EFFICIENT PART LOAD COMBINED CYCLE OPERATIONS

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Other pioneering applications included the gas turbine locomotive, helping the chemical and process industry to recognize the benefits to industrial power generation of a gas turbine with waste heat recovery, the use of a gas turbine for blast furnace blowers burning low Btu blast furnace fuel, and developing the acceptance of the large combined cycle for utility base load application.

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He was educated at Texas A&M in Mechanical Engineering with a B.S. degree and a Master's degree from Purdue University in steam power. He is a Life Fellow Member of ASME.

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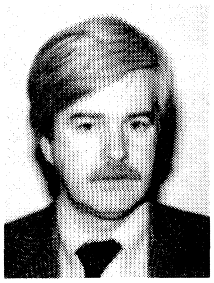

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His career at Riley Stoker has included many aspects of boiler engineering and engineering management, such as fuel burner design, industrial and utility proposal designs, performance definition of all Riley supplied equipment, field data analysis and evaluation, boiler design, boiler analysis and development, instrumentation and controls, and auxiliary systems engineering. Mr. Hunt has authored articles regarding innovative boiler designs, fluidized bed experience and emissions.

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ABSTRACT

During the oil crisis in the 1970s, many electric utilities assigned their base load portion of electric generation requirements to nuclear and coal fired steam plants. The intermediate duty in some cases was assigned to combined cycles or to oil fired steam plants. The peak duty under 1000 hr was assigned in many cases to combustion turbines.

By innovative boiler modification, the peaking combustion turbine can be combined with a steam plant to provide the combustion air for the steam plant boiler with a wide load variation at good efficiency.

In addition to maintaining high efficiency at part load, it increases the plant capacity 6 to 7 percent at full load, decreases NO_x generation, varies the air flow to an existing boiler 10 percent and up to 20 percent using a new combustion turbine and provides a system such that the gas turbine can automatically be removed from the cycle without tripping the steam cycle.

INTRODUCTION

As the proportion of combined cycle and cogeneration power in a utilities grid increases, this new generation capability will be required to accept the full electric utility load requirement. This means that the combined cycle plant will be required to share the extended part load operation of the electric utilities.

Over the past three decades, air conditioning has lowered many utilities' load factor to less than 60 percent. A typical utility in the south might have a load of 100 percent in the heat of the day but only 40 percent at night, as shown in the load duration curve of Figure 1. When the load on the combined cycle can be maintained at a high level, heat rate stays low. However, the daily cyclic load must be generated by some power machinery, so a solution has been to install multiple non-fired combined cycle units and start and stop individual units on a daily basis.

A combined cycle plant with a reheat steam turbine that has novel features enabling efficient load turn down will be analyzed. This plant was installed over 23 years ago, so the advantages of this plant have been demonstrated [1, 2, 3, 4, 5].

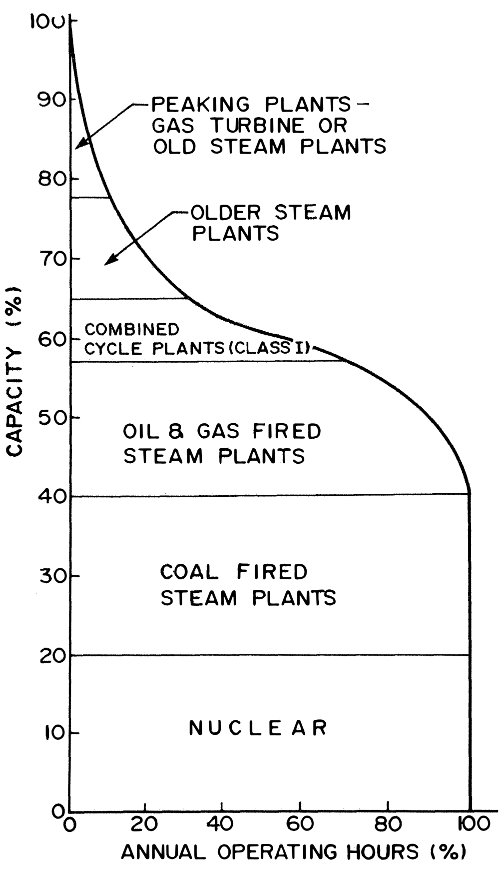

Figure 1. Electric Utility Load Duration Curve.

FEATURES OF COMBINED CYCLES

In this plant, the combustion turbine stays at full load all the time and load reductions are taken in the steam portion of the cycle. In a normal steam plant, reduced load is achieved by lowering the forced draft or induced draft fan speed (flow), so that excess air remaining after combustion stays nearly constant.

With a combined cycle plant, boiler combustion air is set by the gas turbine and reduced steam production usually means higher remaining excess air. In the case of an older, single shaft gas turbine, exhaust flow from the gas turbine remains constant. More recent gas turbines for combined cycles use variable inlet guide vanes (IGVs) to reduce the air flow 20 percent to 30 percent. In some cases, additional flow reduction can be obtained by the use of several rows of variable compressor vanes in conjunction with the IGVs. A two-shaft gas turbine, where the speed can be lowered on the gas generator results in an additional "turn down" for good boiler efficiency at light loads.

A novel feature was applied to the West Texas Utilities plant in that the new boiler was a forced draft system and the forced draft fan was placed ahead of the gas turbine (Figure 2) producing 40 inches of water pressure at the axial compressor flange (supercharged [6]).** In a converted plant or a new plant, maximum air flow variation with a single shaft gas turbine can be achieved with both supercharging and variable IGVs.

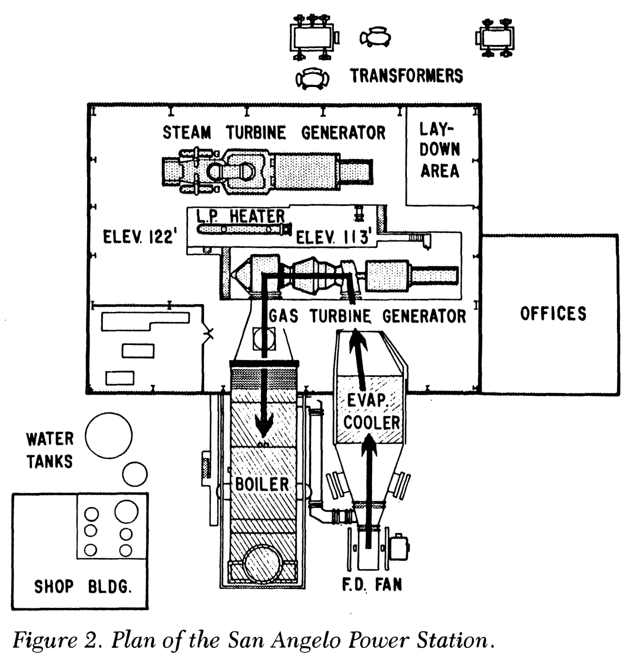

Figure 2. Plan of the San Angelo Power Station.

ANALYSIS OF HEAT AVAILABLE AT TWO STEAM TURBINE LOADS

Using a Mollier chart and plotting the state points, an indication is displayed as to why the West Texas Utilities cycle **with** reheat and a waste heat boiler obtains good part load efficiency. Data was taken from a heat balance at 99584 kW on the steam turbine as a high load point and a part load point of 48,991 kW. The expansion lines are shown in Figure 3 and the significance of the reheat system in making more work available is apparent, especially at part load. Also, the split in work available between the initial turbine (high pressure) and the reheat turbines (intermediate and low pressure) is shown.

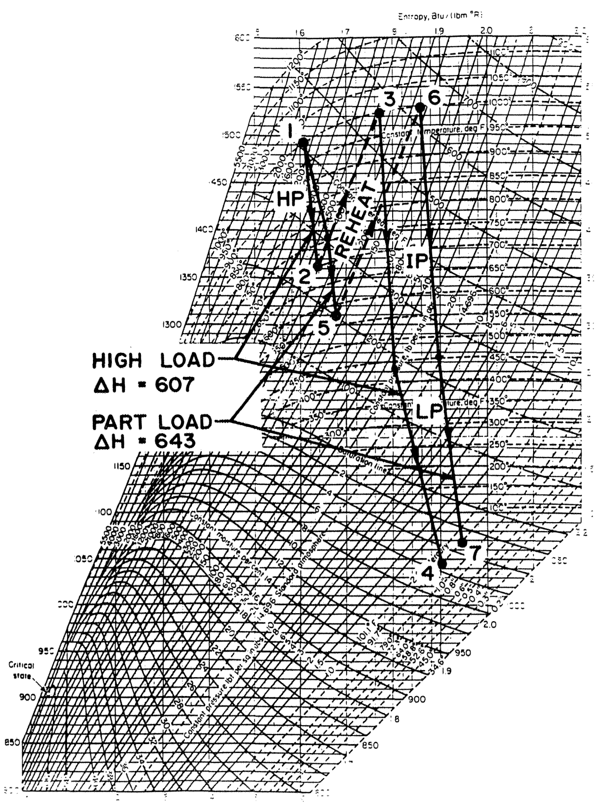

Figure 3. Heat Drop Profile.

In the high load case, steam starts at Point 1 at 1465 psia and expands to Point 2 at 424 psia. It is then piped to the reheat element of the boiler and heated back to 1000°F (Point 3). From there it is expanded to Point 4 at 1.5" mercury vacuum.

For the part load point of 49 MW on the steam turbine, the expansion would again start at Point 1 and expand to Point 5 at 185 psia. After heating to Point 6 in the boiler, the steam would expand to Point 7 at 1.5 in mercury vacuum.

The available work at high load compared to that at part load is approximately the same, i.e., High Load = 607 Btu/lb and Low Load = 643 Btu/lb. However, the proportion of auxiliary load has increased so the output/per pound of steam for the part load point percentage-wise has increased slightly.

SUPERCHARGED GAS TURBINE FOR A COMBINED CYCLE

When supercharging was first proposed by an engineering consulting firm, it was the authors' opinion that this was an appendage that would complicate the combined cycle. Only after detailed design and subsequent operation, did advantages other than variable air flow come to light. These advantages are:

- Increase in gross output in summer of approximately 10 percent.
- By removing the outer layer of air from the fan volute, the fan becomes, and has been used as, a very high powered inertial separator.
- The low inlet velocity of a centrifugal fan and acceleration as the air travels radially makes the centrifugal (compressor) fan less prone to icing conditions. The increased inlet temperature of the air to the axial compressor of some 17°F (evaporative cooler turned off), lowers the relative humidity to 80 percent or less, so that the axial compressor remains ice free.
- Tests in the shop and in the field proved that with 20 in H_2O or more at the gas turbine inlet, the fan could be used as a starting device. When compared to a normal starting motor ramp, turbine inlet temperature was 200°F less when using the fan. *Note:* The gas turbine in this study does not have a starting motor. The only starting device is the super-charging fan.
- Transfer, in an emergency, from a combined cycle mode quickly (less than 10 seconds to separate cycles, ie. steam and gas turbine) is accomplished by opening one "cold" damper (Item 1 of Figure 4.) Closing of "cold" damper, Item 2, and "hot" damper Item 3, can be at a more leisurely pace. *Note*: On three supercharged combined cycles over a period of 20 years, there has been no failure to transfer after the initial startup adjustments.

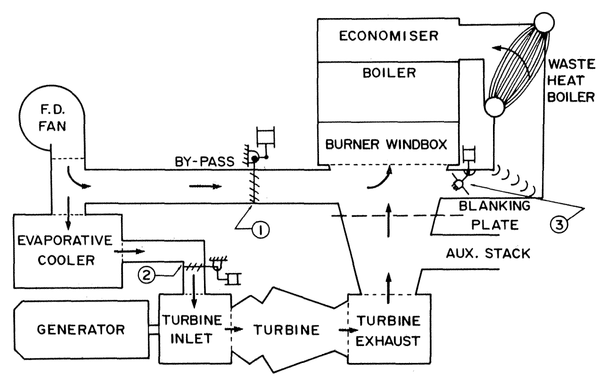

Figure 4. Diagram— Turbine Exhaust Gas and Air Paths.

STARTING THE PLANT FROM A COLD CONDITION

Initial calculations indicated that the gas turbine could be started with the supercharging fan, but, due to the size of the plant, it was decided to supply a starting motor. Field experience proved that the starting motor was unnecessary and the motor and clutch were returned for credit. The major advantage of this method of starting was that the turbine inlet temperature was 200°F less than required with a starting motor (for the same starting time).

It was expected that this decrease in starting temperature would improve "hot parts" life of the gas turbine. A comparison with two of the same model gas turbines in two large combined cycle operations revealed that the value of the "hot parts" sold to West Texas Utilities was half that sold to a Gulf Coast industrial plant (per turbine) over the same 15 years of operation.

Using the supercharged start, there are two methods of bringing the complete combined cycle to full load.

One method is to start the gas turbine and exhaust into the wind box of the boiler at 400°F (Figure 4). This brings the boiler to steaming conditions with a uniform temperature in the furnace. At this point, the boiler burners can be ignited and the boiler brought "on line," as designed.

A second method is to start the boiler in the normal utility manner with the gas turbine bypassed. Once operating on steam only, using the supercharging fan for combustion air, the gas turbine (without a starting motor) is started. The turbine bypass damper (Item 1 of Figure 4) is partially closed to force 20 in H_2O pressure on the inlet of the gas turbine. This forces the gas turbine to ignition speed where a normal start occurs. The rest of the plant is then brought up to full load conditions.

During startup, the bypass damper (Item 3) to the waste heat boiler has been open to bypass some of the exhaust gases to the waste heat boiler section. As the load on the steam turbine is increased, the bypass damper (single plate, Item 3) is modulated closed. This decreases the possibility of a steaming economizer and maintains the proper air to fuel ratio on the furnace. A tight shutoff of this damper is not necessary. The actual boiler flow arrangement is shown in Figure 5.

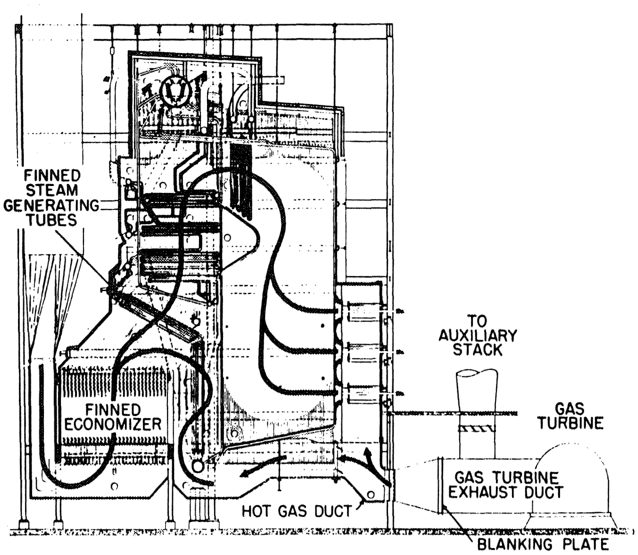

Figure 5. Cross Section of Boiler.

NO_x REDUCTION USING COMBINED CYCLE

NO_x testing was conducted on the Lake Nasworthy (renamed San Angelo) Plant Unit 1 during February 1972. The objectives of the testing were to determine the effect on boiler NO_x emissions when using turbine exhaust gas as the combustion air source, and when firing the boiler in an off stoichiometric mode. Off stoichiometric firing was achieved by taking some burners out of service and leaving their air registers open. This simulates an overfire air system.

Results were not surprising on natural gas firing. Since flame temperatures are lowered when operating "combined cycle" and off stoichiometrically, NO_x emissions were also significantly reduced. Emissions were lowered by almost 50 percent (Figure 6) when operating with turbine exhaust gas as combustion air at normal stoichiometric burner conditions, and the same reduction is shown in Figure 7 with off stoichiometric burner conditions.

STATION ARRANGEMENT

The inside arrangement is shown (Figure 8) with the steam turbine in the foreground and the gas turbine recessed in the background. This minimizes the length of the hot exhaust duct from the gas turbine to the boiler. An outside view of the station is shown in Figure 9 with the supercharging fan shown on the left side.

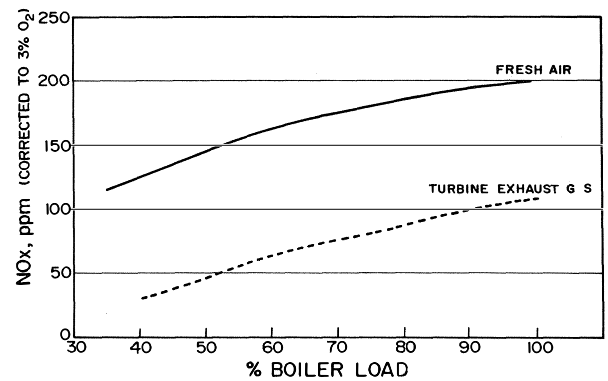

Figure 6. Actual NO_x Emissions for Combined Cycle and Fresh Air Firing—Normal Stoichiometric Firing.

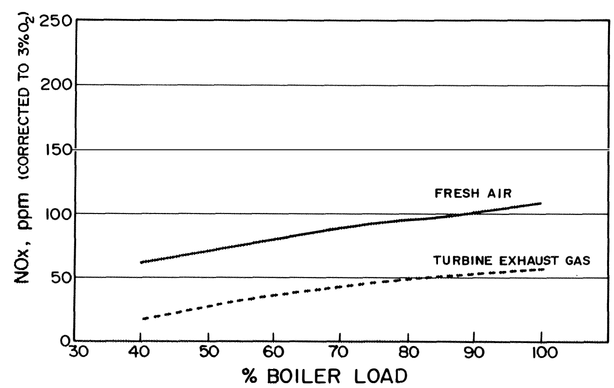

Figure 7. Actual NO_x Emissions for Combined Cycle and Fresh Air Firing—Off Stoichiometric Firing.

OPERATING RECORD

The combined cycle plant was initially started using the gas turbine only as a 25 MW peaking unit, while the steam portion of the plant was being completed. After three years of operation, it was decided to report an operating record beginning at the end of 1969, as shown in Table 1.

The heat rate is reported in higher heating value of natural gas from 1970 through 1976, and 1985 through 1989. For easy comparison to International Standard Organization ratings, the next line reports cycle efficiency in lower heating value.

Normal operation is to keep the gas turbine at full load (rated temperature) at all times and vary the station load with the steam turbine. Thus, the steam turbine operates from overpressure conditions rating of 99584 kW down to 42922 kW with a good efficiency. The average efficiency of the complete combined cycle for the first seven years is 39.28 percent (lower heating value).

When the natural gas was curtailed, the gas turbine operation was stopped as it was only equipped to burn gas.

The normal way to define load factor is the average load for the year divided by the maximum output. For the gas turbine, this is not a good "yardstick" because the cold weather capability can be as high as 40 percent above the nameplate rating. This causes the load factor to be disproportionately low. Even so, the gas turbine operated for 52,500 hr during the seven year period at an average load of 106.23 percent. The steam turbine was taking the load reductions, so its load factor was much lower.

Table 1. West Texas Utilities, San Angelo Power Station Combined Cycle Operating Record.

W-301 GAS TURBINE 1450°F 25.2 MW 1450°F FIRING TEMPERATURE
STEAM TURBINE DESIGN 85 MW 1523 PSIG 1000°F/1000°
TOTAL 110.2 MW

YEARS	1970	1971	1972	1973	1974	1975	1976	1985	1986	1987	1988	1989
STATION HEAT RATE												
BTU/KWHR HHV	9576	9505	9554	9845*	9802*	9,727*	9,568*	10146	9887	9996	9780	9730
AV. EFFICIENCY LHV	39.60%	39.89%	39.69%	38.52%	38.68%	38.99%	39.63%	37.38%	38.36%	37.94%	38.78%	38.97%
OPERATING HOURS												
GAS TURBINE W-301 52,526.2 TOTAL HRS.	7610	8297	7885.8*	5451.7*	7692*	7,586.9*	8,002.8*	7249	7785	7349	7174	7941
STEAM TURBINE 57,424.3 TOTAL HRS.	8393	8377	8431.5	7140.7	8601	8,268.7	8,212.4	7487	8072	7310	7121	7902
LOAD FACTOR												
GAS TURBINE	93%	101%	96.56%	64.39%	88.60%	91.51%*	97.20%*	86.31%	88.39%	84.29%	79.87%	91.59%
AVERAGE LOAD ON GAS TURBINE	27,000KW	26,800	27,138	26,177	26,827	26,414	26.671	26.074	24.865	25.120	24.457	25.257
AVERAGE LOAD AS A PERCENTAGE OF RATED LOAD (106.23% AVERAGE)	107.1%	106.5%	107.3%	103.7%	106.6%	105.7%	106.7%	103.5%	98.67%	99.68%	97.05%	100.23%
STEAM TURBINE	67%	69%	68.65%	52.74%	78.35%	78.55%	77.0%	52.51%	73.89%	49.90%	51.48%	65.60%
SERVICEABILITY FACTOR (Availability)												
GAS TURBINE	87%	95%	95.72%	89.88%**	88.10%**	93.35%**	93.00%*	83.32%	89.56%	91.06%	82.35%	90.91%
STEAM TURBINE	96%	96%	96.65%	83.00%**	94.99%	95.73%	94.57%	93.15%	92.29%	83.73%	81.07%	90.34%
FORCED OUTAGE RATE (E.E.I Basis)												
GAS TURBINE % Total $\frac{.296}{7}$ = .042%	0.0	0.021	0.136	0.0	0.139	0.0	0.0	INFORMATION AVAILABLE UPON REQUEST				
STEAM TURBINE	Not Available					0.0	0.0					

* NATURAL GAS FUEL CURTAILMENT FOR GAS TURBINE, STEAM PLANT OPERATES ON OIL DURING CURTAILMENT

** CHANGED TO ONE MAINTENANCE SHIFT FOR INSPECTION PERIOD RATHER THAN TWO SHIFTS

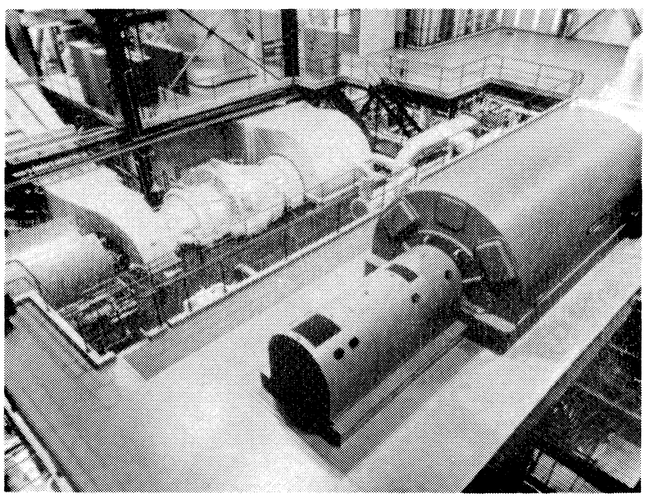

Figure 8. Inside the Station.

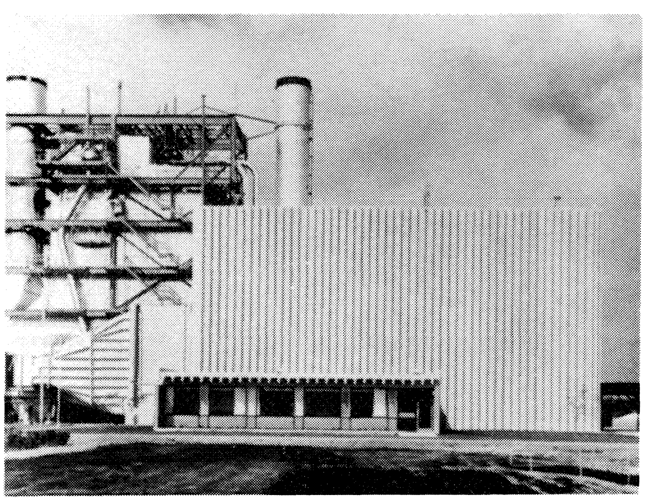

Figure 9. Outside the Station.

The availability numbers were influenced by a change in maintenance procedures.

The seven year average gas turbine forced outage rate (FOR) was 0.04 percent. This low FOR was primarily due to the control system. The system consists of pressure balanced hydraulics with no frictional parts. The oil supply is from the main direct drive lubricating oil pump. Such a system had been very successful on a steam turbine at a Gulf Coast refinery, so its good features were applied to gas turbines.

POTENTIAL APPLICATIONS

The concept described for efficient combined cycle operation at the West Texas plant can be applied by utilities and cogenerators today as the need for a large daily turndown increases.

Using the 75 MW industrial turbine, as being typical of existing utility gas turbines, a comparison of combined cycle heat rates *vs* load was calculated, as shown in Figure 10.

A single 75 MW turbine exhausting into a heavy fired reheat boiler (1450 psig, 1000°F), produces a nominal 300 MW plant output. Without boiler firing, about 100 MW at 8900 higher

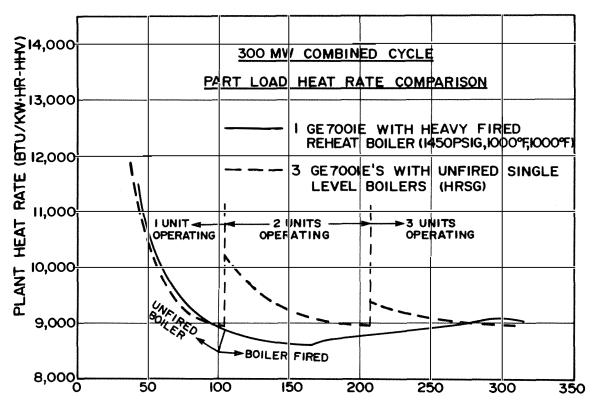

Figure 10. 300 MW Combined Cycle Part Load Heat Rate Comparison.

heating value (HHV) heat rate (42.5 percent LHV efficiency) is achieved. Once the boiler is fired and the supercharger and evaporative cooler started, plant output can be more than tripled to about 320 MW.

Increasing boiler firing initially reduces plant heat rate to 8660 HHV heat rate (44 percent LHV efficiency) at 170 MW when 1000°F superheat temperature is first reached. As the additional 150 MW is generated, heat rate gradually increases back to the unfired boiler level. From a part load perspective, plant load can be reduced to 30 percent of rated before heat rate exceeds its rated level.

Daily power swings of 70 percent of rated can be achieved with the gas turbine base loaded and boiler burners selectively lit/ extinguished.

As a comparison, the same 75 MW industrial turbine, but with a single level HRSG will produce about 100 MW plant output. To reach 300 MWs total, an additional two gas turbines (HRSG units) would have to be started. Therefore, to turn down power to 30 percent at night, two of the three gas turbine combined cycle units would have to be shutdown. Daily startup and shutdown of combined cycle units is not desirable. In addition, between 100 MW and 300 MW plant output, the single gas turbine with the heavy fired boiler has a 400 Btu/kW-hr heat rate advantage over the multiple unfired units.

A potential application of the West Texas combined cycle concept is in the re-configuration of existing power generation equipment. By combining existing peaking units and existing 200 MW steam generators, combined cycles with part load characteristics similar to those of Figure 10 can be created. An added benefit is the replacement of the individual heat rates (11900 HHV gas turbine and 9500 to 10000 HHV steam generator), with a combined 9000 HHV heat rate.

To convert the steam generator to combined cycle usage, air heater surface must be replaced by economizer surface so that exhaust gases instead of steam are used for feedwater heating. The forced draft fan can be moved ahead of the gas turbine to act as a supercharger, as described previously.

Supercharging (increasing the pressure at the gas turbine inlet) increases the gas bending loading on the first row of compressor blades approximately in proportion to the increase in inlet pressure. Thus, 40 in of H_2O static pressure on the inlet would cause a 10 percent increase in gas loading. Approval should be given by the OEM that his compressor will accept this additional load before supercharging is incorporated in any retrofit application. Fourteen industrial gas turbines have been supercharged in power generation for as long as 24 years.

CONCLUSIONS

Twenty three years of successful operating experience have demonstrated that good combined cycle efficiency over a broad load range can be obtained with a gas turbine combined with a heavy fired reheat boiler. Benefits in terms of maintenance cost and NO_x reductions, accompany this combined cycle configuration. This concept should be considered for reconfiguration of existing peaking gas turbine and stand alone steam generators, and for new utility and cogenerator combined cycles.

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