provided the designer with the means to account for all these requirements. Careful application of established procedures and guidelines should result in superior gear and bearing performance. The gears and bearings of the Upgraded engine represent a significant improvement over the Baseline designs, primarily with respect to noise and road load parasitic losses. This is



evident in Figures 15 through 19 which predict the power turbine reduction gear performance.

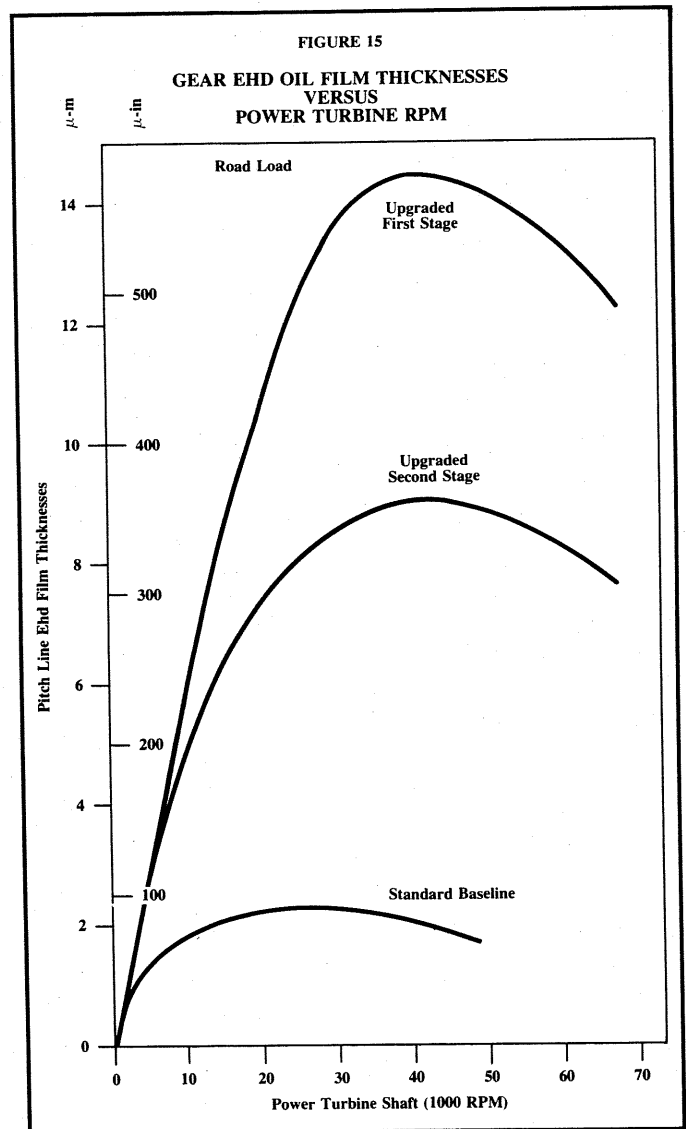
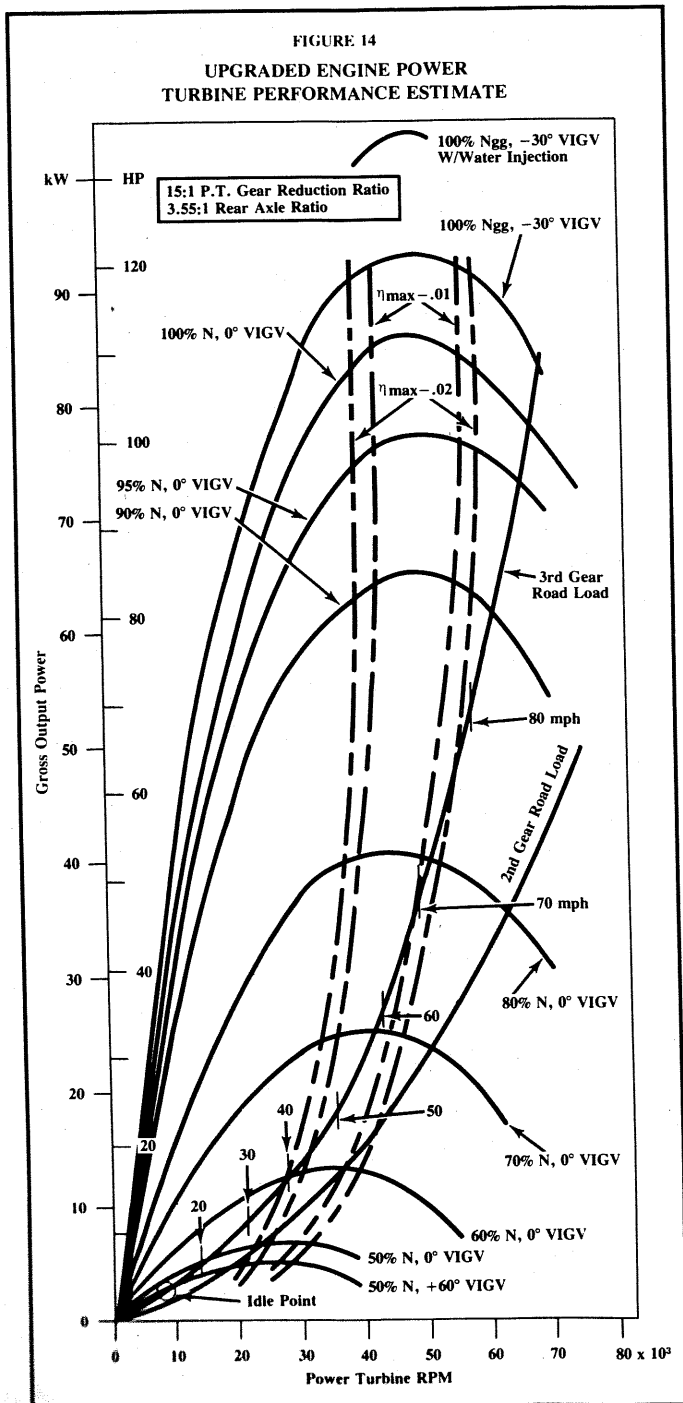
The bearing design is typified by the operating envelope of Figure 20. The complete system shafting and gearing is schematically shown in Figure 21 and basic gear data is given in Table 4.

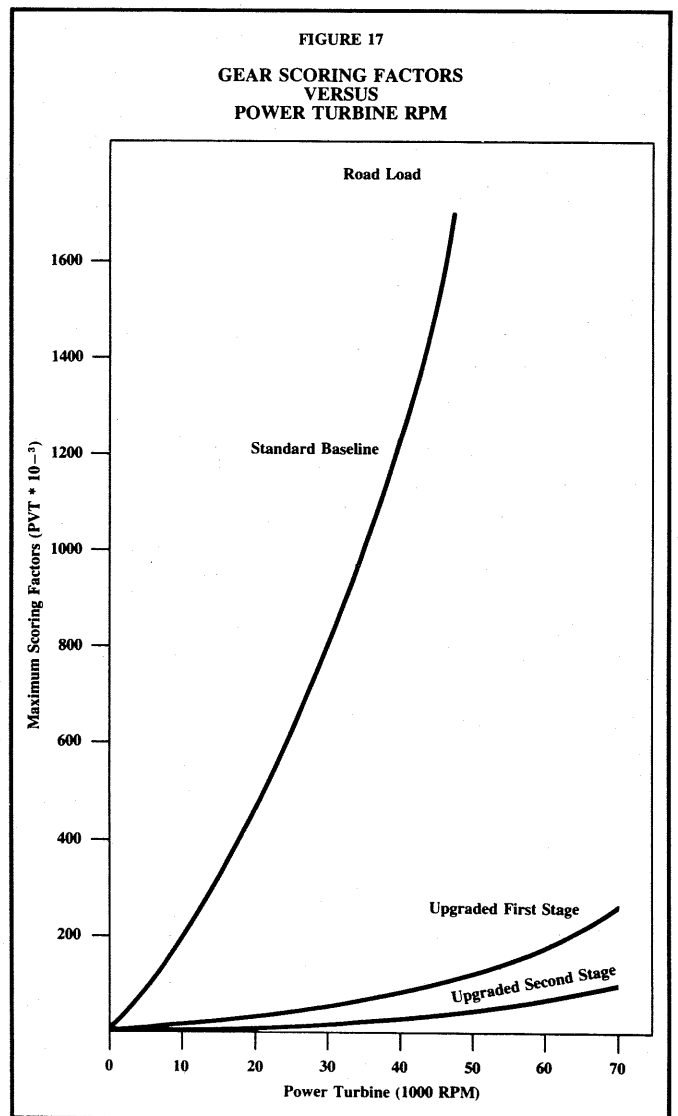
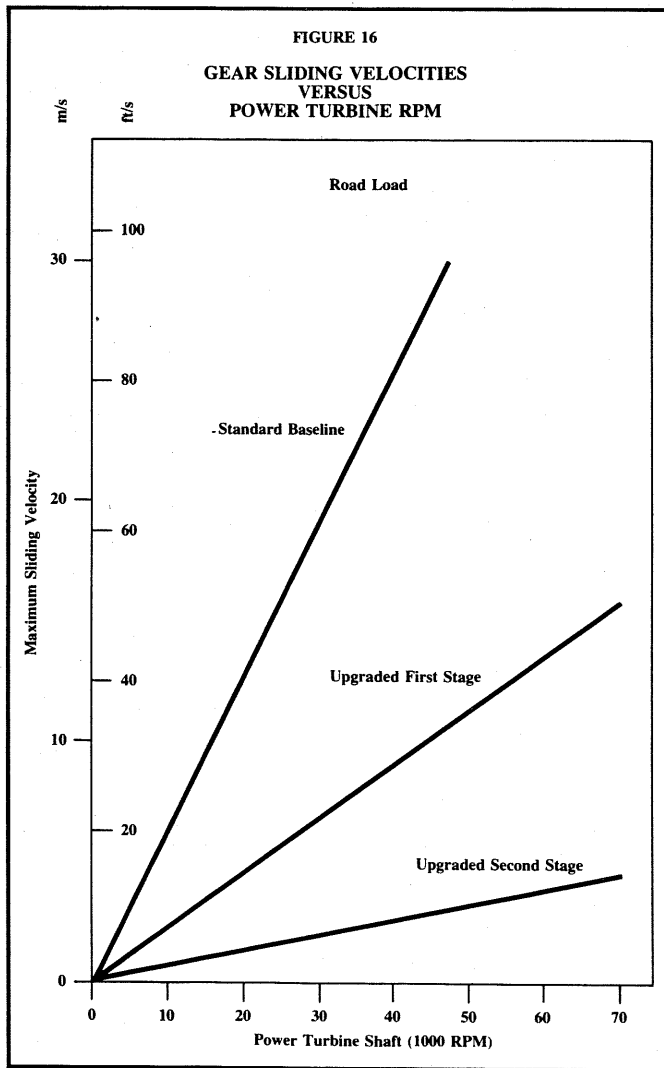
**SUMMARY AND CONCLUSIONS**

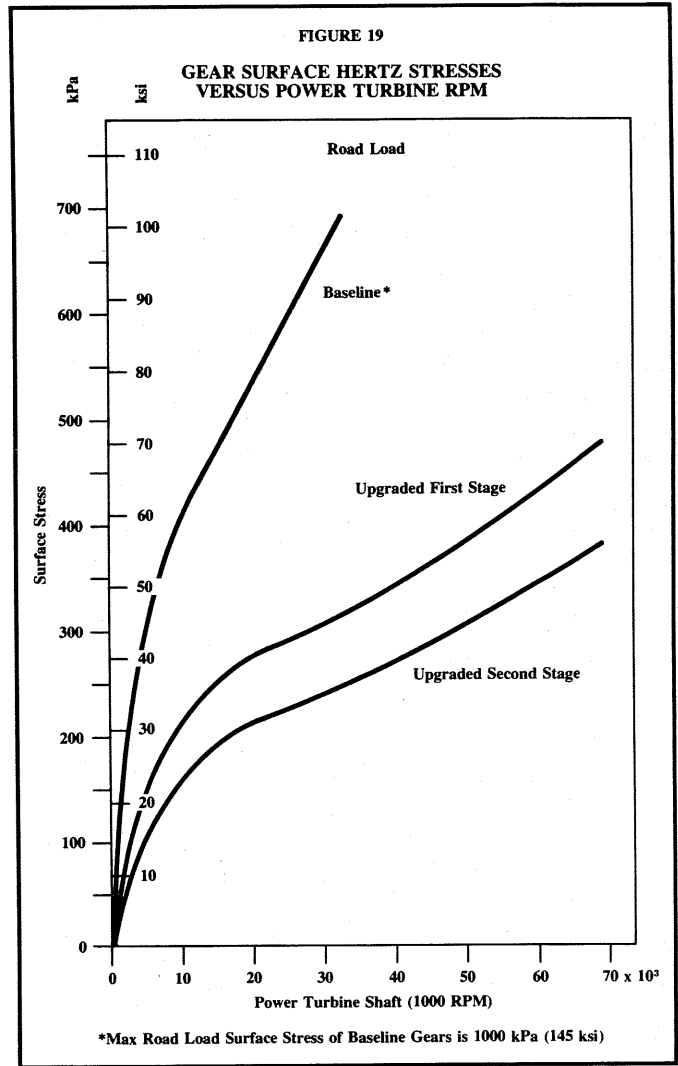
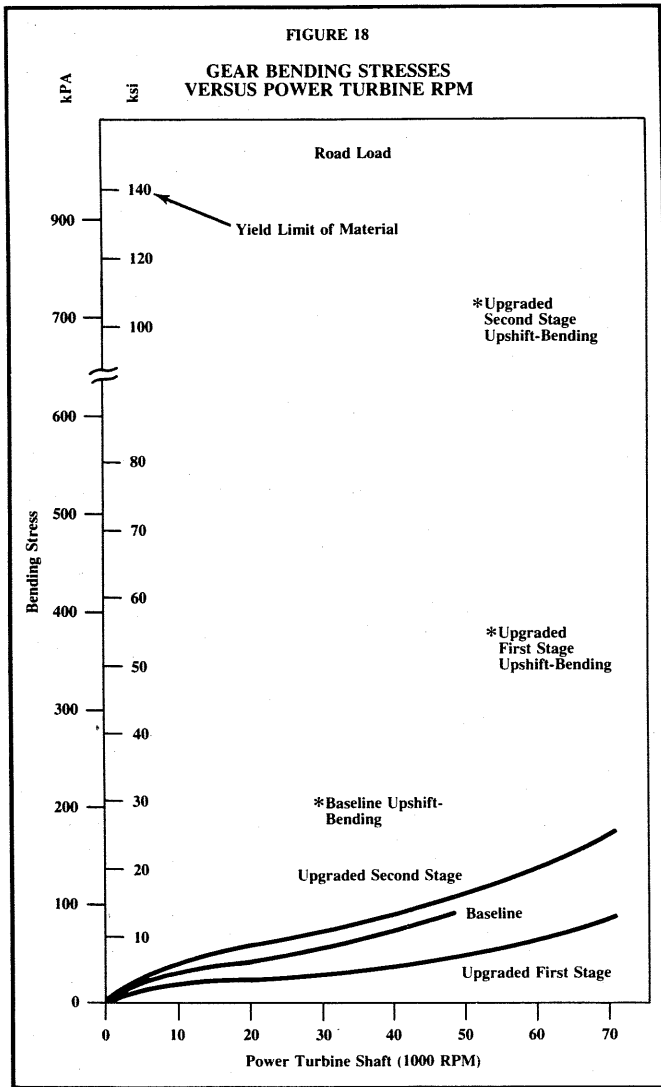
In this paper we have summarized the major factors leading to the selection of power augmentation and a free

rotor configuration for the design of the Upgraded Engine. It has been shown that these features are a rational approach to the development of a viable option to the conventional automotive engine for improved fuel economy and cost. In addition, the engine has been generally described.

An endeavor as extensive as that reported here involves the contributions of many individuals. Although it is impractical to name each of them here, the credit for this work belongs as well to them.





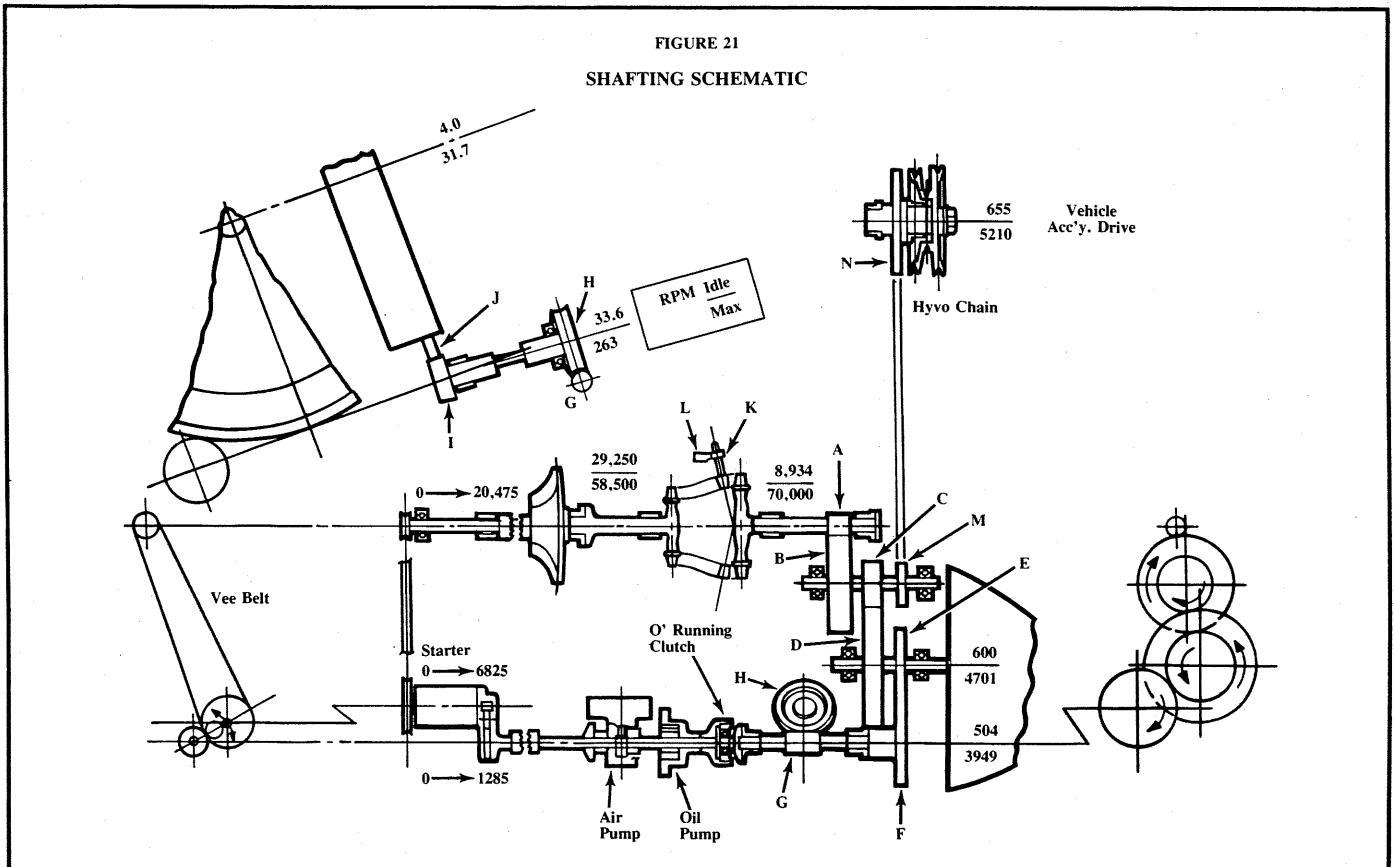
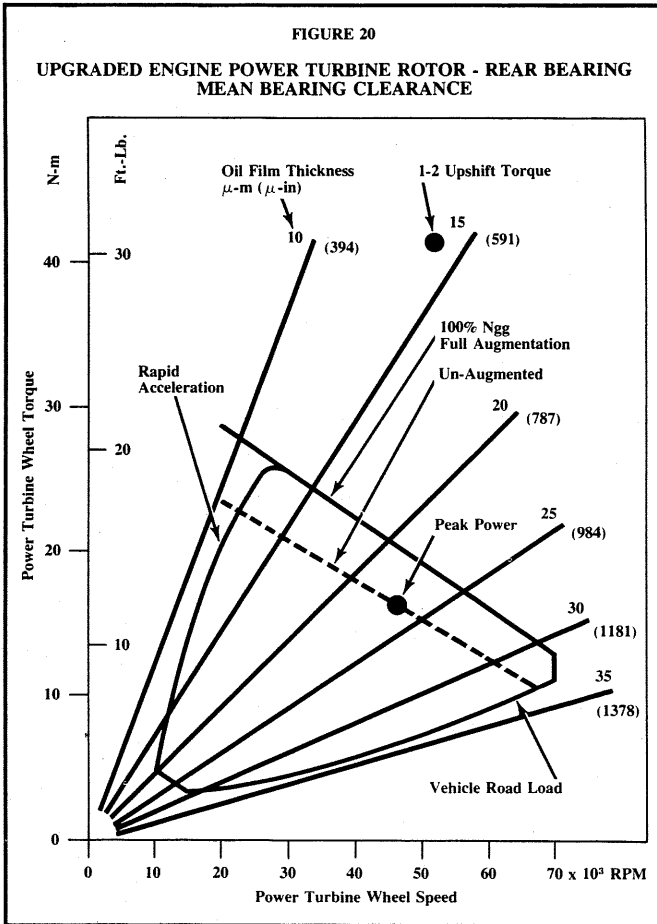


**TABLE 4**

**GEAR DATA**

* No.	Description	Max. RPM	Material	Face Width mm (In.)	No. Of Teeth	Pitch Dia. mm (In.)	Press. Angle Deg.	Location
A	Power Turbine Pinion	70,000	SAE 6150 Steel	28 (1.1)	28	25.4 (1.0)	20	P.T. Reduction
B	Intermediate Gear	12,405	SAE 6150 Steel	25 (1.0)	158	143.5 (5.65)	20	P.T. Reduction
C	Intermediate Pinion	12,405	SAE 6150 Steel	30 (1.2)	72	65.8 (2.59)	20	P.T. Reduction
D	Transmission Input Gear	4,701	SAE 6150 Steel	28 (1.1)	190	172.5 (6.79)	20	P.T. Reduction
E	Accessory Drive Pinion	4,701	Nodular Iron	18 (0.7)	105	102.6 (4.04)	20	Accessory Drive
F	Accessory Drive Gear	3,949	Nodular Iron	15 (0.6)	125	122.2 (4.81)	20	Accessory Drive
G	Worm	3,949	SAE 8620 Steel	44 (1.75)	6	21.3 (.84)	20	Accessory Drive
H	Worm Gear	263	Manganese Bronze	16 (.62)	90	101.3 (3.99)	20	Accessory Drive
I	Regenerator Pinion	263	SAE 1046 Steel	25 (1.0)	38	68.8 (2.71)	20	Accessory Drive
J	Regenerator Gear	32	Cast Iron	15 (.6)	315	571.5 (22.5)	20	Regenerator
K	V.P.T.N. Sector Gear	—	Stainless Steel	8 (.3)	20	27.9 (1.1)	20	Interstage
L	V.P.T.N. Ring Gear	—	Cast Iron	15 (.6)	150	208.3 (8.2)	20	Interstage
M	Driver Sprocket	12,405	SAE 1141 Steel	13 (.5)	21	61.0 (2.4)	—	P.T. Reduction
N	Driven Sprocket	5,210	SAE 1045 Steel	13 (.5)	50	152.4 (6.0)	—	Ancillary

\*See Figure 21



## REFERENCES

1. P.R. Angell and T. Golec, "Upgrading Automotive Gas Turbine Technology/An Experimental Evaluation of Improved Concepts." SAE Paper 760280 February 1976.
2. S. Gray, N. Sparks, and J. McCormick, "The Application of Gas and Oil Lubricated Foil Bearings for the ERDA/Chrysler Automotive Gas Turbine Engine." ASME Paper 76-GT-115 March 1976.
3. J. S. Collman, C. A. Amann, C. C. Matthews, R. J. Stettler, and F. J. Verkamp, "The GT-225 — An Engine for Passenger-Car Gas-Turbine Research." SAE Paper 750167, February 1975.
4. M. R. Galvas, "A Compressor Designed for the ERDA Automotive Gas Turbine Program." NASA TM X-71719.
5. R. J. Roelke & K. L. McLallin, "The Aerodynamic Design of a Compressor Drive Turbine for use in a 75 kW Automotive Engine." NASA TM X-71717.
6. M. G. Kofskey and T. Katsanis, "Aerodynamic Design of a Free Power Turbine for a 75 kW Gas Turbine Automotive Engine." NASA TM X-71714.
7. T. F. Conry and A. Seireg, "A Mathematical Programming Technique for the Evaluation of Load Distribution and Optimal Modifications for Gear Systems." ASME 72-PTG-5.
8. L. S. Akin, "An Interdisciplinary Lubrication Theory for Gears (With Particular Emphasis on the Scuffing Mode of Failure)." ASME 72-PTG-6.
9. E. P. Remmers, "Analytical Gear Tooth Profile Design." ASME 72-PTG-47.
10. Jet Propulsion Laboratory / California Institute of Technology, "Should We Have A New Engine?" JPL SP 43-17 Vol. II

## APPENDIX

### BASELINE ENGINE DESCRIPTION

The 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. In general, the engine was designed to permit concept evaluation, ease component development, and develop manufacturing technology. Integrally cast components, such as turbine wheels and nozzles, are utilized in separate bolt-in sub-assemblies that can be easily changed. Drawings of the basic engine arrangement are shown in Figures A1 through A3.

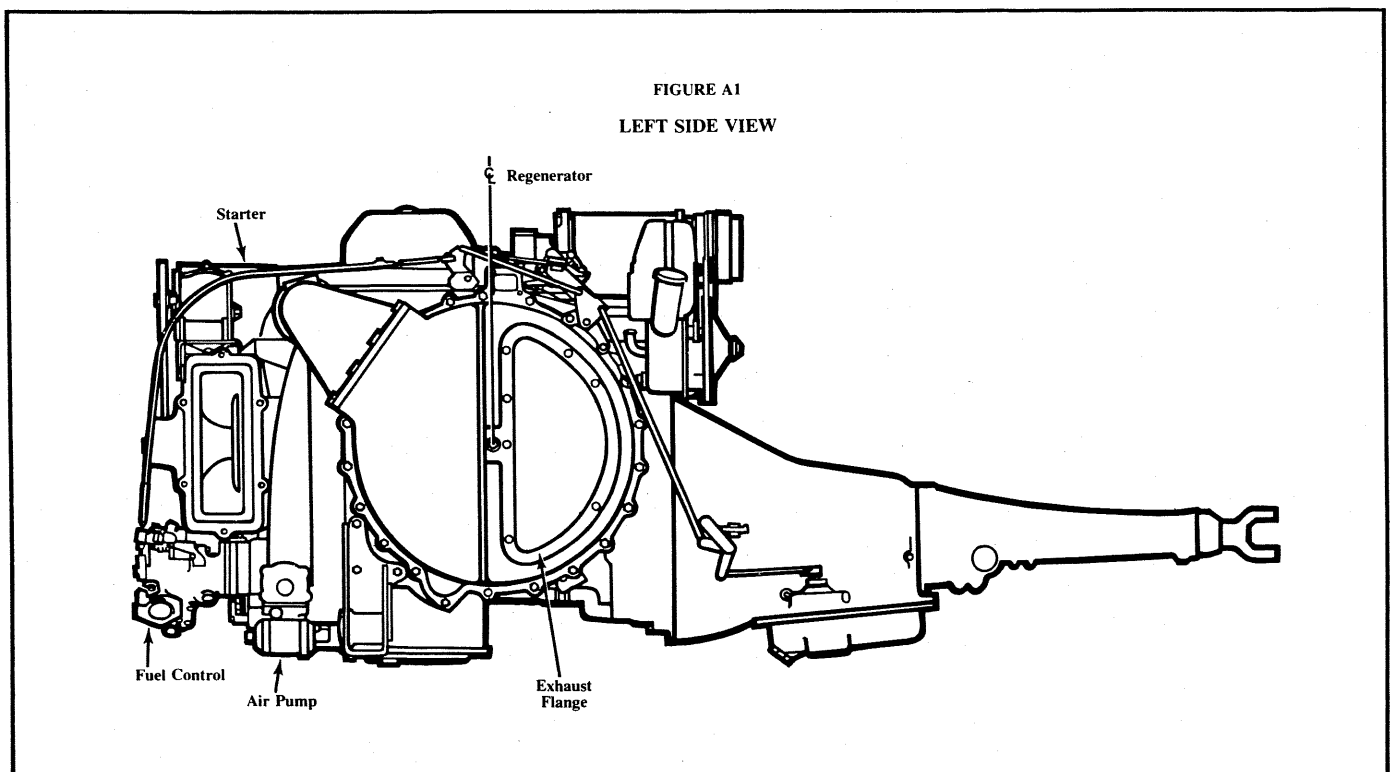


FIGURE A2  
TOP VIEW

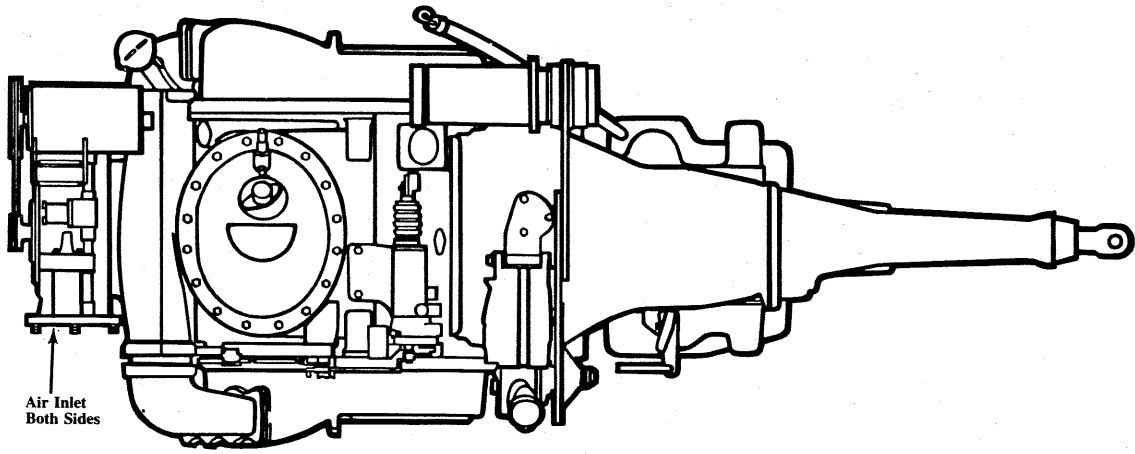
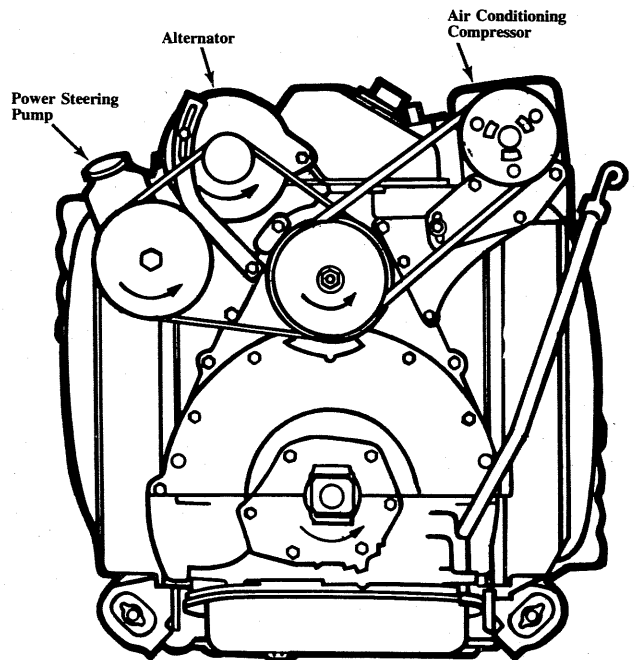
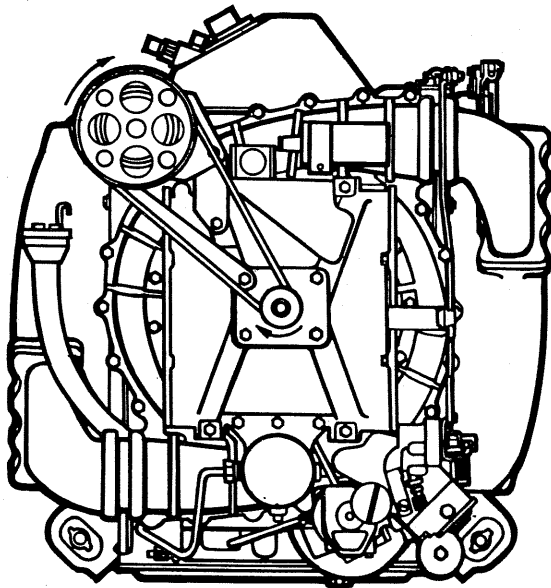


FIGURE A3

FRONT VIEW

REAR VIEW



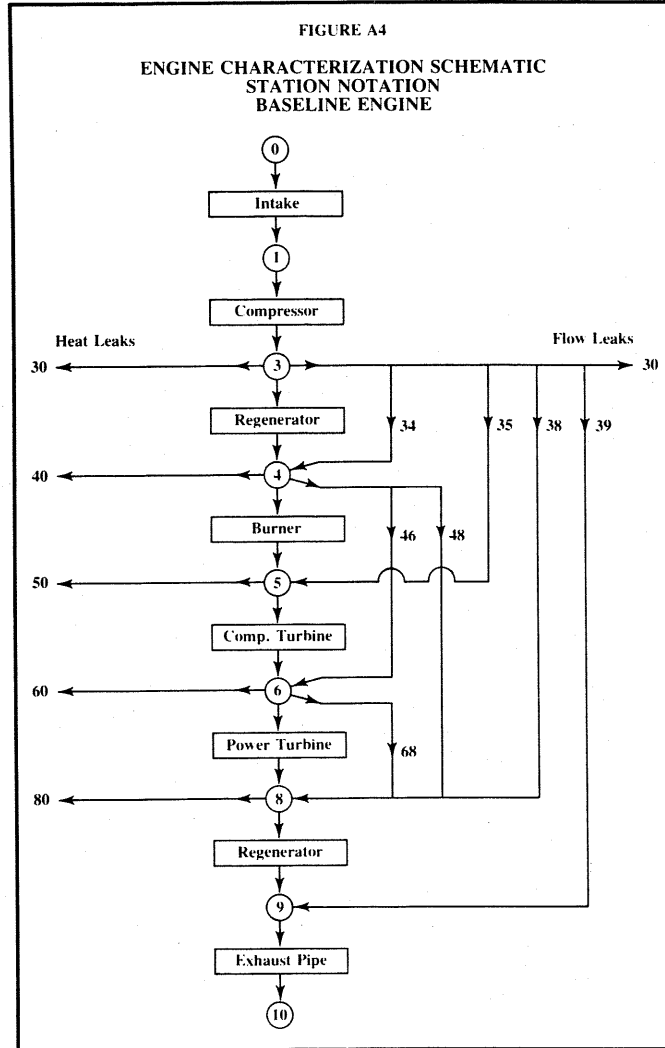
Some 85°F full power characteristics along with other descriptive data are listed in the following:

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"
Matrix Height	3.5"
Stock Thickness	.002"
Overall Dimensions	
Length to Transmission Mounting Flange	35.5"
Width	27.6"
Height	29.9"
Weight, Complete Engine - Dry	600 lb
Gas Generator Acceleration, Idle-to-Maximum Speed	1.2 sec.
Brake Specific Fuel Consumption	.54 lb/HP-Hr
Idle Fuel Flow	
50% Speed	10 pph
40% Speed	8 pph

Complete engine data is given in Table A1.

TABLE A1  
 CHRYSLER CORPORATION  
 SIXTH GENERATION GAS TURBINE  
 ENGINE CHARACTERIZATION  
 (Station Notation Defined In Figure A4)

Gas Generator Speed, RPM	22,877	27,452	32,028	36,603	41,178	43,466	44,610
Compressor Pressure Ratio	1.545	1.865	2.290	2.825	3.500	3.865	4.080
Component Efficiency							
Compressor, 1-2*	.765	.776	.792	.797	.784	.769	.766
Compressor Turbine, 5-6	.76	.78	.80	.82	.84	.85	.86
Power Turbine, 6-8	.67	.67	.68	.685	.69	.695	.70
Burner	.984	.986	.989	.993	.997	.998	.998
Regenerator	.901	.898	.892	.885	.877	.874	.873
Aerodynamic Work, BTU/lb.							
Compressor	22.64	32.87	44.12	56.70	71.79	80.17	84.40
Compressor Turbine	24.57	35.05	46.59	59.53	75.02	83.68	88.01
Power Turbine	8.70	15.42	23.77	32.11	40.02	42.84	48.34
Parasitic Loss, HP							
Gas Generator	1.85	2.47	3.33	4.46	5.86	6.70	7.12
Power Turbine	1.80	2.40	3.50	4.90	5.40	5.90	5.90
Output Power, HP Net	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
Specific Fuel Consumption, lb/HP-hr	1.32	.90	.71	.62	.57	.57	.54
Gas Flow, lb/sec							
Station 1	.790	1.010	1.293	1.620	2.001	2.181	2.274
Station 3	.750	.957	1.224	1.532	1.889	2.059	2.146
Station 4	.772	.986	1.259	1.575	1.942	2.114	2.204
Station 5	.781	.997	1.275	1.596	1.970	2.146	2.238
Station 6	.769	.979	1.253	1.570	1.941	2.116	2.207
Station 8	.787	1.006	1.288	1.613	1.993	2.172	2.266
Station 9	.791	1.012	1.296	1.626	2.010	2.192	2.287
Pressures (Total), psia							
Station 1	14.67	14.65	14.63	14.58	14.53	14.49	14.48
Station 2 = 3	22.66	27.33	33.49	41.20	50.84	56.02	59.06
Station 4	22.48	27.12	33.27	40.97	50.60	55.78	58.81
Station 5	22.08	26.52	32.49	39.98	49.42	54.42	57.42
Station 6	17.07	18.69	20.88	23.38	25.98	27.01	28.42
Station 8	15.03	15.15	15.32	15.55	15.87	16.04	16.16
Station 9	14.74	14.77	14.82	14.90	15.03	15.10	15.15
Temperatures, °F							
Station 1	85.	85.	85.	85.	85.	85.	85.
Station 2	178.0	220.3	266.8	318.6	380.6	414.8	432.0
Station 4	1145.5	1148.4	1150.0	1152.1	1155.1	1158.0	1198.0
Station 5	1431.4	1494.6	1568.3	1647.7	1730.8	1772.3	1850.0
Station 6	1338.0	1362.9	1394.4	1425.6	1453.5	1463.7	1527.5
Station 8	1290.6	1291.1	1292.2	1293.4	1294.4	1294.6	1339.5
Station 9	343.3	383.2	429.5	480.6	540.1	571.2	594.9
Flow Leaks, lb/lb from station i to j							
i j = 20	.00204	.00214	.00218	.00224	.00230	.00235	.00238
25	.00689	.00636	.00585	.00539	.00507	.00498	.00496
34	.035	.035	.035	.035	.035	.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							
i j = 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





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