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EVOLUTION OF HEAVY-DUTY POWER GENERATION AND INDUSTRIAL COMBUSTION TURBINES IN THE UNITED STATES

A. J. Scalzo and R. L. Bannister
Power Generation Business Unit
Westinghouse Electric Corporation
Orlando, Florida

M. DeCorso[1]
Media, Pennsylvania

G. S. Howard[1]
Charlottesville, Virginia

ABSTRACT

This paper reviews the evolution of heavy-duty power generation and industrial combustion turbines in the United States from a Westinghouse Electric Corporation perspective. Westinghouse combustion turbine genealogy began in March of 1943 when the first wholly American designed and manufactured jet engine went on test in Philadelphia, and continues today in Orlando, Florida with the 160 MW, 501F Advanced Combustion Turbine. In this paper, advances in thermodynamics, materials, cooling, and unit size will be described. Many basic design features such as two-bearing rotor, cold-end drive, can-annular internal combustors, CURVIC[2] clutched turbine discs, and tangential exhaust struts have endured successfully for over 40 years. Progress in turbine technology includes the clean coal technology and advanced turbine systems initiatives of the U.S. Department of Energy.

HISTORICAL PERSPECTIVE

Westinghouse, a major supplier of steam turbines since the beginning of the 20th century (Bannister and Silvestri, 1989), obtained the experience it needed to develop land based power generation combustion turbines through its entry into the jet engine business. In March of 1943, the first wholly American designed and manufactured jet engine went on test at Westinghouse, 15 months after obtaining a contract from the U.S. Navy. The engine (designated WE19A) had a thrust of 1130 lb (513 kg) and weighed 827 lb (375 kg). Without any knowledge of German, British or other United States activity, Westinghouse developed the first American jet engine with an axial compressor, an annular combustor, a turbine and a jet exhaust nozzle. The device resembled the Whittle engine developed in England, but there were major differences, in that an axial flow compressor was used, along with a submerged combustion chamber.

An improved version of the engine, the WE19B, was test flown at Patuxent Flight Test Center in January 1944 as a booster unit on a Chance Vought Corsair. It delivered 1365 lb (620 kg) of thrust, weighed 731 lb (332 kg), and one year later, as the J30, was used to power the Navy's first jet fighter, the McDonnell Douglas FH-1 Phantom. Sixty-one (61) Phantom planes were equipped with the J30 engine.

The J34, a 34-in. diameter engine that delivered 3000 lb (1362 kg) of thrust, was the last production aero engine built by Westinghouse. It was used extensively by the Navy in the McDonnell Banshee. Overall, Westinghouse supplied the engines for 1223 Navy jets before exiting the business in 1960.

Westinghouse's experience with land based gas turbines started in 1945 with the development of a 2000 hp gas turbine generator set (W21) that consisted of a single reduction gear, compressor, 12 combustors and turbine (Putz, 1946). A thermal efficiency of 18% lower heating value (LHV)* was obtained. By 1948, Westinghouse built a 4000 hp gas turbine locomotive with the Baldwin Company that used two of these units. Initial operation was on the Union Railroad burning distillate fuel oil. Later operation was on the Pittsburgh and Lake Erie Railroad using residual oil fuel.

In the late 1940s, the U.S. Navy wanted to develop a back-up for the nuclear submarine power plant. Westinghouse received a contract designated "Wolverine" to develop a gas turbine engine for surface, snorkel, and submerged operations. A gas turbine engine of 7500 shaft horsepower was designed to operate in open, semi-closed, and closed cycle modes. In the semi-closed mode, oxygen was supplied to the cycle from the snorkel. While in the submerged mode (fully closed cycle) oxygen was supplied and CO_2 gas was discharged overboard. Engine testing was done at the U.S. Naval Test Laboratory

[1] Westinghouse Electric Corporation Retiree
[2] Gleason Works

*All efficiency values in this paper will be LHV.

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in Annapolis, MD during the early 1950s. While the power plant met all contract requirements, the success of the nuclear submarine program eliminated the need for "Wolverine."

By 1954, gas turbine power plants were becoming a practical and economic reality for particular applications in the power industry. Westinghouse was offering 3 basic gas turbine designs for power generation which had a rotor inlet temperature of 1350°F (732°C) (Putz, 1954). The units were rated at 3500 kW, 5000 kW (with a regenerator), and 15,000 kW (with a regenerator and intercooler). The latter unit was designed for a full-load simple cycle efficiency of 29% compared to the full-load efficiency of 21% for the 5000 kW unit.

West Texas Utilities helped pioneer combustion turbine base-load generation with a 5000 kW unit that was installed in 1952 (Cox, 1957). In 1954, another 5000 kW combustion turbine was installed. The waste heat from this unit was used for feedwater heating until 1959 when it was incorporated into a fully-fired combined-cycle system. This system generated 39 MW in the steam cycle and 5 MW in the combustion cycle.

The Westinghouse single shaft, two bearing, combustion turbine with a reduction gear was first introduced in 1952 as the W81 at 5.5 MW (Stephens, 1952). In 1967, the W251A introduced major innovations such as minimum peak stress turbine blade roots, and cooled low alloy steel turbine discs. Today there are over 300 gear driven units operating in both simple, combined, and cogeneration cycle modes with the current W251 B12 rated at 48 MW.

The first direct drive combustion turbine (W201) at 20 MW was introduced in 1960. A modified W201 combustion turbine was used in 1960 as part of a blast furnace system at U.S. Steel's Chicago Works. The successful development program included design, development, and manufacture of a special casing configuration to efficiently remove compressor discharge flow and deliver it externally to the engine. (Krapf and Stephens, 1958). Gases from the blast furnace were burned and ducted back to the turbine through a special duct and scroll.

In 1962, Westinghouse designed the first "packaged" combustion turbine power plant, which was delivered to the City of Houma. The "packaged" power plant concept is retained today in the ECONOPAC[3] system. This simple cycle package includes the combustion turbine engine assembly; generator and exciter; starting package; inlet and exhaust systems; and plant auxiliary equipment. It is constructed in modules for easy shipment and installation. The ECONOPAC system is also the base for cogeneration and other heat recovery applications.

In 1967, a supercharged direct drive W301 at 25 MW in a heat recovery application was installed at West Texas Utilities San Angelo's Power station and achieved over 39% in combined cycle efficiency, the highest in the U.S. for a number of years.

The first direct drive W501 combustion turbine went into service in 1968 at the Dow Chemical Company with a rating of 42 MW. Many basic design innovations were introduced on the W501A including unique cooled, filtered rotor cooling systems and tilting pad bearings. By 1974, fifty-seven 501s were operating. The 501 technology has grown steadily over 25 years. The current production engine, W501D5 is now being offered with an ISO rating of over 118 MW and the 501F is at 160 MW. The "F" technology includes increases in airflow and firing temperature, improved component efficiencies, and advances in materials, turbine cooling, and dry low NOx. Currently, there are about 300 direct drive units in operation. A complete listing of all models sold is shown in Table 1 where total service hours is seen to exceed 47 million.

For over 30 years, Westinghouse has had licensee arrangements with several combustion turbine manufacturers, the major ones being Fiat Avio in Italy and Mitsubishi Heavy Industries (MHI) in Japan. These partners have contributed to the development of the combustion turbine with Westinghouse, particularly in the 50 Hz market; i.e., the TG50 of Fiat Avio and the MW 701 of MHI, 50 Hz versions of the Westinghouse W501 Model. During the late 1980s and early 1990s, a tri-lateral alliance was formed between Westinghouse, Fiat Avio and MHI to replace the old licensee/licensor arrangement. The alliance provides for joint development of combustion turbines and sharing of technology. The 501F and the 50 Hz 701F are the first engines developed under this partnership arrangement.

WESTINGHOUSE ENGINE TECHNOLOGY EVOLUTION

Westinghouse model designations initially meant the horsepower output in thousands followed by the number of shafts with letters denoting whether the engine was intercooled, regenerated, reheated, or extracted. Therefore, W201RE would mean 20,000 hp, single shaft, regenerated, with cycle air extraction. Later engine evolution, accelerated by materials and cooling improvements, invalidated such designations.

It should be noted that in this paper, the term "firing temperature" for uncooled engines will be defined as the average temperature leaving the combustors and entering the turbine. This neglects the effect of turbine inlet leakage and first stage vane cooling and will be confined to first and second generation engines. For all engines discussed in the third through fifth generations, rotor inlet temperature (RIT) will be used to define the temperature level. RIT is the mixed out average temperature entering the first rotor blade. The basic reason for the change is to include the effects of cooling air used by the row 1 vane.

Westinghouse combustion turbine evolution will be presented as follows:

First Generation - This period of the mid 1940s was heavily influenced by jet engine and steam turbine philosophies and will be confined to the W21 engine.

Second Generation - This period spanned from the late 1940s to the mid 1960s and generated many major design innovations that have been retained in our present designs.

Third Generation - The "Northeast Blackout" of 1965 initiated this generation that saw an explosion of design effort and introduced air cooling, higher firing temperatures, improved materials, and precision cast components. In addition, innovative concepts such as cooled, filtered rotor cooling air and rotor blade cooling and low alloy turbine discs, were introduced.

Fourth Generation - The fourth generation covered the 1970s and the 1980s, a period that was severely impacted by the "Oil Embargo" and the Fuel Use Act. Essentially the combustion turbine market disappeared until the Fuel Use Act was repealed and the Public

[3] Westinghouse Electric Corporation

TABLE 1. WESTINGHOUSE SUPPLIED COMBUSTION TURBINES

Model	No. of Units	Rating (kW)	Installed Capacity (kW)	Oper. Year	Mech. Drive	60 Hz	50 Hz	Service Hours*
W21	5	1,300	6,500	1949	3	5	0	500,000
W31	6	2,200	13,200	1956	6	0	6	600,000
W41	8	3,100	25,000	1961	4	7	1	800,000
W52	14	3,800	52,400	1956	2	14	0	1,400,000
W62	15	5,500	83,000	1961	13	15	0	1,500,000
W72	17	6,300	107,100	1961	17	1	16	1,700,000
W81	30	5,500	155,400	1952	17	30	0	3,070,345
W82	12	6,000	72,300	1962	12	0	12	1,200,000
W92	29	8,200	231,500	1960	27	29	0	2,900,000
W101	83	7,450	607,600	1961	54	70	13	8,306,630
W121	2	8,950	19,000	1959	1	2	0	200,000
W122	1	10,215	10,200	1967	0	1	0	100,000
W171	39	16,425	509,900	1961	1	28	11	2,184,371
W191	182	17,300	3,044,500	1965	14	127	55	8,596,174
W201	3	18,650	62,600	1960	0	3	0	296,022
251	195	49,100	5,626,700	1967	1	147	48	4,322,303
W301	32	23,850	938,100	1964	0	27	5	1,638,369
W352	7	27,900	190,900	1979	4	7	0	665,227
501	227	159,000	13,929,466	1968	0	227	0	7,192,050
701	8	136,900	1,095,200	1992	0	0	8	0
TOTAL	915		26,780,566		176	740	175	47,171,491

* Service hours are estimated as of 12/31/92

Regulatory Policies Act of 1978 (PURPA) was upheld. The latter part of this generation saw the introduction of increased firing temperatures and improvements in materials, cooling and efficiency. The W501D5, introduced in 1983, became the largest and most efficient 60 Hz single shaft combustion turbine in the world at 100 MW and over 33% efficiency for a simple cycle plant.

Fifth Generation - This generation covers the late 1980s to the present and starts with the 501F advanced combustion turbine with an RIT of 2300F (1260°C). Units in this generation will extend the air flow limits of last row turbine blades, the temperature limits of row 1 turbine vanes, and the flow limits of row 1 compressor blades through the use of advanced materials and cooling.

First Generation

The jet engine designers needed a high pressure, high flow facility, for combustor and turbine aerodynamic development. A lightly loaded 23-stage axial flow compressor was designed to meet the needs of the air supply facility. This test compressor together with a 12 can interconnected combustion system and an 8-stage turbine formed the first industrial combustion engine at 2000 hp. The air flow was 35.5 lb/sec (16.1 kg/sec) at a pressure ratio of 5, a firing temperature of 1250°F (677°C), and a power output of 2070 hp at an efficiency of 18 % (Mochel, 1947). The unit was a single shaft, three bearing design with a design speed of 8750 rpm. The external load was driven from the compressor end of the engine.

After a development period, the engine was first applied (1948) as a twin set in a locomotive built by Baldwin and Westinghouse (Stephens and Young, 1989). The engines burned either diesel fuel or residual oil. The locomotive was used as a demonstrator and ran on several railroads, but was never placed in commercial service. Although the locomotive operated at a cost approximately 5% less than a diesel locomotive, the capital costs were not competitive. The locomotive was subsequently scrapped, but both turbines were sold for further operation. The left hand unit was sold to the city of Larned, Kansas as one of the early decentralized peak shaving plants and the right hand unit became the world's first intercooled steam injected plant driving an air compressor at Hopewell, Virginia. The first base-load service of the engine was as a pipeline gas booster pumper on The Mississippi River Fuel Company System in March 1949. This was a dual fuel application burning either oil or gas with the capability of changing to either fuel at full load, an industry first. This particular engine, the first combustion turbine in the world to achieve 100,000 operating hours, was retired after more than 150,000 hours of operation when the gas line was retired.

W21 combustor baskets were contained in individual pipe-like enclosures connecting compressor discharge to turbine inlet, as shown in Fig. 1, and bore little resemblance to the Westinghouse aviation gas turbine design which was an annular combustor, but it used the wall cooling arrangement (wiggle strips) from the aviation design. The wiggle strip concept of wall cooling was unique; it interposed a corrugated strip of metal between two cylindrical sections of combustor that permitted the injection of cooling air, is structurally superior to a single cylinder because the increased section modulus provides superior buckling characteristics, and proved to be a fundamental cooling feature that is still being used in current Westinghouse turbines. Also, casings were horizontally split (like steam turbines) instead of the axial stacking used by the jets. In addition, late production units introduced the CURVIC coupled rotor structure, as shown in Fig. 2, to align discs and transmit torque.

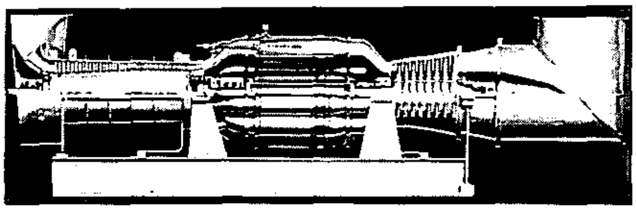

FIG. 1. CROSS SECTION OF W21 COMBUSTION TURBINE

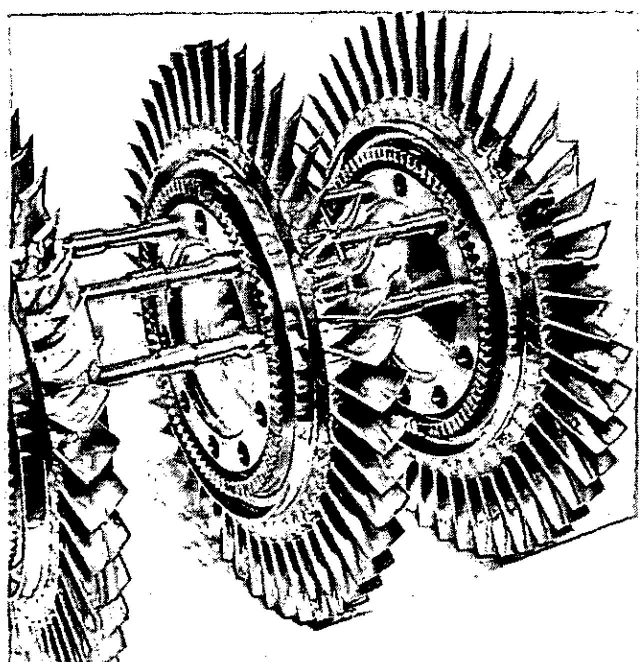

FIG. 2. CURVIC COUPLED TURBINE DISCS

Second Generation

In the late 1940s, Westinghouse formed a design group independent of the aircraft group to develop gas turbines for land based applications. Engineering studies indicated that with the high temperature materials then available a firing temperature of 1350°F (732°C) was feasible for base-load duty (100,000 hr life). In response to a market survey, the first engine developed was a 5000 kW single shaft, simple cycle engine. This engine had a design speed of 5740 rpm, an air flow of about 125 lb/s (57 kg/s) at a pressure ratio of 6.25.

The second generation engine design eliminated the undesirable center bearing and placed the combustion system inside a pressure vessel. Overall, this design had the following improvements:

- Elimination of center bearing removed a fire hazard and simplified shaft alignment.
- Compressor flow from the diffuser provided impingement and convective cooling of the transition duct.
- Flow into this plenum type enclosure attenuated flow disturbances to and from the compressor.
- The annular plenum effect provided damping of combustor pressure fluctuations (This became valuable in later years when limits of combustor stability were approached when using water injection for NOx control).
- Provided for improved cross-flame tube connections.
- Permitted the inclined position for the combustors and transitions, thus, facilitating the center bearing removal by shortening the span length.
- Combustors and cross-flame tubes completely encased in relatively cool compressor discharge air, thereby eliminating a multitude of pressure connections and potential fire hazard.

In addition to the combustion system changes, the 5000 kW (W81) combustion turbine, introduced in 1952, contained the following basic mechanical design features that are illustrated in Fig. 3:

- Individually replaceable turbine and compressor rotor blading without removal of the rotor.
- Inner shrouded diaphragm construction for turbine and compressor stationary blading to permit servicing with rotor in place.
- Support of the exhaust bearing housing 6 tangential struts that rotate the housing during thermal transients and maintain shaft alignment without the need of cooling.

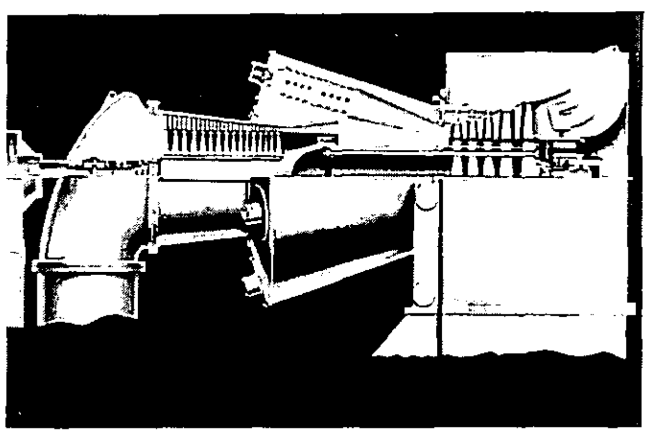

FIG. 3. CROSS SECTION OF W81 COMBUSTION TURBINE

The W81 16-stage compressor design was the result of extensive analytical and laboratory development. While the free-vortex design of the early compressor had the advantage of minimizing the effect of neglecting some terms in the Euler equations, the result was many stages at relatively low blade tip speeds. In addition, theoretical studies indicated that a design of 50% reaction with a flow coefficient of about 0.55 would give a stage of maximum efficiency. By setting up a pre-swirl entering the first rotating blade, the blade relative velocity could be significantly reduced and high reaction obtained. This would allow an increase in blade speed with the consequent reduction in the number of compressor stages required for a given pressure ratio. The vortex pattern selected was a compromise between little change in reaction radially and blade shock losses. This vortex pattern specified the mean stage absolute swirl constant and made the axial velocity relatively constant radially. A positive energy gradient was added from blade hub to tip in the first few stages, the energy addition was kept constant in the middle stages, and the gradient removed in the last stages.

Other design techniques used were to specify flow area blockage factors and to use NACA 65 blade sections on a circular arc mean line for the higher Mach number front stages and NACA (4-digit) sections elsewhere. The resulting design was successful except that the starting surge margin was inadequate; therefore, an interstage bleed was added. The average stage efficiency was about 89%.

The aerodynamic design of the 5-stage turbine was taken mainly from Westinghouse aircraft engine design practice using a free-vortex swirl distribution and constant work radially with a last stage designed for axial velocity outflow. A minimum reaction was maintained at the blade hub sections to prevent suction side separation. All blade sections were designed to provide smooth accelerating flow passages. The average stage efficiency was about 90% and the matching of the compressor and turbine flow characteristics was excellent. The engine produced 5700 kW at the generator terminals at a firing temperature of 1350°F (732°C) and a thermal efficiency of about 21%. The prototype engine, sold to West Texas Utilities, went in commercial service in 1952. This unit was the first utility application of a Westinghouse gas turbine.

Marketing studies indicated there were many differing applications for gas turbines from power generation, transportation, to mechanical drives. Engineering cycle studies were made to determine the best cycle arrangements for the various applications. A summary of those studies at a 1400°F (760°C) firing temperature is shown by Fig. 4. Component efficiencies, pressure losses, and heat exchanger effectiveness for these studies were state-of-the-art.

With the successful completion of the 5000 kW design, a line of mechanical drive engines using the same basic blading as the 5000 kW engine was designed. For pipeline pumping, a unit was designed with regeneration and a free power turbine (Stephens and Bruce, 1955). The regenerative engine produced about 5400 hp at the turbine coupling with an efficiency of approximately 30%. The cycle pressure ratio was 4.2, the compressor speed was 6000 rpm, and the firing temperature was 1350°F (732°C). The regenerative units did not perform in the field as expected, primarily because leakage and pressure loss were greater than expected. This same unit was also modified to be used without regeneration.

To make the engines more competitive with steam plants and other prime movers, it was necessary to use a more efficient cycle. As shown in Fig. 4, a gain of about 12% could be obtained by adding intercooling to the regenerative cycle resulting in 34% efficiency which made it competitive with small steam plants. A 15,000 kW engine was designed as a two shaft, intercooled, and regenerative engine with the load taken from the high pressure shaft. By taking the load from the 3600 rpm high pressure shaft, the low pressure (LP) shaft was made variable speed resulting in little drop off in turbine inlet temperature at part load and, therefore, excellent part load efficiency.

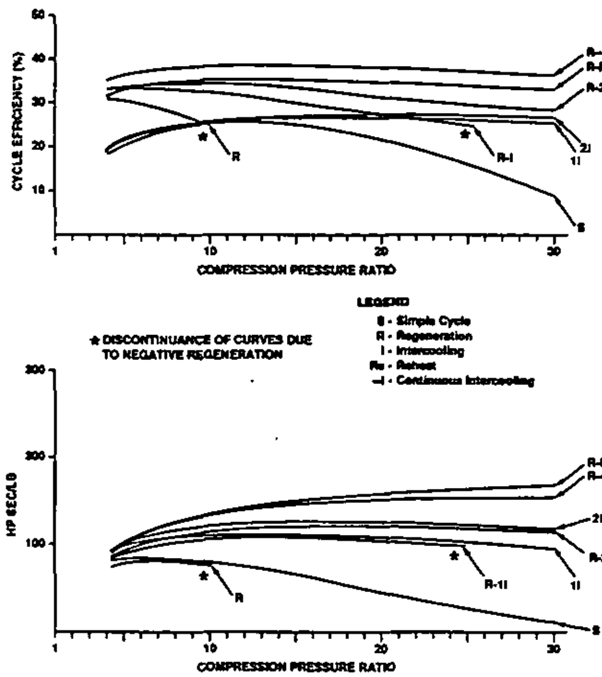

FIG. 4. SUMMARY OF ENGINEERING CYCLE STUDIES AT 1400°F

In 1954, prior to the cancellation of the 15,000 kW project, work was begun on a single shaft simple cycle 3600 rpm gas turbine. This engine was later to become the W201 gas turbine, the first direct drive (no gear) Westinghouse combustion turbine. In the W201, the aerodynamic design of the successful 5000 kW engine compressor was modified slightly by eliminating the energy gradient so that stages could be more easily added or removed from the basic design. In addition, lightly loaded stages for high efficiency were adopted for these engines. Design techniques were also improved by refining empirical factors. As high temperature materials advanced, the allowable turbine inlet temperature was increased, thus improving output and efficiency. By the end of the 1950s, firing temperatures had risen to 1450°F (788°C) at base load. Also, the W31 and the W121 at 3000 hp and 12,000 hp, respectively, both single shaft engines, were designed following the same philosophy developed in the W201.

In the middle 1950s, Westinghouse designed a blast furnace blower to supply 125,000 ft3/min (3540 m3/min) of air at about 2 1/2 atmospheres gauge pressure, and burning blast furnace gas at about 100 Btu/scf (3700 kJ/scm). The blower was a modification of the W201 in which the blast furnace air was extracted from the compressor discharge and the fuel gas supplied to the engine through a shaft driven fuel compressor. The fuel compressor was a modified compressor from the W31 engine. The engine and it's performance are shown in Figs. 5 and 6, respectively. Turbine blading, turbine discs, compressor discs, and recuperator were taken from the 15,000 kW unit. This was a very successful design from an engineering point of view, but the market disappeared when oxygen blown furnaces displaced the old air blown ones.

In the late 1950s, production for the direct drive W201 was initiated. The W201 was the forerunner of the W301 and the W501 line. Also, the W121 was improved to the W171 by eliminating the excess quote margin and increasing firing temperature to 1450F (788°C). In the early 1960s, a zero stage was added to the 14-stage

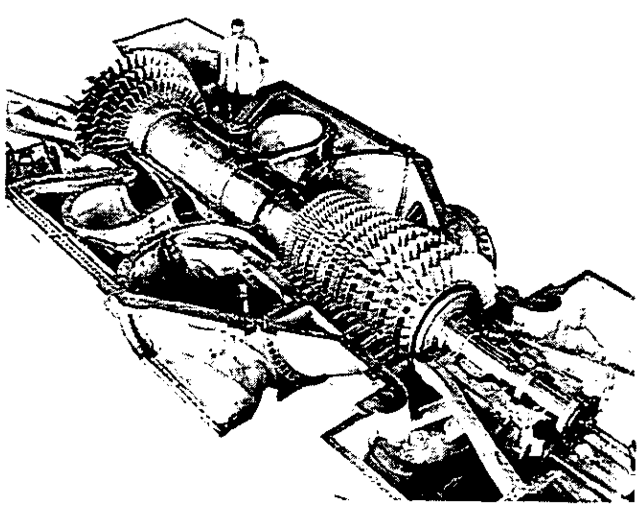

FIG. 5. BLAST FURNACE ENGINE

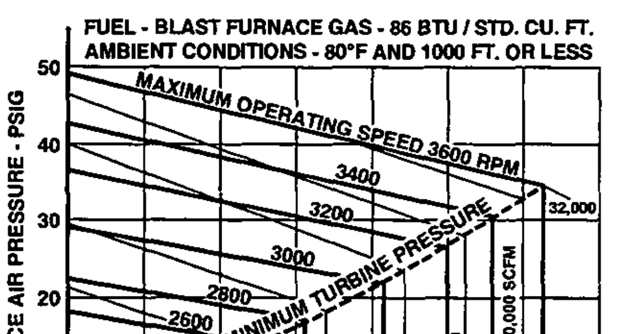

FIG. 6. PERFORMANCE OF BLAST FURNACE ENGINE

W171 compressor to create the W191 and the firing temperature of the W31 was increased to 1450F (788°C) to create the W41.

President Eisenhower's Administration started the "Atoms for Peace Program." An interesting outgrowth of this program was the "Marine Gas Cooled Reactors Program" (designated MGCR) that was expected to power a nuclear powered commercial ship. The power plant was to be a helium cooled reactor as the heat source, coupled with a closed cycled gas turbine. Westinghouse was selected to provide the turbomachinery for the power plant and the design proceeded to development testing and engine design. However, when the reactor development lagged, the program was canceled. The turbomachinery presented no severe aerodynamic problems and was actually much easier to design than air breathing machinery due to the fact that helium is a light monatomic gas and the speed of sound through it is very high. The engine was rated at 20,000 hp, at an expected efficiency of 31%.

Combined gas and steam plants had been developed as early as the locomotive engine, but the 1960s saw the development of the combined cycle. The combined cycle is an ideal thermodynamic solution to the thermal efficiency problem as it combines the best attributes of the Joule-Brayton cycle with that of the Rankine cycle; namely, the addition of heat energy at the high temperature of the gas turbine and the rejection of heat at the low temperature of the steam turbine. In 1967, a Westinghouse supercharged combustion turbine (W301), rated at 25 MW with a firing temperature of 1450F (788°C), was placed in operation at West Texas Utilities San Angelo's Power Station. Hot turbine exhaust gases were used in a heavy fuel, reheat boiler (1500 psi, 1000°F [10.3 MPa, 537°C]) which furnished steam to drive a 85 MW turbine (Cox, Henson, Johnson, 1967). The plant achieved an annual average efficiency of over 39%, the highest in the U.S. for a number of years. In 1990, this plant with over twenty-three years of successful operating experience, continued to demonstrate that good combined cycle efficiency over a broad load range can be obtained with a combustion turbine combined with a heavy fuel, reheat boiler (Stephens et al., 1990).

An interesting feature of this plant was that the forced draft fan of the steam power plant was used to supercharge the gas turbine. Not only did this feature increase the plant capability and efficiency, but the fan was used as a starting device to start the gas turbine by windmilling the compressor up to self-sustaining speed. In addition, the fan could supply the steam boiler while the gas turbine was out of service.

Third Generation

In November 1965, the "Northeast Blackout" ushered in the "Gas Turbine Age" as gas turbine manufacturers were swamped with peaking power plant orders. Development of gas turbines accelerated as technical staffs expanded. Blade and vane cooling, pioneered by the aircraft engine industry, became available to the industrial engine builders. In response to a need for larger, more efficient engines, Westinghouse developed the W251 and the W501 engines.

The 20 MW W251A, introduced in 1967, was a single shaft, two bearing, combustion turbine with reduction gear to accommodate either 60 Hz or 50 Hz for simple or combined cycle applications. Many basic design features proven reliable over many years were retained along with the two bearing, single shaft concept. Most noteworthy of these are the CURVIC coupled turbine discs, tangential exhaust bearing struts, blades and stationary vanes which are field removable with the rotor in place, and a compressor end drive. The evolution of the W251 is shown in Table 2.

The compressor of the W251A was taken from the W191, a rear stage added to allow an increase in engine pressure ratio, and a new 3-stage turbine designed with higher blade speeds to replace the old 5-stage turbine. The first stage turbine vane was cooled with compressor discharge air. The base load RIT was increased to 1575°F (857°C) from 1450°F (788°C) increasing the output power by 16% at constant heat rate. The first engine was placed in commercial peaking service at Detroit Edison in 1967. The W251A introduced the following improvements for the first time in a Westinghouse design:

- New 5-serration, minimum peak stress root design with root extensions to eliminate 3-D stress concentration.
- Precision cast turbine vane segments of cobalt-base material (X45) to eliminate structural weld concerns with the nickel-base welded diaphragm assemblies.
- Single turbine blade ring concept with roll out feature to provide service of turbine stationary parts with the rotor in place.
- Extensive use of turbine cooling including the row 1 vane segment, all three rows of turbine discs, and blade rings together with associated stationary flow path parts.
- Segmented ring segments over the rotor blade tips to allow free thermal expansion.
- Interstage seal housing supported from the vane segments on radially oriented keys to allow independent thermal growth of the seal housing to minimize running seal clearances.
- Use of cooled and filtered compressor discharge air for rotor cooling.
- Use of NiCrMoV turbine disc material.
- Compressor bleed extractions for selected cooling of turbine vanes and disc cavities to provide the lowest cost air for duty required.

The W251AA used the W251A compressor with an added front and aft stage to increase the airflow and pressure ratio while using the same turbine except improved row 1 vane cooling to permit a modest increase in RIT to 1640°F (893°C).

The W501 genealogy begins with the 30 MW W301 which began commercial operation in 1960. Unlike previous smaller, higher speed engines, the W301, like the W201, was directly connected to the generator without a gear.

Table 3 traces the development of the 501 family from the W501A to the current fifth generation 501F model. The W501A was designed to meet a 42 MW need of the Dow Chemical Company by utilizing

TABLE 2. W251 EVOLUTION

Engine	W191	W251A	W251AA	W251B	W251B2	W251B8	W251B10	W251B12
First Startup Date	1961	1967	1969	1971	1973	1978	1983	1990
Power, MW Class	18	20	26	31	34	36	40	48
Rotor Inlet Temp.,F	1450	1575	1640	1806	1806	1911	1985	2100
Air Flow, lb/s	270	270	353	353	353	353	345	372
Pressure Ratio	7	8	10	10	11	11	14	15
No. Comp. Stages	15	16	18	18	18	18	19	19
No. Turb. Stages	5	3	3	3	3	3	3	3
No. Cooled Rows	0	1	1	3	3	3	4	4
Exhaust Temp., F	777	850	850	930	898	962	941	990
Heat Rate, Btu/kWh; (ISO Gas)								
Simple Cycle	13,430	13,775	13,130	12,540	12,265	11,980	10,980	10,600
Combined Cycle	9,600	9,840	9,040	8,600	8,400	7,900	7,400	7,100

the proven, high performance, 15-stage W301 compressor, with an added front and aft stage to increase the air flow and pressure ratio, and a newly designed four-stage turbine to replace the old five-stage turbine. Dow required a cold end drive because their system design was a single train of combustion turbine, steam turbine, and generator with the steam turbine providing combustion turbine starting. Also, the hot end drive was undesirable because a flexible coupling was required. They also preferred two bearings which provided an excellent match with Westinghouse turbine design philosophy. In addition, the W501A introduced the following improvements:

- Variable inlet guide vane to provide exhaust temperature control on heat recovery applications and to improve starting characteristics.
- Individual blade rings with a roll-out feature to provide independent field service of each stator and seal system with rotor in place.
- Unique cooled, filtered rotor cooling concept utilizing "air separator" rotor component to confine rotor cooling air.
- Unique fore and aft blade seal plates to prevent cooling air leakage and service of blades with rotor in place.
- Four-pad, tilting pad bearings to eliminate any possibility of bearing instability from "oil whip" phenomenon.
- Compressor rotor utilizing a hollow shaft with shrunk on discs to reduce stresses and improve rotor dynamics.

The 60 MW W501AA in 1971 used a newly designed, higher flow, higher pressure ratio compressor with the W501A turbine at essentially the same RIT. Another major design change was a reduction in combustor shell diameter in order to facilitate shipping the engine fully assembled, a feature which has been retained on subsequent models. The combustor was set parallel to engine axis, resulting in an s-shaped transition duct. A secondary benefit of this system was a reduction in pattern factor or peak temperature due to increased mixing.

A significant improvement in engine performance was due to the new axial exhaust system. The axial exhaust system allowed waste heat recovery systems to be located directly downstream of the turbine exhaust, avoided high duct losses, and accommodated a longer and more efficient turbine diffuser.

Many engineering studies were made to determine the optimum cycle to minimize the cost of electricity. Some of the more promising ones (intercooled, multiple shafts, reheat, steam injected, water injected at various locations) were studied in great detail. In the final analysis, the simple cycle gas turbine combined with a steam bottoming cycle was the most widely accepted in the market. The missing link was higher firing temperatures. Fig. 7 is a summary of some of those studies. (The question of more efficient cycles is currently being re-visited in the advanced turbine systems program discussed later in the paper.)

As shown by Fig. 7, the combined cycle is superior to the regenerated cycle by a significant amount. The pressure ratio of the gas turbine that yields maximum specific power (power output per

TABLE 3. W501 EVOLUTION

Engine	W501A	W501AA	W501B	W501D	W501D5	W501D5	501F
First Startup Date	1968	1971	1973	1975	1981	1995	1993
Power, MW Class	42	60	80	95	107	118	160
Rotor Inlet Temp.,F	1615	1630	1819	2005	2070	2150	2300
Air Flow, lb/s	548	744	746	781	781	830	960
Pressure Ratio	7.5	10.5	11.2	12.6	14	14.8	14.6
No. Comp. Stages	17	17	17	19	19	19	16
No. Turb. Stages	4	4	4	4	4	4	4
No. Cooled Rows	1	1	3	4	4	5	6
Exhaust Temp., F	885	798	907	982	987	1006	1083
Heat Rate, Btu/kWh; (ISO Gas)							
Simple Cycle	12,600	11,600	11,180	10,925	10,270	10,040	9,590
Combined Cycle	9,000	7,990	7,350	7,025	6,950	6,900	6,600

pound of air flow) also gives optimum combined cycle efficiency. This is a most interesting conclusion and shows that for simple cycle gas turbine engine applications, a high pressure ratio engine is the economic choice, such as for aircraft propulsion. For the same firing temperature levels, the lower pressure ratio gas turbine engine is superior for combined cycle applications. Since the 1970s Westinghouse engines have been designed to be optimum for combined cycle applications and these engines are also excellent for peaking applications, where first cost dominates, since they yield the maximum power for a given amount of hardware. The figure also indicates the dominant role of turbine inlet temperature on both power output and combined cycle efficiency.

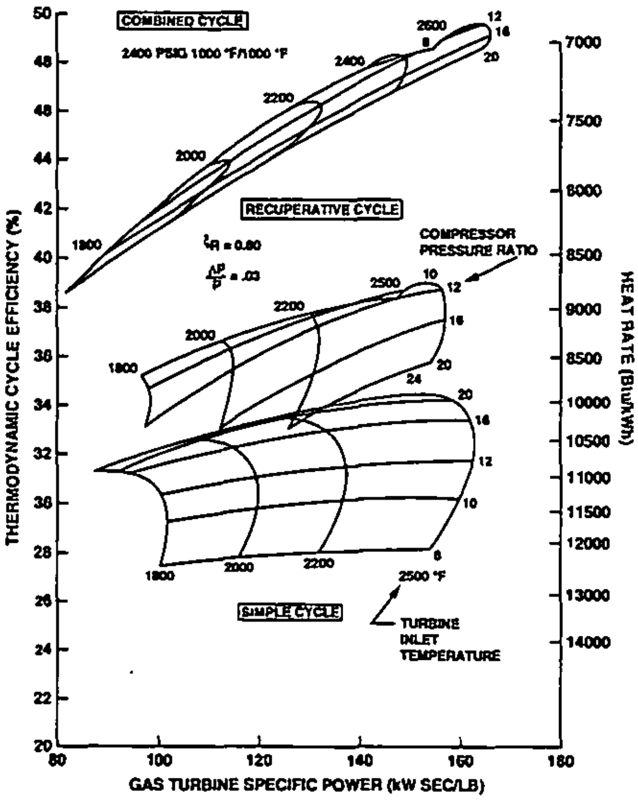

FIG. 7. SUMMARY OF COMBINED CYCLE STUDIES

Fourth Generation

In this era, the W251B used the same compressor as the W251AA, but with RIT increased to 1806°F (986°C). In addition to further improvement in row 1 vane cooling, the row 1 blade and row 2 vane segment were cooled for the first time in a Westinghouse design. At this point material changes were made such as the row 1 blade made as a casting for the first time with cooling passages integrally cast.

The W251B2 represented additional design improvements in the turbine that produced increased performance, while using the same W251AA compressor with no increase in RIT. The continuation of growth represented in this model consisted of adding a single vane with further improvements in cooling, restaggering of the turbine blade path for an increase in pressure ratio, improvements in cooling flow management and an improvement in row 2 vane cooling. The single vane was designed to permit service without removing the casing cover, an industry first.

The W251B8 used the W251AA compressor with performance improvement from an increase in RIT to 1911°F (1044°C) brought about by a redesigned combustion system for improved temperature profile into the turbine, and additional improvements in row 1 vane cooling. Also, the compressor used coated diaphragms for improved performance as well as to retard degradation from ingested contaminants, and rows 1 and 2 turbine vanes and blades were coated for corrosion protection.

The 80 MW W501B, in 1973, was a planned growth step of the W501 model with the principal change being an increase in RIT to 1819°F (993°C). In addition to increased row 1 vane cooling, cooling was added to the row 1 blade and row 2 vane and material was changed, as required, on other rows to compensate for higher gas temperatures. Provision was added for removal of combustors and transition pieces without a cover lift.

The 95 MW W501D was a continuation of the planned growth program for the W501 frame. A further increase in RIT to 2005°F (1096°C) was made possible by advances in turbine cooling technology. Two stages were added to the aft end of the W501B compressor for higher pressure ratio in order to optimize efficiency in combined cycle operations. Cooling was added to the row 2 turbine blade along with increased cooling in upstream stages and material changes as required.

The W501D5 achieved additional performance gains from improvements in component efficiencies, conservation of cooling air energy, and a modest increase in RIT. Compressor performance was improved by increasing the number of diaphragm seal points from two to four, by use of coated diaphragms with improved surface finish, and by restaggering airfoils to improve work distribution. Turbine performance was improved by reducing exit velocity and swirl, by reducing incidence losses, and by better stage work distribution. Stator cooling air was extracted from three compressor bleed points in order to use air at the lowest suitable pressure available. Improvement in detail design reduced internal cooling flow leakage by 12% compared to the earlier W501D. In addition to improving cooling of the first two vanes and row 1 blade, cooling was added to row 2 blade. Other features provided improved reliability and availability by virtue of improvement in parts' life and easier maintenance and inspection such as a single row 1 vane. In the compressor, the aerofoil-shroud joint was redesigned for increased strength in key diaphragm stages. The combustor cooling technique using the standard "wiggle strip" corrugation was improved by the extended lip, as shown in Fig. 8. This extension improved the tenacity of the convective heat transfer coefficient and reduced downstream temperatures by 200°F (111°C) (Scalzo et al., 1983).

The last engine models of this generation were the W251B 9/10 and the W501D5. Both engines used the same basic compressor, the W251 compressor being a 2/3 scale of the W501. In combined cycle applications the engine's potential efficiency approaches 50%. The power capability of these engines are more than 2 1/2 times their

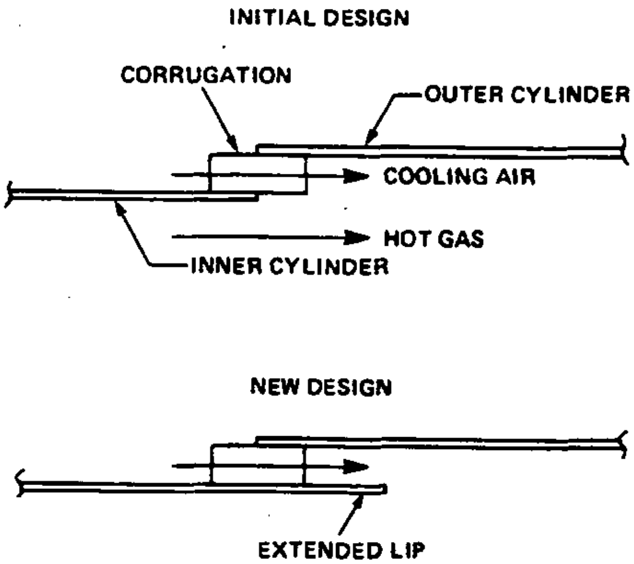

FIG. 8. EXTENDED LIP IMPROVED COMBUSTOR COOLING

third generation antecedents and the efficiency was increased by about 1/3.

The last model engines of this generation utilized streamline curvature calculation methods to solve the Euler equations, with energy loss models developed from a very large experimental data base. The result was compressors with average stage efficiencies approaching 93% and cooled turbines with excellent characteristics.

The power rating of a gas turbine engine is limited by the amount of flow that can be passed by the last stage blade row of the turbine component. Stress capability of the last row turbine blade material at operating temperature and aeroelastic blade limits determines the flow area of the last stage, while leaving axial velocity and density of the exhaust gases, in addition to flow area, determines engine flow. The next generation of engines, the F series and beyond, will be flow limited designs.

Fifth Generation

The fifth generation of Westinghouse gas turbine engines brings us to the present. These engines are designed with the aid of complete 3-D computer codes some of which solve the Navier-Stokes equations. The 501F at 160 MW and 2300°F (1260°C) firing level approaches 55% thermal efficiency in combined cycle applications.

The 501F, shown in Fig. 9, is a 3600 rpm heavy-duty combustion turbine designed to serve the 60-Hz power generation needs for utility and industrial service in the 1990s. It was jointly developed by Westinghouse and Mitsubishi Heavy Industries, Ltd. (Scalzo et al., 1989). Designed for both simple and combined cycle applications, it will operate on all conventional combustion turbine fuels as well as with coal-derived low-Btu gas produced in an integrated gasification combined cycle (IGCC) power plant.

The 501F rotor is of bolted construction supported by two, two-element tilting-pad bearings for load carrying and an upper half fixed bearing. This provides inherent stability of the tilting pad with the

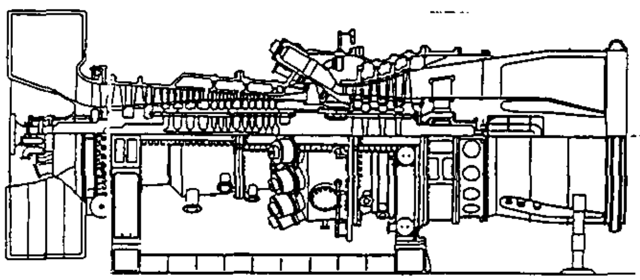
FIG. 9. GENERAL CONFIGURATION OF 501F COMBUSTION TURBINE

reliability of the plain bearing, thus eliminating the top pad fluttering problem that has led to local babbitt fatigue distress. The thrust bearing is a double-acting Kingsbury thrust bearing that uses leading edge groove (LEG) lubrication system.

The compressor rotor is comprised of a number of elements, spigotted and bolted together by 12 through bolts. The turbine rotor section is made up of disks bolted together by 12 through bolts and using CURVIC clutches. This turbine rotor design has amassed over 40 million hours of reliable service in all sizes of combustion turbines. The compressor is a 16-stage axial flow design of 14 pressure ratio that is based on the highly successful W501D5 compressor. A four-stage turbine was selected to maintain moderate aerodynamic loadings even at the increased firing temperature.

Blade rings have been added in the compressor for stages 7 through 16. Similar to those used in the turbine, they have a high thermal response independent of the outer casing, can be aligned concentric to the rotor to prevent blade rubs and minimize clearance, and therefore maximize performance as well as enhance maintainability of stationary parts.

Flow and pressure coefficients of the 501F compressor have been kept similar to the D5 compressor by increasing the mean diameter of the stages to accommodate the 20% increase in flow. In addition, the rear stages of the new compressor have larger diameters to help balance spindle thrust. Interstage bleeds are used for starting and for supplying cooling air to the turbine stationary blading and interstage cooling system. Rotor blades are double circular arc designs in the first four stages. The stators and all other rotor blades are conventional W65 airfoil sections.

The design of the 501F turbine has maintained moderate aerodynamic loadings in spite of the increased inlet temperature by choosing a four-stage turbine with higher peripheral speed compared to the W501D5. The 1st and 2nd stage blades are the free-standing type, while the 3rd and 4th stages utilize integral "Z" tip shrouds. The use of a shrouded system is a departure from past design practice on the 501 series, but has been in use on the W352.

The 1st turbine stationary row consists of precision-cast, single-vane segments. There are precision-cast, two-vane segments in the second turbine station row while the 3rd and 4th turbine stationary rows are precision-cast vane segments three-vane and four-vane segments, respectively.

Cooling circuits for the turbine section, displayed in Fig. 10, are similar to those used on the W501D5. Rotor cooling air is provided by compressor discharge air extracted from the combustor shell. This air is externally cooled and filtered before returning to the torque

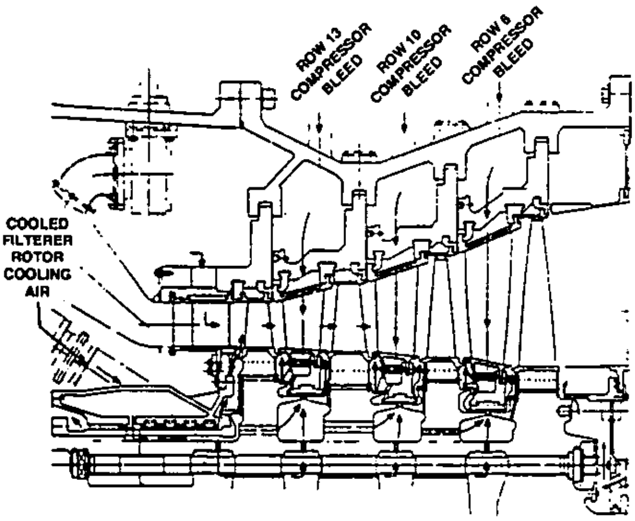

FIG. 10. COOLING CIRCUITS FOR THE 501F COMBUSTION TURBINE

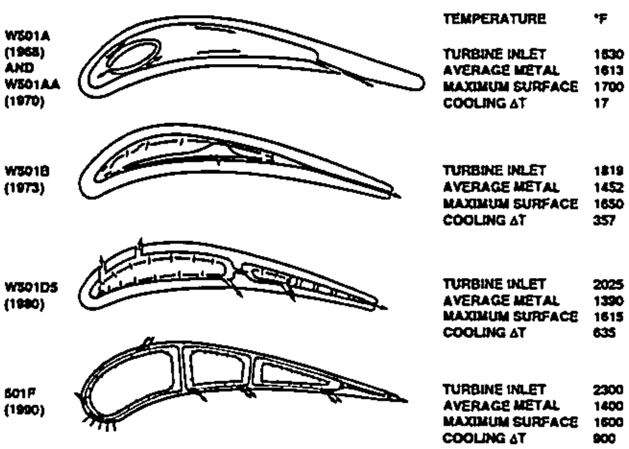

FIG. 11. EVOLUTION OF COOLING TECHNILOGY

tube casing for seal air supply and for cooling of the turbine disks as well as the first, second, and third-stage turbine rotor blades. The row 1 vane cooling design is shown in Fig. 11. This highly effective configuration, which evolved directly from the W501D5 design utilizes state-of-the-art concepts with three impingements inserts in combination with an array of film cooling exits and a trailing edge pin fin system. The first stage blade is cooled by a combination of film cooling and convection techniques via multipass, turbulated, serpentine passages, with pin fin cooling in the trailing edge exit slots. The cooling circuitry is shown in Fig. 12.

The fifth generation also produced improvements in the W251B10 and the W501D5 models. Introduced in 1990, the W251B12 improvements included modest increases in airflow, pressure ratio, and firing temperature. To accommodate these changes, double circular arc compressor blade profiles were used for the first two blade rows, compressor stages were restaggered, and cooling was improved for the first turbine vane and first rotor blade. Current rating is 48

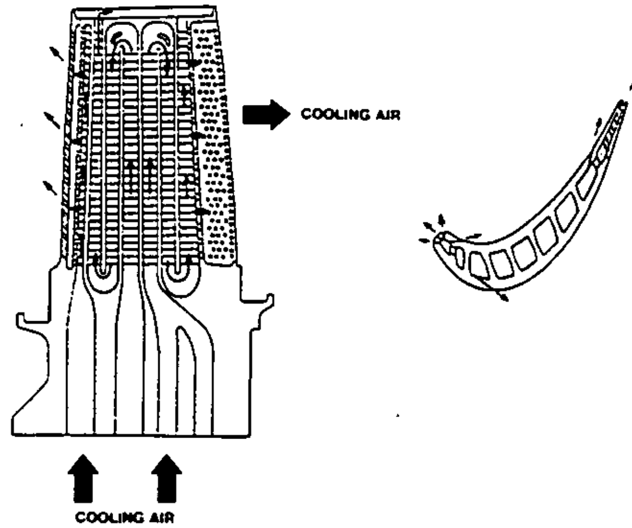

FIG. 12. COOLING CIRCUITRY OF 1ST STAGE BLADE USED IN 501F

MW at a simple cycle efficiency of 32% (Diakunchak, 1989). The improved W501D5, to be introduced in 1995, also will include modest increases in airflow, pressure ratio, and firing temperature. Compressor changes were scaled from the W251B12 and cooling was improved for the first three vanes and the first two rotor blades. In addition, the increase in airflow required that the row 4 blade be changed to a "Z" tip shroud similar to the 501F design. Rating of the improved W501D5 will be 118 MW at a simple cycle efficiency of 34%.

COOLING EVOLUTION

The evolution of the W501 and the W251 was highly dependent upon improvements in cooling technology as shown in Fig. 13. Without cooling, the maximum RIT would be under 1800F (982C). Fig. 11 depicts this evolution for row 1 vane, from the early W501A in 1968 to the latest highly sophisticated cooled design used in the current 501F. Similarly, rotor blade cooling has evolved from spanwise holes of the 501B to a sophisticated multi-pass, turbulated, pin-fin, film cooled design of the 501F. These advances in cooling technology were beyond the published state-of-the-art, hence, a comprehensive development program was required to verify the latest blade and vane cooling designs and to provide a broad base for future designs.

For example, parametric design information on various vane internal cooling features was generated in atmospheric pressure rigs, with the geometry scaled up to effect Reynolds number simulation. (Scalzo, Holden and Howard, 1981). Tests were run over a wide range of test parameters using basic test rigs such as the leading edge rig shown in Fig. 14. The insert nose in the leading edge heat transfer rig can be replaced so as to vary target distance, radius, and hole array. Actual leading edge cooling geometries have been tested as well as a range of geometries for the generation of parametric design information. In addition to atmospheric model tests, design verification tests were run on production vane segments at full operating temperature and

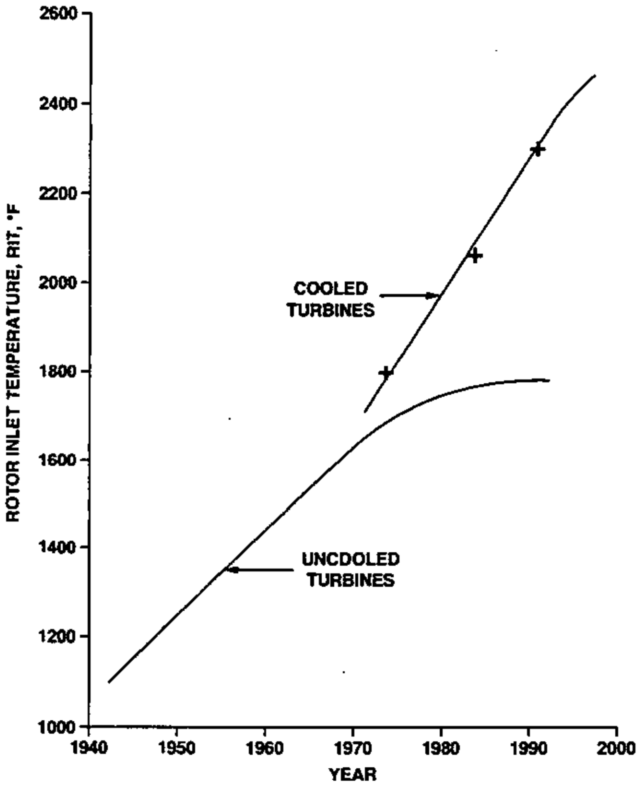

FIG. 13. FIRING TEMPERATURE TREND

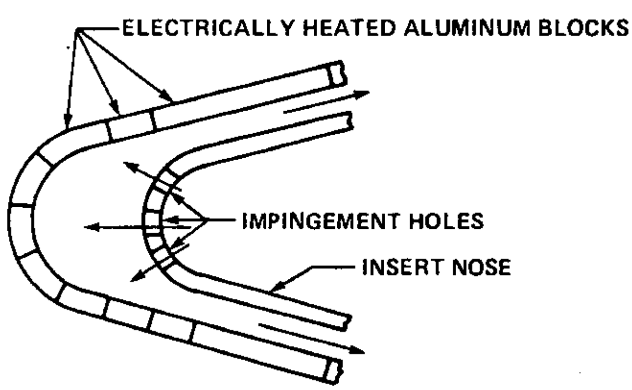

FIG. 14. INSERT NOSE IN THE LEADING EDGE HEAT TRANSFER RIG

pressure in a full-scale vane cooling rig.

Final verification of blade and vane temperatures were obtained from a fully instrumented full-load engine tests utilizing over 1400 pressure and temperature sensors. Comparisons were made to rig data to update the technology data base. Rotating blades and discs were monitored via telemetry (Scalzo, Allen and Antos, 1986; Gabriel and Donato, 1986).

MATERIAL DEVELOPMENT

In the early years (1950s to 1960s) materials were selected from available in-house steam turbine and jet engine experiences. However, since aero engines enjoy a pristine environment, many of these materials were not the best choice for the more corrosive fuels and operating environments of the land based gas turbine. Also, inspection intervals for aero components are more frequent as dictated by governmental regulations, while intervals in combustion turbines are being driven up to 24,000 hours by commercial considerations. For these and other reasons, material selection criteria, problems encountered, and developed solutions, would fill several volumes and are beyond the scope of this paper; however, a few pertinent comments will be made.

Compressor Area

Steam turbine and aero engine experience at the time of the initiation of the Westinghouse combustion turbine (1948 with the W21) dictated using a 12% chrome material for rotating and stationary blading because of its high strength, good corrosion resistance, and superior damping characteristics. This rationale is still valid today with standard strength 12% chrome material the choice for all compressor rotating blades and diaphragms with the exception of first two rotating blades in the 501F which use 17-4 PH, a 17% precipitation hardened stainless steel. Sermetel 5380DP[4] coating is currently used in the compressor to retard corrosion. To inhibit fretting, CuNiIn coating is used on blade roots. Compressor rotor materials have essentially retained the low-alloy steel materials based on steam turbine experience.

Combustor Area

The 25-20 series and 18-8 series stainless steel materials were used in early engines, however, Hastelloy X[5], a high temperature nickel-base alloy, became the standard for combustor baskets and transition ducts in the late 1960s. This material has been highly successful when kept below 1550°F (843°C) metal temperatures. For turbines with the increased firing temperatures used in the 1980s and later, the transition duct material was changed to IN617[6]; however, the standard combustor basket material is still Hastelloy X. IN617 provided a more stable higher strength material that required no pre or post heat treatment as was required by some sheet alloy candidates.

Turbine Stationary

Early designs utilized welded structures in AISI 310, a 25-20 austenitic stainless steel that had excellent resistance both to corrosion and to oxidation at elevated temperatures, but had limited strength capabilities.

A change to the higher strength, nickel-base, IN713 used successfully by aero engines, met with unsatisfactory results because

[4] Sermetech International Incorporated
[5] Hayes International
[6] International Nickel Company

of a lack of oxidation/corrosion resistance. A change to another precision cast nickel-base alloy, U500[7], resolved the corrosion concern, but the integrity of the welded structure was limited.

In 1967, Westinghouse introduced precision cast, colbalt-base, X45 turbine vane segments. This material was a modification of Haynes Stellite 31 used for years in aero engine turbine blades. It was the standard for precision cast turbine vane segments up to 1975, when the W501D was introduced with Westinghouse's ECY768 material in row 1. ECY768, is a cobalt-base alloy with higher creep strength and oxidation/corrosion resistance than standard X45 material. This improved cobalt-base alloy has become the standard for all row 1 vane segments and some selected row 2 and row 3 vane segments. Beginning in about 1975, diffusion and overlay coatings have been used selectively to enhance oxidation/corrosion resistance, as dictated by application.

Turbine Rotating

Various forged nickel-base alloys, and even 12% chrome materials, were used for early combustion turbine blades. Today nickel-base alloy Inconel X-750 remains the choice for many last row blades. Increases in firing temperatures necessitated changing to the higher strength Udimet nickel-base alloys in the mid 1970s. U520 forgings and cast U500 become the standard materials for front turbine stages. (All turbine blades are free-standing except for rows 3 and 4 of the 501F and row 4 of the improved W501D5.)

The 1980s and 1990s saw a move from forgings to higher strength cast IN738 material for front end blading because the complex cooling configurations could not be produced in machined, forged blades. As with the turbine stationary, diffusion and overlay coatings have been used for rotor blades. Future advanced engines will see a move to directionally solidified (DS) and to single crystal (SC) blading together with advanced coatings.

Turbine Discs

Early combustion turbine discs used a variety of materials including AISI 422, 19-9, and Discalloy 24 (Westinghouse-developed, austenitic alloy similar to A286). Selected cooling of the disc rims was required to prevent creep and notch sensitivity type concerns. To avoid these concerns, Westinghouse developed, in the mid 1960s, a unique cooling system for turbine rotors. High temperature, high pressure compressor discharge air is taken outside the engine where it is cooled and filtered before returning to the turbine rotor. This cooled, filtered air provides the rotor with a blanket of protection from hot blade path gases and supplies filtered cooling air to all disc rims as well as cooled airfoils. Filtration eliminates excessive contaminants that could block critical, intricate cooling passages of today's advanced combustion turbines. A reliable steam turbine material, NiCrMoV, was selected because of its high strength and outstanding fracture toughness. When cooled below 750°F (400°C), this material has infinite life because it is below the creep and aging thresholds. Also, its outstanding fracture toughness accommodated standard ultrasonic qualification and, therefore, did not require overspeed spin pits to qualify the forging as is the case for some materials. Since the mid 1960s, NiCrMoV discs integrated into the Westinghouse filtered, cooled air cooling system have generated over 20 million hours of problem free operation.

MECHANICAL EVOLUTION

The evolutionary Westinghouse design philosophy has resulted in operational reliabilities of over 99% and unit availabilities of over 95%. Many of the basic design features have stood the test of over four decades of design evolution. These design innovations were identified in previous sections of the paper; for example, cooled, filtered turbine cooling air with NiCrMoV turbine discs. However, several will be expanded in this section; i.e., single row 1 vane, integral compressor vanes, and minimum peak stress turbine blade roots.

Single Row 1 Vanes

In the early 1970s, the row 1 vane segment of the W251B2 was introduced as a single vane that could be serviced along with baskets and transitions through the combustor basket openings with the casing cover and enclosure roof in place. Other advantages of this design were structural thermal stress at 25 to 50% of a three-vane structure and improved casting quality. (Hultgren, 1981 and Scalzo et al., 1988). Entering a log/log plot displaying the low cycle fatigue (LCF) of a nickel base alloy or cobalt base alloy, LCF is increased by over a factor of 10 when stress is reduced 50%. In addition, a single vane facilitates higher quality castings compared to multi-vane castings. For advanced turbines, the inner shroud is supported from the inner casing to limit axial and circumferential displacements, thus, controlling flow angles.

Compressor Integral Vane

Selected diaphragm rows on current product models incorporate an integral airfoil/shroud design with four point seals on a removable seal carrier as shown in Fig. 15. Critical structural weld joints are eliminated by use of the integral structure. Also, the airfoil thickness/chord ratio is tapered to produce an optimized aero/mechan-

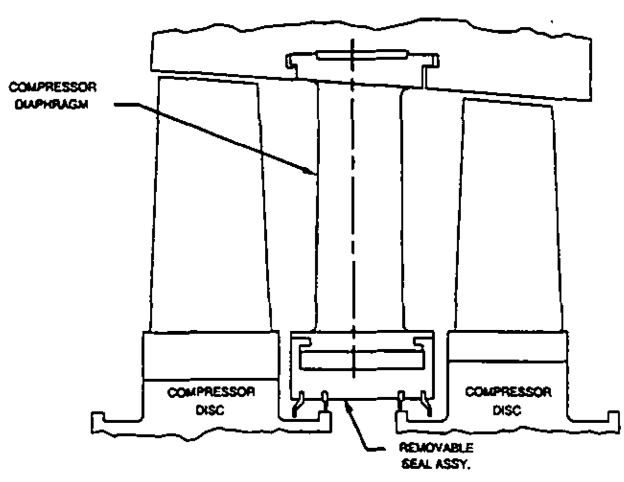

FIG. 15. INTEGRAL DIAPHRAGM DESIGN

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[7] Special Metals Company

ical design; i.e., high at the tip required mechanically and acceptable aerodynamically, and low at the hub, required aerodynamically and acceptable mechanically. Endurance strength of the structure is equal to base-metal properties as shown in Fig. 16. This design also provides for service of compressor diaphragms with rotor in place, as in past designs.

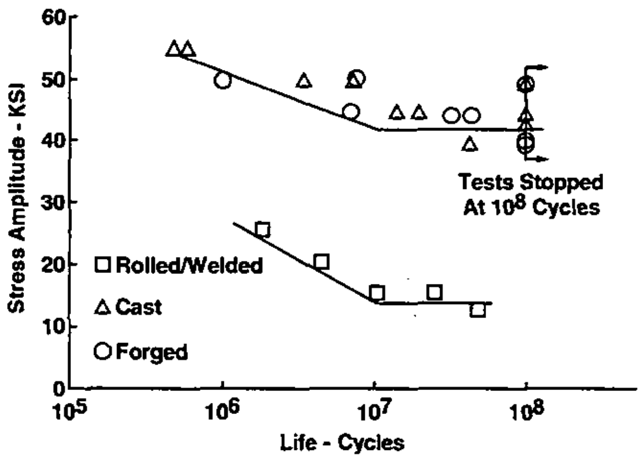

FIG. 16. ENDURANCE STRENGTH OF DIAPHRAGM DESIGNS

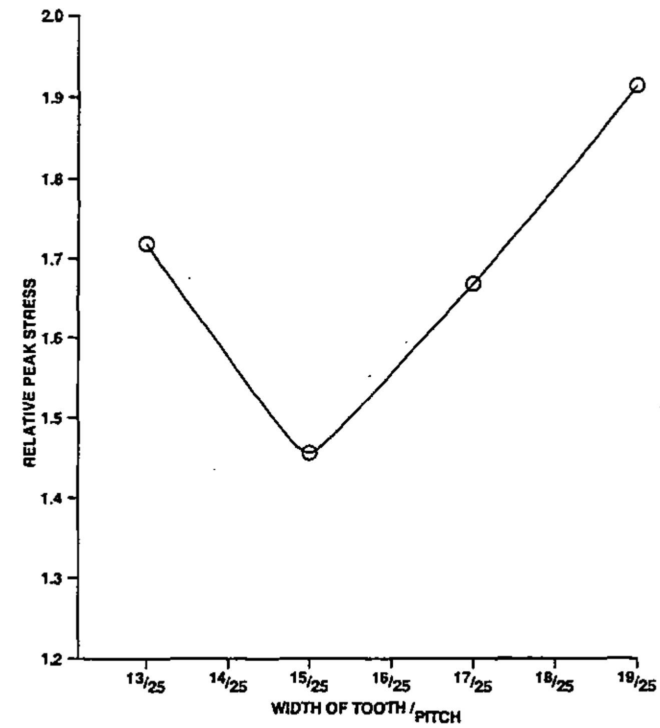

FIG. 17. STRESS VERSUS WIDTH OF TOOTH/PITCH

Turbine Blade Roots

In 1963, one of the top priority design goals was to develop an improved turbine blade root more suitable for higher firing temperatures and increased dynamic loadings expected in the third generation designs. Earlier designs were simply scaled from existing steam turbine and aero technology sources. There was also a desire to incorporate the new root design using NiCrMoV material as described in the preceeding materials section. Three basic objectives were identified; i.e., provide a thermal barrier, eliminate 3-D stress concentration, and minimize peak stresses. Using an extended root or extension isolates the root from flow path temperatures, thus providing a thermal conductive barrier. Also, it eliminates 3-D stress concentration resulting from the transfer of bending or dynamic moments from an airfoil shape to the root neck (Scalzo, 1992).

Minimizing peak stresses required the knowledge of 2-D stress distribution in irregular shapes. Since there were no finite element analysis programs available in the early 1960s, a general method for the determination of 2-D stress distribution in irregular shapes subjected to body and surface forces was developed using the finite difference method to represent the governing fourth order partial differential equation together with suitable boundary conditions. A mainframe computer was used to solve the multitude of equations using the over relaxation technique (Scalzo, 1963). Parametric analyses similar to Fig. 17, developed to determine the optimum tooth shape for minimum peak stress, resulted in Fig. 18. This was used to establish the root configuration for the W251A combustion turbine shipped in 1967, and has been used successfully for every Westinghouse combustion turbine since that time.

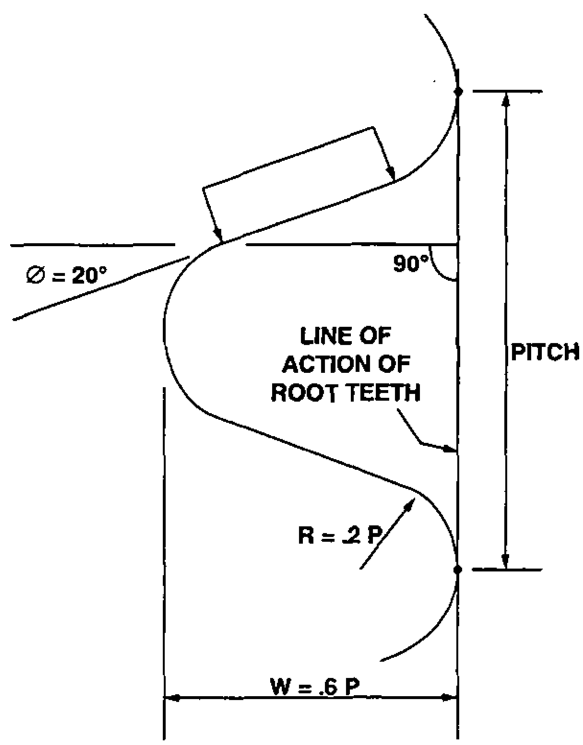

FIG. 18. ROOT TOOTH PROPORTIONS FOR MINIMUM PEAK STRESS UNDER DIRECT LOAD

COMBUSTION DESIGN EVOLUTION

Overall Geometry

In selecting the can-annular design, the W21 designers drew on the early experimental combustor can experience of the jet engine group even though the jets eventually selected an annular combustor. Sets of holes placed at the upstream end provided primary air sufficient for stoichiometric burning and another set of holes placed some distance downstream provided dilution air, bringing the stream temperature down to the required burner outlet temperature (BOT). (Note that BOT is the same value as the "firing temperature" used in the first and second generation.)

All engines have used can-annular combustors except for a silo type combustor installed in the blast furnace unit. The rationale that supports the can-annular design is still valid after 45 years. Advantages of this concept are:

- Shipped fully assembled with engine, thus reducing field cost.
- Facilitated full-scale laboratory testing.
- Can be developed to provide a preferential radial temperature profile, thus satisfying rotating blade requirements.
- Provides a lower "hot spot" by control of circumferential temperature gradients to improve turbine vane life.
- Generates a discrete harmonic content, thus minimizing blade vibratory design requirements.

Fuel Injection

The basic concept of fuel injection and ignition has been maintained from the W21 up to present engines, with the exception of the new advanced low NOx designs. Key elements are a pressure atomizing liquid fuel nozzle with a concentric atomizing air-assist used for starting. Gas fuel is injected via a concentric set of orifices. To prevent undesirable combustor noise, the orientation of fuel injection vectors must be compatible with the essential recirculation stabilization pattern required to anchor the flame front (Scalzo et. al., 1990). In models built after the mid 1970s, mechanical flow dividers were used to meter liquid fuel equally to each combustor. When required for emissions control, water is injected through the air atomizing ports of the liquid nozzle or pre-mixed, steam is injected either pre-mixed with gaseous fuel or via a separate set of orifices.

Ignition

The ignition scheme used on the earliest combustor designs was a spring loaded spark plug in each can which retracted as combustor pressure built-up to operating conditions; and cross-flame tubes provided ignition between cans. This belt and suspenders approach was used because, with the large distance between adjacent cans, cross-firing was not reliable. In later engines, with combustors being nested closer together (W251A model and later), cross firing was more effective and spark ignition was provided for only two cans (for redundancy) and cross-firing lit the others. In early designs the cross-firing occurred through tubes attached to each combustor at the combustor flame zone, with one tube fitting inside the other in a sliding fit. In more recent models this arrangement has been replaced by hard-connected flexible tubing. The retractable spark plug approach remained basically the same with improvements made to the spark plug, the retracting piston, and the electrical power to the plug.

In the earlier engines, flame-on indication was provided by thermocouples protruding into each transition section, which recorded BOT as well as other control input functions. When the BOT increased to that required for the W251AA engine and succeeding models, this arrangement had to be abandoned due to the impact of the higher temperature environment on sensor life. The flame-on detection is now by an optical sensor which views the combustor flame zone through a port. Other control supervisory functions are taken up by thermocouples placed in the blade path downstream of the last turbine blade row. These exhaust blade path thermocouples also monitor each combustor to protect rotating blades against harmful harmonies (Scalzo, 1992).

Special Fuels

Gas turbines have the capability to burn a wide variety of fuels. Combustor and fuel injector design modifications have been able to accommodate fuels which have included: natural gas, blast furnace gas, coal gas, distillate fuel, residual fuel, crude oil, methanol, propane, coal derived liquids, and shale oil (Pillsbury et al., 1974, 1978 and 1979; Seglem and DeCorso, 1980). Burning fuels which have potentially corrosive elements requires that fuel contaminant specifications be set (Wenglarz and Menguturk, 1981). The Westinghouse Research Laboratory was equipped with corrosion evaluation test rigs where actual fuels were burned making accurate determinations of fuel element corrosion levels on the combustor and other turbine hot parts. This work at the Research Laboratory was key to setting the ASTM D2880 and ASME B-133 fuel contaminant specifications now in use industry wide (DeCorso et al., 1971 and Hussey et al., 1973). Use of these various fuels in the future will now be subject to more stringent emission regulations.

Emissions

The first emission concerns to impact combustor designers began with exhaust visibility (smoke) in 1966. As a result, the W171 and W191 combustors were modified to produce a smokeless design (DeCorso, Hussey and Ambrose, 1967). In the early 1970s, EPA began to develop regulations which dealt with NOx emission levels. In local areas of California more stringent emission regulations were passed. In the Los Angeles Air Pollution Control District (LAAPCD) regulations (rule 67) called for particulate emissions of no more than 10 lb/hr (4.54 kg/hr) regardless of plant size, and a NOx regulation which required 32 ppmv from a W501 engine. Particulate levels were minimized by adhering to established fuel specifications (Carl, Obidinski and Jersey, 1975) and limiting sulfur content in the fuel, since the Los Angeles particulate measurement technique counted sulfur compounds. (Many current regulations require NOx to be 25ppmv or lower, regardless of engine size.)

Working in cooperation with the Westinghouse Research Laboratory, a theoretical understanding of NOx formation was gained and a NOx formation model was developed and verified by laboratory and field testing. This model proved to be invaluable in predicting the effect of combustor operating parameters on NOx formation. It was capable of giving NOx emission effects due to combustor operating parameters; and, it also accurately predicted the effect on NOx emission of various fuels, fuel bound nitrogen, water injection, and steam injection (Hung, 1974 and Vermes, 1974). With the NOx model as a guide, combustion laboratory test time was greatly reduced and valuable insight into potential low NOx approaches was provided.

In order to meet the 32 ppm NOx regulation for W501 turbines, water injection for NOx control was designed, tested and proven in the laboratory, and then in a full-scale field test (Ambrose and Obidinski, 1972), in what is believed to be the first application of water injection for NOx control.. Subsequent design and testing also confirmed the use of steam for NOx control, with steam being less effective, pound for pound, than water.

Dry Low NOx

Dry low NOx development schemes were explored in the three areas; i.e., premixed lean combustion, rich-lean combustion, and catalytic combustion. Small scale Research Laboratory tests had confirmed that very low NOx levels were attainable by premixing the air and fuel. Premixing means in this context that the air and fuel are mixed on a molecular level. A practical scheme to use premixing was devised and given the name of "hybrid combustor" because it utilized the dual approach of a premixed fuel/air stream and a pilot flame zone which burned in the diffusion flame mode. The term "hybrid combustor" is now commonly used in the industry to refer to this type of low NOx combustor. An example of an early design which was tested is shown in Fig. 19 (Mumford, Hung and Singh, 1977). This design was the basis for the lean, pre-mix combustor design used in the MW701D combustion turbine (Aoyama and Mandai, 1984). Advanced versions of this design are now operating in the 701F (50Hz scale of the 501F) engines in Japan with NOx emissions less than 25 ppmv. This design is currently being developed further to produce less than 15 ppmv by the late 1990s.

Another approach to NOx reduction utilized the rich-lean burn principle, which did not require premixing and was effective for cases where the fuel contained fuel bound nitrogen (FBN). The premixing or water/steam injection approach is defeated by the presence of significant amounts of FBN. In rich-lean burning, the burning is

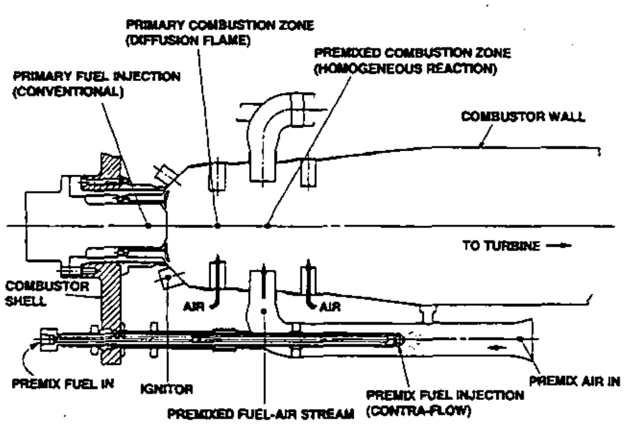

FIG. 19. HYBRID BURNER

staged to occur in a fuel-rich zone where the oxygen deficiency prevents NOx formation; in a second lean zone which follows the rich zone, burning of the products from the rich zone are completed at a temperature low enough to prevent NOx formation. The multi-annular swirl burner (MASB) was developed which had the capability of operating as a rich-lean burner, as well as in the lean burn mode. This combustor was successfully operated at the Waltz Mill, PA coal gas test site (Lew, DeCorso et al., 1981) and is now under development as a topping combustor for an advanced PFB cycle program (Garland and Pillsbury, 1992).

The prospects of ultra low NOx using catalytic combustion was first evaluated in 1971. Tests run at the Research Center confirmed that virtually zero emissions could be attained and that very uniform temperature patterns were possible (DeCorso, Mumford et al., 1977). A very active program included testing of catalytic combustors at full-scale conditions (Hung, Dickson and DeCorso, 1978; Scheihing et al., 1982). Although NOx was in single digits, the mechanical integrity of the substrate was very unsatisfactory. After the oil embargo sharply curtailed market activity, development effort in low emissions tapered off through the early 1980s; however, activity has surged again in the 1990s with advanced designed catalytic systems currently under review.

CLEAN COAL TECHNOLOGY

Combustor design and testing first took place in the mid 1970s under U.S. Government sponsorship for a projected coal gas plant for Public Service of Indiana. Full-scale test facilities were built in which combustors modified for coal gas use were tested with actual coal gas compositions encompassing air blow gasifier and oxygen blown gasifier gas (Pillsbury, 1974). This development supported the joint Westinghouse/Dow Chemical Co. program to burn medium Btu (239 Btu/scf [8890 kJ/scm]) coal gas; first, in a W191 trial engine; and, later in W501D5 turbines at the Dow Chemical Co. IGCC plant in Plaquemine, LA (Hendry and Pillsbury, 1987). These two 104 MW units are operating in a Dow Chemical developed oxygen-blown integrated gasification combined cycle (IGCC) that includes a cold gas cleanup system (Morrison and Pillsbury, 1989). To date, these combustion turbines have operated with average availabilities in excess of 95% (Geoffroy and Amos, 1991).

Most commercial size gasification projects have used high purity oxygen rather than air as the oxidant for the gasifiers. Recent IGCC evaluations have looked at using the combustion turbine air compressor to supply air for the air separation unit. Typically, this air stream is sent to a high pressure air separator unit which produces oxygen for gasification and high pressure nitrogen for combustion turbine NOx control. The diluent nitrogen lowers the flame temperature and, therefore, lowers the NOx.

Other concepts being developed at Westinghouse include incorporating combustion turbines in a coal-fueled combined cycle for second generation pressurized fluidized bed (PFB), and direct coal-fired (DCF) combined cycle configurations. IGCC, PFB and DCF concepts require a fuel gas cleanup system to remove particulates, sulfur, and alkali (Newby and Bannister, 1993). Low NOx levels are obtained through use of advanced combustion processes (Scalzo et al., 1991).

Pressurized Fluidized Bed

Under a DOE-sponsored program, Westinghouse is developing a MASB topping combustor which will be used in the second generation PFB topping combustor (Domeracki, Dowdy and Bachovchin, 1994). The entire combustion air quantity is introduced into the combustor through axial flow, concentric vane rows. The MASB utilizes vitiated air at 1600°F (871°C) for cooling. Thermal NOx levels of less than 10 ppm have been measured for a burner outlet temperature of 2300°F (1260°C).

In addition to the topping combustor, Westinghouse is developing and supplying integrated gas turbine systems that will interface with PFB plants and incorporate the functions of hot gas filtration, alkali vapor removal, hot gas piping and control, and turbine compression and expansion (Newby et al., 1994).

Direct Coal

In a direct coal-fueled concept, ash and sulfur are to be removed from the coal during the combustion process. Air from the combustion turbine driven compressor flows into the pressurized slagging combustor and into a topping combustor. About one-third of the air is directed to the slagging combustor for utilization in the substoichiometric combustion of the coal fuel. Two-thirds of the compressor discharge air goes directly to the topping combustor where the low Btu fuel gas is fired under lean-burn conditions. One advantage of this split is that the majority of the primary combustion process only has to handle one-third of the total air flow required by the combustion turbine. The slagging combustor operates at temperatures high enough to melt or vaporize the inert ash constituents in the coal. Westinghouse has worked with Textron Defense Systems under a DOE contract to demonstrate the feasibility of this concept (Bannister, Newby and Diehl, 1992). Test results confirmed 99% carbon conversion with NOx levels under 50 ppmw. The next step in the development of this process is to integrate various hardware components into an operating cycle in a pilot plant.

CURRENT AND FUTURE TRENDS

In cooperation with U.S. Department of Energy's Morgantown Energy Technology Center, a Westinghouse led team is working on the second part of an 8-year, Advanced Turbine Systems (ATS) Program to develop the technology required to provide a significant increase in natural gas-fired combined cycle power generation plant efficiency (Bannister et al., 1994).

Efficiencies for large natural-gas-fired combined-cycle systems for the utility market have been demonstrated at 54 to 55%. Even though manufacturers will make improvements in the 1990s, pursuing the historic trend, shown in Tables 2 and 3, efficiency levels will reach a plateau for simple and combined cycle plants. Cycle innovations, a 2600°F (1427°C) combustion turbine RIT, reduced cooling air usage, steam cooling, improved component efficiencies, together with improved material/coating systems can achieve combined cycle efficiencies in the 60% range for natural gas-fired utility machines.

ATS using natural gas is to be commercially available by the year 2000. Coal-derived fuel concepts are candidates for the post-2005 power-generation market. The final design will be fuel flexible in that it will operate on natural gas, but also be capable of being adapted to operate on coal, coal-derived, or biomass fuels.

SUMMARY AND CONCLUSION

The Westinghouse evolutionary combustion turbine design philosophy has always maintained reliability and maintainability as major considerations. Many basic design innovations have been retained after over four decades of evolution, thereby attesting to the soundness of their selection. Future advancements in firing temperature, materials, cooling, and cycles must also satisfy reliability/maintainability requirements.

Westinghouse has actively worked with industry and the government for over three decades in developing the use of coal as a clean fuel for power generation. The company is currently actively supporting clean coal technology programs in advanced IGCC, first generation PFB, and second generation PFB plants.

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REFERENCES

Ambrose, M. J., and Obidinski, W.S., 1972, "Recent Field Tests for Control of Exhaust Emissions from a 35 MW Gas Turbine," ASME Paper 72-JPG-GT-2.

Ambrose, M. J., Costello, R. F., and Schreiber, H., 1984, "Utility Combustion Turbine Evaluation of Coal Liquid Fuels," ASME Paper 84-JPGC-GT-13.

Aoyama, K., and Mandai, S., 1984, "Development of a Dry Low NOx Combustor for a 120 MW Gas Turbine," *ASME Journal of Engrg. for Gas Turbines and Power*, Vol. 106, pp. 795-800.

Bannister, R. L., Cheruvu, N.S., Little, D. A. and McQuiggan, G. 1994, "Development Requirements for an Advanced Gas Turbine System," to be presented at the 1994 Turbo Expo Conference, The Hague, Holand.

Bannister, R. L., Newby, R. A. and Diehl, R. C., 1992, "Developing a Direct Coal-Fired Combined Cycle," *Mechanical Engineering*, Vol. 114, No. 12, pp. 64-70.

Bannister, R. L. and Silvestri, G. J., 1989, "Evolution of the Central Station Steam Turbine," *Mechanical Engineering*, Vol. 111, No. 2, pp. 70-78.

Carl, D. E., Obidinski, E. S. and Jersey, C.A., 1975, "Exhaust Emissions From a 25-MW Gas Turbine Firing Heavy and Light Distillate Fuel Oils and Natural Gas," ASME Paper 75-GT-68.

Cox, A. R., 1957, "Gas Turbine Generating Experience of West Texas Utilities Company," ASME Paper 57-GTP-11.

Cox, A. R., Henson, L. B., and Johnson, C. W., 1967, "Operation of San Angelo Power Station Combined Steam and Gas Turbine Cycle," *Proceedings American Power Conference*, Vol. 29, pp. 401-409.

Diakunchak, I. S., "The CW251B12 Gas Turbine Engine," ASME Paper 89-GT-119.

DeCorso, S. M., Hussey C. E., Ambrose M. A., 1967, "Smokeless Combustion in Oil-Burning Gas Turbines," ASME Paper 67-PWR-5.

DeCorso, S. M., Lee, S. Y., Young, W. E., 1971, "Laboratory Procedures for Evaluating High Temperature Corrosion Resistance of Gas Turbine Alloys," *ASME Journal of Engrg. for Power*, July.

DeCorso, S. M., Mumford, S. E., Carrubba, R., and Heck, R., 1977, "Catalysts for Gas Turbine Combustors -- Experimental Test Results," *ASME Journal of Engrg. for Power*, Vol. 99, No. 2.

Domeracki, W. F., Dowdy, T. E. and Bachovchin, D. M., 1994, "Topping Combustor Development for Second, Generation Pressurized Fluidized Bed Combined Cycles," to be presented at the 1994 Turbo Expo Conference, The Hague, Holland.

Gabriel, F. K., and Donato, V., "Telemetry Measurement of Combustion Turbine Blade Vibration in a High Temperature Environment," ASME Paper 86-GT-207.

Garland, R. V. and Pillsbury, P. W., 1992, "Status of Topping Combustor Development for Second-Generation Fluidized Bed Combined Cycles," *ASME Journal of Engrg. for Gas Turbines and Power*, Vol. 114, pp. 126-131.

Geoffroy, G. A. and Amos, D. J., 1991, "Four Years Operating Experience Update on a Coal Gasification Combined Cycle Plant with Two 100 MW Class Gas Turbines," Combined Heat and Power and Independent Power Producers Conference, England.

Hendry, R. L. and Pillsbury, P. W., 1987, "Commercial Demonstration of the Dow Gasification Process in an Integrated Combined Cycle Cogeneration Application," *Proceedings, American Power Conference*, Vol. 49, pp. 149-153.

Hultgren, K. C., 1981, "Use of 3-Dimensional Finite Element Model in Design of First Stage Turbine Vane Segments," CIMAC Paper No. GT-19, 14th International Congress on Combustion Engines, Helsinski.

Hung, W. S. Y., 1974, "Accurate Method of Predicting the Effect of Humidity or Injected Water on NOx Emissions from Industrial Gas Turbines," ASME Paper 74-WA/GT-6.

Hung, W. S. Y., Dickson, W. H. and DeCorso, S. M., 1978, "Preliminary Design Analysis of a Catalytic Ceramic Structure in a Turbine Combustor," ASME Paper 78-WA/GT-10.

Hussey, C. E., Lee, S. Y. and Young, W. E., 1973, "Operation of Gas Turbines on ASTM3-GT Fuel," Manual on Requirements, *Handling, and Quality Control of Gas Turbine Fuel, ASTM STP 531*, American Society for Testing and Materials, pp. 3-21.

Krapf, G. H. and Stephens, J. O., 1958, "Gas Turbines for Blast Furnace Blowing," ASME Paper 58-GTP-14.

Lew, H. G., DeCorso, S. M., Vermes, G., Carl, D., Havener, W. J., Schwab, J., and Notardonato, J., 1981, "Low NOx and Fuel Flexible Gas Turbine Combustors," ASME Paper 81- GT-99.

Morrison, E. M. and Pillsbury, P. W., 1989, "Coal Generated Synthetic Gas," ASME Paper 89-GT-257.

Mumford, S. E., Hung, W. S. Y., and Singh, P.P., 1977, "A Potential Low NOx Emission Combustor for Gas Turbines Using the Concept of Hybrid Combustion," ASME Paper No. 77-GT-15.

Newby, R. A., Domeracki, W. F., McGuigan, A. W. and Bannister, R. L., 1994, "Integration of Gas Turbine Systems into Pressurized Fluidized Bed Combustion Combined Cycles," to be presented at the 1994 Turbo Expo Conference, The Hague, Holland.

Newby, R. A., and Bannister, R. L., 1993, "Advanced Hot Gas Cleaning System For Coal Gasification Processes," ASME Paper 93-GT-338.

Pillsbury, P. W., 1974, "A High Pressure Coal Gas Combustor Testing Program," ASME Paper 74-PWR-11.

Pillsbury, P. W., Cohn, A., Mulik, P. R. and Stein, T. R., 1978, "Investigating Combustion Turbine Burner Performance with Coal-Derived Liquids Having High Full-Bound Nitrogen," ASME Paper 78-GT-126.

Pillsbury, P. W., Cohn, A., Mulik, P. R., Singh, P. P. and Stein, R. T., 1979, "Fuel Effects in Recent Combustion Turbine Burner Tests of Six Coal Liquids," ASME Paper 79-GT-137.

Putz, T. L., 1954, "Gas Turbines for the Power Industry," *Proceedings American Power Conference*, Vol. 16, pp. 293-298.

Scalzo, A. J., 1992, "High-Cycle Fatigue Design Evolution and Experience of Freestanding Combustion Turbine Blades," *ASME Journal of Engrg. for Gas Turbines and Power*, Vol. 114, pp 284-292.

Scalzo, A. J., 1963, "Numerical Solution for the Two-Dimensional Stress Distribution in Irregular Shapes," *Westinghouse Electric Corporation Report*, E-1420, Orlando, FL.

Scalzo, A. J., Allen, J. M., Antos, R. J., 1986, "Analysis and Solution of a Nonsynchronous Vibration Problem in the Last Row Turbine Blade of a Large Industrial Turbine," *ASME Journal of Engrg. for Gas Turbines and Power*, Vol. 108, pp. 591-598.

Scalzo, A. J., Amos, D. J., Bannister, R. L., and Garland, R. V., 1991, "Coal-Fueled Programmes Gather Pace," *Modern Power Systems*, Vol. 11, No. 6., pp. 55-59.

Scalzo, A. J., Antos, R. J., Fukue, I., 1988, "Operating Experience Complements New Technology in Design of Advanced Combustion Turbine," ASME Paper 88-JPG/GT-1.

Scalzo, A. J., Holden, P. C., and Howard, G. S., 1981, "Westinghouse 501D Combustion Turbine Engine," ASME Paper 81-GT-32.

Scalzo, A. J., Howard, G. S., and Holden, P. C., 1983, "Field Test of the W501D, a 100 MW Combustion Turbine," ASME Paper 83-GT-174.

Scalzo, A. J., McLaurin, L. D., Howard, G. S., Mori, Y., Hiura, H., Sato, T., 1989, "A New 150 MW High Efficiency, Heavy-Duty Combustors Turbine," *ASME Journal of Engrg. for Gas Turbines and Power*, Vol. 111, pp. 211-217.

Scalzo, A. J., Sharkey, W. T. and Emmerling, W. C., 1990, "Solution of Combustor Noise in a Coal Gasification Cogeneration Application of 100-MW-Class Combustion Turbines," *ASME Journal of Engrg. for Gas Turbines and Power*, Vol. 112, pp. 38-43.

Scheihing, P. E., Piaia, T. A., Pillsbury, P. W., and Kitaplioglu, O., 1982, "Full-Scale Catalytic Burner Testing with No. 2 Oil for Combustion Turbine Application," ASME Paper No. 82-JPGC-GT-20.

Seglem, C. E. and DeCorso, S. M., 1980, "Alternate Fuels Capability of Gas Turbines For Pipeline and Industrial Applications," 7th Annual Technology Conference and Exposition, Washington, D.C.

Stephens, J. O. and Young, W. E., 1989, "A Super High-Speed Combined-Cycle Locomotive," *Turbomachinery International*, July/August, pp. 26-30.

Stephens, J. O., Little, D. A., Hunt P. J., and Stanaland, R., 1990, "Efficient Part-Load Combined Cycle Operations," *Proceedings 19th Turbomachinery Symposium Texas A&M*, pp. 31-36.

Stephens, J. O., and Bruce, D. F., 1955, "5,000 Horsepower Dual Shaft Gas Turbine Power Plant," *Proceedings American Power Conference*," Vol. 17, pp. 342-349.

Stephens, J. O., 1952, "Gas Turbine Power Plant Designs," *Proceedings American Power Conference*, Vol. 14, pp. 267-271.

Vermes, G., 1974, "A NOx Correlation Method for Gas Turbine Combustors Based on NOx Formation Assumptions," ASME Paper 74-WA/GT-10.

Wenglarz, R. A. and Menguturk, M., 1981, "Use of Cascade and Small Turbine Tests to Determine Erosion of Utility Turbines," ASME Paper 81-GT-52.