STEAM-INJECTED GAS TURBINE ANALYSIS:  
PART I – STEAM RATES  

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ABSTRACT  
This paper presents a three-part analysis of steam-injected gas turbines (simple and reheat) as follows: Part I – Steam Rates, Part II – Steam Cycle Efficiency and Part III – Steam-ReGenerated Heat (RGH). The analysis is based on the same approach used for large-utility steam turbines where one pound of throttle steam is passed through the steam turbine, and where enthalpy points are determined along the steam path. Work output, heat input and turbine efficiency are thus determined from this data. When considering a gas turbine, the steam-injection flow is separated from the main gas stream for analysis. Dalton’s and Avogadro’s laws of partial pressure and gas mixtures are applied. Accurate analysis of heat input, partial pressure expansion and heat-rejection steam-enthalpy points are computed with steam-table software.

INTRODUCTION  
Gas-turbine scholars, users and manufacturers are all becoming more interested in steam injection. Steam injection can take one or more forms. Water can be injected directly into the combustor to control NOx and therein produce steam. Steam can be alternately injected for both NOx control and power augmentation. Water can be evaporated by the compressor-discharge hot air – an example being the Humid Air Turbine (HAT) cycle as reported by Day and Rao (1992). Steam can be mixed with compressor air either at the discharge (GE LM2500 and LM5000) or at extraction (W/MHI 501F and 701F) to improve cooling. Another example of steam injection is the proposed Chemically-Recuperated Gas Turbine (CRGT) cycle where a mixture of steam and reformed natural gas becomes the gas-turbine fuel (Janes, 1989).

Humidified nitrogen return and/or coal-gas fuel provides a form of steam injection for the integrated coal-gas combined cycle (IGCC) as reported by Shell Oil Co. (Bayens and Cremer, 1991). The Department of Energy (DOE) Morgantown is proposing a water-injected regenerative indirect-fired coal-fuel cycle with some of the features of the HAT cycle as suggested by Parsons et al (1991). A further example of steam injection is the Compressed-Air Storage Cycle with Air Humidification (CASH) with or without integrated coal-gas fuel (IOGASH) as projected by Cohn and Nakhamkin (1992).

The injected steam produces more power for a given turbine (rotor) inlet temperature (TIT) in all cases but additional heat input is required to heat the steam to full TIT. There is a need to evaluate steam injection in terms of incremental power output and heat input. Up to now, the task of evaluating limited steam injection for a given turbine design has been left to the manufacturers. This paper provides analysis and insight for the individual turbine engineer because in the future we can expect the manufacturers to make greater design changes to optimize for steam injection for both power augmentation and cooling.

The steam-injected Allison 501-K, GE LM2500 and LM5000 turbines have been in operation for several years. Reliability, capital and operating costs and maintenance considerations have been field-evaluated and the results are positive. The door is now open to future applications and optimization.

PART I - Steam Rates - is presented in this section of the paper to evaluate potential power augmentation for a given amount of steam injection. Incremental output can be determined by dividing the particular injection steam flow (lb/hr) by the steam rate (lb/kW hr) to obtain kW incremental power output as is commonly done for steam turbines - but with the understanding of certain gas-turbine limitations. These limitations will be explained near the end of Part I.

EXISTING STEAM-INJECTION SYSTEMS  
The first steam-injected gas turbines in the mid 1950s piped steam directly to the inside of the combustor liners. Steam was injected into the compressor discharge (CD) air through ports in the...
casing during the 60s. More recently steam has been injected with the fuel (or around the fuel-nozzle tips) to control NOx.

As an example of present designs, a schematic diagram of the high-pressure section of the steam-injected General Electric STIG LM5000 is shown in Figure 1 (Smith, 1989). Seven percent steam by weight of the core air flow is injected in this particular high-pressure design. Turbine modifications are minimal so a nearly standard gas turbine can be used. Half of the steam is injected around the fuel tips and the top part of the compressor discharge. This latter portion flows around the outer part of the annular liner. The other half is injected into the compressor discharge to mix with the CD air and then to flow around the inside ID of the annular liner with a small amount being fed to the shaft sealing area and therefrom to the first-stage rotating blades for improved cooling. The mixture of steam and air on both sides of the combustor liner provides better first-stage nozzle-vane cooling.

Additional steam is injected in front of the low-pressure gas-generator nozzle ring (up to 6.5% of weight air flow) as is noted in Figure 2. This steam is passed through the interior of the nozzle vanes and is then exited at high velocity through nozzle holes at the vanes' trailing edges.

The vanes (high pressure and low pressure) provide the means for injecting the steam evenly to the gas stream at a high velocity so work can be extracted from the steam by the downstream rotating blades. The leading-edge stream exiting the high-pressure vanes expands with the air from full pressure as it helps film cool the vanes.

The power output of the gas turbine is increased from 34 MW to 52 MW as a result of the steam injection, yet the air flow remains about constant.

Intercooling does one other thing: it makes it possible to pass significantly more air mass through the gas generator core by utilizing an oversized low-pressure compressor (LPC). The HPC is essentially a constant inlet-volume flow device and pumps the same volume air rate, but the mass is increased indirectly as the ratio of the absolute LPC outlet temperature divided by the intercooled discharge absolute temperature according to both Charles' and Boyle's Laws ($PV = WRT$). The increase in mass flow for a given core is thus increased by a significant amount.

Figure 1. Steam-injection arrangement of the gas generator high-pressure section of the GE LM5000 full STIG

Figure 2. Steam-injection arrangement of the gas generator low-pressure turbine of the GE LM5000 full STIG

Figure 3. Schematic diagram of the steam-injected intercooled reheat gas turbine
Intercooling is receiving more attention by manufacturers (Leonard, 1992) and is being mentioned for future cycle consideration. Intercooling, however, has no effect on steam-injection power augmentation or heat input to the steam as can be determined by Figure 3.

In all proposed cases cited earlier in this paper, steam is injected downstream of the compressor discharge. In the case of the HAT and CASH cycles, part of the heat of compression of the LPC (intercooler heat) is used to evaporate water at a low partial pressure and low temperature for a very efficient heat-exchange arrangement (second law of thermodynamics).

In the early 50s industrial gas turbines firing at about 1400°F TIT (760°C) required twice as much work to compress the air as was left over to drive the load. Refer to Figure 4. Over the years the firing temperatures have been increased to produce more net power. By the year 1990, the new "F" technology gas turbines had reached TIT levels of 2300°F (1260°C). As a result, the ratio of compressor work to shaft output was reduced to a ratio of about one. The aero engines have dropped to a level of only about 1.5 because of their high-cycle pressure ratios (CPRs) of 30 to 35 and the extra compression work required.

The introduction of intercooling, steam injection and reheat reduces the compressor/output ratio to about .5. Reference is made to both Figures 3 and 4. The significance of this low ratio is important. The air compressor now becomes far less dominant. Compressor efficiency and fouling degradation becomes less significant. The tail (compressor) no longer wags the dog (power output). Steam injection becomes far more significant because power output is increased without the required work of air compression. Water is pumped as a liquid at very low power input and not compressed as a gas at very high power input. The air then becomes a means to heat the steam directly before expansion and stoichiometric conditions are approached for full steam injection.

STREAM SEPARATION

Reference is made to Figure 5 showing a schematic flow diagram of a steam-injected expander turbine. Both gas (heavy arrow) and steam (light arrow) enters the power turbine (PT) at a given TIT. TIT is defined by U.S. heavy duty and aero manufacturers as the average total temperature of the gas and nozzle coolant mixture striking the first-stage rotating blades. The coolant (air and/or steam) and the combustor-outlet gas-stream mixture condition is projected back to the first-stage nozzle inlet at essentially constant entropy to obtain this value. The combustor outlet temperature is therefore some 150 to 175°F (83 to 97°C) higher than the TIT.

The steam part of the working fluid, at its own partial pressure, expands through the PT at a given isentropic efficiency and exits at its own specific temperature and partial pressure enthalpy. The steam, with its sensible heat component, then passes through a heat-recovery steam generator (HRSG). Schematically all or zero steam flow can pass through the portion of the HRSG (A) that produces steam for process use. The other part passes through the part of the HRSG (B) to produce incremental steam for the steam injection. The two streams then meet again schematically to form the stack gas.

Figure 5 is helpful in visualizing stream separation and recovery separation for analytical purposes knowing that in reality there is no such separation or division of flow. There is no change in total enthalpy of the stream.

**Figure 5.** Stream separation schematic diagram

STEAM RATES

The steam-rate calculating procedure can now be presented using the previous background of stream separation and TIT definition. A computer program called "Steam 92" (Strong, 1992) has been used to obtain accurate steam enthalpy and temperature numbers for each point. The ASME steam tables are limited to 1500°F (816°C) and do not give partial pressure data whereas the software provides very accurate answers up to and beyond the TITs encountered by present-day gas turbines.

**Figure 4.** Ratio of compression work divided by shaft output versus TIT

SHRSG A 100% FLOW

H2O IN

PT

H2O IN

H2O IN

H2O IN

HTSG B 0-100% FLOW

TO STEAM INJECTION

TO PROCESS HEATING

Steam Flow

Gas Flow

LOAD
Partial pressures have been used in the calculations. However, full throttle pressure expansion introduces only a small error of .87 BTU/lb (2.02 KJ/Kg) when expanding steam at constant entropy from 450 psia (approximately 30 atmospheres) at 2400°F TIT (1316°C) to 15 psia (one atmosphere) when comparing full pressure to 1% partial pressure (4.5 psia to .15 psia). The expansion-exit temperature variation is only 1.89°F (1.05°C). A computer printout is included in the Appendix showing the entering and exiting data. Partial pressure expansion provides more accurate answers, but full pressures can be used without introducing a significant error.

**Simple-Cycle Gas Turbine**

A schematic diagram of the steam-injected simple-cycle gas turbine is presented as Figure 6. Steam or water flow $Q_3$ enters combustion chamber (CC) at enthalpy $h_6$. The flow is heated to $h_7$ and is expanded to $h_e$. (The particular numbered enthalpy subscripts are used to be consistent with the rest of the paper.) The steam exhausts in most cases to a HRSG. Feedwater is heated from an assumed temperature of 100°F (38°C) ($h = 68$). In the case of steam, the enthalpy rise of the water and steam inside the HRSG plus the enthalpy rise across the CC gives the total enthalpy of $h_7$. The work output is the enthalpy drop across the turbine ($h_7 - h_e$) taking into consideration expansion efficiency.

In the case of water injection, water is heated by incremental fuel from 100°F (38°C) all the way to the CC outlet ($h_e$). The work output remains the same - that is ($h_7 - h_e$).

![Figure 6. Schematic diagram of a steam-injected simple-cycle gas turbine](image)

Steam rates are calculated using the following formula:

$$SR = \frac{34121416}{(h - h_6)} = \text{Lbs/KW Hr} \tag{1}$$

It is important to note that if steam is injected directly inside the combustor liner - or such that it enters the CC - the steam is heated to the combustor outlet temperature. If the steam is injected into the compressor-discharge outlet, the steam is mixed with the CD air and a portion of the steam is used with the air to cool the blades and rotor. In this latter case, the TIT lower temperature takes into account a cooling penalty for the first-stage nozzle vanes. A cooling penalty for the downstream blading and rotor is not taken into account.

![Figure 7. Steam rate versus gas-turbine expansion ratios (low)](image)

Figures 7 and 8 are graphs of Steam Rate (lb/KW hr) versus Expansion Ratio ($P_2/P_1$) for TIT inlet temperatures of 2000 to 2600°F (1093 to 1427°C). Isentropic expansion efficiencies of 88% and 92% have been selected to span typical gas turbines. Expansion ratios start at 3 in Figure 7 and end at 50 in Figure 8.
Referring to Figure 8 and considering a 15 expansion ratio for 2400°F TIT (1316°C) and 90% efficiency, point A is read off the graph to be 5.7 l/h/kW hr (26 Kg/kW hr). Considering a 30 expansion ratio for these conditions, the steam rate drops to 4.8 lbs/KW hr (2.2 Kg/kW hr) (point B). The shaft power output is simply the steam injection flow in lbs/hr divided by the steam rate. If the injection flow is 200,000 lbs/hr (90,718 Kg/hr) for point B, the shaft power developed would be 41.7 MW.

There are some practical considerations such as fixed nozzle area, extra cooling required and reduction in TIT which can reduce this potential output. These considerations will be reviewed at the end of Part I of the paper.

Simple-Cycle Gas Turbine With Topping Steam Turbine

Let us now consider the case where steam is first run through a topping (non-condensing) steam turbine before it is injected into the gas turbine. The steam turbine can be connected in tandem to the outer end of the electric generator, or alternately it can be in the form of a large separately-positioned steam turbine. Reference is made to Figure 9.

Figure 9. Schematic diagram of a steam-injected simple-cycle gas turbine with a topping steam turbine.

The total enthalpy rise is the HRSG enthalpy rise to the steam turbine throttle plus the enthalpy rise from the steam-turbine exit point to the gas turbine TIT. The work output is the enthalpy drop across the steam turbine plus the enthalpy drop across the gas turbine.

The topping steam turbine increases output per unit weight of steam flow and thereby decreases the steam rate. Figure 10 is a graph showing steam rates for various pressure ratios, throttle pressure and TITs. Note that the exhaust temperature of the gas turbine limits the steam temperature as the CPR increases; and therefore the steam rates flatten out above a pressure ratio of about 20. Even so, very low steam rates of 4 lbs/KW hr (1.8 Kg/KW hr) are possible for a TIT of 2400°F (1316°C) coupled to a 2400 psig (166.5 bar) throttle pressure. This steam rate is far better than can be achieved with a 2400 psig (166.5 bar) reheat condensing steam turbine.

When considering a topping steam turbine, the steam rates of the topping steam turbine can be determined from the ASME Theoretical Steam Rate Tables by applying an appropriate expansion efficiency. Alternately steam-table software can be used as was done for Figures 7 and 8. The overall combined steam rate can be readily calculated using the following formula:

\[
SR = \frac{1}{SR_{\text{at}}} - \frac{1}{SR_{\text{gt}}} = \text{Lbs/KW hr} \quad (2)
\]

where \(SR\) = overall steam rate
\(SR_{\text{at}}\) = steam rate of topping steam turbine
\(SR_{\text{gt}}\) = steam rate of gas turbine.

It can be noted that the reciprocal of the steam rate is actually work per unit mass of steam (KWhr/lb steam). These work units can be added together and then converted back to steam rates.

Reheat Gas Turbine
Steam-injection steam rates are lower for reheat (RH) gas-turbines. The steam is reheated in the reheat combustor after partial expansion and more power is produced for a given injection rate. Figure 11 presents a schematic flow diagram of a steam-injected reheat gas turbine. Steam can be injected into both the gas generator combustor (CC) and the reheat combustor (RH).

The steam or water that is injected in the gas generator combustor (CC) is heated to TIT level and is then expanded to produce driving power for the air compressor. Considering appropriate nozzle-area changes and rotating-blade modifications, the gas-generator exit pressure will rise. Additional work is produced by the steam and less by the hot gas, and the expansion ratio is reduced for the fixed power requirements of the air compressor. Reference is made to Figure 12 which presents a T-S diagram illustrating the rise in pressure. Without steam injection the gas is expanded from TIT point 3 to point 4. When steam is injected the gas is only expanded to point 4' and the steam work makes up the difference as the steam also expands from point 3 to point 4'. The shaded area 4' - 4 - 5 - 5' represents the additional work provided by the air side for a given pre-selected exit temperature, point 6. Incremental power is thus produced by both the air side and the steam side of the two stream system.

As an indication of the pressure rise, reference is made to Figure 13 which is a plot of power-turbine...
Figure 11. Schematic diagram of a steam-injected reheat gas turbine

Figure 12. T-S diagram showing effect of steam injection

Figure 13. Power-turbine expansion ratio versus steam-injection rate

expansion ratio versus steam-injection rate. Considering zero steam injection, the power-turbine expansion ratios of different gas generator (GG) TITs are rather low - ranging from 4.6 to 6.25 (Rice, 1982). A steam injection rate of 15% increases the power-turbine expansion ratios to a range of 9.5 to 11 as can be noted. Data taken from various studies and that contained in the GE ISTIG study for the California Energy Commission (Janes, 1989) have been used to prepare this graph. These higher expansion ratios are ideally suited for a three-stage power turbine.

The steam-rate graph of Figure 14, is a plot of steam rate versus overall cycle pressure ratio both with and without a topping steam turbine for various TIT levels. The bottom line, which represents a topping steam turbine, shows steam rates ranging from 3.1 to 3.5 lb/KW hr (1.4 to 1.6 Kg/KW hr) which are about half the steam rates of a 2400 psig 1000°F (166.5 bar 538°C) reheat condensing steam turbine. In other words, twice as much power is produced for a given steam flow.

Figure 14. Steam-rate curves versus expansion ratio of a reheat gas turbine - with and without a topping steam turbine

PRACTICAL CONSIDERATIONS

There are several practical points to consider when dealing with steam-injection for existing gas turbines. New gas turbines being designed from a clean sheet of paper should also take into consideration practical limitations. The steam rates given in this paper do not take into consideration such things as nozzle-area requirements, compressor pressure-ratio and flow changes, changes in compressor efficiency, increases in rotating-blade loading, additional blade-cooling requirements and changes in expansion efficiency. The power turbine of a reheat gas turbine requires more rotor, blade and casing cooling which necessitates different steam-cooling
methods. Some of these points will be discussed in the following paragraphs.

When a gas turbine is steam injected by moderate rates of 2 to 4% weight of air flow, no nozzle-area changes are generally needed. Reference is made to Figure 15 - a typical compressor map taken from an EPRI Report (Eustis and Johnson, 1990). When steam is injected, the compressor discharge pressure will rise, the compressor efficiency will be reduced and the air flow will also fall off. The net results will tend to increase the steam rates given in Figures 7 and 8. At high-ambient inlet-air temperatures, however, the operating points are lower, and compressor efficiency can be improved for a given nozzle area at very little reduction in air flow.

Another limitation example in using Figures 7 and 8 is cited in the case of the new LM6000 gas turbine. This gas turbine, adapted from the GE CF6 dash 80 aero engine (without turbine flow-path modifications) has an exit flow limitation. The power-turbine exit velocity is about Mach .8, and if any steam or water is injected, the TIT has to be lowered to maintain the flow volume. The parametric graph of Figure 16 illustrates this point. The top family of curves represent aero derivative engines and the bottom curves the heavy-duty machines.

Published data is plotted for the LM6000 as points A, B and C as reported by deBiasi (1990) and Johnson (1992). Point A is for the LM6000 without any steam injection. Point B is for this engine for 25 ppm NOx utilizing steam injection where the TIT has to be lowered to maintain the flow volume. The parametric graph of Figure 16 illustrates this point. The top family of curves represent aero derivative engines and the bottom curves the heavy-duty machines.

As a note of interest, the GE Frame 7F and 7FA gas turbines are spotted in Figure 16 as points F and FA. Also point D represents the new GE-90AD engine when adapted as an industrial prime mover (Leonard, 1992). Note points Ar and Dr for reheat gas turbines.

If a gas turbine were designed specifically for steam injection and steam cooling, then all of the above considerations could be taken into account to capture the full potential of steam injection in terms of the lowest possible steam rate. Parts II and III of this paper will deal with cycle-efficiency aspects of steam injection - which are of equal importance.

**CONCLUSION**

This paper, Part I, presents steam rates for various gas turbines with and without a topping steam turbine function. The steam rates for high cycle-pressure ratio and high TIT gas turbines are appreciably lower than steam rates for condensing steam turbines. The steam rate is about 4 lb/KW hr (1.8 KG/KW hr) when considering a CPR of 20, a TIT of 2450°F (1343°C) and when a 2400 psig 1000/1000°F (166.5 bar 538/538°C) RH topping steam turbine is utilized. The graphs presented can be used to obtain estimated steam rates for any given cycle pressure ratio and TIT. Both simple and reheat-gas-turbine cycles are included.

The lowest steam rate is obtained for the reheat gas turbine utilizing a topping steam turbine - for example using extracted steam from a steam turbine. Steam rates as low as 3.1 lbs/KW hr (1.4 KG/KW hr) are projected for the reheat gas turbine with a 2400 psig, 1000/1000°F (166.5 bar 538/538°C) RH topping steam turbine.

This paper provides a procedure for calculating steam-injection incremental power. Useful steam-rate curves are given. Various methods of steam injection have been presented, and practical limitations have been discussed.
ACKNOWLEDGMENTS

Acknowledgments are included at the end of Part III of this paper.

REFERENCES


APPENDIX


