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Title:
Effects of External Boost Compression on Gas Turbine Performance in an Advanced CPFBC Application

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ABSTRACT

When a commercial gas turbine, designed and optimized for natural gas fuel, is employed in an Advanced Circulating Pressurized Fluid Bed Combustor (CPFBC) application, changes occur that affect both the thermodynamic cycle and the performance of the individual components comprising the machine. These effects derive principally from the increased pressure drop encountered between the compressor discharge and the expander inlet, with changes in gas properties and flow rates for the hot combustion products having secondary effects (Ref. 2 and 4). The net effect is that power output can be reduced, and significant design and/or operational compromises may be required for the gas turbine. Application of an external boost compressor can mitigate these effects.

INTRODUCTION

The operation of a gas turbine is based on well defined physical principles, derived from fundamental aerodynamics and thermodynamics. The application of these principles has led to a design evolution that has resulted in the large, efficient gas turbines manufactured for today's competitive power generation market. These machines are optimized for the combustion of natural gas or distillate oil. The thermodynamic cycle and detailed design of components such as compressors, combustors, and expanders, reflects this focus on optimizing both simple cycle and combined cycle performance for these fuels.

BRAYTON CYCLE EFFECTS

The typical design basis pressure drop between compressor and expander for a commercial gas turbine is in the range of 3.5 to 4.0 percent. In a CPFBC application, this pressure drop can be as high as 12 percent, reducing the pressure at the inlet of the expander, and reducing the pressure ratio available for expansion. The combined effect of reducing expander inlet pressure and temperature and/or mass flow is a very significant loss in net power output. Figure 1 illustrates the effects of pressure and temperature on a representation of the Brayton cycle, from an energy/entropy perspective.

Figure 2, below, presents a simplified flow diagram of an advanced CPFBC system. The CPFBC uses compressed air from the gas turbine compressor to fluidize and oxidize the coal/sorbent bed. Hot vitiated (oxygen-depleted) air from the CPFBC is routed to the gas turbine combustor to support topping combustion of low Btu syngas, produced in the carbonizer vessel, enabling the turbine to operate at the full design basis firing temperature. Energy in the gas turbine exhaust produces steam in a HRSG, which is used to power a Rankine bottoming cycle. Additional steam is generated in the CPFBC for use by the bottoming cycle.

Satisfaction of the gas turbine design physical principles alluded to above requires that some adjustment or compensation be made for the increased pressure drop. The compressible flow theory underlying this assertion is explained below, followed by a discussion on several ways to compensate for the effects of the added pressure drop. Thermal performance for the system depicted in Figure 2 is calculated using the ASPEN-SP flow sheet simulator software. Comparative performance is presented in Table A, following a brief description of the configurations that were evaluated.

COMPRESSIBLE FLOW EFFECTS

The basis for the potential reduction in expander inlet temperature is explained by compressible flow theory. At the design condition, flow is sonic, or choked, at one or more discrete sections in the expander flow passages. The relationship between flow rate, temperature, pressure, and the thermodynamic properties of the flowing gas are defined by a specific relationship, as follows:

$\psi = W\ \sqrt{T} / A\ P\ \sqrt{MW} = [\ k/R\ (2/k+1)^{k+1/k-1}]^{1/2}$, where

W = mass flow, lb/sec
T = absolute temp, °R
A = flow area, sq. ft.
P = total press., psia
k = specific heat ratio, Cp/Cv, of the gas
R = universal gas constant, 1.986 Btu/lb-mole °R
MW = Molecular Weight

Note: See Ref. 3 for derivation of this equation.

The value of the critical, or choked flow parameter, (ψ) is a weak function of k and R, and is essentially constant. Thus, the relationship between flow, turbine inlet temperature, turbine inlet pressure, molecular weight, and flow area is uniquely defined. If a gas turbine is designed for a specific application, such as firing natural gas, a suitable turbine nozzle area is provided. If the same machine is used in a CPFBC application, some of the factors in the flow parameter relation change as follows:

1. The gas molecular weight changes in an advanced CPFBC application due to two factors: (a), the combustion of coal produces more CO_2 relative to H_2O compared to the firing of natural gas, and (b) advanced CPFBC applications may result in firing additional fuel to produce steam in the Fluid Bed Heat Exchanger, yielding a gas with a higher proportion of combustion products (less excess air) compared to a standard natural gas fired combustion turbine application. The net change in gas molecular weight in an advanced CPFBC application varies from about 3.5% to 5%, the change on a square root basis is therefore between 1.75% to 2.5%. This effect tends to increase the amount of mass flow that can pass through the turbine nozzles, other conditions being equal.

2. The mass flow of gas through the turbine nozzles in an advanced CPFBC is significantly greater than the mass flow obtained for the same machine in a natural gas fueled application. This increase in nozzle mass flow varies from about 1.3% to as much as 4.8%. The increased mass flow is quantified by comparing the sum of the following mass inputs/outputs in an advanced CPFBC application: fuel (such as coal), sorbent (such as limestone), moisture (such as steam injected into the carbonizer); less solids removed from the system (ash and spent sorbent).

3. Finally, and most significantly, the pressure drop between the compressor exit and the turbine nozzles is increased from a nominal 3.5% in a standard natural gas fueled machine to approximately 12% in a typical advanced CPFBC application.

COMPENSATING FOR INCREASED PRESSURE DROP

The first two factors noted above tend to offset each other. The most significant net effect is caused by the increased pressure drop. This can be mitigated in several ways, as follows:

1. The airflow through the machine may be reduced by partial closure of the compressor inlet guide vanes. This reduces the mass flow through the turbine expander nozzles; it also tends to reduce the power output of the system.

2. The firing temperature, and thus the turbine inlet temperature, may be reduced, as illustrated in Figure 1. This results in a sacrifice of output and efficiency for the gas turbine. The overall impact on the combined cycle in an advanced CPFBC application, which uses both a Brayton cycle and a Rankine cycle, depends on many variables and must be evaluated for each application.

3. The pressure may be increased in several ways: (1) An additional stage may be added to the gas turbine compressor to provide a higher pressure ratio; (2) An external boost compressor may be added to the system; (3) The external circuit pressure drop may be reduced by changing design parameters for the components in the circuit; and (4) The gas turbine compressor may be rematched to operate at a higher pressure ratio. For a constant speed machine, this implies setting the compressor operating point higher on the compressor head/flow characteristic. This requires caution to avoid operation too close to the stall or surge line, at which point compressor operation is unstable and damage may result. In this paper, the use of an external, electric motor driven boost compressor is evaluated.

4. The turbine nozzle area may be increased by about 9%. This is a significant increase for a production machine, and could require redesign and tooling changes.

This paper evaluates integrated system (gas turbine/CPFBC/HRSG/steam turbine) performance as the different approaches noted above are employed to satisfy the choked flow parameter relationship at the gas turbine expander inlet nozzles. Certain system design parameters are fixed, including:

Gas Turbine is a Westinghouse 501F production machine, as built for firing natural gas. Compressor pressure ratio is held constant at 14:1.

Steam Turbine is a generic tandem compound machine with a double flow low pressure section, rated at a nominal 105 MWe with feedwater heaters in service, operating at 1450 psig/1000°F/1000°F, condensing at 2.5 in Hg abs. Steam turbine performance is computed per Spencer, Cotton, and Cannon (Ref. 6). Steam turbine throttle flow is held constant at 100% of rated value.

CPFBC is as shown in Figure 2. Coal is Pittsburgh No. 8, with a carbonizer temperature of 1700°F.

The following configurations are evaluated:

Case A incorporates an electric motor driven boost compressor in the system. The gas turbine compressor discharge air flow is collected and conveyed off-board the machine, and through a regenerative heat exchanger and a trim cooler. The cooled air is boosted in the boost compressor, reheated in the regenerative heat exchanger, then sent to the CPFBC process vessels. The boost compressor replaces the pressure lost in the CPFBC vessels, as well as the regenerative heat exchanger and trim cooler losses. The gas turbine air flow, firing temperature, and nozzle area remain as in the original machine.

Case B also utilizes an electric motor driven boost compressor, but does not incorporate the regenerative heat exchanger and trim cooler. Boost compressor pressure rise is somewhat reduced; even so, drive power requirements are increased due to the significantly higher inlet temperature to the boost compressor. As in Case A, airflow, firing temperature and nozzle area remain unchanged.

Case C does not use a boost compressor. Matching of the design basis flow parameter at the turbine inlet is achieved by partial closing of the gas turbine compressor inlet guide vanes, thereby reducing compressor airflow and mass flow to the turbine nozzles. An 8.2% reduction in airflow is required.

Case D also does not use a boost compressor. Matching of the design basis flow parameter at the turbine inlet is achieved by reducing the turbine inlet temperature by approximately 440°F.

Case E is the final case without a boost compressor. The turbine nozzle area is increased by 9%, compensating for the CPFBC added pressure drop. Airflow and turbine inlet temperature remain the same as in the original design case.

Relative performance for the various cases is presented in table A below.

Table A
Performance of Various Advanced CPFBC Configurations for a Westinghouse 501F Gas Turbine with a Repowered Steam Bottoming Cycle

Case	A	B	C	D	E
CT Mwe	162.0	163.2	139.2	103.7	151.8
ST Mwe	115.0	111.0	111.2	112.1	111.1
Aux Mwe	11.0	12.4	5.3	4.6	5.8
Net MWe	266.0	261.8	245.1	211.2	257.1
η, HHV	45.7	46.2	46.5	43.3	45.8
Boost Fan HP	7700	9650	0	0	0
CT Nozzle A/Ades.	1.0	1.0	1.0	1.0	1.09
Airflow, lb/sec	968	968	889	968	968
Firing Temperature, °F	2465	2480	2455	2020	2460

The results for the cases evaluated indicate that the boost compressor, with or without regenerative heat transfer, provides the best output. The highest efficiency is realized by the reduced airflow case, but at a loss in net output. The reduced turbine inlet temperature case suffers with respect to net output and efficiency. Finally, the case with the increased nozzle area has a modest reduction in output, with efficiency that is close to the boosted cases. Further evaluation of these cases to evaluate capital cost and economics will be undertaken.

CONCLUSION

Use of a standard production gas turbine in an advanced CPFBC cycle application results in the need to compensate for changes to the original design basis Brayton cycle (caused by the increased pressure drop between the compressor and expander), and for effects on the turbine nozzle flow matching. Some of these compensating actions will reduce overall net cycle power output and efficiency. The addition of a boost compressor can effectively rematch the nozzle flow relationship and increase net power while maintaining net efficiency, without increasing the physical nozzle area of the turbine. A significant advantage for this approach is that it avoids modifications and investment for the gas turbine compressor and expander. An external subsystem, comprised of available industrial components, can be configured to suit the system requirements. The ultimate success of this approach depends on the specific application, including the turbine selected, and the cycle configuration and its design parameters.

ACKNOWLEDGEMENT

The effects of gas property and mass flow changes on gas turbine performance were investigated by M. S. Johnson (Ref. 2). The additional effects of increased pressure drop, inherent in the CPFBC cycle, were investigated in a clean coal technology repowering study sponsored by the US DOE, Morgantown Energy Technology Center, and reported on for a CPFBC cycle co-firing coal and natural gas in a Westinghouse 501D5A gas turbine in Ref. 4. Donald Bonk is the Contracting Officer's Technical Representative (COTR) for that study and has supported the repowering study, along with related investigations regarding the use of the boost compression technique. Robert Travers is the COTR for a repowering study in progress, utilizing a Westinghouse 501F gas turbine in an advanced CPFBC configuration.

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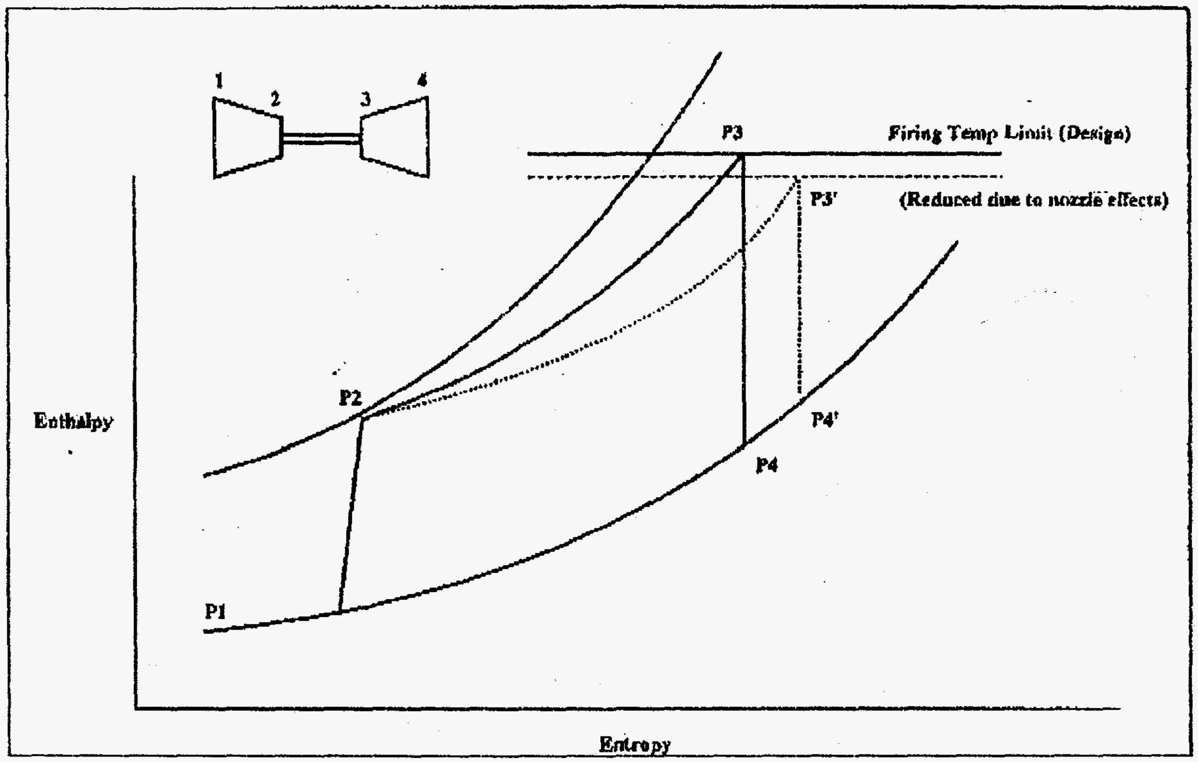
FIGURE 1
BRAYTON CYCLES



1. Cycle P1/P2/P3/P4 is a normal design basis cycle for a typical machine.
2. Cycle P1/P2/P3'/P4' represents a CPFBC gas turbine cycle. Note the reduced turbine expansion ratio and turbine inlet temperature. (Based on maintaining design basis mass flow.)

FIGURE 2

ADVANCED CIRCULATING PRESSURIZED FLUIDIZED BED (PFBC) CYCLE