GE Power Generation

GE Gas Turbine Design Philosophy

D.E. Brandt
R.R. Wesorick
GE Industrial & Power Systems
Schenectady, NY

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D.E. Brandt, R.R. Wesorick
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INTRODUCTION

Several important design philosophies have enabled the GE family of heavy-duty gas turbines to achieve worldwide market leadership. These design philosophies have been important in achieving continuous advances in the state-of-the-art gas turbine technology, and they will continue to guide technological developments. This paper will review the significance of certain GE design philosophies and development objectives for the flange-to-flange gas turbine.

The major elements of this philosophy are the evolution of designs, use of geometric scaling, and thorough preproduction development. The evolution of designs has been highly successful, and this approach will continue to be the basis for further progress. One result of the evolutionary approach is a family of axial-flow compressors whose flow, pressure ratio, and efficiency have been improved in several discrete steps, while retaining the proven reliability of existing designs. The historical development of these compressors will be described. Another result of the evolutionary approach is the MS7001 turbine. It has been improved in performance through six models, the A, B, C, E, EA, F and FA.

A second, highly successful principle of GE's product line has been geometric scaling of both compressors and turbines. Scaling is based on the principle that one can reduce or increase the physical size of a machine while simultaneously increasing or decreasing rotational speed to produce an aerodynamically and mechanically similar line of compressors and turbines. Application of scaling has allowed the development of the product line by the use of proven compressor and turbine designs. Machines such as the MS1002, MS5001, MS6001, and MS9001 were designed utilizing scaling which maintained geometric similarity with counterpart components in MS3002 and MS7001 units. This results in constant temperatures, pressures, blade angles, and stresses. Additionally, important cycle parameters are maintained, such as pressure ratio and efficiency. If the scale factor is defined as the ratio of the diameters, then shaft speed varies as the inverse of that ratio. Linear dimensions vary directly as the scale factor; the airflow and power vary with the square of the scale factor; and the weight varies with the cube of the scale factor (Table 1). The vibratory frequencies of the blading, relative to rotational speed and centrifugal stress levels, are the same for all scaled compressors and turbines. Thus, the application of scaling allows maximum utilization of available experience.

Table 1
SCALING RATIOS

Scale Factor	0.5	1	2
Pressure Ratio	1	1	1
Efficiency	1	1	1
RPM	2	1	0.5
Velocities	1	1	1
Flow	0.25	1	4
Power	0.25	1	4
Weight	0.125	1	8
Stresses	1	1	1
Freq/ROM	1	1	1
Tip Speed	1	1	1

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A third element of the GE design philosophy is thorough development. This involves design analysis, quality manufacturing, testing, and feedback from field experience. This philosophy is evidenced by GE's substantial investment in development and test facilities.

There are several other important considerations which have produced the combination of construction features found in GE-designed heavy-duty gas turbines. For example, the use of relatively common materials such as grey iron and nodular iron in the casings, and low-alloy steel compressor and turbine wheels, allows fabrication in many locales using foundry and forging technology common to several equipment industries.

The subjects of fuel flexibility, packaging, and maintenance are also important design considerations and are discussed in other papers. This paper will focus on the development philosophy of the three major gas turbine elements: the compressor, the combustor, and the turbine.

GAS TURBINE DESCRIPTION

The Gas Turbine Cycle

The gas turbine cycle is a constant flow cycle with a constant addition of heat energy. It is commonly referred to as the Brayton Cycle after George Brayton. Figure 1 illustrates this cycle as it is plotted on temperature entropy coordinates. The constant pressure lines diverge with increasing temperature and entropy. This divergence of the constant pressure lines make the simple cycle gas turbine possible. For all common gas turbines in use today, the lower pressure represents atmospheric pressure, and the upper pressure represents the pressure after compression of the air. Air is compressed from state 1 to state 2 in an axial flow compressor, while heat is added between states 2 and 3 in a combustor. Work is then derived from the expansion of the hot combustion gases from states 3 to 4. Since the expansion from states 3 to 4 yield more work than that required to compress the air from states 1 to 2, useful work is produced to drive a load such as a generator.

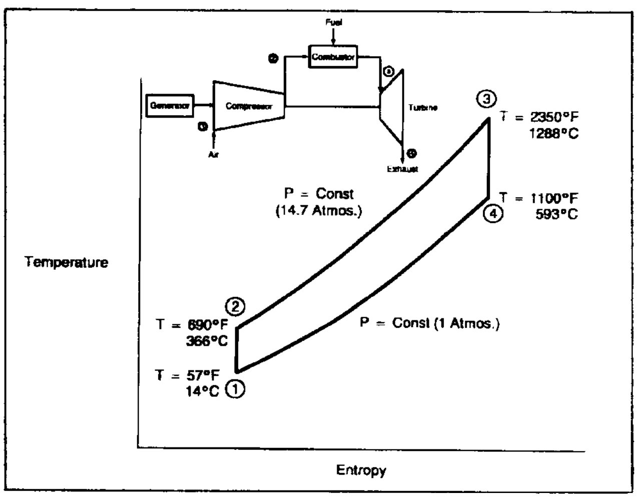

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Figure 1. Ideal Brayton Cycle

Figure 1 illustrates the common open cycle gas turbine which is nearly universal for power generation, mechanical drive, and aircraft applications. Other cycles such as reheat cycles and pumped storage cycles represent variations on that illustrated in Fig. 1.

Gas Turbine Configuration

Figure 2 illustrates an MS7001FA gas turbine. It is typical of all gas turbines in commercial operation today. Gas turbines with multiple shafts, such as the heavy duty MS3002 and MS5002, and aero-derivative gas turbines, are modifications of the configurations shown in Fig. 2. While these modifications require considerable design and mechanical innovation, the basic description of the gas turbine remains unchanged.

In the compressor section, air is compressed to many atmospheres pressure by the means of a multiple-stage axial flow compressor. The compressor design requires highly sophisticated aerodynamics so that the work required to compress the air is held to an absolute minimum in order to maximize work generated in the turbine. Of particular interest in the design of any compressor is its ability to manage stall of its aerodynamic components. In starting the gas turbine, the compressor must operate from zero speed to full speed. It is essential that the varying air flow within the compressor be so controlled that damage does not occur from avoidable stalling during part speed operation, and that stalling is absolutely prevented at full speed. During low speed operation, the inlet guide vanes are closed to limit the amount of air flowing through the compressor, and provisions for bleeding air from the compressor are provided at one or more stages. This reduces the strength of the stalling phenomena during part speed operation, which avoids compressor damage. The compressor aerodynamics are such that at full speed operation, no stalling should occur. Because sufficient margin exists between normal operating conditions and those conditions which would result in stall, General Electric gas turbines do not experience stall phenomena during normal full speed operation.

The combustor of a gas turbine is the device that accepts both highly compressed air from the compressor and fuel from a fuel supply so

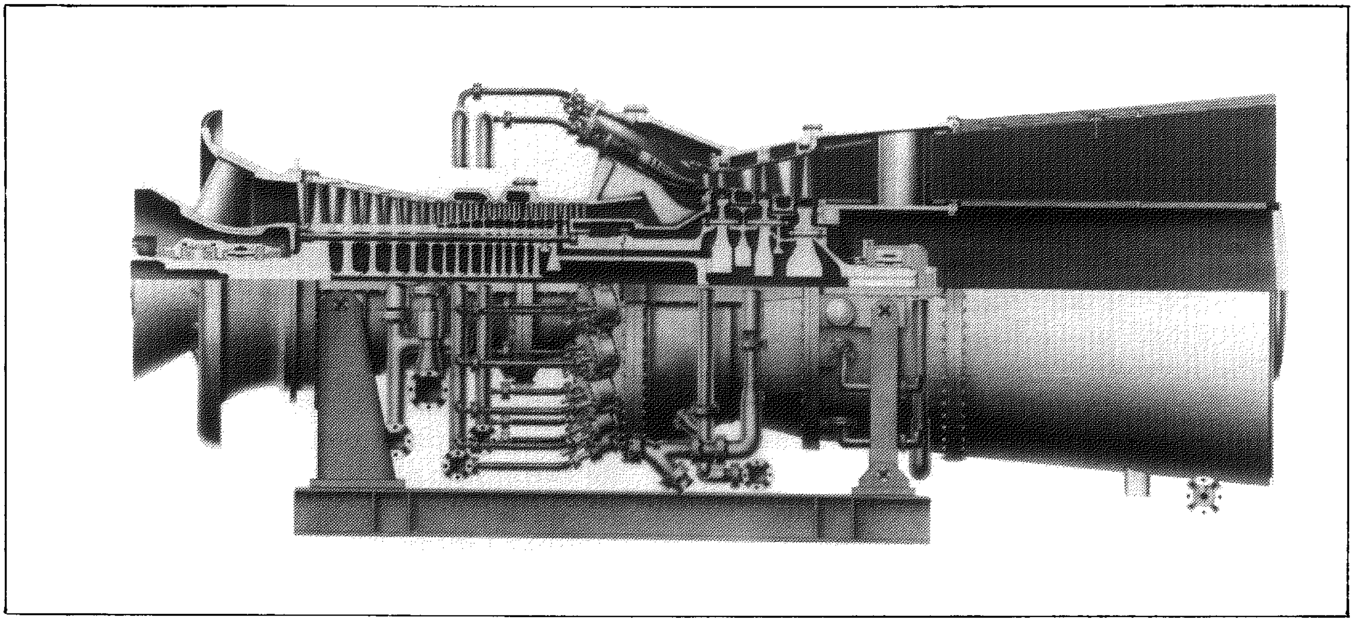

Figure 2. MS7001FA simple cycle gas turbine

that continuous combustion can take place. This raises the temperature of the working gases to a very high level. This combustion must take place with a minimum of pressure drop and emission production. The very high temperature gases flow from the combustor to the first stage turbine nozzles.

It is in the turbine that work is extracted from the high pressure, high temperature working fluid as it expands from the high pressure developed by the compressor down to atmospheric pressure. As the gases leave the combustor, the temperature is well above that of the melting point of the materials of construction in the nozzles and first stage buckets. Extensive cooling of the early stages of the turbine is essential to ensure adequate component life. While the hot gases cool as they expand, the temperature of the exhaust gases is still well above that of the original ambient conditions. The elevated temperature of the exhaust gases means that considerable energy is still available for boiling and superheating water in a combined cycle bottoming plant. It is this use of the exhaust energy that results in the dramatic improvement in cycle efficiencies between simple cycle turbine and combined cycle systems.

AXIAL COMPRESSOR

Aerodynamic Development

GE's experience with compressor design spans several decades. The original heavy-duty design of the axial-flow compressor was based on experience with the development of the TG180 aircraft jet engine during the mid-1940s. In the late 1940s, a prime mover was designed based on the TG180 and intended for use in pipeline pumping and industrial power applications. This prime mover, the earliest model of the MS3002, was a 5000-hp gas turbine with a compressor airflow of 37 kg/sec (81.5 lb/sec). The original MS3002 compressor did not require bleed valves, variable-inlet guide vanes, or variable-angle stator vanes for the turbine to accelerate and operate over a wide speed range without compressor surge. El Paso Natural Gas Company purchased 28 of these turbines which, after 30 years of operation, have accumulated an average of over 200,000 hours each.

In 1955, the design of a new compressor was undertaken to better satisfy the electrical power generation market; this design resulted in higher airflow and higher efficiency. Blade air-foils, an improvement over the NACA 65 series profile, were tapered in chord and camber and specified a root thickness of 13.5% of chord to provide ruggedness. Air extraction ports were added to the fourth and tenth stages to avoid surge while the compressor accelerated to rated speed. This design, used in the original MS5000, produced an airflow rate of 72.4 kg/sec (159.2 lb/sec) and a pressure ratio of 6.78 at 4860 rpm. Compressor airflow was later increased by raising the rotational speed to 5100 rpm and opening the inlet guide vanes, resulting in the basic MS5001M design which has led to today's modern compressors.

Starting with the MS5001M, the family of compressors in GE's present product line has been developed for single-shaft units by increasing the diameter of the inlet stage to increase the airflow and pressure ratio. For the MS5001N, the first three stages of the MS5001M were redesigned, and a stage was added at the inlet. The fixed inlet guide vane was replaced with a variable guide vane to adjust the airflow at start-up and provide higher firing temperature at reduced load for regenerative-cycle and combined-cycle applications. The MS5001N compressor operated at a pressure ratio of 9.8. It was tested at GE's aircraft engine compressor facility at Lynn, Massachusetts, where flow, pressure ratio, efficiency, start-up characteristics, full-speed surge margin, and mechanical integrity were established.

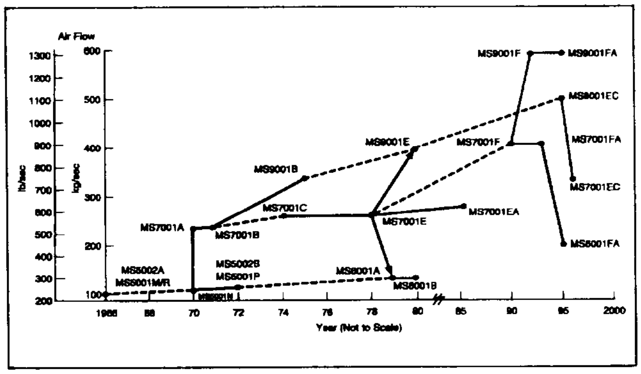

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Figure 3. Growth in compressor air flow

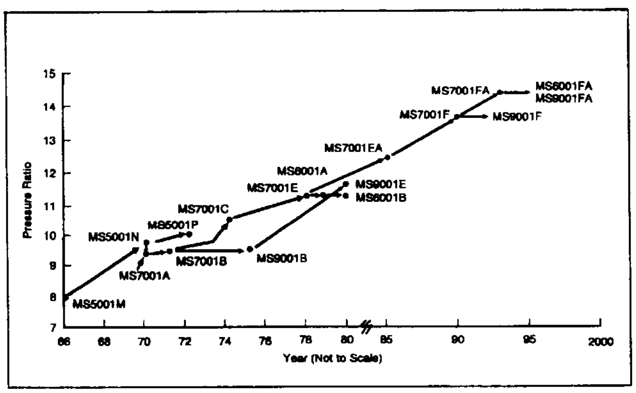

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Figure 4. Growth in compressor pressure ratio

The MS5001N and P, the MS7001A and B, and the MS9001B are essentially the same aerodynamic design, with increases in airflow and pressure ratio shown in Fig. 3 and 4. Figure 5 illustrates the mechanical configurations associated with these compressors. The MS5001N compressor, which runs at 5,100 rpm, was scaled to 3,600 rpm with over a 100% increase in airflow, and used in the MS7001A design. The flow and pressure ratio have been increased further in the MS7001C and MS7001E by redesigning the first four stages. A modification made to the stators of stages 1 through 8 was applied to the MS7001E, MS9001E, and MS6001 to improve underfrequency operation. Figure 6 shows how the power available during underfrequency conditions was improved by this modification. With the current production compressor, this power reduction is unnecessary because of the improved part-speed surge margin in the compressor. The slight fall-off in power results from reduced airflow at lower speed.

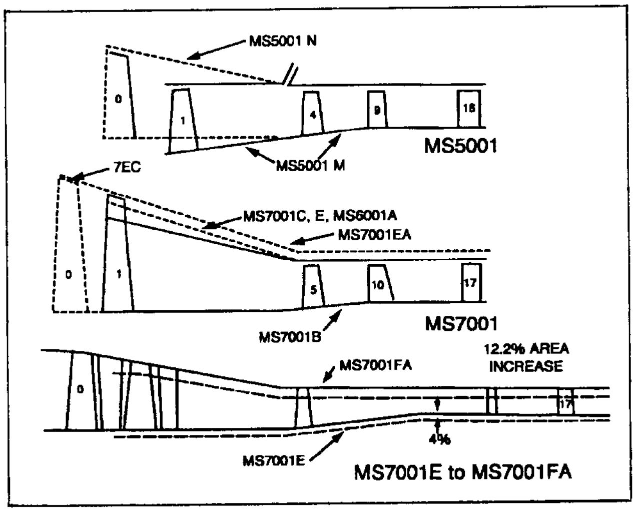

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Figure 5. Evolution of compressor design

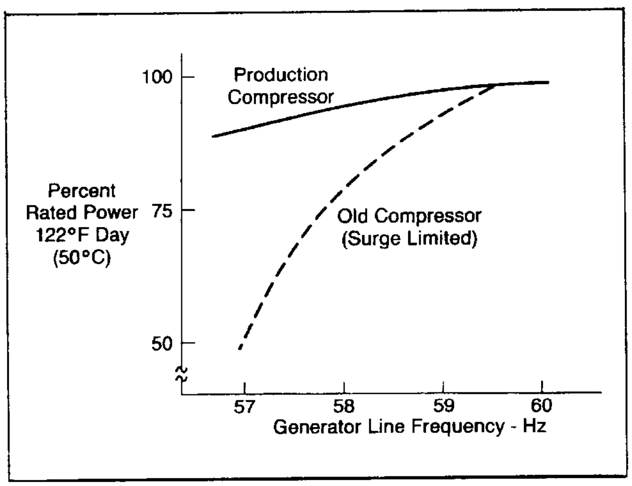

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Figure 6. MS7001 under-frequency power (peak load, hot day 50C (122F)

A further improvement in the output of the MS7001E machine was made by simply increasing the outer diameter of the compressors. This has resulted in a 3.7% increase in flow and a new designation of MS7001EA, as illustrated in Fig. 3, 4 and 5.

In 1986, GE introduced a new gas turbine the, MS7001F, and its derivative, the MS9001F; in 1990, the uprated MS7001FA and MS9001FA (Fig. 7); and in 1993, the scaled MS6001FA. The compressor for the MS7001FA is an axial-flow, 18-stage compressor with extraction provisions at stages 9 and 13. The compressor aerodynamic and mechanical design closely follows the 17-stage MS7001E, but with an additional zero stage. For convenience in maintaining this relationship, the MS7001FA compressor stages are numbered 0 through 17 rather than 1 through 18.

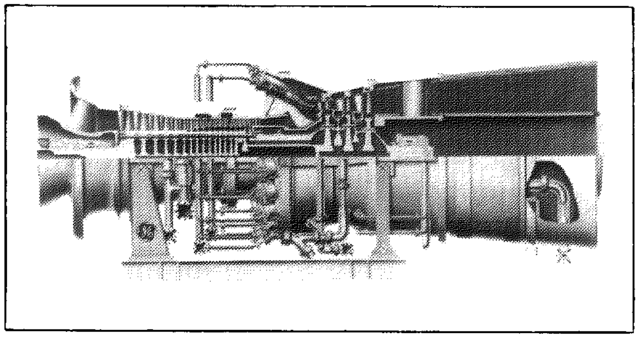

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Figure 7. MS9001FA gas turbine

The MS7001FA compressor was developed by first scaling the diameters of the MS7001E, then increasing the annulus area an additional amount to achieve the desired flow, and lastly adding a 0 stage. As a result, the MS7001FA is aerodynamically similar to the MS7001E, and most of the blading is identical to the MS7001E except for length. Stages 0 and 1 have been designed for operation in transonic flow using design practices applied by aircraft gas turbine designers. As a result of using this conservative design approach, variable stators in addition to variable-inlet guide vanes are not required for surge control. The MS9001FA and MS6001FA are direct scales of the MS7001FA.

The 7EC compressor, although introduced later, uses a similar approach by adding a zero stage directly to the 7EA compressor. As shown in Fig. 5, the aft stages are the same as 7EA. The 9EC is a direct scale of the 7EC for the 50 Hz size.

Table 2 lists some of the parameters of these axial compressors. By starting with an efficient, reliable design and improving this design in a gradual manner, improved overall compressor performance has been achieved without sacrificing reliability or mechanical integrity.

Table 2
COMPRESSOR ROTOR DESIGN PARAMETERS

Unit	Tip Speed Ft./Sec. (m/Sec.)	Compressor Tip Diameter Inches (mm)	N RPM	Turbine Output (MW) Base, ISO, Gas
MS5001P	1092 (333)	49.1 (1247.1)	5100	26.3
MS7001B	1092 (333)	69.5 (1765.3)	3600	60.0
MS9001B	1092 (333)	83.5 (2120.9)	3000	84.7
MS6001B	1114 (340)	50.1 (1272.5)	5100	38.3
MS7001E	1114 (340)	70.9 (1800.6)	3600	75.8
MS7001EA	1120 (341)	71.3 (1811.0)	3600	83.5
MS9001E	1114 (340)	85.1 (2161.5)	3000	116.9
MS6001FA	1282 (391)	56.1 (1425.5)	5235	70.1
MS7001FA	1282 (391)	81.6 (2072.6)	3600	168.4
MS9001FA	1282 (391)	97.9 (2486.6)	3000	226.5
MS7001EC	1227 (374)	78.1 (1983.7)	3600	116.0
MS9001EC	1227 (374)	93.7 (2380.0)	3000	169.2

Thorough testing is essential for the development of modern axial compressors. In GE's manufacturing facility in Greenville, South Carolina, a standard MS7001 compressor (Fig. 8) is used as a loading device for testing prototype gas turbines and as a compressor development vehicle. The facility has been constructed with nozzles for measuring airflow, valves for regulating airflow, and flow straighteners in the inlet duct.

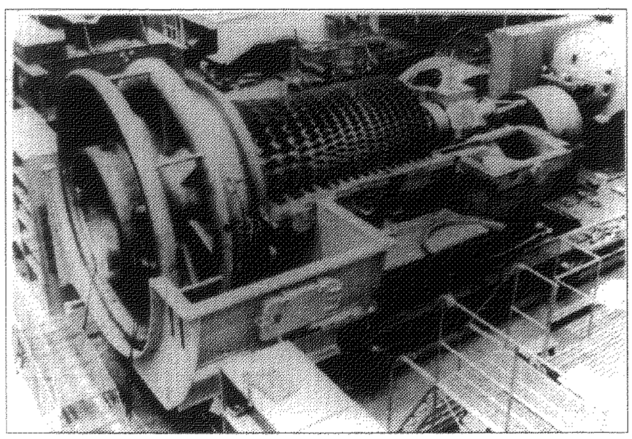

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Figure 8. MS7001 load test of axial-flow compressor

The discharge system includes parallel discharge valves for coarse and fine adjustment of the pressure ratio. Provisions for standard extraction, bleed flow, and flow measurement have also been made. For test flexibility, some of the controls for the load compressor have also been made to protect the equipment in case of trips.

Test measurements include flow in and out of the compressor, inlet and discharge pressures and temperatures, and interstage pressures and temperatures needed to design stage-by-stage characteristics. Dynamic data are measured to evaluate rotating stall, surge, and blade stresses.

Tests are run over a wide range of speeds and pressure ratios to generate a performance map, start-up characteristics, stress data, blade dynamic characteristics, and to design surge margins. Since 1968, seven full-scale compressor development programs have been conducted by GE. Results include computer models which permit design improvement analysis. As a result of these tests, the performance and operating characteristics of GE compressors can be predicted with considerable accuracy throughout the operating range.

Mechanical Construction

GE axial compressors have proven to be durable, stable, and reliable. The design also offers important versatility for optimizing compressor wheel material characteristics, cost, and service conditions.

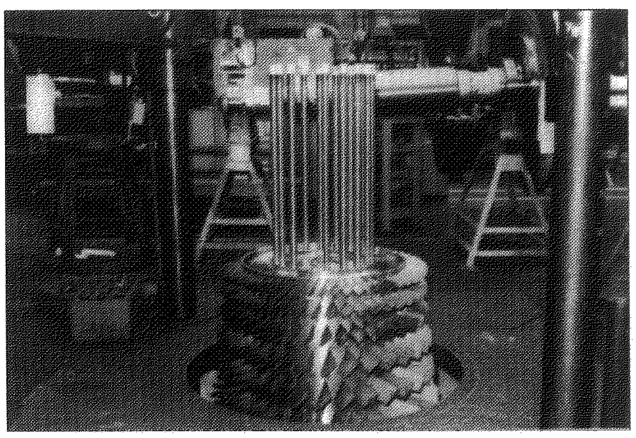

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Figure 9. MS5001 compressor rotor stacking

Each stage of the compressor is an individual bladed disk (Fig. 9). The use of this construction allows some weight reduction by contouring the wheels, thereby reducing the mass which must be accelerated during start-up. The disks are assembled with a number of axial tie-bolts, with the bolt-circle diameter selected to produce a dynamically stiff rotor and good torque transmission. The stiffness and mass of GE rotors insures that the first bending critical speed is above the running speed. The wheels are positioned radially by a rabbet fit near the bore. Axial clearance is provided between the wheel rims to allow for thermal expansion during start-up.

Application requirements have resulted in several important mechanical design features in the axial compressor. In models with air-cooled turbine buckets, the last-stage wheel has been adapted to provide an extraction to supply the necessary cooling air for the turbine and rotor buckets (Fig. 10). The system was designed and carefully tested to extract air without disturbing the main compressor flowpath. The extraction system is a radial in-flow turbine which accepts compressor air at the outer diameter entrance with low-pressure loss, and completely guides the flow to a radial direction so it enters the rotor bore without swirl. The guide slots in the wheel eliminate free-vortex flow in the extraction system, providing aerodynamic stability over the entire range of compressor operation.

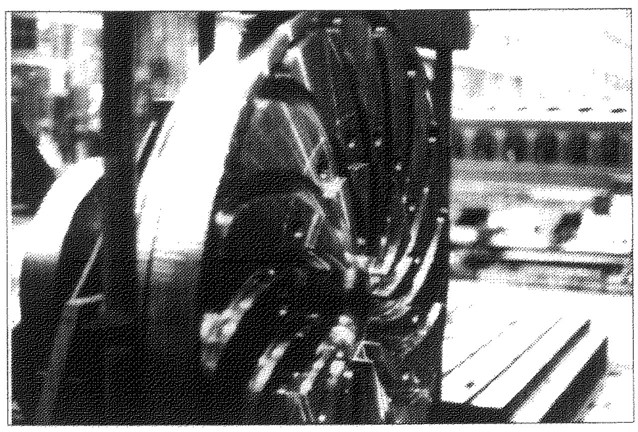

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Figure 10. Last-stage wheel with cooling-air extraction

Higher-cycle pressure ratios produce higher compressor-discharge temperatures; the MS7001E compressor-discharge temperature increased by 31.6C (57F) over the MS7001B when the pressure ratio was raised from 9.6 to 11.5. To compensate for the temperature increase, higher-strength material (CrMoV) is used in the last compressor stage. This material has the high-temperature strength compatible with a wheel life of 30 years at base load.

Gas turbine rotors are designed for may thousands of starts. Start-up and shutdown thermal stress, material properties, and material quality are considered in the design. Additionally, the material quality of each wheel is ensured by very stringent process controls and ultrasonic inspection procedures. Compressor wheels with turbine-grade materials, such as CrMoV, receive high-speed proof testing similar to our long-standing practice for turbine wheels.

Each wheel is spun in a pit after being cooled below its fracture appearance transition temperature (FATT). The wheel is then in a brittle condition and would fail if a serious flaw existed. A hot spin, with the wheel temperature well above the FATT, is also used to enhance the life of the wheel. The speed is sufficiently high to plastically yield the bore, producing a residual compressive stress at the bore when the wheel is brought to rest. During subsequent operation of the machine, the residual stress reduces the bore tensile stress, producing enhanced low-cycle fatigue life (Fig. 11).

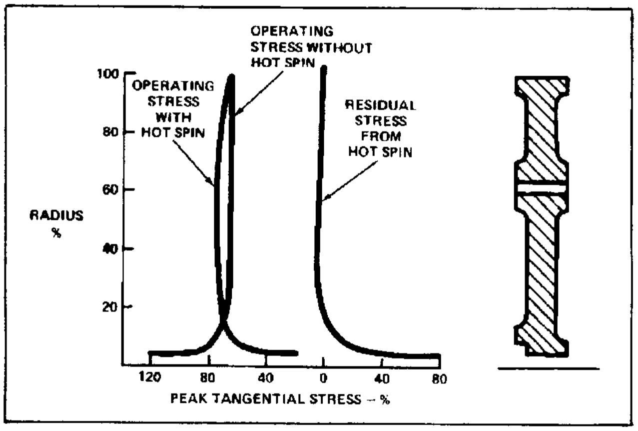

Figure 11. Improvements due to hot spinning

The remainder of the compressor wheels are made of three basic grades of steel, CrMo, NiCrMo, and NiCrMoV the principal alloying elements. Processing of these alloys produces a balance of desired material properties including tensile strength and fracture toughness. Fracture toughness is important for good cyclic life of wheels, especially in low ambient environments. Since 1970, optimization of these materials has resulted in a 35% improvement in fracture toughness (Fig. 12). The banded area shows the evolution of the minimum and maximum observed values for low-temperature fracture toughness.

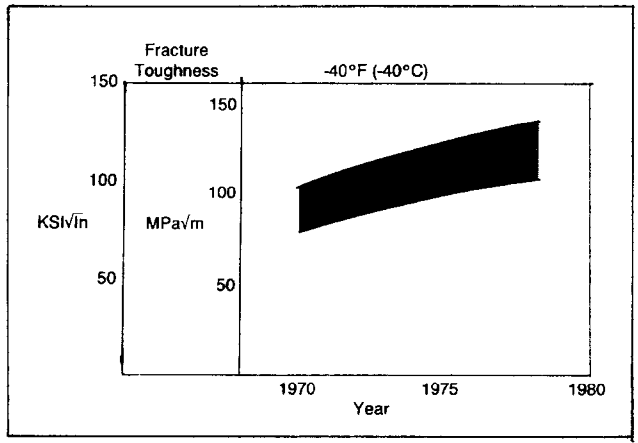

Figure 12. Facture toughness of compressor rotor steels

The same proven alloys and construction techniques have been employed in the MS6001FA, MS7001FA and MS9001FA designs, a very trouble free and reliable design.

MULTIPLE-COMBUSTION SYSTEM

Design

A typical reverse-flow multiple-combustion system, similar to those in most of the GE heavy-duty gas turbines, is shown in Fig. 13. This system is a product of years of intensive development and successful field application. In the combustor, a highly turbulent reaction occurs at temperatures above 1982C (3600F). The essential feature of the combustor is to stabilize the flame in a high-velocity stream where sustained combustion is difficult. The combustion process must be stable over the wide range of fuel flows required for ignition, start-up, and full power. It must perform within desirable ranges of emissions, exit temperature, and fuel properties, and must minimize the parasitic pressure drop between compressor and turbine. The combustion hardware must be mechanically simple, rugged, and small enough to be properly cooled by the available air. This hardware must have acceptable life and be accessible, maintainable, and repairable. GE's reverse-flow, multiple-combustion system is short, compact, lightweight, and is mounted within the flange-to-flange machine on the same turbine base. This multi-combustor concept has allowed full machine size and operating conditions to be applied to combustor systems during laboratory development testing.

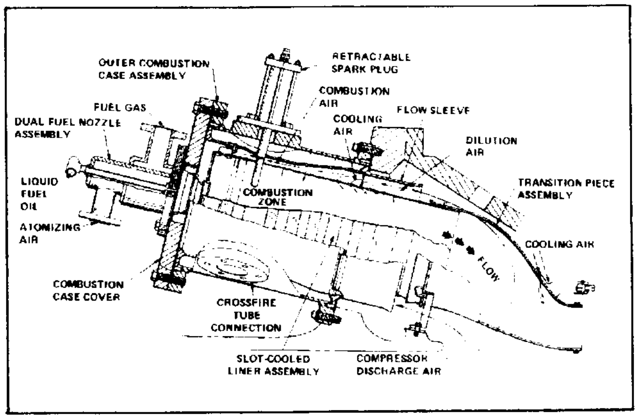

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Figure 13. Reverse-flow combustion system

While model tests are useful for locating areas of high pollutant formation, such models do not allow prediction of other operating characteristics. Since a scale model does not reproduce chemical reactions, heat release rates, and aerodynamic mixing, neither mathematical nor geometric modeling has proven satisfactory for combustion development. In addition, aerodynamic mixing, which is achieved by jet penetration from the walls of the combustor, is more difficult in a larger-diameter combustor burner. For this reason, good emissions performance, which depends strongly on aerodynamic mixing, cannot be predicted from scale model tests. A practical combustor can only be developed in full-scale tests.

Almost all laboratory testing of development work can be done on a single-burner test stand at full operating conditions, with only a fraction of the fuel and air of a complete gas turbine. All GE heavy-duty gas turbines except the MS1002 are designed to use multiple combustion chambers offering significant adaptability such as:

- Small diameter permitting careful control of the airflow patterns for smoke and NOx reduction.
- The design allows control of the gas path profile.
- Combustor diameter can readily be increased to accommodate combustion of the low heating value gas fuels.
- Combustor length can be provided for residual fuels.
- The design is readily adaptable to modifications, such as water injection.
- The components are small enough to be adequately cooled.

As a result, all GE gas turbines, with their fully developed combustion systems, are shipped from the factory fully tuned, precluding the need for start-up adjustments or field testing.

The combustion chamber diameters are not scaled for the different turbine models. Only two combustion liner diameters for non-DLN applications are used for the GE product line: a 268 mm (10.7-in.) diameter for the MS3000, MS5000, and MS6000; and a 358 mm (14.3-in.) diameter for the MS7000 and MS9000. The combustion liners for the MS5001N, MS6001, MS7001B and MS7001E are shown in Fig. 14.

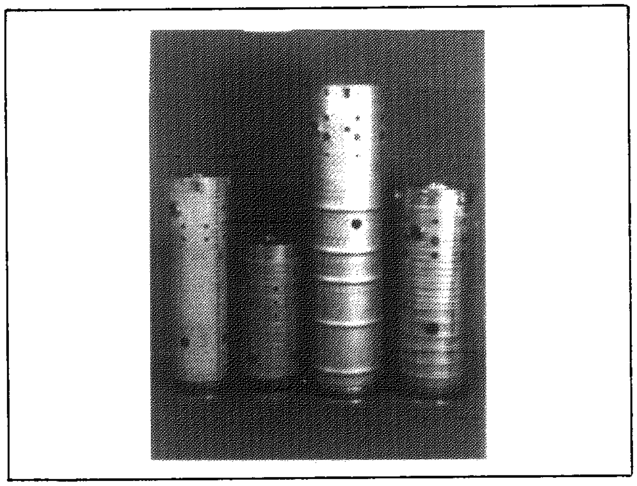

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Figure 14. Combustion liner comparison

The number of combustors, however, is adjusted proportionally to the machine airflow divided by the pressure ratio, e.g., the MS9001E uses 14 combustors compared to 10 on the MS7001E because the 9E airflow is 1.44 times as large.

The MS7001FA combustion system consists of 14 combustion chambers. The liners are constructed in a manner identical to the MS7001E liners but are 30% thicker and 210 mm (8.4 in.) shorter. The MS7001FA liners are constructed of Hastelloy-X material, as are the other product line liners, with the addition of HS-188 in the aft

278 mm (11.1 in.) portion and the application of thermal barrier coating to the internal surface. These additions provide for improved high-temperature strength and a reduction of metal temperatures and thermal gradients. The MS6001FA uses six combustors and the MS9001FA uses 18 .

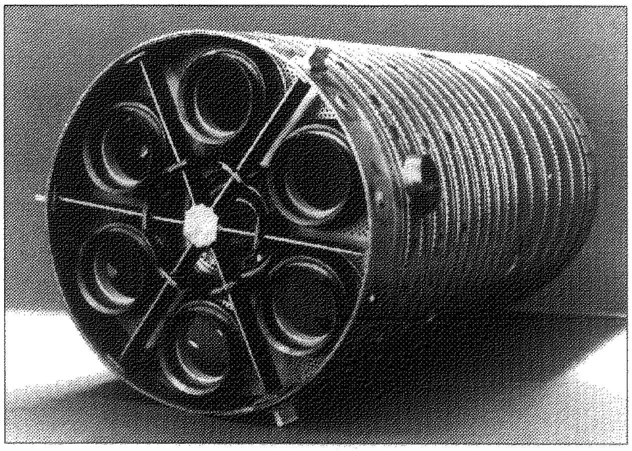

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Figure 15. Combustion liner cap

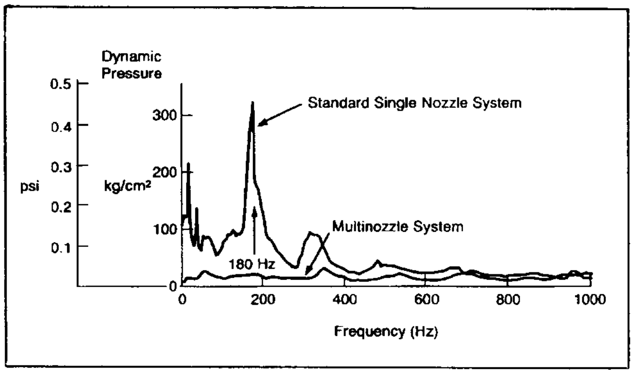

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Figure 16. Multi-and single-fuel nozzle combustion noise

The liner cap is changed from the MS7001E design to accomodate six fuel nozzles instead of one (Fig. 15). This multi-fuel-nozzle arrangement was selected because of the superior field experience with multi-fuel-nozzle systems on an operating MS7001 gas turbine in utility service with water injection for NOx control. This test, confirmed by extensive laboratory full-scale combustion tests, clearly demonstrated the reduced combustion noise (dynamic pressure) level achieved when operating with multi-fuel instead of single-fuel-nozzle systems (Fig. 16). This noise reduction reduced wear in the combustion system so that combustion inspection intervals of a tested machine could have been extended from 3,000 to 12,000 hours. Additionally, multi-fuel-nozzles result in a shorter flame, and the MS7001FA combustion system is 575 mm (23 in.) shorter than the MS7001E system. The six fuel nozzles are mounted directly on the combustion end cover and require no more piping connections than a single fuel nozzle because of manifolding integral with the cover.

The combustion ignition system uses two spark plugs and two flame detectors, along with cross fire tubes. Ignition in one of the chambers produces a pressure rise which forces hot gases through the cross-fire tubes, propagating ignition to other chambers within one second. Flame detectors, located diametrically opposite the spark plugs, signal the control system when ignition has been completed. Because of the relative simplicity and reliability of this technique, it is used in all GE heavy-duty gas turbines.

Fuel is distributed into the combustion chambers by fuel nozzles. For gas, the fuel nozzle is a simple cap with accurately drilled metering holes. Liquid fuels are metered by a positive-displacement, gear-element flow divider. Liquids are either pressure-atomized or air-atomized if better smoke performance is required. Residual fuel and crudes generally require atomizing air to achieve acceptable smoke performance.

The size of the combustion liners provides the space required to completely burn residual fuel. Lighter fuels are also easily burned in these liners. Smaller-diameter GE combustors allow penetration of air jets into the combustor at acceptable pressure drops. Jet penetration is necessary to mix the air with the fuel quickly and obtain complete combustion without forming soot in fuel-rich pockets. The highly stir-red flame produced by these jets also reduces radiation to the liner walls, with beneficial effect on liner life.

The combustion liner is carefully cooled to tolerate high-temperature gases a few millimeters from the combustor liner wall. As firing temperatures increase, more air is needed to combine with the fuel for adequate combustion, and less air is available for liner wall cooling. This has been offset by a more efficient cooling system and by reducing the surface area (length) of the liner. Louver cooling, which has been highly successful and reliable over the years, has been replaced by slot cooling in the turbines with the highest firing temperatures. The slot cooling method reduces liner metal temperatures by 139C (250F) compared to an equivalent louver system, and is the standard cooling method in aircraft gas turbines.

The length of the combustor provides time to complete the combustion reaction for the variety of fuels burned in the turbine and then dilute the combustion products with excess air to form a temperature profile acceptable to the downstream turbine components. The temperature profile of hot gases entering the turbine sections is carefully developed to provide maximum life for the nozzles and buckets. The average radial profile from the combustors will produce lower temperatures near the bucket root where the centrifugal stress is maximum, and at the outer sidewall where nozzle bending stresses are also at a maximum.

The transition piece, which channels the high-temperature gas from the combustion liner to the first-stage turbine nozzle, is small enough to be cooled by air flowing from the compressor. This provides effective cooling of the transition piece for firing temperatures up to 1010C (1850F). The outer portion of the transition piece near the first-stage nozzle is less effectively cooled, and at firing temperatures above 1010C (1850F) jet-film cooling is added.

The MS6001FA, MS7001FA and MS9001FA transition piece is constructed of two major assemblies (Fig. 17), which is unique to these machines. The inner transition piece is surrounded by a perforated sleeve with the same general shape as the transition piece. This perforated sleeve forms an impingement cooling shell causing jets of compressor discharge air to be directed onto the transition piece body. The air, after impinging on the transition piece body, then flows forward in the space between the impingement sleeve and transition piece into the annulus between the flow sleeve and the combustion liner. It then joins additional air flowing through bypass holes provided in the flow sleeve to provide the air for the combustion/cooling/ dilution processes (Fig. 17). The impingement sleeve is fabricated of AISI-304 stainless steel, the transition piece body of Nimonic 263, and the aft frame of cast FSX-414. The internal surface of the transition piece has a thermal barrier coating to minimize metal

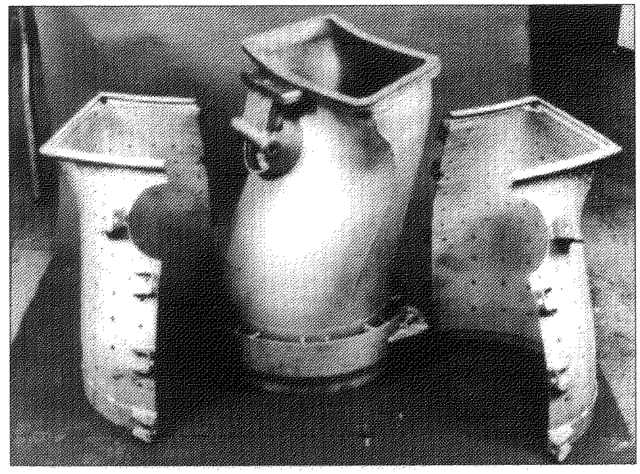

Figure 17. Transition piece

temperatures and thermal gradients.

Higher firing temperatures require combustors that release more energy in a given volume. High volumetric heat release rates, which depend on higher turbulent mixing in the combustor primary zone, are achieved by raising the combustor pressure drop. As mixing has increased in combustors, the turbulence generated by combustion may cause broad-frequency-banded noise. While this is generally "white noise," it is possible for the combustion flame to couple with the acoustic characteristics of the combustor volume or fuel system components to generate unacceptable pure tone frequencies, or acoustic waves.

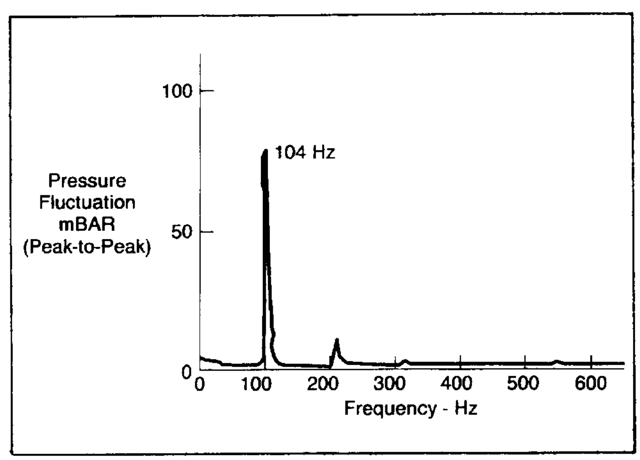

Figure 18. Combustor dynamic pressure spectrum

The characteristic frequency of the waves is established by the combustor geometry or external equipment such as the fuel pump. One example of this phenomenon is shown by a spectrum of the dynamic pressure within a combustion chamber while burning natural gas (Fig. 18). When the chamber pressure near the fuel nozzle rises, the fuel flow is reduced. Conversely, a decrease in pressure near the fuel nozzle causes an increase in fuel flow. Amplified pressure oscillations occur when a low fuel-nozzle pressure drop permits this fuel flow oscillation. These dynamic pressures can be damaging to the combustion hardware. Comprehensive testing under actual operating conditions is necessary to develop systems in which these pure tone frequencies are avoided.

In order to determine an acceptable range of fuel-nozzle pressure drops, stability maps (Fig. 19) have been developed from tests run in our Gas Turbine Development Laboratory. This map is used to select fuel-nozzle designs which ensure stable system operation.

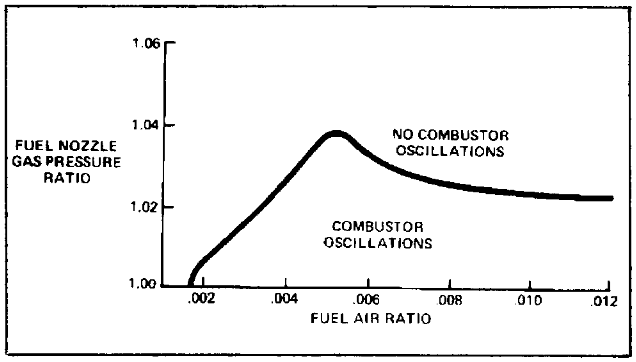

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Figure 19. Combustor dynamic pressure stability (gas fuel)

Development Testing

The Gas Turbine Development Laboratory has six test stands which operate at full machine conditions in either simple-cycle or regenerative-cycle configurations. The stands are equipped to inject water, steam, or inert gas for emissions reduction. Tests may run using gaseous or liquid propane, methane, distillates, blended residuals, or heavy residual fuels. A low heating value fuel facility is also available with the capability to blend fuel and inert gases for a heating value range of 3353 to 4098 kJ/m^3 (90 to 110 Btu/ft^3). The main test bay is shown in Fig. 20. Since laboratory testing of combustion components and systems can be performed under full machine conditions, we are able to achieve excellent correlations between laboratory and field performance.

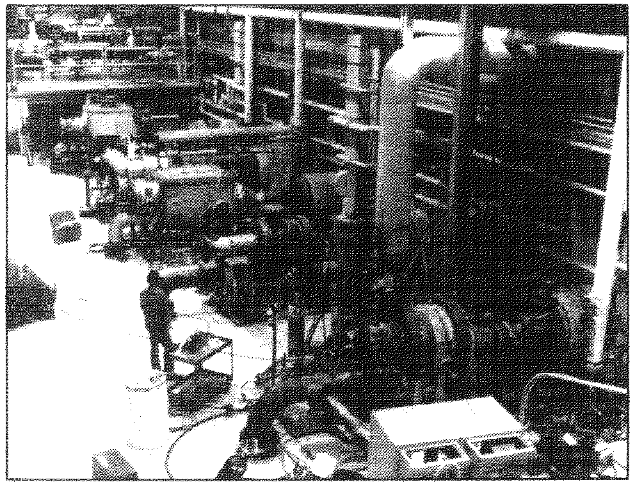

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Figure 20. Gas turbine development laboratory main test bay

Additional cold-flow testing is conducted at the GE Research and Development Center on scale models of all new combustion systems (Fig. 21). These models are used to measure the flow distribution from the compressor discharge diffuser to the individual combustion chambers. Model testing is useful for measurements of static pressure recovery and flow visualization to ensure flow stability in the vicinity of the combustion chamber.

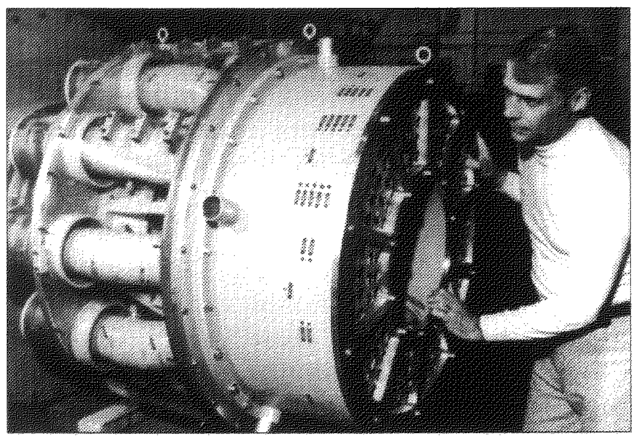

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Figure 21. Combustion system scale model

After laboratory development of combustors, testing is completed on a production turbine at full load conditions. This turbine is extensively instrumented to evaluate the combustor performance and to permit comparison with the results of the single-burner test. Measurements are made of the gas temperature profile at the entrance to the first-stage nozzle, metal temperatures and vibratory response of the hardware, combustor pressure drop, and dynamic pressures in combustors, fuel lines, and atomizing-air piping. Lightoff, cross-firing, and control characteristics are also measured. Emissions from the turbine exhaust are determined, including smoke and particulate matter, to compare with laboratory tests and theoretical predictions. Water and steam injection systems are tested to determine the amount of water or steam required to meet emissions standards. Years of gas turbine combustor development experience have shown that this combination of laboratory and machine testing is essential to the production of a reliable combustion system.

Dry Low No_x Development

GE Power Generation's Dry Low NO_x (DLN) development is a multi-faceted program to provide combustors, controls, and fuel systems that significantly reduce emissions from both the current gas turbine product line and existing field machines. There are many programs that provide products to meet current emissions codes and prepare for more stringent requirements in the future. The available DLN products for the MS6001B, 7001EA, and 9001E machines are designed to meet 15 ppmvd at 15% O_2 of NO_x. This DLN technology has been extended to produce equivalent products for the MS7001FA and MS9001FA class machines. More advanced DLN systems are being developed to meet 9 ppmvd at 15% O_2 of NO_x.

The Dry Low NO_x system is a sophisticated system that requires close integration of a staged, premixed combutor, the gas turbine's SPEEDTRONIC™ controls, and the fuel and associated sytems. Thus, there are two principal measures of performance. The first one is emissions– the base load levels of NO_x and CO that can be achieved on both gas and oil fuel, and how these levels vary across the load ranges of the gas turbine. The second measure is operability– the smoothness and reliability of combustor mode changes, the ability to load and unload the machine without restriction, the capability to switch from one fuel to another and back again, and the response to rapid transients (e.g., generator breaker open events or rapid swings in load). GE's design goal is to make the DLN system operate so that the gas turbine operator does not know such a system is installed, i.e. it is "transparent" to the user." To date, a significant portion of the design and development effort has focused on operability.

The Dry Low NO_x combustor, shown in the cross section in Fig. 22, is a two-staged premixed combustor designed for use with natural gas fuel and capable of operation on liquid fuel. As shown, the combustion system comprises four major components: fuel injection system, liner, venturi, and cap/centerbody. These are arranged to form two stages in the combustor. In the premixed mode, the first stage serves to thoroughly mix the fuel and air and to deliver a uniform, lean, unburned fuel-air mixture to the second stage.GE Dry Low NO_x combustion systems are currently operating in 60 field machines, As of June '94, they have accumulated over 200,000 operating hours.

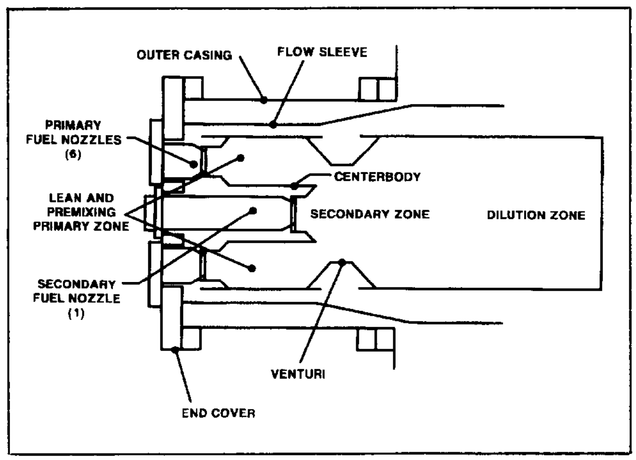

Figure 22. Dry low NO_x combustor

TURBINE

Background

Increasing firing temperature has been the most significant development thrust for turbines over the past 30 years. Baseload firing temperature capability has increased from 816C (1500F) in 1961, when the MS5000 package power plant was introduced, to 1288C (2350F) today in the MS6001FA, MS7001FA, and MS9001FA machines. The base rated power of the MS7000 has increased since the first model was shipped in 1971, from the 46-MW MS7001A to the 83.5-MW MS7001EA. Seventy percent of this increase has been accomplished through higher firing temperature; the remainder from increases in airflow because of compressor developments.

Higher turbine firing temperatures are achieved by improved nozzle and bucket materials and by the air-cooling of this hardware. Concurrent development in alloy corrosion and oxidation resistance and bucket surface protection systems have played a significant role in supporting firing temperature increases.

Aerodynamics

GE gas turbines are characterized as a high energy-per-stage design, which requires a high stage pressure ratio. This results in the two or three turbine stages typical of GE heavy-duty gas turbines, instead of the five stage low energy-per-stage design common in competing machines.

The temperature of the first of three high energy-per-stage buckets will be approximately 55C (100F) lower than the first of five low energy-per-stage buckets. As shown in Fig. 23, for a given wheel speed, firing temperature, and turbine output, higher energy-per-stage turbines have fewer stages than lower energy-per-stage turbines . This results in a larger energy drop (hence reduction in temperature) per stage

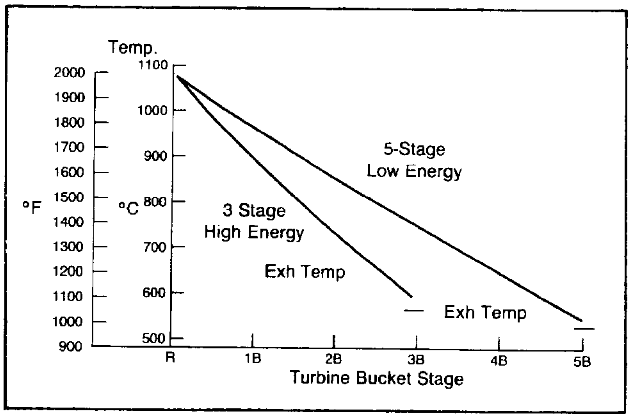

Figure 23. Bucket metal temperatures

and, therefore, lower bucket metal temperatures.

Since high energy-per-stage turbines have inherently lower metal temperatures for a given firing temperature, it follows that less cooling air needs to be supplied in order to provide satisfactory metal temperatures and component lives. The greater amount of cooling air which must be supplied to a lower energy-per-stage turbine bucket imposes a greater performance penalty upon that design. Conversely, for the same cooling airflow, the high energy-per-stage turbine bucket will inherently have a lower metal temperature, and hence, longer life.

Turbine Cooling

The thermal efficiency and specific output of a gas turbine are strongly influenced by two cycle parameters, pressure ratio and firing temperature (Fig. 24). Thermal efficiency increases up to stoichiometric firing temperature levels and pressure ratios of 50:1 or 60:1, in an ideal cycle where losses for turbine cooling are not considered. Since superalloys begin to melt at about 1200C (2200F), the hot-gas-path components must be cooled to maintain metal temperatures well below this temperature. For this rea-

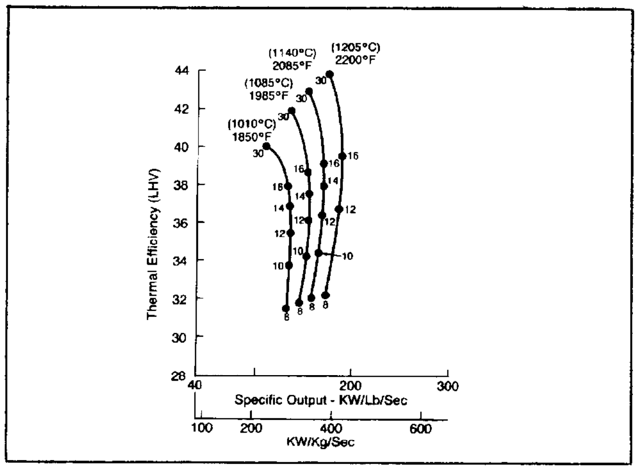

Figure 24. No compressor extraction flow (ideal flow)

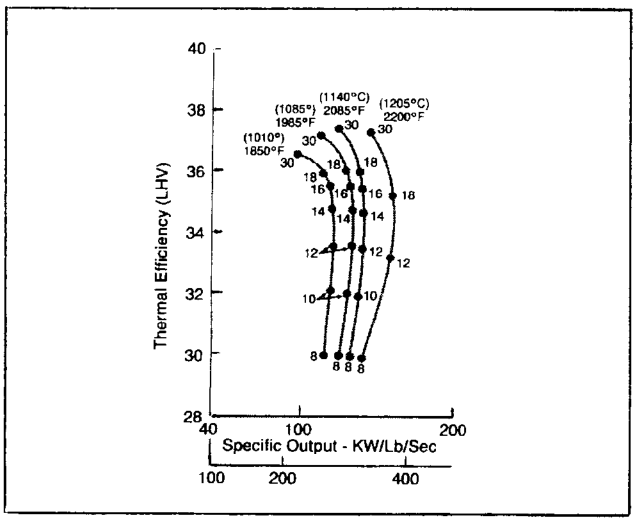

Figure 25. Compressor extraction flows as needed (real flow)

son, air is extracted from the compressor and used to cool these components.

While substantial performance gains can be realized by increased firing temperature, a comparison of Figs. 24 and 25 shows that performance improvements are also possible at fixed firing temperatures by use of higher-temperature materials to reduce cooling losses. More efficient cooling systems will also improve performance.

Beginning in the early 1960s, air-cooled first-stage nozzles were introduced into GE heavy-duty designs. Nozzle metal temperatures were maintained at about 843C (1550F), as firing temperatures were raised to take advantage of stronger bucket alloys. By the late 1960s, turbine baseload firing temperatures were near 910C(1670F), and significant firing temperature increases depended on cooling first-stage buckets. With future increases in mind, the MS7001 was designed to be readily adaptable to bucket cooling.

Several important criteria were selected for air-cooled turbines. First, the bucket air-cooling circuit is entirely internal to the rotor, starting with radially inward extraction from the inner diameter of the compressor gas path. As the compressor acts as a centrifuge for dirt, the internal extraction point minimizes the amount

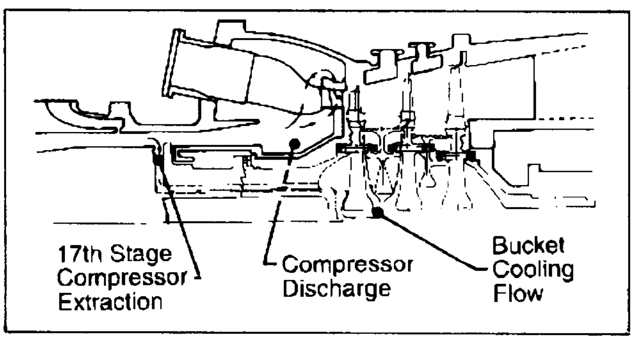

Figure 26. Internal cooling circuit

of foreign matter taken into the cooling circuit. The internal circuit, shown in Fig. 26, eliminates the need for additional seals or packings between the rotor and stator to contain the cooling air, producing the highest possible integrity of the circuit. Second, metering of the air is accomplished by the buckets themselves because the cooling circuit has a much greater flow area than the bucket cooling holes. This provides the highest pressure drop for efficient heat transfer in the bucket. Additionally, metering of air at the buckets allows the cooling flow to increase if the buckets are damaged. This allows ash-forming heavy fuels to be burned without concern for external plugging of the bucket cooling system.

Cooled buckets and advanced air-cooled first-stage nozzles were shipped in MS7001B turbines beginning in 1972. A baseload firing temperature of 1004C (1840F) was established, 106C (190F) higher than the MS7001A uncooled bucket design. In the FA model of the MS7001, nozzle and bucket cooling have been further developed to provide a baseload firing temperature of 1288C (2350F). The first-stage bucket is convectively cooled via serpentine passages with turbulence promoters formed by coring techniques during the casting process (Fig. 27). The cooling air leaves the bucket through holes in the tip as well as in the trailing edge. The second-stage bucket is cooled by convective heat transfer using STEM (Shaped Tube Electrode Machining) drilled radial holes with all the cooling air exiting through the tip. The first-stage nozzle contains a forward and aft cavity in the vane, and is cooled by a combination of film, impingement, and convection techniques in both the vane and sidewall regions (Fig. 28). There are a total of 575 holes in each of the 24 segments. The second-stage nozzle is cooled by convection. The advanced cooling techniques applied in the MS7001FA turbine components are the result of extensive aircraft engine development, as well as correlative field testing performed on cooled components in current production heavy-duty machines. In addition, hot cascade tests were performed on MS7001FA

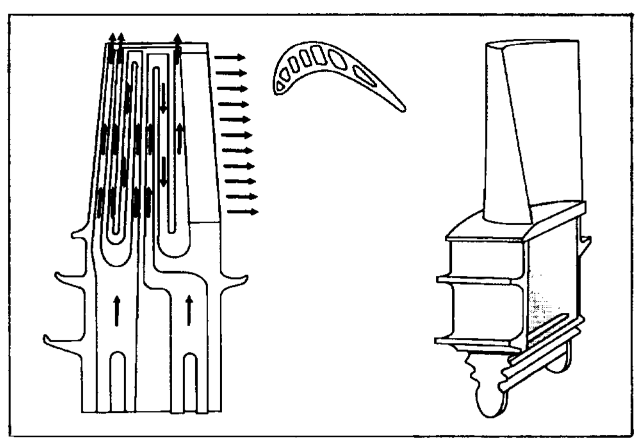

Figure 27. First-stage bucket cooling passages

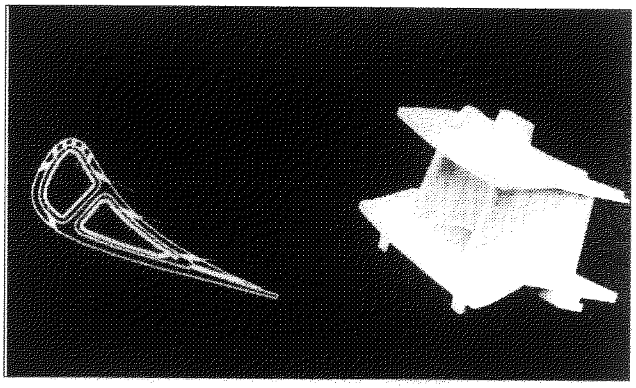

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Figure 28. First-stage nozzle cooling

first-stage components to validate the heat transfer design assumptions.

Bucket Design

Buckets are subjected to a gas force which provides torque to the rotor. Relatively small variations in these gas forces can cause bucket vibration. Coincidence of resonance between these periodic gas forces and bucket natural modes must be avoided at full operating speed; however, resonance cannot be avoided at all speeds, particularly during starting and shutdown. Effective vibration control is required, therefore, to produce reliable turbine designs. All GE-designed turbines incorporate two

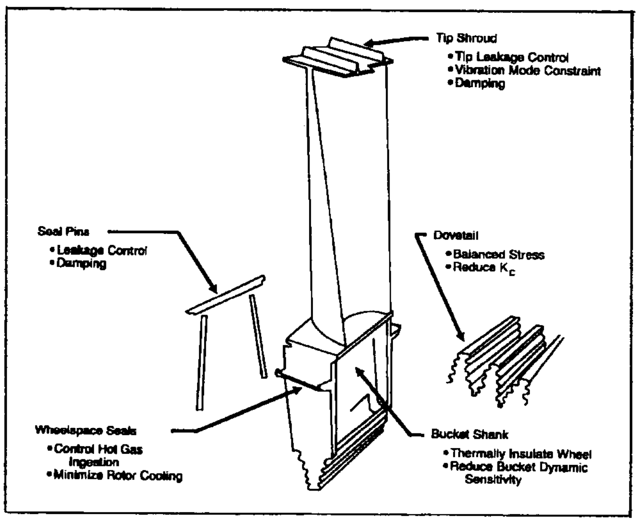

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Figure 29. MS6001 second stage bucket

important features to suppress resonant vibration—the long-shank bucket and the bucket tip shroud. Fig. 29 shows these features on the MS6001B bucket.

The bucket shank, which joins the bucket airfoil and the dovetail, is a significant fraction of the overall bucket length. Damping is introduced near the bucket midspan by placing axial pins underneath the bucket platform between adjacent buckets. On first-stage buckets, the damping provided by these pins virtually eliminates all vibration involving tangential motion and significantly reduces vibration in other modes. The shank has a second important advantage in providing an effective thermal isolation between the gas path and the turbine wheel dovetail. The dovetail is maintained at a low temperature, and because the shank is a uniform, unrestrained section, stress concentrations in the dovetail are minimized.

The integral tip shroud is the second major vibration control feature of GE-designed buckets and is used on the second and third stages. Individual bucket shrouds are interlocked to form a continuous band during operation. The natural tendency for the buckets to untwist under centrifugal load is used to force the mating faces of adjacent shrouds together, providing coulomb damping. The tip restraint provided by the continuous shroud band totally eliminates the most sensitive mode of vibration, the first flexural.

Service experience now provides a factual record. Since 1962, when shanks were introduced, no bucket of this design has experienced a vibration failure in the dovetail or wheel rim. The long shank and tip shroud remain remarkable innovations in vibration suppression.

The development of turbine stages which are vibration free requires sophisticated interaction between the aerodynamic design and testing disciplines. For free-standing buckets the calculation of frequencies is relatively routine; however, the amplitude of vibration response of buckets to aerodynamic stimulus is not easily determined without extensive test correlations. When the complexities of variable boundary conditions at platform and tip shroud are introduced into the assembly, analytical predictions become even more uncertain. Extensive test experience is required, therefore, to produce a reliable design.

Several test techniques are used to ensure adequate margin against vibration. For simple stationary-bench testing, a bucket is mounted on a heavy mass and driven at natural modes by a harmonic external force. Such tests provide useful data on expected modes, frequencies,

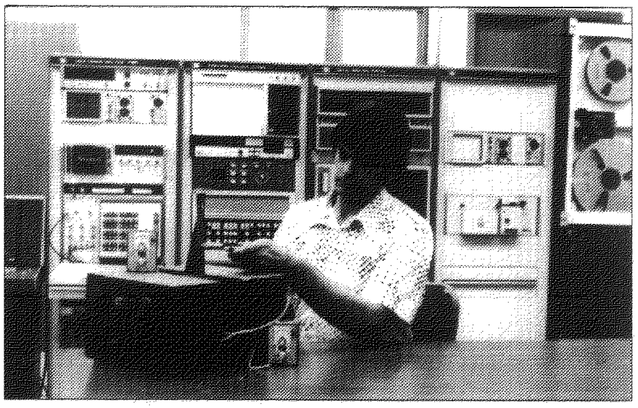

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Figure 30. Fourier analysis of bucket impulse excitation

and optimum strain gauge locations in preparation for wheelbox tests. More extensive information on bucket mode shapes is determined by Fourier analysis of impulse excitation (Fig. 30).

Wheelbox testing is one of several important steps used to produce reliable turbine designs. The wheelbox (Fig. 31) is one of the major test facilities in GE's Gas Turbine Development Laboratory. It is a large evacuated chamber in which full turbine stages are run throughout the operating speed range in order to determine bucket vibration response. Gas-force excitation is simulated by an array of nozzles which direct high-velocity air jets at the buckets. This facility is capable of handling the full range of rotor sizes produced and can operate over a speed range of zero to 7,500 rpm. Vibration data from

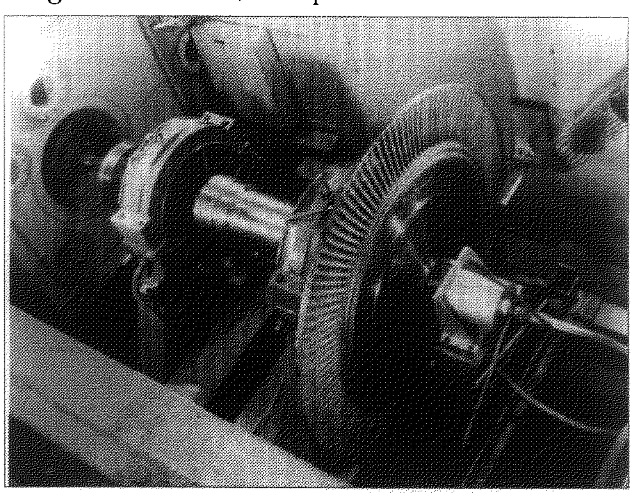

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Figure 31. Wheelbox facility

strain gauges mounted on the buckets is fed through slip rings into the processing facility, where both tape recordings and on-line analyses are accomplished.

MATERIALS

Design Stress and Material Properties

The nature of the design process requires serious consideration to the relationship between predicted machine conditions such as stress, strain, and temperature, and the capability of the component materials to withstand those conditions. Engineers will utilize the most appropriate analytical methods and the most precise mechanical and thermal boundary conditions in the design effort. They will then modify the analytical results by factors of safety, correlations, or experience to arrive at the specific value for stress and temperature for assessing

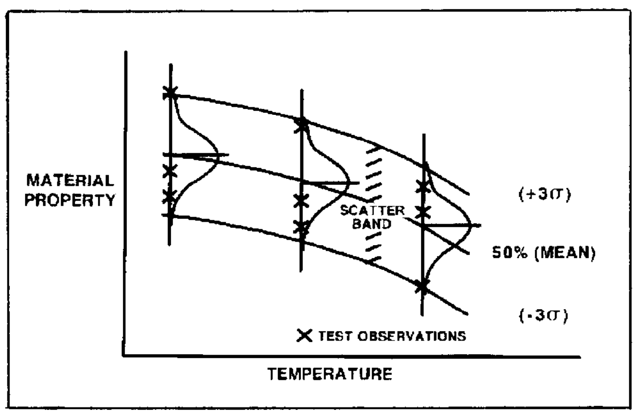

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Figure 32. Statistical nature of material properties

component life. This value is understood to be a reasonably close and conservative approximation. It is of particular significance that this value is specific, and that it becomes the standard against which the design and materials are measured to judge acceptability.

Figure 32 illustrates the variability of material properties. If many tests are run at a specific temperature, a scattering of the property about some mean value is noted. It should also be noted that there is finite probability (generally greater than 5%) that values for the measured property can fall outside of the scatterband of actual data. This characteristic of material properties requires the engineer to determine just what value of the properly will be used to judge the acceptability of the design. Should the average value, the lower scatterband value, or some other value be used? It is clear that a proper and detailed understanding of the properties possessed by the materials of construction is required if a component is to be properly designed.

The GE gas turbine designer goes to great lengths and considerable expense to develop information similar to that shown in Fig. 32. More than ten million dollars over the past 20 years has been invested to develop a large body of data so that the behavior of the critical materials of construction can be described with considerable confidence. In characterizing a material property, our practice is to obtain data from several different heats, to account for chemistry variations; from several heat-treat lots, to account for heat-treat variables; and from several sources (cast-to-size bars, test slabs, and actual parts), to account for grain size and other part-related variables. Once all this is accomplished, a material property value is typically selected so that at least 99% of the sample at a given temperature will have a greater strength than that utilized for life prediction. This prudent approach in evaluating life is the foundation of ensuring reliability of the product.

Since the general nature of material behavior variability has been addressed, it is appropriate now to discuss several specific material behavior topics that are significant to the gas turbine design engineer and the user. Discussion of these topics, creep/rupture and fatigue— will aid the operator in understanding the operating and repair options associated with the gas turbine, especially in nozzle, bucket, and combustion hardware.

Creep/Rupture

Fig. 33 represents the classic creep/rupture strain-versus-time relationship characteristic of metallic materials. This characteristic is important whenever a material is operating under stress at temperatures greater than 50% of the melting temperature (measured on the absolute scale), as is the case with the high-temperature components of a gas turbine. The designer historically has utilized data such as those portrayed in Fig. 33 to establish the design criteria. If distortion was important (as in a nozzle deflecting downstream into the buckets), a creep strain criterion would be chosen. If actual

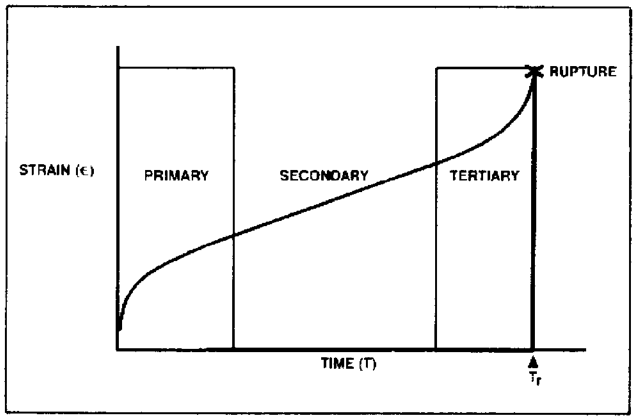

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Figure 33. Strain accumulation during the standard creep test (constant stress and temperature)

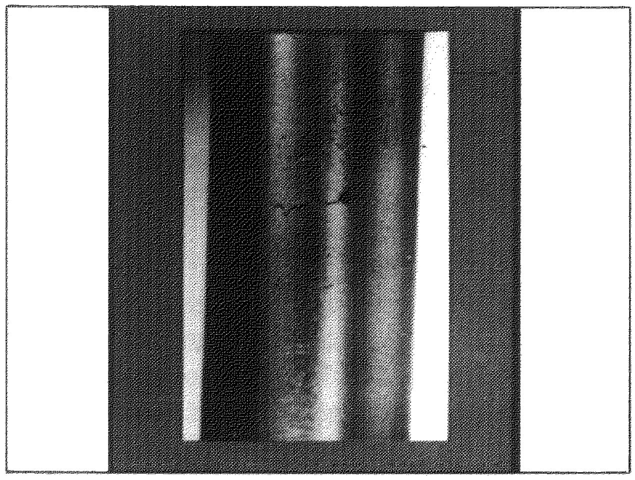

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Figure 34. Surface cracking in IN-738 (after 1.2% creep strain at 732C, 1350F)

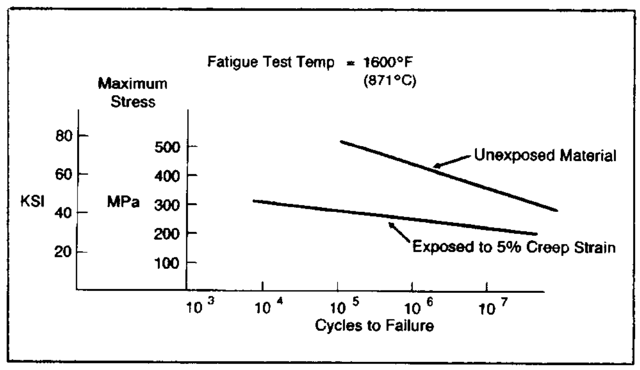

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Figure 35. Effect of pre-exposure in air on 871C (1600F) high-cycle fatigue life of cast IN-738

separation was important (as in bucket vane separation), then time to rupture would be the chosen criterion.

Research by GE gas turbine materials engineers has shown that rupture time, shown in Fig. 33, is not in itself a failure criterion. Figure 34 illustrates the degree of cracking developed in the cast nickel-base superalloy IN-738 when it has accumulated 1.2% creep strain at 732C (1350F). This cracking developed well before actual rupture of the test specimen. We have observed that creep cracking develops in nickel- and cobalt-base superalloys at approximately the onset of the tertiary stage of creep (see Fig. 33). For this reason, a time-to-rupture criterion is not utilized when designing against failure; instead, a creep strain criterion is chosen to avoid creep cracking. This criterion follows from the recognition that multiple loading modes occur in a gas turbine, and that creep-induced damage has a deleterious effect upon fatigue life, as illustrated in Fig. 35.

Thermal Fatigue

Thermal fatigue is the single most frequent cause of machine repair or failure, and understanding it requires substantial analytical, experimental, and metallurgical effort. Cracking and crack-induced failures of nozzle and combustion hardware are prime examples of this phenomenon. Thermal fatigue-induced cracking finds its genesis in the operationally induced transient and steady-state gradients that are most generally associated with cooled hardware. Neither can be eliminated, but their impact can be mitigated by judicious design and careful operation.

Figure 36 illustrates a typical nozzle vane pitch cross section with lines of constant temperature superimposed. The significant consideration is the thermal gradients in the part in combination with the temperature. Both the thermal stress and the temperature associated with this gradient cause fatigue damage during both transient and steady-state operation. Thus, this gradient must be evaluated with much care in order to achieve an acceptable design. Figure 37 illustrates a strain-versus-temperature trajectory for a cooled part after normal operation of a gas turbine from start-up through full load to shutdown. Note that the maximum strains do

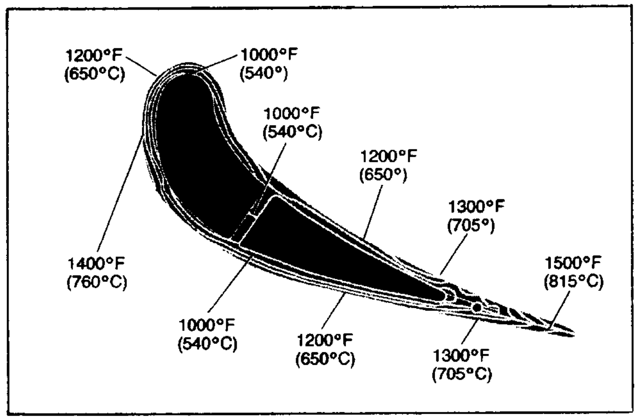

Figure 36. Cooled nozzle vane showing isotherms (typical)

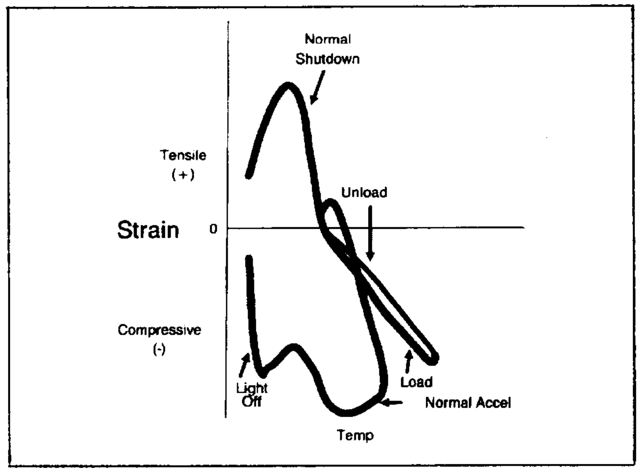

Figure 37. First-stage bucket leading edge strain/temperature variations (normal start-up and shutdown)

not coincide with the maximum temperature of the cycle. For this reason, complex material-testing procedures must be utilized to properly understand the thermal fatigue requirements of a given design and control sequence.

Start/Stop Transient Effects

The control functions provided with the GE gas turbines are set to limit the impact of the start/stop cycle. The duration and severity of light-off spikes are controlled so that only low strains develop in turbine components without impeding light-off and cross firing. Acceleration and fired shutdown functions are also designed to have a minimum impact upon part life. Great effort has been expended to understand the impact of start/stop cycles on cyclic life. Field tests on an MS5002 unit and the MS9001E prototype incorporated a variety of start/stop characteristics to explore their impact upon cyclic life. Fully instrumented hot section components were incorporated to provide experimental correlation. The results of these efforts clearly demonstrated that the major deleterious cyclic effect is caused by machine trips, especially trips from full load. Fig. 38 compares the impact upon strain range for a normal start/stop cycle with a cycle containing a full-load trip. While a full-load trip is not catastrophic in itself, the resultant life reduction is equivalent to that of approximately 10 normal shutdowns. A reduction in fatigue life by a factor of 10 is substantial

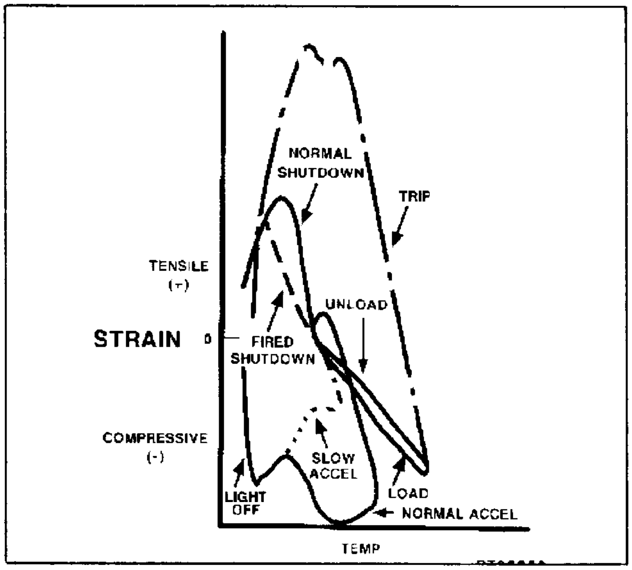

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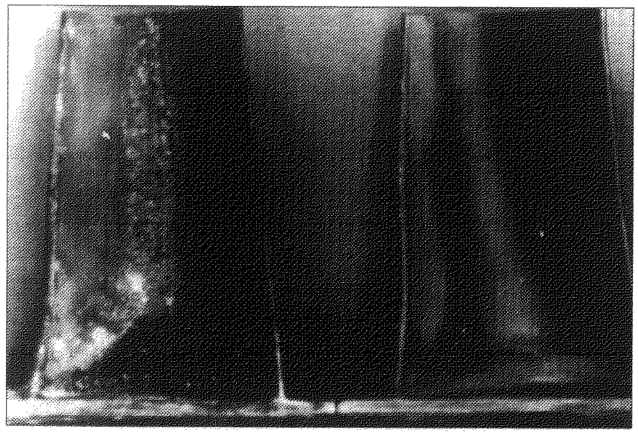
Figure 38. Leading edge strain/temperature (first-stage turbine bucket)

and certainly warrants careful and detailed attention to those machine factors that cause trips, especially the control, fuel, and auxiliary systems. Slowing the acceleration adds an additional 60% on the fatigue life of nozzles and buckets.

Corrosion Resistance Development

A two-pronged program was implemented in the 1970s to improve the corrosion resistance of the buckets. The first approach was to increase the corrosion resistance of the base alloy itself, while still satisfying strength requirements. This program resulted in introduction of IN-738. The second program was the development of the first generation of long-life coatings. In the mid to late 1970s, platinum-chromium-aluminide diffusion-type coatings were introduced. These alloy and coating improvements have increased corrosion resistance ninefold over the base alloy used in the late 1960s, and they have increased the range of permissible fuels.

Coated and uncoated IN-738 buckets are shown in Fig. 39. These two buckets were run simultaneously in an MS5002 located in the Arabian desert, one of the most corrosive environments in the world. These buckets operated for 24,731 hours in a unit burning sour gas with



Uncoated PtAl Coated RDC26882

Figure 39. First-stage turbine buckets (coated and uncoated IN-738 – 25,000 service hours)

3.5% sulfur. The terrain surrounding the site contains up to 3% alkali metals which frequently contaminate the inlet air during dust storms. In this environment the Pt-Cr-Al coating doubles the corrosion life of the IN-738 bucket.

Although the combination of IN-738 and Pt-Cr-Al coatings has offered a substantial improvement in corrosion resistance, improvements continue in first-stage bucket materials and manufacturing processes, with the intent of producing machines of increased performance capability and greater fuels flexibility. Two recent developments have been phased into production, the first is Vacuum Plasma Spray (VPS) coatings, the second is GTD-111 bucket alloy.

The GE patented vacuum plasma spray coatings are overlay-type coatings, which offer better control of coating composition than diffusion coatings. These coatings were laboratory tested for mechanical strength and corrosion resistance and also rainbow field tested where a number of coatings were run side by side on the same machines for comparative evaluations. All of these data established that VPS coatings are extremely attractive for improving bucket corrosion resistance. With full qualifications of this process, GE has introduced this coating into first-stage bucket production. VPS coatings were further improved by the addition of an aluminide coating to both external (airfoil) and internal (cooling) surfaces for first stage MS6001, MS7001, and MS9001 buckets (all models). The aluminide layer improves oxidation resistance.

Improvements continue in bucket alloys, the most recent of which is GTD-111 in equiaxed, directionally solidified, and single crystal forms. This alloy increases metal temperature capability with equal or better strength than IN-738 and displays comparable corrosion resistance. Much of the development work on this alloy was done in the late 1970s, and it is now our standard first-stage alloy for all designs; in the MS6001FA, MS7001FA and MS9001FA it is used on all three stages.

Mechanical Properties of Coatings and Substrate

Much analysis has been done toward understanding the effect of our VPS coatings on substrate mechanical properties. It has been determined that these coatings have little or no effect on substrate tensile or creep behavior. Vacuum plasma spray coatings have their largest impact on low-cycle fatigue (LCF). The GE-patented coatings can, in some cases, cause 2-to-1 life improvements compared to similar uncoated materials, as shown in Fig. 40. Without exception, life improvements have been observed in cases where the VPS coating exhibits superior ductility. Optimizing corrosion resistance of

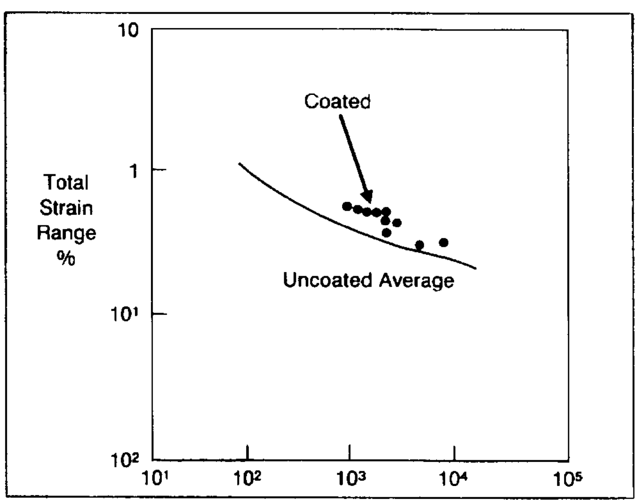


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Figure 40. IN-738 low-cycle fatigue at 1600F (871C)

coatings does not lead easily to ductile compositions. Current and future work is aimed at overcoming this obstacle by identifying coating compositions which have high corrosion resistance while maintaining acceptable levels of ductility.

PROTOTYPE TESTING

The history of instrumented testing under loaded conditions began in 1965 with the MS5001 at the Schenectady plant outdoor test site and again in 1968 with the MS3002 on the factory load test stand. This was followed by a fully instrumented MS7001A prototype unit tested at the LILCO Shoreham utility site. In 1971, at the compressor load facility in the Greenville plant, the MS7001B was tested with over 1300 channels of instrumentation. Again, in 1974 and 1976, this facility was used for testing the MS7001C and the MS7001E with comparable instrumentation. In 1979-80, prototype testing of the MS6001 was accomplished with two instrumented units. One had limited stator sensors and was tested in Montana at a Montana-Dakota Utility Company site, and the other unit, with almost 2200 channels of instrumentation, was load tested at Schenectady. In the span of one and a half years of testing, the unit achieved 235 fired starts and over 281 fired hours of operation while generating over five million kWh of electricity. The MS9001E design was tested at a customer site in Germany in 1980 and 1981. In 1982, the second prototype was tested at an Electricity Supply Board Company site in Dublin and at customer sites in Germany and Ireland.

During the 1980s, the design of the MS7001F gas turbine was supported by a three-phase test program:

- Phase I – Fundamental studies and component tests
- Phase II – Factory prototype tests
- Phase III – Field prototype test

The Phase I effort included the development and application of advanced analytical methods and computer techniques to accurately predict three-dimensional viscous fluid dynamics, boundary layer heat transfer, dynamic response of blading, dynamic response of complex systems, and complex material behavior. Where practical, the results of these advanced analytical tools were checked on models and components to ensure the accuracy of the predictions. Examples include hot cascade testing of the first-stage nozzle; liquid crystal studies of the first-stage nozzle and bucket to verify heat transfer assumptions; flow testing of the rotor cooling circuit and other components; materials behavior testing under calculated strain/time/temperature cycles; dynamic response wheelbox testing of all turbine buckets; exhaust system flow testing; and maintainability studies. A major effort permitted complete and thorough development of the combustor prior to actually operating the machine. Field testing of selected materials and configurations was included in Phase I to gain manufacturing and operating experience.

Phase II was largely aimed at verifying the compressor performance and obtaining component and system performance and operating data. During this phase, a full compressor map was developed, including surge margin. Also during this phase, extensive rotor and stator instrumentation was included to measure temperatures, pressures, hot-gas-path profiles, blading dynamic behavior, and system dynamic behavior.

The Phase III test involved a full-load test at a customer site. The primary objective was to verify all design and performance parameters. Metal, cooling circuit, and gas path temperatures; cooling circuit and cycle pressures; and component and system dynamic behavior were all determined under both transient and steady-state conditions. Cycle and emissions performance were also determined under normal steady-state conditions.

Each of the component and system data bases developed during Phase II and Phase III were compared with the analytical predictions before the MS7001F design was fully validated for commercial application. This testing involved investigations with firing temperatures of 1288C (2350F), justifying the uprate of the MS7001F at 150 MW to the MS7001FA at 166 MW. It also justified the MS9001FA rating of 226 MW and MS6001FA rating of 68.8 MW.

A similar sequence of prototype testing has been completed for the MS9001F. The first prototype machine was tested at Greenville in 1991. It is now commercial at Electricite' de France (EDF) in Paris, France having completed its full load, fully instrumented prototype tests in 1992.

"F TECHNOLOGY OPERATING EXPERIENCE"

As of June 1994, the MS7001F prototype unit at Virginia Power has accumulated more than 22,000 hours in combined cycle operation with a reliability level of 98%. Twenty additional 7F technology units are now in service, yielding similar performance. Clearly this experience has been due in large part to the stability and quality of the design process used to create this family of gas turbines.

SUMMARY

Reliable heavy-duty gas turbines have resulted from GE's design philosophy, based on a firm analytical foundation and the experience of years of gas turbine operation in the field. On this basis, successful designs are carefully scaled to larger or smaller size. Scaling has been used to produce similar designs that range from 25 to 200 MW. Evolution of proven designs has resulted from improved components and materials which have been applied prudently and carefully to increase power and thermal efficiency. Finally, designs are carefully tested and demonstrated in extensive development facilities, and by fully instrumented prototype machines in order to provide full confirmation of the design under actual operating conditions.

LIST OF FIGURES

LIST OF TABLES

For further information, contact your GE Field Sales Representative or write to GE Power Generation Marketing

GE Industrial & Power Systems

General Electric Company
Building 2, Room 115B
One River Road
Schenectady, NY 12345

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