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Evolution of the Central Station Steam Turbine in the United States

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ABSTRACT

In 100 years the steam turbine has evolved from an experimental device to the major source of electrical generation. Cycle innovations such as regenerative feedwater heating and reheat, despite much earlier theoretical conception, have achieved widespread acceptance and maturity when applied to steam turbines. However, despite astonishing advances during this 100 year period, two fundamental design philosophies, drum rotor reaction stages and disc and diaphragm impulse stages, have remained viable concepts.

Within the United States, steam conditions have increased from 155 psig, 410°F (1.07 MPa, 210°C) in 1900 when the first central station unit was installed to 5000 psig, 1200°F/1050°F/1050°F (43.5 MPa, 649°C/565°C/565°C) in 1960, concomitant with unit size increases from 1500 kW to 1,300,000 kW, while plant heat rates have decreased from 35,000 Btu/kWh (36,925 kJ/kWh) to about 8800 Btu/kWh (9285 kJ/kWh). This heat rate improvement reflects not only the increases in steam conditions and the adoption of more complex cycles, but also improvements in blading efficiency and reductions in a variety of parastic losses. In addition, advancements in controls have been a factor in achieving viable designs.

This paper reviews the evolution of the central station steam turbine primarily from a Westinghouse perspective. Advances in the areas of steam conditions, cycles, efficiency improvements, unit size and configuration, materials, design analyses, and verification techniques are described. Development of the nuclear turbine is also reviewed. Progress up to the present day is given including turbine life extension possibilities, use of artificial intelligence, new turbine designs, unit capacity and use of advanced steam conditions.

HISTORICAL PERSPECTIVE

For practical purposes, it was the basic inventions of Parsons (England) and de Laval (Sweden) in the 1880s and 1890s that introduced steam turbines as a prime mover. Also, to their names should be added Rateau (France), Curtis (United States) and Ljungstrom (Sweden) for their later fundamental concepts.

Dawn of the Electric Age

Practical steam turbine inventions were well tuned to coincide with the development of direct current (d.c.) electric dynamos that were being used to power arc-lighting street systems. Edison, who developed the incandescent filament lamp by 1879, is responsible for the first central station to provide commercial electric lighting service. This was the Pearl Street Station in New York City, which started operation in September 1882, serving 1284, 16-candle-power d.c. lamps, approximating a 72 kW total load. Similar plants followed rapidly in other cities, as well as to serve other districts of New York City. A large number of small plants were needed, since d.c. transmission was limited economically to small areas. Competition developed overnight in a rapidly expanding market. For example, Pearl Street Station's load alone, increased tenfold in three years (Campbell, 1962).

George Westinghouse, with his Union Switch and Signal Co., was one of those competitors. In 1885 Westinghouse acquired rights to manufacture and sell the Gauland and Gibbs European transformer, and then he proceeded at once to develop alternating current (a.c.) distribution. The a.c. system quickly became a primary factor in stimulating the development of electrical power for industry and for traction. Transmission economy made practical the concentration of generation in units and stations, and of ever increasing size, located strategically with respect to condensing water supply.

Steam turbines with ratings of 1000 kW were introduced into the United States about the beginning of the 20th century. At that time there were numerous d.c. and a few a.c. electric generating stations in use, all with steam engine or hydraulic turbine prime movers. Some d.c. generators were directly connected to engines.

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Among the pioneers of d.c. machinery development were Thomson and Houston, of Philadelphia. Their patents and services were first acquired by the newly formed American Electric Co. In 1883 this concern was re-incorporated and moved to Lynn, MA as the Thomson-Houston Electric Co. Meanwhile, Edison had formed several companies to exploit his incandescent lamp, including the Edison Lamp Co., the Edison Machine Works and the Edison Electric Light Co., all of which were merged in 1889 with the Sprague Electric Railway and Motor Company to form the Edison General Electric Co. Then in turn, this company was merged with the Thomson-Houston Electric Co. in 1892 to become the General Electric Co. at Lynn.

George Westinghouse incorporated the Westinghouse Electric Co. in 1885, to exploit a.c. electrical developments, after having in 1881 set-up the Westinghouse Machine Co. for steam and gas engine manufacture. These organizations were merged in 1907 to form the Westinghouse Electric and Manufacturing Co., which later became the Westinghouse Electric Corp.

The Allis Chalmers Co. was established by a consolidation of several concerns in 1901, whose principal products had been boilers, mining machinery, steam engines and pumps. Electrical machinery was added by acquisition of the Bullock Electric Co., in about 1904 (Christie, 1937).

Start of the Turbine Business in the U.S.

Westinghouse turbine development followed after the acquisition in 1895 of rights to manufacture straight reaction type turbines under Parsons' patents (Keller and Hodgkinson, 1936). Allis Chalmers attempted several approaches beginning in 1902, and in 1904 proceeded with Parsons' designs under agreement with a British Turbine Advisory Syndicate. The following year Allis Chalmers secured rights to manufacture directly under Parsons' patents; thus, their early machines differed little from the first Westinghouse turbines. General Electric, in 1897, acquired rights to exploit the impulse type turbine patents which had been awarded to Curtis in 1896.

All Westinghouse and Allis Chalmers station steam turbine have had horizontal shafts. Early units used reaction blading throughout. The principal difference between the early turbines of Westinghouse and Allis Chalmers concerned details of blade construction, and location of the low pressure balance piston.

In about 1907 Westinghouse substituted a velocity compounded (Curtis type) impulse wheel in place of a large number of short and relatively high-leakage high-pressure reaction stages; this decreased turbine length considerably, made practicable the use of more control valves leading to improved light load efficiency and better adapted the unit construction to increasing steam pressure and temperature (Keller and Hodgkinson, 1936).

The first American-built Westinghouse turbine, 120-kW, was shipped in 1897 to the Nichols Chemical Co. Turbine serials 5, 6, and 7, (300 kW units), were installed during 1899 in the plant of the Westinghouse Air Brake Co. (Morgan, 1950). Westinghouse turbine serial 8, rated 1500 kW at 1200 rpm (the unit actually produced 2000 kW) was installed in 1900 by the Hartford Electric Light Co. (Figure 1) as this country's pioneer central station turbine. Steam conditions were 155 psig, 410°F (1.97 MPa, 210°C) with a steam rate of 19 lb/kWh (8.6 kg/kWh); the turbine efficiency was 58%. After eight years of service, it was replaced by a 3500 kW Westinghouse turbine-generator.

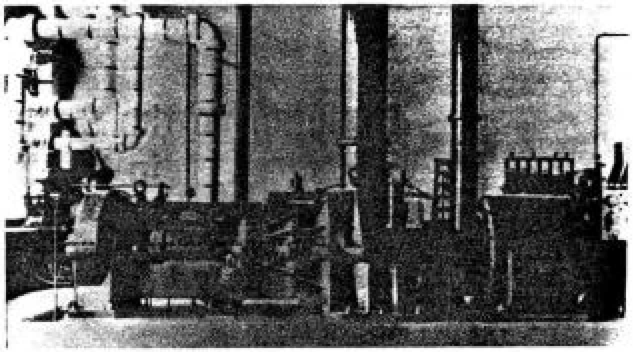

FIG 1 HARTFORD ELECTRIC CO.'S 1500 kW STEAM TURBINE-GENERATOR

The first General Electric turbine, shipped in 1903, was a 720 rpm, horizontal shaft unit rated at 1500 kW. General Electric then shifted quickly to vertical shaft turbines, and concentrated on them for some years for ratings 500 kW and upward. (Robinson, 1937).

Arguments of relative merit between horizontal and vertical turbines were heated; the principal points favorable to vertical units included use of minimum floor space and capability for quick start-up. On the other hand, accessibility for overhaul was poor, they were limited in capacity and were not suitable for large heat drops, to say nothing of complex regeneration and reheat which ultimately entered the picture.

By 1908, General Electric was changing over to horizontal designs. The last vertical unit, 20MW, was shipped in 1913. Their first sizable horizontal turbine was a 1500 kW, 1800 rpm unit for the Public Service Electric Co. of New Jersey.

Evolution of the Industry

The evolution of steam turbine power generation in the U.S. is summarized in Figure 2. It is marked by a steady increase in steam pressure and temperature, and a consequent increase in plant thermal performance. During the first fifty years of the 20th century, inlet steam pressure and temperatures increased at an average rate per year of 43 psi (0.3 MPa) and 13°F (7°C), respectively. Plant thermal efficiency peaked at about 40% with the Eddystone 1 unit. Fossil-fired steam turbine-generator size peaked at 1300 MW in the 1970s with TVA's two Cumberland units.

In striving for higher efficiency, turbine designers turned to more complex cycles. By the early 1920s, regenerative feedheating was well established using two or four bleed points. Reheat cycles came into use in the mid 20s. At the throttle temperature of 700°F (371°C), that was current when the pioneer 600 and 1200 psi (4.1 to 8.3 MPa) units went into service, reheat was essential to avoid excessive moisture in the final turbine stages.

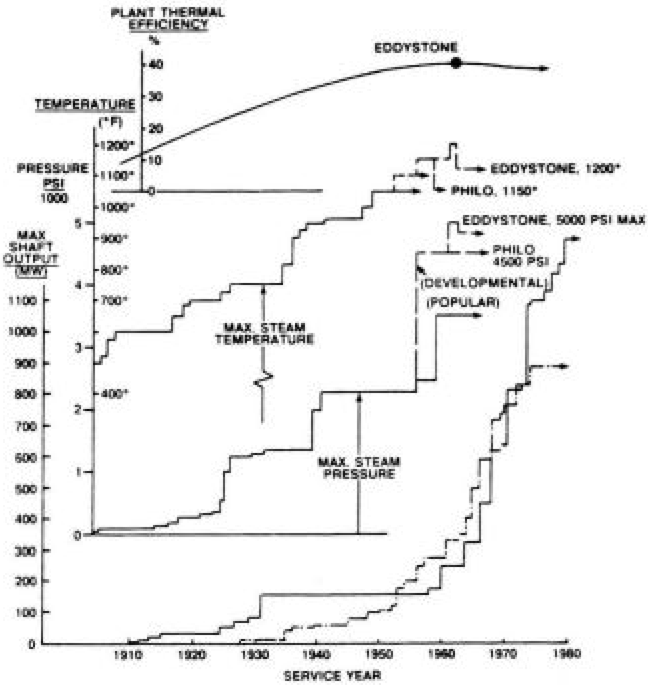

FIG 2 HISTORICAL DEVELOPMENT OF STEAM TURBINE-GENERATORS IN THE U.S.

As a result of rising throttle temperatures in the 1930s, and the control complexity of reheat units, reheat fell out of use. Reheat was reintroduced to improve plant efficiency in the 1940s and second reheats appeared in the early 1950s. By the late 1940s, exhaust area was a major limitation on size. The earliest answer was the double-flow, single-case machine. Cross-compounding introduced in the 1950s represented a big step forward. Now the speed of the low-pressure unit could be reduced. Hence, the last stage diameter could be greater, yielding more exhaust annulus area.

Until the early 1920s, generators limited all but the smaller direct connected turbine to low rotative speeds, and through much of that period, exhaust vacua were rather high by today's standards. A good portion of the larger units then served 25-cycle traction power systems and operated at 750 or 1500 rpm, some as low as 500 rpm. Sixty-cycle generators were built for 720, 900, 1200 and 1800 rpm, with 3600 rpm then only to 5000-kW output. As long as these conditions prevailed, there was no difficulty in providing ample last stage annulus area for low exhaust steam leaving velocity and exhaust loss. With the moderate steam conditions employed in complete expansion turbines by 1925, exhaust stage volumetric flow was 150 to 250 times that of the inlet steam.

After 1925, electric utilities essentially stopped adding 25-cycle generating capacity. Traction systems adopted 60-cycle power, their remaining 25-cycle demand was supplied through 60/25 cycle converters. Interest in the U.S. was directed towards 1800 and 3600 rpm units, or a cross-compound combination of the two, for 4- and 2-pole 60-cycle generators (Reese, 1953).

Early design studies made during the 1920s indicated that the smaller physical size of the 3600 rpm unit gave it an advantage over the slower speed machines. Around 1920, the 3600 rpm units were constructed in maximum sizes of 7500 kW and were operated with inlet steam conditions of 250 psig (1.7 MPa) and 550°F (288°C). By 1930, the 3600 rpm single-cylinder condensing turbines were limited to a capacity of 10 MW. At this time, a tandem type 3600 rpm unit was developed for a nominal rating of 15 MW. During the 1930s, the increased pressures and temperatures made the advantages of the 3600 rpm unit even more obvious, so a single-cylinder unit was developed for a nominal rating of 20 MW. During this same period, the 50 MW, (nominal rating) 3600 rpm tanden compound unit was developed. In the late 1930s, 3600 rpm topping units were developed which became the forerunners of the high pressure (HP) elements used in the large reheat tandem and cross-compound units developed in the late 1950s (Reese, 1959) and 1960s.

CYCLES

It should be noted that the various thermal cycles employed with steam turbines were conceived and reduced to practice, prior to the application of this prime mover. For example, Weir patented a regenerative feedwater heating cycle in 1876 (Harris, 1984). Reheaters were employed by various designers and builders of multiple-cylinder steam engines with the first application attributed to Bourne in 1859 (Thurston, 1900). The reheat typically involved a steam to steam heat exchange using steam at boiler discharge condition in the receivers between cylinders of the compound or multiple cylinder engines. Steam was also supplied to the jacketed cylinder heads to reduce the cooling of steam from the cylinder walls. The discharge steam from the jackets and steam extracted from the receivers were used for feedwater heating (Hood, 1907).

The steam turbine, being a steady-flow engine, rather than a non-flow or semi-flow engine like the steam engine, is uniquely suited to the application of reheat and regenerative feedheating. Consequently, the concepts have achieved a high degree of refinement and acceptance in steam turbine practice.

Rankine Cycle

The Rankine cycle, employed in the first central station steam plants, consisted of a compression of liquid water, heating and evaporation in the heat source, expansion of the steam in the prime mover and condensation (heat rejection) of the exhaust steam. There was a continuous expansion of the steam with no internal heat transfer and only one stage of heat addition.

To review this cycle, refer to Figure 3, a representation of the Rankine cycle on a T-s (Temperature-entropy) chart; the area FABCDE represents the heat added to the cycle. The heat rejected corresponds to the area FADE. The area ABCD represents the cycle work. It can be seen that by increasing the area below line ABC (increasing the pressure and/or temperature) and by decreasing the area below line AD (decreasing the heat rejected by lowering exhaust temperature), the efficiency of the cycle can be improved.

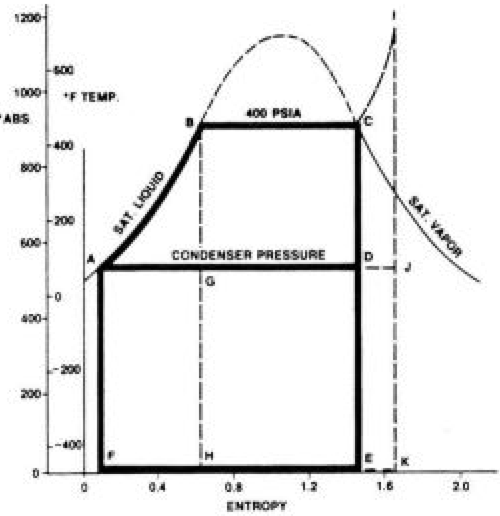

FIG 3 RANKINE CYCLE - T-s DIAGRAM

On Figure 3, line AB represents the liquid heating, line BC represents the evaporation, line CD represents the expansion and line DA represents the heat rejection of a theoretical saturated steam Rankine cycle. Since the compression power is so small, it was ignored on the figure.

The efficiency of the steam cycle can be improved if the steam were superheated, that is, heated beyond its saturation temperature. This is illustrated by the dashed line on Figure 3 where the final steam temperature is 700°F (371°C), rather than 460°F (239°C) with the saturated steam cycle. The line CI represents the superheating. The incorporation of CI has improved the ratio of the work (area above the heat rejection line) to the heat added (area below the heat added line) in the cycle, so the thermal efficiency is improved.

Regenerative Cycle

Suppose that the liquid heating line AB of Figure 3 could be eliminated. Since the ratio of the area above line AG to the area below line AB is poorer than the ratio of the area above line DG to the area below line BC, the thermal efficiency would improve if the heat addition from A to B were eliminated. The regenerative Rankine cycle is the means of eliminating all or part of the external heating of the liquid water to its boiling point.

In this cycle a small amount of expanded steam is extracted from a series of pressure zones during the expansion process of the cycle, line CD of Figure 3, to heat liquid water in a multiplicity of heat exchangers to a higher temperature, point B.

The theoretical and the practical regenerative cycles reduce both the heat added to the cycle (because the external heating of the liquid is either reduced or eliminated completely) and the heat rejected from the cycle. The cycle work is reduced as flow is continuously removed during the expansion to preheat the liquid. However, the ratio of work to heat added is improved.

Reheat Cycle

In the reheat cycle, the steam, after partially expanding through the turbine, is returned to the reheater section of the boiler where additional heat is added to the steam to increase its temperature. After leaving the reheater, the steam completes its expansion in the turbine. However, pressure losses in the reheater and reheat piping reduce the efficiency improvement.

Figure 4 is a representation of a theoretical reheat cycle on a T-s diagram. If the throttle steam at point D expands only a slight amount to point E and is reheated to F, the heat addition temperature of the reheat energy is very high. However, a very small amount of heat is added, so it has a very small effect in raising the combined heat addition temperature of the total boiler heat input, line ABCD and line EF. In contrast, an initial expansion from point D to point I (the expansion indicated by the dashed line) has a lower effective heat addition temperature, line IJ, than EF. However, much more heat is added during IJ and so it has greater weight in the total heat addition to the cycle.

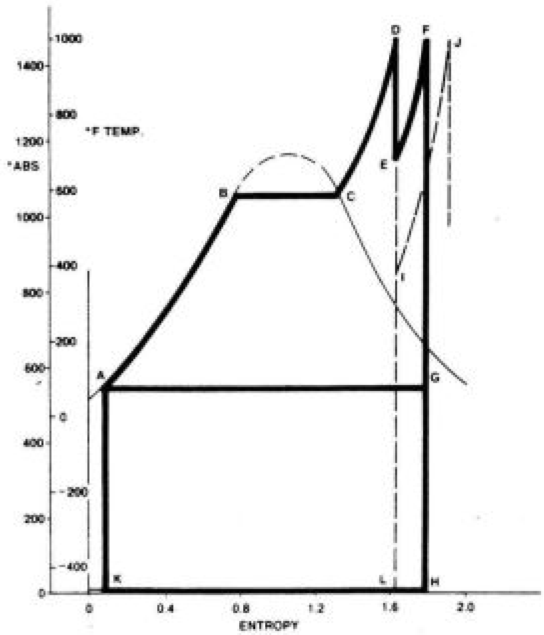

FIG 4 REHEAT CYCLE - T-s DIAGRAM

For plants with practical final feedwater temperatures and number of heaters, the optimum reheat pressure is about 25% of the throttle pressure (Reynolds, 1952). The incorporation of reheat in a practical regenerative Rankine cycle improves heat rate or efficiency by 4% to 5%, depending upon the throttle steam conditions.

In most reheat regenerative cycles, the highest pressure feed heater extracts steam at the cold reheat point. When an additional stage of feedheating is incorporated, which extracts steam above the cold reheat point, the cycle efficiency improves by 0.5% to 0.7% depending upon the throttle pressure.

The application of reheat allowed the use of higher throttle pressures than were practical with regenerative Rankine cycles by reducing the amount of moisture that is formed during the turbine expansion. For example, the highest practical throttle pressure on a regenerative Rankine cycle was 1800 psig (12.4 MPa) at a throttle temperature of 1050°F (565°C). In contrast, reheat units are in operation with supercritical throttle pressures of 3500 psig (24.1 MPa) at throttle temperatures of only 1000°F (538°C).

The widespread adoption of reheat in the early 1950s resulted in relatively rapid increases in throttle pressure from 1450 psig (10 MPa), to 1800 psig (12.4 MPa), to 2000 psig (12.8 MPa) and finally 2400 psig (16.6 MPa). Throttle temperature - reheat temperature combinations were 1000°F/1000°F (538°C/538°C), 1050°F/1000°F (565°C/538°C), and 1050°F/1050°F 565°C/565°C). A number of 2400 psig, 1100°F/1050°F (16.6 MPa, 593°C/565°C) units were also built.

Multiple Reheat. If single reheat improves efficiency, why not adopt double or even triple reheat? The theoretical improvement in efficiency from the addition of the second reheat is about half of that which results from the addition of single reheat.

Unless the throttle pressure is high enough, double reheat would result in superheated exhaust. In practice, double reheat has been employed only on plants with supercritical throttle pressures (above the critical pressure of 3208 psia (22.1 MPa). A third stage of reheat would increase the cycle efficiency by about half of the efficiency improvement obtained by the second reheat. The gains become so small that they do not warrant the added cost and complexity.

The first supercritical double reheat units had steam pressures considerably above 3500 psig (24.1 MPa) and steam temperatures well above 1000°F (538°C). The 120 MW, 4500 psig, 1150°F (31.03 MPa, 621°C) Philo 6 unit with reheats to 1050°F (565°C) and 1000°F (538°C) and the 325 MW, 5000 psig, 1200°F (34.5 MPa, 649°C) Eddystone unit with two reheats to 1050°F (565°C) were the pioneering supercritical units. U.S. utilities have purchased a total of 26 supercritical double reheat units and over 150 supercritical single reheat units in sizes ranging from 250 MW to 1300 MW.

When a second reheat is incorporated in a 3500 psig, 1000°F/1000°F (24.1 MPa, 538°C/538°C), single reheat cycle, plant efficiency increases by 1.6 to 2.0%. In each case, the highest pressure feedheater extracts steam from the cold reheat line of the first reheat. The majority of double reheat units have steam conditions of 3500 psig, 1000°F/1025°F/1050°F (24.1 MPa, 538°C/552°C/565°C), while most of the supercritical single reheat units operated at 3500 psig, 1000°F/1000°F (24.1 MPa, 538°C/538°C).

All of the aforementioned cycle variations, superheating, regenerative feed heating, increased initial pressure, and reheat, are employed to raise the heat addition temperature and thereby the cycle efficiency. An additional benefit may be a reduction in some cycle loss mechanisms.

Moisture Separation and Steam Reheat. The introduction of nuclear energy in the 1950s produced a radical change in the cycle and related hardware as compared to the advanced technology of the fossil applications. Thermodynamic concepts similar to fossil units, but with special variations, were developed to meet reliability and efficiency concerns (Artusa, 1967).

Cycle with External Moisture Removal. Because of the high moisture, 20 to 25%, that would be formed in a direct expansion to condenser pressure, the early nuclear plants used an external moisture separator at pressure levels ranging from 10% to 25% of the inlet throttle pressure. This arrangement reduced the turbine exhaust moisture to a satisfactory level of 13% from the standpoint of blade erosion and resulted in a 2-1/2% improvement in thermal efficiency over a simple regenerative cycle. Figure 5 shows a typical expansion line with external moisture removal as indicated by the dotted line from point A to point B.

The expansion line is plotted on an enthalpy-entropy, h-s diagram. The discontinuities on the expansion line between point B and point H, as shown at points C, D, E, F and G, indicate the use of internal (blade path) moisture removal devices. Similar discontinuities are shown on the other expansion lines of Figure 5. While the cycle, defined by the expansion lines connecting points A-B and B-H, is often referred to as non-reheat cycle, the removal of moisture (dotted line from point A to point B) is thermodynamically equivalent to reheating.

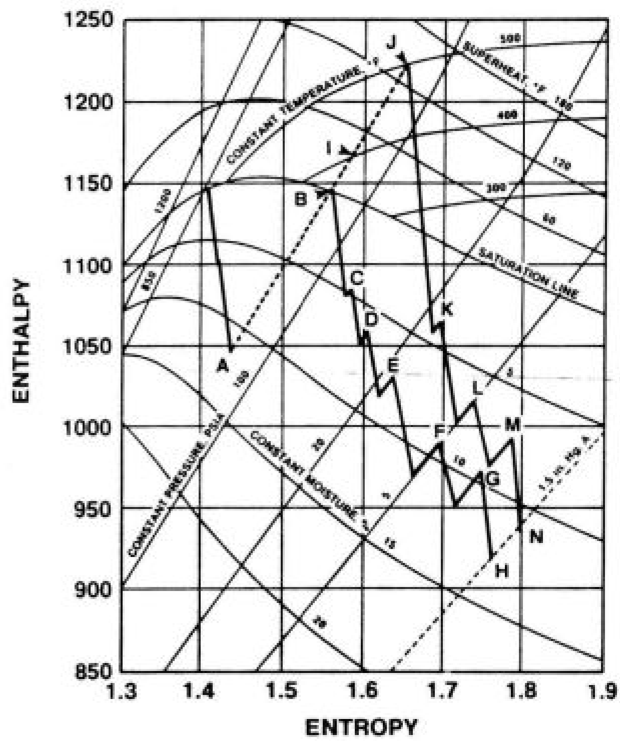

FIG 5 NUCLEAR CYCLE EXPANSION LINE

Steam to Steam Reheat Cycle. The later generation nuclear plants (and practically all current applications) made use of steam to steam reheating immediately after external moisture separation. Steam from either the throttle (single stage reheat) or a combination of throttle and extraction steam (two-stage reheat) is used to reheat the separator exhaust steam to a temperature that is about 25°F (14°C) below the saturation temperature of the heating steam.

The portion of the reheat cycle is denoted by the dotted line between points B and J. For the two-stage reheat cycle, the dotted line between points B and I relates to the the first stage (extraction steam) reheat and the portion between I and J relates to the second stage (throttle steam) reheat. With single-stage reheat the dotted line between points B and J relates to reheating with throttle steam alone.

Based solely on ideal considerations, the use of steam-to-steam reheat results in a loss in theoretical cycle efficiency or very little gain. However, the reduction in downstream stage moisture results in improved turbine efficiency. As a result, the application of single-stage reheat improves the cycle efficiency by 1-1/2 to 2% (depending upon the throttle pressure) over a regenerative cycle with external moisture separation only. The gain from single reheat can be increased by an additional 0.3 to 0.5% by adopting two-stage reheat.

MATERIALS

Metallurgy has played a key role in steam turbine development. Around 1910, inlet steam conditions were still below 200 psig (1.4 MPa) and 500°F (260°C) and the turbines were constructed of cast iron. The adoption of cast and forged steel parts after 1912 paved the way for increasing inlet steam conditions quickly. Alloy development with consideration of long-life, high-temperature service got under way around the mid 20s, as did the development of the low-carbon, 12% chrome alloy (Mochel, 1952).

By the early 1920s, throttle pressures were 200-300 psi (1.4 - 2.1 MPa), and throttle temperature increased to 600°F (316°C). As temperatures rose above 800°F (427°C), creep became a major material problem. Among the many alloying agents investigated, molybdenum proved effective. Using carbon-moly steels, designers pushed temperatures beyond 900°F (482°C) by the late 1930s.

To go from 950°F (510°C) to 1000°F (538°C), chromium was added to the carbon-molybdenum alloy. To increase the throttle temperature from 1000°F (538°C) to 1050°F (565°C) turbine manufactures considered the use of austentic steel. Austentic steels were used for the high-temperature components, as applicable, for the Public Service Electric and Gas Co.'s 2400 psig, 1100°F/1050°F (16.6 MPa, 593°F/ 565°F) units which were placed in service in the early 1960s (Baker, 1962) and Philadelphia Electric Co.'s Eddystone 1 unit. For example, on Eddystone 1, austentic steel was utilized for the inlet stop and control valves, inlet piping, nozzle chambers and inner cylinder of the superpressure (SP) turbine. The outer casing operating at 1000°F (538°C), is ferritic steel.

It should be noted that the 215 MW Westinghouse supplied Cleveland Electric Illuminating Company's Avon 8 unit, 3500 psi, 1100°F/1050°F (24.1 MPa, 593°C/ 565°C) was designed specifically to utilize conventional CrMoV steel in those features exposed to 1100°F (593°C). The CrMoV rotor was later replaced with a 12Cr ferritic rotor to eliminate oxidation. With proper design considerations and accumulated manufacturing experience and fabricating experience, it should be possible to use ferritic material at 1100°F (Franck and Carlson, 1961).

In an assessment of Eddystone 1, after it had been in operation for twenty-four years, it was noted that most of the troubles encountered in the the plant were not caused by the high pressures and temperatures (Chamberlain, 1983). A number of the difficulties were in areas where limited or no experience existed. The chief metallurgical reasons for derating Eddystone 1 in the early 1960s were fireside corrosion in the boiler and thermal fatigue cracking in the boiler stop valves (Jaffee, 1979). Excessive fireside corrosion, experienced in the superheater and reheater surfaces of the 1200°F (649°C) boiler, was brought under control by reducing the steam inlet temperature to 1130°F (610°C). The pressure was reduced from 5000 psi (345 MPa) to 4800 psi (33.1 MPa) to keep the steam flow to the SP turbine constant. Boiler stop valve cracking was eliminated after the valve was redesigned and the material changed.

In 1982, the stop and control valves of the Eddystone 1 SP turbine were replaced after 130,000 hours of operation. Availability of these components represented an unique opportunity to study the long-term metallurgical changes and the mechanical deformations which occurred in the austentic steel turbine valve parts (Argo et al., 1984).

Since the early 1960s advanced steam conditions have not been pursued. In the 1960s and early 1970s, there was little motivation to continue lowering heat rates of fossil-fired plants due to the expected increase in nuclear power generation for baseload application and the availability of relatively inexpensive fossil fuel. Therefore, the metallurgical development required to provide material "X" for steam conditions higher than Eddystone was never undertaken.

It was projected (Franck, 1957), that when material "X" became available, that steam conditions of 7500 psig, 1400°F/1200°F/1200°F (51.7 MPa, 760°C/ 649°C/649°C) would be used for the next advancement in steam plant design. The estimated plant heat rate was 7700 Btu/ kWh (8125 kJ/kWh).

Advances in metallurgy can be used to improve the reliability and efficiency of today's steam turbines. For example, a new high-temperature material developed for 1100°F (593°C) application could have applications in existing or future turbines operating at temperatures below 1100°F (593°C). As a specific example, development of clean steels, produced by secondary refining techniques, has increased the reliability of low pressure (LP) turbines (Jaffee, 1986). Today integral disc rotors made by forging or welding have replaced shrunk-on nuclear LP rotors which are subject to stress corrosion cracking at the keyways that hold the discs to the rotor shaft.

EFFICIENCY IMPROVEMENTS

Cycle Effects

As noted in the discussion of cycles in an earlier section, the use of regenerative feedwater heating, Figure 3, increased the cycle efficiency. However, since only a finite number of stages of feedwater heating are feasible and each incremental increase in final feedwater temperature produces a smaller improvement in efficiency, there is both a thermodynamic and economic optimum feedwater temperature corresponding to a given throttle pressure. See Figure 6.

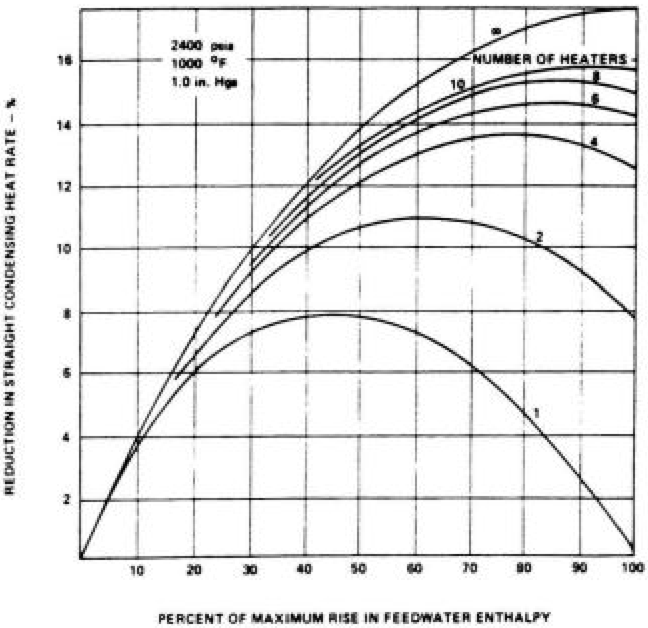

FIG 6 REGENERATIVE CYCLE HEAT RATE EFFECT

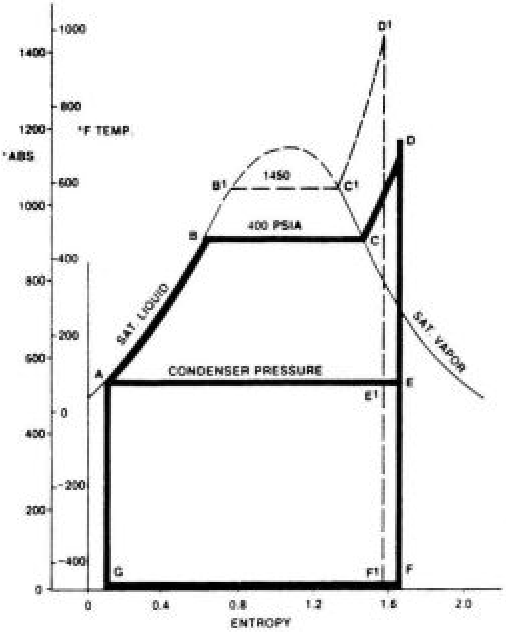

FIG 7 THROTTLE PRESSURE VARATIONS - T-s DIAGRAM

On actual rather than ideal cycles, with a given unit rating, feedwater heating results in an increase in turbine inlet flow and a reduction in exhaust flow. The end result is an increase in efficiency of the initial stages because of reduced leakage and secondary flow losses associated with the larger blade heights as well as an improvement for the exhaust stage (lower leaving loss).

Steam Conditions

Figure 7 illustrates the effects of raising inlet pressure and temperature. Increases in both pressure and temperature increase the cycle available energy and, therefore, the ideal efficiency. However, increases in pressure reduce the blade heights of the initial stages and decreases efficiency, offsetting a portion of the ideal improvement, unless unit rating is increased commensurately. Consequently, increases in throttle pressure must be judiciously evaluated.

A recent paper (Silvestri et al., 1988) presented possible heat rate improvements for increasing steam pressure and temperature to such elevated conditions as 10,000 psi, 1400°F/1400°F/1100°F (70 MPa, 760°C/760°C/593°C). Based upon the potential heat rate improvements, there was no reason to elevate the turbine steam conditions beyond 7000 psi, 1400°F/1400°F/1100°C (48.3 MPa, 760°C/760°C/593°C).

For the past 10 years, Electric Power Research Institute (EPRI) has funded a series of programs aimed at determining the feasibility of developing, designing, and building an advanced pulverized coal steam plant. Westinghouse, involved in one of the initial studies, recommended a turbine-generator designed for steam conditions of 4500 psi, 1100°F/1050°F/1050°F [31 MPa, 593°C/565°C/565°C] (Bennett et al., 1981). These steam conditions represent a modest departure from today's current practice (Bannister et al., 1982).

Increases in throttle and/or reheat temperature, as well as incorporation of reheat, result in reduction of turbine exhaust moisture and consequently moisture losses, which are not reflected in the ideal cycle representations of Figures 3, 4, and 7. Moisture separation and steam to steam reheat increases the efficiency of LWR cycles because of the reduction in moisture losses.

Size

Increases in unit rating result in lower leakage and secondary flow losses in the initial stages. This is reflected in the lower maximum throttle pressures that are thermodynamically and economically feasible on smaller size units.

Improved Components

The major improvement in turbine cycle efficiency, neglecting cycle and steam conditions effects, has resulted from blading improvements. The blading improvements have resulted from reductions in profile losses, end-wall losses, secondary flow losses, leakage losses (flow bypassing the blading and disturbance effects when the leakage re-enters the blade path), and incidence losses.

Utilization of tapered twisted designs (vortex blading) for the longer blades reduced the incidence losses on the innermost and outermost portions of the blades.

Improvements in exhaust hoods have eliminated the positive static pressure loss between the last stage exit and the turbine exhaust flange (Seglem and Brown, 1960). Current designs recover some of the velocity leaving the last stage so that the blade exit static pressure is lower than the static pressure at the exhaust flange (Brown et al., 1963).

The two classical types of blading, impulse and reaction, correspond to stages with 0% pressure in the rotating blade and 50% of the stage pressure drop in the rotating blade. Actual impulse stages incorporate a small amount of reaction at the base of the rotating blade which increases with increasing diameter. In LP stages with long blades, the differences between impulse and reaction stages are academic, because of the resulting low reaction at the base and high reaction at the tip. Stages with substantial variations in reaction are often called vortex blading.

The different design philosophies, impulse versus reaction, are more correctly identified by the drum versus disc and diaphragm construction, rather than the classical impulse versus reaction classification.

Constant vs. Sliding vs. Hybrid Throttle Pressure Operation

The comparative performance of full-and partial-arc turbines for constant and sliding pressure operation is shown in Figures 8. Also shown is the performance of the partial-arc design with hybrid operation (Silvestri et al., 1972). Traditionally U.S. units have been partial-arc designs operating at constant throttle pressure. The increased need to cycle units has led to the widespread adoption of sliding pressure operation on new units and backfitting on older units. About 170 units in the U.S. have been designed or modified for sliding pressure operation. All but about 20 of these units employ hybrid operation.

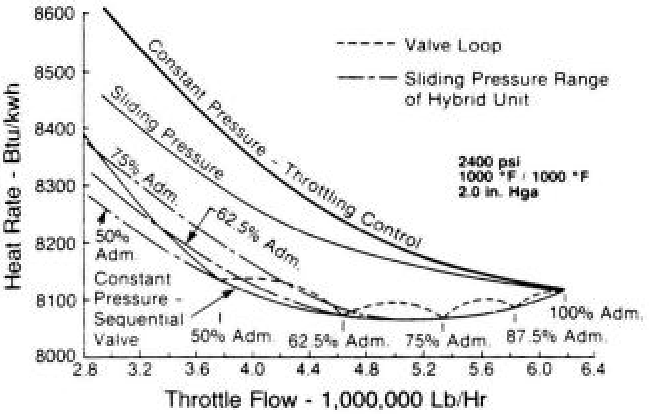

FIG 8 HEAT RATE COMPARISON OF THROTTLE PRESSURE VARIATIONS

In addition to the efficiency improvements associated with sliding pressure (combination of effects related to cycle available energy, HP element part-load efficiency, feedpump and feed-pump drive power and boiler temperature droop), there is less change in first stage temperature during load changes, resulting in lower thermal stress as shown on Figure 9.

Vortex Blading

While the total pressure at inlet and the static pressure at exit of a stage are essentially constant along the blade height, the pressure between the stationary and rotating blades increases from the base to the tip as shown on Figure 10. Centrifugal force

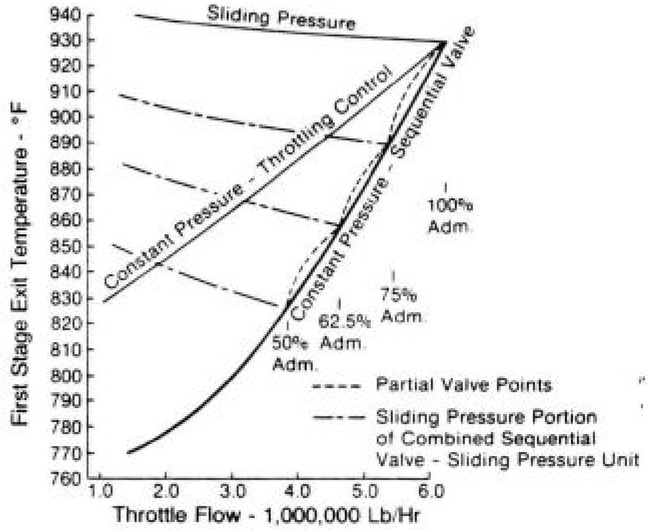

FIG 9 FIRST STAGE EXIT TEMPERATURE CHARACTERISTIC

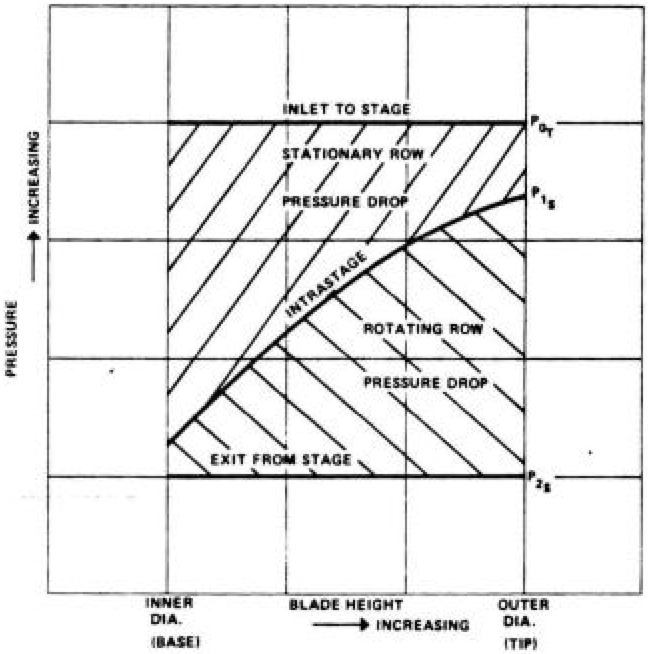

FIG 10 PRESSURE DISTRIBUTION OF A TURBINE STAGE

acting on the steam leaving the stationary row tends to accelerate the steam toward the tip while radial equilibrium results in a resisting pressure force, acting in the opposite direction. Consequently, the pressure increases as one moves from base to tip in order to maintain radial equilibrium.

This is true for any stage. Since the pressure drop is greater at the base, the exit steam velocity from the stationary blade is greater at the base than at the tip. Because of this variation in pressure drop or reaction, especially on the longer blades, the rotating blades are twisted to match the resulting variations in inlet steam direction.

Tuned vs. Untuned Blades

The stages in the HP, intermediate pressure (IP) and most of the LP elements have untuned blades in which material damping limits the vibratory build-up due to resonance. The untuned blades are designed for reliable operation, even if run under resonance conditions in the fundamental modes of operation. The untuned blades may be parallel section designs or may be tapered twisted vortex designs, depending upon the blade length and the base to tip ratio of the stage.

Tuned blades have low fundamental frequencies and are long, tapered, twisted and generally used in the last three rotating rows of the LP element. Efficient aerodynamic design is incompatible with the strength required to withstand resonance. Therefore, the blades are tuned so that the lower natural frequencies avoid integral multiples of running speed. Tuning may be accomplished by joining the blades in groups by shrouds and/or lashing wires. The shrouds or lashing wires also limit the vibration amplitude of the individual blades, thereby reducing the stress level.

In some designs, tuned, free-standing blades (no shrouds or lashing wires) are used. Free-standing blades have much larger chords and a larger radial taper than designs with lashing wires or shrouds, and do not experience the efficiency losses that result from lashing wires.

The length of the last-row blade and the corresponding exhaust annulus of the stage, which has typically increased in increments of about 25%, is shown in Table 1 for the majority of currently operating units.

TABLE 1

3600 RPM DESIGNS

Blade Length (in.)	Exhaust Angulus Area (ft^2)
20	26.2
23-23.5	32.9-33.9
25-26	41.1-43.4
28.5-30	53.7-55.6
31-34	66.1-67.4

1800 RPM DESIGNS

Blade Length (in.)	Exhaust Angulus Area (ft^2)
38-40	95.3-105.7
43-45	123.8-131.3
52	172.4-176.4

NUCLEAR TURBINES

Westinghouse built the first turbine-generator unit for nuclear power application, which was placed in service in 1957 at Duquesne Light Co.'s Shippingport Station. Initially rated at 60 MW, it had a maximum turbine capability rating of 100 MW . This unit was the first of a series of 1800 rpm units which were developed from a base design and operating experience with fossil machines dating back to the 1930s.

Inlet steam conditions for Shippingport at maximum load were 545 psig (3.8 MPa), dry and saturated, with a flow of 1,300,000 lb/h (590,200 kg/h). This unit, Figure 11, was a single-case machine with a 40-in. (1016 mm) long last-row blade.

Pressurized water-cooled reactors supply dry and saturated steam in the pressure ranges of 400 to 1000 psig (2.8 to 6.9 Ma). The major problem associated with these conditions is the increasing amount of moisture as the steam expands. If the steam at dry and saturated conditions was allowed to expand normally, the moisture content at the turbine exhaust, as discussed previously, could be 20 to 25%, depending upon the inlet pressure and vacuum conditions. Excessive moisture contributes greatly to both turbine blade erosion and blade efficiency losses.

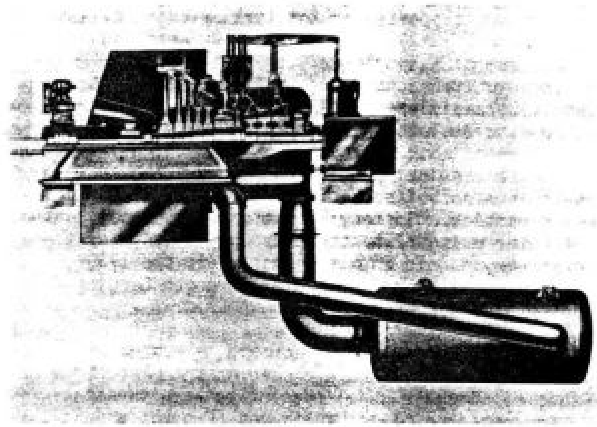

FIG 11 DUQUESNE LIGHT CO.'S SHIPPINGPORT UNIT

By 1973, thirteen 1800 rpm Westinghouse nuclear turbine-generators, three of which had ratings in excess of 800 MW, were in operation. Eventually, Westinghouse placed fifty-seven 60-Hz and six 50-Hz nuclear turbine-generators into operation with the largest unit having a 1325 MW output.

The low inlet pressure and comparatively small heat drop resulted in large volumetric flows at both the inlet and the exhaust. The large physical size of an 1800 rpm turbine provides the necessary inlet and exhaust annulus area.

For maximum cycle efficiency, a mechanical moisture separator should be used at the turbine stage which give nearly identical moisture percentages at the exit of the HP and LP turbine elements (Carlson, 1955). The separator for the Shippingport unit was a stationary, centrifugal type. The quality of steam entering the LP element was in the order of 99 percent.

The second generation of nuclear turbines provided reheat at the crossover zones for improved thermal performance. This design combined the moisture separator and reheater, using a mechanical type separator followed by a shell and tube reheater which extracted steam from ahead of the main stop valve for heating.

While the earlier designs used wire mesh pads for the separation stage, subsequent designs have used chevron vanes. The steam is reheated to approximately 100 to 135°F (56 to 75°C) superheat.

Westinghouse's nuclear LP turbines use both 40-in. (1016 mm) and 44-in. (1118 mm) last-row blades. Experience with the 40-in. (1016 mm) blade dates back to the early 1940s. The first 44-in. (1118 mm) blade was introduced in 1959 on the Eddystone 1 (Campbell et al., 1957). Through development testing and field investigation, including telemetry tests in the field, a new generation of 44-in. (1118 mm) blades was developed in the early 1980s. Included in the developed process was the creation of the ruggedized design 44-in. (1118 mm) LP turbine blades (Pigott and Warner, 1986 and Davids et al., 1988).

A nuclear turbine's control stage and nozzle chambers are designed to permit partial arc admission through sequential operation of the control valves. All stationary blades are carried in separate blade rings which are supported in the turbine outer cylinder (Williamson, 1973). Low inlet steam conditions permit the use of a single-wall cylinder design. All blade rings located in the high-moisture zones are provided with stainless steel cladding and moisture removal drainage slots.

In the LP turbine, the inner cylinders are supported independent of each other to permit freedom of thermal expansion. In the 1960s and 1970s, a built-up type of rotor using high-strength alloy discs, was shrunk and keyed onto an alloy shaft. In the 1980s this design was replaced by a high-strength forging with integral discs.

Since moisture is a major concern, a number of erosion control techniques are used in the LP turbine. For example, adequate axial spacing between the stationary and rotating blades minimizes LP blade erosion. Moisture is removed at all extraction points in the moisture region. Moisture in these areas is removed by centrifugal force towards the outer blade surfaces where it accumulates and is carried off by the extraction steam flow.

Nuclear turbines designed for use with a boiling-water reactor will be radioactive. The possibility of building up radioactivity in the turbine due to the accumulation of corrosion products over a period of time is real. Also, a fuel rod rupture could result in highly radioactive materials entering the turbine. Therefore, internal wiredrawing-type leakage paths ordinarily unimportant in normal steam turbine design have to be eliminated to the greatest practical extent. Where it is impossible to eliminate, the surfaces forming the leakage paths should be faced with erosion resistant materials deposited by welding.

CONTROLS

Turbine speed control was originally achieved with a fly-ball governor with subsequent addition of springs. Later refinements, relayed governor for example, substituted oil pressure from a servomotor controlled by a pilot valve to move the valves. The hydraulic governor used a special centrifugal pump instead of flyweights as the speed-sensing element. The relayed governor and the hydraulic governor are referred to as mechanical-hydraulic types. Current governors and control systems are electrohydraulic designs that utilize digital computers.

Throttle, governor, interceptor and reheat stop valves are opened by hydraulic cylinders while springs are used to achieve closure. Over the years a number of other systems have been added to the general category of controls.

Controls may be identified as protective or emergency systems and regulating systems for controlling flow (load) or speed either individually or in combination. Designated instruments or sensors provide the information for control actuation, equipment shutdown or activation of alarms. Moreover, some control systems perform both the protective and the regulating function.

In addition to turbine valving, protective systems can activate alarms or trips, which relate to low condenser-vacuum, exhaust temperature, low bearing and main oil pressure, rotor position, differential expansion, and overspeed. During start-up and load changes, thermocouples monitor metal temperature so that loading rates and the accompanying thermal stresses can be monitored and controlled.

With the application of reheat, interceptor and reheat stop valves were added to the throttle and governor valves used at the main steam inlet because of the large stored steam volume in the reheater and associated piping. The interceptor valves of fossil units are plug types, like the throttle valves, while the reheat stop valves are clapper designs. Some LWR turbines employ fossil type interceptor valves at the LP turbine inlet while other designs uses butterfly valves as both reheat stop and interceptor valves.

To regulate flow and consequently load, turbines use either throttle control where all of the governor valves and the associated first stage nozzles are active at all loads (full-arc admission) or nozzle control where the valves operate sequentially and supply individual sectors or groups of nozzles (partial-arc admission).

Nozzle control results in superior part-load performance, about 95% of maximum load and below, while full-load performance is slightly poorer than throttle control. The more severe operating environment of the nozzle control designs results in poorer blade aspect ratios as well as additional losses associated with operation when the active arc of admission is below 100%.

Operation with sliding or variable throttle pressure significantly improves part-load performance of full-arc admission designs. Part-load performance is also improved on partial-arc admission design by a combined or hybrid mode that involves high-load operation at constant throttle pressure with sequential valve operation. As load is reduced, when a particular valve point is reached, further load reductions are achieved by holding valve position constant and varying throttle pressure. The optimum transition point for transferring from sliding to constant pressure operation occurs when half the governor valves are wide open and half are closed, 50% admission, and corresponds to about 70% load.

UNIT SIZE

Installation of very large capacity turbine units once carried along an element of prestige and glamour. It could be justified on large systems for genuine investment economy. Concentration of capacity in a small number of very large units led to savings for buildings, machinery foundations, piping, condensing plant and switching.

Many factors combined to make unit size growth possible, a major item being the development of new and larger exhaust stage blading, and the more extensive compounding of such last stages for parallel low pressure steam flow paths.

Large volumetric steam flows made possible the application of double-flow steam path design, and the resulting elimination of the problems of balancing steam pressure thrust forces. Furthermore, the large volumetric flows enable the use of longer, more efficiently proportioned blades which contribute to improved overall performance.

Turbines with large volumetric steam flows usually require fewer stages of blading than smaller capability units operating with the same steam conditions. The larger volumetric steam flows permit optimized blading to be used at larger mean diameters, resulting in higher blade speeds and more desirable stage velocity ratios with fewer stages. Fewer stages avoid having to excessively increase the shaft bearing centers, even though the large volumetric steam flows and larger heat drop per row require the use of wider, heavier blade sections.

The weight-per-kilowatt for large turbines compared with smaller ones is markedly reduced. To illustrate this, the weight-per-kilowatt for a 1000 MW unit is 5.57 lb (2.53 kg) or 30 percent less than the corresponding 7.90 lb (3.59 kg)/kW for a 380 MW unit.

Remember that the first commercial steam turbine-generator in the U.S. had a rating of 1.5 MW. In the 1910 to 1920 period, Westinghouse was manufacturing units in the 30 MW to 70 MW range (Johnson, 1919). By 1945, however, the median size unit sold in the U.S. was still only 100 MW. By 1967 the median size unit had increased to 700 MW with a peak of 1300 MW for several fossil units placed into service in the 1970s.

Westinghouse's largest nuclear unit has a 1325 MW output. Westinghouse has built 850 MW tandem-compound and 1080 MW cross-compound fossil steam-turbine generators.

DESIGN ANALYSIS/PRODUCT PERFORMANCE

The past fifteen years has been a period of significant advancements in the technology of steam turbine-generators. Major advances have been made through the coupling of laboratory testing, field testing, and operating experience with mathematical modeling and computer-aided design.

Westinghouse uses three basic types of computer programs: design, analysis and performance programs. The design programs are generally overly complicated for analysis purposes. Appreciable reductions in computer calculation time can be realized with simpler models to analyze the equipment under design and off-design conditions. Many analysis programs are suitable for performance predictions in the myriad application of a particular component (e.g., a particular blade style). Finally, performance prediction programs, which sacrifice some accuracy, are economical, run quickly and are simple to use. Each type meets the needs of particular classes of users.

For example, in designing untuned blades, which employ standardized blade profile, using a variety of widths and lengths (with individual manufacturers using a variety of blade styles), performance predictions are based on cascade tests of a particular row or rotating tests of a group of stages. Then, mean diameter simulations will very accurately represent the performance of untuned blades. Blade lengths, profiles and orientation of the tuned blades vary considerably from stage to stage. Therefore, performance prediction methods, are based on modeling of loss mechanisms derived from cascade tests and rotating tests.

Untuned blades can also be designed individually and applied in groups between extraction points. Before the introduction of electronic computers, design procedures solved the design parameters of tuned blading stage-by-stage. With computers a more complete solution of the three-dimensional flow field within turbine elements is made. Design requirements such as numbers of stages, blade lengths, inlet and exit angles, and flow areas are more precisely determined. Better matching of fluid angles and blade angles reduces flow losses and excitation, and has resulted in improvements to both reliability and efficiency.

Another significant design advance is the use of finite element analysis. This procedure, which allows the accurate determination of stresses, deflections, vibratory frequencies, and vibratory mode shapes of blades and other structural components, has been verified in our laboratory. Also, calculated deflection pattern of a blade for a complex mode of vibration has been compared to the visualization of that same mode as determined through a laser beam hologram.

Thermal performance programs, such as the Generalized Performance and Heat Balance Program, are used to predict turbine cycle performance by modeling blading performance and the operating characteristics of equipment typically depicted on a heat balance. Both design and off-design performance can be modeled realistically for a variety of blading types and blading performance prediction models of varying complexity.

During the last decade, Westinghouse has used a ruggedized design approach to HP, IP (Conway et al., 1986) and LP turbines (Pigott et al., 1982). Ruggedized LP turbine blading has more robust proportions than earlier generations and the ability to operate over a wider range of operating conditions (e.g., exhaust pressures up to 8.0 in. HgA [0.271 bars] as compared to earlier designs that were limited to 5.0 to 5.5 in. HgA [0.169 to 0.186 bars]) with greater reliability and superior thermal performance. Many of today's LP designs have free-standing rotating blades in the L-0 and L-1 stages.

With a computer, a myriad of activities related to the design, analysis, and testing of the various turbine components, and the prediction of their performance during operation can now be made. For example, an instrumentation package is available today that will monitor the static and dynamic history of a LP turbine rotor, and then calculate its expected service life based on its power plant operation (Bannister et al., 1983).

Among the more leading edge technologies that have become viable tools because of computers is on-line diagnostics based upon artificial intelligence (AI). In this instance, rule bases are used to monitor equipment operation and alert operators if equipment distress is present, allowing corrective actions to be taken before serious damage occurs.

In order to achieve this, an on-line diagnostic system, located at the power plant, gathers turbine-generator data using an advanced monitoring system referred to as the Data Center. The Data Center transmits the information to the centralized Westinghouse Diagnostic Center (located in Orlando, FL) where AI programs translates the sensor inputs into a diagnosis, and a recommended corrective action, if an abnormal condition is diagnosed.

Today, there are a total of 14 individual on-line AI based diagnostic systems in commercial operation or on order, applied to old and new units, both fossil and nuclear. Westinghouse provides the centralized AI service to their utility customers 7-days-a-week, 24-hours-a-day. AI programs available include turbine and generator diagnosis, controls, and nuclear secondary and fossil steam chemistry.

PRODUCT AND DEVELOPMENT TESTING

There has always been close cooperation between utilities and the manufacturers of steam turbine-generators in developing improved designs. Westinghouse has used the laboratory and the field to verify existing and new designs, and generate new technology. A major step in laboratory facilities occurred in 1955 with the opening of a laboratory devoted towards research and development work on central station turbines (Fischer, 1955).

Air cascade test facilities have allowed us to evaluate potential blade airfoil shapes for efficiency and to evaluate effects of parameters such as pitching, airfoil shape, surface roughness, and edge thickness on flow losses. The surface waves of flowing water has been used to simulate two-dimensional unsteady gas flow in turbine blading. When applied in a rotating stage, unsteady forces can be evaluated and interactive effects identified in a rotating water table analog.

As a recent example, during the 1970s, it became evident that the turbine environment itself was a major cause of some blade and disc problems. Fatigue testing of blading and rotor materials in various laboratory environments was expanded to adequately define material properties. Theoretical studies and field probe tests verified that harmful concentrations of corrodents can occur in the blade path at the point where initial condensation takes place (called the salt solution zone).

Due to the large size of modern turbines, it is impractical to provide full-scale testing of all designs. This is especially true for large LP turbine designs, since the steam flow required and the powers developed prohibit such testing. Westinghouse, however, took an industry leading position on this question in the 1970s by using an unique combination of both model and selected full-size testing which not only utilized the theory of turbine scale model testing, but also answers the question of how to handle the unscalable variables (e.g., moisture).

During the 1970s, Westinghouse developed the design for two new LP blading test facilities. One facility was used to verify final product design, while the other was used to screen and test LP blading designs prior to commitment for use in the product. The product verification facility was placed into operation during 1975 at the Chester plant of the Philadelphia Electric Co. With a steam supply of 750,000 lb/hr (340,500 kg/h), LP turbines with blade lengths of up to 40-in. (1016 mm) were evaluated at speeds up to 4320 rpm (Steltz et al., 1977).

To verify tuned blade integrity, various measurement techniques were developed to establish resonant frequencies. Shakers are used to excite blades, or frequencies can be measured following an impact. But these techniques can only establish static frequencies which differ from rotating frequencies (Owens and Trumpler, 1949). A major step at Westinghouse in 1934 involved use of an optical system for measuring blade vibration during turbine operation. It used a beam of light that was directed into a hollow shaft and then reflected radially through a transverse hole to a small mirror mounted near the tip of a blade. This blade-mounted mirror reflected the light back to the central mirror and out through the shaft to a viewing screen or photographic film (Kroon, 1941).

General advancements in instrumentation by the mid-1950s dictated the use of strain gages to measure rotating blade vibration. Transmission of the signals from the rotating shaft to the analysis instrumentation was accomplished through a slip ring assembly.

Telemetry was first introduced at Westinghouse in 1958. At that time, the transmitter was very large, about 3-in. (76 mm) in diameter and 12-in. (305 mm) long. It could be installed only in the center of the turbine shaft, which for field application, represented little improvement over the use of slip rings.

With the introduction of miniature transmitters that were 1/2-in. (12.7 mm) in diameter and length, interest was again generated in developing the application for steam turbines (Donato et al., 1981). Development progressed sufficiently to permit field testing of 3600 rpm machines, on which the majority of the tests were conducted over the past fifteen years. With the advent of computers and the availability of Fast Fourier Analysis, the analysis and interpretation of the vibration data was simplified.

Recently, Westinghouse has developed a blade vibration monitor (BVM) which is used in the field to measure the vibration of rotating blades. Using two sensors per blade row (mounted radially over the rotating blades), the BVM system will sense axial and tangential deflections. The ability to observe the motion of all the blades, in an instrumented row, now exists.

LIFE EXTENSION

Today, with U.S. utilities having to rely on older steam turbine-generators, for both base load and cycling, retrofitting existing units is an attractive utility option for extending the life of existing units versus installing new capacity. By incorporating new technology into existing designs, it is possible not only to extend the unit's life, but also increase its availability and thermal efficiency (Barsness et al., 1984).

Since 1983, Westinghouse has performed turbine-generator life evaluation/ modernization studies for over 100 units. This does not include outage-related concerns which require immediate engineering evaluation. The units have ranged in age from 18 to 35 years and in size from 17 MW to over 1000 MW (Davis and Dayal, 1987). The life evaluation scope has ranged from one critical component, such as a turbine rotor or cylinder, to total plant remaining life assessment.

Although each unit evaluation has been unique, most often the candidate unit has seen 5 or more years of cyclic duty (two-shift or load-follow) and is expected to see at least an additional 20 years of cyclic operation. The typical candidate unit has performed reliably with acceptable efficiency for the greater part of its operating life.

Westinghouse's experience in performing turbine-generator life evaluations has indicated that life extension for mature units is both technically feasible and economically attractive. For example, a Westinghouse retrofit of a 15-year coal-fired plant resulted in a power output increase of nearly 20%.

CURRENT AND FUTURE TRENDS

During the late 1960s, in the U.S., there was a retrenchment in steam temperatures with practically all units using maximum steam temperatures of 1000°F (538°C), the notable exception being double reheat units with ascending steam temperatures of 1000°F/1025°F/1050°F (538°C, 552°C, 565°C) (Bannister et al., 1987). The last double reheat units were installed in the 1970s. Most of the capacity ordered in the last 10 years has had steam conditions of 2400 psig, 1000°F/1000°F (16.6 MPa, 538°C/538°C).

The slow growth in electrical demand since 1973 has impacted the purchase of large units. As the result of a number of concerns (Bennett et al., 1981), including concerns about the availability and reliability of large units, current trends indicate that most utility unit sizes for fossil plants will be in the 150 MW to 500 MW output range in the foreseeable future.

If EPRI and other organizations are successful in developing the technology for viable turbine designs at pressures in the 4500 to 5000 psi (31 to 34.5 MPa) range and steam temperature in the 1100°F to 1200°F (593°C to 649°C) range, unit sizes must increase to at least 450 MW to retain a substantial fraction of the improvement resulting from the increased pressure and temperature. More realistic units size would be 700 MW and above.

Today, the long power plant construction cycle, which adversely affects the total financial investment, and the uncertainties of the regulatory process have led to a standstill in commitment for new nuclear and large fossil plants in the U.S. While research efforts are proceeding on advanced LWRs and simplification of the reactor system, the reluctance to commit new nuclear capacity remains because of the uncertainty with the licensing process.

Enactment of the Public Utility Regulatory Policies Act (PURPA) of 1978, new environmental requirements and other regulatory and economic developments have altered the ways in which U.S. utilities function. Also, the scrutiny of purchasing and system decisions have placed increasing cost pressures on utility operations. The days of utilities being able to pass all construction and operating costs on to the consumer are gone forever.

Cogeneration, which is the sequential production of power and process heat from a single fuel by using some of the heat that is left over from power generation, is back in vogue again due to PURPA. This act requires utilities to purchase power from the independent power producer at the utilities' avoided cost (cost to a utility to construct and generate or purchase the power from another source). Nominal ratings for a steam turbine-generator used in a cogeneration cycle are 20 MW to 180 MW. At 20 MW inlet steam conditions are 600 psi, 750°F or 825°F (4.1 MPa, 399°C or 441°C). At 90 MW, inlet steam is 1450 psi, 950°F (10 MPa, 510°C). At 180 MW the steam conditions are probably 1800 psi, 1000°F/1000°F (12.4 MPa, 538°C/538°C).

Combined cycle plants, in which the exhaust gas from combustion turbine-generators is used in a heat recovery boiler to produce steam at nominally 1200 psi, 900°F or 950°F (8.3 MPa, 482°C or 510°C) to drive a steam turbine-generator, is another option for power producers today.

Combined cycles may be designed so that approximately two-thirds of the total plant power output is generated by the combustion turbines and one-third by the steam turbine. Nominal steam turbine-generator ratings are between 50 MW to 150 MW. Combined cycles can also be operated with direct combustion of coal in a pressurized fluid bed (PFB), or in an atmospheric fluid bed (AFB). Steam conditions for a turbine in this cycle could be as high as 2400 psi, 1000°F/1000°F (16.6 MPa, 538°C/ 538°C). The AFB has also been applied on conventional steam plants to reduce stack emissions.

The trend, therefore, in the foreseeable future for new steam turbines is for moderate inlet steam conditions and capacity. The element of industry prestige and glamour, once associated with elevated steam conditions and large capacity turbines, has disappeared.

CONCLUSIONS

As the nation's dependence on electric energy began in 1882 at the Pearl Street Station, the steam engines used in the first U.S. central station required 10 lb (4.5 kg) of coal to produce a kWh. As the first steam turbine central stations were built, the coal rate dropped to around 8 lb (3.6 kg) to produce a kWh. However, by 1932 the coal rate had dropped to 1.49 lb (0.68 kg). In 1947, only 0.84 lb (0.38 kg) was required to produce a kWh. Today, a coal rate of 0.71 lb (0.32 kg) is achievable for a station thermal efficiency of 40%.

The 20th century has been a period of major changes in the size, steam conditions and cycles applied with steam turbines. Today, a fossil-fueled turbine will use 7,000,000 lb (3,178,000 kg) of steam-per-hour while a large nuclear steam turbine could require 17,000,000 lb (7,718,000 kg) of steam-per-hour.

Many of the early units were single-case designs, but with the increase in unit size, multiple-flow exhausts were adopted with separate shells for the HP and LP sections. The number of multiple exhausts has been increased from two to as many as eight on some cross-compound 3600/3600 RPM designs. The concept of multiple-flows was extended to the IP section on larger reheat units, and in some cases, units above 600 MW, even the HP section has two parallel flows.

Turbines built at the beginning of this century could generate 2 MW of electricity, less than the requirements for a single multi-story office building today. By the late 1950s, unit size had reached 325 MW and several fossil-fired units in the 500 MW to 800 MW size range were in operation by 1970. A number of fossil units in the 900 to 1300 MW size range became operational in the 1970s. (A 1300 MW unit can generate enough electricity to supply the residential needs of over 4 million people.)

Nuclear turbines, excluding the early demonstration plants, have ranged in size from 400 to 1325 MW and used multiple LP exhausts, four in the lower size range and six in the larger sizes. While some units in the 400 to 500 MW range used a single-flow HP section, most have adopted double-flow configurations for this element.

Over the last nine decades, Westinghouse and other U.S. steam turbine manufacturers have made many step changes in the basic design of central station steam turbines. New technology and materials were developed to support the industry's elevation of steam conditions, optimization of thermal cycles and unit capacity. Even though there is a return in the industry to more moderate steam conditions and unit size, improvements in turbine design procedures and verification enhance a manufacturer's ability to design new and modified turbine elements for specified operating ratings, cycles and back pressures, with a high degree of confidence considering both performance and reliability.

Westinghouse analytical procedures have been validated and calibrated through the use of many specialized laboratory test facilities. Turbine elements have been verified by scale-model and full-scale tests in laboratory test vehicles, as well as by extensive field testing.

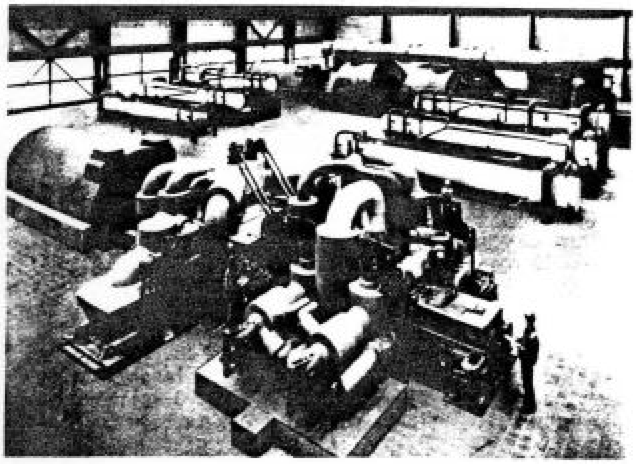

FIG 12 PHILADELPHIA ELECTRIC CO.'S EDDYSTONE 1

This is a continuously changing business. Technology evolved over the past decade has resulted in new knowledge which has been applied to new turbine designs and retrofit products used in the life extension market. Even Eddystone 1, Figure 12, which has continued to operate at 4700 psi, 1130°F/1050°F/1050°F (32.4 MPa, 610°C/565°C/565°C), still the world's most advanced steam conditions, is being considered for a plant life extension of 20 to 25 years. This would extend the plant's retirement date to the years 2015 to 2020. Not bad for a prototype unit put into service in 1959.

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