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CATALYTIC COMBUSTION FOR THE AUTOMOTIVE GAS TURBINE ENGINE

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REA

ABSTRACT

A study of fuel injectors to provide a premixed prevaporized fuel-air mixture and an evaluation of commercial catalysts were performed as part of a program leading to the demonstration of a low-emissions combustor for an automotive gas turbine engine. At an inlet temperature of 800 K, a pressure of 5×10^5 Pa and a velocity of 20 m/s a multiple-jet injector produced less than ± 10 percent variation in Jet-A fuel-air ratio and 100 percent vaporization with less than 0.5 percent pressure drop. Fifteen catalytic reactors were tested with propane fuel at an inlet temperature of 800 K, a pressure of 3×10^5 Pa and inlet velocities of 10 to 25 m/s. Seven of the reactors had less than 2 percent pressure drop while meeting emissions goals of 13.6 gCO/kg fuel and 1.64 gHC/kg fuel at the velocities and exit temperatures required for operation in an automotive gas turbine engine. NO_x emissions at all conditions were less than 0.5 ppm. All tests were performed with steady-state conditions.

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SUMMARY

A study of fuel-air premixing-prevaporizing systems and commercial catalysts was performed as part of a program leading to the demonstration of a low-emissions combustor for an automotive gas turbine engine. The goals included a fuel preparation system which would supply a fuel-air mixture which was uniform to within ± 10 percent of the mean fuel-air ratio, with 90 percent fuel vaporization and with no autoignition. The catalytic reactor was required to produce emissions which were low enough to meet the most stringent proposed U.S. automotive standards. The overall pressure drop for both systems was to be less than 3 percent, with 1 percent allowed in the fuel-air preparation system and the remainder in the catalytic reactor.

The engine conditions included a combustor inlet temperature range of 970 K at full speed to 1210 K at idle, a combustor exit temperature of 1310 K at all speeds, and a pressure range of 1.5×10^5 Pa at idle to 4.5×10^5 Pa at full speed. The maximum available inlet temperature in the test rig was 800 K; however, it was possible to duplicate the other operating parameters.

For the fuel-air preparation studies, a pressure of 5×10^5 Pa was considered the most likely to cause autoignition. Four fuel injector configurations were tested at this pressure, at a temperature of 800 K, and a reference velocity of 20 m/s with Jet-A fuel. Jet-A is a typical gas turbine fuel. A multiple-jet injector produced the required fuel-air distribution and vaporization with only 0.5 percent pressure drop. No autoignition occurred in a 35-cm length at these conditions; however, at the higher inlet temperatures of the regenerative gas turbine engine, autoignition will probably be the most serious problem to be solved.

For the catalyst evaluation, a pressure of 3×10^5 Pa was chosen to represent the regenerative gas turbine engine pressure range, and propane fuel was used to avoid autoignition and vaporization problems. Fifteen catalytic reactors were tested at an inlet temperature of 800 K and with reference (inlet) velocities of 10 to 25 m/s. Seven reactors met the steady-stage emissions goals of 13.6 g CO/kg fuel and 1.54 g HC/kg fuel with less than 2 percent pressure drop at the velocities and exit temperatures required for engine operation. Inlet temperatures of 500 to 625 K were required before the catalysts were sufficiently active to

combust the propane fuel.

INTRODUCTION

This paper describes studies which are intended to lead to the demonstration of a catalytic combustor for an automotive gas turbine engine.

The Lewis Research Center has established an EFDA-supported program with the goal of demonstrating an improved automotive gas turbine engine which operates with the combustor conditions shown in table 1 (ref. 1). With a turbine inlet temperature of 1310 K for all engine speeds, extensive use of ceramic components should not be required. The engine conditions of table 1 were defined by a computer analysis which predicted a combined EPA city-highway driving cycle fuel economy of 26.2 miles/gallon in a compact car (ref. 1). Also included in table 1 are reference velocities computed from the combustor inlet conditions for several combustor diameters. U. S. automotive emissions standards are defined for operation over a city driving cycle; for this cycle, the engine of reference 1 would have a fuel economy of about 22 miles/gallon. The most stringent standards proposed by the 1970 Clean-Air Act are 0.4 g NO_x/mile, 3.4 g CO/mile, and 0.41 g HC/mile. The Lewis program includes the evaluation of the catalytic combustor with the goal of achieving emissions which are half those of the most stringent proposed standards.

Catalytic combustion has the principal advantage over gas-phase combustion that lower combustion temperatures and therefore, lower NO_x emissions are possible. The importance of combustion temperature to NO_x emissions is illustrated by figure 1 which is based on the prediction of a well-stirred reactor computer program (ref. 2). The assumptions made in preparing figure 1 included a vehicle fuel consumption of 0.0775 kg/km (about 22 miles/gallon fuel economy), a combustor operating with an inlet temperature of 970 K at a pressure of 5×10^5 Pa and no nitrogen bound to the fuel. Thermal NO_x emissions are shown as a function of the maximum combustion temperature achieved before dilution to an exit temperature of 1310 K. It is clearly advantageous to burn at the lowest possible temperature to limit NO_x emissions. The lowest temperature at which combustion can take place is determined by the lean flammability limit, which for hydrocarbon fuels occurs between flame temperatures of 1600 K (ref. 3) and 1800 K (ref. 4). Thus, for good stability, peak flame temperature should probably be maintained near 1900 K. Figure 1 shows that at this temperature, steady-state NO_x emissions should be only half the 0.4 g/mi proposed as a standard; lean premixed combustors have in fact been demonstrated with emissions low enough to meet this standard (ref. 5).

A way to react fuel and air at even lower flame temperatures is to use catalysis to augment the reactions. Catalytic combustion at reaction temperatures of 1300 to 1400 K has been shown to give good com-

bustion efficiency and produce thermal NO_x emissions which were several orders of magnitude below those of conventional combustors (refs. 6 to 9).

A schematic view of one possible catalytic combustor configuration is shown in figure 2. Fuel is injected upstream of the reactor to vaporize and mix with the inlet air. Autoignition must be avoided in this premixing section because the flame would tend to seat on the fuel injector and burn with locally high temperatures; high NO_x emissions and possible component damage could result.

The catalyst bed, or reactor, may consist of several sections, each made of a different kind of catalyst; it is desirable to use a catalyst which is active at low temperatures for the inlet while subsequent sections need to be selected for good oxidation efficiency. All sections must be durable at the reactor operating conditions. Downstream of the catalytic reactor a thermal reaction zone may be provided to permit catalytically-initiated reactions to continue. Such reactions have been shown to contribute as much as 20 percent of the total reactor temperature rise for some conditions (ref. 9). Finally, dilution of the reacted products to achieve the turbine inlet conditions may be necessary. Some method of starting the combustor will also be required, although none is shown in figure 2.

Successful fuel-air premixing systems have been demonstrated in earlier studies (ref. 10) and the potential of catalysis applied to low-emissions combustion has been shown (ref. 6 to 9 and 11). However, the development of the technology required to demonstrate a complete catalytic combustor has yet to be achieved, and an evaluation of both fuel-air preparation and catalyst performance in terms of particular engine requirements has not been made. The Lewis catalytic combustion program includes both fuel-air preparation studies and catalyst testing. The emissions goals are to be achieved while operating at the conditions shown in table 1 and with a combustor pressure loss of no more than 3 percent. Arbitrarily, a goal of 90 percent fuel vaporization and a fuel-air ratio uniform to within ± 10 percent with no more than 1 percent pressure drop across the fuel preparation section was chosen.

Steady-state tests were performed in a 12-cm diameter combustion test rig at inlet temperatures of 450 to 800 K, pressures of 3 and 5×10^5 Pa, catalyst inlet velocities of 10, 15, 20, and 25 m/s and fuel-air ratios from 0.01 to 0.026. Jet-A, a typical gas-turbine fuel, was used in the fuel-air preparation studies, while vaporized propane fuel was used in the evaluation of catalysts to avoid the vaporization and auto-ignition problems associated with typical gas-turbine fuels.

FUEL PREPARATION STUDIES

Test Configuration

Fuel preparation systems were tested in the rig shown schematically in figure 3. The inlet airflow rate was measured with an ASME standard

square-edged orifice. The air was heated to temperatures as high as 900 K in a non-vititating preheater. The test section inlet duct was 10 cm in diameter with a 7.6 cm diameter insert to increase the velocity of the air. This section was followed by a straight-walled diffuser to increase the diameter to 12 cm. The Jet-A fuel flow was measured by two turbine flowmeters in series; the fuel was injected into the highest velocity region of the inlet duct (see fig. 3) to provide good atomization.

The fuel-air mixture was sampled 35.6 cm downstream of the fuel injector. Two probes were traversed along perpendicular diameters to obtain the spatial fuel distribution by iso-kinetic sampling at several positions. The sample from each probe was allowed to react in a catalyst bed maintained at 1030 K in an oven. The reaction products were then analyzed to determine the concentrations of CO, CO₂ and HC. These concentrations were used to compute the local fuel-air ratio at the sampling position. The degree of vaporization was also measured with these same probes using the spillover technique described in references 10 and 12.

Eighteen meters of electrically heated (410 to 450 K) 0.5 cm diameter stainless steel tubing connected the sampling probes with the exhaust gas analyzers. CO and CO₂ concentrations were measured with Beckman Model 315B nondispersive infrared analyzers and unburned hydrocarbon concentration was measured with a Beckman Model 402 flame ionization detector. Water vapor was removed with a Hankinson Series E refrigeration-type dryer before the sample was analyzed for CO or CO₂, and corrections were made to obtain the actual, wet-basis, concentration. The concentrations of unburned hydrocarbons and CO were negligible.

The temperature and static pressure were measured upstream of the fuel injector and at the sampling station as shown in figure 3. A single Chromel-Alumel thermocouple was used in both locations.

Downstream of the sampling station, hydrogen was injected into the fuel-air stream to produce a flammable mixture. The mixture was burned to avoid discharging fuel into the atmosphere. A water quench cooled the exhaust before it discharged through a back-pressure valve.

Test Injectors

Four fuel injectors were evaluated: a multiple-jet injector, a splash-groove injector, a simplex pressure atomizer, and a Sonicore nozzle. The multiple-jet and splash-groove injectors are air-blast atomizers; that is, they rely on the relative momentum between fuel and air for atomization. The simplex pressure atomizer uses fuel pressure for atomization. The Sonicore nozzle is a commercially-available injector which depends on a high-velocity external air stream for atomization.

The multiple-jet cross-stream injector used 28 orifices of 0.37 mm diameter injecting into zones of approximately equal duct cross-sectional areas (see fig. 4(a)). The splash-groove injector (fig. 4(b)) was developed by Ingebo (ref. 13). Fuel is injected through 21 passages to impinge upon conical surfaces of the nozzle. Fine droplets are formed which are rapidly atomized by the air flow. The simplex nozzle was a Monarch 70-degree angle spray nozzle with a flow rating of 0.013 m³/hour at 6.8×10^5 Pa differential pressure. The Sonicore nozzle (fig. 4(c)), Model 125MA, used an external air supply at 6.5×10^5 Pa pressure to provide a high-velocity airstream. The airstream impinges on the resonator cap, and fuel injected into the airstream forms a cone-shaped finely-atomized spray. The fuel-air ratio distribution provided by the splash-groove injector, the simplex nozzle, and the Sonicore nozzle were determined both with and without the concentric air swirler shown in figure 4(d). The 30-degree vane angle induced a swirl in the air flow to improve mixing between the fuel and the air.

Test Conditions

Tests were performed at inlet temperatures varying from 450 K to 800 K, a pressure of 5×10^5 Pa, fuel-air ratios of 0.01 and 0.025 and a reference velocity of 20 m/s. The reference velocity is computed from the inlet air temperature, pressure, and mass flow rate using the 12 cm diameter; thus, this velocity corresponds with that at the inlet to the catalytic reactor. The pressure and reference velocity are representative of those in table 7; however, the inlet temperature was not duplicated due to preheater limitations. The test pressure was chosen to be slightly higher than that in Table 7 in order to simulate the most severe condition for autoignition.

Results

The spatial fuel distribution, degree of vaporization and pressure drop were determined for each injector.

The fuel distribution along one diameter of the duct for the splash-groove injector is given in figure 5. The injector alone produced a very non-uniform profile; however, when the 30-degree-vane air swirler was mounted concentric with the injection, the fuel distribution became nearly uniform. Local values of the fuel-air ratio, calculated from the gas-analysis carbon balance, agreed within 10 percent of the average value determined from the metered fuel and air flows. Results obtained over the traverse perpendicular to that shown were essentially the same, and a nearly uniform profile was also obtained with the simplex and Sonicore nozzles when tested with the 30-degree-vane swirler. The multiple-jet injector gave a uniform profile without the necessity for air swirl. Tests conducted at inlet air temperatures of 500 and 600 K at both 0.01 and 0.025 fuel-air ratios suggested that these variables had no effect on the

uniformity of the mixture profile.

The effect of inlet air temperature on the degree of vaporization at the center of the duct is given in figure 6 for all four nozzles tested. None of the vaporization data were obtained with swirler-assisted nozzles. At any temperature, the best performance results were obtained with the multiple-jet and splash-groove injectors, although for the inlet temperatures of interest for this application, vaporization should approach 100 percent for all injector configurations.

The multiple-jet cross-stream injector, which used no air swirler, achieved the required fuel-air mixture uniformity and vaporization with only 0.5 percent pressure drop. The pressure loss for each of the other injectors was also about 0.5 percent when no air swirler was used, but was about 1 percent when the 30° air swirler was added.

CATALYST EVALUATION

Test Configuration

Catalysts were evaluated in the same 12 cm diameter combustion test facility used to test fuel-air preparation systems. Commercial grade propane fuel (91 percent pure by volume) was introduced as a vapor into the airstream through a single-orifice injector (ref. 9) 150 cm upstream of the test section. The resulting fuel-air ratio profile at the test section inlet was found to be uniform within ± 10 percent of the mean.

Inlet and exit pressures were measured at wall static taps located 9 cm upstream and 12 cm downstream of the test section, respectively. For the test conditions of these experiments, total pressures are only about .2 percent higher than the static so no total pressure measurements were made. The pressure drops across the test section was measured with a differential pressure transducer between the inlet and exit static location.

A description of the catalyst evaluation test section is given in figure 7. The inlet temperature to the test section was taken as the average reading of the thermocouples in the 8- thermocouple array labelled number 1 in figure 7. Individual catalyst elements were located between pairs of the thermocouple arrays labelled 4-8 as shown in figure 7. The housing containing the catalyst elements was uncooled to limit heat loss; a downstream section containing a gas sampling probe and an additional 12-thermocouple array was water cooled, however.

The gas-sampling probe was also water cooled and had five 1.5 mm-diameter sampling orifices located in the center of equal cross-sectional areas as shown in figure 7. The use of small orifices along with copious

water cooling was found in previous testing to be necessary to insure that the CO concentration in the probe and sampling line is frozen at the sampling-station value.

In addition to the gas analysis instruments described in the Fuel-Air Preparation Section, these experiments also used a Thermo-Electron Model 10A Chemiluminescent analyzer to measure total NO_x emissions.

Test Catalysts

Twelve catalysts from four manufacturers were evaluated either singly or in combination to form a total of fifteen reactors. A description of the reactors is given in table 2 and specifications of the individual catalyst elements appear in table 3. Typical catalyst elements are pictured in figure 8. The two Johnson Matthey catalysts were 7.6 cm long and used a corrugated metal substrate wound into a cylinder as shown in figure 8; the remaining elements were all 2.5 cm long and had ceramic substrates. All catalysts were either platinum, palladium, or a mixture of the two.

Tests Performed

Two series of tests were made on the reactors: (1) a determination of the catalyst activation temperature (i. e., the inlet temperature at which the catalyst became sufficiently active to react most of the fuel), and (2) measurement of reactor emissions and pressure drop.

The catalyst activation temperature was found by increasing the inlet-air temperature in increments of 25 K, starting at 500 K, with a pressure of 2×10^5 Pa and a catalyst inlet velocity of 10 m/s. This pressure and velocity correspond approximately to those existing when the engine is being cranked during start-up. At each temperature, propane was introduced at an equivalence ratio of 0.3 while the reactor thermocouples were monitored. For inlet temperatures below the catalyst activation temperature, a small temperature rise across the bed (of as much as 15-20 K) was generally observed; however, when the inlet temperature was increased to the catalyst activation temperature, the bed and exit temperatures began to increase steadily for a period of about 2 or 3 minutes until a temperature rise of several hundred degrees across the bed was achieved.

To evaluate catalyst performance, exhaust emissions and pressure drop were measured with an inlet propane-air mixture temperature of 800 K, a pressure of 3×10^5 Pa and a range of inlet velocities of 10 to 25 m/s. It was not possible to simulate the combustor inlet temperature shown in table 1; however, the pressure and velocities are fairly representative of those at all engine speed settings.

Results

The catalyst activation temperatures are given in table 2. The G12P and G24D reactors were not available for this part of the testing. Testing was discontinued at 625 K with the OCL6 and OCDS reactors when they produced only 15-20 K temperature rise. Activation tests performed on reactors which had only one kind of element showed that activation occurred at relatively low temperatures (500-550 K) with Pt catalysts (E11 and JM1 reactors) or when the catalyst was a mixture of equal amounts of Pt and Pd (G24 reactor). Higher temperatures were required for activation when the catalyst was either Pd (reactors E14, G17, and JM2) or a mixture with more Pd than Pt (reactors OCL6 and OCDS). Reactors E1, E2, E3, E4 and G1 had low-activation temperature elements at the front of the reactor and high-activation-temperature elements following. Table 2 shows that these combinations had the low activation temperatures characteristic of their leading elements. Because propane was used for these tests, the activation temperatures which were determined may be higher than would result if Diesel, gasoline, or Jet-A had been used.

When the reactor performance and emissions were determined at 800 K inlet temperature, NO_x emissions were found to be less than 0.5 ppm for all test conditions. Actual values were difficult to measure accurately because the lowest range of the analyzer was 2.5 ppm full scale, while it was necessary to use a 100 ppm calibration gas. NO_x emissions should be well below levels required to meet the 0.4 g NO_x /mile standard, at least for propane fuel which has no fuel-bound nitrogen. Only the conditions for which the HC and CO emissions met the standards remained to be found.

Each reactor can be characterized by a minimum operating exit temperature. This temperature is the adiabatic reaction temperature (established by the fuel-air ratio and inlet temperature) at which the reactor must be operated to achieve the CO and HC emissions goals. Reaction temperatures higher than this minimum result in lower emissions of CO and HC. Although the automotive exhaust emissions standards are established for operation over a specific cycle which includes start-up and transient operation, conditions not duplicated in these tests, reference emission indexes can be computed from the standards to provide guide in determining the minimum operating temperature. Assuming a vehicle fuel consumption of 0.0775 kg/km (about 22 miles/gallon fuel economy), 3.1 g CO/mile translates to 27.3 g CO/kg fuel and 0.41 g HC/mile becomes 3.29 g HC/kg fuel. The emissions goals for the program were half of these reference values, or 13.6 g CO/kg fuel and 1.64 g HC/kg fuel. A combustion efficiency of about 99.5 percent is required to achieve these goals.

Both pressure drop and minimum exit temperature are functions of the inlet velocity; therefore, the tradeoff between pressure drop and minimum exit temperature can be illustrated by plotting the re-

sults for each reactor as shown in figure 9. The pressure drop in figure 9 is that which was measured at a reaction temperature of 1400 K. The lines representing each inlet velocity were drawn arbitrarily through the results for the bulk of the best performing reactors tested. For inlet velocities of 10 and 15 m/s, the performance variation among all except the E11 and G24D reactors suggested only the effects of the tradeoff between pressure drops and minimum operating temperature. At higher velocities greater differences appeared. At 25 m/s the minimum exit temperature for reactors E11, E4, G1, OCL6 and OCDS was in excess of 1550 K and was not determined.

One of the functions of a plot like that of figure 9 is to help in determining if the reactor can be operated at conditions which are consistent with both the program goals and the engine requirements. For example, a combustor with no dilution air flow and a 16 cm diameter reactor has catalyst inlet velocities of 15-17 m/s (table 1). The reactor exit temperature is required to be 1310 K with no dilution flow. Figure 9 shows that none of the reactors tested have minimum exit temperatures as low as 1310 K at 15 m/s and with less than 2 percent pressure drop. However, if 25 percent of the combustion air is used to dilute the products from the reactor, the reactor exit temperature increases and the inlet velocity decreases to the values shown in table 4. From figure 9 it can be determined that seven of the reactors tested would meet the program goals for both pressure drop and emissions while operating at the reactor conditions of table 4.

The results for reactor E11 (which used a Pt catalyst) illustrates that a reactor which activates at a low temperature (table 2) will not necessarily have good performance at higher temperatures (see fig. 9). In contrast, reactor E14 (which used a Pd catalyst) required a relatively high temperature to become active but performed well at higher temperatures. The best attributes of both reactors were obtained when elements from each reactor were combined to form reactor E1.

DISCUSSION

This study has successfully demonstrated both fuel-air preparation systems and catalytic reactors capable of operating within the pressure drop and emissions restraints of an improved automotive gas turbine engine. However, the results obtained cannot be applied directly to engine operation because of a number of considerations.

The inlet temperature at which the tests were conducted was limited by preheat capabilities to 800 K while the engine combustor inlet temperature varies from 970 to 1210 K. This higher temperature can be expected to affect the performance of the combustor in at

least two ways. The first effect is the increased tendency for autoignition of the fuel upstream of the catalyst. Although the fuel-air ratios of interest are well below the lean flammability limit and thus should not be susceptible to autoignition, locally richer mixtures exist near the fuel injector before complete mixing has taken place. Thus, mixing will have to be achieved in less than the ignition delay time. The autoignition delay time decreases exponentially with increasing inlet temperature and linearly with pressure (ref. 14). For the combustor conditions of table 1, the maximum inlet temperature occurs at the minimum pressure; as a result, the predicted range of ignition delay times is only 10.8 ms at 100 percent speed to 12.2 ms at idle. To keep the residence time with a flammable mixture below these values at the velocities shown in table 4, it is necessary that the mixing distance be less than 13.8 cm.

The second effect of operating at higher inlet temperatures should be a reduction in the reactor pressure loss. The pressure drop through the reactor is primarily due to frictional losses which are proportional to the product of mass flowrate and average reactor velocity. As the inlet temperature increases, the mass flowrate decreases with constant reference velocity, and the average reactor velocity decreases because the reactor temperature rise is less for the same exit temperature. This effect of inlet air temperature may be offset by the losses incurred by the mixing of dilution air with products downstream of the reactor, however.

None of the catalytic reactors used in this study were subjected to durability tests. Each reactor was tested for 2 to 3 hours, and only the GL2P reactor showed loss of activity after that time. Final selection of catalysts will have to be made on the basis of both performance and durability, however.

An evaluation of catalyst durability involving catalytic combustion at 1500 K exit temperature for a period of 1000 hours has been performed by Engelhard Industries under contract to Lewis (ref. 15). Parametric tests were performed with propane fuel at the start of durability testing and with both propane and Diesel fuel at the completion of 1000 hours of operation. The tests showed that after a period of high-temperature operation combustion of Diesel fuel may remain very efficient at conditions for which propane combustion efficiency has decreased drastically. The combustor exit temperatures shown in table 1 are lower than those tested by Engelhard, so loss of activity may not be as severe as that suggested by the Engelhard results. Tests of selected catalysts at the required reactor conditions for the desired lifetime will be needed before final catalyst selection can be made.

The Engelhard results suggest that Diesel fuel may be easier to combust than propane fuel. An additional advantage of Diesel fuel over propane is that its catalyst activation temperature is lower (ref. 16). On the other hand, NO_x emissions with Diesel fuel

may be higher as a result of fuel-bound nitrogen (ref. 17). Clearly, catalyst evaluation will have to be performed with typical gas turbine fuels like Jet-A or Diesel.

No transient tests were made in these studies. The ability of catalytic reactors to respond to changes in inlet conditions will be crucial to the success of the concept. It is likely that catalytic substrate materials which have high thermal conductivities and low thermal capacities such as metal or silicone carbide will be required to insure both rapid warm-up and quick response. Tests will also have to be made to determine if engine transients will induce conditions favorable to autoignition in the fuel-air mixing section.

The emissions produced during start-up and transient operation may be considerably higher than those produced at the steady-state conditions described in this paper. Because the emissions standards are based on operation over a cycle involving both start-up and transient conditions, a full evaluation of catalytic combustor feasibility cannot be made until tests can be performed at these conditions.

The practical operation of a catalytic combustor in an engine will require a method of heating the catalyst to its activation temperature for cold-engine-start conditions. Possible approaches include the use of a preburner or upstream torch with a spark igniter, an electrical heater to raise the catalyst substrate temperature directly, a downstream burner with a spark igniter, and possibly even the use of hydrogen fuel for starting. Whatever system is chosen would operate only until the combustor inlet temperature reached the catalyst activation temperature.

While this study has shown that catalytic combustion may be feasible for steady-state operation, final evaluation of the concept will depend on engine demonstration. For a full engine demonstration to take place considerably more technology development will be required, and engine demonstration with start-up and transient operation may well reveal additional problem areas requiring attention.

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TABLE 1. - AUTOMOTIVE REGENERATIVE GAS TURBINE ENGINE:

HYPOTHETICAL COMBUSTOR INLET CONDITIONS FOR AN IMPROVED ENGINE (Ref. 1)

Speed, % of full	Airflow, kg/s	Fuel flow, 10^{-3} kg/s	Pressure, 10^5 Pa	Combustor inlet temperature, K	Combustor exit temperature, K	Reference velocity (m/s) for possible combustor diameter		
						14 cm	16 cm	18 cm
50 (idle)	0.127	0.455	1.43	1210	1310	19.8	15.1	12.0
60	.204	1.135	1.91	1140	1310	22.4	17.2	13.6
70	.254	1.705	2.28	1100	1310	22.6	17.3	13.6
80	.336	2.77	2.92	1045	1310	22.1	16.9	13.4
90	.409	3.84	3.56	1005	1310	21.2	16.3	12.8
100	.517	5.41	4.50	970	1310	20.5	15.7	12.4

Table 2. - CATALYTIC REACTOR DESCRIPTION

Note: The JMPt and JMPd elements are 7.6 cm long. All other elements are 2.5 cm long. A complete description of each element is given in Table 3.

Reactor designation	Catalyst elements				Catalyst activation temperature, K
	1	2	3	4	
E11	E1128	E1128	E1128	E1128	500
E14	E1412	E1412	E1412	E1412	580
E1	E1128	E1412	E1412	E1412	540
E2	E1128	← JMPd →			530
E3	E1128	G512:3.2	G524:290	G524:290	530
E4	E1128	OC3.6:N	OC1.8:1.6	OC1.8:1.6	550
G12P	G512:P	G512:P	G512:P	G512:P	NA
G17	G517:290	G517:290	G517:290	G517:290	600
G24	G524:290	G524:290	G524:290	G524:290	530
G24D	G524D:200	G524D:200	G524D:200	G524D:200	NA
G1	G512:3.2	G524:200	G524:200	-----	540
JM1	-----	← JMPt →			525
JM2	-----	← JMPd →			620
OC16	OC1.8:1.6	OC1.8:1.6	OC1.8:1.6	OC1.8:1.6	>625
OCDS	OC3.6:S	OC3.6:S	OC3.6:S	OC3.6:S	>625

N.A.: Not available for these tests.

TABLE 3. - DESCRIPTION OF CATALYST ELEMENTS

All elements are 12 cm diameter

Element designation	Catalyst manufacturer	Manufacturers designation	Catalyst	Catalyst loading, kg/m ³	Substrate manufacturer	Substrate composition	Cell density, cells/cm ²	Cell shape	Element length, cm
E1128	Engelhard	EVD 1128	Pt	*	Corning	Cordierite	35	Sine Wave	2.5
E1412	Engelhard	EVD 1412	Pd	*	Corning	Cordierite	35	Sine Wave	2.5
G512:3.2	W. R. Grace	Davex 512A	Pt/Pd:1/1	2.0	General Refractories	Mullite	10	Circular	2.5
G512:P	W. R. Grace	Davex 512A	Pt/Pd:1/1	5.3	Pure Carbon	Silicon Carbide	30	Sine Wave	2.5
G517:290	W. R. Grace	Davex 517	Pd	2.5	W. R. Grace	Cordierite	45	Square	2.5
G524:200	W. R. Grace	Davex 524A	Pt/Pd:1/1	3.5	W. R. Grace	Cordierite	31	Rectangular	2.5
G524:290	W. R. Grace	Davex 524A	Pt/Pd:1/1	3.5	W. R. Grace	Cordierite	45	Square	2.5
G524D:200	W. R. Grace	Davex 524A	Pt/Pd:1/1	6.5	W. R. Grace	Cordierite	31	Rectangular	2.5
JMPt	Johnson Matthey	-----	Pt	5.3	Johnson Matthey	Metal	62	Sine Wave	7.6
JMPd	Johnson Matthey	-----	Pd	5.3	Johnson Matthey	Metal	62	Sine Wave	7.6
OC1.8:1.6	Oxy-Catalyst	-----	Pt/Pd:1/2	1.8	General Refractories	Mullite	34	Circular	2.5
OC3.6:S	Oxy-Catalyst	-----	Pt/Pd:1/2	3.6	Norton	Silicon Carbide	11	Circular	2.5

*Proprietary

TABLE 4. - CATALYTIC REACTOR INLET CONDITIONS FOR IMPROVED ENGINE

DOWNSTREAM DILUTION, 25% OF TOTAL COMBUSTOR AIRFLOW

REACTOR DIAMETER, 16 cm

COMBUSTOR EXIT TEMPERATURE, 1310 K

Speed % full	Total air flow, kg/s	Total fuel flow, 10^{-3} kg/s	Reactor air flow, kg/s	Reactor f/a	Reactor inlet temperature, K	Reactor exit temperature, K	Pressure, 10^5 Pa	Reactor inlet velocity, m/s
50 (idle)	0.127	0.455	0.0952	0.00478	1210	1350	1.43	11.4
60	.204	1.135	.153	.00742	1140	1375	1.91	12.9
70	.254	1.705	.190	.00895	1100	1385	2.28	12.9
80	.336	2.77	.252	.0110	1045	1400	2.92	12.7
90	.409	3.84	.307	.0125	1005	1405	3.56	12.2
100	.517	5.41	.388	.0140	970	1425	4.50	11.8

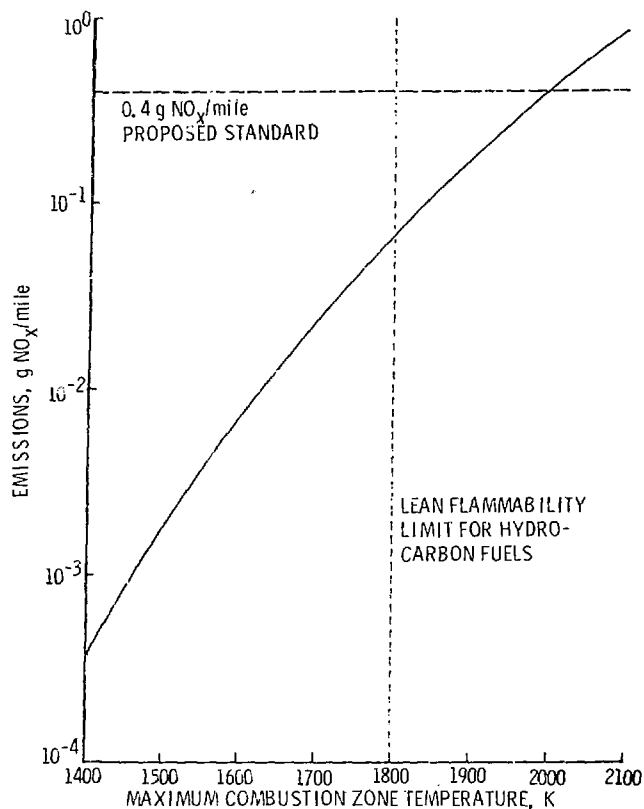


Figure 1. - Predicted thermal NO_x emissions from a well-stirred reactor (ref. 2). Combustor inlet temperature, 970 K; combustor exit temperature, 1310 K; combustor pressure, 5×10^5 Pa; vehicle fuel consumption, 0.0775 kg/km.

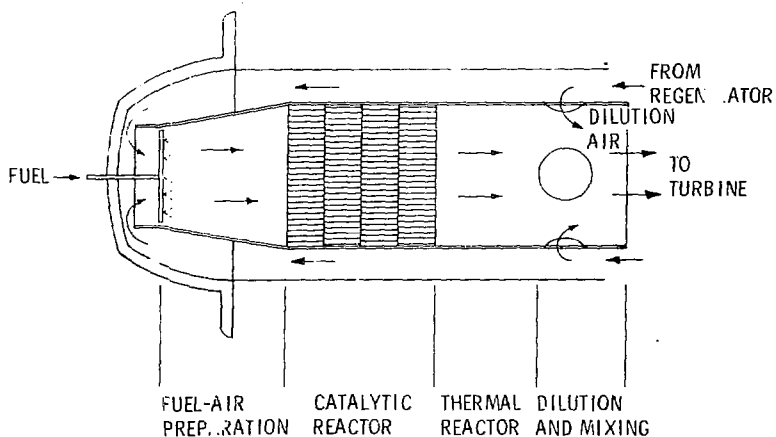


Figure 2. - Schematic representation of a catalytic reactor.

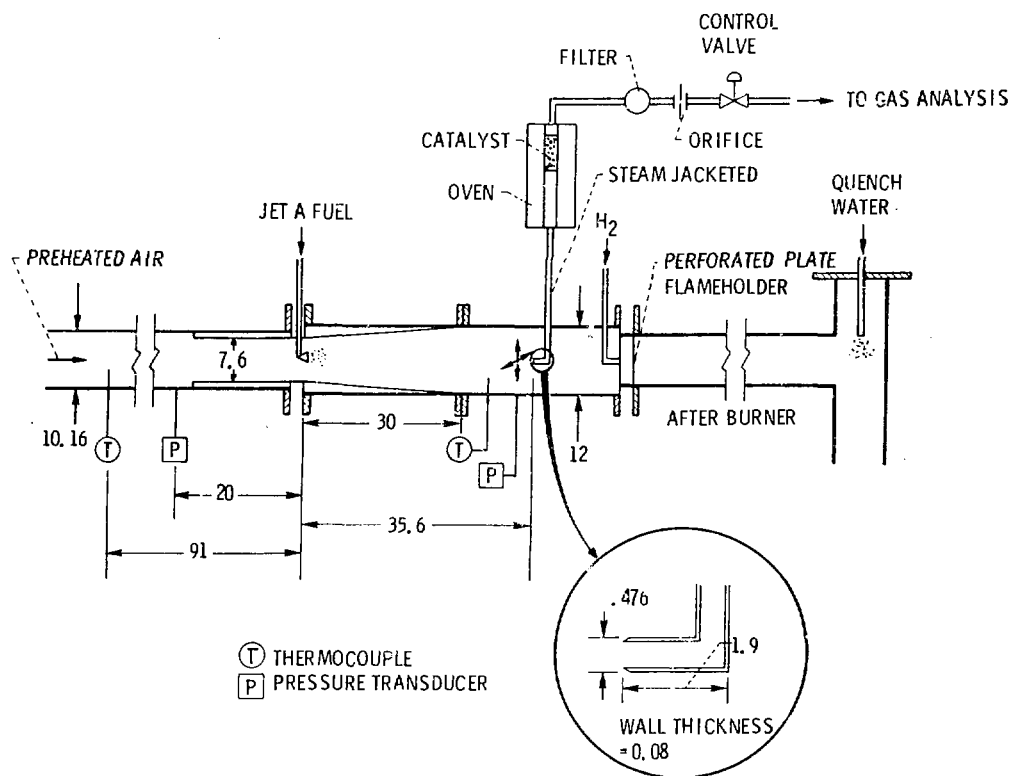
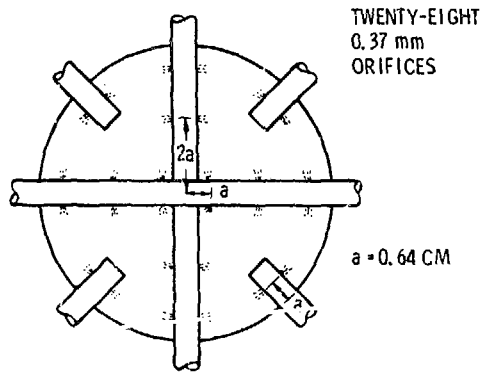
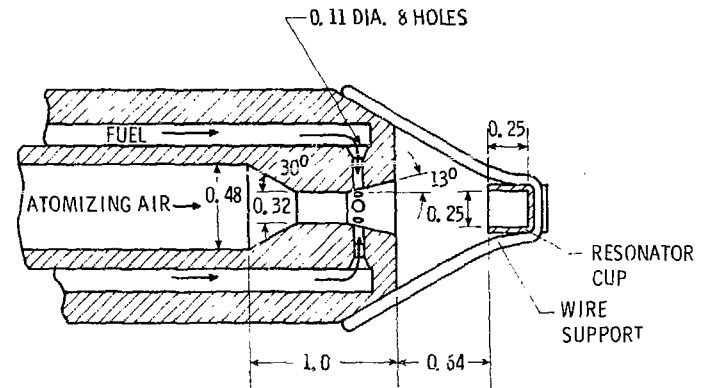


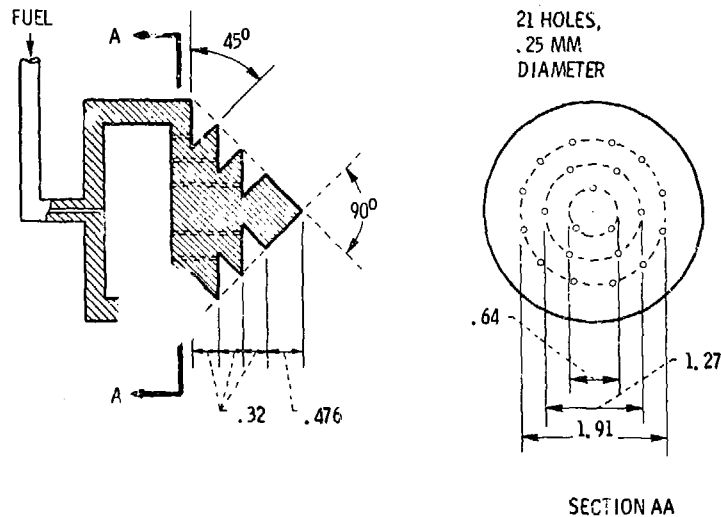
Figure 3. - Rig schematic. (Dimensions in cm.)



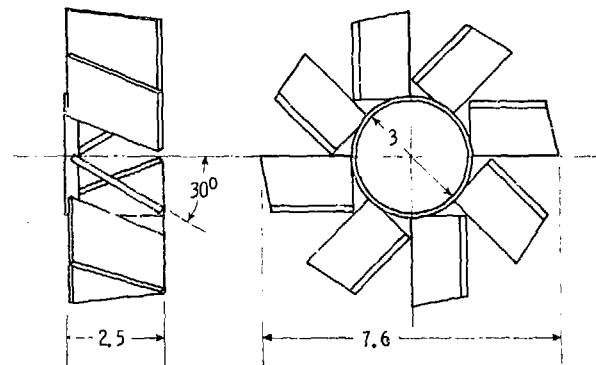
(a) Multiple-jet cross-stream injector.



(c) Sonicore injector.



(b) Splash-groove injector.



(d) Swirler used optionally with splash-groove injector, sonicore injector, and simplex pressure-atomizing nozzle.

Figure 4. - Concluded.

Figure 4. - Test injectors. (Dimensions in cm.)

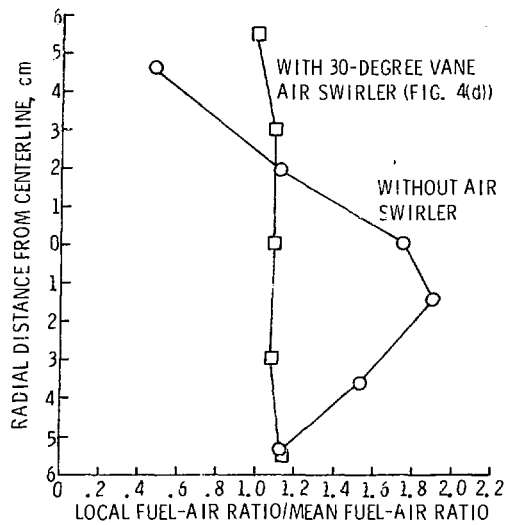


Figure 5. - Spatial fuel distribution. Inlet temperature, 800 K; pressure, 5×10^5 Pa; reference velocity, 20 m/s; mean (metered) fuel-air ratio, 0.01; splash-groove fuel injector.

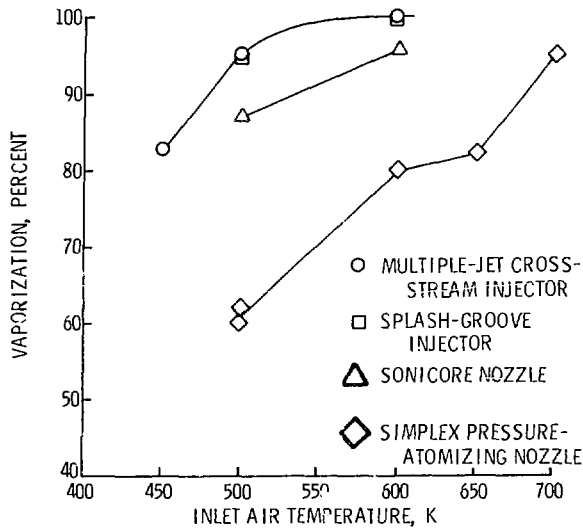


Figure 6. - Effect of inlet air temperature on degree of vaporization. Pressure, 5×10^5 Pa; reference velocity, 20 m/s; fuel-air ratio, 0.010.

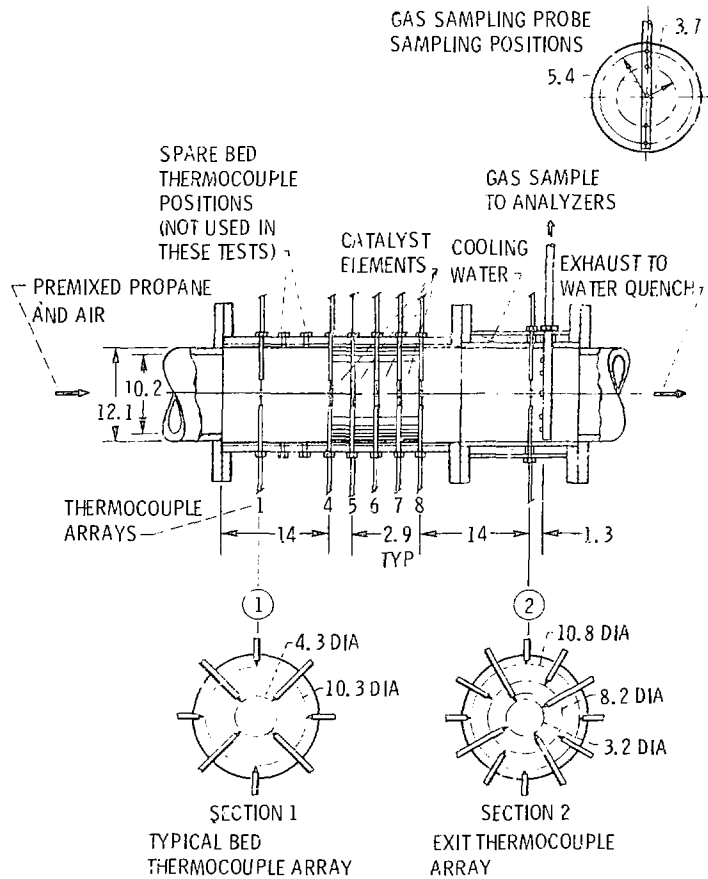


Figure 7. - Catalyst evaluation test section. (Dimensions in cm.)

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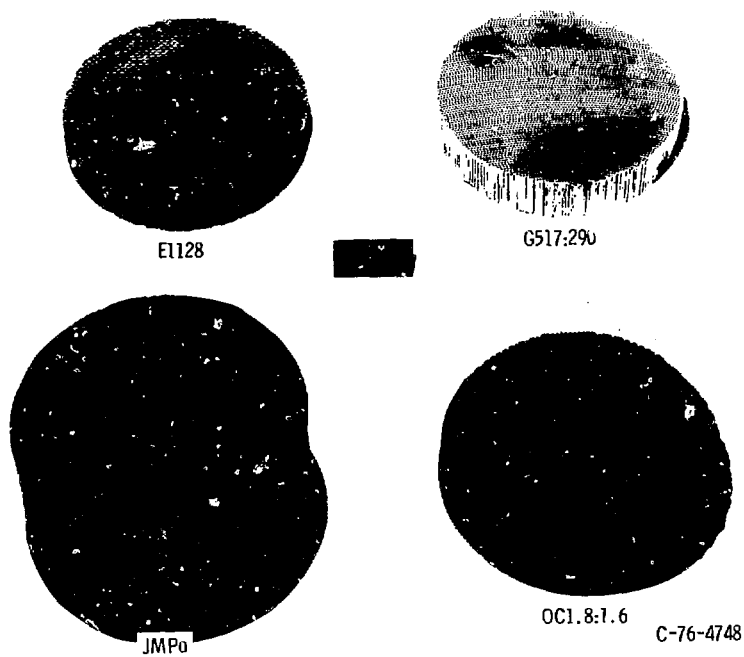


Figure 8. - Typical catalyst elements.

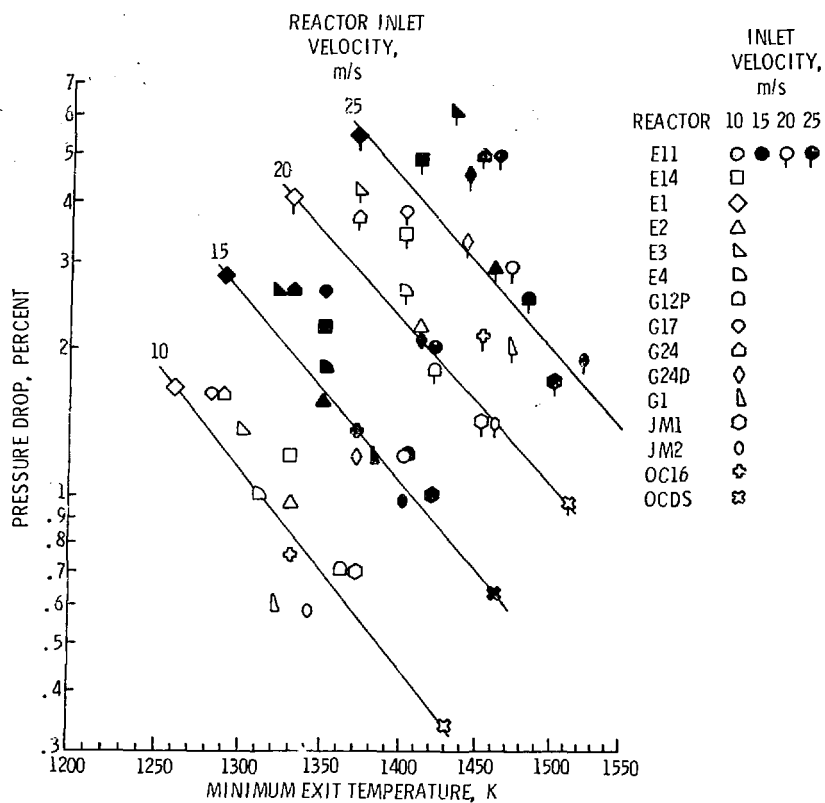


Figure 9. - Tradeoff between pressure drop at 1400 K and minimum required exit temperature. Inlet mixture temperature, 800 K; pressure, 3×10^5 Pa; propane fuel.

THE ERDA/CHRYSLER UPGRADED AUTOMOTIVE GAS TURBINE ENGINE -
EMISSION CONTROL SYSTEM

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ABSTRACT

This paper describes the compact, low-cost, fixed-geometry combustor and the related engine/fuel control system parameters which should enable the ERDA/Chrysler Upgraded Engine to meet emission goals of .26 HC, 2.1 CO, and .25 NO_x (gr/km).

Low emissions are achieved by combustion control in a system which does not sacrifice economy or driveability. Torch ignition of a very lean premixed charge is utilized at steady speeds with rich operation during compressor stage acceleration transients. Unleaded gasoline has been the principal development fuel, but emission control has been demonstrated with several potential alternate fuels. This combustion system should be applicable to advanced, more efficient, higher cycle temperature gas turbine engines which will be candidates for future production.

The Upgraded Engine design has a pressure ratio of 4.1:1, 1050°C turbine inlet temperature, 0.69 kg/s airflow, and power of 92 kW with use of variable inlet guide vanes and inlet water injection.

INTRODUCTION

Chrysler began a combustor development program in January, 1971, aimed specifically at reducing NOx emissions of its sixth-generation, 112kW (150HP), regenerative gas turbine engine from approximately 3.1 gr/km (5 gr/mi) to less than .25 gr/km (0.4 gr/mi). Previous work, without regard to NOx control, had shown that HC and CO could be held to near-zero levels.

A combustor producing less than 1.8 gr/km of NOx had been developed by November, 1972, when Chrysler was awarded Contract 68-01-0459 through the AAPS Division of EPA. The sixth-generation engine with this improved combustor was released as the "Baseline Engine" for performance of the contract, while proprietary combustor development efforts continued. Periodic demonstrations [1] [2] of improved technology have been made during the contract, which in January, 1975, was transferred to ERDA and designated E(11-1)-2749.

The contract includes the design, development, and demonstration of a new engine (Upgraded Engine) using improved technology to minimize emissions, fuel consumption, and cost. As the engine was being designed, Baseline Vehicle demonstrations were conducted using a fixed geometry combustor which showed promise of meeting the Upgraded Vehicle emission goals without performance or economy penalties, and with only nominal penalties in size and cost.

Emission control philosophy at Chrysler has always given priority to prevention of formation rather than aftertreatment clean-up. Due consideration is also required for production feasibility, durability, size, cost, and performance effects.

*NOTE: [] numbers in brackets designate references at end of paper.

Lean, premixed combustion is the only known means to prevent formation of undesirable pollutants. The regenerative, automotive gas turbine must, however, operate with a wide range of fuel/air ratios. The most often suggested means of achieving this wide range of low-emissions operation has been the use of variable-geometry combustors. But such combustors don't meet the requirements of production feasibility, durability, size, or cost.

Many other combustor concepts have been dependent on relatively high (5-7%) pressure loss across the combustor. Because of its strong influence on fuel economy, Chrysler has established a 3% limit on combustor pressure loss, with 2% as a goal.

Since all combustor development at Chrysler is done on engines, the first requirement of any concept is the ability to start a cold engine. And since cold start emissions are included in the Federal Test Procedure, it must be done cleanly. This is an area often overlooked when combustor concepts are being suggested.

The background of the combustion system now released for the Upgraded Engine is summarized in Reference [3] and the engine design is described in Reference [4]. This paper provides additional combustor and control detail. However, because most of the development has taken place on the Baseline Engine, much of the data reported herein is identified with that engine.

COMBUSTOR ASSEMBLY

The combustor utilizes a single, fixed-geometry, premixer tangentially entering a fixed reaction chamber (burner tube). A torch ignitor enters through the dome of the reaction chamber at a right angle to the premixer discharge. A conventional dilution zone cools the combustion products to the desired turbine inlet temperature. See Figure 1. (Reference [5] provides a discussion of combustion processes and presents alternate design concepts).

Premixer/Injector

The premixer is a short section of carefully placed swirl, counterswirl, and wall-sweep openings with a fuel injector at the upstream end. The injector includes eight small fuel tubes in line with the first row of premixer air slots. These tubes merely place the fuel in the proper location in the premixer; doing nothing in themselves to atomize the fuel. The maximum fuel pressure drop across the injector is only 70 kPa (10 psi). The incoming airflow atomizes the fuel off the tips of the injector tubes. During cold starting, the fuel starts burning in a droplet-diffusion flame within the premixer. As burner inlet temperature rises (regenerative engine), the fuel droplets begin vaporizing in the premixer while engine fuel requirements diminish. Careful development of premixer and injector detail has resulted in a burner which converts itself from droplet-diffusion flame to premixed reaction within a few seconds after first reaching idle speed.

During normal running, the premixer provides a uniform charge into the reaction chamber and is generally immune to flashback. This premixer accomplishes these tasks using approximately 45% of the total Upgraded Engine airflow. Since no mechanical flame-holders are used, it will tolerate a flashback condition, should it occur, without damage.

Torch

The torch provides a continuous jet of flame to ignite the premixed main charge in the reaction chamber. It consists of a small chamber with its outlet tube at right angles to the premixer flow. A pressure atomizing, 140 kPa (20 psid), fuel nozzle sprays into the chamber where compressor discharge air enters tangentially, forming a strong swirl action. About 2% of total airflow is used in the torch. Combustion is initiated by a single automotive spark plug firing continuously at 160-180 cps from a transistor-triggered induction coil.

Torch combustion is a droplet-diffusion process at an equivalence ratio near unity. Emission of NO_x is low due both to the non-regenerated inlet airflow and the low mass flow (high dilution ratio). CO and HC are consumed in the main reaction process except during "flame-out" conditions when they are minimized by the overall dilution ratio.

Reaction Chamber

The reaction chamber, or burner tube, is convection cooled and provides sufficient recirculation and residence time to complete the combustion process. By introducing the premixed charge tangentially, strong vortex recirculation is established, per-

mitting stable operation at premixed fuel air ratios as lean as .010 F/A. Because the cooler chamber walls tend to quench the reaction, leaving locally high CO levels, a secondary wall constriction is provided to turn the surface flow into the main vortex where reaction will be completed. The conventional dilution zone admits the remaining airflow which was not introduced through either the premixer or torch.

The chamber is located and retained by the premixer and torch tubes which in turn are rigidly mounted to the insulated burner cover. The outlet end is a slip fit into the vortex chamber, permitting assembly misalignment and thermal expansion. It is fabricated from 1.5 mm thick RA 333 material which will provide satisfactory service at 1150°C.

CONTROL SYSTEM

The Upgraded Engine utilizes an integrated electronic control system [6] for fuel, variable power turbine nozzles, inlet guide vanes, and water injection. The fuel system schematic is shown on Figure 2 and the fuel schedule on Figure 3.

Fuel Schematic

The electrically driven pump at the fuel tank provides a constant 80 kg/h (180 lb/hr) flow at up to 550 kPa (80 psi). The electrohydraulic fuel metering valve has a built-in pressure-drop regulator to maintain a constant differential pressure of 140 kPa (20 psid) across the metering orifice. This, in effect, also maintains a constant differential pressure across the torch nozzle.

The entire pump flow passes the torch nozzle and all except torch flow passes through the metering valve body, where excess is returned to the fuel tank through the regulator. This provides cooling to these areas where vapor formation would affect the fuel metering, essential when using volatile fuels, such as gasoline, in a low pressure system.

The built-in fuel flowmeter aids in bench calibrating each system and is a sensor providing feed-back to the control unit for closed-loop scheduling of start-up and acceleration fuel flow. It does not indicate engine fuel consumption because torch flow is not included and injector bleed flow is included.

The bleed orifice provided at the fuel injector returns fuel to the tank. This permits excess fuel to flow to the injector, provide cooling, and thus minimize prevaporization within the

injector. It also avoids the need for a fuel metering valve capable of accurately metering low flows. In addition, in combination with the normally open solenoid, it provides a convenient means to control transient acceleration NOx emissions. During decelerations engine air backflushes fuel from the injector tips through this orifice, along with metering valve leakage, to help minimize emissions of unburned hydrocarbons.

The check valve in parallel with the fuel metering valve combines with the 3-way solenoid valve to purge fuel from the injector and lines at shut-down. Engine air pressure during compressor coast-down enters the injector tubes and flushes the fuel back to the tank.

Fuel Schedule

Refer to Figure 3. The control is programmed for a start and acceleration schedule which represents the maximum allowable fueling level. During these modes, the metering valve opens with increasing speed (airflow) such that maximum acceleration is maintained without exceeding turbine stage temperature limits. Note that the metering valve does not open until a significant (5-10%) speed signal has been obtained. The injector bleed solenoid is closed for maximum fueling.

During steady-speed, road load operation the metering valve acts as a governor, adjusting fuel flow to correct deviations from demanded speed. However, the injector bleed is open, so the metering valve is flowing a much higher fuel flow as indicated by the dashed line near the acceleration schedule line.

During moderate driving with gradual changes of accelerator position, the bleed solenoid remains open, thereby greatly reducing the maximum fuel actually available to the burner. Thus, combustion remains lean and NO_x formation remains low. This is sufficient for most accelerations in the Federal Emission Test Cycle. But some accelerations require a more rapid engine response, and the bleed solenoid is closed when there is a significant demand for acceleration (demanded speed more than five percentage points above actual speed). This causes the burner to receive its richest mixture and the gas generator to accelerate at its maximum rate. By avoiding governor modulation of fuel flow in the range of combustion mixtures from .030 to .060 F/A, NO_x formation is greatly reduced. Although high NO_x concentrations are still encountered during maximum fueling, the mass contribution is low because of very short time durations.

Torch fuel is not regulated by the integrated control. It is proportional to the square root of the pressure drop across the nozzle. This pressure drop consists of the regulated drop across the metering valve plus flow losses in the system. Thus, torch fuel flow rises only slightly through the speed range. Since torch air flow is approximately proportional to engine speed, the resulting mixture is quite rich for good cold light-off and is lean at maximum speed. It runs slightly rich at idle to provide the necessary flame intensity for good ignition of the pre-mixed charge and stability at low combustion levels.

For engine decelerations, the metering valve closes, shutting off premixer fuel. The torch continues to burn, ready to relight

the reactor.

Variable Power Turbine Nozzles

The power turbine nozzles are modulated by an electrohydraulic actuator to maintain a constant turbine outlet (regenerator inlet) temperature as engine speed changes. However, when significant gas generator accelerations are demanded, these nozzles are opened a few degrees more. This causes the compressor turbine to absorb a greater proportion of the pressure ratio available and increases the engine airflow, which in turn permits a higher fuel rate to be used for the same maximum transient turbine inlet temperature. Consequently, the gas generator accelerates much more quickly. Mass emissions of NO_x are reduced because the increased airflow rate is more than offset by the decreased time at maximum fuel rate.

Variable Inlet Guide Vanes

The influence of variable compressor inlet guide vanes on engine emissions has not yet been fully determined. They can be used either to throttle or augment airflow and may be useful in controlling engine cycle temperature at part throttle or during engine braking.

Water Injection

Water injection is used only at wide open throttle (less than 1% of normal operation) to augment maximum power. On the Baseline Engine, with its droplet-diffusion combustor, NO_x was reduced with increasing water injection while no effect was evident on HC or CO.

RESULTS AND DISCUSSION

Sixth-Generation Engine - Baseline Vehicle

The combustor/control concept has been developed on the sixth-generation Baseline Engine, having much higher airflow and a lower cycle temperature than the Upgraded Engine. Steady-speed emissions from idle to 80% Ngg are very low. Above that speed, NO_x emissions rise rather sharply as mixture strength exceeds .030 F/A. However, driving the Federal Emission Cycle at 1814 kg (4000 lb) inertia load requires speeds in excess of 80% for only a few seconds of the total cycle. This system has resulted in 1975 "3-bag weighted" cold cycle Baseline Vehicle emissions of 1.59 HC, 5.31 CO, and 0.28 NO_x (gm/km). While almost meeting the NO_x goal established for the Upgraded Vehicle, the unburned hydrocarbons and carbon monoxide levels significantly exceed their goals.

Furthermore, very low emissions are obtained during operation at other normal driving conditions. Typical steady, road-load operation at all speeds from 50 to 100 km/hr (30 to 60 mph) produces lower exhaust hydrocarbons than the intake air contains, carbon monoxide less than 0.7 gm/km, and NO_x less than 0.1 gm/km. Over the Federal Highway Fuel Economy Cycle this system produced Baseline Vehicle emissions of virtually zero HC, 0.8 CO, and 0.08 NO_x (gm/km). Obviously, the higher values on the emission test cycle are a result of the very transient operation over that specified driving pattern. Approximately 150 gas generator acceleration-deceleration cycles are encountered in each test.

Gas generator accelerations produce NO_x spikes due to momentary operation at near stoichiometric mixtures. As described above, these spikes are minimized by the fuel and power turbine nozzle control systems. The cycle results show satisfactory achievement of NO_x control.

Almost half of the carbon monoxide is generated during engine braking periods. With the power turbine nozzle vanes reversed, the power turbine wheel is putting work (heat) into the engine. This heat is recovered by way of the regenerators resulting in much less fuel needed to maintain the gas generator at idle speed. Thus, the burner is operating much leaner (cooler) than at curb idle, with the subsequent higher CO. (Engine braking is stronger than typical gasoline engines). Much of the remaining CO is produced at curb idle conditions which total 30% of the emission cycle time. Some CO is also generated at the flame-out and re-light points for each significant compressor deceleration.

Most of the unburned hydrocarbons accumulate from fuel passed just after a flame-out and just before a re-light. Each gas generator deceleration thus produces two spikes. While maintaining a minimum fuel schedule to keep the burner lit during decelerations has been shown to eliminate these spikes, it entails unacceptable penalties in fuel consumption and driveability. Limited test cell efforts have shown potential for clean flame-outs and re-lights by careful attention to torch details and mixture strength. Some additional hydrocarbons are also passed during the engine braking periods due to marginal combustion stability at the very lean mixture involved.

Fuels

This concept has been developed on today's fuel . . . unleaded gasoline. But to be an acceptable emission control system, it must be compatible with future fuels. Tests with diesel, broad cut, methanol, isopropyl alcohol, and ethyl alcohol as well as various mixtures of these fuels have all shown emission results similar to gasoline, although torch mixture adjustment is required to compensate for major differences in a fuel's stoichiometric ratio. See Table 1 for some sample emissions. These results suggest general compatibility with most fuels.

Evaluation of an unrefined liquid fuel obtained from pyrolysis of coal resulted in high NO_x levels and excessive fouling of the torch chamber. This fuel had high levels of chemically bound nitrogen, residue, and ash. Further processing would be necessary to make this an acceptable gas turbine fuel.

Upgraded Engine

No transient emission testing has been conducted to date on the Upgraded Engine due to emphasis on power plant development. However, excellent emissions have been achieved at steady-state speeds (Table 2).

This broader range of low-emission operation is a result of the higher cycle temperature. As cycle temperature increases, a greater portion of the airflow can be used in the premixer before reaching excessive CO levels at idle. See Figure 4. This results in leaner operation at all speeds and, for this engine, combustion temperature is just reaching the level of rapid NO_x

formation at wide open throttle (not augmented). Idle CO level is basically a function of reaction temperature and residence time, modified by wall cooling and mixing intensity. Since burner inlet air is about 85°C hotter in the Upgraded Engine, a leaner mixture must be used for the same reaction temperature. Residence time is increased somewhat by maintaining identical reaction tube volume for the Upgraded Engine which has 40% lower airflow than the Baseline Engine. Furthermore, the hotter burner inlet air also means less wall quenching of the reaction mixture. And finally, for the same CO concentration, mass emissions of CO are still 40% lower due to the engine airflow. These conditions combine to permit a cooler reaction zone at idle than used on the Baseline Engine.

It is expected, then, that the Upgraded Vehicles which are 230 kg lighter than the Baseline Vehicles will be able to meet the CO and NO_x emission goals since they should require only 60% as much fuel and air (nearly twice the fuel economy) as the Baseline vehicles and emission concentrations will be similar.

Mass emissions of unburned hydrocarbons will also be reduced due to the higher cycle temperature and lower airflow, but some additional refinement of the torch and control system may also be necessary to meet the HC goal. Since this is solely a transient problem, its extent won't be known until vehicle tests have been conducted.

CONCLUSIONS

The regenerative, automotive gas turbine will be able to meet stringent emission standards for a 1585 kg weight class vehicle by utilizing a fixed-geometry combustor with a choice of several fuels. This will be achieved by combustion control rather than exhaust aftertreatment, without sacrificing fuel economy or driveability, and with only a small penalty in cost and weight. There will be no incentive for consumer modification or adjustment which would defeat this emission control. It will be effective for all normal driving situations including highway cruise.

Realization of this potential in production passenger cars still depends on satisfactory achievement of competitive fuel economy and overall cost. Only then can definite plans be made for the conversion of facilities to produce gas turbines. Thus, a great development effort remains before the gas turbine can be the automotive engine of the future.

/daa

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TABLE 1

ALTERNATE FUELS

Emission effects of several fuels with advanced, premixed combustor on the Baseline Engine.

<u>Fuel</u>	<u>Idle-704°C P.T. Outlet Temp.</u>			<u>Acceleration*</u>
	<u>HC₆ (Net)</u> <u>ppm</u>	<u>CO</u> <u>ppm</u>	<u>NOx</u> <u>ppm</u>	<u>NOx</u> <u>ppm</u>
Unleaded Gasoline	0	97	3.8	18
No. 1 Diesel	0	50	4.6	32
Ethyl Alcohol	0	53	1.7	11
Isopropyl Alcohol	0	64	3.0	14
Methyl Alcohol	1.5	330	0.8	11
Coal Derived	0.4	180	32.0	70

* Peak observed NOx during gas generator acceleration from idle-704°C to 70% speed.

TABLE 2

STEADY-STATE EMISSIONS-UPGRADED ENGINE

Unleaded Gasoline Fuel

750°C Power Turbine Outlet Temperature

<u>Speed</u>	<u>Net* Concentration-ppm</u>		
	<u>HC₆</u>	<u>CO</u>	<u>NO_x</u>
50% (Idle)	0	190	3.6
60%	0	16	2.9
70%	0	0	2.4
80%	0	0	2.2
90%	0	0	3.6
95%	0	0	4.8
100%	0	0	8.3

* Net = Exhaust - Inlet (Exhaust concentrations less than inlet are reported as zero).

LIST OF FIGURES

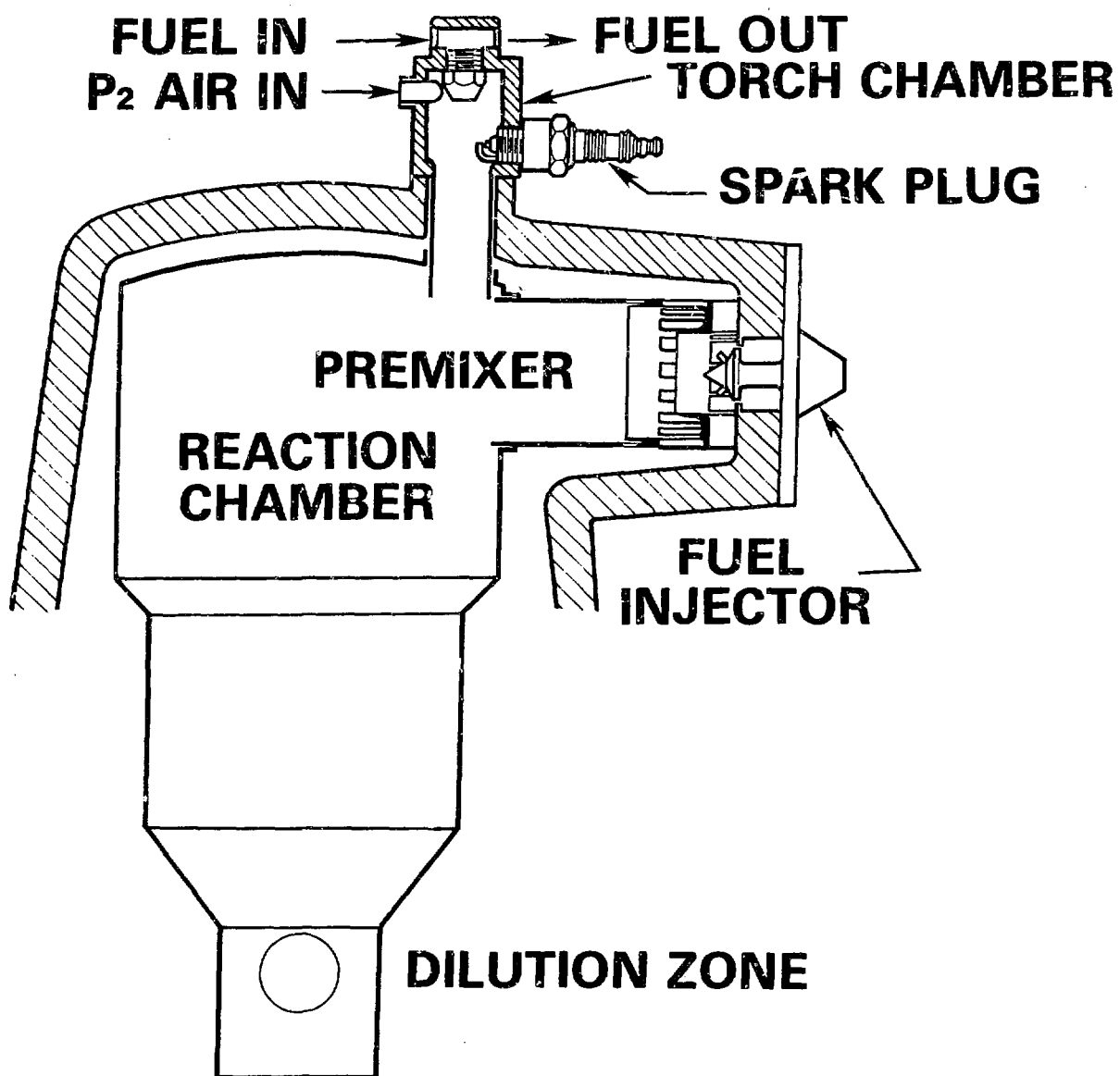
Nogle, T.D.

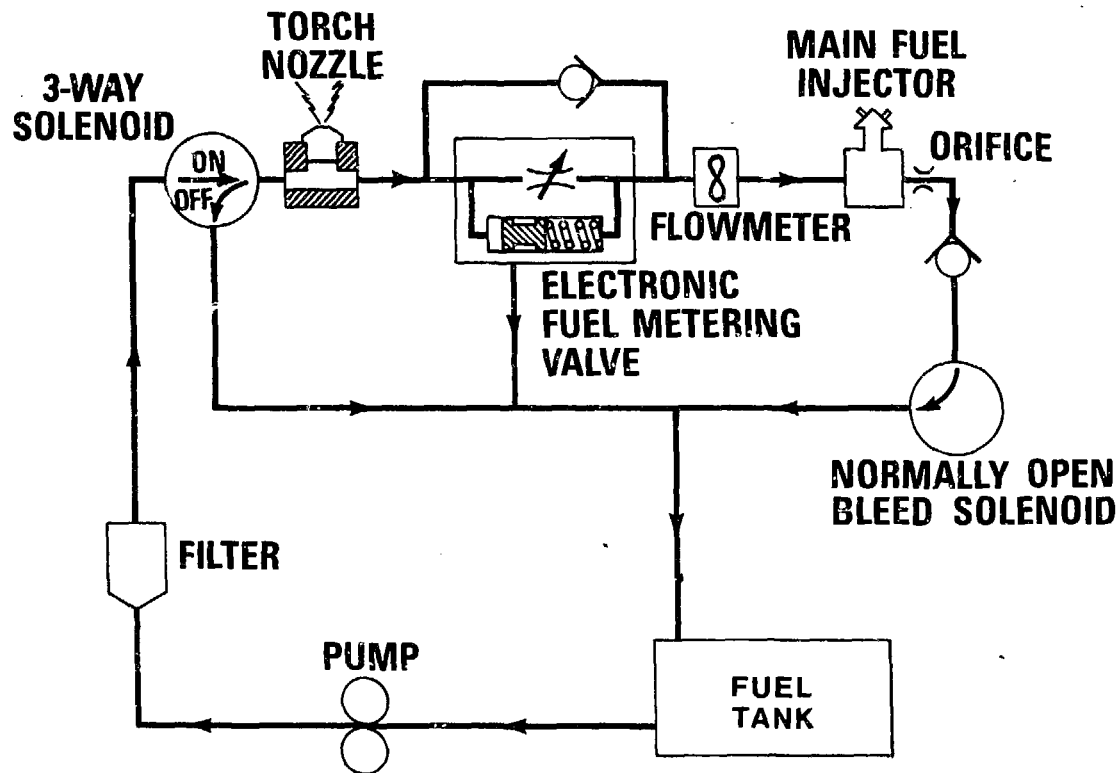
Figure 1: COMBUSTOR ASSEMBLY, ERDA/Chrysler Upgraded gas turbine engine.

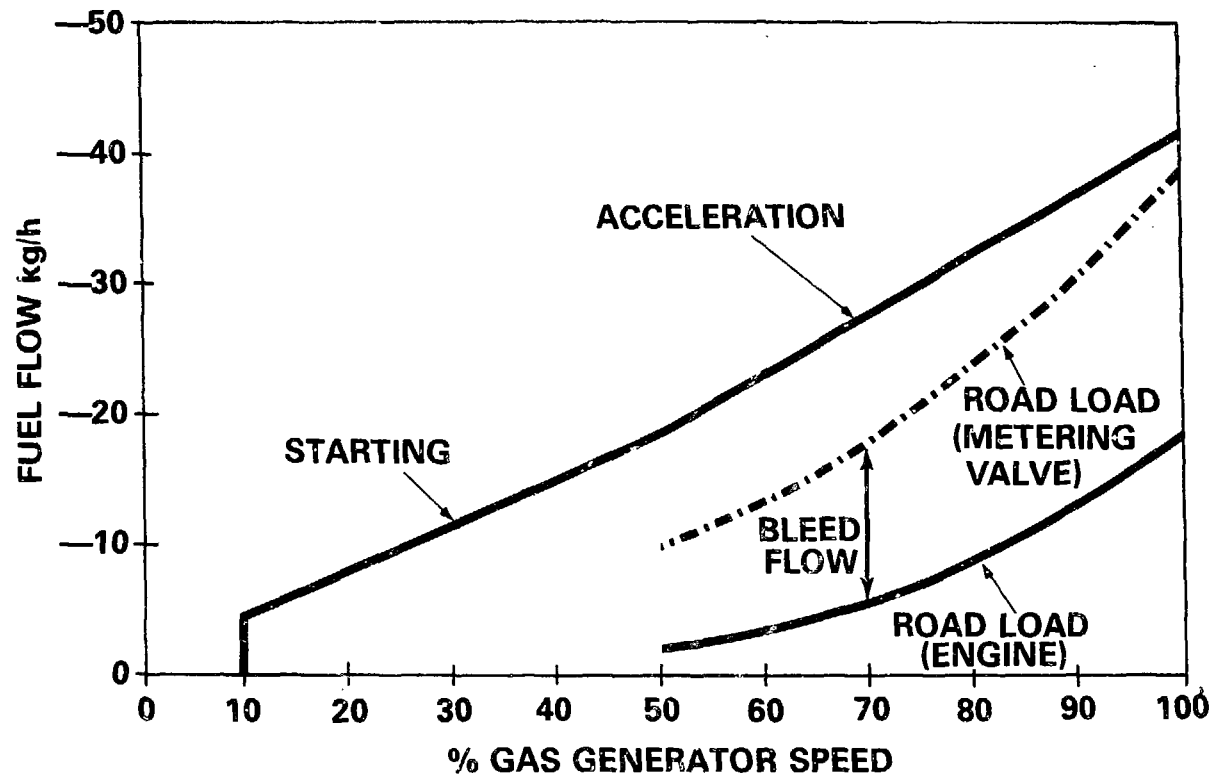
Figure 2: FUEL SYSTEM SCHEMATIC, ERDA/Chrysler Upgraded gas turbine engine.

Figure 3: FUEL SCHEDULE, ERDA/Chrysler Upgraded gas turbine engine.

Figure 4: REACTION ZONE COMPARISON, Baseline reaction temperatures and residence time versus Upgraded.







REACTION ZONE COMPARISON

BASELINE ENGINE

UPGRADED ENGINE

3520 cm³

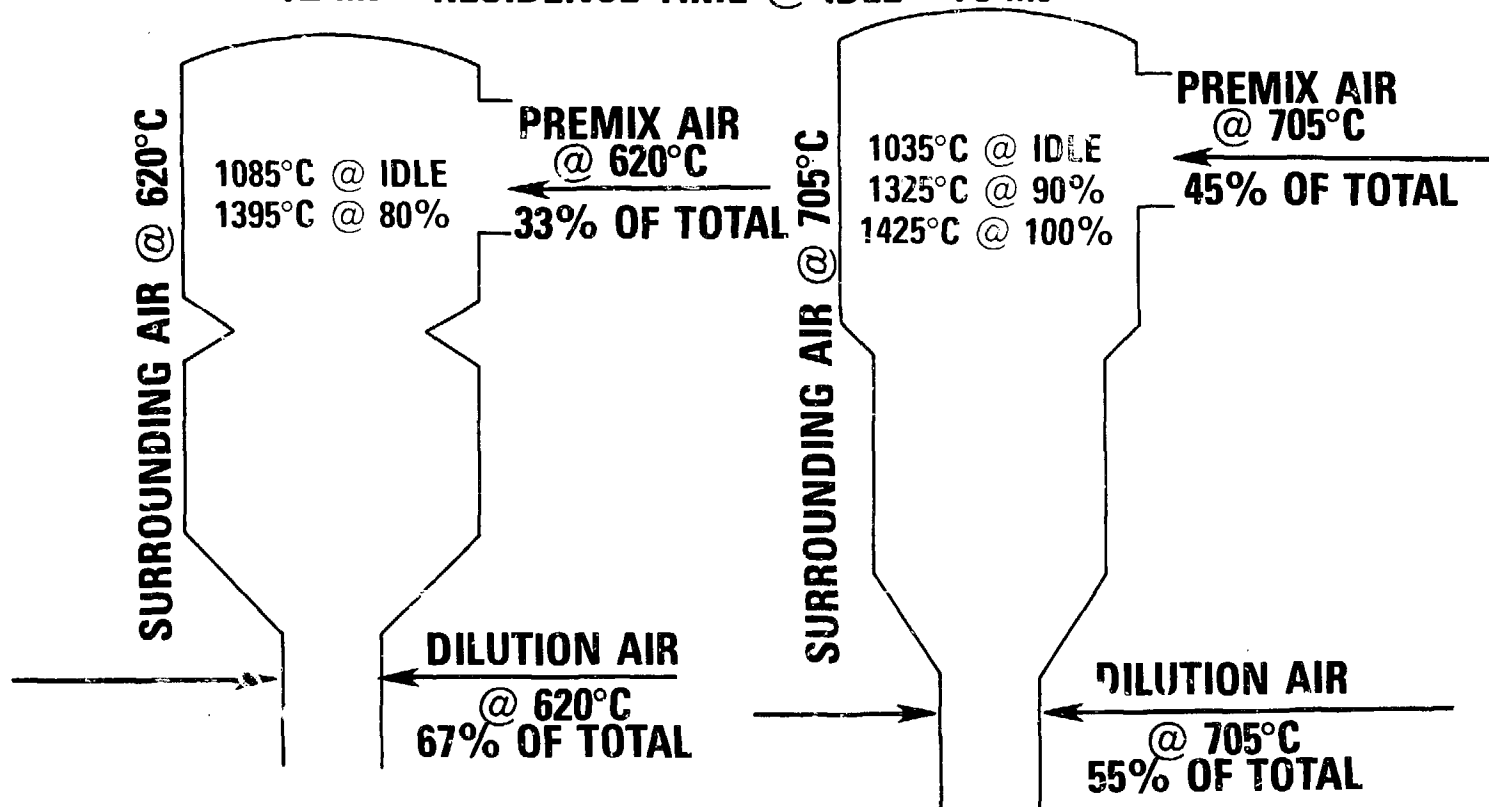
REACTION VOLUME

3520 cm³

12 ms

RESIDENCE TIME @ IDLE

16 ms



THE AUTOMOTIVE GAS TURBINE - STATE OF DEVELOPMENT AND PROSPECTS

P. Walzer

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Wolfsburg

Federal Republic of Germany

Fourth International Symposium on Automotive Propulsion
Systems, April 17 - 22, 1977, Washington D.C.

ABSTRACT

At Volkswagen a 55 kW and a 100 kW automotive gas turbine have been tested. Efforts are continuing on the development of low emission combustion concepts and on turbine components made from high temperature ceramics. Based on the state of technology and possible future improvements, it is predicted that advantages in the fuel economy for heavier cars will be attained, that the Muskie standards can be achieved, and that a wide variety of fuels can be used. It appears that higher production costs will have to be accepted.

1. INTRODUCTION

Future power plants for automobiles must combine high fuel economy, low pollutant emissions, multifuel capability and reasonable production costs. At Volkswagenwerk AG continuing efforts are being concentrated on further improving the conventional automotive engines in these directions. At the same time the possibility that other unconventional power plants could better solve the requirements is being investigated. In the following

the state of development of automotive gas turbines is described. Based upon accomplished results and anticipating improvements which should be attainable in further development steps, the advantages and disadvantages of gas turbines compared to the conventional automotive engines are discussed.

2. STATE OF DEVELOPMENT

A 55 kW and a 100 kW gas turbine have been tested. The prototypes of these engines have been developed in cooperation with WILLIAMS RESEARCH CORP., Michigan, USA. Both gas turbines are two shaft engines with regenerative air preheating. Table 1 gives the most important design data. The 55 kW gas turbine represents an earlier state of technology (1). This engine has a maximum turbine inlet temperature of 1184 K and a pressure ratio of 3.7. The engine operates without variable geometry. At full load a specific fuel consumption of 388 g/kWh is attained. The fuel consumption rises to 660 g/kWh at 30 % load. In the 100 kW engine the maximum turbine inlet temperature is raised to 1283 K and the pressure ratio to 4.5. This engine has variable inlet guide vanes at the power turbine. At full load the specific fuel consumption is 310 g/kWh the fuel consumption rises to 375 g/kWh at 30 % load.

Figure 1 shows the 55 kW gas turbine. The gasifier shaft and the power turbine shaft are arranged concentrically; both shafts drive into a common front mounted gear box. Power absorption by the compressor is utilized for braking by means of connecting both shafts with an overrunning clutch. Figure 2 shows the installation in the rear engine compartment of a VW-Microbus. The standard three-speed automatic transmission VW 1600 is used. The air inlet ducts of the standard air cooled Otto engine have been connected in the gas turbine installation directly to the compressor inlet. The exhaust gases leave the turbine through two short gas ducts. Figure 3 shows the 100 kW gas turbine on the test rig. The engine is equipped with an electronic control, which provides that within the tolerable temperature limits at all steady loads the engine runs with the lowest possible fuel consumption, that in instationary operation load changes are attained in the shortest possible time, and that during coasting braking is available. Figure 4 shows the installation of this gas turbine in a passenger car. Again a three speed automatic transmission is used. Air inlet ducts are provided at both sides near the top of the fenders. The exhaust gases are ducted along the bottom of the car to the rear end. Power turbine exit gases are used in the heater. The car has power steering and power brakes.

Figure 5 shows this experimental passenger car. It was converted for this purpose from a Ro 80, which is a Wankel engine powered car manufactured by AUDI NSU. The vehicle test weight is 1700 kg. The acceleration time from zero to 100 km/h (62.5 mph) is 14 s, a distance of 130 m is covered in the first 10 s. The response is fast enough to be able to follow the US City Cycle driving pattern within the prescribed speed tolerances. In the US Combined Cycle a fuel economy of 14 mpg has been attained. From a distance of 7.5 m a noise level of 57 dB(A) at idling and 66 dB(A) at maximum gasifier speed has been measured. The car can be driven with gasoline, Diesel, JP4, and methanol fuel.

For the purpose of investigating the low emission capability of the gas turbine combustion process, a 3 year research program was carried out, which was partially sponsored by the German Ministry for Research and Technology (2). Figure 6 shows some of the combustion systems developed during this program. On the left side a conventional diffusion flame type combustion chamber is shown. Using such a combustion system the 100 kW gas turbine vehicle has attained 3.4 g/mile CO, 0.4 g/mile HC and 2 g/mile NO_x . In order to attain lower NO_x levels, combustion systems with prevaporizing and premixing are necessary. Depending on the NO_x goals more or less complicated chamber designs and reaction zone controls are necessary. In the middle a premix chamber is shown which reduces the NO_x emission to

1 g/mile. On the right side a premix chamber with two stage combustion is to be seen. In this concept the first stage operates during low loads and deceleration while the second stage is added during higher loads and acceleration. In both reaction zones the fuel air mixture burns near the lean flame limit. With such a two stage combustion system the Muskie standards were attained on the combustor test rig in a 7 point simulation of the US City Cycle. As a next step the control has to be adjusted before this combustion system can be tested in the vehicle.

Besides developing more efficient components, raising the process temperatures would also aid in improving gas turbines. A substantial temperature increase, however, can be realized only if components from new high temperature materials can successfully be developed. In Figure 7 different turbine components made from ceramics are shown. The materials are silicon nitride and silicon carbide, fabrication techniques include hot pressing, injection molding, and slip casting. The parts have been developed together with the ceramic industry in a second research program, which is also partially sponsored by the German Ministry of Research and Technology (3). So far stationary components such as flame tubes and nozzle rings have been tested up to the gas temperatures and temperature cycles of metallic gas turbines. Prototype allceramic rotors with hot pressed Si_3N_4 hubs and reaction sintered Si_3N_4 rims

have survived circumferential speeds up to 450 m/s in cold spinning and gas temperature shocks of 500 K/s. Rotors with metallic hubs and simplified hot pressed Si_3N_4 blades have endured speeds up to 420 m/s in cold spinning. So far this development work is suffering from a broad scatter and an unsatisfactory reproducibility of the test results. It appears however that by continuing development efforts at least the stationary components can be developed successfully. In the case of the rotors such a prediction still has a high degree of uncertainty since the question of economic fabrication techniques is not yet answered.

3. FUTURE PROSPECTS

In the following the prospects of gas turbines as power plants for passenger cars will be discussed. Criteria are seen in the fuel economy, the pollutant emissions, the multifuel capability, and the production costs. The predictions are based on the state of technology already established and on improvements which seem to be attainable in future development work.

FUEL ECONOMY

An analog simulation technique which includes models for the gas turbine, the transmission, and the vehicle was used in investigating the fuel economy potential (4). The results are

given in Figure 8. The upper diagram shows the engine power which has to be installed in order to give the vehicle an acceptable acceleration performance. The lower diagram shows the fuel economy in the US Combined Cycle. The fuel economy shown for conventional engines corresponds to a large extent to EPA measurements. The fuel economy which can probably be attained with gas turbines is shown as a grey band. Gas turbines using today's technology can attain fuel economies near the lower border of this range. High temperature gas turbines using ceramic components should attain a fuel economy near the upper border. In comparing the fuel economy of the different engine types considerable fuel economy advantages for gas turbines in heavier cars are obvious.

As already mentioned the described 1700 kg experimental gas turbine car has attained a fuel economy of 14 mpg. It is felt that within the given temperature limits the fuel economy range in the figure can be met with this engine by further component improvements and by introducing variable geometry at the compressor.

POLLUTANT EMISSIONS

Figure 9 shows CO-NO_x correlations which have been measured with diffusion flame type combustors and with premix combustion concepts. The HC emission values are not shown as this pollutant normally does not exceed the tolerated limit if the CO limit is met. The tolerated operation ranges for 1.5 g/mile NO_x and

0.4 g/mile NO_x are shown as dark bands. According to the analog simulation described above future metallic gas turbines attaining fuel economies of 15 mpg in the US City Cycle would operate in the indicated ranges. Therefore it is most likely that future gas turbines can meet the Muskie standards. However, two stage premix combustion systems together with more complex fuel controls will be needed.

These predictions apply to gas turbines working with today's turbine inlet temperatures. For gas turbines with higher process temperatures the engine simulation and the extrapolation of combustion chamber test results indicates that the Muskie standards are attainable up to the desired ceramic turbine inlet temperatures of 1623 K. Beyond this temperatures the tolerable combustion chamber operation range has to be restricted.

MULTIFUEL CAPABILITY

Gas turbines are especially suited for multifuel operation for numerous reasons.

- The continuous combustion process at constant pressure allows fuels with a broad range of octane ratings, vapor pressures, distillation curves ignition characteristics, and heating values.

- The high amount of excess air allows the burning of fuels with widely differing stoichiometric values.
- The reaction volume is formed as a free boundary which adjusts itself to the reaction volume requirements of the different fuels.

Accordingly the gas turbines developed so far could be operated with a broad variety of fuels without influencing the performance of the engines. Thus, based on the refinery structure of today, 60 - 70 % of the crude oil can be used in gas turbines.

PRODUCTION COSTS

In addition to fuel economy, pollutant emissions, and multi-fuel capability the production costs will be decisive for the serial introduction of an unconventional power plant. To get reliable production cost predictions, however, is extremely difficult. On the one side the mass fabrication technology for these new engines is not yet developed; on the other side the designs of the engines so far have not been optimized for low production costs.

These difficulties and uncertainties should be kept in mind when studying the cost projections in Figure 10. The upper diagram again gives the relation between necessary engine power

and vehicle weight. The lower diagram gives production cost relations between the different engine types. The production costs of a 37 kW Otto engine are taken as a basis. The cost estimates of the gas turbines are based on a detailed cost break down of the research engines developed so far. At low vehicle weights a mass production of 400 engines a day, at higher vehicle weights 200 engines per day have been assumed. The results indicate that the production costs of two shaft gas turbines with today's process temperatures would be at the upper border of the gas turbine band. Ceramic gas turbines will have smaller air flow which will reduce the volume of the expensive heat exchanger and probably mean less control complexity. For those engines therefore a reduction in production costs can be expected. The lower border of the cost band should be attainable with single shaft ceramic gas turbines. In a car application the latter engines however will need an inexpensive and highly efficient continuously variable transmission.

4. CONCLUSIONS

At Volkswagen two research gas turbines have been developed and tested in cars. Besides improving these engines by continuing component development, work is being concentrated on low emission combustion systems and on ceramic turbine components. Based on the state of technology and possible future improve-

ments, it is predicted that advantages in the fuel economy of the heavier cars will be attained, that the Muskie standards can be achieved, and that a wide variety of fuels can be used. It appears that higher production costs will have to be accepted.

5. ACKNOWLEDGEMENTS

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- (3) Walzer, P., Köhler, M., Rottenkolber, P.: Hochtemperaturkeramik für Automobilgasturbinen, MTZ 37 (1976) 11
- (4) Walzer, P., Klarhoefer Ch., Köhler, M.: Fuel Economy Study of Future Metallic and Ceramic Automotive Gas Turbines, ASME paper 76-GT-104

CAPTIONS

- Fig. 1 Automotive Gas Turbine VW-GT 70
- Fig. 2 VW-GT 70 in Microbus
- Fig. 3 Automotive Gas Turbine VW-GT 150
- Fig. 4 VW-GT 150 in Ro 80
- Fig. 5 Experimental Gas Turbine Car Ro 80
- Fig. 6 Combustion Chamber Concepts
- Fig. 7 Ceramic Gas Turbine Parts
- Fig. 8 Fuel Economy Potential of Gas Turbines
- Fig. 9 CO_2 - NO_x Ranges for Different Combustion Systems
- Fig. 10 Production Cost Projection for Gas Turbines

Schriftgröße

Linienbreite

siehe auch VW 010 04

Für Dia: 5 x 5 cm
Maskenausschnitt: 23 x 35 mm

7 mm für Bildtitel
5 mm für Text und Bild-Nr.
5 mm für Indizes und Exponenten für Bildtitel
3,5 mm für Indizes und Exponenten für Text

3,7 mm für hervorzuhebende Teile
0,5 mm für Hauptteile
0,35 mm für Nebenteile
0,25 mm für Mittel-, Maß- und Schraffurlinien

	VW - GT 70	VW - GT 150
max. power	55 kW (75 HP)	100 kW (136 HP)
spec. fuel consumption		
- at max. power	388 g/kWh (0.63 lbs/HPh)	310 g/kWh (0.50 lbs/HPh)
- at 30 % power	660 g/kWh (1.07 lbs/HPh)	375 g/kWh (0.61 lbs/HPh)
air flow	0.59 kg/s (1.30 lbs/s)	0.84 kg/s (1.85 lbs/s)
pressure ratio	3.7	4.5
turbine inlet temperature	1184 K (1674 F)	1283 K (1850 F)
size	0.156 m ³ (5.5 ft ³)	0.275 m ³ (9.7 ft ³)
weight	170 kg (375 lbs)	210 kg (463 lbs)

ambient 288 K, 1 bar; 2 % pressure loss at air inlet and at gas exhaust;
engine auxiliaries



Engine Design Data

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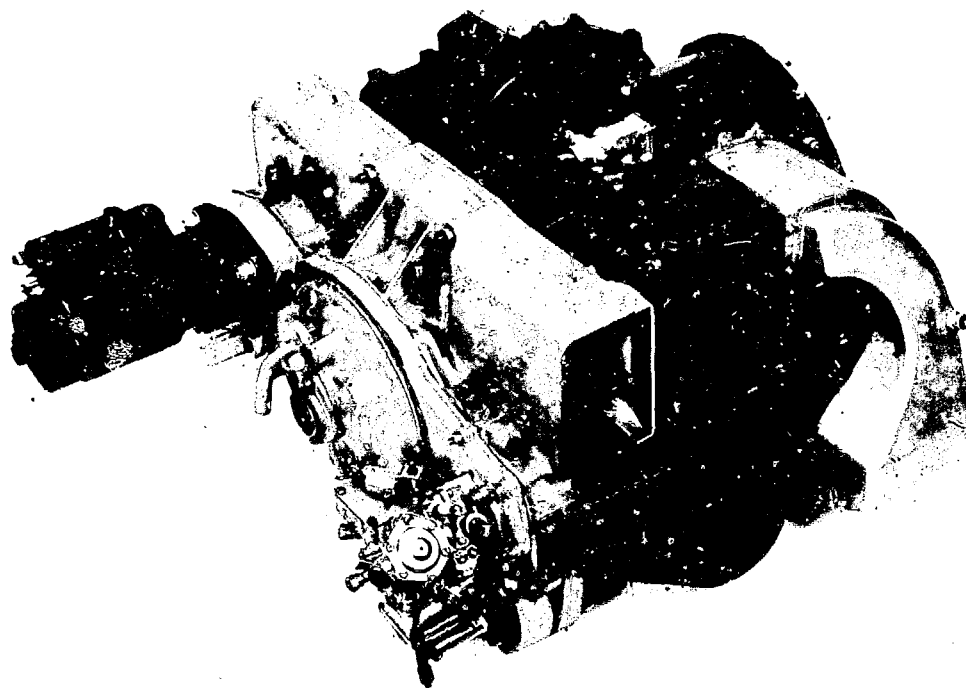
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AUTOMOTIVE GAS TURBINE
VW-GT 70

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VW-GT 70 IN MICROBUS

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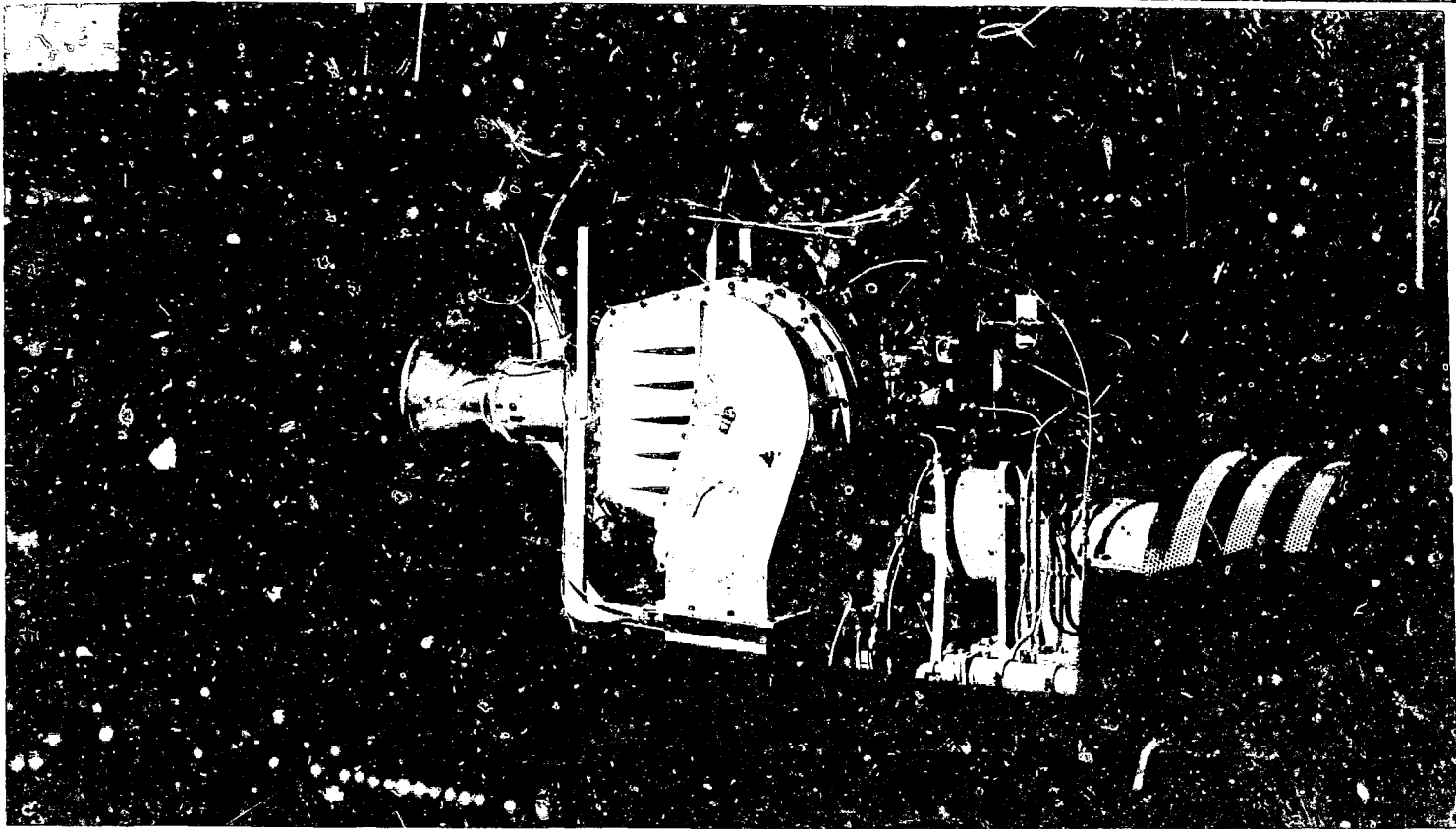
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AUTOMOTIVE GAS TURBINE
VW-GT 150

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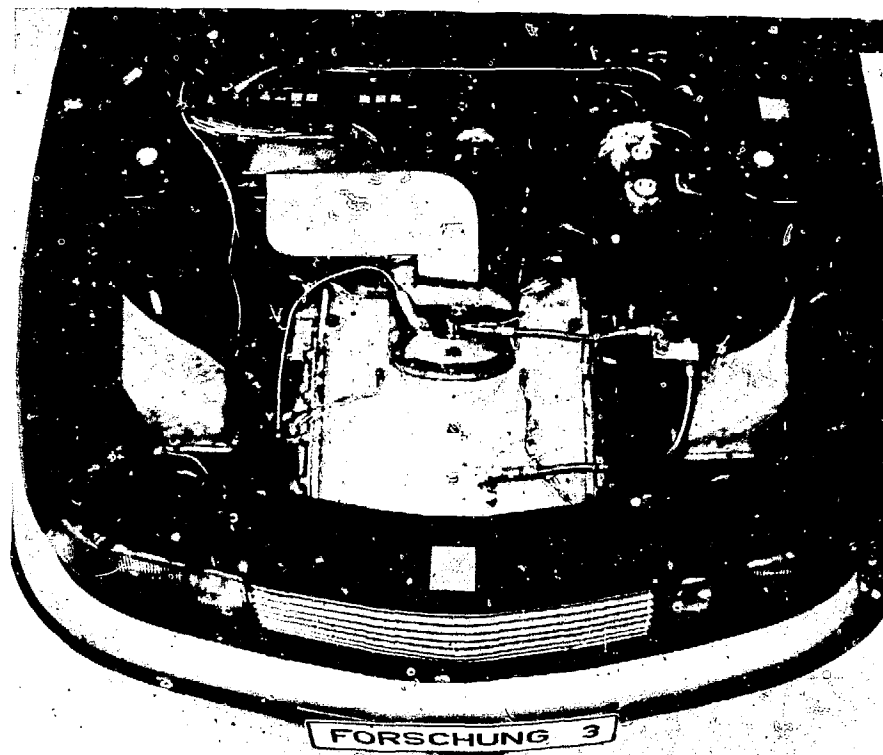
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VW GT 150 IN R0 80

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Schriftgröße

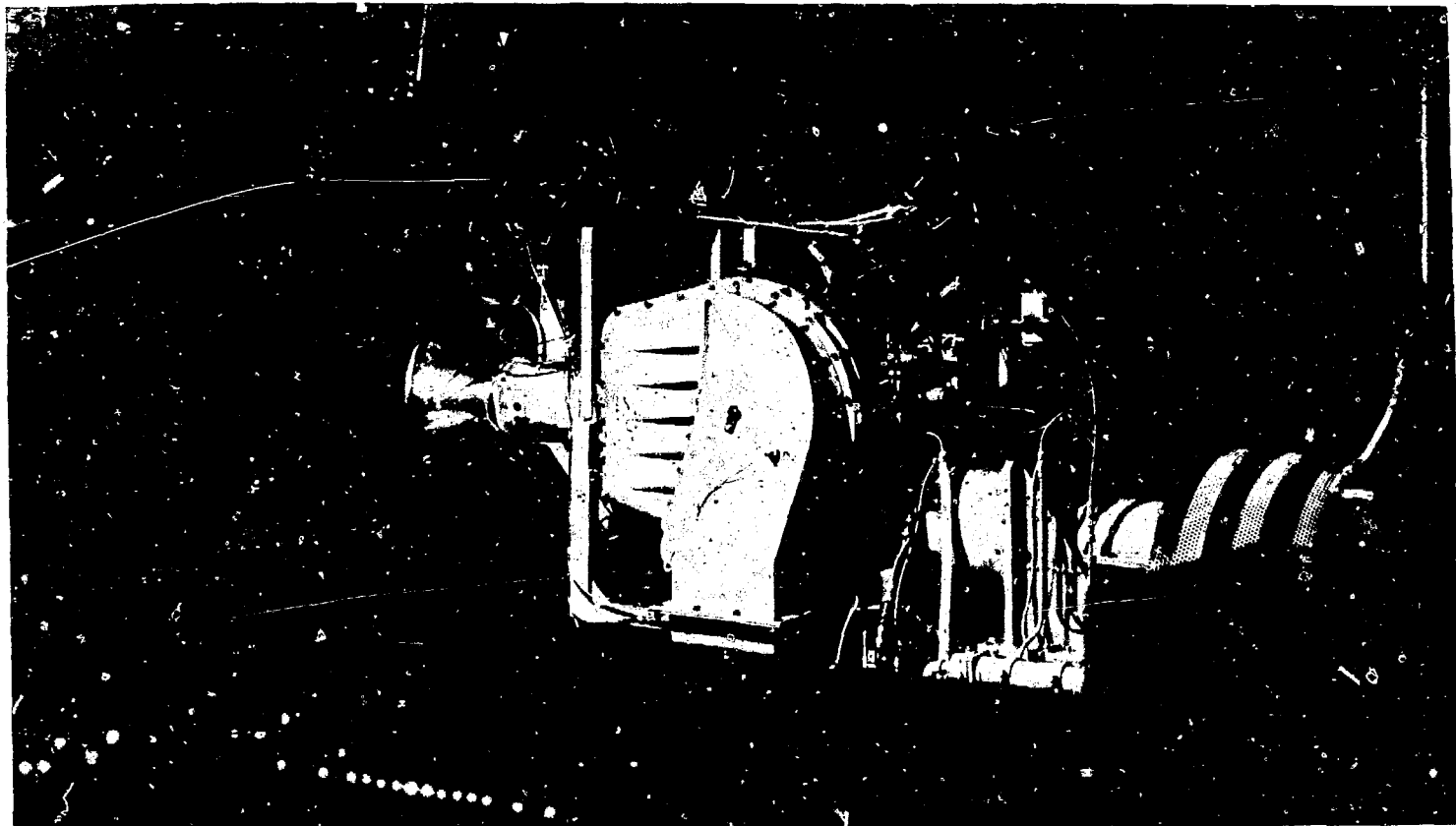
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AUTOMOTIVE GAS TURBINE
VW-GT 150

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Schriftgröße

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EXPERIMENTAL GAS TURBINE
CAR R080

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Linienbreite

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- 0,35 mm für Nebenteile
- 0,25 mm für Mittel-, Maß- und Schraffurlinien



DIFFUSION
FLAME



SINGLE STAGE
PREMIX



TWO STAGE
PREMIX



COMBUSTION CHAMBER CONCEPTS

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Schriftgröße

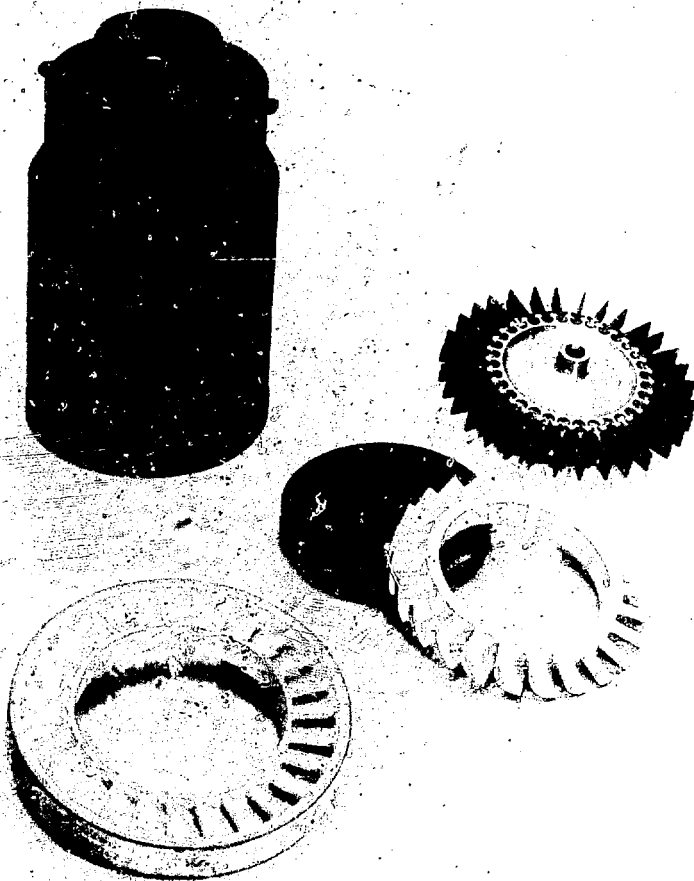
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CERAMIC GAS
TURBINE PARTS

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44 für Industrie und Exportation für die Zukunft

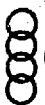
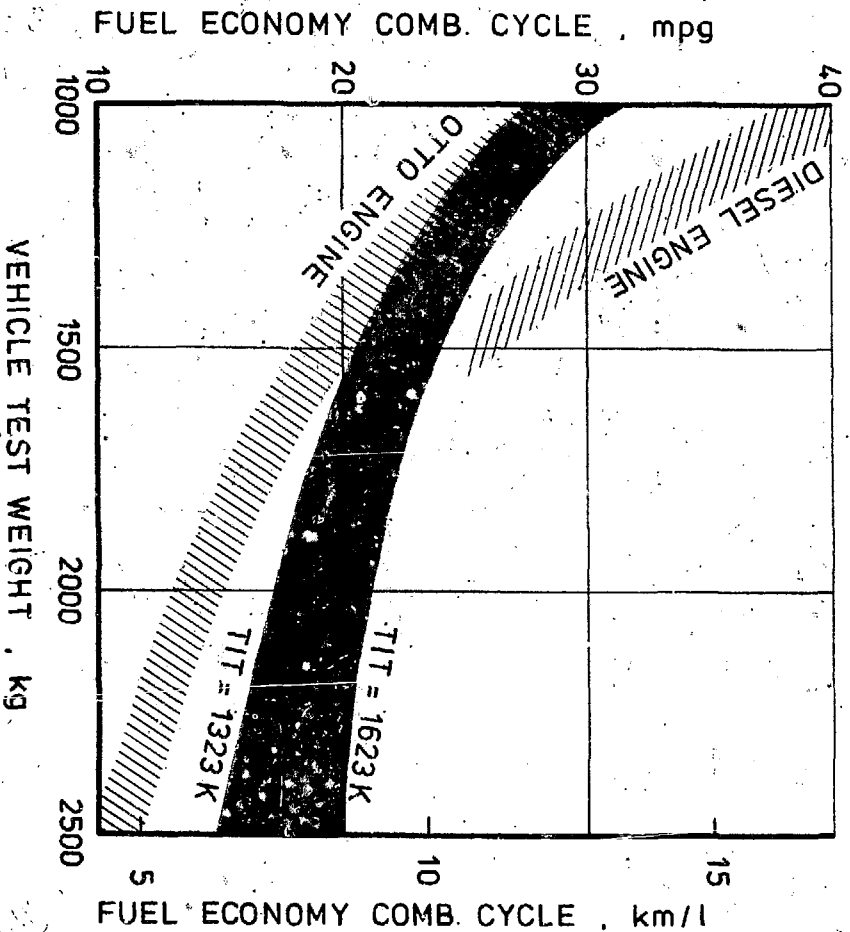
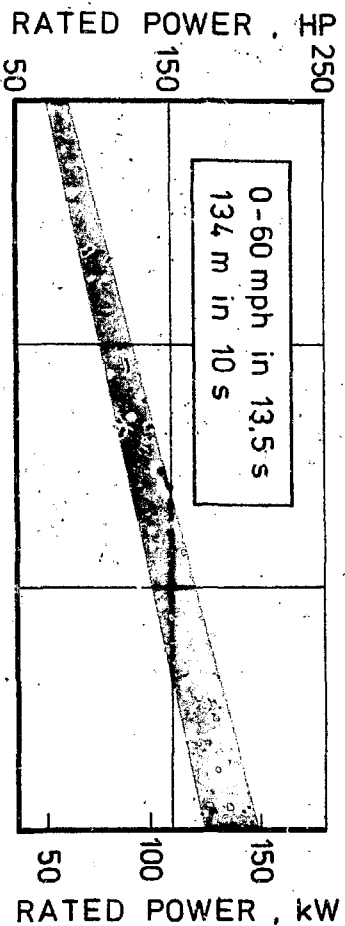
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FUEL ECONOMY POTENTIAL OF GAS TURBINES

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Schriftgröße

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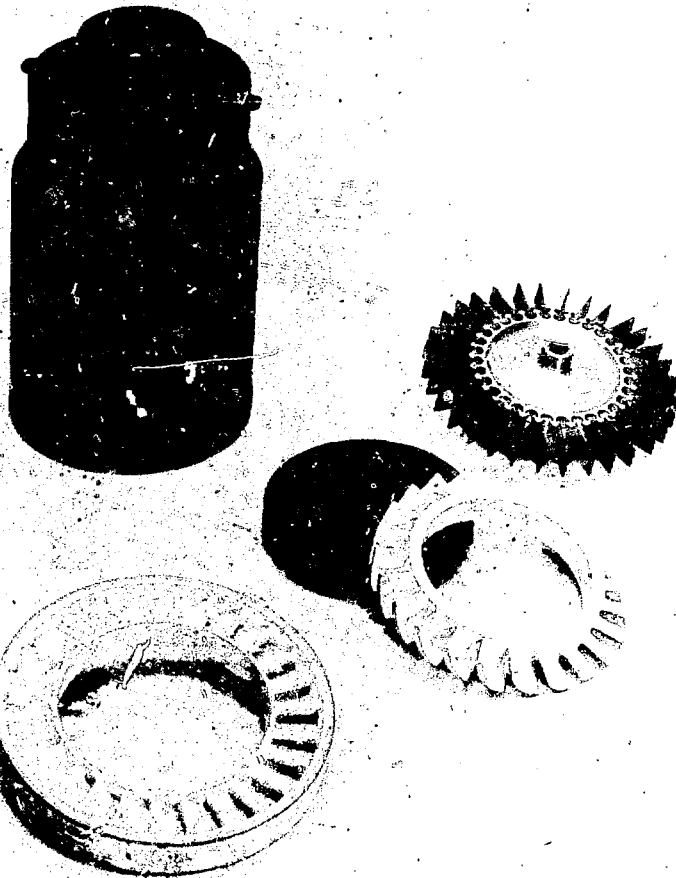
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CERAMIC GAS
TURBINE PARTS

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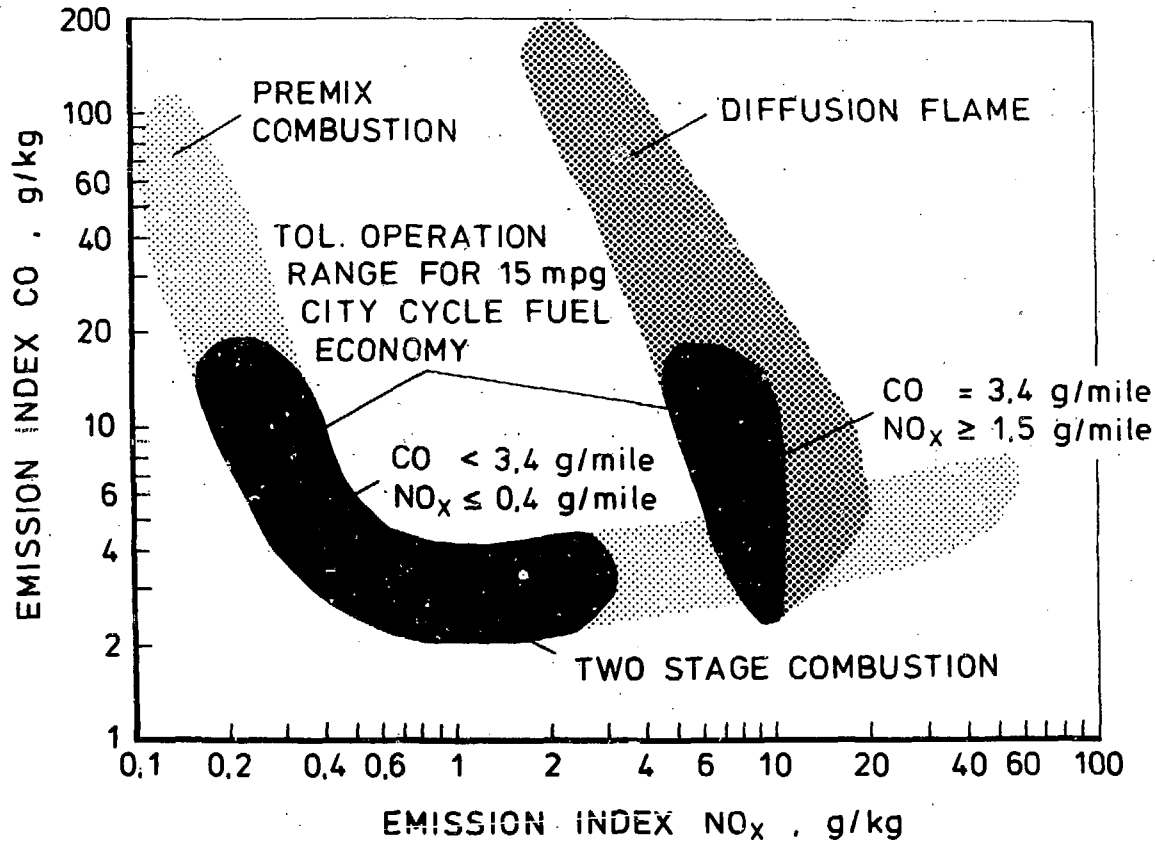
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CO-NO_x RANGES FOR DIFFERENT COMBUSTION SYSTEMS

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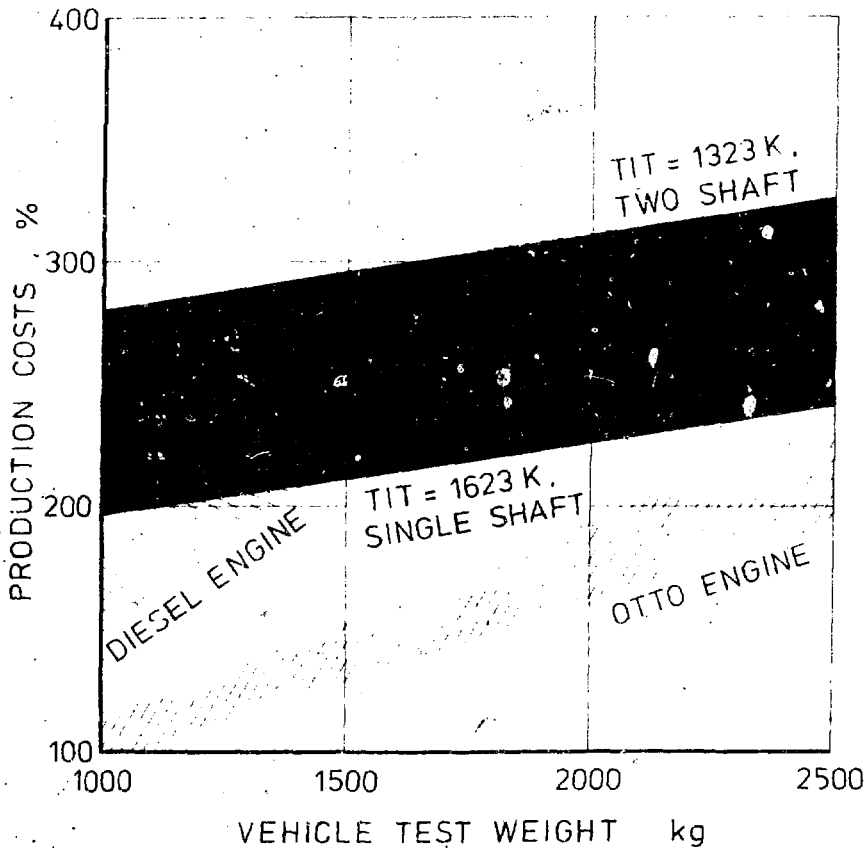
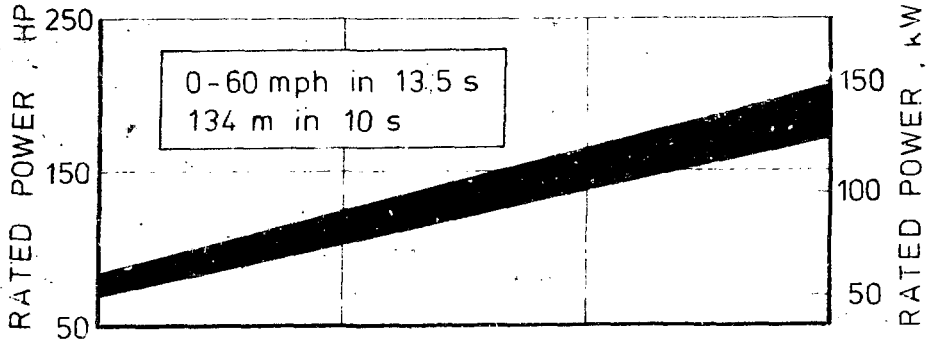
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PRODUCTION COST PRO-
JECTION FOR GAS TURBINES

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THE FIAT 6803 GAS TURBINE ENGINE - DESIGN AND DEVELOPMENT

Paper to be presented at 1977 ERDA - Fourth International Symposium on
Automotive Propulsion System/Low Pollution Power System Development
(April 18-22, 1977)

Washington DC - USA

by

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FIAT - S.p.A. - Torino

ABSTRACT

This paper describes the mechanical design, components test, engine performance and vehicular application of the FIAT 6803 two shafts, regenerative gas turbine engine.

The FIAT 6803 engine, designed for application on trucks and buses, has a maximum rating of 260 KW (350 HP). The adopted solution is with a single stage centrifugal compressor, two axial turbines, power turbine variable nozzle, two ceramic regenerators, and is electronically controlled.

The engine performance and the mechanical behaviour are on the whole satisfactory, although much work remains to be done in order to reduce fuel consumption. A great deal of emphasis is being placed therefore on improvements of combustion system, aerodynamic components efficiency and gasifier matching.

The main problems encountered within the mechanism of the engine concern the regenerators sealing system durability and the reliability of the variable nozzle mechanism. In fact much work is still required in regenerators area.

After completion of the test evaluation program, one unit has been mounted on an experimental bus for road tests. The drivability is good. The exhaust emissions are satisfactory.

INTRODUCTION

The vehicular gas turbine engine has been the object of a remarkable research effort by FIAT from the early 1950's.

In 1954 a simple cycle 180 KW engine was developed and installed on a vehicle for road demonstration.

The activity was subsequently directed to deepen the knowledge, both theoretical and experimental, of the gas turbine problems. Two models were developed of the two-shaft recuperative type with variable geometry nozzle for the power turbine. One of these engines followed the pattern with the power turbine upstream of the compressor's turbine, the other with the power turbine downstream.

At that stage a great deal of emphasis was placed to improve the ability to design the fluidodynamic components. The methods of calculation were verified and improved by an intense experimental activity on compressors, turbines and combustors.

At the beginning of 1970 the design of the 6803 engine for installation on an experimental bus was begun. Engine tests started at the beginning of 1974, the bus evaluation in the early 1976. The goal of the program is the acquisition of a direct experience of the problems and technologies connected with the gas turbine vehicular application.

The main effort has not, therefore, been directed to the design improvements as compactness and cost, but rather in order to achieve a unit which would allow the widest acquisition of experimental informations.

To this effect the engine has been designed following a modular technique which allows several engine arrangement variations.

The design detail drawings and construction stages have been developed together with an extensive experimentation of the fluidodynamic and mechanical components.

Presently two engines are testing, the first on a bus for road evaluation, the second on a test bench for endurance and reliability improvements. Tests on components, and in particular the combustor evaluation with regard to the emissions of pollutants, are in progress.

The purpose of the paper is to describe the thermo-fluidodynamic and mechanical design of the Fiat 6803 gas turbine engine, its installation on a bus and the performance achieved.

GENERAL LAYOUT

The Fiat 6803 gas turbine engine has been designed to be installed on trucks or buses. Its maximum power is 260 KW (fig. 1).

The engine is a two-shaft regenerative gas turbine interstage variable nozzle. As can be seen in fig. 2, the engine is also provided with a device to mechanically connect the power turbine with the gas generator (power-transfer) in order to verify the ability of this system for braking and control purposes.

The engine main section is shown in fig. 3.

The main housing, is the principal structural member of the engine, to which the following six major subassemblies are assembled:

- the gas generator assembly with ancillary equipments: fuel and oil pumps, variable nozzle actuator pump, starting system and the hydraulic pump for the automatic transmission;
- the two regenerators, fitted at both sides of the engine, with the inherent sup-

- port system, the drive mechanism and the seals;
- the combustor, located at the bottom side of the housing, with injector and igniter plug;
 - the power turbine variable nozzle;
 - the power turbine with reduction gear train. Incorporated in this unit is the clutch for the hydraulic actuated power-transfer device.

Each subassembly is a self-contained unit and can be separately removed from the engine, except the power turbine variable nozzle which can be removed only after the gasifier unit has been removed: this solution allows partial inspections and the possibility to modify several components without disturbing the remainder of the engine.

THERMODYNAMIC CYCLE

The air enters the compressor through a double entry air intake, then from the compressor diffuser outlet passes through the cool half of the regenerators to feed the combustor. From the combustor the hot gases are conveyed to the gasifier turbine by a vortex chamber. After the turbine stages the gases pass through the hot half of the regenerators and are finally exhausted.

The operating conditions at the design point are shown in table 1.

TABLE 1 - FIAT 6803 operation conditions at the design point (sea level and $300^{\circ}\text{K} = 27^{\circ}\text{C}$)

Output power	260 KW (350 BHP)
Output speed	2920 rpm
Air flow	2.4 Kg/s

Compressor pressure ratio	4.15
Turbine inlet temperature	950°C
Gasifier speed	33,300 rpm
Power turbine speed	27,500 rpm
Compressor efficiency	0.785
Compressor turbine efficiency	0.90
Power turbine efficiency	0.85
Combustor pressure loss	3.5%
Regenerator effectiveness	0.85
Regenerator speed	18 rpm
Regenerator leakage	5%
Fuel	No. 2 Diesel oil

AERO-THERMO DESIGN

Compressor

The centrifugal compressor impeller aerodynamic design has been carried out on the basis of the Dallenbach (1) and Katsanis (2) theory, consisting in the analysis of the hub-to-shroud and blade-to-blade equilibrium equations.

Afterwards the computing methods, based on the jet-flow and diffusion ratio concept of the R. Dean (3), were performed for check purposes. These methods are perhaps less mathematically valid, but able to describe more realistically the rotor flow which, being widely separated, is not actually similar to the hypothetical potential flow of the preceding methods. Likewise the diffuser (of the channel type) has been designed on the basis of the blade-to-blade analysis.

The present compressor design, defined throughout a large amount of experimental

research, includes a 22 blade rotor with radial exit (tangential velocity at outlet diameter is equal to 450 m/s) and a 13 channels diffuser (fig. 4). A low diameter ratio of the vaneless diffuser was selected to obtain a near straight surge line, favourable to a good gasifier matching.

At present a great deal of investigations is being performed both on the rotor and on the diffuser to locate the gasifier matching curve in the compressor maximum efficiencies range.

The total to static characteristics of the current compressor configuration are shown in fig. 5 compared with the first configuration.

Regenerators

Ceramic type regenerator cores of 685 mm outlet diameter and 76 mm thickness have been chosen. A lot of research work has been devoted to the design and development of the rubbing seals, the main problems being the leakage and wearing rates of the solid lubricant layer, plasma sprayed on the metallic pads.

To deeply investigate this subject a regenerator test rig has been built, able to perform the seals tests in real operating conditions. Fig. 6 shows the values of leakage and driving torque, obtained on the fixture, for the current seals and for the new type with preloaded pads.

Before the rig tests, an important experimental program has been performed to evaluate the characteristics of different kinds of solid lubricants.

Nevertheless a lot remains to be done to improve the life of the rubbing pads, which until now is far too short in comparison of other manufactures claims.

Combustor

The combustor of the 6803 engine is a conventional can type with three rows of holes and film cooling fences for the primary and secondary air and finally for the dilution air: a large swirler, useful to obtain a wide stability range of combustion

is provided.

For the geometry assessment an important experimental program has been performed using the combustors test rig.

The results of the experimental investigations, as gas temperatures distributions as the combustor exit and wall temperatures are shown in fig. 7. The pressure losses are fairly low (3,5% at the design point), no carbon deposits are generated in the whole range of operating conditions.

Recently, development work to optimize the combustor emission characteristics is being carried out using a specific test fixture, which exactly reproduces the fluidodynamic engine working conditions (fig. 8). Fig. 9 shows a new type of combustor fitted with the turbine inlet vortex chamber.

Turbines

The hot gases expansion occurs through two single stage axial turbine rotors: the first to drive the compressor, the second for the power generation. The rotors rotate in opposite directions.

The blade design has been performed employing computing methods derived from the initial Baumgartner concept (4), which was afterwards implemented using the radial equilibrium equations. The basic flow pattern concept applied in both wheels design is the free vortex, which is particularly suitable in this case, because of the relatively small height/diameter cascade ratio. The blade losses evaluation is based on the Sodeberg correlation.

The efficiencies of the gasifier turbine are shown in fig. 10. The exhaust diffuser geometry results exhibit a static recovery equal to above 50% of the kinetic inlet energy.

Control system

The control of the engine operates essentially on the fuel flow and on the geometry of the power turbine variable nozzle. It is performed by an Electronic Control Unit (E.C.U.) able to define the proper fuel flow and the variable nozzle blade setting versus the accelerator pedal and the engine working conditions.

The hydraulic actuator of the variable nozzle inlet vanes is electrically controlled by the E.C.U.

The E.C.U. logic, shown in fig. 11, is alike the others applied in similar engines. It allows two operating modes: the "slow" and the "fast" control.

In the case of "slow" power variation demands by the driver, the fuel flow and the power turbine variable nozzle are controlled in such a manner as to maintain the gas generator Turbine Inlet Temperature (T.I.T.) at the maximum value between the maximum gas generator r.p.m. and 90% r.p.m.. Below of this value the T.I.T. is decreased in order to maintain the regenerators inlet temperature within the safety limits.

"Fast" power increase requests are complied by a quick opening and closing of the variable nozzle together with a larger fuel flow supply. The TIT control, in this phase, is shown in the dotted line of fig. 12, shifted upwards of about 80°C with respect to the slow control line.

When the variable nozzle is opened nearly all the enthalpy drop is accomplished through the first turbine, therefore a fast acceleration of the gas generator occurs. The following quick closing of the nozzle allows the power turbine to pick up the higher available power with a very short delay with respect to the driver demand.

When the driver quickly releases the accelerator pedal, the control system sets the nozzle vanes in reverse position, thus achieving a braking effect.

The E.C.U. is provided with programs for the starting and shut-off procedures and with safety devices.

The power-transfer system control has not yet been realized at the present.

MECHANICAL DESIGN

Engine housing

The engine main housing (fig. 13) is an one-piece nodular iron casting, providing the connection of several modules as the gas generator, the power turbine unit, the combustor subassembly and the oil sump. The regenerators accommodations are provided at both sides of the housing comprising the support and drive system, the seals platform and the regenerators cover flanges.

The engine bulkhead dividing the high from the low pressure side is fitted with power turbine variable nozzle support flange. The inner housing surface is supplied with linerless preformed insulation panels directly glued on the metal. The panel surface is hardened to avoid erosion from the gases.

Gasifier and ancillaries equipments

The gasifier module includes the compressor, the compressor turbine, and the reduction gear train for the ancillary equipments actuation.

The bearing support is in aluminum alloy casting (A-U5NZr), shielded with thermally insulating panels as used in the housing.

The centrifugal compressor impeller is an aluminum alloy investment casting, fitted on the gas generator shaft by means of a polygon profile correction as a self-centering device. The compressor wheel vibration characteristics have been checked using laser holographic techniques (fig. 14).

The compressor diffuser is an aluminum alloy casting, finished by a numerical control machine.

The gasifier turbine wheel (fig. 15) is integrally casted with the blades in INCO

713 LC. It is electron beam welded to the shaft.

The first stage nozzle is integrally casted in Stellite 31 with the gas duct which is fitted with support rings for the turbine inlet vortex chamber at one end, and at the other end for the interstage. The vortex chamber is made in Hastelloy X sheet. A gear train, fitted on the engine front, provides the actuation of the oil and fuel pumps and the regenerators drive.

The starting is made by means of an 8 KW electric motor which drives the gear train through an electro-magnetic clutch. When the clutch is disconnected, the electric motor can independently drive an auxiliary pump to feed the lubricating oil system before the engine start or to provide an oil flow to cool the bearings after the engine is stopped.

Regenerators

The driving and suspension system of the regenerator cores is made by an outside metallic ring with intermediate foil springs to compensate the thermal expansion and to transmit the driving torque (fig. 16).

The metallic ring is externally shaped with two races for suspension rollers and is fitted with a toothed ring meshing with the driving pinion. The radial position in the housing is provided by the rollers mating with the ring races. (Fig. 17)

One roller is mounted on a fixed pivot, the second on springed pivot, the third is integrally machined with the driving pinion connected to the driving shaft of the regenerator by a polygon profiled joint. The regenerators are axially located by the rubbing seals supporting the resulting gas thrust on the core.

The sealing system consists of metallic rubbing pads plasma sprayed with solid lubricant (CaF₂+NiO₂) and flexible metal foils. The present and the upgraded configurations are shown in fig. 18.

The second configuration has been studied in order to give some pressure on the rubbing pads even at the low engine speeds.

The rubbing pads installation in the engine housing is shown in fig. 19.

Combustor

The combustor unit consists of the combustion chamber, the injector and the igniter plug, all fitted to an external cap, flanged to the main housing. The injector is a single nozzle air assisted type.

In steady-state, the air is supplied by the compressor diffuser exit with a dynamic intake. During the starting sequence the air is supplied by the brake bus system. Several igniter plugs were tested. The high energy system has been finally selected.

Variable power turbine nozzle

The variable power turbine nozzle module (fig. 13) consists of 27 Hastelloy X precision cast vanes supported by a ring holder, flanged to the bulkhead.

The vanes are fitted with pinion gears meshing with a ring gear supported by the holder. In order to reduce friction and wear the vanes stems are spray coated with solid lubricant and the ring gear is supported by suitable graphite pads.

The ring gear is actuated by an externally located hydraulic rotary actuator connected, through an Oldham coupling, to the stem of one vane. The vanes angles range from -12° (min throat area) to $+93^{\circ}$ (reverse position) with respect to the design angle. The holder bears, by means of the vanes bushes, the external wall of the interstage duct. The internal wall of this duct is clamped by three airfoil shaped struts fitted to the walls with a suitable degree of freedom in order to allow differential thermal expansions.

Power turbine

The power turbine module consists of the rotor, the shaft and bearings (fig. 20), the

exhaust diffuser and the main gearbox. The rotor bearing housing is an aluminum alloy casting (A-U5NZr) externally shielded by a suitable thermal insulation.

The turbine blades are integrally casted with the rotor in INCO 713 LC as the gasifier turbine. The shaft is electron beam welded to the rotor.

The reduction gear train is double staged and made by two couples of helical gears with a resulting gear ratio of 9,45 to 1. In the gear box is also housed the power-transfer clutch.

Lubrication

Oil for lubrication and cooling is provided by three pumps, mechanically driven by the gasifier shaft: the first pump supplies the oil to the bearings and the gears, the second takes the oil from the sump to the cooling radiators and supplies the pressure to actuate the power-transfer clutch, the third scavenges the oil from the auxiliary gear box pan.

The oil complies with DERD 2487 specifications. The oil sump capacity is 30 litres.

Fuel system

The fuel is supplied to the nozzle by a gear pump through a metering device which controls the nozzle pressure ($p=30$ bars at the design point) as a function of the compressor discharge pressure and according to an electrical signal from the E.C.U.. An electrically drive boost pump supplies the pressurized fuel during the starting sequence.

The fuel is Diesel oil number 2.

Bearings

The gasifier and power turbine shafts are carried by hydrodynamic bearings fitted to an oil damped flexible support.

The hydrodynamics bearings were selected because of their reliability and quietness characteristics; nevertheless a solution adopting roller bearings is being investigated, mainly with the aim of reducing the friction losses.

The material selected for the bearing supports is cast aluminum in order to avoid engine soak-back problems after engine stop, because of the good thermal conductivity of this material.

The external thermal insulation and the cooling of the internal surface, provided by the oil discharged from the bearings, assure a steady state temperature of the bearings well below the safe values.

ENGINE TEST

The test bench results of the engine performance are shown in fig.21.

The main functional parameters are plotted in fig. 12 versus the gasifier speed.

A test bench exhaust emissions survey has been performed on the engine in its current configuration, not yet optimized from the emission point of view.

The emissions levels according to the "13 modes" procedure for heavy duty vehicles are: $HC = 3.4 \text{ gm/HP.h}$, $CO = 24 \text{ gm/HP.h}$, $NO_x = NO + NO_2 = 2.64 \text{ gm/HP.h}$.

The results regarding nitrogen oxides are plotted in fig. 22 versus gasifier speed.

The high values of hydrocarbons and carbon monoxides are due to a not yet satisfactory combustion at partial loads; a test program for improving the combustion process in these conditions is underway.

VEHICLE INSTALLATION

The bus is a modified intercity current model normally powered by a 200 KW Diesel engine. (Fig. 23).

The chassis modifications were made in order to allow the engine installation in the rear end and the body modifications in order to provide the passages for the air inlet and exhaust gases ducts and the installation of the accessories and auxiliary units in the vehicle central zone (fig. 24).

The engine is connected to the wheels by means of an Allison HT740CT four speed automatic transmission fitted with an hydraulic coupling. The coupling has a very low torque characteristic at power turbine low speed and is completely and automatically clutched when the power turbine speed exceeds 5000 r.p.m..

The engine and transmission oil coolers and the vehicle accessories (coolers, fan, brake air compressor, alternator, power steering pump) have been installed in the central section of the bus to achieve a better weight balance and to improve the engine accessibility. The power to drive accessories is supplied by the power turbine through an hydrostatic transmission.

Devised as a travelling test bench the vehicle is supplied with a complete system of survey, with visualization and recording of all the main functional parameters of the engine and the vehicle.

Vehicle performance

The tractive force characteristics of the vehicle are shown in fig. 25.

The acceleration curves of the vehicle at full load (total weight 16.000 Kg=35000lb) are plotted in fig. 26 as function of time.

The drivability is satisfactory.

Noise

Although the vehicle is only roughly treaded from the point of view of the acoustic insulation, a survey of the emitted noise has been performed following the EEC n°73/350 specifications for heavy-duty omologation (full load acceleration from constant speed of 3/4 of max speed in II gear = 45 Km/h, with fully loaded vehicle) the noise levels obtained are 82 dB (A); EEC limits are 91 dB (A).

ACKNOWLEDGMENT

The authors wish to express their gratitude to dr. G. Savonuzzi for his leadership in the design and development of the 6803 gas turbine engine.
Finally, the authors wish to thank the FIAT Research Centre for permission to publish this paper.

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T. Katsanis: Use of arbitrary quasi - orthogonals for calculating flow distribution on a blade-to-blade surface in a turbomachine (Lewis Research Center).
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The FIAT 6803 Gas Turbine Engine - Design and Development

ILLUSTRATIONS

- Fig. 1 - Fiat 6803 gas turbine engine
- Fig. 2 - Schematic general arrangement
- Fig. 3 - Fiat 6803 gas turbine main section
- Fig. 4 - Centrifugal compressor on test rig
- Fig. 5 - Compressor performance map.
- Fig. 6 - Regenerator loadings and torque characteristics
- Fig. 7 - Temperature distribution at combustor exit and wall temperature
- Fig. 8 - Combustor test rig simulating engine fluidodynamic flow conditions
- Fig. 9 - New type of combustor with turbine inlet volute
- Fig. 10 - Gasifier turbine efficiency
- Fig. 11 - Fuel and variable power turbine nozzle control system schematic
- Fig. 12 - Performance parameters versus gasifier speeds
- Fig. 13 - Main housing fitted with power turbine variable nozzle ass'y.
- Fig. 14 - Comparison of parallel elastic lower vibration mode
- Fig. 15 - Gasifier rotor assembly with bearings
- Fig. 16 - Ceramic core regenerator elastic suspension and driving
- Fig. 17 - Regenerator suspension and driving system
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- Fig. 19 - Rubbing pads installation in the engine housing
- Fig. 20 - Power turbine rotor assembly with bearings
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- Fig. 22 - Emission map of the engine
- Fig. 23 - Fiat 6803 powered bus
- Fig. 24 - Vehicle installation
- Fig. 25 - Tractive force characteristics
- Fig. 26 - Vehicle road performance

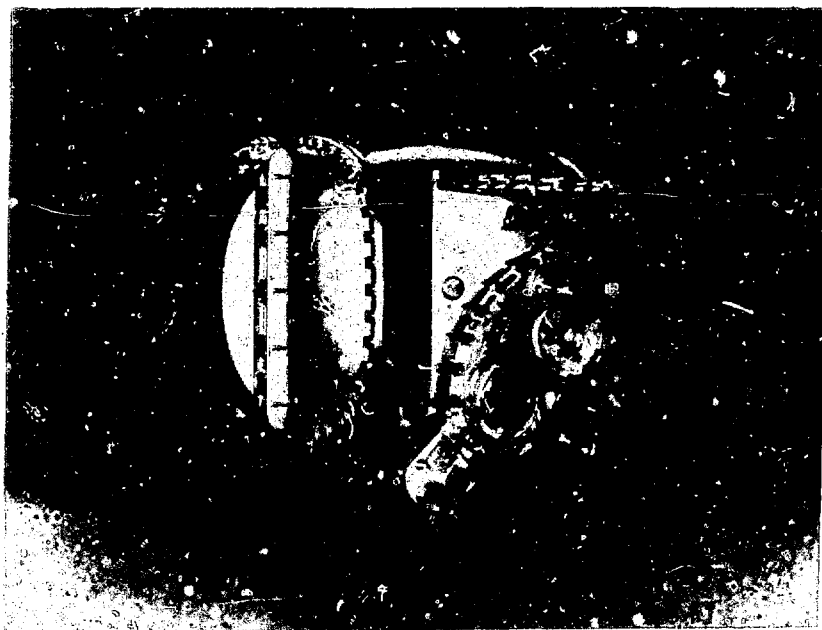


Fig. 1 - Fiat 6803 gas turbine engine

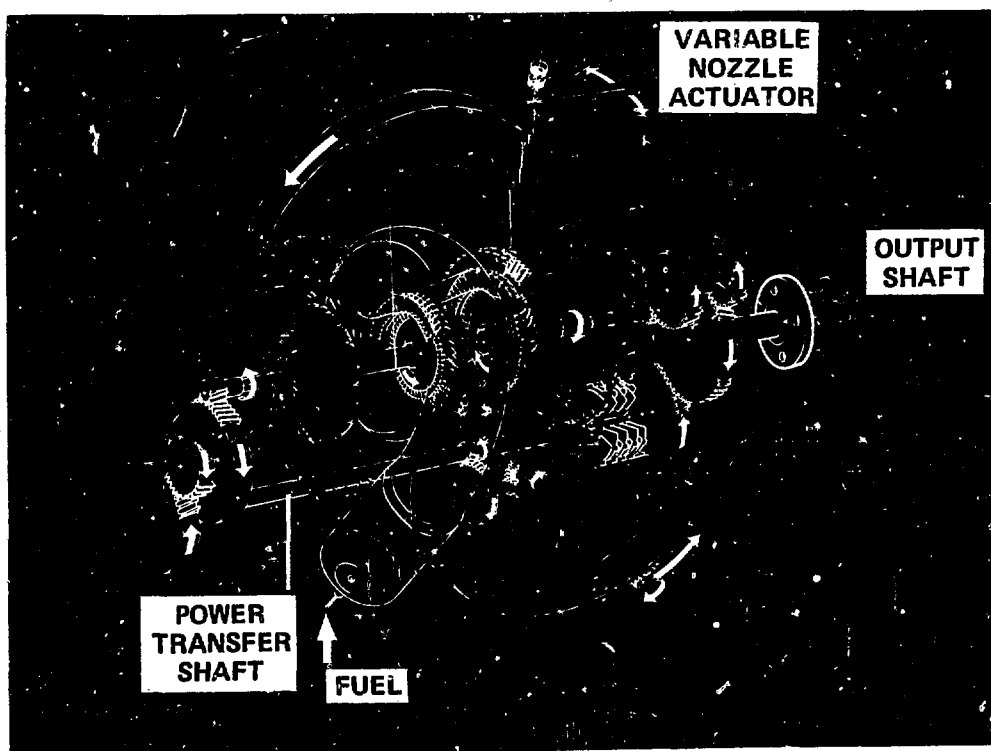


Fig. 2 - Schematic general arrangement

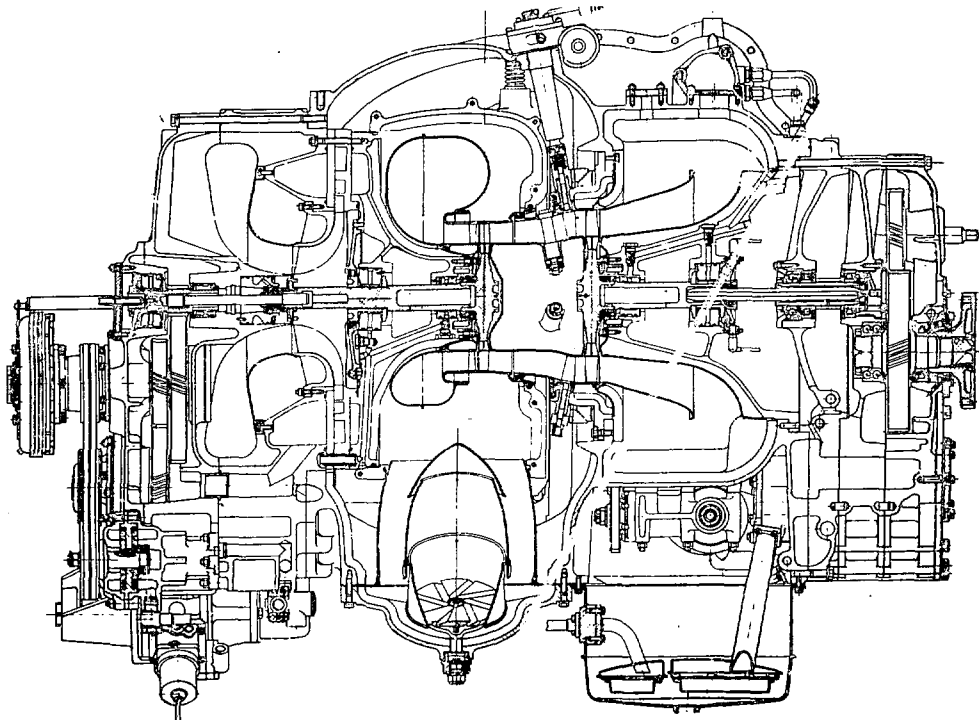


Fig. 3 - Fiat 6803 gas turbine main section



Fig. 4 - Centrifugal compressor on test rig

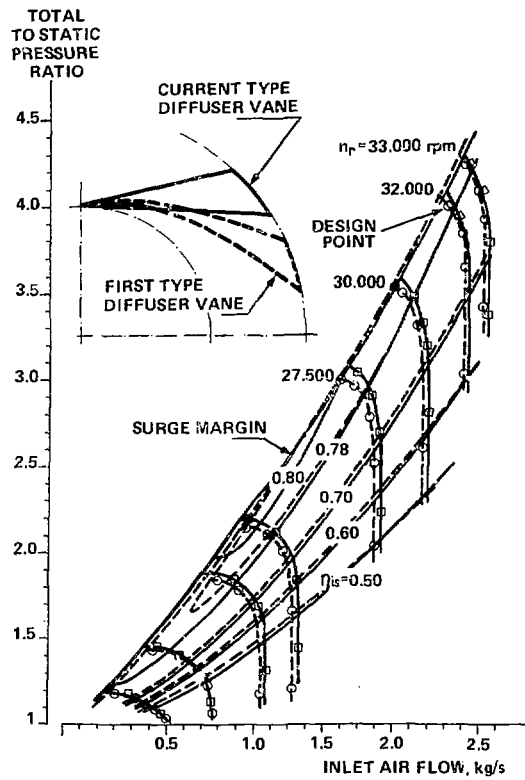


Fig. 5 - Compressor performance map.

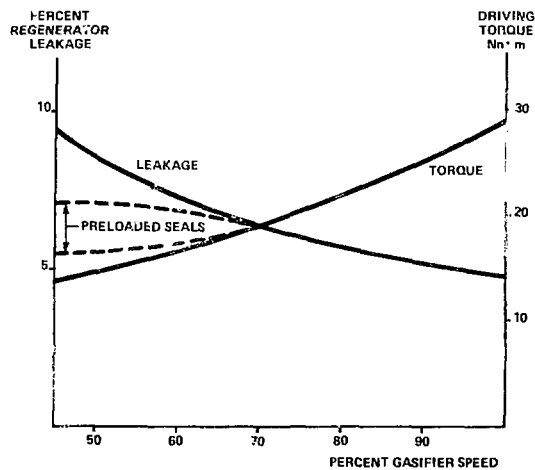


Fig. 6 - Regenerator leakage and torque characteristics

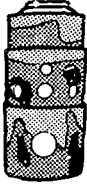
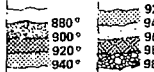
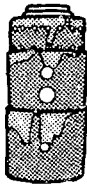
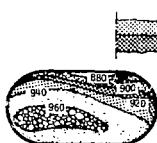
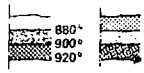
6803 COMBUSTOR CHAMBER		GAS TEMPERATURE DIFFERENCES		PRESSURE LOSS $\Delta P/P_0$
		$t_{max} - t_{min}$ °C	$t_{max} - t_{1/2}$ °C	
FIRST COMBUSTOR CHAMBER 	WALL TEMPERATURES  DISCHARGE GAS TEMPERATURE 	124	65	4.61
CURRENT COMBUSTOR CHAMBER 	WALL TEMPERATURES  DISCHARGE GAS TEMPERATURE 	103	20	1.5

Fig. 7 - Temperature distribution at combustor exit and wall temperature

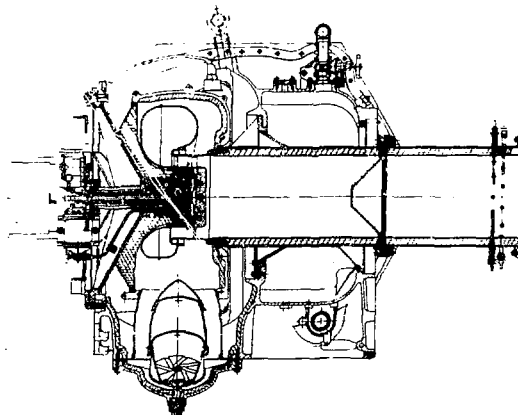


Fig. 8 - Combustor test rig simulating engine fluidodynamic flow conditions



Fig. 9 - New type of combustor with turbine inlet volute

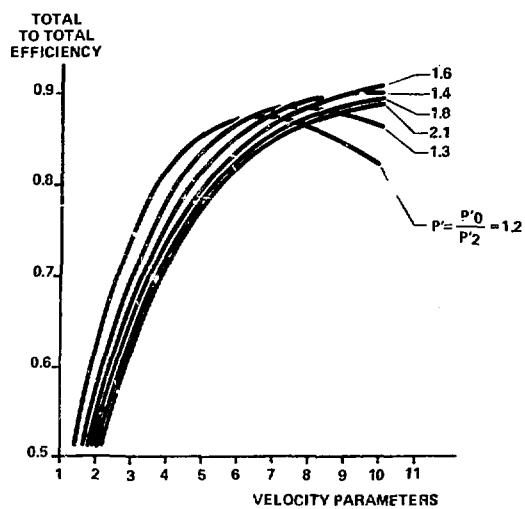


Fig. 10 - Gasifier turbine efficiency

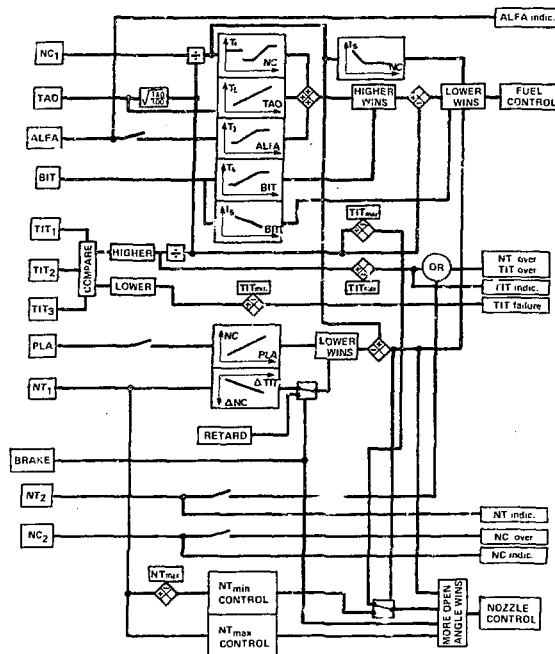


Fig. 11 - Fuel and variable power turbine nozzle control system schematic

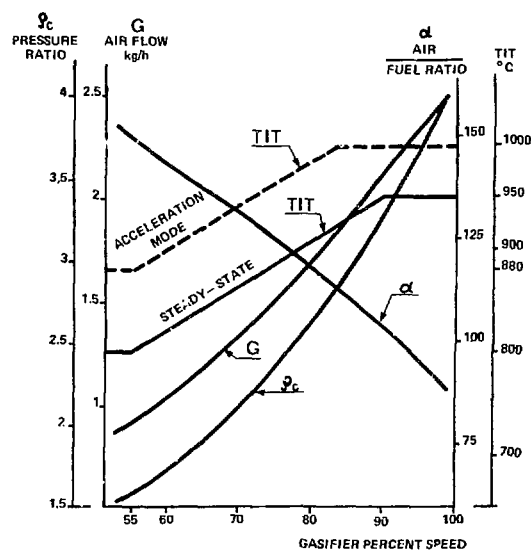


Fig. 12 - Performance parameters versus gasifier speeds

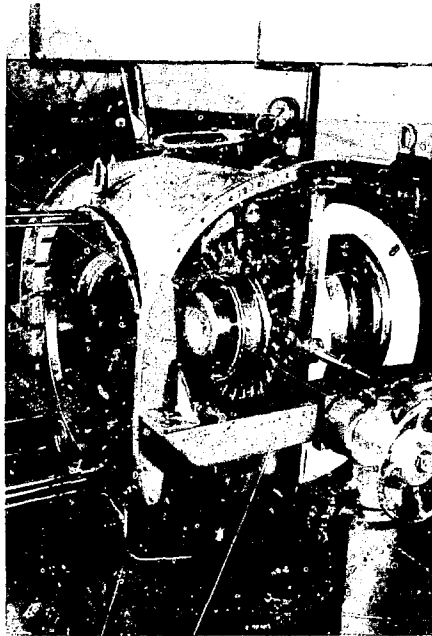


Fig. 13 - Main housing fitted with power turbine variable nozzle ass'y



Fig. 14 - Compressor impeller olografic laser vibration mode

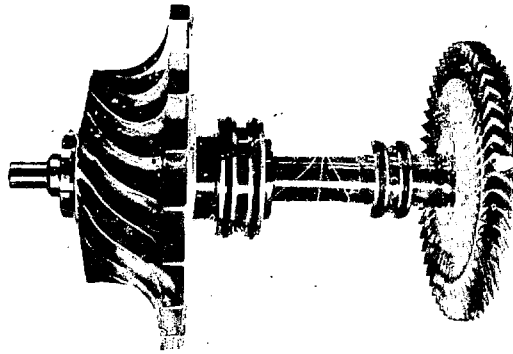


Fig. 15 - Gasifier rotor assembly with bearings

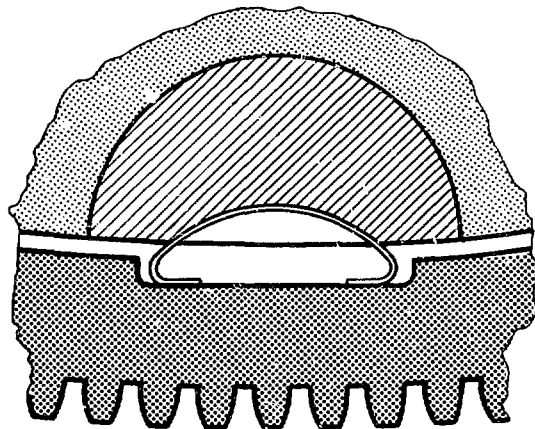


Fig. 16 - Ceramic core regenerator elastic suspension and driving

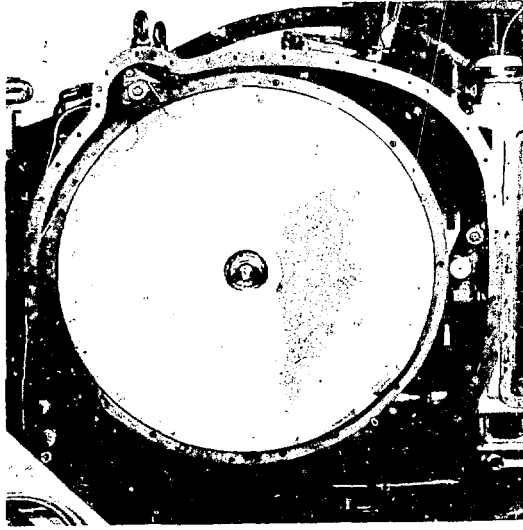
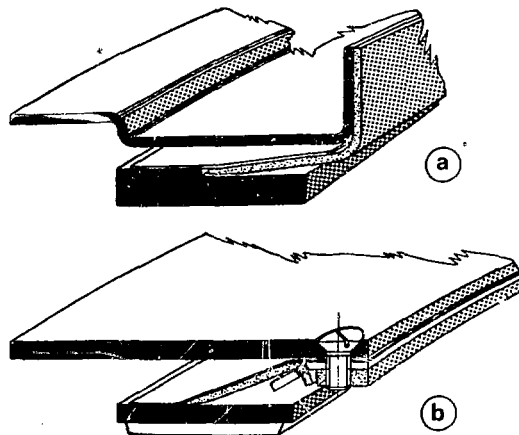


Fig. 17 - Regenerator suspension and driving system



- a) PRESENT CONFIGURATION
- b) UPGRADED PRELOADED CONFIGURATION

Fig. 18 - Regenerator sealing systems



Fig. 19 - Rubbing pads installation in the engine housing

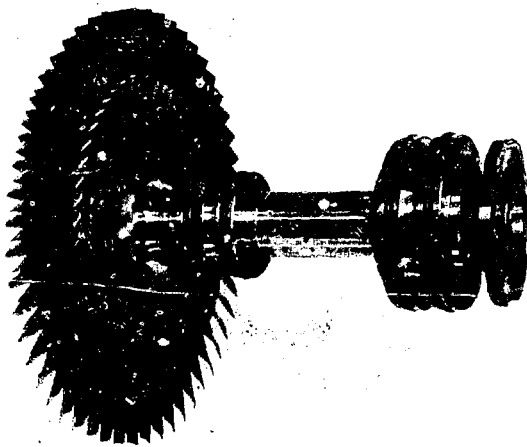


Fig. 20 - Power turbine rotor assembly with bearings

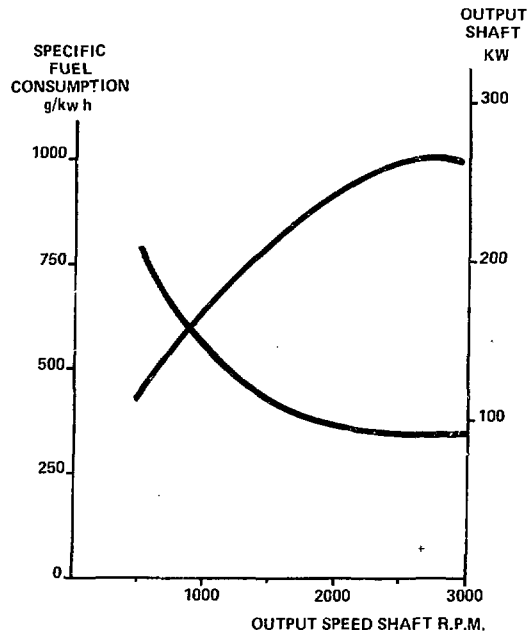


Fig. 21 - Engine performance

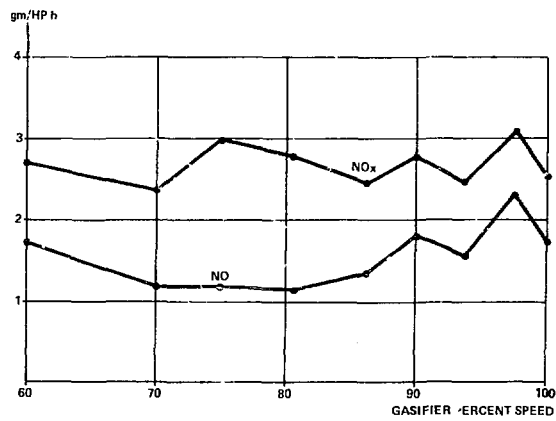


Fig. 22 - Emission map of the engine

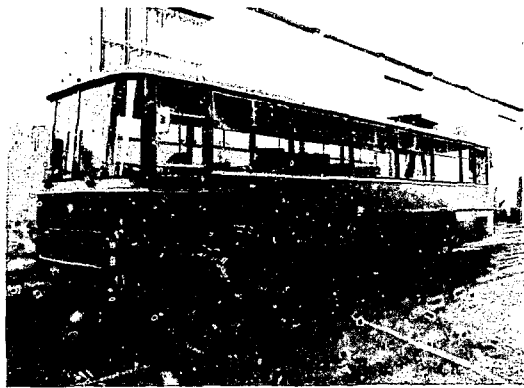


Fig. 23 - Fiat 6803 powered bus

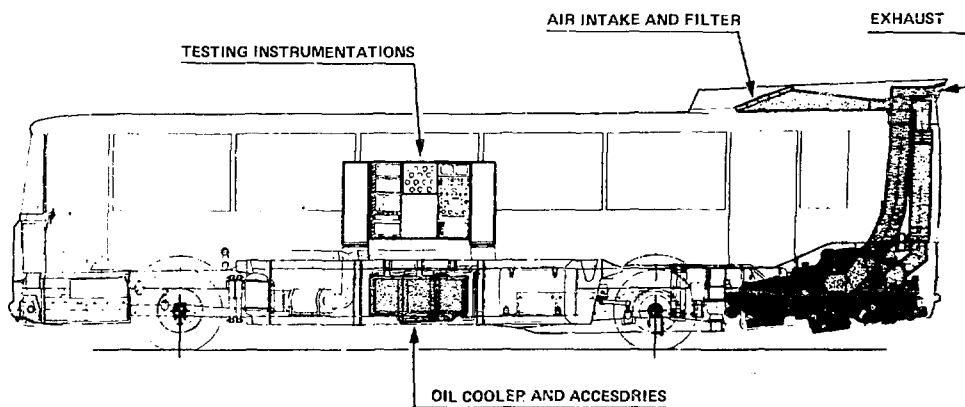


Fig. 24 - Vehicle installation

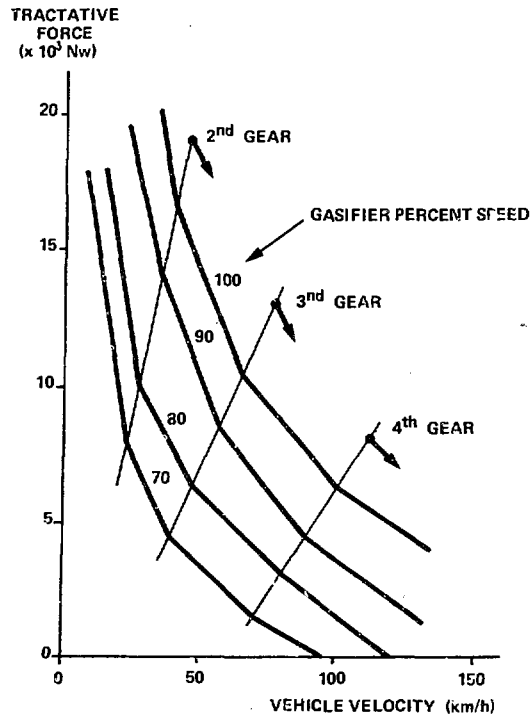


Fig. 25 - Tractive force characteristics

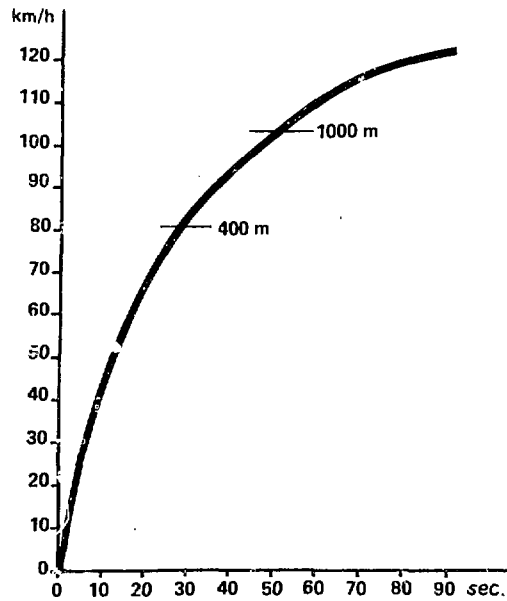


Fig. 26 - Vehicle road performance

COMBUSTION PERFORMANCES
OF INTERMITTENT FUEL INJECTION OF GAS TURBINE

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ABSTRACT

This paper deals with the control and combustion of automotive gas turbines, and describes the results of our combustion tests conducted to find out how an inexpensive on-off control system could be applicable to conventional gas turbines. To obtain stable continuous combustion, the frequency of an on-off control system producing intermittent fuel injection should be approximately 40 Hz or more. In our tests of intermittent fuel injection, the stable combustion zone was expanded by damping the pulsation of fuel. Further, intermittent fuel injection rendered it possible to decrease the fuel flow in ignition. Emission performance proved about the same as with conventional control systems.

FOREWORD

The gas turbine is one of the best prospects for automotive engines of the future from the standpoint of environmental protection and conservation of natural resources.

Among the problems involved in putting the gas turbine to practical use as an automotive engine is the problem of initial cost. One factor that renders the gas turbine high in cost is the use of super alloy; another is the fuel control system. To make the fuel control system available at a low cost is essential before the gas turbine can be put to practical use as an automotive engine.

We believe that inexpensive electronic gasoline injection systems, widely used recently on passenger cars, may be applicable to the gas turbine. There is perhaps no other instance in which such an intermittent fuel supply system and an intermittent injection method have been applied to the gas turbine.

To achieve application of the intermittent injection method to the gas turbine, we have conducted a variety of tests and have gained insight into its applicability. The test results are described in this paper.

The tests have been conducted on the following three items:

- (1) Combustion stability
- (2) Inflammability and blow-out limit
- (3) Exhaust emission performances

The stability of combustion is an important factor which affects driveability and which greatly contributes to the vibration and noise of

the engine components. Therefore, we have conducted:

- (1) Visual observation of flame
- (2) Measurement of pressure fluctuation in combustor
- (3) Measurement of flame luminosity fluctuation

PULSE WIDTH MODULATION (PWM) FUEL CONTROL SYSTEM

The purpose of controlling the gas turbine is to efficiently obtain necessary output from the engine, with mechanical or thermal protection provided for the engine. The fuel control system, calling for highly accurate control, involves a heavy cost.

To control the fuel flow, the fuel passage area of a measuring valve already has been changed by a servomechanism. If a fuel flow on-off control system is applicable to fuel control in the gas turbine, a more inexpensive measuring valve and control circuit will suffice. Furthermore, such a pulse-controlled method will be easy to handle.

The on-off control utilizes only two positions--on or off--and the fuel flow is controlled by changing the ratio of "on" time to one cycle time. PWM is a control method which changes the time average of a controlled variable by changing the ratio of "on" time to "off" time, under a constant and stable frequency (Cf. Fig. 1). Fig. 2 shows the fuel flow when the on-off valve is utilized.

CONTINUOUS INJECTION AND INTERMITTENT INJECTION

Unlike a reciprocal engine, the gas turbine carries on continuous

combustion with continuous fuel supply. Fuel pressurized by the fuel pump is sprayed from the fuel injector into the combustor after being measured by the control valve. Near the nozzle of the fuel injector, the fuel forms a spinning film in the shape of a hollow cone; the spinning film, as it becomes distant from the nozzle, grows thinner and undulates until it is separated into minute droplets and scattered away. These droplets, grouped together, form spray.

In a continuous fuel supply, the flow of droplets injected from the nozzle is considered constant.

Suppose that the constant supply of fuel to the fuel injector is discontinued temporarily, such as for 0.5 second. Then the flow of fuel injected from the nozzle also will undergo a temporary change. In other words, a nearly constant distance between droplets will increase.

If this distance is sufficient to permit droplet flame propagation under the given conditions, the flame can occupy the same position in a stable system of coordinates. If this distance is increased, the fuel (droplet) supply to the flame under formation will be delayed so that the flame front will recede from the nozzle for as long as the lag time.

Where the flame front is stabilized after that point will, of course, be determined not only by the size and velocity of droplets passing through the space near the flame front but also by the air-fuel mixture ratio, air flow and temperature there. In the actual combustor, droplets forming groups affect one another in an intricate manner, and both droplet combustion and gas combustion simultaneously take place. Such a condition makes analysis difficult.

TEST METHOD AND EQUIPMENT

Fig. 3 shows the test equipment. The test rig is a cylindrical can type combustor.

JIS No. 1 kerosene has been used as fuel, and preheating of the air at the combustor inlet has not been performed.

To judge the stability of the combustion process -- that is, to see whether combustion reaction as a whole is occurring smoothly -- a total pressure probe has been installed in the liner in order to record, by a photocorder, the output signal of the pressure transducer.

The flame luminosity has been measured by a phototransistor (its circuit is indicated in Fig. 4). The flame luminosity is related to the stability of the flame as well as to an increase or decrease in free carbon in the flame and is useful in comprehending the condition of the flame after droplet combustion. After the fuel is pressurized to 30 kg/cm^2 , the fuel flow is established by the on-off valve and is supplied to the fuel injector. In order to examine how the damping of the fuel pulsation produced by the on-off valve affects combustion, two system circuits have been provided between the on-off valve and the fuel injector.

A copper tube has been used for one circuit, and a slightly elastic rubber tube for the other. Simultaneously recorded by the photocorder are: (1) the output signal from the PWM controller for opening or closing the on-off valve; (2) the pressure of fuel immediately in front of the fuel injector inlet; (3) pressure in the combustor; and (4) the output signal of the circuit of the flame luminosity measurement.

TEST RESULTS

(1) Fuel Flow and Output Signal from Controller

Fig. 5 illustrates the relationship between the fuel flow and the on-off valve actuating output signal. The fuel flow is, of course, determined by the duty percentage, and it also is affected by frequency.

(2) Visual Observation Results

Fig. 6 shows the visual observation results; Fig. 6a, b and c indicate the results of observation when the air flow is changed. The line of cycle time 0 on the axis of abscissa represents an example where a valve other than the on-off valve is used to control the fuel flow.

In the classification of stability, a flame which flickers in its entirety has been designated as an "oscillating flame"; a flickering flame perceived to have a greater difference between brightness and darkness has been designated as a "strongly oscillating flame." The boundary between these two zones is not definite.

The visual observation results have indicated that when frequency of approximately 40 Hz or more is used for the output signal, stable combustion is obtainable. However, the blow-off zone was sharply increased with an increase in cycle time, and the engine was in the idling condition in the vicinity of the fuel flow at 3g/second. As this state is frequently utilized, improvement was needed.

Fig. 7 shows test results in the case where a rubber tube is used in the pipeline between the on-off valve and the fuel injector to decrease fuel pulsation. Use of the rubber tube has greatly improved the characteristics of blow-off.

(3) Measurements of Pressure in Combustor

Fig. 8 presents the measurements of pressure in the liner. In this figure, shown in $\bigcirc$ is the pressure fluctuation rate (%) defined by the following formula:

$$\text{Pressure fluctuation rate} = \frac{2(P_{\max} - P_{\min})}{P_{\max} + P_{\min}} \times 100 (\%) \dots\dots\dots (1)$$

This chart reveals a trend considerably resembling the visual observation results. A pressure fluctuation rate of zero indicates a stable zone; a rate in the 10's, an "oscillating flame" zone; a rate of 20 or more, a "strongly oscillating flame" zone.

(4) Measurements of Flame Luminosity

Fig. 9a, b and c show the test results on flame luminosity when intermittent injection is performed. In comparison with Fig. 9a, Fig. 9b shows a high value of flame luminosity in spite of the fuel flow having been decreased to 60%. Compared with Fig. 9c, Fig. 9b shows the same fuel flow but indicates that the maximum flame luminosity in intermittent spray combustion is increased two or three times.

(5) Results of Exhaust Emission Analysis

The exhaust emissions CO, HC and NO_x in intermittent spray combustion have been examined. The measurements obtained did not reveal any remarkable changes in any of the three substances.

CONSIDERATION OF CONTINUOUS COMBUSTION IN INTERMITTENT INJECTION

In the gas turbine's combustor, spray combustion and gas combustion

occur side by side. To gain continuous stable combustion, the space density of droplets in each type of combustion must be kept within set limits in the place of combustion. To form the flame, a sufficient supply of oxygen must be furnished and flame propagation between droplets must be possible as well.

As mentioned above, there is a close relationship between the fluctuation rate of pressure in the combustor and the flame stability visually observed. This indicates that intermittent spray causes not only the movement of the flame front but also the cyclic fluctuation of the mass burning velocity. Of course, this fluctuation also results from combustion load intensity. However, the fact that the flame luminosity during intermittent injection is always greater than that during continuous injection indicates not simply the fluctuation of combustion load intensity but also a rapid increase in carbon particles and a change in the flame distribution in the combustor.

The measurements of exhaust emissions have provided us with data indicating that intermittent injection causes a slight reduction in NO_x. Within the test scope, it may be said that intermittent injection only slightly affects exhaust emissions, including CO and HC.

Results pertaining to ignition and blow-off also have been obtained from intermittent injection combustion tests. Improvement in ignition performance by intermittent injection may be due to the easily ignitable air-fuel mixtures produced in the neighborhood of the spark plug, even when the fuel flow is comparatively small, as a result of a fluctuating spray angle caused by the fluctuation of fuel pressure.

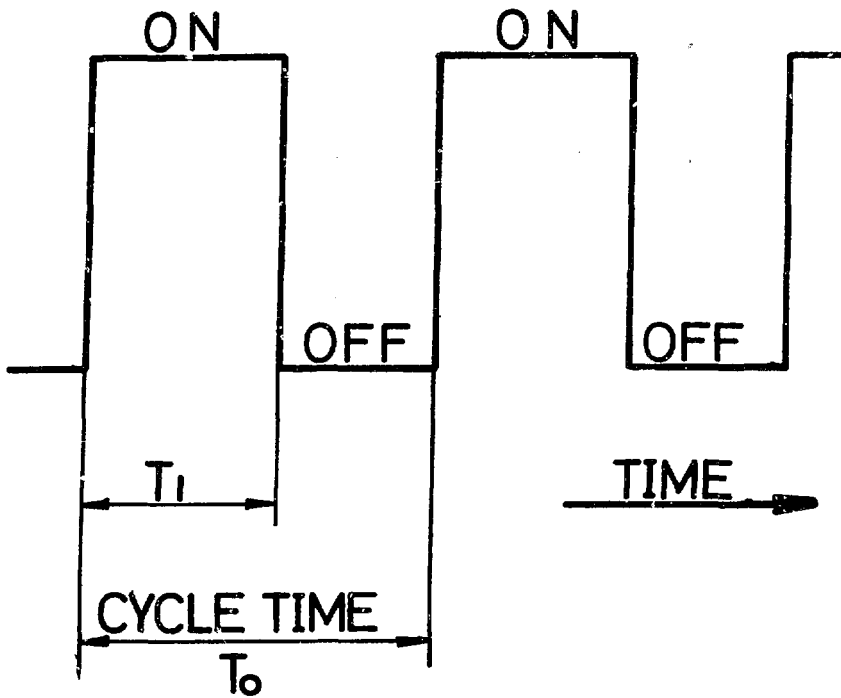
To prevent blow-off characteristics from deteriorating, a rubber tube

installed in the fuel pipeline for damping has proved quite effective and is an important point in enabling the gas turbine to be put to practical use for automotive engines.

CONCLUSIONS

The PWM fuel control system under development by Nissan has a good possibility of practical application with a frequency of more than 40 Hz. Utilization of the PWM fuel control system will greatly improve the ignition performance. By damping the pulsation of the fuel flow, the blow-off performance will be protected from deterioration.

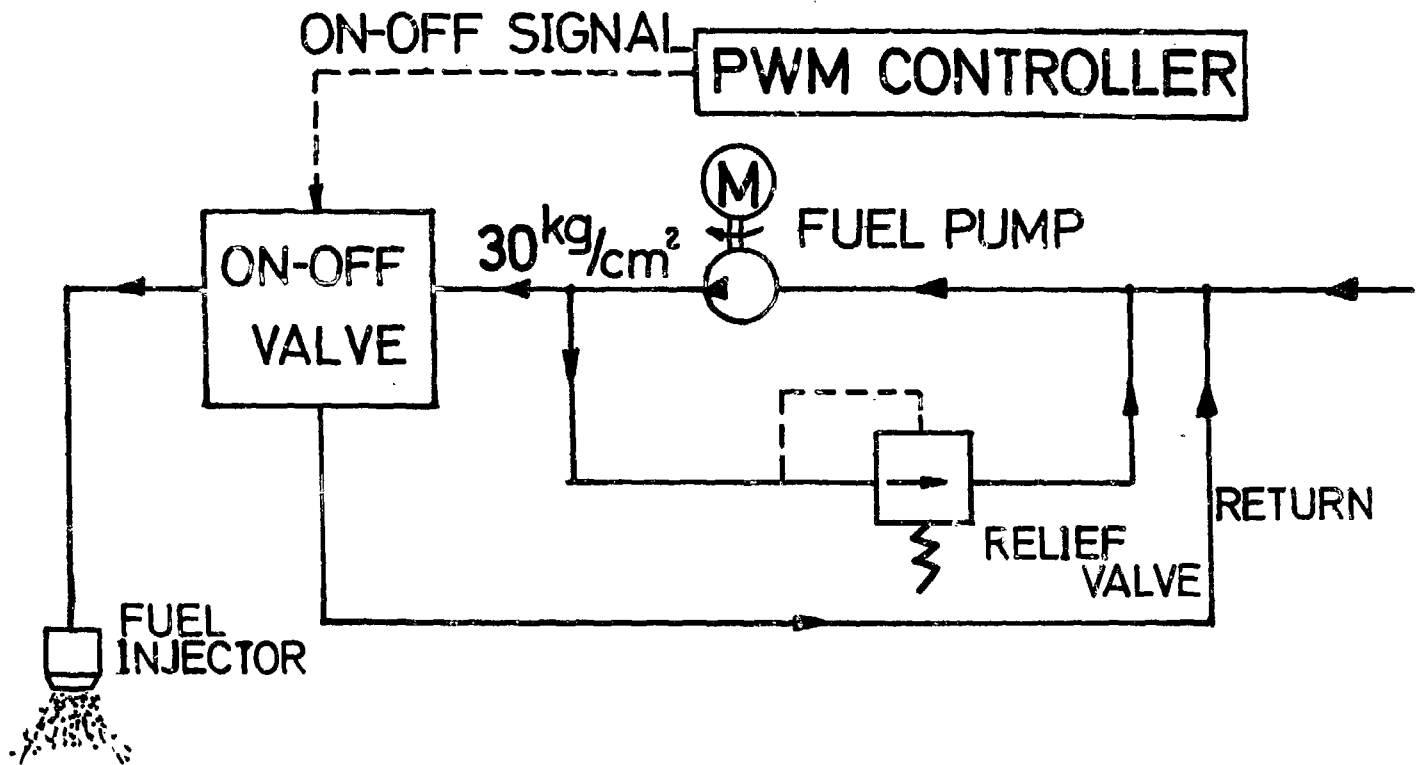
At any rate, use of the PWM control system on the gas turbine will mark one further step toward application of the gas turbine as an automotive engine based upon cost reduction and performance improvement.



$$\text{Duty} = \frac{T_1}{T_0} \times 100 (\%)$$

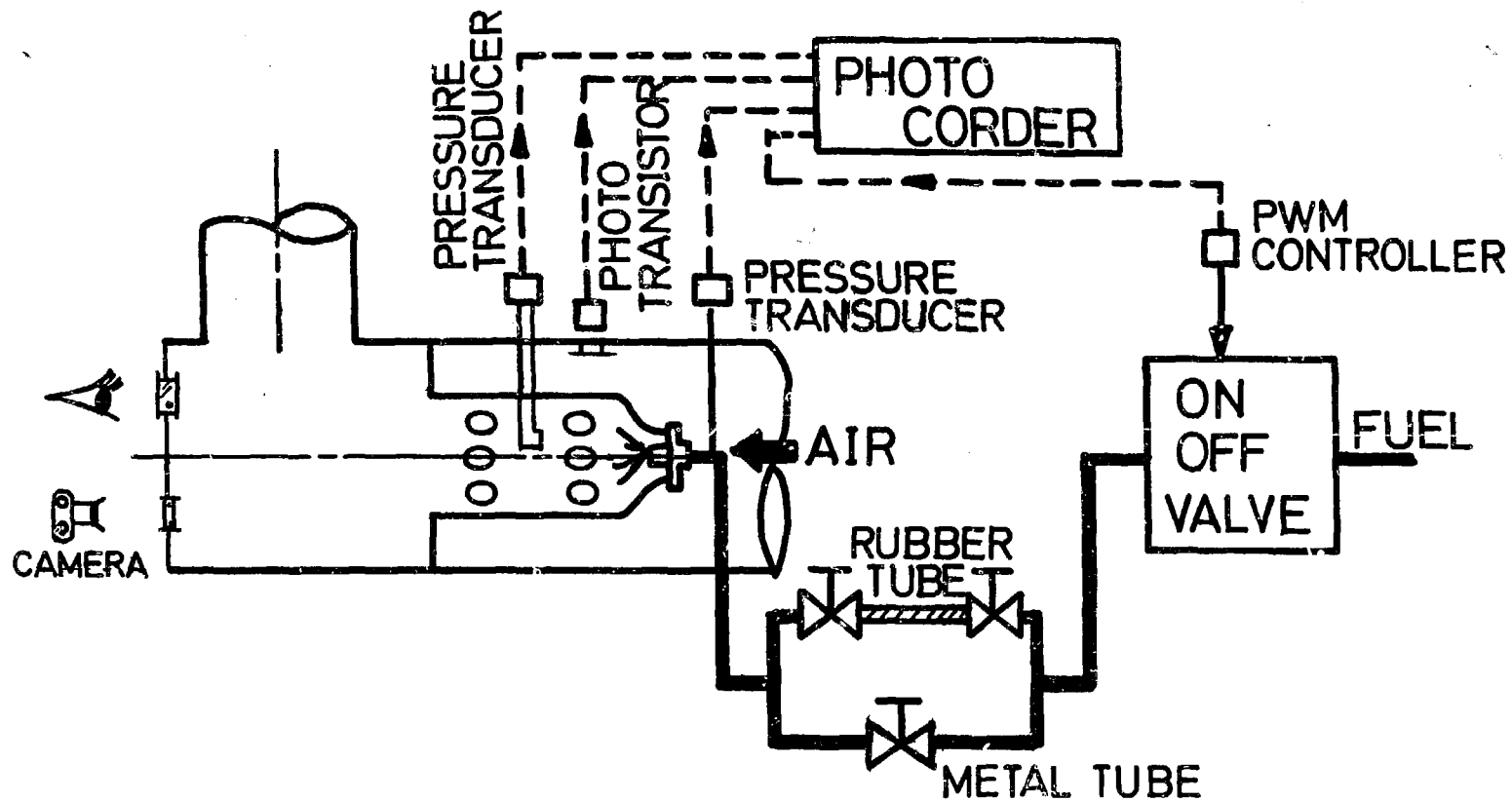
PWM CONTROL

Fig-1



FUEL FLOW SYSTEM

Fig-2



SCHEMATIC OF TEST FACILITY

Fig-3

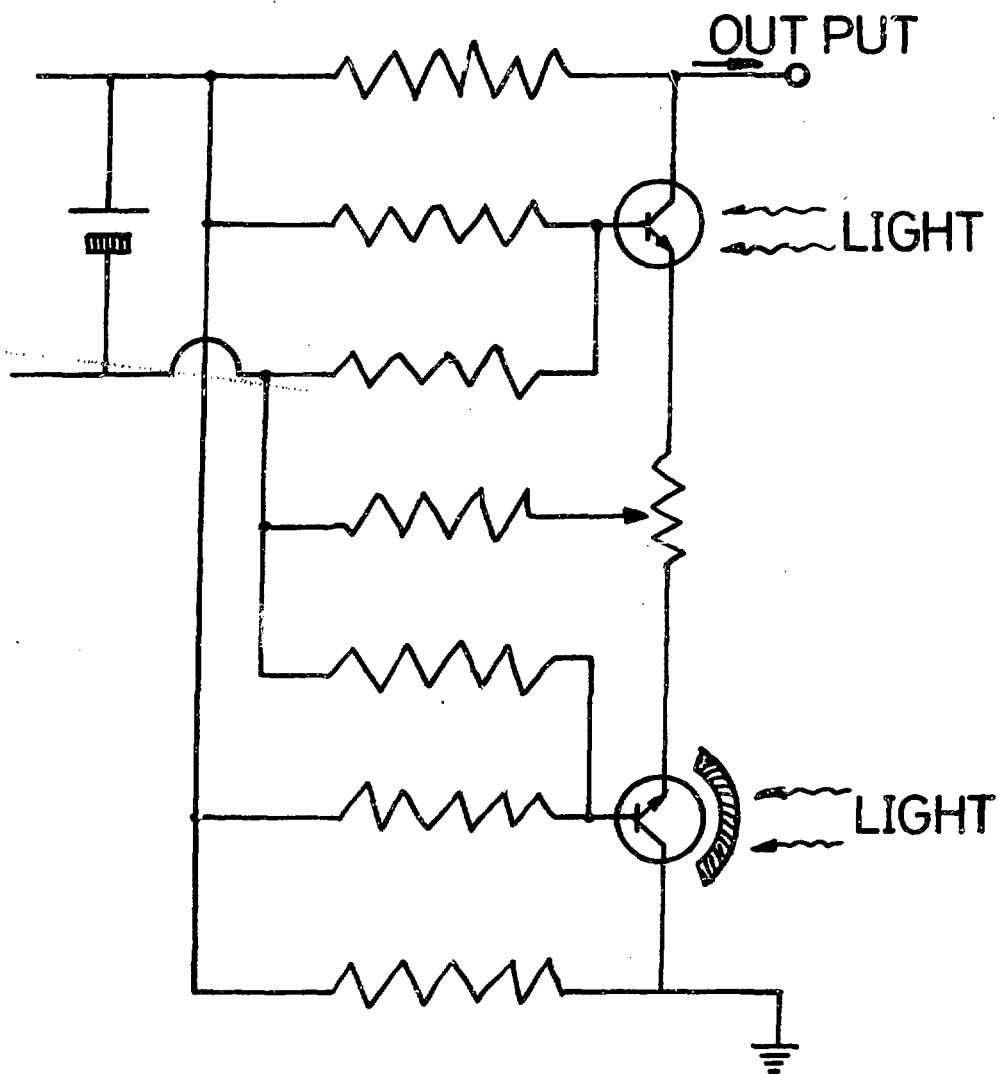
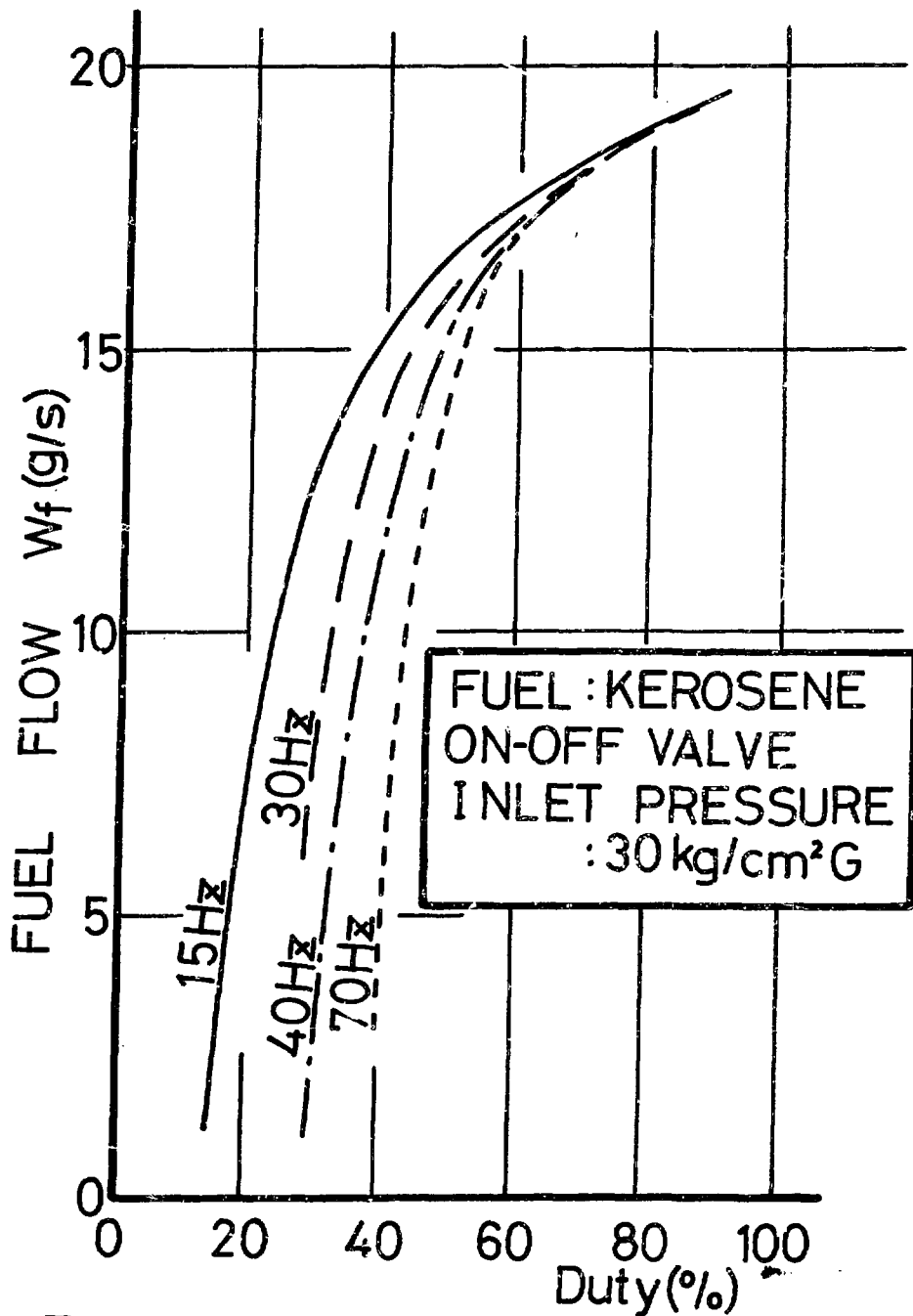
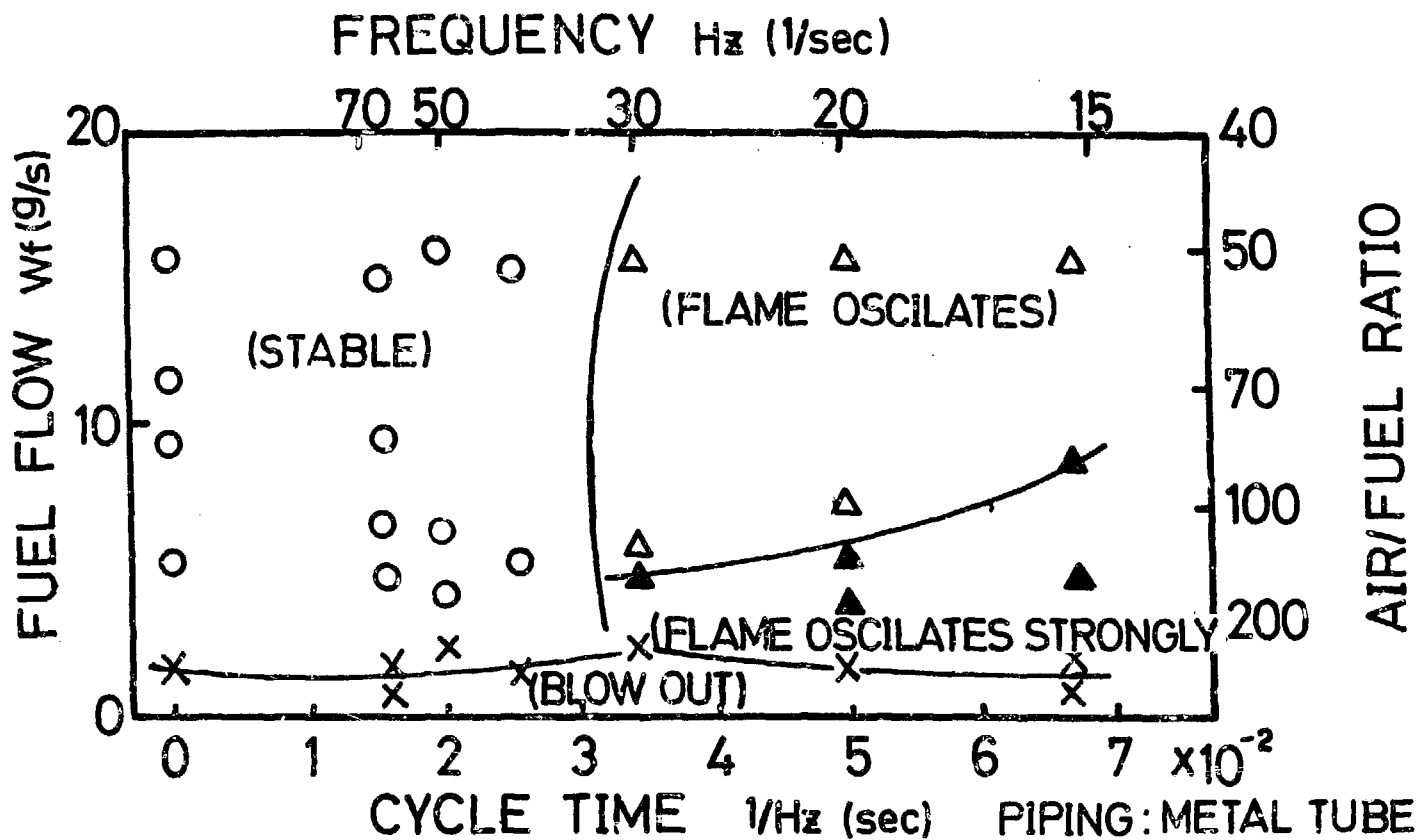


PHOTO TRANSISTOR CIRCUIT
Fig-4



FUEL FLOW BY PWM CONTROL

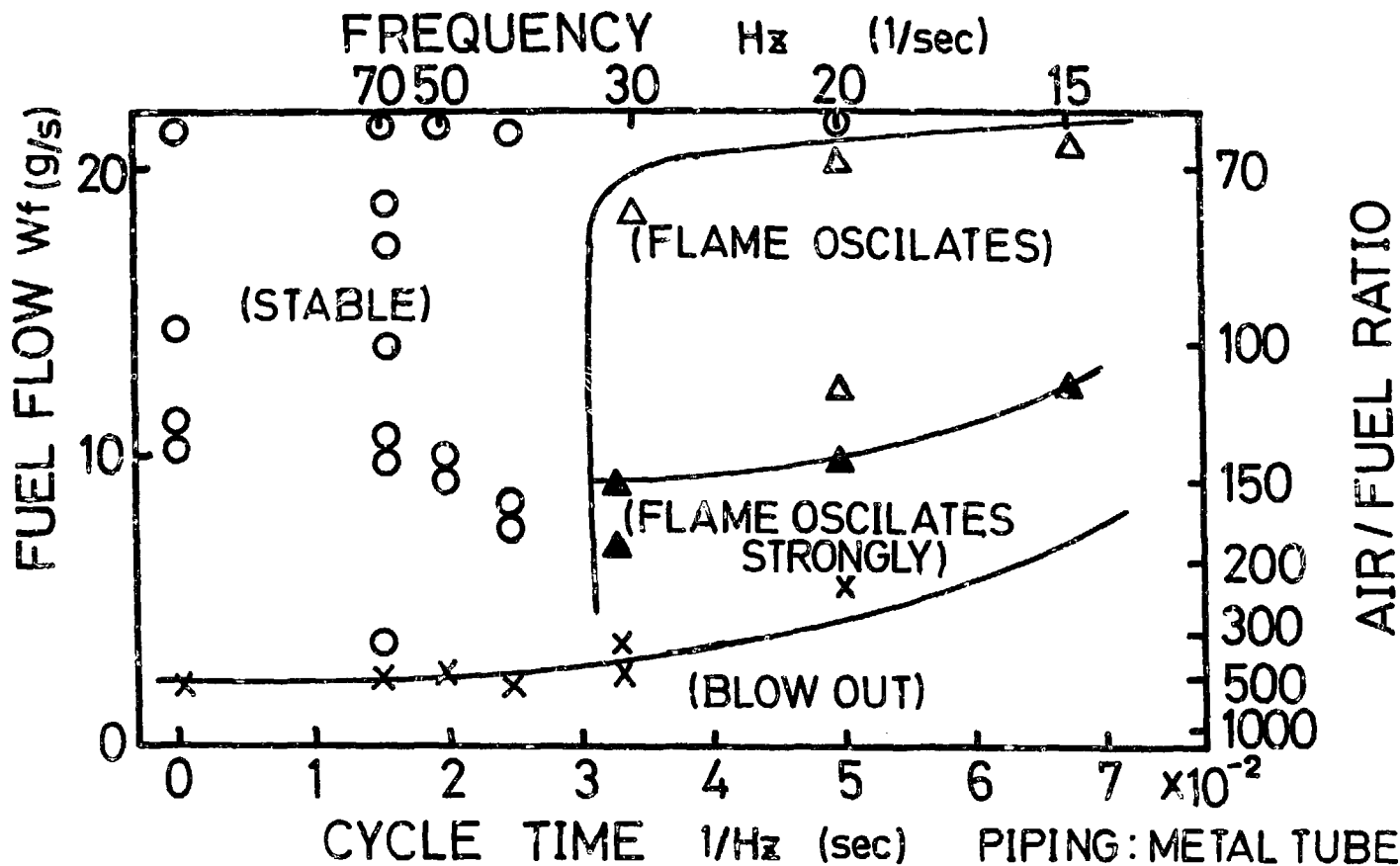
Fig- 5



ON-OFF FREQUENCY AND COMBUSTION STABILIZATION

Fig-6a

(AIR FLOW $W_a = 0.8 \text{ kg/sec}$)



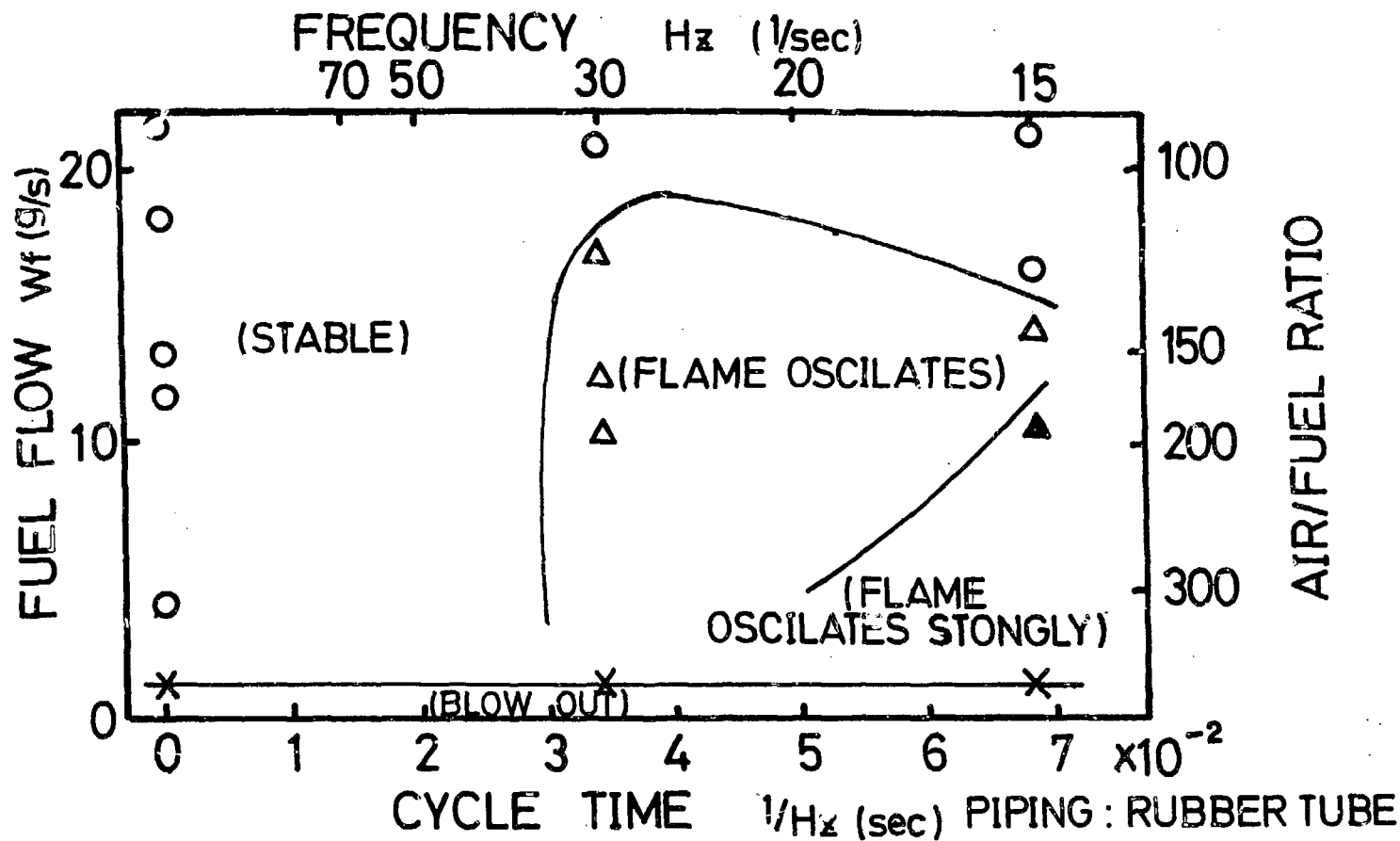
ON-OFF FREQUENCY AND COMBUSTION STABILIZATION

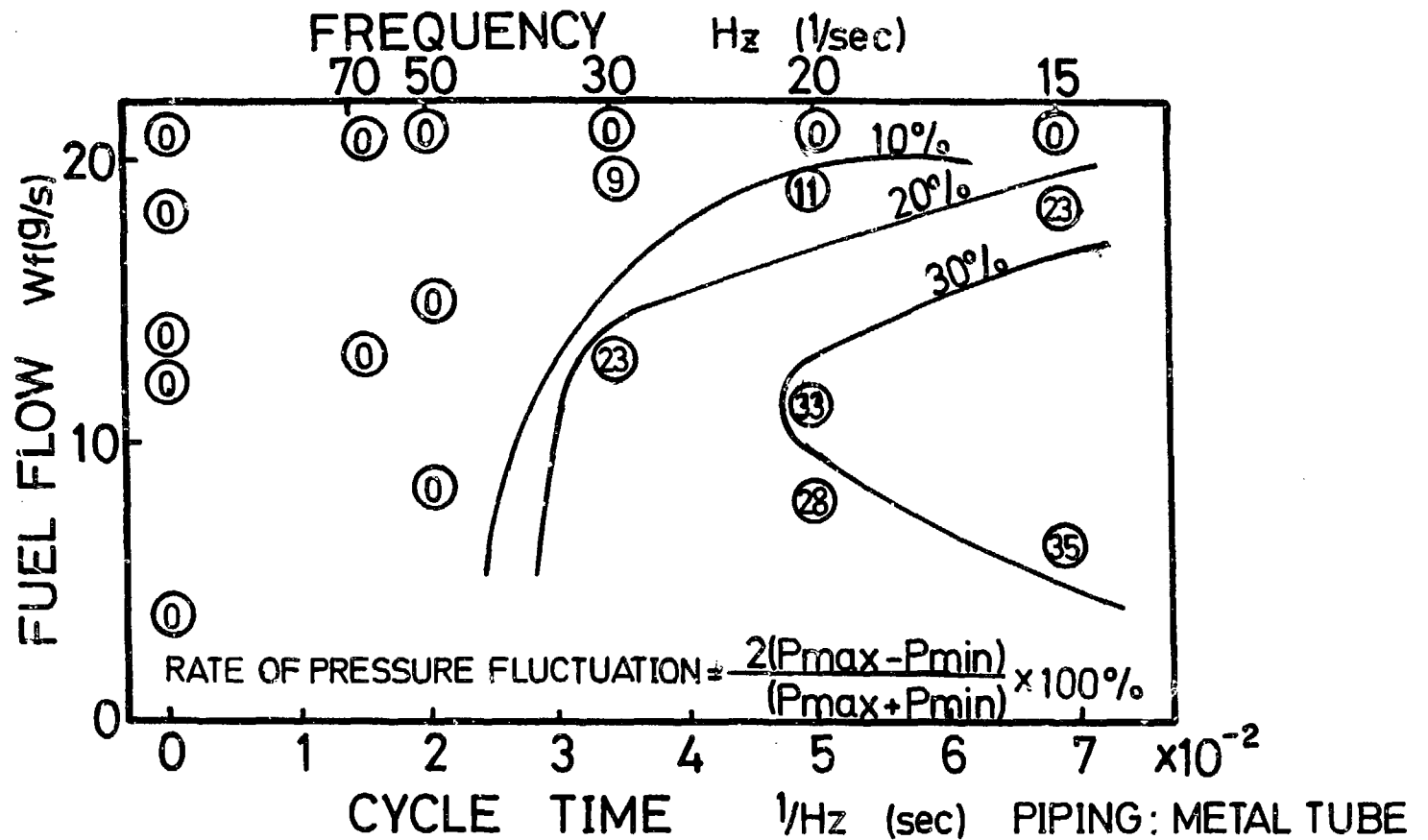
Fig-6b

(AIR FLOW $W_a=1.4\text{kg/sec}$)



(AIR FLOW $W_a = 2 \text{ kg/sec}$)





ON-OFF FREQUENCY AND RATE OF PRESSURE FLUCTUATION
IN A COMBUSTOR

Fig-8

(AIR FLOW $W_a = 2 \text{ kg/sec}$)

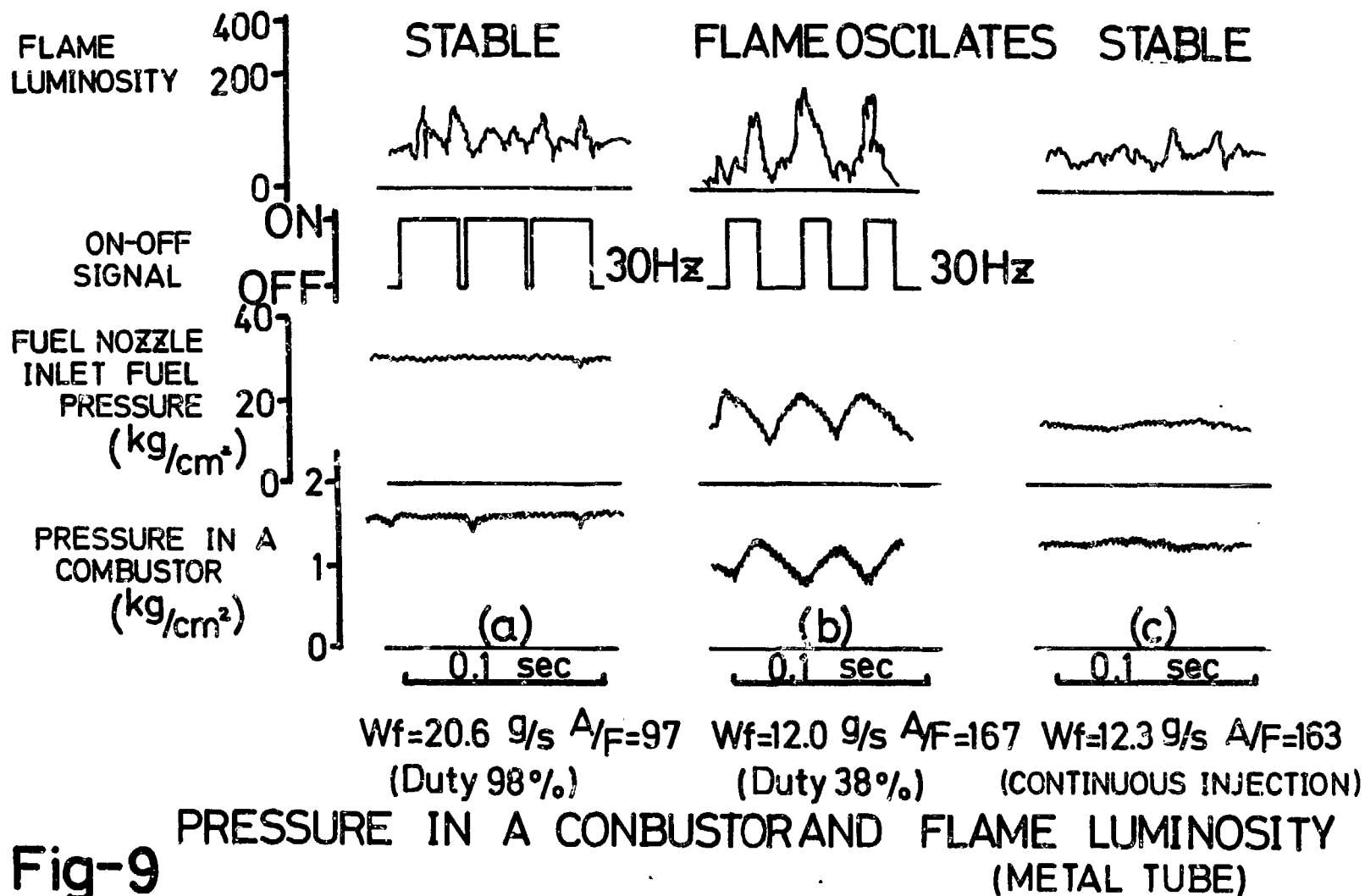


Fig-9

PRESSURE IN A COMBUSTOR AND FLAME LUMINOSITY (METAL TUBE)

Session 3b

THREE SHAFT TURBINE-TRANSMISSION SYSTEMS
FOR VEHICULAR APPLICATIONS

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During the years a number of different gas turbine types have been designed and tested for vehicle applications. Each one has been aimed at better characteristics and lower price. At United Turbine in Sweden, a subsidiary to Volvo AB, a gas turbine system, called the KTT system is being developed. (Ref. ERDA Ninth Summary Report, Nov. 17, 1975, page 96-117).

Recent publications have revealed some of the elements of the KTT concept. This system was conceived primarily for vehicular applications. Special emphasis has been on resolution of the chronic problems associated with small vehicular gas turbines: high part load and idle fuel consumption, high cost, and acceleration lag. These difficulties have discouraged and retarded early implementation of production programs. The KTT approach appears to have potential to resolve or mitigate these problems to the point where serious consideration should be given to production.

I Objective

The purpose of this paper is; to highlight the versatility of the KTT system, to show how such an engine-transmission system can satisfy the installation requirements of light vehicles, and to indicate its potential for use in multiple engine installations in larger, heavy duty vehicle applications.

Gas turbine advocates have long believed that the inherent compact, light weight, and smooth operating characteristics of gas turbines would ultimately give the gas turbine an important place in vehicle powerplant markets. Numerous

prototype engines have been built and tested. Many more have been designed and analyzed. The important problem areas have been identified; approaches to solutions formulated.

The three shaft KTT system represents a new approach which offers considerable promise for vehicular gas turbine engines. Analysis, design and prototype tests to date on the three-shaft KTT system show it to have a remarkable compatibility with the requirements for small vehicles such as low idle and part load fuel consumption, rapid acceleration and low cost.

II Review of the KTT System

A close integration of the three-shaft turbine engine with the car transmission functions is the basic idea of the KTT system. The configuration involves three independent turbine stages (Fig. 1). The first stage drives the compressor (gas generator rotor); the second stage drives the output shaft and thus the load. This is the same principle as in a two shaft engine. The third stage, the auxiliary turbine, drives a planetary gear system, and, through this, can deliver power to both the gas generator rotor and the output shaft.

The key features of this system are:

- . The three-stage turbine design for a given rated power and pressure ratio results in reduced size, weight, inertia, and tip speed of the turbine wheels. This in turn results in reduced stress and overall package size. It can also open the road for early introduction of ceramic rotors (Fig. 2).
- . Alternatively the three stages can be used for a higher design pressure ratio without increasing the stresses or sizes of the turbine stages compared to a two shaft engine. A higher pressure ratio results in a substantial improvement in part load fuel economy (Fig. 3).

- . The hydraulic torque conversion function (where conversion losses are rejected through the oil cooler) is eliminated from the transmission and put into the turbine system. The aerodynamic torque conversion accomplished by the free power turbine and the auxiliary turbine (acts as a rotating exhaust diffuser for the power turbine) make possible stall torque ratios over 4:1. Conversion losses are in this case recovered by the auxiliary turbine and the regenerator.
- . Use of variable geometry with the three-shaft KTT system also makes it possible to keep TIT (Turbine Inlet Temperature) close to optimum levels at part load and idle. Since changing the power split between the three turbine stages allows a certain freedom when designing the compressor turbine and power turbine, these can be designed for high efficiency at low load. This plus the higher loss recovery as per above all act to reduce the idle and part load fuel consumption.
- . The lower inertia compressor turbine wheel increases acceleration capability of the gas generator. This in turn permits a lower idle speed without sacrificing the time to accelerate to a certain speed. This then further decreases idle fuel consumption.
- . The gear arrangement also permits an immediate transfer of torque and inertia energy from the auxiliary turbine rotor system to the gas generator shaft if some sort of variable drive is connecting these two systems.

In addition, if this optional use of a continuously or step-wise variable drive between the auxiliary turbine and the power turbine is combined with the planetary gear system, reaction torque will be supplied to the output shaft during acceleration.

- . The close integration of turbine-transmission functions results in an overall simplification, and reduction in size and weight of the entire power system package. This, in turn, enhances the installation flexibility of the system. The schematic illustration of Fig. 4 shows the system compared to other suggested turbine transmission systems for cars: a two-shaft gas turbine engine with a 3 speed automatic transmission, and a one-shaft gas turbine with an IVT transmission.
- . The KTT package is about 40% smaller in size than the comparable piston engine-transmission package and also the two-shaft gas turbine-transmission package. Estimates show that this could correspond to more than 10% weight reduction for the entire vehicle.

Not only does the size reduction make it easier to install the engine package in the vehicle and still accomodate different styles and functional designs, but the corresponding weight reduction also makes further reductions in fuel consumption possible.

Regarding the development work, it can be mentioned that component testing was initiated toward the end of 1975. This has included test of sub-systems as well as individual components. Operation of the turbine-transmission functions between the power turbine, auxiliary turbine and the power transfer shaft to the gas generator has been demonstrated. Test and development of the complete prototype engine was started late in 1976.

The results to date support the performance and operation of the KTT system predicted from the earlier analyses and design studies.

III Installation Studies

Increasing attention is being directed toward vehicle installation investigations. The initial prototype engine development is being done on a version of the KTT engine called Mark I. To ease development problems, this engine has two regenerator cores, one on each side. This will, among other things, provide a more symmetrical temperature distribution in the engine. The Mark II engine is a single regenerator version of the KTT system. It represents a projected future production engine. Vehicle installations for these two engines are discussed here.

A. Mark I Engine In Typical Compact Vehicle

The general arrangement and gas flow path of the Mark I engine has been shown in Fig. 1. Figures 5, 6 and 7 show the engine in the engine compartment of a typical compact automobile.

In general this car is in the 3000 lb weight class with 104- inch wheel base and suitable for four passengers. The standard engine for the car has 130 cu. in. displacement, and develops 100 HP at 5250 rpm. This engine plus automatic transmission and radiator weighs about 520 lbs.

The Mark I in its initial version will develop about 125 HP and weigh about 390 lbs. including the transmission. According to preliminary predictions, fuel economy should be improved up to 20% relative to a comparable piston engine.

B. Mark II In Typical Compact Vehicle

The Mark II engine is expected to be a production version of the KTT system. It has been designed specifically for a compact car or a light, high performance vehicle.

Special attention was given to reduced manufacturing cost, lower installation volume, and increased installation flexibility. Either front-, rear-, or four-wheel drive can be used. Because the KTT turbine-transmission system is so short, it can be mounted with the turbine rotor axis perpendicular to the vehicle axis (cross-wise) and the regenerator toward the passenger compartment. The burner and accessories are then easily accessible from the front; the engine shape nicely fits within a low hood contour.

Figure 8 shows a mock-up of the engine in a compact vehicle engine compartment. Figures 9 and 10 are installation sketches showing the Mark II engine in a conventional rear-wheel (optional four-wheel) drive arrangement and in a front-wheel drive arrangement respectively. The vehicle shown is the same as described before.

Figure 11 shows the engine in a four-wheel drive, jeep-type vehicle.

The Mark II engine-transmission, will be rated in the 75 to 100 HP range and weigh about 280 lbs. This corresponds to 10% reduction in vehicle weight. This permits additional reductions in size and weight of vehicle and, in turn, reductions in power required to drive the vehicle. This all results in further improvements in fuel economy and emissions.

IV Future Installations

The KTT system really represents a family of turbine-transmission power packages (Fig. 12). The basic elements of the system can be optimized and combined to match the specific applications involved. With this flexibility applications can range from single shaft electric generator sets, to marine propulsion systems, to small passenger cars, to heavy military vehicles.

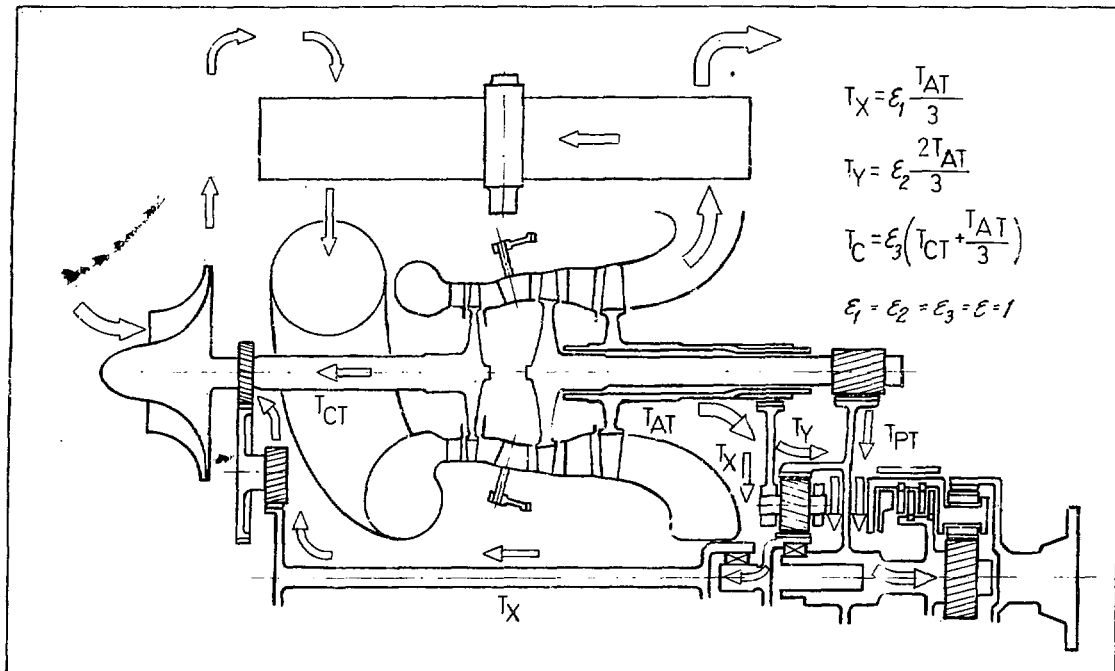
Because of the small size and high power density of the basic KTT system, the growth in application development can be pursued in another direction -- multiple or dual power packages.

The possibility of using dual KTT gas turbine engines in heavy duty trucks and buses is illustrated in Fig. 13. Again the small size and flexibility of the KTT package leaves more space for additional payload. Simultaneously the high power density gives improved vehicle performance.

V Conclusions and Outlook

Preliminary assessment of the KTT system for installation in compact vehicles continues to support the conviction that gas turbines will ultimately have an important impact on the vehicle market.

Projections of possible market growth (Fig. 14) indicate significant penetration of the military, heavy duty, and industrial vehicle markets by the 1985-1990 time frame. Important penetration of the light vehicle and passenger car market will follow by about five years.



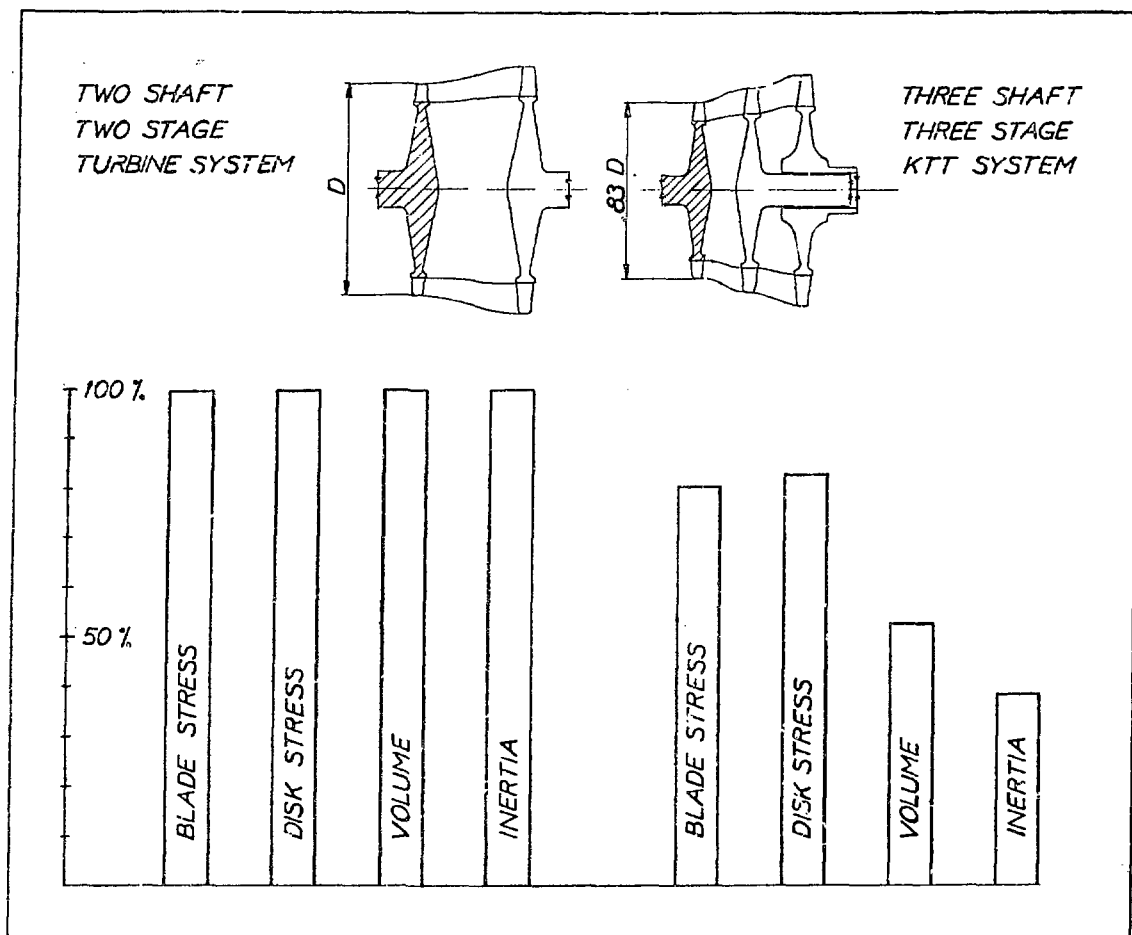
T_{CT} = Torque - Compressor Turbine

T_{AT} = Torque - Auxiliary Turbine = $T_X + T_Y$

T_{PT} = Torque - Power Turbine

ϵ = Gear Ratio

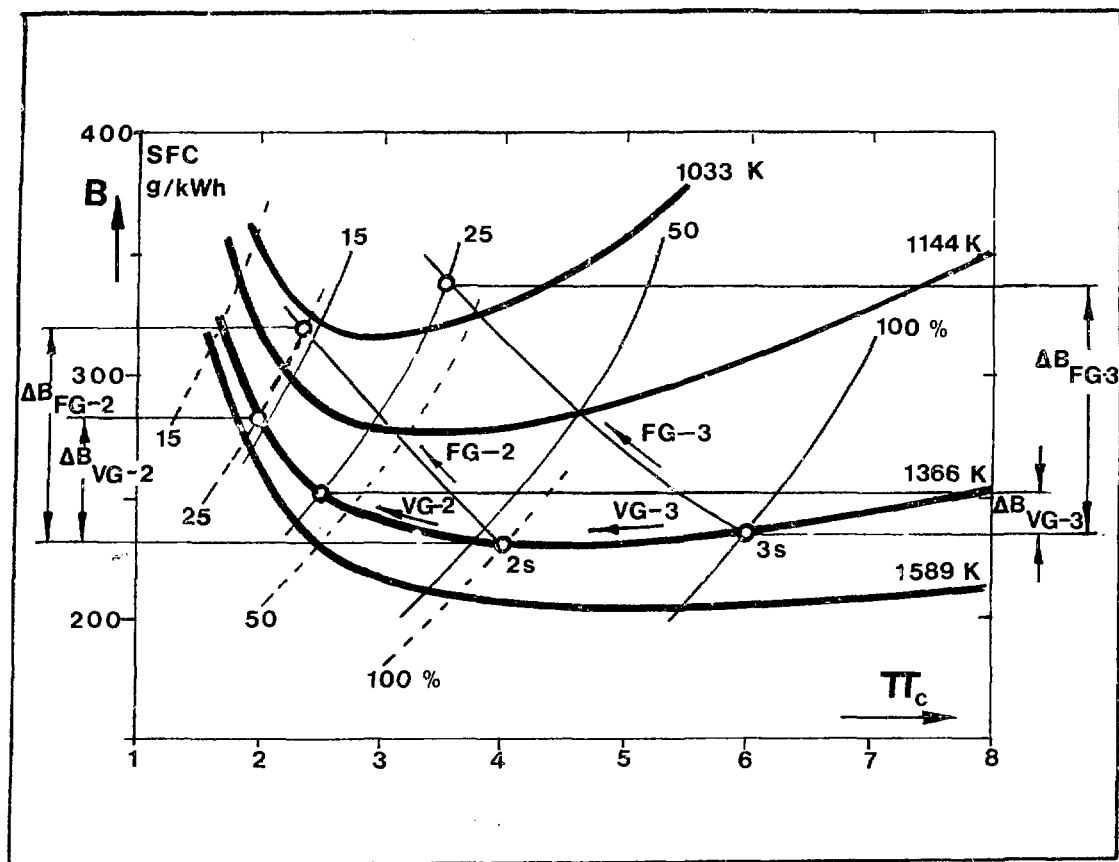
Fig. 1 Schematic Drawing of Three-S' aft KTT System with Single Regenerator



Two-Shaft Turbine
Rotor System

Three-Shaft KTT Turbine
Rotor System

Fig. 2 Relative Dimensions, Stresses and Inertia for
Rotors in Two-Shaft and Three-Shaft KTT Systems



———— Three-Shaft Characteristic

----- Two-Shaft Characteristic

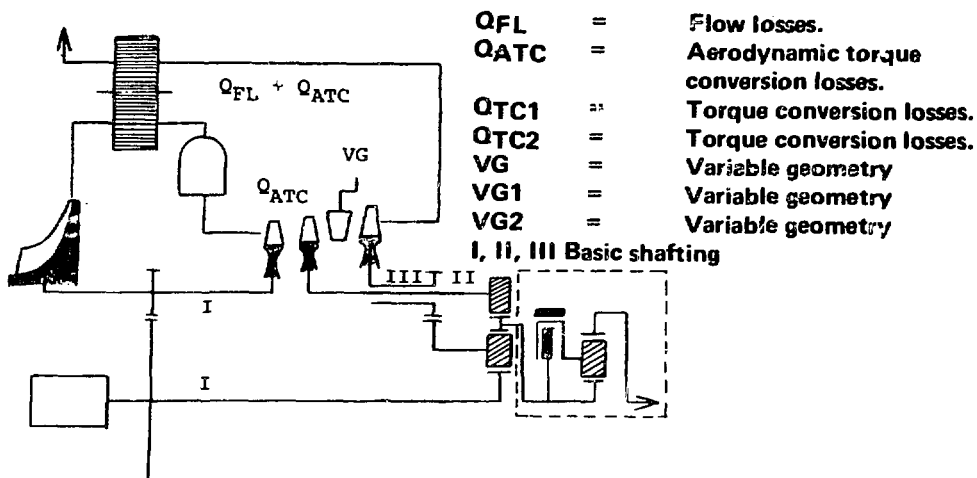
VG-2;VG-3 = Variable Geometry, 2- and 3-Shaft Machines, Respectively

FG-2;FG-3 = Fixed Geometry, 2- and 3-Shaft Machines, Respectively

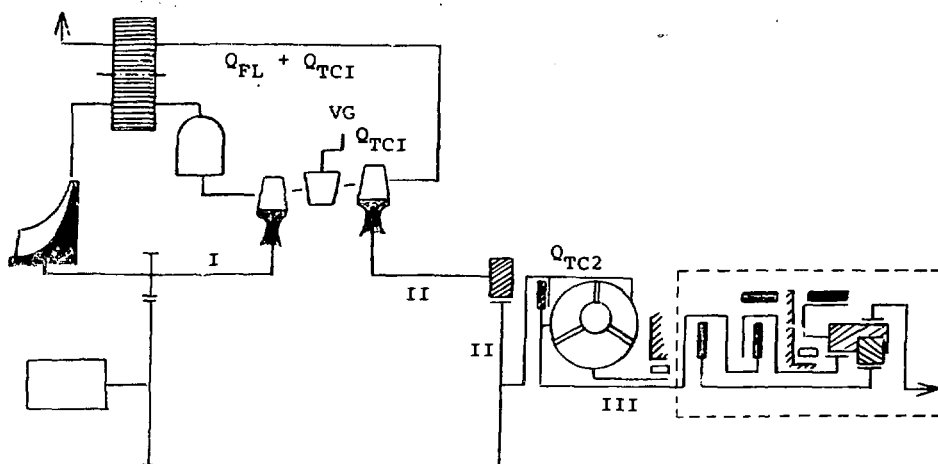
B = Specific Fuel Consumption

π_c = Compressor Pressure Ratio

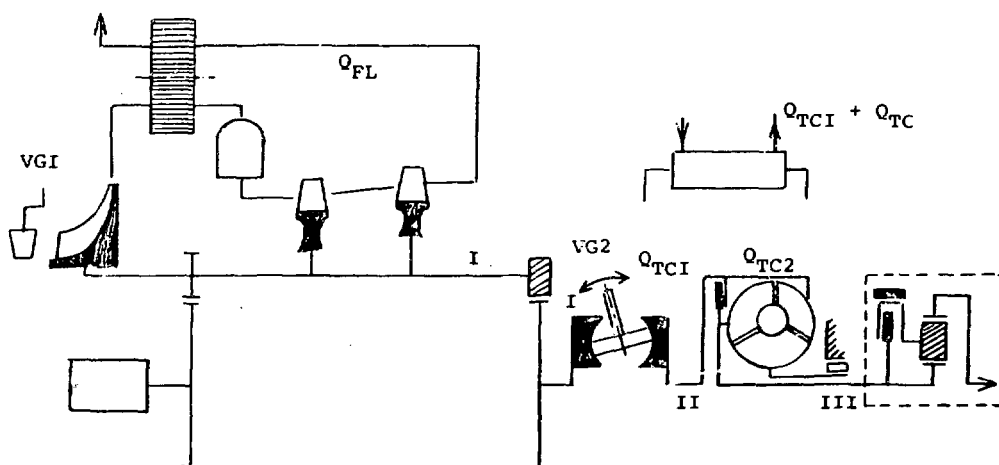
Fig. 3 Influence of Design Pressure Ratio, Variable Geometry (VG) and Fixed Geometry (FG) on Part-Load SFC



a. The KTT System representative of a 3-shaft turbine configuration.



b. Typical 2-shaft free power turbine with automatic 3-speed transmission.



c. Single shaft turbine arrangement plus continuously variable transmission.

Fig. 4 Different Turbine-Transmissions for Car Application

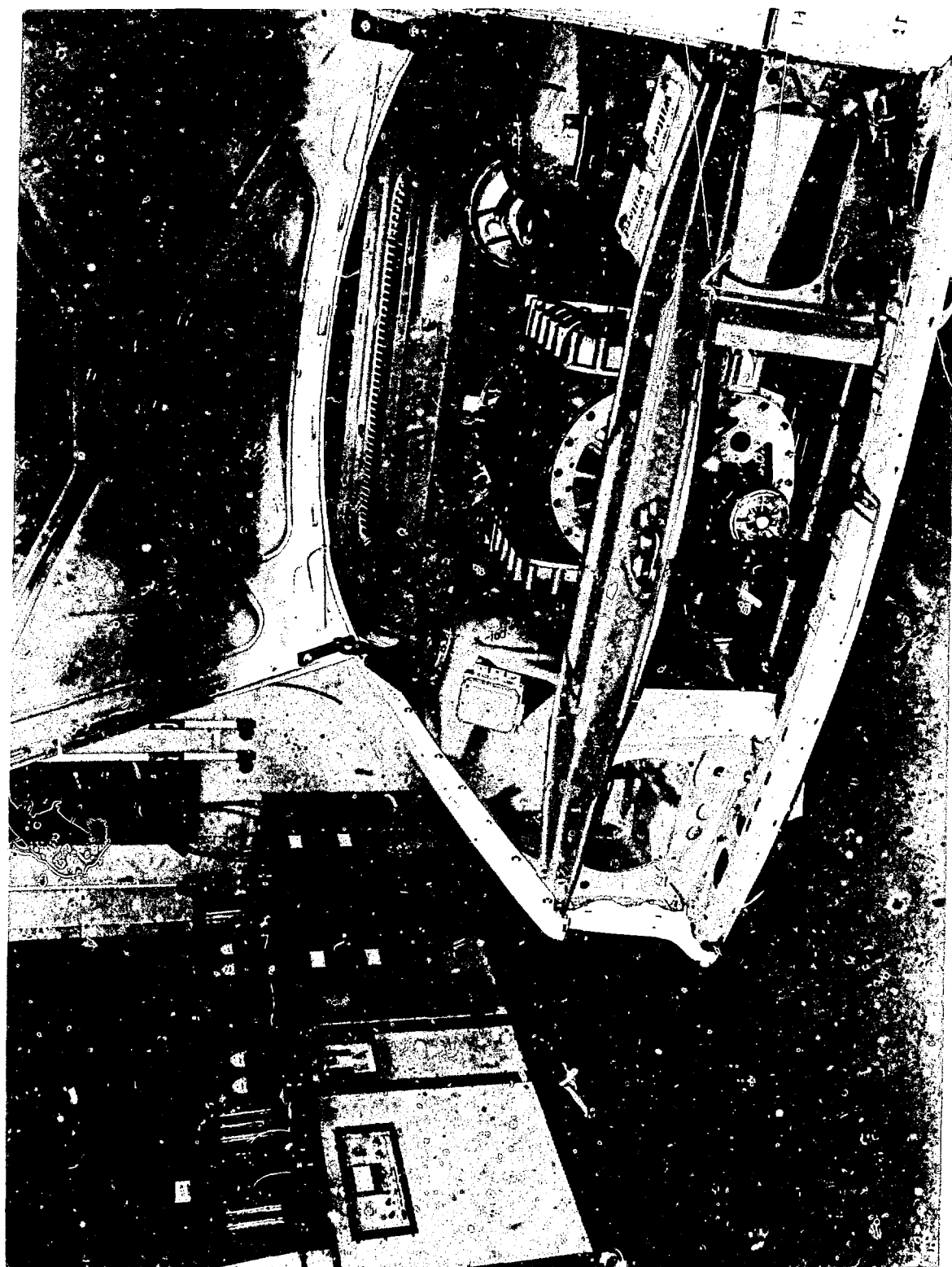


Fig. 5 Mark I KII engine in a Compact Vehicle

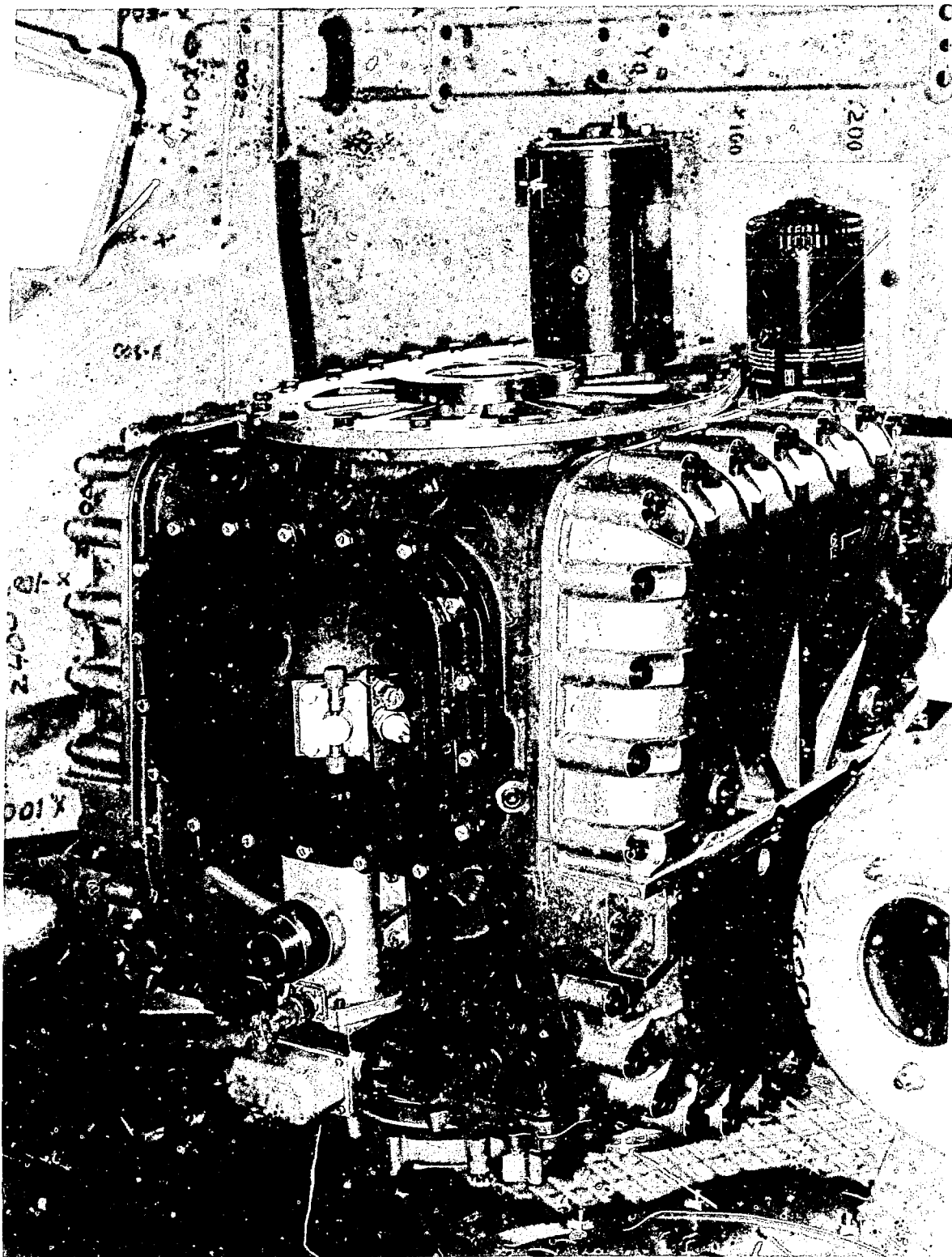
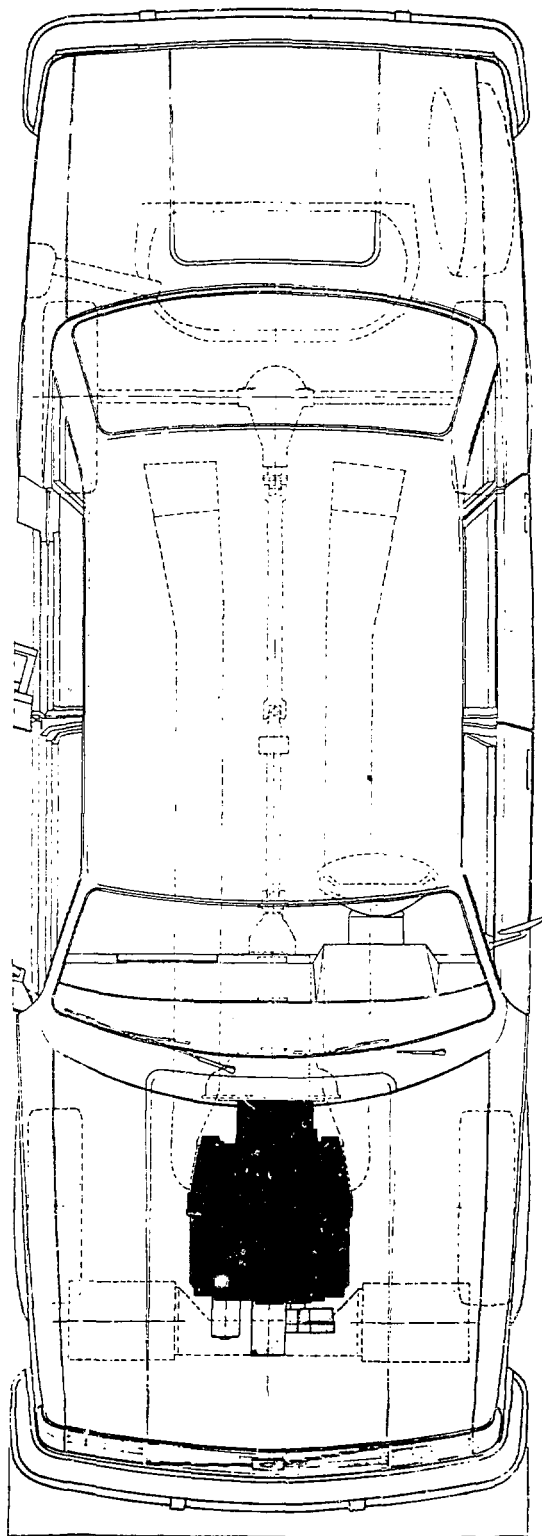
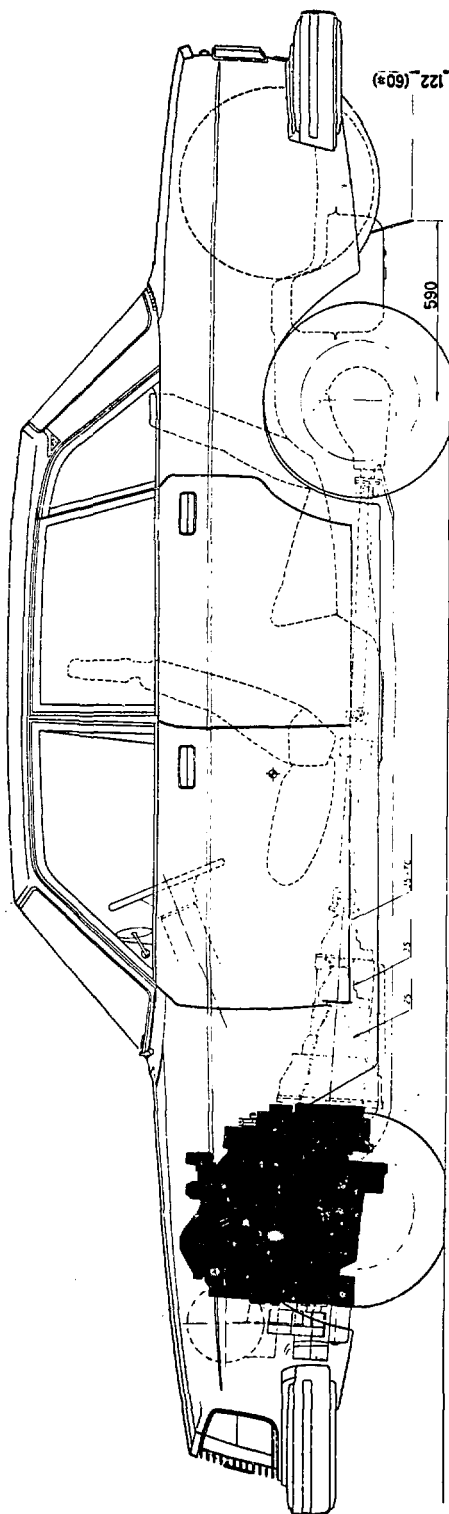


Fig. 6 Mark I KTT Engine in Engine Compartment of
Typical Compact Car

Fig. 7 Installation sketch for 800 Watt 120V



1207



085

122 (60°)

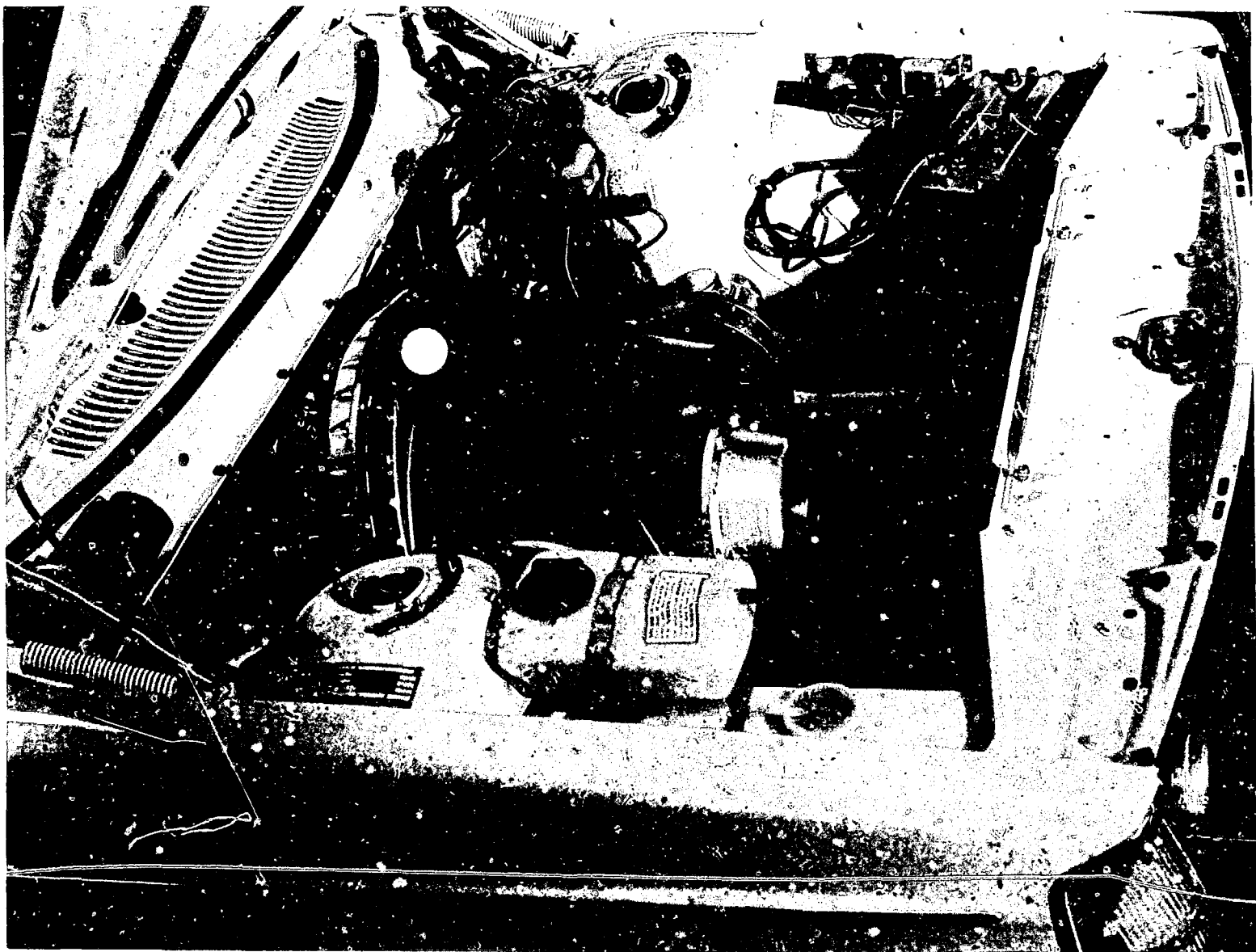


Fig. 8 Mockup of KTT Mark II in Compact Vehicle Engine Compartment

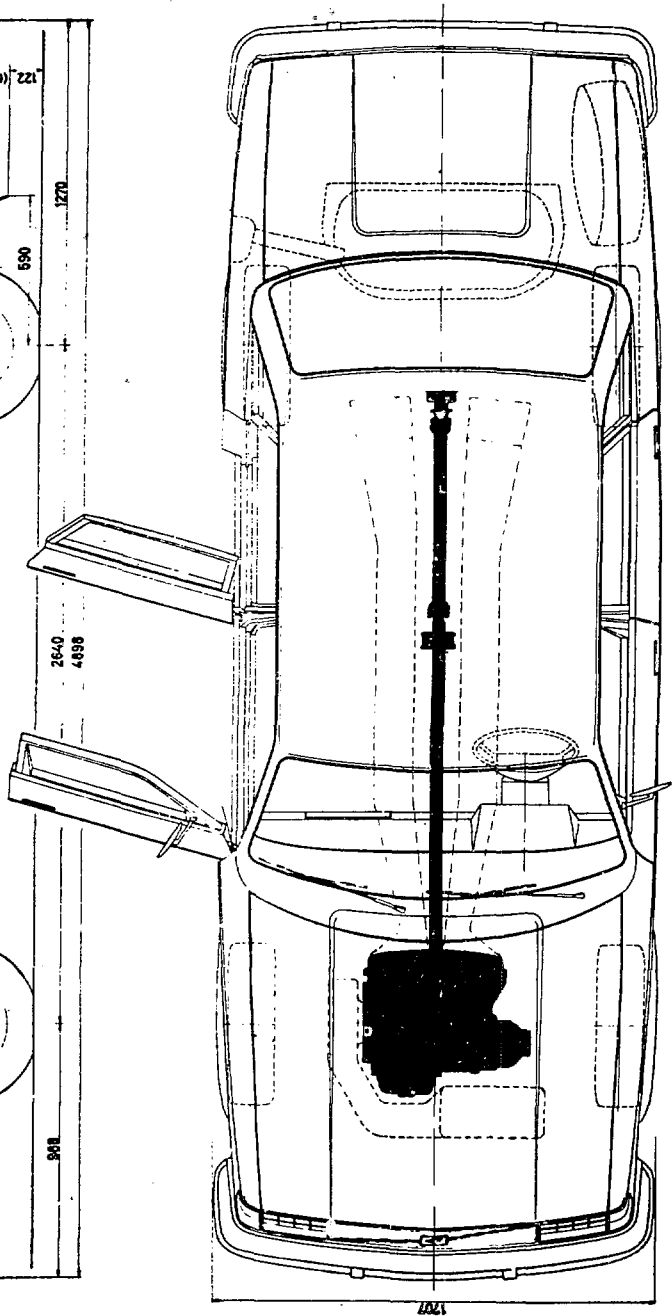
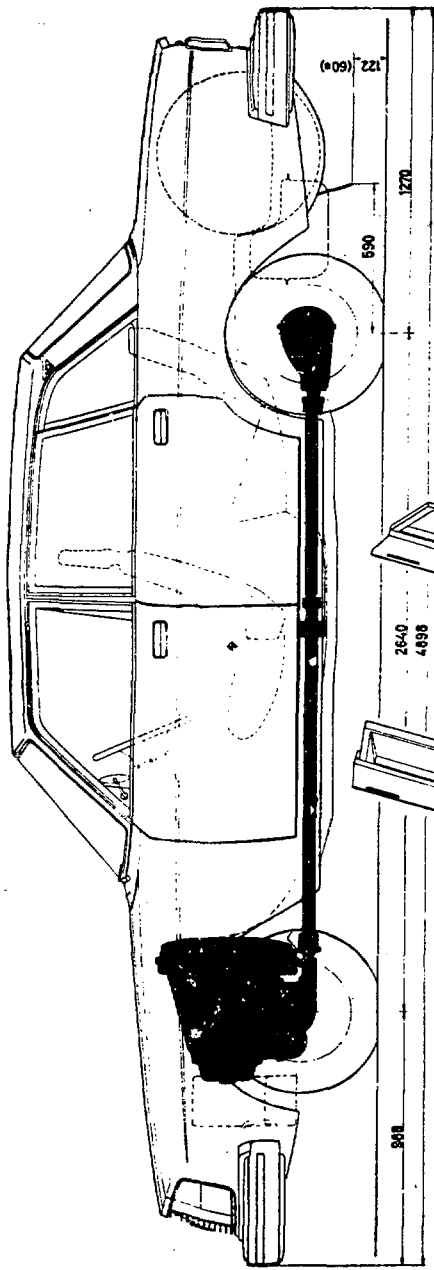


Fig. 9 Installation Sketch of KTR Mark II Engine - rear- or Four-wheel Drive Arrangement.

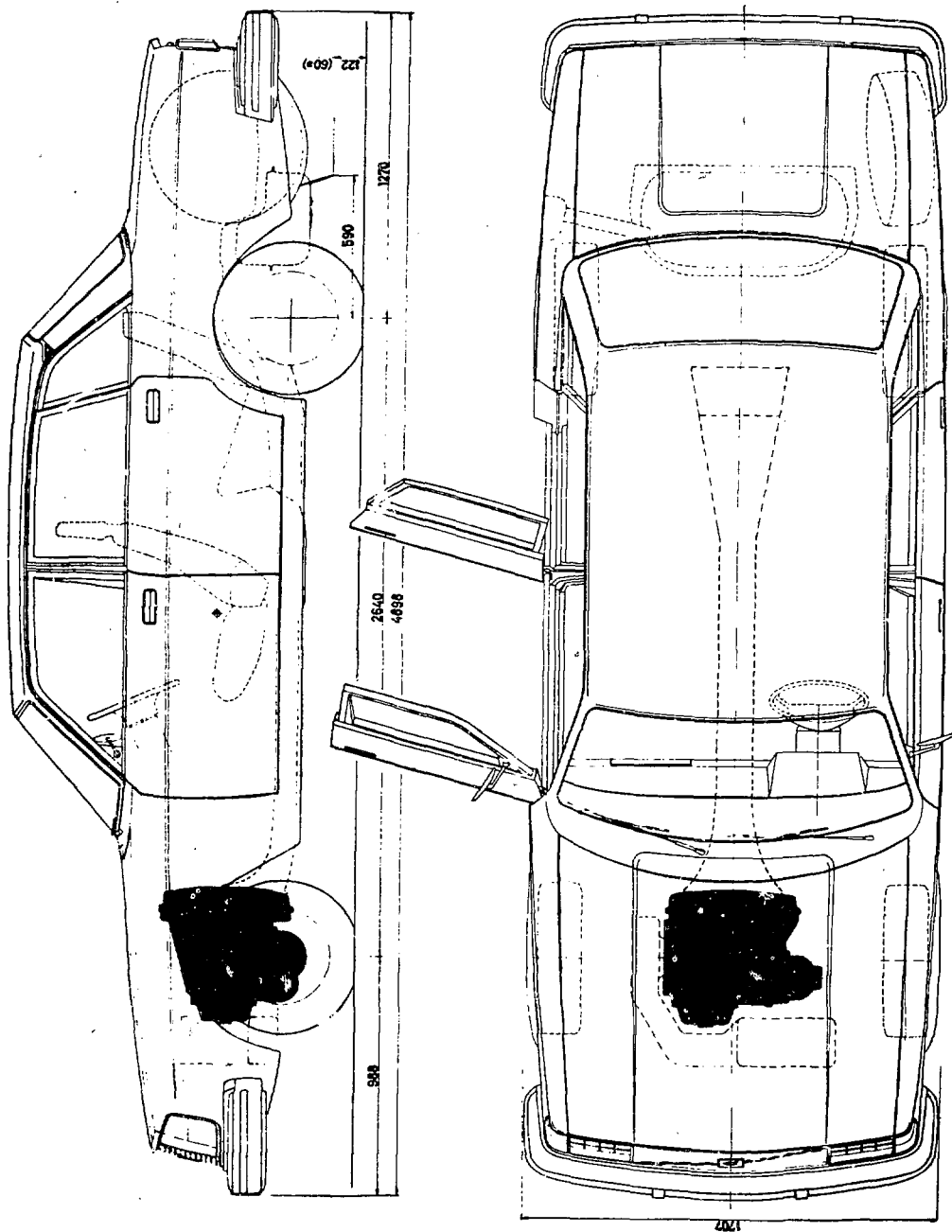


Fig. 10 Installation Sketch Showing KIT Mark II engine in front-wheel Drive Arrangement

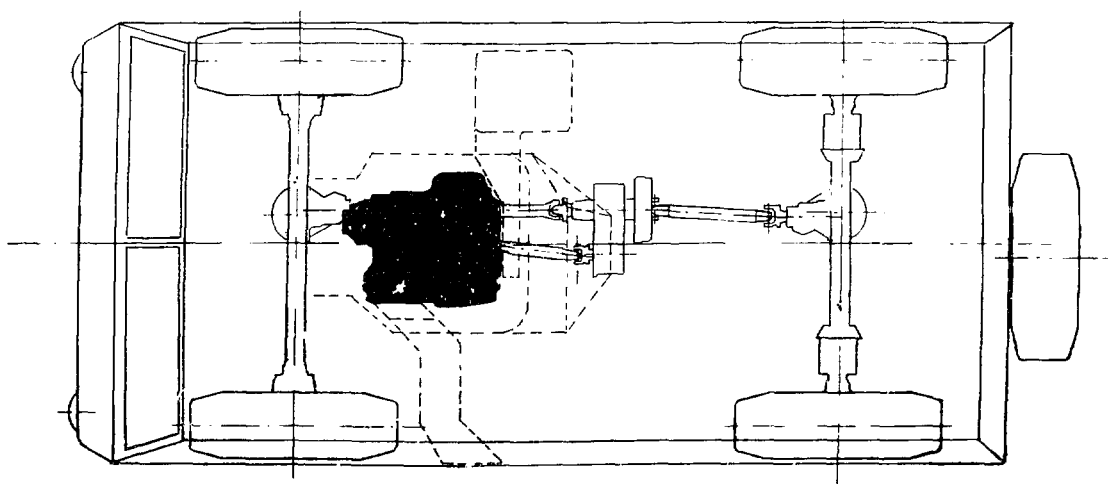
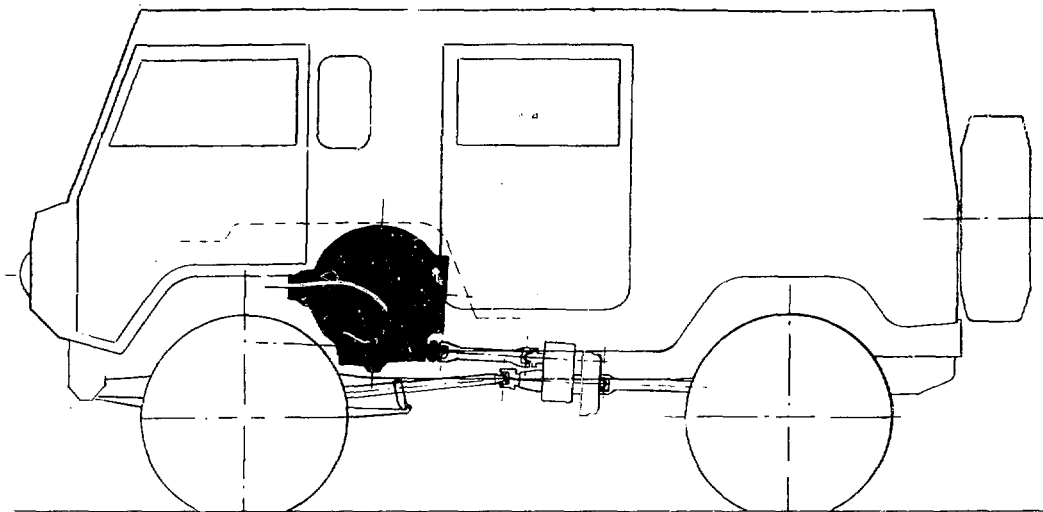


Fig. 11 Installation Sketch of KTT Mark II in a Four-Wheel Drive, Jeep-Type Van

KTT TURBINE FAMILY 3SH, 2SH, 1SH

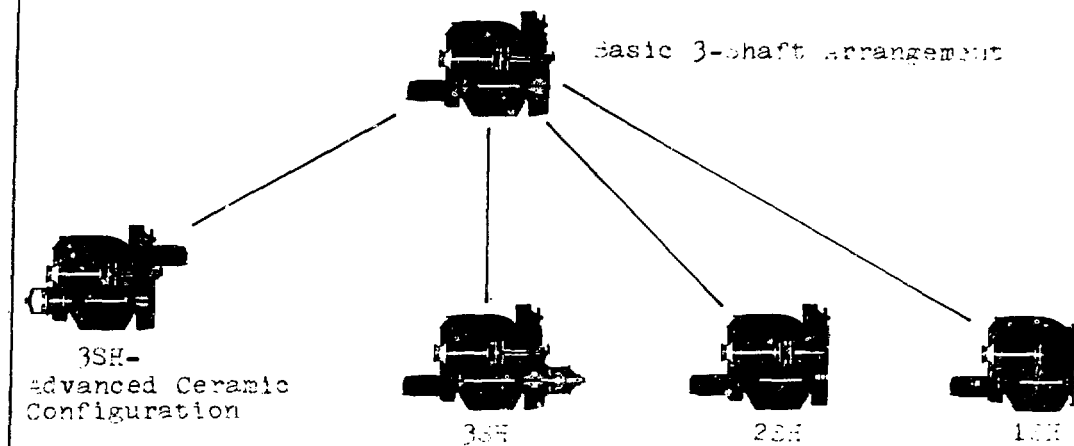


Fig. 12 KTT Family of Power Systems

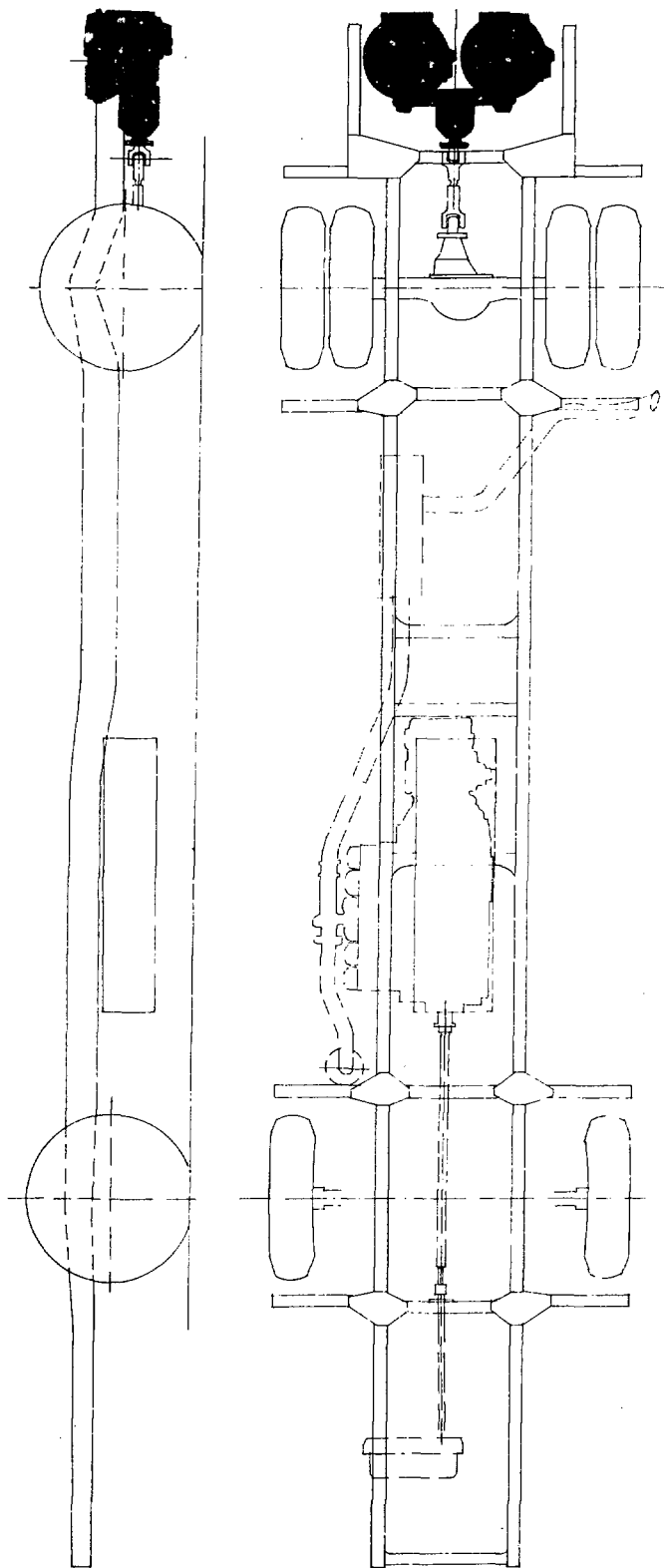


Fig. 13 Dual 750 Power Cylinder in Heavy-Duty Vehicle

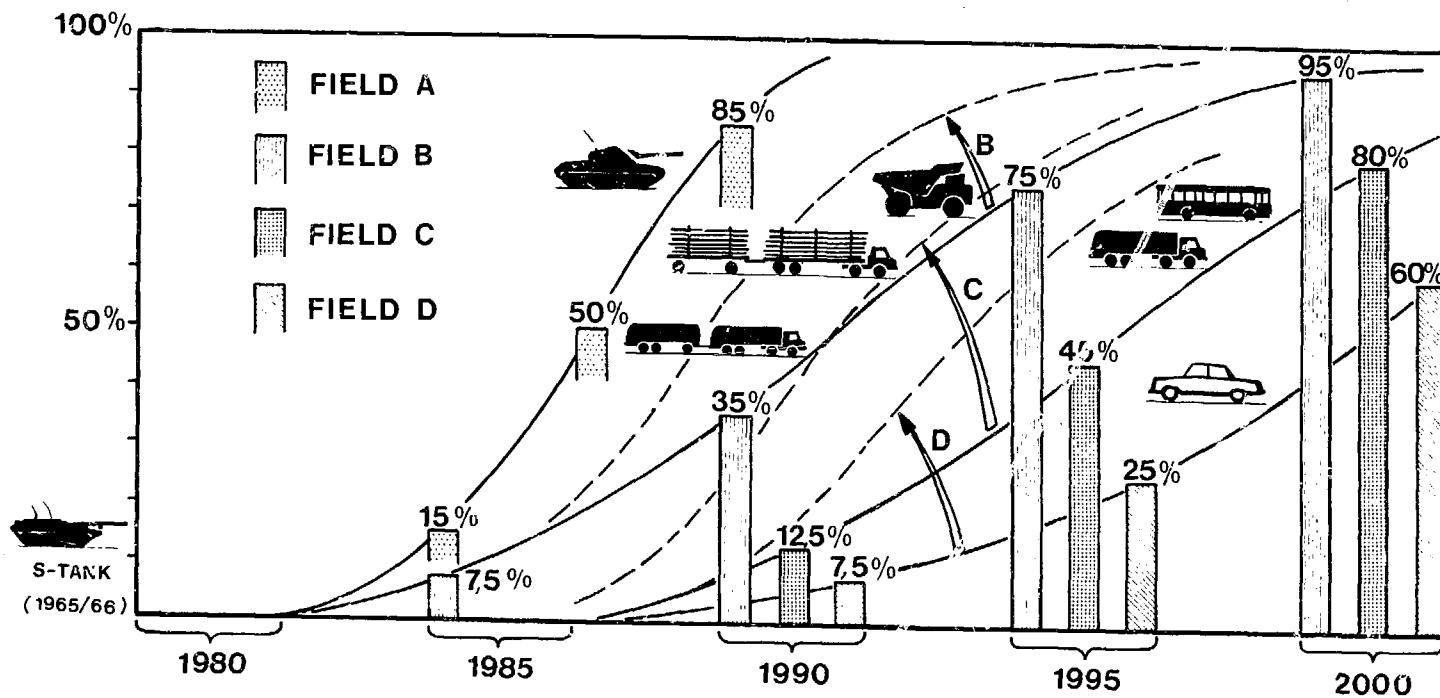


Fig. 14 Projections of Possible Market Growth for Vehicular Gas Turbines

"CERAMIC GAS TURBINE TECHNOLOGY:
MATERIALS FOR TURBINE ROTORS"

By: Arthur F. McLean
Lewis R. Swank
James C. Uy

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INTRODUCTION

This paper reviews some of the general background and highlights of the emerging technology of ceramic turbines and, in particular, discusses some of the material developments associated with the critical, highly-stressed turbine rotor.

Automotive companies and gas turbine manufacturers have R & D programs underway to investigate the application of ceramics for advanced gas turbine engines covering the spectrum of power range from small vehicular turbines to large electricity-producing gas turbines. The motivation to apply ceramics to the turbine is to increase turbine inlet temperature with resulting improvements in engine efficiency and specific power, and to replace costly nickel-chrome superalloys with potentially low cost ceramics. The other attributes of the gas turbine include continuous combustion with low exhaust emissions, multi-fuel capability, potentially lower maintenance, smooth vibration-free production of power, low oil consumption, good cold starting capability and rapid warm-up time. Because of this potential for the ceramic gas turbine, automotive companies, gas turbine manufacturers and ceramic materials companies are active in developing ceramics technology.

Probably the most significant program that has been described publicly is the Ford program which is supported in part by both the Advanced Research Projects Agency and the Energy Research and Development Administration. The overall program objective is to draw together and develop ceramic designs, materials, processes, and test and evaluation methods to establish the usefulness of brittle materials and show this by demonstrating that uncooled ceramic components for an experimental, high temperature, vehicular gas turbine engine can survive 200 hours of duty cycle operation comprising 175 hours at 1930°F, 25 hours at 2500°F and a number of transient starts and shutdowns. In addition, ERDA's current objectives are focussed on continued development of ceramic technology for advanced gas turbine engines.

SUMMARY STATUS OF STATIONARY CERAMIC COMPONENTS

Figure 1 schematically shows the hot flow path of the experimental turbine engine used in the program. All of the high temperature components in this flow path were designed for ceramics. In the case of regenerators, a number of companies had previously investigated the use of ceramics. For example, twin Corning ceramic regenerators made of lithium-aluminum-silicate were used in Ford's industrial turbine program, and had accumulated thousands of hours of operating experience. Under certain conditions, however, the lithium-aluminum-silicate material underwent chemical attack associated with the sulfur content in the fuel or sodium (salt) ingested into the engine. Resolution of this problem now seems well along in that new materials are being developed in a separate program sponsored in part by ERDA. Approximately 300 hours per month are being accumulated on each of several Corning regenerator cores made of a new chemically resistant aluminum-silicate material. To date one such core has survived testing for over 6700 hours and another has been tested for over 2500 hours at 1800°F indicating its suitability for a high temperature gas turbine.

The hot flow path components which are subject to the peak cycle temperature of 2500°F and which are made out of superalloys in today's gas turbine, comprise the combustor, turbine inlet nose cone, 1st and 2nd stage stators, 1st and 2nd

stage turbine rotors, and 1st and 2nd stage rotor tip shrouds. The ceramic nose cone, stators, rotors and rotor tip shrouds shown are of the design D configuration which was the outcome of considerable earlier development. This last D configuration uses common stators and rotors for both turbine stages. While this reduced aerodynamic efficiency, it accelerated ceramic development by providing common parts and tooling; the common tooling was especially important since the stator was injection molded in one piece in contrast to earlier iterations where it was assembled from individual vanes.

As a result of continued developmental iterations, considerable progress has been made on the stationary ceramic flow path components as summarized by the test results shown in Table I which give encouragement to meeting the program objectives for the stationary components.

THE HIGH TEMPERATURE CERAMIC TURBINE ROTOR

While development of stationary ceramics is encouraging, the real breakthrough from superalloys to ceramics rests with success of the ceramic turbine rotor. This is by far the most difficult task in the ARPA/ERDA/Ford or any ceramic turbine program. These accurately and complex-shaped parts are uncooled, yet have to withstand hostile through-gas at temperatures of up to 1371°C (2500°F), rotational speeds of up to 64,000 rpm and thermal up-shocks and down-shocks associated with engine starts and shutdowns.

Preliminary Design and Materials

In the preliminary phase of the program, various combinations of rotor design/material/fabrication processes were considered including:

- . Direct hot pressing of an integral silicon nitride rotor.
- . Ultrasonic machining of a rotor from a hot pressed silicon nitride billet
- . Pseudo-isostatic hot pressing of an injection molded silicon nitride preform.
- . Hot pressing through conformable tooling of preformed silicon nitride blades and hub.
- . Fabrication of a dense silicon carbide blade ring by chemical vapor deposition.
- . Electrical discharge machining of a rotor from a hot pressed silicon carbide billet.

On the one hand, silicon carbide was preferred over silicon nitride because of its higher dissociation temperature and therefore expected higher use temperature. On the other hand, silicon nitride has a lower thermal expansion so that, for the same thermal boundary conditions, thermal stresses and associated strength requirements are lower. All things considered, and especially since it was further along in development, silicon nitride was selected as the primary material candidate to meet the program goals.

Further preliminary design/materials/fabrication process work on the fully dense silicon nitride rotor concept revealed some problem areas. One was the difficulty of fabricating a fully dense rotor from silicon nitride and the other was the relatively poor creep and stress rupture for the rotor blades. In the blades where temperatures are highest, the use of fully dense silicon nitride presents a problem because the material strength dropped off at high temperature. This was due to the incorporation of a magnesia hot pressing additive required to form fully dense silicon nitride. To overcome these problems, a duo-density rotor approach was conceived.

Duo-Density Rotor

In this concept, the blade ring would be made of reaction sintered silicon nitride and could be fabricated either by injection molding or slip casting. These methods do not require additives which degrade the strength of hot pressed material. The hub would be fabricated of hot pressed silicon nitride which has the high strength required for the bore of the rotor. The bore region operates at a relatively low temperature and hot pressed material retains most of its room temperature strength at these temperatures. The duo-density concept is illustrated schematically in Figure 2. The blade ring is attached to the hub by diffusion bonding; two approaches are under development for this. In the two-piece approach, the blade ring attaches directly to the hub with one diffusion bonded joint. The other approach uses three pieces in which an intermediate bonding ring is formed between the blade ring and the hub. While this extra bonding ring provides for minor material composition changes, it does necessitate two diffusion bond joints. Recent experience is showing that the two piece approach produces a better bond and, of course, is simpler.

Rotor Temperatures and Stresses

Based on known engine conditions, rotor design geometry, and rotor material physical properties, the rotor thermal and stress gradients can be calculated.

The rotor is subjected to an environment which imposes upon it severe thermal and mechanical stresses. The hot gases, from which work is extracted, heat the blades as they flow by the blades. The attachment bolt passes through the bore of the rotor. The bolt requires cooling to prevent overheating. It is cooled by flowing compressor discharge air down a hole in the bolt. The bore of the rotor radiates heat to the attachment bolt. The combination of heat input from the blades and heat loss to the bolt sets up a thermal gradient in the disc. The thermal gradient is illustrated in the isothermal contour map of Figure 3. The temperatures shown are for the maximum power condition at 100% speed (64,240 rpm) and turbine inlet temperature of 1371°C (2500°F). The temperature gradients shown in Figure 3 impose thermal stresses in the disc and in addition stresses are imposed by the centrifugal loads. The combined centrifugal and thermal stress contour map of the disc is shown in Figure 4.

The rotor blades are slightly cooled by the heat flowing from them into the rotor disc. This sets up a thermal gradient in the blades. The temperature contour map of the blades for 100% speed and turbine inlet temperature of 1371°C (2500°F) is shown in Figure 5. The temperature gradients shown impose thermal stresses in the blades and, in addition, stresses are imposed by centrifugal loads. The combined stress contour map of the blades at the camberline is shown in Figure 6.

These stress and temperature maps are typical of those used during design iterations which led to the final design of the turbine rotor. The actual maps shown in Figure 3 through 6 are for the final design. The materials selected for the rotor therefore must be able to withstand these temperatures and stresses.

Probabilistic Design Technique

Based on the calculated temperatures and stresses in the rotor and the measured strengths of the rotor materials, estimates of the rotor reliabilities can be made at various engine conditions.

Ceramic materials lack ductility to redistribute locally high stresses. Flaws that occur in ceramic materials may act as stress concentrations and raise the stresses above the local calculated value of stress. The locally high stresses may cause the flaw to grow rapidly causing failure of the part. Failure in ceramics is therefore dependent on the flaw and stress distribution in the part and requires a statistical approach to the calculation of reliability. The statistical approach used in this paper was first proposed by Weibull[1] and is shown in Equation 1.

$$\text{Reliability} = e^{-V \left(\frac{\sigma}{\sigma_0} \right)^m} \quad (1)$$

In this equation, σ is the local calculated stress and σ_0 and m are parameters which show the probability of having a certain strength at that local point. Weibull showed that the characteristic strength σ_0 is identical to the ultimate strength of materials in the classical theory as the Weibull modulus m increases indefinitely. This provides a physical interpretation of the characteristic strength. The Weibull modulus m is a constant relating to material homogeneity. The larger the value of m , the more homogeneous the material.

For any structure, equation 1 is integrated over the volume of the structure using the local value of the maximum principal stress. Volumes in which the local value of the maximum principal stress is compressive are neglected to the integration as suggested by Manson[2] because structural ceramic materials generally do not fail in compression. For complex structures, a computer program is needed to perform the integration numerically. In the case of the rotor blades and disc, a computer program has been prepared to compute reliability. This computer program and the temperatures and stresses shown in Figures 3 through 6 were used to compute rotor reliabilities reported later in the materials development section of this paper.

The analytical techniques and computer programs developed give a rapid means of evaluating the reliability of a turbine rotor made of any specified material and provide feedback to the materials engineer to aid him in materials development.

MATERIALS DEVELOPMENT FOR CERAMIC ROTORS

In the duo-density approach, the blade ring is currently a monolithic reaction sintered silicon nitride of 2.7 g/cc density, and the hub is a hot pressed silicon nitride which contains a hot pressing additive. Joining of the blade ring to the hub by diffusion bonding is accomplished simultaneously during the hot pressing operation, which has been termed "hot press bonding" to distinguish it from regular hot pressing. Much recent effort has been devoted to refining this hot press bonding technique with the objectives of minimizing damage to the blade ring and improving the uniformity of bond strength within a rotor and from rotor to rotor. Most of these refinements involved improvements in "blade fill" (the support for the blade ring), uniformity of radial and axial pressures during press bonding, reduction of bending stresses, and optimization of pressures.

Rotor blade rings are currently being made by injection molding, although methods to slip cast blade rings and complete rotors have also been developed. Studies of nitriding and molding compositions have contributed to the increase in density from 2.2 to 2.7 g/cc and to improvement in creep resistance. Currently, the Weibull strength has also been improved to a characteristic MOR of 35,000 psi at a Weibull modulus of 10.

For the hot pressed hub, many known grades of silicon nitride have been considered. Because of the unavailability of known high strength silicon nitride in the powder form, work with such material was limited to press bonding of available preformed hubs to blade rings. In-house development has concentrated on utilization of the commercially available CP-85 grade of silicon nitride powder from Advanced Materials Engineering in England.

Parametric Study of Rotor Hub Material

Recently, a material parametric study was initiated to optimize Weibull strength and resulting rotor reliability. In order to achieve high rotor reliability, the proper combination of characteristic MOR and m is required; for example, a material with a low characteristic MOR, but a high m , may be more desirable than a material with a much higher MOR but slightly lower m .

This study involves the following experimental parameters: powder purity, additive content, and powder milling conditions. The effects of the variables were analyzed by means of a saturated fractional factorial analysis.

Table II shows the six variables and their two levels investigated. The dummy variable was included to indicate if any strong interaction or unknown variable existed. Duplicate runs were made in experiments 1 and 8 to evaluate the reproducibility of results. The matrix as shown represents the very minimum amount of experimental data in this initial work for factorial analysis, and should be considered preliminary information.

From each of the ten billets (7.6 cm. in diameter by 2.5 cm. thick) listed in Table II, approximately forty "A" size¹ MOR test bars were machined. Half of the bars were tested at 871°C (1600°F) and the balance were tested at 1204°C (2200°F) in four-point bending using a 0.953 cm. upper span and a 1.905 cm. lower span. This resulted in statistical strength distributions at two selected temperatures. From the MOR data such as shown in Table III for one example, Weibull parameters were calculated by means of a "most likelihood estimator"[3] program. Weibull parameters at 704°C (1300°F) and 1371°C (2500°F) were obtained by extrapolation assuming a straight line. Use of these parameters plus thermal and elastic properties such as shown in Table IV, in conjunction with the stresses shown in Figures 4 and 6, permitted the estimation of the overall rotor short-time reliability, as tabulated in Table V. These reliability numbers were used in Table VI to evaluate the matrix of Table II.

The results of the above factorial analysis indicated that, since the reproducibility of reliability results is 0.19299 as shown in Table VI, only the variables ranked 1 to 3 are significant because their effects are larger than 0.19299. The small effect of the dummy variable indicates little interaction among the variables and absence of a strong, unknown variable. The best material would thus be made by tungsten carbide milling and contain 5% magnesia. Table VI shows also that the results can be strongly influenced by the test bar machining source. It is important to point out that these preliminary results are presented here to illustrate the method of approach rather than to present definitive conclusions.

¹ 3.2 cm. long x 0.635 cm. wide x 0.318 cm. thick

Based on the current state-of-the-art in materials development, Table V shows that the required short-time reliabilities are well within reach. It is important now to tighten quality control to ensure the reproducibility of material properties and confirm the results of the analytic techniques by means of experimental testing for actual rotor reliabilities, while continuing to improve material properties and design for increasing the rotor reliabilities.

Future Materials Study

In the future, this parametric study will be expanded to include more variables, more levels of each variable, and a larger matrix. In addition, both short-time reliability and lifetime reliability will be considered. Lifetime reliability is expected to be predicted by two methods.

1. Statistical method based solely on strength distribution in short-time mechanical tests.
2. Proof test method in which actual hardware will be proof loaded to calculated proof stress ratios for ensuring a certain minimum life.

CONCLUSIONS

Ceramic gas turbine technology is receiving widespread attention for potential application over the power spectrum of gas turbine engine.

In the ARPA/ERDA/Ford program, stationary ceramic components for a vehicular ceramic turbine engine have been designed, fabricated and tested with encouraging results.

The ceramic turbine rotor is the most difficult problem and a duo-density silicon nitride rotor approach has been conceived and is under development. Temperatures and stresses in this rotor for the full speed operating condition have been predicted and probabilistic analysis techniques have been employed to assess rotor reliability. Based on the current state of materials development, the calculated reliabilities for the rotor are very encouraging.

An initial parametric study of hot pressed silicon nitride, used in the highly stressed rotor hub, has been completed. From this study, the starting powder and hot pressing additive can be selected for best rotor reliabilities.

Nomenclature

b	=	width of specimen (cm)
C _p	=	specific heat ($\frac{\text{joules}}{\text{kg} \times ^\circ\text{K}}$)
E	=	Young's modulus (GPa)
G	=	shear modulus (GPa)
h	=	height of specimen (cm)
L ₁	=	length between supports (cm)
L ₂	=	length between load points (cm)
m	=	Weibull modulus
MOR ₀	=	characteristic modulus of rupture (MPa); the characteristic value is the modulus of rupture value at which 63.2 percent of the specimens have failed.
n	=	number of test samples
V	=	volume of the structure (cm ³)
α	=	thermal expansion (10 ⁻⁶ /°K)
k	=	thermal conductivity ($\frac{\text{watts}}{\text{m} \times ^\circ\text{K}}$)
σ	=	maximum principal stress at each point in the structure (MPa)
σ ₀	=	characteristic strength (MPa [cm] ^{3/m})
ν	=	Poisson's ratio

References

1. Weibull, W., "Statistical Theory of Strength of Materials", Proceedings of Royal Academy of Engineering Science, No. 151, 1939.
2. Manson, S. S., "Thermal Stress and Low-Cycle Fatigue", McGraw-Hill, New York, 1966, p. 293.
3. Mann, N. R., Schafer, R. E., Singpurwalla, N. D., "Methods for Statistical Analysis of Reliability and Life Data", Wiley, New York, 1974.

Status of Design D Stationary Ceramic Turbine Components

<u>Component</u>	<u>Best Hours of Survival (Without Failure)</u>	
	<u>1930°F</u>	<u>2500°F</u>
Reaction Bonded SiC Combustor	175 hours	26 hours
Reaction Sintered Si ₃ N ₄ Nose Cone	220 hours	5 hours
Reaction Sintered Si ₃ N ₄ Stator	175 hours	5 hours
Reaction Bonded SiC Stator	145 hours	Not yet tested
Reaction Sintered Si ₃ N ₄ Tip Shrouds	245 hours	5 hours

Table I

Table II
Factorial Analysis Matrix

Experiment	Number	Powder	% MgO	Ball Type	Wet/Dry	Time	Machining Source	Dummy Variable
1a	1011	CP85	5	WC	Wet	72	B Co.	+
1b	1012	CP85	5	WC	Wet	72	B Co.	+
2	1029	CP85	5	Al ₂ O ₃	Wet	48	t Co.	-
3	1010	CP85	2	WC	Dry	72	T Co.	-
4	1030	CP85	2	Al ₂ O ₃	Dry	48	B Co.	+
5	1018	KBI	5	WC	Dry	48	B Co.	-
6	1017	KBI	5	Al ₂ O ₃	Dry	72	T Co.	+
7	1019	KBI	2	WC	Wet	48	T Co.	+
8a	1014	KBI	2	Al ₂ O ₃	Wet	72	B Co.	-
8b	1016	KBI	2	Al ₂ O ₃	Wet	72	B Co.	-

Powder: Si₃N₄ powder, either as-received A.M.E. CP85 or after magnetic separation and air reclassification of the CP85 powder by KBI.

% MgO: Weight percent MgO hot pressing additive.

Ball Type: Type of milling balls. WC balls were 0.95 cm size and Al₂O₃ balls were 1.9 cm size. In a gallon size milling jar, 30,000 g. wc were used with 800 g. Si₃N₄ in dry milling, and with 2000 g. Si₃N₄ in wet milling; 6000 g. Al₂O₃ balls were used with 800 g. Si₃N₄ in dry milling and with 2000 g. Si₃N₄ in wet milling.

Wet/Dry: Wet milling with methanol or dry milling

Time: Milling time in hours

Machining Source: Name of shop which machined the MOR test bars.

Dummy Variable: A fictitious variable to indicate whether there is a strong interaction among the real variables or whether there is a strong variable which has not been considered.

Table III

Sample MOR Data (Billet 1018) (MPa)

871°C (1600°F)

499	529
509	547
598	535
567	515
447	473

1204°C (2200°F)

382	404	364	357
366	388	339	362
331	379	366	377
413	364	380	372
364	388	377	370

Table IV

Material: Hot Pressed Silicon Nitride

Designation: Billet 1018

Density: 3.17 g/cc

Composition: 2.2 Mg, 0.8 Al, 0.6 Fe, 0.2 Ca

	Property	Unit	Temperature, °K					
			300	530	810	1090	1370	1540
Thermal Properties	MOR ₀	MPa	617	617	617	566	429	303
	m _w	-	11.6	11.6	11.6	11.6	16.6	22.9
	α	10 ⁻⁶ /°K	1.84	2.39	2.87	3.20	3.46	3.66
	k	$\frac{\text{watts}}{\text{m} \times ^\circ\text{K}}$	29.4	25.9	22.5	19.0	15.9	13.8
	C _p	$\frac{\text{joules}}{\text{kg} \times ^\circ\text{K}}$	745	950	1100	1209	1372	1360
Elastic Properties	E	GPa	294	294	294	292	291*	286*
	ν	-	0.256	0.254	0.251	0.246	0.239*	0.221*
	G	GPa	117	117	117	117	117*	117*

*Extrapolated value or calculated from extrapolated values

Table V
Weibull Parameters and Calculated Reliabilities

<u>Number</u>	<u>Characteristic MOR, MPa</u>		<u>Weibull Slope</u>		<u>Reliability</u>		
	<u>871°C</u>	<u>1204°C</u>	<u>871°C</u>	<u>1204°C</u>	<u>Ring</u>	<u>Hub</u>	<u>Rotor</u>
1011	500	375	16.3	22.3	0.99190	0.99970	0.99160
1012	509	360	8.7	16.7	0.99190	0.90810	0.90074
1029	449	324	10.5	16.5	0.99190	0.90027	0.89297
1010	639	368	21.7	11.2	0.99190	0.99917	0.99097
1030	471	346	5.0	21.1	0.99190	0.11365	0.11273
1018	541	381	11.6	19.1	0.99190	0.99582	0.98775
1017	593	422	11.6	15.6	0.99190	0.99796	0.98987
1019	725	420	9.4	12.7	0.99190	0.99633	0.98826
1014	451	299	4.8	17.2	0.99190	0.01503	0.01491
1016	485	290	6.2	14.9	0.99190	0.31256	0.31003

Table VI
Factorial Analysis

Level	Variable					Machining Source	Dummy Variable
	Powder	% MgO	Ball Type	Wet/Dry	Time		
CP85	0.73571						
KB!	0.78209						
KBI	0.04638						
2% MgO		0.56361					
5% MgO		0.95419					
5% MgO		0.39058					
WC			0.97829				
Al ₂ O ₃			0.53951				
WC			0.43878				
Wet				0.74747			
Dry				0.77033			
Dry				0.02286			
48 hours					0.74543		
72 hours					0.77237		
72 hours					0.02694		
B Co.						0.55228	
T Co.						0.96552	
T Co.						0.41324	
Dummy (+)							0.75926
Dummy (-)							0.75854
Dummy (+)							0.00072

Ranking of strength of variables at their indicated level changes:

Rank:	1	2	3	4	5	6	7
Variable:	Ball Type	Machining Source	% Mgo	Powder	Time	Wet/Dry	Dummy
Effect:	0.43878	0.41324	0.39058	0.04638	0.02694	0.02286	0.00072

Average Reproducibility in Experiments 1 and 8 =

$$1/2(0.99160 - 0.90074) + 1/2(0.31003 - 0.01491) = 0.19299$$

THE "ALL-CERAMIC FLOW PATH" OF FORD'S 200 HORSEPOWER VEHICULAR TURBINE ENGINE

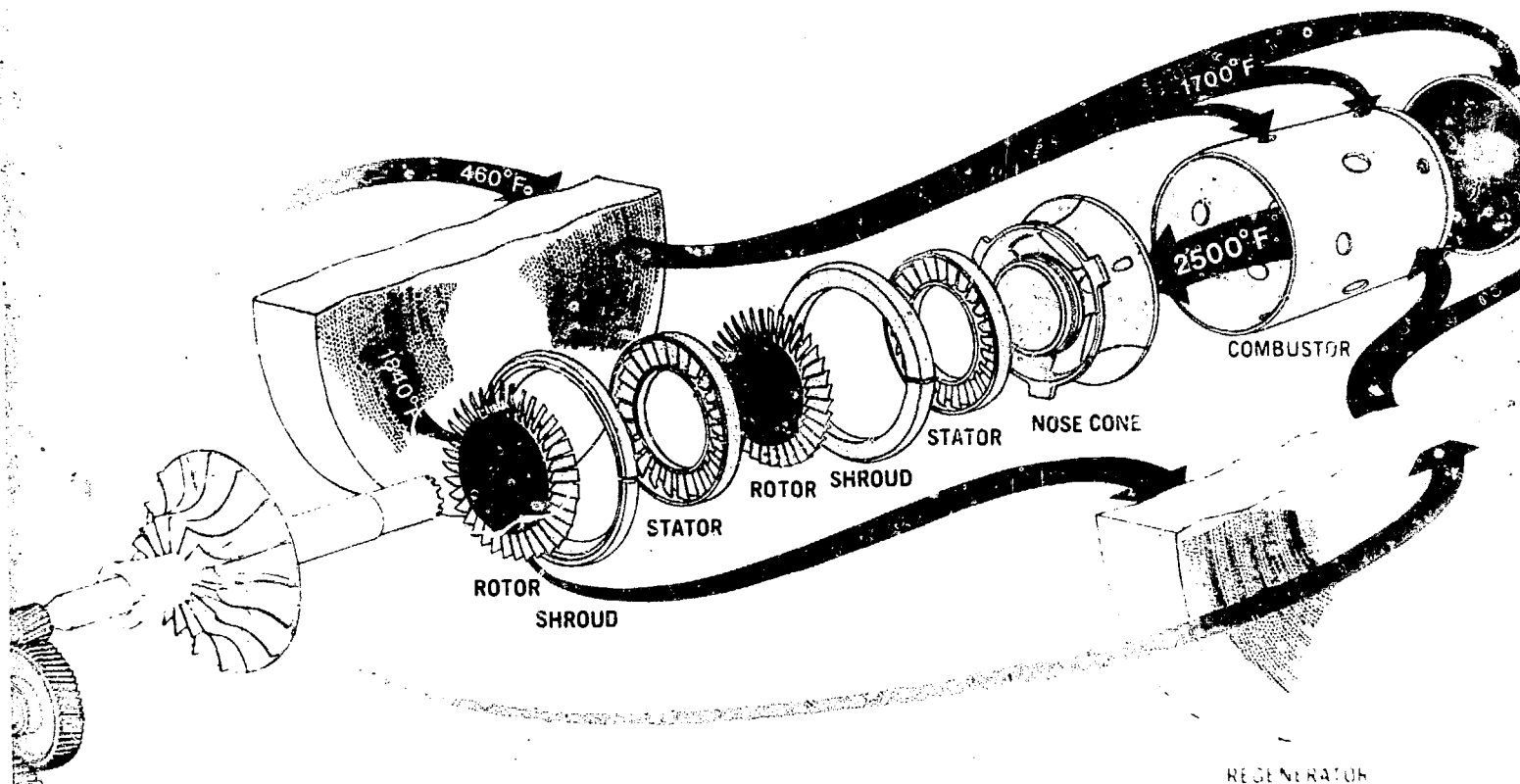
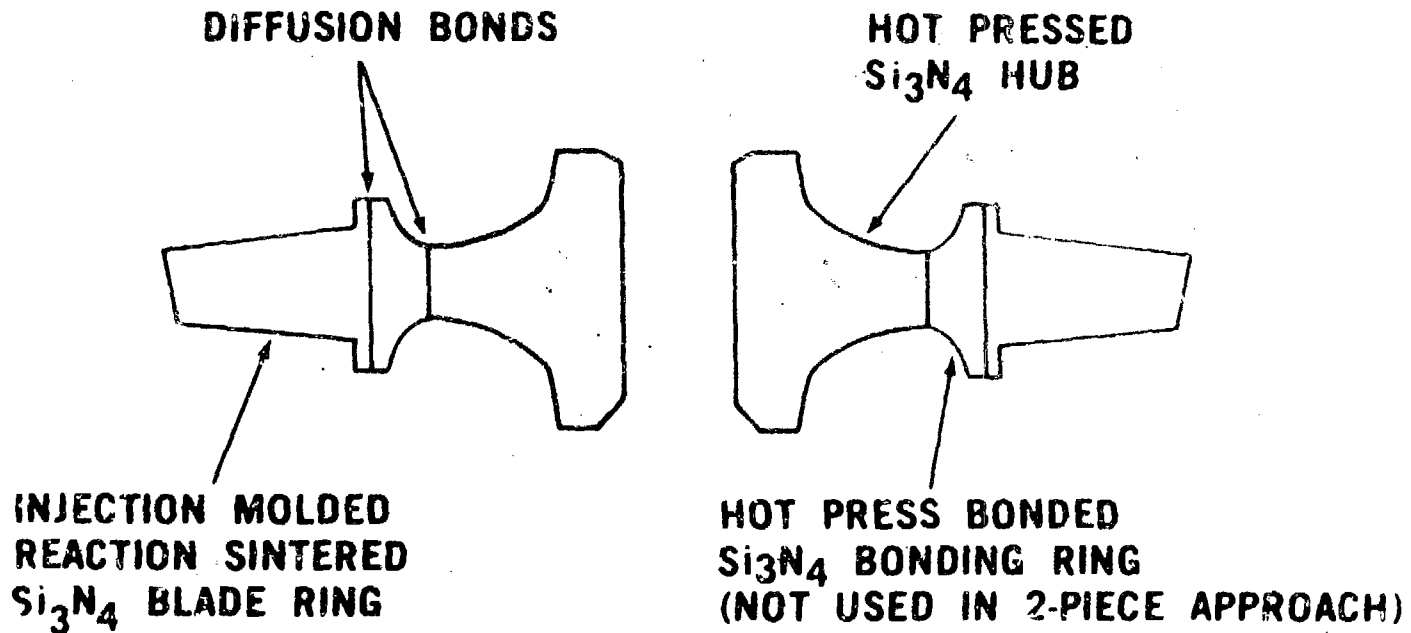


Figure 1

DUO-DENSITY SILICON NITRIDE TURBINE ROTOR



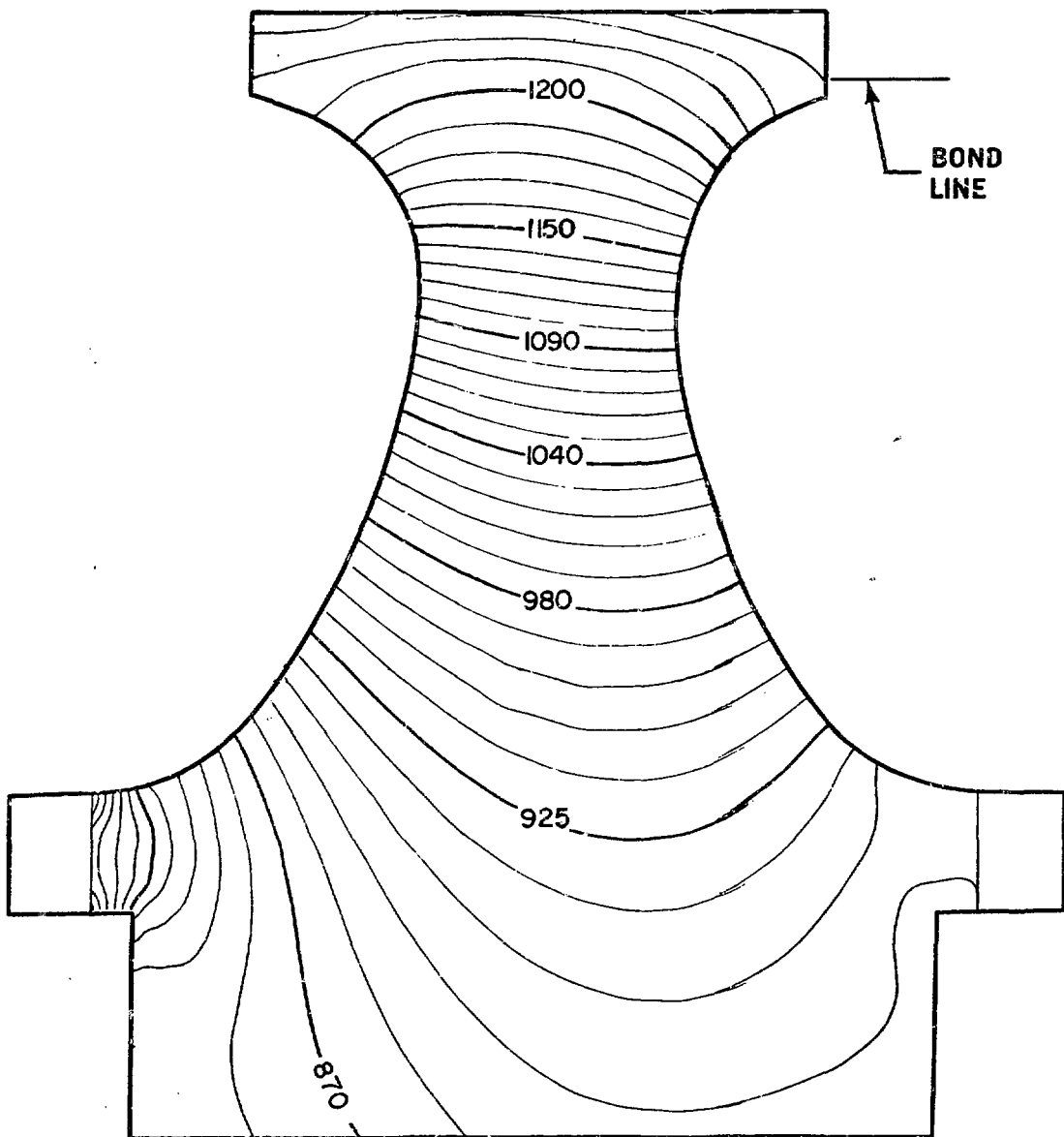


Figure 3 Temperature Contour Map at 1371°C TIT and 100% Speed - First Stage Turbine Hub

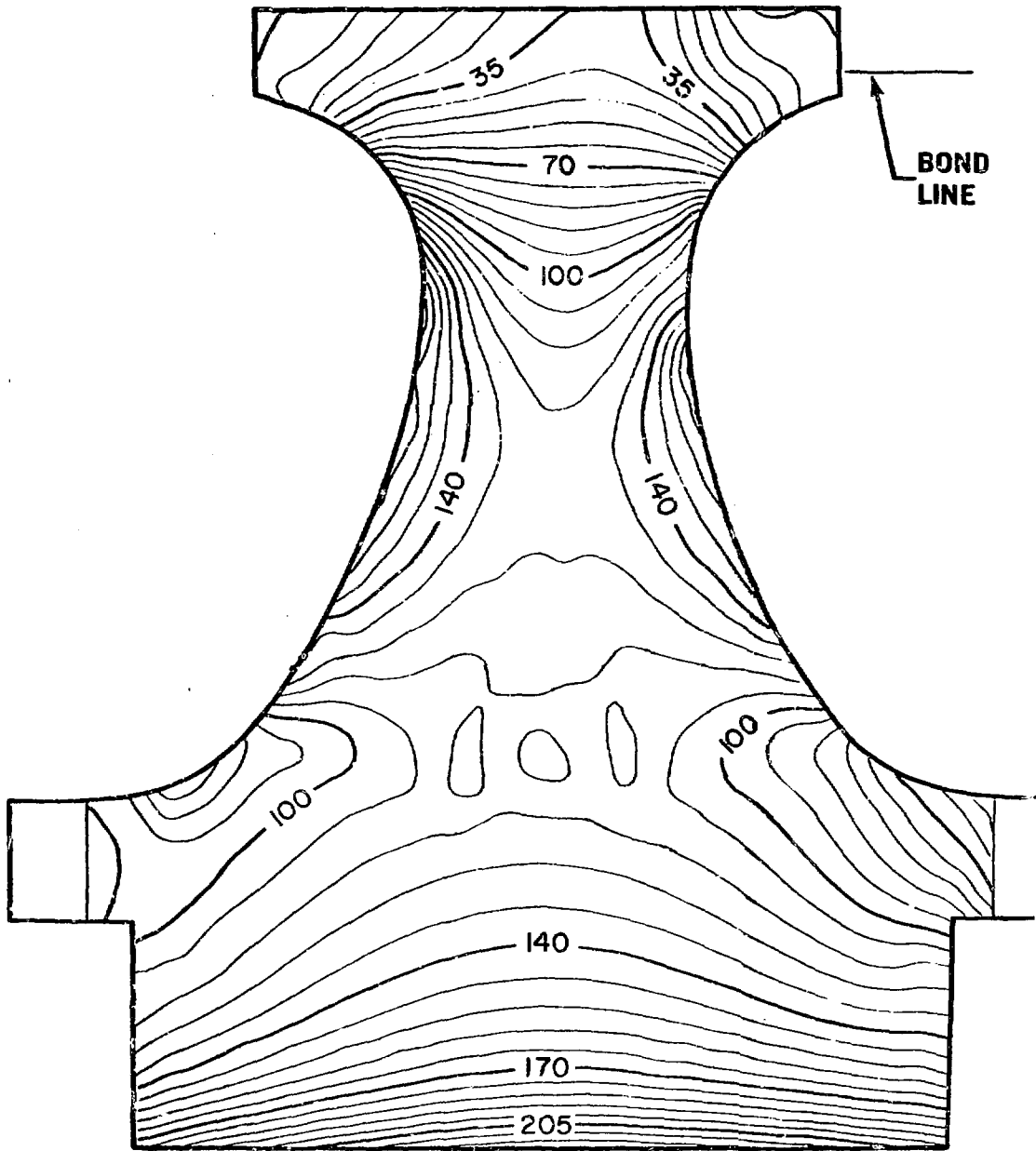


Figure 4 Contour Map of Maximum Principal Tensile Stresses
at 1371°C TIT and 100% Speed - First Stage Turbine Hub

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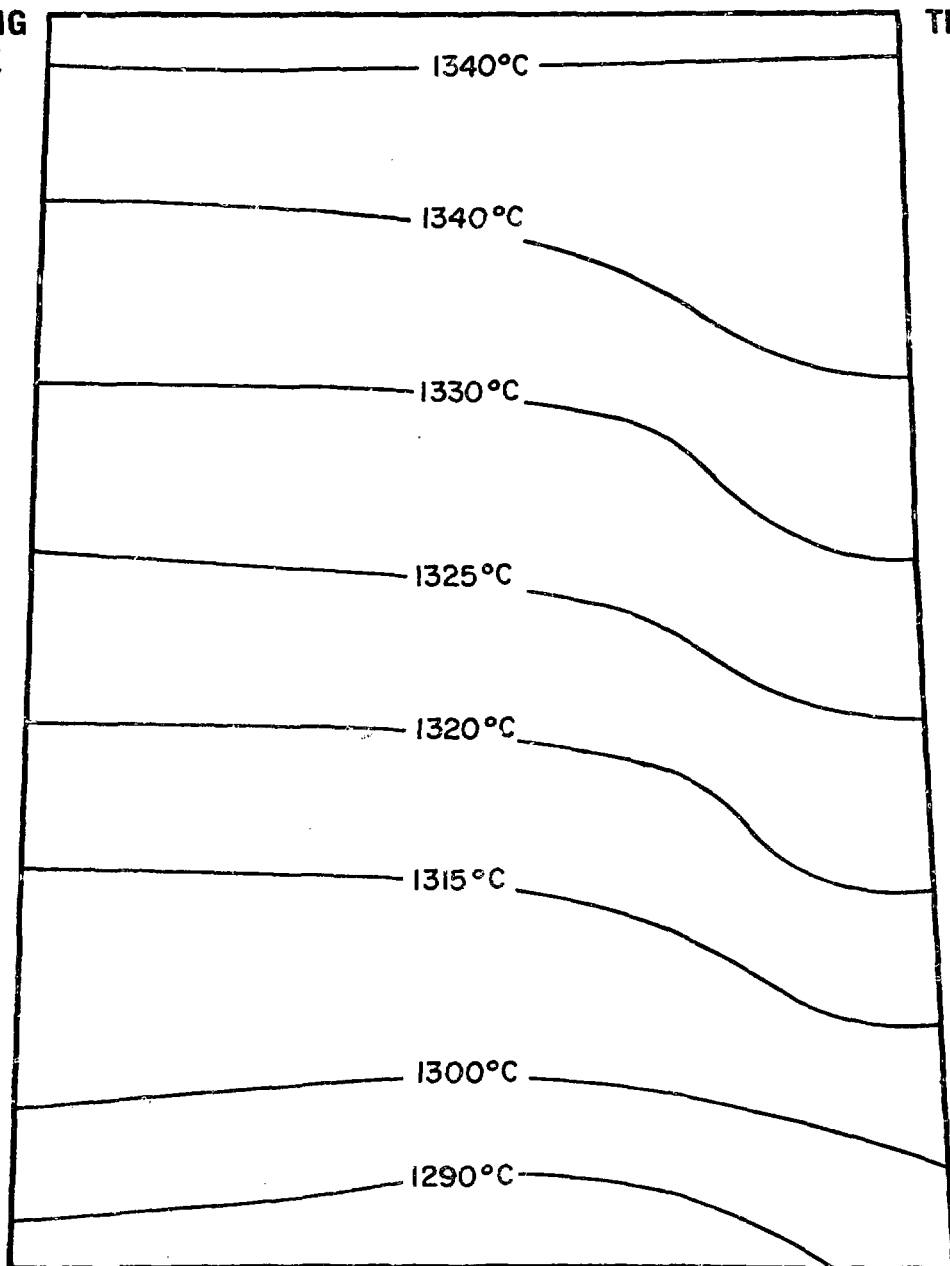


Figure 5 Temperature Contour Map at 1371°C TIT and 100% Speed -
First Stage Tu-bine Rotor Blade Design D'

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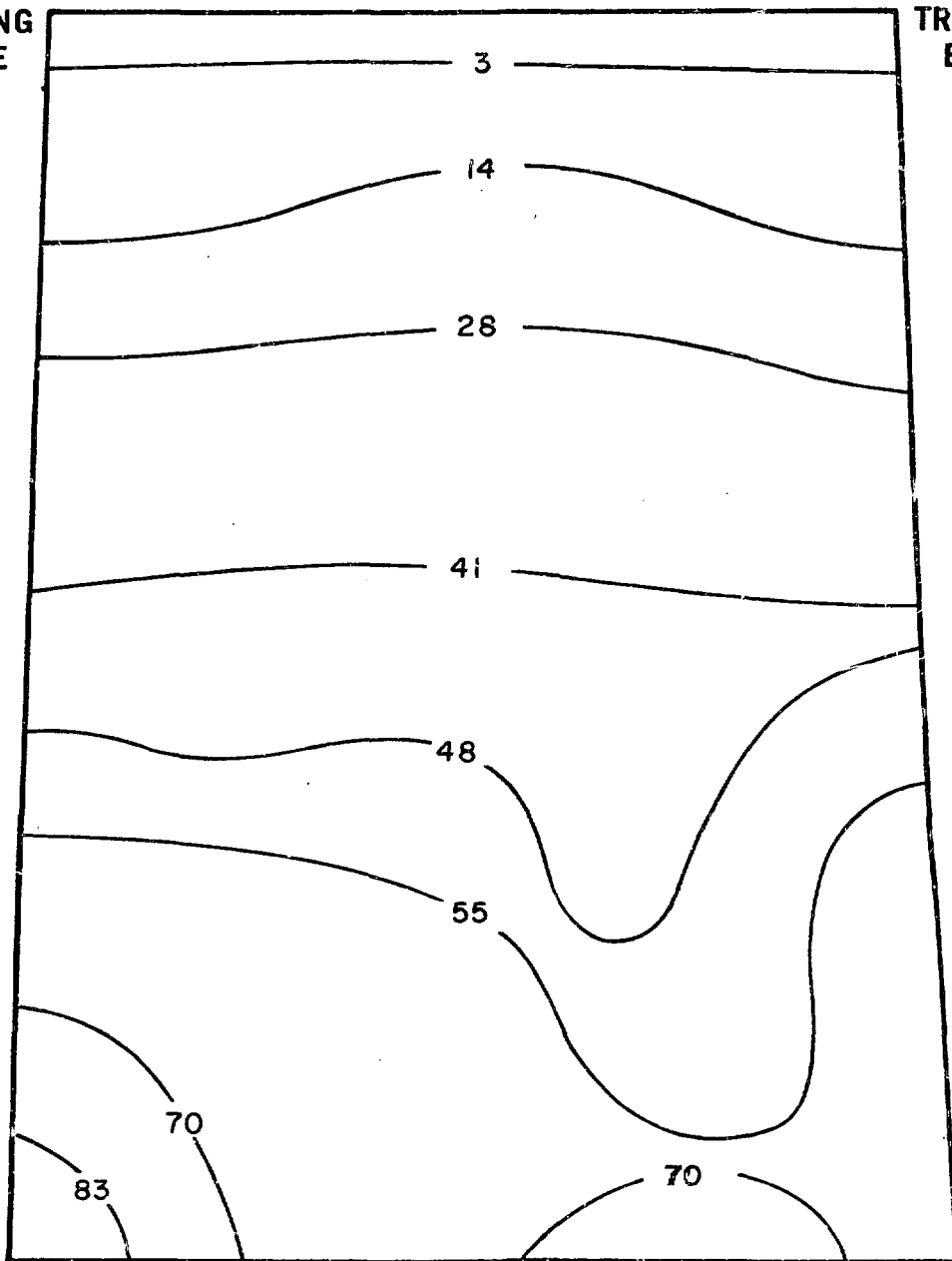
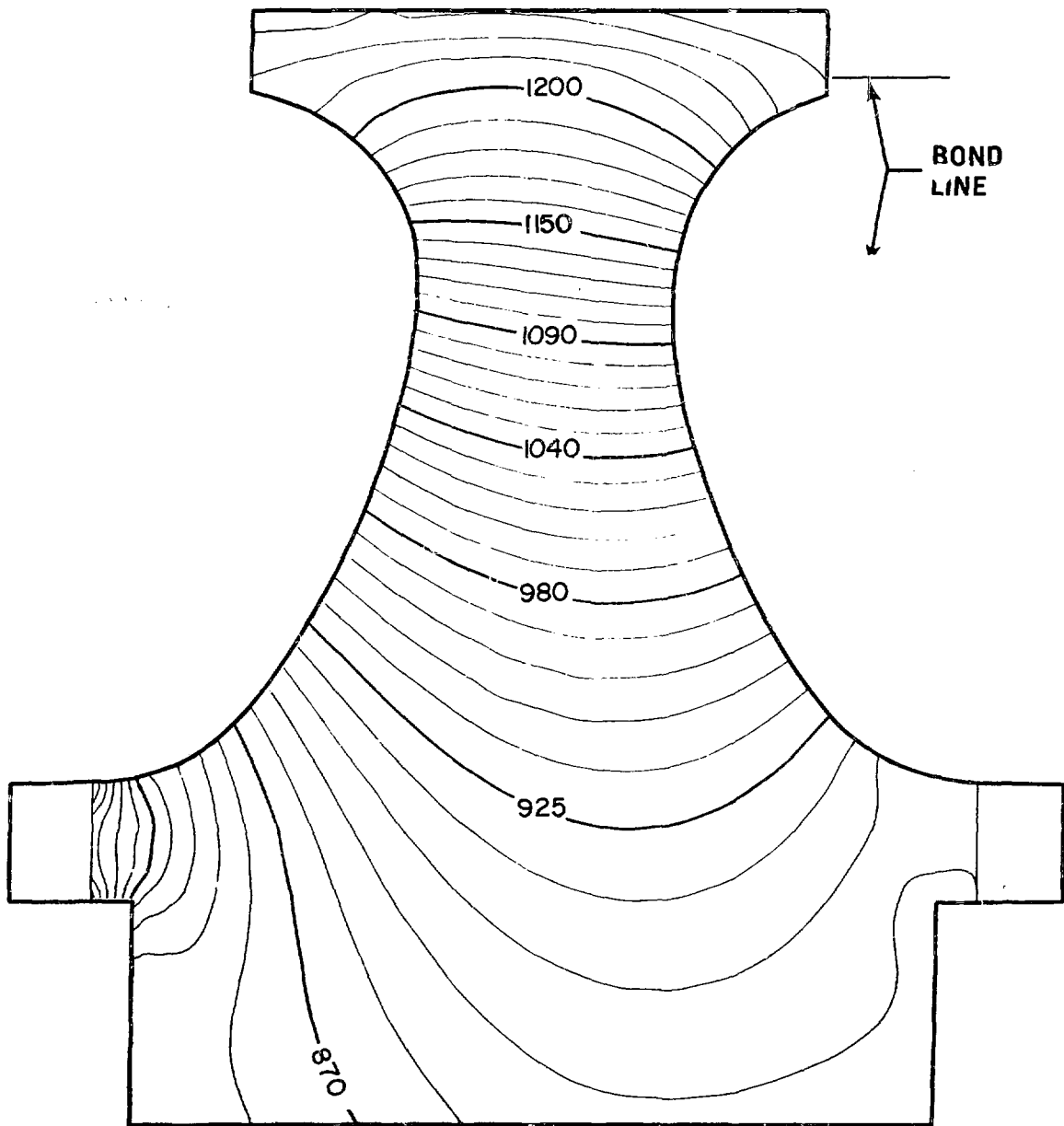
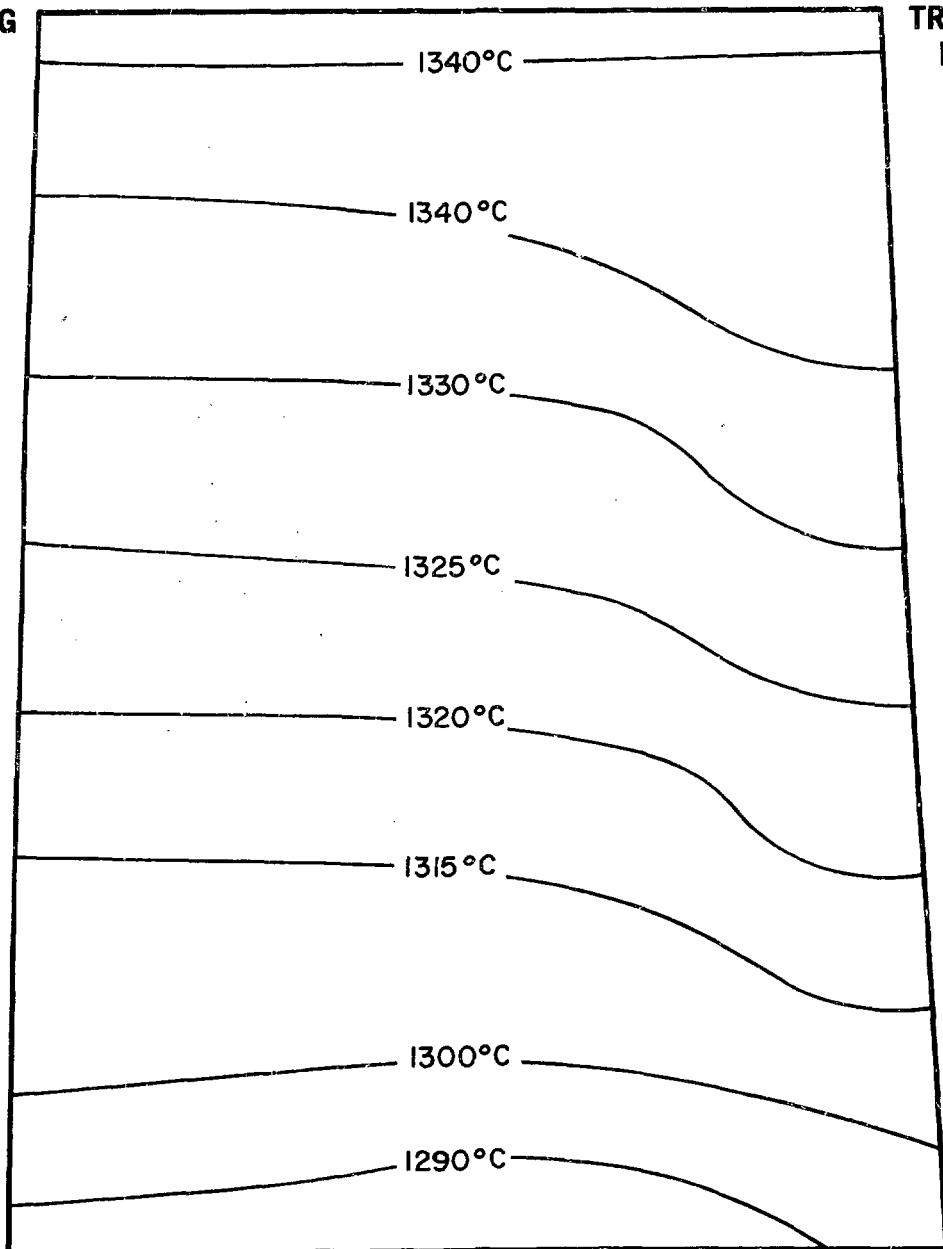


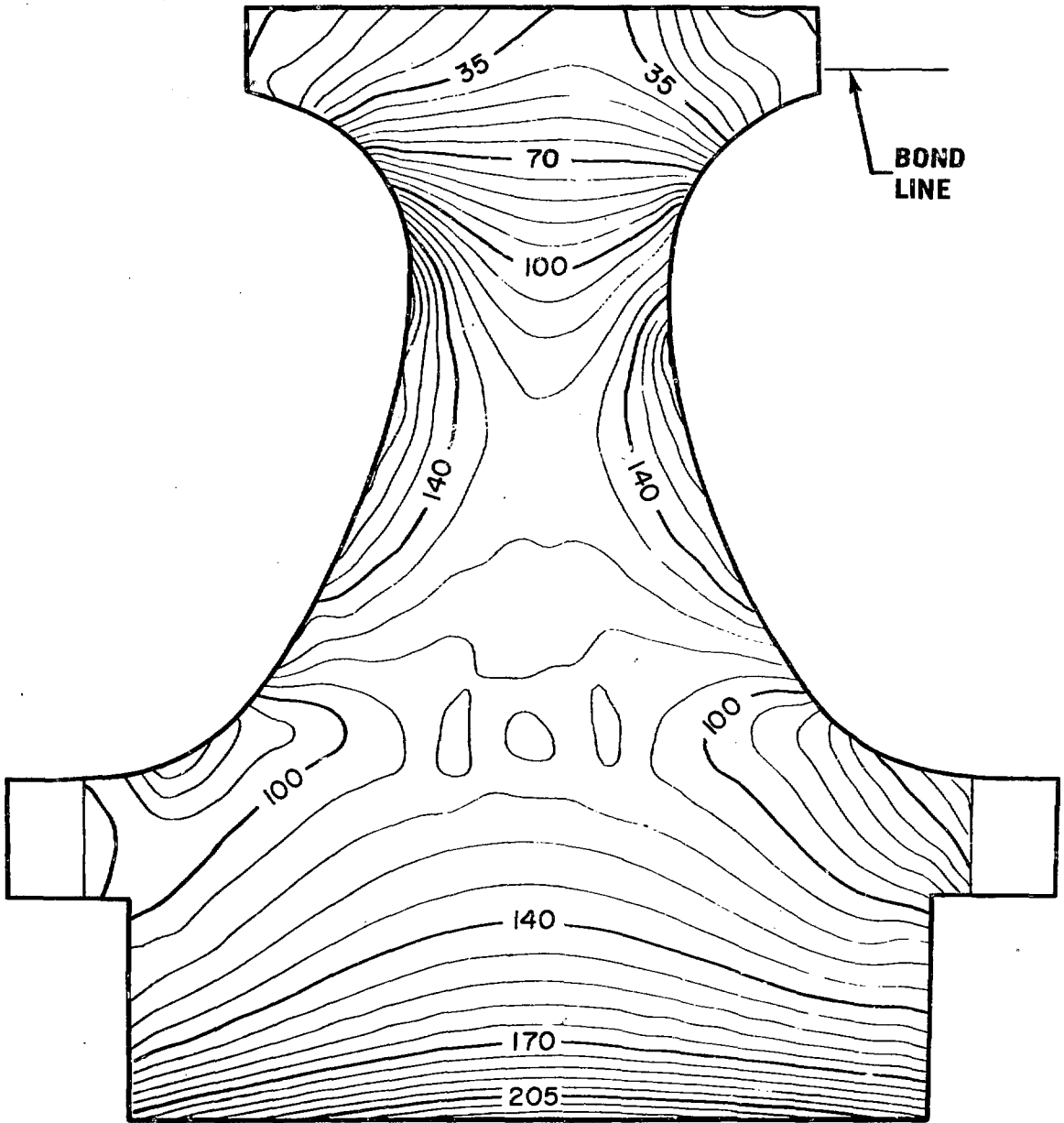
Figure 6 Contour Map of Maximum Principal Tensile Stresses at
Camberline - 1371°C TIT and 100% Speed - First Stage
Turbine Rotor Blade Design D'



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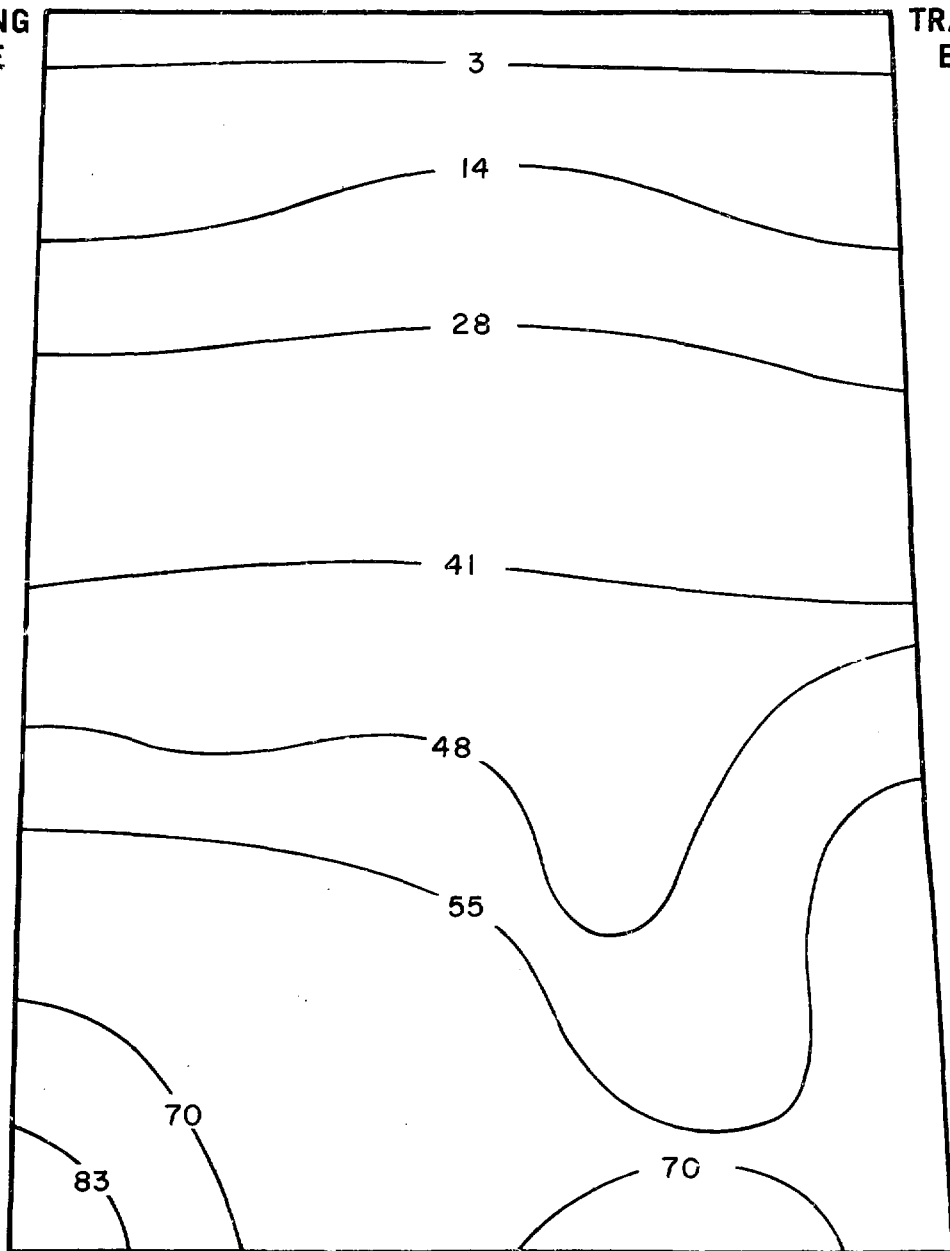
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GAS TURBINE ENGINE APPLICATION
IN TRANSIT COACHES

by

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Richard Tarkir

Booz, Allen & Hamilton, Inc.
Transportation Consulting Division

GAS TURBINE ENGINE APPLICATION IN TRANSIT COACHES

ABSTRACT

An investigation of gas turbine engine application in transit coaches was conducted by Booz, Allen Applied Research, as part of the Transbus program, to determine the viability of the gas turbine engine as a potential power source for transit coaches. During the Transbus engineering and public demonstration testing, the three Transbus prototypes that were equipped with gas turbine powerplants exhibited numerous advantageous operational characteristics that warranted further investigation. Consequently, an in-depth survey was conducted of gas turbine engine manufacturers whose products may be suitable for installation in transit coaches. Their engine products were evaluated to determine their merits, if any, against those of the conventional diesel engine used in transit coaches. While several concerns are actively engaged in research and development of automotive turbine engines, only Detroit Diesel Allison Division (DDAD) of General Motors Corporation has a product that is currently developed to a state where volume production can be seriously considered.

In addition to research and development engines, nearly 100 DDAD gas turbines have been field-tested in trucks, transit coaches, intercity coaches, marine craft, and

industrial electrical generator sets and air compressors. Three DDAD gas turbine engines were installed in the Transbus prototype coaches manufactured by Truck and Coach Division of General Motors. This engine was selected for testing in Transbus because of the gas turbine's apparent advantages and demonstrated success in heavy trucks. Since 1972 test engines in 24 trucks of ten different makes have operated in widely divergent service in all parts of the country. In addition, eight Greyhound intercity motor coaches powered by gas turbines have operated throughout the country. In both of these applications, the turbine engine has demonstrated greatly improved reliability and fuel consumption rates compared with earlier turbines. In fact, the fuel consumption is nearly competitive with that of the diesel engine. In the Transbus prototypes, the gas turbine engines were extremely reliable and maintenance free; however, fuel consumption was much higher than contemporary diesel engines. The extended periods of idle and extensive part-load operation, both inherent to transit coach service, account for a measure of the high fuel consumption. Improvements have been made since these coaches were evaluated and further improvements are scheduled for reducing brake specific fuel consumption (BSFC) in the transit coach duty cycle to where it will be competitive with diesels.

GAS TURBINE ENGINE APPLICATION IN TRANSIT COACHES

INTRODUCTION

Development of the gas turbine engine as a potential power source for automotive vehicles began in the early 1950's. By 1976 three manufacturers were actively pursuing the development of small (150HP to 650 HP) turbine engines. The major thrust of this development centered on heavy-duty truck application because it comprised the largest potential market in the small engine power range and because the heavy-duty truck power requirements corresponded to gas turbine engine characteristics. Since 1972 test engines in 24 trucks have operated in widely divergent service in all parts of the country.

Some engine development, however, has been directed toward transit coach application and in the early 1970's, under the Urban Mass Transportation Administration Transbus program, a gas turbine engine was tested in one of the prototype buses. During the program, this engine exhibited the following operational advantages over the conventional diesel engine in transit coach application:

- . Reduction of installed weight and volume
- . Elimination of cooling radiator, fan, and attendant piping

- . Cleaner exhaust emissions
- . Lower noise level
- . Vibration-free operation
- . Reduced oil consumption
- . Improved serviceability
- . Improved reliability
- . Improved vehicle performance for given power rating
- . Greater engine braking capability
- . Superior cold weather starting
- . Broad multifuel capability

Because of the gas turbine's apparent advantages and demonstrated success in heavy trucks, an in-depth survey of gas turbine engine manufacturers was conducted to determine the suitability of the turbine engine for transit coach application.

This paper presents an overview of the survey, and traces the development of the only small gas turbine engine sufficiently developed to be considered for near-term volume production and subsequent application to transit coach service.

EVALUATION OF ENGINE MANUFACTURERS

Currently in the United States three companies — Chrysler Corporation, Detroit Diesel Allison Division (DDAD) of General Motors Corporation, and Industrial Turbines International (ITI), a consortium — are actively engaged in the development of small (150 HP to 650 HP) gas turbines that can realistically be considered engines for application to automobiles, trucks, and buses. The efforts of these small turbine engine developers, listed in Table 1 with the reported engine status, were monitored and evaluated on the following bases:

- . Demonstrated reliability
- . Fuel consumption comparable with diesel engines
- . Design that can be mass produced at a cost comparable with diesel engines
- . Realistic production plans
- . Product improvement potential.

Table 1

Activities Involved in Developing Improved
Gas Turbine Engines

Chrysler Corporation	ERDA Baseline Engine (150 HP) and Upgraded Engine (123 HP) under active development, 2-shaft regenerative
Detroit Diesel Allison Division	300 HP, 400 HP and 500 HP, 2-shaft regenerative engine under active development for bus, truck, industrial application
Industrial Turbines International (Air Research/Mack/KHD)	Comprehensive program for truck and industrial applications, 450-600 HP range

The Chrysler engine evolved as a passenger car engine and does not produce sufficient power for transit coach service. In 1962 this engine was extensively field-tested in 50 prototype automobiles. The development of this engine is continuing, with U.S. Government financial aid, in areas that can best be described as "advanced technology." No field testing, however, has been conducted on the current design engine. Should the advanced design engine demonstrate its viability in passenger cars, it could be upgraded to an engine suitable for heavy-duty service. However, considering the design, development, and tooling effort that would be required, the engine could not be available on a mass-production basis for 8 to 10 years.

The ITI consortium engine is in the 450 HP to 600 HP range and is considered too large for transit coach propulsion. While this engine is being designed as an eventual replacement for truck diesel engines, it is in the initial, unproven stages of development and prototype engines will not be available for 2 to 3 years. Production engines could not be available for 4 to 6 years.

The DDAD turbine engine has been developed as a replacement for the diesel engines manufactured by that division. In the Transbus program, the turbine engine demonstrated its potential as a viable transit coach powerplant, and current production plans will make the engine available within 2 years.

DETROIT DIESEL ALLISON DIVISION ENGINE

Description and State of Development

For over 20 years DDAD has been energetically developing small gas turbine engines to compete in many commercial applications with piston diesel engines. Specifically, they have concentrated on engine applications in large trucks and buses, although field evaluation has included construction equipment, marine equipment, and electrical power generation units. DDAD's goals were to develop a practical, cost-effective turbine engine to meet the requirements of large, heavy-duty ground vehicles. This program has resulted in a family of three engine sizes covering the power range from 300 HP to 500 HP (SAE rating). The smallest of these engines is designated as the GT-404, and engines incorporating the latest design changes for improved performance are designated as the -3 series. This engine exhibits the following advantages over the diesel engine currently used in transit coaches.

The GT-404 gas turbine, is a two-shaft, regenerative gas engine featuring a power transfer system. A rigid block assembly, constructed of cast iron, serves as the main structural support member for the engine. It houses, in a modular fashion, the burner, gasifier section, power section, regenerators, and the reduction and accessory drive gearing. The modular design, shown in Figure 1, allows easy service and unit replacement of the various

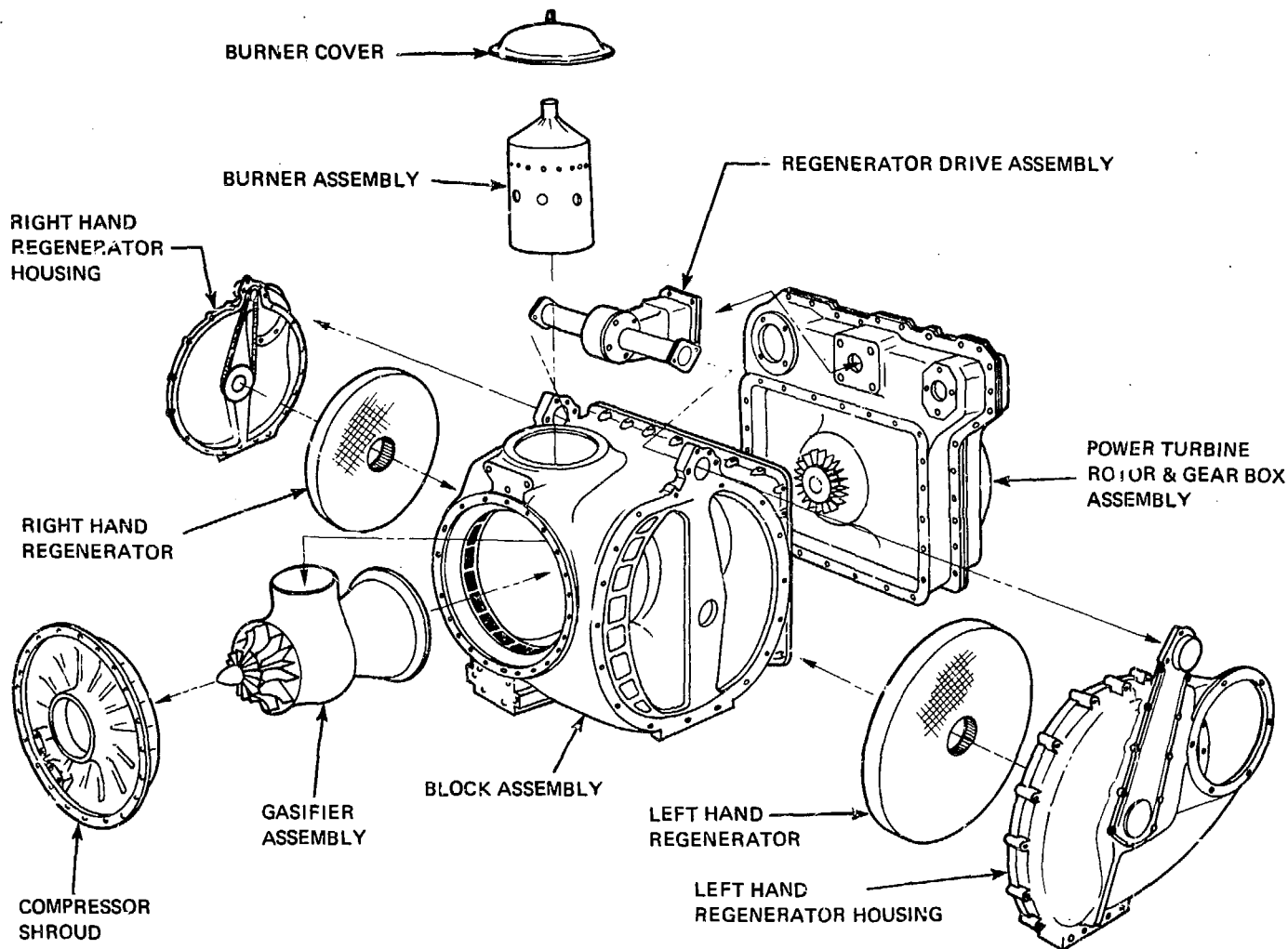


Figure 1
Modular Construction of GT-404 Gas Turbine Engine

sections. The engine controls are electronic and are remotely mounted from the engine block itself. The engine is available with a rated maximum output shaft speed of 2880 rpm.

The GT-404 engine's normal dry weight of 1750 pounds is approximately 650 pounds lighter than a comparable diesel engine, the DDAD 8V-71, and its basic size is very similar. On an average vehicle-installed basis, this weight savings advantage is about 1000 pounds, and the GT-404 can be installed in virtually any vehicle accommodating the 8V-71.

The performance capabilities of the current GT-404-3 engine are shown on the characteristic curve in Figure 2.

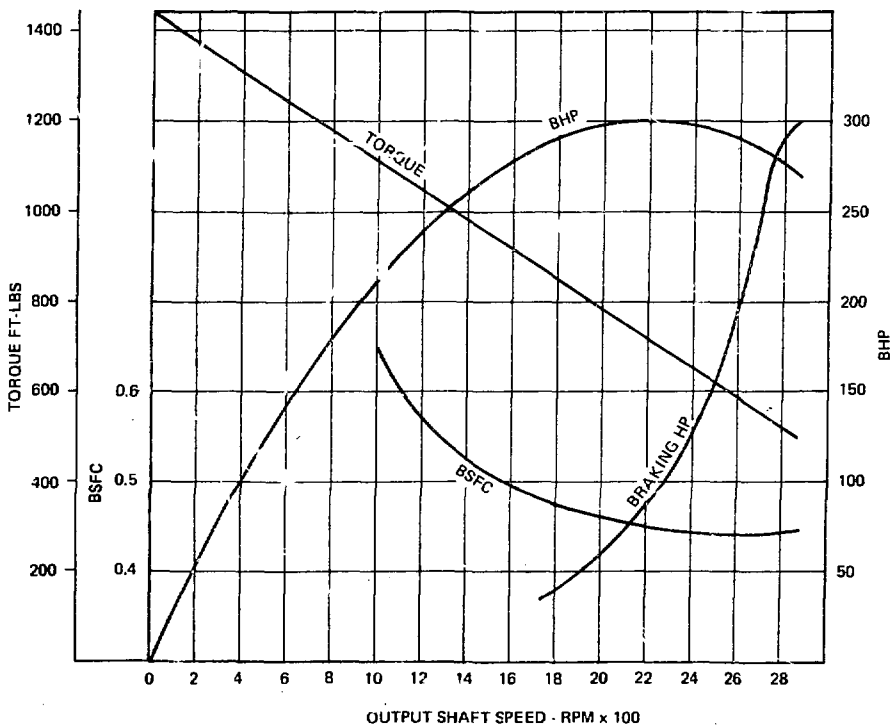


Figure 2
Performance Capabilities of GT-404-3 Engine

Maximum torque is produced at power turbine (output shaft) stall (0 rpm) condition. With the high torque rise characteristic, fewer number of transmission gear ranges are required—generally five or less in trucks, and four or less in buses. With an automatic transmission, a torque converter is not required and a fluid-coupling can be used. The dynamic braking capability of the engine is equal to the rated power at maximum output shaft speed and is effective in each transmission gear range. The fuel consumption (BSFC) characteristics of the engine are shown with BSFC as a function of engine output shaft speed at 100 percent gasifier speed (maximum power capability). This fuel consumption does not compare with the DDAD 8V-71 diesel engine which has a BSFC of less than 0.40 at 100 percent power and which maintains this efficiency at low engine speeds.

The GT-404 engine consumes only about 1 quart of oil per 20,000 to 30,000 miles compared with about 1 quart per 800 miles for a diesel engine in transit coach application. In addition, the turbine engine oil requires changing every 250,000 miles compared with 20,000 miles for a diesel engine. This results in considerable savings for the turbine engine.

The GT-404 engine emits less noise than a comparable diesel engine. Results of an exterior noise test, conducted

to the SAE test standards, on trucks demonstrate that the noise level of the GT-404 engine is 11 dBA lower than a standard diesel-powered truck. This 11 dBA lower reading, which represents nearly a three-fourths reduction in sound pressure over that of a diesel, would reduce transit coach noise levels to about 75 dBA. Development effort is continuing toward a still further reduction of the noise level to meet future noise attenuation requirements.

Major pollution elements of the GT-404 engine in transit coaches would be minimized because highly efficient, low-pressure, continuous burning with large amounts of excess air produce almost 100 percent combustion. The odor is almost undetectable, exhibiting only a slight kerosene odor at engine idle, and exhaust smoke is virtually undetectable.

The gas turbine's ability to start quickly at low temperature is superior to any conventional powerplant. The GT-404 engine has demonstrated its ability to start, without aids, in temperatures well below 0° F. However, batteries must be reasonably well-charged and diesel fuel #1 must be used as diesel fuel #2 begins to cloud (solidify) at about +20° F.

The GT-404 engine does not require a water-based cooling system because it is internally cooled by the

atmospheric air passing through the engine and by the lubricating oil which is cooled through a small oil-to-air heat exchanger. The elimination of a water-based cooling system greatly decreases engine maintenance and downtime, thereby reducing a major maintenance cost area in transit coaches.

The gas turbine engine in general requires less maintenance than a diesel engine because of fewer wearing parts and almost vibration-free operation. All moving elements in the basic turbine engine are rotary in motion, compared with reciprocating components in diesel engines. No water hoses or pipes, drive belts, or other elements that tend to be unreliable are exposed on the exterior of the engine. Increased oil change intervals, use of self-cleaning inertial air filters, absence of the liquid cooling system, absence of a manifold exhaust system, extended brake life, and probable extended life of components tend to reduce maintenance costs and significantly increase vehicle availability.

The fuel system of the GT-404 engine can operate on a wide range of petroleum-based fuels including: diesel #1, diesel #2, furnace oil, JP fuels, kerosene, and gasoline. The diesel fuels are most commonly used because of their higher energy content and ready availability. Gasoline is the least preferred fuel because the lead component tends

to deposit and foul the turbine, nozzles, and regenerators, thereby reducing engine performance. But it can be used in an emergency situation. When synthetic or other fuels become available in quantity, the fuel handling and control system of the GT-404 could be modified or redesigned to accommodate these fuels.

Field Test Summary

DDAD has manufactured approximately 100 GT-series gas turbines for field evaluation in trucks, buses, boats, electrical power generation units, and other applications. These engines include the GT-404 and GT-505 in both the -2 and -3 configurations.

Pilot models of the engine began going into service in 1972 for extensive field evaluation. The engines have been tested in 24 trucks from ten manufacturers, eight motor coaches from MCI-Greyhound, coaches from GMC Truck and Coach Division, Transbus prototypes, various watercraft, and industrial applications.

Consignment engines are currently operational with Greyhound on the East Coast and West Coast, Binswanger Trucking in Los Angeles, Freightliner Corporation and Consolidated Freightways in Portland, Acadian Marine Rentals in New Orleans, Terminal Transport in Atlanta, Gardner-Denver in Quincy, Illinois, a Hatteras yacht operating in

the waters off New Jersey, GMC Truck and Coach Division of General Motors in Pontiac, Michigan, and Detroit Diesel Allison in its Indianapolis-based field-test vehicles.

The major effort at DDAD, in the development of the GT-404 engine, has been directed toward heavy-duty trucks. The field experience with turbine engines in transit coaches has been limited to the Transbus prototypes and several development engineering models assembled by Truck and Coach Division that never experienced real transit service. The most extensive coach experience with turbine engines is in the Greyhound fleet. However, it must be recognized that the duty cycle and service requirements for intercity coaches such as Greyhound is substantially different from transit coaches.

The six GT-404 turbine-powered Greyhound coaches have logged well over one million miles since mid-1975 and have currently been fitted with -3 pre-production engines. These -3 engines are capable of operating at 0.51 BSFC as compared with 0.54 BSFC of the -2 engines used in the Transbus prototypes, and DDAD is updating the -3 engines with the latest burner improvements, and other developments to lower the BSFC to 0.45. Even without the improved engine modifications, the fuel penalty sustained by Greyhound has averaged only slightly over 1 mpg for the turbine engine compared

with the diesel. The turbine engine has also virtually eliminated engine overheating and other cooling system-related problems, which account for 50 percent of Greyhound's road failures with diesel engines. Brake life on the turbine-powered Greyhound coaches has been extended by more than 50 percent due to the engine's regenerative braking system. Elimination of the engine vibration-induced cracks in refrigerant lines and fittings has improved air conditioning system reliability in turbine-powered coaches, and elimination of engine vibrations and reduction of the powerplant weight have improved coach structural integrity. The largest number of engine problems Greyhound experienced with the turbine engines were with the various electrical controls.

Because the turbine engine does not have a water base cooling system to provide hot water for coach heating, Greyhound mounted an exhaust gas-to-water heat exchanger above the engine at the rear of the coach. This system is augmented by a boiler element located in the heater plenum. This auxiliary heating system is used infrequently, only during very cold ambient temperatures. In transit coaches the exhaust gas-to-hot-water heating system will be inadequate by itself because transit coaches require higher heating due to the frequent door openings and there is less heat available in the exhaust gas to power the system due to the lower load factor of the engine.

In summary, Greyhound is more than pleased with their turbine engine experience and view the engine as a technical breakthrough for intercity coach transportation. Top management personnel of Greyhound have publicly stated on several occasions that they are prepared to incorporate gas turbines in their entire fleet of coaches when the engine becomes available in production.

Production and Development Plans

DDAD has established a plan to place the GT-series of turbine engines into production, but the plan has not been accepted or funded by the management of General Motors at this time. It is contingent upon successful attainment of several engine development milestones which relate to fuel consumption and engine durability and production cost contingencies that must be attained. DDAD has stated, "that when the gas turbine engine is available in production, the cost of a turbine engine will be within 20 to 25 percent of an equivalent power size diesel engine."

Cost of the complete turbine powerplant will be on the order of 20 percent over the diesel, considering that the cost of the muffler and radiator required for the diesel will be offset by the coach cabin heating system required for the turbine transit bus. Several potential engine component suppliers have reportedly stated a willingness

to accept a monetary loss in the low-volumed early production to ensure participation in the forecasted high production volumes of the early 1980's.

The major development effort on the turbine engine is being directed toward improving the fuel economy. However, since the turbine inlet temperature (TIT) has the greatest single effect on fuel consumption and engine life, engine durability is a function of the BSFC (where TIT is increased without blade cooling or improved materials). The durability goal is to double the engine life currently rated at 5,000 hours by 1981.

The improvement goal of 0.40 BSFC scheduled for later production engines is not quite equal to the current fuel rate of diesel engines. However, if the relative costs of fuel, maintenance, and labor do not change, the gas turbine will be less expensive to own and operate than a diesel engine in a heavy truck because of the turbine's extended life and other advantages.

In spite of the favorable outlook for the gas turbine in heavy trucks and intercity coaches, the engine's future in transit coaches is less certain. Even with the fuel consumption improvements scheduled by DDAD, the gas turbine will still be far from competitive with the diesel engine at idle and part-load. Figure 3 shows a comparison

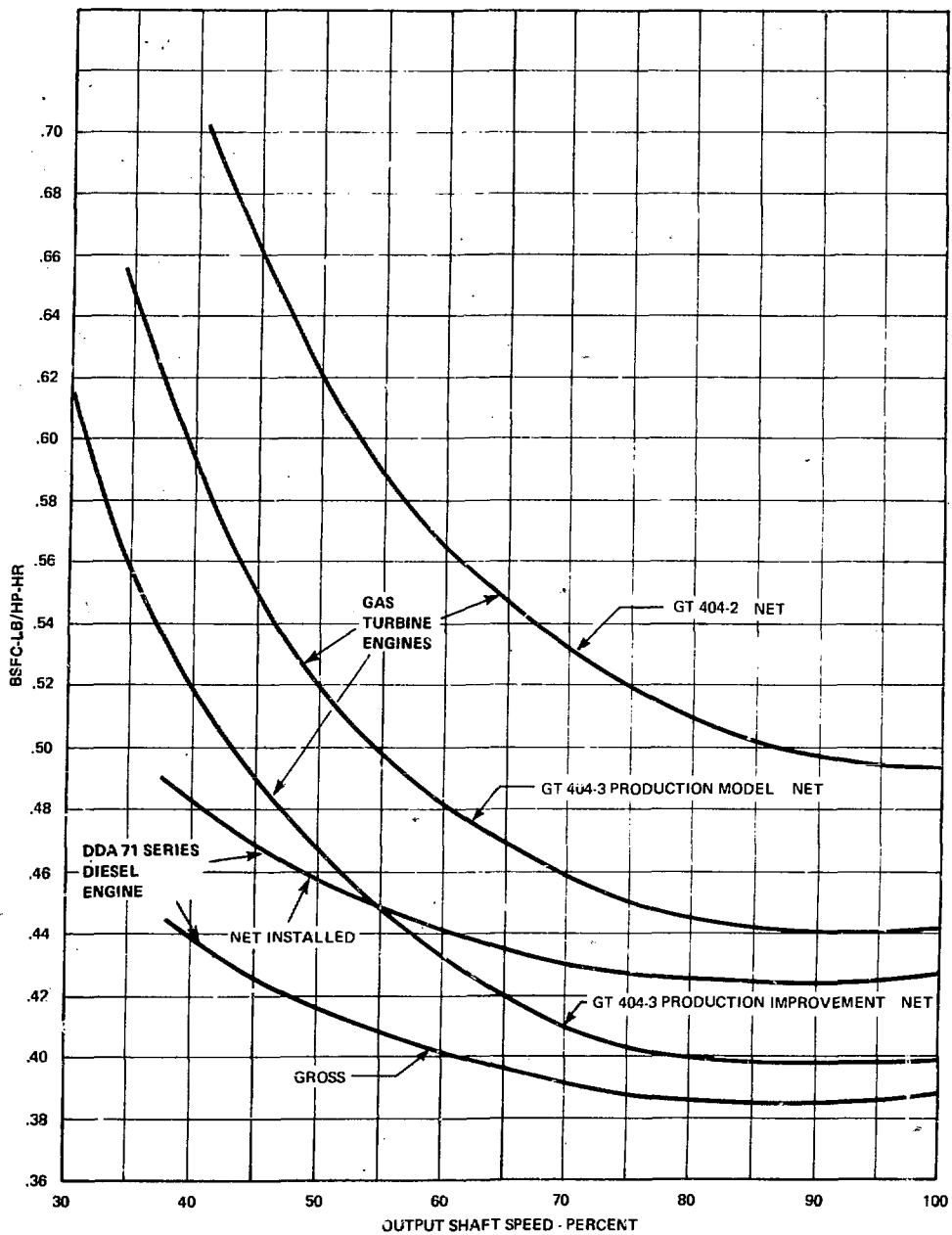


Figure 3
Full Power Fuel Consumption Comparison

of gross fuel consumption characteristics for the diesel and turbine engines. The gross fuel curve for the diesel engine indicates gross engine BSFC data and does not reflect the installed losses from inlet and exhaust, cooling system losses, and fan losses. The net installed curve for the diesel engine reflects a 10 percent adjustment to include these losses. The poorer fuel economy of the turbine engine at idle and part-load conditions can be observed from the data curves. Transit coaches can be idle up to 35 percent of the time, and virtually all of their operation is at low speeds. Hence, the advantageous aspects of the turbine must offset this fuel penalty to justify its use in transit coaches.

DDAD has several developments underway to reduce turbine engine fuel consumption and to adapt the engine to the transit coach application.

Projection of Current and Turbine Power Coach Costs

Table 2 projects the cost of incorporating turbine engines in transit coaches. The established baseline current (1973) bus fleet data is shown in the first column of Table 2. The 1976 coaches, however, are different in some major respects from the "typical" coach of the 1973 transit fleet. In addition, fuel costs have increased at a rate far higher than "normal" inflation (other costs, such as labor, overhead,

and repair parts have escalated at the same relative rate since 1973 and have been carried over in the cost projection). The second column of Table 2 reflects the projected cost of operating a "typical" 1976 model transit coach. The third column of Table 2 reflects the projected cost of operating a "turbine-powered" 1976 production coach in the 1976 operating environment.

In making operating cost projections for new coaches and engines, the data sources available are:

- Booz, Allen experience with the coach system, or component, on prototype vehicles, if the system or component is new
- Transit industry experience, if the component is the same as on current buses
- Engineering judgment and supplier claims, if the component is new to both the industry and Booz, Allen.

Table 2
Changes in Maintenance, Servicing and
Consumables Cost for Complete Replacement of Fleet

CATEGORIES	CURRENT BUS FLEET (1973) \$/MILE	CURRENT BUS AS BUILT IN 1976		MATURE TURBINE (GT404-3 PI) IN 1976 BUS	
		\$/MILE	BASIS	\$/MILE	BASIS
REPAIRS TO REVENUE EQUIPMENT:					
- ACCIDENTS	1.39	1.72	SAME	1.72	SAME
- POWER PLANT	1.83	2.27	8V-71	1.22	REDUCED MAINTENANCE
- BODY AND DOORS	1.68	2.08	SAME	2.08	SAME
- SUSPENSION/CHASSIS/ REAR AXLE/DRIVE TRAIN	1.47	1.82	SAME	1.82	SAME
- BRAKES	1.13	1.79	FMVSS 121 SYSTEM	1.60	REGENERATIVE ENGINE BRAKING
- TRANSMISSION	.82	1.26	V-730 MORE COMPLEX	1.26	SAME
- ELECTRICAL, STARTER, GENERATOR	1.11	1.38	SAME	1.38	SAME
- AIR CONDITIONING	.64	.79	SAME	.99	AC/SAME, ELECTRIC OR OTHER HEATING SYSTEM WILL COST MORE
- FRONT AXLE/STEERING	.49	.69	POWER STEERING	.69	SAME
- AIR SYSTEM	.37	.51	INCREASED CAPACITY FOR FMVSS 121 BRAKE SYSTEM	.51	SAME
TOTAL FOR REPAIRS	10.93	14.31		13.27	
FUEL	3.60	8.86	HIGHER FUEL COST	9.82	INCREASED FUEL CONSUMPTION
OIL	.16	.20	SAME	.21	REDUCED OIL CONSUMPTION
TIRES	1.28	1.98	INCREASED PERFORMANCE	1.84	REDUCED VEHICLE WEIGHT
TOTAL COST	15.97	25.35		24.94	

SOURCE: BOOZ, ALLEN APPLIED RESEARCH: CURRENT 1973 BUS COSTS BASED ON 1973 APTA OPERATING COST DATA, A SURVEY OF 16 MAJOR PROPERTIES, AND ENGINEERING ESTIMATES, WHICH UTILIZE TEST DATA, COMPONENT MANUFACTURER'S DATA, AND JUDGEMENT BASED ON OBSERVATIONS OF VEHICLES AND DRAWINGS OF EQUIPMENT. 1973 COSTS UPDATED TO 1976 DOLLARS.

Production Cost Impacts

Production cost and selling price are not directly relatable. Selling price is a function of production cost plus market conditions, profit margin, and the time selected by the manufacturer over which tooling and production engineering costs are written off. It is, therefore, unrealistic in this analysis to attempt to predict selling price. However, since DDAD has made available anticipated production costs for the turbine engine, based on 1976 dollars, consistent production cost estimates are possible and are shown in Table 3.

Table 3
Production Cost Comparison
Assuming Equal Amortization of Manufacturer Start Up Costs* 1976 Dollars

SYSTEM CATEGORIES	1976 PRODUCTION BUS**	TURBINE ENGINE 1976 BUS
• BODY STRUCTURE, DOOR AND GLAZING	10,527	10,527
• SUSPENSION, INCLUDING AXLES AND STEERING	7,705	7,705
• ENGINE	6,465	8,081
• DRIVELINE AND SUPPORT SYSTEMS (not affected by engine) - Transmission Accessories/Mounts/Wheels/Drive Shafts/Misc.	13,511	13,511
• DRIVELINE AND SUPPORT SYSTEMS (affected by engine) - Exhaust/Aspiration Air/Cooling	1,175	250
• ELECTRICAL	1,544	1,544
• AIR CONDITIONING/HEATING	4,772	5,877
• INTERIOR TRIM	4,292	4,292
• SEATS	3,948	3,948
• DESTINATION SIGNS/BUMPERS/MISC.	8,083	8,083
SUBTOTAL	62,022	63,818
G&A AND PROFIT (10%)	6,202	6,382
TOTAL	68,224	70,200

*These are cost, not selling price, estimates

**Current production bus used as a baseline is not typical of fleet-in use, but of a fully equipped 1976 model bus.
Source: Booz, Allen Applied Research

The turbine engine in the early years of production is projected to cost 20 to 25 percent more than 8V-71 diesel engine. The projected production cost of a turbine-engined transit coach, however, is projected to be \$70,000 in 1976 dollars, or only 2.9 percent more than \$68,224 for the typical diesel-powered coach currently in production.

CONCLUSIONS

The operational cost impact analysis indicates that transit coaches equipped with the production improvement gas turbine engines (GT-404-3 PI) will be essentially equal in total operating cost to current diesel-powered transit coaches, without including any added value for:

- . BSFC performance gain potential beyond 1981
- . Multifuel capability
- . Ability to accommodate heavier middle-distillate fuels
- . Improved performance
- . Conformance with environmental standards
- . Perceived environmental improvement
- . Reduced noise
- . Reduced gaseous emissions
- . Elimination of exhaust odor and smoke.

The tangible and intangible value/benefit of these attributes can, based on currently proposed regulations,

lower the operating cost of a turbine-powered transit bus by well over 1 cent/mile.

If the current government regulatory trends continue into the 1980's, the above factors will become increasingly important and realistic monetary value could be assigned to these benefits offered by the gas turbine engine. The current higher fuel consumption of the gas turbine will continue to be a disadvantage of the engine, particularly if the cost of fuel continues to escalate at a faster rate than other costs. The multifuel capability and the ability to burn a lower grade middle-distillate fuel may offset this disadvantage, especially if nonpetroleum-based fuels become readily available.

Beyond 1981, the fuel consumption of the gas turbine engine will continue to improve relative to the diesel engine. Improved technologies under development by engine manufacturers at this time in all engine component areas indicate that a fuel economy equal to that of the diesel engine at all speed/load ranges (with the exception of idle) may be achieved by the mid-1980's.

The major factors indicated by the cost impact analysis are the improved reliability and reduced maintenance costs of the gas turbine engine, offsetting the increased fuel

consumption. The turbine engine used in the analysis was the DDAD GT-404-3 PI model. This improved engine, with a rated MTBO of 10,000 hours, is tentatively scheduled for volume production in 1981, which is the earliest reasonable date to expect widespread use of turbine engines in transit coaches. This GT-404-3 engine is projected to have a transit service operational fuel consumption penalty of about 19 percent over contemporary diesel engines. The GT-404-3 PI is a dramatic improvement over the GT-404-2 turbine engine which was used in the GM Transbus prototype and which demonstrated a fuel penalty of 37 percent and a MTBO rating of 3,000 hours. The first production model engines, designated GT-404-3 PM, cannot be considered truly viable transit coach powerplants even with the 4,000-hour MTBO due to the projected 34 percent operational fuel penalty.

The procurement cost of the turbine engine, as stated previously, will initially be about 20 percent higher than the contemporary diesel engine. As the production volume of the engine is increased in the 1980's, the cost of the engine should be reduced and may be comparable with the diesel engine. Cost of the diesel, moreover, may be adversely affected by additional equipment required as a result of environmental regulations. The anticipated fuel consumption improvements will be attained by increasing the TIT and possibly the variable inlet geometry. The higher TIT will

necessitate incorporating improved materials in the nozzles and turbine blades and may also require blade cooling. These changes and the additional complexity of the variable inlet mechanism will increase the cost of the engine somewhat, but it must be presumed that the competitive nature of the engine market will maintain the selling price of the gas turbine within a range commensurate with the advantages it has to offer for vehicular power.

In summary, it appears that the gas turbine engine will become competitive and, in fact, superior to the diesel engine in transit coach operation in the 1980's.

"IMPROVED HEAVY DUTY GAS TURBINE ENGINE"

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ABSTRACT

An improved heavy duty gas turbine engine program plan has resulted from a study contract issued by NASA Lewis Research Center, Cleveland, Ohio using funds supplied by the Energy Research and Development Administration (ERDA). Program objectives are to demonstrate improved fuel economy [(213 mg/W·h (0.35) specific fuel consumption by 1981], conformance to current and projected Federal noise and emission standards and to demonstrate a commercially viable engine.

Study results show that increased turbine rotor and regenerator inlet temperatures, using ceramic materials, contribute the greatest amount to achieving fuel economy goals. Further, improved component efficiencies (for the compressor, gasifier turbine, power turbine and regenerator disks) can also show significant gains in fuel economy. Fuel saved in a 500,000 mile engine life, risk levels involved in development and engine related life cycle costs for fleets (100 units) of trucks and buses were used as criteria to select work goals for the planned program.

INTRODUCTION

Gas turbine engines offer an alternate power plant selection for trucks, buses, and other heavy duty engine applications. The diesel engine now is the prime power plant in these heavy duty vehicles and has established an impressive record for reliability, economy of operation and flexibility of operation. The gas turbine must also achieve a similar level of performance, durability and economy to become a complimentary power plant for heavy duty applications.

An industrial gas turbine (IGT) (see Figure 1) has been under development at General Motors Corporation for more than fifteen years and is rapidly approaching a stage of production readiness. This engine is being developed in two power sizes. The 404 engine is rated at 224 kW (300 hp) at 29.4°C (85°F) and 152 (500 ft) altitude. The 505 engine is rated at 291 kW (390 hp). A third member of this engine family is planned, the 605, to be rated at 347 kW (465 hp). Several million miles of bus, truck, boat, and other field service has now been accumulated on the 404/505 all metal engines. See installations in Figures 2 and 3. Valuable lessons on performance, durability and operating costs have been learned as introduction to production of this family of engines is considered.

Recent emphasis on fuel economy has produced improvements in fuel consumption for all types of highway and off-highway vehicles. Gasoline and diesel engines have incorporated new features (leading to improvements in fuel economy) and the IGT 404/505 gas turbine engines must also be improved to meet target fuel economies in the late 1970's and the 1980's. This consideration has led to the "Improved Heavy Duty Gas Turbine Engine" program initiated with NASA Lewis Research Center, Cleveland, and the Energy Research and Development Administration (ERDA) in 1976. This Phase I study program has three definite objectives. These are to

- Improve specific fuel consumption (sfc) from 274 mg/W·h (0.45) [installed] to 213 mg/W·h (0.35) [installed] by the year 1981

- Satisfy current and projected government noise and emission regulations
- Develop a commercially viable engine for use in trucks and buses

The suggested method to achieve improved fuel economy in the IGT is to increase turbine and regenerator inlet temperatures and improve component efficiencies of the compressor, turbines, and regenerator. The increased turbine and regenerator temperatures are to be achieved by the introduction of ceramic materials (non-cooled). An inherent advantage of the gas turbine is lower noise and emission characteristics than diesel and gasoline engines. Meeting current and projected noise and emission standards should be easily achieved. Demonstrating a commercially viable engine can only be achieved by future demonstrations of required performance, durability and costs. This first study phase has the objective of quantifying levels of improvement to be pursued, to plan technical activities, and establish a feasible program schedule to achieve these improvements.

In the Phase I study program many factors were considered. Utilizing existing engines and capitalizing on the availability of spare parts was important. This would promote early acceptance for production of new ceramic components and minimize program cost. Selection of work to be performed had to be established recognizing time, funding, and feasibility of technical accomplishment. The final selection of work had to be based upon reduced fuel consumption and life cycle costs (engine related) for a customer of the gas turbine engines. Evaluation of these life cycle costs involves assumptions

relative to fabricability, reliability and costs of ceramic parts which introduce an unknown 'risk' to the final assessments made in the study.

ANALYSIS PERFORMED AND RESULTS OF STUDY PROGRAM

Approach to Study

The approach to establishing a viable program was to make design assessments of candidate engine improvements, evaluate reductions in fuel consumption, assign production and maintenance costs for candidate engines and establish fuel saved and life cycle costs for a fleet of buses or trucks for each level of engine improvement. If fuel saved was significant and life cycle costs improved, the individual developments would be worthwhile and new components should receive serious consideration for early introduction to production engines.

Assessment of Candidate Improvements

An assessment of sensitivity to change in component efficiencies, engine pressure losses, engine heat rejection, cooling air requirements, engine operating temperature, regenerator inlet temperature, engine pressure ratio and leakages was conducted. Table 1 summarizes the sensitivities compiled showing the change in parameter required to obtain 1% improvement in specific fuel consumption (sfc). Having these sensitivities, technical analysis was applied to establish which parameters could be improved with development and improved materials. Two basic routes for improvement were selected as follows.

- Improved component efficiencies
- High turbine inlet temperature (through use of ceramic materials)

Changing pressure ratio would not be effective because lower pressure ratio leads to compromised part power engine operation. Only very small improvement in heat and power losses relative to the current engine can be achieved. Small reductions in cooling air flows and leakages will be strived for although increasing turbine inlet temperature will make it difficult to keep cooling air flows at the same level as in the current engine. Pressure losses are largely a function of overall engine geometry (turns, expansions, heat exchanger matrix and inlet and exhaust) and it is desirable to maintain maximum geometric similarity with the current engine because of costs and time to accomplish changes in the basic block, covers and gearbox.

Specific components selected for improvement were the compressor, turbine (compressor drive), the power turbine and the regenerator. It was known that increasing turbine inlet temperature would require lower engine airflow for a given engine horsepower output. Further, lower engine airflow leads to smaller compressor and turbine sizes with the associated loss in component efficiency from decreasing size. Figure 4 shows a projection of loss in compressor, turbine (compressor drive) and power turbine efficiencies with decreasing airflow. Also shown is the projected improvements achievable in these components with a rigorous development program. Two to three years will be required to fully develop these advanced technology components. Figure 5 shows a regenerator effectiveness plot. Improved effectiveness is achieved by decreasing airflow while maintaining the same frontal airflow area in the current regenerator matrix. The lower curve represents the current metal matrix, the center curve is a current thin wall

(0.0889 mm [0.0035 in.] to 0.1143 mm [0.0045 in.]) triangular ceramic (AS) matrix, and the upper curve represents a thin wall rectangular matrix (projected to require two to three years to develop) which has the potential of both higher effectiveness and lower matrix pressure drop. A prime benefit achieved in the change from metal to ceramic regenerators is the increased temperature capability of the ceramic. The current metal regenerator is limited to a maximum steady state temperature of 774°C (1425°F) and a transient temperature of 913°C (1675°F). The ceramic regenerator can achieve a 982°F (1800°F) steady state temperature and a transient temperature of 1093°C (2000°F). This increased temperature capability is particularly important at part power operation of the engine since the best sfc is achieved by maintaining turbine inlet temperature constant over the power range of the engine and letting the regenerator inlet temperature increase as power level decreases. In the current engine, turbine inlet temperature is reduced starting at approximately 80% power (to accommodate the 774°C [1425°F] metal limit), whereas the ceramic regenerator with a 982°C (1800°F) capability will not require reduced part power turbine inlet temperature until a level of near 1260°C (2300°F) is obtained. In addition, the lower conduction loss of the ceramic provides a favorable part power effectiveness.

In summary, improvements in component efficiencies and regenerator effectiveness can substantially improve the engine fuel consumption and should be vigorously worked on in the proposed program.

HIGHER TURBINE INLET TEMPERATURE

The current engine turbine inlet temperature is 1002°F (1835°F).

Table 1 shows that a 12°C (22°F) increase will give a 1% improvement in sfc at this level of turbine inlet temperature. This is not a linear function, but increasing temperature does provide improved sfc over a considerable range of temperature. Changing this parameter yields the greatest single improvement in sfc so long as it is not necessary to change another parameter (such as cooling air from the compressor). The incorporation of ceramic materials in the high temperature components offers the means for achieving the improved sfc. Silicon carbide and silicon nitride are the two leading candidate ceramic materials for the high temperature vanes, blades, combustor, turbine inlet plenum and the stationary turbine tip shroud rings.

Analysis of metal high temperature parts was performed to establish limiting turbine inlet temperatures which render the metal no longer usable. In metal components it was found that either chemical stability (oxidation, sulfidation), creep strength or tensile strength were a limiting factors at given operating temperatures. Figure 6 shows limits for components plotted against gasifier turbine inlet temperature, regenerator inlet temperature and power turbine inlet temperature. The metal regenerator, gasifier turbine inlet vanes and tip shrouds are the initial metal components that should be replaced with ceramic materials. Table 2 shows the turbine inlet temperatures at which other metal components must be replaced. Figure 7 shows these ceramic components in the IGT engine. Table 2 identifies logical engine development temperature levels

for advancing to the target sfc levels. These engine temperature levels (1038°C [1900°F], 1132°C [2070°F], 1204°C [2200°F] and 1271°C [2500°F]) were used in the balance of the study to calculate engine performance, design of components and for cost analysis.

IMPROVED ENGINE PERFORMANCE LEVELS

Initial planning specified use of the current engine with the introduction of ceramic components and improved component efficiencies as they became available. Assessment of component efficiency improvements suggest a two to three year development period for compressor, turbine and regenerator efficiency development. A thin wall ceramic regenerator can be introduced into the current engine along with ceramic vanes, tip shrouds, turbine blades and a ceramic turbine inlet plenum. Thus, the current engine can be used for development at turbine inlet temperatures to 1132°C (2070°F). (At this temperature level, the power turbine must be replaced because of limiting blade creep strength and turbine aerodynamic loading.) This permits two to three years development before a significant flow path change can be accomplished. A new compressor and turbine flow path would then be introduced with improved component efficiencies at the 1204°C (2200°F) gas turbine operating temperature and this technology would be applicable to 1371°C (2500°F). This timing dictated that a technology discontinuity occur between the 1132°C (2070°F) and 1204°C (2200°F) engines.

For comparison of fuel savings and life cycle costs, the engine horsepower selected for engine sizing was 224 kW (300 hp) (the same as the 404-3 engine). Performance calculations were then made using the preceding turbine inlet temperatures, component efficiency improvements and constant horsepower. Table 3 shows the airflow for constant horsepower, sfc at the various turbine inlet temperatures, and other engine component parameters. It is noted that the target 213 mg/W·h (0.35) sfc is achieved at a temperature level between 1204°C (2200°F) and 1371°C (2500°F). Figure 8 shows how sfc varies for the baseline engine and the increased cycle operating temperatures over a complete operating power range from 0 to 224 kW (300 hp). It is noted that the ceramic regenerator temperature limit yields a much flatter sfc curve in the 50% (112 kW [150 hp]) to 100% (224 kW [300 hp]) power range where most heavy duty engine usage occurs. Also, it is worthy of note that the turbine is very stable to low power levels and yields a torque-speed curve that is ideal for many off-design operating conditions. Figure 9 shows this torque characteristic (compared to a diesel engine) which experience has shown to be very valuable on icy roads, wet roads, for use on grades and for selecting a less complex transmission when compared to a diesel engine.

An engine feature maintained in all the engines analyzed is the power transfer unit. This device has several advantages as follows.

- Braking capability equal to maximum engine horsepower
- Single shaft operation capability (particularly valuable in gen set installations)
- Power turbine overspeed control

- Lower transient turbine temperature excursions compared to most turbine engines

This power transfer feature is considered to be important to help the turbine successfully compete against current heavy duty power plants.

All engine performance presented is shown for an SAE standard day rating of 29.4°C (85°F) and 152 (500 ft) altitude. Gas turbines are sensitive to ambient inlet temperatures as shown in Figure 10. Attention is directed to the feature that lower ambient inlet temperatures produce increased power and lower specific fuel consumption for a given turbine engine. The average United States temperature is 13.3°C (56°F).^{*} This lower average ambient air temperature (compared to the SAE Rating temperature) means an average improvement in fuel economy of approximately 5% above the values shown herein.

PERFORMANCE — NOISE AND EMISSIONS

Current and projected regulations on noise and emissions for heavy duty vehicles/engines are shown in Table 4. These regulations are subject to change, but can materially affect the current gasoline and diesel heavy duty power plants if they are not changed. The production diesel

^{*}Reference: The Weather Almanac, 1974, edited by James E. Ruffner and Frank E. Bair, p. 189.

engine will find it difficult to meet the 1980 and 1983 California standards without compromising fuel economy. However, it is believed that Federal standards may override state standards and may continue to be much less stringent than state standards.

The noise standards may become restrictive for the diesel with encapsulation a possible solution to the truck noise problem. This will cause increased acquisition and maintenance costs.

Noise and emission values were calculated for each of the four study engine cycles [1038°C (1900°F), 1132°C (2070°F), 1204°C (2200°F) and 1371°C (2500°F)]. Because of reduced airflow and lower exhaust velocities, noise was found to be reduced below current turbine engine noise values of 75-77 dBA at 15.2 m (50 ft) sideline measuring locations. The exhaust noise will be reduced by over 5 dBA at the 1371°C (2500°F) turbine inlet temperature. Even the 75 dBA noise at 15.2 m (50 ft) sideline regulations of some cities and states can be met by these improved turbine engines (with no special noise insulation or encapsulation). It is noted that the normal air inlet filter and regenerator disks are very effective noise suppressors for compressor and turbine noise.

Emission values obtained at each of the improved engine cycle temperatures are shown in Table 5.

These emission levels were obtained for the 13 mode Federal Heavy Duty Diesel Cycle. HC, CO and smoke all reduce as a result of higher cycle temperatures. The primary pollutant of concern is NO_x. This would

only be of concern if a Federal Standard of less than 10.0 is legislated in 1979 and will only very slightly exceed the projected 1980 California standard. Work for pollutant (NO_x) reduction will be performed in the proposed program to achieve a minimum NO_x margin of 25% to accommodate deviations obtained in production combustors.

VEHICLE CHARACTERISTICS WITH IMPROVED ENGINES

Having performance established for the improved engines, typical truck and bus usage with these power plants was evaluated. The truck selected was a 31,751 kg (70,000 lb) gross weight unit with 9.5 m^2 (102 square foot) frontal area, 0.2983 tire revolutions per metre (480 revolutions per mile), a 4.886 axle ratio and a Fuller RT9509A nine speed transmission. The bus was a typical Greyhound of 16,329 kg (36,000 lb) gross vehicle weight, 6.9 m^2 (74 square feet) frontal area, 0.3088 tire revolutions per metre (497 revolutions per mile), an axle ratio of 4.330 and a DDA HT-740CT four speed transmission. In each vehicle, appropriate accessory loads and driveline losses were assigned for various vehicle speeds. Each vehicle was then run through the General Motors vehicle computer analysis for typical runs from Los Angeles to Salt Lake City and from Chicago to Boston. Figure 11 shows the plots of kilometres per litre (miles per gallon) versus rotor turbine inlet temperature for both the truck and the Greyhound bus. Note is made of the discontinuity in the curves between 1132°C (2070°F) and 1204°C (2200°F) as predicted earlier in the paper.

Particular note is made in the improvement made in fuel consumption. Figure 12 shows fuel used in a typical 804,672 kilometres (500,000 mile) engine life for the baseline engine and fuel savings for the study engines.

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The gallons saved for one truck with a 1204°C (2200°F) engine multiplies to 3,060,000 gallons (72,857 barrels) for a 100 truck fleet. It is believed that fuel savings shown are extremely significant and will have a meaningful impact on conservation of petroleum products.

LIFE CYCLE COSTS

Engine drawings were generated for each engine cycle temperature (1038°C [1900°F], 1132°C [2070°F], 1204°C [2200°F] and 1371°C [2500°F]) with emphasis placed on ceramic components and supporting metal parts. Figure 13 is a general arrangement drawing showing the 1371°C (2500°F) configuration study engine. Sketches of each unique part were made and pricing for ceramic components was performed by the Carborundum Company, Niagara Falls, New York. Costs for ceramic parts to produce 6000 engines per month were supplied. Harrison Radiator Division of General Motors Corporation supplied quantity costs for regenerators and seals and Detroit Diesel Allison priced all other engine parts required. These prices were then used to generate engine acquisition and maintenance costs in the life cycle cost study.

Life cycle costs were calculated for both trucks and buses. A 100 tractor truck fleet and a 100 coach bus fleet was chosen for these studies. A 804,672 km (500,000 mile) engine life was used. An average of 14,484 km per month (9,000 miles per month) was used for the trucks and 26,554 km per month (16,500 miles per month) for the buses. A mature

engine schedule of maintenance was applied for each engine. In this maintenance schedule it was assumed that ceramic and metal parts would have the same reliability since no data base is available for ceramic parts in production gas turbine engines. Road applied maintenance included oil and fuel. Shop applied maintenance included predicted items (oil change, oil filter, fuel filter, and other predicted component parts). Sixty percent of shop maintenance was in-frame, forty percent was out-of-frame. Of the 40% out-of-frame, 10% was overhaul and 5% was major repair. Fuel costs were calculated based upon \$0.40 per gallon (current price) and \$0.60 per gallon (projected 1980 price).

Data gathered for vehicle performance, maintenance material and labor and engine acquisition costs were then entered into a life cycle cost computer program and actual life cycle costs calculated.

The current all metal 404 engine was used as a baseline engine for comparison purposes with the improved engines. Figure 14 shows a plot of life cycle costs relative to the baseline engine life cycle costs. This figure is for the line haul truck fleet. Figure 15 is for the Greyhound bus fleet. Note the effect that \$0.40 and \$0.60 fuel has on relative costs. Note the discontinuity between 1132°C (2070°F) and 1204°C (2200°F) caused by the introduction of components with improved efficiencies. Figure 16 summarizes gains in fuel savings and costs for the study engines.

The most obvious conclusion drawn from the life cycle cost analysis was that only 1% improvement is achieved in going from a 1204°C (2200°F)

turbine rotor inlet temperature to a 1371°C (2500°F) turbine rotor inlet temperature. This is caused by fuel cost being the primary cost driver and the fact that so little improvement in specific fuel consumption is achieved in this change in operating temperature level.

It is observed that high risk is introduced by the use of ceramic materials at the 1371°C (2500°F) cycle temperature level. At this temperature level, the single stage gasifier turbine rotor blades, two stages of power turbine rotor blades and all three stator vane rows (one gasifier and two power turbine stages) are made of ceramic materials.

Also, variations in costs of ceramic components can impact severely the acquisition costs and thus total life cycle costs. Analysis was performed where ceramic components were priced 10% to 25% over those levels used in the basic analysis and it was found that total life cycle costs at 1371°C (2500°F) actually increased over the baseline engine costs. Costs are very important and must be closely monitored as engine development proceeds. (It is acknowledged that ceramic component prices could also be less than the base prices obtained.)

Based upon the low improvement in life cycle costs and high risk with today's technology with ceramic materials in going from 1204°C (2200°F) to 1371°C (2500°F) it was decided that the program recommended would have a maximum cycle temperature that would produce the target sfc of 213

mg/W·h (0.35) at 100% power. It is also recommended that flexibility be maintained in the program to permit re-evaluation of this conclusion as progress is made toward meeting program objectives and experience is gained in working with ceramic flow path components.

PROGRAM PLAN EVOLVED FROM STUDY

Two primary means of improving fuel economy evolved from the study conducted. These were:

1. Improved component efficiencies are feasible and can achieve approximately one-third the specific fuel consumption improvement desired. Improved efficiency in the compressor, both gasifier and power turbines and in the regenerator should receive emphasis in the program.

2. Increased engine cycle and regenerator inlet temperatures are feasible using ceramic materials and can achieve approximately two-thirds the specific fuel consumption improvement desired.

This is achieved by increasing turbine rotor inlet temperature from 1062°C (1835°F) to 1241°C (2265°F) and regenerator inlet temperature from 774°C (1425°F) to 982°C (1800°F).

Naturally, it is desirable to initiate this program using the engine being prepared for production and introduce improvements as soon as possible.

Steps to achieve engine fuel consumption improvements as soon as possible are suggested. The first step is to introduce ceramic regenerators, ceramic gasifier turbine vanes and ceramic stationary tip shrouds into the engine and increase turbine rotor inlet temperature to 1038°C (1900°F).

The second step is to introduce additional ceramics into the current engine for the turbine inlet plenum and ceramic gasifier turbine rotor blades and increase turbine rotor inlet temperature to 1132°C (2070°F) (the current power turbine temperature limit for minimum demonstration testing).

While these two steps are in progress, improved efficiencies for the compressor, turbine and regenerator disks should receive emphasis with rig testing and engine evaluations as practical. Estimates of time required to achieve these steps is two to three years. (This recognizes funding and technical constraints.) The highest risk element is considered to be the rotating gasifier turbine rotor with ceramic blades. It is recommended that an alternate air-cooled rotor be carried as a parallel program in case development problems with the ceramic bladed rotor assembly are not solved in time to meet the proposed schedule.

The third step in the program will be an advanced engine with new, improved compressor, turbine and regenerator components and maximum use of ceramics. This engine would demonstrate the target sfc of $213 \text{ mg/W}\cdot\text{h}$ (0.35). Table 6 summarizes the steps for introduction of ceramic components. Table 7 summarizes the temperature steps, engine airflow, horsepower and specific fuel consumption that will be obtained. Test stand and vehicle demonstrations will be made with each of the last three steps in the program. Figure 17 shows the predicted sfc versus power to be achieved by the recommended engines in the proposed program. The timing required for the third step is recommended to be three years. However, it is desirable to perform the

design and initial fabrication of the advanced engine during the third year of the program thus giving a five year overall program. Figure 18 shows a schedule with various steps identified for reference. On this schedule, the vehicle demonstrations with the improved engines will be made either in a line-haul truck or a bus as appropriate.

SUMMARY

This study program has identified a program plan which should meet the objectives of improving sfc from 274 mg/W·h (0.45) to 213 mg/W·h (0.35) by 1981, and should produce demonstration engines meeting Federal noise and emission regulations. Sufficient durability running of engines is planned in the program to demonstrate components that can progress to a commercially viable engine for use in trucks and buses.

Technology generated in this program will be applicable to automotive turbines and will represent a significant step forward for commercial gas turbine engines.

ACKNOWLEDGMENTS

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Beuford C. Hall	Technical Director for Phase I Study Program
Roger C. Dycus	Project Direction and Coordination
Samuel R. Thrasher	Regenerator Design and Analysis
James R. Wooten	Performance Analysis
Terry Knickerbocker	Performance Analysis
Carlton Curry	Life Cycle Cost Analysis
John Wertz	Design Analysis
James Lunsford	Design Analysis
John Hayes	Design Analysis
David Decker	Design Analysis
Dr. Peter Heitman	Materials

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Mr. William Geotte	Technical Program Manager - (Replaced Mr. Beremond 12/1/76)

Table 1

IGT 404 PM Baseline Engine Performance Parameters and Sensitivities

Turbine Inlet Temperature = 1002°C (1835°F)

Brake Specific Fuel Consumption, BSFC = 268 mg/W·h (0.441 lb/bhp-hr)

<u>Parameter Name</u>	<u>Parameter Value</u>	<u>Parameter Sensitivity⁽¹⁾</u>
<u>Component Efficiencies</u>		
Compressor	82.4%	+0.66
Combustor	99.9%	+1.00
Gasifier Turbine	87.0%	+0.92
Power Turbine	89.7%	+1.32
Regenerator Effectiveness	89.8%	+0.92
<u>Pressure Losses ($\Delta P/P$)</u>	11.56%	-0.80
<u>Leakage and Cooling Flow</u>		
Turbine Piston Ring Leakage	0.17% ⁽²⁾	
Turbine Rotor Cooling	1.54%	
	1.71%	-0.62
Overboard Leakage (Splitlines and Main Bearing Seals)	1.21%	-0.74
Regenerator Leakages	5.12%	-1.11
Block and Flow Path Cooling	3.56%	-1.66
Gasifier Turbine Nozzle Cooling	1.00%	-1.66
<u>Power Losses</u>		
Gasifier Driven Accessories (Fuel Pump, Oil Pump, Regenerators)	6.7 kW (9.0 hp)	-2.5 kW (-3.3 hp)
Gearbox Windage and Friction	9.0 kW (12.11 hp)	-2.2 kW (-3.0 hp)
Gasifier and Power Turbine Bearings	-2.2 kW (2.91 hp)	-2.4 kW (-3.15 hp)
	17.9 kW (24.02 hp)	
<u>Heat Losses</u>		
Into Lubrication System	5.3 kW (300 BTU/min)	-7.1 kW (-405 BTU/min)
External	10.6 kW (600 BTU/min)	-7.1 kW (-405 BTU/min)
<u>Engine Pressure Ratio</u>	4.0	-0.34
<u>Engine Cycle Temperature</u>		
Turbine (Rotor) Inlet	1002°C (1835°F)	12°C (+22°F)

(1) Sensitivities shown are the change in the parameter value to produce a 1.0% reduction in engine BSFC

(2) Values are percent of supply point airflow

Table 2
Introduction of Ceramic Components into Study Engines

Engine Cycle Temperature, °C (°F)	1002 (1835)	1038 (1900)	1122 (2070)	1204 (2200)	1371 (2500)
Regenerator Disk and Seals	M	C	C	C	C
Gasifier Nozzle and Tip Shroud	M	C	C	C	C
Plenum	M	M	C	C	C
Gasifier Rotor Blades	M	M	C	C	C
Single Stage Power Turbine	M	M	M		
Two Stage Power Turbine Nozzles				C	C
Two Stage Power Turbine Blades				M	C
Exhaust Diffuser	M	M	I	C	C
Combustor	M	M	M	M	C

M - metal component

I - improved metal component

C - ceramic component

Table 3
Performance Analysis of Baseline and Study Engines (100% Power)

CYCLE PARAMETER AT MAX POWER						
Gasifier Rotor Inlet Temperature	°C	1002*	1038	1132	1204***	1371***
	(°F)	(1835)	(1900)	(2070)	(2200)	(2500)
Compressor Pressure Ratio		4.00	4.00	4.00	4.00	4.00
Compressor Airflow	kg/sec	1.57	1.57	1.33	1.15	0.96
	(lb/sec)	(3.45)	(3.45)	(2.93)	(2.54)	(2.12)
Compressor Efficiency	(%)	82.4	82.4	82.2	82.9	82.2
Gasifier Turbine Efficiency	(%)	87.0	87.0	86.4	87.1	86.6
Power Turbine Efficiency	(%)	89.7	89.7	88.9	89.2	88.8
Burner Efficiency	(%)	97.6	97.6	97.6	97.8	97.8
Regenerator Effectiveness	(%)	88.9	90.6	93.3	96.2	97.0
Regenerator Leakage	(%)	4.92	4.92	5.75	5.26	6.29
Turbine Cooling (Wheels & Seals)	(%)	1.71	1.83	2.63	3.04	3.66
Block & Turbine Shroud Cooling	(%)	3.56	3.56	3.56	3.56	3.56
Gasifier Vane Cooling	(%)	1.00	0.0	0.0	0.0	0.0
Overboard Leakage	(%)	1.21	1.21	1.42	1.29	1.54
Total Pressure Loss**	(%)	11.8	13.0	12.5	10.3	9.5
Mechanical Losses	kW	17.9	17.9	17.9	15.9	15.9
	(hp)	(24.0)	(24.0)	(24.0)	(21.4)	(21.4)
Shaft Horsepower	kW	224	224	224	224	224
	(hp)	(300)	(300)	(300)	(300)	(300)
Specific Fuel Consumption	mg/N·hr	268.25	258.52	245.74	217.16	209.25
	(lb/hp-hr)	268	259	246	217	209

*Baseline IGT 404

**Includes inlet and exit loss

***Technology advancement

Table 4
Noise and Emission Regulations

Noise*	1977	1978	1979	1980	1982	1983
dBA at 15.2 m (50 ft) Greater than 4536 kg GVW (10,000 lb)	-	83	83	83	-	-
Emissions—kg/kW-hr (gm/bhp-hr)						
<u>Federal</u>						
HC	-	-	0.002 (1.5)	0.002 (1.5)	-	-
NO _x	-	-	-	-	-	-
HC + NO _x	0.02 (16)	0.02 (16)	0.01** (10)**	0.01** (10)**	**	**
CO	0.05 (40)	0.05 (40)	0.0003** (0.25)**	0.0003** (0.25)**	**	**
<u>California</u>						
HC	0.001 (1.0)	0.001 (1.0)	0.002 (1.5)	0.001 (1.0)	0.001 (1.0)	0.0007 (0.5)
NO _x	0.010 (7.5)	0.010 (7.5)	0.010 (7.5)	-	-	-
HC + NO _x	0.007 (5.0) ⁺	0.007 (5.0) ⁺	0.007 (5.0) ⁺	0.008 (6.0) ⁺⁺	0.008 (6.0) ⁺⁺	0.006 (4.5)
CO	0.034 (25)	0.034 (25)	0.034 (25)	0.034 (25)	0.034 (25)	0.034 (25)

*Many City and State Regulations are more rigid (75 dBA in 1980)

**To be established

⁺Option

⁺⁺Optional 5HC + NO_x with no separate HC std.

Table 5
Emissions for Study Engines

kg/kW-hr (gm/bhp-hr)	Standards		1038°C (1900°F)	1132°C (2070°F)	1204°C (2200°F)	1371°C (2500°F)
	Federal	California				
	1977	1980				
HC	-	0.001 (1.0)	0.0004 (0.3)	0.0003 (0.2)	0.0004 (0.3)	0.0003 (0.2)
NO _x	-	-	0.005 (3.9)	0.007 (5.3)	0.005 (4.0)	0.008 (6.0)
HC + NO _x	0.022 (16.0)	0.008 (6.0)	0.006 (4.2)	0.007 (5.5)	0.006 (4.3)	0.008 (6.2)
CO	0.054 (40)	0.034 (25)	0.002 (1.7)	0.002 (1.2)	0.002 (1.7)	0.001 (1.0)

Table 6
Introduction of Ceramic Components in Recommended Program

Gasifier Rotor Inlet Temperature	°C	1002	1002	1038	1132	1241
	(°F)	(1835)	(1835)	(1900)	(2070)	(2265)
Regenerator Hotside Inlet Max Temperature	°C	774	866	899	954	982
	(°F)	(1425)	(1590)	(1650)	(1795)	(1800)
Introduction of Ceramic Components						
Regenerator Disk and Seals		M	C	C	C	C
Gasifier Nozzle Assembly		M	M	C	C	C
Gasifier Tip Shroud		M	M	C	C	C
Plenum		M	M	M	C	C
Gasifier Rotor Blades		M	M	M	C	C
Two Stage Power Turbine Nozzle		*	*	*	*	C
Two Stage Power Turbine Rotor Blades		*	*	*	*	M
Exhaust Diffuser		M	M	M	M	C
Combustor		M	M	M	M	C

M - Metal

C - Ceramic

* - Single Stage Metal

Table 7
Recommended Program Engines

Performance Demonstration Engines
 29.4°C (85°F) 152 m (500 ft) altitude

		Baseline Engine	Initial Tests	Step 1	Step 2	Step 3
Gasifier Rotor Inlet Temperature	°C	1002	1002	1038	1132	1241**
	(°F)	(1835)	(1835)	(1900)	(2070)	(2265)
Compressor Airflow	kg/sec	1.56	1.56	1.56	1.56	1.37*
	(lb/sec)	(3.45)	(3.45)	(3.45)	(3.45)	(3.03)*
Shaft Horsepower	kW	224	219	231	261	287
	(hp)	(300)	(293)	(310)	(350)	(385)
Specific Fuel Consumption	mg/W·h	274	268	262	243	213
	(lb/hp-hr)	0.45	0.44	0.43	0.40	0.35
Test Stand Data		Yes	Yes	Yes	Yes	Yes
Vehicle Demonstration				Yes	Yes	Yes

*Improved Technology Components

**Demonstration of HDGTE Performance Goal [213 mg/N·h (0.35) sfc]

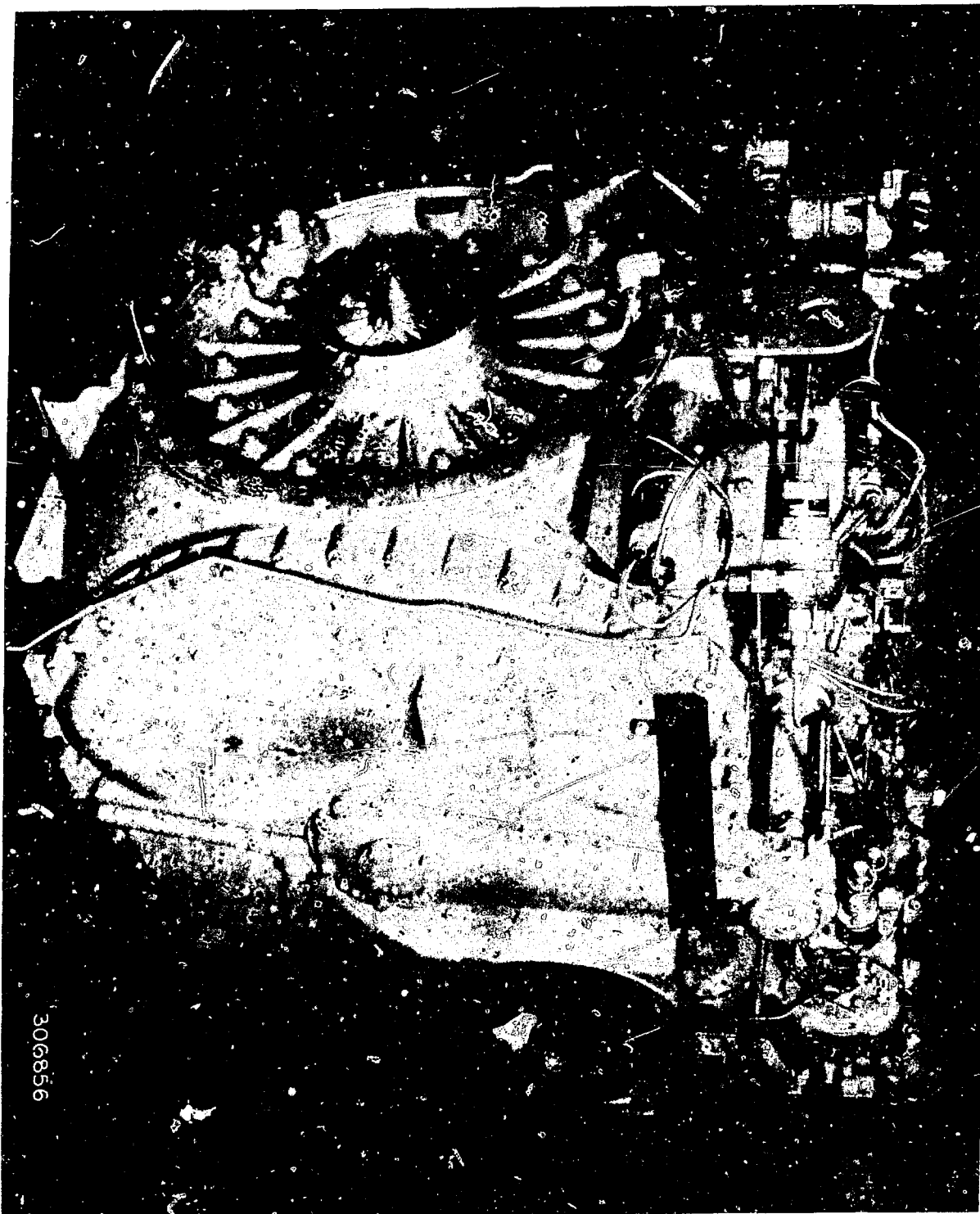


Figure 1. Detroit Diesel Allison Industrial Gas Turbine.



Figure 2. Chevrolet Titan 90 truck powered by GT 505-3 engine.

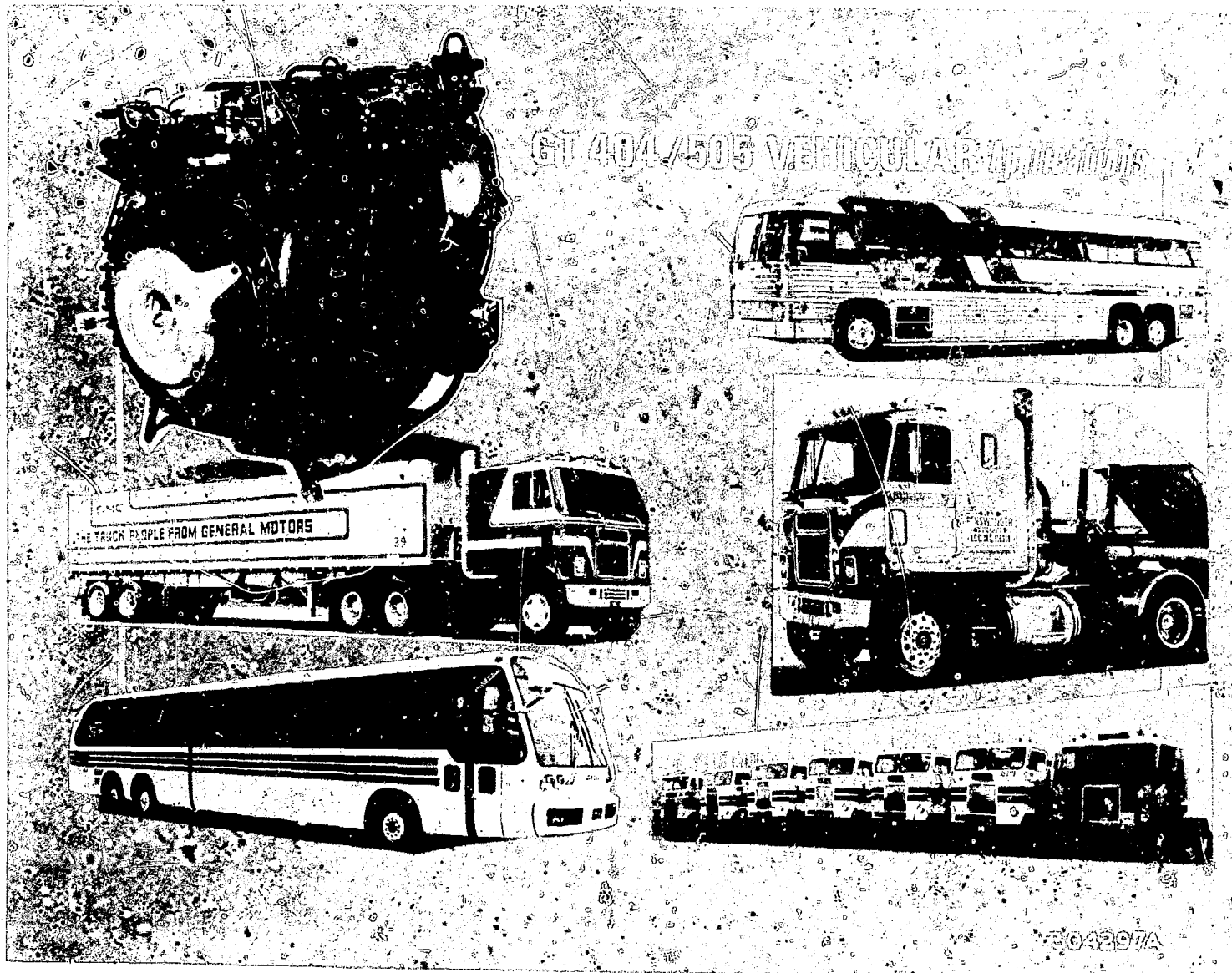


Figure 3. GT 404-3/505-3 applications.

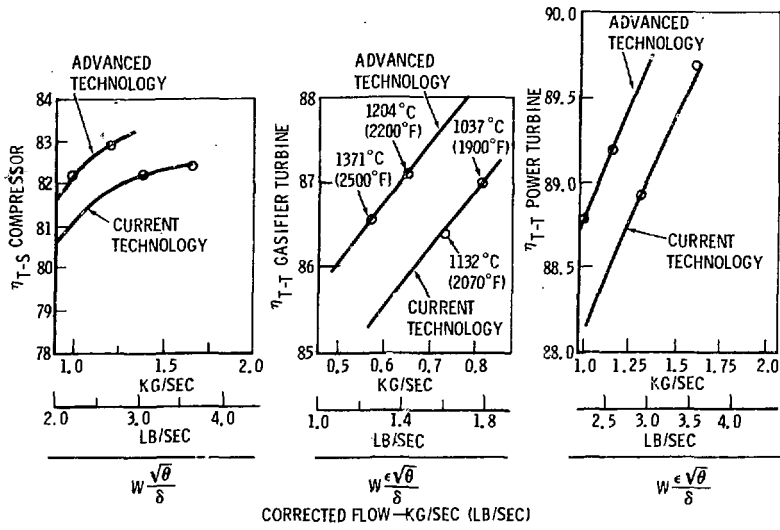


Figure 4. Effects of size on component efficiency.

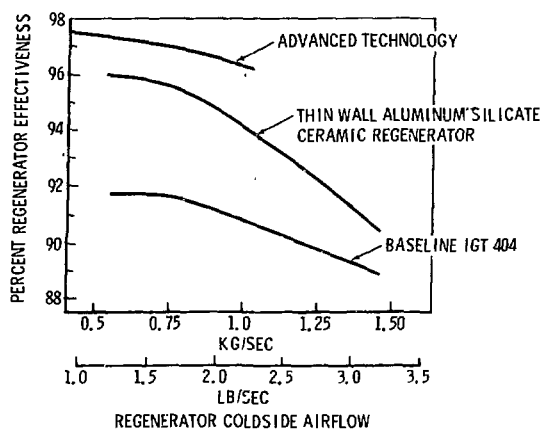


Figure 5. Effectiveness for metal and ceramic regenerator disks.

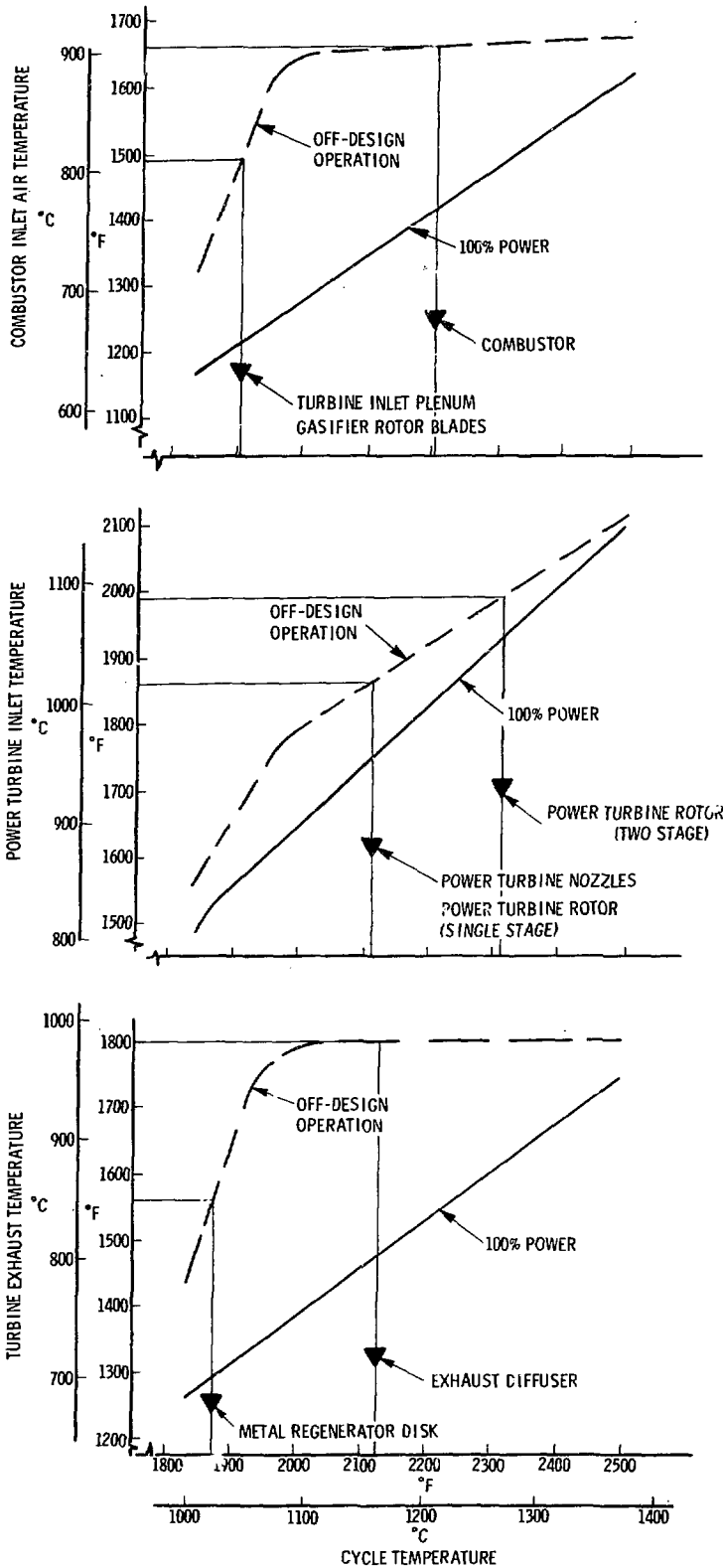


Figure 6. Limiting turbine inlet temperatures for metal engine.

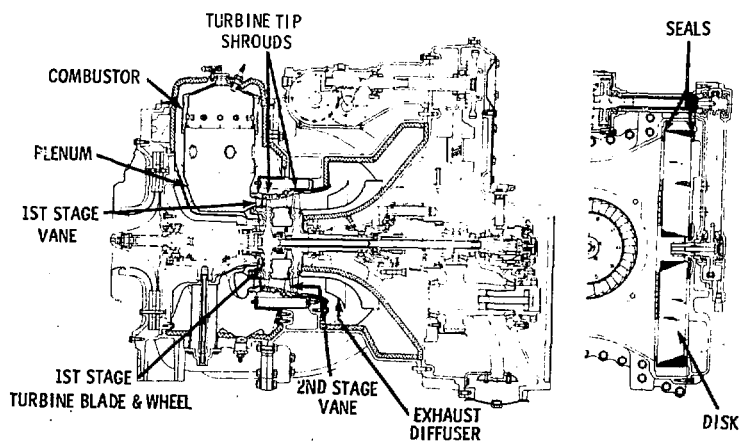


Figure 7. Applications of ceramics to IGT.

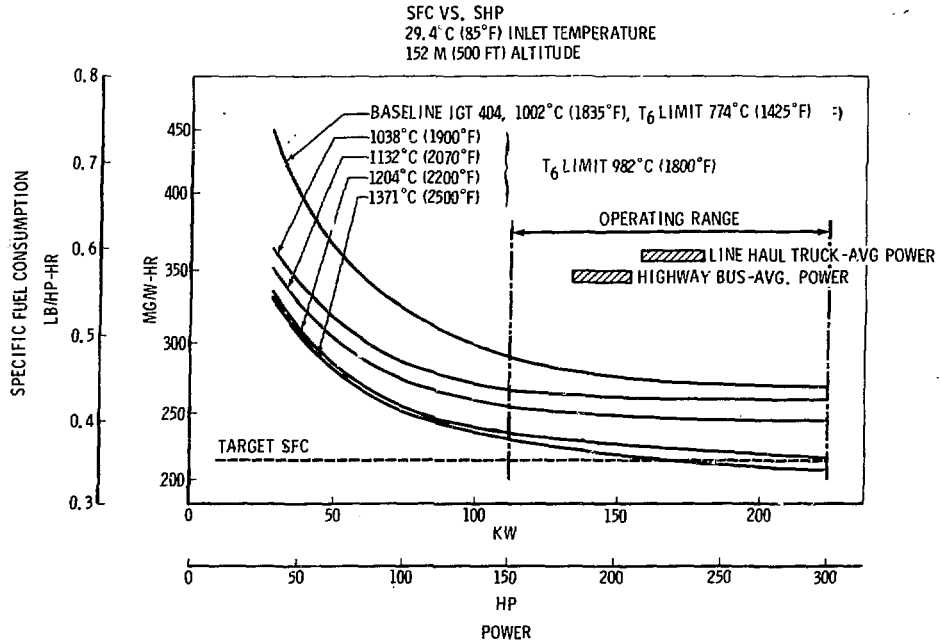


Figure 8. SFC versus horsepower for baseline and study engines.

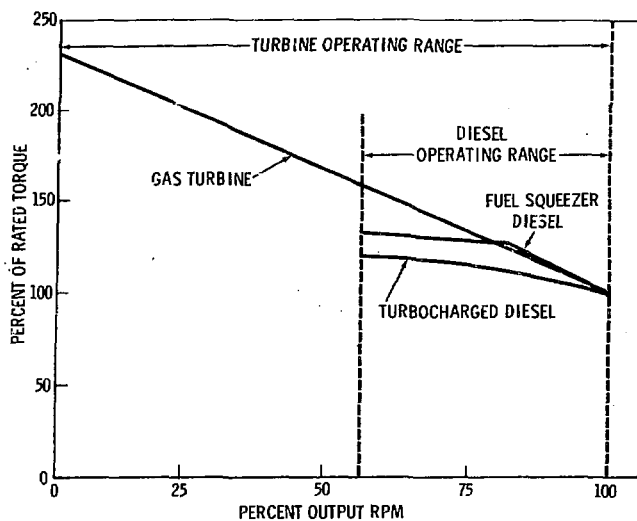


Figure 9. Comparison of maximum available diesel and gas turbine torque.

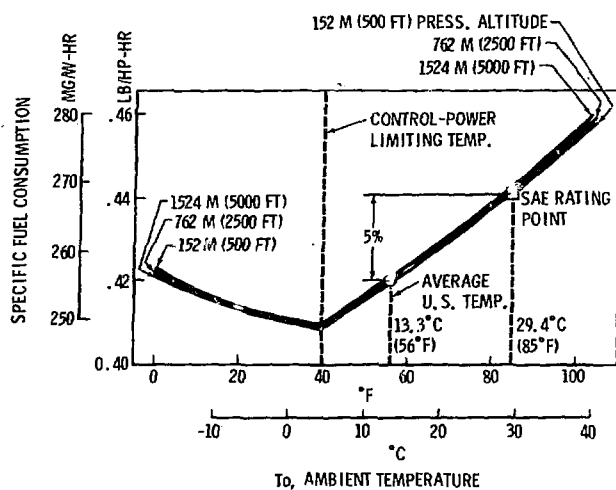


Figure 10. Inlet temperature for a heavy duty gas turbine engine.

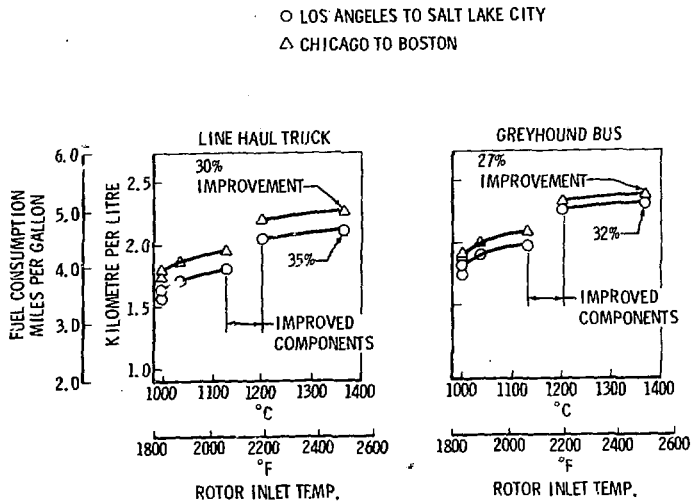


Figure 11. Kilometres per litre (mi per gal) vs increasing cycle temperature for study engines.

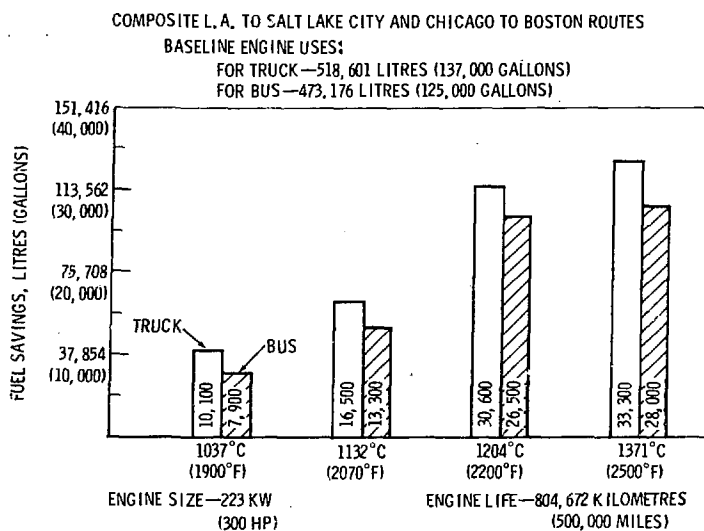


Figure 12. Fuel savings for study engines.

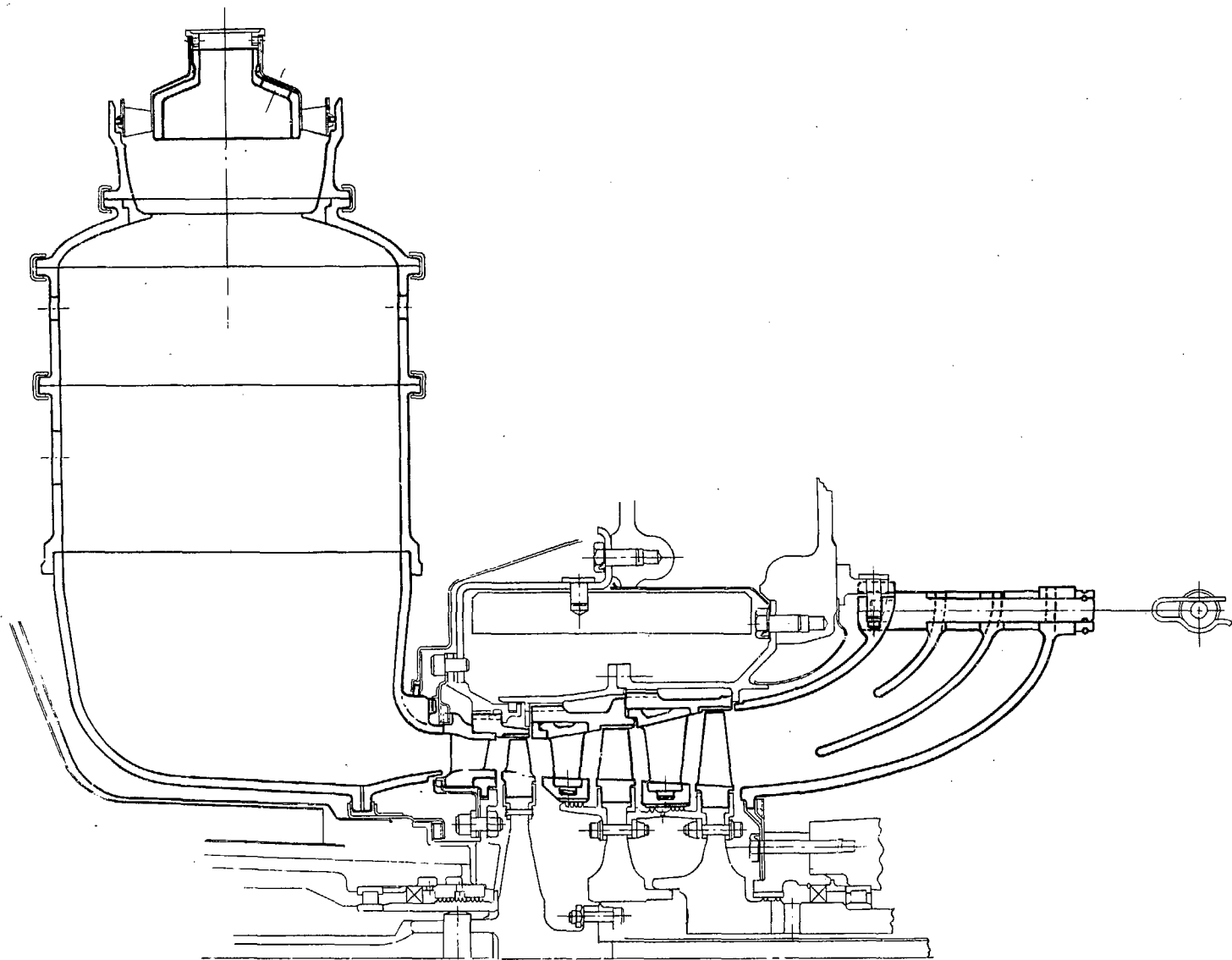


Figure 13. Combustor and turbine general arrangement for 1371.1°C (2500°F) engine.

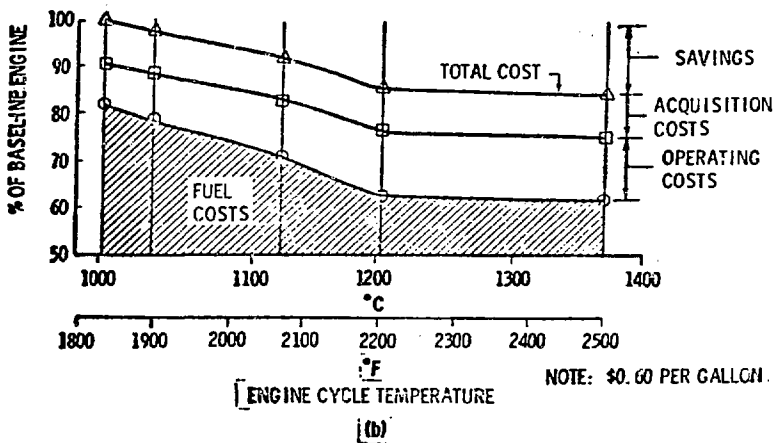
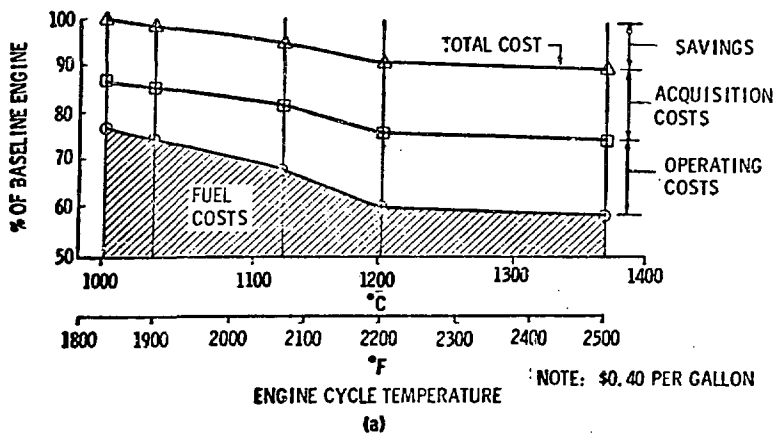


Figure 14. Line haul truck life cycle cost results.

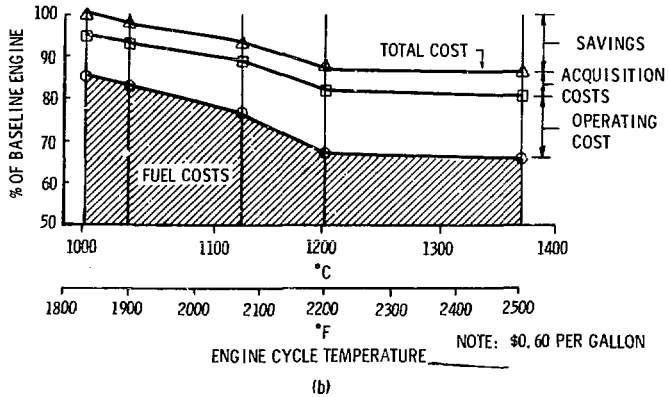
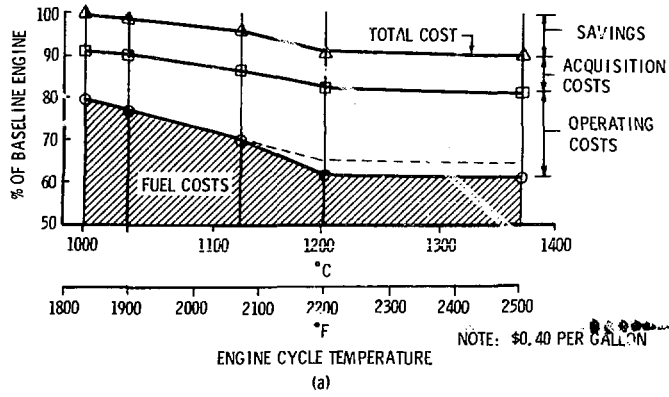


Figure 15. Highway coach life cycle cost results.

	1002°C (1835°F) (BASELINE) ENGINE	1204°C (2200°F) ENGINE	1371°C (2500°F) ENGINE
TEMPERATURE CHANGE	← 185°C (365°F) →	← 148.9°C (300°F) →	
REDUCTION IN ENGINE SFC			
DUE TO TEMPERATURE	← 12.1% →	← 3.0% →	
DUE TO IMPROVED COMPONENTS	← 7.0% →		
	TOTAL 19.1%		
FUEL SAVINGS IN ENGINE LIFE			
31751.5 KG (70,000 LB) GVW TRUCK	← 115,834 LITRE (30,600 GAL.) →	← 10,221 LITRE (2,700 GAL.) →	
GREYHOUND COACH	← 100,313 LITRE (26,500 GAL.) →	← 5,678 LITRE (1,500 GAL.) →	
ENGINE MANUFACTURING COST INCREASE	← 22.8% →	← 4.7% →	
LIFE CYCLE COST REDUCTION			
TRUCK AND COACH	← 9% →	← 1% →	
NUMBER OF ROTATING CERAMIC PARTS	← 1 →	← 3 →	
RISK FACTORS (COSTS, STRUCTURAL, CONTROL, MATERIALS, FAB)	← MEDIUM →	← HIGH →	

Figure 16. Summary of study engines—fuel savings and costs.

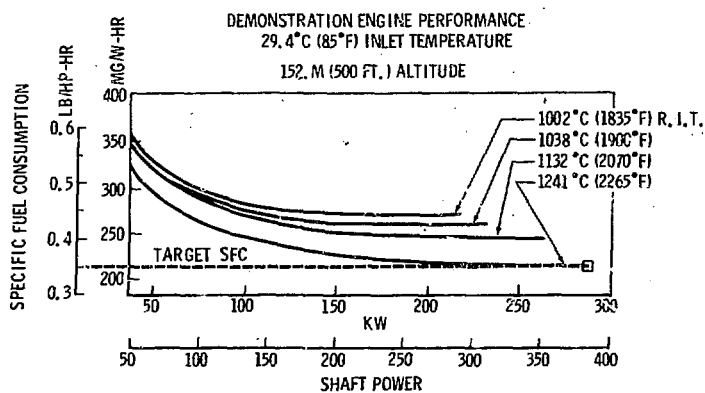


Figure 17. Proposed engines—sfc vs shaft horsepower.

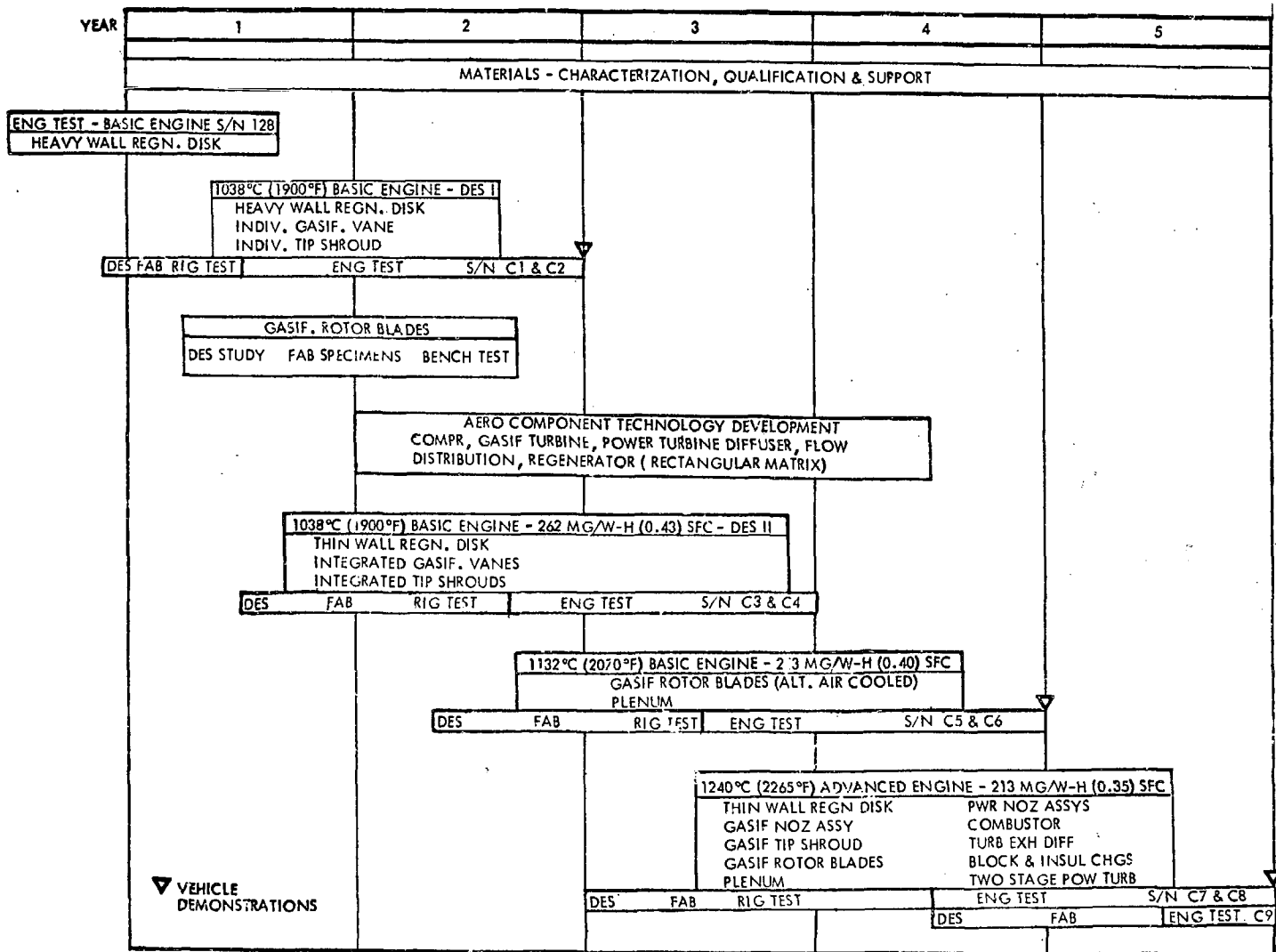


Figure 18. HEAVY DUTY GAS TURBINE ENGINE PROGRAM SCHEDULE

REVIEW OF PROOF TESTING ACTIVITIES
IN
STRUCTURAL CERAMICS

by

E. M. LENOE

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REVIEW OF PROOF TESTING ACTIVITIES IN
STRUCTURAL CERAMICS

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ABSTRACT

Important structural ceramics currently exhibit considerable variability in mechanical behavior. Therefore successful application of ceramics in heat engines requires a thorough understanding of the source of this variability and insight into those experimental procedures which can be applied to minimize strength scatter. This paper briefly reviews the various techniques to monitor component quality during manufacture, to qualify parts for further evaluation, and to proof test in order to guarantee minimum desired strength levels.

INTRODUCTION

Starting in mid 1971 Ford Motor Company in partnership with ARPA,* embarked on an extensive program to design, evaluate and demonstrate an all ceramic vehicular engine. Encouraged by the positive technical achievements of the past few years, The Office of Highway Systems, Division of Transportation Energy Conservation of ERDA** (ERDA-TEC) is attempting to exploit the civilian sector related technology of the ARPA program. Accordingly, ERDA is now participating in the funding of Ford activities in ceramic vehicular engine development. In particular this paper describes the ERDA-TECH task on

*Defense Advanced Research Projects Agency

**Energy Research and Development Administration

development of proof testing methodology. Currently, this study is underway both at the Army Materials and Mechanics Research Center and the Turbine Development Department of Ford Motor Company.

In reality, from the initiation of the Ford ceramic engine program, proof testing concepts have played an integral role. This is clearly illustrated by Figure 1, indicating that an iterative design, analysis and test evaluation sequence was the basis of the study. Materials and process development were directed by results of simple mechanical and thermal screening tests as well as various rig tests. As examples, a rotating, alternate heating and air quench thermal shock apparatus proved not only economical but invaluable in providing insight into development of improved reaction sintered stator and rotor vane designs. A simple cantilevered blade bend failure test gave data on uniformity of ceramic component properties and a further index to measure progress in process technology. Blade flexure tests were applied to both stators and rotors in process development. Stator vane proof testing is currently based on static preloading of each vane to screen out defective parts. The proof test load level was established by correlation with observed survival times in the high temperature flow path qualification rig. The stator shrouds were also subjected to proof testing by imposing external pressures of 100 psi in a special fixture. An elementary globar furnace provided capability to impose radial thermal gradients across the bond line of duodensity ceramic rotors whereas curvic couplings were initially screened by means of a specially designed axial loading device. In addition, ceramic stationary and rotating hot flow path components were subjected to various rig tests which attempt to simulate critical aspects of the engine environment.

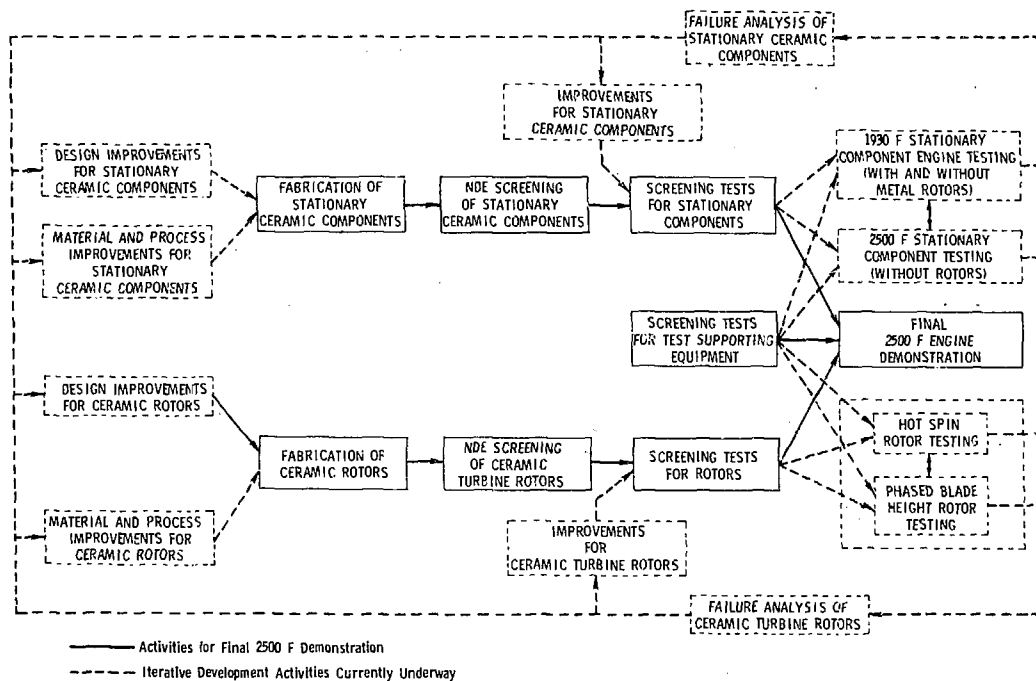


FIGURE 1. Illustrating Iterative Program and Component Screening Tests

The purpose of these numerous procedures was to achieve rapid feedback on materials, process and design deficiencies and moreover to provide a stock-pile of engine components which successfully survived these screening tests. Engine durability tests are to be performed on an assembly of veteran, pre-screened parts.

Notwithstanding these systematic component qualification and screening procedures, it is important to realize that to date no formal, quantitative proof testing methodology has been developed for actual engine parts. Thus this paper presents preliminary analytical findings regarding application of proof testing.

Prior to discussing the various test rigs, it is well to consider fabrication procedures since they are important in controlling the introduction of defects into ceramic engine components. In this paper for the remaining discussion emphasis is on the ceramic rotor.

FORMING METHODS

There are many techniques to form ceramic powders into useable shapes, including isostatic pressing, flame spraying, injection molding, warm die molding, slip casting, extrusion, and thin film forming (1 thru 4).^{*} Selection of the forming method depends on component design requirements such as size, shape and mechanical properties criteria. Each method has its advantages and drawbacks and often times it is necessary to combine techniques to produce a satisfactory article. For example, the all ceramic turbine rotor currently being evaluated at Ford Motor Company, used injection molding, slip casting and hot pressing techniques.

Ford engineers have used injection molding to form complex shapes with

^{*} Numbers in brackets designate References at end of paper.

close manufacturing tolerances such as rotor blade rings, stators and nose cones. A moldable compound is formulated by adding polymeric materials (20 to 40% by volume) to silicon powder. When heated, this provides a viscous material which can be forced under moderate pressures into a metal die cavity to form the desired part. The molded part is solidified by cooling, extracted from the mold, reheated under controlled conditions to remove the polymeric vehicle and subsequently reacted with nitrogen to convert the component to silicon nitride. Such parts can be successfully produced with typical densities ranging from 2.2 to 2.75 gm/cm³.

The so-called duodensity turbine rotor concept is the primary fabrication technique being developed at Ford Motor Company. This concept utilizes the high strength of hot pressed Si₃N₄ (HPSN) in the hub region where stresses are highest but temperatures are moderate and so creep problems caused by use of an MgO densification aid are minimized. Reaction sintered Si₃N₄ (RBSN) which can be readily formed into complex airfoil shapes by either injection molding or slip casting, is used for the blade rings. Since RBSN does not contain the MgO densification additive, it exhibits better creep resistance at the higher blade gas stream temperatures. Although the reaction-sintered material possesses lower strength, it is also of lower density and is adequate for the turbine blades because the stress levels are considerably lower than those in the hub region.

The duo-density concept relies on the ability to successfully bond these two varieties of Si₃N₄ (i.e. HPSN & RBSN) by hot pressing into an integral turbine rotor. Significant bonding parameters and the results of over 700 hot press bonding experiments have been reported elsewhere

CERAMIC TURBINE ROTOR FABRICATION

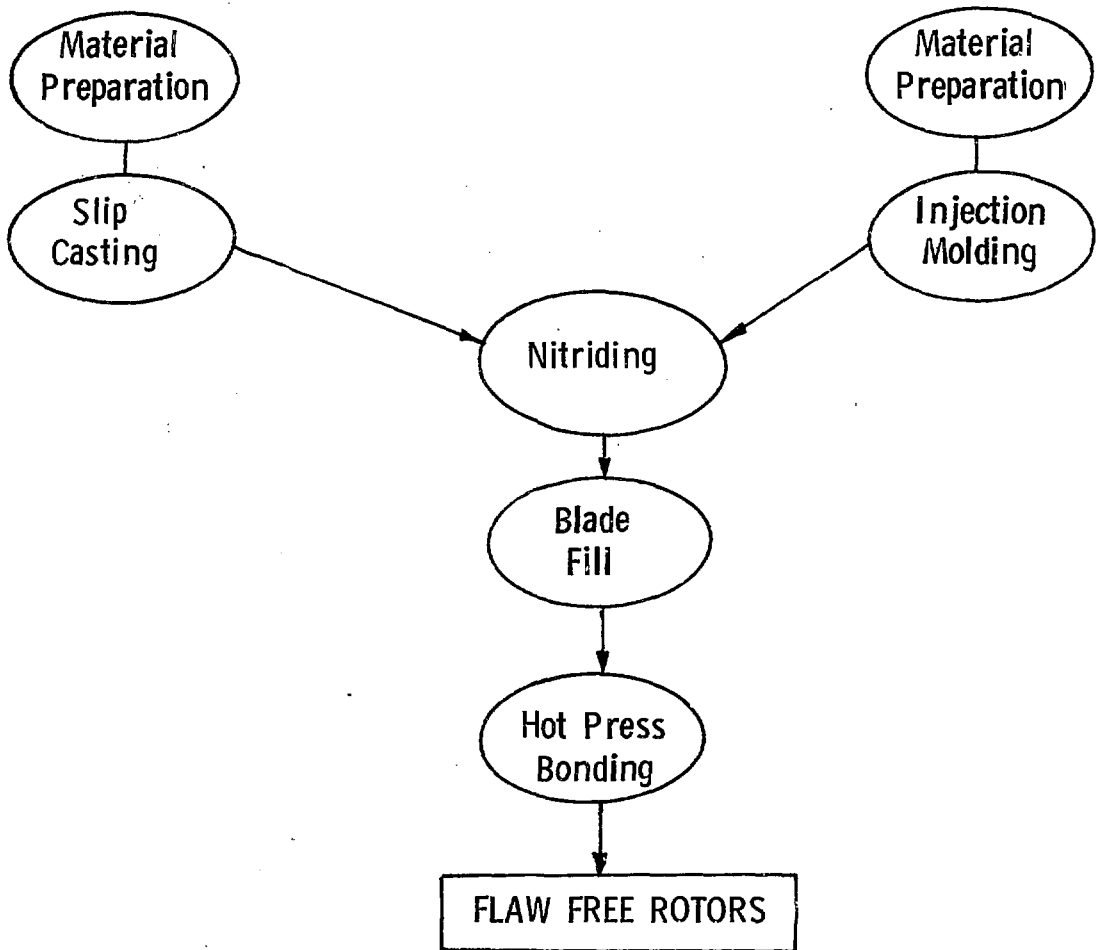


FIGURE 2. Ceramic Turbine Rotor Fabrication Flow Chart

FIGURE 2

POTENTIAL DEFECT INTRODUCTION

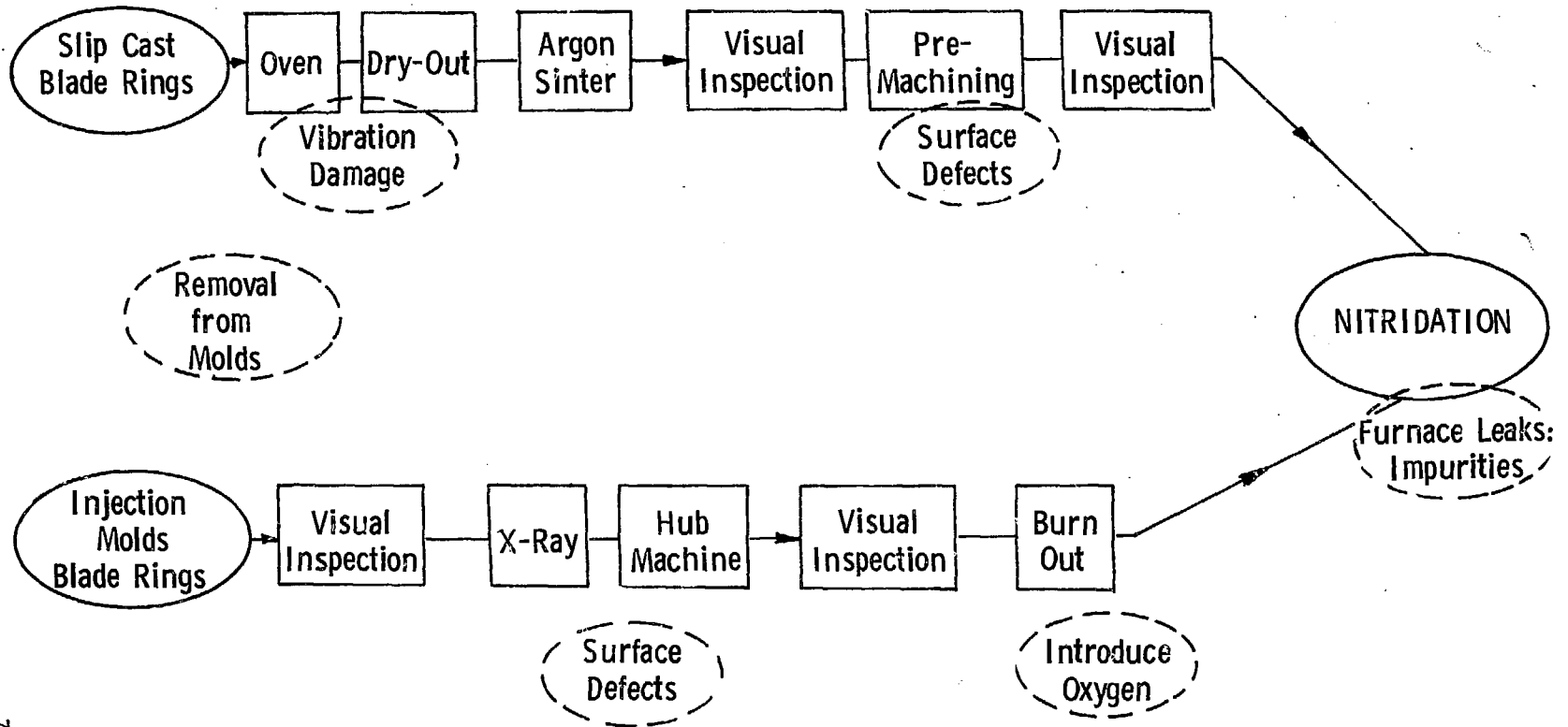


FIGURE 3. Potential Defect Introduction

[5, pages 113 thru 153]. It is evident that the method to produce the ceramic component controls the nature and distribution of microflaws and defects introduced into the component. Figures 2 and 3 are intended to illustrate some of the features of the manufacturing technology and the relationship to potential defect introduction. Referring to Figure 2, for instance, material preparation, the nitriding process, techniques for blade fill and subsequent removal of the fill as well as the details of the hot press bonding techniques all have a marked controlling influence on the quality of the finished rotor. Figure 3 illustrates in somewhat more detail other potential defect sources. Obviously it is necessary to introduce a variety of materials and screening tests to provide process controls and also to evaluate actual component quality.

It is not sufficient to prepare mechanical properties test specimens using the same starting materials, nitriding cycles, etc. As a further illustration of this point, consider removal of the blade ring from the molds. A different procedure is required for the slip cast versus the injection molded RBSN. The injection molded blade ring tool is quite intricate in design and pull angle, ground part-lines, surface finish, distribution of heat, pressure and mold gate design are important features in producing a high quality component. In order to systematically develop appropriate process controls, component evaluation techniques are necessary. These have included thermal shock tests and the blade flexure test previously mentioned. The blade evaluation procedure has demonstrated its utility in several ways.

TURBINE BLADE FLEXURE TESTS

In this apparatus blades or reaction sintered silicon nitride blade

rings can be mechanically loaded in bending as shown in Figure 4. The blade ring is accurately mounted on an indexing fixture and each blade can be displaced by an instrumented loading arm until failure occurs. Variations in fracture load from blade to blade within one blade ring can be assessed as well as variation in fracture loads from one blade ring to another. Results to date indicate a characteristic failure load of approximately 90 lb. which is equivalent to a maximum tensile stress of 35,000 psi and a Weibull modulus of about 9. These values agree reasonably well with four-point MOR* test data of flexure test specimens. A strain gauge calibration of a ceramic blade under such a bending load indicates that 100 lb. load is equivalent to about 39,000 psi tensile stress in the blade outer fibers. A high temperature version of this fixture is currently being designed.

The primary function of the blade flexure test is to evaluate changes in the fabrication process and provide timely feedback. Such feedback has prompted a current program to refine the nitriding step in the rotor fabrication process to produce blade rings of more homogeneous microstructure.

As a further example of the usefulness of the apparatus, characteristic blade failure loads ranged from 69 to 123 lbs. for injection molded material and from 69 to 130 for the slip cast material. Based on limited data, it appears that there is no significant difference in the room temperature strength of these two fabrication processes. However, two injection molded blade rings of 2.7 g/cc density were tested before and after a 200 hour furnace soak in air at 1900°F. In this case, failure loads of the thermally treated blades were slightly higher than the untreated blades. This is contrary to results obtained with a slip cast 2.8 g/cc material where a strength

*Modulus of Rupture as determined in flexure tests.



FIGURE 4. Blade Flexure Apparatus

degradation occurred after thermal exposure. The difference in behavior between the 2.8 g/cc slip cast and the 2.7 g/cc injection molded materials is attributed to the effect of unreacted silicon present because of incomplete nitridation of the higher density slip cast material. It is evident that such data is invaluable in further process developments.

COLD SPIN TESTS FOR BLADES AND BLADE RINGS AND ROTOR HUBS AND ROTORS

Cold spin testing of rotor blades and finished turbine rotors is carried out in a vacuum spin rig shown schematically in Figure 5. Rotor blades or blade rings are bonded with epoxy to a hub which is then attached to a shaft and inserted into the spin pit, and the chamber subsequently evacuated. A fragile ring coated with a conductive paint surrounds the blade ring. Blade failure will break the continuity in the conductive paint and trigger a strobe light enabling a pictorial record. Of interest is the fact that recently injection molded blades have been cold spun successfully to 80,000 RPM.

This apparatus is also used to evaluate complete rotors and rotor hubs as well as blade rings and segments. During the Ford project, over 1,000 press bondings of duodensity rotors have been completed. Fabrication improvements in the hub regions have progressed from flat-sided hubs to fully contoured hubs made simultaneously in the hot press bonding operation. Over 110 cold spin tests have been conducted and some rotors have exceeded full speed requirements.

Test procedures for hubs have been described elsewhere [6,7]. Hubs have been tested to destruction and photographed at burst speeds. HPSN hub failure speeds have shown a characteristic failure speed of 115,965 RPM.

SPIN PIT

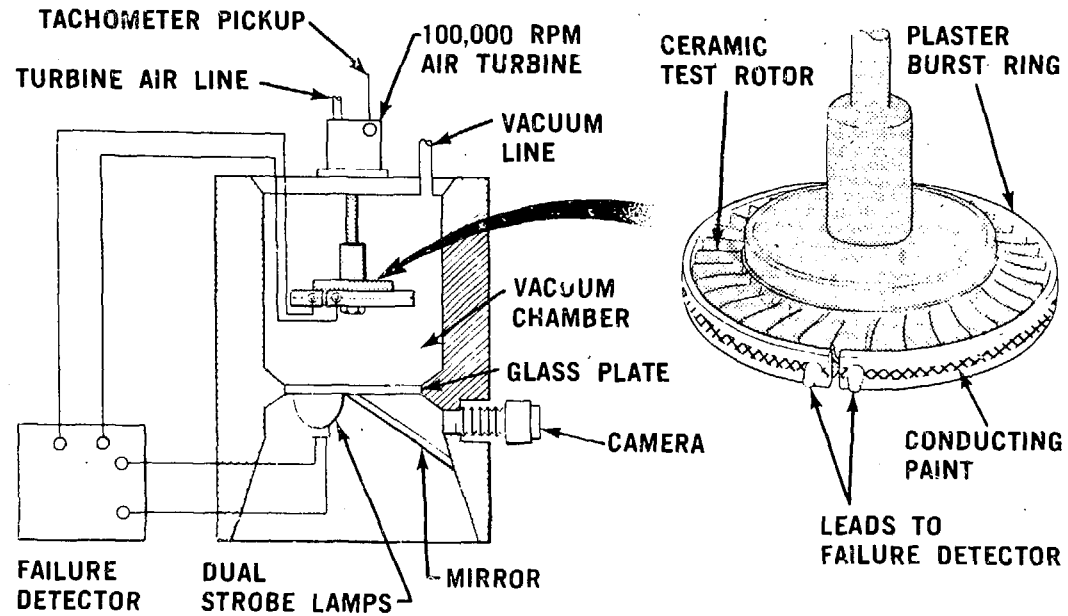


FIGURE 5. Cold Spin Rig

Several successful attempts have been made to obtain correlations of predicted and observed failure distribution for hub burst speeds. Commercial HPSN and Ford materials made from 2% MgO Si_3N_4 have been evaluated. The purpose of the correlation study was to check the validity of the use of Weibull theory (8) as applied to brittle materials by calculating a failure distribution versus speed and comparing it to the experimentally determined failure distribution. Generally the calculated failure distributions fall within the 90 percent confidence band of the experimental failure distributions. This indicates reliable correlation between calculated and experimental results and confirms use of Weibull theory in the prediction of rotor hub failure distributions under the cold spin test conditions.

HIGH TEMPERATURE TEST RIGS FOR COMPONENT EVALUATION

Now let us briefly review the various high temperature test rigs and test techniques for evaluation of the stationary ceramic components and the turbine rotors. Figure 6 is a schematic of a combustor test rig which essentially comprises an assembly of ducting to direct incoming air to the test combustor in a manner similar to the engine. Incoming air can be pressurized to five atmospheres by a laboratory compressor system and heated up to 1200°F and further by a separate upstream combustor to 1700°F. The ceramic combustor is mounted in a manner similar to the engine and the observation window, shown in the schematic, can be used to observe combustor operation at high temperature. Another rig similar to the combustor test rig is used to test combustors, nose cones, stators and/or rotor tip shrouds up to equivalent turbine inlet temperatures of 2500°F. The primary difference between these test rigs and the engine is that they do not use a high speed rotating shaft, nor do they use regenerative heat exchangers. However care

2500°F Flowpath Qualification Rig

Non - Rotor Steady - State Configuration

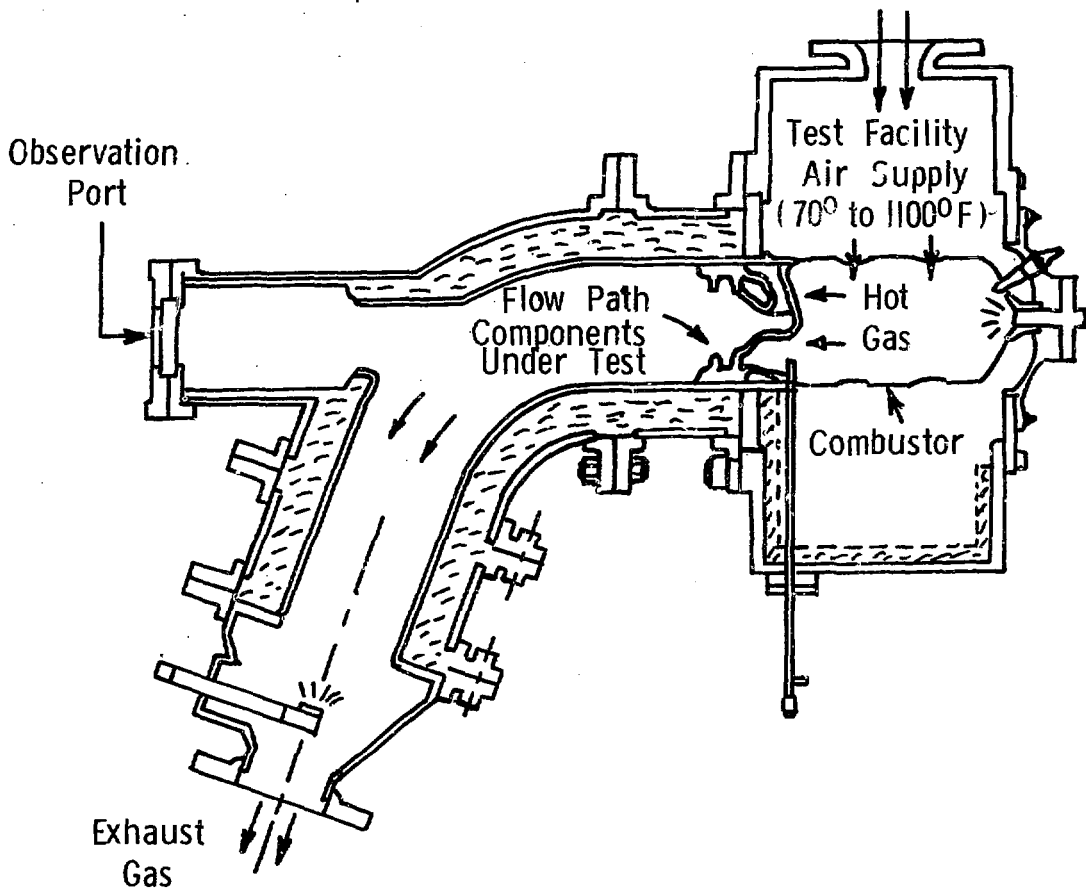


FIGURE 6. 2500°F Flow Path Qualification Rig

Hot Spin Rig

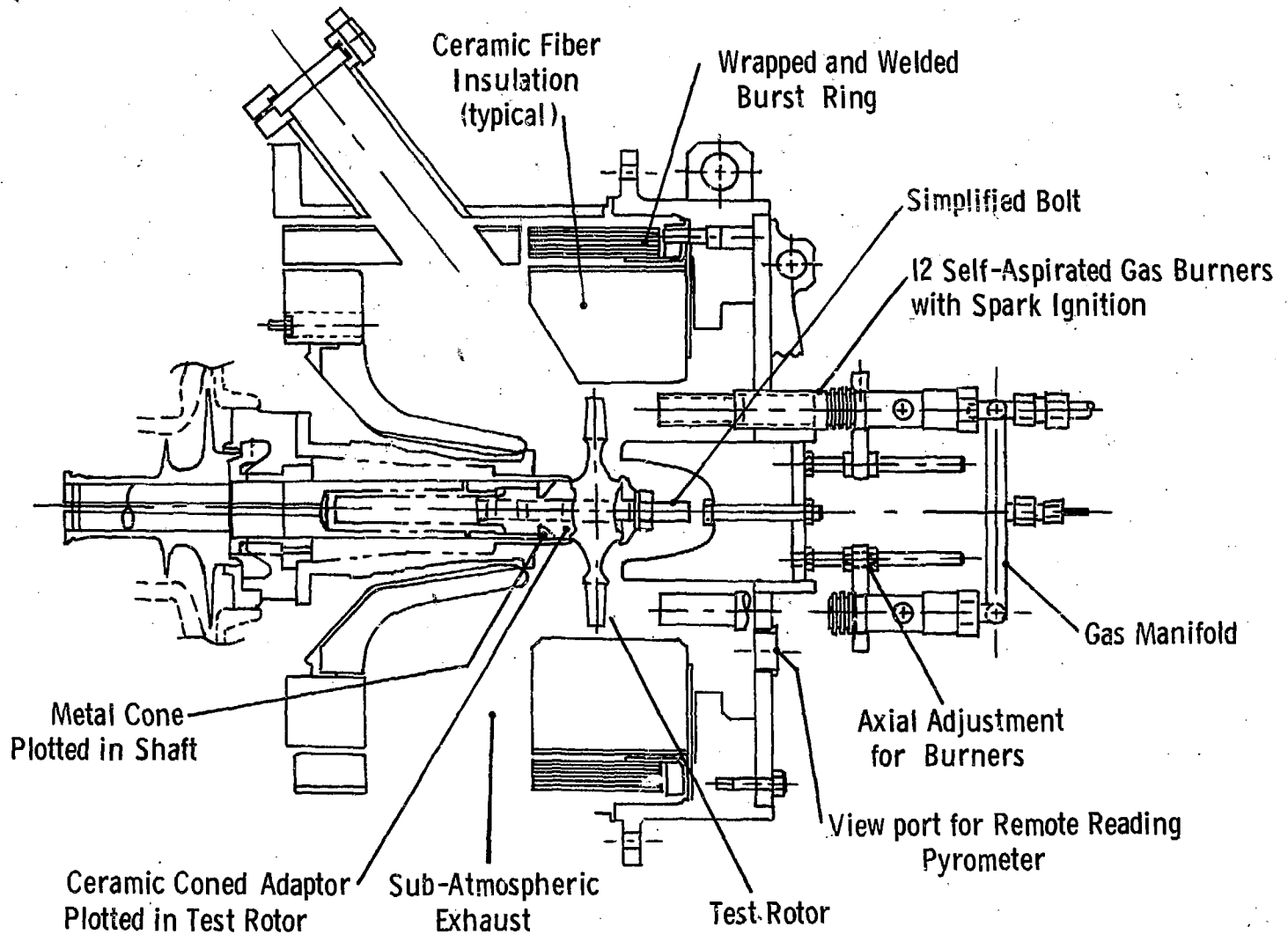


FIGURE 7. Schematic of Hot Spin Rig

has been taken in the design to closely simulate thermal conditions of the ceramic components under test.

Hot spin testing of ceramic turbine rotors is a further step which more closely approximates testing conditions experienced in an engine. A hot spin rig was designed, as shown schematically in Figure 7, to subject ceramic turbine rotors to both centrifugal and thermal stresses. The rig was designed to test-to-failure and yet minimize both the acillary damage to the apparatus and associated turn-around time. The rig uses engine hardware for the rotors, high speed shaft and reduction gear. A motoring dynamometer drives the output shaft, which through the gear box, drives the high speed shaft and the rotors. Twelve gas jets evenly spaced in a circle are used to heat the rotor blades and rotor rim. The rotor environment is surrounded with fibrous insulation which serves to insulate the outer metal housings from high internal operating temperature and which also serves as a readily replaceable "catcher" material to absorb pieces of rotor on failure. Two quartz windows are provided for observation during testing and also for sighting optical pyrometers for setting operating temperatures. Recognizing the difference in heat flux from the 12 gas jets as compared to that of the engine combustor, the test procedure is to use the pyrometer to set a given rotor rim temperature which is equivalent to the desired turbine inlet temperature in the engine. While this means a higher turbine inlet temperature in the hot spin test rig, it is considered to more closely simulate local hub temperatures in the bond regions of the rotor. Currently two such hot spin test rigs are operational which provide capability for testing ceramic turbine rotors to failure in an air-atmosphere at rotor rim temperatures in excess of 2200°F or equivalent turbine inlet temperatures in excess of 2500°F. This

capability, which provides for a turn-around time of one-test-per week is considered a unique step forward in the development of high temperature, ceramic turbines.

In addition to the combustor rig and hot spin test facilities, several Ford Model 820 ceramic turbine engines are used in the program for testing ceramic components; these engines have been designed to facilitate experimental flexibility for testing ceramic components. There are a number of unique ways of operating the engines during such testing. First, the engine can be operated as a complete engine using either metal or ceramic turbine rotors; the latest designs for rotor mounting provide for metal rotor/ceramic rotor interchange. Second, the rotor can be completely removed from the engine and, being a single shaft engine, the high speed shaft and compressor can be driven from a laboratory motoring dynamometer. Third, the high speed shaft including the turbine rotors and compressor can be removed from the engine and an external laboratory air supply ducted to the engine at the normal compressor discharge location. In this third configuration, the regenerators are driven independently by variable speed electric motors. These last two arrangements have proven to be extremely useful for testing and developing the stationary ceramic components under very realistic conditions. The standard 820 ceramic turbine engine is designed to operate on a temperature schedule which increases turbine inlet temperature from 1930°F at 55% speed to 2500°F at 100% speed. This schedule was selected to prevent turbine exit temperature or regenerator inlet temperature from exceeding 1850°F, which is considered the limit for the aluminum-silicate material used for the regenerators. The difference between the turbine inlet and turbine exit (regenerator inlet) temperature is, of course, a measure

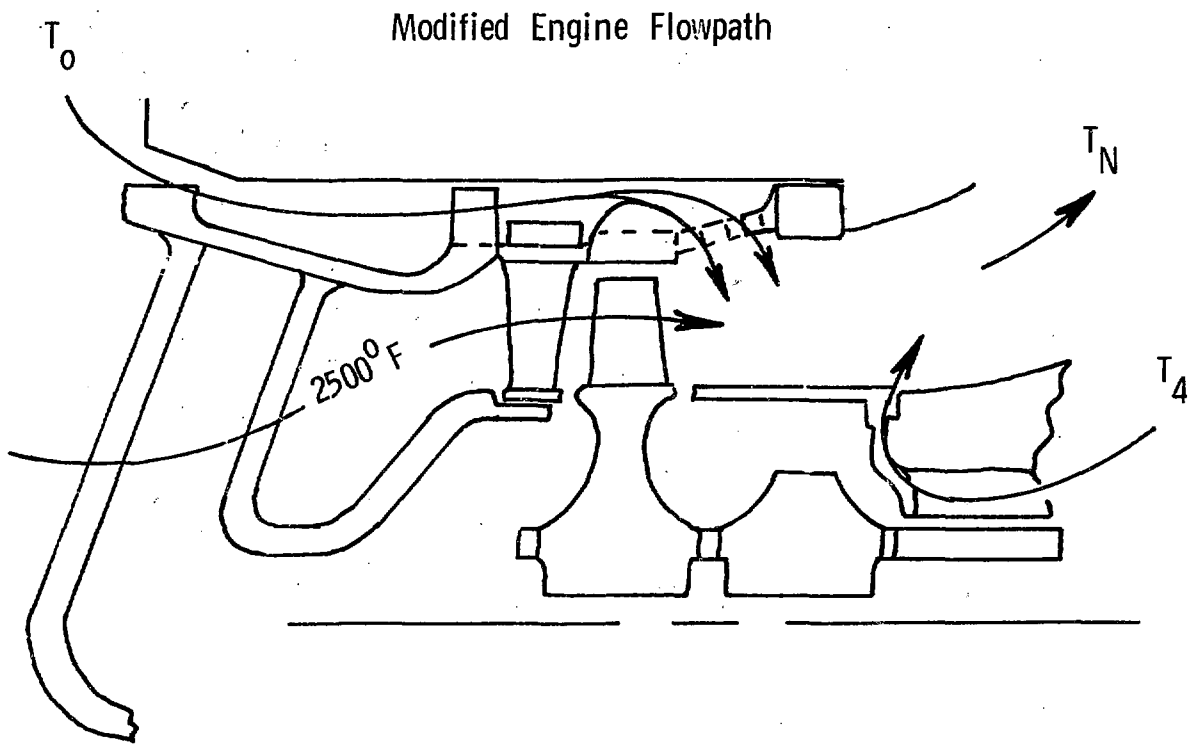


FIGURE 8. Modified Engine Flow Path

of the work taken out by the turbine. While this turbine inlet temperature schedule is meaningful for a fully developed engine, it has distinct drawbacks during the ceramic hardware developmental stage. For example, it is desirable during development to engine test ceramic rotors at full turbine inlet temperature (2500°F) over a range of speeds rather than at only 100% speed, because the rotors may fail at less than 100% conditions. Alternatively, it is desirable to engine test rotors during development with excessive tip clearance or with less than the full complement of blading. In either case, the temperature drop through the turbine stages would be less than design values. Accordingly for such testing, turbine stages would operate at less than design values. Therefore it is desirable to be able to test with 2500°F turbine inlet temperatures, take little or no work out of the turbines, yet limit the turbine exit temperatures to 1850°F to protect the regenerator materials from over-temperatures. A special engine modification accomplishes this, by-passing the turbines and combustor with some of the regenerated compressed air and remixing at the turbine exit. Figure 8 shows a schematic of the modified flowpath incorporating a by-pass outside of the stator and tip shrouds to reduce the turbine exit gas temperature prior to entering the aluminum-silicate regenerator. With this modification, it has been possible to test ceramic turbine rotors at turbine inlet temperatures as high as 2650°F without exceeding the regenerator inlet limit of 1850°F.

PRELIMINARY ANALYSIS OF PROOF TESTING

With regard to brittle fracture a statistical approach is mandatory in order to account for observed variability in ceramic properties. The most commonly applied theory is the Weibull [8] approach to estimate fast fracture

reliability. Slow crack growth considerations to estimate ceramic component life have also been developed and applied [9 thru 14]. Techniques to account for the influence of proof test truncations of properties into reliability estimates have also been derived [15 thru 19]. It is important to recall that analytical predictions are based on various approximations to physical properties as well as to strength and slow crack growth parameters of materials and the boundary and loading conditions in the components. Thus the adequacy of the mathematical modeling must be established. It is imperative to develop insight into the controlling influences of these numerous assumptions and conditions. Therefore systematic analysis as well as reproducible experimental rig tests are required.

Analytical methods expounded to date have been aimed at components subjected to relatively simple stress states wherein the principal stress orientation and magnitude are fairly well known. Application of the elementary theories to simple situations appears to have been fruitful. [14,20] However when definition of the problem becomes more dubious, less success has been demonstrated. [21,22]

Any attempt to apply proof test considerations to a practical situation is of interest since improvements in engineering methodology are largely dependent on lessons learned from past mistakes. In this instance, [21 thru 24] unknown or uncontrolled boundary conditions and loading conditions negated attempts to apply simple engineering methodology.

Therefore it is imperative to systematically calibrate and evaluate the behavior of the various test rigs and also to complete detailed analyses of critical test conditions. Preliminary results have been obtained relating to rotor proof tests at Ford Motor Company. It is axiomatic that to be

effective, a proof test should be as simple as possible and yet closely simulate the critical stress conditions. However, for the vehicular engine under consideration it is interesting to note that the hub stresses at full power consist of approximately 50% centrifugal and 50% thermal stresses, whereas the blade stresses at the same design point are virtually all centrifugal. Obviously this introduces problems with regard to cold spin, versus hot spin versus static imposition of a radial thermal gradient across the rotor. The exact distribution of stresses throughout the component is also of importance. Considering each elemental volume of the stressed rotor, the stress severity factor will vary from element to element. Furthermore in the case of time dependent loadings, the orientation of principal stress throughout the component, will necessarily vary with time. A key engineering judgment then is the isolation of critical conditions during the operational cycle. This further emphasizes the importance of rig testing to substantiate such judgment as well as to systematically calibrate the apparatus.

Application of proof test analysis to the short term, fast fracture problem is relatively straight forward. Considering a cold spin test Ford engineers have also applied slow crack growth considerations, again using available data, and the critical rotor element (in the neck region of the rotor) to estimate the overspeed requirement to attain desired rotor life of 25 hours with a survival probability of 90%. The result was an overspeed of 68%. It is noteworthy that the calculation was determined to be extremely sensitive to input statistical data, further emphasizing the necessity for a rather extensive properties characterization program.

The requirement for similarity of stress conditions is clearly demonstrated by considering the isotherms and stress states depicted for the rotor both in the engine and in the hot spin rig [Figures 9, 10 and 11]. The difference in temperature distribution at 100% design speed and 2500°F turbine inlet temperature, steady state conditions in the engine is contrasted to the hot spin rig with blade ring rim at 2300°F in Figure 9. While temperature conditions in the critical rotor hub neck bond region are well simulated in the hot spin rig, peak temperatures are higher in other portions of the hub. In general the overall stress symmetry is affected and peak stresses are slightly higher in the hot spin test. More importantly, the volume of stressed material is affected. Based on these preliminary analyses the failure rates in the hot spin rig is anticipated to be higher than in an engine test. However it must be emphasized that these estimates are based on preliminary data and further characterization of ceramics is required.

Furthermore symmetry of stress distributions in the hot spin rig is based on assumed boundary conditions which remain to be experimentally verified. However it is likely that the rotor hub temperature gradients will differ in the engine as contrasted to the hot spin test rig. This difference can largely be attributed to differences in the rotor mounting. For instance the hot spin mounting shaft has a lower temperature material capability accordingly necessitating approximately a factor of three in bolt cooling mass flow rates which obviously imposes different thermal gradients under the two test conditions.

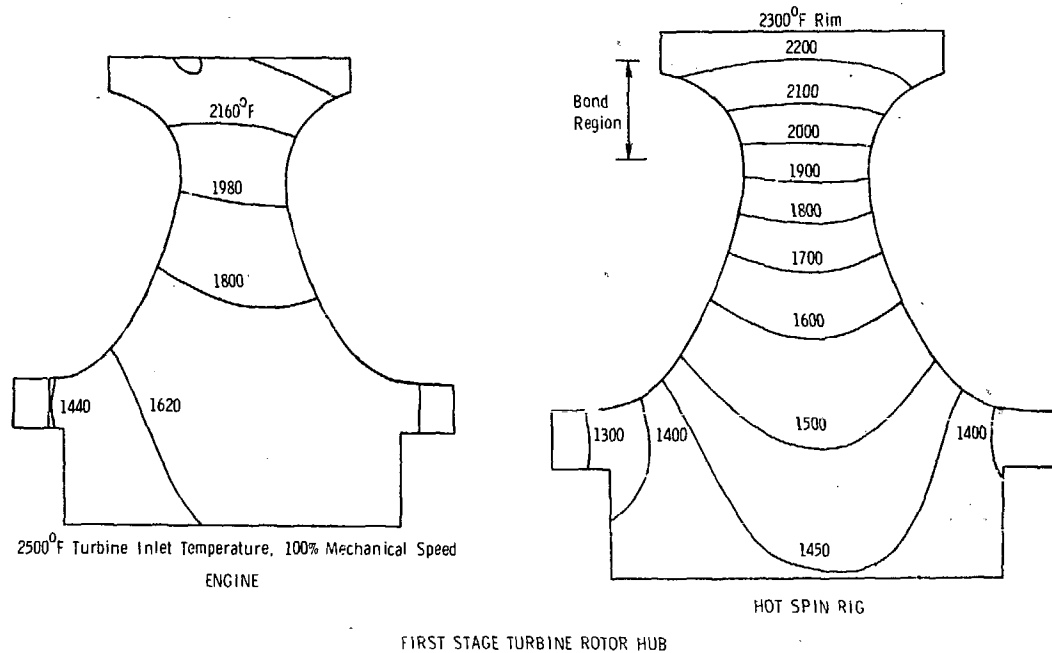


FIGURE 9. First Stage Turbine Rotor Hub Isotherms in the Engine and in the Hot Spin Rig. (Courtesy Ford Motor Company)

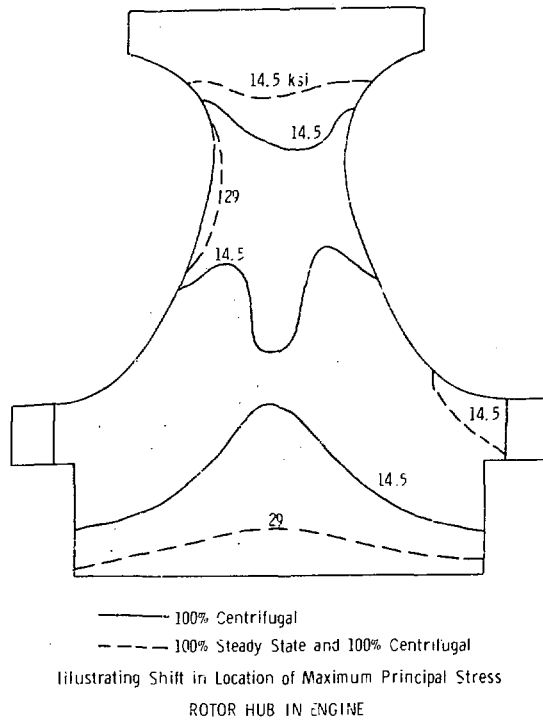
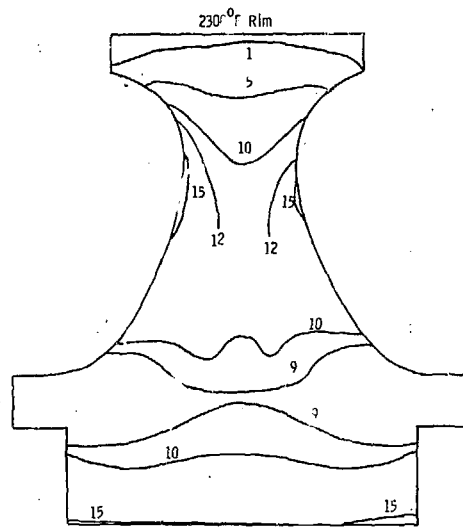
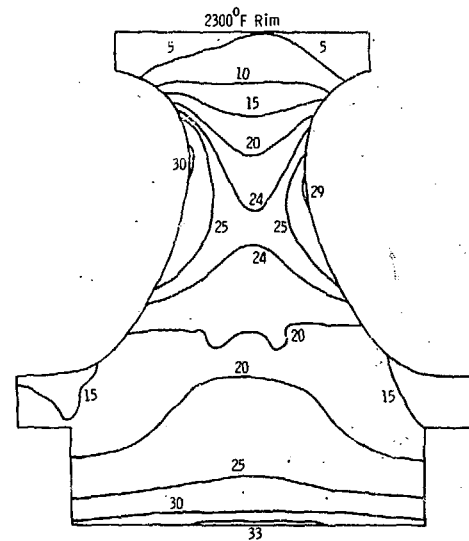


FIGURE 10. Illustrating Shift in Maximum Principal Stresses, Thermal and Mechanical, of Rotor Hub in Engine (Courtesy Ford Motor Company)



Thermal and Centrifugal Stresses at 24,000 rpm
ROTOR IN HOT SPIN RIG



Thermal Stresses at 24,000 rpm Plus Centrifugal Stresses at 64,240 rpm
ROTOR IN HOT SPIN RIG

FIGURE 11. Thermal and Centrifugal Stress of Rotor Hub in Hot Spin Rig
(Preliminary Analysis, Courtesy Ford Motor Company)

CONCLUSIONS

In the past six years the various types of silicon nitrides as well as silicon carbides have been significantly improved and alternate materials such as siliconized SiC have been developed. While more progress is a reasonable anticipation, it would be too optimistic to expect an equal rate of progress in the next few years. Making the conservative assumption that strength and uniformity of these classes of materials will not change significantly in the near future it is timely to explore alternate ways to enhance the quality of ceramic components. Proof testing is a practical method of eliminating defective parts in a particular lot of components (for example in the ceramic grinding wheel industry). Therefore, our next goal is to systematically explore economic methods to proof test ceramic engine components, and verify the proof testing methodology via tests.

Within the next year, hot spin and engine rig test data on high temperature evaluation of ceramic rotors will be available. Fast fracture as well as preliminary durability data are anticipated. These results will permit assessment of the design procedures and may well stimulate further studies. As an example, further evaluation of material stability, of tensile behavior, and general nonlinear stress-strain response at high temperatures, as well as combined stress failure behavior may be required, especially for the highest temperature exposures. Therefore, the later portions of our study of proof testing are reserved for such efforts.

ACKNOWLEDGEMENTS

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NET SHAPE INTEGRAL AND CERAMIC BLADED WROUGHT SUPERALLOY GAS TURBINE ROTORS

Bryant H. Walker

ABSTRACT

With ever increasing demands for more efficient, low cost automotive propulsion systems, many development programs have been initiated to improve the gas turbine engine. This paper summarizes two such programs utilizing the GATORIZING™ forging process to fabricate (1) net shape integrally bladed and (2) ceramic bladed wrought superalloy turbine rotors.

Discussed first is the demonstration of fabricating a net shape integrally bladed rotor which is both aerodynamically efficient and economically feasible. Included are material processing and rotor fabrication techniques, component evaluation, and projected cost analyses for mass production.

Reviewed secondly is the development and evaluation of a hybrid superalloy rotor with ceramic blades. This includes design, fabrication, and testing of the blade root attachment, culminating with the hot spin test of a fully bladed rotor.

INTRODUCTION

Two programs currently being conducted by the Government Products Division of Pratt & Whitney Aircraft are aimed at demonstrating unique techniques for producing uncooled gas turbine rotors. However, the approaches to achieving this objective are significantly different in the two programs. In one program, sponsored by the Energy Research & Development Administration (ERDA) [Contract E(11-1)-2637], the approach is to produce integral wrought

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superalloy rotors utilizing the GATORIZING isothermal forging process. In the other program sponsored by the Defense Advanced Research Projects Agency (Contract N00019-74-C-0484), the GATORIZING process combined with diffusion welding are used to produce hybrid ceramic bladed superalloy disks. This paper presents additional details of these programs and discusses their current status.

INTEGRAL WROUGHT SUPERALLOY ROTOR PROGRAM (ERDA/NASA)

It has been well identified¹ that one of the cost drivers in an automotive gas turbine is the cost of the turbine rotors. Pursuant to investigating lower cost manufacturing techniques this two phase program was initiated to demonstrate the feasibility of using an isothermal, superplastic forging (GATORIZING) process to produce wrought integrally bladed turbine rotors. The rotor selected as the demonstration model throughout the program was the gasifier-turbine rotor for the ERDA/Chrysler baseline gas turbine engine.

In Phase I, the superalloy IN-100 was selected for demonstrating, (1) the feasibility of low cost - close tolerance forging, using the GATORIZING process, and (2) the ability of the wrought superalloy to meet mechanical property requirements for the targeted design limits of the rotor.

To fabricate the rotor, a two-step forging sequence was selected. The first step produces a nonbladed overthick preform to ensure proper metal distribution for the second step. During the second step the preform is forged to a finish dimension disk and the blades extruded onto the rim of the disk to form the integral bladed rotor. Figure 1 shows an actual billet, preform and rotor fabricated using this two-step sequence. Figures 2 and 3 are cross-sectioned views of the TZM molybdenum preform and final form tooling used to isothermally forge the preform and rotor shown in Figure 1.

To optimize the forging parameters various temperatures and strain rates were evaluated to effect proper die fill and resultant response to heat treatment. The optimum forging parameters

¹ United Aircraft Research Laboratories Report K-97017-4, "Manufacturing Cost Study of Selected Automotive Gas Turbine Concepts," dated August 1971.

were found to be a constant strain rate of 0.1 in./in./min (0.25 cm/cm/min) with the preform forged at 1900°F (1038°C) and the final rotor forged at 2000°F (1093°C).

During the forging parameter optimizations, definite limitations to the processes were formulated. Very close tolerance forgings were easily produced; however, there are restrictions on the blade design to assure forgeability as evidenced by lack of blade fill during the second forging step. These restrictions were found to be an interrelated function of blade length, thickness, and radial taper in cross-sectional area. With these parameters quasi-quantified, the airfoil of the Chrysler design was modified to allow complete fill of the blade die cavities. Figure 4 shows the final rotor configuration in the as forged condition. To attain the finished part, shown in Figure 1, the only machining was that required to deburr the forging and grind the blades to the correct length.

Having attained a reproducible forging sequence demonstrating close tolerance forging of the rotor, a heat treatment study was conducted to optimize the mechanical properties of the rotor. This study was aimed at establishing a single heat treatment which would achieve a compromised microstructure combining the high tensile strengths typical of a fine grain structure with the good rupture life of large grained materials, while maintaining an adequate low-cycle fatigue (LCF) life. The heat treatment selected to achieve this optimum combination of properties was a double solution at 2150°F (1177°C) and 1950°F (1066°C) followed by precipitation at 1600°F (871°C) and 1800°F (982°C), and aging at 1200°F (649°C) and 1400°F (760°C).

Material capability curves were generated from several rotors heat treated using the described heat treatment. These curves indicated that neither the stress rupture nor the low-cycle fatigue properties met the targeted values of a 100 hr, 1750°F (955°C) stress-rupture strength of 17.5 ksi (121 MN/m²), and a 5000 cycle LCF life at 1700°F (927°C) and a 0.5% strain range. The typical properties were, however, close to the target goals [14.5 ksi (100 MN/m²) and 3200 cycles respectively] and were considered as good a combination of properties as could be achieved within the limits of the investigation using the wrought superalloy IN 100.

To ascertain the projected cost of producing an automotive rotor utilizing the GATORIZ-ING process, a manufacturing process and complete facility suitable for volume production of rotors was projected. Based on this study, volume production of the rotors is within current state-of-the-art at a selling price of \$51.77 per rotor at the rate of one million per year. The impact of alternate production rates is considered during Phase II of the program.

Based on the results of Phase I, the Phase II effort was directed at (1) producing turbine rotors with blades that would meet aerodynamic efficiency requirements of the baseline engine, and (2) to evaluate an alternate alloy, IN 792, as to its capability to meet, in wrought form, the targeted stress rupture and LCF values.

In an effort to produce turbine rotors that would equal the desired 78.8% efficiency, a tradeoff study was conducted on turbine rotor performance versus ability of the GATORIZING process to produce various blade shapes. Through iterative design and single blade forging trials, an optimum tradeoff was obtained resulting in a 78.3% efficient rotor. This was an acceptable design and could easily be improved by modification of the stator exit gas angle to ensure convergence in the blade row.

To fabricate rotors with the redesigned blades, new blade die inserts were fabricated using a unique electro chemical machining (ECM) technique. This approach was selected to hopefully maintain tight contour tolerances from insert to insert. It was also felt this technique would prove very cost effective in meeting the high volume machining requirements of the projected production facility studied in this program. Unfortunately, this ECM technique required many iterations to obtain an acceptable contoured die insert and significantly delayed the program. However, it is felt in a production facility this method of machining would be very cost effective.

While the ECM iterations were being conducted, the program efforts were concentrated on evaluation of the IN 792 alloy. This evaluation consisted of characterizing optimum extrusion and forging parameters to effect proper response to heat treatment and yield acceptable mechanical

properties. The elimination or control of this structure would have required a forging and heat treating study beyond the scope of this program with no guarantees as to resulting mechanical properties.

Due to the out of scope effort required to solve the IN 792 problems, the decision was made to use another alloy, AF2-1DA, capable of meeting program goals and requiring minimal optimization studies.

To conserve remaining program funding, AF2-1DA forging stock was processed using an all inert powder processing technique and extruded using parameters developed under an Air Force Materials Laboratory contract (F 33615-70-C-1387). The material is currently being forged, using Phase II tooling, into integral bladed rotors for subsequent heat treatment and mechanical property evaluations.

The manufacturing cost study compiled during Phase I of the program will be updated and expanded to include alternate quantities of 100,000 and 10,000,000 units per year. This update will reflect further process refinements obtained during Phase II of the program.

CERAMIC BLADED WROUGHT SUPERALLOY DISK PROGRAM (DARPA/NAVAIR)

This two phase program was planned to develop and demonstrate the attachment of ceramic blades to a wrought superalloy disk. A unique approach was devised utilizing superplastic, isothermal forging (GATORIZING) to form a disk around the ceramic blade roots.

During Phase I, analytical and experimental design studies of selected candidate attachment schemes were conducted. Primary candidates chosen for further evaluation were reduced to practice on a sample basis to establish optimum fabrication parameters and their respective load carrying capabilities. Interactive evaluation yielded an optimum attachment combining the GATORIZING forging process with diffusion bonding (welding).

In this attachment scheme, illustrated in Figure 5, the root sections of the hot pressed and diamond ground silicon nitride blades are sandwiched between two nickel base superalloy AF2-1DA disk halves in a forging-diffusion welding cycle. The flexibility of this scheme is depicted in Figure 6 with the primary attributes of the attachment listed as follows:

1. Attainment of compressive surface-to-surface contact between the ceramic blade root and superalloy disk throughout its intended operational regime
2. Utilization of a compliance layer eliminating ceramic/wrought alloy interaction during fabrication and enhancing surface-to-surface contact
3. Simplicity of fabrication and incorporation of bore entry air cooling when required
4. Attachment of the ceramic blades directly to the disk or attach the ceramic blades to an intermediate wrought alloy blade root for attachment to the disk by conventional means.

To establish a meaningful design criterion for the room temperature strength capability of the optimized attachment, twelve single blade attachments were spin tested to failure. A peaked distribution of data resulted and a Weibull plot of the P/A ultimate stress levels was made and shown here in Figure 7. This plot reveals a mean stress of 37.4 ksi (258 MN/m²) and a 99.5% lower bound stress of 20.5 ksi (141 MN/m²) above which 99.5% of the failures would occur for a given sample size.

With this three sigma (99.5%) lower bound design point established, a demonstration test was conducted to complete Phase I of this program. This test consisted of spin testing three attachment specimens to 100% of the design stress level without failure. Two attachments were spun to 60,000 rpm and tip speeds of 2029 ft/sec (618.4 m/sec) while the third was spun at 40,000 rpm and a tip speed of 1487 ft/sec (453.2 m/sec).

The objective of Phase II of this program is to demonstrate the elevated temperature capabilities of the attachment by spin testing a fully bladed rotor at state-of-the-art temperatures. To accomplish this objective, several single and multi-bladed rotors were designed and spin tested at various temperatures and speeds. These tests led to the establishment of a quantified program milestone of spinning a fully bladed rotor for 50 hours at a blade temperature of 2250°F (1232°C) and 45,000 rpm.

The first step toward attaining this milestone was the design and testing of a single blade rotor. This specific rotor was spun without failure for 66 hours at 45,000 rpm and a blade tip temperature of 2250°F (1232°C). During the test, the rotor was subjected to 10 thermo-mechanical cycles and 97 isothermal LCF-dwell cycles. The success of this test led to a similar spin test of the 29 bladed rotor shown in Figure 8. Two blades were omitted to accommodate thermocouple instrumentation of the disk and blades. Following calibration, this rotor was spun for 54 hours at 45,000 rpm, blade temperature of 2250°F (1232°C), and saw 12 thermo-mechanical cycles without failure.

To meet the Phase II milestone, the 30 bladed rotor shown in Figure 9 was fabricated and tested. Spin testing was accomplished without failure of the rotor for 50 hours at 45,000 rpm, a blade tip temperature of 2250°F (1232°C), and included 4 thermal and 10 mechanical cycles.

The total success of this program has led to the extension of this work to include additional spin testing of a fully bladed rotor up to blade temperatures of 2500°F (1371°C).

SUMMARY

Two programs using unique techniques to fabricate high-performance gas turbine rotors have been discussed. One approach utilizes the GATORIZING forging process to produce wrought integral bladed rotors which are both aerodynamically efficient and economically feasible for large volume production. The other technique combines the GATORIZING process

with diffusion welding to produce a hybrid ceramic bladed superalloy rotor. This hybrid rotor has the potential to allow significant increases in turbine inlet temperatures without introducing the requirement for air cooling; a much more efficient engine would thus result.

ACKNOWLEDGEMENTS

The author wishes to express his thanks to the Energy Research and Development Administration and to the Defense Advanced Research Projects Agency for funding these programs, and to the respective program monitoring agencies NASA and NAVAIR. Special thanks also go to the respective program task groups for the successful completion of each of these programs.

FD 104851

Figure 1. Two-Step Superplastic Forging Sequence to Form Integrally Bladed Turbine Rotors

FD 72643B

Figure 2. Cross-Section of Preform Tooling

FD 74448C

Figure 3. Cross-Section of Final Form Tooling

FD 78238A

Figure 4. Fully Bladed Turbine Rotor in As Forged Condition

FD 110264A

Figure 5. Fabrication Process for the Ceramic Wrought Attachment

AV 104957

Figure 6. Flexibility of the Ceramic Blade Metal Disk Attachment Scheme

FD 110268A

Figure 7. Room Temperature Spin Test Results of Ultimate Strength for the Ceramic Wrought Alloy Attachment

FD 110268

Figure 8. Instrumented Ceramic Bladed Rotor Hot Spin Tested for 50 Hours at 2250°F

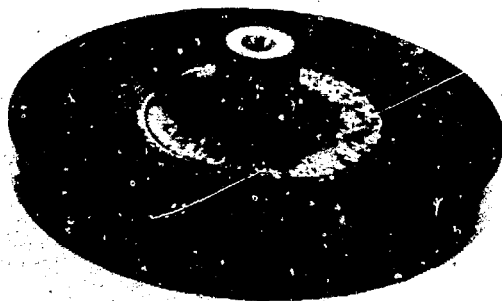
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Figure 9. Fully Bladed Hybrid Rotor Spin Tested at 2250°F Blade Temperature for 50 Hours at 45,000 rpm

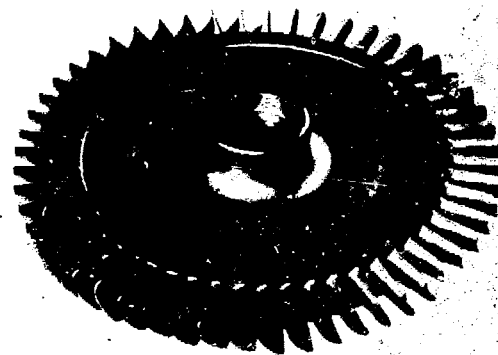
Approach:



Billet



Preform



Net Shape Rotor

*Figure 1. Two-Step Superplastic Forging Sequence
to Form Integrally Bladed Turbine Rotors*

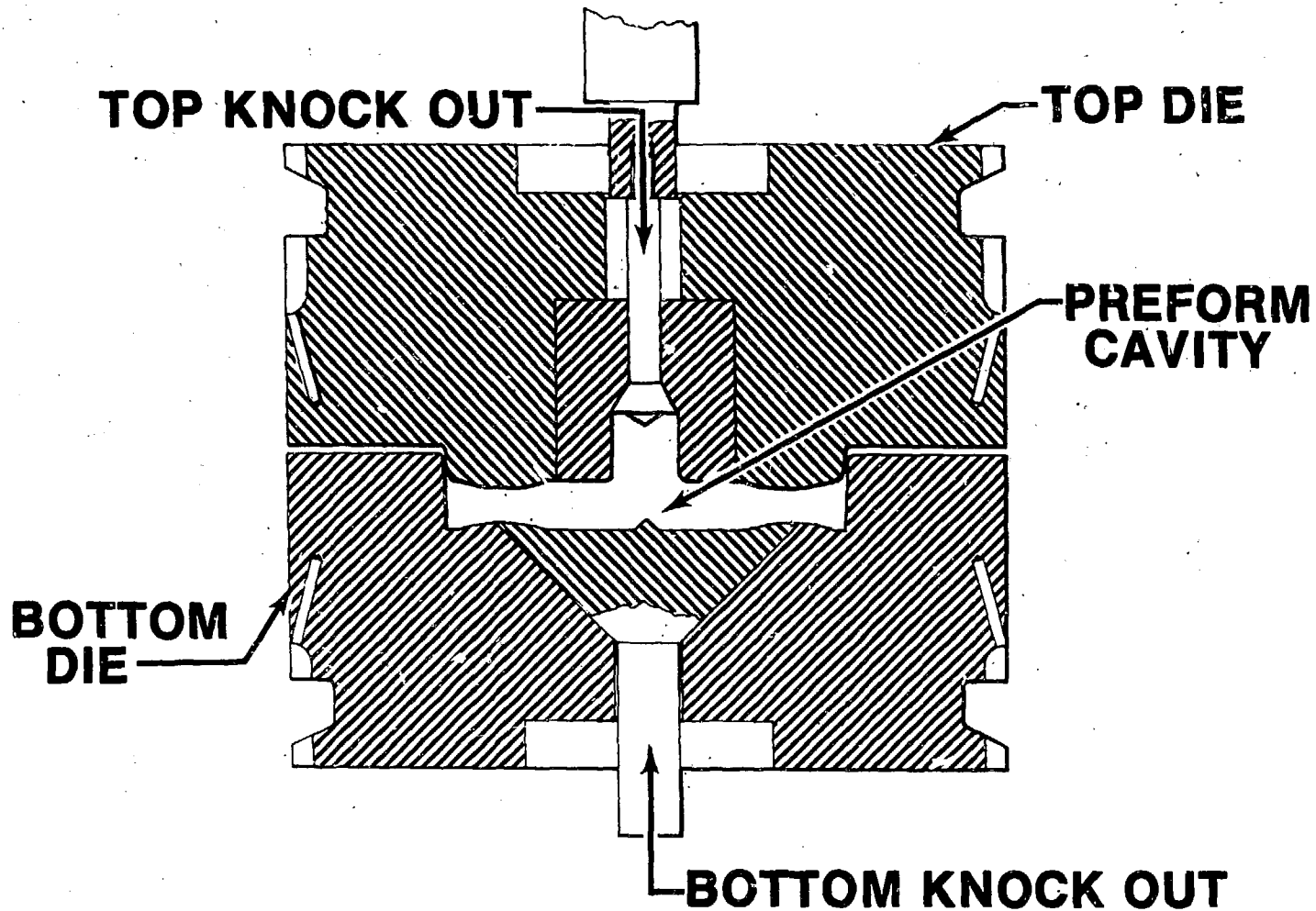
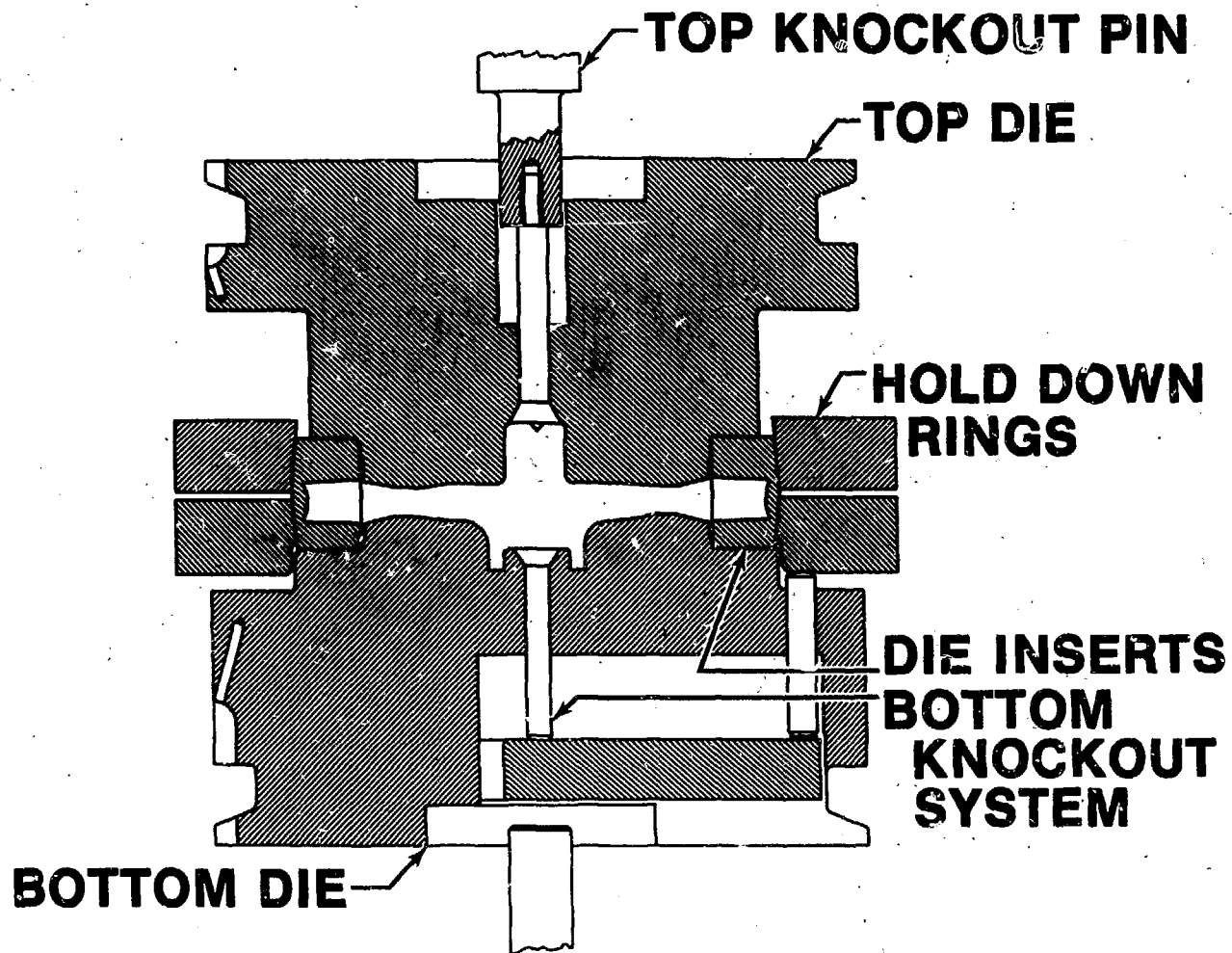


Figure 2. Cross-Section of Preform Tooling



FD 74448C
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0391

Figure 3. Cross-Section of Final Form Tooling

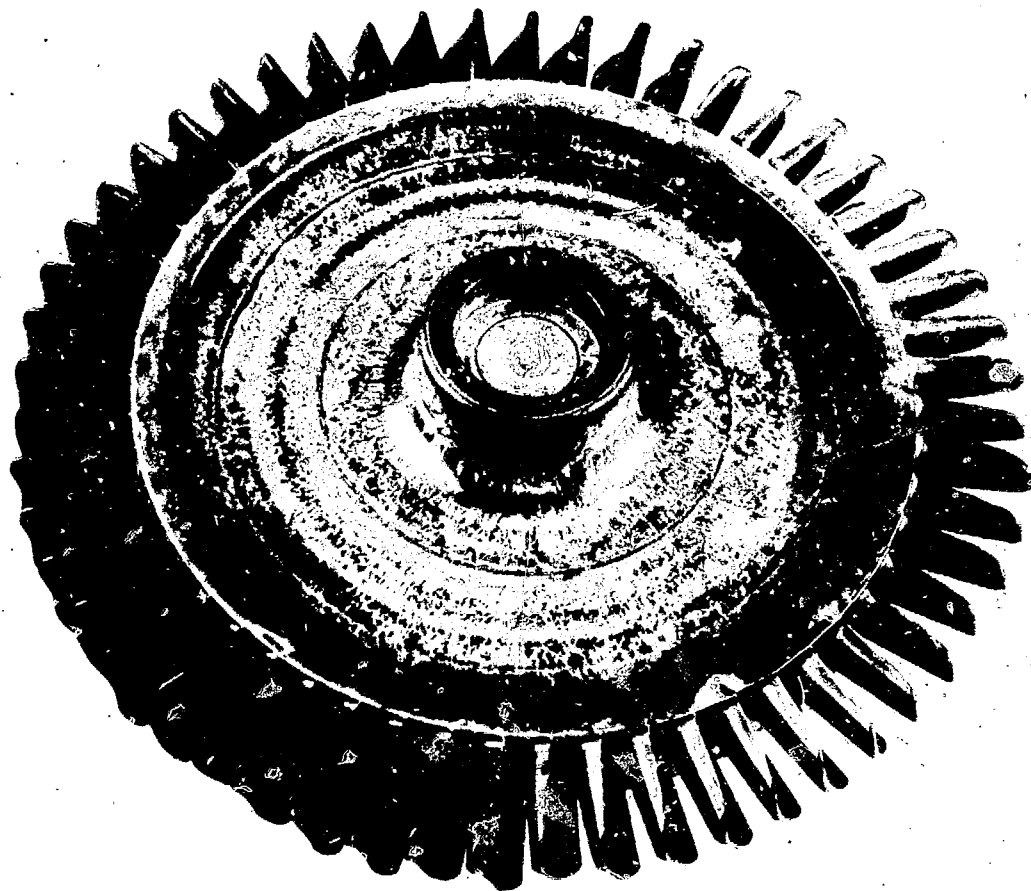
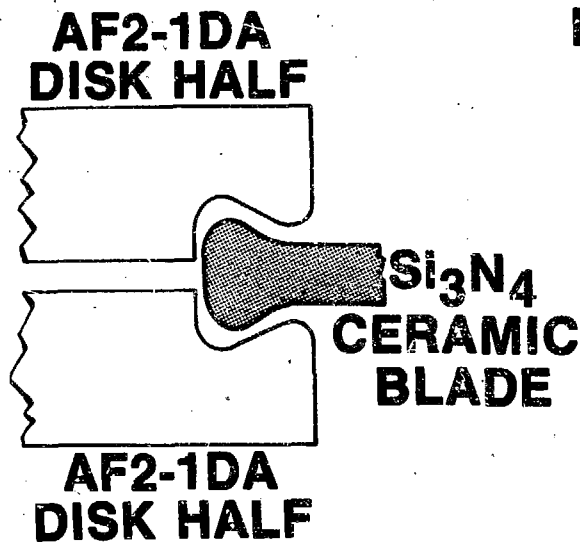
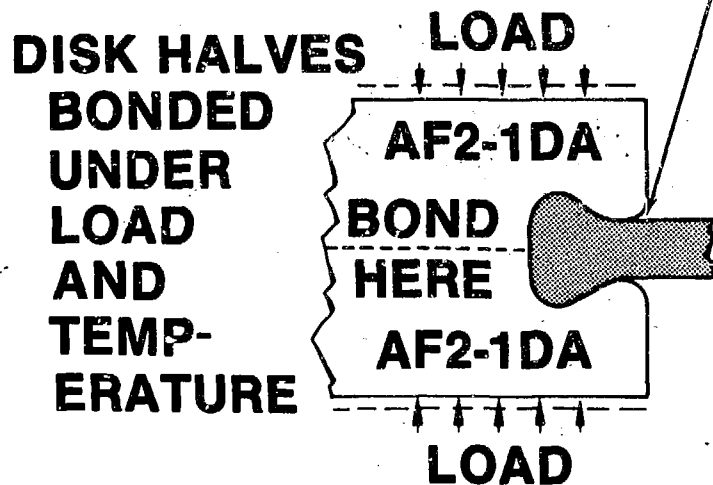
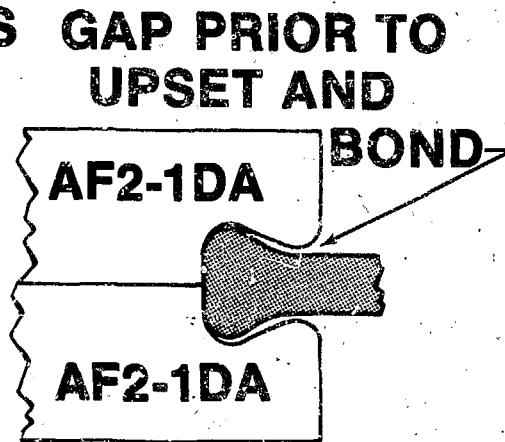


Figure 4. Fully Bladed Turbine Rotor in As Forged Condition



**DISK HALVES
PLACED
TOGETHER
PRIOR TO
UPSET/
BONDING**



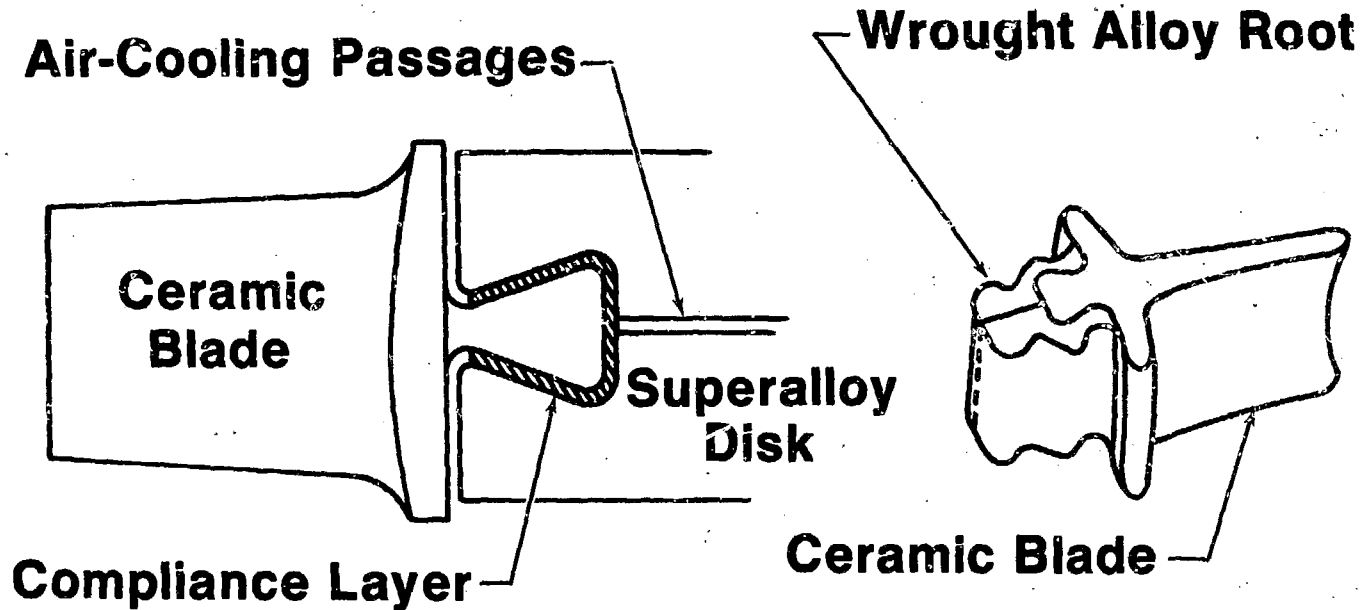
**DISK HALF MATERIAL
FORGED TO MATCH THE
CONTOUR OF CERAMIC
BLADE ROOT
(NO CERAMIC/WROUGHT
BOND IS FORMED. AN
INTERFERENCE FIT IS
OBTAINED BETWEEN
CERAMIC AND
WROUGHT ALLOY).**

Figure 5. Fabrication Process for the Ceramic Wrought Attachment

**Attach Ceramic Directly
To Wrought Disk**

Or

**Attach Ceramic To
Wrought Alloy Root**



AV 104857
772401
T983

*Figure 6. Flexibility of the Ceramic Blade Metal
Disk Attachment Scheme*

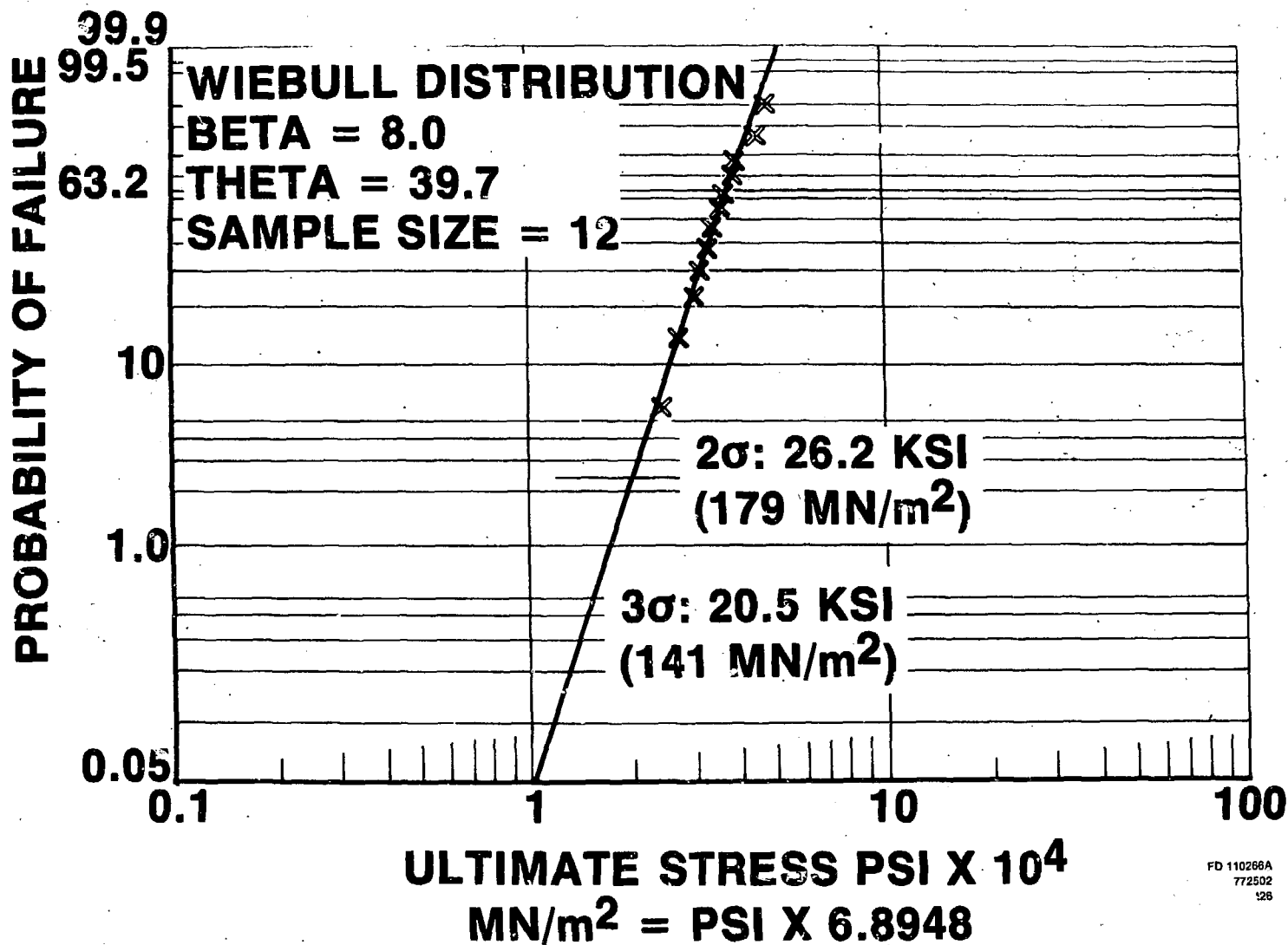


Figure 7. Room Temperature Spin Test Results of
Ultimate Strength for the Ceramic
Wrought Alloy Attachment

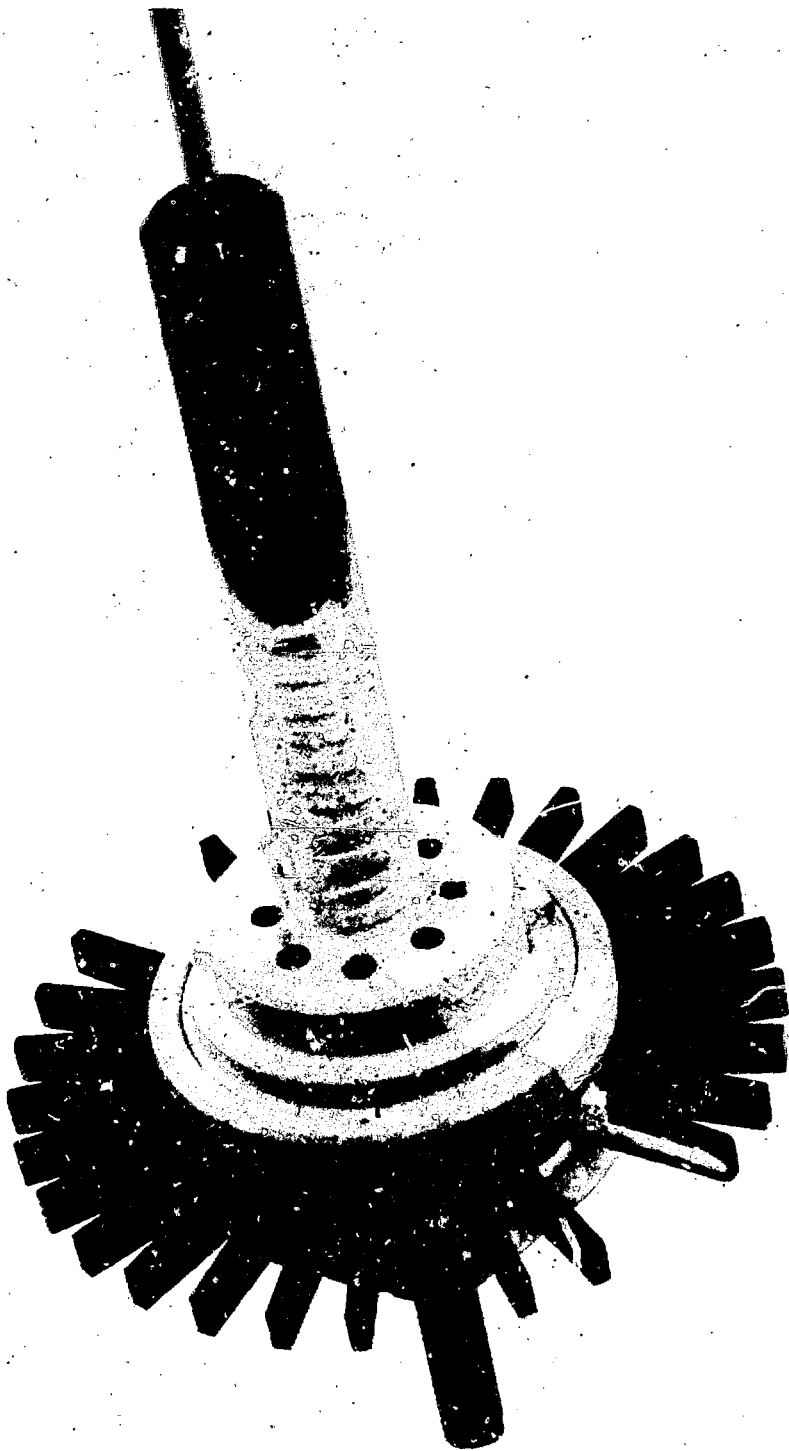


*Figure 8. Instrumented Ceramic Bladed Rotor Hot
Spin Tested for 50 Hours at 2250°F*



*Figure 9. Fully Bladed Hybrid Rotor Spin Tested at
2250°F Blade Temperature for 50 Hours at
45,000 rpm*

COMPLIANT FOIL AIR BEARINGS FOR HIGH PERFORMANCE AUTOMOTIVE GAS TURBINES

S. Gray*

INTRODUCTION

While the objective of achieving a successful gas turbine for an automobile is not new, this engine has received some emphasis in recent years as a possible alternate power plant which potentially offers increased performance (less fuel consumption), reduced pollutants, and competitive costs.

Table 1 prepared in the 1960's lists the important criteria which have to be met in achieving the successful gas turbine powered vehicle [1]**; these criteria have not changed significantly today.

Even as the present generation of gas turbines progresses through development and application, thought is being given to the next generation of Advanced Gas Turbines for smaller vehicles. It is anticipated that a program will be required to develop smaller, faster, and higher temperature engines of less than 100 HP output and constructed from some novel materials.

The purpose of this paper is to discuss the part that the compliant foil air bearing is playing in supporting the Performance, Reliability, and Cost goals of a current engine and its anticipated contribution to advanced engines.

Much of the discussions and projections are based on current work with the MTI Hydresil™ compliant foil air bearing in the Chrysler/ERDA Upgraded Automotive Gas Turbine and on technology support work sponsored by NASA/ERDA.

THE COMPLIANT FOIL BEARING IN THE AUTOMOTIVE GAS TURBINE

The MTI Hydresil, hydrodynamic, compliant foil air bearing as used at the turbine position in the gas generator section of the two-shaft, Chrysler/ERDA Upgraded

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**Numbers in brackets designate references at end of paper.

TABLE 1

IMPORTANCE OF ENGINE CHARACTERISTICS RELATIVE TO THEIR
INFLUENCE ON PERFORMANCE AND ECONOMY OF OPERATION⁽¹⁾

ENGINE CHARACTERISTIC	RATING
LOW FUEL CONSUMPTION.	10
HIGH RELIABILITY.. . . .	9
LOW INITIAL COST.. . . .	9
LOW MAINTENANCE COST.	7
HIGH POWER/WEIGHT RATIO	6
SMALL BULK.	6
RAPID THROTTLE RESPONSE	6
LOW NOISE LEVEL	6
INHERENT TORQUE MULTIPLICATION	5
POWER AT HIGH ALTITUDES	5
LOW VIBRATION LEVEL	4
GOOD ENGINE BRAKING	4
WIDE FUEL RANGE	4
LOW ATMOSPHERIC POLLUTION	3
GOOD COLD STARTING	2
LOW LUBRICATING OIL CONSUMPTION	2
LOW ELECTRICAL INTERFERENCE	1
LOW INFRA-RED RADIATION	1

(KEY: 1 TO 10 IN RISING ORDER OF IMPORTANCE.)

Turbine is illustrated schematically in Figure 1(a). Details of the foil configuration are given in Figure 1(b). The gas generator rotor operates at 58,500 rpm with an air bearing temperature of about 500 F.

The original justifications for the use of a foil bearing at the hot turbine position are listed in Table 2 [2].

The Chrysler program for the Upgraded Engine has reached the stage of approximately 50-60 hours of operation on the engine test bed at full speed and temperature and preliminary checks in the car; this follows simulator development testing of the gas generator rotor-bearings system at MTI.

The illustration of the engine in the car is shown in Figure 2, and Figures 3 and 4 show the foil bearing in the engine and actual engine hardware.

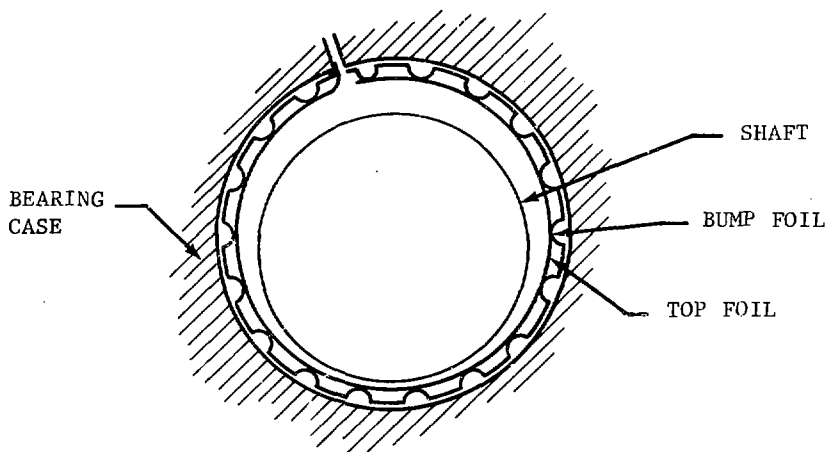
Currently there is considerable interest in the potential for compliant air bearings with both flexible and rigid surfaces for turbo-machinery, particularly as speeds and temperatures continue to increase. This is evident from the number of bearing concepts being disclosed, some of which are shown in Figure 5. Most of these have been reduced to practice.

CONTRIBUTIONS OF THE HYDRESIL FOIL AIR BEARING TO THE AUTOMOTIVE GAS TURBINE

The contribution of the Hydresil foil bearing to the current and future advanced automotive gas turbines will be discussed in the context of Performance, Reliability, and Potential Cost -- much of the discussion is on the basis of recent experience with the Chrysler/ERDA Upgraded Gas Turbine.

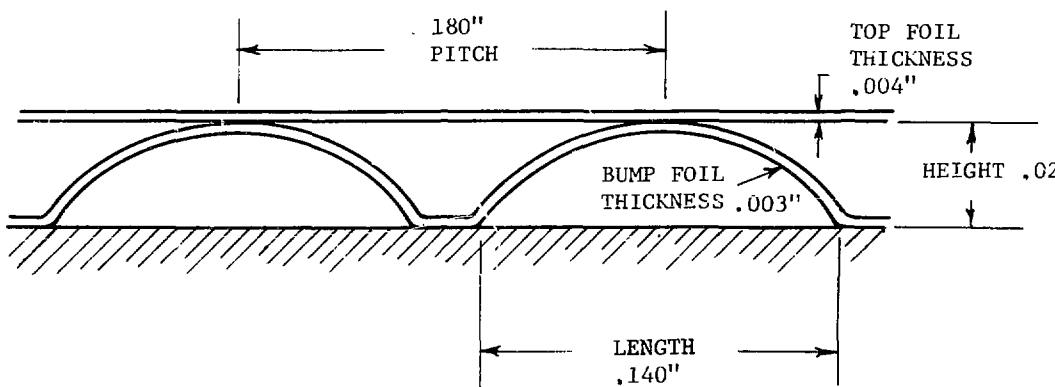
Engine Performance Gain from Reduced Bearing Power Losses

The Upgraded Engine is a derivative of the previous more powerful and lower speed Chrysler automotive engine now referred to as the Baseline Engine. Figure 6 gives a direct comparison of the turbine ends of the gas generators for the two engines which includes the change from an oil journal to the foil air bearing.



(a) SCHEMATIC

BEARING DIAMETER: 1.5"
 LENGTH: 1.5"
 STIFFNESS: 20,000 lb/in.



(b) DETAILS OF BEARING

FIG. 1 HYDRESIL FOIL JOURNAL BEARING - UPGRADED ENGINE

TABLE 2

ADVANTAGES OF AIR FOIL BEARING IN UPGRADED GAS GENERATOR

FOIL AIR BEARING	● REDUCED FRICTIONAL POWER LOSS. ● REDUCED OIL SUPPLY REQUIREMENTS. ● ELIMINATION OF OIL SEAL AND FIRE HAZARD AT TURBINE END. ● COMPATIBILITY WITH LOW COST HOUSING MATERIALS. ● POTENTIAL FOR DEVELOPMENT TO HIGHER SPEEDS AND TURBINE INLET TEMPERATURES.

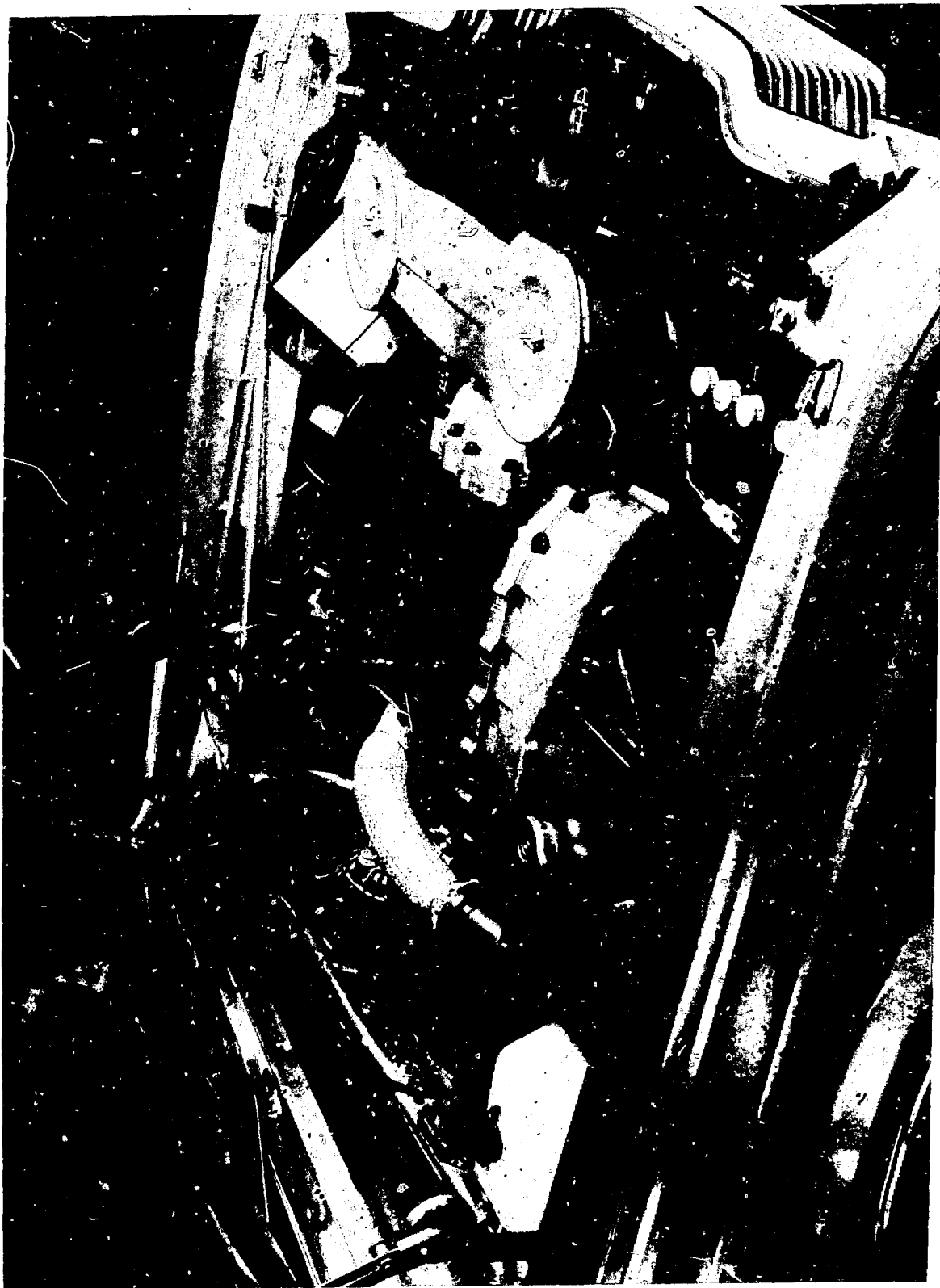


FIG. 2 CHEVSELE/EBDA UPGRADED ENGINE IN CAR

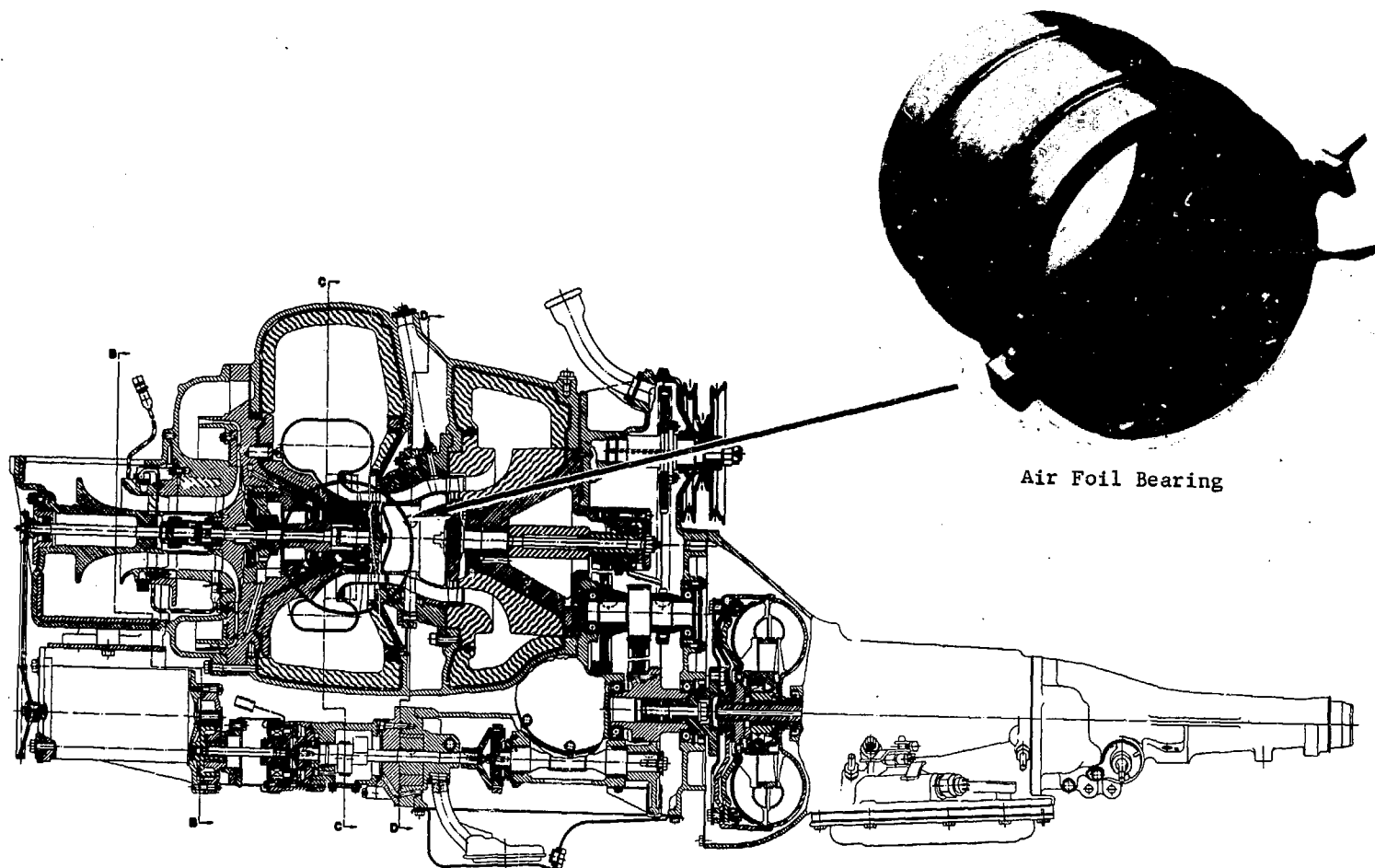


FIG. 3. HYDRESIL FOIL BEARING IN GAS GENERATOR OF CHRYSLER ENGINE

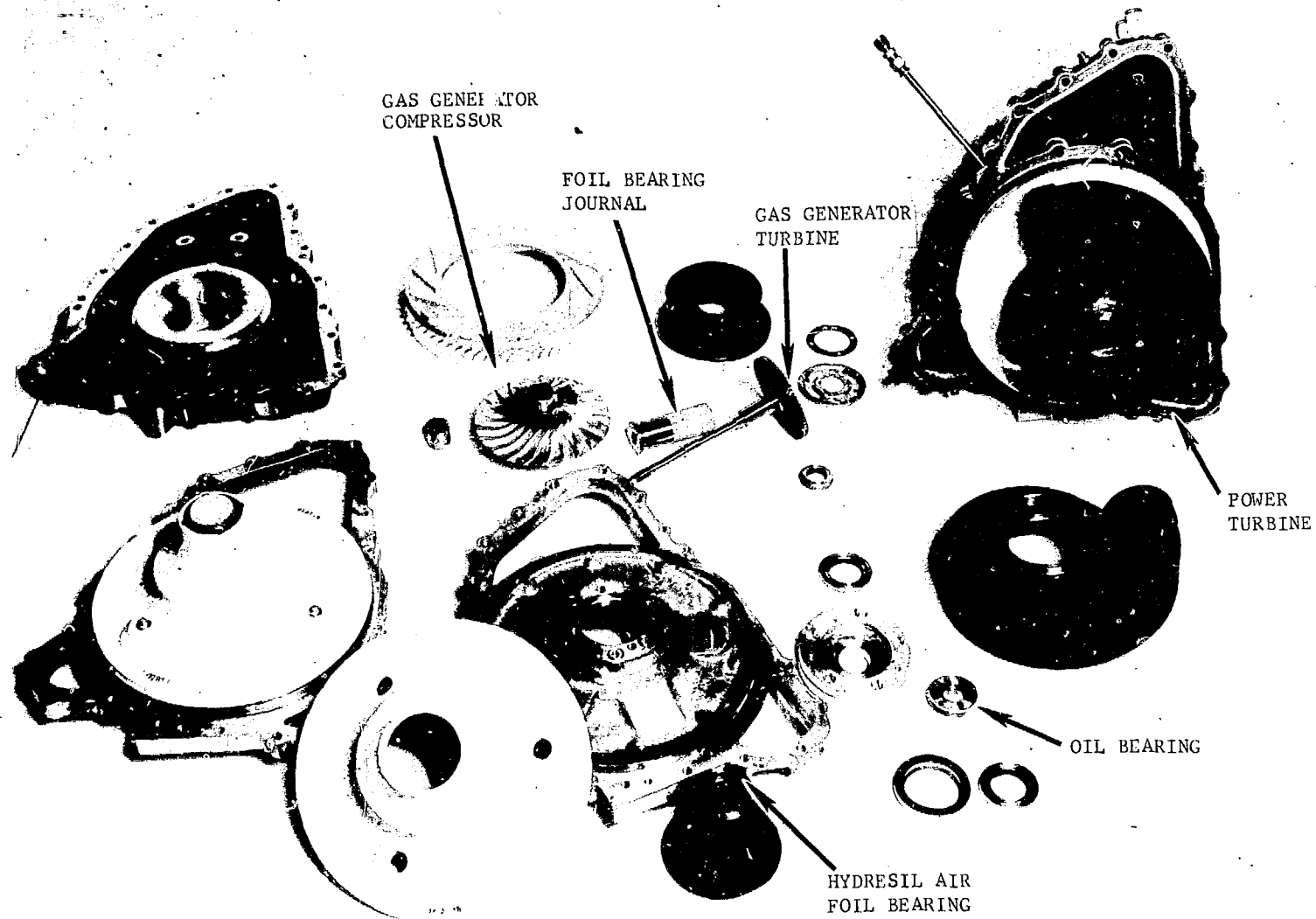
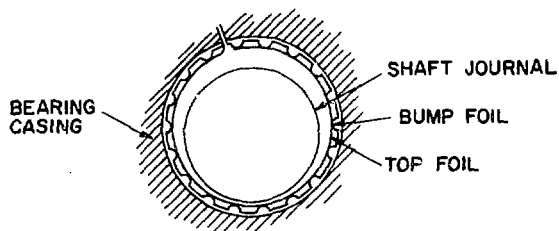
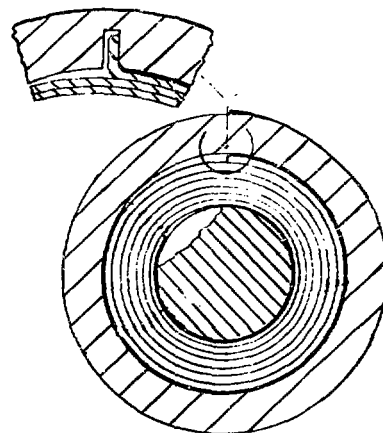


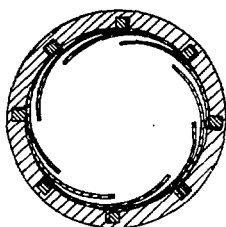
FIG. 4. ENGINE PARTS SHOWING FOIL BEARING AND JOURNAL



HYDRESIL SINGLE PAD
FOIL BEARING



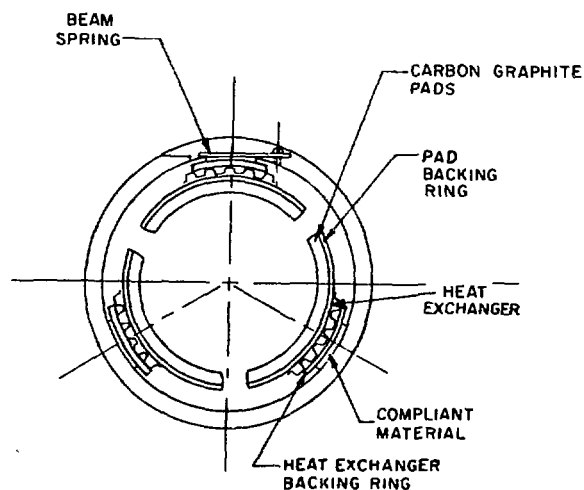
SPIRALLY WOUND FOIL BEARING (3)



MULTI-LEAF FOIL JOURNAL BEARING (4)



3-PAD TENSION FOIL BEARING (5)



3 RIGID PADS - COMPLIANTLY MOUNTED
BEARING (6)



CANTILEVER MOUNTED RESILIENT
PAD THRUST BEARING (7,8)

FIG. 5. EXAMPLES OF FOIL AND COMPLIANT BEARING CONCEPTS

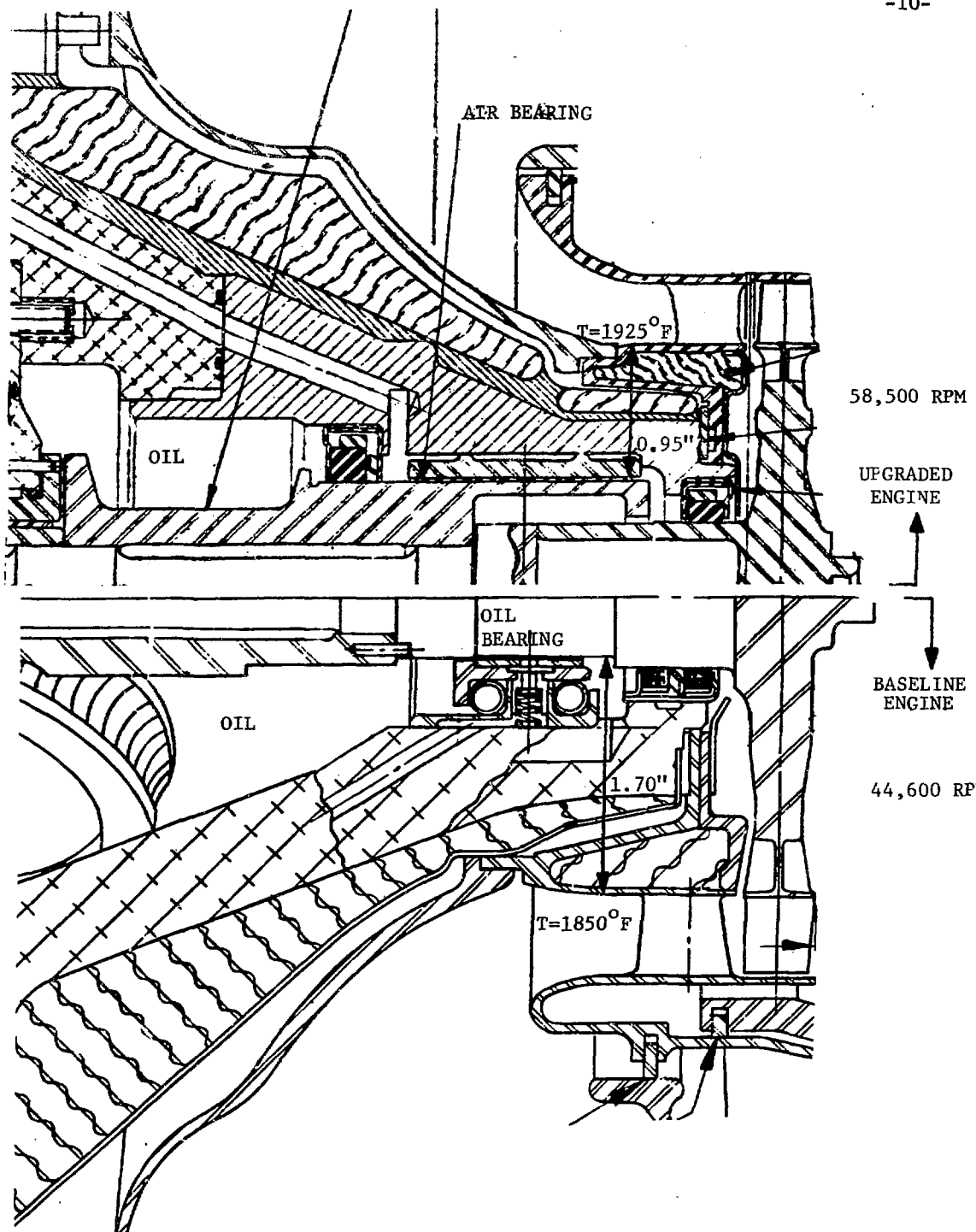


FIG. 6. COMPARISON OF GAS GENERATOR TURBINE BEARINGS

Despite the increase in bearing size and speed of operation, the air bearing requires 1.0 less horsepower than the smaller oil bearing to overcome frictional drag when operated at the new higher speed. If foil air bearings are used at all positions in the gas generator, as shown schematically in Figure 7, then an additional power saving of 4.0 HP is achievable. This additional savings, detailed in Table 3, also includes influences such as seal changes and elimination of shear pumping losses related to oil in the housings. These power savings are certainly not insignificant for engines in the 50-100 HP output range.

The type of foil thrust bearing applicable to this analysis is shown in Figure 8.

Engine Performance Gains from Increased Cycle Temperatures

Returning to Figure 6, the increase in the turbine inlet temperature to achieve greater performance and the shorter heat flow path to the turbine bearing will be noted for the Upgraded Engine. These trends are expected to continue for the advanced gas turbine.

The operating temperature of the oil at the turbine bearing in the Baseline Engine is approximately 300 F and reaches approximately 400 F at soak back. It is apparent that any significant increases in these temperatures which must be anticipated in smaller, higher performance engines will give rise to oil coking problems affecting engine life and reliability. The benefits from the elimination of the oil at this bearing are thus significant.

A careful thermal analysis made for the Upgraded Gas Generator design with the foil air bearing [2] was important in arriving at the optimum design of the bearing area and of the cooling air arrangement. Figure 9 shows the thermal map which was developed. The corresponding temperatures to those given above with the foil air bearing are 500 F operating and 650 F at soak back, these temperatures are quite acceptable in the foil bearing and can be exceeded in

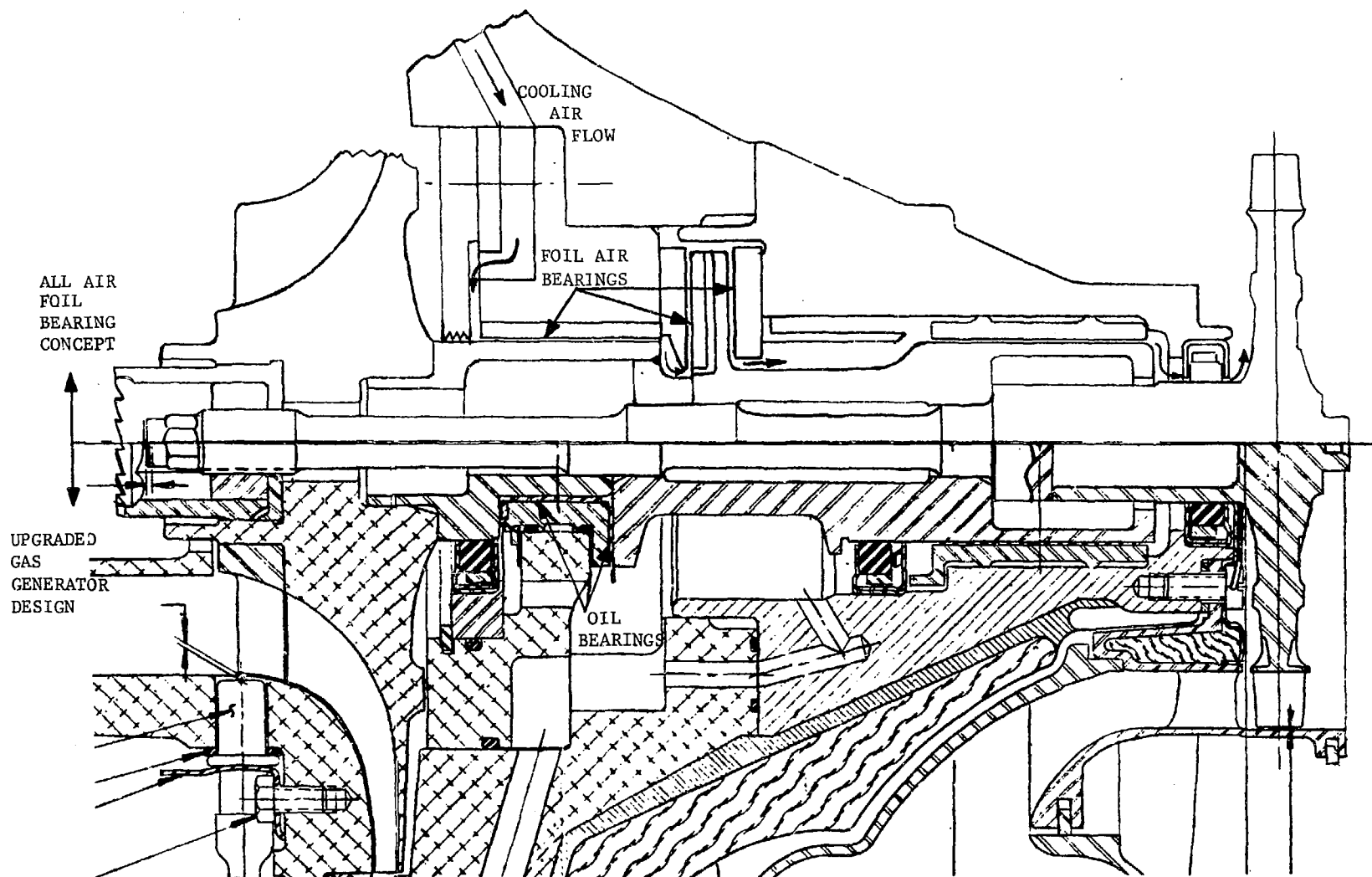


FIG. 7. CONCEPT FOR ALL AIR FOIL GAS GENERATOR

TABLE 3

FRICION HP POWER SAVING OVER UPGRADED GAS GENERATOR
WITH ALL AIR FOIL BEARING SYSTEM

SOURCE OF CHANGE	HORSEPOWER		POWER SAVINGS HP
	AIR	OIL	
COMPRESSOR END JOURNAL BEARING	0.1	1.1	1.0
MAIN THRUST BEARING	0.5	1.6	1.1
REVERSE THRUST BEARING	0.1	0.3	0.2
MID-SHAFT SEAL	-	1.5 ⁺	0.8
COMPRESSOR SEAL	0.01	1.5 ⁺	0.8
"PUMPING" IN CAVITY (ESTIMATE)	0	0.1	0.1
	TOTAL SAVINGS		4.0

+ Based on complete oil film

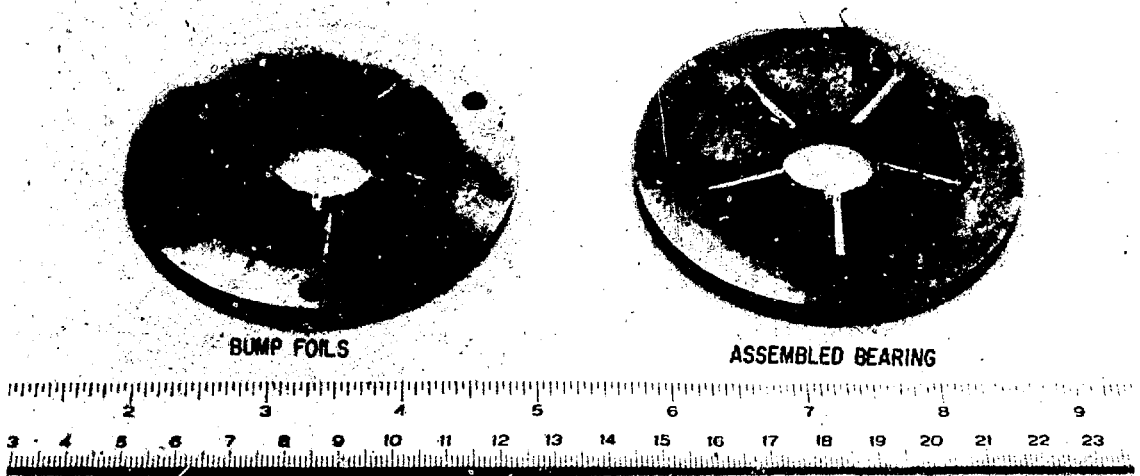


FIG. 8. AIR FOIL THRUST BEARING

AIR BEARING ENLARGEMENT

NOTES:
ALL VALUES ARE METAL
TEMPERATURES UNLESS
OTHERWISE INDICATED.
VALUES WITH IDENTIFYING
LETTER ARE SOAKBACK
TEMPERATURES
TEMPERATURES, °F

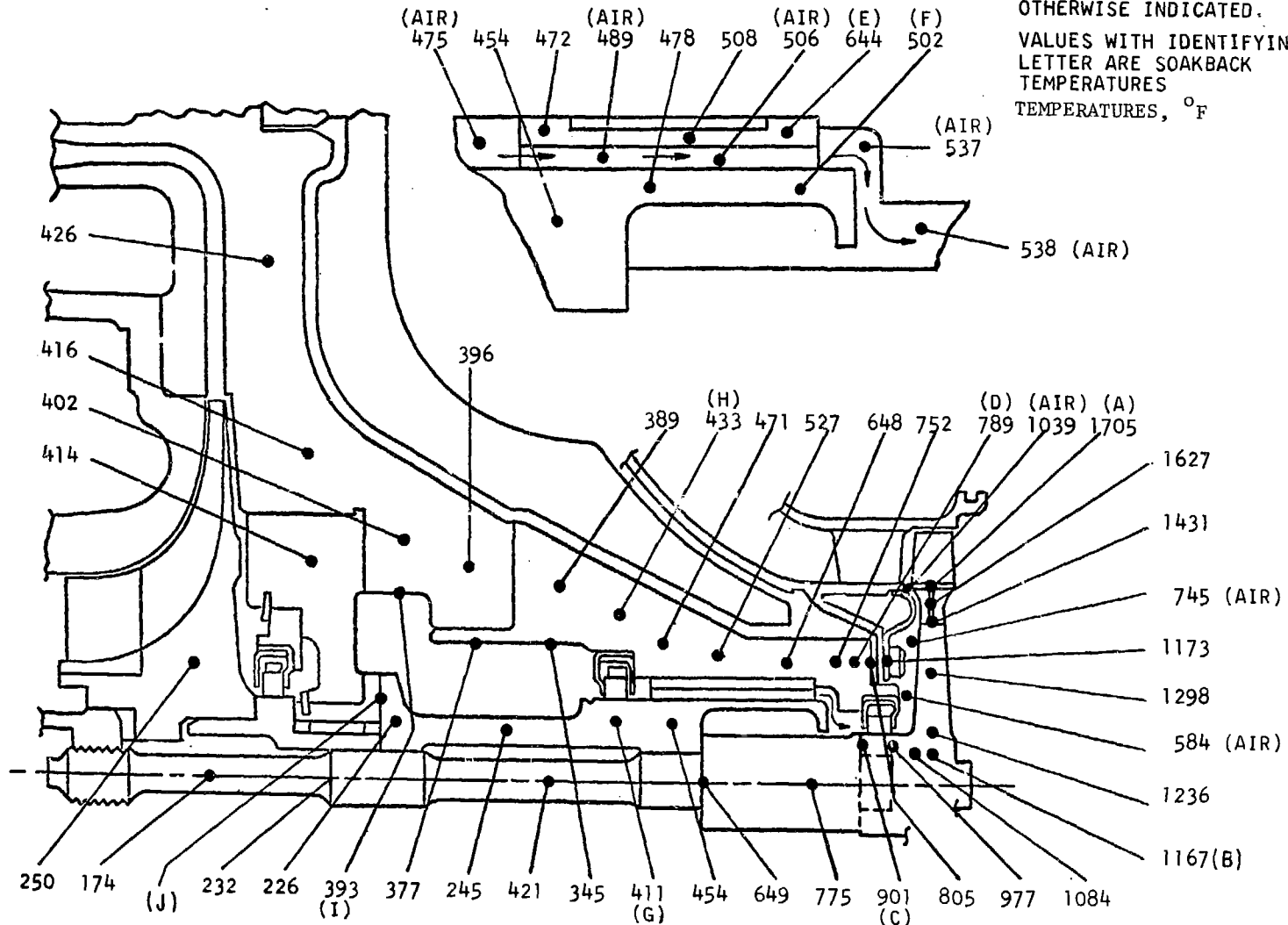


FIG. 9. THERMAL MAP - UPGRADED GAS GENERATOR

future designs thus permitting higher turbine inlet temperatures and further performance gains.

Determination of Bearing Performance and Reliability from Simulator Testing

The gas generator simulator for the Chrysler/ERDA Upgraded Engine played an important part in the development work ahead of the first engine built in determining bearing-rotor performance. It is now being used in parallel with engine testing in developing reliability criteria for the bearing-rotor system.

The simulator under high temperature testing is shown in Figure 10, and Figure 11 shows the rotor and bearings.

Recent activities have emphasized simulation of vehicle shock and maximum rotor unbalance in the testing, and future work will explore the effect of bearing clearance tolerances on performance.

Figure 12 illustrates the results of testing in which shock loads of up to 16 g were imposed on the simulator with the rotor at high speed. This shock level is representative of combined road shock and vehicle slew loading. Also shown in Figure 12 are the rotor orbits measured outboard of the dummy turbine wheel with unbalance levels of 0.006 oz. in. and 0.010 oz. in. at both wheel planes. These small orbits include manufacturing run-out errors of approximately 0.0005 in. T.I.R. The 0.010 oz. in. level of unbalance is the maximum anticipated from balance deterioration during vehicle operation and prior to engine servicing.

In both the shock and unbalance tests, no surface damage was experienced in the air bearing.

Cost and Manufacturing Considerations

While major emphasis has been on developing the bearing system for the engine, continuing thought has been given to bearing cost and manufacturing requirements for the high volume production necessary for a viable product.

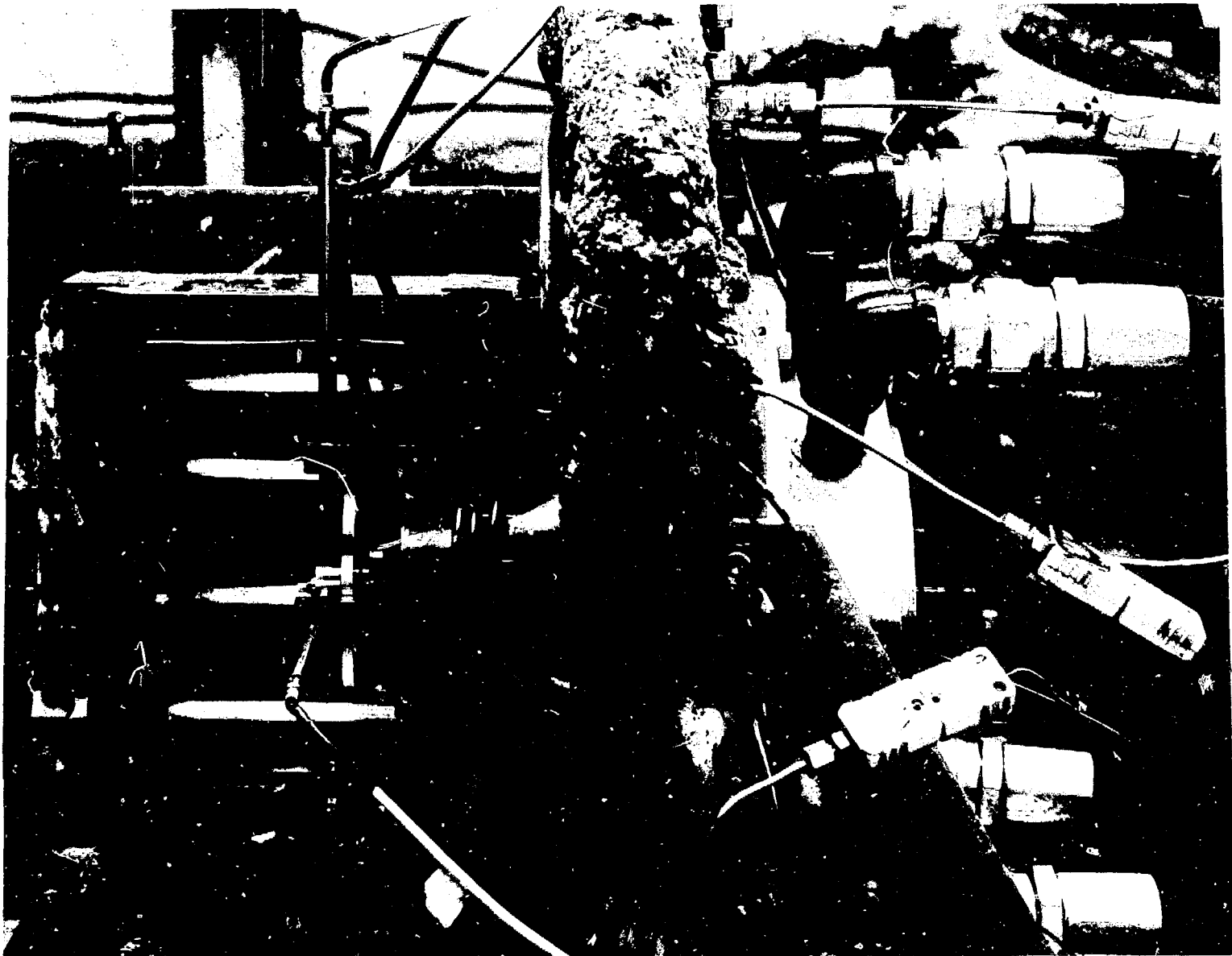


FIG. 10 HIGH TEMPERATURE TESTS IN SIMULATOR

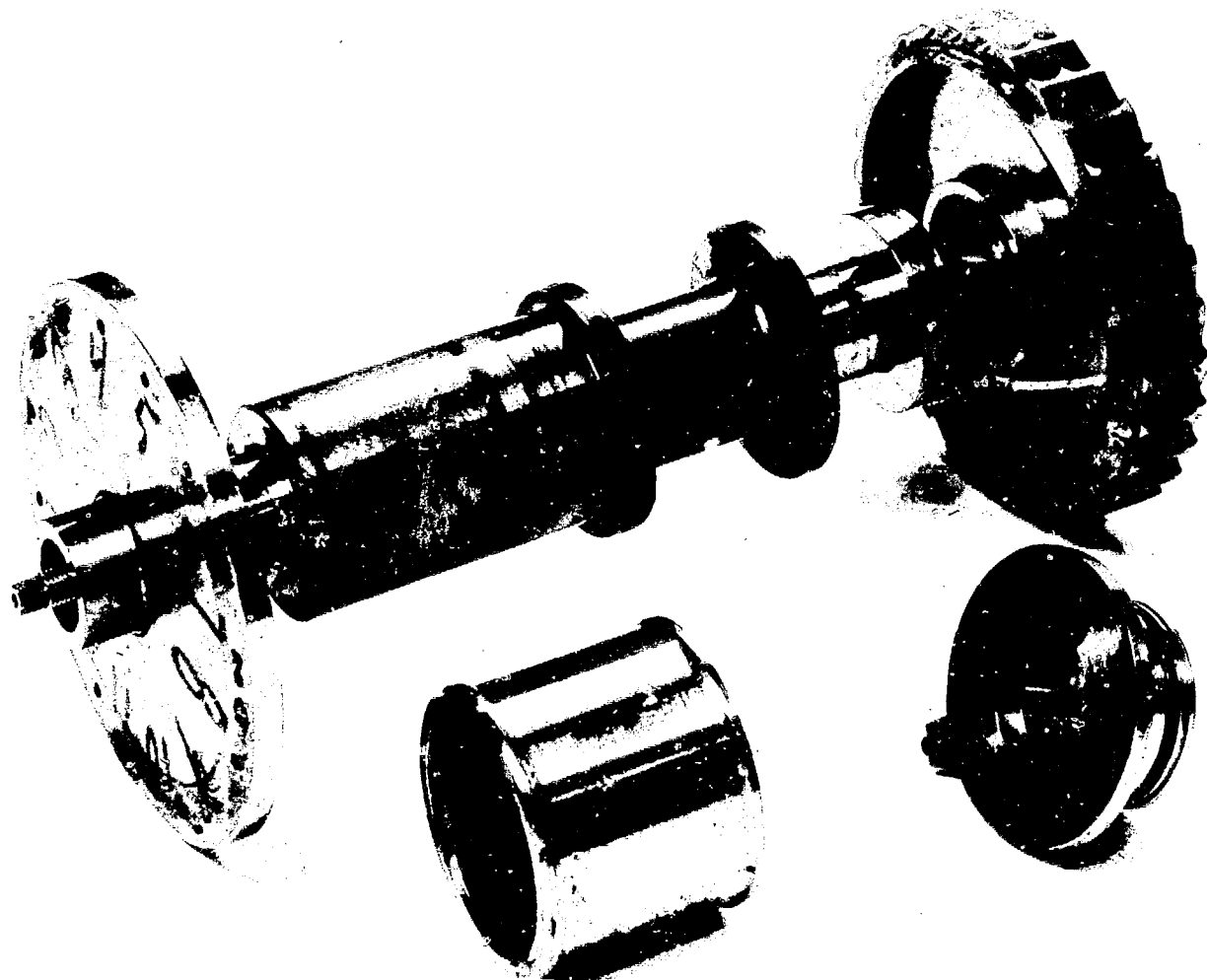
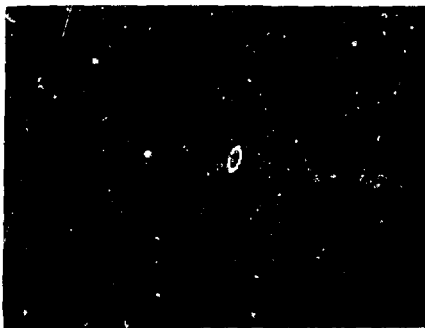
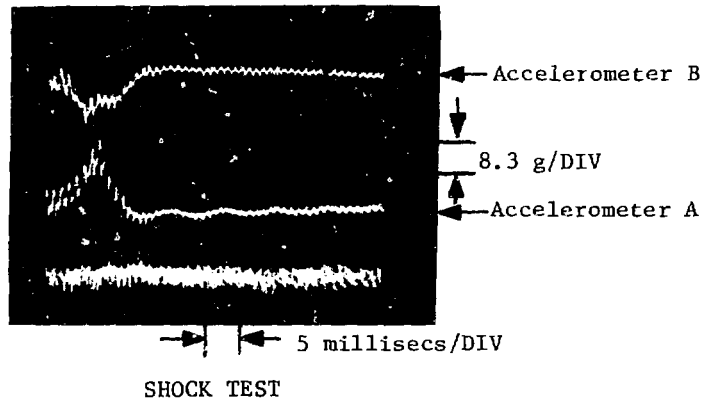
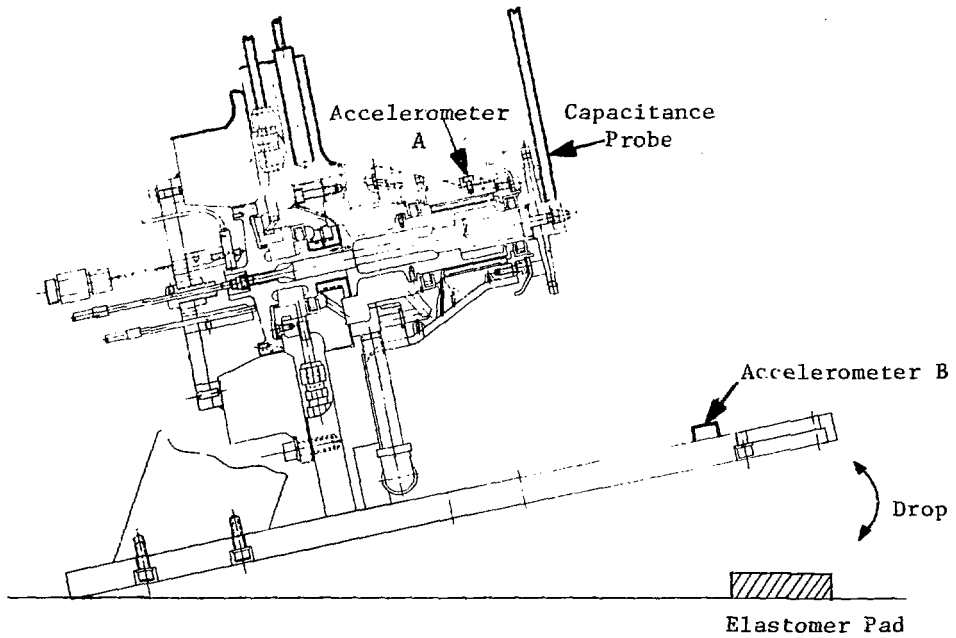


FIG. 11. SIMULATOR ROTOR AND BEARINGS



0.006 oz-in.
Unbalance in Two Planes

TURBINE END
ROTOR ORBITS



0.010 oz-in.
Unbalance in Two Planes

FIG. 12. SHOCK AND ROTOR UNBALANCE TESTS IN SIMULATOR
AT 90° SPEED

A foil bearing is not a one-for-one replacement for an existing oil bearing, and other engine and system items are subject to changes which affect cost. These changes are listed below:

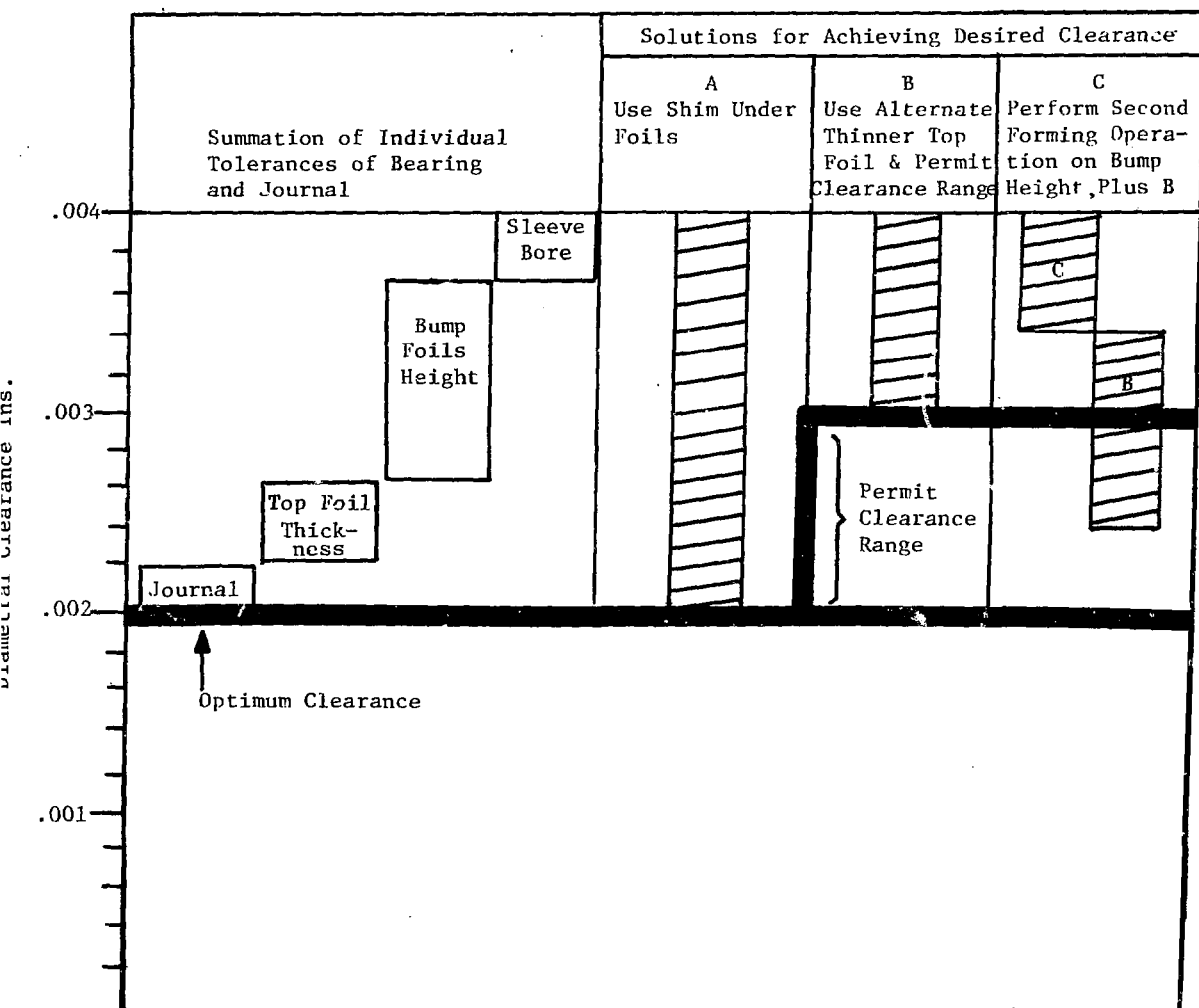
- Simple less critical single pad bearing
- New rotor sleeve material and design
- Reduced oil system requirements
- Potentially simplified seals
- Lower cost bearing housing
- Reduced insulation

Most of these changes are advantageous from a cost savings viewpoint, and a preliminary cost review at Chrysler was positive in this regard.

Other cost saving studies involving replacement of rolling element and oil film bearings systems with foil bearing systems [9, 10] have also given quite positive results.

Looking specifically at the Hydresil foil air journal bearing, which has the virtue of the minimum number of parts, this is presently being built in prototype quantities without the benefit of "Detroit" automotive high-volume manufacturing engineering techniques being applied. One critical factor influencing cost is the control of the cold bearing clearance. This clearance is affected by tolerances on the rotor journal, foil members (this enters twice), and bearing sleeve bore.

In the simulator testing, diametral clearances ranging from 0.0015 in. to 0.0035 in. have been used. Bearings for the engine are being set up with 0.002 in. clearance; and as mentioned before, further work is underway to establish a satisfactory clearance range. Figure 13 shows the present situation on



Assumptions

Rotor and Sleeve Bores Used as Manufactured. All Adjustments Made on Foil Members or with Shims.

FIG. 13. MANUFACTURING CONTROL OF FOIL JOURNAL BEARING OPERATING CLEARANCE

bearing-rotor-sleeve tolerances and three methods suitable for automatic manufacturing and gauging which can be used for clearance control in high volume production.

In these methods rotors and sleeve parts are used "as manufactured" and all adjustments are made on foil elements, consequently interchangeability and servicing requirements can be met.

BASIC TECHNOLOGY WORK DIRECTED TO THE ADVANCED AUTOMOTIVE GAS TURBINE

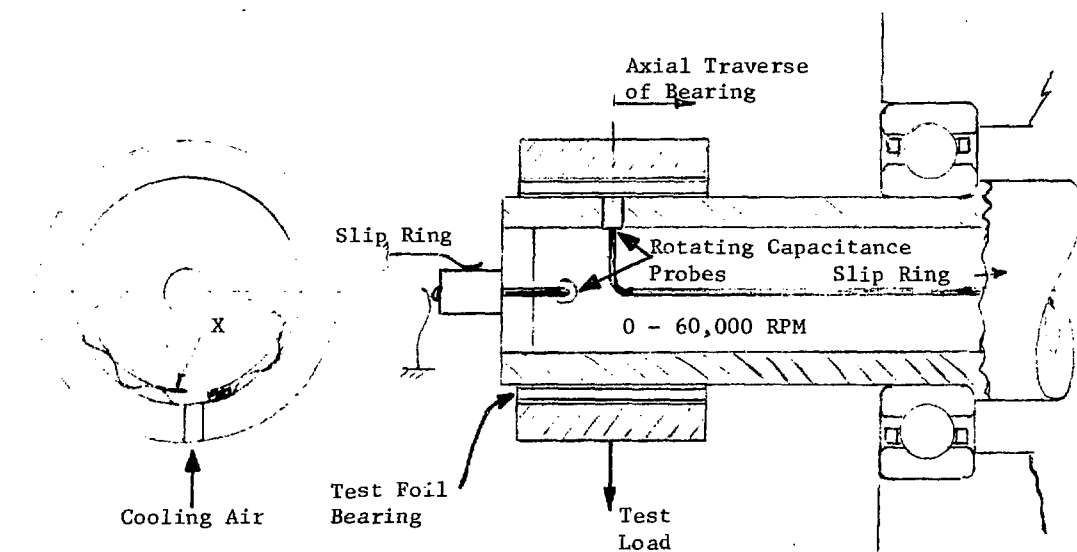
A continuing technology activity is essential in order to ensure the foil air bearing will be ready for the requirements of the future advanced gas turbine, particularly in the areas of: (a) increasing load capacity beyond the present 25 psi level to permit the use of small bearings, (b) developing higher temperature bearing materials and coatings, and (c) achieving higher reliability under severe long-life service conditions.

Some technology work is currently in progress under NASA/ERDA sponsorship aimed at gaining a better technical understanding of the foil bearing characteristics which will lead to increased load performance, and for achieving bearing materials and coatings suitable for high temperature services.

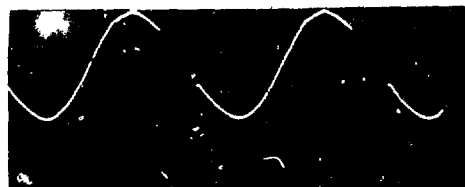
Foil Bearing Technology

Figure 14 illustrates results from recent work with the Chrysler sized foil air journal bearing in which the minimum film thickness is measured at speed and under load in both the circumferential and axial directions using rotating capacitance probes mounted in the journal.

The wide minimum film thickness bands of the compliant foil bearing relative to a rigid air bearing illustrated in Figure 14 are indicative of the potential for higher load capacity and tolerance to shock loading of the foil bearing. Also illustrated is the detection of foil excitation from a poorly positioned

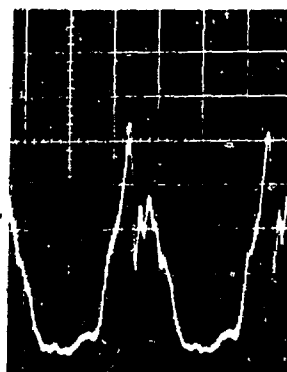


TEST ARRANGEMENT



Reference Data with Rigid Air Bearing

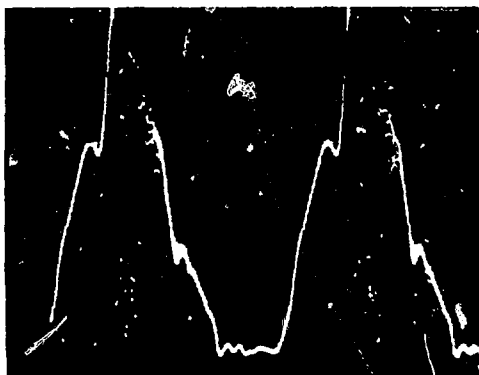
N=14,300 RPM
Load=11.9 lb.
Min Film = 375 μ in.



N=45,000 RPM
Load=43.2 lb.

One Revolution

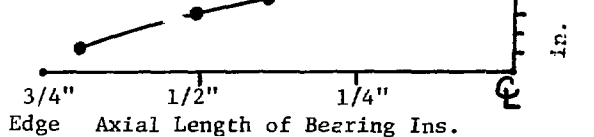
Film Thickness of Foil Bearing



Excitation of Top Foil at X by Cooling Air

N=30,000 RPM
Load=9-1/4 lb.

Min. Film Thickness



Axial Min. Film Thickness Scan

FIG. 14. TECHNOLOGY DATA - AIR FILM THICKNESS OF FOIL BEARINGS

cooling air supply. Studies such as these coupled with good analysis give directions for the achievement of greater bearing performance.

Material and Coating Technology

The Upgraded Engine is operating with a bearing constructed from Inconel X-750 with a commercial MoS_2 type dry film lubricant coating for meeting the start-stop sliding contact requirements. The mating journal sleeve is made of stainless steel with a flame sprayed and ground chrome carbide surface.

This combination has proved to be satisfactory in simple rig tests of 11,000 contact cycles, in simulator tests at temperature, and in the limited engine testing. There is, however, some evidence that the loading on the foil bearing surface during sliding contact is higher than originally assumed. This is probably the result of engagement and acceleration of the rotor by the starter motor. There is a continuing need for improved coatings and materials, and the problem becomes increasingly severe as the temperatures are increased.

The present technology effort applicable to the advanced gas turbine is directed towards foil bearing materials and coatings suitable for 1000 F operating and 1200 F soak back temperatures. Base materials of Inconel X-750 for the foils and A-286 for the journals have been selected and the major emphasis is on coatings.

Table 4 details the range of candidate coatings and combinations which have been selected, applied to material coupons, and subjected to the following static oven tests:

300 hours at a test temperature of either 1200 or 1000 F

10 temperature cycles

This screening has been very encouraging in that an adequate number of candidate coatings passed the test. Many of the coatings which were unsuccessful failed during the temperature cycling.

TABLE 4. SELECTED COATING COMBINATIONS FOR STATIC SCREENING

FLAT (SHAFT) A-286	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22
FOIL INCONEL X-750	TiC (sputtered)	B ₄ C (sputtered)	CrB ₂ (sputtered)	Ni-Cr Bonded CrB ₂ (Plasma Sprayed)	Ni-Cr Bonded Cr ₂ O ₃ (Plasma Sprayed)	Borided A-286	Tribaloy 800 Ni-Aluminide Undercoat (Plasma Sprayed)	Metco J ₂ C ₂ with a Binder (Plasma Sprayed)	Linde Cr ₃ C ₂ with a Binder (Plasma Sprayed)	Cr ₂ O ₃ (sputtered)	WC (sputtered)	NASA PS-101 (Plasma Sprayed)	Cr ₃ C ₂ (sputtered)	Electrolyzed A-286	Silicon Nitride (sputtered)	Al ₂ O ₃ (sputtered)	Al ₂ O ₃ (Plasma Sprayed)	CdO and Graphite with Sodium Silicate (Fused Coating)	Ni-Co (Electroplated)	Nitrided A-286	NASA PS-100	Silicon Nitride Suspended in Electroless Nickel
A (sputtered)	(1)	(3)																	(26)			
B (sputtered)		(2)																	(27)			
C (sputtered)			(4)	(6)		(9)	(10)															
D (sputtered)				(5)																		
E (sputtered)					(7)									(21)								
F (sputtered)			(8)				(19)			(20)												
G (sputtered)								(11)		(13)												
H (sputtered)									(12)		(14)											
I (Fused Coating)											1000F											
J (sputtered)													(18)									
K (Fused Coating)															(22)							(31)
L (sputtered)									(24)								(25)					
M (Fused Coating)																						
N (Chemical Adherent Coating)																		(32)				
O (Uncoated Foil)												(17)										

Table 5 details coatings and combinations selected from above which will next be given S.E.M. and X-ray defraction examinations prior to being used in dynamic bearing tests. The dynamic start-stop cycle testing of the coatings will be done on partial arc bearings under 2 psi and 5 psi loading at temperature in the test fixture shown in Figure 15.

Emphasis in the materials and coatings work has been placed on using thin coatings (5,000 angstroms) applied by sputtering since this is particularly attractive with the thin foil materials, and also on coating both the journal and foil surfaces.

Figure 16 shows equipment which will be used in future parametric studies aimed at optimizing the sputtering procedures of the most promising coatings.

Looking further to the future and the trend towards complete ceramic components in the engine, it is encouraging to note that, in general, ceramics make good bearing surfaces. Further thoughts on all ceramic bearings will be found in Reference [11].

THE SINGLE SHAFT AUTOMOTIVE GAS TURBINE

It is not clear at this time whether success for the advanced automotive gas turbine will be achieved with a single or multi-shaft design since many design parameters are involved [12]. An example of a single-shaft gas turbine configuration is shown in Figure 17 [13].

Two significant features demonstrated in this Ford design are: (a) the introduction of many ceramic components for the high operating temperatures and (b) the use of a simple spur gearing power output train at the cooler compressor end of the rotor.

The use of this gearing arrangement whether on a single or multi-shaft design results in radial loading of the foil bearing at the turbine end and may

TABLE 5

COATINGS AND COMBINATIONS SELECTED FROM STATIC OVEN SCREENING TESTS

Shaft → Foil ↓	CrB ₂ (P.S.)*	Tribaloy 800 (P.S.)	Linde Cr ₃ C ₂ (P.S)	Cr ₂ O ₃ (Sp.)**	NASA PS106 (P.S.)	Kaman Des
TiC (Sp.)	①	⑮		⑯		②
Cr ₂ O ₃ (Sp.)	⑰	⑤		③		④
Kaman Des		⑱		⑦	⑳	⑥
CdO + Graphite (fused)			⑧	⑭	⑬	
Si ₃ N ₄ (Sp.)	⑨	⑰				
Uncoated Foil					⑩	
Cr ₂ O ₃ (Sp.) + Ag (Sp.)				⑪		
TiC (Sp.) + Ag (Sp.)			⑫			

*Plasma Sprayed

**Sputtered

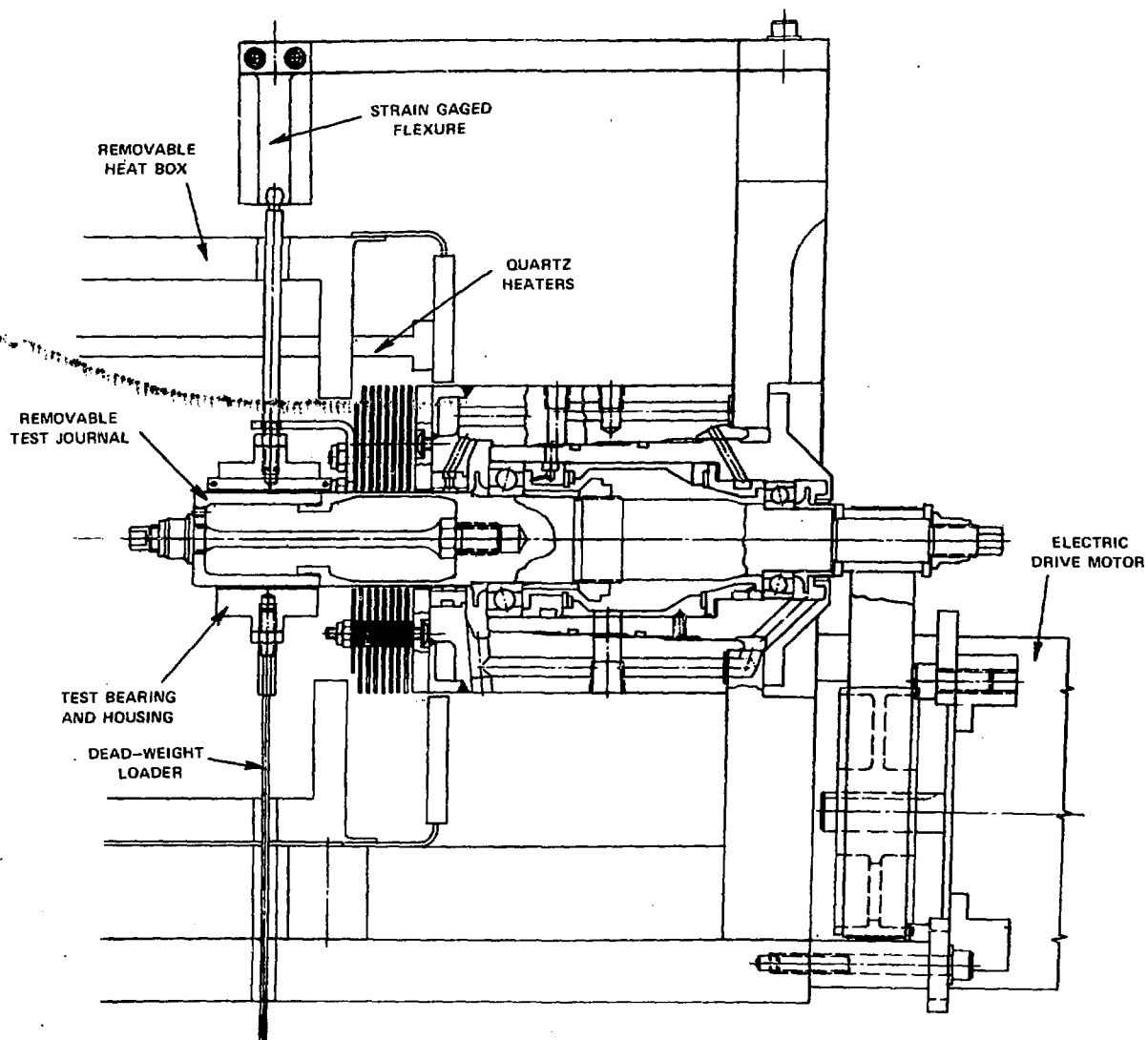


FIG. 15. FOIL JOURNAL BEARING MATERIALS TEST RIG

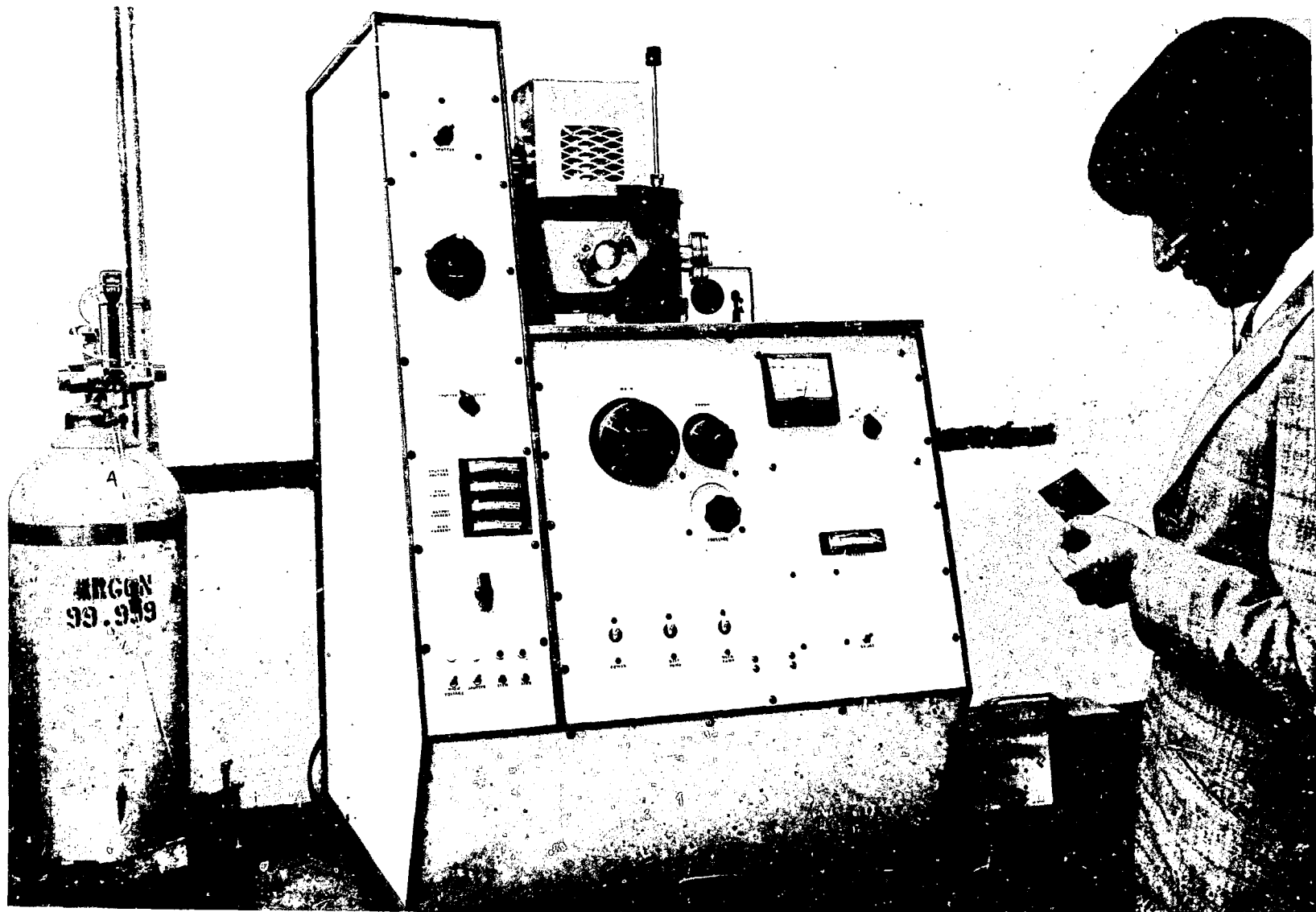


FIG. 16. THIN COATING SPUTTERING EQUIPMENT

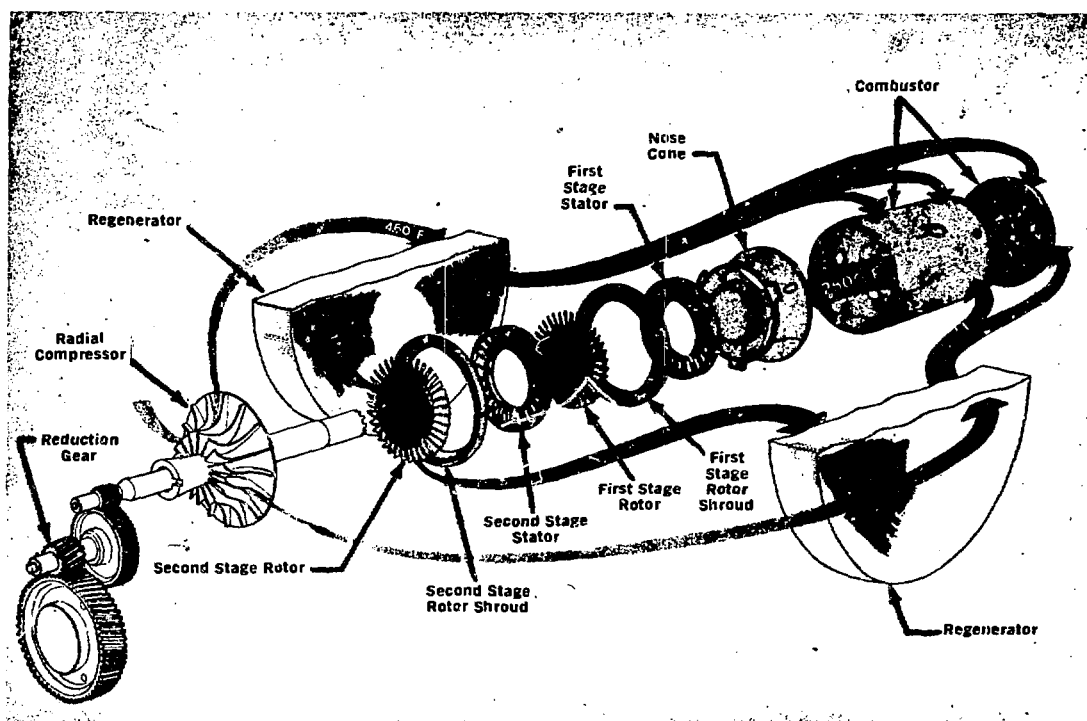


FIG. 17. FORD SINGLE SHAFT VEHICULAR GAS TURBINE

necessitate the use of an undesirably large bearing. The use of a planetary gear system with balanced radial loading would be beneficial to the bearing designer.

Figure 18 shows the simulator rotor and bearings for the Ford engine during preliminary testing with foil bearings.

BEARING REQUIREMENTS FOR THE ADVANCED GAS TURBINE - CONCLUSIONS

The gas turbine designs of the future are anticipated to continue the trend towards lower horsepower and smaller size to match smaller cars and have higher cycle temperatures and rotational speeds to achieve maximum engine efficiency and performance.

It is evident that there is little possibility of using oil lubricated bearings particularly at the hot position(s) unless perhaps a more novel overhung compressor-turbine concept is introduced.

Table 6 gives the suggested bearing operating specifications for a hypothetical advanced engine.

In comparing these specifications with the progress reported with the compliant foil air bearing, it is evident that a good base has been established which makes the foil bearing a very strong contender; however, further technology work is needed.

Specific areas where further foil bearing system work is necessary to achieve the targets of optimum Performance, Reliability, and Potential Cost include the following:

- continued effort to increase bearing load capacity
- continued work on high temperature materials and coatings
- bearing design refinements for increased tolerance to thermal gradients and rotor distortions

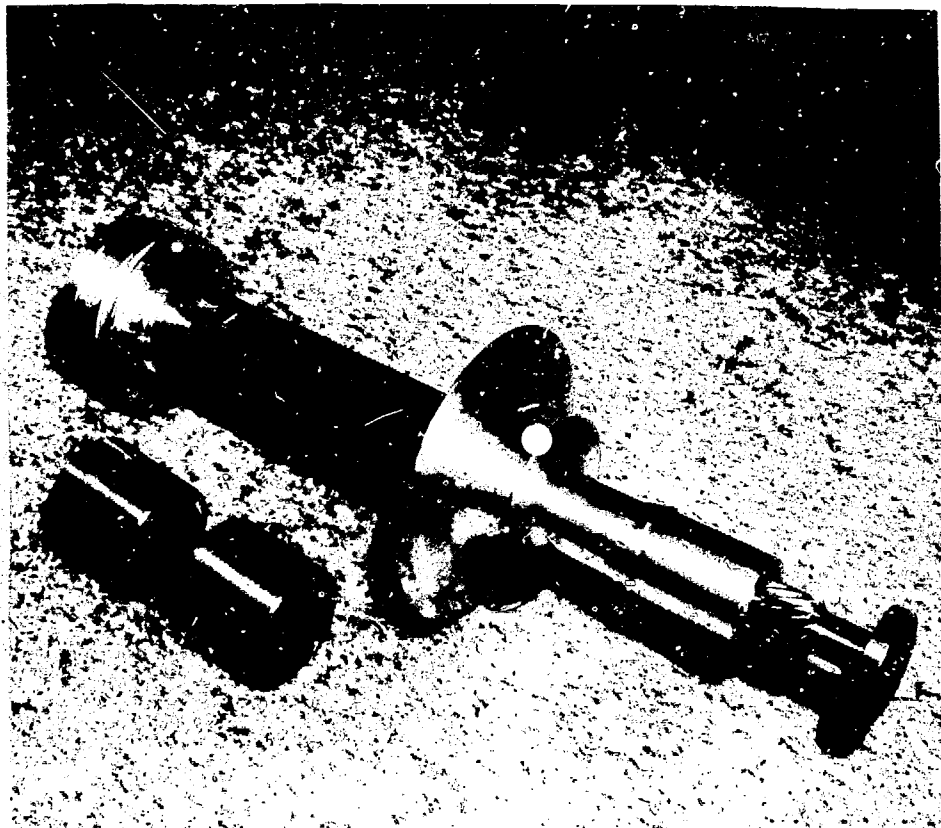


FIG. 18. FORD SINGLE SHAFT GAS TURBINE SIMULATOR
WITH HYDRESIL BEARINGS

TABLE 6

SUGGESTED BEARING OPERATING SPECIFICATION
FOR ADVANCED AUTOMOTIVE GAS TURBINE

PARAMETER		JOURNAL BEARING	THRUST BEARING
SPEED (MAXIMUM)	rpm	80,000 - 100,000	
STEADY-STATE TEMPERATURE	°F	1000 (TURBINE)	550
SOAK-BACK TEMPERATURE	°F	1200 (TURBINE)	600
BEARING SIZES	in.	3/4 to 1-1/2 DIA.	3.0 O.D.
	in.	1.0 to 2.0 L	1.0 I.D.
STEADY-STATE LOADING	psi	15	20
PEAK LOADING	psi	40	40
COOLING AIR TEMPERATURE	°F	500	
TEMPERATURE GRADIENTS	°F/in.	200	100
ROTOR UNBALANCE	oz-in.	.010 at Two Planes	

- simplified, effective, and low power loss seals
- continued cost studies and manufacturing refinements of the bearing system.

ACKNOWLEDGEMENTS

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Session 5

THE DIESEL IS "THE" ENGINE FOR HIGH ANNUAL MILEAGE LIGHT DUTY VEHICLES

by
C.C.J. French*

ABSTRACT

While the percentage market penetration by the diesel engine into the private passenger car field in Europe has been relatively small, its use in taxi service is dominant and the use of diesel engines in the light delivery van is widespread.

For the late 1970's and 1980's, with emphasis on good fuel economy and low exhaust emissions, the diesel engine has unique advantages to offer over all alternatives. Under high annual mileage conditions, the reduction in fuel costs is sufficient to more than offset the increased first cost, and hence there are economic advantages to be coupled to the technical and ecological ones.

This paper supplements previously published data which established the superior fuel economy of the diesel engine by additional data on six cylinder and on turbocharged four cylinder diesel engines of equivalent power output to typical European gasoline engines.

It examines the disadvantages of the diesel engine and suggests that some of these are over-rated and that others are susceptible to treatment.

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THE DIESEL IS "THE" ENGINE FOR HIGH ANNUAL MILEAGE LIGHT DUTY VEHICLES

by
C.C.J. French

The diesel engine has had a long and distinguished career as a power plant. Engines are in use covering the power range from under 5 horsepower to 50 000 horse power - a much wider range than any other type of prime mover. In surface transport almost all railways which are not electrified are now powered by diesel engines, and the World's heavy duty trucks are also diesel powered.

In Europe and Japan diesel engines are used almost exclusively in the medium truck field and have been widely used for light duty trucks and vans, while it is becoming rarer to find a gasoline engined taxi in Europe. Despite these trends, the diesel engine has failed up to now in the United States to penetrate downwards below the heavy truck market apart from the agricultural tractor where the penetration over the past twentyfive years or so has been spectacular.

With World-wide legislation calling for low levels of exhaust emissions however and with requirements for improved fuel consumption, arising as a result either, as in most parts of the World from high fuel prices, or as in the United States from fuel economy legislation, a new look is being given to the gasoline engine and to other alternatives.

In the Author's view, the results of these investigations have shown that

certainly for the next ten years, and probably for longer, the only practical alternative to the gasoline engine and its further developments is the diesel engine. Work on vehicle transmissions to improve economy is desirable and may be applied to most types of engines, and work on gas turbines and on Stirling engines should certainly be encouraged, but breakthroughs are required in the following areas:-

Gas Turbines	-	Efficiency
		First Cost
		Durability
		High Temperature Materials
Stirling Engines	-	First Cost
		Bulk
		Control
		Sealing
		High Temperature/High Pressure Materials

As a result of these problem areas, the Author does not see any significant penetration of the light duty vehicle market by either of these engines before the nineteen nineties at the earliest.

We are left therefore with improvements in gasoline engines or with the diesel engine as a means of improving the fuel economy of light duty vehicles in the nineteen eighties. One must certainly not overlook the possibility of improvements in gasoline engines, and here the most promising area would appear to be the use of high compression ratios, detonation being suppressed by ignition timing retard and by the use of weak mixtures when approaching full throttle operation, or by avoiding full throttle operation entirely.

Such retardation of timing to prevent detonation at current compression ratios is of course common practice, but there is now an intense interest in very high compression ratio operation. Values up to 15:1 with 98 Octane gasoline are currently being investigated, with somewhat lower values for

lower Octane fuels. References (1), (2), (3)*.

While the development of a successful gasoline engine of this type would have an important influence on the future of alternative power plants, its likelihood is affected by a number of problems. Of these, probably the most imponderable is future legislation on lead content of gasoline, although this type of engine may also prove to be very sensitive to deposit build-up in the cylinder leading to severe detonation, preignition, and failure. In the United States we already have a requirement for lead free gasoline, and there is growing pressure in Europe for a reduction in current lead levels. In view of the high refinery energy requirement for the production of high Octane gasoline in the absence of lead containing additives, a reduction in lead levels is likely to lead, as in the United States, to the need to employ gasoline of no higher than 91/92 Octane, and this may well drive down the operating compression ratio, and hence the fuel economy of "high compression" engines which have to use this fuel, to a point at which they are no better than the current gasoline engines. There may also be problems in operating engines of larger cylinder size, since the chambers may then not be sufficiently compact for detonation-free operation.

In this scenario we are left with the diesel engine as the only practical alternative to the gasoline engine for some years to come - a period of ten years at least, and possibly much more. We thus have to ask, "What are the advantages of the diesel engine?", "What are its disadvantages?", and "If its advantages are so compelling, why is it not already in much wider use?".

ADVANTAGES AND DISADVANTAGES OF THE DIESEL ENGINE

So much has already been written about the advantages and disadvantages, and the Author's Company have carried out and published an extensive survey into these for EPA (4), that it is not necessary to do more than summarise them here and to discuss how the disadvantages can be overcome.

* Numbers in brackets designate references at end of paper.

Fuel Economy

The outstandingly good fuel economy of the diesel engine as compared with the gasoline engine is the main reason for employing direct injection diesel engines in buses and trucks. In light duty applications, where an engine is required which can operate over the same wide speed range as the gasoline engine, high speed indirect injection engines are currently universally employed. These engines have a disappointingly high fuel consumption at high speeds, due to high mechanical friction and heat losses, Fig. 1, but as speed is reduced, and especially at low loads, the absence of intake throttling and the thermodynamic advantages of the use of a very weak mixture, result in an increasingly larger gain in fuel economy resulting from the use of the diesel engine, Figure 2.

It has been suggested that considerable improvements in the fuel consumption of current gasoline engines could be made by improvement in combustion and/or fuel injection, by improved ignition, and by alterations to inlet manifold-ing and valving. In this way, gasoline engine fuel economies could approach those of diesel engines. While improvements in all engines are possible and indeed should be strived for, it has yet to be demonstrated that the gains just mentioned are achievable in practice. Considerable improvements have been demonstrated under steady state conditions on the test bed, but the changes in mixture strength which are necessary to achieve acceptable driveability have destroyed the improvements.

It has also been suggested by a number of authorities that the improvements in fuel economy which result from the use of the diesel engine arise largely from the use of a diesel engine which is of a much lower power than that of the corresponding gasoline engine, which is as a result heavily throttled and has a much poorer mechanical efficiency. There have, in the Author's view, never been any good grounds for this view when considering vehicles driving over a reasonable duty cycle which includes urban and city driving as opposed to continuous high speed operation.

With any acceptable power/weight ratio for a light duty vehicle - a minimum of about 50 bhp/ton - such a driving cycle involves large amounts of inefficient low power operation, and this effect is not offset by relatively minor changes in power/weight ratio.

That this is so has been demonstrated by the fact that the prime factor which controls fuel economy when driving a fixed duty cycle is vehicle weight, with power/weight ratio having only a second order effect which is easily offset by differences in gearing.

While this argument has failed to convince everyone in the past, two recently published sets of data should be conclusive to all. A recent Ricardo paper (5) has included data on a Vauxhall Viscount car fitted with gasoline and diesel engines of the same power - but of a rather low power/weight ratio. The results - Table 1 - have confirmed a substantial albeit reduced advantage for the diesel engine.

<u>Conditions</u>	<u>Fuel Consumption</u> Litre/100 km (mile/gal)		
	1.8litre	3.3litre	3.17litre
	<u>Gasoline</u>	<u>Gasoline</u>	<u>Diesel</u>
Steady speed			
30 mile/h 3rd gear	8.9 (31.7)	9.4 (30.0)	- -
30 mile/h top gear	7.7 (36.6)	- -	5.9 (47.6)
40 mile/h	9.0 (31.2)	9.0 (31.2)	6.5 (43.4)
50 mile/h	10.2 (27.7)	9.5 (29.9)	7.6 (37.3)
60 mile/h	10.7 (26.3)	10.7 (26.3)	8.5 (33.1)
US Federal emission cycle	13.0 (21.5)	17.8 (15.9)	11.1 (25.4)
US Federal durability circuit	9.5 (29.7)	11.6 (24.4)	7.6 (37.3)
Round trip (300 miles)	12.05 (23.4)	13.5 (20.9)	8.5 (33.1)

Table 1 - Comparisor of Fuel Consumption

The other data concerns the diesel version of the VW Golf car (6). Here the 1.1 litre gasoline and the 1.5 litre diesel engines have almost identical

performance, and the fuel economy advantages of the diesel engine are in line with all previous data with a gain of 25%-40% in fuel economy, depending on the duty cycle.

Of these improvements, some eleven percent arises from the higher volumetric calorific value of the fuel, but the remainder arises from improved cycle efficiency.

Currently Legislated Exhaust Emissions

The diesel engine, especially in its indirect injection form, has inherently low levels of gaseous emissions which may be compared with those from emission controlled gasoline engines - Table 2.

		Anticipated Federal Limit for 1980	Diesel 2 litre	Gasoline	
				1.4litre*	2 litre**
Unburnt hydrocarbons	HC	0.41	0.4	2.8	0.17
Carbon monoxide	CO	3.4	1.5	2.9	2.2
Oxides of nitrogen	NOx	2.0	1.5	2.8	1.7

Table 2 - Exhaust Emissions g/mile - C.V.S. Cycle

* Standard engine with controlled carburation, ignition, and crankcase ventilation.

** Engine fitted with exhaust gas recirculation, oxidising catalyst box and air pump.

Hydrocarbon levels are very low and although it may be necessary to take additional steps to ensure production compliance, the legislation limit of 0.4 g/mile should be achievable with a 3500 lb car and a similar target for a 4500 lb car would appear to be attainable, perhaps with hang-on devices.

Under any conditions where the exhaust smoke would be acceptable, the CO levels are very low and well within any anticipated legislative limits.

The problem area is of course NOx. While the current levels for an uncontrolled exhaust emissions car fitted with an indirect injection diesel engine are somewhere under 2 g/mile for a vehicle weight of 3500 lb, injection timing control programmed for load and speed should give production compliance at a legislative limit of 1.5 g/mile, and 1.0 g would appear to be a reasonable target for the future with some exhaust gas recirculation.

A future target of 0.4 g/mile would however appear to be unattainable without a technical breakthrough. The problem, which is similar to that faced by fuel injection, stratified charge gasoline engines, arises from the presence of heterogeneous combustion and from the use of lean mixtures. With heterogeneous combustion, combustion takes place in the fuel/air vapour around the drops of fuel and tends to stoichiometric and hence to high flame temperatures under all load conditions. In addition, as a result of the need to run always with overall lean mixtures, there are very low levels of CO and HC in the exhaust, which makes it impossible to employ "three-way" catalysts to reduce the NOx to N₂ while oxidising HC and CO.

While research and development work aimed at the achievement of 0.4 g/mile is continuing, it would be unrealistic to base the future of the diesel engine on the gamble that this will succeed, and the demonstrated fuel advantages of the diesel engine are such that in the Author's opinion the diesel engine should not be debarred by unrealistic NOx emission legislation limits unless medical evidence is available to back these limits.

Other Exhaust Emissions

Distillate fuel as used in the engines currently discussed has a sulphur content of somewhere in the region of 0.5%. Almost all of this is burnt during combustion and appears in the exhaust as SO₂, although there may also be small amounts of SO₃. If it becomes necessary to reduce these levels, it would be necessary to reduce the sulphur level in the fuel, which requires to be done in the refining process. Such a process is possible, but would result

in an increase in fuel costs, and with the very large quantities involved the disposal of the resulting sulphur might pose some logistic problems.

All diesel engines emit much larger quantities of particulate material than do gasoline engines. Typical figures for such emissions are shown in Figure 3 and occur even when there is no visible smoke in the vehicle exhaust. It can be seen from this data that the particulate materials in diesel exhaust exceed those from equivalent gasoline engines by an order of magnitude or even more, and there is some evidence that these levels may be further increased by the use of exhaust gas recirculation. While the reduction of these levels would be a serious problem if any future legislation set a requirement comparable to that of current gasoline engines, recent work has shown that some reduction is possible by the use of exhaust treatment devices. Relatively simple filter/afterburners can give a reduction of particulate levels by 50%, and with development further improvement might well be possible.

The odour of the exhaust from light duty diesel engines is different from that of gasoline engines. It should not however be confused with that which was experienced with grossly over-fuelled truck diesel engines at a time when there was no legislation to prevent the emissions of smoke from these vehicles.

Experience indicates that the steps which are taken to reduce HC levels also reduce exhaust odour, and that the afterburning devices mentioned in the previous paragraph also have a dramatic effect in odour reduction. While it is necessary to ensure freedom from fuel leaks on the vehicle so that the driver and passenger do not smell liquid fuel, an emission controlled diesel engine car should not cause annoyance from exhaust odour, provided that its fuelling quantities are properly set and the engine is well maintained. In this regard it is no more demanding than its gasoline engine counterpart. That large numbers of diesel engine taxis and other vehicles are employed in European cities without complaint is an indication that, with still further improvement, the odour levels in city streets with increased numbers of diesel engine

vehicles should be acceptable.

Engine Weight

Unless turbocharging is employed, the diesel engine is appreciably heavier for the same power output than the gasoline engine. This increased weight arises not from any increase in mechanical loadings since the peak cycle pressures are similar, but from the inability of the diesel engine to burn more than 70%-75% of the air charge. A diesel engine of about the same power has therefore to be about 30% larger in swept volume with a corresponding increase in weight.

This increase in bulk and weight may not however be so large when compared with United States automobile gasoline engines, since in many cases these are not rated up to their potential, and it is possible for a diesel engine of the same swept volume to be rated at the same power as the gasoline engine.

Turbocharging offers considerable advantages for weight reduction. To enable an acceptable torque curve shape to be produced, it is essential for the turbocharger to be matched to the engine at as low an engine speed as possible. This would result in excessive boost pressure and hence excessive mechanical loading of the engine at speeds in excess of this, and a wastegate is therefore employed to "spill" exhaust gas from before the turbine to maintain a constant boost pressure. Small turbochargers are now becoming available, suitable for engines of down to about 1.5 litres swept volume, and the turbocharging of such small diesel engines can be considered.

Problems arise from turbocharging a gasoline engine due to the need either to use a higher Octane fuel or to reduce the compression ratio of the engine to prevent detonation. This can give a penalty in fuel economy at part load. In the case of a diesel engine however, combustion imposes no such limits, and it is in fact essential to maintain the same compression ratio as

is employed on the naturally aspirated engine in order to give good cold starting and acceptable noise levels at idle and low load, and to prevent white smoke and excessive odour in the exhaust following a cold start.

Typical performance curves from a turbocharged 2.1 litre Opel engine are given in Fig. 4 and Tables 3 and 4, and show that when the vehicle is driven over the United States Emissions cycles, the turbocharged engine gives little if any change in either exhaust emission levels or in fuel economy.

<u>Conditions</u>	<u>Fuel Consumption</u>			
	Litres/100 km (mile/gal)			
	<u>N/A</u>		<u>T/C</u>	
Steady speed				
30 mile/h	5.1	(55.0)	5.6	(50.2)
40 mile/h	5.5	(51.6)	6.0	(47.4)
50 mile/h	6.5	(43.8)	7.4	(38.4)
60 mile/h	7.4	(38.0)	8.7	(32.6)
US Federal Emissions cycle	9.0	(31.4)	9.2	(30.6)
US Federal Durability circuit	6.73	(42.0)	6.87	(41.1)
Round trip (300 miles)	9.2	(30.6)	9.7	(29.1)

Table 3 - 2.1 Litre Opel Engine Fuel Consumption

<u>Conditions</u>	<u>g/mile</u>		
	<u>HC</u>	<u>CO</u>	<u>NOx</u>
Naturally aspirated	0.57	1.25	1.45
Turbocharged	0.21	1.20	1.21

Table 4 - Opel Engine Naturally Aspirated & Turbocharged

While the torque curve shape is acceptable, there is of course no boost pressure available at cranking and idling speeds, and in fact the compressor acts as a "throttle" at very low speeds. If clutch engagement occurs at these

speeds, the torque could be inadequate, and with low levels of power/weight ratio, hill starting performance of the vehicle would be unsatisfactory. This would not of course be a problem with an automatic transmission.

It is desirable to have a boost responsive maximum fuel stop to prevent overfuelling and black smoke. Theoretically, acceleration times are affected by the inertia effect of the turbocharger which requires time to accelerate when a higher boost pressure is called for. In practice however, quite extensive driving of several different turbocharged diesel engined cars has not shown this to be a problem.

First Cost

The diesel engine is bound to be more expensive than the gasoline engine. There are two reasons for this. Firstly, as already explained, the diesel engine is larger and heavier, and cost is a function of weight. If turbocharging is used as an alternative, the turbocharger is an expensive piece of precision machinery and the turbine itself is made of expensive material. To a first order, the cost of the turbocharger in these sizes is approximately equal to the extra cost of the equivalent larger engine.

The second reason for increased cost is the fuel injection equipment which is also an expensive piece of precision machinery and which costs appreciably more than an emissions controlled carburettor with a set of ignition equipment. While it is desirable to reduce this cost, for purposes of emissions control the fuel injection equipment is tending to further complication with a requirement for timing variation with load and speed, and it is perhaps unrealistic to hope for any appreciable cost reduction.

Vehicle Operating Economics

As a result of the increased first cost which could be of the order of 50% more than the cost of the gasoline engine, it is necessary to take advantage of the improvement in fuel economy, and hence in vehicle fuel costs, in order to

make an economic case for the adoption of the diesel engine. Fuel cost is therefore vital, and it is harder to make an economic case in any country such as the United States whose fiscal policies result in very low unit costs of fuel. On the other hand, American cars tend to be heavier and to use automatic transmissions and air conditioning which results in much increased fuel consumption, and this helps to reduce the balance.

Under all conditions today, the break-even mileage beyond which there is a saving in total cost due to the use of the diesel engine is somewhere between 20 000 and 30 000 miles; Figure 5. Under stop-start taxi service conditions with high unit cost fuel, the break-even point is even lower, which together with high annual mileages accounts for the widespread use of diesel taxis in Europe.

Cold Starting

The cold starting abilities of the diesel engine are often criticised by non-diesel users. With high heat losses, the indirect injection engine is not a very good starter under extremely cold conditions, and the problem is exaggerated in the small engine by a worsening surface to volume ratio. With heater plugs however in the combustion chamber, which are essential for such engines, good starting can be achieved at down to about -20°C , and lower with ether-containing starting fluids.

The only problem is the 20-30 seconds required for warming up the plugs before starter engagement, but work is in hand which should lead to the use of quick warm-up plugs which might be used without preheat and which, if they were left engaged for a short time after starting has been achieved, should eliminate white smoke upon cold starting, which results from the passage of unburnt fuel through the engine.

Noise

While diesel engines are commonly thought of as noisy, in fact the average

drive-by noise of diesel engined motor cars is only about 2 dBA higher than that of average gasoline engined vehicles. While this will require a greater effort to reach future noise legislation limits, the driver is not in fact conscious of the fact that a diesel engine is fitted, apart from when the engine is idling.

While the idling noise is subjectively loud, its decibel level is low but cannot of course be compared with that of the throttled gasoline engine which is often undetectable. Major modifications to the engine structure and/or a complete enclosure would appear to be the only way to obtain the required large reduction in noise when idling, but penalties in first cost and/or maintenance accessibility might well result.

CURRENT USE OF LIGHT DUTY DIESEL ENGINE

The diesel engined passenger car has a much longer history than is generally known, dating from at least as far back as Citroen and Mercedes in the 1930's. Over the years, Mercedes-Benz has been the strongest protagonist, followed fairly closely by Peugeot. Today, however, there are many others either in production or about to be produced, as shown in Table 5.

Engines are also widely used in light duty trucks and utility vehicles in Europe and Japan, and the annual production is certainly in excess of 300 000 and probably of 400 000. With these production numbers, which have been growing rapidly over the past few years, it is clear that there is a wide experience of their use in most parts of the World and that this experience, together with the excellent reputation of the diesel engine in terms of long life and reliability, is leading to increased market penetration, and that at times such as now, with increasing relative fuel costs, this will result in a much wider use of the diesel engine in high annual mileage vehicles.

FURTHER DEVELOPMENT

While the fuel economy advantages of the diesel engine are thus well-established, it is clear that there will be further improvements in the fuel economy of the gasoline engine. The use of higher compression ratios has already been discussed, but other improvements may also be possible. One can ask therefore - "What is the scope for corresponding improvements in the diesel engine?".

A number of areas are currently being studied, and each would appear to offer scope for improvement:-

Engine Friction

As has already been explained, the losses due to mechanical friction, as shown in Figure 6, are excessive (7). By minimising the size of engine auxiliaries this may be reduced, but it is also possible that it may prove feasible to reduce piston and ring friction, which is the largest single item.

Heat Losses

Heat losses during compression, combustion, and the expansion stroke, are a drain on the thermal efficiency of the cycle, and strenuous efforts are being made to reduce them either by reducing gas velocities or by the use of insulation. The reduction of gas velocities is possible but tends to a less flexible engine and hence to difficulties in covering a wide speed range, as described in the next section. Reduction in heat losses would of course give a further improvement, arising from the reduction in the power of the cooling fan even as in the case of insulating the exhaust duct where there was no change in cycle efficiency.

Pumping Losses

These take two forms; Figure 6. Firstly is the charging of the cylinder and the exhausting of the spent charge. With valve sizes at their limit and

with little to be gained in this size of engine from the use of four valves per cylinder, any possible gains in this area seem to be small. In the indirect injection engine however there are also pumping losses which result from pumping air into the pre or swirl chamber, depending on type. While gains are possible by changes in the relative volume of or by an increase in the throat area, this results in a distortion of the torque curve shape which is undesirable.

Reduction in Compression Ratio

The compression ratio is set in light duty diesel engines by considerations of acceptable cold starting, acceptable noise when starting and at low load, and by the absence of misfiring when starting and at high speed and light load, both of which could result in unacceptable hydrocarbon levels and exhaust odour. Potential improvements in these areas, arising perhaps from ignition aids, could be taken, or alternatively the compression ratio which is currently excessive from the point of view of fuel economy, could be lowered.

The Use of Direct Injection

Direct injection is used almost exclusively in almost all types of diesel engine, apart from very small or from high speed, light duty engines. It offers better starting, lower heat losses, lower thermal loading, and a better fuel economy with an advantage of 8-10%. Clearly, it would also be used in light duty engines if it were at all possible, and the engine performance data of Fig. 7 for such an engine, obtained by Ricardo some ten years ago, are of interest. The reasons why it is not used are many, but they all arise from the need to operate the engine over a wide speed range, both in order to match the transmission of a conventionally engined vehicle and to obtain an adequate engine power output without excessive engine size.

Direct injection engines employ a number of holes in their fuel injectors since the fuel has to be "taken to the air", and this leads to a fixed hole size. When a wide speed range has to be covered, with the fuel line pressure varying

as the square of the fuel velocity and hence of the engine speed, pressures will either be excessive at high speeds or too low at low speeds, depending on the orifice size chosen. As a result there is considerable difficulty in getting a good engine performance over a wide speed range.

With indirect injection however, the air is "taken to the fuel", and a single hole nozzle with a lifting obturator is employed which gives an ability to have a smaller orifice area at lower speeds, and hence good combustion is easily achieved over a wide speed range.

Direct injection engines also tend to be noisy, especially at high speeds, and this difference in noise can be as great as 5 dBA; they have higher maximum cylinder pressures, and the nozzle holes are small and hence in small cylinder sizes more prone to blocking and requiring more frequent maintenance.

In general the NOx emissions from direct injection engines tend to be higher than with indirect injection, as do the hydrocarbon levels, especially in small cylinder sizes. Some recently published results by AVL (8) with direct injection have however shown NOx levels approaching those of current vehicles with indirect injection. No evidence of noise level acceptability or of injection system life is currently available however.

Clearly, with so much to be gained, a number of companies are investigating the use of direct injection, employing various alternative systems, and success with one or another of these would be likely to lead to use in service.

CONCLUSIONS

Diesel engines are widely used in light duty service throughout the World where they have proved acceptable to their operators.

They offer a fuel consumption advantage of 25% to 30% as compared with

gasoline engines in combined urban/city cycles and a greater advantage in start-stop taxi service.

The cost differential is inherent, but with current fuel costs it is offset in the first 20 000 to 30 000 miles (less in start-stop taxi service), and beyond this there is an economic gain to the operator arising from the use of the diesel engine.

For a 3500 lb inertia class vehicle a NOx limit of 1.5 g/mile should be achievable with programming of the injection timing, and 1.0 g/mile with the use of some degree of exhaust gas recirculation. Heavier vehicles would give roughly pro rata results.

The imposition of tighter NOx levels than this would effectively bar the diesel engined light duty vehicle from the market place.

At the required hydrocarbon levels, exhaust odour is not likely to be a serious problem, but it should be further reduced by exhaust treatment devices which may have to be developed to meet possible future particulate legislation.

Engine noise poses problems which are only marginally more severe than with gasoline engines but subjective noise at idling is intrusive and difficult to reduce.

Other Alternative Power Plants are too far away in the time scale to affect the introduction of the diesel engine. The most likely contender is the high compression gasoline engine which may however prove to have problems of endurance, under deposit build-up conditions, and which may be no more efficient than current gasoline engines when operating, as may be essential, with low or lead free gasoline.

The Author's overall proposition is that "The Diesel is 'THE' engine for high annual mileage light duty vehicles" - proven, widely used, dependable, efficient, and giving a long life with low maintenance costs.

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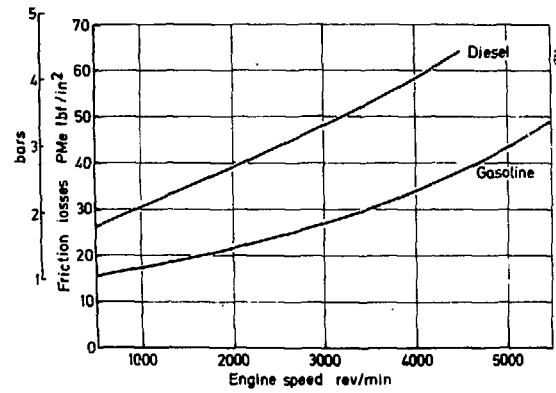
Manufacturer	Capacity	No. of Cylinders	Bore & Stroke mm	Max. Speed rpm	Power Output kW	Weight kg	kg/kW	kW/litre
<u>British Leyland</u>	1.5	4	73 x 89	4000	30	185	6.2	20.0
	1.8	4	80.2 x 88.9	4250	33.8	174	4.5	21.5
	2.5	4	88.9 x 101.6	3500	47.7	268	5.62	18.9
<u>Citroen</u>	2.175	4	90 x 85.5	4500	45.9	206	4.49	21.0
<u>Nissan</u>	2.164	4	83 x 100	4000	48.4	186	3.78	22.4
<u>Mercedes-Benz</u>	1.968	4	87 x 83.6	4200	40.5	193	4.77	20.6
	2.192	4	87 x 92.4	4200	44.1	195	4.42	20.0
	2.404	4	91 x 92.4	4200	48.4	203	4.19	20.1
	3.0	5	91 x 92.4	4000	58.8	234	3.98	18.9
<u>Opel</u>	2.1	4	88 x 85	4400	44.1	199	4.51	21.0
<u>Peugeot</u>	1.357	4	78 x 71	5000	33.5	152	4.54	24.7
	1.948	4	88 x 80	4500	44.7	170	3.8	23.0
	2.112	4	90 x 83	4500	48.4	185	3.82	23.0
<u>VW</u>	1.5	4	76.5 x 80	5000	37	130	3.51	24.7
<u>Rover/MSA</u>	2.28	4	90.6 x 88.9	4000	46.2	243	5.26	20.0
	3.43	6	90.6 x 88.9	4000	65	270	4.15	19.0
<u>Ford</u>	2.34	4	93.7 x 85.6	3600	46	234	5.1	19.5
	3.54	6	93.7 x 85.6	3000	59	325	5.5	16.7
<u>Fiat</u>	2.45	4	93 x 90	4200	51.4	215	4.18	21.0
	3.68	6	93 x 90	4200	77.2	275	3.56	21.0

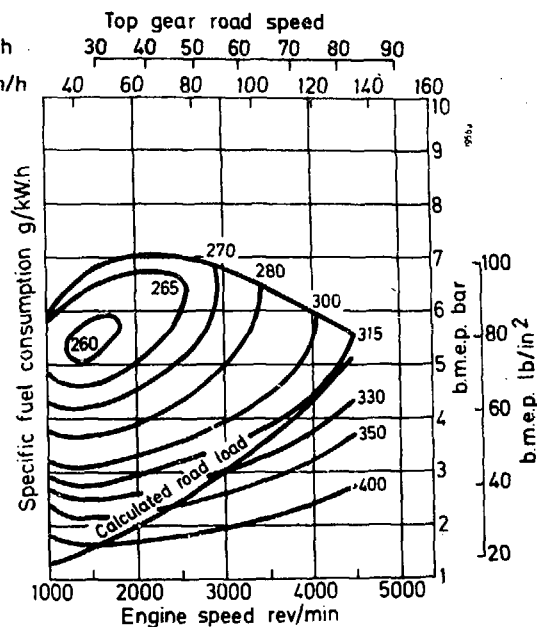
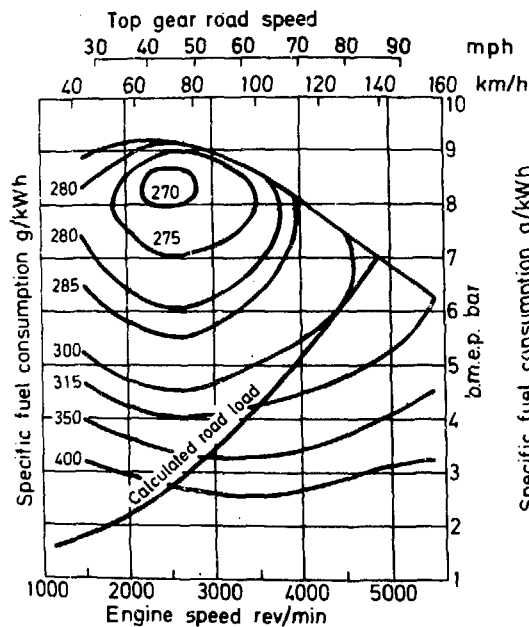
Table 5 - Current Light Duty Diesel Engines

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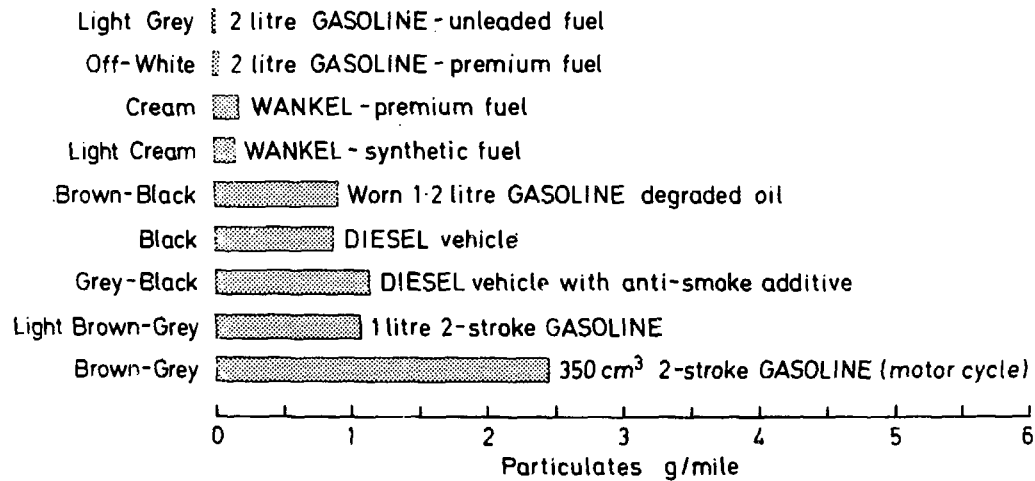
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- Fig.2 Fuel Consumption Maps of (a) 1.618 litre gasoline and (b) 2.11 litre Diesel Engines with calculated Road Load.
- Fig.3 Automobile Particulate Emissions.
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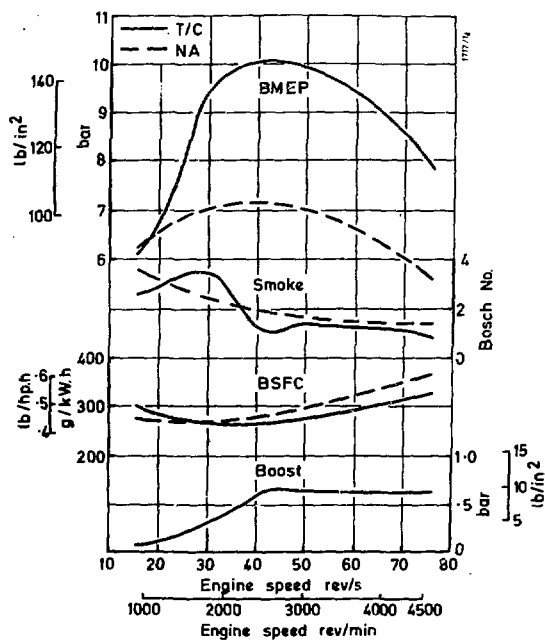
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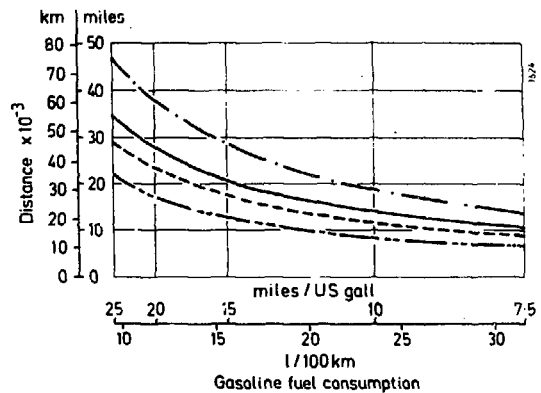


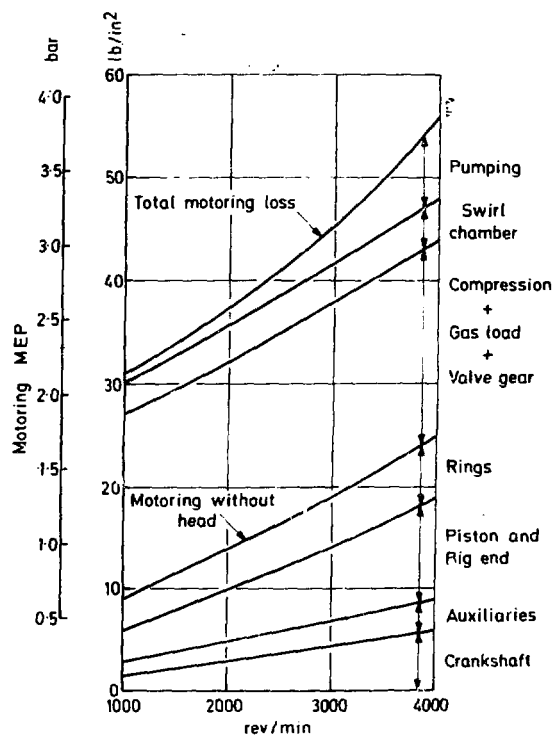
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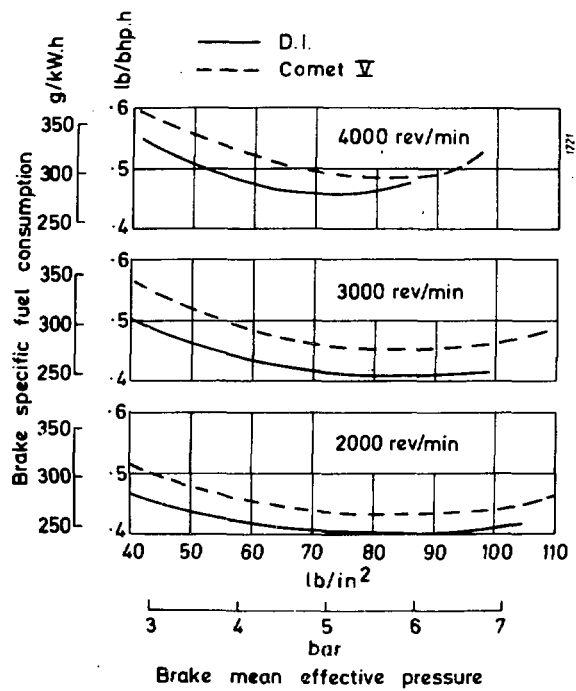




Diesel fuel economy ratio	
60%	75%
—	—
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\$400 diesel cost difference	
\$250 diesel cost difference	







GASEOUS EMISSIONS CONTROL FOR HEAVY DUTY DIESEL ENGINES :
WORK DONE BY FIAT RESEARCH CENTER

Paper for Presentation at the
NATO/CCMS Fourth International Symposium on Automotive Propulsion Systems
to be held in Washington D.C. - April 17-22, 1977

by

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GASEOUS EMISSIONS CONTROL FOR HEAVY DUTY DIESEL ENGINES :
WORK DONE BY FIAT RESEARCH CENTER
TO MEET THE PROPOSED LIMIT OF 5 GR/HP.h FOR NO₂ + HC

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I N T R O D U C T I O N

The adoption of very stringent levels for heavy duty Diesel engine emissions, particularly the 5 gr/HP.h of NO₂ + HC of the California legislation for 1977, required a great research effort to find out methods suitable for exhaust contaminant abatement.

The research in this field by Fiat Research Center started in 1971. The first part of the study was released at the 1975 SAE Off-Highway Vehicle Meeting (1).

Results of a number of test methods were presented including such individual variables as : -

- Injection timing retard.
- Exhaust gas recirculation (E.G.R.).
- Exhaust gas treatment by catalytic converter.
- Fast injection rate.
- Fast injection rate plus injection timing controlled by both engine speed and load.
- Fast injection rate plus modulated E.G.R.

Though nothing final can be said about the reliability of these methods, the last two have met the proposed 5 gr/HP.h NO₂ + HC limit but at low running hours.

This paper reports the second part results of the Fiat Research Center (F.R.C.) study on heavy duty Diesel engine emissions.

In the time elapsed between the start and the end of the research, California A.R.B. allowed a choice between two targets for HC and NO₂.

One choice is HC = 1 g/HP.h, NO₂ = 7.5 g/HP.h; the other is the previously stated one : HC + NO₂ = 5 g/HP.h. F.R.C. selected as target for the research what was felt to be the more stringent : 5 g/HP.h NO₂ + HC emission level.

The emission control methods reported here are : -

- Exhaust gas recirculation : a deeper investigation.
- Exhaust gas recirculation plus supercharging to the original power level.
- Variable compression ratio.
- Dual fuel (gasoil and methanol).

The work was performed on different direct injection engines, both naturally aspirated and turbocharged, whose characteristics are given on Table 1. Test methods used are in accordance with SAE Standards (2, 3, 4).

The HCs were measured with an F.I.D. having a sampling line heated to 170° C. For the CO and NO measurements an N.D.I.R. analyzer was used and the smoke level was measured with a Hartridge Mk 3 smoke meter or a Bosch smoke meter (using a photoelectric cell). The test procedure is in accordance with the U.S.A. gaseous emissions control legislation (5).

Specifications of the fuel used for the tests are shown in Table 2.

EXHAUST GAS RECIRCULATION

As is well known the air charge dilution with an inert gas (N_2 , CO_2 , H_2O vapour, etc.) greatly reduces NO_x emission in both spark and compression ignition engines. Exhaust gas recirculation (E.G.R.) is the easier technique to obtain intake air charge dilution; this technique, in fact, is fully applied on spark ignition engine emission control systems.

To make a detailed investigation of the possibilities offered by E.G.R. the Fiat 8060 engine was selected (see Table 1), since it could be expected that this standard engine having low HC + NO_2 emissions, would meet the target with E.G.R. alone. In fact the 8060 engine, with carefully tuned injection system and, in particular, with reduced nozzle sac volume shows the following emission levels (13-mode California cycle) : -

CO	=	5.6	g/HP.h
NO_2	=	7.4	"
HC	=	.8	"
HC + NO_2	=	8.2	"

During the tests, the intake charge (air + recirculated exhaust gas) was kept constantly at $40 \pm 45^\circ C$ to separate the temperature effect from the charge dilution effect. The emission and performance results are expressed versus E.G.R. weight fraction, i.e. the recirculated exhaust gas mass flow divided by intake charge (air + exhaust gas) mass flow :

$$Q = \frac{G_r}{G_a + G_r}$$

Q = E.G.R. weight fraction

G_r = recirculated exhaust gas mass flow (kg/h)

G_a = air mass flow (kg/h)

Obviously : % E.G.R. = $Q \cdot 100$

The tests were performed increasing E.G.R. at two different engine speeds : intermediate (1900 R.P.M.) and rated (3200 R.P.M.). At each speed, four constant loads - 2%, 25%, 50% and 75% of the maximum load without E.G.R. - were selected. Other tests were performed at full load versus E.G.R. with the smoke kept constant at the original level.

The purpose of the research work was to : -

- Evaluate NO_x reduction achievable as a function of % E.G.R. and the influence of E.G.R. over CO, HCs, smoke, fuel consumption and performance.
- Establish by balancing positive and negative effects, the best % EGR for the 13 modes of the California cycle.
- Compare the E.G.R. approach with other NO_x control techniques.

Influence of E.G.R. on emissions and performance

The results of the tests at constant loads of 25% and 75% of maximum load are shown in Figs. 1 and 2. The results are roughly the same for the other loads.

By increasing % E.G.R., at every speed and load, it is possible to identify the following trend : -

- Strong NO_x reduction.
- Notable increase of HC, particularly at low loads.
- Notable increase of smoke, particularly at high loads.
- Small increase of specific fuel consumption.

The NO_x reduction, at constant % EGR, is influenced mainly by the load. At both speeds, with E.G.R. = 15% the percent reductions of NO_x are 60% at 75% of maximum load, 42 ÷ 43% at 25% of maximum load (see Figs. 1 and 2). It is possible to give an explanation of this phenomenon considering that as the load is decreased the air to fuel ratio increases.

As a consequence, at the same % E.G.R., the inert gas fraction, responsible for the NO_x reduction, is smaller at low load. A mathematical model was developed to work out the composition of the exhaust gas and of the intake charge (see appendix A). This way, it is possible to correlate NO_x reduction not only against % E.G.R. but also against % O_2 in the intake charge (the % O_2 is, in some way, inversely proportional to the inert gas fraction in the intake charge).

As shown in Figs. A3 and A4, if NO_x reduction is plotted against % O_2 the correlation is better.

The trade-off between HCs and % E.G.R. is influenced by speed and load : in particular, the HCs increase, as % E.G.R. increases is stronger at higher speeds (see Figs. 1 and 2).

Because of the excess air available smoke and CO are not influenced by E.G.R. at low loads, but they are strongly influenced at higher loads. This fact is clearly shown in Fig. 3 where smoke is plotted against air-to-fuel ratio at 25% and 75% of maximum load, varying % E.G.R. over a wide range. The replacement of air with exhaust gas has no influence on smoke at lower loads (excess air still high), but has strong influence at 75% of maximum load (approaching stoichiometric air-to-fuel ratio). The same considerations may be made for CO as well as smoke emissions.

Maximum available power in Diesel engine is smoke-limited, as known.

The smoke constraint in presence of E.G.R. at full load involves serious power drops as shown in Fig. 4. In this figure are indicated the results of the tests performed at full load and constant smoke level versus % E.G.R. The % NO_x reduction obtained with % E.G.R. at par is of the same magnitude for both speeds; for example, with $Q = 0.10$ it is 55%. However, the negative effects induced by E.G.R. are worse at higher than at lower speeds as can be seen in Fig. 4. In particular, with $Q = .10$ the power drop is 10% at the intermediate and 20% at the rated speed; the fuel consumption is not influenced by E.G.R. in the first case whereas it increases sharply in the second.

Study of the optimum % EGR for the 13-modes of the California cycle

As stated before the aim is to meet the $\text{NO}_2 + \text{HC}$ 5 g/HP.h standard for '77.

The % E.G.R. for every mode must then be chosen with the purpose of minimizing $\text{NO}_2 + \text{HC}$.

The specific mass emissions of $\text{HC} + \text{NO}_2$ are plotted in Figs. 4, 5 and 6 against E.G.R. % for all the modes of the 13-mode cycle. There is no mention of the idle condition since E.G.R. at idle is not useful from the point of view of $\text{NO}_x + \text{HC}$ emissions as found in a previous study (6).

As pointed out for NO_2 , with regard to $\text{NO}_2 + \text{HC}$ reduction the usefulness of E.G.R. increases with the load (Fig. 5 and Fig. 6).

Moreover due to the fact that as E.G.R. is increased NO_2 decreases but HC increases, at the rated speed and low load, the minimum $\text{NO}_2 + \text{HC}$ emission is achieved with E.G.R. = 20% (see Fig. 6).

With regard to CO, this is influenced by E.G.R., see Figs. 1, 2 and 4, but the CO emission of the standard 8060 engine (5.6 g/HP.h) is well below the 25 g/HP.h limit. Therefore, some increase in CO emission, induced by the E.G.R. technique, can still be accepted and yet meet the '77 California limits. Then, the choice of the optimum % E.G.R. can be independent of CO emission.

The smoke level is, on the contrary, a constraint to the maximum $\text{NO}_2 + \text{HC}$ reduction achievable by E.G.R. in two modes : -

- At rated speed, 75% of max. BMEP, % E.G.R. has to be $\leq 12\%$ (see Fig. 2).
- At intermediate speed, 75% of max. BMEP, % E.G.R. has to be $\leq 28\%$ (see Fig. 2).

Finally, as shown previously, E.G.R. at full load and constant smoke level brings forward power reductions proportional to the pre-set % E.G.R. (see Fig. 4).

13-mode California cycle results with modulated E.G.R.

The above considerations allow the definition of the % E.G.R. as a function of BMEP with the aim of minimizing HC + NO₂ emissions without worsening the smoke level. This function is plotted in Fig. 7 for the two speeds of the cycle.

The smoke level constraint leads to an unavoidable choice : -

- if the same power curve against R.P.M. of the standard engine is desired E.G.R. has to be cut off at full load.
- If the maximum reduction of NO₂ + HC is desired, it will be necessary to have E.G.R. also at full load, but in this case a power drop proportional to the % E.G.R. must be accepted (see Fig. 4).

With this in mind, six California cycles were worked out by calculation with the following assumptions : -

Cycle 1 : modulated E.G.R. for maximum NO₂ + HC reduction at every mode : E.G.R. cut-off at full load.

Cycles 2, 3, 4, 5, 6 : modulated E.G.R. for maximum NO₂ + HC reduction at every mode; E.G.R. at full load (constant smoke level) and power drops of 5%, 10%, 15%, 20%, 25% respectively.

Emission results, performance and % E.G.R. fixed at full power are plotted in Table 3 : the possibility of meeting the 5 g/HP.h NO₂ + HC standard using E.G.R. is evident.

As % E.G.R. increases at full load, the HC + NO₂ emissions remain practically constant at 3.8 + 3.9 g/HP.h up to a given extent (cycles 3, 4, 5); at higher % E.G.R., NO₂ + HC shows a tendency to increase (cycle 6).

The results of cycles 1 and 3 are worthy of note : cycle 1 shows that it is possible to achieve a 45% HC + NO₂ reduction with regard to the emissions of the standard engine (without power loss or smoke and fuel consumption increase at full load) while cycle 3 shows that the maximum obtainable advantage with the

E.G.R. technique applied to the 8060 Fiat engine is an emission of about 3.8 g/HP.h of NO_2 + HC over the 13-mode cycle. In fact higher % E.G.R.s, whether at intermediate or at full load, bring forward higher NO_2 + HC emissions and further power losses.

Comparison of E.G.R. with injection retard technique for NO_2 control

As known, injection timing retard is a useful technique for achieving NO_x emissions reduction. On the 8060 engine this technique gave the best results with 9 crankshaft degrees retard. Such results are shown in Table 4 (13-mode California cycle) and are compared with those obtained by the fully modulated E.G.R. in the assumption of obtaining the same power curve with both techniques. Table 4 shows that with E.G.R. it is possible to achieve a further HC + NO_2 reduction of 30% over the injection timing retard technique. Carbon monoxide emissions are the same, fuel consumption is better with E.G.R., particularly at rated speed (see Table 4 and Fig. 8). On the other hand, E.G.R. shows substantial smoke level increases at part loads (see Fig. 8).

EXHAUST GAS RECIRCULATION AND SUPERCHARGING

The exhaust gas recirculation technique offers the possibility of obtaining a remarkable reduction of NO emissions without negative effects on smoke and efficiency provided the air-to-fuel ratio (α) is greater than 20+22 (see Fig. 3).

The 8060 engine test results, reported in the previous paragraph, show that it was necessary, for maximum NO₂ + HC reduction to modulate E.G.R. at all operating conditions. In particular, it is necessary to fix a % E.G.R. of 25-30% at full load, with remarkable power losses (Fig. 4 and Table 3).

Consequently, if it is desired to take full advantage of the E.G.R. method with no adverse effects on smoke and efficiency, a drop in specific power of 25+35% has to be accepted. E.G.R. plus supercharging then looks particularly interesting : operation at high air-to-fuel ratio should assure an effective control of smoke and should allow the use of higher % E.G.R. at full load without power losses. It was decided to investigate this on the Fiat 8210 engine whose characteristics are shown in Table 1. This engine has higher NO₂ + HC base line emissions than the 8060. Therefore, more sophisticated methods have to be used in trying to meet the 5 g/HP.h NO₂ + HC California limit. The baseline exhaust emissions of the 8210 engine according to the 13-mode cycle are :

CO	=	5.4	g/HP.h
NO ₂	=	12.6	"
HC	=	.9	
NO ₂ + HC	=	13.5	

To study the effect of supercharging + E.G.R. an independent compressed air supply was used. Therefore, the easier and cheaper way to perform the tests at very high % E.G.R. was to increase the absolute back pressure well over the absolute intake pressure.

For this reason, some tests were made to study the influence of the back pressure on exhaust emissions, performance and smoke. These tests were performed at both intermediate and rated speeds, at full load and with an absolute intake pressure as adopted for the following test with E.G.R. The back pressure was increased, by means of a throttle, over the barometric pressure up to the absolute pressure necessary to obtain a % E.G.R. of 30-35%. The results shown in Figs. 9 and 10 bring to light the following points :

- The influence of the back pressure on HC + NO₂ emissions is very small.
- Smoke and CO increase considerably as back pressure increases.
- The back pressure increase from a value as that obtainable with a turbocharger up to the value adopted for these E.G.R. tests makes specific fuel consumption worse : 10% higher at the rated speed and 6% higher at the intermediate speed (Fig. 10).

In conclusion this test procedure - E.G.R. obtained by increasing the back pressure - is more severe than recirculation at the intake of the compressor when using a turbocharger. In fact, results are worse from the view point of efficiency, smoke and CO but practically the same for NO₂ + HC.

The aim of the research was to bring to light the influence of % E.G.R. on emissions, fuel consumption and smoke at different intake pressure levels and different injection timing.

The work was performed over a wide range of speeds and loads, including those of the 13-mode cycle, the purpose being to work out, for the more interesting solutions, the emission results according to the California cycle.

Influence of inlet pressure on emissions, performance and smoke when operating with high E.G.R. rates

The tests were conducted at very high load (20% more than the max. BMEP of the standard 8210 engine) and at the intermediate speed, with three different levels of inlet pressure (1.50, 1.75 and 2 kg/cm² absolute). The % E.G.R. was progressively increased up to 20+25%, measuring fuel consumption, emissions and smoke. The results are shown in Fig. 11 : at the same % E.G.R., increasing the inlet pressure, smoke, CO and specific fuel consumption are reduced while NO₂ + HC is increased. However, at the same smoke level (for example, 2.5 Bosch units) minimum NO₂ + HC emissions are reached at the highest inlet pressure. In fact, at this pressure value it is possible to increase the % E.G.R. without the smoke level getting worse. The smoke constraint at 2.5 BU imposes a maximum % E.G.R. of only 1% at 1.5 kg/cm² absolute inlet pressure instead of 20% E.G.R. at 2 kg/cm² absolute inlet pressure.

When the inlet pressure is increased to the first step (from 1.5 kg/cm² to 1.75 kg/cm²) there is a big gain on NO₂ + HC emissions, whereas with the second step (from 1.75 kg/cm² up to 2 kg/cm²) the gain is lower. This means that, in terms of maximum NO₂ + HC reduction induced by E.G.R., it is convenient to operate at an equivalence ratio (ϕ) of about 1.5, as is clearly shown in Fig. 12.

Above $\phi = 1.4+1.5$ there are no smoke problems and the E.G.R. can be increased achieving substantial NO₂ + HC reductions.

Influence of E.G.R. on emissions, performance and smoke when operating with different injection timings

The tests were run at the speeds and loads of the 13-mode cycle with absolute inlet pressures of 1.5 kg/cm^2 (intermediate speed) and 2 kg/cm^2 (rated speed). At 2% and 25% of maximum BMEP, much lower inlet pressures were imposed, considering that the equivalence ratios are still high enough, for the purpose of simulating the operation with a turbocharger. Exhaust back pressures were fixed in such a way as to obtain the desired % E.G.R. and keep constant the ratio between the absolute intake and back pressure at both speeds.

The injection timing was changed around the optimum setting for % E.G.R. = 0 at every speed and load; % E.G.R. was increased until 2.5 - 3 Bosch units were reached or no further reductions on NO_2 + HC emissions occurred.

In Figs. 13, 14 and 15 the results of the tests performed at 75% of full load are shown. Similar results, only quantitatively different, have been obtained with the other test conditions.

It is possible to operate with % E.G.R.s as high as 40%+50% (particularly at part load) with acceptable smoke levels: NO_2 reductions are very high in this case. Hydrocarbon emissions and specific fuel consumption are very little influenced by E.G.R., while carbon monoxide emissions increase, as % E.G.R. increases, but still attain low absolute levels.

The influence of injection timing on the trade-off between NO_2 emissions and % E.G.R. is remarkable. By advancing the injection timing, NO_2 emissions increase (see Fig. 13) but it is still possible to set higher % E.G.R. without the smoke getting worse (see Fig. 14). Therefore, the evaluation of the influence of E.G.R. and injection timing on NO_2 + HC emissions must be made at constant smoke level. For this reason, NO_2 + HC versus % E.G.R. has been plotted in Figs. 16 and 17, at constant injection timing and smoke level. In these graphs, following a line at constant smoke level, for example 2.5 BU, it is possible to see how the NO_2 + HC emissions decrease by advancing the injection timing and at the same time increasing the % E.G.R.: this trend is more remarkable at the rated speed (Fig.17),

but less at the intermediate speed (Fig. 16). In conclusion, for every operating condition (identified by speed, load, inlet and exhaust pressure etc.) and for the constant smoke condition, there is a pair of values for injection timing (φ_0) and % E.G.R. which minimizes $\text{NO}_2 + \text{HC}$ emissions. If is the injection timing optimized for the naturally-aspirated engine without E.G.R., it will be $\varphi_0 \geq \varphi_s$ whatever the load, the speed, etc. With this injection timing (φ_0) the negative effects on CO and fuel consumption, are very small (see Fig. 15) compared with condition E.G.R. = 0.

13-mode California cycle results with modulated E.G.R. and supercharging

The latter results allow the calculation of emissions (according to the 13-mode cycle) to be performed with different % E.G.R.s, injection timing and inlet pressure laws versus speed and load.

The calculations were worked out at the same power of the N.A. engine and at 2.5+3 BU smoke level, and the more significant results are shown in Table 5.

Supercharging and modulated E.G.R., for minimum $\text{NO}_2 + \text{HC}$ emissions, with both modulated injection timing as a function of speed/load and injection timing as a function only of speed, is an effective way to meet the '77 California Standard (see cycles 1 and 2 of Table 5).

The following considerations can be drawn from the emissions and performance results reported in Table 5 and Fig.18 :

- There is little to gain with injection advance as function of both speed and load. A proper choice of injection advance as function only of the speed allows satisfactory results to be obtained.

- Further emission reductions can be ensured by increasing the inlet pressure (see cycle 3, Table 5).
- Adoption of a suitable turbocharger can improve specific fuel consumption by 5.8% over the figure given in Table 5 as evidenced in Fig. 10.

VARIABLE COMPRESSION RATIO

An investigation was conducted on high specific power and limited bulk Diesel engines and it included the evaluation of an 8210 turbocharged engine fitted with variable compression ratio pistons developed by Teledyne Continental Motors on a patent by B.I.C.E.R.I. (British Internal Combustion Engine Research Institute). A schematic of said piston can be seen in Fig. 19 and its working principle (Fig. 20) can be easily understood by observing that the position of piston shell A (outer member) and piston pin carrier B (inner member) relative to each other is controlled by the amount of oil in the upper and lower chambers. Such control is a consequence of the balance between peak firing pressure and relief valve opening pressure.

Movement between A and B in the exhaust and intake strokes is controlled by a fixed orifice O. Oil discharged from both relief valve and control orifice returns directly to the sump. So, as the peak combustion pressure increases over a pre-fixed value (in our application : 125 kg/cm^2) the compression ratio will reduce from a maximum to a minimum (in our case, 18:1 and 12:1 respectively). Though the primary target of the work was a system evaluation, tests were also made on gaseous emissions both at maximum performance and retarded injection timings. Results of this investigation can be seen in Figs. 21 and 22 which show absolute emission levels for both California 13-mode cycle engine speeds. As can be seen, NO_2 emission levels against b.m.e.p. are more or less constant (2200 R.P.M.) or are decreasing (1600 R.P.M.) commencing when the compression ratio reduction has just started.

Fig. 23 shows the effect of injection timing reduction : 4.8 gr/BHP.h for $\text{NO}_2 + \text{HC}$ are reached with 18° crankshaft static injection timing. As readily apparent, to get this result a substantial loss of power (about 20%) must be accepted, but the performance is still gaining over the original engine with invariable compression ratio.

These results have been obtained using a compressed air supply because a suitable turbocharger was not at hand. The limitation of maximum combustion pressure, and indeed peak combustion temperature, is responsible for the effective control of NO emissions encountered, but some role could be possibly attributed to combustion chamber geometry, outer piston material (steel) and the injection law resulting from the very high quantity of fuel injected. The influence of these parameters has not been investigated but it seems realistic to consider them as secondary with respect to compression ratio variation.

From the above considerations, it may be concluded that V.C.R., though originally devised for other purposes, shows promising prospects in emission control provided the increased construction complexity and maintenance are not problem areas. Naturally, because of our limited V.C.R. experience deeper investigation is needed in this direction; at this stage the intent of our report is only to point out the added potential of this solution.

GASOIL-METHANOL DUAL FUEL OPERATION WITH SUPERCHARGING

The possibilities both for gasoil consumption reduction and exhaust emission control, by means of gasoil-methanol dual fuel feeding, have been investigated on a 8210 supercharged engine. Methanol's high heat of evaporation is the physical factor which has been considered of main interest because :

- It avoided using the heat exchanger which is normally needed for reducing the high temperature of the compressed air.
- It could allow some benefit for NO_2 emission control through some limitations of the peak combustion temperature, as a consequence of the reduced temperature of the inlet air.

Moreover, having the dual fuel operation shown very effective possibilities for smoke abatement (11), a further benefit for NO_2 reduction could be obtained by injection timing retard without incurring in power losses due to smoke limits. Fig. 24 shows a simplified scheme of the methanol feed system, while gasoil was injected by the original equipment injection pump of the engine.

The tests were performed with the following aims :

- Establish the maximum percent of methanol to be adopted without incurring in engine malfunction and/or unacceptable HC emissions.
- Verify how far the controlled emission limits may be lowered using the methanol-gasoil fuel combination plus injection advance retard, without unacceptable smokiness.

The dual fuel operation was actuated by reducing gasoil delivery and adding methanol up to the original power level. Results of one of such tests can be seen on Fig. 25, where gaseous emissions, smoke, exhaust gas temperature, peak combustion

pressure, etc. are plotted against the percentage of load obtained with methanol. Here, the maximum allowable load with methanol is about 28%, beyond which there is notable increase in HC emissions.

Fig. 26 shows the engine power curve at 28% load by methanol at the highest speed and holding the carburettor throttle in the same position even for lower speeds : for comparison, performance with gasoil alone is also indicated : smoke and exhaust gas temperature are reduced by a wide margin and total efficiency is slightly improved.

The load is reduced from 100% to 72% by excluding methanol only, so operation at loads lower than 72% is at gasoil alone.

Fig. 27 shows smoke, emissions, exhaust gas temperature, gasoil and methanol delivery, bhp and total efficiency versus static injection timing at 2200 R.P.M. (rated speed). With dual fuel operation, optimum injection timing for emission reduction is 12° b.t.d.c. : $\text{NO}_2 + \text{HC}$ of 4,8 gr/bhp.h according to the California cycle is reached without power loss and with a slight smoke increase, while on gasoil alone 5.7 gr/bhp.h of $\text{NO}_2 + \text{HC}$ with a power loss of about 15% was obtained.

The histograms (Fig. 28) give a direct comparison between the more significant results obtained with gasoil alone and with dual fuel under both maximum performance and minimum emission injection timing conditions.

While from an experimental point of view the above results may be considered of great interest, for practical applications some problems arise.

Of these problems, the hardest to solve - not considering intake system design changes, which are possible - is ensuring the necessary uniform distribution of methanol to the cylinders : as evidence of how true this is, some combustion chamber erosion in piston n. 1 were experienced.

Additionally, as for the other experiences described here, not much can be said on long term testing reliability and repeatability.

CONCLUSION

The results of the investigation presented in this paper allow the following conclusions to be drawn with regard to the possibilities of meeting the 5 g/HP.h emissions limit of $\text{NO}_2 + \text{HC}$ for the different types of engines tested.

8060 Engine

- Modulated E.G.R. is an effective method for ensuring an emission level of 4.5 g/HP.h with little losses in performance and fuel consumption (see Table 3). Further $\text{NO}_2 + \text{HC}$ reductions are possible but with remarkable power penalties.
- The modulated E.G.R. technique is, on the whole, more effective than the injection timing retard method (see Table 4 and Fig. 8).

8210 Engine

This engine shows higher $\text{NO}_2 + \text{HC}$ baseline emissions : more sophisticated techniques are needed. Fig. 29 shows the methods allowing the 5 g/HP.h $\text{NO}_2 + \text{HC}$ emission level to be met. Some data are taken from a previous paper (1) reporting the results of the first part of our research program.

As shown in Fig. 29, the best results can be obtained with modulated E.G.R. plus supercharging. However, the E.G.R. technique presents many problems, particularly as regards reliability and the E.G.R. modulation is not easy to handle.

On the balance of the positive and negative effects the more reliable method to meet the $\text{NO}_2 + \text{HC}$ standard is to increase the injection rate in conjunction with modulated injection timing.

Such a possibility is offered by the adoption of a pump with high injection rate, such as the Fiat DRF (Distributore Rotante Fiat) pump with its electronic control for injection advance. A picture of this distributor type pump is shown in Fig. 30; its operation was explained in the previously quoted paper (1).

8210 turbocharged engine with variable compression ratio

By means of compression ratio variation controlled by the peak combustion pressure, both very high specific outputs and low NO_2 + HC emissions are obtainable (see Fig. 23).

Dual fuel (Gasoil-methanol) operation on 8210 turbocharged engine

NO_2 + HC emission levels lower than 5 gr/HP.h without performance losses and with smoke still at fully acceptable levels were reached using the two fuels. With gasoil only, a minimum of 5.7 g/HP.h NO_2 + HC emission level was possible but with a b.h.p. loss of about 15 percent. (See Fig. 28).

As mentioned earlier, all results are experimental and were recorded from basic laboratory work.

Only after a more specific research on emission variations against operation time will it be possible to state what emission levels are needed at low hours to stay within the specified limits even after long term testing.

Since single units of the same engine can give notable difference in emission levels, more specific work is needed on engines of the same family to evaluate emission variations and associated affecting parameters.

A P P E N D I X A

A very simple mathematical model to work out the composition of the intake charge has been developed for a better understanding of the E.G.R. technique.

The mathematical model has been developed with and without intercooling of the recirculated exhaust gas. For the sake of brevity, the explanation of this model is referred to the set-up without intercooler.

In Fig. A-1 a scheme of the E.G.R. system is shown. The molar flow of the different components through sections 1 and 4 are (assuming the air to be composed only of O_2 and N_2 and in the hypothesis of complete combustion) :

Section 1 : through this section passes air flow G_a (kg/h) composed of :

- a (moles/h) O_2
- a • 3.76 (moles/h) N_2
- x (moles/h) H_2O (due to ambient humidity)

Section 4 : the flow of recirculated exhaust gas G_r (kg/h) is composed of :

- b (moles/h) CO_2
- c (moles/h) H_2O
- d (moles/h) O_2
- e (moles/h) N_2

In the hypothesis of complete combustion the equation of combustion is :

$$\begin{aligned}
 \text{(EQ.1)} \quad nCH_\lambda + (a + d) O_2 + (a \cdot 3.76 + e) N_2 + b CO_2 + (c + x) H_2O = \\
 (n + b) CO_2 + \left[\frac{n\lambda}{2} + (c + x) \right] H_2O + \left[a + d - \left(2n + \frac{n\lambda}{2} \right) \frac{1}{2} \right] O_2 + \\
 (a \cdot 3.76 + e) N_2
 \end{aligned}$$

where n is the molar fuel flow (moles/h) and λ is the atom ratio H/C of hydrocarbon CH_λ .

The molar fuel flow can be expressed as a function of mass fuel flow C (kg/h):

$$(EQ.2) \quad n = \frac{C}{12.01 + 1.008 \lambda}$$

However, the intake oxygen flow (moles/h) can be expressed as a function of C and x :

$$(EQ.3) \quad a = \frac{\alpha C - x 18}{32 + 3.76 \cdot 28.016}$$

Where α is the air-to-fuel ratio and x is a function of absolute humidity m (kg.H₂O/kg dry air) :

$$(EQ.4) \quad x = \frac{\alpha C}{18} \cdot \frac{m}{1 + m}$$

The mass fractions of the intake charge components in section 2 can be expressed as :

$$(EQ.5) \quad f(H_2O) = \frac{(1 - \theta)}{C \alpha} (x + c) 18$$

$$(EQ.6) \quad f(CO_2) = \frac{(1 - \theta)}{C \alpha} b 44$$

$$(EQ.7) \quad f(O_2) = \frac{(1 - \theta)}{C \alpha} (a + d) 32$$

$$(EQ.8) \quad f(N_2) = 1 - [f(H_2O) + f(CO_2) + f(O_2)]$$

The values of b, c and d in previous equations 5, 6, 7 and 8 can be calculated from the following :

$$(EQ.9) \quad b = \frac{n \omega}{1 - \omega}$$

$$(EQ.10) \quad c = \frac{n\lambda\omega + 2x\omega}{2(1 - \omega)}$$

$$(EQ.11) \quad d = \frac{\omega}{4(1 - \omega)} \quad (4a - 4n - n\lambda)$$

In equations 9, 10, and 11 : $\omega = \frac{\alpha Q}{\alpha + 1 - Q}$ and x has to be calculated from EQ.4.

This way, it is possible to work out from EQ.5, 6, 7 and 8 the exhaust gas composition (in the hypothesis of complete combustion) as a function of fuel flow (C), % E.G.R. (Q), air-to-fuel ratio (α) and ambient humidity (m) for a given fuel (λ).

Fig. A-2 shows the results of this calculation for the 8060 engine. At the same % E.G.R. the inert gas fraction $[1 - f(O_2)]$ at the intake increases as the load is varied from 25% to 75% of full load.

Plotting the % NO reduction as a function of $f(O_2)$ (Fig. A-4) the variation induced by the load is very much lower than plotting % NO reduction versus % E.G.R. (Fig. A-3).

This fact proves that the main parameter controlling NOx is the inert gas fraction at the intake and that it is better to study E.G.R. technique as a function of % O_2 at the intake rather than as a function of % E.G.R.

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- FIG. 1 - Emissions, smoke and specific fuel consumption as a function of % E.G.R. at 25% of max. BMEP. (8060 engine, inj. system tuned for low emissions).
- FIG. 2 - Emissions, smoke and specific fuel consumption as a function of % E.G.R. at 75% of max. BMEP. (8060 engine, inj. system tuned for low emissions).
- FIG. 3 - Smoke as a function of equivalence ratio and % E.G.R. (8060 engine, inj. system tuned for low emissions).
- FIG. 4 - Emissions, smoke and specific fuel consumption as a function of % E.G.R. at full load and constant smoke level. (8060 engine, inj. system tuned for low emissions).
- FIG. 5 - NO_2 + HC specific emissions on the modes of the California cycle as a function of % E.G.R. (intermediate speed). (8060 engine, inj. system tuned for low emissions).
- FIG. 6 - NO_2 + HC specific emissions on the modes of the California cycle as a function of % E.G.R. (rated speed). (8060 engine, inj. system tuned for low emissions).
- FIG. 7 - Modulated E.G.R. over the California cycle : % E.G.R. as a function of load and speed (8060 engine).
- FIG. 8 - Specific fuel consumption and smoke with two different techniques for NO_x emission control (8060 engine) :
- . inj. timing retard
 - . modulated E.G.R.
- FIG. 9 - The effect of exhaust back pressure on emissions and smoke (8210 engine, full load).
- FIG. 10 - The effect of exhaust back pressure on specific fuel consumption (8210 engine, full load).
- FIG. 11 - The effect of inlet pressure on emissions, smoke and specific fuel consumption with E.G.R. (8210 engine).

- FIG. 12 - Equivalence ratios at full load as a function of inlet pressure and % E.G.R. (same operating conditions of Fig. 11).
- FIG. 13 - HC and NO₂ emissions as a function of % E.G.R. and inj. timing (8210 engine, rated speed, 75% of full load).
- FIG. 14 - Smoke and HC + NO₂ emissions as a function of % E.G.R. and inj. timing (8210 engine, rated speed, 75% of full load).
- FIG. 15 - BSFC and CO emissions as a function of E.G.R. and inj. timing (8210 engine, rated speed, 75% of full load).
- FIG. 16 - The effect of % E.G.R. and inj. timing on HC + NO₂ and smoke (8210 engine, intermediate speed - full load).
- FIG. 17 - The effect of % E.G.R. and inj. timing on HC + NO₂ and smoke (8210 engine, rated speed - full load).
- FIG. 18 - Modulated E.G.R. over the California cycles reported in Table 5 : % E.G.R. as a function of load and speed (8210 engine).
- FIG. 19 - Variable compression ratio piston.
- FIG. 20 - V.C.R. engine - Compression ratio and peak combustion pressure as a function of BMEP.
- FIG. 21 - The effect of inj. timing on no absolute emissions at intermediate speed. (T.C. 8210 engine with V.C.R.).
- FIG. 22 - The effect of inj. timing on no absolute emission at rated speed. (T.C. 8210 engine with V.C.R.).
- FIG. 23 - The effect of injection timing on NO₂ + HC specific emissions, smoke, B.S.F.C. and B.H.P. (T.C. 8210 engine with V.C.R.).
- FIG. 24 - Scheme of the methanol feed system. (T.C. 8210 engine).
- FIG. 25 - The effect of methanol on emissions, smoke, exhaust temperature, peak combustion pressure, head vibrations. (T.C. 8210 engine).

- FIG. 26 - Performance, smoke, fuels delivery total efficiency with both gas oil and dual fuel. (T.C. 8210 engine).
- FIG. 27 - The effect of injection timing on emissions, smoke, exhaust gas temperature, total efficiency at rated speed. (T.C. 8210 engine).
- FIG. 28 - The Table shows the results of the 13-mode cycle with standard and retarded injection timing. (T.C. 8210 engine).
- FIG. 29 - Comparison of different methods of emission control on the N.A. 8210 engine. (13-mode California cycle).
- FIG. 30 - Isometric view of the Fiat DRF pump.
-
- FIG. A-1 - Scheme of the E.G.R. system.
- FIG. A-2 - Intake charge composition as a function of % E.C.R. (8060 engine, intermediate speed).
- FIG. A-3 - NO reduction in percent as a function of % E.G.R. (8060 engine).
- FIG. A-4 - NO reduction in percent as a function of % O_2 . (8060 engine).

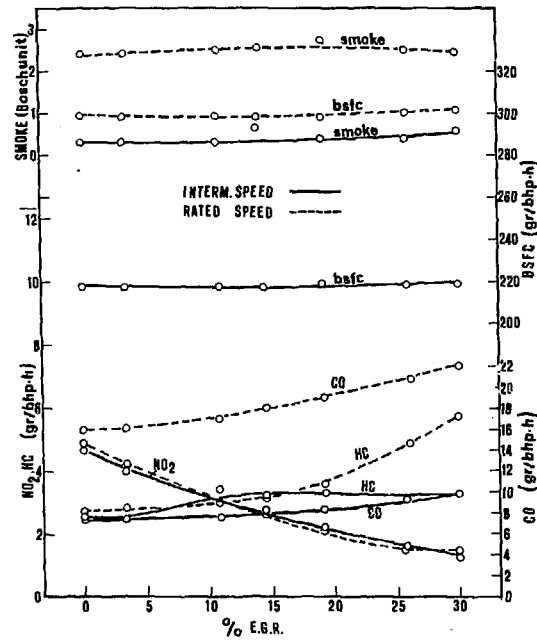


FIG. 1 - Emissions, smoke and specific fuel consumption as a function of % E.G.R. at 25% of max. BMEP. (8060 engine, inj. system tuned for low emissions).

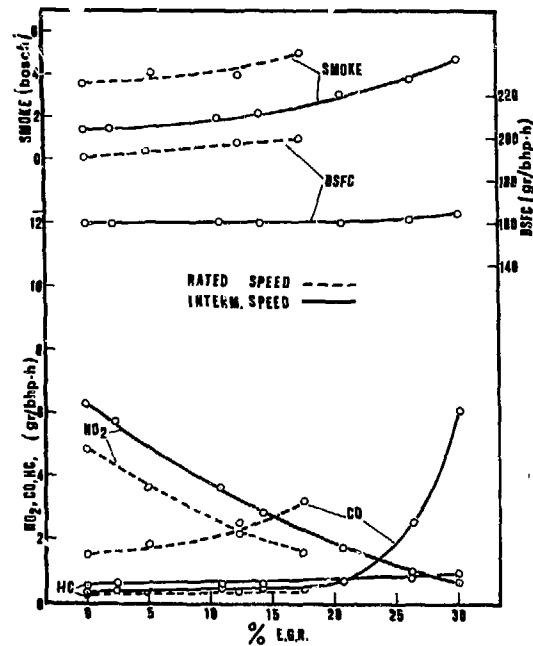


FIG. 2 - Emissions, smoke and specific fuel consumption as a function of % E.G.R. at 75% of max. BMEP. (8060 engine, inj. system tuned for low emissions).

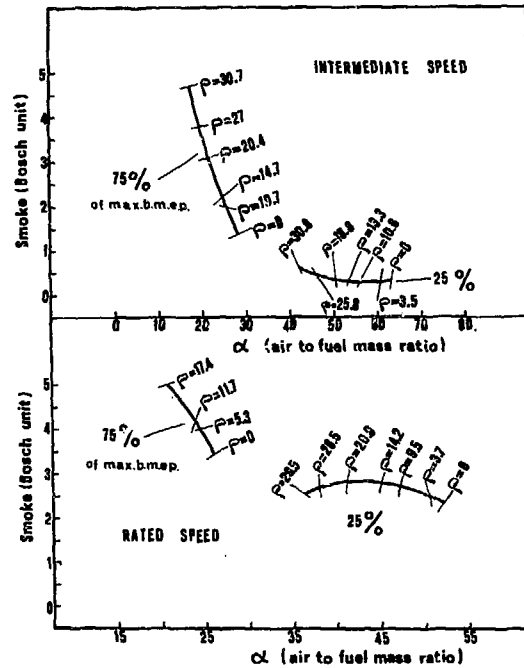


FIG. 3 - Smoke as a function of equivalence ratio and % E.G.R. (8060 engine, inj. system tuned for low emissions).

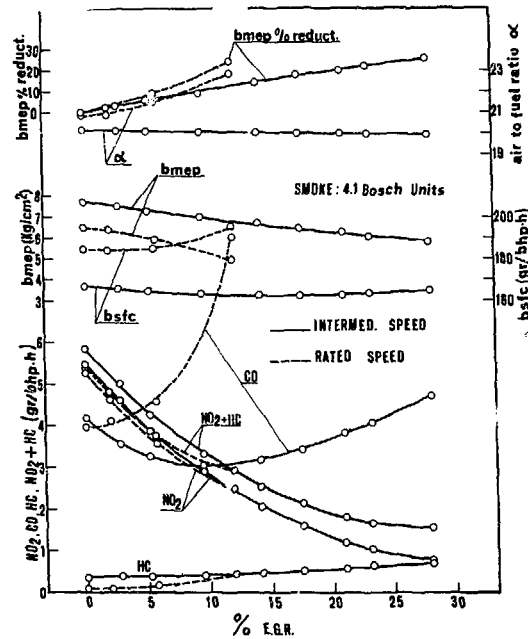


FIG. 4 - Emissions, smoke and specific fuel consumption as a function of % E.G.R. at full load and constant smoke level. (8060 engine, inj. system tuned for low emissions).

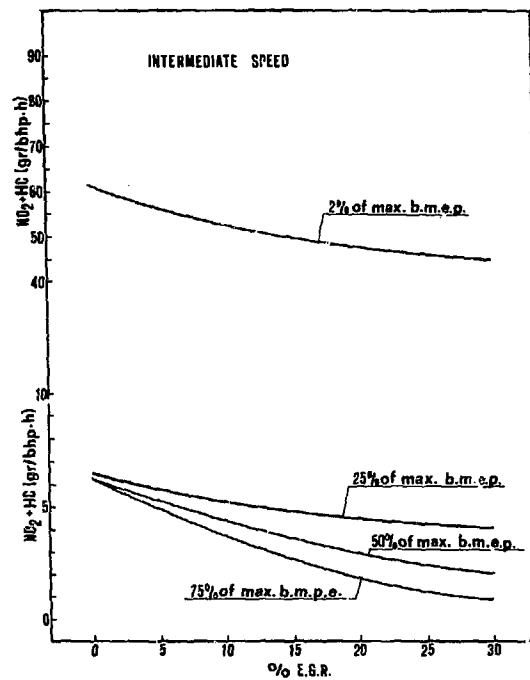


FIG. 5 - NO₂ + HC specific emissions on the modes of the California cycle as a function of % E.G.R. (intermediate speed). (8060 engine, inj. system tuned for low emissions).

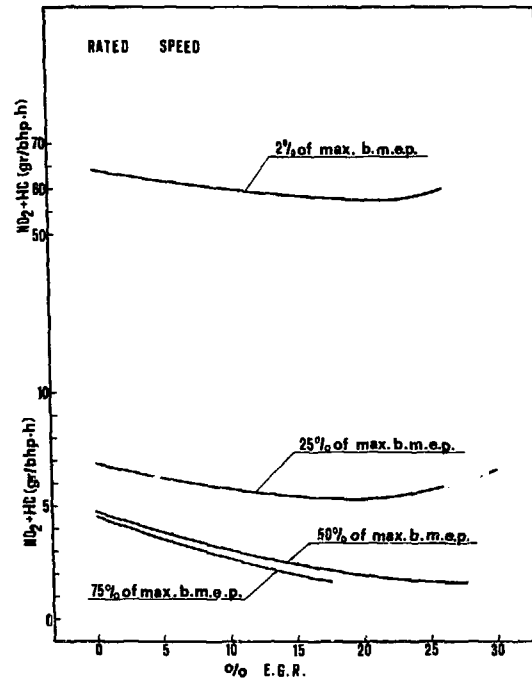


FIG. 6 - $\text{NO}_2 + \text{HC}$ specific emissions on the modes of the California cycle as a function of % E.G.R. (rated speed). (8060 engine, inj. system tuned for low emissions).

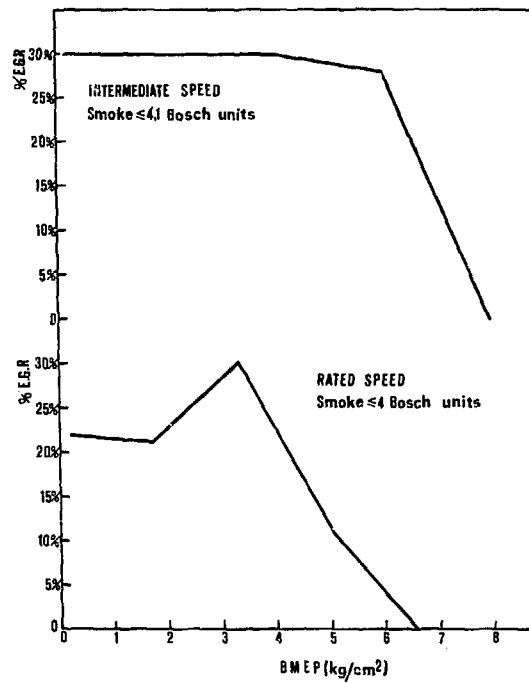


FIG. 7 - Modulated E.G.R. over the California cycle : % E.G.R. as a function of load and speed (8060 engine).

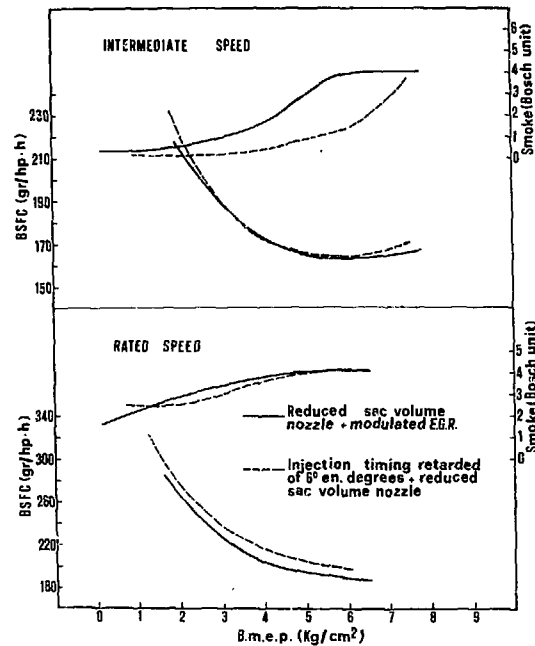


FIG. 8 - Specific fuel consumption and smoke with two different techniques for NO_x emission control (8060 engine) :

- . inj. timing retard
- . modulated E.G.R.

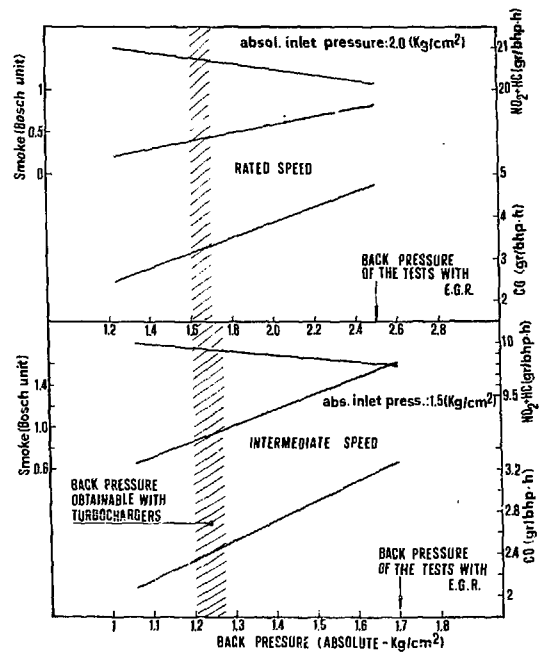


FIG. 9 - The effect of exhaust back pressure on emissions and smoke (8210 engine, full load).

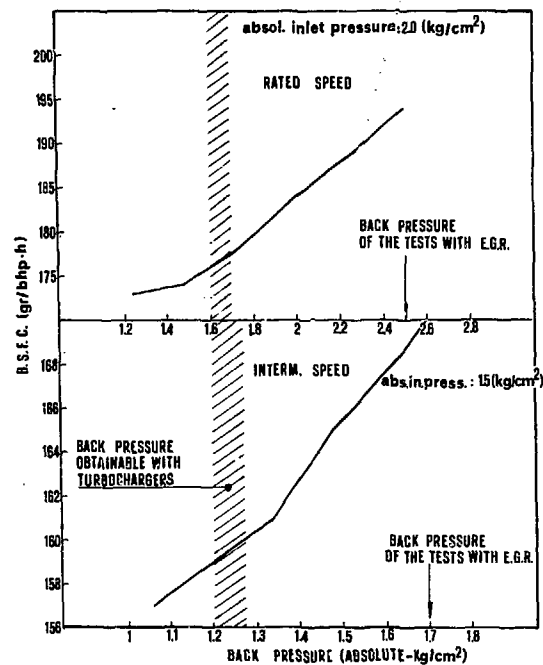


FIG. 10 - The effect of exhaust back pressure on specific fuel consumption (3210 engine, full load).

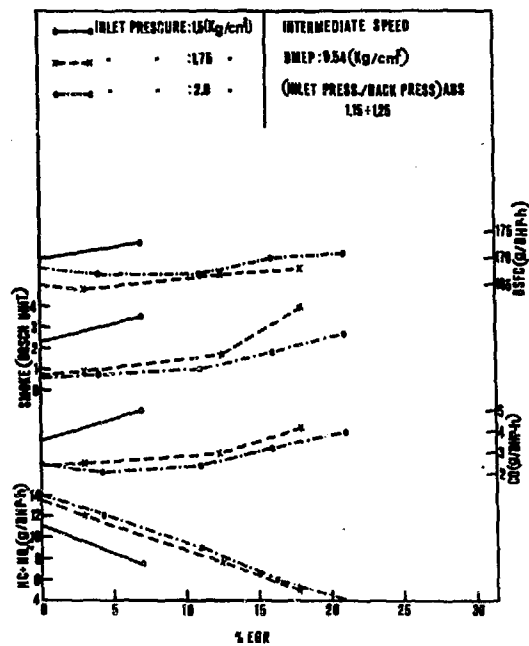


FIG. 11 - The effect of inlet pressure on emissions, smoke and specific fuel consumption with E.G.R. (8210 engine).

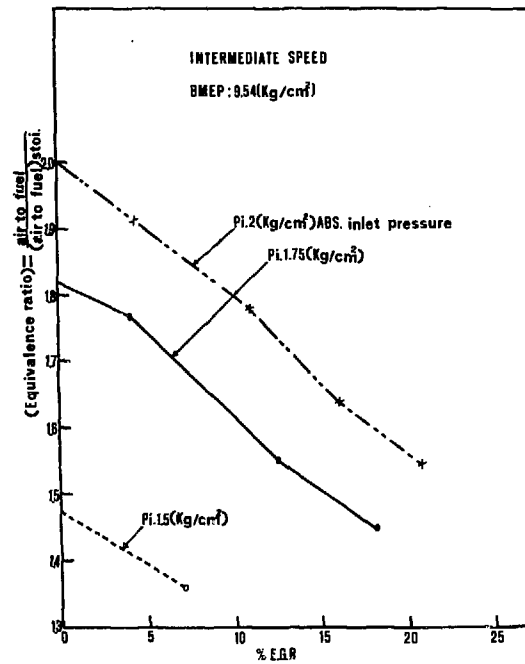


FIG. 12 - Equivalence ratios at full load as a function of inlet pressure and % E.G.R. (same operating conditions of Fig. 11).

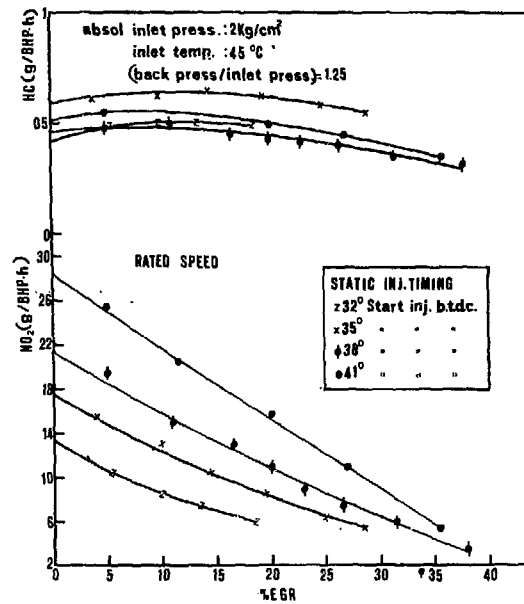


FIG. 13 - HC and NO₂ emissions as a function of % E.G.R. and inj. timing (8210 engine, rated speed, 75% of full load).

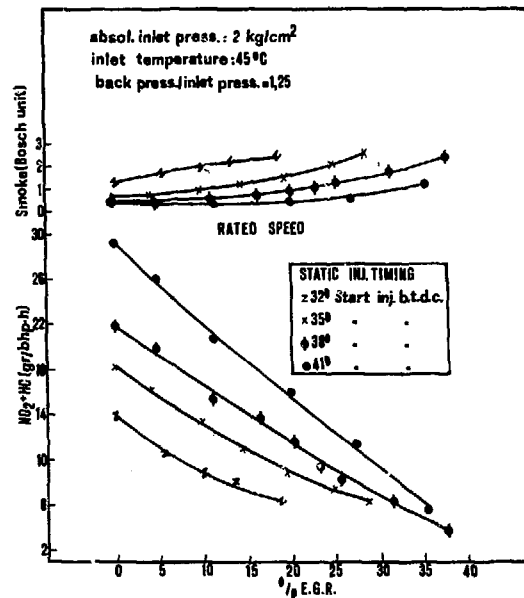


FIG. 14 - Smoke and HC + NO₂ emissions as a function of % E.G.R. and inj. timing (8210 engine, rated speed, 75% of full load).

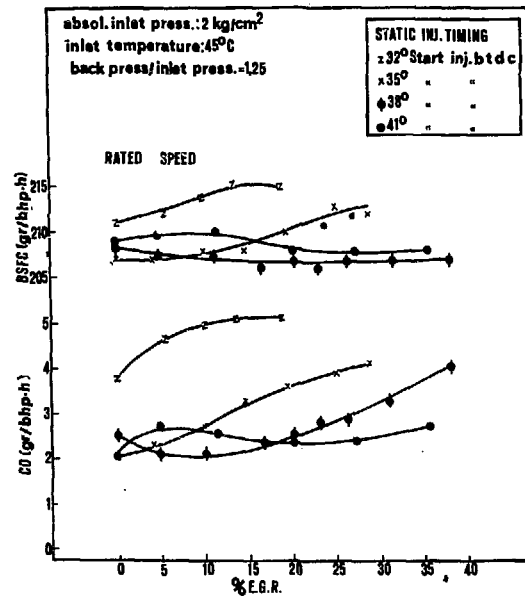


FIG. 15 - BSFC and CO emissions as a function of E.G.R. and inj. timing (8210 engine, rated speed, 75% of full load).

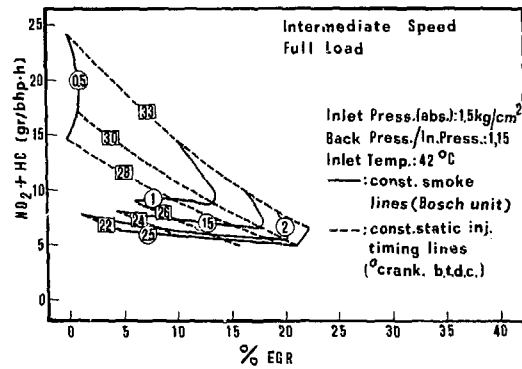


FIG. 16 - The effect of % E.G.R. and inj. timing on HC + NO₂ and smoke (8210 engine, intermediate speed - full load).

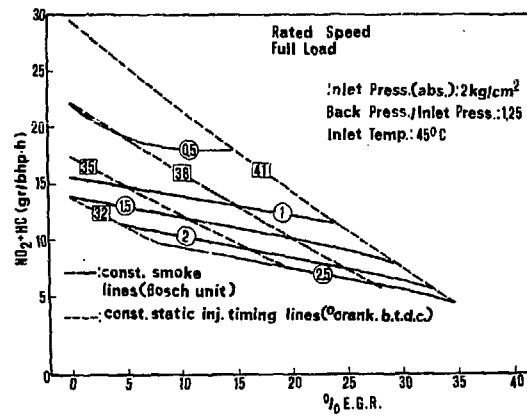


FIG. 17 - The effect of % E.G.R. and inj. timing on HC + NO₂ and smoke (8210 engine, rated speed - full load).

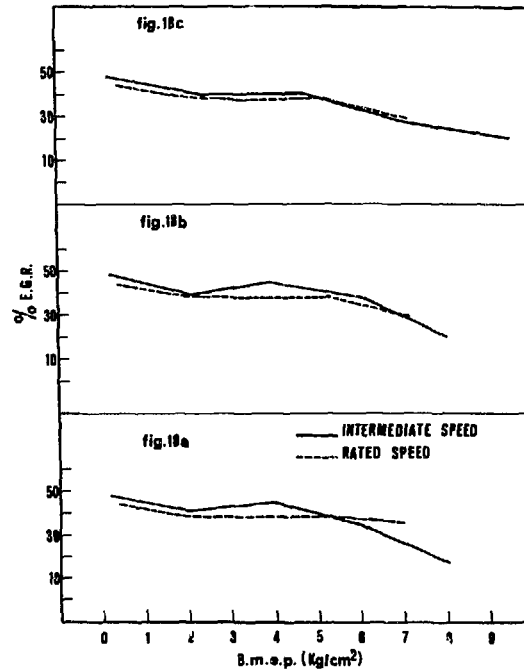


FIG. 18 - Modulated E.G.R. over the California cycles reported in Table 5 :
% E.G.R. as a function of load and speed (8210 engine).

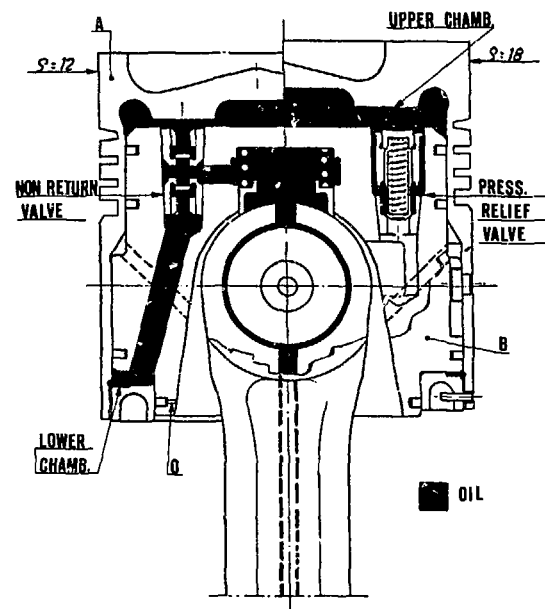


FIG. 19 - Variable compression ratio piston.

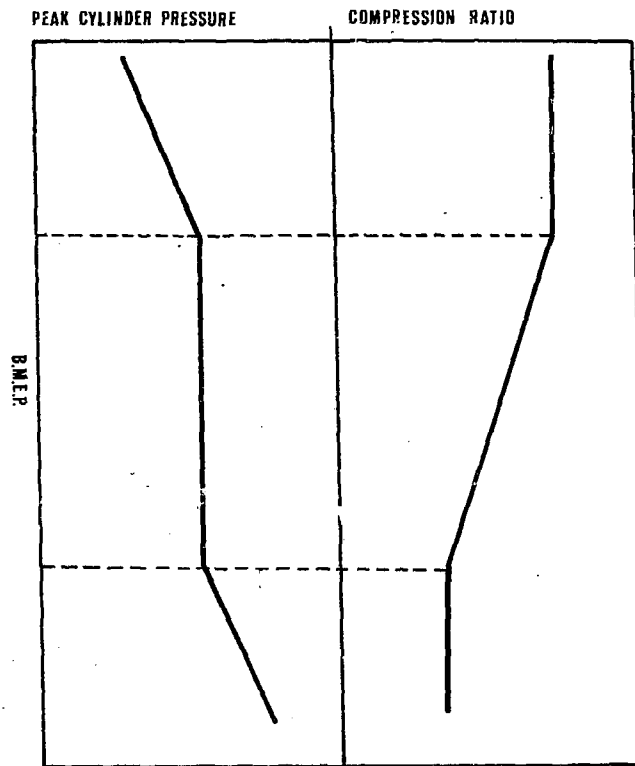


FIG. 20 - V.C.R. engine - Compression ratio and peak combustion pressure as a function of BMEP.

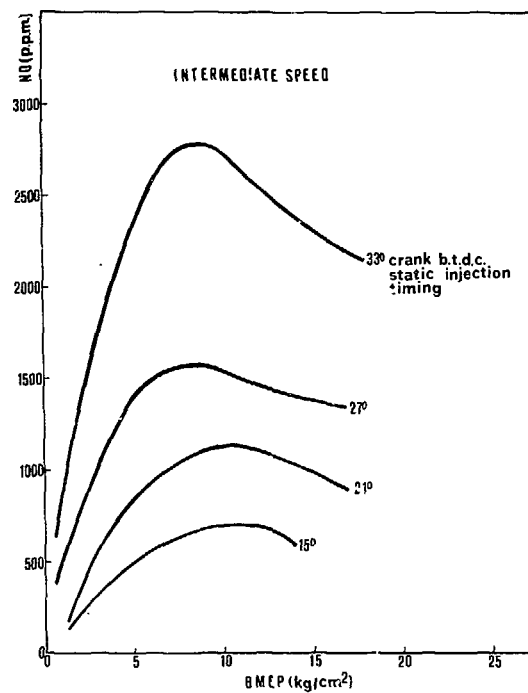


FIG. 21 - The effect of inj. timing on no absolute emissions at intermediate speed. (T.C. 8210 engine with V.C.R.).

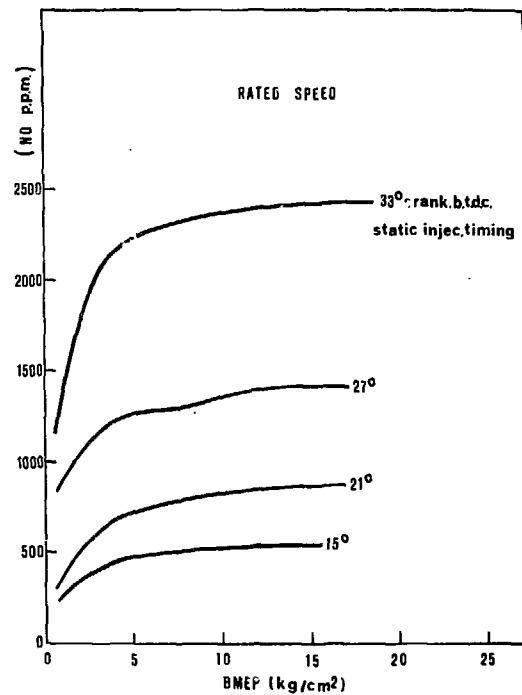


FIG. 22 - The effect of inj. timing on no absolute emission at rated speed.
(T.C. 8210 engine with V.C.R.).

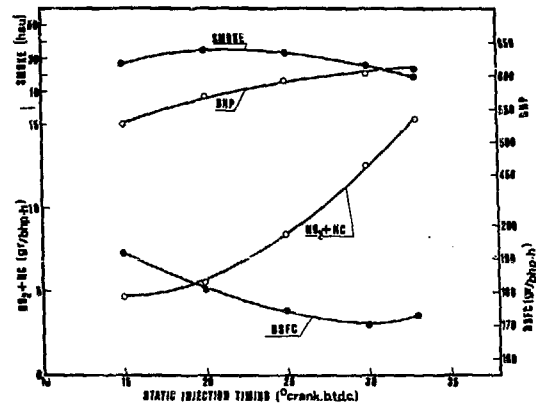


FIG. 23 - The effect of injection timing on NO₂ + HC specific emissions, smoke, B.S.F.C. and B.H.P. (T.C. 8210 engine with V.C.R.).

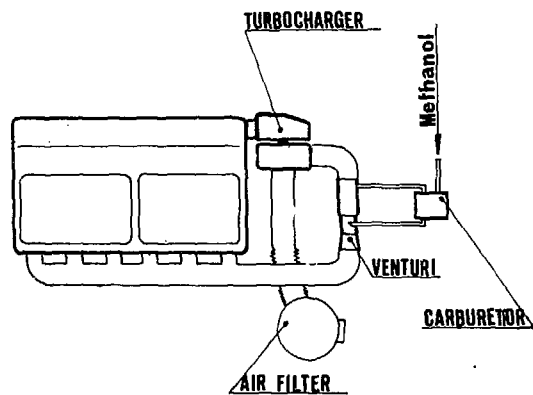


FIG. 24 - Scheme of the methanol feed system. (T.C. 8210 engine).

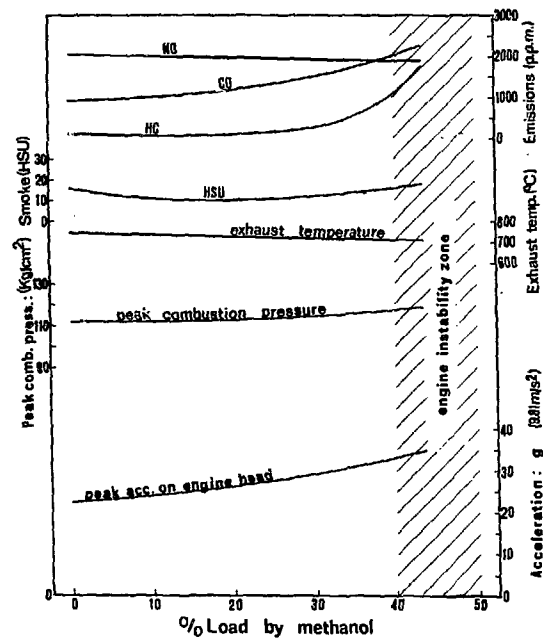


FIG. 25 - The effect of methanol on emissions, smoke, exhaust temperature, peak combustion pressure, head vibrations. (T.C. 8210 engine).

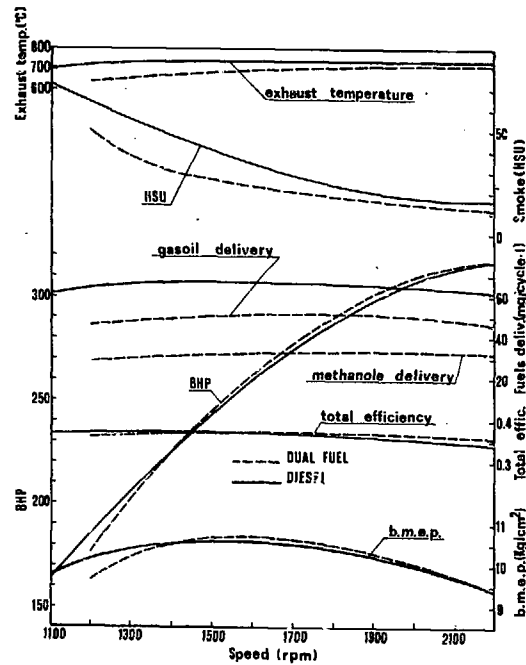


FIG. 26 - Performance, smoke, fuels delivery total efficiency with both gas oil and dual fuel. (T.C. 8210 engine).

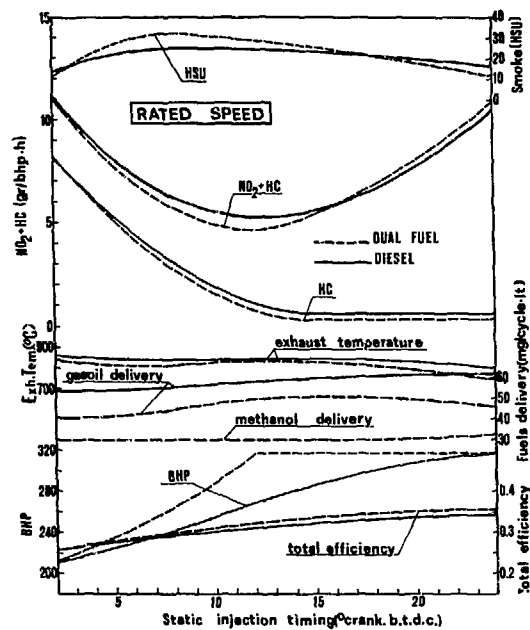


FIG. 27 - The effect of injection timing on emissions, smoke, exhaust gas temperature, total efficiency at rated speed. (T.C. 8210 engine).

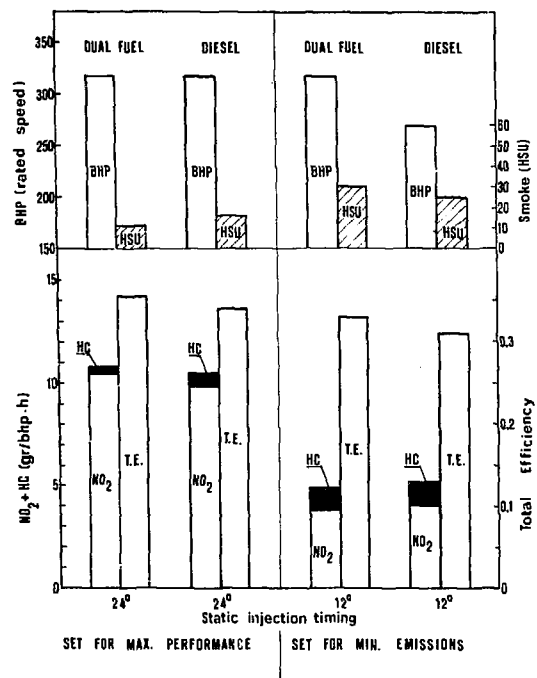


FIG. 28 - The Table shows the results of the 13-mode cycle with standard and retarded injection timing. (T.C. 8210 engine).

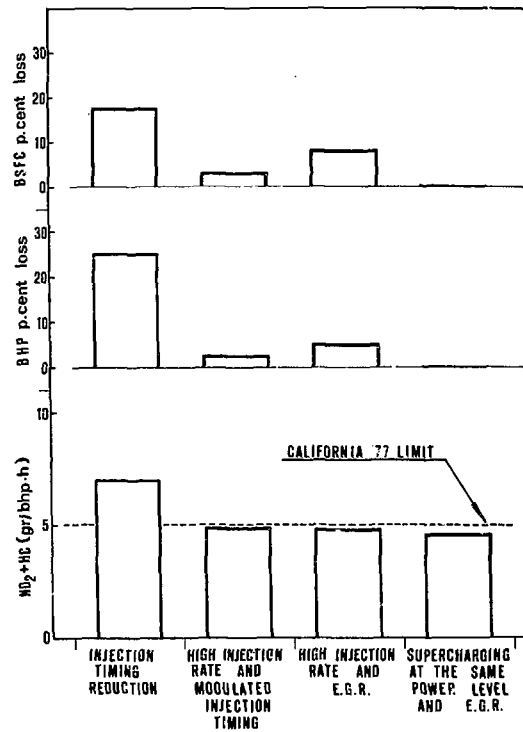


FIG. 29 - Comparison of different methods of emission control on the N.A. 8210 engine. (13-mode California cycle).

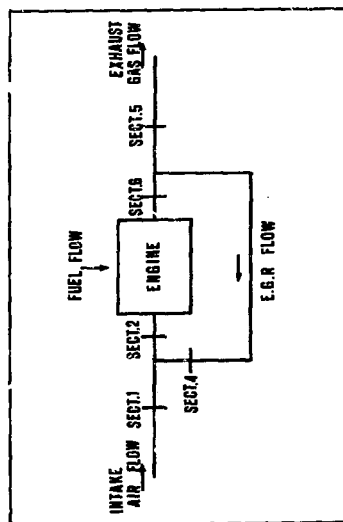


FIG. A-1 - Scheme of the E.G.R. system.

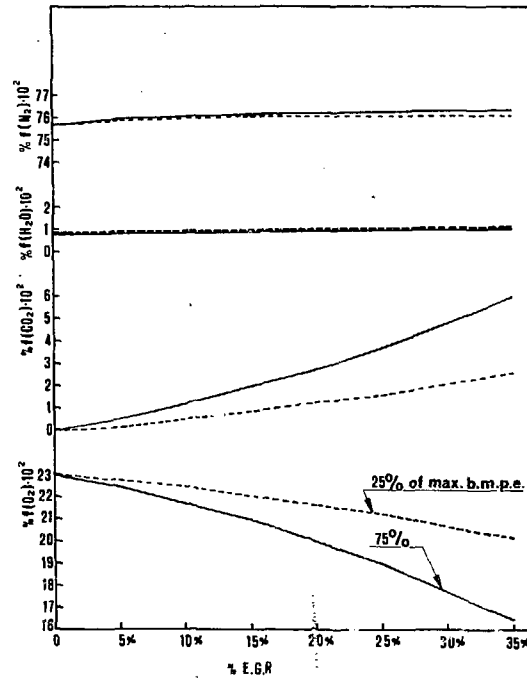


FIG. A-2 - Intake charge composition as a function of % E.G.R. (8060 engine, intermediate speed).

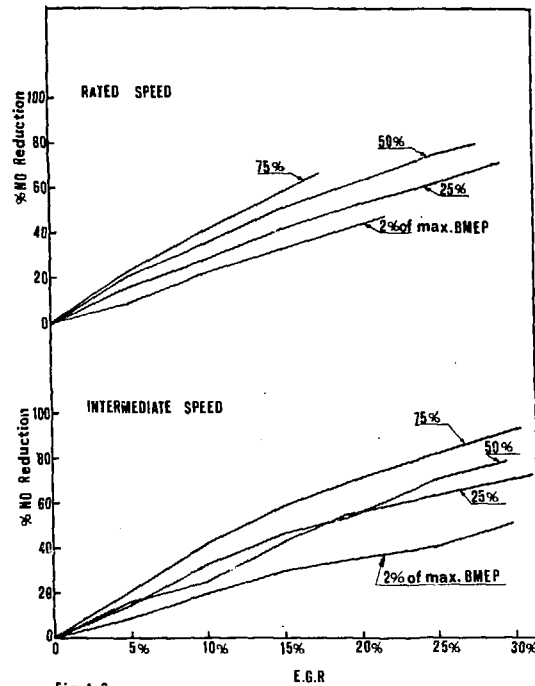


Fig. A-3

FIG. A-3 - NO reduction in percent as a function of % E.G.R. (8060 engine).

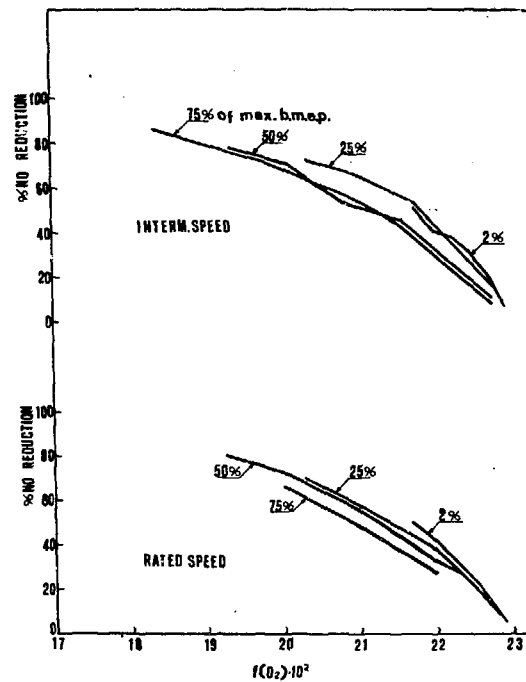


FIG. A-4 - NO reduction in percent as a function of % O_2 . (8060 engine).

TABLE 1 : Specifications of the Fiat D.I. Diesel Engines Used

	8060	8210 NAT.ASPIRATED	8210 TURBOCHARGED
Displacement (cm ³)	5.183	13.798	13.798
Bore (mm)	100	137	137
Stroke (mm)	110	156	156
N° of cylinders	6	6	6
Compression ratio	17:1	16:1	15:1
Speed of Max. BMEP (R.P.M.)	1.800	1.300	1.600
Max BMEP (kg/cm ²)	7,8	8,3	12,0
Rated speed (R.P.M.)	3.200	2.200	2.200
B.M.E.P. at rated speed (kg/cm ²)	6,7	7,4	10,6

TABLE 2 : Specifications of the Fuel Used

	CEC RF-03-T-74	FEDERAL REGISTER (June 30, 1975)	GASOIL USED FOR TESTS
Gravity, 15 °C	0.830 ± 0.005	0.8251 ± 0.8063	0.832
Distillation range			
50 percent point	245 min. °C	195.8 + 234.7 °C	265 °C
90 percent point	330 ± 10 °C	223.6 + 256.9 °C	331 °C
E.P.	370 max. °C	---	348 °C
Cetane	54 ± 3	48 + 54	55.5
Viscosity at. 37.8 °C	3 ± 0.5 cst	1.6 - 2.0 cst	2,80 cst
Total sulfur	0.4 ± 0.1% wgt	0.05 + 0.20 %	0.50 %
Flash point	55 min. °C	34.7 °C	64 °C
Cloud point	-7 max. °C	---	---
Aniline point	69 ± 5 °C	---	70 °C
Conradson residual on the percent	0.2 max. % wgt	---	0.08 %
Ash content	0.01 max % wgt	---	0.01 %
H ₂ O content	0.05 max % wgt	---	0
Copper foil corrosion at 100 °C	1 max.	---	0
Acidity	nil mg KOH/g	---	---

TABLE 3 : The Effect of E.G.R. over the 13-mode California cycle.
8060 engine with injection system tuned for low emissions.

13 MODE CALIF. CYCLE		CYCLE No 1 CYCLE No 2 CYCLE No 3 CYCLE No 4 CYCLE No 5 CYCLE No 6					
		NO E.G.R.	WITH E.G.R. WITH E.G.R. WITH E.G.R. WITH E.G.R. WITH E.G.R.				

TABLE 4 : Comparison of Different Techniques for NO₂ + HC Emissions Control over the 13-Mode California Cycle (8060 Engine)

8060 engine with injection system tuned for low emissions

		6° INJ. RETARD	FULLY MODULATED E.G.R.	
INTERMEDIATE SPEED	B.H.P.	82,3	82,1	CALIFORNIA 1977 LIMITS
	SMOKE (BOSCH UNIT)	3,7	4,1	
	B.S.F.C. (g/BHP.h)	171	165	
	E.G.R. %	-	4	
RATED SPEED	E.H.P.	112,6	112,4	
	SMOKE (BOSCH UNIT)	4,1	4,0	
	B.S.F.C. (g/BHP.h)	198	183	
	E.G.R. %	-	4	
13 MODE CALIF. CYCLE	CO (g/BHP.h)	8,25	8,39	25
	NO ₂ " "	4,13	2,76	
	HC " "	1,93	1,51	
	HC+NO ₂ (g/BHP.h)	6,06	4,27	5

TABLE 5 : The Effect of E.G.R. and Supercharging over the 13-Mode California Cycle (8210 Engine)

		<u>CYCLE N° 1</u>	<u>CYCLE N° 2</u>	<u>CYCLE N° 3</u>
		- FULLY MODUL. E.G.R. (see Fig. 18a)	- FULLY MODUL. E.G.R. (see Fig. 18b)	- FULLY MODUL. E.G.R. (see Fig. 18c)
		- INJ. TIMING CONTROLLED BOTH BY SPEED AND LOAD	- INJ. TIMING CONTROLLED BY SPEED ONLY	- INJ. TIMING CONTROLLED BY SPEED ONLY
		- INLET PRESSURE :	- INLET PRESSURE :	- INLET PRESSURE :
		1,5 kg/cm ² ABS. AT INT. SPEED	1,5 kg/cm ² ABS. AT INT. SPEED	2,0 kg/cm ² ABS. AT INT. SPEED
		2,0 kg/cm ² ABS. AT RATED SPEED	2,0 kg/cm ² ABS. AT RATED SPEED	2,0 kg/cm ² ABS. AT RATED SPEED
INTERMEDIATE SPEED	B.H.P.	160,9	160,3	159,6
	B.S.F.C. (g/BHP.h)	169	168	173
	SMOKE (BOSCH UNIT)	2,8	2,6	3,0
	EX. GAS TEM. (°C)	535	520	455
RATED SPEED	B.H.P.	239,8	242	242
	B.S.F.C. (g/BHP.h)	196	194	194
	SMOKE (BOSCH UNIT)	2,4	2,7	2,7
	EX. GAS TEM. (°C)	540	520	520
13 MODE CALIF. CYCLE	CO (g/BHP.h)	4,99	4,72	4,93
	HC " "	0,68	0,66	0,65
	NO ₂ " "	3,64	3,93	3,58
	HC+NO ₂ " "	4,32	4,59	4,23

Paper for the Fourth International Symposium on Automotive Propulsion
Systems/Low Pollution Power Systems Development.

Washington, D.C. April 17-22, 1977

"Turbocharging the "Squish Lip" Low Pollution Diesel Engine."

by

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Abstract

The oxides of nitrogen emitted by diesel engines can be reduced by retarding injection timing. However, with conventional direct injection engines, this results in serious degradation in smoke and specific fuel consumption. The application of the "Squish Lip" combustion system to a naturally-aspirated engine has been shown to allow significant injection retard without such penalty. It is now confirmed that a similar trend occurs for the "Squish Lip" system applied to engines turbocharged for higher output.

The development is described of a four-stroke, six-cylinder engine of 5.8 litre (354 cu.in.) capacity, turbocharged and charge-cooled, to meet U.S. Federal Heavy Duty Vehicle Emissions Legislation. The engine is based on a new unit for World-wide manufacture, the turbocharged vehicle version being optimised for smoke and fuel consumption. With "Squish Lip" combustion, injection retard of up to 20 degrees crank angle is possible within acceptable smoke limits, reducing NO by 70% over the 13-mode cycle. Lightload advance is required to eliminate misfire and attention to injector nozzle geometry further reduces unburnt hydrocarbons. The

retarded timing reduces maximum cylinder pressure by 40% with benefit to mechanical loading and engine noise.

Introduction

The demands on prime movers, particularly for automotive applications, have never been greater. The requirement for units giving high power from small package size, driveability and reliability has been met by reciprocating engines using gasolene or diesel fuels. The more durable diesel engine became predominant in heavy duty applications, particularly where efficient use of fuel was demanded, in those countries with no national oil resources, notably Europe. With the dramatic increase in World oil prices since 1973 and concern over World oil resources, fuel consumption has become a premium in all countries.

The very success of these engines in automotive applications has led to concern for the environment, particularly in the urban arena where vehicle and human population density is highest. Visual and audio emissions are immediately obvious and vehicle smoke and noise have been the subject of legislation in the developed countries for many years. More recently, attention has been given to the toxic gaseous emissions of unburnt hydrocarbons, oxides of nitrogen and carbon monoxide resulting in stringent legislation primarily in the U.S.A. Some emissions, notably exhaust odour, are not as yet regulated owing to the difficulty in analysing their complex nature and devising appropriate instrumentation.

The Case for the Diesel Engine.

Vehicles today require a low pollution propulsion system with minimum fuel consumption without compromising the performance, package size, reliability and durability. Both gasolene and diesel engines have been demonstrated to comply with exhaust emission limits. However, the thermodynamic advantage of the compression-ignition engine at part load allows the diesel engine to return fuel consumption 25-50% better than the

equivalent spark-ignited gasoline engine. (Ref.1)* The wide spread of results corresponds to the vehicle duty cycle and the percentage of part load running (when the gasoline engine is throttled). The lower figure stems from a high proportion of highway operation and the greater saving from stop-start urban driving such as city taxi-~~cab usage~~.

In this situation, the higher first cost of the diesel engine is quickly repaid. It is also worth noting that a proportion of the cost differential is not endemic to diesel operating characteristics but stems from premium features inherited from its heavy duty ancestry. The diesel engine has a longer life expectancy, which, if adopted across the board for mobile applications, would bring significant savings to national and World material resources. In addition, refining crude oil for diesel engine use optimises oil resources, in that 1.08 gallons of crude are required for every gallon of diesel rather than 1.21 gallons for gasoline (2).

For all reciprocating engines, the overall engine size is closely related to swept volume and so competitive power units may be compared on specific power, kW/litre (HP/cu.in.). The combustion mode adopted dictates the maximum air utilisation and engine speed. The conventional gasoline engine operates at around stoichiometric, unity equivalence ratio, giving an inherent advantage over the diesel operating at a maximum of 0.85 ratio. Despite this differential, the indirect injection diesel, which can match the gasoline engine operating speed range, is finding increasing application in light vans and automobiles replacing gasoline engines. For the intermediate area of medium trucks, the differential on specific power between gasoline and direct injection diesel engines is wider and more problematic, owing to the speed limitation of the conventional D.I. diesel, at about 3000 rev/min. In countries where fuel costs have always been high, the diesel is paramount, although in the U.S.A. gasoline engines

* Numbers in brackets designate References at end of paper.

are still predominant in Class 4-8 trucks (14001-33000 lb. gross vehicle weight). However, the specific power of the naturally-aspirated diesel engine can be enhanced by turbocharging, leading to an attractive alternative to the gasoline engine if low pollution standards can be maintained.

The Direct Injection Diesel Engine

Perkins Engines markets D.I. diesel engines within the power range 30-225 kW (40-300 HP) for vehicle, agricultural, industrial and marine applications World-wide. For light and medium trucks, a family of four- and six-cylinder engines is available, having 0.97 litre/cyl. (59 cu.in/cyl.), 98.4 mm.(3.875in.) bore and 127mm.(5in.) stroke. The six-cylinder engine, 5.8 litre (354 cu.in.), gives 89.5kW (120 HP) at 2800 rev/min in naturally aspirated form, and 116 kW (155 HP) at 2600 rev/min when turbocharged and charge-cooled. (Table 1)

The engines are optimised for minimum fuel consumption and smoke to meet U.K. and European Regulations (3)(4). Toroidal bowls (Figure 1) are used in the pistons to intensify air swirl from directed inlet ports and to promote squish from the minimum piston crown/cylinder head clearance. To cope with the increased loadings, the turbocharged version incorporates premium features such as nitrided crankshaft, larger gudgeon pin, and valve seat inserts. The thermal loading on components is eased by piston jet cooling and charge cooling of the inlet air. The air-to-air charge cooler mounted ahead of the vehicle radiator reduces compressor outlet air temperature to 45°C (113°F) at the inlet manifold for 20°C (68°F) ambient temperature (Fig.2). The charge cooler also acts as an emission control feature in that smoke, NO and fuel consumption are all reduced by the cooler charge air.

Although optimised for smoke, the gaseous emissions as measured on the 13-mode Heavy Duty Vehicle cycle specified by the U.S. Environmental

Protection Agency are unfavourable, apart from the carbon monoxide. The high NO levels stem directly from the smoke optimisation. Soot is formed in diesel combustion (Fig.3) during the diffusion burning phase, increasing with temperature and with the amount of fuel burnt during this phase, but reducing with increase in diffusion burning rate(5). The high swirl D.I. engine utilises an advanced timing giving a relatively long ignition delay and hence a large amount of fuel is introduced into the cylinder during this period. Mixing is rapid owing to the high swirl and squish, and a large proportion of the fuel burns pre-mixed and smoke-free. The violent combustion during pre-mixed burning also generates a high rate of pressure rise and combustion noise. Ignition delay is shorter on the turbocharged engine owing to the raised compression pressure and temperature, and more advance is required to reduce smoke. As expected the high swirl D.I. engines emit high NO, 13-mode mean values being 17.5g/kWh (13 g/BHP.h) for the naturally-aspirated engine and 34 g/kWh (25 g/BHP.h) when turbocharged.

The situation with regard to emission of unburnt hydrocarbons (HC) is reversed with 13-mode values of 3.4 and 3.1 g/kWh. (2.5 and 2.3 g/BHP.h) for naturally-aspirated and turbocharged engines respectively. This is readily understood as the primary cause of HC is from fuel issuing from the injector nozzle sac when the cylinder gas temperature is too low for combustion. With similar sac volumes (0,8 cu.mm.), the turbocharged engine has lower specific HC because of higher brake power. Further, minor advantage is obtained from the higher compression temperature.

The Perkins "Squish Lip" System

At the 2nd. NATO/CCMS Conference on Low Pollution Power Systems in 1974, Bertodo described initial developments of a new low pollution system devised at Perkins Engines termed the "Squish Lip" Combustion System (6). On a three-cylinder engine of 2.5 litre (152 cu.in) capacity, the addition

of re-entrant lips to the piston bowl allowed considerable injection timing retard without degradation in smoke or fuel consumption. With retarded timing, NO emissions drop dramatically. Subsequently, the system has been tested on 4, 6 and 8-cylinder engines (7) varying from 0.8 - 1.3 litre/cylinder (50-80 cu.in./cyl.) with detailed investigation of the controlling parameters (8). It is recorded that the naturally aspirated six-cylinder engine referred to above achieved 13-mode emission levels of 4.91 g/kWh (3.66 g/ HP.h.) NO and 1.82 g/kWh (1.36 g/ HP.h.) HC with peak smoke values less than 5% capacity measured on the USPHS meter. These development engine figures serve to illustrate the ability of the "Squish Lip" system to reduce high swirl D.I. engine emissions under naturally aspirated conditions.

It is apparent that combustion is significantly changed in the "Squish Lip" engine, despite the fact that the fuel injection equipment is not substantially altered and indeed has remarkably little effect compared to experience on toroidal bowl engines. However, the air motion and consequently the mixing of fuel and air is radically changed by the piston bowl shape. The inward radial squish velocity generated in the clearance between piston crown and cylinder head is greatly increased by the extended bowl lips, approximately in proportion to the squish area. Swirl from the directed inlet port is unchanged during induction, but the rotational velocity is increased on transfer of the air to the piston bowl owing to the reduced radius of gyration. With current "Squish lip" designs, swirl velocities at TDC are 10-20% higher than their toroidal counterparts. The nett result is a vigorous rolling toroidal vortex of high turbulent intensity into which fuel is sprayed from the 4-hole nozzle.

With injection retarded back as far as TDC, the ignition delay is extremely short - difficult to measure accurately but certainly less than five degrees crankangle. The quantity of fuel injected during the delay period is

small and the initial pre-mixed peak of the heat release diagram is often difficult to determine. Characteristic of the "Squish Lip" system is the following diffusion burning period (Fig.3) which is very rapid being dependent on the air/fuel mixing rate. It is the high rate of diffusion burning that precludes the soot formation normally associated with such retarded timings in D.I. engines. It is also likely that the vigorous air/fuel mixing retained in the piston bowl contributes to a reduction of the combustion tail (Fig.3) which can contribute to smoke when combustion again becomes kinetic rate-controlled as the temperature reduces later in the cycle.

The fast diffusion burning rate also enables the system to retain efficiency at retarded timings, although some degradation is experienced at the high rotational speeds of vehicle ratings and the very retarded timing associated with very low NO emissions. The retarded timing increases the tendency towards light load misfire and consequent HC emission, particularly with fuel pumps of constant and of injection (CEI) characteristics. A light load advance device (LLA) is required to limit HC. In addition, startability may be affected as the timing at cranking speed is now retarded from optimum, particularly if the conventional speed advance is retained for a wide vehicle engine speed range. The "Squish Lip" system then requires conventional fuel injection equipment but with a complex timing plan, which may demand speed advance, light load advance and start advance.

Further characteristics of the system include a smoother pressure diagram with much reduced rate of pressure rise. Combustion noise is reduced to a minimum, and bare engine noise can be dramatically reduced depending on the ranking of the noise sources. Lower peak pressure reduces mechanical loading and gives potential for lighter, lower friction components. Although mechanical loading on the piston is lower, the bowl

lip geometry leads to higher mechanical and thermal stresses requiring particular attention and often compromise between performance and stress optimum designs for a given piston material.

Turbocharging the "Squish Lip" Engine

As can be seen from the previous discussion, initial "Squish Lip" development has been concentrated on naturally aspirated engines. In any case, the relevance of the "Squish Lip" system to the turbocharged engine which already has a high degree of turbulence is not immediately apparent. Also the very high NO emissions experienced with the toroidal bowl piston implied that considerable derate might be required to bring these under control. However, the attraction of turbocharging with its increased specific power led to development trials with the aim of producing an efficient, low pollution power system as an alternative to the gasoline engine in medium trucks, primarily for the U.S. market. The response of the turbocharged engine to the "Squish Lip" system was so encouraging that development of production specifications was undertaken to comply with current and 1979 U.S. Federal Heavy Duty Vehicle Regulations. A Federal certificate has been obtained for current legislation using a first generation "Squish Lip" system and development is well advanced for 1979 using a second generation system. The target rating was fixed at 104 kW (140 HP) at 2600 rev/min (10% derate from the toroidal specification), but with torque back-up increased from 15 to 25% (Fig. 4) to suit the characteristic of Class 6 trucks at 8850 - 11800 kg (19501 - 26000 lb.) gross vehicle weight.

First Generation "Squish Lip" System

Current U.S. Federal legislation requires the combined emissions of NO and HC not to exceed 21.4 g/kWh (16 g/ HP.h.) on the 13-mode cycle. In addition, limits are placed on the acceleration, peak and lugging smoke achieved on the transient Federal smoke cycle of 20, 50 and 15% opacity

(USPHS meter) respectively. For the production turbocharged engine, reduction of 50% was required in combined NO and HC. Whilst the steady state smoke was comfortably inside the limit, careful attention would be required for the transient operating mode owing to turbocharger lag on acceleration. Action had already been required on the specification to meet European E.C.E. regulations (4) for free acceleration smoke, and a boost control device used to limit fuel delivery below maximum torque speed.

In 1975, production fuel injection equipment incorporating the known requirements of a "Squish Lip" system, namely light load advance and start advance, was not available. It was evident that compromise would have to be made and that with the interim specification it would not be possible to meet in production the lowest emission levels then projected. Direct control of HC would not be possible with unchanged injector nozzle geometry and the full retardation potential of the "Squish Lip" system could not be used without promoting excessive light load misfire and adverse startability.

The piston bowl chosen had the mild re-entrant features shown in Fig. 1 of 67° flank angle and throat/bore diameter ratio of 0.48. The system allowed retardation at rating back to 10° BTDC compared to 25° BTDC for the toroidal specification without impairing smoke control. At this timing, the NO emission dropped from 1750 ppm to 630 ppm, substantially the same reduction that would be expected from retarding the toroidal specification.

The fuel injection pump used was a 1 x 11mm plunger distributor type having constant beginning of injection (CBI) characteristics, and excess fuel for starting. Startability was retained by deleting the normal speed advance of 8° crank angle, and this deletion together with CBI prevented light load misfire. The resultant HC emission over the 13-mode cycle was unchanged at 3.1 g/kWh (2.3 g/ HP.h.). NO emissions were down to 10.6 g/kWh (7.9 g/ HP.h.), a reduction of 70% from the toroidal

specification. Compression ratio had only minor effects within the range 15-18:1 and of the nozzle variables only protrusion had significant effect. Final values of 16:1 compression ratio and increased protrusion of 2.5 mm were used.

The desired increase in torque back-up reduced the lugging smoke margin at maximum torque speed, and the turbocharger match was reconsidered to give more boost at lower speeds, turbine area reducing from 14.5 cm^2 (2.24 in^2) to 10.3 cm^2 (1.6 in^2). Further gains in boost were obtained by the adoption of a new single-pass charge cooler design, styled for Class 6 trucks, reducing the maximum pressure drop from 25 kN/m^2 (3.5 lbf/in^2) to less than 7 kN/m^2 (1.0 lbf/in^2).

With the turbocharger matched lower down the speed range, inevitably the pumping losses at full speed are increased and some fuel consumption penalty is experienced at rating. This was partially offset by the inclusion of the central pip in the piston bowl eliminating a relatively stagnant area, the overall penalty becoming 3% at rating (Fig.4), but very small in actual truck operation. In comparative tests in a Claas 6 truck, the "Squish Lip" engine returned 34% better fuel consumption than an equivalent V8 gasoline engine, as detailed in Table II. Acceleration and hill climbing were also improved.

Peak pressure loading on the engine is reduced from 13.4 MN/m^2 (1950 lbf/in^2) to 11.4 MN/m^2 (1650 lbf/in^2), Fig.5. Following finite element calculations, the piston was unchanged apart from the bowl modifications and no lip cracking problems have been experienced. The performance of the ring pack is unaltered, the three rings above gudgeon pin giving an oil consumption at rated speed of 0.4-0.5% of fuel.

The resulting engine specification gives moderate emissions (Table III) with little penalty in truck performance or economy. There was no increase in component complexity from the production specification

optimised solely for smoke and fuel economy. However, it was apparent that this situation could not be maintained as exhaust emission legislation tightened. A second generation "Squish Lip" system would be required with considerable increase in complexity, particularly in fuel injection equipment.

Second Generation "Squish Lip" System

For 1979 model year, U.S. Federal legislation tightens to 13.4 g/kWh (10 g/ HP.h) combined NO + HC of which HC must not exceed 2.01 g/kWh (1.5 g/ HP.h.) Peak acceleration smoke reduces from 50 to 35% opacity. The results from the first generation specification indicate that specific controls are required for HC and further retardation to provide production margin for NO emission.

It was already known from naturally aspirated engine tests that further retardation maintaining smoke control was possible by decreasing flank angle and throat/bore diameter ratio (8) effectively increasing the air/fuel mixing rate. Initial development was undertaken on a 40° bowl offset 4.4 mm. (0.175 in.) with throat/bore diameter ratio 0.36. The parallel programme of piston stressing and proving described below showed that a 50° central bowl was advantageous for lip stresses and this compromise was adopted (Fig.1).

At rating, optimum dynamic timing for economy was 9° BTDC but further NO reduction was possible by retarding back to TDC maintaining smoke control. Despite a remarkable retardation of more than 20° crank angle from the toroidal bowl timing the smoke level is still below 5% opacity (Fig.6). It can be seen that there is increasing fuel consumption penalty for low NO particularly at the highest speed, 2600 rev/min shown. A central pip to the bowl had little effect in this case. However, the "Squish Lip" piston has effectively solved the rival demands of low smoke and low NO.

By 1976, improved turbochargers and fuel injection equipment were available. The new turbocharger with higher compressor efficiency allowed relaxation of turbine area to 12.5 cm^2 (1.93 in^2). With improved boost control on the fuel pump and attention to fuel delivery levels, acceleration smoke has been reduced to 10-12% opacity with peaks of 25%. The fuel pump itself retained the same pumping action and CBI characteristic, but was now available with mechanical governor, and facility for light load advance and start advance. Injection rate was increased by using a 12 mm. plunger.

The control of HC was approached in two complementary ways : optimising injector geometry and operation; and devising a timing plan for the fuel pump. Injector optimisation will aim for a clean, sustained injection, free from dribble or secondary injection with a minimum of fuel left in the nozzle sac. The injectors chosen had low inertia needles, and an opening pressure of 250 bar (3600 lbf/in^2) was used to give sharp cut-off and maintain a high injection rate. Due to the crucial importance of nozzle geometry, considerable work was carried out with sacless nozzles (Fig.7) but because the volume of the holes is still high, comparable results could be obtained with conventional nozzles at a reduced sac volume of 0.51 mm^3 . The latter were chosen owing to the unproved durability of the sacless nozzle.

The derivation of an optimum timing plan demands painstaking exploration of the whole load-speed spectrum at a succession of injection timings. A theoretical timing plan can then be put together manually or by computer to be married up with the fuel pump capability in terms of speed advance and light load advance. The resulting compromise must give the final plan (Fig.8) consistently with minimum scatter. By these two control methods, development HC results can be as low as 1.07 g/kWh (0.8 g/HP.h) on the 13-mode cycle, although further compromise for NO leads to

a realistic production mean of about 1.5 g/kWh (1.1 g/HP.h.). The equivalent mean for NO emission using the timing plan is 8.7 g/kWh (6.5 g/HP.h.), as shown in Table II.

The specification is again relatively insensitive to compression ratio and nozzle variables (excepting sac volume and protrusion). The major influencing variable is injection timing, but development well inside the legislative limits allows normal static timing ($\pm 2^\circ$ crank angle) to be retained, avoiding the complications in production and service of dynamic timing. The sensitivity to diesel fuel quality and Cetane Number is less than the toroidal specification, and exhaust odour is lessened by the reduction in unburnt HC. Further reduction in diesel odour will require major breakthroughs in the understanding of the chemistry and measurement techniques of the many trace compounds involved (9).

Mechanical Loading

The retardation has dramatic effect on the peak cylinder pressure (Fig.5). The toroidal maximum pressure of 13.4 MN/m² (1950 lbf/in²) is reduced to 8.3 MN/m² (1200 lbf/in²). This is lower than that of the naturally aspirated engine, consequently the peak loading is reduced on all drive-train components leading to increased safety factors on piston boss/gudgeon pin, connecting rod, bearings, crankshaft, damper and gears.

It can be seen from the pressure diagram in Figure 5 that a much smoother cycle results from the combustion start after TDC. The maximum rate of pressure rise reduces from 1000 kN/m² per $^\circ$ CA (150 lbf/in²) to 300 kN/m² per $^\circ$ CA (45 lbf/in²) giving significant reduction in combustion noise. The effect on bare engine noise levels depends on the ranking of combustion

to mechanical and other noise sources, and the measures taken to reduce transmission through, and radiation from the engine structure. In the present case, bare engine noise attenuation at 1 m. under anechoic conditions was 3 dBA at 2600 rev/min, 7 dBA at 2000 rev/min, and 8 dBA at 1000 rev/min from the conventional engine.

Piston Integrity

The thermal loading on the engine components is generally lower, apart from the piston bowl lips. (The turbocharged engine does not experience increased cylinder head temperatures nor valve sinkage as has been found on naturally aspirated "Squish Lip" engines owing to the improved air/fuel ratio). The piston bowl lip is critical owing to the combination of thermal and mechanical stresses (Fig. 9) peaking just under the lip on the thrust-non-thrust axis.

Two- and three-dimensional finite element temperature and stress calculations followed by piston fatigue life predictions allowed optimisation of the piston geometry. Adoption of the 50° flank angle central bowl limits the maximum temperature to 325°C and the maximum thermal stress to 46 MN/m^2 (3 ton/in^2). Mechanical stresses are also reduced by the central bowl, and the bowl undercrown was thickened to reduce the tendency of the piston to bend about the gudgeon pin promoting tensile stress at the lip on the gudgeon pin axis and compressive stress on the thrust-non thrust axis. Computer predictions of piston life showed that a monolithic piston in conventional aluminium alloy would be acceptable, using a 0.0017% failure rate to first visible crack from 100,000 miles of urban driving as the acceptance criterion.

The predictions have been backed up by conventional test bed and vehicle endurance programmes together with thermal cycling rig tests and accelerated engine tests specifically for piston bowl cracking. The

accelerated tests are conducted on a special test bed shown in Fig.10 on which the six-cylinder engine fitted with the "Squish Lip" system alternatively drives and is driven by the V8 engine. The test engine runs at rated speed, fuel delivery alternating between maximum and zero on a 5 second/5 second cycle. The V8 engine alternatively motors the test engine and absorbs the power using an exhaust brake. In 750 hours of cycling, the pistons experience more thermal load cycles than would be achieved in a truck engine in 100,000 miles of urban driving. Because of turbocharger lag following the rapid fuel increase, the test cycling conditions are about three times more severe on piston fatigue life than cycles up to the steady state loading at rating. Piston designs and materials can be evaluated and crack propagation studied in a matter of weeks.

Conclusions

1. The "Squish Lip" system can be applied successfully to turbocharged high swirl direct injection engines.
2. The turbocharged "Squish Lip" diesel presents a low pollution power system of comparable specific power to the gasoline engine but with superior fuel economy.
3. Comparative tests against the original gasoline engine in a U.S. medium truck showed 34% better fuel consumption and improved acceleration and hill-climbing.
4. U.S. Federal Heavy Duty Vehicle emissions limits can be met in development and in production.
5. Injection retard of more than 20° crank angle reduces NO emissions by 75%.
6. Peak cylinder pressure is reduced by 40%, and rate of pressure rise by 70%, giving engine noise attenuation of 3-8 dBA over the engine speed range.
7. A complex timing plan is required to limit HC emission. The fuel injection pump requires speed advance, light load advance, start

advance and boost control.

8. Piston bowl geometry requires optimisation for performance and stress to achieve a durable, economic design.

Future Prospects

1. Significant economy improvement is expected if a limited speed range is adopted for the engine with increased torque. Further gains are likely if injection rate can be increased and when the air motion and air/fuel mixing are fully understood from theoretical and practical studies.
2. Higher ratings, normally limited on turbocharged engines by peak pressure limitations, are possible for the "Squish Lip" diesel, but increases in piston bowl lip temperatures and stresses may require a change from a monolithic aluminium alloy piston to an inserted-lip design.
3. For lower NO emission levels, dynamic injection timing will be required in production and service. Further retardation potential is available and exhaust gas recirculation may be applied, both at the expense of fuel economy.
4. The minimum level of HC emission that can be met in production from a bare engine are being rapidly approached. Severe penalties would result from the addition of downstream reactors/convertors.
5. There is unlikely to be further scope for reduction of combustion noise at source, and attention must be directed to the vibration transmission paths and noise radiation from the engine structure.
6. Exhaust odour will no longer reduce with further reduction in HC, Further research into the chemistry and measurement of the trace compounds involved is required, before odour can be controlled.
7. It is clear that the socio-economic limits of vehicle pollution control are at hand. The feasibility of volume production of power systems to meet the emission levels proposed for the next decade is

in doubt. However, there can be no doubt that the penalties involved, in terms of limited fuel, material, and human resources, will be large. A reappraisal of society's requirement for low pollution power systems seems overdue in relation to national and World resources.

Acknowledgements

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TABLE I

Author - C. Goodwin

HIGH SPEED D.I. ENGINES OPTIMISED FOR SMOKE AND FUELCONSUMPTION

	NATURALLY ASPIRATED	TURBOCHARGED AND CHARGE-COOLED
No. of Cylinders	6	
Swept Volume litre/cu.in.	5.8 (354)	
Bore mm. (in.)	98.4 (3.875)	
Stroke mm. (in.)	127 (5)	
Combustion System	Toroidal	
Rating kW (HP) @ rev/min.	89.5(120) @ 2800	116(155) @ 2600
Fuel Pump	Rotary CEI 2x9mm Plungers	Rotary CBI 1x11mm. Plunger
Peak Pressure kN/m^2 (lbf/in ²)	10(1450)	13.4 (1950)
NO	17.4 (13)	34 (25.2)
CO	13-mode g/kWh	13.7 (10.2)
HC	(g/ HP.h.)	3.1 (2.3)
NO + HC	20.8 (15.5)	36.9 (27.5)

TABLE II

Author - C. Goodwin

"SQUISH LIP" AND GASOLINE ENGINE TRUCK PERFORMANCE

ENGINE	ACCELERATION TIME (SEC)		MAX. SPEED Km/h (MPH)	FUEL CONSUMPTION litre/100 Km (M/U.S. GAL)
	0-48 Km/h (0-30 MPH)	0-96 Km/h (0-60 MPH)		
V8 GASOLINE 5.4 litre (330 cu.in.)	19.6	96.7	103 (64)	42.0 (5.6)
T6 "SQUISH LIP" DIESEL 5.8 litre (354 cu.in.)	11.6	86.7	103 (64)	27.6 (8.5)

FORD F-600 Truck

9800 Kg (21,600 lb) gross vehicle weight

ROUTE FARMINGTON - BAY CITY - FARMINGTON, MICHIGAN

TABLE III

Author - G. Goodwin

TURBOCHARGED ENGINE EMISSIONS

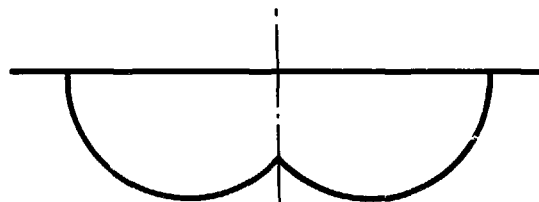
	NO _x	CO	HC	NO _x + HC
TOROIDAL BOWL	34 (25.2)	13.7(10.2)	3.1(2.3)	36.9(27.5)
1st GENERATION "SQUISH LIP"	10.6(7.9)	4.4(3.3)	3.1(2.3)	13.7(10.2)
2nd GENERATION "SQUISH LIP"	8.7(6.5)	3.9(2.9)	1.5(1.1)	10.2(7.6)

13-mode cycle results - g/kWh (g/ HP.h)

CAPTIONS FOR ILLUSTRATIONS

- Fig.1 Piston Bowl Profiles
- Fig.2 Charge Cooler System
- Fig.3 Heat Release in Diesel Combustion
- Fig.4 Turbocharged Engine Power Curves
- Fig.5 Turbocharged Engine Cylinder Pressure
- Fig.6 Effect of Timing on "Squish Lip" Performance
- Fig.7 Reduced Sac and Sacless Nozzles
- Fig.8 Fuel Pump Timing Plan
- Fig.9 "Squish Lip" Piston Temperature and Stresses
- Fig.10 Accelerated Engine Cycling Tests

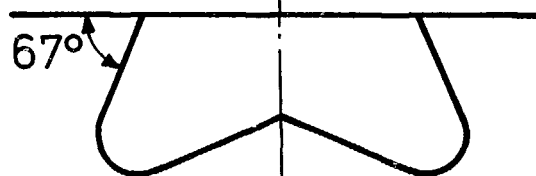
TOROIDAL
BOWL



THROAT DIA.
BORE DIA.

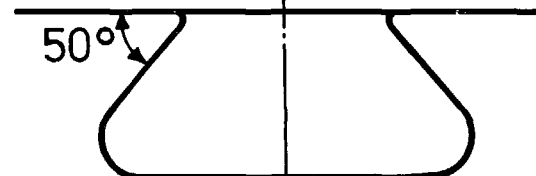
0.68

1st. GENERATION
"SQUISH LIP"



0.48

2nd. GENERATION
"SQUISH LIP"



0.36

Fig.1 Piston Bowl Profiles

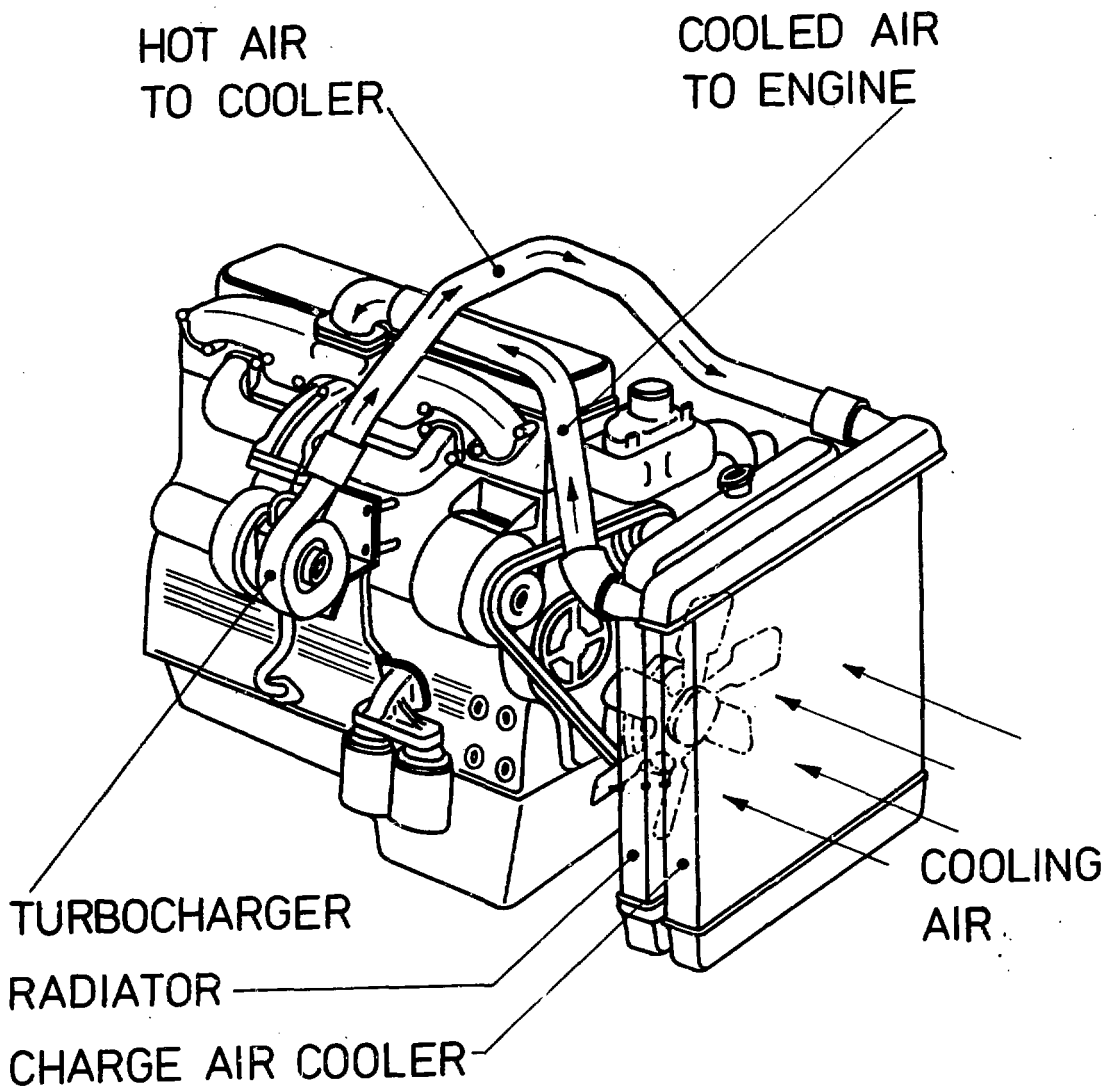


Fig.2 Charge Cooler System

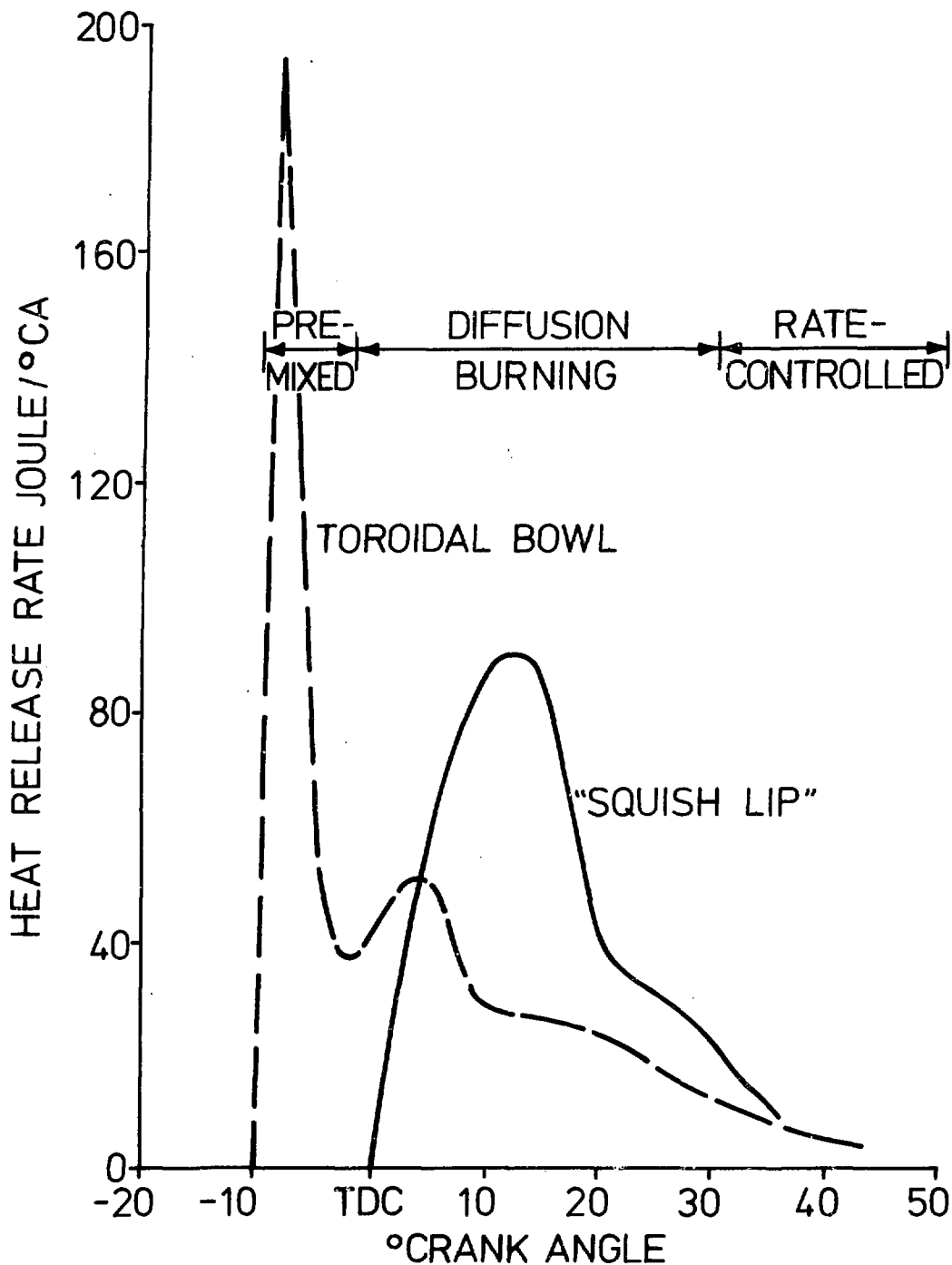


Fig.3 Heat Release in Diesel Combustion

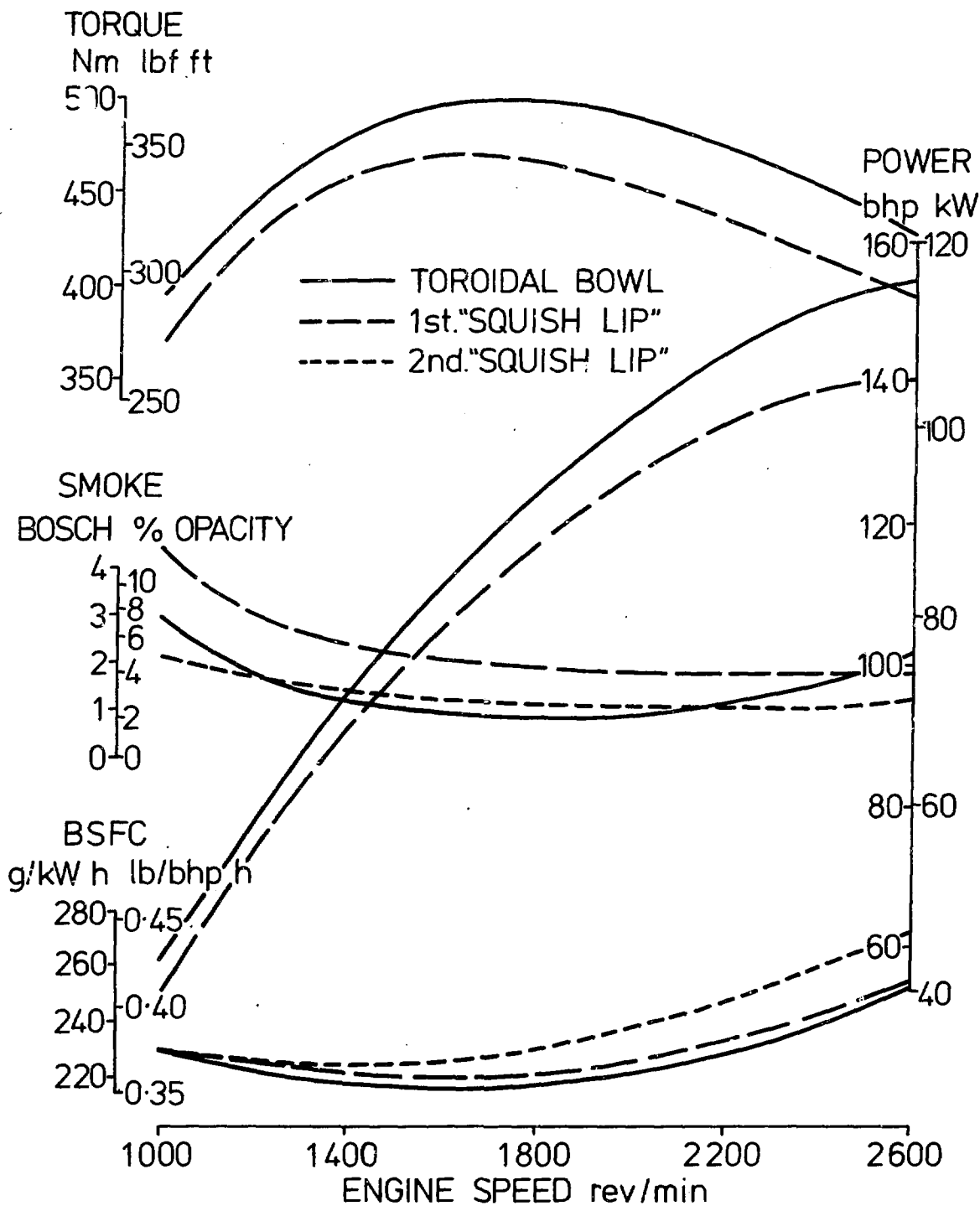


Fig.4 Turbocharged Engine Power Curves

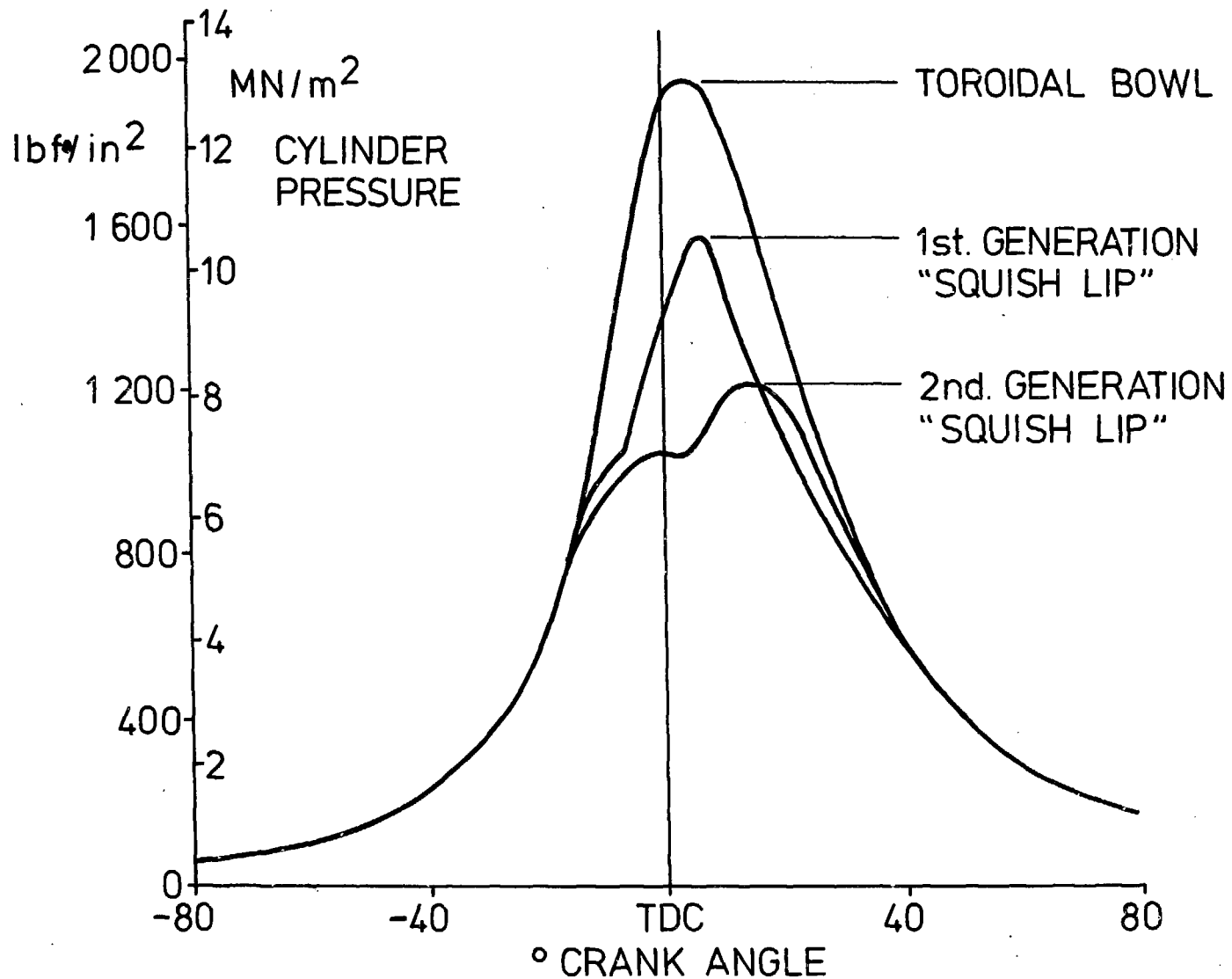


Fig.5 Turbocharged Engine Cylinder Pressure

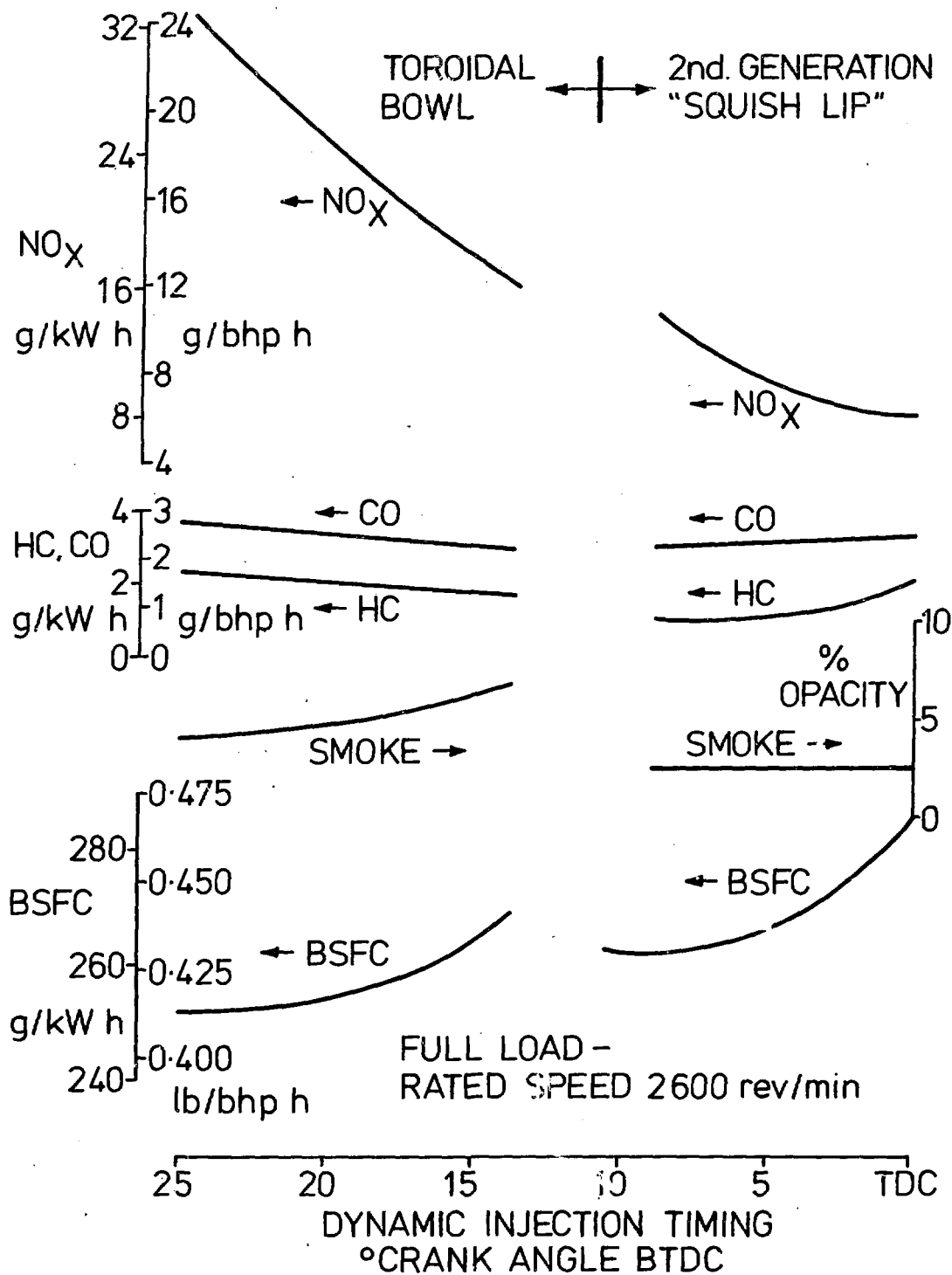
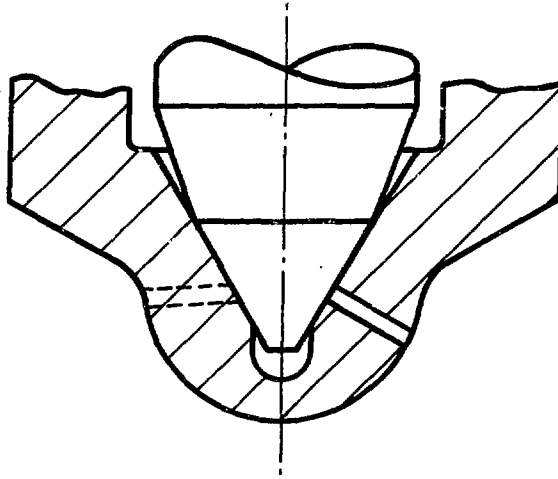


Fig.6 Effect of Timing on "Squish Lip" Performance

SAC LESS



REDUCED SAC

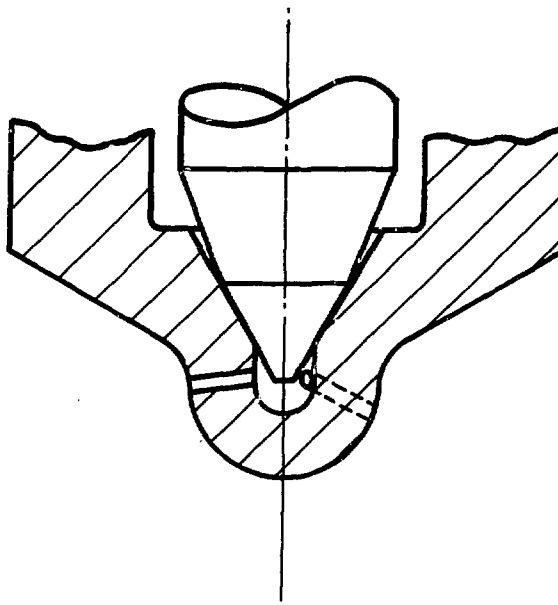


Fig.7 Reduced Sac and Sacless Nozzles

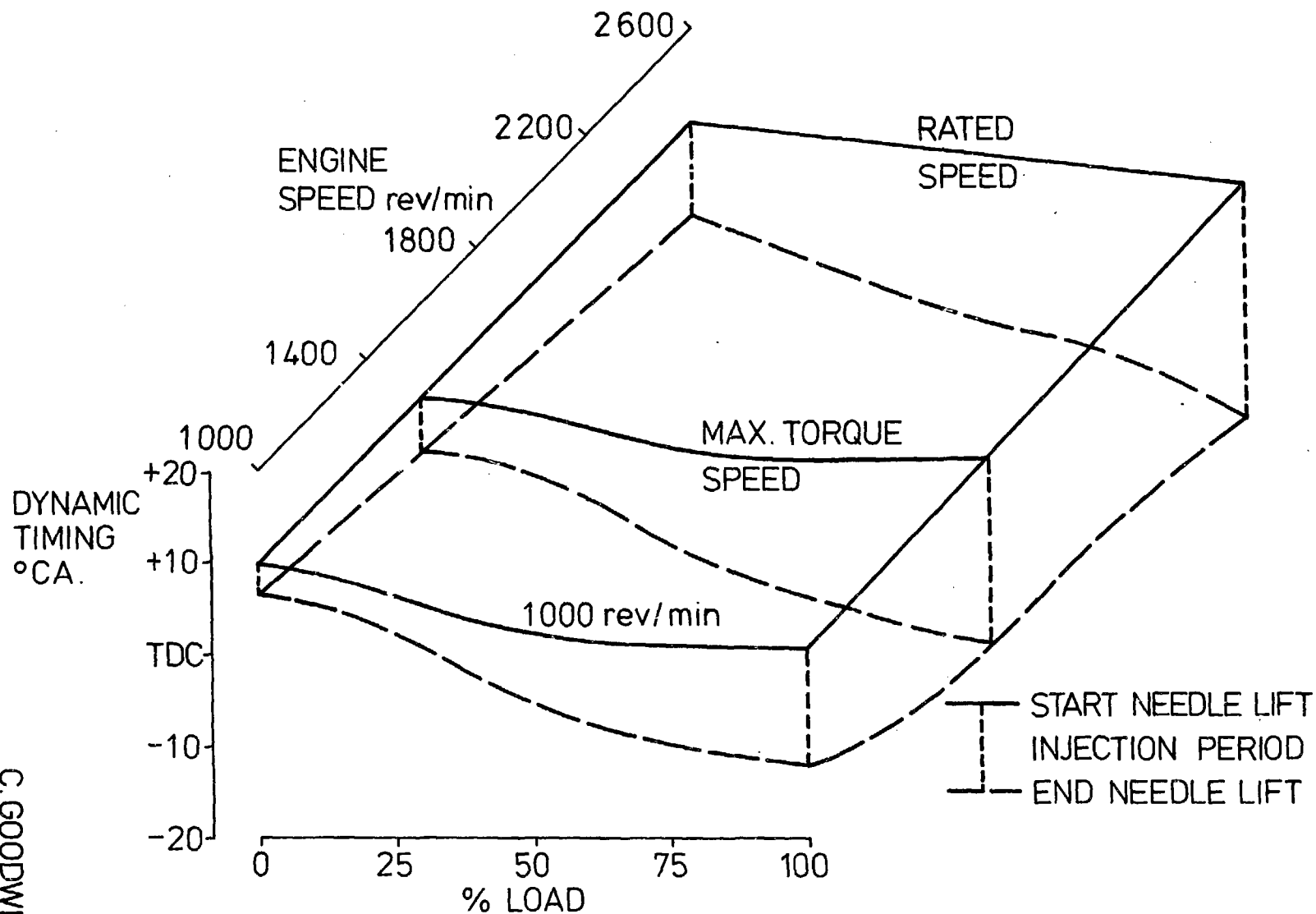


Fig.8 Fuel Pump Timing Plan

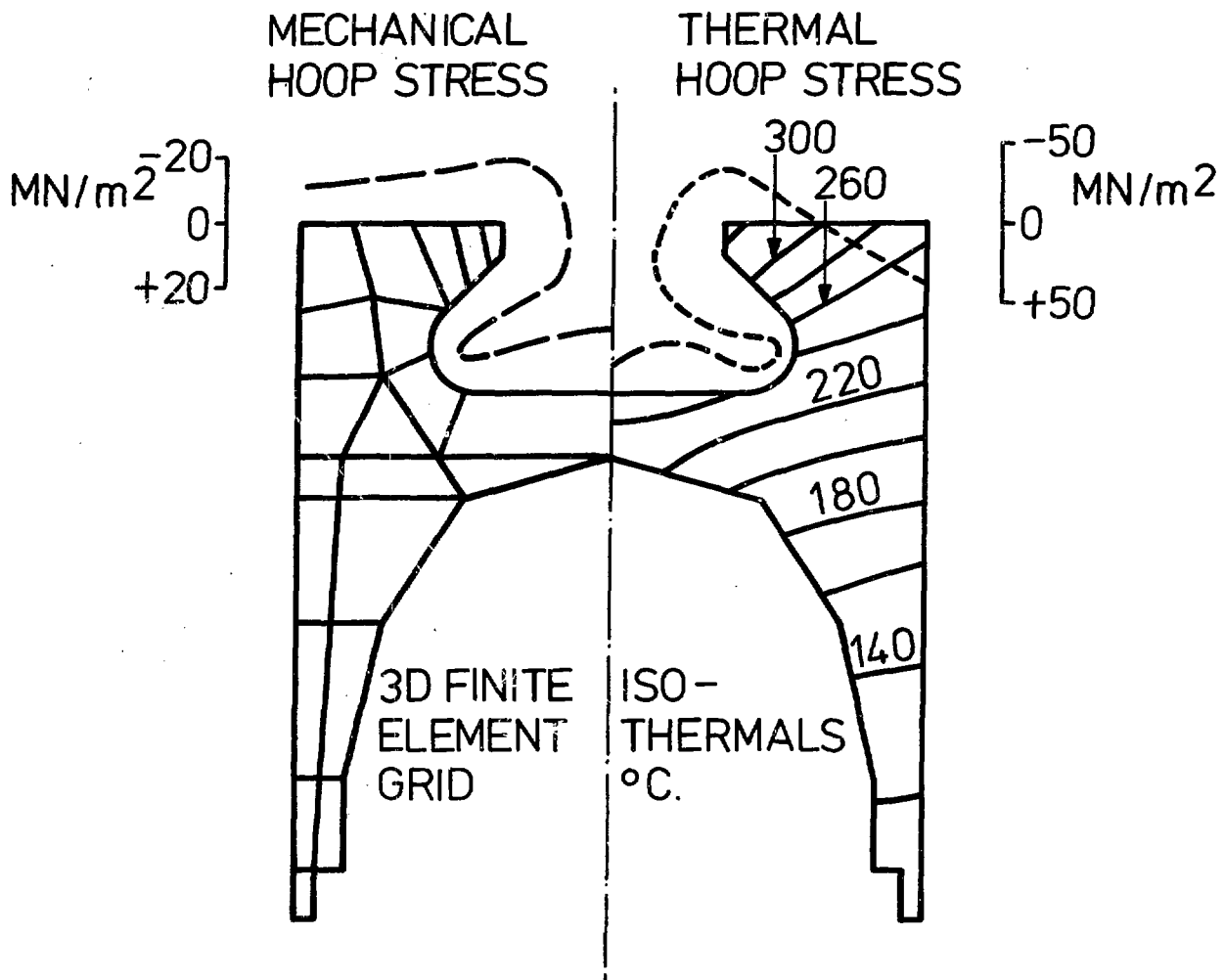


Fig.9 "Squish Lip" Piston Temperature and Stresses

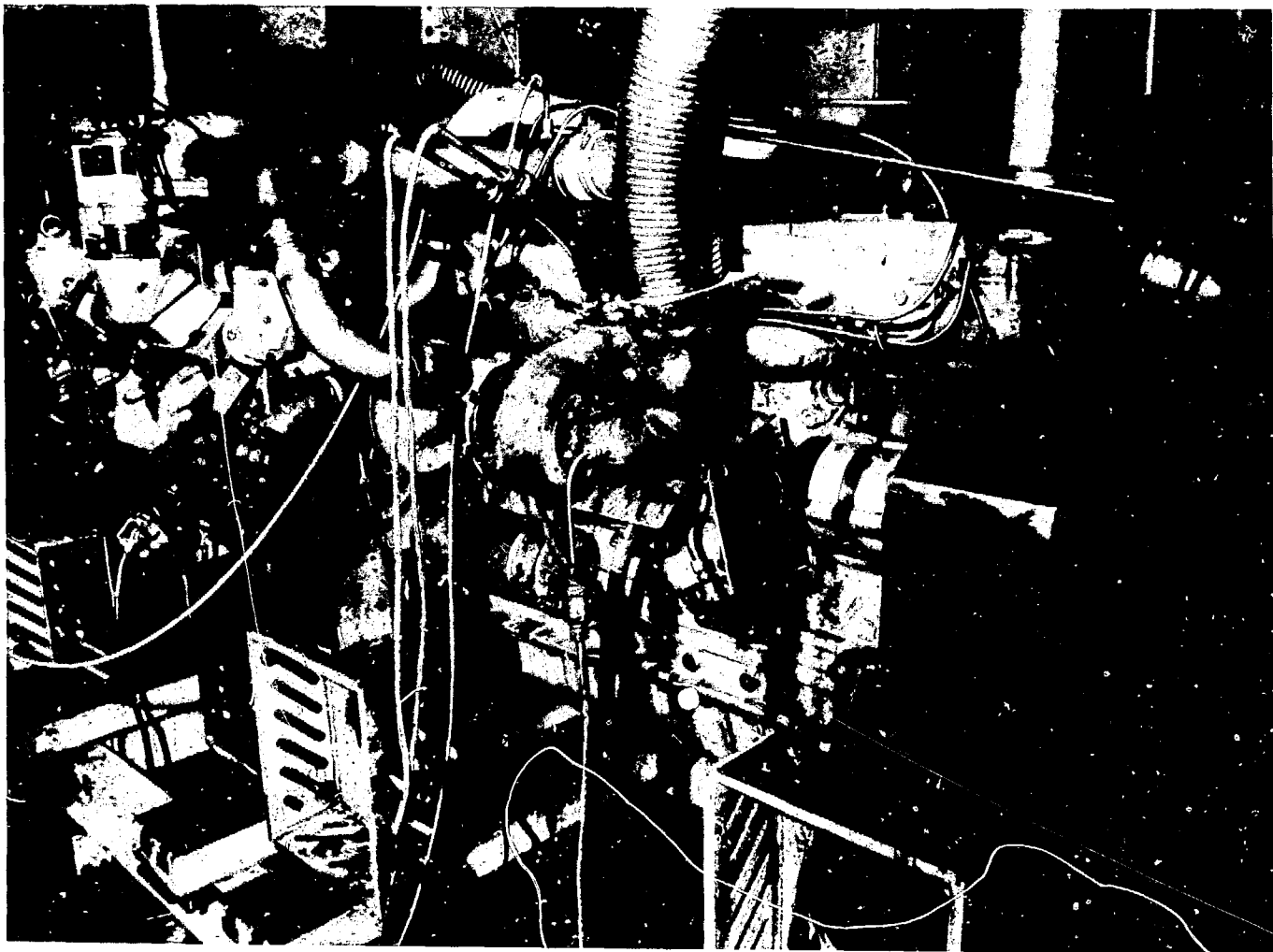


Fig.10 Accelerated Engine Cycling Tests

A FLASH RADIOGRAPHIC TECHNIQUE APPLIED TO
FUEL INJECTOR SPRAYS

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LOW POLLUTION POWER SYSTEMS DEVELOPMENT

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Abstract

A flash radiographic technique, using 50 ns exposure times, has been used to study the pattern and density distribution of a fuel injector spray. The experimental apparatus and method are described. An 85 kVp flash X-ray generator, designed and fabricated at the Lawrence Livermore Laboratory, is utilized. Radiographic images, recorded on standard X-ray films, are digitized and computer processed.

INTRODUCTION

The motion of a spray of fuel droplets is an important area in stratified charge, diesel, and gas turbine engine design. Sprays of interest for these combustors are often thick. (In a thick spray the processes of vaporization, drag, nucleation and break up of a drop are strongly coupled to those of neighboring drops. In a thin spray the interactions between neighboring drops are much weaker). In many cases optical techniques cannot be used to study thick sprays because the spray is optically dense or the environment is thermally non-uniform. Some of the limitations of optical techniques can be overcome by moving up the electromagnetic spectrum and using soft x-rays as the probing medium. X-rays can easily penetrate the most dense sprays and are not affected by thermal gradients. Submicrosecond exposure times are available that can "stop the motion" of high velocity sprays.

Experimental Apparatus

The flash x-ray source chosen for this study was designed and fabricated at the Lawrence Livermore Laboratory. The generator is a triaxial transmission line similar in circuit design to the voltage vector inversion pulseders of Blumlein¹. The generator consists of three basic parts: a transmission line, a low-inductance detonator switch at one end, and an evacuated diode at the other end. A peak cathode current of 37 kA and a peak anode voltage of 85 kV have been measured. The current pulse width is 50 ns (FWHM). The x-ray spectrum consists of the K lines of copper superimposed on a bremsstrahlung background. The effective source spot size is about 1.1 mm in diameter.

A simple diagram of the apparatus used in this experiment is shown in Figure 1. The diode tip of the x-ray source is labeled *a* in Figure 1. A 25.4 μm (0.001 in) thick Mylar window, labeled *b*, is used to separate the vacuum surrounding the x-ray diode from the experimental chamber. (The vacuum chamber is necessary to minimize x-ray absorption.) The fuel injector nozzle, labeled *c*, injects into air at standard temperature and pressure. A film cassette, labeled *d*, is made from a standard cardboard film holder whose front is cut out and replaced with black polyethylene 50.8 μm (0.002 in) in thickness. The film used is Kodak Type AA*. No screens are used with the film in order to enhance the response to low energy x-rays. The source to object distance is 838 mm and the object to film distance is typically 19 mm.

* Reference to a company or product name does not imply approval or recommendation of the product by the University of California or the U. S. Energy Research and Development Administration to the exclusion of others that may be suitable.

The fuel injector used for these measurements is a shower head type with a fixed area pintle and is designed for a nominal opening pressure of 2.6 MPa (370 psi). Fuel is supplied to the nozzles from a ROOSA MASTER jerk pump. The jerk pump is turned by a 1/3 hp dc motor rotating at 1000 rpm. A magnetic pickup on the shaft between the motor and pump is used to time the x-ray source.

A stainless steel tube connected to an air-driven ram collects the spray pulses until the picture is to be taken. The ram is pulled just prior to the spray pulse that is to be photographed. This apparatus is used so that prior sprays do not wet the x-ray film. It also assures that a combustible mixture does not build up in the experimental vault (safety precaution).

Experimental Method

The geometric resolution of the apparatus is related to spot size by the relation

$$R = sd/\ell$$

where R = geometric resolution (mm)

ℓ = source to object distance (mm)

s = object to image distance (mm) and

d = spot size (mm).

For these experiments R is typically 0.03 mm.

The liquid fuel used in these experiments is a 75% diesel - 25% trichloroethylene mixture. This liquid is chosen because it has a high x-ray absorption cross section. The trichloroethylene content was varied from 10 to 25 percent and this does not affect the spray properties. An alternate procedure is to use a leaded or iodized fuel, such as a one percent solution of iodine in gasoline or diesel fuel. Preliminary tests show that this fuel yields excellent radiographic pictures (but is more hazardous than the trichloroethylene mixture).

One radiograph may be taken per spray pulse. The timing of this radiograph is varied so that a series of spray pictures ordered in time, is obtained.

Experimental Results

Figure 2 shows a magnified copy of a typical flash radiograph. (Some of the sharpness of the original radiograph is lost in the copy.) The picture is taken 0.1 ms after the pintle opens. Several spray streams can be seen emerging from the nozzle.

The original radiograph is scanned by a PDS Scanning Microdensitometer using a 50 μm square collimated light beam. The specular density versus x-y position is digitized and processed on a CDC 7600 computer with a program called TIXY. This program produces plots and data files of film density as a function of position. The user controls contour levels and displayed area. Each output scan is a close approximation to a single scan of the original radiograph.

For comparison with calculations one may not need all the information in the original radiograph. In this case the digitized data may be Fourier analyzed to remove the high frequency structure in the data. The resulting data may then be processed with TIXY. Figure 3 is a plot made by TIXY with the smoothed data revealing contours of constant film density. In Figure 3 there are 5 equally-spaced contour levels.

A "scan rectangle" is also shown in Figure 3. The film density in this scan rectangle is averaged in the y-direction and plotted versus the scan direction (x-direction); see Figure 4. The width and orientation of the scan rectangle are chosen by the user.

In Figure 4, regions of low film exposure correspond to thick spray regions. The exact film calibration is obtained by exposing droplets of known size to the x-ray beam. In this way the film density may be related to fuel density; this calibration (shown in Figure 5) may be used to interpret the data in Figures 3 and 4. Recall that the x-ray beam is absorbed by fuel that is in either the liquid or vapor state. For calibration purposes we relate the film density to the equivalent thickness of liquid fuel that would cause the same absorption. This technique does not distinguish between liquid and vapor fuel.

Conclusions

Flash radiography can yield information about the density, pattern, and droplet distribution in sprays. In many cases radiographs yield data which cannot be obtained with optical techniques because the spray is optically dense or the environment is thermally non-uniform. Sub-microsecond exposure times that can easily stop the spray motion are available.

Figure Captions

- Figure 1. Experimental arrangement: (a) Flash x-ray generator, (b) 25.4 μm mylar window, (c) fuel injector nozzle, (d) film cassette.
- Figure 2. Magnified copy of a typical flash radiograph. The spray from a pintle nozzle is shown .1 ms after the pintle opens.
- Figure 3. Isodensity plot of computer processed radiograph. Contours of constant film density are plotted. Five equally-spaced levels are shown in this plot.
- Figure 4. Density scan of computer processed radiograph. The film density in the scan rectangle (shown in Figure 3) is plotted. Film density is directly related to the spray thickness.
- Figure 5. The x-ray film density is related to the thickness of a liquid fuel column that would cause the same absorption. This calibration may be used to interpret Figures 3 and 4.

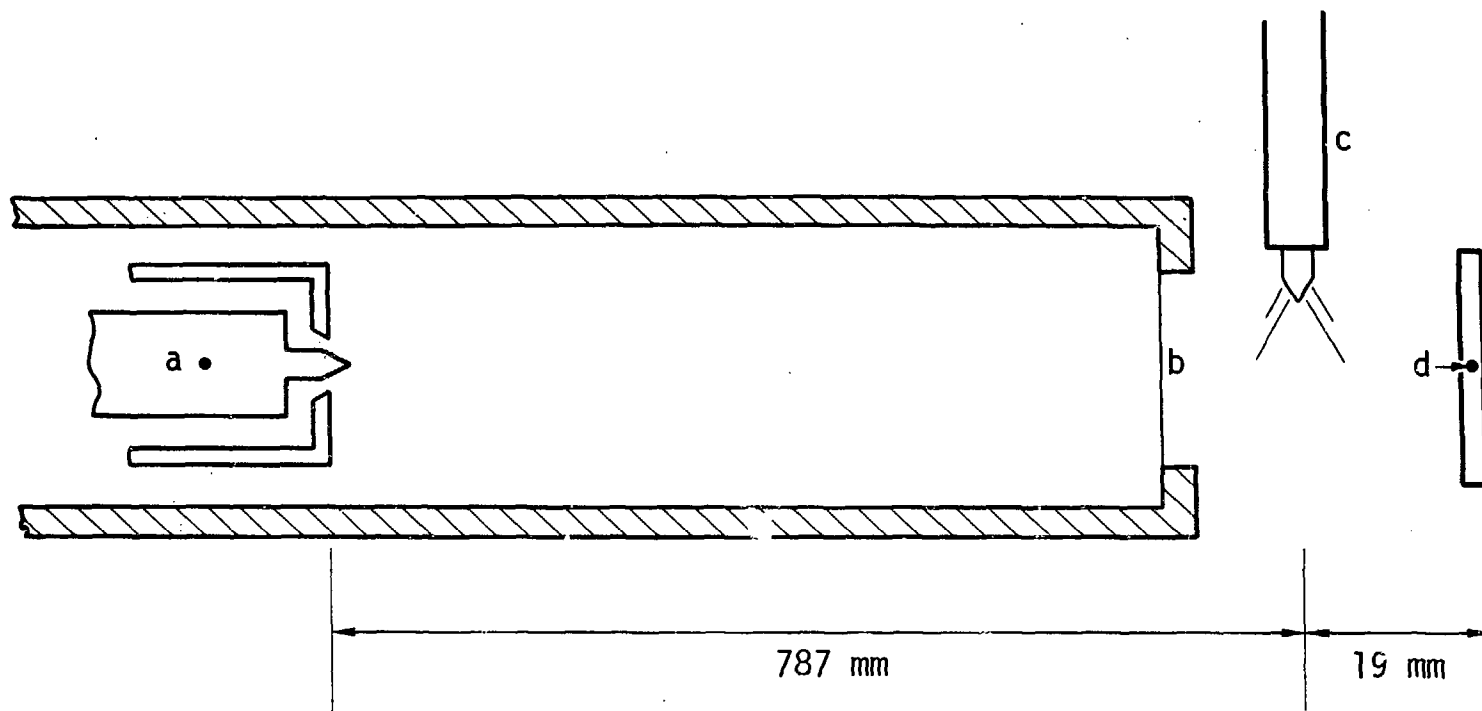
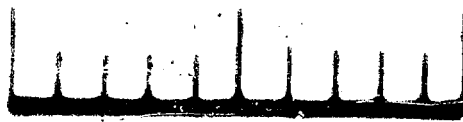


Figure 1. Experimental arrangement: (a) Flash x-ray generator, (b) 25.4 μm mylar window, (c) fuel injector nozzle, (d) film cassette.



1 cm



Figure 2. Magnified copy of a typical flash radiograph. The spray from a pintle nozzle is shown .1 ms after the pintle opens.

IDENT: PINTLE-1-18-77 #1

FILE: PINTLE#1F

401 ROWS

DX= 50 MICRONS

281 COLUMNS

DY= 50 MICRONS

112691 POINTS

SCAN RECTANGLE:

INITIAL ROW: 120

INITIAL COLUMN: 30

FINAL ROW: 125

FINAL COLUMN: 274

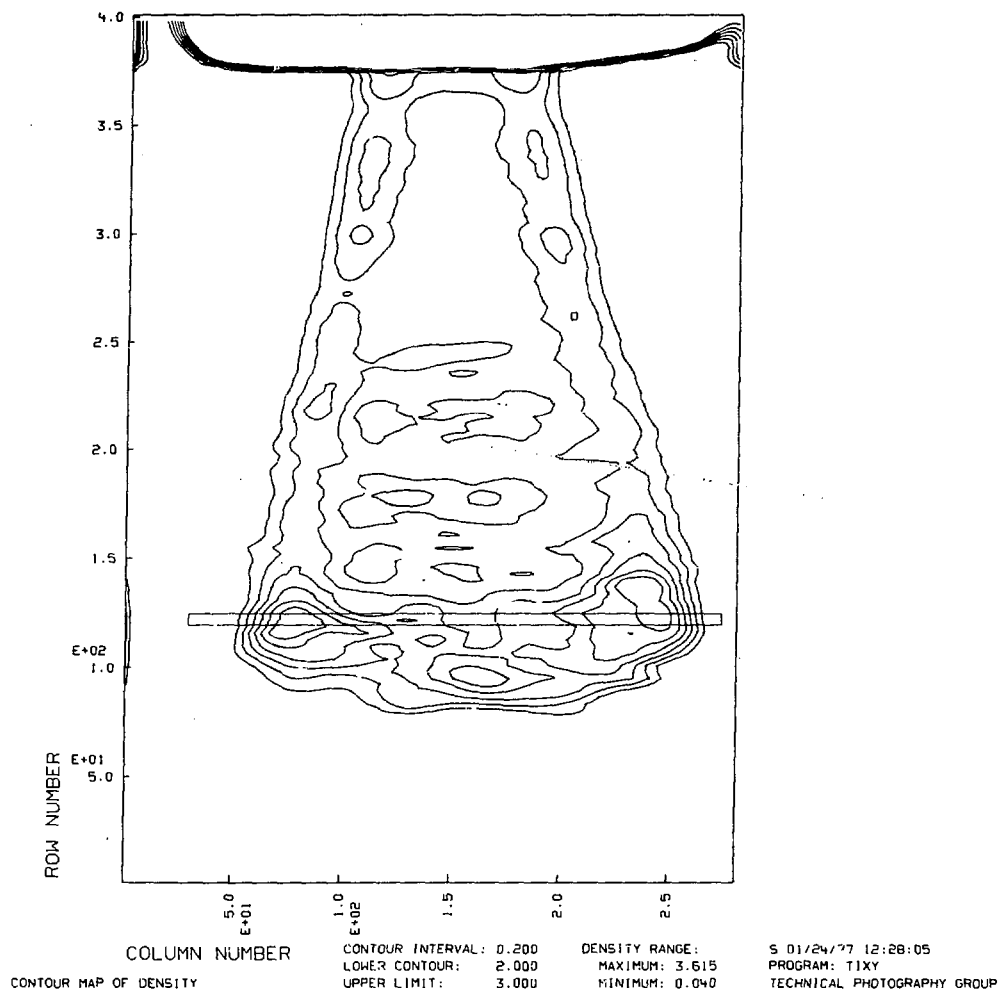
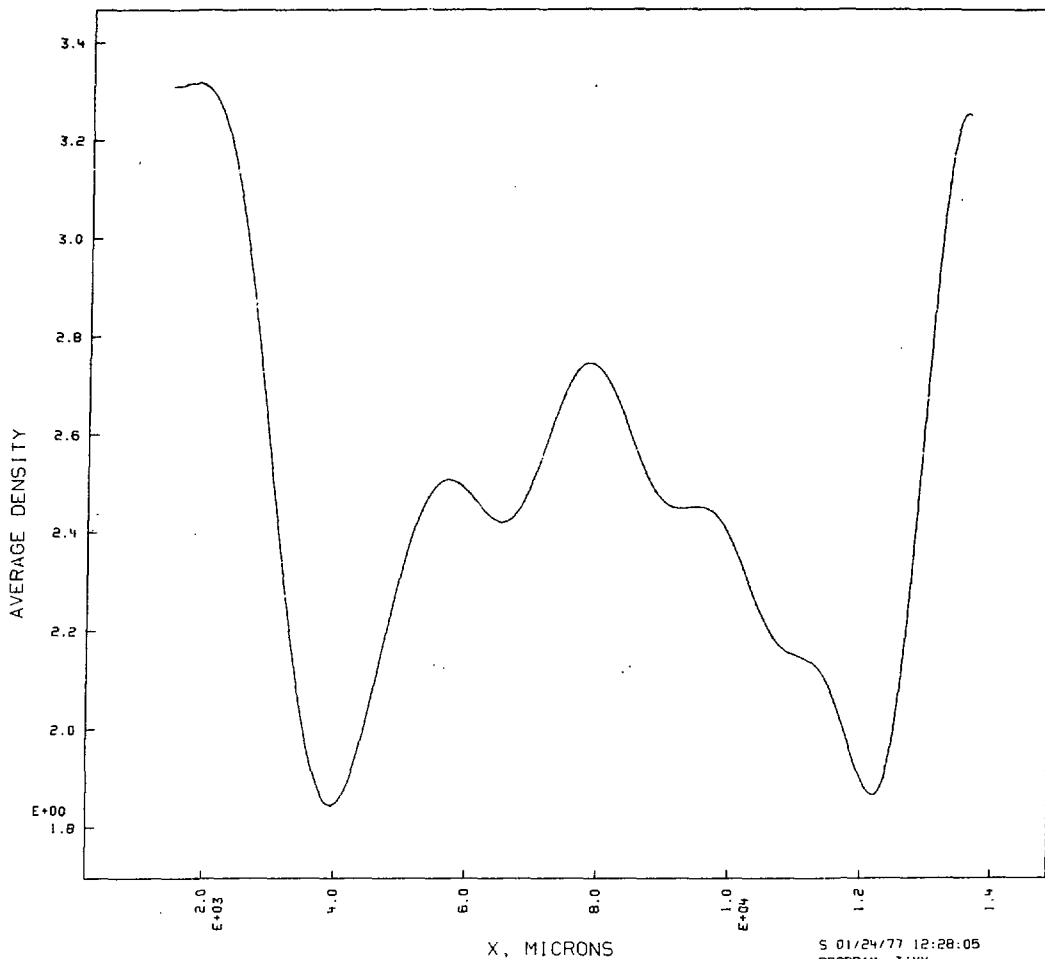


Figure 3. Isodensity plot of computer processed radiograph. Contours of constant film density are plotted. Five equally-spaced levels are shown in this plot.

IDENT: PINTLE-1-18-77 #1

FILE: PINTLE-1F 401 ROWS DX= 50 MICRONS
281 COLUMNS DY= 50 MICRONS
112681 POINTS

SCAN DIRECTION: +X ROWS AVERAGED: 6
INITIAL COLUMN: 30 INITIAL ROW: 120
FINAL COLUMN: 274 FINAL ROW: 125



SCAN OF DENSITY

S 01/24/77 12:28:05
PROGRAM: TJJY
TECHNICAL PHOTOGRAPHY GROUP

Figure 4. Density scan of computer processed radiograph. The film density in the scan rectangle (shown in Figure 3) is plotted. Film density is directly related to the spray thickness.

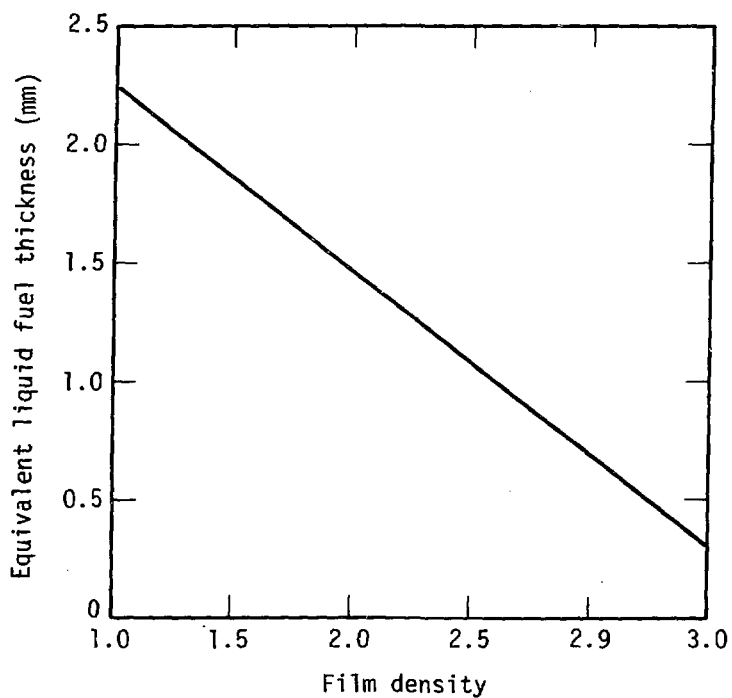


Figure 5. The x-ray film density is related to the thickness of a liquid fuel column that would cause the same absorption. This calibration may be used to interpret Figures 3 and 4.

**ERDA NATO/CCMS
Fourth International Symposium
Automotive Propulsion Systems
Washington, D.C.
April 18-22, 1977**

AN ADVANCED, LOW EMISSION DIESEL ENGINE CONCEPT

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Project Engineer

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 **TELEDYNE CONTINENTAL MOTORS**
General Products Division

A new automobile diesel engine concept has been developed to the preliminary engine design level and demonstrated by simulating vehicle tests with a computer model using steady state engine dynamometer data. The preliminary design is a six cylinder, swirl chamber diesel of 209 CID and 130 GHP. This concept engine weighs 495 pounds and employs turbocharging, variable compression ratio, high prechamber-main chamber volume ratio and exhaust gas recirculation. An existing automobile diesel was modified to simulate the concept engine and steady state engine tests were conducted. Test results were converted to urban cycle results for a 3000 pound GVW vehicle through the use of a computer model. Emission results (Grams/Mile) are .21 NO_x, .24 HC, and 1.24 CO. Fuel consumption (Miles/Gallon) is 30.3 urban cycle and 36.4 highway cycle. Vehicle cycle results at 3700 pounds GVW meet all emission requirements and exceed the future 27.5 MPG requirement by ten percent. Multifuel capability and 175 PSI BMEP power levels were also demonstrated.

AN ADVANCED, LOW EMISSION DIESEL ENGINE CONCEPT

Stephen H. Hill

Joseph L. Dodd

INTRODUCTION

Current and forecasted energy shortages have created a need for a more efficient power plant for light duty vehicles. This power plant must also be capable of meeting strict exhaust emission standards in order to maintain air quality levels. The twin, and frequently contradictory, requirements of high fuel economy and low emission levels have focused attention on an engine which has served reliably and economically in heavy duty applications—the diesel.

Although previous diesel installations in passenger cars have demonstrated the fuel economy and durability advantages of this combination, these vehicles have suffered from low power, excessive bulk, high cost, poor cold starting, high noise levels and excessive emissions, particularly oxides of nitrogen (NO_x).

The solution to these problems was envisioned to be a Variable Compression Ratio (VCR) diesel engine which maintains the fuel economy and durability of the diesel while providing excellent performance, lightweight, reduced bulk, low cost, excellent cold starting, reduced noise level and very low emission levels. This engine could also have the capabilities of being converted from an existing gasoline engine using the same basic engine structure and burning a broad range of fuels.

Recognizing the need for such an engine and the potential of a VCR diesel to satisfy that need, the Energy Research and Development Administration (ERDA) awarded a contract to TCM/GPD to further investigate the feasibility of a light duty diesel, to design such an engine and to demonstrate its potential in a 3,000 pound vehicle. The following goals, continued on Page 2, were established by ERDA:

Fuel Consumption (Miles/Gallon)	25 - Urban cycle <i>minimum</i> 35 - Highway cycle <i>minimum</i>
Exhaust Emissions (Grams/Mile)	0.41 HC <i>maximum</i> 3.40 CO <i>maximum</i> 0.40 NO _x <i>maximum</i>
Acceleration (0-60 MPH)	17.5 seconds <i>maximum</i>
Distance traveled from a standing start in ten (10) seconds	440 feet <i>minimum</i>
Merge Maneuver (25-70 MPH)	20 seconds <i>maximum</i>
Pass Maneuver (50-80 MPH)	17 seconds <i>maximum</i>
Engine Specific Weight	4.5 pounds/HP <i>maximum</i> including accessories

The arrangement of this paper will follow the major steps of the study with conclusions and recommendations following Section V.

REQUIREMENTS ANALYSIS

Model Formulation

The TCM/GPD Mobility Model was originally developed to investigate the dynamics of vehicle performance, especially the transient behavior of military vehicles with particular emphasis on the dynamics of engine and drive train performance.

This model was modified to simulate a turbocharged diesel in a compact car operating in the following modes:

- A. Acceleration from a standing point.
- B. Acceleration in merging traffic.
- C. Acceleration, DOT high speed pass maneuver.
- D. Federal Urban Driving Cycle - Exhaust Emissions and Fuel Economy.
- E. Federal Highway Driving Cycle - Fuel Economy.

The model calculates the response of the vehicle system by assuming that torques and rates of acceleration remain constant for a short increment of time during which the incremental change in speed can be calculated and added to the value of the previous increment.

The model is organized to accept a variety of vehicle, engine and transmission characteristics stored in external files. In this manner, the parameters of the vehicle and engine under study can be quickly changed without affecting the basic model. A flow diagram of the model is shown in Figure 1.

All the values calculated by the model can be stored in a separate file and recovered at a later time. This makes it possible for the analyst to provide a variety of reports from each run of the model and with plotting routines, produce a variety of graphic as well as tabular outputs. This flexibility enables the user to quickly review the results of design changes and make decisions which will result in improved vehicle performance.

The study which addressed these goals was performed as follows:

- I. **Requirements Analysis**—A computer model was adapted to translate vehicle performance goals to engine requirements. The model was then verified with test data. Model parametric studies and background research were used to define and evaluate alternate configurations.
- II. **Preliminary Engine Design**—Using the requirements analysis results, the preliminary design of a 209 CID diesel engine was completed.
- III. **Emission Testing**—An Opel passenger car diesel engine was modified to simulate the 209 CID concept engine and engine dynamometer development testing was conducted to demonstrate emissions and fuel consumption potential. An exhaust gas recirculation system was constructed and additional testing was conducted to very low emission levels.
- IV. **Vehicle Cycle Simulation**—Engine dynamometer test data was evaluated against vehicle performance goals using the computer model, and parametric studies were conducted to select the optimum configuration.
- V. **Multifuel and High BMEP Testing**—Additional testing was conducted on unleaded gasoline to demonstrate multifuel capabilities. High BMEP testing was conducted to demonstrate full power potential.

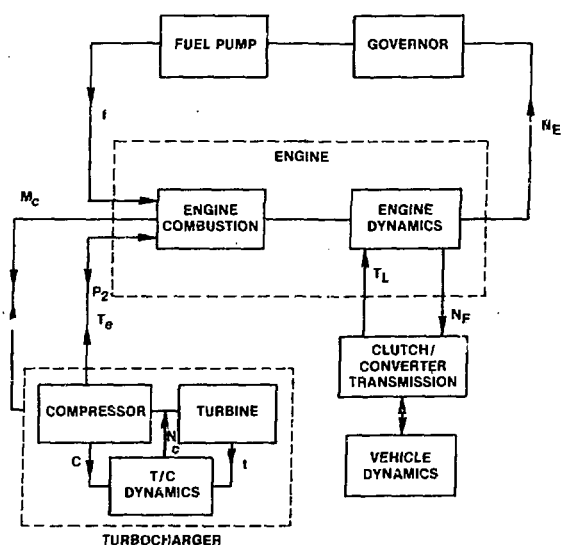


Figure 1—Flow Diagram of Turbocharged Diesel Engine Simulation Model.

Model Verification

The accuracy of the computer model was verified by comparing vehicle test results to computer model outputs generated from engine dynamometer test data. A Mercedes 300D was instrumented and all the acceleration tests were performed. Chassis dynamometer tests were performed for fuel consumption, emissions and wheel horsepower baseline data. The engine was dynamometer tested to determine the engine performance characteristics. Engine and drive train polar moments of inertia were measured.

Data from the above tests was used to verify the computer model. Figure 2 presents the computer simulation of the Mercedes 300D wide open throttle acceleration. The car test results are superimposed on the computer plotted prediction. Excellent correlation of results is demonstrated.

NO_x emissions and fuel economy data correlation with the model simulation is shown in Table 1. Test data shown includes chassis dynamometer results from an independent laboratory and EPA published results.

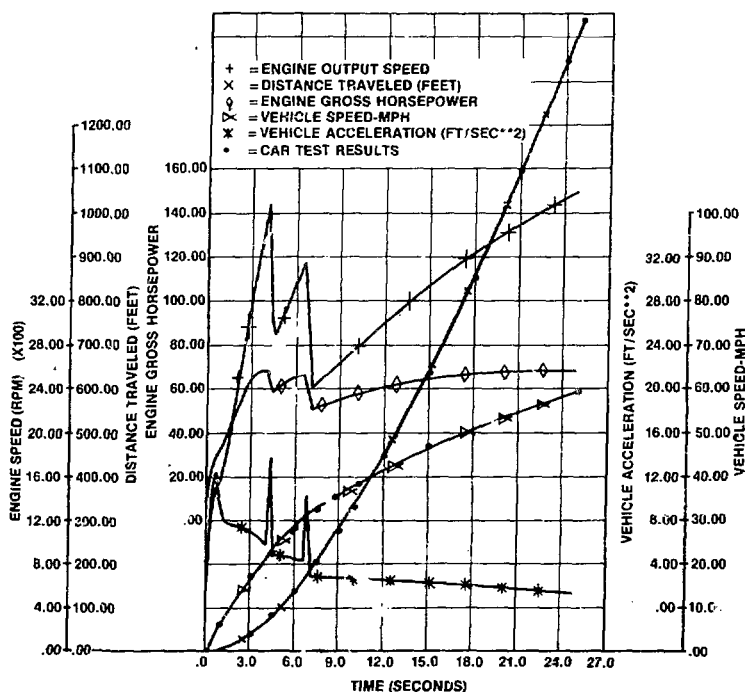


Figure 2—Performance Model Verification.

Table 1—Mercedes 300D/Model Verification

	Mile/Gallon		Grams/Mile Urban NO _x
	Urban	Hwy.	
Vehicle Test	22.2	27.7	2.01
EPA 1975	24	31	1.6
EPA 1976	22	28	1.8
Model Simulation	22.6	27.4	1.73

The model was verified a second time using turbocharged Opel 2100D data. Steady state engine data was input to the model. The simulation results are compared to actual vehicle test results in Table 2.

Table 2—Opel 2100D Turbocharged/Model Verification.

Urban Cycle Results

	MPG	Grams/Mile		
		NO _x	HC	CO
Vehicle Test	25.6	1.21	.22	1.21
Model Simulation	26.3	1.22	.22	1.18

Parametric Studies

During this phase of the program, TCM/GPD conducted trade-off studies on the ERDA specified compact passenger car. These studies were used to quantify the relative impact due to each parameter on vehicle performance, start-up, acceleration, vehicle range, emissions and fuel economy.

The studies resulted in the optimization of the specification for the various system components to assure compliance with the total system requirements defined by the "Preliminary Prototype Vehicle Specification for Compact Passenger Car" dated March 1975.

Parameters for four engine sizes with performance ratings of 85, 100, 115 and 130 gross horsepower were entered into the computer mobility model for simulation in the 3,000 pound compact car. Four DOT acceleration and pass maneuver cycles were conducted at each of the four engine ratings. Interpolation curves over the performance range of 85 to 130 gross horsepower indicated the minimum power required to meet each specification. The results, as shown in Figures 3 and 4, indicate the following:

1. Wide open throttle acceleration of 0 to 60 MPH in 17.5 seconds, at 3400 LB GVW requires only 95 GHP (Figure 3).
2. Wide open throttle acceleration from 0 to 440 feet in 10 seconds, 3400 LB GVW requires almost 130 GHP (Figure 3).
3. The merge maneuver acceleration from 25 to 70 MPH in 20 seconds, at 3400 LB CVW requires 100 GHP (Figure 4).
4. The pass maneuver acceleration from 50 to 80 MPH and pass in 17 seconds, at the maximum GVW of 3700 LB requires only 85 GHP (Figure 4).

The DOT acceleration requirement of 440 feet in 10 seconds, from a standing start, is the most severe. This one specification had the greatest influence on the size and the 130 GHP performance rating of the AD-209, Automotive Diesel Engine.

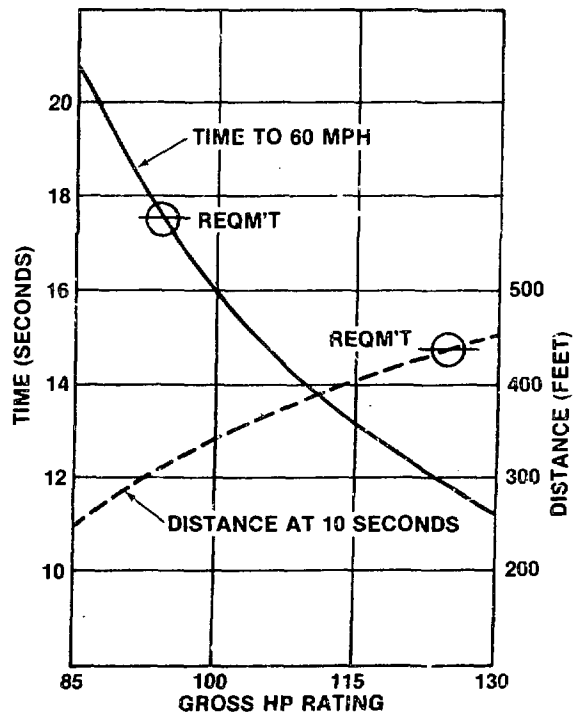


Figure 3—Acceleration from a Standing Start.

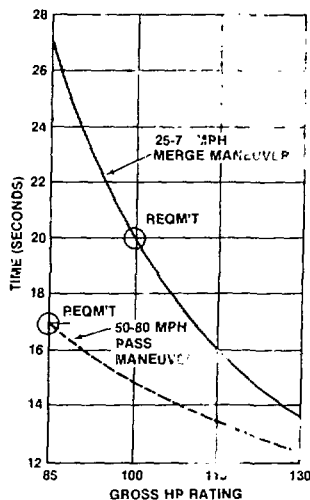


Figure 4—Acceleration, Merge & Pass Maneuvers.

Emissions Evaluation

A comparative analysis of indirect injection diesel engines was conducted to determine the NO_x emission characteristics of these engines. Excellent correlation was found among many different engines when the emissions were compared to the air/fuel ratio in the prechamber. Figure 5 shows the results of this study which demonstrates that the higher the air/fuel ratio in the prechamber, the lower the NO_x emissions. It is also apparent that for any given overall air/fuel ratio in an engine, the air/fuel ratio in the prechamber increases as the ratio of prechamber-main chamber volume increases. This is demonstrated in Figure 6.

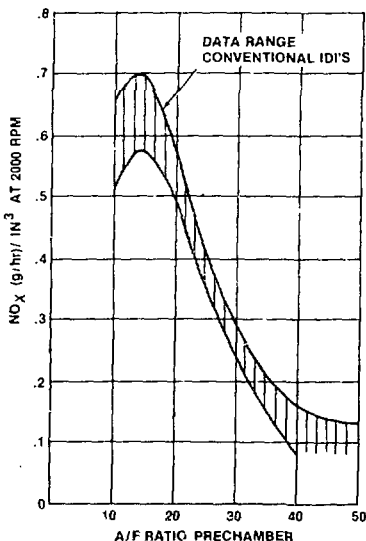


Figure 5— NO_x Emissions, Conventional IDI Engines.

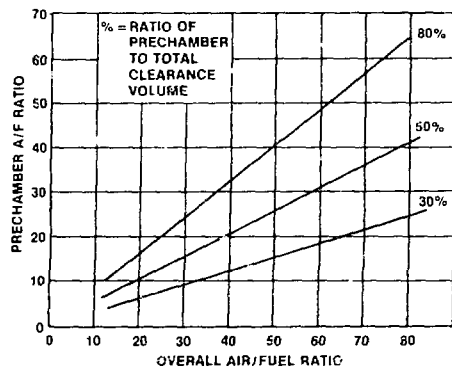


Figure 6—Prechamber Versus Overall Air/Fuel Ratio.

Various prechamber-main chamber volume ratios up to 80:20 were programmed into the computer model. Figure 7 shows the three NO_x versus prechamber air/fuel ratio curves introduced into the computer. Curve Number 1 is the mean of the shaded curve from Figure 5. It represents NO_x data from current IDI diesel engines with prechamber ratios from 40% to 70%, optimized for best smoke and fuel consumption only.

Curve Number 2 in Figure 7 represents projections with optimized injection characteristics, timing for emissions and approximately 5% exhaust gas recirculation. Curve Number 3 is the same as Curve Number 2 except with additional EGR, up to a total of approximately 15%.

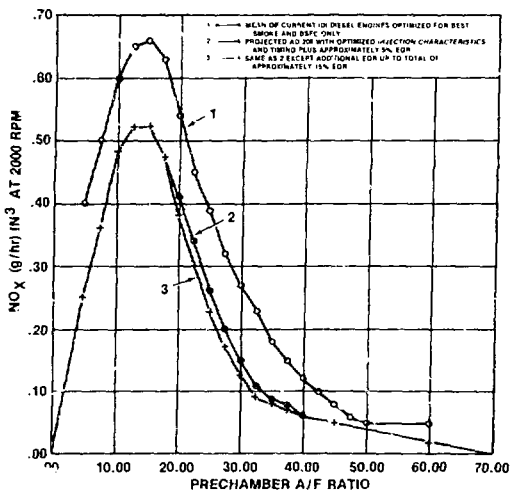


Figure 7—Specific NO_x Emissions.

The vehicle performance model was utilized to run the Federal Urban and Highway cycles with the following variables.

Car Weight

Prechamber Volume Ratio

Rear Axle Ratio

NO_x Curves Shown in Figure 7

The projected NO_x versus prechamber air/fuel ratio profile established (Curve 3, Figure 7) is shown in Figure 8 in comparison to the range for conventional IDI engines.

The parametric studies resulted in the following fuel economy and emissions predictions for the 130 horsepower engine in the ERDA compact car:

Fuel Economy - 28.4 Urban & 35.9 Highway MPG

NO_x Emissions - .39 Grams/Mile Urban

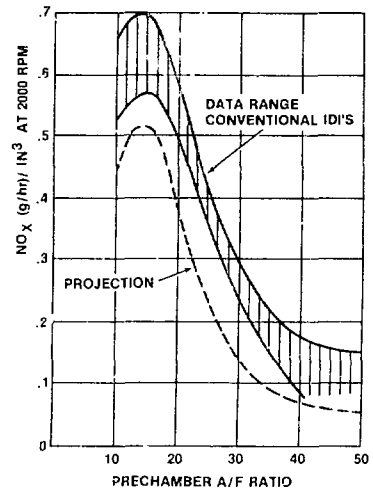


Figure 8—Projected NO_x Profile.

PRELIMINARY ENGINE DESIGN

Based on the results of the requirements analysis task, the preliminary design was completed for an engine that can meet the goals for fuel consumption, emissions and specific weight.

Features required to meet the program goals are as follows:

- A. **Variable Compression Ratio.** VCR is required to maintain a high compression ratio at light load for hydrocarbon control, with lower compression ratios at high load for controlled cylinder pressures. Controlled cylinder pressures permit operation at high power levels with lightweight construction for reduced weight and bulk. The use of high compression ratio at light load also reduces noise level, improves cold starting and enhances multifuel capabilities. A VCR piston schematic is shown in Figure 9 and an explanation of the VCR principle is placed in the Appendix.
- B. **Swirl Type Prechamber with a High Percentage of the Total Clearance Volume in the Prechamber.** This combustion system gives low emission levels and broad fuel tolerance with reduced noise levels and cylinder pressures. Providing a high percentage of the total clearance volume in the prechamber results in higher air/fuel ratios at initial combustion and reduced NO_x.

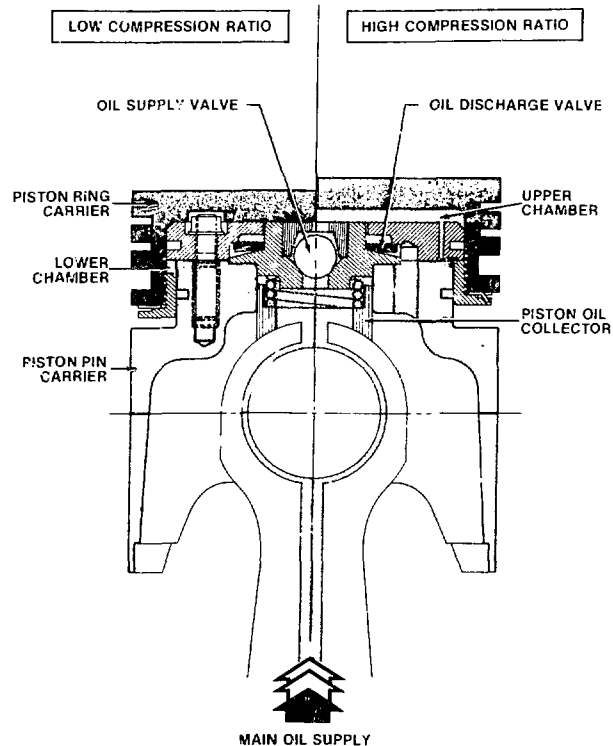


Figure 9—VCR Piston Schematic.

- C. Turbocharging.** Turbocharging maintains high air/fuel ratios for NO_x control and provides additional combustion air for high power operation.
- D. Exhaust Gas Recirculation.** EGR is required to achieve very low NO_x levels.
- E. Variable Injection Timing.** Variable injection timing is required to optimize HC and NO_x emissions while maintaining good fuel economy.
- F. Lightweight Castings.** Lightweight castings are required to meet the weight goals.

Table 3--AD 209 Specifications.

Bore	3.622 In (92MM)
Stroke	3.386 In (86MM)
Displacement	209 In ³ (3.4L)
Compression Ratio	24:1/12:1
Rated HP	130 GHP at 3200 RPM
Peak Torque	235 LB-FT at 2000 RPM
Weight, dry	495 LBS (Estimated)
(Includes fan, starter & alternator)	

Basic engine specifications and appearance are shown in Table 3 and Figure 10 respectively.

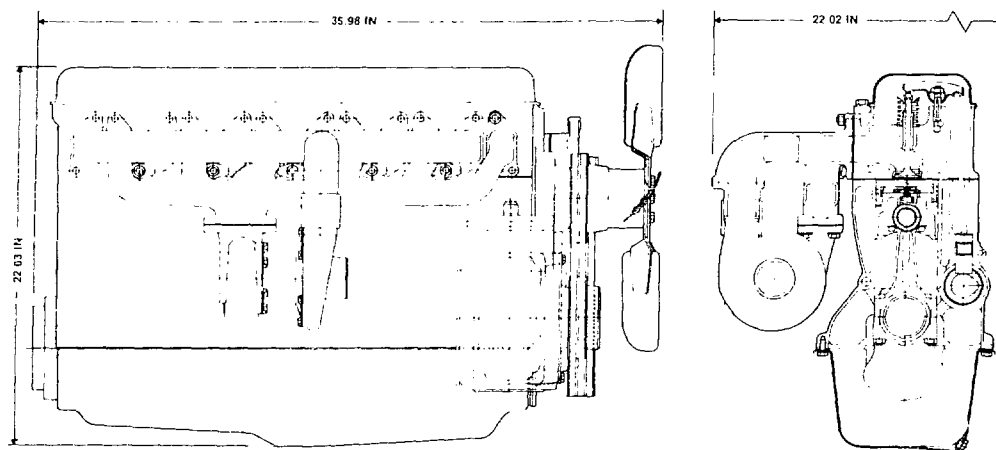


Figure 10--AD-209 Engine Concept.

Engine Size & Weight

The weight of the AD-209 engine was estimated by a sectional analysis of the various castings and catalog weight estimates for accessories. The bare, dry engine weight is estimated to be 149 pounds. The specific weight (pounds per gross horsepower) is presented in two ways in Table 4.

Conventional practice is to include the starter, alternator and fan in the engine weight. On this basis, the specific weight of the engine is 3.8 pounds per horsepower.

The ERDA specifications identifies the engine weight as including the additional engine auxiliaries noted in Table 4. On this basis, the specific weight of the engine is 4.48 LB/HP. The ERDA goal is 4.5 LB/HP.

Table 4—Engine Weight Estimate.

Bare, Dry Engine (Incl. Flywheel)	449 LBS.
Auxiliaries	
Starter	32 LBS.
Alternator	10 LBS.
Fan	4 LBS.
Subtotal	495 LBS.
Specific Weight	3.8 LB/HP
Battery	30 LBS.
Air Cleaner	5 LBS.
Radiator	20 LBS.
Oil	10 LBS.
Water	20 LBS.
Belts	2 LBS.
ERDA TOTAL	582 LBS.
SPECIFIC WEIGHT	4.48 LB/HP

A size comparison of the AD-209 engine with comparable gasoline engines in compact cars was made. Table 5 summarizes the results. Note that the AD-209 is not as wide nor as high as the comparison engines. The length dimension of 35.98 inches is longer; however, this dimension can be reduced by changing the fan adaptor shown in Figure 10.

EMISSION TESTING

The objective of the emission testing phase was to develop emissions and fuel economy data that would define the performance and potential of the AD-209 diesel engine concept. This objective was accomplished by modifying an existing automobile diesel engine to simulate the AD-209.

Steady state dynamometer tests were conducted to develop performance data and guide further modification. Data from promising configurations were later evaluated against vehicle performance goals through the use of the TCM/GPD computer model.

The Opel 2130D engine was selected for modification and test for three reasons:

1. It is typical of current, state-of-the-art automobile diesels.
2. It uses the same combustion system (Ricardo Mark V) and is similar in bore and stroke to the AD-209 concept engine.
3. It can be successfully turbocharged (1)*.

Specifications for the Opel 2100D engine are placed in the Appendix (2) (7)*.

Baseline tests were conducted on the engine as received. The engine was then turbocharged and modified for increased compression ratio and prechamber-main chamber volume ratio. Static injection timing was varied manually from TDC to 18° ATDC. Testing was conducted from 600-3200 engine RPM at loads of 0-100 PSI BMEP. Test variables were evaluated by replacing or modifying engine components. The range of variables tested during this phase is listed in Table 6.

A discussion of the effect of selected test variables on emissions and/or fuel consumption follows.

* Numbers in parentheses designate references at end of the paper.

Table 5—Size Comparison With Current Gasoline Engines.

AUTOMOBILE	Disp. [In ³]	L	W	H
Dart, Duster, Scamp	225	33.20	24.50	32.20
Gremlin, Pacer	232	34.47	23.69	25.84
Nova, Camaro, Omega, Ventura	250	33.00	28.00	27.00
Maverick, Granada, Monarch, Comet	200	35.91	25.11	27.66
ERDA Car with Diesel	209	35.98	22.02	22.30

Table 6—Test Variables.

VARIABLE	RANGE
Hot Compression Ratio	23.0:1 - 29.5:1
Prechamber-	
Main Chamber Volume Ratio	57:43 - 77:23
Prechamber Throat Size ...	Standard & 15% Oversize
Turbocharger Turbine Housing	
A/R Ratio**	.25, .36 & .48
Injection Pump Plunger Diameter	.9 and 11 MM
Injection Line Inside Diameter	.063 & .084 Inch
Piston Rings	Standard, Head Land & Improved Oil Control
EGR	0 - 55 Percent

**NOTE: A/R ratio is an expression of relative size.

A/R ratio is calculated by dividing the area of the volute by the radius of the volute.

Effect of Turbocharging

Turbocharging supplies additional combustion air resulting in higher air/fuel ratios for NO_x reduction and permitting the addition of more fuel for higher power output. In addition, it was anticipated that the increase in air motion and temperature provided by turbocharging would improve hydrocarbon control and fuel economy through better combustion.

The turbocharger utilized is the wastegated AiResearch T-3 unit in compressor trim "C". Turbocharging did increase air/fuel ratio at the same power level as shown in Figure 11.

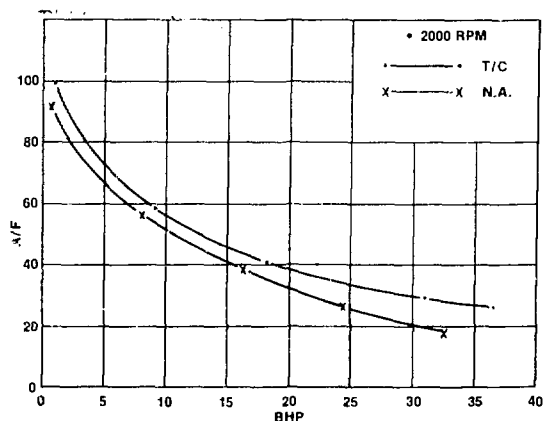


Figure 11—Effect of Turbocharging.

Turbine housings with A/R ratios of .25, .36 and .48 were evaluated during the test program. A/R ratio should be sized so as to maximize the pressure drop across the engine, thereby minimizing negative work or creating positive work in the pumping loop and decreasing fuel consumption. The effect that turbine housing A/R ratio had on fuel consumption and NO_x is shown in Figures 12 and 13. The .36 A/R ratio turbine housing resulted in the best overall fuel consumption and NO_x over the engine speed and load range tested.

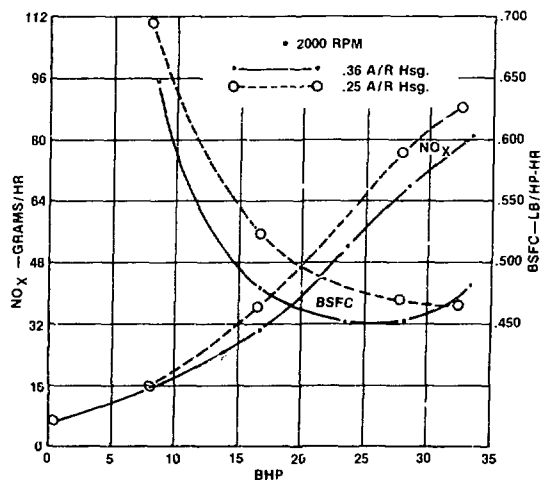


Figure 12—Effect of Turbine Housing Size.

Effect of Injection Timing

One of the primary techniques for NO_x control is injection timing. Late injection timing reduces peak cylinder pressures and temperatures, and residence time of the combustion gases at high temperatures. Reduction of temperature and time available for reaction slows NO_x formation rate.

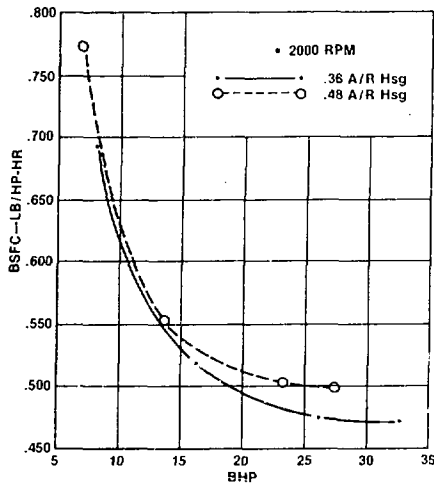


Figure 13—Effect of Turbine Housing Size.

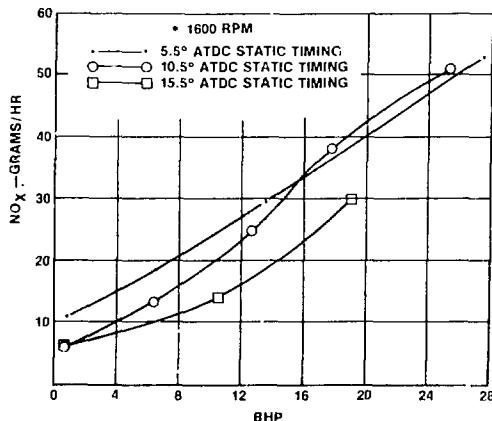


Figure 14—Effect of Injection Timing on NO_x .

Test results showing the effect of injection timing on NO_x and HC emissions are shown in Figures 14 and 15. NO_x is substantially reduced with injection timing retard. At high loads, the increase in fuel consumption as a result of retarded timing causes an increase in NO_x as a response to air/fuel ratio. HC are affected very little by up to 10° of injection timing retard. However, additional retard causes a large increase in hydrocarbons particularly near no load and full load.

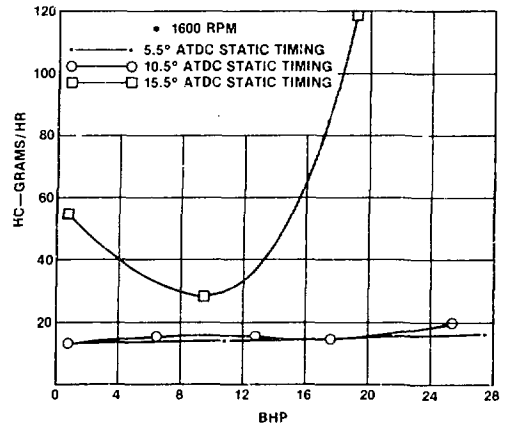


Figure 15—Effect of Injection Timing on HC.

Effect of Compression Ratio

Prior test results indicated that improved hydrocarbon control would be essential to meet the desired emission goals. The control technique utilized had to be effective at the late injection timings dictated by NO_x requirements and under the low engine speed, light load conditions prevalent in the light duty vehicle test. Within the framework, it was felt that increased compression ratio should be evaluated.

Compression ratio changes were accomplished with the following special components:

1. Thinner head gasket.
2. "Head Land" piston rings.
2. Pistons without swirl cavities in the dome.
4. increased volume prechambers (swirl chambers).

As predicted, an increase in compression ratio did give better hydrocarbon control as indicated in Figure 16. The increased compression ratio also decreased NO_x levels at certain engine speeds and loads as shown in Figure 17. This decrease in NO_x is primarily due to an increase in air/fuel ratio as a result of lower fuel consumption (See Figures 18 and 19.)

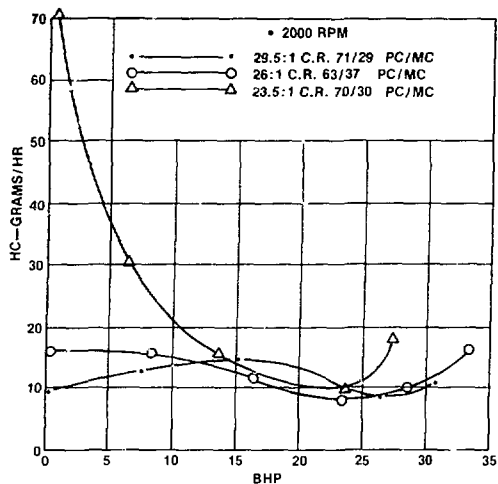


Figure 16—Effect of Compression Ratio on HC.

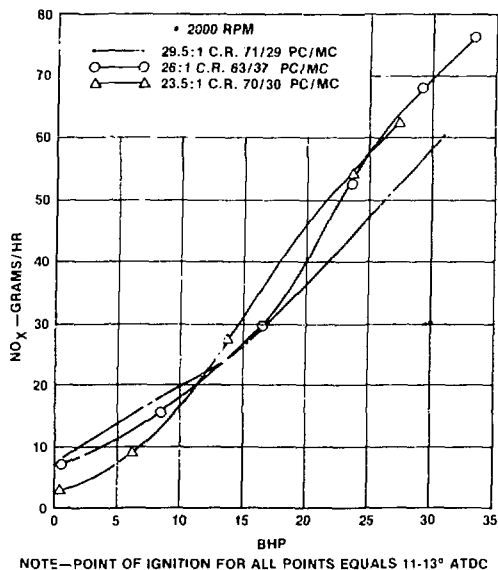
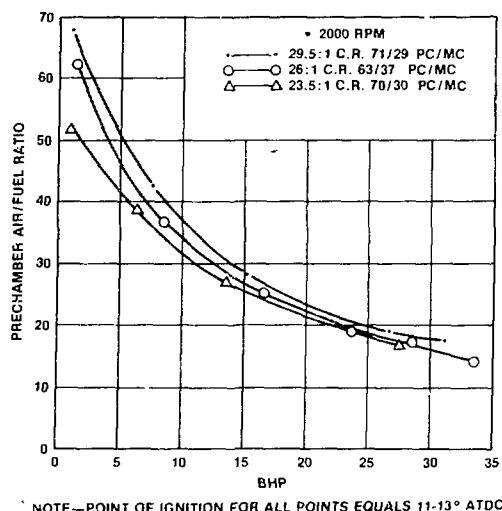
Figure 17—Effect of Compression Ratio on NO_x.

Figure 18—Effect of Compression Ratio on Prechamber Air/Fuel Ratio.

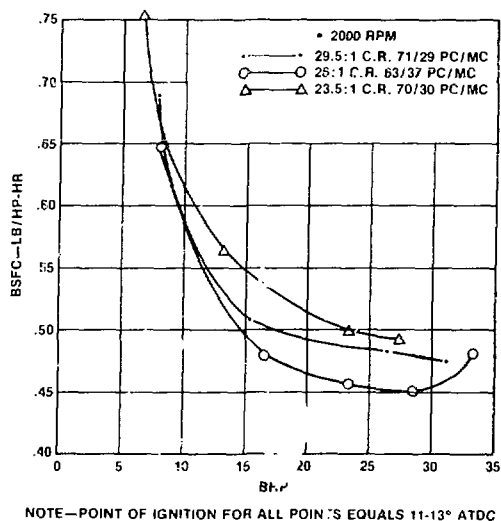


Figure 19—Effect of Compression Ratio on BSFC.

The increase in NO_x at light loads is a consequence of the shape of NO_x response to prechamber air/fuel ratio. It is assumed that the increased compression ratio resulted in higher combustion pressures and temperatures which took precedence over the air/fuel ratio effect at light load, resulting in an increase in NO_x. The improved hydrocarbon control and fuel consumption resulting from the increased compression ratio permitted additional injection timing retard further decreasing NO_x.

Effect of Prechamber-Main Chamber Volume Ratio

Variations in prechamber-main chamber volume ratios affect NO_x levels by varying air/fuel ratio in the prechamber where most of the combustion occurs. For the same compression ratio and fuel consumption level, an increase in prechamber-main chamber volume ratio will increase (lean) the air/fuel ratio in the prechamber. Changes in prechamber-main chamber volume ratio were accomplished by installing a larger prechamber and/or decreasing main chamber by one of the methods used to increase compression ratio.

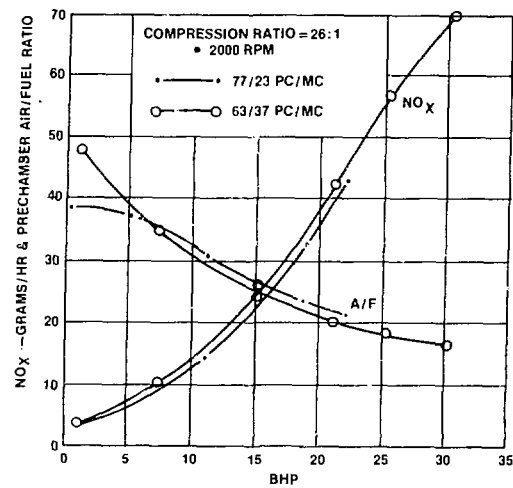


Figure 20—Effect of Prechamber-Main Chamber Volume Ratio.

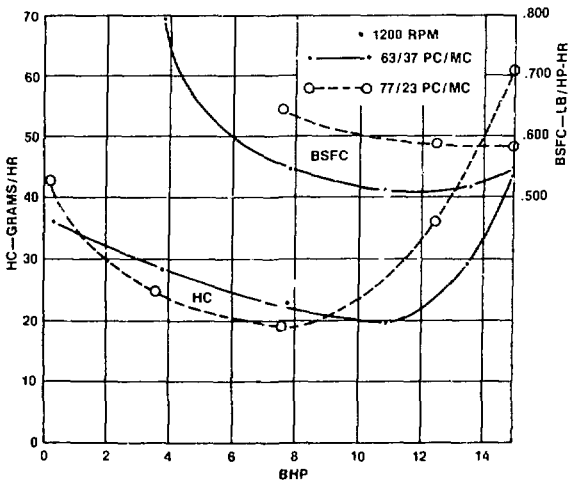


Figure 21—Effect of Large Prechamber.

The effect of prechamber-main chamber volume ratio on prechamber air/fuel ratio and NO_x is shown in Figure 20. The 77/23 PC/MC build was accomplished by installing a larger prechamber. Prechamber throat size and injection nozzle location were the same as the smaller standard prechamber. Use of this non-optimized prechamber resulted in an increase in overall air/fuel ratio. Therefore, the test results do not indicate the magnitude of prechamber air/fuel ratio increase and NO_x decrease that would be expected for a prechamber-main chamber volume ratio increase of 22 percent. However, the trend is as predicted.

Use of the large, non-optimized prechamber had a very negative effect on fuel consumption and HC as shown in Figure 21. It was felt that a portion of the fuel consumption increase was due to the additional work required to pump more air into the larger prechamber through the standard size throat. Therefore, a large prechamber with a 15 percent increase in throat size was evaluated.

Effect of Large Prechamber Throat

Typical test results with the large throat prechamber are shown in Figure 22. Fuel consumption has decreased significantly and some reduction has been made in NO_x , particularly at high load. At light load, NO_x has increased slightly. However, fuel consumption and NO_x are still well above results obtained with the small (63/37 PC/MC) prechamber. These test results indicate the importance of extensive performance development to optimize new prechamber designs. Although the large prechamber did give indications of NO_x reductions, the test program time framework did not permit the additional development required to realize this potential.

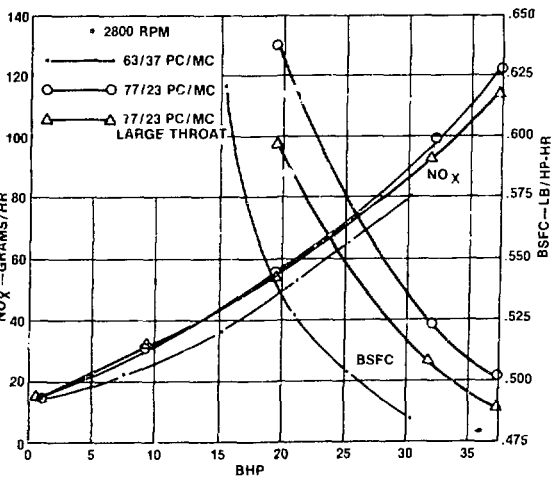


Figure 22—Effect of Large Prechamber Throat.

Injection Rate

Although a swirl chamber engine is relatively insensitive to injection system variations, other investigators (3) have found some effect on emissions as a result of changes in injection rate. Injection rate was varied here by replacing the 9 millimeter plunger diameter injection pump with a unit having a plunger diameter of 11 millimeters. Test results generated at the same static injection timing are shown in Figure 23. Air/fuel ratio is also plotted to indicate that the change in NO_x is not due to a change in air/fuel ratio. NO_x is increased although the effect on NO_x becomes less as speed and load decrease.

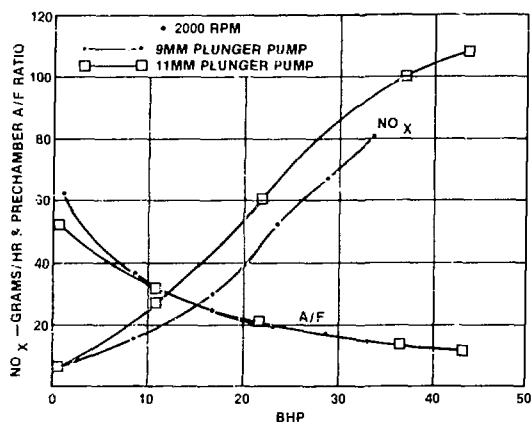


Figure 23—Effect of Injection Rate.

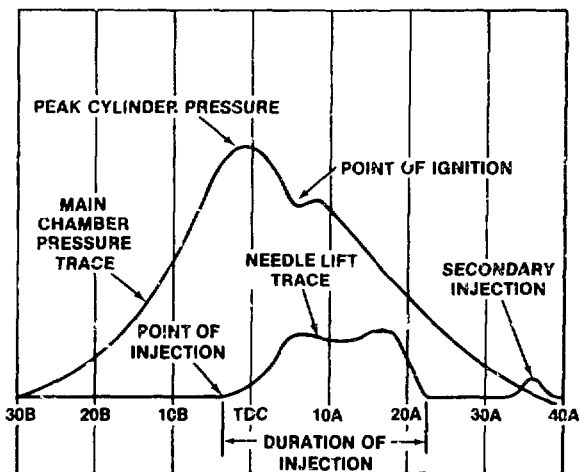


Figure 24—Pressure Card.

Effect of Secondary Injections

The Opel injection system gave satisfactory performance when used with the standard engine. However, after engine modification and at retarded injection timings, pressure cards made during development testing indicated that secondary fuel injections were occurring, particularly at high load. A typical pressure card is shown in Figure 24. A secondary injection is an untimed injection nozzle opening or fuel injection as a result of reflected pressure waves. In Figure 24, the secondary injection is occurring at approximately 36° ATDC. Secondary injections are known to be a significant source of emissions.

To decrease secondary injections, the fuel injection line inside diameter was reduced from .084 inch (2.1 MM) to .063 inch (1.6 MM). This change eliminated virtually all secondary injection indication on needle lift traces throughout the engine speed and load range tested. The effect on HC emissions is shown in Figure 25. Large reductions in HC resulted even in areas where secondary injections were not detected from the needle lift trace.

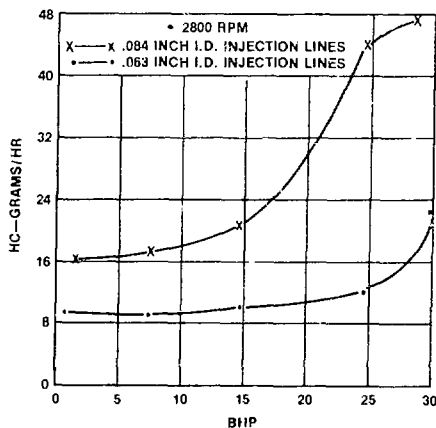


Figure 25—Effect of Secondary Injections.

Reduction in injection line inside diameter should not be construed as a cure-all for injection secondaries. Each injection system must be individually, hydraulically balanced for the fuel delivery schedule required. The fix used here was verified on an injection test bench before being installed on the engine.

Exhaust Gas Recirculation

EGR reduces NO_x by increasing the heat capacity of the intake charge and thereby decreasing peak cycle temperatures and NO_x formation. EGR also decreases the oxygen

content of the intake charge, thereby decreasing the amount of oxygen available for NO_x formation at combustion. EGR has an additional benefit in a diesel engine which affects all mass emission rates. Since a portion of the exhaust flow is directed back into the engine cylinder, exhaust mass flow out the tailpipe is decreased. A comparative analysis (Figure 26) of prechamber fuel-air ratios versus BHP at constant 2000 RPM indicates the limitations of diesel cycle lean range EGR application. The prechamber fuel-air ratio is very lean at low BHPs and becomes richer with increasing HP and increasing fuel flow because airflow remains virtually constant. At any BHP level the prechamber fuel-air ratio may be increased as high as stoichiometric, approximately .062 (16 to 1 air-fuel ratio) by the substitution of EGR for excess air. At or above a fuel-air ratio of stoichiometric, EGR may not be applied for there is no excess air in the combustion process. Diesel operation in the extreme rich fuel-air ratio range above stoichiometric produces excessive fuel consumption and smoke.

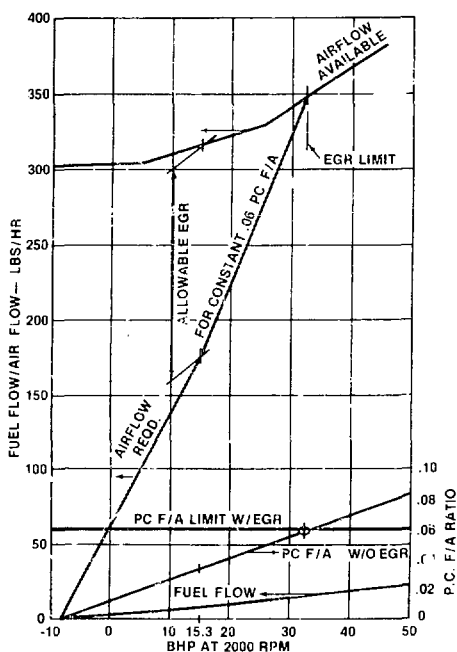


Figure 26—Opel 2100D EGR Limits.

A further comparative analysis (Figure 27) of smoke versus NO_x reduction with varying amounts of EGR at the 15.3 BHP point of Figure 26 reveals that with no EGR there is 70% excess air. Smoke is approximately 2.5% opacity. The prechamber fuel-air ratio is noted to be .036 and NO_x emissions are 29.5 grams per hour.

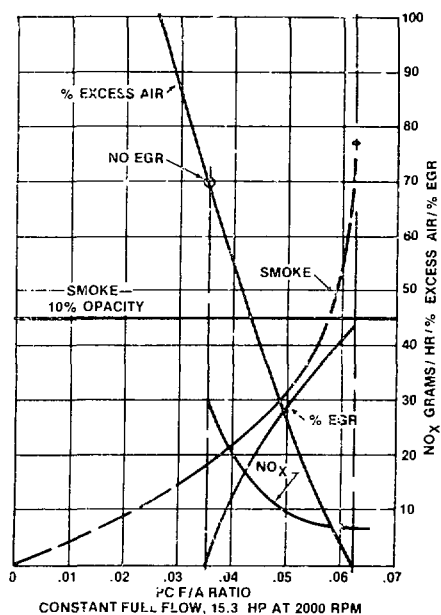


Figure 27—Opel 2100D Effect of EGR on NO_x vs Smoke.

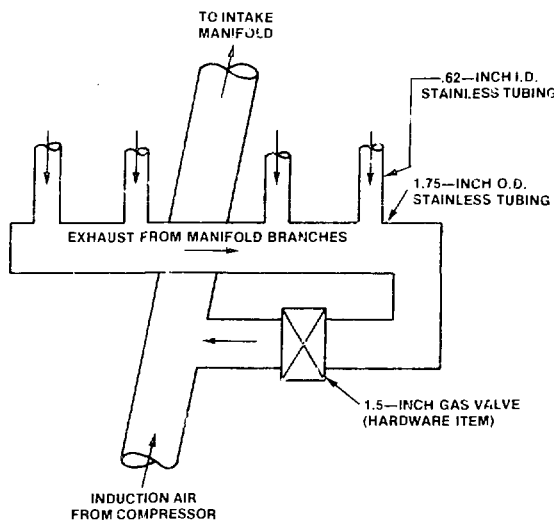


Figure 28—EGR Schematic Lab System.

As EGR is applied, excess air is reduced and prechamber fuel-air ratio rises toward .062 (stoichiometric). NO_x emissions fall steeply at first and then gradually toward the last, whereas smoke rises gradually at first and steeply toward the last. This dual curve characteristic yields an extreme early reduction in NO_x emissions with EGR before

smoke rises eventually to the 10% opacity point. This case shows EGR may be limited to approximately 38% for 10% opacity smoke with a reduction in NO_x from nearly 30 to about 7.5 grams per hour, or by 75%.

A schematic of the EGR system used in testing is shown in Figure 28. With this system, EGR rate is governed by the gas valve position and the pressure differential between the exhaust manifold and the intake manifold or compressor discharge. At light loads when air/fuel ratio is high, exhaust pressure is significantly higher than intake manifold pressure and high rates of EGR occur. As engine load becomes greater and air/fuel ratio becomes lower, intake manifold pressure increases faster than exhaust pressure and EGR rate decreases until it reaches zero when exhaust and intake pressures are equal. Therefore, the EGR rate characteristics provided by this system are exactly what is required for a diesel engine. In a vehicle, this system could consist of a connector between intake and exhaust manifolds containing a properly sized orifice to limit maximum EGR rate, and a check valve to prevent back flow.

The effect of EGR on NO_x and HC is shown in Figures 29 and 30. Note that NO_x and HC are plotted on a mass basis so that both the concentration level and reduced exhaust flow effects are indicated. NO_x drops sharply with increasing EGR rates. HC levels decrease slightly and then increase rapidly with increasing EGR rates except at higher loads, where HC levels increase with increasing EGR. Fuel consumption was unaffected by EGR except for a slight decrease at a small number of test points. Note that EGR was not used at high loads. Since the driving force for this EGR system is exhaust-intake pressure differential, no EGR can occur at high loads. The use of EGR at load levels requiring low (rich) air/fuel ratios would be expected to have a very negative effect on HC and fuel consumption.

The EGR system used in initial development testing was constructed to allow maximum flexibility in testing and, as a result, was quite large and bulky. The large amount of surface area resulted in a significant amount of heat loss through radiation and a consequent reduction in EGR temperature. Maximum heat loss was calculated to be 8000 BTU per hour, a level which could be achieved easily with a small heat exchanger in a vehicle. In order to evaluate the effect of EGR temperature, a new, smaller EGR system was constructed. This configuration resulted in EGR temperatures near exhaust temperatures. The effect of this EGR temperature change is shown in Figures 31 and 32. These figures depict test points with the small EGR system data plotted over the effect of EGR plots shown previously in Figures 29 and 30. NO_x data shows no significant change. HC data with zero EGR has increased considerably as a result of oil consumption problems. At higher load, the increase in HC as a result of thirty percent EGR is as expected. At medium load, zero and forty-two percent EGR give the same HC result. With forty-eight percent EGR, HC increase is less than expected. At

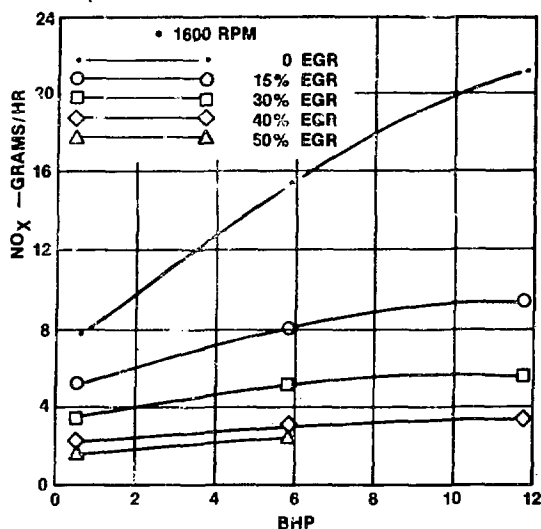


Figure 29—Effect of EGR on NO_x .

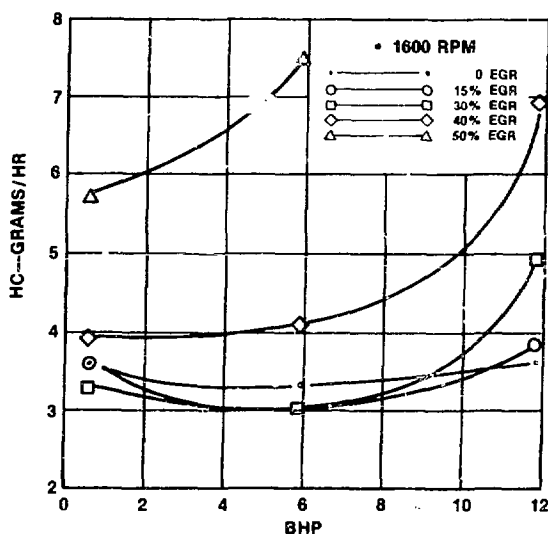


Figure 30—Effect of EGR on HC.

light load, HC levels decrease with thirty-eight percent EGR and decrease further with fifty percent EGR. It is assumed that these changes in HC response to EGR are the result of increased intake manifold temperatures with the small EGR system. Fuel consumption was also affected more frequently with this EGR system. At light load test points, small decreases in fuel consumption were most frequent. At higher

loads where EGR was available, small increases in fuel consumption were seen. The overall effect on fuel consumption is judged to be slightly favorable.

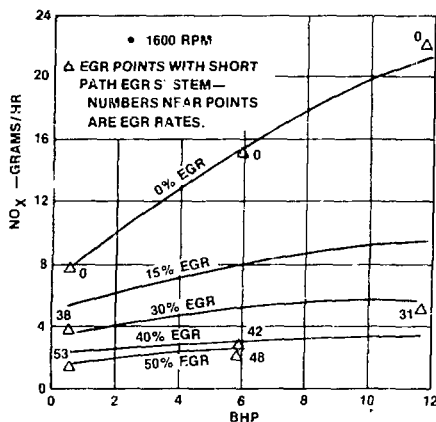


Figure 31—Effect of EGR Temperature on NO_x.

In summary, the following list of EGR advantages is presented:

1. With fixed injection timing, EGR can reduce NO_x drastically with little effect on other emissions or fuel consumption.
2. The use of EGR reduces the amount of injection timing retard required to control NO_x thereby decreasing other emissions and fuel consumption.
3. EGR reduces mass flow through the engine thereby decreasing manifold pressure, peak cylinder pressure and all mass emission rates.
4. The increase in intake manifold temperature with EGR decreases ignition delay. Decreased ignition delay should assist light load hydrocarbon control and decrease rates of pressure rise and noise levels.
5. The same techniques by which EGR reduced NO_x formation should also decrease the formation of other oxides thereby decreasing diesel odor.

Low Emission Configuration

The best overall emission results were obtained with an engine configuration employing turbocharging, 29.5:1 compression ratio, 71:29 prechamber-main chamber volume ratio, improved injection system and piston ring package, and EGR. Modifications to the standard Opel 2100D to obtain this configuration are listed in Table 7.

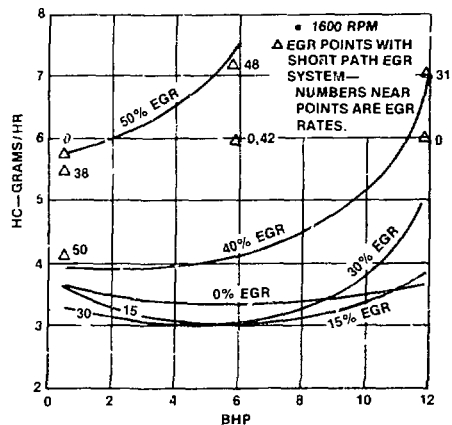


Figure 32—Effect of EGR Temperature on HC.

Table 7—Modifications to Opel 2100D Engine.

CYLINDER HEAD
Valve stem seal - Sealed Power P/N ST-2010
Cylinder head gasket - McCord P/N X-8077
PISTONS
Modified for head land piston ring
(See illustration in Appendix)
Swirl cavities filled with weld
PISTON RINGS (Sealed Power)
Top (head land) groove - P/N XCCC-158
Second & fourth grooves - P/N CF-3556
Third groove - none
Bottom groove - side rails (2) - P/N XCF-3558
Expander - P/N XCF-3557
FUEL INJECTION SYSTEM
.063 inch inside diameter injection lines
TURBOCHARGER
AiResearch Model T-3 wastegated in compressor trim "C"
with .33 A/R turbine housing
EGR
Exhaust gas recirculation from 0-50 percent

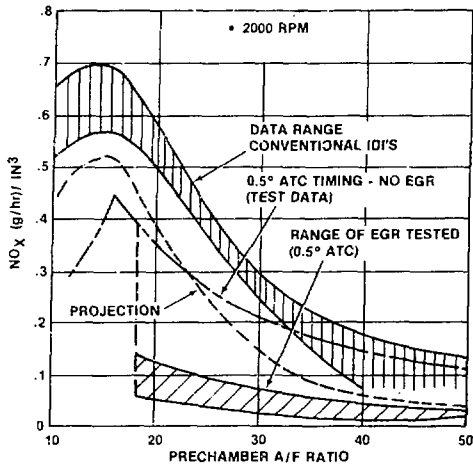


Figure 33—NO_x Test Results Compared to Prediction.

Placed in the Appendix are plots of NO_x, HC, CO and BSFC at three engine speeds for: (a) the baseline as received configuration; (b) the low emission configuration at 5.5° ATDC static injection timing; and (c) the low emission configuration with EGR at 0.5° ATDC static injection timing. EGR rate was selected by fixing the control valve in the seventy percent open position resulting in EGR rates from zero to forty percent.

When compared to baseline data, the low emission configuration modifications result in substantial reductions in NO_x; particularly in the important low speed and load areas. HC and CO are similar to baseline results except at high speed where some increase occurs. Fuel consumption has increased as much as ten to fifteen percent at high loads as a result of retarded injection timing.

The application of EGR reduces NO_x further in the low speed and load areas. This decreases the amount of injection timing retard required thereby decreasing NO_x at high load, and HC, CO and BSFC everywhere. An exception is the increase in HC and CO at 1200 RPM - mid load with EGR, indicating that EGR rates should be decreased in this area. Fuel consumption with EGR is similar to baseline results except at 2800 RPM where it is five percent greater.

Figure 33 is a plot comparing actual test data to the projection shown in Figure 8. The emission levels achieved in testing are well below the projected level required to meet the NO_x goal.

VEHICLE CYCLE SIMULATION

Steady state engine dynamometer data was entered in the computer model input files for vehicle simulation over the urban cycle. The 126 CID engine data is converted to 209 CID

engine data by the model. The computer model notes the instantaneous BHP required. It then reduces this BHP requirement to a level of 126/209, at the same engine speed, to find data at an equivalent BMEP as would be realized with the larger engine. The computer model then goes into the 126 CID memory files for the data required. It applies this data to the program logic to find the interim emissions. These interim emissions are then multiplied by 209/126 to bring the emissions to the level which would be seen with the larger engine.

Simulation results at two injection timings, with and without EGR, are shown in Table 8.

Table 8—Simulation Results.

TIMING °ATC	EGR	MPG FUELECON.	GMS/MI NO _x HC	CO
0.5	NO	33.3	.73 .15	.83
5.5	NO	31.1	.64 .28	1.23
0.5	YES	33.2	.17 .18	.89
5.5	YES	31.1	.41 .29	1.28

3000# GVW, 209 CID, 71% PC, 29.5:1 CR, 3.23:1 Axle Rat

The use of a 3.23:1 axle ratio resulted in several occurrences of low prechamber air/fuel ratio and associated smoke. The effect of axle ratio on prechamber air/fuel ratio is shown in Table 9.

Table 9—Effect of Axle Ratio on P.C. Air/Fuel Ratio.

Number of Seconds [In 1369 Seconds], When P.C. Air/Fuel Ratio is at/or Below.

AXLE RATIO	14/1 [.07 F/A]	16/1 [.06 F/A]	18/1 [.055 F/A]	20/1 [.05 F/A]
3.23	0	3	10	23
3.55	0	2	7	14
3.905	0	0	5	11
4.30	0	0	4	7

10% Opacity Smoke

The effect of axle ratio on urban cycle emissions and fuel economy is shown in Table 10. Optimum axle ratio is 3.905:1.

Table 10—Effect of Rear Axle Ratio.

AXLE RATIO	MI/GAL. FUELECON.	GMS/MI NO _x	HC	CO	SMOKE
3.23	33.2	.17	.18	.89	Decreasing ↓
3.55	31.8	.19	.20	1.04	
3.905	30.3	.21	.24	1.24	
4.30	28.7	.24	.29	1.47	

3000# GVW, 0.5° ATC, 209 CID,
71% P.C. 29.5:1 CR with EGR

The effect of increasing vehicle weight is shown in Table 11. Note that the 3700 pound vehicle easily meets the future 27.5 MPG requirement.

Table 11—Effect of Vehicle Weight-Urban Driving Cycle.

TEST WEIGHT POUNDS	MI/GAL. FUELECON.	GMS/MI NO _x	HC	CO
3000	30.3	.21	.24	1.24
3400	29.1	.24	.24	1.23
3700	28.3	.26	.24	1.23

209 CID, 29.5:1 CR, 71% P.C.
3.905 Axle Ratio, W/EGR 0.5° ATC

The best overall results with a 3000 pound test weight, 3.905:1 axle ratio, 0.5° ATDC static injection timing and EGR meet all requirements as shown below:

	MPG		GRAMS/MI NO _x	HC	CO
	URBAN	HIGHWAY			
Result	30.3	36.4	0.21	0.24	1.24
Goal	25	35	0.40	0.41	3.40

MULTIFUEL AND HIGH BMEP TESTING

Multifuel Testing

Multifuel capability is a very desirable feature in an engine since it permits most efficient and economical utilization of fuel resources. In a diesel engine, the multifuel capability allows substitution of more readily available fuels in areas or circumstances where diesel fuel is uneconomical or unavailable.

Since time did not permit surveying a variety of fuels, testing was conducted with unleaded gasoline, generally acknowledged as the worst case, commonly available fuel for diesel usage. The test engine was not modified from the low emission configuration with the exception of static injection timing which was advanced five degrees. This change was necessary to compensate for the different pumping characteristics and longer ignition delay of the less viscous gasoline. Advancing static injection timing five degrees advanced the point of injection approximately three degrees when compared to DF2 test points.

The Opel 2100D injection pump plunger is lubricated by the fluid being pumped. The pump manufacturer indicated that gasoline did not provide sufficient lubrication to the plunger and seizure might occur in an hour or less of running. In order to generate sufficient data for analysis, it was decided to conduct a majority of the testing on a mixture of 95% unleaded gasoline and 5% lubrication oil (Shell Rimula - 20W20). Previous testing with this mixture by other investigators had shown no significant decrease in injection pump plunger life. It was also desired to evaluate performance with a mixture of 90% unleaded gasoline and 10% DF2, such as might occur if a diesel vehicle were fueled with gasoline in an emergency. The last fuel to be tested was straight unleaded gasoline for comparison to test results with the two mixtures. A summary of the test points and fuel mixtures is in Table 12. Cetane number was determined from advertised octane number and charts of the octane-cetane relationship for unleaded gasoline (4). Cetane numbers for mixtures were interpolated from a chart on the effect of additives to leaded regular grade gasoline (5).

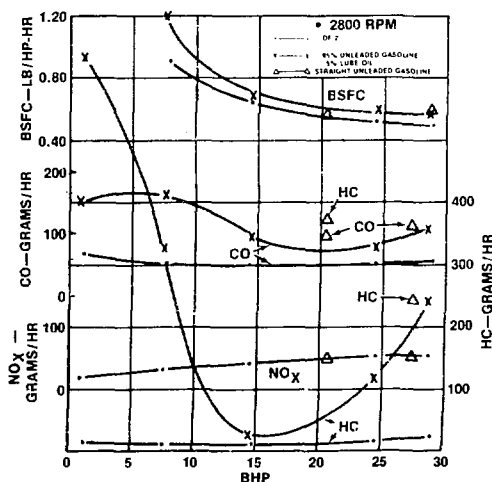


Figure 34—Performance with Gasoline.

Use of the standard Opel glow plugs was required to start the engine in the 50°F ambient air. The engine successfully completed the test schedule outlined in Table 12. Typical test results are compared with DF2 test results in Figures 34 and 35. Fuel consumption has increased approximately 10 - 20%. NO_x is unchanged. CO has increased by a factor of 2 - 3 and HC has increased greatly at the extremes of the load range with a 100% increase at mid range. Turbine inlet temperatures have increased slightly. Ignition delay is unchanged except at high load where a slight increase is indicated. Smoke is reduced by 50%.

Results from two test points on straight unleaded gasoline are also plotted with similar characteristics except for a large increase in mid load HC, and an increase in smoke levels to near DF2 results. Some increase in noise and vibration was observed although no measurements were made.

The significance of these results is not the performance described in the figures but the fact that the schedule outlined in Table 12 could be successfully completed indicating the true multifuel capability of this combustion system.

Table 12—Multifuel Test Schedule.

RPM	BHP	FUEL	CETANE NO.
3200	24.3	95% Unleaded Gasoline	17.5
2800	28.7	&	
2800	24.5	5% 20 Weight Lube Oil	
2800	14.6		
2800	7.7		
2800	1.1		
2400	10.5		
2000	22.6		
2000	15.2		
2000	7.6		
2000	1.1		
1600	12.0		
1200	12.0		
1200	9.1		
1200	4.3		
1200	0.4		
600	0.2		
3200	23.7	90% Unleaded Gasoline	17
2000	11.8	&	
2000	7.8	10% DF2	
1200	9.1		
3200	22.4	Straight Unleaded Gasoline	15
3200	15.2		
2800	27.4		
2800	20.5		
1200	9.4		

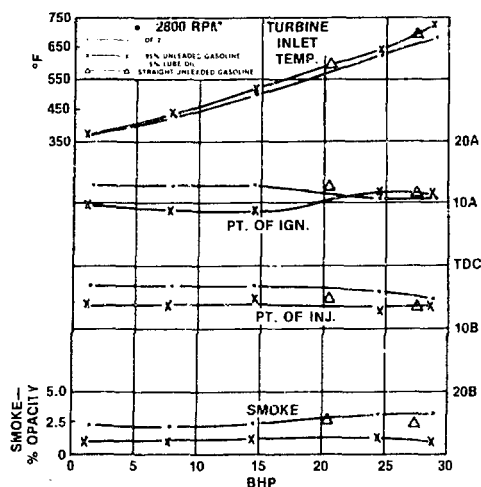


Figure 35—Performance with Gasoline.

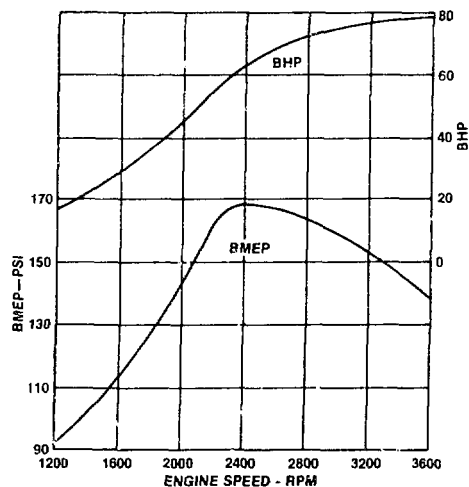


Figure 36—Proposed Full Load Curve-Low Emission Configurations.

High BMEP Testing

The objective of high BMEP testing was to define the high power capabilities of the low emission configuration of the Opel 2100D engine. The test engine was in the low emission configuration with two exceptions: (1) an eleven millimeter diameter plunger injection pump was installed to furnish the additional fuel required to achieve high power levels and (2) the pistons were modified by removing .080 inch from the top surface to simulate low compression ratio with VCR. The piston modification resulted in a compression ratio of 17:1 and a prechamber-main chamber volume ratio of 40:60. A possible

full load performance curve is shown in Figure 36. Higher power levels were reached at some speeds (175 PSI-BMEP at 2400 RPM), but are omitted in Figure 36 to produce a smooth power curve.

The eleven millimeter diameter plunger injection pump was used for high power tests as a matter of expediency. It has already been shown that the use of this pump will increase light duty NO_x levels. Because of time limitations, an optimized pump giving the best emissions and adequate high power fuel delivery was not available for test. The injection pump manufacturer indicated that the nine millimeter plunger diameter pump could probably be modified to meet the fuel delivery requirements.

A short test was conducted to quantify the effect of the eleven millimeter plunger on emissions. Consecutive tests with the nine millimeter and eleven millimeter plungers were conducted on the 29.5:1 compression ratio engine. One hundred, fifty and two percent load points were run at speeds of 1200, 1600, 2000, 2400 and 2800 RPM. Seventy-five and five percent loads were evaluated at 600 RPM. All points except the 100% load points were evaluated with and without EGR. Injection timing was varied manually to give the same point of ignition. With the eleven millimeter plunger, NO_x increased twelve percent (+12), CO increased four percent (+4), and HC decreased thirty-one percent (-31). A significant part of the decrease in HC resulted from the elimination of trace secondary injections occurring at 2400 RPM with the nine millimeter plunger. Ignoring this point gave a sixteen percent decrease (-16) in HC with the eleven millimeter plunger. The

larger plunger decreased HC, fuel consumption, and smoke at medium and high loads. At light loads, the larger plunger caused slight increases in HC, fuel consumption and smoke. The overall reduction in HC suggests that injection timing could probably be retarded further resulting in a very small NO_x penalty for the use of the eleven millimeter plunger.

In order to determine what emission levels could be expected in heavy duty applications, emission results were calculated for the thirteen-mode heavy duty emission test. Thirty-two hundred RPM was used as the rated speed and twenty-four hundred RPM was used as the peak torque speed. Emissions were calculated with and without EGR. Results are shown in Table 13.

Table 13—Thirteen Mode Gaseous Emissions.

	NO _x	GRAMS/HP-HR NO _x + HC	HC	CO
With EGR	2.26	—	0.46	2.85
Without EGR	2.74	—	0.49	2.68
Current Standard	—	16	—	40
Proposed Standard	—	10	1.5	25

The engine easily meets current or proposed standards and EGR would not be required.

CONCLUSIONS AND RECOMMENDATIONS

Conclusions

Conclusions concerning the potential of the AD-209 lightweight diesel concept will be discussed in the areas of emission, fuel economy, durability, size and weight, and producibility.

Emissions

The AD-209 concept has demonstrated the ability to meet any current and known projected limits on gaseous emissions. The techniques used to achieve emission reduction are sufficiently flexible to permit tailoring an engine to less strict emission requirements. At the one gram per mile NO_x level, a minimum improvement of ten percent in fuel consumption would be realized. Because the engine employs lean air/fuel ratios and modulated EGR, it is believed that the level of unregulated emissions such as smoke, sulfates and odor, could also be low.

engined cars of 110-150 BHP and 3000 - 3500 pounds curb weight, and the AD-209 engine in a 3200 pound curb weight vehicle. All vehicles use automatic transmissions. When compared to the average of the five gasoline cars, the AD-209 results in a 95% improvement in urban fuel economy and a 65% improvement in highway fuel economy. When compared to the best gasoline engine results, the AD-209 results in an 80% improvement on the urban cycle and a 50% improvement on the highway cycle.

In previous comparisons to gasoline engines, the diesel has been penalized due to the higher energy content of diesel fuel. This penalty is not appropriate to real world circumstances. On a volume basis, diesel fuel is less expensive to buy and requires less energy to produce than gasoline. At current prices, the diesel user will pay ten to fifteen percent less per gallon than the gasoline user. Viewing the situation from an energy standpoint, the production of diesel fuel requires five percent of the energy in the crude oil, whereas the production of gasoline requires ten percent (6). Using either viewpoint, any penalty should be attached to gasoline. In addition, the multifuel capabilities of the AD-209 should permit the use of low cost, broad cut fuels. The improved fuel consumption of the diesel engine provides a secondary benefit. The use of diesel engines will permit compliance with legislated fuel consumption standards while maintaining more adequate vehicle size for family use.

Durability

The reliability and long life of diesel engines is well known. VCR pistons have accumulated over 30,000 hours of testing on the AVCR-1360 engine including six successful 400 Hour NATO Endurance Tests. A reduced cost version of the VCR piston is currently under development and has successfully completed over 200 hours of endurance testing.

Size & Weight

The AD-209 has been shown to be equivalent or reduced in size when compared to six cylinder gasoline engines of comparable displacement. The AD-209 projected weight is less than current gasoline engines of comparable power. The unique characteristics of VCR make this light weight possible. The use of VCR permits operating at high power levels with controlled peak cylinder pressure, thereby allowing the engine basic structure to be very light compared to conventional diesel engines.

Producibility

Because the AD-209 engine is similar in construction to current gasoline engines, extensive modification to existing production lines would not be required to produce the AD-209.

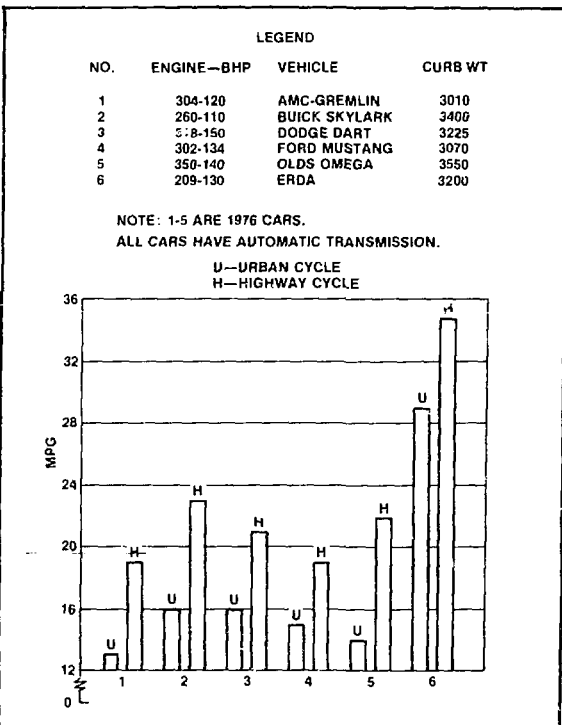


Figure 37—Fuel Consumption Comparison.

Fuel Economy

A comparison of urban and highway cycle fuel economies is shown in Figure 37. The vehicles listed are five 1976 gasoline

The unique advantages of VCR would also permit converting an existing gasoline engine to diesel operation. Because of the simplicity of conversion to diesel production, an engine utilizing the AD-209 concept could be introduced much more quickly and cheaply than other alternate engines now under consideration. The similarity of the diesel engine to gasoline engines would limit the vehicle changes required to a very few minor items.

Recommendations

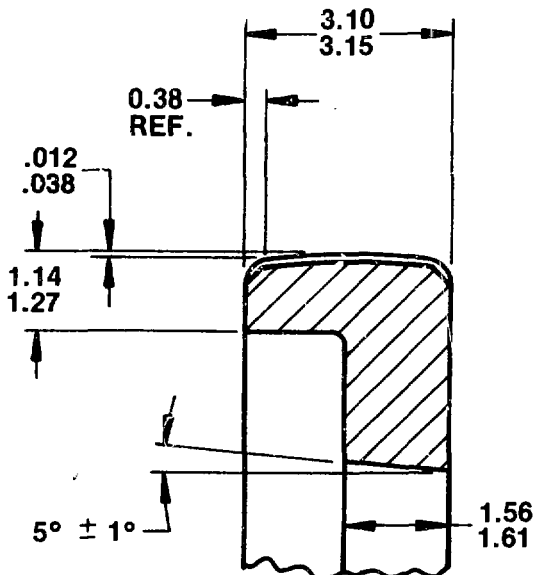
It is recommended that additional work be conducted in the following areas to further develop and demonstrate this engine concept:

1. Design, develop and procure VCR, EGR and injection pump hardware to permit engine installation and test in a vehicle. Conduct vehicle tests to further verify simulation results.

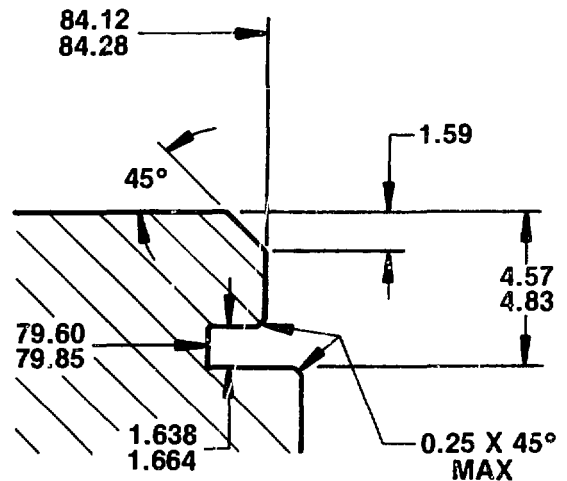
2. Convert an existing gasoline engine to a VCR diesel.
3. Define unregulated emission levels. If required, conduct development testing to reduce unregulated emissions.
4. Conduct long-term durability testing.
5. Conduct cold start tests.
6. Conduct more extensive multifuel testing and develop emission control systems for use with these fuels.
7. Conduct cost studies to determine production costs.

APPENDIX

SEALED POWER HEAD LAND RING



**HEAD LAND RING
CROSS SECTION
(CHROME PLATE FACE)**



**PISTON MACHINING
FOR HEAD LAND RING**

VCR PISTON PRINCIPLE

The Continental® piston is an automatic, hydraulically actuated assembly. It consists of two main parts; the piston ring carrier and the piston pin carrier.

The piston pin carrier, connected to the connecting rod; travels a definite path, whereas, the piston ring carrier is free to move within predetermined limits relative to the pin carrier. This movement provides a variable height from the center of the piston pin to the top of the piston crown. The variation in compression ratio is achieved through a change in combustion chamber clearance volume. Movement of the piston ring carrier is restrained hydraulically by engine lubricating oil in the upper and lower chambers.

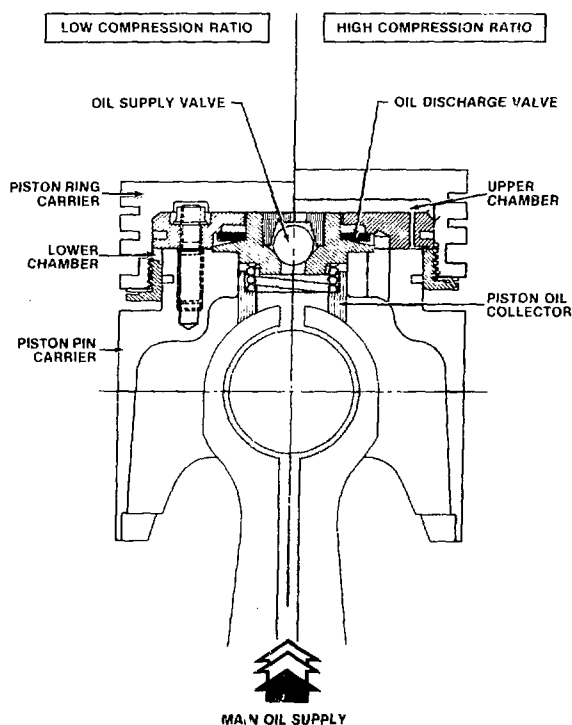
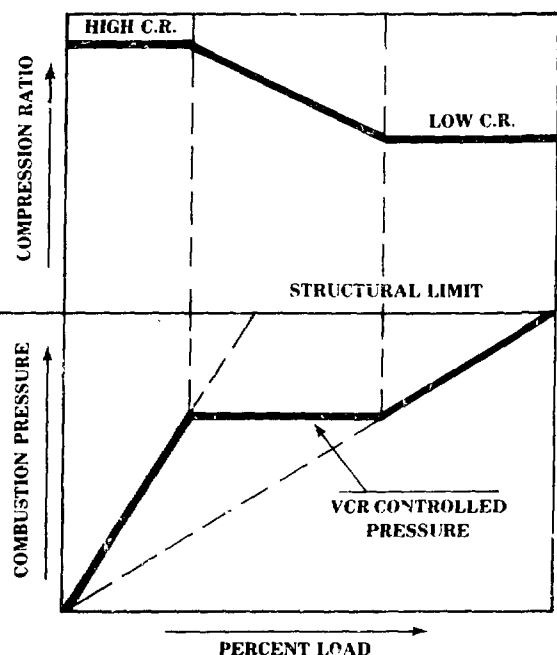
Oil from the lubricating system is fed through the connecting rod to the non-return piston oil supply valve by the piston oil collector. As the upper chamber fills with oil, the piston ring carrier moves to its extreme position or maximum

compression ratio. Oil also enters the lower chamber through the orifice connecting the two chambers.

The piston remains in the high compression ratio mode until power demand increases and the force of combustion pressure causes the pre-set oil discharge valve to release oil from the upper chamber. The VCR piston regulates combustion pressure until the low compression ratio is reached.

Inertia forces tend to force the ring carrier to move upwards during the end of the exhaust and the beginning of the intake strokes.

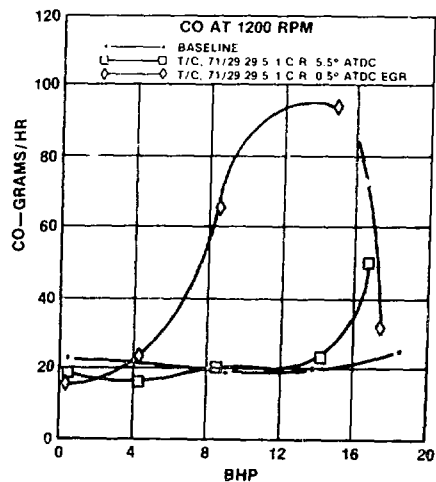
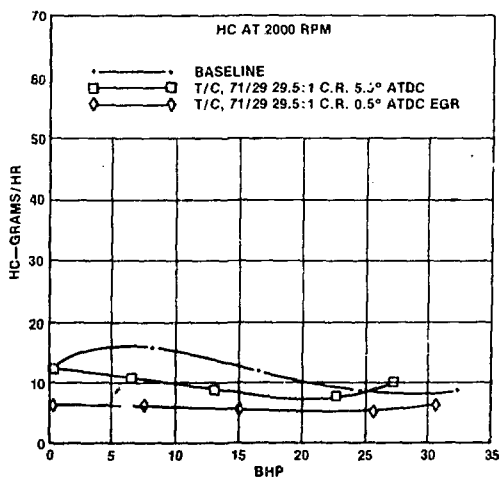
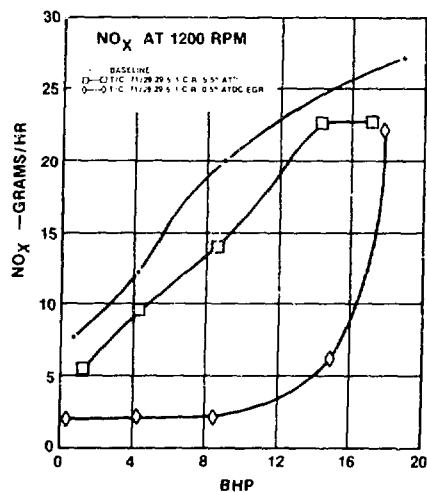
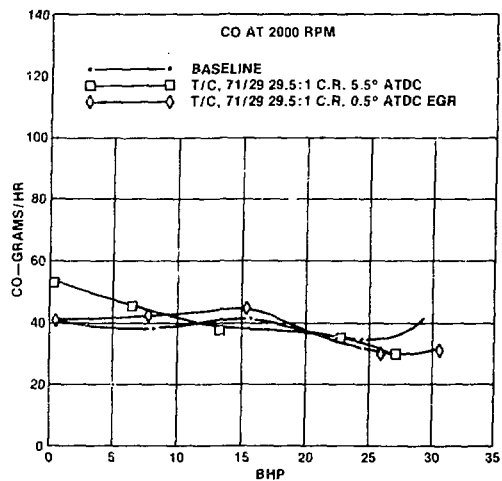
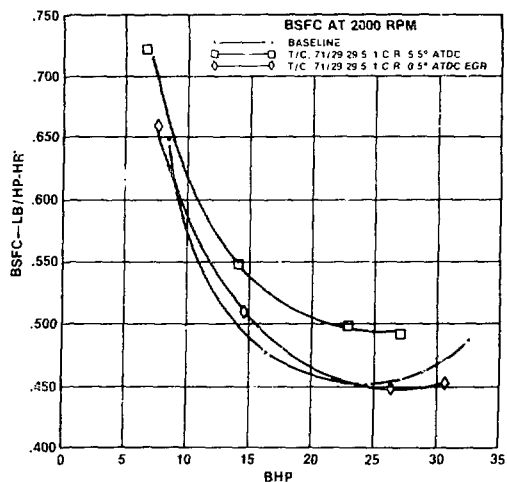
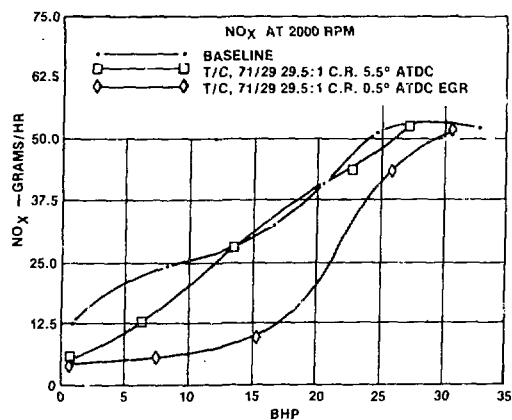
Oil in the lower chamber, which has a fixed orifice to the upper chamber, controls the amount of movement between the piston pin carrier and ring carrier during the exhaust and intake strokes. As combustion pressures are reduced, the ring carrier returns to the higher ratio until equilibrium is achieved.

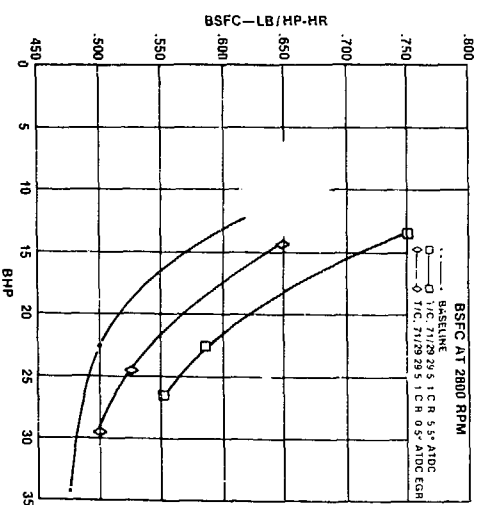
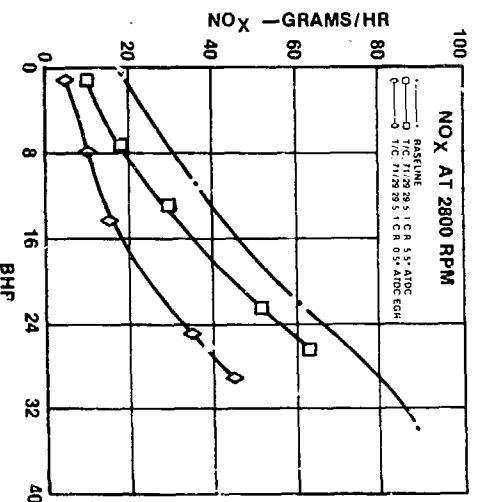
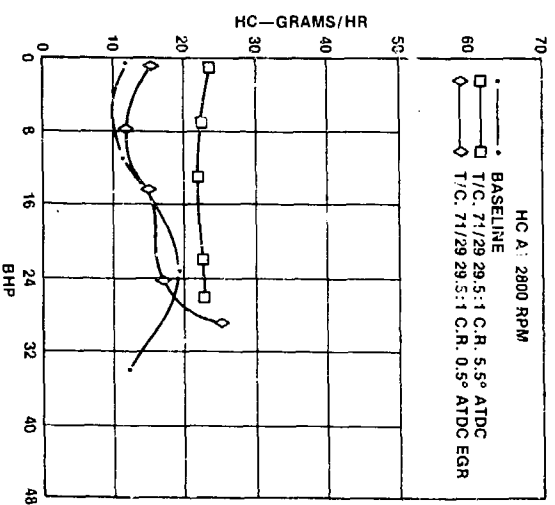
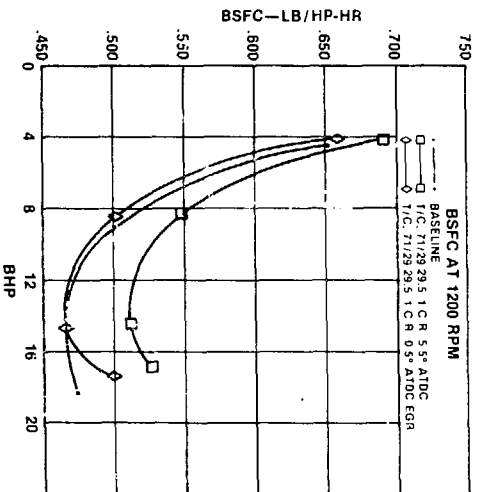
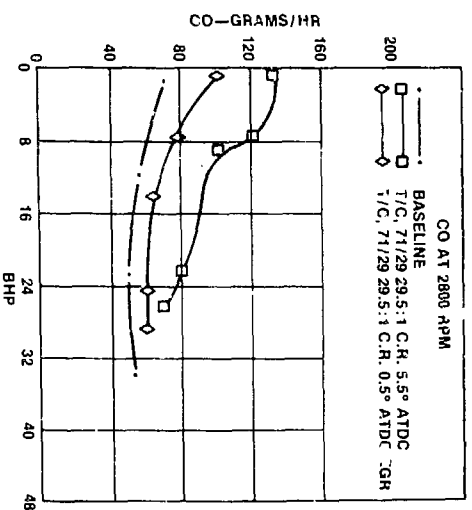
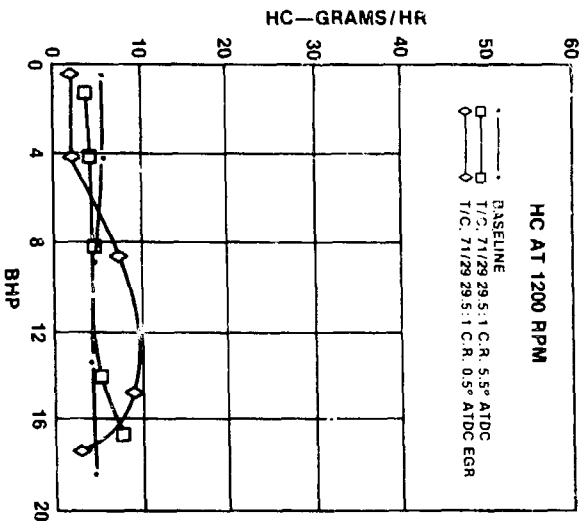


OPEL 2100D SPECIFICATIONS

Designation	Data
Operating Mode	4-Stroke Swirl Chamber Diesel Engine
Number of Cylinders	4
Arrangement	In Line
Bore - mm	88 (3.465")
Stroke - mm	85 (3.346")
Swept Volume, Effective - cm ³	2068 (126.2 Cu/Inch)
Power According to	
DIN 70 020 - Metric HP/RPM	60/4400
GMC Test 20 - Metric HP/RPM	68/4400
Torque According to	
DIN 70 020 - kgf.m/RPM	12.0/2500
GMC Test 20 - ft-lbs/RPM	92/2500
Compression Ratio	22:1
Prechamber-Main Chamber Volume Ratio	50:50
Average Piston Speed at 4500 RPM	12.5 m/sec.
Average Combustion Pressure	7.3 KGF/cm ² (103.83 PSI) at 2400 RPM
Combustion System	Ricardo "Comet" Mark V
Valves	Overhead, Driven by Camshaft & Rockers
Camshaft	Cylinder Head, Overhead, With 5 Bearings
Crankshaft	5 Bearings
Timing Control	Through Timing Sprockets & Chain
Cylinders	In Engine Block Without Sleeves
Cooling	Water Circulation by Maintenance Free Circulation Pump

NO_x, CO, HC & BSFC DATA





DEFINITIONS

NO _x	—Oxides of Nitrogen
HC	—Total hydrocarbons as measured with a heated flame ionization detector.
Point of Ignition	—Point at which main chamber pressure curve is deviated upward from motoring curve.
Point of Injection	—Point at which needle lift curve deviates from horizontal.
Duration of Injection	—End of injections minus start of injection expressed in degrees crank angle.
Peak Cylinder Press	—Main chamber peak pressure.
"Head Land" Piston Ring	—An "L" shaped top piston ring located very close to the piston dome to minimize crevice volume.
TCM/GPD	—Tei Jyne Continental Motors General Products Division.
ERDA	—Energy Research and Development Administration.
A/F	—Intake air mass divided by intake fuel mass.
EGR	—Exhaust Gas Recirculation.
VCR	—Variable Compression Ratio.
CID	—Cubic Inch Displacement.
BMEP	—Brake Mean Effective Pressure.
RPM	—Revolutions Per Minute.
BSFC	—Brake Specific Fuel Consumption.
AD-209	—Advanced Diesel concept engine of 209 cubic inch displacement.
[°] B or ATDC	—Crank angle degrees before or after top dead center position of piston.
PSI	—Pounds per Square Inch.
DF2	—Number 2 Diesel fuel.
BHP	—Brake Horsepower.
GHP	—Gross Horsepower.
MPG	—Miles Per Gallon.

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DIESEL ENGINE AND CONSERVATION

Roy Kamo

Cummins Engine Company, Inc.

Abstract

The diesel engine is rapidly gaining recognition and acceptance in the automotive field primarily because of its excellent fuel economy and good emission characteristics. For many years, the diesel engine was used solely in the heavy duty trucking industry, but it is currently attracting renewed interest in the medium and light duty truck and even in the passenger car. How long the diesel engine will continue to serve in the world's automotive conservation efforts will be determined largely by the success of the advanced diesel programs now underway. Some of the important programs discussed are the high specific output, compounding, adiabatic, minimum friction, and combustion improvement concepts. The advanced diesels resulting from those works are presented in terms of their potential capabilities and limitations regarding performance, sociability and their influence on conservation.

Introduction

The contemporary diesel engine has played a major role in providing one of the most efficient heat engines known to the world with moderately low emission characteristics. Progress is being made by industry and other organizations and institutions to further improve on the performance and sociability of the diesel engine. Dieselization of the medium and light duty trucks is just beginning and dieselization of passenger cars is seriously being considered by major automotive manufacturers all over the world.

The high speed passenger car and light truck diesel engines are invariably of the prechamber type, while medium duty and larger trucks use direct injection diesel engines. The heavy duty truck diesels are normally turbocharged for emission and efficiency reasons. The medium duty, the light duty and even the passenger car diesel engines are considering turbocharging for added performance.

Although the paper considers all types of diesel engines for transportation, the emphasis is on the larger heavier duty truck engines. The diesel engine performance and sociability characteristics are presented for continued consideration as vehicular powerplants of the future.

Prechamber Diesel Engine

There are many types of prechamber diesel engines. Today, these prechamber engines are used primarily in the smaller high speed engine class. Figure 1 shows a cross section of one of the well-known prechamber diesel engines which is used all over the world.

Several advantages of the prechamber diesel engine over the direct injection version are:

- Higher speed operation
- Excellent emission characteristics
- Lower noise levels
- Lower cylinder pressures
- Less stringent injection requirements

In spite of these advantages, the prechamber diesel has decided drawbacks when compared to the direct injection engine; namely:

- 5-10% fuel consumption penalty
- Poorer cold start capability
- Higher piston thermal loading

The above considerations indicate why the prechamber engine is preferred in passenger car application. The emission characteristics are quite similar to those of a stratified charge engine and are shown in Figure 2. The fuel consumption and performance are shown in Figure 3.

Direct Injection Diesel Engine

There are two major types of direct injection engines; namely: high swirl and quiescent chamber. The high swirl versions are commonly used in Europe and employ a jerk pump injection system. The quiescent chambers, like those used by Cummins, employ a high pressure injection system. The high swirl version requires air motion for mixing the fuel spray being issued from a relatively large hole injection nozzle. The number of injection holes is generally four and is less than the quiescent chambered counterpart. The high swirl cannot be obtained without increasing pumping losses through the intake port.

The emission characteristics of the two systems are rather similar and the fuel consumption and emissions trade-off may slightly favor the quiescent chamber. The differences between the high swirl and quiescent chambered systems are compared in Figure 4 and Figure 5.

High Output Diesel

Since the air-fuel ratio in a diesel engine is somewhat leaner than the Otto gasoline engine counterpart, for equal displacement the diesel engine develops less power. To increase the power output, more air must be pushed into the combustion chamber so that more fuel can be injected.

The diesel engine can be easily turbocharged to increase its power output and reduce its specific weight. Unlike the Otto gasoline engine, only air is boosted into the combustion chamber, and the injected fuel is tailored to the desired load. Danger from preignition or knock is non-existent. The air-fuel ratio in a turbocharged engine is generally leaner, i.e. 24:1 as compared to 19:1, for the naturally aspirated diesel. Thus, less smoke and cleaner exhaust emission can be expected from the turbocharged diesel.

Figure 6 illustrates the three classifications for increasing the BMEP in a diesel engine. Each advancing stage in increasing BMEP introduces a new problem which has to be coped with separately. For example, as boost is increased, the peak pressure in the combustion chamber may endanger the mechanical integrity of the engine. Thus, the compression ratio must be reduced or the injection timing retarded. In either case, other problems could again develop if the compression ratio is reduced or the injection timing is retarded.

Nonetheless, in the U.S.A., the majority of diesel engines are turbocharged for reasons of performance and sociability, which includes noise levels. Thus far, the turbocharged diesel has been able to provide the improved performance and still meet the sociability legislations.

Otto Competitive Diesel

In spite of its outstanding fuel economy, the diesel engine has not been used in this country in the passenger car or the light duty vehicle class because of the following major reasons:

- Cost
- Weight
- Lack of performance
- Noise
- Odor and smoke

The above problems can be attributed to the fact that the diesel engine must operate at higher air-fuel ratios and lower speed than the Otto engine. Thus, supercharging is immediately suggested. However, in order to keep the peak cylinder pressure low so that heavy engine construction need not be resorted to, a low compression ratio diesel is the next suggestion. This, from Figure 6, is the class III approach in the class I BMEP range. A combustion aid will be necessary for cold starting. Aftercoolers are not needed in this category, as indicated in Figure 6. Thus, an Otto competitive diesel suggested here for the light duty vehicle embodies the following principles:

- Low compression ratio
- Combustion aid for starting and light load operation
- Supercharge with pressure exchanger such as the

Drown Boveri Comprex, because of:

- excellent low load/speed fuel economy
- excellent transient response
- built-in exhaust gas recirculation (EGR) for NO_x reduction
- excellent smoke characteristics

Indeed, the initial design concept of such a powerplant has been made and surprisingly versatile performance seems to be possible. The expected performance and sociability parameters are tabulated and compared with the existing diesels as well as the Otto gasoline engine counterpart in Figure 7. The essential components of the proposed engine are shown in Figure 8. A similar concept has been built and tested under ERDA support (Reference 1) to Continental Motors on a VCR (variable compression ratio) piston engine using a conventional turbocharger. The VCR piston diesel engine concept is shown in Figure 9. Excellent performance and sociability results are reported.

Compounding

In May 1975 ERDA¹ (Reference 2) presented their targets for each and every classification of engine and vehicle manufacturer to reduce fuel consumption in order to meet the objectives of America's project independence. This calls for a 20-23% fuel reduction for trucks and buses. Figure 10 shows the ramifications of such a goal. It is quite obvious the contemporary turbocharged aftercooled diesel engine will have a difficult time meeting ERDA targets. Therefore, some form of compounding becomes necessary. The turbocharged engine may be called a compound engine in a true sense, but compounding herein is limited to mean engines utilizing a drive system back to the crankshaft.

When one looks into the energy balance of a conventional diesel engine, the energy split is roughly equal for brakehorsepower, exhaust and cooling system as shown in Figure 11. Although optimizing of injectors, modifying swirl, etc. may improve the brakehorsepower a few percent, the largest available energy is in the exhaust and the cooling system. The cooling system energy also includes the frictional heat dissipated through the oil cooler via the engine cooling system. The above facts suggest the following modifications to the diesel engine so that it may continue to serve the needs of the world as the most efficient heat engine:

- Exhaust energy
 - turbocompounding
 - bottoming cycles
- Cooling water
 - turbocompounding with adiabatic combustion chambers
 - minimum friction engine design

A low pressure power turbine turbocompound engine is depicted in Figure 12. Although the overall power of the engine is increased 18%, only a 9% reduction in fuel economy is realizable because of the additional pumping work incurred by the reciprocator. Several immediate advantages are recognized in the turbocompound diesel engine:

- Retarded timing for good NO_x control since exhaust energy can be recovered

¹Energy Research Development Administration

- Ease of EGR, if necessary, since exhaust manifold pressures are generally greater than the intake manifold pressure
- It provides an exhaust energy recovery system for the adiabatic engine concept

Adiabatic Turbocompound Engine

The adiabatic turbocompound engine simply insulates the engine combustion and exhaust system as shown in Figure 13. The engine is allowed to run "hot" without:

- Any water cooling
- Any air cooling
- Minimum oil cooling only

Metal surface temperatures in the combustion chamber could approach 2000°F and thus advanced ceramic materials could be used to advantage.

The in-cylinder work of the adiabatic engine improves somewhat, but the greater contribution to the overall cycle is achieved through exhaust energy recovery via the turbocompound engine system. Thus, extremely low fuel consumption can be expected. Since the engine requires no cooling system, considerable reductions in overall installed weight, cost, and size can be realized (Reference 3).

In addition to those obvious advantages indicated above, the adiabatic engine can be expected to provide the following spin-offs:

- Multi-fuel capability
- Combustion efficiency improvement
- Combustion and mechanical noise
- Reduction in maintenance
- White smoke clear up
- Lower compression ratio operation

Some of the problem areas associated with the adiabatic turbocompound diesel engine will be:

- Lubrication
- Wear
- Reliability/durability

The adiabatic turbocompound engine opens up a new dimension in mechanical engine design, combustion, and engine tribology. However, early indications are that significant gains can be made in the BSFC-BSNOx¹ tradeoff.

Bottoming Cycle

The adiabatic turbocompound engine, in spite of exhaust energy extraction via the turbocompound device, still possesses considerable exhaust energy upon expansion. This is so because the initial turbine inlet temperatures are considerably higher than for today's turbocharger engines (1485°F as compared to 1150°F). Consequently, temptation presents itself to harness the sensible exhaust heat via a bottoming cycle.

Although several forms of the bottoming cycle have been considered in the past, the best known one is the Rankine bottoming cycle (Reference 4). Upon successful harnessing of the Rankine bottoming cycle, an additional 13 to 17% improvement in fuel consumption can be expected. This should reduce the BSFC down to around 0.24 lb/hp-hr, which is approaching the thermodynamic air standard cycle efficiency.

¹ Brake Specific Fuel Consumption-Brake Specific Oxides of Nitrogen

Diesel Engine and Emissions

The above presentation illustrated what can still be done to the diesel engine to preserve its leadership role in the world's conservation effort. Although superior to the Otto gasoline engine in terms of exhaust emission, the diesel engine has difficulty meeting achievable emission levels of the closed cycle external combustion Stirling cycle and the continuous combustion Brayton cycle.

The competitive position of the diesel based engines in terms of emissions (BSNO_x) versus brake specific fuel consumption is shown in Figure 14. There are several ways still available for reducing diesel engine emissions. They are:

- Combustion improvement
- Engine mechanical efficiency improvement
- Retarded timing
- Cooled inlet air
- Exhaust gas recirculation

How the above parameters affect the BSFC-BSNO_x tradeoff is shown in Figure 15.

In this paper, only NO_x was considered. The reason being the carbon monoxide and unburned hydrocarbon emissions are not considered a problem in the diesel engine. This can be attributed to its excess air heterogeneous combustion characteristics.

Conclusions

The diesel engine has been shown to possess moderately good emission characteristics at excellent fuel consumption. The fuel consumption and emissions can be further reduced by the adiabatic turbocompounding of the diesel based powerplant. The Rankine bottoming cycle offers further improvements, but its practicality needs to be demonstrated. It is expected that the application and usage of the diesel engine will continue to expand and continue to contribute to the world's conservation effort.

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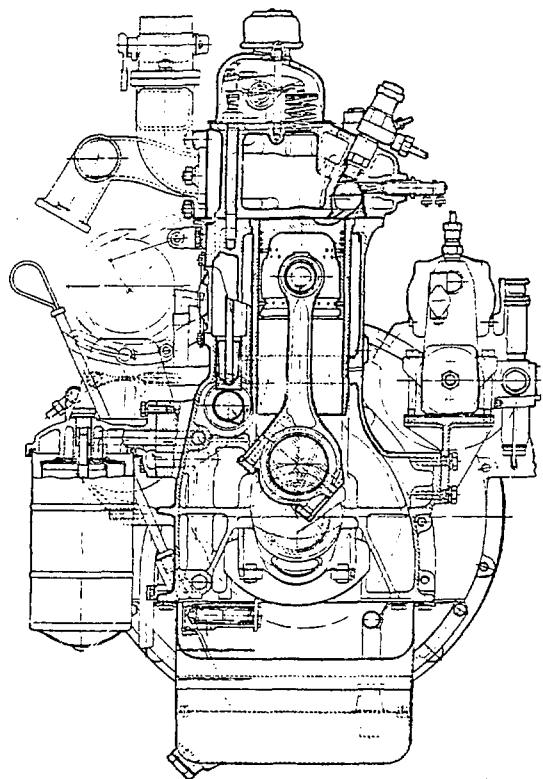


FIGURE 1. TYPICAL SWIRL PRECHAMBER ENGINE

FIGURE 2.

TYPICAL EMISSIONS FOR 2 LITER PASSENGER
CAR DIESEL OPERATED ON CVS CYCLE.

HC 0.4 G/MILE

CO 1.5 G/MILE

NO_x 1.5 G/MILE

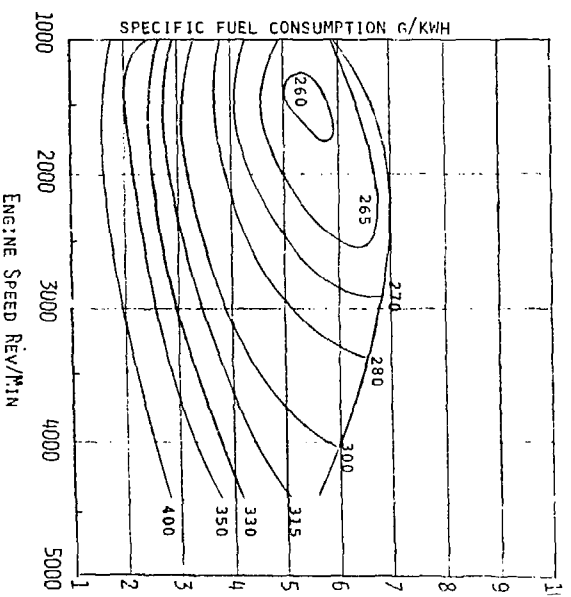


**FIGURE 3. SMALL SWIRL TYPE PRECHAMBER
ENGINE PERFORMANCE**

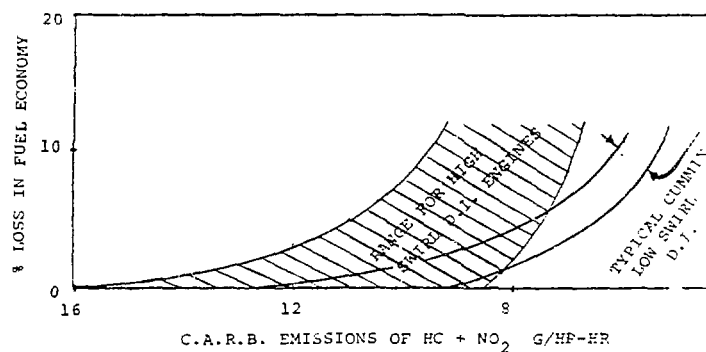


FIGURE 4. FUEL CONSUMPTION LOSS FOR HIGH SWIRL AND LOW SWIRL D.I. ENGINES

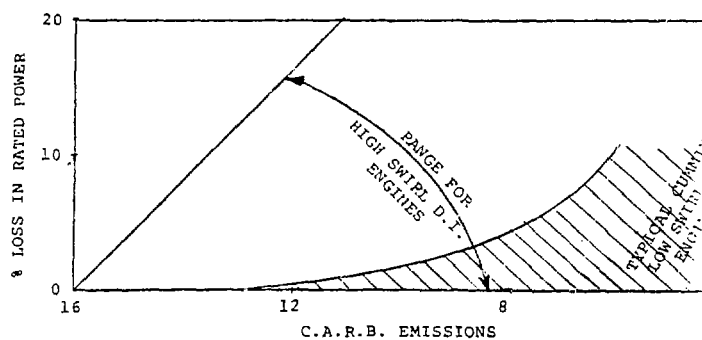


FIGURE 5. POWER LOSS FOR HIGH SWIRL AND LOW SWIRL D.I. ENGINES

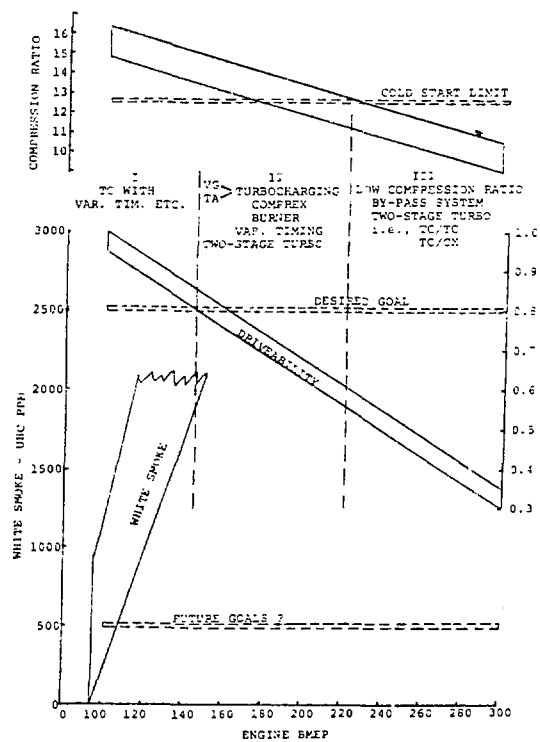


FIGURE 6. THREE CLASSIFICATIONS OF HIGH OUTPUT DIESEL ENGINE

FIGURE 7
ESTIMATED PERFORMANCE AND EMISSION TARGETS
OF PROPOSED ENGINE

	<u>CURRENT DIESEL</u>	<u>PROPOSED DIESEL</u>	<u>CURRENT GASOLINE</u>
SPECIFICATION:			
DISPLACEMENT, CU. IN.	134	134	134
VEHICLE WEIGHT, LB	3000	3000	2900
SPECIFIC VEHICLE WEIGHT (LB/HP)	46	30	25
PERFORMANCE:			
FUEL ECONOMY, MPG*	20-25	20-25	13
HORSEPOWER @ RPM	65 @ 4200	100 @ 4200	116 @ 5000
ACCELERATION, 0-60 MPH (SEC.)	26	15	13.7
MAX. SPEED, MPH	84	100	104
EMISSION:			
HYDROCARBON, G/M	0.35	0.4	2.7
CARBON MONOXIDE, G/M	1.79	2.0	32.3
NOx AS NO ₂ , G/M	1.55	0.7	3.6

*OVER FEDERAL DRIVING CYCLE

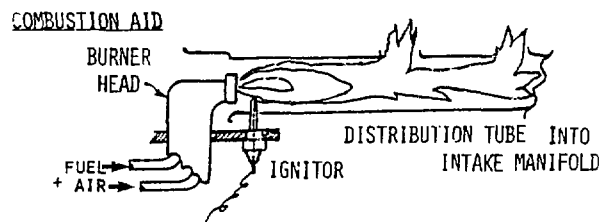
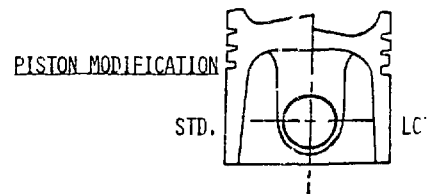
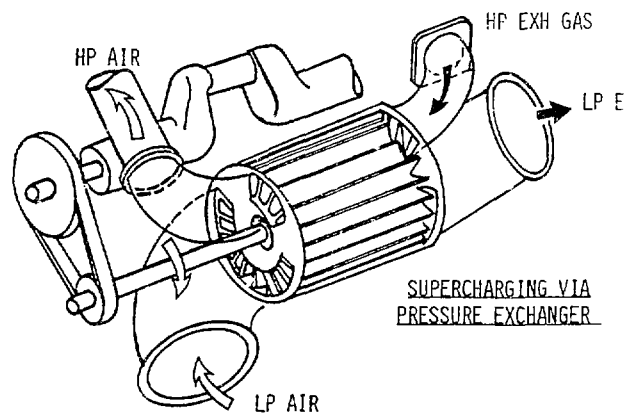


FIG. 8. BASIC LOW COMPRESSION RATIO (LCR) ENGINE CONCEPT

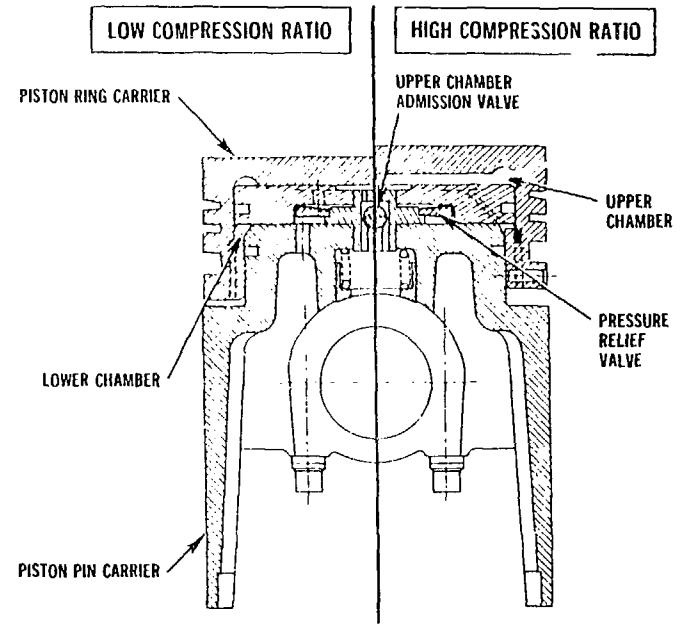
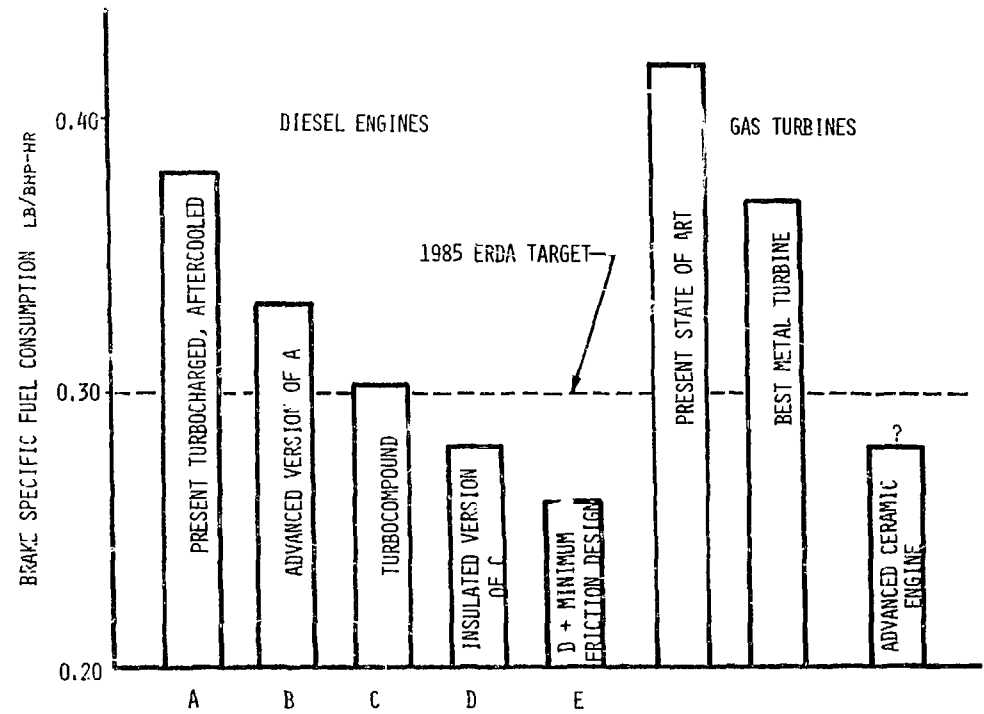


FIGURE 9. CONTINENTAL MOTORS VARIABLE COMPRESSION RATIO PISTON



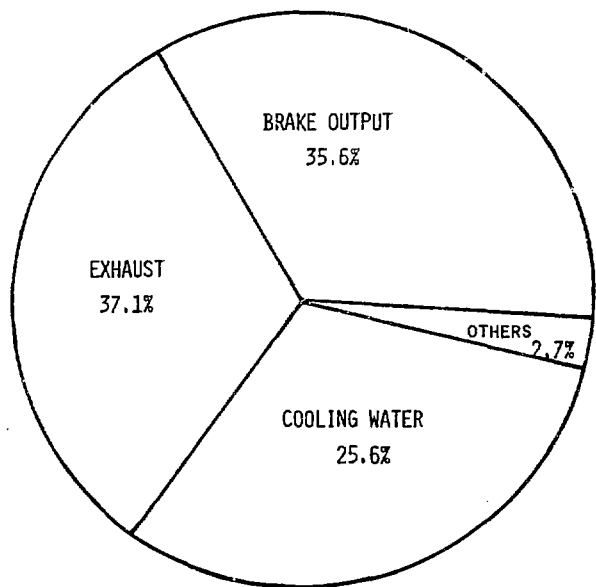


FIG.11. TYPICAL ENERGY BALANCE OF
DIESEL ENGINE

FIGURE 12
TURBOCOMPOUND ENGINE
WITH
LOW PRESSURE POWER TURBINE

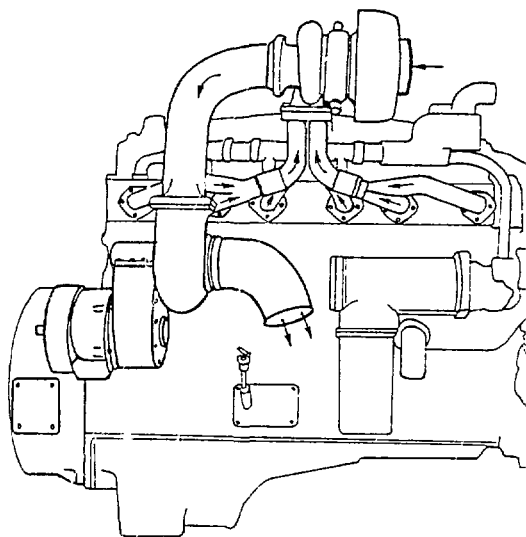
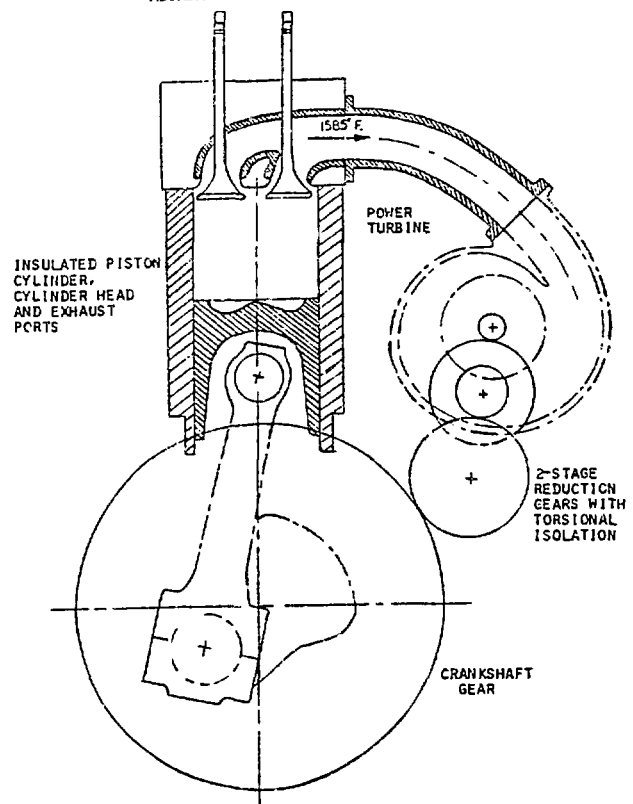


FIGURE 13 TOTAL ENERGY RECOVERY VIA CUMMINS
ADIABATIC TURBOCOMPOUND ENGINE



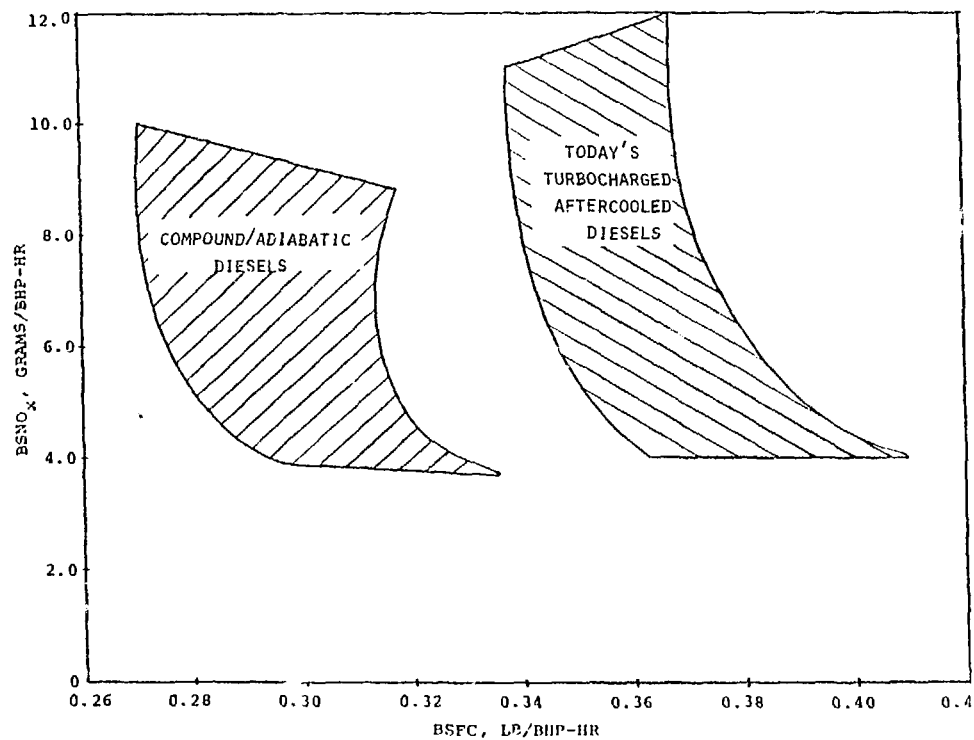


FIGURE 14. NO_x EMISSION RANGE OF DIESEL BASED ENGINES

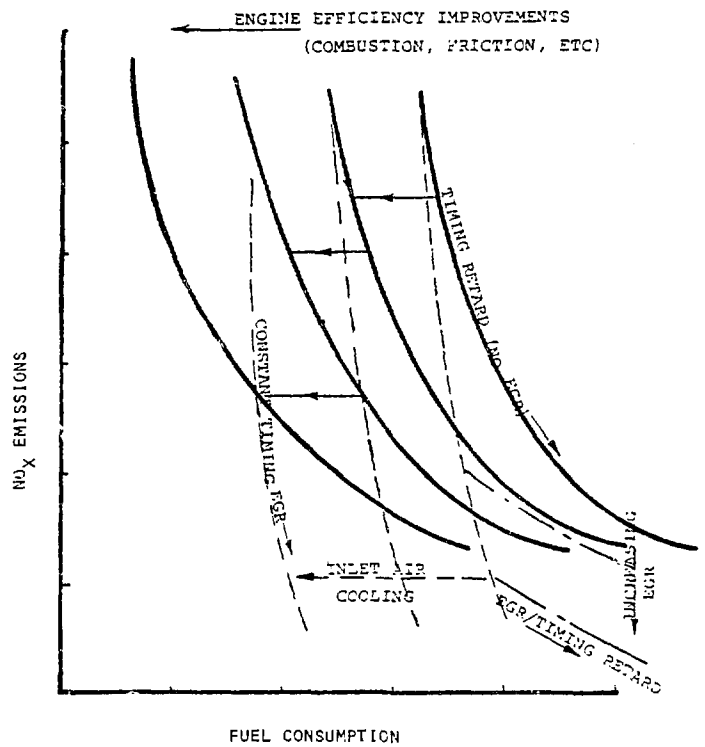


FIGURE 15. APPROACH TO LOW EMISSION DIESEL ENGINE