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CHRYSLER'S GAS TURBINE CAR

Laboratory Procedures
and
Development Methods

G. DeClaire and A. H. Bell
Chrysler Corp.

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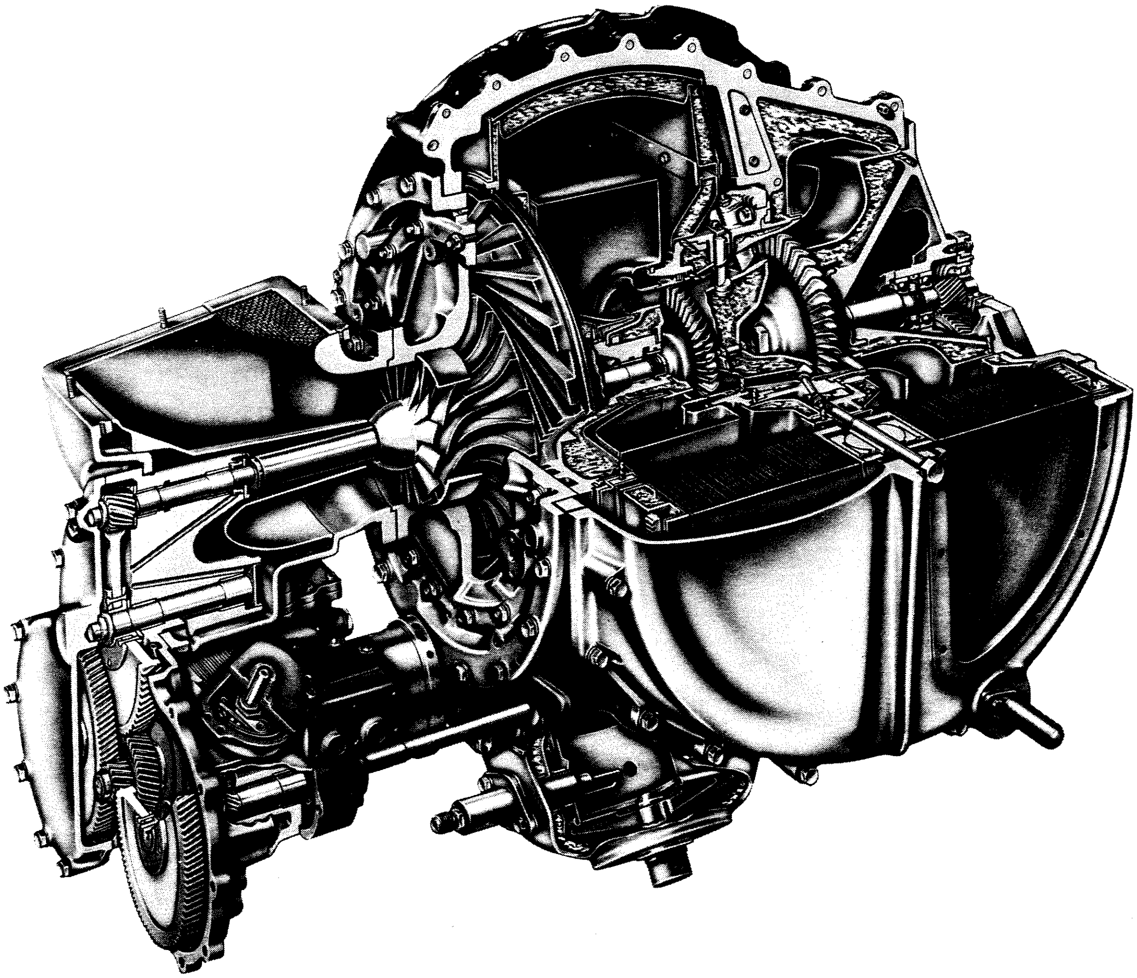

Fig. 1 - Chrysler Gas Turbine Engine

INTRODUCTION

The laboratory procedures and development techniques employed by Chrysler for its automotive gas turbine are bound to have similarities with those used on current reciprocating engines intended for the same application. Both power plants are of the internal combustion type, burning fuel in air; both must meet the wide range of a passenger car power demands and fit in basically the same engine compartment; and, finally, both are road tested in the same place in the same manner by essentially the same people. However, our interest does not lie in those similarities, but rather in the differences. These differences will become increasingly evident in the discussion that follows; in the facilities required, in the instrumentation required, in the varied types of component testing, and in the engine testing itself.

LABORATORY AND FACILITIES

The major portion of the gas turbine development laboratory is located at the Engineering Staff facilities of Chrysler Corporation in Highland Park, Michigan. The gas turbine laboratory is separated from the other engineering departments, although frequent use is made of those facilities which are common in the development of any automobile prime mover or vehicle. Among them are, for example: Cold Room, Sound and Vibration, Mechanical, Electrical, Lubrication, Rubber and Plastics Laboratories, and Inspection. The experience of the personnel of these departments is solicited in solving many problems not particularly related to the gas turbine alone. The Research Departments of Metallurgy, Chemistry, and Physics contribute continuously to their respective fields in the development program. In addition, limited and special component and powerplant test facilities are located at the Research Laboratories in our Greenfield Plant in Detroit, Michigan.

Test and development functions are performed in basic test cells. General size, arrangement, personnel safety, and equipment safeties of all cells are similar, but the amount and type of instrumentation and test facilities are dependent upon the function being performed. Eight dynamometer cells, including two at the Greenfield Plant, are fully equipped for powerplant testing and completely air conditioned for control of engine inlet temperature. Eight component test cells, including two at Greenfield, equipped with motoring or absorption drives where applicable, are used for compressor, turbine, burner, regenerator, fuel systems, passage flow, accessories, and general mechanical development. A more detailed description of the facilities available in any one cell will be discussed, as necessary for clarity, under subsequent component headings.

The gas turbine laboratory has its own compressed air supply. Two commercial air compressor units of approximately 2000 hp (total) are capable of supplying about 36,000 lb/hr at 50 psig. This air is supplied to all test cells at Highland Park with air measuring devices and pressure regulating valves in each cell. A centrifugal compressor assembly from an earlier Chrysler gas turbine powerplant, driven by two 413 cubic inch piston engines, provides an economical capability of delivering approximately 12,000 lb/hr at 45 psig for the Greenfield facilities. (Fig. 2).

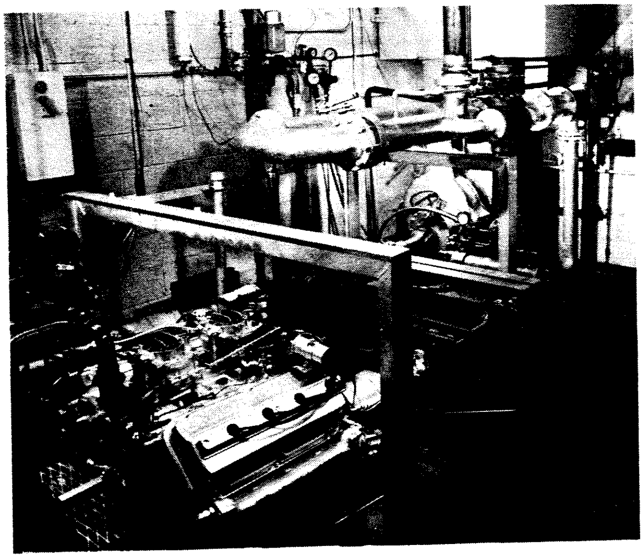

Fig. 2 - Laboratory Air Supply - Greenfield Plant

An air conditioned room furnished with adequate inspection equipment, a precision balancing machine, and assembly benches and fixtures is maintained for general assembly, dimensional recordings, and parts analysis and review.

The general laboratory area includes a car hoist, stockroom, and miscellaneous gaging and fixturing equipment.

A complete machine shop, including a pantograph machine capable of producing aerodynamic blading, is maintained at the Greenfield Plant.

INSTRUMENTATION

One definition offered by Webster for the word "instrument" is as follows: "a device for measuring the present value of the quantity under observation." Although somewhat dramatic, this definition bears out the importance of the device since subsequent data presentation based on measurements with this device must lead to conclusions and program direction. In over 15 years of automotive gas turbine development, these devices have played a very major role and the effort spent in instrumentation development has often exceeded that applied directly to the turbine or its components. In many cases a device has to be "invented", this invention being dictated by accessibility and the extremely small size of flow passages, and the lack of a commercial source for our particular requirements.

A classification of instrumentation types is somewhat arbitrary and dependent upon the field of endeavor -- this field being defined as general aero-thermo-mechanical development. Let us include primary or sensing elements and secondary or read-out elements in the general field of instrumentation. It is obvious that accuracy and dependability of both elements are necessary. Standard commercial types, as well as special "invented" instruments, are available to measure both steady state and transient phenomena. Equipment is also available to calibrate or evaluate these devices.

Standard laboratory instrumentation used in all test cells on powerplants or components for mechanical, aerodynamic, and thermodynamic evaluation include the following:

1. Power Absorption
Full electric or eddy current dynamometers with associated speed and torque measuring systems are available. Calibration is checked at regular intervals.

2. Pressure
Routine pressure sensing, in either liquids or gases, is performed with conventional static pressure taps .020 - .040 inches diameter or with pitot type total pressure probes. These routine measurements are usually limited to regions of low Mach number and low velocity gradients. Each cell has a minimum of twenty 60 inch and two 120 inch manometers, and portable banks are available. Pressure transducers are employed for transients and the output read with light beam oscillographs.

3. Temperature Sensing
Chromel-alumel thermocouples are generally used although more exotic materials are available for extreme temperature sensing. Random samples from all purchased reels of wire are calibrated in a furnace against a Bureau of Standards master for millivolt output.

4. Temperature Recording
Dual range self balancing potentiometers are standard. Periodic calibration of the sensing and read-out systems ensures an overall accuracy of ± 1%. Light beam galvanometer oscillographs are employed for transient temperature measurements.

5. Air Flow
Flow nozzles and sharp-edged orifices are used exclusively for accurate flow measurements. These are calibrated against laboratory masters for flow coefficient. The masters are determined by a complete total pressure traverse across the nozzle throat from the passage center through the boundary layer to the wall. Calibration accuracy is within ± 0.5%. Partial flow venturi tubes have been used with various degrees of success. Air rotameters are used in tests where accuracy is not of prime importance.

6. Speed
Gas generator, compressor, and turbine component speeds are sensed with reluctance type pick-ups. The output of the pick-up is fed into commercial digital type counters -- read-out being to the nearest 10 rpm. Light beam oscillographs are used for transients.

7. Fuel Flow
Precision rotameters for steady state fuel flow measurements are periodically calibrated for different fuels against a laboratory master to within ± 0.5%.

8. Vibration and Noise
Commercial accelerometer and velocity type pick-ups and microphones are employed with oscilloscopes, oscillographs, and tape recorders. Outputs are analyzed with the aid of band pass filters and wave analyzers.

Special instrumentation, i.e., that not used routinely in the course of development, is outlined below. Because of the nature and detail of some of the work, a more complete description of the techniques involved is omitted here and may be discussed under a particular component for reasons of clarity.

1. Torque Measurements

Cradling of the compressor turbine nozzle in a cold air fixture on anti-friction bearings is employed for determining torque reaction.

Cradling of air turbine drive nozzles on hydrostatic air bearings is used for determining minute torques in bearing and seal fixtures.

Cradling of a special speed increaser gear box is used in compressor testing for determining compressor work.

2. Pressure Sensing

The types of total, static, and flow angle sensing probes available to the aerodynamicist are well covered in the literature. It is sufficient to say that all have been tried at some time or other and the choice is based on accessibility, physical space, environmental temperature, and mechanical reliability.

Calibration for flow angle, Mach number and shock effects is tedious and time consuming, as is their installation in powerplants or components. Once the measuring station is selected and the probe type defined, the engineer invariably demands a radial and partial circumferential traverse and tram. This requires the use of remotely operated equipment for reasons of safety. Fortunately, commercial equipment is available and has been purchased for this purpose.

A more sophisticated piece of equipment, the hot wire anemometer, is used to measure the flow variations as a function of time at any one point in a flow passage. The use of this device has thus far been limited to compressor impeller flow studies and to cold flow turbine and cascade studies.

3. Temperature Sensing

Special temperature probes in regions of high temperature and Mach Number present a similar development story. Calibration for velocity, convection, radiation, and conduction is difficult and is dependent upon mass flow and environment. The response or characteristic time of a thermocouple, so important in transient studies, depends upon these same factors. For these studies, thermocouples are installed in their environment and a cold air jet is squirted on the thermocouple with the gas temperature at equilibrium. The cold jet is stopped, simulating a step function input, and the thermocouple output is recorded on a light beam oscilloscope and the characteristic time is thus computed.

The measurement of temperature gradients in impellers and turbine wheels in engine operation is necessary for minimum inertia disc design. Therefore, slip rings have been designed, fabricated, and developed for this work.

4. Fuel Flow

Transient fuel flows are measured with commercial turbine type flow meters. Care must be exercised in the insertion of this device in the fuel system plumbing so as not to upset the scheduling of the fuel control.

5. Strain Measurements

Conventional stress coat and strain gage techniques are employed for general powerplant structural members. Dynamic stress determination in impellers and turbines requires the use of special high temperature gages and slip rings.

6. Exhaust Gas Analysis

Because of the large excess of air associated with the gas turbine combustor and its very high combustion efficiency, exhaust gas analysis requires a more sophisticated array of instrumentation than used with conventional piston engines. The Chemical Research people assume complete responsibility for this analysis, using non-dispersive infrared analyzers for the determination of CO_2 and CO with an accuracy of ± 2 ppm. Unburned hydrocarbons are measured with a flame ionization analyzer with an accuracy of ± .025 ppm. The phenol disulfonic acid method is used to determine the concentration of oxides of nitrogen.

The above discussion is only intended to introduce the reader to the wide scope of gas turbine instrumentation requirements. Many more pages and words could be devoted to the how's and why's of data acquisition.

COMPONENTS

As a development program is planned, a designer first wants to know how good a job he has done and it is only through testing that he will get the answer. Since the gas turbine is a continuous flow engine, it is possible to test major parts independently for their performance parameters. The compressor, the turbines, the burner, the regenerator, the accessory drive system, and fuel control and all the rest of the devices which can be independently tested, are major components. Logically, then, the laboratory must start with component testing and development. As this utilizes half of our test facilities, we feel it deserves special emphasis here.

A 300 hp dynamometer that has increasing torque capacity with decreasing speed, and direct coupled power absorption at speeds of 50,000 rpm, is not exactly common place. To top it all off, a large capacity, controllable, stable hot air supply is needed. In our early days of testing, it was not unusual to lose a number of hours of delicate probe results because the capacity and stability of the air supply were not sufficient and the unit drifted off the test setting.

Our component testing started some 15 years ago, and the fixtures and rooms developed to do this work have been used continuously to develop the parts for our current engine. Many of our engineers feel that it is just as difficult to develop a completely satisfactory component stand facility as it is to develop a whole powerplant. These facilities, their flexibility and independence, are necessary to improve engine parts continuously.

It is easy to plan a component test, but quite another thing to execute it. First, the test fixture should have all the environmental features of the engine with none of its restrictions. It must be sturdy and reliable, for nothing is quite so frustrating as the interrupting of a complex test by failure of an incidental part. Consequently, the fixture design incorporates as many commercial components as possible, well proven in conventional usage. But the supplier's first reaction usually tells us how far from the shelf we really are, "We've often wondered about our stuff running at these speeds and loads. We'll be glad to sell you some parts - let us know how they work! "

Compressor

Compressor testing was first attempted in a standard dynamometer test cell, but the requirements of the compressor are so specific that this testing is now done in a room designed especially for this purpose. The requirements of this cell were determined to be:

1. Speeds to 70,000 rpm
2. Drive power to 600 hp

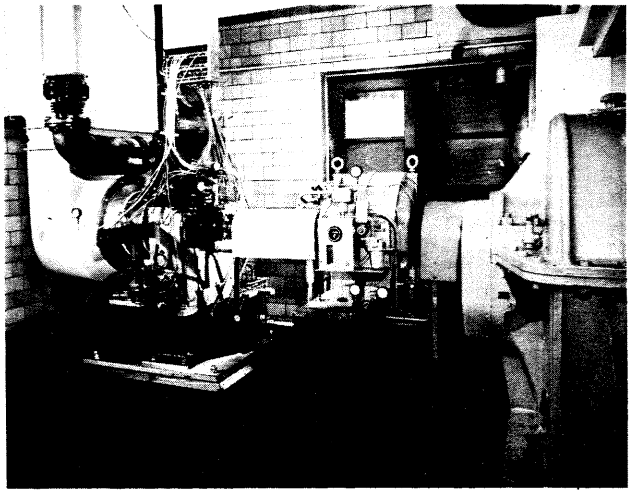

Fig. 3 - Compressor Test Cell

The compressor cell is shown in Fig. 3. Not seen in this figure is a 1,500 hp constant speed electric motor which supplies power for the laboratory air system and for the drive gear box at the extreme right-hand side of the picture. The gear box increases the speed from 720 rpm to 3,600 rpm at this output shaft, and is rated at 1,100 hp. Following the gear box is an eddy current coupling which is used to control speed. This particular coupling is rated at 600 hp and controls speed effectively at power values down to about 1% of the rating. Following the magnetic clutch is a cradled set of turbine engine reduction gears. The arm protruding through the side is used to measure torque to the compressor fixture while the box itself produces the 20.3 to 1 speed increase required. Above the fixture is a manometer junction and electric outlet manifold. The usual jungle of mixed wires and manometer lines is apparent.

Each compressor assembly has its own group of specialized equipment. 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. Tests are also made with flow visualization tufts, smoke filaments, vibration pick-ups, clearance rub rods, and boundary layer trip wires. Fig. 4 shows the compressor control console. At the extreme left of the picture is the controller for the remote probes.

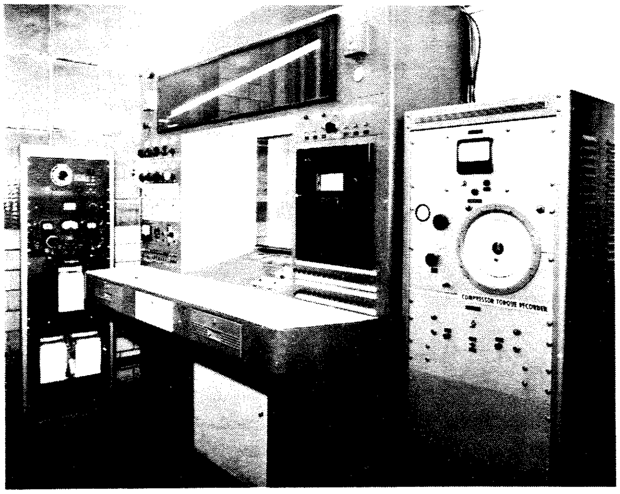

Fig. 4 - Compressor Test Cell - Control Console

Test data are programmed for a digital computer to obtain standard compressor parameters such as were shown in Ref. 1. This reference presents a complete chronological history of the various designs leading to the current compressor. Compressor efficiency is based on a pressure-temperature relationship rather than a pressure-torque relationship because evaluation of losses in the gear box has given less consistent results throughout the program. Torque-based efficiency has been both higher and lower than temperature-based efficiency within a given performance test.

Because a compressor development program requires more than a simple comparison of one compressor with another, each assembly is fitted with its own unique instrumentation to follow the fluid stream through the compressor and evaluate the losses caused by wall friction, blade wakes, zones of separation and other phenomena which cause flow maldistribution. These investigations give an insight into the various disturbances in the compression process and allow intelligent changes in hardware, using the best features of many build-ups to arrive at an optimum combination. In this way, we have been able to improve the maximum compressor efficiency from 0.742 to 0.835 over a 15-year period, and we have justified the original choice of a radial type compressor for the automotive gas turbine.

In addition, compressor testing is done on complete engines. As the current engine was first tested, a tendency to surge during rapid accelerations through the mid speed range was encountered. An engine was built which had simultaneous air flow measurements in each of the 29 compressor diffuser passages. Measurements showed a marked maldistribution caused by rather small non-symmetrical conditions within the housing plenum. An immediate fix for this problem was made by introducing small balancing restrictions in selected diffuser discharge passages. Because of the resulting small loss in efficiency, this was regarded as a temporary fix until the plenum could be increased. A further gain in compressor stability was made later by modification of the compressor diffuser nose.

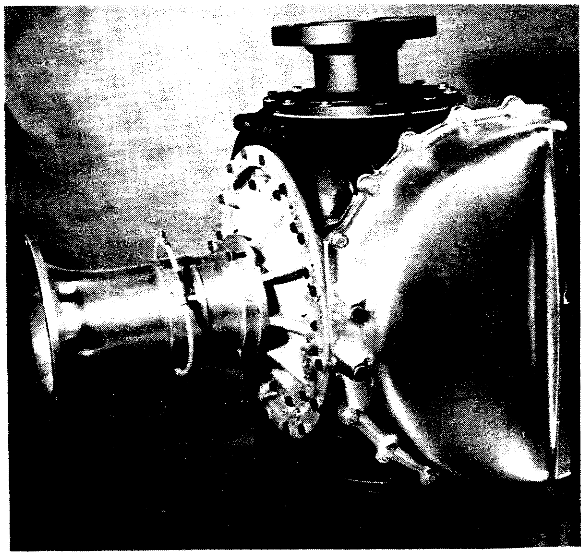

Fig. 5 - "Engine" Compressor Test Fixture

An "engine" compressor test fixture was built as shown in Fig. 5. The fixture is made of engine parts and completely duplicates the environment in which the compressor normally operates.

Regenerator

The regenerator used in the Chrysler turbine is a product of a very intensive development effort. It is not our intention to discuss this entire detailed history concerning all of the different paths pursued with resultant successes and pit-falls, but to concern ourselves with the present product and general development techniques.

The regenerator will be considered as an assembly, including matrix or core, static and rubbing seals, inlet and outlet flow passages, and drive system. Overall performance of this assembly must include thermodynamic behavior, leakage or sealing, wear, friction, compatibility of the rubbing seals, power required, bearings and drive system, ease of manufacture and assembly, and endurance. It becomes apparent that the final selection involves consideration of all these requirements.

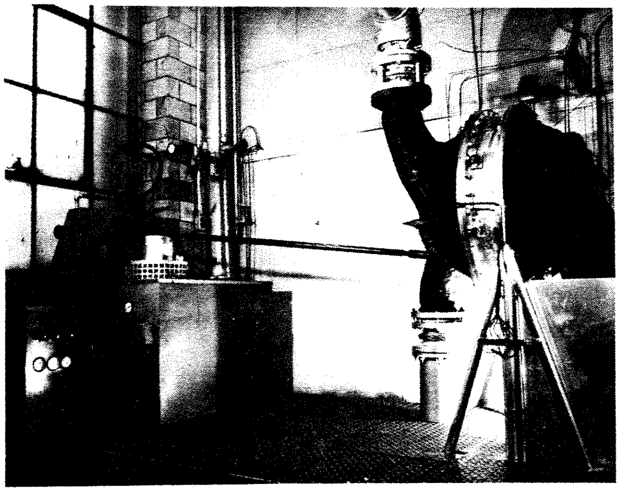

Fig. 6 - Regenerator Test Cell

A number of fixtures have been conceived and built. The general development fixture, Fig. 6, has been designed to allow duplication of engine environment including inlet and outlet flow passages. Overall assembly parameters can be determined in this fixture. Regulated air is supplied from the laboratory air system and is heated in an engine type burner to compressor outlet temperature. The air flows through the high pressure side of the core, is throttled to turbine outlet pressure, reheated in another burner to turbine outlet temperature and then flows through the low pressure side of the core. (Fig. 7). The core is driven by a variable speed air motor which is cradled for determining torque requirements.

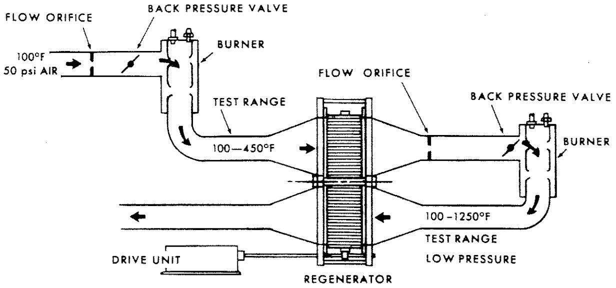

Fig. 7 - Regenerator Test Fixture Schematic

A sub-component type fixture is used for evaluating the heat transfer characteristics and thermal shock resistance of matrix samples. The fixture will accept small rectangular matrix sample. Periodic flow is induced by fixing the sample and valving hot and cold gases in a counter-flow arrangement for controlled time intervals.

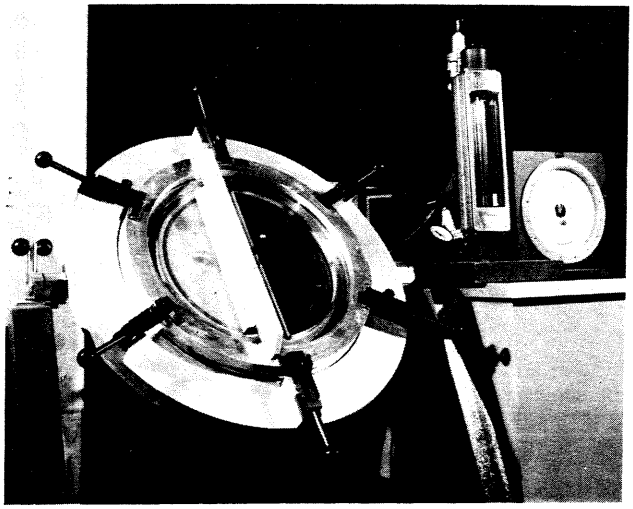

Fig. 8 - Regenerator Static Seal Test Fixture

Other fixtures are used to evaluate the static seals and matrix bonding at room temperature and elevated pressures. (Fig. 8 and 9).

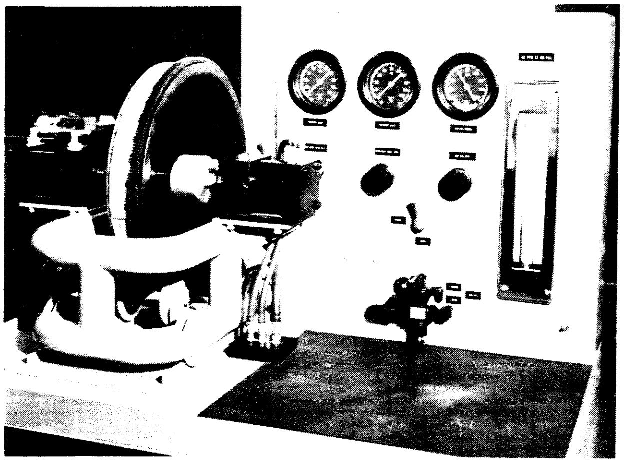

Fig. 9 - Regenerator Core Leak Tester

The thermodynamic performance of the regenerator can be best 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 for measuring performance and flow distribution as affected by inlet and outlet flow ducts. Both radial and circumferential temperatures and pressure distributions are measured. Matrix heat storage utilization factor, as influenced by core rotational speed, is determined for all flow conditions and is used in selecting optimum regenerator drive ratio.

The control of leakage of the regenerator assembly is a prime factor in powerplant performance. The level of leakage influences output and SFC, and variations from assembly to assembly make compressor-turbine matching a nightmare. The assembly leakage can be divided into matrix carryover, matrix bonding, rubbing seals, and static seals.

Carryover leakage is a function of core speed and is about 0.75% of total airflow for this powerplant.

Matrix leakage is a function of degree of bonding and is dependent upon fabrication techniques. The ability to produce consistent, completely bonded cores was evolved over a period of years. The process was completely developed within the Research group between the personnel of the Metallurgical Department and Gas Turbine Laboratory. The selection of the presently used matrix stock, .002 in. AISI 430 stainless steel, was dictated by performance goals, strength, oxidation, bondability, fabrication, and cost. Original matrix shapes were produced in the laboratory. All cores are wound on a machine designed by Chrysler (Fig. 10) and controlled atmosphere furnace brazing was initially done in Metallurgical Research.

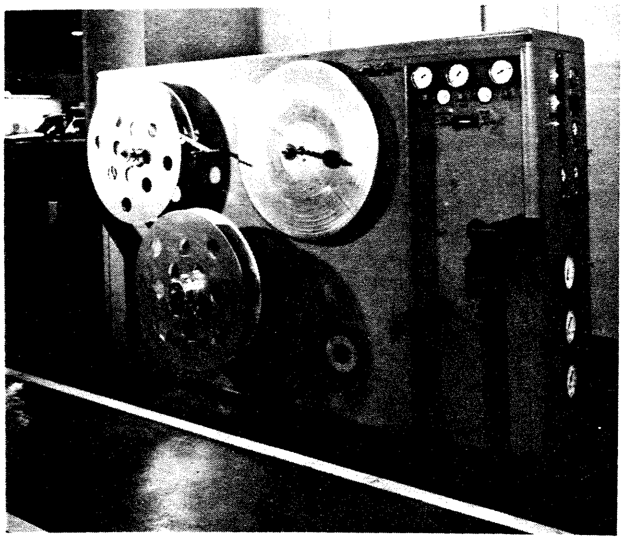

Fig. 10 - Regenerator Core Winding Machine

The factors that influence reliable core production are numerous and somewhat inter-dependent. Competitive security prohibits more than just a casual inkling into these factors at this time.

Stock strip straightness and passage uniformity, winding tension, distribution and method of applying the brazing alloy, fixturing and support in the brazing furnace, furnace atmosphere, rate of furnace heating, brazing temperature, and cooling cycle are the major parameters that were determined. The controls established for these parameters are felt to be tolerant enough for economical production. Matrix bonding, free of distortion, is essentially complete, and leakage of any 2 inch diameter section is less than 5 lb/hr at 50 psig.

The rubbing and static seals presented an even more perplexing problem than core fabrication. Our selected sealing arrangement from the high to low pressure areas requires a seal diametrically across the matrix and over one half of the core circumference, as well as a circumferential seal in the high pressure section on the hot side to prevent by-pass. The outer seals operate in gas temperatures of 550°F maximum and the inner seals in a gas temperature of 1250°F maximum. The former must seal a pressure difference of 50 psi maximum and do the job unlubricated. Specific grades of graphite are excellent for friction, wear, and conformability, if loads, reaction surfaces, and temperatures can be controlled. Relatively low strengths (requiring judicious mechanical design), brittleness, cost, and rapid oxidation above 1100°F limit its application. Graphite is presently used for the outer seals and inner rim seals, but not for the inner crossarm because of the unfavorable operating environment. Early program engines "wore out" a cross arm within 100 hours of operation. Temperature, material stability, matrix surface, low coefficient of friction, and abradability or wear-in requirements imposed a serious challenge to the metallurgist. The latter requirement is the result of the somewhat spherical shape a disc-type regenerator assumes when subjected to a temperature gradient. The results achieved by our Research Metallurgists in the development of a proprietary material are borne out in this power plant. Endurance in excess of 2000 hours has been achieved on complete seal assemblies with little wear other than break-in.

The static seals are those "D" shaped members that bridge the gap between the engine housing and rubbing seals. The regenerator assembly is installed in the engine housing with reasonably tolerant axial clearance and these seals must float in this region and adjust for machining tolerances, core thermal deflection, and thermal expansion differences. They are an assembly of sheet metal constructed to resist thermal and mechanical fatigue.

The wear of the rubbing seal, its compatibility with its wear surface, and the torque required to rotate the core depend upon the coefficient of friction and normal load. The power required, although small, is still a parasitic engine load and is not considered lightly. The normal load is dependent upon the pressure forces acting on the rubbing and static seals. The geometry of the seals dictates these forces somewhat straight-forwardly, except for the gradient between the rubbing seal and matrix. Pressure gradients on both rim and crossarm have been measured and the seals have been changed in such a manner as to ensure a maximum of 15 psi unit seal loading. Core torque is measured in both the general fixture and power plant and is less than 80 lb-ft per core.

Though each component of the regenerator assembly is developed and checked in its own type of fixture for sealing ability, the total leakage of the assembly at operating temperatures and pressures is the proof of the pudding. The absolute leakage of a regenerator in power plant operation is next to impossible to determine. Energy balances and compressor-turbine flow parameter comparisons are dependent upon so many measurements -- each subject to obvious inaccuracies -- that the estimated leakage is at best a "ball park" figure, good only for comparison purposes. Yet, the general regenerator fixture has the ability to determine leakage with extreme accuracy.

This statement obviously requires some explanation. Early overall performance test procedures in the general fixture consisted of running simulated powerplant road load points from idle to full load. Engine conditions of air flow, pressure, and gas temperature were stabilized at each test point. Leakage was determined by measuring the inlet air flow before entering the first burner and adding to it the fuel added in the burner and then measuring the air flow leaving the high pressure side of the core before it entered the make-up burner. This system was theoretically sound. Practically, it would determine assembly leakage to within ± 1% if each flow nozzle was accurate to within ± 0.5%. The primary flow nozzle was in a cold air stream and its calibration remained essentially fixed with time. The secondary nozzle, however, was located in the hot gas leg and subjected to scaling, thermal distortion, and abrasion of both nozzle and pressure sensors. Frequent disassembly for recalibration was time consuming and expensive and one never could be certain when the nozzle calibration began to deviate. Besides, a leakage determination within ± 1% at best was hardly acceptable with total leakage in the neighborhood of 2.5%. (fixture tests would yield an answer of 1.5 to 3.5%). At this level of leakage, each 0.1% is hard to come by, but worth digging for. A simple scheme was then devised to improve this accuracy by an order of magnitude with the lone realistic assumption that assembly leakage is a function of pressure and temperature only and independent of mass flow. Mass flow primarily affects core effectiveness and has only a second order effect on core thermal distortion. Thus, for leakage evaluation, the flow nozzles are replaced with small calibrated orifices, and tests are run at simulated powerplant points of pressure and temperature, but with only 10% of engine air flow. Measuring accuracies remain the same, but now the difference flow determined by this method is 10 times more accurate. Effectiveness tests are run, of course, under engine flow conditions.

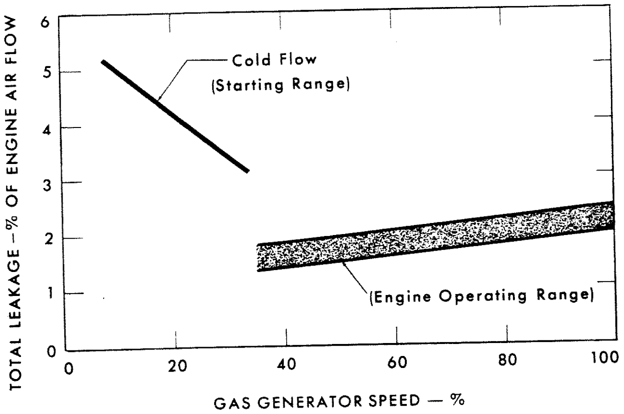

Fig. 11 - Regenerator Air Leakage -vs- Gas Generator Speed

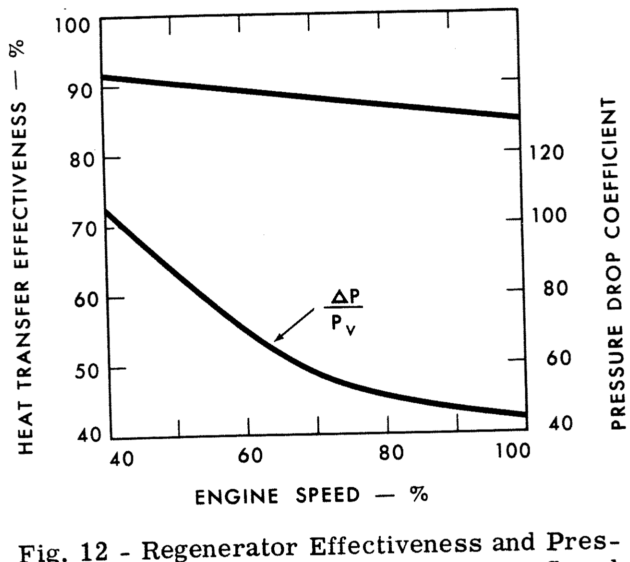

Fig. 12 - Regenerator Effectiveness and Pressure Drop -vs- Gas Generator Speed

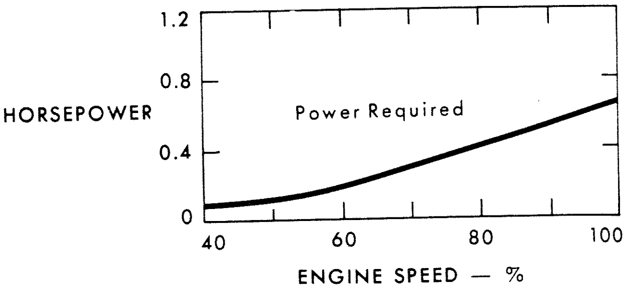

Fig. 13 - Regenerator Drive Power Required -vs- Gas Generator Speed

Total assembly leakage of less than 3% is the rule rather than the exception. In summary, leakage is 0.75% carryover, 0.5% static seals, negligible matrix leakage --the remainder being between core and rubbing seals. Figures 11, 12 and 13 show the overall regenerator performance

The manufacturing tolerance on all parts is within conventional sheet metal and machine tool capabilities. The whole assembly is placed in the engine housing with enough clearance to be referred to as "farm machinery" by non-regenerator personnel.

Burner

The burner, fuel nozzle, and igniter comprise an interacting system, with changes in either the physical environment or in any of the components likely to affect the system performance. Therefore, to achieve satisfactory operation, a complete combustion system must be developed. The underlying fundamental concepts governing combustion system operation are understood, but they cannot be accurately predicted. Thus, development becomes an art rather than a science, with final system evaluation and analysis performed in complete power plant assemblies. However, valuable information regarding operating characteristics can be obtained by studying the components individually.

Burner air flow investigations are made using plastic model burner tubes flowing room temperature air. Smoke tracers are used for observation of the mixing processes that are important for obtaining maximum combustion efficiency and proper temperature distribution.

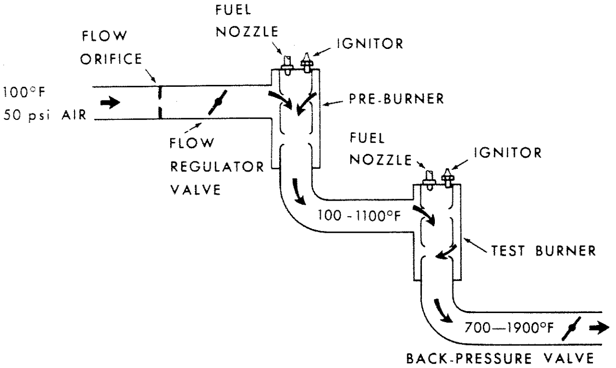

Fig. 14 - Burner Research Fixture - Schematic Flow Diagram

Testing then proceeds to the full scale burner test fixture which consists essentially of two burners in series (Fig. 14). A primary burner simulating engine regenerators supplies heated air to the test burner, allowing duplication of any desired engine operation condition. The temperature rise in the primary burner is significant and complete combustion is necessary so as not to affect the burning process in the test burner. The primary burner, a commercial unit, can be seen in Fig. 15. The test burner is complete with its surrounding engine environment. This fixture, used for steady state burner evaluation, contains viewing ports for visible observation of the combustion process.

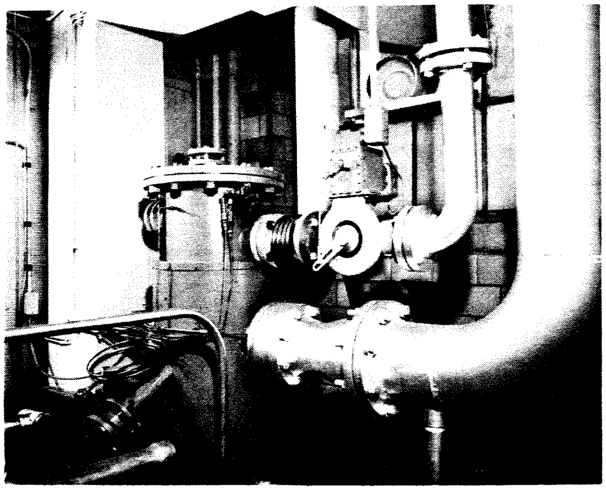

Fig. 15 - Burner Test Cell - Primary Burner

After satisfactory steady state burner operation is obtained, testing is concentrated upon the more stringent requirements of vehicle powerplant operation, including transient conditions, low ambient temperatures, and multi-fuel capability. These tests are conducted using complete engines in test cells and vehicles, as well as with specialized fixtures based on powerplant hardware.

A non-regenerative engine housing burner test fixture (Fig. 16) is used for determining the starting cycle and low temperature requirements of fuel atomization, ignition, and combustion quality with the required range of fuels. The fixture is a complete engine without a power turbine wheel, modified so that the exhaust gases do not pre-heat the air prior to combustion.

A regenerative engine housing burner fixture is used as a test bed for both transient and steady state engine operation (Fig. 17a and 17b). The fixture is a complete engine without a power turbine wheel and with provision for regulating the amount of air pre-heating prior to combustion.

The parameters considered in combustion system development are: combustion efficiency, exit temperature distribution, pressure drop, metal temperatures (life consideration), ability to use various types of fuels, ignition characteristics (including very low ambient temperatures), carbon formation tendencies, and exhaust smoke and odor. Lean limit, a common parameter in burner testing, is not of importance since the fuel flow is stopped on engine decelerations, extinguishing the flame.

Combustion efficiency is obtained using either the temperature rise or the exhaust gas analysis method. The temperature rise method requires very accurate measurement of the actual temperature of the gases through the burner. The far more reliable exhaust gas analysis method involves the chemical analysis and measurement of the products of combustion.

Air temperature measurements are made at the burner exit, as well as within the burner, to evaluate the temperature distribution and the burning process. Both stationary and traversing thermocouple arrangements are utilized. The initial design goal to maintain the turbine

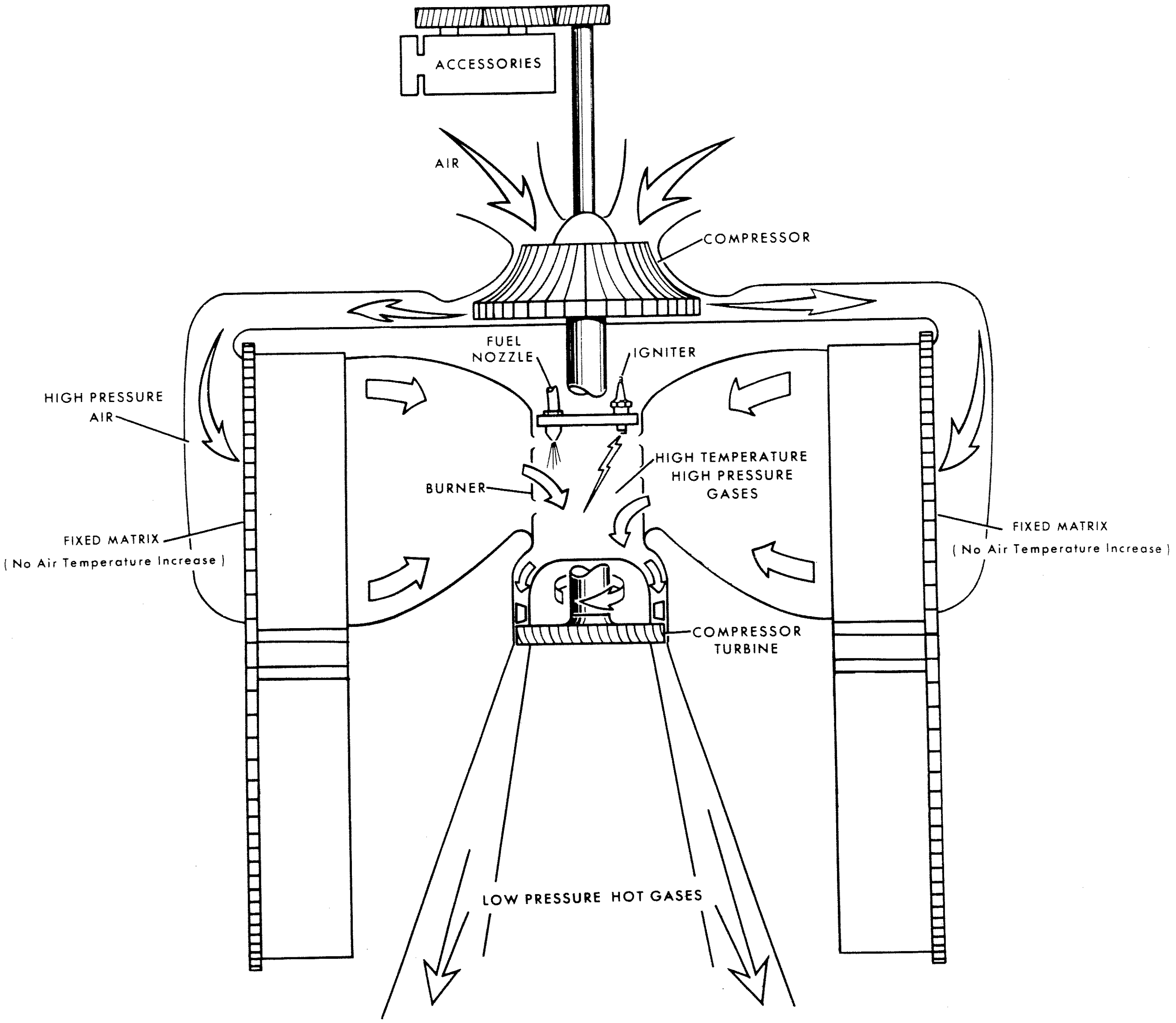

Fig. 16 - Engine Burner Test Fixture - Starting Cycle Development

inlet temperature variation below 50°F has been surpassed; we now have less than a 25°F temperature variation at any given engine condition.

Burner liner metal temperatures are used for determining the amount of burner wall cooling required for adequate life as well as for evaluating the progression of the burning process by noting the temperature distribution along the burner length. Exhaust odor has been determined to be extremely sensitive to liner temperature due to end quenching of the flame.

Both the gas temperature distribution and metal temperature limitations have become more difficult with the burner liner used in our current engine because of the relatively shorter primary combustion and secondary mixing zones.

The ignition characteristic is very important, since delayed ignition can precipitate regenerator failure by wetting the core with raw fuel which then burns on the core face. The problem exists not only at initial light-off at low ambient temperatures with high viscosity fuels, but also during vehicle operation, since the fuel flow is stopped during deceleration and re-light must occur the instant fuel flow is resumed for satisfactory engine response. The most important factor for quick ignition of the fuel-air mixture is proper location of the electrical spark for all conditions of operation, and it is dependent upon the degree of fuel atomization provided by the fuel nozzle. A high energy spark source could project the spark to the proper location in the burner but it is costly and short-lived during continuous operation. Therefore, a low energy spark source is used with the spark gap positioned by trial to optimize performance and life. An air pump supplies the fuel nozzle atomizing air and the ignitor electrode cooling air. The ignition timing problem is eliminated by having continuous sparking ignition.

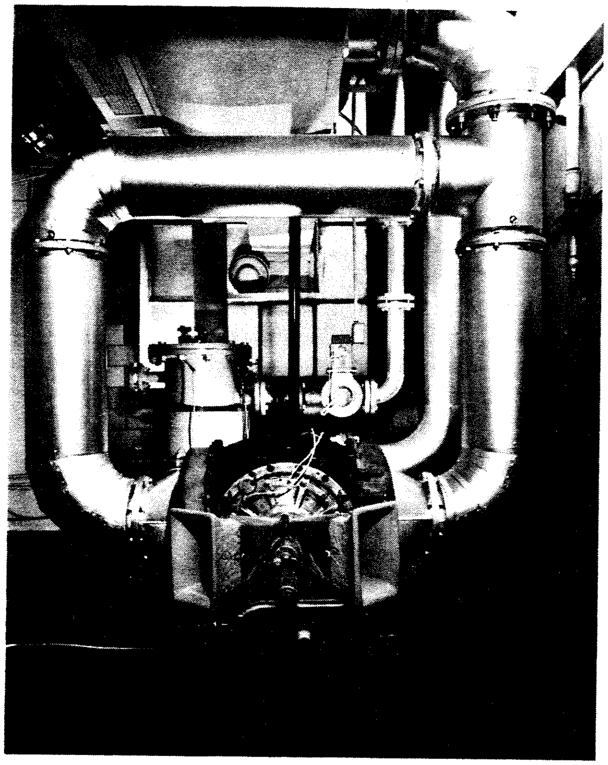

Fig. 17a - Engine Burner Test Fixture and Test Cell

The remaining evaluation parameters of carbon formation, exhaust smoke, and exhaust odor are inter-related, and are best observed using complete powerplant assemblies operating throughout the speed and load ranges. There is little air pollution with turbine engine exhaust because of the completeness of the combustion process. The CO_2 concentration varies from 0.8 to 1.9%, CO from 50 to 200 ppm, and the unburned hydrocarbon concentration is from 1 to 10 ppm.

Turbines

Our early turbine testing was described rather thoroughly in ASME paper 57-GTP-10 (Reference 2) which covered principally the techniques associated with a single stage turbine test. Its treatment of instrumentation and testing procedure is still an up-to-date description of our methods.

The turbine section, however, consists of a compressor turbine first-stage followed by a power turbine second-stage, and we now generally use a combined stage fixture in development tests (Fig. 18a and 18b). The reason for this is the high degree of interdependence of the second stage on the performance of the first. The discharge of the first stage is necessarily at a high velocity and a small radius for reasons of low wheel 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 for reasonable leaving loss. Consequently, the transition region between the two stages is difficult.

It is a region which, because of unavoidable diffusion, tends to build up a heavy boundary layer and results in large power turbine nozzle secondary flows. It is also a region which must have considerable length with consequently high mixing loss in order to avoid severe diffusion pressure gradients. Because of this, the first-stage discharge cannot be simply considered a pressure level available to a power turbine section. Rather it is a flow which, 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 (Reference 2). This relies on dynamic similarity between fix-

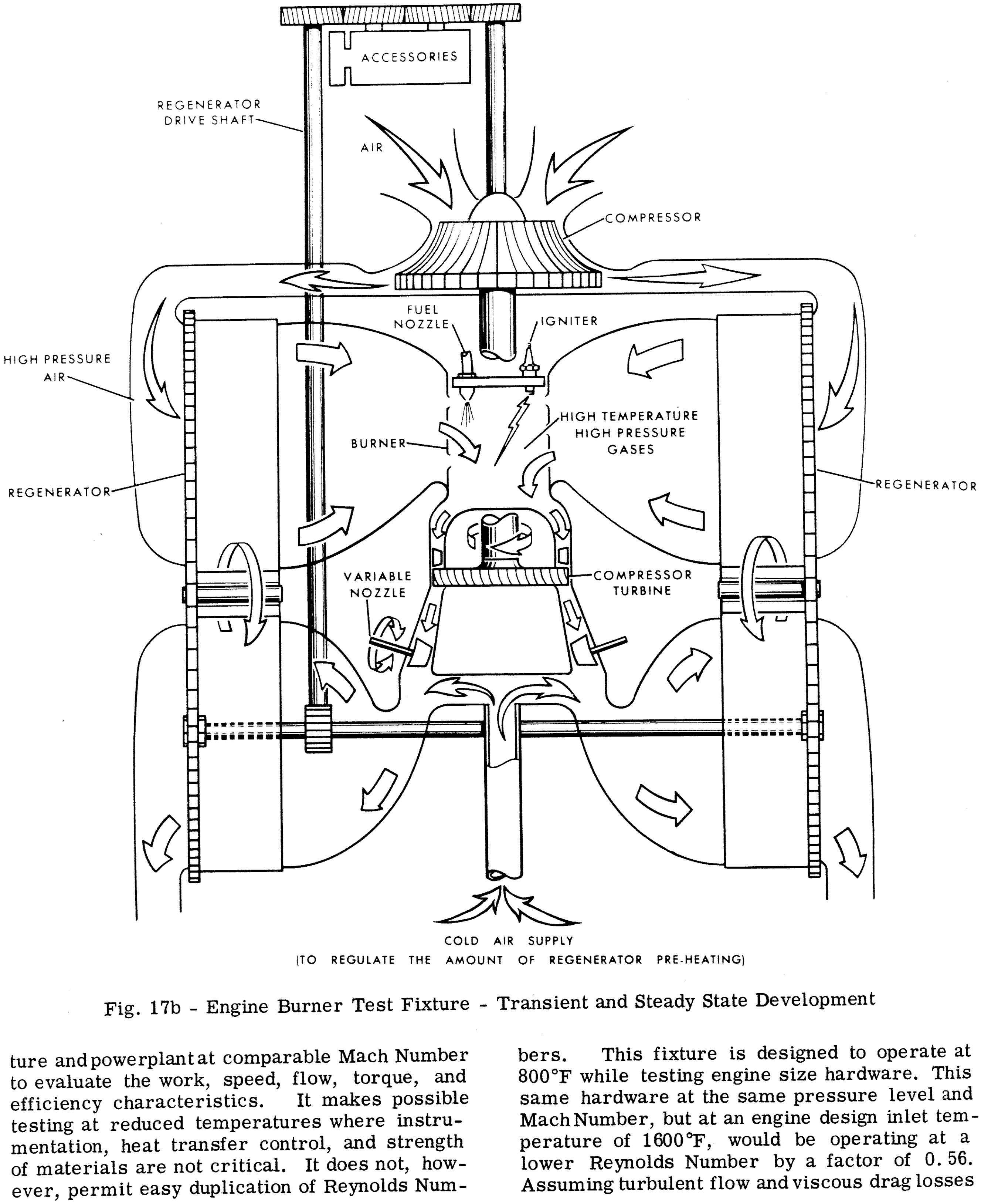

Fig. 17b - Engine Burner Test Fixture - Transient and Steady State Development

ture and powerplant at comparable Mach Number to evaluate the work, speed, flow, torque, and efficiency characteristics. It makes possible testing at reduced temperatures where instrumentation, heat transfer control, and strength of materials are not critical. It does not, however, permit easy duplication of Reynolds Numbers. This fixture is designed to operate at 800°F while testing engine size hardware. This same hardware at the same pressure level and Mach Number, but at an engine design inlet temperature of 1600°F, would be operating at a lower Reynolds Number by a factor of 0.56. Assuming turbulent flow and viscous drag losses to be inversely proportional to the Reynolds Number ratio to the one-fifth power, such losses should be lower by 12% in the fixture. However, mixing losses due to trailing edge thickness and secondary flows would be about the same in either situation. The overall efficiency resulting from the sum of all losses will be therefore only slightly different. This is given by way of justifying the fact that our test experience has indicated no significant efficiency difference at temperature levels from 400°F to 1200°F.

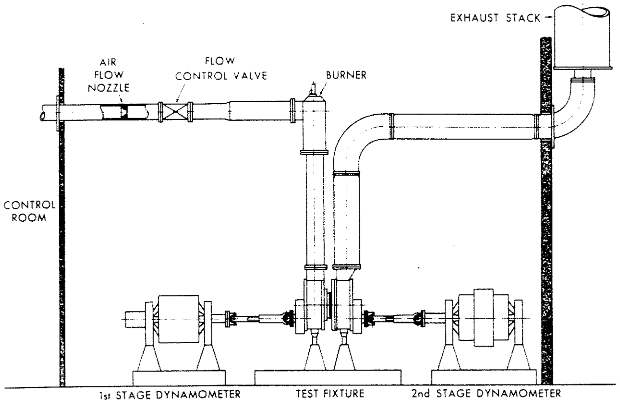

Fig. 18a - Combined Turbine State Test Fixture - Test Cell Arrangement

The measurements made in order to determine the parameter characteristics are weight flow, turbine wheel torques, total pressure at the turbine inlet and exit planes, and turbine inlet and outlet total temperature. In addition, wall static pressure measurements are made for continuity check calculations as well as for indications of diffusion.

Turbine torque is measured on the low speed dynamometers. The gear box and turbine wheel shaft bearing losses are calibrated and added. In addition, the measured torque is increased be a gear mesh loss factor which we have found to be about 1%.

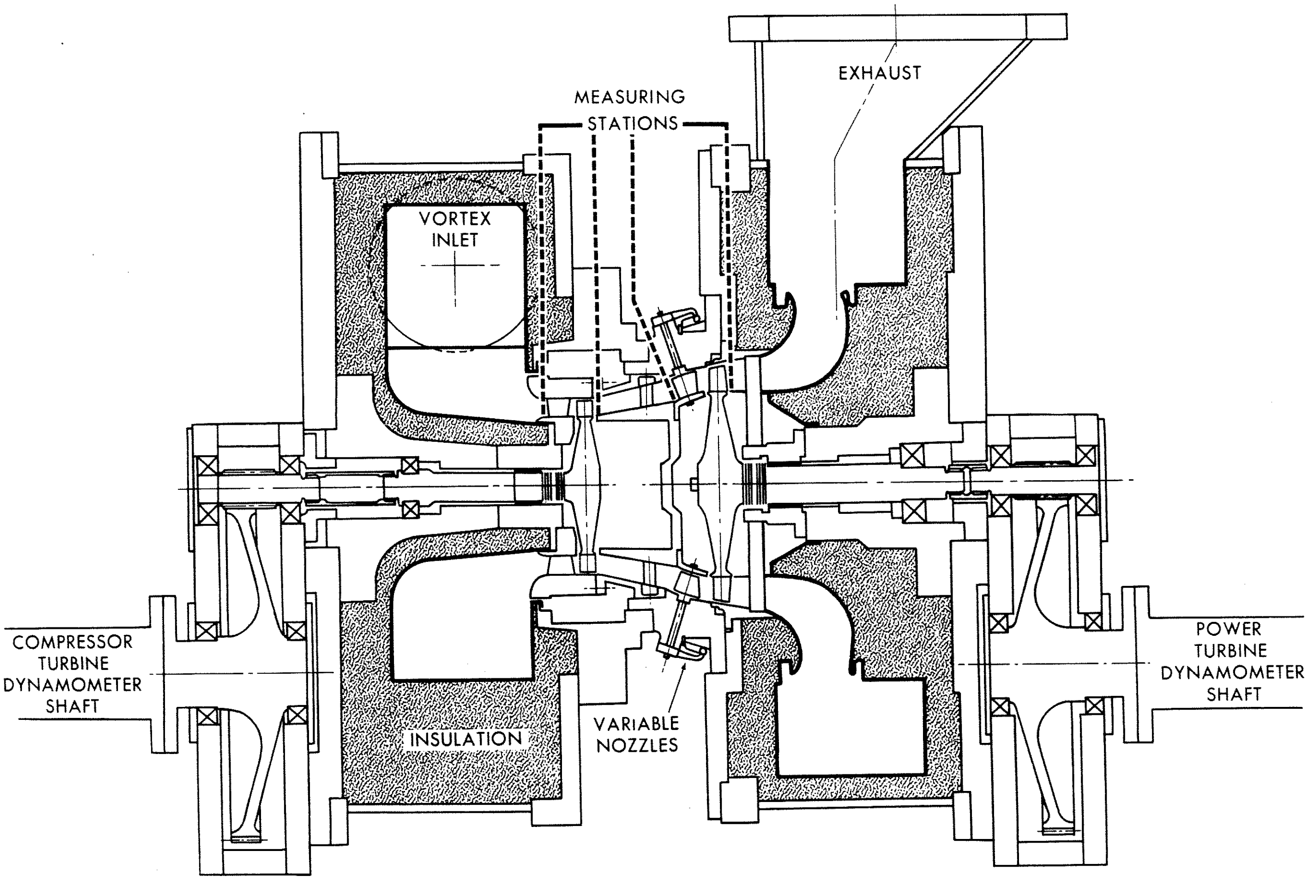

Fig. 18b - Combined Turbine Stage Test Fixture

An important feature of the fixtures is the thorough insulation behind the turbine inlet ducting walls which allows testing with uniform inlet temperature. To assure this further, the burner is set well ahead of the fixture and the connecting pipe is a well-insulated mixing tube. With such care, it has been found that three .06 inch diameter high-recovery thermocouple probes set at mid passage in the inlet annulus are sufficient for a good inlet temperature measurement. The probes are mounted in remote actuators for radial surveys and angular alignment checks.

Radial total pressure and flow angle surveys are made at the various stations of interest with one-hole cylindrical probes. These are mounted in manually-operated remote actuators and are made of .060 inch diameter tubing with a hole no larger than 20% of the diameter. Greater flexibility is obtained by using inter-changeable cylindrical probes for both temperature and pressure.

Determining flow angle with a one-hole probe can be done very accurately by tramming. This is somewhat slow, but our experience with three-hole probes indicates that, when they are of a size to be practical and reliable, they afford too much blockage in a 0.5 inch high annulus. Circumferential traversing has been avoided because the mechanism is generally too bulky to be used effectively on anything other than single element tests.

The procedure for running a single-stage turbine map was explained in Reference 2. There the weight flow, torque, work and efficiency parameters were examined at a constant speed parameter. In running data for a combined-stage turbine test, this concept is extended by setting a speed ratio between the power turbine and compressor turbine, in addition to the compressor turbine speed parameter. Under these conditions, overall pressure ratio is varied to cover the desired range. Then, since our power turbine has a variable stator, the procedure must be repeated for a range of angle settings. This outlines a rather extensive test program which establishes a fine basis for powerplant performance prediction. However, it makes any development program unreasonably long. Consequently, a test program consisting of a few judiciously selected points has been evolved for this latter use. For these, the overall pressure ratio is set at a powerplant point of interest (allowing for parameter testing corrections), the compressor turbine speed is fixed, and the variable power turbine nozzles are adjusted until the work of the compressor turbine is sufficient to run the compressor for the point chosen. The power turbine is then brought to a maximum power speed (a rather flat curve) and the data are recorded. Generally, this is repeated with the power turbine nozzles somewhat opened or closed for a variation of first-stage to second-stage pressure ratio split. The four equivalent engine conditions of idle, low speed, moderate speed, and maximum power have thus been covered to evaluate a turbine design.

The data analysis is formal, to a certain extent, in that it is loaded into a computer program for various efficiency and parameter calculations. Further analysis into wheel inlet conditions, relative Mach Number, and flow angles is then made with the assistance of a more involved program. This adds the requirement for input constants such as areas and blade row efficiency which are adjusted until there is agreement with the overall performance measurements. The probe surveys are also scrutinized for significant gradients and insight into three dimensional flow.

In a combined stage turbine test two other important flow elements are also analyzed. The first is the vortex scroll at the inlet to the first stage nozzles, for which both loss coefficient and discharge uniformity are checked. The instrumentation used for this is a single total pressure probe in the inlet pipe along with radial survey probes at the first-stage nozzles entry. The other flow element is the exhaust diffuser, with the second-stage wheel exit static taps and radial total pressure surveys at its inlet and the turbine section outlet plenum pressure probes at its discharge. These provide the necessary instrumentation for performance under actual wheel discharge conditions.

It is worth emphasizing that single-stage tests have been of considerable value and have allowed us to develop test methods and refine our aerodynamic design techniques. However, with confidence in our abilities in these areas, we find that a development program will be more effective if it proceeds directly to the combined stage concept.

MECHANICAL DEVELOPMENT

This general subject is meant to include all those components that convert an aero-thermodynamic concept into a mechanical machine. This will also include the mechanical development techniques for two of the aerodynamic components: compressor and turbines.

By necessity, this automotive powerplant must have accessories, shafts and supporting bearings, accessory gears and output reduction gears, and a housing to contain its components.

Housing - Engine housings, which have been designed and made of both sheet metal and castings, are submitted to hydrostatic proof testing before any initial first build. Standard techniques of stress coat and strain gages are applied in questionable regions under room temperature conditions and later checked in powerplants while measuring temperature gradients.

Accessories - The accessories of the Chrysler turbine powerplant are in general not too much different from those found on conventional reciprocating engines. Because of this, many of the development and endurance functions have been turned over to the general engineering Mechanical Laboratory in order to capitalize on their experience. The disadvantage of furnishing a high speed reduction drive ratio is somewhat offset by an operating speed range ratio of 2.5:1 instead of the 10:1 experienced with piston engines. Since all of our accessories are gas generator driven, extreme care must be exercised in minimizing polar moment of inertia and drive power requirements of both components and bearings.

The accessory and gearing system includes the following: fuel control, air pump, starter-generator, lubrication pump, and power turbine nozzle actuator.

A. Fuel Control

Many types of scheduling fuel controls have been tried during the course of years. Midway in the program a cooperative development agreement was reached with a reliable and experienced supplier of aircraft fuel controls - Bendix Corporation. Although the general automotive industry was not new to Bendix, this particular product application was. Aircraft quality, production methods, and subsequent costs were well understood, but totally unacceptable. Our long-range target cost -- that of a four-barrel carburetor -- accepted in good faith but with mixed feelings, is now considered feasible.

The requirements of the fuel control are quite basic:

(1) Allow the driver of an automobile to have complete power control of the vehicle at all times under all driving conditions through the use of an accelerator pedal that controls engine output.

(2) Safeguard the engine from destructive overtemperature and overspeed.

The vehicle and powerplant combination is essentially a transient machine. Thus, the control is a transient device except during rare periods of quiescent or steady state operation which occurs most frequently at idle and full speed. Bendix's first approach was to tailor a conventional aircraft control to our fuel requirements. Gas generator response from idle to full speed within 1.5 seconds far exceeded control response and produced undesirable instabilities. Concepts were readjusted many times and the final product of a mechanical-speed-governing, compressor-pressure-biased, acceleration-fuel-scheduling control, has proved reliable and is of automotive design philosophy. There are, however, a few inovations, similar to those found on carburetors, that have been added to the basic control to cope with the extremes of ambient conditions. One of them compensates acceleration fuel flow for ambient pressure. In addition, the presence of the regenerator presents a variable burner inlet temperature that depends upon load, amount of regeneration, transients, and whether or not the engine is "cold or hot". All first daily starts, for instance, are non-regenerative. A burner inlet temperature sensing probe, in conjunction with a variable area metering orifice, performs this compensating control function.

The powerplant is the only test bed that will fully evaluate control performance, but much preliminary work can be done on the fuel control test stand (Fig. 19). This is a steady state fixture and is primarily used for control calibration and endurance testing. It consists of a variable speed drive, a fuel supply and filter system similar to that of a vehicle, an air pump, and controlled air pressure regulators to simulate compressor discharge pressure and to provide a back pressure in the control outlet 5 psi above compressor pressure. Speed, fuel flow, air pressures, fuel pressure drop across the governor and metering orifices, governor slopes, and throttle angle are determined. Idle and maximum speed are set by running the proper control speeds and setting the idle and maximum speed stops until the governor droop curve intersects the fuel required to run. Maximum speed governing is maintained at ± 0.2%. Acceleration fuel flows are calibrated at steady state speeds with the throttle wide open and are checked against simulated temperature levels of the burner inlet temperature - variable area orifice probe. Our requirements of flame-out during deceleration requires zero fuel flow at closed throttle (except at idle governing, of course).

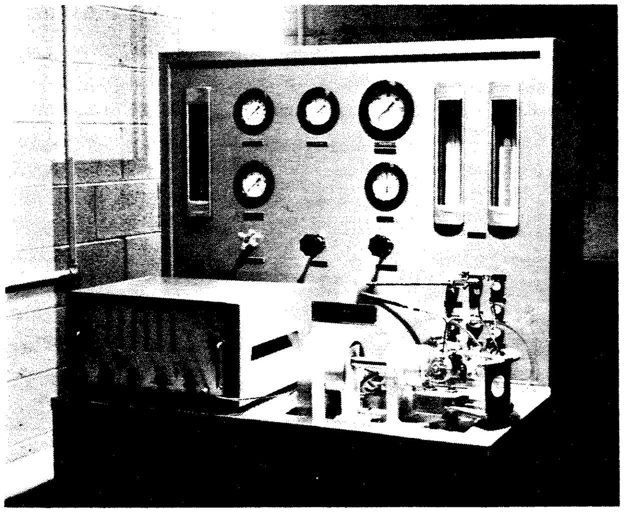

Fig. 19 - Fuel Control Test Stand

B. Air Pump

Since the fuel nozzle is classified as a low pressure air atomizing nozzle, the use of an air pump is indicated. A variety of air pump concepts have been tried, but the conditions of 500°F maximum air temperature in the absence of lubrication always lead us back to the reciprocating air pump, as sacrilegious as it may sound. The pump is a supercharger, accepting compressor air and boosting it 5 psi, since control outlet fuel and air pressure requirements are approximately 5 psi above compressor discharge pressure. Its size is indicated by starting requirements so that it must be regulated by dumping at speeds above idle. The major problem in its development were ring and cylinder material compatibility (unlubricated) and piston design for fatigue life and minimum reciprocating force. Air pumps are bench-tested for starting cycle performance and endurance.

C. Starter-Generator

The selection of a combined starter-generator unit was made on the basis of size, inertia, and engine arrangement. This unit, although adequate for our application, is of aircraft quality and is expensive. Development endurance and standards checking is now the responsibility of the general engineering Electrical Laboratory. The primary development problem has been brush life, the requirements of brush quality for long life and low resistance for starting being in direct opposition, as you would expect. Our experience with this unit has certainly readjusted our thinking for future units even though brush life is now in the 25,000-mile class.

D. Lubrication Pump

The lubrication pump is a modified production power steering pump with a few functions added. Oil from this pump at 100 to 1,200 psi is used for power steering, engine lubrication, transmission controls, servos and lubrication, and nozzle actuator motivation. Most of the development problems have been associated with quality control and noise.

E. Nozzle Actuator

The power turbine nozzle actuator is an engine control accessory and does what its name implies. Its four positions or ranges for idle, economy, maximum power, and braking are modulated as a function of engine load and car speed. The engine load is sensed by throttle position and car speed is sensed by transmission governor pressure. The "idle" position sets the nozzles for minimum fuel flow and creep torque; "economy" for maximum allowable turbine outlet temperature; "maximum power" for rated turbine inlet temperature; and "braking" for engine braking torque. Development problems have been primarily concerned with stability, reaction time in and out of braking, and flow requirements. The piston is sized for about 100 lb. of force at 100 psi and will swing the power turbine nozzles through approximately 90° of travel in less than 0.5 seconds. A bench fixture (Fig. 20) is used for calibration and is similar to any hydro-mechanical fixture equipped with pump, gages, flow meter, dial indicators and pressure transducers.

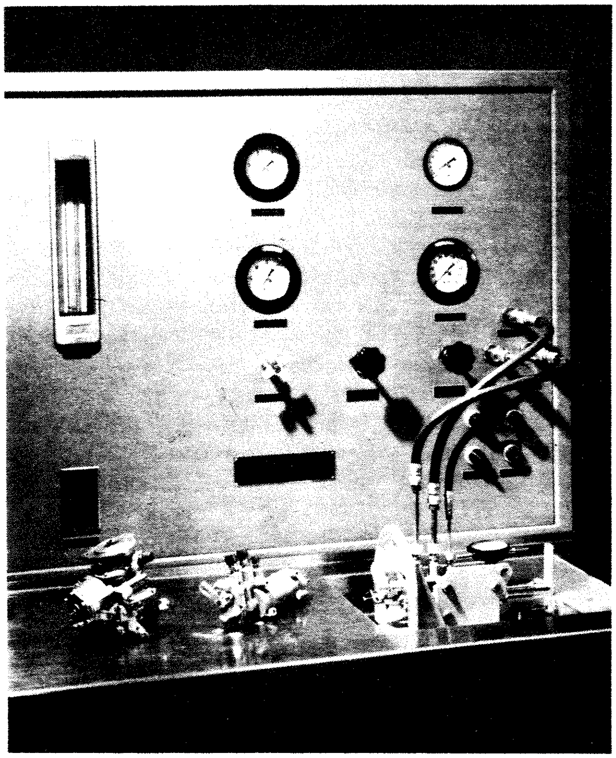

Fig. 20 - Nozzle Actuator Test Fixture

F. Accessory Drive

The remaining parts of the accessory system consist of gears, shafts, splines, shaft seals and primarily sleeve bearings of typical automotive design. A complete accessory test stand (Fig. 21) is available for overall performance and endurance testing, and this fixture is also concerned with reducing power requirements and with lubrication studies. Tests are run with various viscosity oils to simulate Type "A" transmission fluid viscosity down to -20°F and up to 300°F.

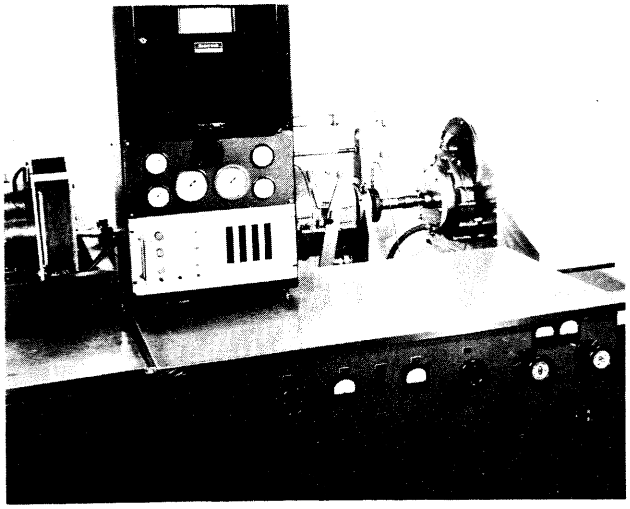

Fig. 21 - Complete Accessory Test Fixture

G. Reduction Gear

The final power reduction gear drive is a single-stage helical gear set. The only serious problem associated with these gears and their bearings has been one of noise which is attributed to manufacturing inaccuracies. The normal development procedures of involute modifications, crowning, lapping, etc., have been tedious but somewhat rewarding. A modified "four-square" fixture is used for checking capacity and efficiency of a gear set. Two boxes are mounted back-to-back with their high speed shafts coupled. A measured torque is locked in low speed shafts. The external power required to run the assembly is a measure of the losses.

H. Gas Generator

The gas generator shaft and bearing system has always been a challenge to the student of vibration and stability. Two distributed masses of different ratio of polar to diametral moments in inertia that rotate at high speed under light bearing loads is a fairly complex system. As soon as an engine design is conceived, a model of the gas generator housing, shaft, shaft seals and bearing arrangement is made with dummy compressor and turbine wheels of the proper mass and moment of inertia. This shaft system is driven by a cold air turbine and the nozzle for this turbine is cradled in hydrostatic air bearings to measure drive torque for determining windage and bearing losses. Engine gas pressures are simulated where necessary and provisions are made for applying thrust loads pneumatically. Oil pressure and temperature, seal pressure, speed, radial and thrust loads, and bearing clearance are the major parameters controlled for performance and stability studies. Instrumentation consists of gages and transducers for pressure, thermocouples for temperature, reluctance pick-ups for speed and along the shaft for deflection, frequency and phase angle measurements. Radial load is induced by controlled amounts of unbalance. Static shaft run-out at each pick-up location is held to within .0005 inch and two flats of .002 inch and .005 inch depth are machined at each location to serve as timing marks and for calibration. Pick-up output is displayed on an oscilloscope (and recorded for posterity with a camera).

The final gas generator shaft and bearing arrangement is pretty much as originally designed except for a few subtleties. Pilot sizes, fits and clamping loads were adequate, but the original shaft system exhibited three undesirable characteristics: (1) bearing losses were higher than desired, (2) a critical speed with relatively high deflection amplitudes existed in the midspeed range, (3) high shaft speed oil film whirl could be induced at any speed as a function of oil temperature and bearing clearance. These problems were all interdependent and had to be dealt with simultaneously. Bearing capacity was more than adequate (except under conditions of half speed whirl), but L/D bearing ratios were reduced, bearing clearance adjusted, and thrust washer size reduced to minimize bearing losses. These changes also helped half speed whirl, but this whirl was not completely eliminated until the development of tapered land journal bearings. The critical speed, high deflection problem was a little more difficult to cope with. Increasing shaft size or stiffness to raise the natural frequency out of the operating range did not appear attractive for reasons of assembly and bearing size. Damping, and the bearing supports themselves, offered some interesting avenues of approach. The oil film spring rates of journal bearings were relatively high, but, if these rates could be changed and damping acquired at these points, a change in system performance could be anticipated. This was the approach that was pursued. The final solution was a low spring rate support for the compressor bearing with damping, all confined within 0.4 inch of axial length and not too clearly shown in the engine cross section, we hope. The critical spot is essentially at the same speed, but maximum deflections are less than .002 inch T. I. R.

The mechanical performance of the compressor and turbine wheels requires continual monitoring since design modification for efficiency and inertia improvements are never-ending. In addition, accumulation of powerplant and vehicle endurance hours keeps bringing up little reminders. This is a universal problem, and our techniques are the same as those of anyone else who is serious about gas turbines. Material properties and casting quality (in our case) must be controlled. A room temperature spin pit (Fig. 22) that is evacuated to less than 50 microns absolute pressure is available for prototype spin testing. The part is suspended from a single bearing on a thin wire and can be driven to over 100,000 rpm by a cold air turbine. The type of test is reasonably informative for compressor impellers as far as centrifugal stress is concerned, but is of little value for turbine wheel investigation because of the absence of temperature level and gradient.

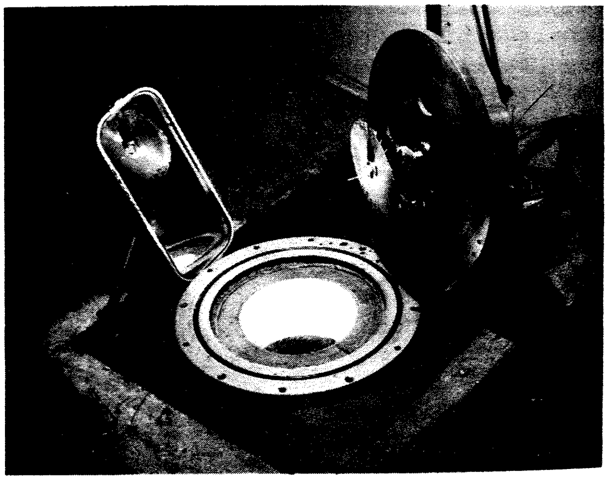

Fig. 22 - Spin Test Pit

Component fixtures orpowerplants are used for dynamic temperature and strain measurements. Slip rings were developed to measure the temperature gradients in compressor and turbine wheels, and slip ring development took many months of fixture time. Brush and ring wear, contact resistance, and extraneous noise generation presented the biggest problems, but these problems are not as serious in temperature measurements as they are in strain measurements. The temperature gradients in both compressor and turbine wheels have been determined and have dictated design modifications.

Dynamic strain measurements present a tougher and more tedious problem and are concerned primarily with blade elements and their vibrational modes. This general approach is standard in the turbine field, with variations, and will be discussed briefly. Bench tests are first initiated to determine modes of vibration, nodes, major frequencies, damping, and magnitude and distribution of alternating stress, which depend upon the magnitude of the unknown forcing function. We have had the most success with a sonic white noise generator for an exciter and a wave analyzer for interpretation. Polyvinyl chloride is used to obtain dust patterns of nodal lines, and strain gages are employed for stress determination. Enough energy can be supplied with an air jet to fatigue blades and to get a feel for allowable alternating stress level.

With this bench test background, high temperature strain gages are attached to the test part, and are installed in an engine or fixture with slip rings and run through speed and temperature ranges. Specifically in the case of our power wheel, the turbine engine is a white noise generator. A whole range of alternating stress frequencies, most of insignificant magnitude, has been measured. Major stresses occur at the first order bending and torsion modes of the blade, and are induced by the reduction gear and by various aerodynamic forces. Reduction gear quality improvement and isolation from the power turbine shaft can be beneficial. Aerodynamic forces are primarily the result of power turbine nozzles and support struts. A literature search indicates that random spacing can help but is no cure-all. Since it is virtually impossible to move this system out of the operating range, modifications to the blade to reduce the vibrating stress level and live with the problem maybe the only practical solution.

POWERPLANT DEVELOPMENT

The powerplant is available to the laboratory long before component development is completed. This is not a paradox since component testing never ceases and the effect of the engine arrangement on both aerodynamic and mechanical interaction cannot be determined fully from fixture tests alone. Powerplants are continuously updated during the development phase to incorporate the latest modifications aimed at improving individual components.

The success of any such venture must begin with an organized system of assembly, disassembly, and record keeping. Unless we have knowledge of all the parts involved, the conditions under which they were tested, and what happened to them during operation, an evaluation of engine behavior is virtually impossible. Much of the learning depends upon a statistical accumulation of data. All pertinent dimensions of fits, runouts, and clearances necessary to determine wear, distortion, or creep of critical parts are recorded for all components and subassemblies. These include such things as bearing clearances, turbine blade and impeller shroud tip clearances, assembly torques, gasket compression, unbalance, antifriction bearing clearances, and gear back lash. Turbine wheel hubs,

Fig. 23 - Powerplant Test Cell Control Console

for instance, are carefully indented, microscopically, at intervals across two diameters. The dimensions between indentations are measured optically to within 0.0001 inch and are recorded periodically during the life of the wheel. These data furnish short-time plastic deformation and long-time creep information.

The eight dynamometer test cells used in development of our turbine are similar in arrangement (Fig. 23 and 24). The test area is separated from the control area for reasons of noise isolation and personnel safety. Normally, three of the cells are used for 24-hour endurand, 6 days a week. One or two are used for vehicle powerplant preparation and checkout, and the remainder are used for general or specific performance development.

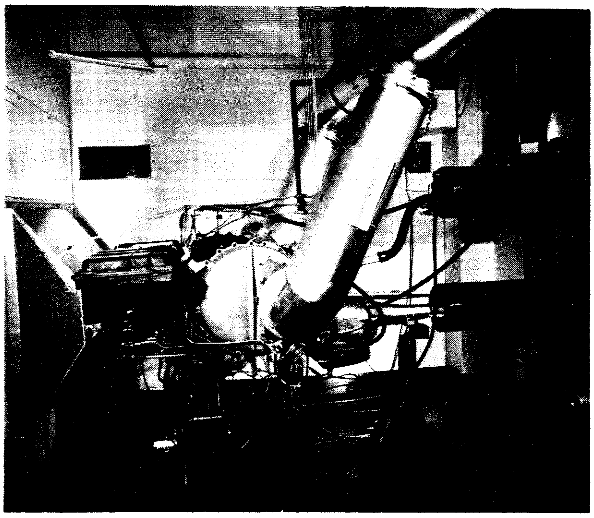

Fig. 24 - Powerplant Test Cell

A. Endurance

The object of endurance testing is quite obvious. The overall powerplant is subjected to a specific cycle of operation for periods of anywhere from 100 hours to 300 hours between inspections. Failed or malfunctioning parts are modified and replaced, if necessary, and endurance continued to an arbitrary 2,000-hour goal on all parts. The selection of an endurance cycle has been the subject of conflicting opinions for many years, but this problem is no different from any other automotive laboratory simulated endurance test. Everyone wants to duplicate customer vehicle experience or develop some meaningful accelerated life test. Our concept of a dynamometer endurance cycle is the result of years and miles of experience at the Chrysler Proving Grounds and on the highways of this country. Steady state operation in a vehicle is the exception rather than the rule and we favor a large percentage of transients or acceleration time as being more representative and certainly most severe. A typical endurance cycle is outlined in the Appendix.

B. Performance

Performance testing covers a multitude of general and specific development techniques. Instrumentation ranges from relatively standard equipment for general powerplant tests to sophisticated devices in particular component tests. Each test point may require recording as few as 50 individual parameters for general testing or may exceed 100 pieces of data for a specific test. All engine tests are run with transmission since we consider it part of the powerplant package and since the oil pan acts as the lubrication sump. Output data are referred to the prop shaft flange.

Overall performance tests represent the end products of horsepower, SFC, air flow, and engine response. Measurement of these parameters is straightforward. In addition, state properties of temperature and pressure are measured before and after each aerodynamic component in an attempt to obtain thermodynamic correlation. The selection of type and number of sensing elements and their position in the gas stream is the result of experience, as is the interpretation of the data. The one parameter that is difficult to determine is internal leakage, i. e., gas that by-passes the turbine stages. Nevertheless, we feel that a comparison between measured compressor weight flow and the computed first-stage turbine flow parameter, as obtained from measured total temperature and pressure ahead of the stage, gives a sensible comparative measure of cycle leakage, if not an absolute value. Typical performance curves for the current Chrysler gas turbine are shown in Figures 25a through 25f.

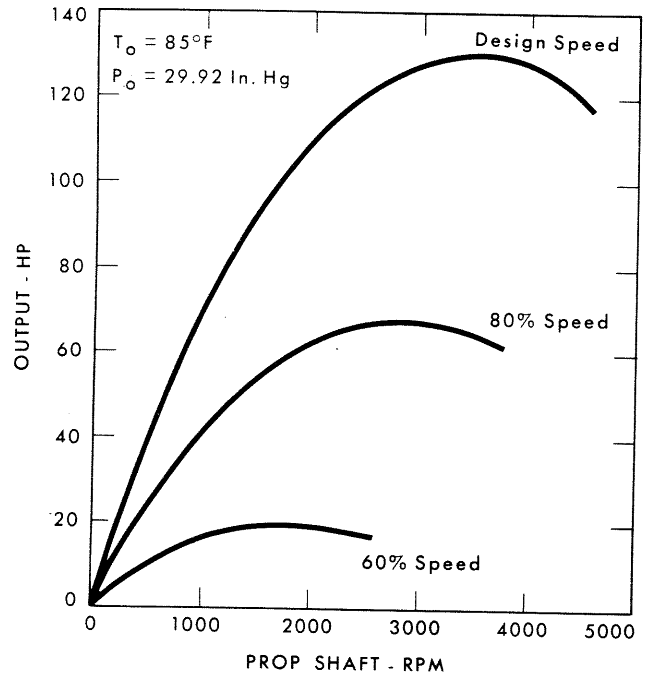

Fig. 25a - Engine Horsepower -vs- Prop Shaft rpm at Various Gas Generator Speeds

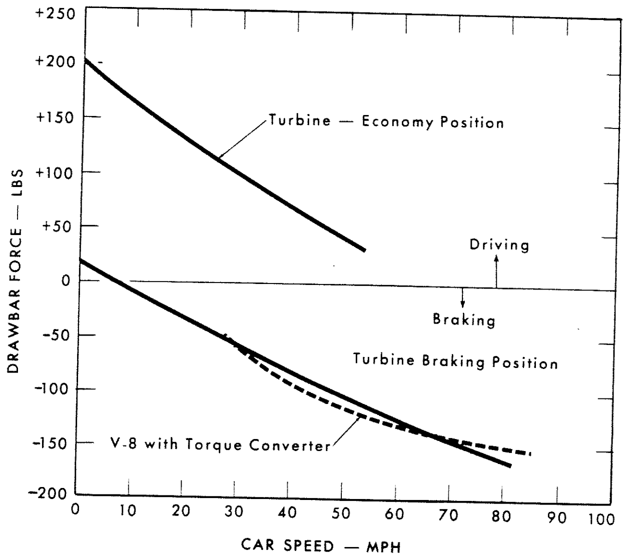

Fig. 25c - Effect of Power Turbine Nozzle Position at 50% Generator Speed

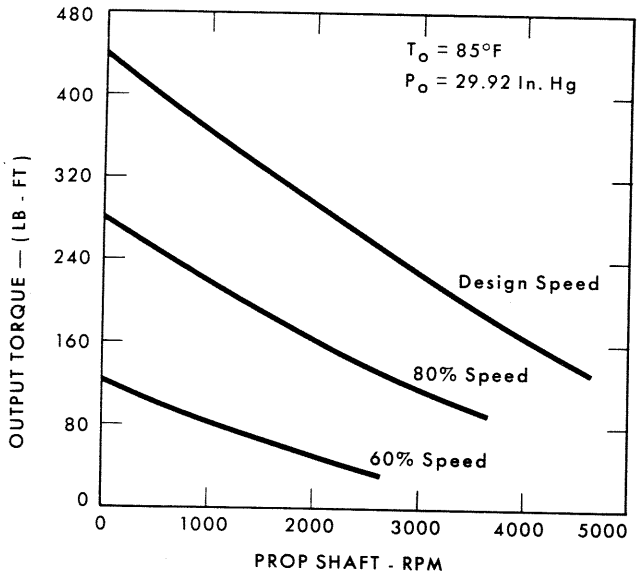

Fig. 25b - Engine Torque -vs- Prop Shaft rpm at Various Gas Generator Speeds

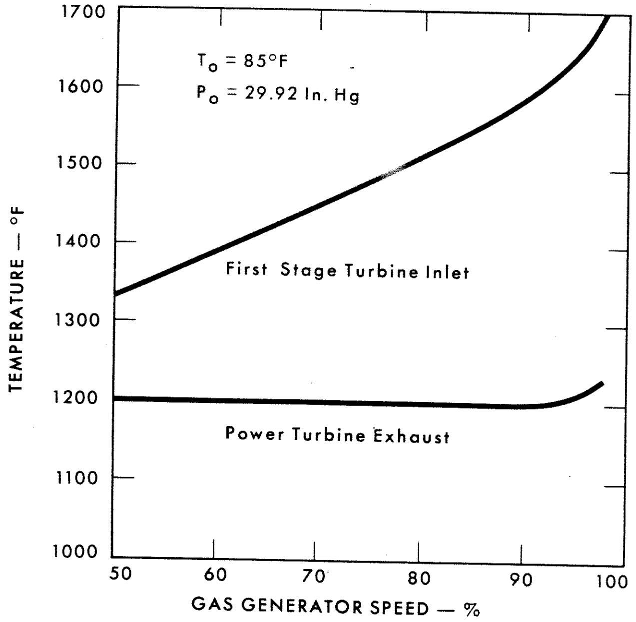

Fig. 25d - Turbine Temperatures -vs- Gas Generator Speeds

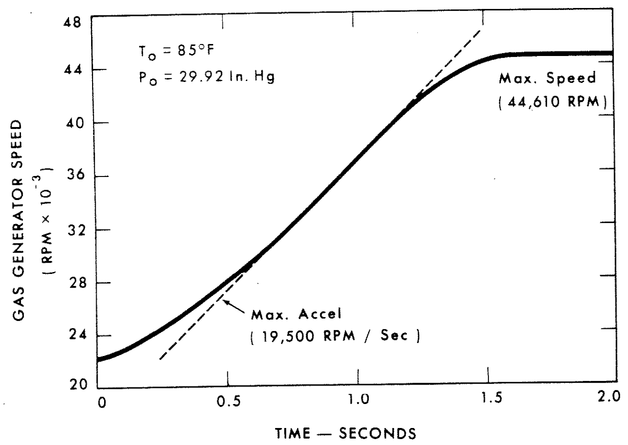

Fig. 25e - Gas Generator Response

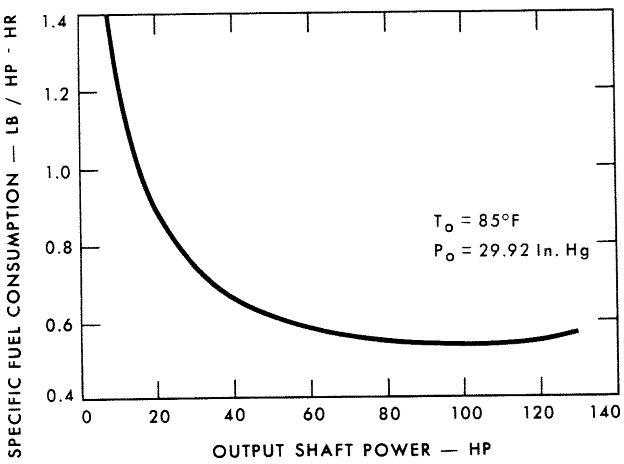

Fig. 25f - Specific Fuel Consumption -vs- Engine Output Power

Performance tests for a specific purpose may have a variety of forms, each with a particular objective. Component modifications or improvements, as determined from fixture tests, are incorporated into a powerplant for evaluation of mechanical, aerodynamic or transient behavior. Every aerodynamic component has at one time or another been completely surveyed in a powerplant. The instrumentation and techniques employed are analogous to those used in fixture tests, and the engine results obtained are compared with those fixture tests. Significant differences form the basis for additional investigations.

The principal factors that influence component performance deviations in a powerplant and that are difficult to simulate in a fixture are: heat transfer, gas temperature distribution, elastic thermal distortion, transients, internal leakage, and the effects of interaction.

The structural and insulating portions of powerplants and associated lubrication system have been completely swamped with thermocouples to determine temperature gradients, heat flow paths, and operating clearances. These temperatures are recorded during starting transient, steady state, operating transients, and shut-down soak. These measurements are well worthwhile, as indicated by the following development changes:

(1) Heat transfer to the compressor discharge air before reaching the regenerators had a detrimental effect on SFC which required redesign of flow paths and insulating liners.

(2) The construction of turbine wheel shrouds was modified to balance strength requirements against transient blade tip clearances.

Compressor turbine bearing soak-back temperatures presented a difficult problem when rapidly shutting an engine down from full load under conditions of high ambient temperature. Bearing soak-back temperature approached 600°F under these conditions. This temperature level caused three things to occur: (1) incipient lead sweat from the bearing matrix, (2) accumulative varnish formation on the journal and (3) panic. The first two resulted in a reduction of bearing clearance that eventually led to failure. The third produced a rapid modification of heat flow paths into and out of the bearing by revising the major conduction areas.

The engine operating conditions under which all components, aerodynamic and mechanical, must function positively as a family are the transients. Gas generator acceleration time on the order of 1.5 seconds makes fixture transient studies impossible. This problem reduces to one of a complex study of fuel control and nozzle actuator behavior, compressor surge, and burner combustion.

The fuel control must schedule fuel as a function of ambient conditions, speed, and burner inlet temperature in such a manner as to provide the maximum allowable acceleration turbine inlet temperature. The nozzle actuator must set the power turbine nozzles in their optimum position. All final control development and stability studies are performed on a powerplant.

Burner response to acceleration demands had to be determined in a powerplant. Spark plug location and fuel nozzle performance contribute to quality of combustion, flame length, and liner transient temperature. The two special fixtures described in the "Burner" sections are essentially modified powerplants.

The Chrysler turbine of 130 hp has a comparatively low compressor turbine output potential at idle, and parasitic loads due to accessories, bearing losses, oil drag, etc. impose a rather severe requirement on this element. In addition, these same loads hinder gas generator acceleration and part load SFC. In the initial development phase of this powerplant, inadequate drainage from the accessory box accounted for a 10% loss in maximum output with proportionate penalties at part load. This deficiency was not obvious except for abnormal oil heat rejection. An engine was modified for starting with cold air jets on the impeller blades. The engine accessories were removed and these functions were supplied from room facilities, and the engine lubrication drain system was separated for observation. Accessory and drain components were added a piece at a time and modifications were made until a satisfactory system was obtained. This procedure produced a vivid, eye-opening demonstration of the effects of parasitic loads on output and SFC.

Cold starting tests are conducted in the cold room facilities where powerplants and vehicles are soaked for 24 hours at -20°F. The primary problems at this temperature are battery capacity, starter output, engine cranking requirements, fuel viscosity, and fuel quality. Although battery capacity at -20° is recognized to be marginal, starts at -20° are made in about 15 seconds to idle, combustion occurring within the first 1 to 3 seconds. Engine cranking requirements depend upon oil viscosity (Type A) and, again, accessory and bearing power required. Care must be exercised with the quality of fuel used. Commercial grade Diesel No. 2 fuels, for example, have a wide range of pour points and cloud points, and while pour point depressants have been used with limited success, they do not alter the cloud point.

In summary it can be stated that, even though the turbine lends itself to detailed component studies, the powerplant, in part or in its entirety, is still the final proving board.

VEHICLE TESTING

Two of the Chrysler gas turbine prototype cars have been assigned to the Research group, and these are used exclusively for studies peculiar to a turbine installation. Many problems have already been evaluated and ironed out in the dynamometer development phase but, like any component, they must be re-evaluated in a vehicle. Again, transient behavior studies absorb the majority of test time, since no amount of engineering recordings will ever replace that "seat-of-the-pants" feel. The concepts of throttle position versus engine speed and transmission shift points, for instance, were developed from vehicle tests. Similarly, intake filtration requirements and location had to be determined from car experience.

One of the most noticeable differences between turbine and piston-powered vehicles is in the character of engine-produced noise. The gas turbine and reciprocating engines are both excellent white noise generators; the former favoring the mid and high frequency range and the latter the low and mid frequency range of the audio spectrum. Typical piston and turbine engine vehicle installations, with appropriate sound treatment have produced essentially equivalent total sound energy. Unfortunately, the human ear is more susceptible to frequency than it is to total energy. Since the reciprocating engine has been the standard for so many years, the gas turbine presents a different sound to the public, but not one that has to be objectionable.

We choose to consider turbine noise sources as belonging to any one of four categories: (1) intake, (2) accessories and gears, (3) engine housing and (4) 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 non-resonating reinforced plastic, lined with sound absorbing material, and are devious enough to "hide" the eye of the impeller. Though velocities are low and pressure losses minimized, the transition section from the ducts to the compressor inlet must be handled with extreme care in order to present the proper velocity distribution to the impeller. For example, when an early interim inlet duct (Fig. 26) adversely affected compressor surge conditions during transients, the inlet was adapted to the suction side of the laboratory air supply and a simple demonstration with tufts or threads indicated gross separation at the compressor inlet transition radius. Fairing this radius with clay (Fig. 27) and further traversing with a total pressure and flow angle probe, as well as observing the circumferential wall static pressure distribution, produced a satisfactory transition inlet.

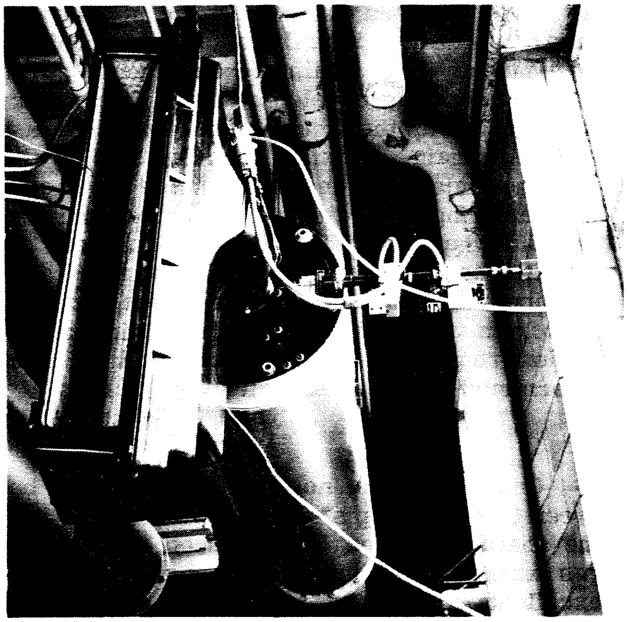

Fig. 26 - Air Intake and Filter System -- Original Model

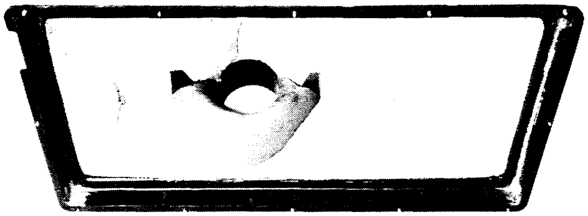

Fig. 27 - Air Intake and Filter System -- Transition Modification

Our experience to date indicates that the accessory system presents the toughest noise suppression problem. The major accessory components responsible for objectionable noise level and frequency are the starter-generator, the air pump, and the drive gears. Particular attention was given to armature unbalance, antifriction bearing selection, and cooling fan design of the gas generator. The air pump reciprocating forces were minimized and the tortuous pump porting was faired experimentally to reduce air noise. Quality control was exercised to obtain good dynamic gear tooth action as required by high gear pitch line velocities and associated speed reduction. Finally, the use of plastic gears proved extremely beneficial in reducing gear noise.

The engine housing by itself makes no noise, but rather serves as a transmitter. Sources of noise are: (1) internal velocity-induced air noise that is transmitted "through" the housing, (2) air forces that resonate portions of the structure, and (3) forcing functions from rotor unbalance and gearing that cause structure resonance and housing displacement on its mounts. Ideally, a dense structure with inherent damping would adequately handle the first two problems, but weight and cost cannot be compromised. The cast structure of this engine, however, does lean in this direction, with large flat areas avoided to prevent structure resonance. The primary effect of gas generator rotor unbalance is not one of bearing capacity and life, but is rather concerned with noise. First order shaft speed -- 300 to 750 cps -- has the potential of producing a very pure, objectional tone. The magnitude of unbalance and resulting forcing period, along with the mass of the engine and spring rate of its mounts, can be considered analogous to a loud speaker. The rotor bearing and shaft system previously described, though stable and damped, still requires dynamic balancing to within 0.010 oz.-in. at each bearing to minimize this speaker effect.

General exhaust system noise with this vehicle had been objectionable, but the problems were quite simple and solutions or corrections were quickly applied. Duct sizes were made adequate to assure low velocities. However, pipe inlet and outlet conditions, dictated by chassis clearance requirements, were conducive to flow separation. These separations produced noise-generating eddys and local high velocities that resulted in the pipe flowing only partially full. The pipe discharge flow induced abnormal and objectionable dust cloud formation, especially on dirt roads. Transition radii were modified and turning vanes added to the pipe where necessary. The ducts were also stiffened to eliminate "oil-canning" and some sections were treated accoustically.

SUMMARY

The development techniques that have been used in testing and refining the Chrysler turbine are extremely broad in spectrum. In many cases, the description of those methods has been generalized since further detail would only have been applicable to this powerplant. To those in the field of gas turbines, the reporting of all of these techniques may seem elementary and repetitious; however, in a field as new as the small gas turbine, there is no better way than to rely on a tedious and detailed development program. Such a comprehensive program, which in the beginning may seem prohibitively long, is -- in the final analysis -- the shortest path.

APPENDIX

Endurance Cycle (One half hour test time)

1. Start and idle for 3 minutes.
2. Accelerate to 80% speed. Run steady state for 2 minutes.
3. Return to idle and accelerate to 90% speed. Run steady state for 5 minutes.
4. Return to idle and accelerate to 95% speed. Run steady state for 5 minutes.
5. Return to idle and accelerate up to design speed. Run steady state for 5 minutes.
6. Return to idle and run acceleration tests up to design speed for 3 minutes. (Idle, accelerate - idle, accelerate... etc. - 6 cycles).
7. Run idle for 2 minutes.
8. Shutdown for 5 minutes. Allow turbine inlet temperature to cool below 600°F before starting.

Prop Shaft Speeds (rpm)	
80%	2200 - 2600
90%	2700 - 3200
95%	3300 - 3700
Design	3800 - 4500

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