Check for updates

THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS
345 E. 47th St., New York, N.Y. 10017

The Society shall not be responsible for statements or opinions advanced in papers or discussion at meetings of the Society or of its Divisions or Sections, or printed in its publications. Discussion is printed only if the paper is published in an ASME Journal. Papers are available from ASME for 15 months after the meeting.

Printed in U.S.A. Copyright © 1994 by ASME

94-GT-435

GAS TURBINE AND COMBINED CYCLE TECHNOLOGIES FOR POWER AND EFFICIENCY ENHANCEMENT IN POWER PLANTS

Meherwan P. Boyce, Cyrus B. Meher-Homji, and A. N. Lakshminarasimha
Boyce Engineering International Incorporated
Houston, Texas

Downloaded from http://asmedigitalcollection.asme.org/GT/proceedings-pdf/GT1994/78866/V004T10A020/2405145/v004t10a020-94-gt-435.pdf by guest on 28 January 2022

ABSTRACT

A wide variety of gas turbine based cycles exist in the market today with several technologies being promoted by individual Original Equipment Manufacturers. This paper is focused on providing users with a conceptual framework within which to view these cycles and choose suitable options for their needs. A basic parametric analysis is provided to show the interdependency of Turbine Inlet Temperature (TIT) and Pressure Ratio on cycle efficiency and specific work.

NOMENCLATURE

$\dot{m}$ =	Mass Flow Rate, lb/sec or kg/sec
h =	Enthalpy, Btu/lb or KJ/kg
Q =	Heat added, Btu/Sec or KJ/sec.
η =	Efficiency
r_p =	Compressor Pressure Ratio
T =	Temperature, °F or °C
LHV =	Lower Heating Value
γ =	Ratio of Specific Heat
Cp =	Specific Heat at Constant Pressure, BTU/lb °R or KJ/KgK

Subscripts

c =	Compressor
t =	Turbine, total
cyc =	Cycle
Opt =	Optimum
1,2,3,4 =	Station designation
f =	Fuel
a =	Air

INTRODUCTION

The oil crisis and current environmental awareness have driven industries towards energy efficient operation and conservation. In spite of this trend, the demand for power is on the rise. It is estimated that US will add 113 GW of new generating capacity in the next decade. Of this total, 20 GW is expected to come from simple cycle gas turbines and 52 MW from combined cycle configurations [Collins, 1993]. In order to increase power production, simple cycle gas turbine operators can increase power output by retrofit modifications such as installing evaporative and inlet coolers and by using steam injection. Some are opting for cogeneration and some for combined cycle operation. Large aging steam turbine utilities are repowering by adding gas turbines and Heat Recovery Steam Generators (HRSG) to the operating cycle.

In this paper, a basic overview of GT cycle parameteric analysis is covered to give the user a background of key cycle operating parameters. Examples of various retrofits available for gas turbine power enhancement with examples are provided. Combined cycle and repowering cycles will also be covered. The objective is to provide the user with a conceptual understanding of cycle parameters and power augmentation methods.

[1]Fellow, ASME
[2,3] Member, ASME

Presented at the International Gas Turbine and Aeroengine Congress and Exposition
The Hague, Netherlands — June 13–16, 1994

GENERAL GAS TURBINE PARAMETERIC ANALYSIS

Application of the first law to the air-standard Brayton cycle shown in Figure 1 gives us the following relationships [Boyce M.P. 1987, Boyce M.P. and Chen, M. 1978]:

Work of compressor

$$W_c = \dot{m}_a (h_{t_2} - h_{t_1}) \tag{1}$$

Work of turbine

$$W_t = (\dot{m}_a + \dot{m}_f)(h_{t_3} - h_{t_4}) \tag{2}$$

Total work output

$$W_{cycle} = W_t - W_c \tag{3}$$

The heat added to the system

$$\begin{aligned} 2Q3 &= \dot{m}_f \times LHV(Fuel) \\ &= (\dot{m}_a + \dot{m}_f)(h_{t_3}) - \dot{m}_a h_{t2} \end{aligned} \tag{4}$$

Thus the overall thermal cycle efficiency

$$\eta_{cycle} = \frac{W_{cycle}}{2Q_3} \tag{5}$$

A simplified approach to calculating the overall cycle efficiency can be obtained from the above relationships if one were to make certain assumptions such as (1), (2) specific heat (C_p) and the specific heat ratio (γ) remain constant throughout the cycle, (3) the pressure ratio in both the compressor and the turbine is the same, (4) all components operate at 100 percent efficiency. With these assumptions, the following relationships for efficiency can be obtained.

$$\eta_{cycle} = 1 - \frac{1}{r_p^{(\gamma-1)/\gamma}} \tag{6}$$

$$\eta_{cycle} = 1 - \frac{T_1}{T_2} \tag{7}$$

$$\eta_{cycle} = 1 - \frac{T_4}{T_3} \tag{8}$$

Thus, a cursory inspection of equations 6, 7, and 8 indicate that overall efficiency of a cycle can be improved by increasing the pressure ratio, decreasing inlet temperature and increasing the turbine inlet temperature. These relationships, however, do not indicate the effect on the work output of the unit or the inaccuracies which are inherent in them at high pressure ratios, high turbine inlet temperatures, and with inefficiencies in the components.

To obtain a more accurate relationship between the overall thermal efficiency and the turbine inlet temperatures, overall pressure ratios, and output work, the following relationships were obtained. For maximum overall thermal cycle efficiency, the following relationship provides the optimum pressure ratio at a given inlet temperature, turbine inlet temperature, and component efficiencies.

$$(r_p)_{opt} = \left[\sqrt{\left(\frac{T_3\eta_t}{T_1 + \eta_t T_3 - T_3}\right)^2 - \left(\frac{\eta_c\eta_t T_3^2 - \eta_t\eta_c T_1 T_3 + \eta_t T_1 T_3}{T_1^2 + \eta_t T_1 T_3 - T_1 T_3}\right)} \right]^{\frac{\gamma}{\gamma-1}} \tag{9}$$

This equation reduces to:

$$(r_p)_{opt} = \left(\frac{T_3}{T_1}\right)^{\frac{\gamma}{\gamma-1}} \tag{10}$$

if one assumes $\eta_c = \eta_t = 1.0$. However, this simplified equation is not very accurate. The optimum pressure ratio for maximum output work is given by the following relationship:

$$(r_p)_{opt} = \left[\left(\frac{T_1}{T_3}\right)\left(\frac{1}{\eta_c\eta_t}\right)\right]^{\frac{\gamma}{2-2\gamma}} \tag{11}$$

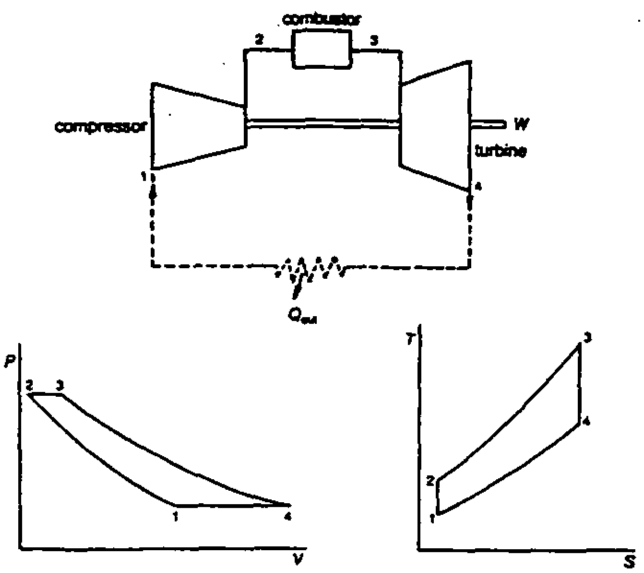

Figure 1. The air-standard gas turbine cycle.

Downloaded from http://asmedigitalcollection.asme.org/GT/proceedings-pdf/GT1994/78804/V004T10A020/2404143/v004t10a020-94-gt-453.pdf by guest on 26 January 2022

Equations 9 and 11 give a relatively accurate analysis, but the two values do not necessarily equal each other. The discrepancy between these equations and the more complete analysis is due to the fact that in the above equations the following assumptions were made: (1) $m_f << m_a$, (2) that C_p and γ are constant throughout the system, and (3) the pressure ratio is constant throughout the system . The advantage of equations 9 and 11 is that a fast, quick analysis can be obtained.

REVIEW OF PRACTICAL CYCLES

In all cycles compared in this section, the inlet temperature, inlet pressure (open cycles only), and the various efficiencies of the components are kept constant. In this manner, an accurate comparison between the cycles can be obtained.

The efficiency of the compressors has been assumed to be 85 percent and the turbine efficiency to be 87 percent. These numbers have been chosen as being practical taking deterioration into account. Combustor efficiency is assumed at 97 percent, with a pressure loss of about 5% from exit of compressor to turbine inlet. Efficiencies have been kept constant; consequently, even though particular designs may not match these efficiency numbers, the *trends* obtained here are still valid for various cycles.

1. Simple Cycle (Brayton Cycle)

This is the most common type of cycle used in gas turbines today. Analysis of this cycle indicates that an increase in turbine inlet temperature and compressor pressure ratio of the turbine increases the cycle efficiency. This is shown in Figure 2 [Hopkins, 1991] with specific power output as ordinate. Specific power output is an indicator of gas turbine size or power output based on mass flow rate considerations. The higher the specific power output, the smaller the size of the gas turbine. Optimum pressure ratio varies with the turbine inlet temperature from an optimum of about 12:1 at a temperature of 2100 °F(1149°C) to a pressure ratio of about 15:1 at a temperature of 2400 °F. (1316 °C.). As depicted in Figure 2, a pressure ratio of 14:1 for a temperature of 2300 °F. (1260°C.) would be the most optimum based on size, thermal efficiency and power output.

If we are to assume that the type of compressor used is an axial flow design, typically fifteen to twenty stages are used with a pressure ratio of 1.13 to 1.17 per stage. While 1.14 is a conservative stage design pressure ratio, 1.16-1.17 pressure ratio per stage compressors are of more recent origin. The stage loading (Pr. Ratio/Stage) can be higher for aeroderavative engines. A higher stage loading implies fewer compressor stages and therefore lower associated costs.

In any gas turbine design, a compromise is struck between higher pressure ratios per stage and stage efficiency. While current day Advanced Gas Turbines (AGTs) have 2300-2350 °F. (1260-1288°C.) firing temperatures, turbines with a firing

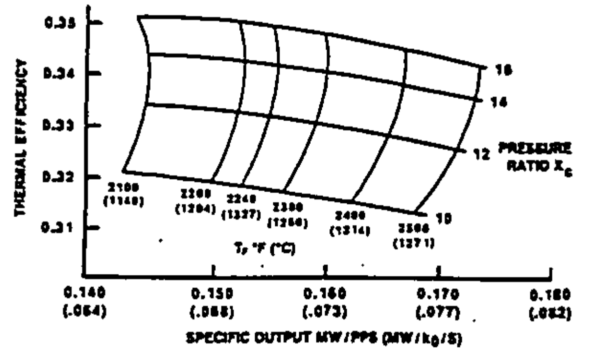

Figure 2. Performance map showing the effect of pressure ratio and turbine inlet temperature on a simple cycle. (Hopkins, 1991)

temperatures of 2100 °F.(1149 °C.) are prevalent. An important point to be noted is that as the compressor pressure ratio and firing temperature increase, the amount of cooling air required to cool the turbine also increases which penalizes the cycle efficiency. The bleed air flow in AGTs is typically in excess of fifteen percent of the air mass flow rate.

Figure 3 [Hopkins, 1991] shows the effect of bleed air extraction on heat rate and power output. In order to improve the starting characteristics, the high pressure ratio compressors almost always have variable stators in their front stages.

While single shaft machines are common in utility applications, split shaft machines are more common in mechanical drive applications in process industries, although there are a number of split-shaft aeroderivatives in utility applications. The advantage of the free turbine or split shaft gas turbine is mainly in the area of high torque, variable speed and improved off-design performance.

Figure 4 shows a typical performance map of a split-shaft simple cycle gas turbine.

2. The Reheat Cycle

The reheat cycle produces more work per pound of air than the other cycles. The reheat cycle, as shown in Figure 5, consists of a two-stage turbine with a combustion chamber before each stage. The assumptions made in this study are that the high pressure turbine drives the axial compressor, power turbine drives the load, and that the gas leaving the high pressure turbine is then reheated to the same gas generator firing temperature. This reheat cycle has an efficiency which is slightly less than that encountered in a simple firing cycle, but produces about a 35 percent increase in the shaft output power, as shown in Figure 6. However, advanced reheat cycles which use high pressure ratio compressors up to 30:1, and high firing temperature of the order of 2300 °F(1260°C) using high technology combustors can produce in excess of 150 MW and

Downloaded from http://asmedigitalcollection.asme.org/GT/proceedings-pdf/GT1994/78866/V004T10A020/2405145/v004t10a020-94-gt-435.pdf by guest on 28 January 2022

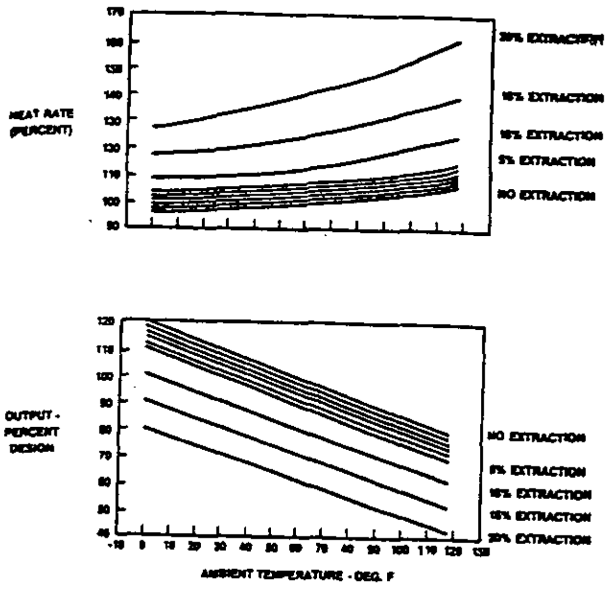

Figure 3. Effect on output and heat rate of air extraction. (Hopkins, 1991)

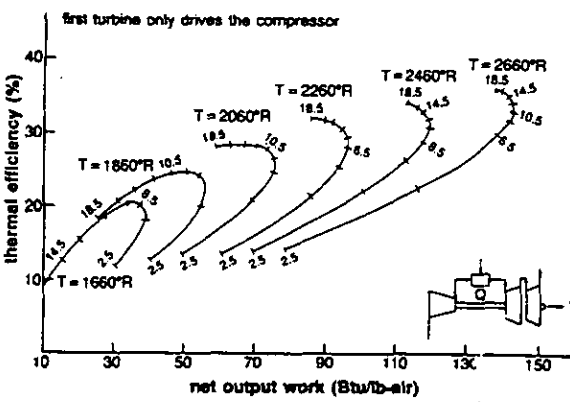

Figure 4. Performance map showing the effect of pressure ratio and turbine inlet temperature on a simple cycle split shaft system.

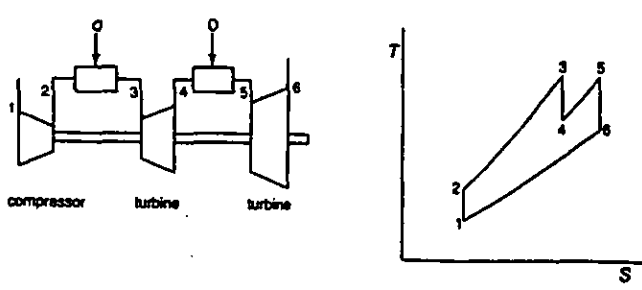

Figure 5. The split shaft reheat gas turbine cycle.

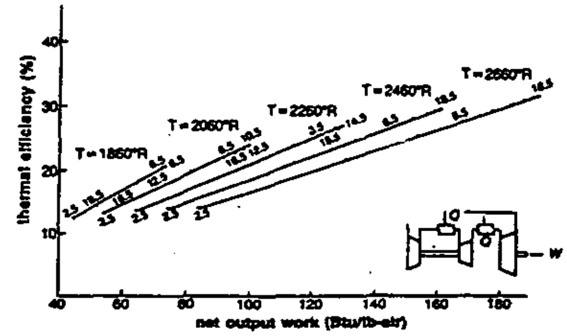

Figure 6. Performance map showing the effect of pressure ratio and turbine inlet temperature on a split shaft reheating cycle.

still have high efficiency of the order of 38 percent. However, more recently with the advancement of technology single shaft reheat cycles have been possible.

To obtain both an increase in the power output and the thermal efficiency, the natural conclusion would lead us to reheat, regenerative cycle. In this cycle, the hot gases leaving the power turbine are used to heat the cooler air leaving the compressor. The combination gives us an increase in efficiency of about 40 percent and an increase in power output of about 35 percent, as indicated in Figure 7.

The reheat regenerative cycle has been put to use in Compressed Air Energy Storage (CAES) plants such as the 110MW McIntosh (Alabama) facility. In this cycle, air is first compressed during nights and weekends when electricity demand is low and stored in large underground caverns. During peak power demand periods, the compressed air is released and fired for expansion in hot gas expanders or turbines. The expander train operates in reheat cycle with the hot gases leaving the HP turbine refired before entry into LP turbine.

Many other cycles such as the regenerative and a combination of regenerative, reheat and intercooled cycles are possible, but are not that common. To sum up, it is obvious that high compressor pressure ratios and turbine inlet temperatures increase power output and cycle efficiency.

3. The Intercooled Simple Cycle

A simple cycle with intercooler can reduce total compressor work and improve net output work. Figure 8 shows the simple cycle with intercooling between compressors. The assumptions made in evaluating this cycle are that the compressor interstage temperature equals the compressor inlet temperature, compressor efficiencies are the same, and pressure ratios in both compressors are the same and equal to $(\sqrt{P_2/P_1})$.

Downloaded from http://asmedigitalcollection.asme.org/GT/proceedings-pdf/GT1994/78866/V004T10A020/2405145/v004t10a020-94-gt-435.pdf by guest on 28 January 2022

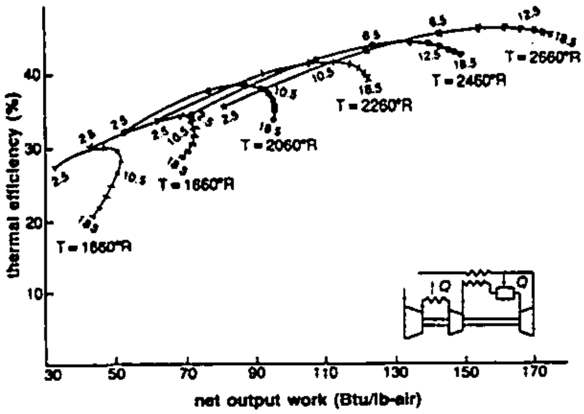

Figure 7. Performance map showing the effect of pressure ratio and turbine inlet temperature on an intercooled regenerative cycle.

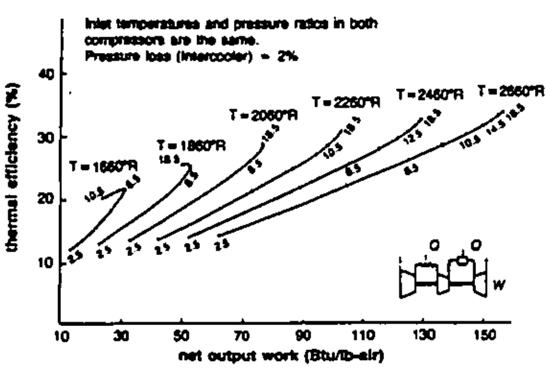

Figure 8. Performance map showing the effect of pressure ratio and turbine inlet temperature on an intercooled cycle.

The intercooled simple cycle reduces the power consumed by the compressor. A reduction in consumed power is accomplished by cooling the inlet temperature in the second or other following stages of the compressor to the same ambient air and maintaining the same overall pressure ratio.

POWER AND EFFICIENCY AUGMENTATION TECHNOLOGIES

Gas turbine power and efficiencies degrade with time. The main reasons for degradation of gas turbine performance are:

a. Ambient temperature, pressure and humidity variations.
b. Gas turbine fouling.
c. Gas turbine erosion and tip clearance increase.

The effects of fluctuations in gas turbine performance due to ambient conditions and fouling are temporary in nature. The effect of erosion on gas turbine performance is permanent. Installing evaporative coolers and inlet chillers are designed primarily to offset ambient temperature effects. Improvements obtainable from periodic gas turbine washing to alleviate fouling problems are well known. Steam injection which is primarily used for power augmentation and reduction in Oxides of Nitrogen in flue gas emissions in some cases can also be used to offset permanent degradation of gas turbines caused due to erosion and tip clearance increases. The modeling and evaluation of gas turbine performance deterioration has been covered by Lakshminarasimha et al (1993) Compressor fouling problems are discussed by Meher-Homji (1990)

1. Evaporative and Inlet Cooling systems

Under a given set of operating conditions, gas turbines are constant volume flow rate machines. Thus, if the density of air entering the machine increases, such as during a cold day, gas turbine mass flow rate increases, boosting the power and efficiency of the machine. When installed in gas turbine inlet systems, evaporative coolers and inlet chillers produce additional power during hot days. Reduction of power from gas turbine during hot days could be as much as 25 percent. Evaporative coolers do not require input energy and hence are very attractive.

Figure 9 [Hopkins, 1991] shows the effectiveness of a typical evaporative cooler, having an 85 percent efficiency of cooling, to increase the power output and decrease the heat rate. The figure clearly indicates that evaporative coolers are effective under hot ambient and low humidity conditions. Below 60°F(15.5°C), however, the effectiveness of evaporative coolers decrease. Poor quality of water in the cooler can increase the susceptibility of compressor fouling and also corrosion. Thus, it is important that the quality of water used in the cooler be pure.

Inlet chillers are more expensive than simple evaporative coolers. However, inlet chillers are not limited by low humidity conditions. Inlet chillers can reduce the inlet air temperature to near 32°F(0°C) and are only limited by inlet icing problem, coolant flow rate and heat transfer effectiveness of the coil. Several utilities have begun to use inlet cooling devices to improve gas turbine performance. Studies have shown that the payback of the initial investment depends upon two main factors: a) average annual weather conditions and b) size of the gas turbines. Investment can be recovered typically within three years. A review of inlet cooling approaches can be found in a paper by Meher-Homji and Mani [1983].

2. The Steam Injection Cycle

Steam injection has been used in reciprocating, as well as gas turbines, for a number of years. Steam injection increases power output, improves efficiency, and reduces exhaust emissions. Corrosion problems are the major concern in such a system. The advantages however, far outweigh the problems. In cogeneration and combined cycles, steam injection is an attractive option as it is directly available from the HRSG. At

Downloaded from http://asmedigitalcollection.asme.org/GT/proceedings-pdf/GT1994/78866/V004T10A020/2405145/v004t10a020-94-gt-435.pdf by guest on 28 January 2022

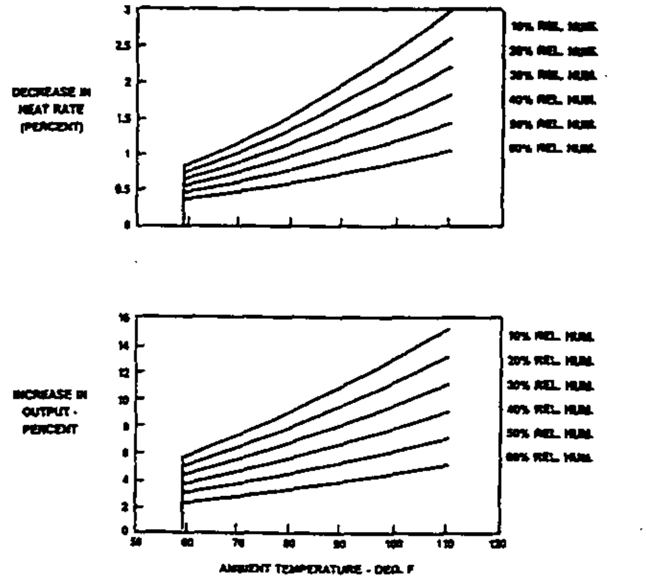

Figure 9. Effect of evaporative cooling on output and heat rate (Hopkins, 1991).

times of reduced steam demand, additional energy can be generated by injecting it into the gas turbine. Also, if the gas turbine performance degrades, steam injection can be utilized to offset lost performance.

There are many methods of water/steam injection. In the simplest method water is injected into the compressor discharge air where, due to high compressor discharge temperature, it gets converted into steam and the steam gets superheated. The steam increases the mass flow rate through the turbine thus producing additional work. As the steam is injected downstream from the compressor, there is no increase in compressor work however, one must be careful of surge problems during startup and shutdown. Injection at compressor discharge is done for power augmentation. Steam can also be directly injected into the head of the combustion chamber inlet. A typical steam injection cycle is shown in Figure 10. The effect of steam injection on power output and heat rate under different ambient temperature variation conditions is depicted in Figure 11[Fogg, 1991].

During constant Exhaust Gas Temperature mode of operation in single shaft machines, compressor inlet guide vanes need to be modulated with increase in steam injection rate. The effect of VIGV modulation and steam injection on power production enhancement is shown in Figures 12 [Burnham et. al., 1988]. The effect of steam injection under different modes of gas turbine operation, such as constant firing temperature mode, constant power output mode and constant fuel input mode, on both power output and heat rate are shown in Figures 13 and 14 respectively [Little et. al, 1988].

Under design operating conditions, exhaust gas temperature spreads remain constant as the steam injection rate is increased, as seen in Figure 15a. Under EGT control and VIGV modulation, at high steam injection rates, EGT spread can dramatically increase, as shown in Figure 15b. An increase in EGT spread is deleterious to hot section health and should be maintained within predetermined limits for safe operation. In a typical gas turbine where large quantity of steam is injected into the gas turbine gas path for producing additional power and efficiency, steam injection occurs at multiple locations in a turbine such as the compressor discharge port, fuel nozzle and through the first stage turbine nozzle vanes. For some operators, this is an alternative to a full cogeneration plant which uses both gas and steam turbines.

As can be seen from the Figure 10 described earlier, steam

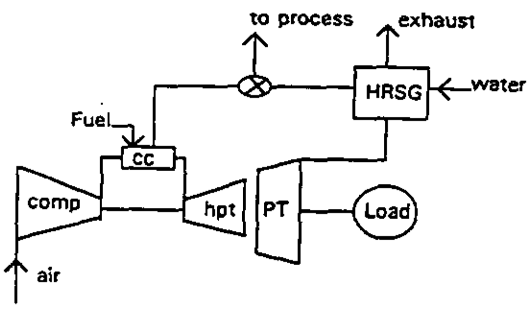

Figure 10. A typical steam injection cycle.

for this mode comes from the Heat Recovery Steam Generator (HRSG), as in the case of the cogeneration plant. However, unlike a cogeneration plant which involve an additional cost of steam turbine, the steam is directly injected into the gas turbine to produce additional work. The advantage of this cycle is that it offers fully flexible operating cycle, since the amount of steam injected can vary with load requirements and steam availability.

Three aeroderivative gas turbine in which this cycle has been successfully implemented are the General Electric LM1600, LM2500 and LM5000. Table 1 summarizes published GE performance data for these gas turbines with and without STIG option [Fogg, 1991].

	LM1600	LM2500	LM5000
Dry Power (MW)	13.0	21.5	33.1
Dry Thermal Efficiency (%)	34.0	35.0	36.0
STIG Power (MW)	16.1	26.3	51.9
STIG Thermal Efficiency (%)	40.0	38.0	43.0
Fuel Nozzle Steam Flow (PPH)	9,700.0	19,700.0	39,100.0
Comp Disch Steam Flow (PPH)	22,300.0	20,300.0	44,900.0
LPT Nozzle Steam Flow (PPH)	0.0	0.0	55,400.0

Table 1. Summary of STIG and Dry Aeroderivative GTCycle Performance (Fogg, 1991)

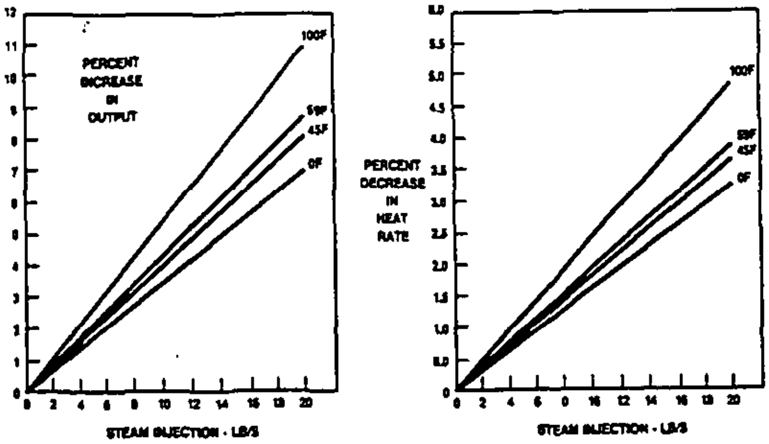

Figure 11. Effect of steam injection on output and heat rate (Fogg, 1991)

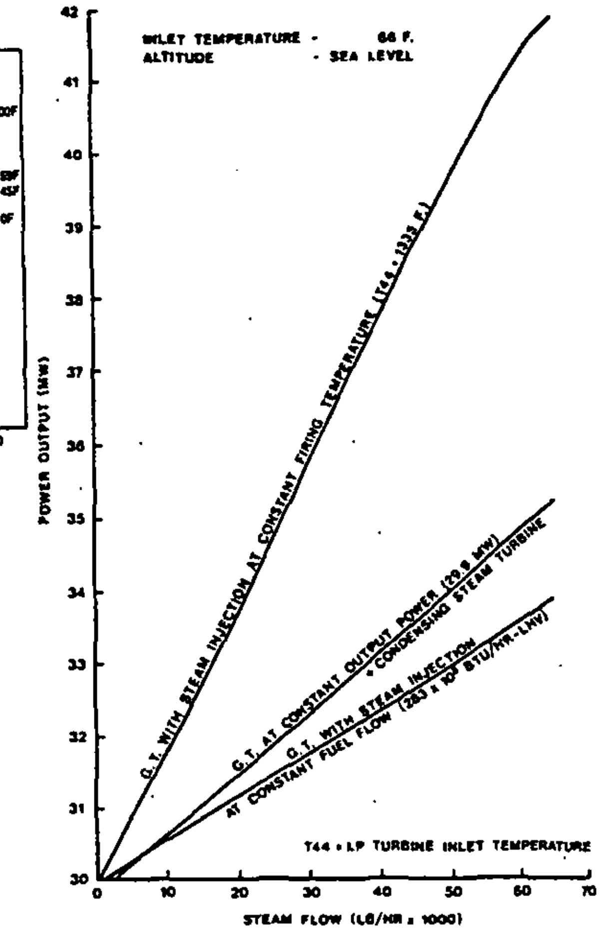

Figure 13. Gas turbine generator output with steam injection. (Little, et. al, 1988)

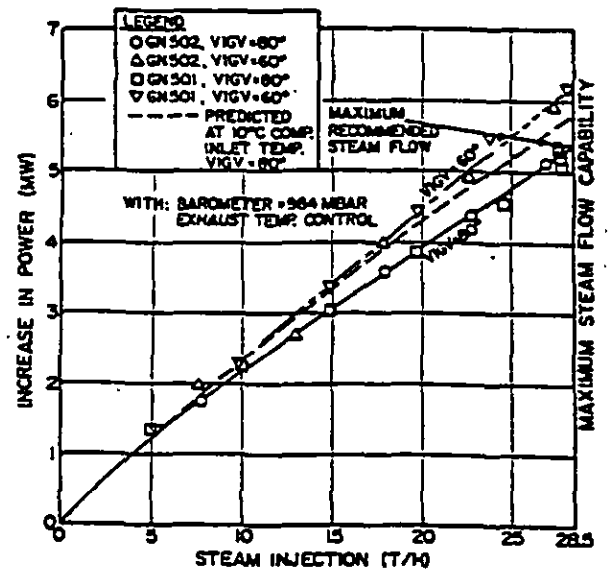

Figure 12. Measured gas turbine power increase with steam injection. (Burnham et. al, 1988)

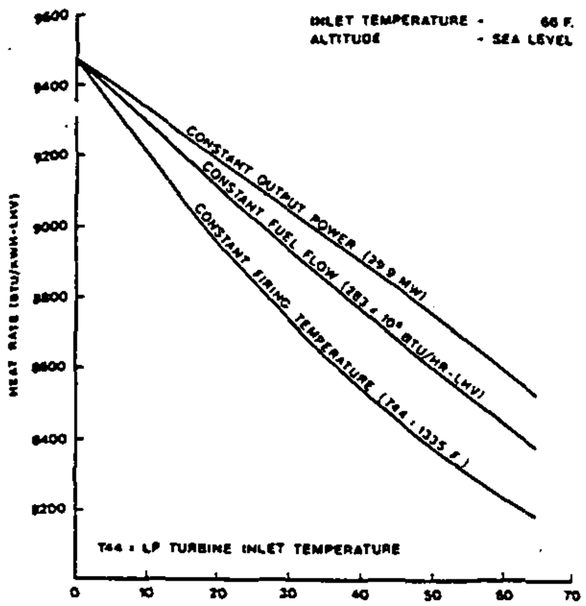

Figure 14. Gas turbine thermal efficiency with steam injection. (Little et. al, 1988).

Downloaded from http://asmedigitalcollection.asme.org/GT/proceedings-pdf/GT1994/78866/V004T10A020/2405145/v004t10a020-94-gt-435.pdf by guest on 28 January 2022

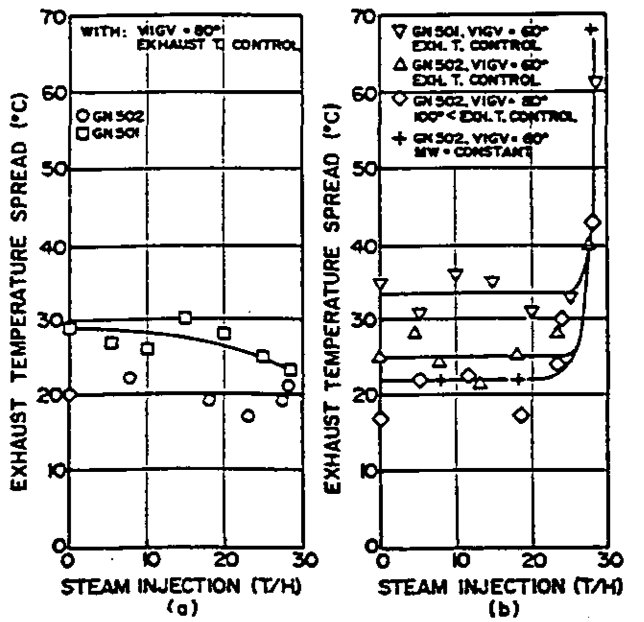

Figure 15. Exhaust temperature spread with steam injection. (Little et. al, 1987)

The above table indicates that close to 32, 000 PPH (14,546 Kg/Hr) of steam is injected into LM1600, 40,000 PPH (18,182 Kg/Hr) of steam into LM2500 and as much as 139,400 PPH (63,364 Kg/Hr) into LM5000 gas turbine. In all the three cycles described above, steam is injected into compressor discharge as well through a specially designed fuel nozzle, into the combustor. The LM5000 is unique among these STIG cycles in that steam is injected into the power turbine.

The steam is injected into the hollow LPT nozzle vanes and it escapes through the pores in the pressure surface and the trailing edge nozzle vanes. The increase in the power output and the increase in power output per pound of steam injected per second is tabulated below:

Model	Steam lb/sec	MW dry	MW STIG	MW Increment	MW/(lb/sec Steam)
LM1600	8.89	13.0	16.1	3.1	0.349
LM2500	11.10	21.5	26.3	4.8	0.432
LM5000	38.72	33.1	51.9	18.8	0.486

Table 2. Comparison of Various STIG Cycles

One of the important advantages of steam injection is that it reduces the level of oxides of nitrogen in the exhaust This is accomplished by the steam being injected in the combustor of the turbine. This reduces the oxygen content of the fuel-air mixture, thus increasing its heat capacity, which in turn reduces the temperature of the combustion zone, reducing the NOx formed. Under maximum steam flow rate conditions, NOx levels at exhaust is reduced to less than 7 ppm. The maximum steam flow rate for this case was around three (3) times the fuel flow rate.

As shown in Figure 16 [Burnham et. al., 1987], NOx has been reduced by close to 98 percent. On the other hand, the CO emissions increase with increase in steam injection flow rate. Near design operating conditions, an optimum acceptable level of emission of both NOx and CO can be obtained by controlling the steam injection flow rate.

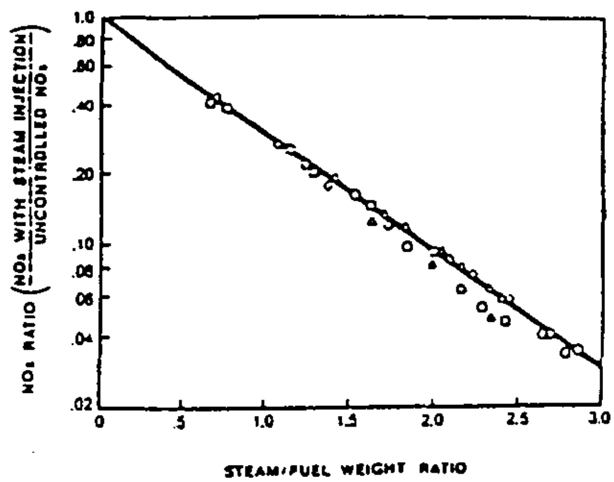

Figure 16: NO_X reduction ratio with steam injection.

Under simple cycle and low turbine inlet temperature, the amount of compressor work is almost twice that of the useful shaft power output obtainable from gas turbines. However as the firing temperature increase, this ratio decreases, indicating a reduced power consumption from the compressor. At current day firing temperatures of about 2300° F (1260°C), this ratio is almost one. As shown in Figure 17 [Rice, 1993], with steam injection and increased firing temperature this ratio becomes less than 1 to 0.5 indicating a jump in power output and efficiency possible from simple cycle machines.

3. <u>Humid Air Cycle</u>

The Humid Air Turbine (HAT) cycle is another promising power cycle. HAT utilizes a multiple shaft gas turbine in which the compressor is intercooled to reduce compressor power consumption and increase cycle efficiency. The hot water from the intercooler and the compressor discharge air which, after getting reheated by gas turbine flue gases, is mixed in a saturator, and the humid air combined with fuel are heated before expansion in a turbine. In another version of this power cycle, the gas turbine is fueled by a coal gasification process and the cycle is termed IGHAT. Studies have shown that the HAT cycle has been found to be efficient in heavy duty gas turbine operation producing more than 200 MW.

Downloaded from http://asmedigitalcollection.asme.org/GT/proceedings-pdf/GT1994/78866/V004T10A020/2405145/v004t10a020-94-gt-435.pdf by guest on 28 January 2022

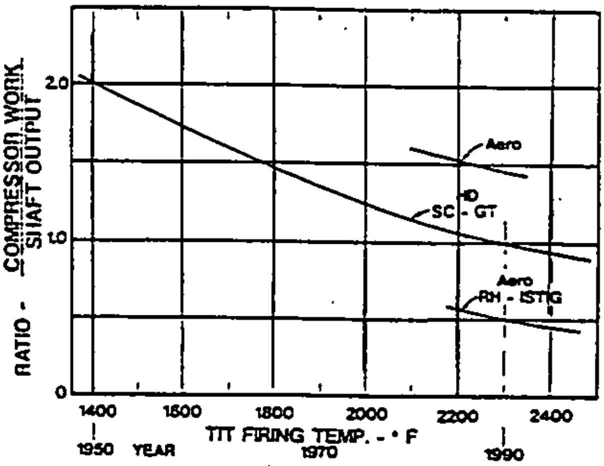

Figure 17. Ratio of compression work divided by shaft output versus TIT. (Rice, 1993)

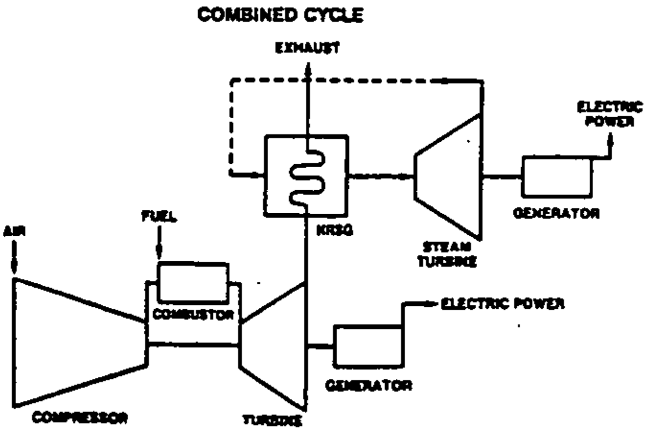

Figure 18. A typical combined cycle.

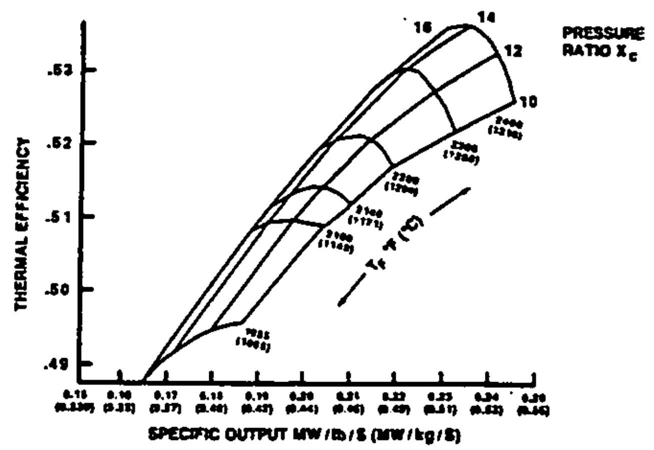

Figure 19. Combined cycle analysis.

THE RANKINE-BRAYTON CYCLE (Combined Cycle)

The combination of the gas turbine (Brayton Cycle) and the steam turbine (Rankine Cycle) is very attractive especially for the electric utilities and process industry where steam is being used. This cycle is often termed a *combined cycle* is as it uses both a gas turbine (Brayton cycle) and steam turbine(Rankine cycle). As shown in Figure 18 the hot exhaust gas from the turbine is used in a Heat Recovery Steam Generator (HRSG) or a boiler to produce steam, which is expanded in a steam turbine to produce additional power.

Figure 19 shows the results of a cycle analysis from a typical combined cycle. It can be easily seen that current day firing temperatures and pressure ratios, thermal efficiencies of combined cycle plant can exceed 50 percent. Thus, from a thermal efficiency point of view, a cogeneration plant operating in a combined cycle mode is more efficient than a simple cogeneration cycle. For a given plant size, economics however plays a key role in making a choice of an optimum cycle.

Combined cycle can be derived by adding a steam source and a steam turbine utilizing steam generated by gas turbine exhaust or by adding a gas turbine as a source of hot exhaust gas for steam production or for feed water heating in an existing steam power plant. Modern combined cycle power plants are of the former type. In many utilities where the steam turbine plants are over 30 years old, repowering has become an attractive option to boost thermal efficiency and to control emissions. Three feasible repowering approaches are shown in Figures 20-22 [Chase et. al. 1991]. Modern thermal power plants operating in a combined cycle mode are Integrated Coal Gasification Combined Cycle (IGCC) in which gas turbines are fired by coal gas. Combined cycles will become of increasing importance as coal has to be utilized by the power industry. There are several technologies that will emerge including the Integrated Gasification Combined Cycle, Pressurized Fluidized Bed Combustion (PFBC), and Externally Fired Cycles (EFCC). Advanced combined cycles using these technologies will attain efficiencies between 53 - 55 percent.

Based on the steam generation method, there are three basic types of combined cycle. These are:

1. Unfired heat recovery cycle
2. Supplemental fired heat recovery cycle.
3. Exhaust fired cycle.

In the unfired heat recovery cycle, power produced by steam turbine would be a maximum of a third of the total generating capacity of the combined cycle. In the supplemental fired heat recovery cycle the steam turbine power output would amount to a maximum of two third the plant capacity and in the exhaust fired cycle the total steam power production could be as much as 60 to 80 percent plant capacity. Figure 23 summarizes the

Downloaded from http://asmedigitalcollection.asme.org/GT/proceedings-pdf/GT1994/78866/V004T10A020/2405145/v004t10a020-94-gt-435.pdf by guest on 28 January 2022

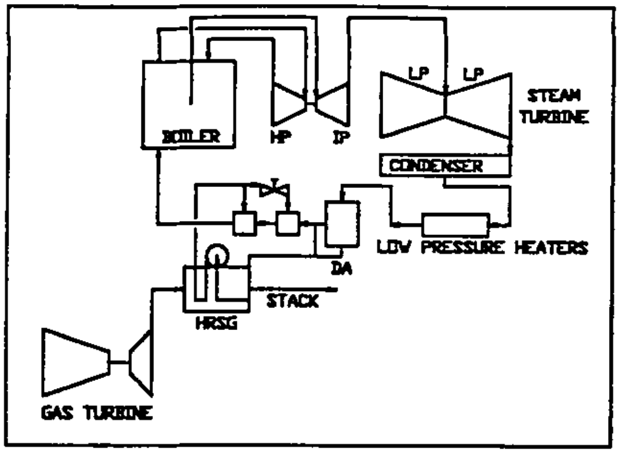

Figure 20. Feed water testing re-powering using HRSG (Chase et. al, 1991)

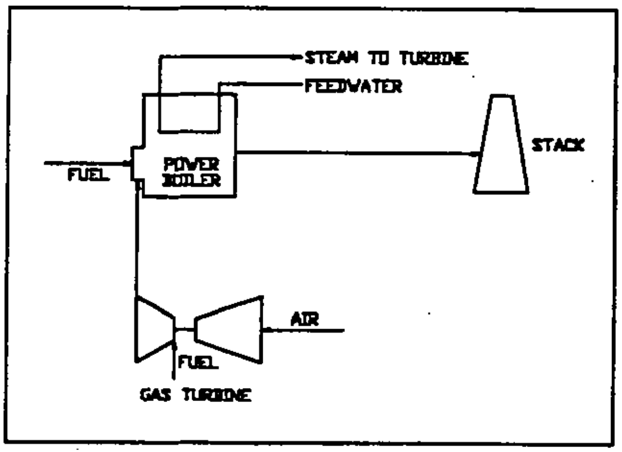

Figure 21. Feed water heater re-power using economizer. (Chase et. al, 1991)

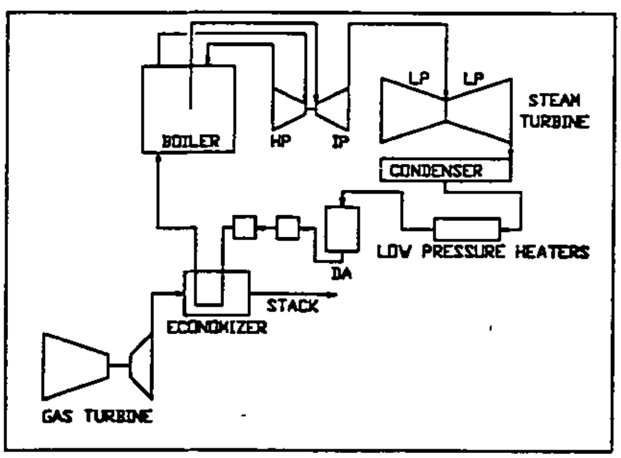

Figure 22. Boiler re-powering (Chase et. al, 1991)

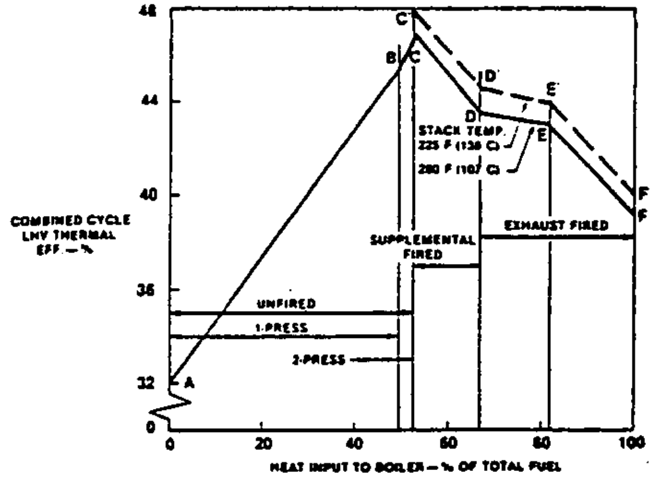

Figure 23. Combined power cycle performance (Tomlinson, et. al,1983)

performance of the above three cycles [Tomlinson, et. al. 1983]. The figure compares the combined cycle efficiency as a function of the fraction of the total heat input to the boiler. Point A represents a simple cycle efficiency. Operating zones of each type of cycle is clearly marked in the Figure. The point of maximum heat recovery 'C' is limited by the stack gas temperature, which is usually 280°F(138°C)

Ambient conditions affect both gas turbine and combined cycle performance. Figure 24[Tomlinson, 1983] depicts the variation in power, fuel consumption and process heat of a fired boiler combined cycle as a function of ambient air temperature.

The effect of variation of combined cycle heat rate as a function of exhaust gas temperature at various gas turbine specific power and gas turbine heat rate points are shown in Figure 25 [Tomlinson, 1983]. At a given gas turbine specific power, the combined cycle heat rate remains nearly constant as the exhaust gas temperature increases. Increase in specific work decreases the combined cycle heat rate and thus increases combined cycle thermal efficiency.

The effect of exhaust gas temperature on specific steam system output is summarized in Figure 26 [Tomlinson, 1983].

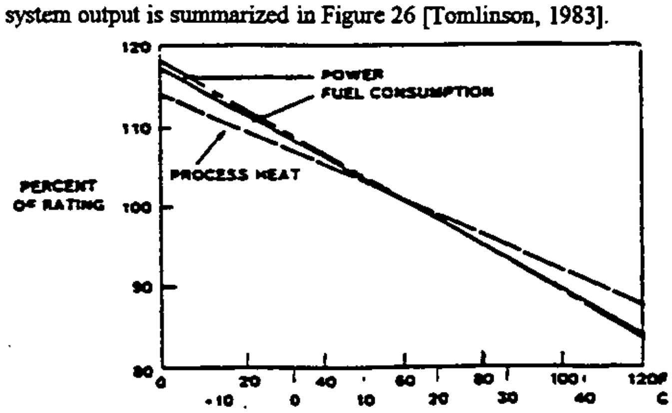

Figure 24. Fired boiler power and heat system performance variation with ambient air temperature. (Tomlinson, et. al, 1983)

Downloaded from http://asmedigitalcollection.asme.org/GT/proceedings-pdf/GT1994/78866/V004T10A020/2405145/v004t10a020-94-gt-435.pdf by guest on 28 January 2022

SUMMARY

This paper has presented various gas turbine cycles and parametric analyses showing the interrelationships of firing temperature and pressure ratio. Approaches for enchancing power and efficiency are listed including inlet cooling, steam injection and HAT cycles. The power generation technology of choice for the next few decades will be various modes of combined cycle plants.

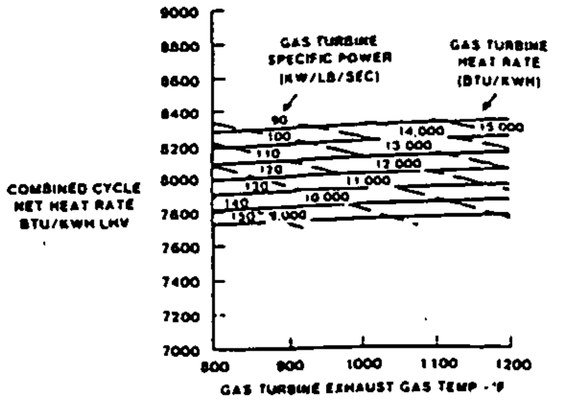

Figure 25. Gas turbine performance effect on exhaust fired power cycle performance. (Tomlinson, et. al, 1983)

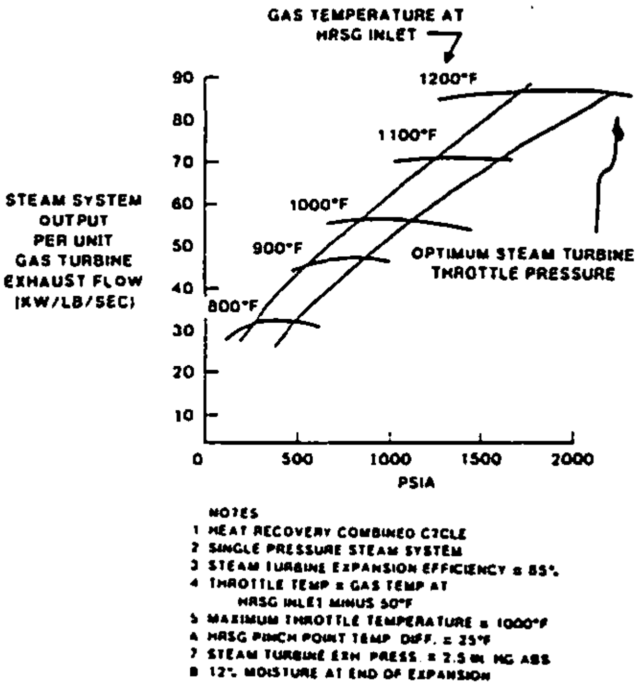

Figure 26. Exhaust gas temperature effect on optium heat recovery combined cycle steam pressure. (Tomlinson, et. al, 1983)

REFERENCES

Boyce, M.P., 1987 Gas Turbine Engineering Handbook, Gulf Publishing.

Boyce M.P. and Chen, M., 1978, "Optimization of Various Gas Turbine Cycles," *Proceedings of the 3rd Texas A&M Turbomachinery. Symposium,* College Station, Texas.

Burnham J.B., Giuiliani M.H. and Moeller D.J., 1987, "Development, Installation and Operating Results of a Steam Injection System (STIG™) in a General Electric LM5000 Gas Generator," *Journal of Gas Turbines and Power,* pp 257-262.

Collins, S., 1993, "Advanced Gas Turbine Cycles - Center piece of Today's Power Cycles," *Power.*

Chase D.L., Kovacik, J.M., and Stoll, H.G., 1991 "The Economics of Repowering Steam Turbines, GE Turbine State-Of-the-Art Technology Seminar," *G.E. Turbine State-Of-the-Art Technology Seminar.*

Fogg H.E., 1991, "GE Aeroderivative Gas Turbines - Design and Operating Features," *GE Turbine State-Of-the-Art Technology Seminar.*

Hopkins J.E., 1991, "GE Gas Turbine Performance Characteristics," GE Turbine State-Of-the-Art Technology Seminar,

Lakshminarasimha, A.N., Boyce, M.P., Meher-Homji, C.B., 1994, "Modelling and Analysis of Gas Turbine Performance Deterioration," Transactions of the ASME Journal of Engineering for Gas Turbines and Power, Vol 116, No. 1, pp 46-52.

Little, D.A. and Rives J.P., 1988, "Steam Injection of Frame 5 Gas Turbines for Power Augmentation in Cogeneration Service," ASME 88-GT-51.

Meher-Homji, C.B., 1990, "Gas Turbine Axial Compressor Fouling: A Unified Treatment of its Effects, Detection and Control," 4th ASME Cogen Turbo Conference, Also in International Journal of Turbo and Jet Engines, Vol.9, pp. 311-334, 1992.

Meher-Homji, C.B., Mani. G., 1983 "Combustion Gas Turbine Power Enhancement by Refrigeration of Inlet Air," *Proceedings of the Industrial Energy Conservation Technology Conference,* Houston, TX.,

Tomlinson, L.O and Lee D.T., 1983, "Combined Cycles," Sawyer's Gas Turbine Handbook, 3rd Edition.

Rice, I.G., 1983, "Steam-injected Gas Turbine Analysis: Part I – Steam Rates" ASME 93-GT-132.

(Little et. al, 1988).

Downloaded from http://asmedigitalcollection.asme.org/GT/proceedings-pdf/GT1994/78866/V004T10A020/2405145/v004t10a020-94-gt-435.pdf by guest on 28 January 2022