CHRYSLER'S AUTO TURBINE TRAVELS THE LONG ROAD HOME

Twenty years of development and experimentation, thousands of road testing miles, and a unique market research method are proving that a passenger-car gas turbine is practical, but room for improvement still exists.

The development of a gas turbine engine suitable for passenger-car use presents many complex technical problems in relatively unexplored areas. Chrysler engineers faced the task of producing components in the small sizes required for automobile use with efficiencies equal to those being obtained in large engines. It was necessary to build a highly effective heat exchanger with low pressure drop. Low cost, high-strength, heat-resistant materials were a mechanical and economic must. Also flexibility of operation, low noise level, adequate engine braking, and reasonable gas generator acceleration time were required. Exhaust gas temperatures had to be low, and the engine had to be light, reliable, and easy to maintain. From the cost aspect, an automobile turbine would have to be competitive with conventional reciprocating engines.

The reliability, efficiency, and minimum service requirements common to all gas turbine usage are augmented in the passenger-car application by additional conditions, which are quite foreign to most gas turbine history and experience. These unique conditions include:
1. Wide load variation, with stringent requirements on operating efficiency over the range.
2. Engine idle for a large portion of the duty cycle.
3. High dynamic response needs.
4. Strict noise level limitation.
5. Cost and production situations unparalleled in previous turbine experience.

Cycle considerations

In approaching the problem of what type of gas turbine is most suitable for passenger-car application, a number of possibilities present themselves, including: simple cycle, regenerative cycle, single shaft, free turbine, reheat, intercooling, and supercharged cycle. Cost, simplicity, and ease of manufacture are predominant factors in establishing an automobile engine design, and therefore rule out the use of such refinements as a supercharged cycle, reheat, and intercooling.

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**FIG. 1** Cycle efficiency of a regenerative gas turbine engine.

**FIG. 2** Fuel consumption for constant turbine inlet temperature.
FIG. 3  Gas flow and temperature values in Chrysler regenerative gas turbine engine.

It was decided that a regenerative cycle offered advantages that would be worth the penalty of increased complexity and cost. Since a passenger-car turbine must be capable of operating over a wide load range, with the major portion of operation in the low end of that range, the use of the simple cycle would result in prohibitively high fuel consumption. The regenerative cycle, on the other hand, exhibits the most pronounced advantage at light loads. The lower the pressure ratio, the larger is the proportion of energy available for regeneration.

The question of single or multiple rotor arrangement was also decided on the basis of load variation. A single-shaft gas turbine coupled to the drivetrain through a manual transmission would require excessive driver skill and dexterity. The useful speed range of the gas turbine is about 2.5/1 compared to about 10/1 for the reciprocating engine, and the torque change throughout the range is abrupt. A free turbine, utilizing a separate power turbine geared to the driveline, provides approximately the same characteristics as a reciprocating engine with a torque converter, since the two gas-coupled turbine rotors behave much like the elements of a torque converter. The only limitation on engine response to load changes then is the rate at which the gas generator can change speed.

Next, limiting values for cycle parameters had to be selected for the regenerative, free turbine cycle arrangement. These were largely determined by metallurgical considerations. Briefly, turbine inlet and outlet temperatures are the two factors most affected. The inlet temperature is limited in part by the relatively un Pressed ducting and stator vanes of the compressor turbine, and to a greater degree by the compressor turbine wheel blade and disc strength characteristics. Thus, the working range for the inlet temperature is defined by the choice of materials. The inlet temperature is identified as $T_\text{in}$. The exit temperature from the power turbine is an important limiting parameter because it is also the maximum regenerator inlet temperature, (identified as $T_\text{g}$), and is determined by the materials used for the heat exchanger (regenerator) matrix and attendant equipment. These two factors essentially establish a value for turbine temperature ratio ($T_\text{out}/T_\text{in}$), since neither temperature can exceed its limit for an extended period of time for given material selections.

The cycle pressure ratio is primarily influenced by three factors: 1. Cycle optimization: There is an optimum ratio for best cycle efficiency of a regenerative engine (Fig. 1). This value, however, is usually exceeded to favor part-load operation. 2. Component complexity and cost: The single-stage radial compressor is currently the simplest and lowest in cost, and works well in a regenerative engine. With such a unit, however, it is difficult to maintain high component efficiencies above a 4/1 pressure ratio. Fortunately, the two simple stages of the engine adequately fulfill the cycle expansion needs for this ratio. A higher ratio would require multistage turbines. 3. Rotor inertia: This important factor depends greatly on impeller and turbine wheel diameter, and places another
Variable power turbine nozzle blades maintain constant inlet temperature for best efficiency.

**Part-load considerations**

The design of the Chrysler turbine was notably influenced by the fact that a passenger-car engine spends a great percentage of its life at loads that are only a fraction of the design capacity.

The first clue to an efficient gas turbine is offered by the effect of maximum gas temperatures on cycle efficiency and specific fuel consumption at any operating load factor. As shown in Fig. 2, maintaining a constant turbine inlet temperature \( T_s \) at all loads results in a higher efficiency for the regenerative engine than the more conventional characteristic where \( T_s \) decreases at reduced output. One way to keep \( T_s \) constant is to vary one of the key restriction points in the engine as a function of load. This was done at the power turbine nozzles. By means of a hydraulic scheduling actuator, the nozzle blades are gradually closed from the design point setting as load is reduced, thus reducing that share of the overall pressure ratio that is allotted to the compressor turbine stage, and preventing \( T_s \) from increasing as rapidly as it would if all the turbine geometry was fixed.

With a regenerative free turbine powerplant and the material limitations mentioned, it is impractical to maintain \( T_s \) constant for all operation. As engine load and gas generator speed are reduced, the total turbine work decreases markedly and so does the turbine temperature-drop. With a constant \( T_s \), the low pressure regenerator inlet temperature \( T_s \) would rise quickly above its permissible maximum. The nozzle schedule is, therefore, established in such a way that one of these temperatures is always at its limit for any road load condition.

Fig. 3 shows design point selection for the automobile gas turbine, and thermodynamic values at various points in the cycle. Fig. 4 shows the interior arrangement of the engine components. Fig. 5 shows the ex-
ternal configuration as viewed from the compressor end.

**Powerplant arrangement**

After entering the engine through the intake filter and silencer assembly, the air passes through the intake elbow and axially into the inducer (Fig. 6) and impeller. The impeller discharges the air radially into the diffuser (Fig. 7), which channels it into the space between the engine housing and the chamber surrounding the burner. The air then passes through a semi-annulus around the front of the generator core, and is returned through the front half of the core. The heated air passes inward and down to the combustor, along the outside of the burner tube. The flow reverses through the annular slots and radial orifices in the tube, setting up a complex vortex flow pattern to stabilize combustion over the range of operation.

The hot gases, mixed to a uniform temperature \( T_s \), are guided by a scroll to set up a vortex flow to the compressor turbine nozzle and wheel. The high-velocity gases leaving the wheel are guided and diffused to the power turbine nozzle and its variable blades and then to the power turbine wheel. The exhaust gases are discharged radially, flowing outward through the rear half of the two regenerators, where heat energy is recovered for reuse in the cycle.

**Compressor** — The cast-aluminum impeller has 30 blades, 15 of which are partial, or splitter, blades. It is necessary to have a large blade area available to spread the aerodynamic load in order to prevent separation of blades and hub. Inlet eye blockage dictates the use of splitter blades, so that there are only 15 blades in the inlet, and aerodynamic stalling in the inducer required that a separate blade row be added in front of the main impeller. This blade row does virtually no turning at the design point; its function is to accept the wide flow angle variation over the engine operating range and thus prevent overloading the following blade surfaces.

**Turbines** — The type of turbines to be used in the compressor and in the power stages of the engine was determined primarily by such factors as simplicity, ease of manufacture, low inertia, and efficiency. The nature of the application dictated two stages: that is, a power turbine separate from the compressor turbine. Also, the power turbine should follow the compressor turbine in a flow sense. The power turbine must operate over a wide range of speeds from stall to maximum, as required by the vehicle speed. This results in a wide range of flow angles leaving the power stage, especially when compared with the compressor turbine. It is preferable to have the relatively constant output of the compressor turbine fed into an interstage passage and second-stage nozzles, and the widely varying output of the power stage fed into an exhaust diffuser, than vice versa. Since the compressor turbine handles the major share of the combined turbine output, the loss in efficiency that would result from a poor inlet flow would be unacceptable. However, it was possible to design the power turbine so that the velocity energy remaining in the stage discharge is considerably smaller than in the discharge from the compressor turbine. When the discharge flow is then led through the exhaust diffuser and a portion of this velocity energy recovered as static pressure, the effect of the dis-
Rotary disc regenerator is simple, self-cleaning.

Charge on the efficiency of the engine can be minimized when the power turbine is the final stage.

The aerodynamic design of the two stages was carried out by a conventional analysis which suitably approximated the three-dimensional character of the main flow, and blade contours were selected that demonstrated satisfactory pressure diagrams. The velocity diagrams shown in Fig. 8 summarizes the overall aerodynamic design.

**Regenerator** — The rotary disc type of regenerator was selected because of its compactness and simplicity. In addition, it is inherently self-cleaning because of the reversal of flow and the cyclic fluctuation of the core temperature during each cycle, so that it reduces fouling by combustion deposits.

The effect of regenerator thermodynamic performance and leakage on overall engine performance was studied at length. It should be noted that good engine performance requires high effectiveness and low pressure losses, but that these are conflicting conditions for establishing regenerator geometry. It was found that effectiveness and pressure loss ratios of 0.9/1 and 0.021/1, respectively, at idle, and 0.85/1 and 0.045/1 at full load were a satisfactory compromise to obtain good performance and a reasonable regenerator size, which resulted in acceptable leakage. This level of performance is achieved with an average Reynolds value of approximately 200, a length to hydraulic diameter ratio of about 100/1, a utilization factor of 0.25, and an axial conductance factor of approximately 0.003.

**Burner** — A single can-type burner (Fig. 9) of reverse-flow configuration was chosen as the most acceptable compromise between compactness and low cost. This type offers good ac-

A design point loss of $1\frac{1}{2}$–2% of maximum cycle pressure has been achieved.

**Component and engine testing**

*Compressor testing* — The first attempts at compressor testing were conducted in a standard dynamometer cell, but the requirements of the small-size compressor needed for the turbine engine were so specific that a special cell (Fig. 10) was built for the purpose. Here speeds up to 70,000 rpm, can be achieved, with drive power up to 600 hp. Use is made of static pressure taps, remotely controlled miniaturized total pressure and flow angle probes, total temperature probes, and occasionally hot-wire anemometer probes. Other test techniques include: flow visualization tufts, smoke filaments, vibration pick-ups, clearance rub rods, and boundary layer trip wires. Test data are programmed for a digital computer to obtain standard compressor parameters.

Compressor efficiency is based on a pressure-temperature relationship rather than on a pressure-torque relationship because evaluation of losses in the gearbox has given less consistent results throughout the life of the program. Torque-based efficiency has been both higher and lower than temperature-based efficiency within a single given performance test.

Over the 15-year development period the corrective use of the test data obtained and processed has resulted in improving the early maximum compressor efficiency of 0.742% to the 0.835% efficiency of the present unit in the gas turbine used in the Chrysler test cars of today.

Compressor testing is also done on complete engines to study the effect of other components, such as the diffuser and plenum, on compressor performance. Results of tests on the present engine will appear as improvements or refinements in future versions.

*Turbine testing* — Early test procedures were based on single wheel techniques; however, the new engine turbines were tested on a combined-stage fixture arranged as shown in
Fig. 11. The reason for this is the high degree of interdependence of the second (power) stage on the first. The discharge of the first stage is necessarily at a high velocity and a small radius for reasons of a low polar moment of inertia. The power turbine, on the other hand, requires a large diameter for good torque ratio as well as low through-velocities to achieve reasonable leaving loss. The transition region between the two stages tends to build up a heavy boundary layer that results in large power turbine nozzle secondary flows. It is also a region that must have considerable length, with consequent high mixing loss, in order to avoid severe diffusion on pressure gradients. For this reason the first-stage discharge cannot be simply considered a pressure level available to the power turbine section. It is a flow that, by its boundary layer and turbulence, as well as its pressure and velocity characteristics, determines significantly the level of the power turbine performance.

The combined-stage fixture design is based on the concept of parameter testing. It relies on the dynamic similarity between the fixture and the powerplant turbine sections at comparable Mach number to evaluate the work, speed, flow, torque, and efficiency characteristics. The fixture makes possible the testing of equipment at reduced temperatures, where instrumentation, heat transfer control, and strength of materials are not critical. Compensation of data values must be made, since the fixture is designed to operate at 800 F while testing engine-size hardware, and does not permit easy duplication of Reynolds numbers.

The measurements made in order to determine the parameter characteristics are:

- Weight flow.
- Turbine wheel torques.
- Total pressure at inlet and exit planes.
- Turbine inlet and outlet total temperature.
- Wall static pressure.

The data analysis is formal, to some
Development of a burner is an art, not a science.

degree, in that it is loaded into a computer program for various efficiency and parameter calculations. Further analysis of wheel inlet conditions, relative Mach number, and flow angles is then made with the assistance of a more involved program. This adds the requirements for input constants, such as areas and blade row efficiency, which are adjusted until there is agreement with the overall performance measurements.

In the combined-stage turbine test two other important flow elements are also analyzed. These are: first, loss coefficient and discharge uniformity of the vortex scroll at the inlet to the first-stage nozzles; and second, radial total pressure of the exhaust diffuser at the inlet of the second-stage wheel, and the plenum pressure at its outlet.

Burner testing — Although the fundamental concepts governing combustion system operation are understood, they cannot be accurately predicted.

Thus, development of a burner is an art, rather than a science, with final system analysis performed using complete powerplant assemblies.

The parameters considered in combustion system development are:

- Combustion efficiency.
- Exit temperature distribution.
- Pressure drop.
- Metal temperatures (life consideration).
- Fuel adaptability.
- Ignition characteristics (including low ambient temperatures).
- Carbon formation tendencies.
- Exhaust smoke and odor.

Lean limit, a common parameter in burner testing, is not of importance in an automobile engine since the fuel flow is stopped during engine decelerations, thus extinguishing the flame.

A nonregenerative engine housing burner test fixture (Fig. 12) is used for determining the starting cycle and low-temperature requirements of fuel atomization, ignition, and combustion quality within the required range of fuels. The fixture is essentially a complete engine without a power turbine wheel, modified so that the exhaust gases do not preheat the air prior to combustion.

A regenerative engine housing burner test fixture (Fig. 13) is used as a test bed for both transient and steady-state engine operation. This fixture is a complete engine, less power turbine wheel, with provisions for regulating the amount of air pre-heating prior to combustion.

Regenerator testing — For test and evaluation purposes the regenerator, as used in the current engine, was considered an assembly, including matrix or core, static and rubbing seals, inlet and outlet flow passages, and drive system. Overall performance data of such an assembly must include:

- Thermodynamic behavior.
- Leakage or sealing.
- Wear.
- Friction.
- Compatibility of the rubbing seals.
- Power requirements.
- Bearings and drive system.
- Ease of manufacture.
- Endurance (life consideration).

The general development fixture
Turbine Exhaust Brews “Instant Heat”

In the Chrysler turbine car exhaust gases are substituted for liquid coolant in heat exchanger of modified forced air heating and ventilating system.


Cold weather testing demanded that a practical heater be included in the turbine car’s equipment. The optimum heat source available was the engine. The design objectives: utilization of the engine’s hot gases; a heating capacity and air distribution comparable to a conventional coolant fluid heater; economy of operation; and maximum use of existing heater components and basic design features.

The use of the turbine gases as the energy source for car interior heating fulfilled two of the design objectives. The desirable “instant heat” feature of a fuel-fired heater was retained, since turbine gases heat up even as the engine is being started.

The basic heater unit is a modification of the forced air and ventilating system used on Plymouth and Dodge. It uses an axial blower to draw in fresh air from outside the car and pass it over the heat exchanger section of the car heater. Capacity of the blower is 225 cfm with the car at a standstill. The heater also functions as a defroster and as a ventilator.

The hot gases were tapped from the exhaust side of the power turbine, prior to passage through the left side regenerator. This position causes no power loss to the engine, and provides gas at approximately 900 F at the desired low pressure. Tapping the exhaust after passing through the regenerator would have been unsatisfactory, since during winter operation at an ambient -10 F the exhaust temperature is approximately 100 F and at too low a pressure level.

Due to the relatively high temperature of the gases passing through the heat exchanger section (Fig. 1) as compared to liquid coolant temperatures, it was decided that temperature control of the heater would be achieved by throttling of the hot gases. This is accomplished by a thermostatic hot gas control valve mounted on the engine side of the dash. Hot gas flow to the heater core is controlled by this valve, whose initial position is set by positioning of the temperature control lever on the instrument console. Thereafter, the flow of hot gas to the heater core is maintained and regulated at the desired heater discharge temperature level by means of a temperature sensing element located in the airstream on the discharge (passenger) side of the heater core.

The hot gas valve is automatically closed by means of a temperature control valve actuator and linkage whenever the heater control push buttons are in the “off” or “cool” positions, regardless of the position of the temperature control lever. This is a safety precaution to eliminate the possibility of an extreme temperature buildup in the heater core with subsequent discharging of hot blasts of air on the car occupants when switching from “off” or “cool” to “heat” or “defrost.” A pressure switch and solenoid, mounted on the heater box, control the feed of engine pressure to the temperature control valve actuator. In the “cool” position engine pressure is removed from the pressure switch, thereby de-energizing the solenoid. In the “off” position engine pressure is removed from the solenoid. For either of these conditions, engine pressure is removed from the control valve actuator, and the spring-loaded temperature control valve immediately closes, shutting off hot gas flow to the heater core. The hot gas valve is also designed to close if the temperature of the capillary heat sensing bulb in the discharge flow of air should reach an excessive value.
Test fixtures are used to evaluate static seals.

(Fig. 14) has been designed to allow duplication of the engine environment, including inlet and outlet flow passages. Regulated air is supplied by the laboratory air system and is heated in an engine-type burner to the compressor outlet temperature. The air flows through the high-pressure side of the core, is throttled to turbine outlet pressure, reheated in a second burner to turbine outlet temperature, and then flows through the low-pressure side of the core. The core is cradled to determine torque requirements, and is rotated by a variable-speed air motor.

The thermodynamic performance of the regenerator is expressed as effectiveness and pressure drop versus Reynolds number, which is a function of gas generator speed. The general fixture has pressure and temperature instrumentation installed at both core faces, measuring performance and flow distribution as affected by inlet and outlet flow ducts. Both radial and circumferential pressures can be measured. Matrix heat storage utilization factor, as influenced by core rotational speed, can be determined for all flow conditions, for use in selecting the optimum regenerator drive ratio.

Other test fixtures are used to evaluate the static seals and matrix bonding at room temperature and elevated pressures. A subcomponent type of fixture is used for evaluating matrix heat transfer characteristics and thermal shock resistance.

Accessory testing — Since all the accessories are driven by the gas generator, very close attention had to be paid to factors influencing their collective moment of polar inertia, and to its minimization.

The accessory system includes the fuel control system, air pump for the fuel nozzle, starter-generator, lubrication pump, and power turbine noz-
zzle actuating system. Also included are all associated gears, shafts, and bearings.

A number of problems were encountered in the accessory area. The air pump, which accepts compressor air and boosts it 5 psi for use in the atomizing burner nozzle, had to operate at 500 F without lubrication. The major problem to be solved was one of material compatibility between cylinder and piston.

Another design problem involved the power turbine nozzle actuator. Its four positions, or ranges, for idle, economy drive, maximum power, and engine braking are modulated as a function of engine load and vehicle speed. The engine load is sensed by throttle position, and car speed is sensed by the automatic transmission pressure. The "idle" position sets the blading (Fig. 15) for minimum fuel flow and creep torque; "economy" for maximum allowable turbine outlet temperature; "maximum" for rated turbine inlet temperature; and "braking" for engine braking torque. The development problems were chiefly concerned with stability, reaction time in and out of braking, and flow requirements. The actuating piston is sized for about 100 lb of force at 100 psi, and will swing the nozzle blades through approximately 90 deg of travel in less than 0.5 sec.

Engine testing — Results of some of the tests run on two of the prototype turbine cars are shown in Figs. 16 to 22.

Both the gas turbine and the conventional reciprocating engine are excellent white noise generators; the former favoring the mid- and high-frequency ranges and the latter the low- and mid-frequency ranges of the audio spectrum. Since the reciprocating engine has been the accepted standard for so many years, the turbine engine presents a different, but not necessarily objectionable, sound to the public ear.

Turbine noise sources were divided into four categories for exploration: intake, accessories and gears, engine housing, and exhaust.

Intake noise is due to inlet air velocity and airborne impeller reactions. These have been attenuated to an acceptable level by providing large-area, low-velocity inlets, and by directing the air through suitable geometry ducting that includes the air filters. The ducts are made of nonresonating reinforced plastic, lined with sound absorbing material, and are devious enough to hide the eye of the impeller.

The accessory system presented the toughest noise suppression problem. The major components responsible for objectionable noise level and frequency were the starter-generator, the mitter. Sources of noise are: internal velocity-induced air noise that is transmitted through the housing, air forces that resonate portions of the structure, and forcing functions from rotor unbalance that cause structure resonance and displacement of the housing on its mounts. The cast housing structure was designed to avoid large flat areas that would tend to resonate. First-order shaft speed — 300 to 750 cps — had the potential of producing a very pure, objectionable tone. It was necessary to

**FIG. 15 Variable power turbine showing idling, power, and braking blade positions.**

balance the shaft system dynamically within 0.010 oz-in. at each bearing to minimize this effect.

Exhaust system noise was reduced to acceptable levels by using a duct size adequate to ensure low velocities, by employing turning vanes and suitable transition-radii to eliminate noise-generating eddies, by stiffening the ducts to prevent "oil canning," and by acoustical treatment of some sections of the system.

**Materials and construction**

Among the new low-cost materials used on the turbine engine are: a...
Engine uses a series of iron-based superalloys and heat resistant iron-aluminum alloys.

series of iron-base superalloys, heat-resistant iron-aluminum alloys, and high-temperature rubbing-seal materials.

Engine housing assembly — This unit is the structural support for the powerplant, and contains the major components — the gas generator, burner, power turbine nozzle, power turbine, regenerators, and compressor.

The housing (Fig. 23) is a 2-piece casting of nodular iron of a composition similar to SAE 60-40-10. Creep, stress-rupture and short-time tensile properties are comparable to those of annealed low carbon cast steel up to 1200 F.

Engine housing burner chamber — The heated air from the regenerators is kept separate from the compressor discharge air by the burning camber (Fig. 24). It is welded from a ductile iron-aluminum alloy (Chrysler designation CRM-4), of high oxidation resistance at elevated temperatures. It is sealed to the regenerator and burner openings by ceramic fiber packing and gaskets. Insulation on the inside minimizes heat losses that

![Graph](image1)

**FIG. 16** Engine output and rpm at various gas generator speeds.

![Graph](image2)

**FIG. 17** Engine torque and rpm at various gas generator speeds.

![Graph](image3)

**FIG. 20** Torque performance characteristics. Ratio of stall to design point is 2.3/1.

![Graph](image4)

**FIG. 21** Acceleration performance. "Jump" start is made with gas generator at maximum speed.
could reduce the thermodynamic cycle efficiency. The rear section forms the turbine exhaust chamber, guiding the flow from power turbine outlet to the regenerators. Both chambers are insulated with a lining of ceramic fiber wool.

Gas generator assembly — This unit (shown in Fig. 25) consists of the gas generator rotor (Fig. 6) with the associated stators and flow passages, and its supporting housing. The spinner at the front end of the rotor is made of aluminum. The following inducer is an investment casting of alloy 17-4PH stainless steel to withstand high vibratory and centrifugal stresses and to provide good corrosion resistance.

Compressor impeller — The radial-flow compressor impeller, which is subject to high centrifugal stresses, is produced as a premium-quality plaster mold precision casting of a high purity aluminum alloy, C355-T61. The impeller hub is a stainless-steel casting. The shroud ring is of stainless steel.

Compressor turbine wheel — This rotor has been used as a single casting and as a steel hub with blade ring. In the former the entire wheel was of the CRM-3 alloy; in the latter the blades alone were of CRM-3. The turbine wheel operates at about 1500 F. The integral compressor turbine nozzle and shroud (Fig. 26), which is subject to temperatures of about 1700 F, is produced as a CRM-3 casting.

Gas generator housing — This consists of the air intake housing, a sand casting (Fig. 7) of 356-T6 aluminum alloy, whose outer flange provides the attachment to the engine housing; the cast diffuser plate of the same material, which forms the rear casing of the compressor; and the bearing support, a cast-iron cone piloted and bolted to the diffuser plate. Both compressor bearings are sleeve-type, and pressure lubricated. The impeller hub is sealed by a graphite ring. The welded sheet-metal burner vortex is made of wrought ductile iron-aluminum alloy CRM-4.

Power turbine nozzle assembly — This unit (Fig. 27) forms the gas flow path between the two turbine
Rotor is built as one-piece unit of CRM-3 superalloy.

wheels and supports the 23 variable nozzle blades and their operating mechanism. The variable nozzle blades, which are subjected to temperatures up to 1400 F, are machined from CRM-3. The power turbine shrouds are thin-wall centrifugal castings of 430 stainless steel.

Power turbine wheel — This rotor has been built as a one piece unit of CRM-3, and as a blade ring type using the CRM-3 superalloy only for the blades. It operates at about 1200 F at stresses considerably higher than those of the compressor turbine wheel. A steel floating-spline shaft connects the power turbine to the straddle-mounted pinion of the reduction gear assembly.

A cast nodular iron housing (Fig. 28) supports the power turbine rotor and the reduction gear assembly. Four struts behind the turbine wheel center it to the power turbine nozzle assembly, which forms the shroud for the wheel.

Regenerators — Each of the two regenerators is built up around the heat-transfer matrix, or core, as shown in Fig. 29. The core has an effective diameter of 15.5 in. and a thickness of 3.0 in. The matrix is formed of AISI 430 stainless-steel strip, 0.002 in. thick by 3.0 in. wide, corrugated and wound spirally on a hub to the required diameter. A thin rim and flanges, of 430 stainless steel, are used to act as a pressure wall and to furnish a sealing surface around the core. The core assembly is copper brazed in a hydrogen atmosphere.

The ring gear, of cast iron, is attached to the cooler of the two core flanges by an arrangement that prevents thermal distortion of the core from affecting gear tooth contact and
endurance. The drive pinion is of nitrided iron. This gear combination operates satisfactorily at temperatures that can exceed 400°F, without lubrication. Gear ratio is 2115/1 to rotate the core at 22 rpm maximum.

The seal assemblies divide the core across the matrix into semicircles, the front half with high-pressure air from the compressor, and the rear half with low-pressure air from the turbine exhaust flowing in the opposite direction. Face-type rubbing seals are used around the periphery of the low-pressure section on both sides of the core, providing a low-pressure passage directly through the matrix. The seal operates in contact with the core rim, and the seal cross-arm acts across the diameter. The seals operate in an environment up to 1200°F.

High-temperature rubbing seals are required to operate under severe sliding conditions at relatively high pressure and with no possible lubrication. The major requirements of satisfactory seal materials are:

- Low coefficient of friction over the range of service temperatures.
- Low wear rate, but good wear-in characteristics.
- Dimensional and structural stability over the range of service temperatures.
- Adequate strength and mechanical properties.
- Adequate oxidation resistance.
- Compatibility with matrix material.
- Low cost and ease of manufacture.

This combination of characteristics was one of the major problems encountered in the design of the automotive turbine. Graphite-base materials had proved to perform adequately up to 900°F, but wore rapidly at higher temperatures. Other commercially available materials were found to operate well in the upper temperature range but were unacceptable at lower temperatures.

The problem was solved by the development of a new material (CRM-