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CHRYSLER'S GAS TURBINE CAR

Powerplant Design Characteristics

W. I. Chapman
Chrysler Corp.

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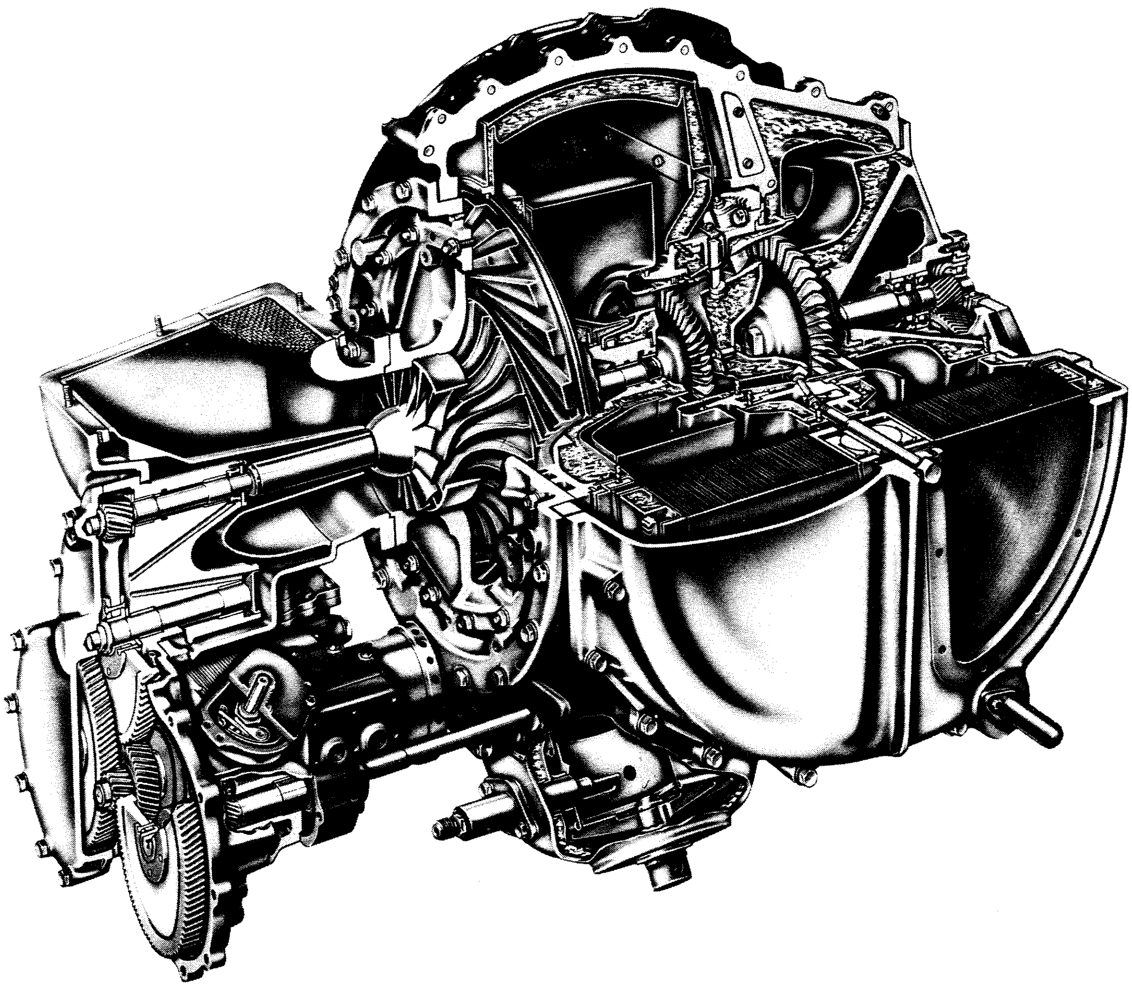

Fig. 1 - Chrysler Gas Turbine Engine

INTRODUCTION

This paper is concerned with the design considerations of an automotive gas turbine engine for passenger car application. It is perhaps not widely recognized, outside of the automotive industry, that throughout the entire spectrum of mechanical devices, there is no other instance in which an apparatus of similar complexity is so completely and continuously operated by a relatively untrained layman as is the conventional passenger car. It is also true, and perhaps just as widely overlooked, that the automotive powerplant is unique among prime movers in the variety of the demands to which it is subjected during a typical "duty cycle." From trivial to full load, cold and hot, on and off, bounding, turning, climbing, mistreated, neglected -- and after all this, cursed if it emits the slightest complaint.

And now into this awesome domain moves the turbine designer. He must not only devise a contrivance that will satisfy these requirements, but work in direct competition with a machine which has been doing just that for a number of decades, and with eminent success.

It will be the prime objective of this paper to illustrate how these and other factors have influenced the various choices that were made in the design of the Chrysler automotive gas turbine.

POWERPLANT ARRANGEMENT

The Chrysler automotive gas turbine powerplant is the simplest gas turbine that can meet the automotive requirements, with a single gas generator rotor, separate power turbine, single can-type burner, and dual disk type regenerators to get low specific fuel consumption.

The major components are arranged symmetrically in the powerplant assembly as shown in Figure 1. The gas generator rotor and

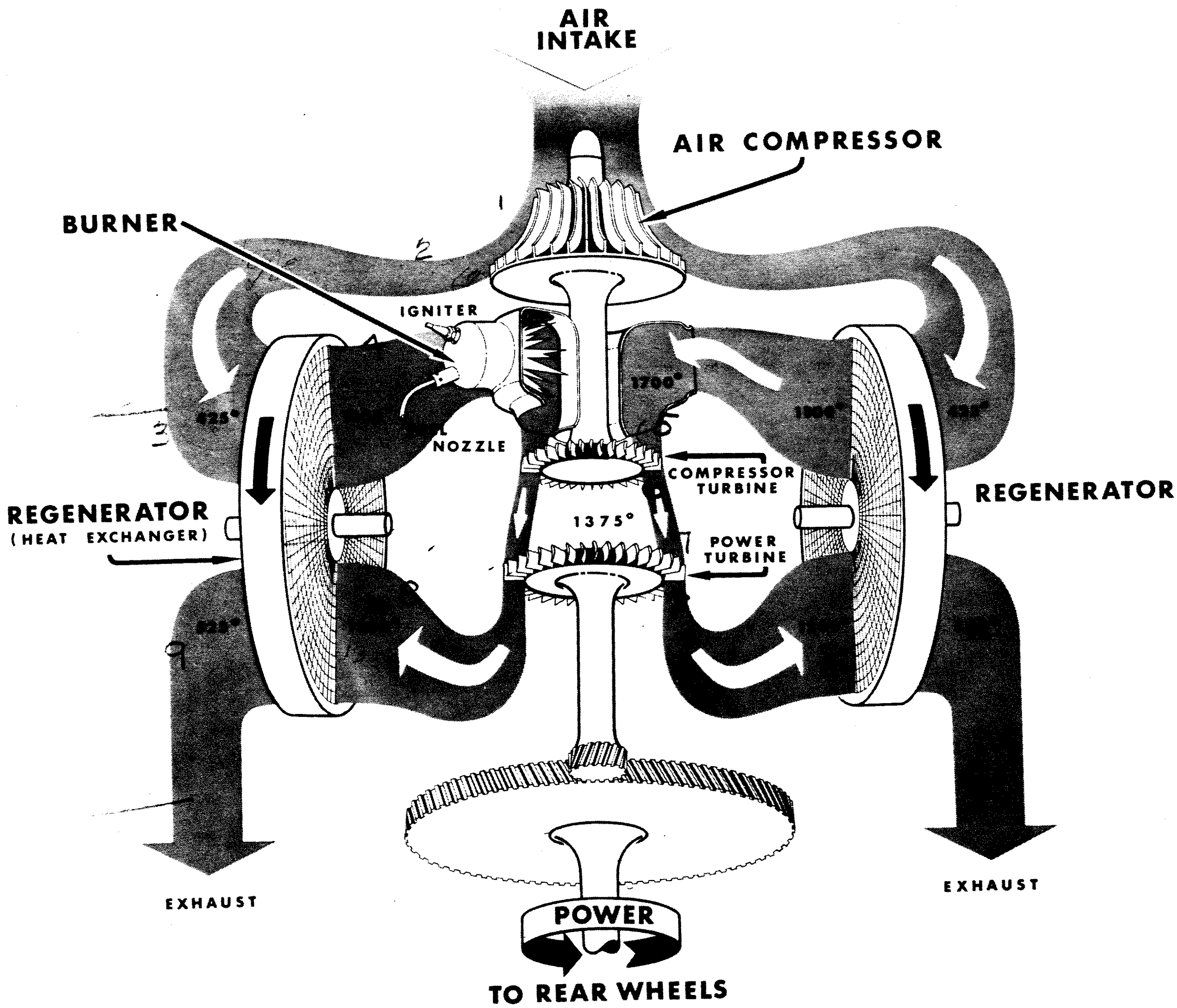

Fig. 2 - Schematic Flow Diagram

power turbine rotor are collinear and form the axes of symmetry in the horizontal and vertical planes. The two disk-type regenerators are mounted in vertical planes on each side of the rotors, rotating on a transverse axis through the rotor axis. This arrangement with dual regenerators provides much better structural symmetry and improved flow paths to and from the regenerator cores compared to the single-disk-type regenerator used on previous Chrysler gas turbines. The single combustor is located between the regenerators at the bottom of the engine for best fitting in the space available in the engine compartment. The accessories are driven from a gear box mounted in front of the compressor intake, and the single-stage reduction gear provides a vertical offset down to meet the desired drive line location.

The air flow through the engine follows the flow path shown in Figure 2. After entering through the intake filter and silencer assembly, the air passes through the intake elbow and axially into the impeller. The impeller discharges the air radially into the diffuser. The diffuser channels dump the compressed air into the space between the engine housing and the chamber surrounding the burner. The velocity of the air leaving the channels causes it to circulate throughout this space on its way to the regenerator cores. It passes through the semi-annulus around the front of the regenerator 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, are guided by a scroll to set up a vortex flow to the first-stage turbine nozzle and wheel. The high velocity gas leaving the wheel is guided and diffused to meet the variable nozzle blades that direct the flow to the second stage or power turbine wheel. The power turbine exhaust is diffused and discharged radially, flowing outward through the rear half of the two regenerator cores, where heat energy is recovered for reuse in the cycle. The cooled gases are collected by converging ducts and passed to the rear of the car by dual exhaust pipes.

POWERPLANT COMPONENTS

Compressor

The compressor and its historical development have been previously described in Reference 1.

The centrifugal compressor has won a strong position in automotive applications because of its ruggedness, moderate rotational inertia, simplicity, and its favorable low axial length and radial discharge. While the axial compressor is generally superior in efficiency, it appears to be a less desirable configuration because of greater number of precision parts and appreciably higher inertia and cost.

The basic compressor can produce a pressure ratio of 4.5:1 and has a maximum efficiency of 84 per cent with a specific speed of 72. This compressor has been very successful in its engine applications, but one of the areas of difficulty has been keeping up with the ever-growing demand for more flow range.

The cast aluminum impeller has thirty blades, with fifteen of them being partial, or splitter, blades. It is necessary to have a large blade area available over which to spread the aerodynamic loading in order to prevent separation. Inlet eye blockage dictates the use of splitter blades so that there are only fifteen blades in the inlet, and aerodynamic stalling in the inducer required that a separate blade row be added ahead 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.

Inducer stall causes a loss in compressor efficiency, creates unnecessary noise, and can be a source of mechanical vibration. Because of the high aspect ratio of the inducer blades and the generally high stress levels involved, the inducer is a steel casting. The inlet section of the main impeller has been subject to fatigue problems, and it has been deemed necessary to fit a steel shroud over this portion of the blades. In order to maintain contact between the blades and the shroud, a fairly severe shrink fit is required. Both the separate inducer and the shroud add to the inertia of the rotor and will be eliminated with further development. An attempt to scallop the impeller back face, similar to radial turbine practice, met with impeller blade vibration difficulties. Further development of the impeller requires greater knowledge of the flow to gain efficiency, flow range, and low inertia without sacrificing life.

The diffuser consists of a short vaneless region followed by a vaned area. The vaneless region is conventionally used to assure subsonic flow at maximum speed so that the vanes operate in absence of supersonic flow. By use of channel model tests, it was found that a diffuser with approximately 3° divergence on one pair of walls and 5° on the other was optimum. In order to take maximum advantage of these tests, the vane pick-up area was designed to provide uniform flow to the throat. It is obvious that there is not unanimous agreement as to the character of the flow in the diffuser area, as can be seen in the variety of configurations used by the industry. A large portion of the compressor losses occur in this area, and it seems certain that major improvements in efficiency will show better control of the diffuser flow. The number of diffuser channels, 29, is relatively large to keep the overall diameter of the compressor as low as practical.

In the present engine, the discharge from the compressor diffuser empties into the rather irregular shaped housing on its way to the regenerator. Somewhat more trouble with this arrangement was encountered than originally expected. The desirability of a uniform static pressure at the compressor discharge was, of course, recognized, but quantitative evaluation was not available until the first housings were received. Although considerable improvement has already been achieved, additional gains in this area are anticipated.

Turbine

The turbine stages of an automotive gas turbine engine have the same fundamental objectives as the compressor; that is, simplicity, reliability, efficiency and low inertia. One may legitimately ask, therefore, why axial stages are used for the turbines when a radial stage was selected for the compressor. The reason is to be found in the fact that many more stages of axial compression are required for a given pressure ratio than are needed for an axial turbine working at the same pressure ratio. Thus low inertia and simplicity are almost impossible to obtain with the axial compressor which might otherwise be chosen for higher efficiency. But why the difference in the number of stages? The answer is twofold.

First of all, any steady-flow compressor operates under an adverse pressure gradient; that is, a rising pressure in the direction of flow. This adverse pressure gradient limits the ability of a single rotor to produce pressure efficiently. In a turbine, on the other hand, the pressure falls in the direction of flow and thus the gradient is favorable. This enables a single turbine rotor to handle much more work than a comparable compressor rotor and do it efficiently. In other words, it is the action of friction in the thin boundary layers covering the flow surfaces and the effect that this friction has on the boundary layers that is crucial in the decision to use axial turbines and a radial compressor

Secondly, the number of stages in either a turbine or compressor is intimately associated with the Mach Number of the gas stream. Turbine stages working in a hot environment enjoy high velocities at a given Mach Number due to the high velocity of sound in the gas. High velocity levels permit large momentum changes that produce large work outputs. Thus, in this sense, a single stage axial turbine can handle more work than a single stage axial compressor. It turns out, moreover, that axial turbines have additional advantages for the automotive engine that will be discussed below.

We shall consider now specifically the choice of turbine stages for the subject engine. First of all, the nature of the application dictated two stages; that is, a power turbine separate from the compressor turbine. Secondly, 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 stage, especially when compared to the compressor turbine. It is far preferable to have the relatively consistent 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. The reason, of course, is that the efficiency of a stage depends on the character of the entering flow. Since the compressor turbine handles the major share of the combined turbine output, the loss in efficiency resulting from a poor inlet flow would be unacceptable. It is possible, on the other hand, 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 an exhaust diffuser and a portion of this velocity energy recovered as static pressure, the effect of the discharge on the efficiency of the engine can be minimized when the power turbine is the final stage.

Thirdly, the two turbine stages should be axial stages. Consider the four basic requirements of simplicity, reliability, high efficiency, and low inertia.

(a) A turbine wheel that is an integral casting with the flow surfaces unmachined is basically simple, whether of an axial or radial design.

(b) It has been well-established that axial turbines can be reliable as, for example, in the aircraft industry.

(c) It has further been established that axial turbines are efficient (Reference 2).

(d) Axial turbines can generally have lower inertia. Turbines do their work by changing the tangential component of gas velocity. The product of this component change and the blade velocity characterizes the power output. Since highly stressed radial turbines generally require radial blade elements, the wheel inlet tangential component cannot be much greater than the blade speeds. An axial turbine inlet, on the other hand, can have a tangential component as much as twice the blade speed. Add to this the fact that the radial turbine exit diameter is smaller than the inlet (and thus contributes less to the tangential momentum change) and we see that for a given output, axial turbine blades can work at a lower blade speed and therefore a smaller diameter and lower inertia for the rotor.

Consider further the effect on the interstage geometry produced by the use of a radial compressor turbine. The discharge of a radial turbine is an annular area of a relatively small outer radius. Whether the following stage is radial or axial, the interstage passage is then tortuous in that it must guide the flow to the much greater mean flow radius of the inlet to the second stage. This added complication to the interstage is undesirable.

Finally, the use of variable power turbine nozzle geometry requires that the power turbine wheel be able to accept flow over a wide range of inlet angles. The leading edge of an axial turbine wheel blade can be designed to accept such flow much more efficiently than is possible with the restrictions that are inherent in a radial turbine blade shape. For all of these reasons a single stage of axial turbine was selected for both the compressor turbine and the power turbine.

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 which demonstrated satisfactory pressure diagrams. The velocity diagrams shown in Figure 3 summarize the overall aerodynamic design. (Since the general considerations leading to the design of the compressor are available in Reference 1, it was not felt necessary to repeat them in this paper. The corresponding turbine data, however, have not been previously published so that an outline of the various factors considered in developing these particular velocity diagrams is included here).

Going through the velocity diagrams in order, the first station is the inlet to the compressor turbine nozzles. This velocity is determined both by the annulus area of the following turbine wheel and the character of the preceding transition from the burner. The transition from a single burner tube to a nozzle annulus is awkward unless a vortex chamber is used to provide uniform entry conditions into the turbine stage. The vortex chamber produces swirling flow which reduces the degree of turning that remains for the nozzle blade row. The axial velocity component at the nozzle inlet is essentially determined by the air flow and the turbine wheel annulus area, while the tangential component is the result of the vortex chamber geometry.

The next station is the nozzle outlet where the velocity is limited by the Mach Number level at which the designer desires the element to

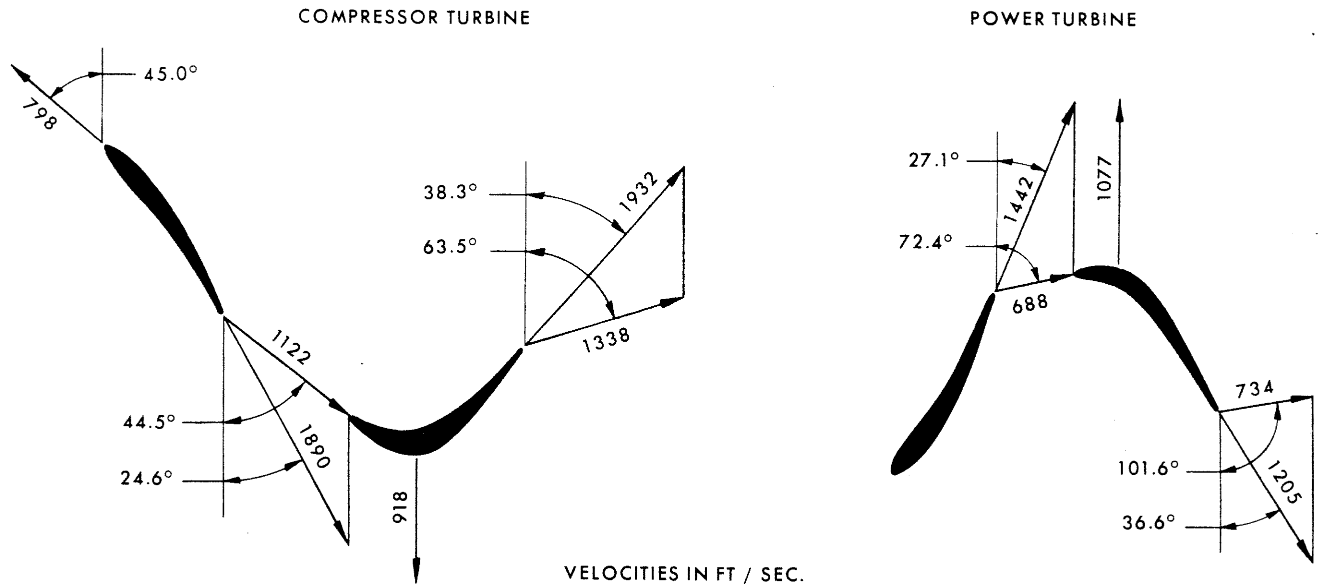

Fig. 3 - Turbine Stage Velocity Diagrams

operate. As previously noted, the output of an axial stage is proportional to the product of the tangential velocity change and the wheel blade speed. A low inertia wheel results from low blade speed (small mean diameter) in conjunction with large tangential velocity change. Thus we chose a nozzle exit velocity that was limited by a Mach Number of essentially 1.00 at the blade root to give a maximum possible tangential velocity within the preselected limits imposed on the flow. With this tangential velocity a value of the parameter $\Delta C_u/U$ as high as 2.5 at the wheel mean radius was found possible, and a correspondingly low value of wheel inertia was obtained.

The remainder of the stage velocity diagram is now virtually determined. The rotational speed of the turbine has been set by the compressor, and the allowable stress at the turbine blade root then determines the flow annulus area. The amount of swirl in the flow leaving the wheel is limited by the allowable friction losses in the subsequent interstage passage, which in turn limits the tangential velocity change across the wheel. This velocity change and gas mass flow determine the blade speed necessary to produce the work required of the turbine. The blade speed and rotational speed determine a wheel diameter and thus the basic size of the wheel. This completes the data for the velocity diagrams.

The other size factor of the wheel, the axial width, is kept as small as practical, again primarily for inertia reasons. In order to keep the width of the wheel small, it is necessary to use a large number of blades so that loading requirements are not exceeded. The torque produced by the gas flow is developed as a mean net pressure on the blade surfaces. A maximum blade loading criterion implies a minimum blade area on which the pressure load acts. A combination of wheel axial width and number of turbine blades must be determined which satisfies this minimum area requirement. There is, of course, a maximum number of blades that a designer can accept, primarily because of blockage of the flow passage due to trailing edge thickness of the blades. This thickness in turn is determined by the minimum permitted by manufacturing processes as well as by thermal stress considerations. When these factors were combined, a 53-blade wheel of about 0.50 inch width was selected.

The interstage transition is the next portion of the flow path. The configuration used is almost entirely determined by other components.

Thus the inlet is at a small radius because of inertia requirements on the compressor turbine; the outlet is at a large radius because of stall torque requirements of the power turbine. In fact, it is interesting to note that one particular operating point, namely break-away of the car from a standstill, is heavily responsible for this much of the interstage geometry. In addition to the obvious influence of the two turbine wheels, the interstage design is determined by the requirements for accommodating variable power turbine nozzles, and for restricting the overall length of the engine. Sophisticated aerodynamic design of this passage is almost impossible due to the disturbed nature of the flow leaving the compressor turbine. The instantaneous velocity components of this flow fluctuate as a result of the finite number of blades on the turbine and the mechanical clearances required around the rotating wheel. This is an undesirable inlet to a diffusing passage and the detailed design must necessarily be tested and developed for the conditions imposed by engine operation.

The requirements for the power turbine are different enough from the compressor turbine that the description of the velocity diagram becomes more devious. Specifically, since the design speed of the power turbine is not predetermined, the wheel annulus area is determined by an unknown wheel speed as well as by the stress level. We find, then, that the main character of the velocity diagram is determined by the work output to be achieved, as available from the pressure remaining for expansion following the compressor turbine, and the discharge conditions from the exit of the power turbine wheel. As noted above, the discharge from the power turbine covers a wide variety of velocity magnitudes and distribution over the operating range of the engine. The limitations of the ability of the exhaust diffuser to recover velocity pressure require the designer to minimize in a general sense the velocities leaving the power stage. If we say that the exit conditions set the level of through-flow velocity and thus the wheel annulus area, then the wheel blade root stress sets the rotational speed. The remainder of the data necessary for the velocity diagram, including setting the wheel diameter, is then a compromise to obtain the required power with an acceptable inertia and outlet swirl characteristics.

So far the discussion of turbine design has centered about maximum power operation. It is natural that the designer is concerned with the ability of the engine to develop full output, especially since, for instance, the gas generator can be at maximum output during 80 per cent to 90 per cent of the duration of a full throttle car acceleration. Nevertheless, an automobile spends the vast majority of its running time at low load, and its success depends on how well it performs as a road vehicle. Therefore, it would be quite impressive to be able to report that both turbine stages were carefully analyzed in the design stage to optimize overall vehicle performance. The fact remains that the requirements for the transonic compressor turbine were so stringent that it was originally designed with all attention focused on inertia and design point operation. There certainly must be some good fortune in the fact that the stage operates satisfactorily ing rapid transients. There is a considerable gap between analytical studies of a power turbine stage operating with variable nozzles delivering power through a three-speed transmission to the wheels of a vehicle under a multitude of road conditions, and the actual performance of the final hardware. There is no question but that the present design of the power turbine stage leans heavily on the performance calculations at maximum output and equally heavily on accumulated previous empirical experience. Just how much improvement in vehicle performance can be attained in turbine stage design is one of the future developments to be explored during the next few years.

Regenerator

A high performance heat exchanger is required for a relatively low pressure ratio gas turbine engine to achieve outstanding fuel economy. The thermodynamic performance criteria used in evaluating heat exchangers are heat transfer, pressure loss, and leakage. Each of these are important factors in determining the overall performance of the powerplant and, in addition, the size of the heat exchanger package is important for vehicular applications.

Basically, all heat exchangers can be classed as either recuperative, in which heat energy is conducted through a wall separating the fluids, or regenerative, in which a heat storage element is alternately exposed to each fluid. Consideration has been given to the stationary recuperator. It has advantages in lower leakage and greater flexibility in powerplant design. The heat transfer surface requirements are similar to those of a regenerator, but the ducting required for gas-to-gas counterflow is difficult to achieve without size, weight, and fabrication penalties. Crossflow configurations are simpler to devise but, due to their inherently lower effectiveness, are larger and heavier still.

The rotary disk type regenerator was selected for 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, reducing fouling by combustion deposits.

The study of regenerators is complicated by the fact that no simple analytical expressions are available for heat transfer, pressure loss, and leakage. This necessitates an experimental program for determining empirical relationships of regenerator thermodynamics to the many geometric and physical properties of the regenerator and fluid streams. However, the problem may be simplified by using dimensionless groupings of the quantities involved.

Heat transfer is characterized by the effectiveness which is the ratio of heat energy actually transferred to the maximum heat potential. Similarly, the pressure losses are characterized by the pressure loss ratio which is the sum of the ratios of pressure drop to inlet pressure for both the hot and cold sides of the regenerator. For the conditions applicable to vehicular regenerators, effectiveness and pressure loss ratio are found to be principally functions of the passage shape, the Reynolds Number, the passage flow length to hydraulic diameter ratio, the utilization factor (which is the ratio of fluid to matrix heat capacity rates), and the axial conductance factor (which is the ratio of axial thermal conductance to gas heat capacity rate).

The effect of regenerator thermodynamic performance and leakage on overall engine performance was studied. 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. Therefore, it was found that effectiveness and pressure loss ratios of .90 and .021 respectively at idle and .85 and .045 at full load were a satisfactory compromise for good performance and reasonable regenerator size which resulted in acceptable leakage. This level of performance is obtained with an average Reynolds Number of about 200, a length to hydraulic diameter ratio of about 100, an utilization factor of about 0.25, and an axial conductance factor of about 0.003.

The actual regenerator geometry was determined from the parametric values using mechanical considerations such as stresses and deflections in the regenerator, seal geometry, adaptability to the powerplant configuration, and the methods of mounting and driving the regenerator. These subjects will be discussed later.

Burner

A single can-type burner of reverse flow configuration is used as the best compromise between the requirements of compactness and low engine cost. Such a burner offers excellent accessibility and ease of replacement, and has been used in all automotive turbine engines designed by Chrysler to date.

Important differences in design requirements exist between regenerative and simple cycle non-regenerative gas turbine combustors. For instance, convective temperature differences available for cooling and maintaining safe metal temperatures are obviously much reduced in the regenerative cycle due to the higher burner inlet temperature; this same hot environment creates special fuel nozzle reliability problems. It is also important to note that quantities of carbon formation considered tolerable in many current simple cycle combustors would cause irreparable damage to a rotary regenerator if particles were to become wedged in the seals. In addition, demands of regenerative cycles for low pressure losses give rise to differences in aerodynamics such as jet-to-stream momentum ratios and aerodynamic-to-combustion pressure loss ratios. Therefore, direct application of non-regenerative burner design practice to the regenerative cycle will not yield optimum performance.

Our early work started with the adaptation of currently available knowledge. Development programs soon indicated, however, that these designs provided insufficient volume, especially in the primary region. Later combustors incorporated larger volumes with moderate overall intensities. In addition, the wide range of input rates required for flexible engine operation causes broad overlapping and shifting of primary and secondary zones, thus preventing realization of zone concepts. Under these conditions, the secondary zone serves as a turbulent mixing and combustion region at high inputs and a gradational dilution zone at low inputs. A separate dilution zone entirely downstream of the combustion volume admits the remaining air.

The wall temperatures of the liner surrounding these combustion zones must also be carefully considered. An adherent cooling film has been found to be a most desirable method of obtaining effective control of these temperatures. In the reverse flow arrangement used, simple stamped louvers cannot provide the degree of control necessary for effective cooling. An annular gap arrangement, however, has been successfully developed, and excellent control over the admission of cooling air is obtained by varying gap width and axial spacing. These cooling gaps effect a circumferentially uniform temperature and a sufficiently low axial gradient so that thermal stress problems are minimized. It is to be further noted that the stress concentrations characteristic of stamped louvers are not present with this design. These features, in combination with the excellent effectiveness of the annular cooling arrangement, insure long liner life.

The flame stabilizing mechanism has been developed around the conventional toroidal, reversed flow of combustion gases common in most can type combustors. The strength of the recirculation can be controlled conveniently by the size and position of punched hole jets with some dependence upon the volume and strength of the upstream cooling annulus flow, and some reaction of the fuel spray momentum and shielding air. The configuration as developed has been very satisfactory, the rich and lean limits never being exceeded at any engine operating condition. Ignition, which is also strongly influenced by the stabilizing pattern, is controlled by spatial location of the spark.

Pressure losses associated with combustors have been in the neighborhood of 5% of cycle pressure for aircraft practice. However, regenerative engines of relatively low pressure ratio place a premium on pressure loss from any source; hence, the combustor has received special efforts along this line. The burner configuration has been designed with the minimum interaction between the liner and surrounding environment consistent with compactness. Sensitivity of the combustor performance to manufacturing tolerances and asymetries of environmental flow are thus virtually eliminated. These conditions also minimize pressure losses due to aerodynamic irregularities in the combustor surroundings. The losses are hence due mainly to the liner itself. A design point loss of 1-1/2 to 2% of maximum cycle pressure is being used with satisfactory performance.

The efficiency of a combustor varies considerably with geometry, fuel type, and ambient conditions. However, our experience with this burner has shown efficiencies in excess of 98% for all types of steady-state operation. This results in extremely clean exhaust emissions and freedom from odor, soot, smoke, and smog producing constituents.

Regenerative engines can be expected to have a high tolerance for heavy fuels because of high burner inlet temperatures, and the ability of the Chrysler engine to operate on a wide variety of fuels is one of its many advantages. An air atomizing nozzle further enhances this capability because of its generally superior ability to break up all grades of fuel.

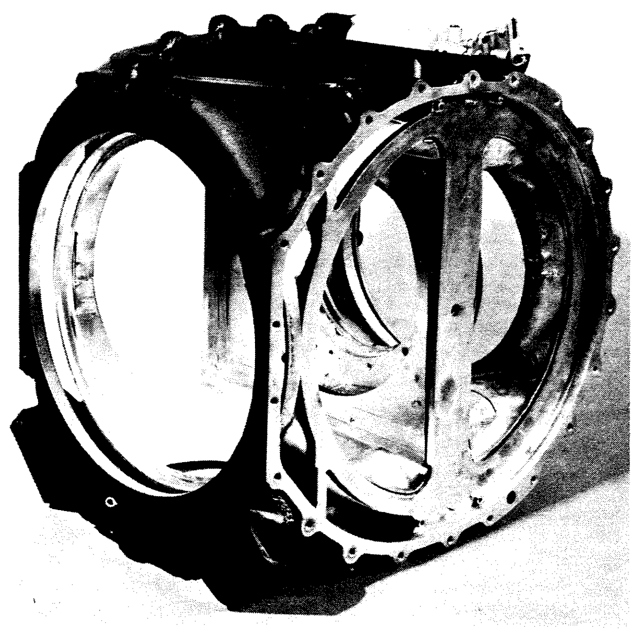

Fig. 4 - Engine Housing Assembly

POWERPLANT CONSTRUCTION

Engine Housing Assembly

The engine housing is the structural support for the powerplant, containing the major components -- the power turbine nozzle, gas generator, power turbine, regenerators, and burner assemblies.

The housing, Figure 4, is a two-piece cast iron construction, split vertically along the axis of the rotors. This construction permits the housing to be cast in two relatively simple castings, with minimum coring. The front section of the housing ahead of the bulkhead acts as a pressure vessel, containing the compressor discharge flow to the regenerator. The housing is shaped to carry the pressure loads as a membrane to the greatest extent practical to minimize stress and deflection.

The heated air from the regenerator is kept separated from the compressor discharge air by the burner chamber, Figure 5.. This is a sheet metal assembly that is clamped between the flanges that form the regenerator seal platforms when the two halves of the housing are bolted together. It is sealed to the regenerator and burner openings by ceramic fiber packing and gaskets. This burner chamber is of sheet metal since the pressure difference it must retain is small, less than 1 psi. It is insulated on the inside to minimize heat loses that can reduce thermodynamic cycle efficiency. Compressor discharge air flows around the burner chamber on its way to the regenerators, washing the entire interior of the high pressure part of the housing. This minimizes the temperature gradients and stresses in the housing, and reduces the metal temperature enough to permit use of conventional materials.

Fig. 5 - Engine Housing Burner Chamber

The rear section of the housing forms the turbine exhaust chamber, guiding the flow from the power turbine outlet to the rear half of the regenerator cores. This chamber is also insulated to reduce heat loss from the cycle and minimize housing temperature and stresses, with natural convection from the outer wall providing adequate cooling here. Both chambers are insulated with a lining of ceramic fiber wool and/or blanket of varying thickness to suit the space available and insulation requirements. This insulation is retained by thin sheet metal liners that form the flow passages, preventing the hot gases from disturbing the insulation, and conversely retaining the insulation to keep it out of the moving parts. The insulation is installed as a semi-permanent assembly by bending tabs and tack welding the liners in place.

Power Turbine Nozzle Assembly

The power turbine nozzle assembly forms the gas flow path between the two turbine wheels and supports the 23 variable nozzle blades and operating mechanism. It is attached to the housing bulkhead where it is surrounded and cooled by air at compressor outlet temperature up to 450°F, while the temperature through the flow passage might be up to 1,400°F.

The design objectives were to minimize heat loss from the assembly for best cycle efficiency, to maintain close nozzle blade and power turbine wheel tip clearance, and to maintain accurate nozzle operating gear fits with temperature. Objectives were attained by use of a double wall construction shown in Figure 6.

Pressure and mechanical loads are carried by the nozzle support. The shrouds forming the annular flow passage are thin so that the thermal response time is similar to that of the blades. The inner shroud is located by three radial struts from the outer support, and front and rear covers for the inner shroud retain insulation. The variable blades pivot between spherical surfaces formed by these inner and outer shrouds. Each blade is supported by an integral radial shaft from the outer housing. A spur gear sector fastened to the outer end of each shaft engages a face type ring gear to vary the blade angle in unison.

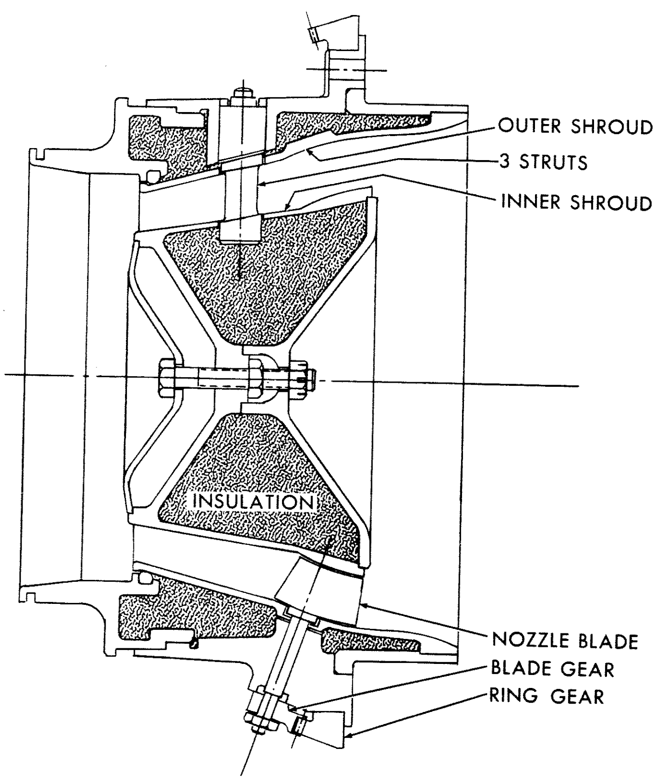

Fig. 6 - Power Turbine Nozzle Assembly

Gas Generator Assembly

The gas generator assembly shown on Figures 7 and 8 consists of the gas generator rotor with the associated stators and flow passages as well as the necessary supporting structure.

The gas generator rotor (Figure 9) consists of two major parts, the compressor impeller assembly and the compressor turbine rotor. As assembled, the impeller and turbine wheel are overhung from opposite ends of the shaft. The shaft is flash-butt-welded to the turbine wheel casting to form an integral unit, then finish machined. Both journals are located on this shaft, and an extension forms a tie bolt for attaching the impeller assembly. The impeller assembly is comprised of the cast aluminum impeller, cast steel inducer, steel shroud ring, aluminum spinner, and steel impeller hub assembled in a sequence of shrink fits. This assembly is piloted and clamped to the turbine shaft by the tie bolt. A tubular drive shaft for the accessory case is driven from three lugs on the nut. The rotor assembly is balanced dynamically in two steps, first the turbine wheel and shaft, making balancing corrections on the turbine disk, then the complete assembly, with corrections on the impeller inlet hub and back face.

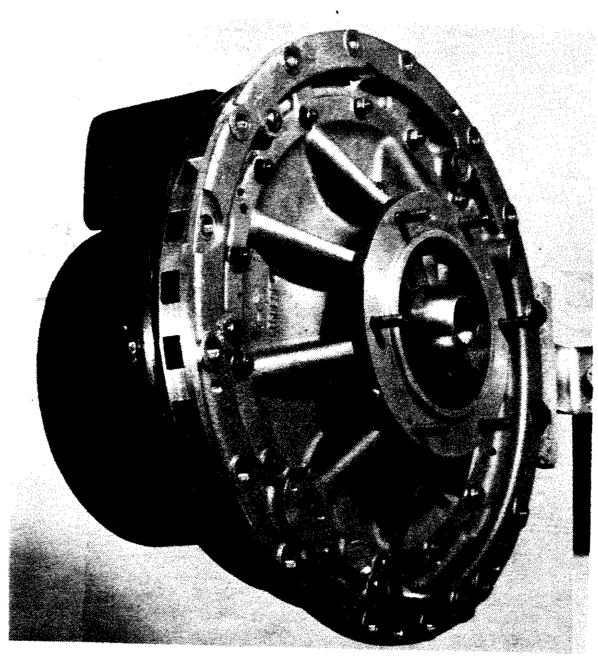

Fig. 7 - Gas Generator Assembly - Front View

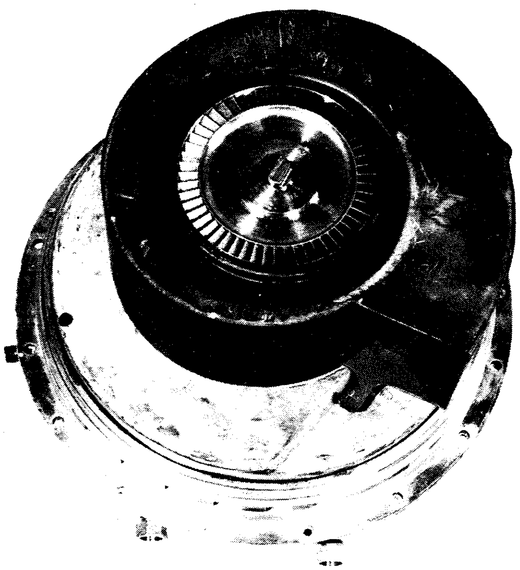

Fig. 8 - Gas Generator Assembly - Rear View

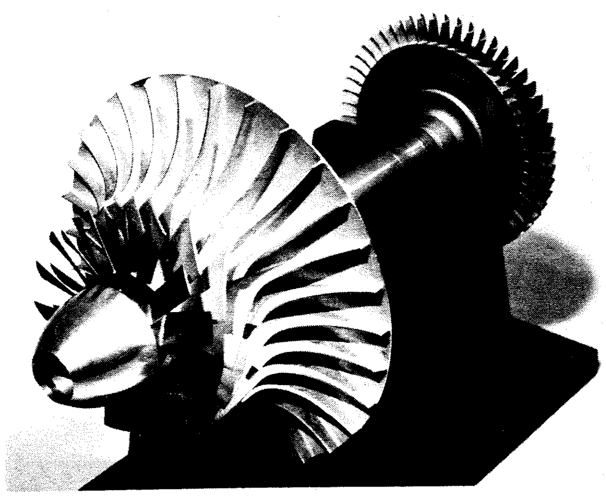

Fig. 9 - Gas Generator Rotor

The gas generator rotor is supported by a structure (Figure 10) consisting of the air intake housing, an aluminum casting which is the forward casing of the compressor and whose outer flange provides the attachment to the engine housing; the cast aluminum diffuser plate which forms the rear casing of the compressor and contains the compressor diffuser channels; and the bearing support, a cast iron cone piloted and bolted to the diffuser plate. Mounted in each end of the bearing support are the rotor bearings, both of which are sleeve-type. The front bearing is contained in a short cylindrical bronze housing, the rear face of which forms a tapered land thrust bearing, and which is flexibly mounted to the bearing support. Oil is supplied to both bearings through internal drilled passages from the air intake housing. The impeller hub is sealed by a graphite ring acting as a controlled gap seal. The turbine wheel hub is sealed by a labyrinth formed by annular teeth machined in the shaft. This seal is pressurized by air from the compressor discharge through internal drilled passages.

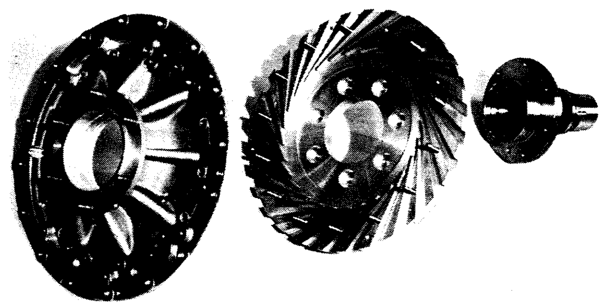

Fig. 10 - Gas Generator Housing Components

The compressor turbine nozzle assembly (Figure 11) is located from the bearing support and clamped by a ring nut. The blade ring is a precision casting with an integral inner shroud and conical support, with a two-piece outer shroud assembled after final flow test and machining. A double seal ring mounted for this shroud engages an extension of the power turbine nozzle assembly, forming the pressure seal on installation.

The welded sheet metal burner vortex is piloted by the turbine nozzle inner shroud and attached to the bearing support. The diffuser plate and bearing support structure are insulated from the hot gases in the burner chamber by insulation contained within sheet metal liners.

the general procedure in Reference 4. Aluminum plates are cemented to the web of the gear to provide damping and reduce noise.

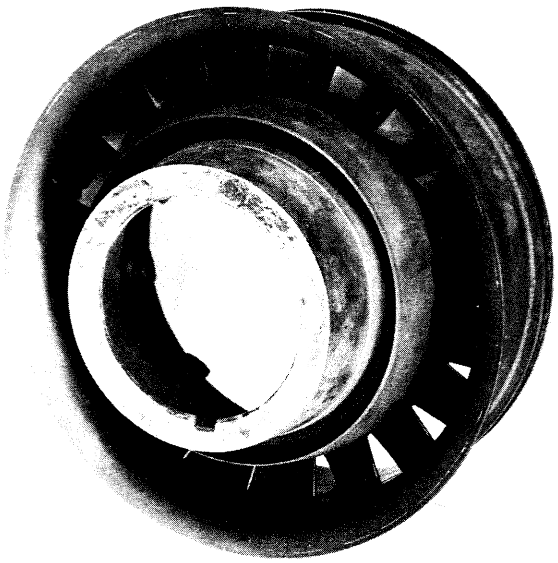

Fig. 11 - Gas Generator Turbine Nozzle Assembly

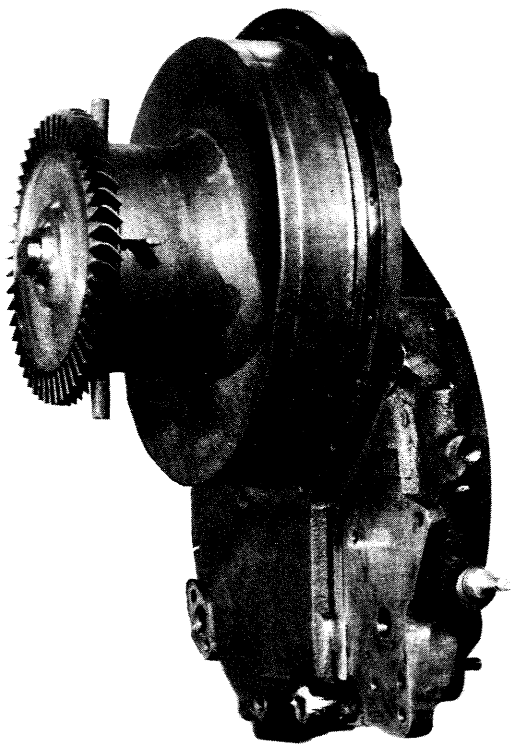

Fig. 12 - Power Turbine Assembly - Side View

Power Turbine Assembly

The power turbine assembly (Figures 12 and 13) consists of the power turbine rotor, reduction gear assembly, and supporting housing.

The power turbine wheel is permanently attached to the shaft by a center tie bolt (Figure 14) and this assembly is balanced dynamically, making corrections on the turbine disk. The rotor is supported by a steel-backed bushing at the front and a ball bearing at the rear, and is sealed at the wheel hub by a pressurized labyrinth. A floating spline shaft connects the power turbine rotor to the reduction gear pinion. The 18-tooth pinion and the 175-tooth gear (Figure 15) are each straddle-mounted on two ball bearings. The gears are of 30° helix angle with a profile contact ratio of 1.5 and a helical overlap of 3.0. The normal diametral pitch is 20 and the operating pressure angle is 16° 38'. The gears are designed for balanced life following

A cast iron housing supports the power turbine rotor and the reduction gear. The cover for the reduction gear cavity is doweled to the housing and the bearings line bored. Four struts behind the turbine wheel center it to the power turbine nozzle assembly, which forms the shroud for the wheel.

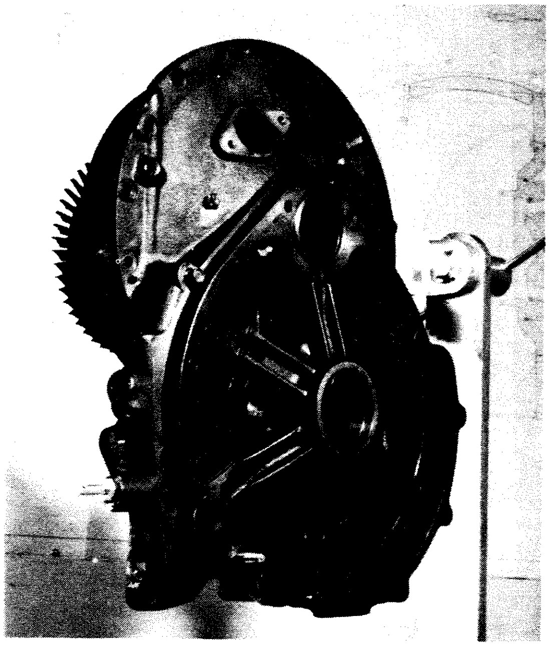

Fig. 13 - Power Turbine Assembly - Rear View

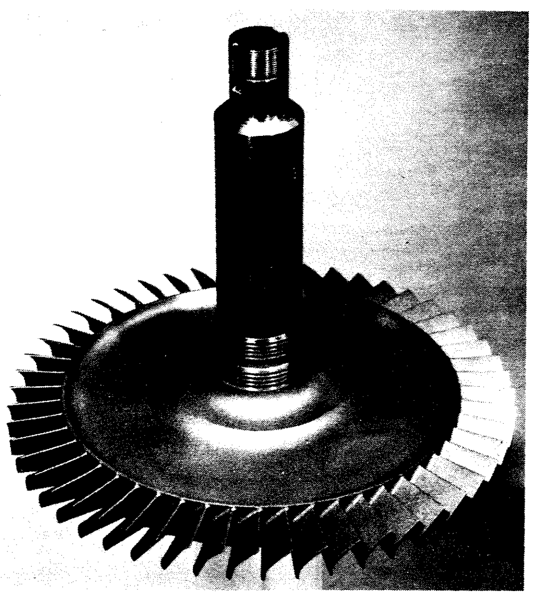

Fig. 14 - Power Turbine Rotor

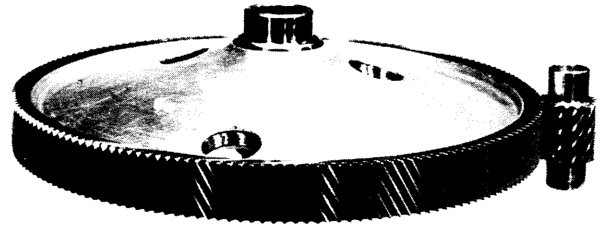

Fig. 15 - Reduction Gears

Regenerator Assembly

Each regenerator assembly is built up around the heat transfer matrix, or core, as shown on Figure 16. The core has an effective diameter of 15.5 inches and a depth of 3 inches. The matrix is formed of AISI Type 430 stainless steel strip, .002 inch thick by 3 inches wide, corrugated and wound spirally on a hub to the

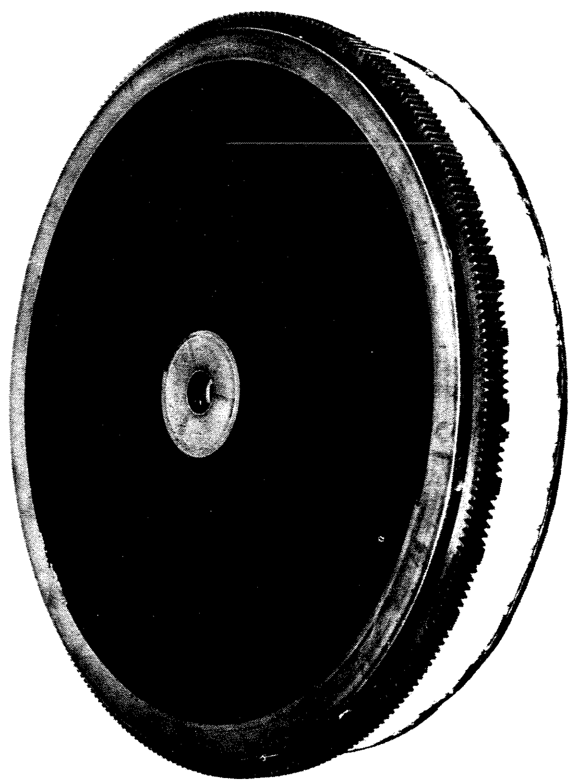

Fig. 16 - Regenerator Core Assembly

required diameter. A thin rim and flanges are added to act as a pressure wall and furnish a sealing surface around the core. The hub and rim parts are designed to minimize stresses in the matrix, allowing it to distort into a shallow spherical shape with the cyclic temperature gradients inherent during operation. The rim flanges are designed to roll with the matrix as it distorts, maintaining a smooth contour over the face of the core assembly for sealing purposes. The core assembly is copper brazed in a vacuum or hydrogen atmosphere. Then the core faces are finished smooth and flat, without burrs on the ends of the folded strip, to provide a smooth surface for sealing. The ring gear is mechanically attached to the cooler of the two flanges by an arrangement that prevents the thermal distortion of the core from affecting gear tooth contact and endurance. Cast iron ring gears, mounted in this manner and mating with nitrided pinions have proven satisfactory even though they operate unlubricated at temperatures over 400°F.

The seal assemblies divide the core across the matrix into semi-circles, the front half with high pressure air from the compressor, and the rear half with low pressure air from the power 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 core. This type of sealing arrangement capitalizes on the use of a smooth surface in one plane that can be easily sealed. The seal operates in contact with the core rim and on the matrix across the diameter, called the seal crossarm. The rubbing seal must deflect to follow the spherical distortion of the core face, maintaining continuous contact with the core. Materials for this type of seal have been developed and tested by Chrysler as discussed in Reference 3.

The rubbing seals are connected to the powerplant housing and regenerator cover by flexible metal diaphragms, which must provide clearance to accommodate the thermal expansion and distortion of the core with respect to the housing while permitting only minimum leakage. At the same time, the seal assembly must be pressure-balanced to give a small positive contact force at the rubbing surface with the leakage pressure gradient that exists. This contact force must be small to minimize friction, drive power, and wear, and should be distributed over the rubbing surface to prevent local wear. This pressure balancing can be accomplished by careful design of the flexible metal seal, and by modifying the pressure gradient on the rubbing surfaces with grooves and channels.

Total leakage of high pressure air to the low pressure passage is less than 3 per cent of engine air flow and remains nearly constant throughout the operating range. About 0.75 per cent is due to the rotation of the core, with the matrix volume retaining air at different pressures as it leaves the two passages, resulting in a net loss from the high pressure passage. If core speed were reduced, this carryover would decrease, but there would be a loss in effectiveness due to heat storage effects. In addition, higher core speeds will increase the power required to drive the core. All these effects have been considered in the final selection of core speed and gear ratio, which are respectively 22 rpm maximum and 2115 to 1.

The regenerator core is assembled into the regenerator cover with drive pinion and seals attached to the flange on each side of the engine, and the core is located at the center by a tie bolt between the cover and housing. A spherical bushing is used to provide rotation with axial movement and tilting, allowing the cores to float between the seals with minimum friction.

Burner Assembly

The burner tube (Figure 17) is 12 inches in length and approximately 5 inches in diameter, having a volume of 250 cubic inches. A transition section is provided in the burner vortex between the burner tube outlet and the turbine nozzle inlet, acting as an extended mixing zone. This extension also serves to minimize turbine inlet temperature gradients caused by long flametails burning into or past the dilution slots. Such flametails might arise under conditions of severe burner overload, i.e., cold starting followed by immediate full power requirement with the regenerator not yet at full temperature.

The construction of the burner tube assembly is of maximum simplicity consistent with aerodynamic and life requirements. The assembly consists of three tubular sheet metal sections with the upstream ends flared for aerodynamic cleanliness and rigidity. These sections are fastened into an assembly by means of robust, radially oriented pins, securely positioning the parts axially and concentrically. A controlled tolerance slip-fit of the pins in the liner sections permits unrestrained radial expansion. Care is taken to position the pins essentially flush with the wall, preventing flameholding tendencies which might result in burn-off of the pins.

Circumferential wall cooling slots are provided by the annular gaps between the individual sections, with sufficient overlap preventing separation of the cooling air. Vorticity for recirculation and flame stabilization is provided by a swirler mounted in the top of the tube, which extends over the whole burner diameter except for an annular cooling gap. Thus, complete blanketing of the burner dome with air is assured, and it is believed that this feature greatly contributed to the absence of carbon formation in the present design. Extreme accuracy in concentricity of the tube with the burner annulus is not required due to the generous annular cross-sectional area. The burner liner is allowed to float axially by a small amount between shoulders on the swirler and vortex

Fig. 17 - Burner Components

respectively, allowing for axial expansion. Experience with this design has shown no sensitivity to small variations in axial depth of the swirler.

Combustion and mixing air orifices are arranged in four axial rows, producing very active mixing by formation of vortices in a plane perpendicular to the burner axis. The remaining air is admitted to the dilution mixing zone by four slots entirely downstream of the primary and secondary combustion zones. Primary and secondary combustion zone orifices as well as the dilution slots are of punched construction and are not only simple to produce but also insensitive to small manufacturing variations from an aerodynamic standpoint.

A single fuel nozzle distributes the fuel into the primary zone of the liner by spray momentum as well as burner air pattern effects. The nozzle is of the air-atomizing type where the fuel is broken up by means of an air-stream in the nozzle itself. The air atomizer will perform equally well on gasoline, JP4, or diesel-grade fuels because of the flexibility and control offered by the air-jet type of atomizing. The required nozzle air differential pressure is quite low and is readily accommodated by a small piston-type pump of conventional design, supercharged with compressor discharge air. The advantages offered by the air-atomizing nozzle are numerous:

(a) The turndown ratio is extremely high, (b) nozzle pressure drops are low, allowing fuel supply by commercial quality positive displacement pumps, (c) the nozzle passages are large and will not plug easily, (d) atomization is considerably finer than with pressure atomizing nozzles, especially at part loads where good atomization is particularly important for high burner efficiency, (e) starting is aided by fine atomization at low flows, (f) multi-fuel capacity is excellent, as the nozzle is negligibly affected by fuel viscosity, and (g) nozzle performance is quite insensitive to small variations in manufacturing tolerances.

Ignition is provided by a single conventional type spark plug with tubular electrodes. This allows for the passage of cooling air from the nozzle air pump through the electrodes to provide extended life.

An automatic pressure-operated fuel drain valve is provided to drain any liquid fuel from the burner section on shutdown. This occurs only if there is a malfunction preventing combustion.

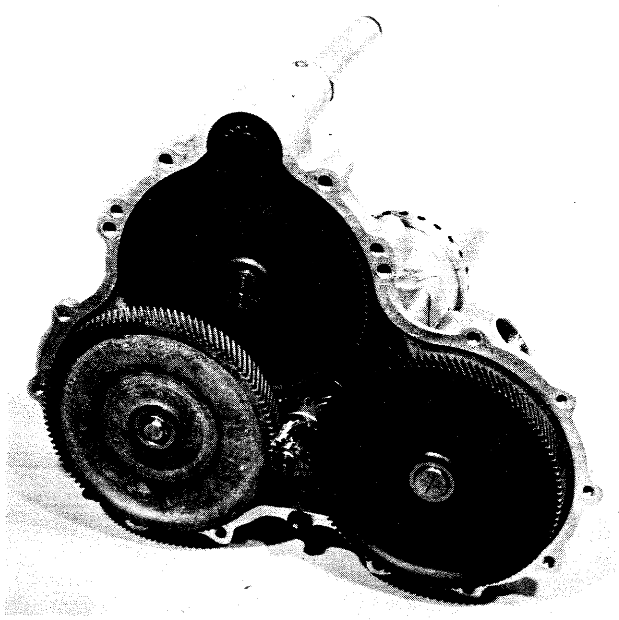

Fig. 18 - Accessory Drive Assembly

Accessory Drive Assembly

The main accessory gear case (Figure 18) is mounted in front of the gas generator, supported from the compressor inlet elbow. The accessory pinion is driven from the front of the gas generator rotor through a tubular shaft with resilient joints to provide for torsional isolation, shaft deflection, and alignment tolerances. A controlled gap graphite seal, pressurized by compressor air, seals the pinion shaft from the compressor inlet air and provides an air supply to help scavenge the gear box.

The accessory pinion and the idler gear below it are each supported on a pair of steel-backed bushings which are finish bored in assembly. The idler gear drives the starter-generator, at 19,957 rpm maximum, by a gear mounted on the starter-generator shaft. A coaxial pinion on this shaft drives two low speed gears (identical to the idler gear) which drive the remaining accessories at 2,984 rpm maximum.

The fuel control and pump are driven by the gear on the right side (viewed from the drive end of the engine) which is supported by two steel-back bushings finish bored in assembly. The gear on the left side is mounted on a shaft supported by a steel-backed bushing and a needle bearing. The sleeve bushing serves also as a transfer bushing to supply oil to a hole in the center of the shaft. A crossed axis helical gear is machined in the shaft to drive the ignition breaker point assembly and the tachometer through a mating nylon gear. The shaft terminates with a spline to drive the worm gear and oil pump, while the air pump, which supplies air for the fuel nozzle, is driven by a cam keyed to this shaft. The central oil hole in the shaft supplies pressure oil to the needle bearing, the air pump cam, and to the spline (to prevent fretting).

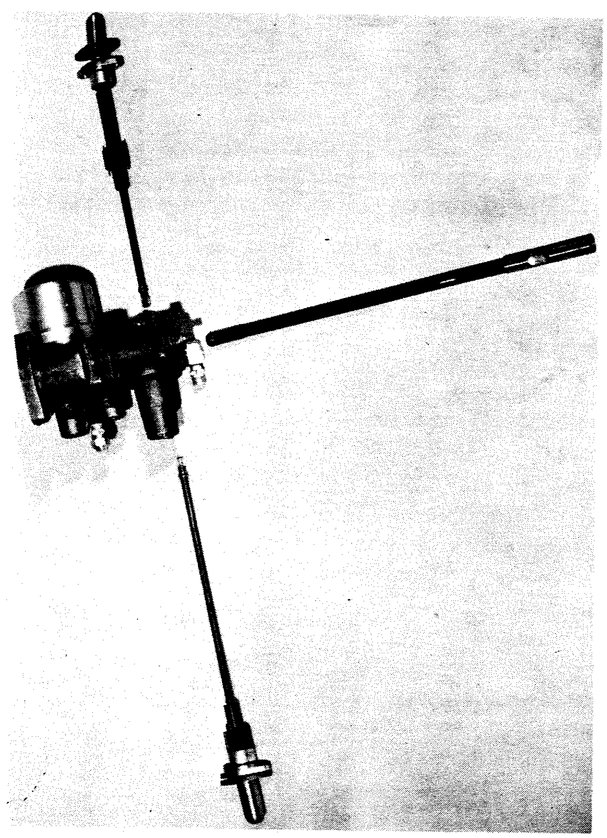

Fig. 19 - Regenerator Drive and Oil Pump Assy.

The spline drives a 17 inch long shaft which is flexible enough to provide for deflection and alignment tolerances between the front accessory housing and the regenerator drive and oil pump assembly (Figure 19) mounted on the power turbine housing. The regenerator drive is a worm gear box with a four-thread hardened steel worm driving a 32-tooth manganese bronze worm wheel. The worm wheel is splined to cross shafts which drive the two 16-tooth regenerator pinions which in turn drive the regenerator core ring gears without any lubrication of the gear mesh. Misalignment between the worm gear box and the regenerator pinions is accommodated by bending of the cross shafts. The worm gear box is mounted on the oil pump assembly which is driven by a spline on the worm shaft.

Transmission Assembly

A modified version of the "Torqueflite 8" transmission without torque converter (Figure 20) is used to provide the torque multiplication and reverse gear necessary for satisfactory application of the engine to the car. Both front and rear transmission oil pumps are removed since they cannot provide the required control pressure with the vehicle at standstill, and the engine cannot be push-started. Oil is supplied for transmission lubrication and control from the engine oil pump. The torque converter bell housing is removed from the transmission case and its mounting function is served by a cast iron adapter plate which also replaces the front pump housing.

The control valve body is modified for various reasons as follows: (a) flow control valve is changed to a throttling valve to be compatible with external oil source, (b) shift valves are changed to provide automatic upshift in "Low" position for power turbine overspeed protection, (c) modifications for friction element capacity control, and (d) manual shift valve changed to eliminate neutral, locking the transmission by engaging forward and reverse elements simultaneously for "Start-Park" and "Idle".

POWERPLANT SYSTEMS

Lubrication and Hydraulic System

A combined lubrication and hydraulic system is used for the gas turbine, transmission, and power steering gear, as shown schematically in Figure 21. Automatic transmission fluid, Type A, adequately meets the requirements for all these functions, permitting the use of the combined system. A single pump is used, driven by the gas generator, drawing oil from the transmission sump. The pump itself (Figure 19) consists of the rotor and insert from a production 0.96 cubic inch slipper-type power

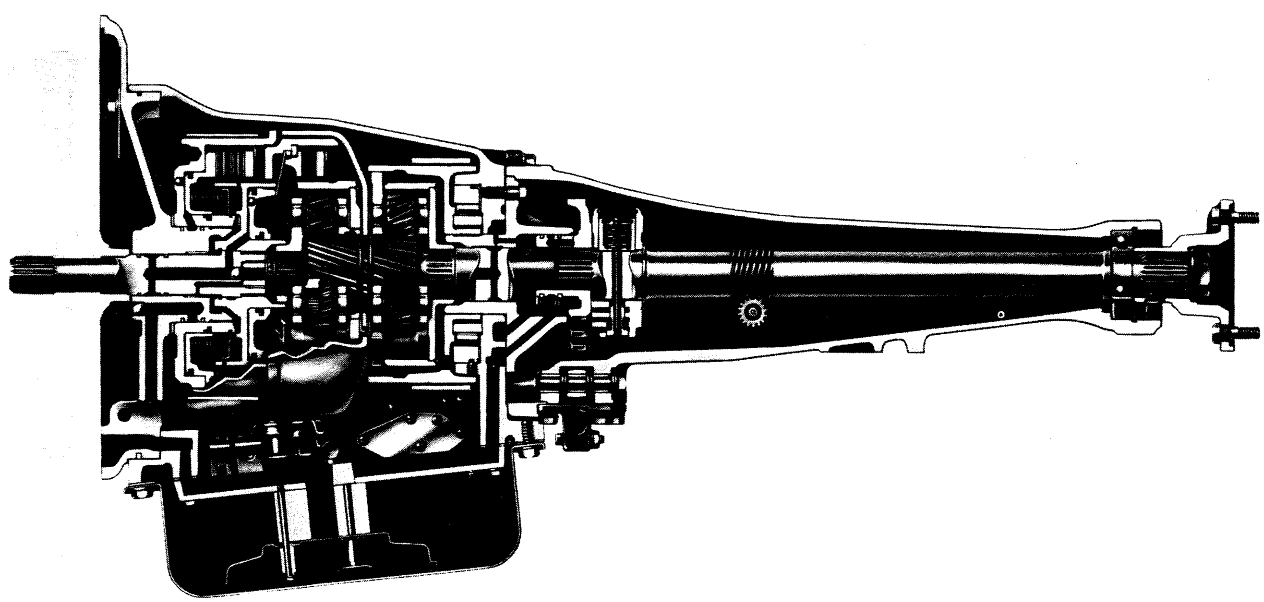

Fig. 20 - Turbine Car Transmission Assembly

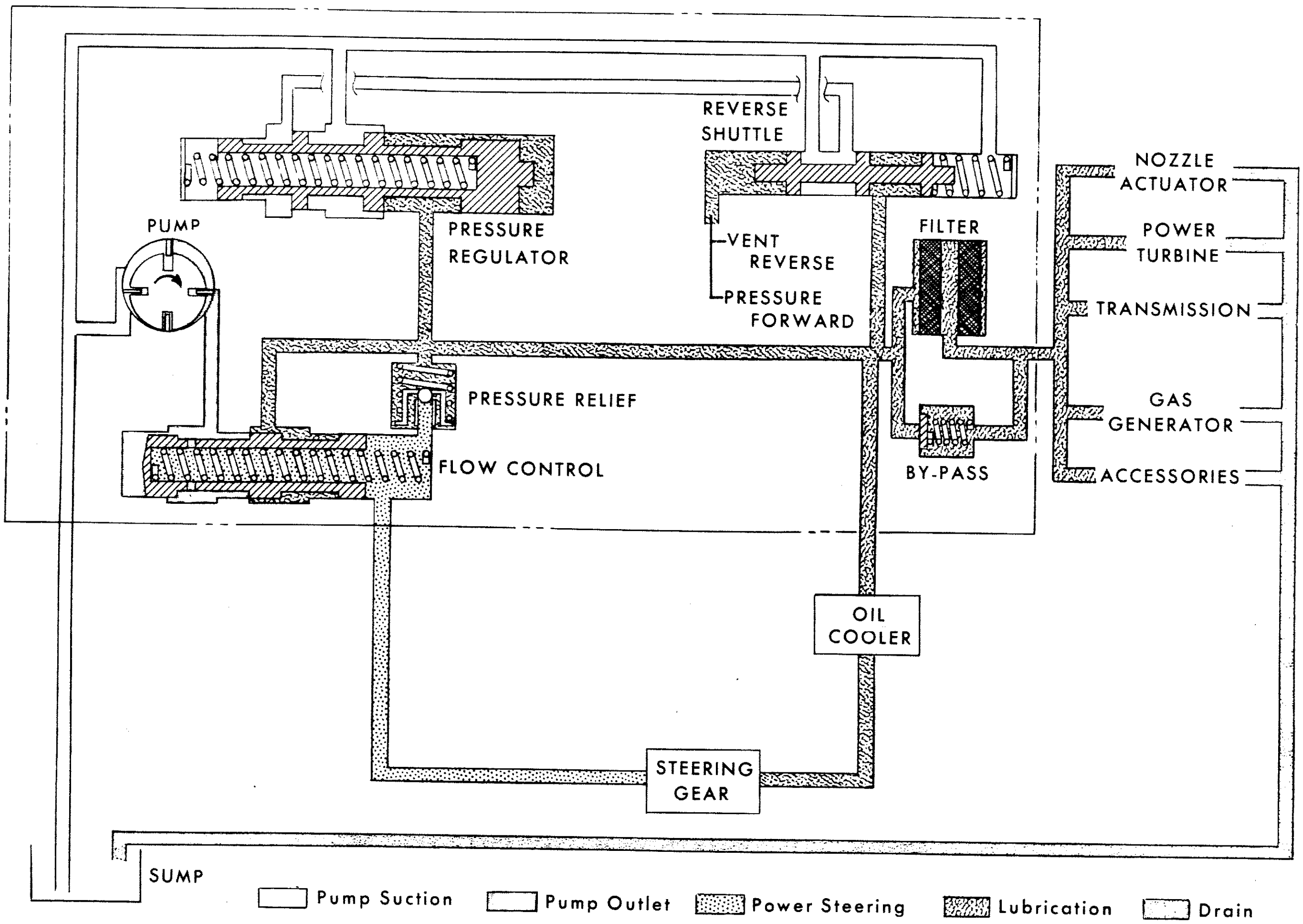

Fig. 21 - Lubrication and Hydraulic System

steering pump in a cast iron housing with a full-flow replaceable element oil filter and five valves; filter by-pass, lubrication circuit pressure regulator, steering circuit flow control, steering circuit high pressure relief, and a relay valve for boosting lubrication circuit pressure in reverse gear only to 160 psi for transmission torque capacity.

The pump outlet feeds the power circuit first at pressures up to 1050 psi as determined by steering requirements. The return from this circuit and the bypass from the flow control valve are regulated at a nominal 100 psi with full pump flow available to the lubrication circuit at all times. Drilled passages in the housings supply oil to the power turbine, regenerator drive worm gear box, and transmission, with external steel lines to the gas generator, accessory gear box, and power turbine nozzle actuator. An oil-to-air cooler in the power steering return line keeps the oil temperature below 300°F. All sections drain back to the transmission sump, with a breather on the power turbine housing to vent the air leakage from the shaft seals.

Fuel System

The fuel system consists of a Bendix Model DE fuel control assembly together with integral fuel pump, regenerator temperature compensator, altitude compensator, solenoid shut-off valve, Delavan air atomizing fuel nozzle, and

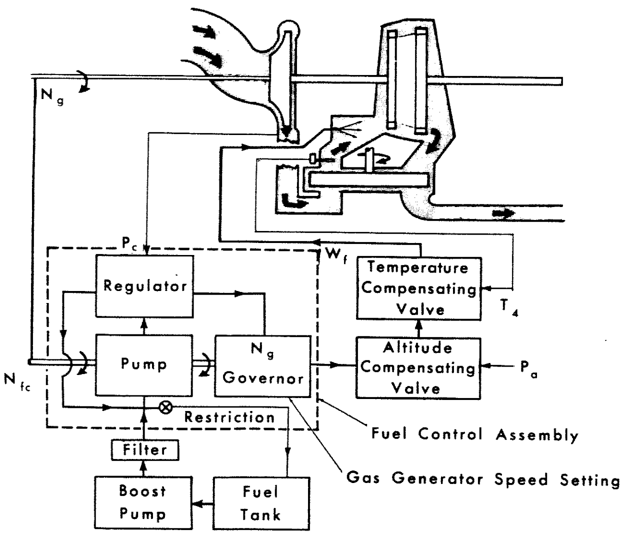

Fig. 22 - Fuel System Flow Diagram

air pump, as shown on Figures 22 and 23. In addition, an electric boost pump is used to prevent vapor lock problems with the wide variety of fuels used. The fuel control assembly contains a fuel pump, governor, pressure regulator, and metering orifice. During constant speed operation, the accelerator pedal sets the spring load on the governor, which regulates fuel flow to the burner fuel nozzle to hold speed as a function of pedal position. During gas generator acceleration, fuel flow is scheduled by the pressure regulator and orifice as a function of compressor discharge pressure, with compensation for regenerator outlet temperature and altitude to hold turbine inlet temperature approximately constant. During deceleration the governor throttle valve closes completely, shutting off all fuel to the nozzle, thus achieving the most rapid gas generator deceleration possible. As the gas generator rotor slows to the lower speed setting, the governor throttle valve opens and permits the fuel flow required to hold this speed. Fuel flow is automatically controlled during the engine start cycle and is not affected by accelerator pedal position until the gas generator reaches idle speed. The gas generator idles at 19,500 rpm when the transmission control is in "Start-Park". In "Idle" or any drive position, a mechanical actuated fast idle stop maintains the idle speed at 22,500 rpm to afford quick response in normal driving or maneuvering. Figure 24 illustrates typical control scheduling performance.

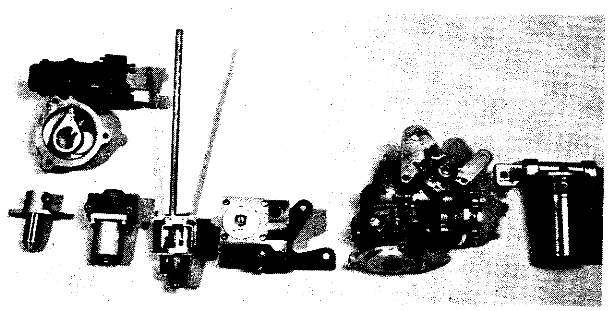

Fig. 23 - Fuel System Components

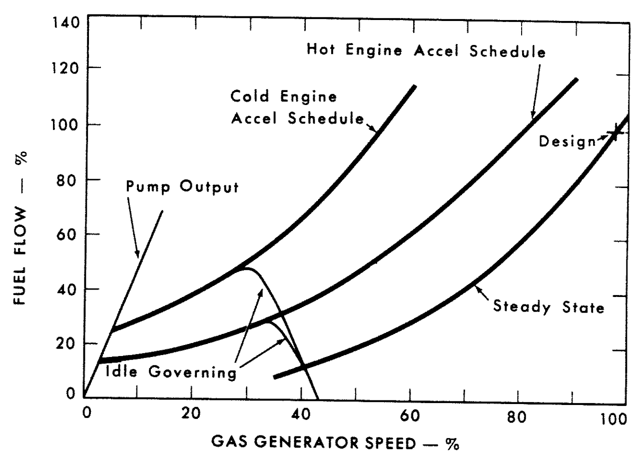

Fig. 24 - Fuel Control Schedule

Fuel is sprayed into the burner through a Delavan air-atomizing type nozzle, and the air to atomize the fuel is supplied by an engine driven air pump. The pump is a positive displacement, double acting, reciprocating piston unit with an external by-pass regulator valve. A solenoid valve is located between the fuel control and the nozzle to shut off fuel to stop the engine.

Control System

A mechanical linkage from the accelerator pedal sets the fuel control governor, transmission throttle valve, and power turbine nozzle actuator in unison as a function of pedal position. The fuel control and transmission operation have been described previously.

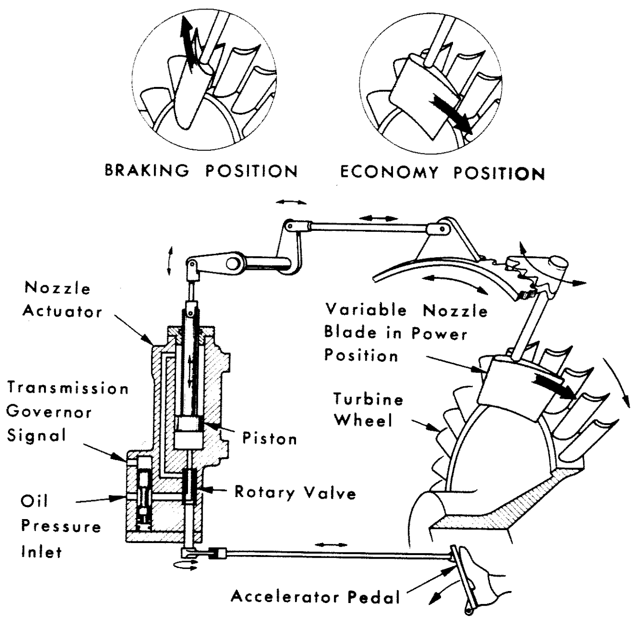

Fig. 25 - Variable Power Turbine Nozzle System

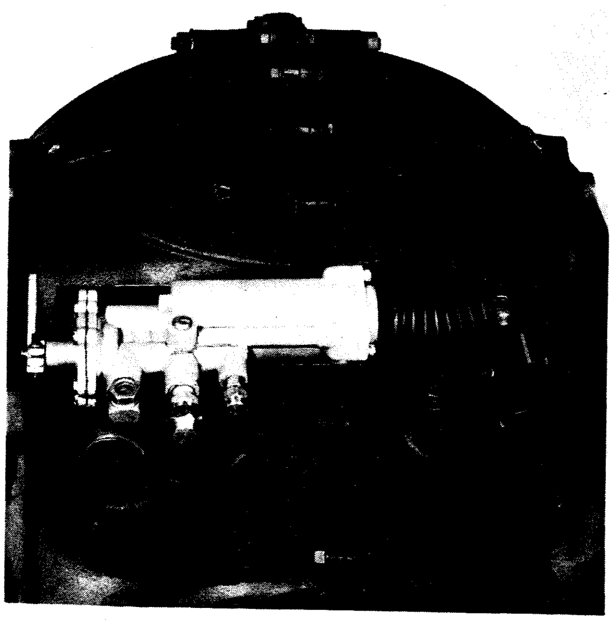

Fig. 26 - Power Turbine Nozzle Actuator

The power turbine nozzle actuator (Figures 25 and 26) is a cam controlled hydraulic servo unit which receives hydraulic power from the lubrication pump. It schedules nozzle blade position as a function of pedal position (equivalent to gas generator speed at steady state) permitting the engine to deliver high performance at optimum cycle conditions over the full power range without exceeding safe temperature limits. At starting or idle the nozzles are open with the blade directing the gas flow in an essentially axial direction to reduce turbine temperature and creep torque. As the acceleration pedal is depressed, the blades are closed to improve part load efficiency, with the setting varying to hold the turbine outlet temperature approximately constant. This gives the lowest specific fuel consumption possible within the temperature limitations of the engine materials. Figure 27 illustrates the power turbine nozzle schedule.

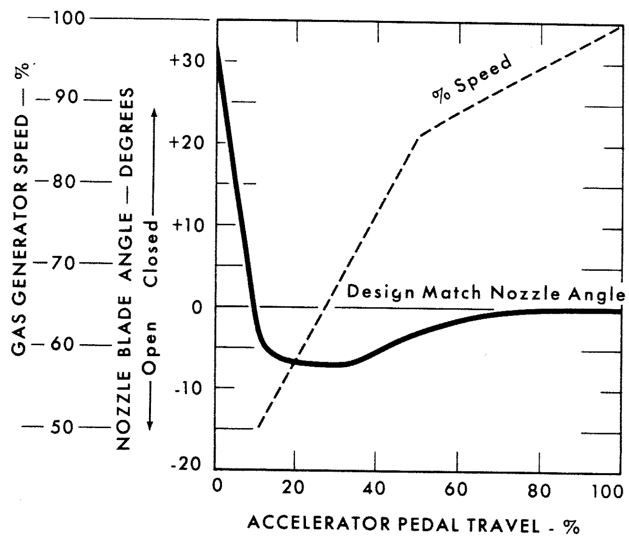

Fig. 27 - Power Turbine Nozzle Schedule

With the accelerator pedal all the way to the floor, the nozzle actuator reaches the rated power setting. Each engine is then matched to operate at rated temperature at rated gas generator speed by adjusting the linkage between the nozzle actuator and the blade ring gear. This adjustment corrects for minor variations in engine hardware, permitting each engine assembly to deliver its maximum safe output.

To provide engine braking, the actuator utilizes a pressure signal from the transmission governor. With the vehicle moving faster than 15 mph, releasing the pedal positions the nozzle blades to a reverse angle, directing the gas flow against the rotation of the power turbine wheel to retard the car. If the vehicle is standing still or moving at less than 15 mph when the pedal is released, the actuator merely positions the blades to the idle setting.

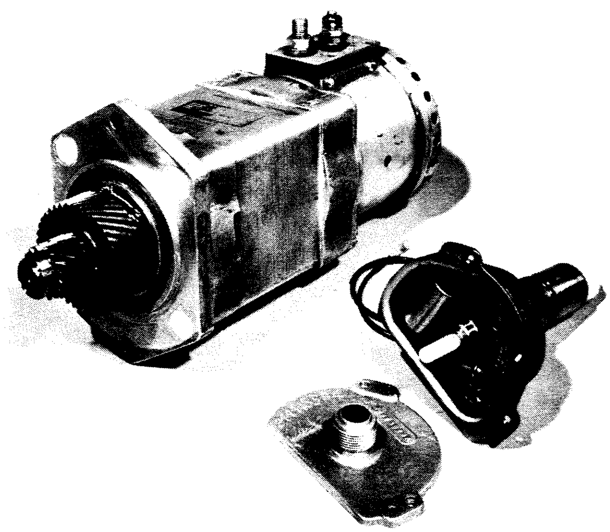

Fig. 28 - Electrical System Components

Electrical System

A starter-generator unit (Figure 28), mounted on the rear of the accessory case, functions as a 24-volt motor during the starting cycle and drives the gas generator and accessories until low idle speed is attained. At this self-sustaining speed, the starter is converted electrically to a generator, which then supplies 12-volt direct current power to the car electrical system. Utilizing two 12-volt, 59-ampere-hour automotive batteries connected in series as the power source, this system will provide adequate cold starting characteristics down to -20°F. Relays are provided in the circuit so that batteries are connected in series to provide 24 volts for starting and in parallel for 12-volt operation in the engine operating range.

An ignition unit (Figure 28), mounted on the front of the engine, is driven from the low speed accessory shaft. It has ignition breaker points which, in conjunction with a standard automotive ignition coil, fires a shielded igniter 80 to 200 cps when the gas generator is rotating. Although the igniter does not have to keep firing to maintain combustion, each time the gas generator decelerates "flame-out" occurs and the fuel must be reignited at the lower speed setting. This is achieved simply by operating the ignition system continuously.

Intake System

The intake system has three main functions: get air into the engine, keep foreign materials out, and absorb any air-borne noise. It should accomplish these things with minimum pressure loss and temperature rise, as in any air-breathing automotive engine. The problems associated with a gas turbine inlet, however, are more severe than with a reciprocating engine since the equivalent of the engine cooling air must also be processed through the intake system. The car intake system is shown in Figure 29 and is described below.

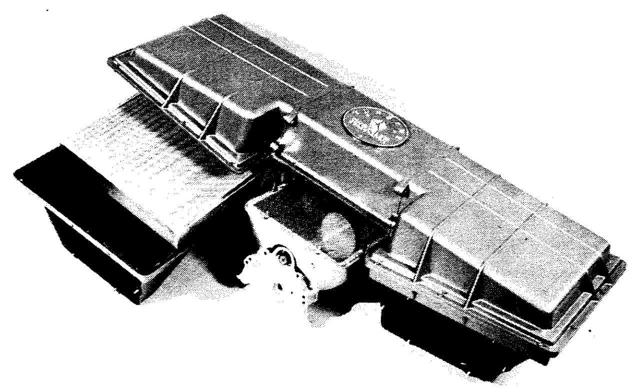

Fig. 29 - Turbine Car Intake System

The principal foreign materials to be removed are rain and splash, air-borne dirt, and any small parts that might drop into the intake elbow during servicing. Water is removed by the baffles in front of the duct opening just behind the front grille, and any remaining droplets would either evaporate or be absorbed on the paper filters which also serve to remove dust and dirt. Large solid particles are prevented from entering the duct by screening the inlet around the splash baffle. Finally a screen is placed over the entrance to the intake elbow for protection during servicing. The shape of the ducting is strongly influenced by the space available and a desire to have several bends for efficient sound absorption. This has resulted in flow passages that required internal vanes to obtain a compressor inlet velocity distribution within acceptable limits.

By constructing the intake ducting out of plastic, the temperature rise of the air is negligible. The pressure drop, however, is a significant 4 inches of water at maximum air flow, approximately half of this being the drop in the paper filters. There are two filter elements -- each having an active frontal area of about 13 inches by 15 inches and a depth of 2 inches. The paper is folded to obtain 7 pleats per inch along the 15 inch dimension, giving about 5,500 square inches of paper filter area. If it were feasible to increase this area significantly, the pressure drop could be correspondingly less. This experience to date, however, has indicated that the size and effectiveness of the filter has been satisfactory for passenger car service.

Exhaust System

The car exhaust system is shown in Figure 30. Two cast aluminum convergers, bolted to the regenerator covers, collect the exhaust gases from the regenerators and direct them into the ducts. Dual ducts are used, one for each regenerator, with a minimum flow area of 45 square inches total. The ducts are wide and shallow to fit under the floor, then curve over the rear axle, finally flaring downward to discharge the gases to the rear in the conventional manner.

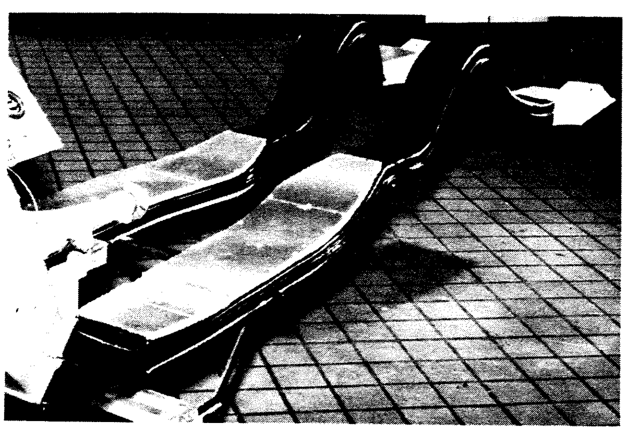

Fig. 30 - Turbine Car Exhaust System

SUMMARY

The Chrysler gas turbine powerplant has been designed for passenger car applications. The major components, compressor, turbines, regenerators, and burner, were selected to meet performance requirements — power, acceleration, braking, response, and fuel economy. The powerplant is built around these components as a compact assembly, emphasizing simplicity, ease of manufacture and assembly, and low production costs, while retaining the durability and reliability expected in automotive applications.

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