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An Integrated Steam/Gas Turbine-Generator for Combined-Cycle Applications

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ABSTRACT

Combined-cycle power plants have been built with the gas turbine, steam turbine, and generator connected end-to-end to form a machine having a single shaft. To date, these plants have utilized a nonreheat steam cycle and a single-casing steam turbine of conventional design, connected to the collector end of the generator through a flexible shaft coupling. A new design has been developed for application of an advanced gas turbine of higher rating and higher firing temperature and exhaust gas temperature with a reheat steam cycle. The gas turbine and steam turbine are fully integrated mechanically, with solid shaft couplings and a common thrust bearing. This paper describes the new machine, with emphasis on the steam turbine section where the elimination of the flexible coupling created a number of unusual design requirements. Significant benefits in reduced cost and reduced complexity of design, operation, and maintenance are achieved as a result of the integration of the machine and its control and auxiliary systems.

INTRODUCTION

The GE single-shaft 107F turbine is a combined gas and steam turbine, fully integrated as a single prime mover for power generation in a combined-cycle plant. A cross section is shown in Figure 1. The "107" nomenclature is broken down in the following fashion: the 1 indicates a single gas turbine; the 0 has no significance in this case; and the 7F indicates the gas turbine model. As a single-shaft unit, the steam and gas turbines are coupled together and drive a single generator.

Single-shaft combined-cycle units have the advantages of lower cost and less complexity of controls compared to multishaft plants of the same rating. in which the gas and steam turbines are not connected and drive separate generators. Single-shaft machines are not well suited for a phased installation requiring operation of the gas turbine in simple-cycle mode for a period of time before the steam cycle equipment is added. Such machines are also not suitable for

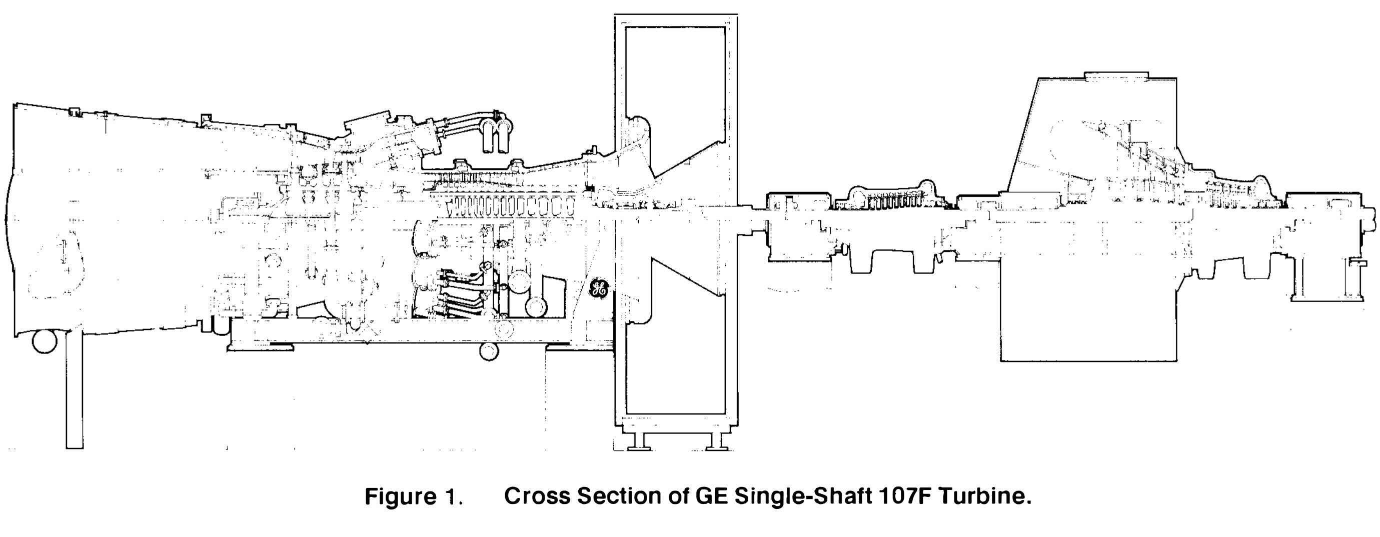

Figure 1. Cross Section of GE Single-Shaft 107F Turbine.

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repowering projects that use an existing steam turbine-generator or foundation.

Previous single-shaft machines have used gas and steam turbines of relatively conventional design, with a flexible coupling at some point in the shaft to permit independent thermal expansion of the rotors. The new single-shaft 107F has a single thrust bearing, and the gas turbine, steam turbine, and generator rotors are connected with all solid couplings. A common control system integrates operation of the inlet steam control valve with the gas turbine fuel and inlet guide vane controls for coordinated startup, normal speed and load control, and overspeed protection. The lubricating oil and hydraulic control fluid systems are also common.

Gas and steam turbine design practices have, in some ways, developed along different lines. Gas turbine practice places heavy emphasis on achieving short production cycle times through design standardization, assembled shipment, and packaged auxiliary systems. Physical sizes have increased, but not to the point where assembled shipment is no longer feasible. Large steam turbines, on the other hand, have for many years been too large for assembled shipment to be even considered as a design objective. Historically, a continual increase in utility steam turbine ratings required engineering emphasis to be on the development of large multiple-casing, tandem-compound machines, where little standardization of the product could be achieved. The design of the integrated combined-cycle machine has benefited from the best of both design practices.

There are several reasons why it has seldom been possible to pre-engineer standardized steam turbines to the extent that is common practice with gas turbines. First, it is a basic fact that steam turbine performance is greatly influenced by the magnitude of the exhaust loss, which is a function of the velocity of the steam exhausting to the condenser. The temperature of the cooling water available at a specific plant location determines the condenser pressure, and, therefore. the specific volume of the exhaust steam. Achieving the optimum exhaust velocity at different plant sites, therefore, requires that the annulus area of the turbine exhaust be matched to the volume flow of the exhaust steam. Therefore, at best, an otherwise standard design would require at least two, and preferably three, predesigned exhaust module selections. The other major reason why steam turbines have not been standardized is the fact that it is not possible to standardize the turbine in the absence of a standardized overall plant design. Numerous possibilities exist for the station designer to optimize rating and cycle parameters, particularly with regard to extractions for feedwater heating, air preheating, and feed pump drive turbines, resulting in a turbine design that is almost always specific to each particular plant.

Combined-cycle plants, however, with little or no requirement for steam extraction, and especially single-shaft applications, have the potential for a high degree of standardization with a steam turbine, standard except for exhaust annulus, matched to a standard gas turbine. The throttle steam temperature and pressure, and the net plant output will vary with the specific fuel, site location, and ambient conditions, similar to practice with simple-cycle gas turbines.

CYCLE AND PERFORMANCE

Pre-engineering power generation equipment requires standardization of design, beginning with the thermodynamic cycle and cycle parameters. Economic and equipment design studies led to the selection of a reheat, three-pressure steam cycle with nominal throttle conditions of 1450 psig (9996 kPa), 1000 °F (538 °C), with reheat to 1000 °F (538 °C); intermediate turbine admission at the reheat pressure of 360 psig (2482 kPa); and low-pressure admission at 50 psig (345 kPa). The most efficient recovery of gas turbine exhaust heat in a combined cycle is achieved with full feedwater heating by exhaust heat rather than by extraction steam. The absence of extraction steam feedwater heating removes one variable that is commonly the subject of economic optimization in conventional Rankine-cycle plants, at the expense of standardized equipment design.

The MS7001F gas turbine exhaust temperature is approximately 1100 °F (593 °C) at baseload conditions, nearly 100 °F (56 °C) higher than that of gas turbines previously applied with nonreheat steam cycles. This higher exhaust gas temperature increases the thermodynamic gain which can be achieved with a reheat steam cycle. Selection of three pressure levels with a reheat cycle avoids the compromise of a two-pressure-level system, in which the lower admission pressure is less than the thermodynamically optimum reheat pressure and higher than that required for efficient recovery and utilization of low-level exhaust gas energy. The three-pressure reheat cycle has a thermal efficiency advantage of about one percent over two-pressure reheat and two percent over two-pressure nonreheat. Reheat has the additional important advantage for a single-shaft unit of reducing the moisture content of the steam in the low-pressure turbine stages, permitting use of longer last-stage buckets, and therefore, a more compact single-flow rather than double-flow low-pressure turbine.

Figure 2 is a cycle schematic diagram. Cycle and performance parameters are summarized in Table 1.

Table 1

CYCLE AND PERFORMANCE PARAMETERS NATURAL GAS FUEL

Rating	
Net Plant Power	228 MWe
Net Plant Heat Rate	6,632 Btu/kWh (6997 kJ/kWh)
Thermal Efficiency (LHV)	51.5%
Steam Cycle Conditions	
Throttle	
Pressure	1450 psig (9996 kPa)
Temperature	1000 °F (538 °C)
Reheat	
Pressure	360 psig (2482 kPa)
Temperature	1000 °F (538 °C)
LP Admission	
Pressure	50 psig (345 kPa)
Temperature	500 °F (260 °C)

Notes:

1. Site conditions = 59 °F (15 °C), 14.7 psia (101.4 kPa), 60% RH

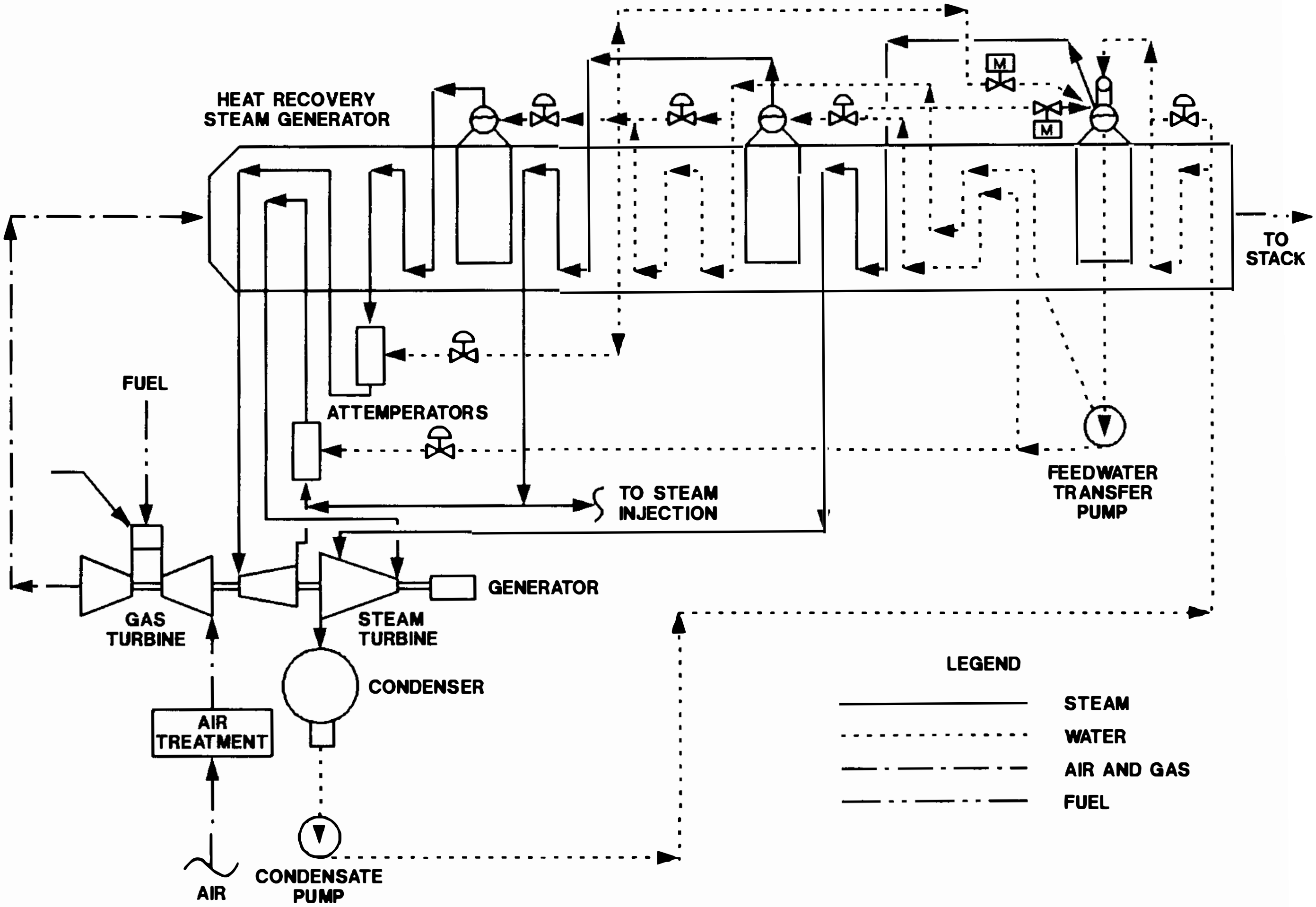

Figure 2. Cycle Schematic.

2. Steam turbine exhaust pressure = 1.5 in. HgA (3.38 kPa)
3. Steam turbine last-stage bucket length = 33.5 in. (850 mm)
4. Performance is net plant with allowance for equipment and plant auxiliaries
5. Gas turbine steam injection for U.S. Environmental Protection Agency New Source Performance Standards NO_X requirements

OVERALL EQUIPMENT ARRANGEMENT

Previous single-shaft machines have used a nonreheat, single-casing steam turbine in the arrangement shown in Figure 3. Both the gas and steam turbines are of essentially conventional design, requiring that the generator be in the middle of the machine because neither turbine is capable of transmitting the torque produced by the other through its rotor to the generator. The steam turbine must be physically moved out of the way to remove the field from the generator. This is practicable because of the small size of the single-casing turbine and the small number of piping connections, both due to the fact that the cycle is nonreheat. The steam and gas turbines have their own thrust bearings to maintain the proper axial position of rotors relative to casings and stationary nozzles. A flexible coupling is required somewhere in the shaft between the steam turbine and the gas turbine to permit the steam and gas turbine rotors to expand and contract axially independently of each other. The usual location for the flexible coupling is between the steam turbine and the generator, because the steam turbine rating is typically only half that of the gas turbine, and it is desirable to minimize the torque transmitted by the flexible coupling for reliability reasons.

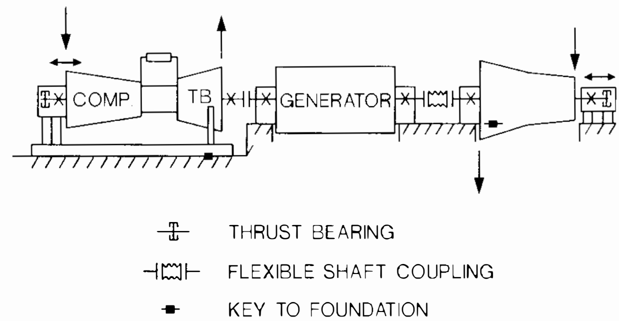

Figure 3. Configuration of Previous Single-Shaft Units.

The arrangement of the single-shaft 107F is shown in Figure 4. The gas turbine is on one end of the machine, its exhaust directed axially into the heat recovery steam generator (HRSG). The generator is on the other end, facilitating ease of removal of the field for maintenance. The steam turbine is in the middle, exhausting

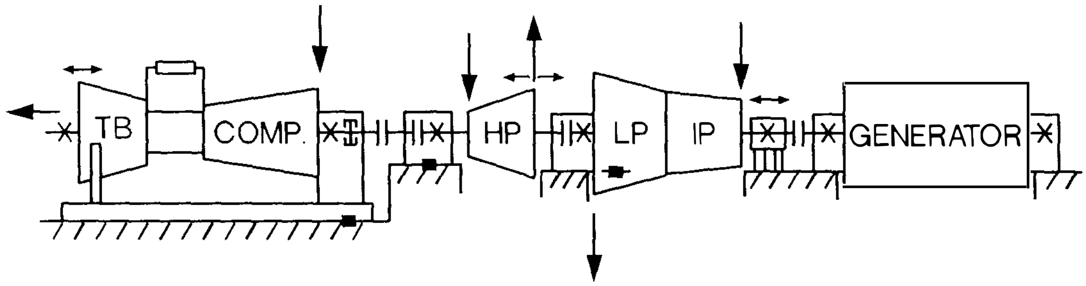

Figure 4. Configuration of the Integrated Design.

downward to the condenser below the foundation. The single thrust bearing is at the gas turbine compressor inlet, a point that does not move axially relative to the foundation. This arrangement places no special requirements on the design of the generator, and permits use of identical gas turbines for simple-cycle, multishaft combined-cycle, and single-shaft combined-cycle applications. The steam turbine, however, has the special requirements of transmitting the torque produced by the gas turbine, and accommodating differential expansion considerably greater than usual for a machine of its rating.

GAS TURBINE

The General Electric MS7001F gas turbine is an advanced design, heavy-duty machine for electric power generation. In the initial planning of its development, it was anticipated that combined-cycle installations would represent an increasingly significant portion of new generating capacity, and that the design must be thermodynamically optimized for high combined-cycle thermal efficiency and must be mechanically suitable for both multishaft and single-shaft applications. At a firing temperature of 2300 °F (1260 °C), the pressure ratio of 13.5 is optimum for maximum specific power, defined as the ratio of power output to compressor inlet mass flow. Maximizing specific power for a given firing temperature results in a design having the highest combined-cycle thermal efficiency and lowest installed cost for simple-cycle operation.

At the gas turbine exhaust end is an axial exhaust diffuser. The thrust bearing is located on the opposite end at the inlet to the compressor, where the machine is anchored axially to the foundation, providing good clearance control in the compressor. This arrangement, plus the fact that the thrust bearing is conservatively sized for thrust loading in either direction, permits the identical machine to be either directly coupled to the generator for simple-cycle or multishaft combined-cycle applications, or to the steam turbine rotor for single-shaft combined cycle.

The gas turbine nominal rating of 150 MWe is ideally matched with the largest single-exhaust-flow reheat steam turbine utilizing the longest available last-stage buckets. The combined-cycle nominal rating for the single-shaft unit is 225 MWe, and is a desirable size for addition of new generating capacity for many utilities, whether intended for baseload or daily start-and-stop operation.

The 50-Hz model, the single-shaft 109F, has a nominal rating of 315 MWe.

STEAM TURBINE

Overall Description

The steam turbine has two casings, a single-flow high-pressure (HP) section and a single-flow combined intermediate/low-pressure (IP/LP) section. For a single-shaft machine, design features that minimize length are important. Reheat permits use of a single-flow exhaust with the longest available last-stage buckets without excessive erosion due to moisture. Impulse stage design with wheel and diaphragm construction is particularly well suited because fewer stages and shorter length are required for the steam path compared to reaction stage designs.

Single-shell construction is used in both the HP and IP sections. Throttle steam pressure varies with load and the steam turbine operates with the inlet steam valves normally fixed in the full-open position. The inlet section is designed for full-arc admission without a control stage. The simple shell geometry in this region minimizes the thermal stress imposed by cyclic operation.

The inlet stop and control valves are located off the shell below the operating floor. There are no intercept or reheat stop valves since they are not required for overspeed protection, as will be discussed in a subsequent section. The absence of an external crossover and shell-mounted valves, and the fact that all steam piping connections are made to the lower half, eliminates the need for bolted piping connections and facilitates removal of the shell upper half for disassembly of the machine.

With the low dynamic thrust of impulse-stage design and the opposed-flow arrangement of the casings, the unbalanced thrust of the steam turbine is quite low. Packing diameters have been established such that the steam turbine thrust is less than that of the gas turbine and in the opposite direction, so that the thrust bearing loading of the combined machine is less than that of the MS7001F gas turbine in simple-cycle applications.

The compressor and hot gas path of the MS7001F gas turbine are standard for all applications. With gas and distillate fuels, and with the standard three-pressure reheat steam cycle, standard high-pressure and intermediate-pressure steam paths can be applied in most cases. However, varying requirements for gas turbine steam injection for emissions control may require small adjustments to the steam path for some applications.

The exhaust flow of combined-cycle steam turbines with low-pressure admissions is large, being as much as 125% of the inlet mass flow. The typical fossil-fuel-fired steam turbine with extractions for feedwater heating has an exhaust flow of only 70 to 80% of throttle flow. Although this large exhaust flow adds to the cost of the steam turbine and increases the exhaust loss relative to more conventional designs, the multipressure steam cycle without extraction for feedwater heating results in the lowest overall operating cost for combined-cycle applications.

As with any condensing steam turbine, the annulus area of the exhaust must be matched with the condenser pressure. Three low-pressure section designs, with 26-, 30-, and 33.5-in. (660-, 762-, and 850-mm) last-stage buckets, permit an economic selection of cost versus plant output to be made for the cooling water temperature expected at a particular site. Figure 5 shows the steam turbine output plotted against backpressure for the three low-pressure designs. All three are designed for the same inlet volume flow to match the exhaust pressure of the intermediate section at the low-pressure admission point.

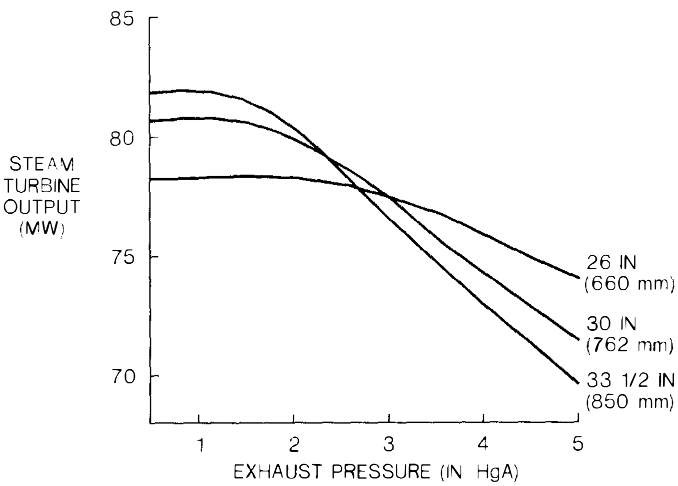

Figure 5. Steam Turbine Output as a Function of Exhaust Pressure and Length of Last-Stage Buckets.

Special Design Considerations

Eliminating the flexible coupling and positioning the steam turbine between the gas turbine and the generator created several special challenges in the design of the steam turbine.

Thermal Expansion Movements. Unusually large differential expansion occurs between the rotor and stationary components of the steam turbine in the IP/LP section. This is the result of the location of the thrust bearing and the fact that it is stationary with respect to the foundation, both conditions dictated by the need for the thrust bearing to maintain proper axial clearances in the gas turbine as well as in the steam turbine. Good thermodynamic performance requires that the large differential movements not result in excessive stage- and end-packing leakage flows, and that proper diaphragm-to-bucket spacing is achieved in the full operating temperature condition.

As a basis for comparison, consider the turbine in Figure 6, which is the design configuration for multishaft applications with the MS7001F gas turbine. The exhaust hood is fixed axially to the foundation near where the connection is made to the condenser. The thrust bearing is located in a sliding middle standard, which moves with the expansion and contraction of the IP/LP casing, moving the rotor and HP casing with it, and maintaining axial alignment between them. The front end standard also moves axially with expansion and contraction, so that the machine is fixed axially to the foundation at only one point.

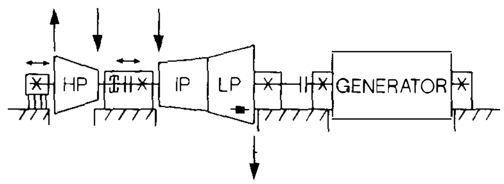

Figure 6. Steam Turbine Configuration for Multishaft Applications.

A turbine of this configuration could not be coupled directly to the gas turbine by simply removing the thrust bearing because the rotor and casings would move in opposite directions with thermal expansion, creating very large relative movements, excessive internal clearances, and poor positioning of buckets relative to the diaphragms.

Several aspects of this design are important to note, however:

1. The turbine is keyed axially to the foundation where the exhaust hood connects to the condenser through an expansion joint, avoiding shear loading on the expansion joint.
2. Opposed-flow orientation of the casings minimizes the net unbalanced thrust force.
3. The thrust bearing is located near the inlet to the high-pressure section, permitting use of interlocking "high-low" type packings to minimize leakage flow where the volume flow is smallest and clearance control is most important.
4. On each side of the thrust bearing, the rotor and its casing expand in the same direction and from the same point, minimizing differential expansion. Therefore, close axial spacing of the moving buckets relative to the stationary nozzles is maintained, even in stages some distance from the thrust bearing. Also, because differential expansion is small, high-low type packings can be used in the IP as well as the HP section.

In the steam turbine configuration for the single-shaft 107F, as shown in Figure 4, the thrust bearing is near the high-pressure inlet, and differential expansion is small, permitting use of packings with high-low teeth. Opposed-flow orientation is retained, but both casings are turned around so that they both expand in the same direction as the rotor, minimizing differential expansion. The casings are both fixed to the foundation and have relative motion between them. This configuration was chosen after consideration of alternative arrangements on the basis of mechanical suitability and thermodynamic performance.

The rotor in the IP section has greater total movement than does the casing because it expands from the thrust bearing, which is farther away than the point from which the casing grows. Differential expansion is too great in this section to permit the use of high-low packings. However, with this configuration, although the total rotor growth is greater than that of the casing, the large axial spacing between buckets and nozzles occurs in the cold condition, and is reduced to the optimum for best efficiency in the full-temperature operating condition.

Rotor Flexibility. Satisfactory lateral and torsional natural frequencies of the rotors, and proper alignment and distribution of load between bearings, are dependent upon satisfying well-established criteria for rotor flexibility. The short HP rotor is much stiffer than that of the comparable steam turbine design for multi-shaft applications because of the larger shaft diameter required to transmit the torque produced by the gas turbine. Satisfactory rotor flexibility was achieved by designing the steam turbine with only three journal bearings. A four-bearing design, with each rotor supported in its own bearings, would normally have been preferred because of the advantage in greater ease of factory and field balance. However, an additional bearing on the high-pressure rotor in this case would have required an impractical increase of nearly 9 ft (2.7 m) in length of the machine, to achieve both a satisfactory value for the first critical speed of the very light HP rotor, and sufficient flexibility of the rotor section between the small bearing on the HP rotor and the large, adjacent bearing on the LP rotor.

Thermodynamic Performance. The thermodynamic performance of the single-shaft 107F has been carefully analyzed and compared to the multishaft plant steam turbine design. Stage-packing and end-packing leakage flows are somewhat greater for the single-shaft design because of the use of noninterlocking packing teeth in the IP section, and larger shaft diameters in the HP and IP sections. Larger journal diameters increase the associated bearing losses. These losses are offset, however, by performance gains due to elimination of the pressure drop associated with reheat stop and intercept valves, and the elimination of the steam turbine thrust bearing and the two journal bearings associated with the additional generator in the multishaft plant. Overall, the performance of the single-shaft machine is essentially identical to that of the separate gas and steam turbine-generators in the multishaft plant design.

Other Configurations

As has been described, the single-shaft 107F is a standard design, with three choices of low-pressure turbine sections to match the exhaust annulus to site conditions. There may be applications for single-shaft machines that require more exhaust annulus area than can be provided in a single-flow design with the longest last-stage buckets available. In locations where a backpressure below 1 in. HgA (3.38 kPa) can be achieved, a double-flow exhaust may be economically justified. Also, some processes for gasifying coal in an Integrated Coal Gasification Combined-Cycle (IGCC) plant produce steam in addition to that produced from the gas turbine exhaust heat, and a steam turbine of considerably higher rating, and therefore, larger annulus area is required.

Figure 7 shows the steam turbine configuration that would be used with a double-flow low-pressure design. The high-pressure and intermediate-pressure sections are combined in one opposed-flow casing, a compact design that has been used in more than 600 reheat turbines manufactured by the author's company since 1950. The HP/IP section expands toward the low-pressure turbine, which is independently anchored to the foundation, requiring axial expansion bellows in the crossover pipe to accommodate a movement of approximately 1.5 in. (38 mm). This is twice the movement of the crossover bellows of a conventional double-flow turbine, but comparable to the movement that has been successfully accommodated in very large six-flow turbines.

The double-flow design would have some moderate disadvantages compared to the single-shaft 107F. Larger rotor diameters are required at the HP and IP inlets, with some adverse effect on starting and loading capability. The external crossover must be removed to disassemble the turbine for inspection and maintenance. On the basis of equal exhaust annulus area, the double-flow steam turbine has a performance penalty of about 0.75% (0.25% on the total plant), primarily due to pressure drop in the crossover, volume-flow effects in the double-flow low-pressure section, and larger differential expansion in the low-pressure section. This must be more than offset by lower exhaust loss to justify the double-flow design on a performance basis alone.

CONTROLS AND OVERSPEED PROTECTION

The single-shaft 107F controls are primarily normal gas turbine controls, the requirements for control of steam valves in single-shaft plants being very simple. In normal operation, plant output is controlled through the gas turbine fuel control valve, and the inlet steam control valve is maintained in the full-open position. The steam turbine acts as a fixed orifice, and boiler pressure varies with load, assuming the level required to make throttle flow equal to the rate of steam production in the HRSG. Off-line, normal speed control is through the gas turbine fuel valve, with the steam valve only brought into action for overspeed protection.

Where steam is available from another unit or an auxiliary boiler, the single-shaft 107F may be started by driving the steam turbine with steam. In this mode of operation, the unit is started with speed and acceleration initially controlled with the steam control valve. As speed is increased and the gas turbine begins to produce positive torque for acceleration, control is gradually transferred to the gas turbine fuel valve. Above 80% of full speed, control is entirely with the gas turbine, and a fixed steam valve position is maintained to provide cooling flow to prevent overheating of the steam turbine low-pressure section.

Since failure of the flexible coupling in previous single-shaft machines would disconnect the steam turbine from its load, independent overspeed protection of the steam turbine was required. With solid rotor couplings, independent protection is not required. The normal gas turbine speed governor and emergency trip com-

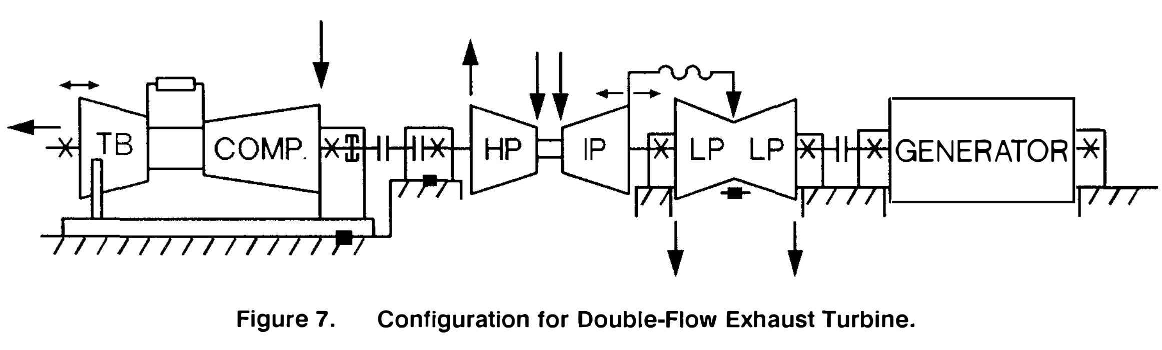

Figure 7. Configuration for Double-Flow Exhaust Turbine.

pose the first and second lines of defense against destructive overspeed. The governor operates on the fuel control only, unless speed rises above 103% of rated speed, at which point the steam control valve is closed in proportion to further speed increase. A power load unbalance system acts to rapidly close the fuel and steam control valves upon sudden occurrence of a significant imbalance between mechanical power and electrical load. The principle is the same as that used on conventional reheat steam turbines, but in this case, mechanical power is derived as the sum of two signals, one representing fuel flow to the gas turbine, and the other, a stage pressure in the steam turbine.

All conventional reheat steam turbines require intercept and reheat stop valves to shut off flow of steam from the reheater on sudden loss of electrical load. Because of the large rotor inertia of the combined machine and the power required to drive the gas turbine compressor, overspeed protection of the single-shaft 107F is achieved without the need for intercept and reheat stop valves. The full volume of steam in the reheater and hot and cold reheat piping can be allowed to expand through the reheat turbine to the condenser without causing speed to rise to the setpoint of the emergency trip. Furthermore, fast closing of the secondary and low-pressure admission valves is not required. Fast-closing steam valves are expensive components, and their elimination represents a significant savings in maintenance as well as in initial cost. In addition, with the absence of any valves between the reheater and the condenser, safety relief valves to protect against overpressure in the reheater are not necessary.

CONCLUSION

The single-shaft 107F represents an important new development in power generation equipment, combining advanced gas turbine technology with a modern reheat steam turbine design, integrated into a single prime mover. Traditional gas turbine design philosophy, emphasizing reduction in lead time for engineering, manufacturing, and construction, and rapid start and stop capability with highly automated operation, has been combined with steam turbine technology for the design of large, multiple-casing machines. The result is a turbine-generator unit with high thermal efficiency, suitable for either baseload or daily start-and-stop operation, and requiring less cycle time from order to commercial operation than conventional fossil-fueled plants. The single-shaft combined-cycle plant can offer greater reliability and lower cost in all combined-cycle applications that do not require separate operation of gas and steam turbine-generators, including those where future addition of a coal gasification plant is a consideration. For these applications, the single-shaft combined-cycle machine is a viable option for the addition of generating capability in blocks of approximately 225 MWe.

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