Degradation Effects on Combined Cycle Power Plant Performance—Part I: Gas Turbine Cycle Component Degradation Effects

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This paper describes the effects of degradation of the main gas path components of the gas turbine topping cycle on the combined cycle gas turbine (CCGT) plant performance. First, the component degradation effects on the gas turbine performance as an independent unit are examined. It is then shown how this degradation is reflected on a steam turbine plant of the CCGT and on the complete combined cycle plant. TURBOMATCH, the gas turbine performance code of Cranfield University, was used to predict the effects of degraded gas path components of the gas turbine have on its performance as a whole plant. To simulate the steam (bottoming) cycle, another Fortran code was developed. Both codes were used together to form a complete software system that can predict the CCGT plant design point, off-design, and deteriorated (due to component degradation) performances. The results show that the overall output is very sensitive to many types of degradation, especially in the turbine of the gas turbine. Also shown is the effect on gas turbine exhaust conditions and how this affects the steam cycle.
[DOI: 10.1115/1.1519271]

Introduction

Combined-cycle power plants achieve very good economic performance. This economic performance is dependent on several factors; prominent among them are first cost, high efficiency, low maintenance cost, and good availability.

Good availability is partly influenced by downtime. It is possible to reduce downtime by having detailed knowledge prior to any inspection of the faulty components of a gas turbine. It is, therefore, very important to know how the CCGT power station behaves if different components of the plant are degraded. It is important to establish how various faults manifest themselves in terms of variations of measurable parameters such as temperatures and pressures. Furthermore, a more detailed knowledge of the engine will allow the user to take some of the maintenance action when it is necessary. This will also help reduce operating costs.

This paper describes how common faults, affect CCGT plant performance. In this paper, the degradation of the gas turbine components is examined, while the other bottoming plant (steam turbine plant) were kept at its original design point conditions. Much has been written on the effects of gas turbine degradation, but there seems to be a scarcity of material on the effects of degradation on combined cycle performance.

Gas Turbine Performance Deterioration

The main gas path components of the gas turbine, namely compressor and turbine, will degrade with engine use, [1–5], which then results into engine performance deterioration.

The focus of this paper is the simple-single shaft gas turbine coupled with a single-pressure HRSG bottoming cycle, of which a schematic diagram shown in Fig. 1. More complex steam cycles are currently in service, but the results of the present work will be a useful guide.

Contributed by the International Gas Turbine Institute (IGTI) of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS for publication in the ASME JOURNAL OF ENGINEERING FOR GAS TURBINES AND POWER. Paper presented at the International Gas Turbine and Aeroengine Congress and Exhibition, New Orleans, LA, June 4–7, 2001; Paper 2001-GT-0388. Manuscript received by IGTI, December 2000, final revision, March 2001. Associate Editor: R. Natole.

The gas turbine of this unfired cycle is an engine of conventional design. This engine has the following (design point) specifications:

inlet mass flow	=	408.66 kg/sec
compressor pressure ratio	=	15.2
turbine entry temperature	=	1697.80 K
exhaust mass flow	=	419.4 kg/sec
exhaust temperature	=	871.24 K
power	=	165.93 MW
Thermal efficiency	=	35.57%

In order to cover the widest range of faults that may occur in the gas turbine, it was assumed that each component may degrade separately, and then all components were assumed to degrade together, to establish the nature of the faults and to assess if they are additive or not. The faults investigated were the following:

1. compressor fouling,
2. compressor erosion,
3. compressor isentropic efficiency degradation,
4. turbine fouling,
5. turbine erosion,
6. turbine isentropic efficiency degradation,
7. compressor and turbine fouling,
8. compressor and turbine erosion, and
9. compressor and turbine isentropic efficiency degradation.

For a clearer and complete discussion of the terms (fouling and erosion) mentioned above and for their mechanism, the reader may wish to examine other work, [1,2].

Because the combustion system is not likely to be a direct cause of gas turbine performance deterioration, [1], it was assumed not to degrade for the following reasons:

1. The faults in combustion chamber that affect overall performance are rare in comparison to those faults that may occur in the compressor and turbine.
2. Any malfunction in the combustion chamber would mean increased emissions, which is not allowed by environmental laws in many places.

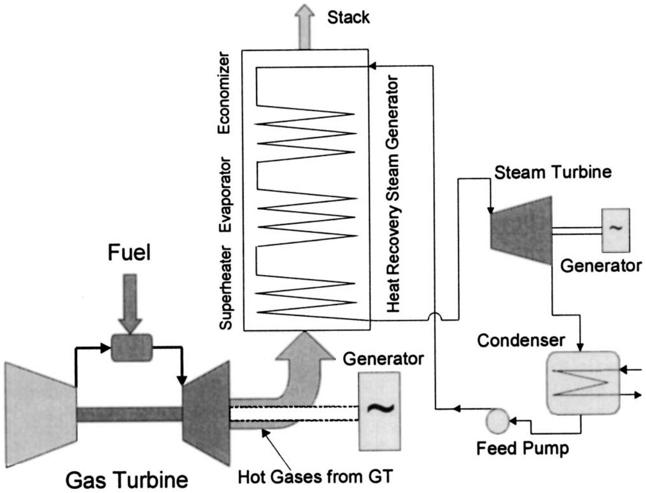

Fig. 1 Schematic diagram of a single pressure CCGT power plant

Faults Representation

In the current study, in order to predict the gas turbine degradation effects, faults implanted on each component of the gas turbine are defined in the following way:

Fouling. Compressor or turbine fouling is represented by reduced flow capacity at the inlet of the component plus a reduction in the component isentropic efficiency. By doing so, it is assumed that there is a blockage in the inlet area of the component due to fouling along with a decrease in the component isentropic efficiency due to surface roughness, for example.

Erosion. Compressor erosion is represented by a lower inlet mass flow capacity and a reduction in compressor isentropic efficiency. On the other hand, turbine erosion is represented by an increased flow capacity plus a reduction in the turbine isentropic efficiency, [2].

These two phenomena are represented by changing the so-called nondimensional mass flow Eq. (1) of the component maps (Table 1).

$$\frac{\dot{W}\sqrt{T_i}}{PA}=\text{constant} \tag{1}$$

Component Efficiency Degradation. This is modeled by reducing the component isentropic efficiency of the appropriate map and keeping all other parameters at their design point (DP) levels. In this case, it was assumed that the component isentropic efficiency may decrease from its DP value due to any reason, such as blade tip rubs or FOD.

To assist the reader observing these faults in a more clearer and readable way, these faults are summarized in Table 1.

Table 1 Representation of component degradation

Fault	Represented by	Range
Compressor fouling	drop in Γ	0.0–(−5.0%)
	drop in η_C	0.0–(−2.5%)
Compressor erosion	drop in Γ	0.0–(−5.0%)
	drop in η_C	0.0–(−2.5%)
Turbine fouling	drop in Γ	0.0–(−5.0%)
	drop in η_T	0.0–(−2.5%)
Turbine erosion	rise in Γ	0.0–(+5.0%)
	drop in η_T	0.0–(−2.5%)
FOD	drop in η_C and η_T	0.0–(−5.0%)

Table 2 Component isentropic efficiency variation with degradation

Physical Fault	Nondimensional Mass Flow Change (A)	Isentropic Efficiency Change (B)	Ratio A:B
Compressor fouling	$\Gamma_C\downarrow$	$\eta_C\downarrow$	~1:0.5
Compressor erosion	$\Gamma_C\downarrow$	$\eta_C\downarrow$	~1:0.5
Compressor corrosion	$\Gamma_C\downarrow$	$\eta_C\downarrow$	~1:0.5
Turbine fouling	$\Gamma_T\downarrow$	$\eta_T\downarrow$	~1:0.5
Turbine erosion	$\Gamma_T\uparrow$	$\eta_T\downarrow$	~1:0.5
Turbine corrosion	$\Gamma_T\downarrow$	$\eta_T\downarrow$	~1:0.5
Foreign object damage	$\Gamma_{C/T}\downarrow$	$\eta_{C/T}\downarrow$	~1:2.0
Thermal distortion	$\Gamma_T\uparrow\downarrow$	$\eta_{C/T}\downarrow$	~1:2.0
Blade rubbing	$\Gamma_C\downarrow$ & $\Gamma_T\uparrow$	$\eta_{C/T}\downarrow$	~1:2.0

Gas Turbine Degradation Simulation

Before starting any degradation simulations it is necessary to establish the base line (design point performance) of the plant. This base line performance point is represented by (0.0) value on all deterioration graphs shown below. Once the design point has been identified, then the magnitude of faults that represents a physical fault of the component in consideration, mentioned in previous section, to be implanted on each component has to be established.

Unfortunately, although there is a lot of work published on the subject of gas turbine performance deterioration, [1–4], the applied degradation magnitude to each component, when simulating gas turbines deterioration performance, in most cases is either arbitrary or based on some published experimental results. Therefore, in present study the values mentioned by [1] and [5] were taken as a guidelines form which the implanted faults were estimated. Table 2, [5], with some modifications, shows a summary of how component isentropic efficiency changes vary with degradation. These values were applied in all calculations to the appropriate components.

Throughout this work it was assumed that there was no equipment washing or any type of maintenance taken on the gas turbine until the deterioration reaches 5% from the original design point performance.

Gas Turbine Degradation Simulation Results

The most important gas turbine deterioration simulation results are briefly presented graphically in Figs. 2 through 5. It is worth noting here that the term (compressor degradation) used in all figures means either compressor fouling or erosion, or both.

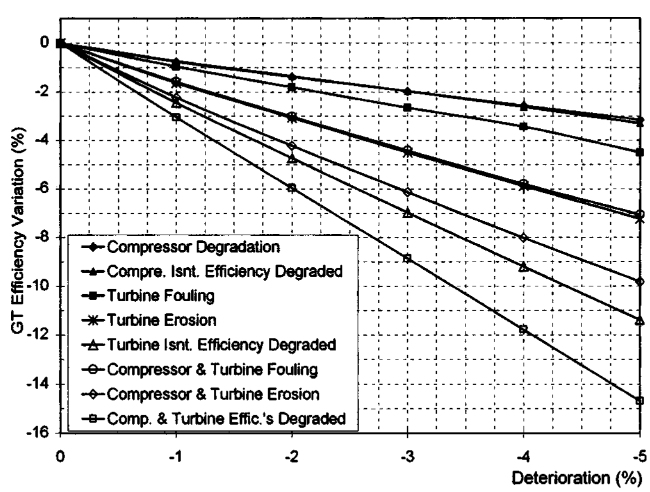

Fig. 2 GT efficiency variation with component deterioration

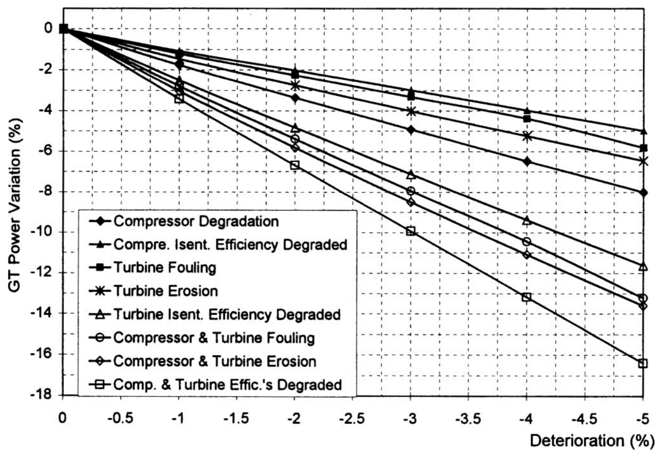

Fig. 3 GT power variation with component deterioration

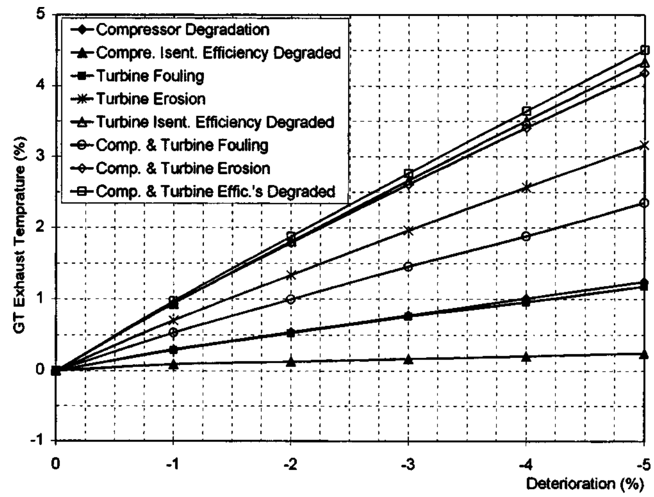

Fig. 5 GT exhaust temperature variation with component deterioration

Figure 2 shows the degradation effects on gas turbine efficiency. It can be observed that the compressor degradation whether it be fouling and/or erosion gave the lowest deterioration in gas turbine overall efficiency. The value of efficiency deterioration was about -3.1% with compressor degradation by 5%. This implies that, gas turbine overall efficiency is less sensitive to compressor degradation than to the degradation of other components, as it will be shown later.

On the other hand, as expected, combined compressor and turbine isentropic efficiencies degradation gave the most severe effect in overall gas turbine efficiency. As it can be seen, the plant efficiency deterioration was approximately -14.6% (from original design point) with only 5% degradation.

Although the expectation was to see more reduction in plant thermal efficiency when both compressor and turbine are either fouled or eroded together as this fault implies a reduction in both component isentropic efficiency in addition to fouling or erosion, the results obtained was not so. This can be explained as follows,

1. It appears that the ratio of isentropic efficiency degradation to the component fouling or erosion applied in each case when simulating component fouling or erosion is 0.5:1.0 (see Table 1). This means that the value of component isentropic efficiency degradation along with 5% of fouling or erosion is 2.5%. Whereas in the situation of simulating the same case (5% degradation) of only component isentropic efficiency degradation the value used was 5%.

2. Secondly, In the case of both components fouling, the increased pressure ratio through the turbine caused by fouling increased the power output of the turbine. This led to a higher plant power output and a higher thermal efficiency than the case where only the component isentropic efficiency was degraded.

3. Similarly the combination of decreased compressor mass flow with an increased turbine flow capacity, due to erosion by 5%, led to a higher (about 2.8%) reduction in the plant's overall efficiency in comparison to the case mentioned above (fouling in both components). This is exactly the opposite of what was mentioned earlier. I.e., the decreased pressure ratio through the turbine resulted in a lower power output of the turbine, and hence a reduced overall power output of the engine which then reflected on the engine's overall efficiency.

4. Finally, this also applies in the case where only the component isentropic efficiencies are reduced, the mass flow capacity is approximately at its DP value and hence the same pressure ratio approximately. The obvious result of this shortfall in component efficiencies will be a reduction in the overall power output and efficiency of the plant.

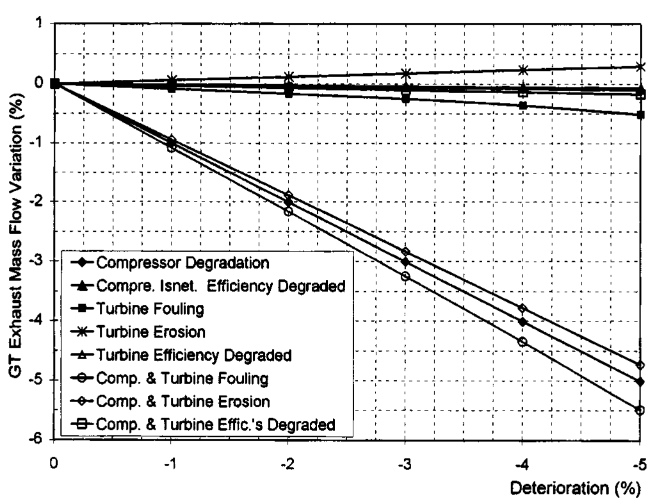

Fig. 4 GT exhaust mass flow variation with component deterioration

Figure 3 below shows different component degradation effects have on gas turbine power output. The compressor inlet mass flow is reduced by 5% due to degradation. This resulted in a reduction of about 8% in engine power output. This is similar to that observed elsewhere, [1], which was 7%. It is worth mentioning here that, in this case of Diakunchak [1] the compressor isentropic efficiency at fouling of 5% was 1.8%, while in the present case it was assumed to be 2.5%. This led to a reduction in engine power output, as stated above, by 1% higher than the one quoted, [1].

Figures 2 and 3 show that the degradation in turbine isentropic efficiency has a higher effect on engine efficiency and power than that of compressor isentropic efficiency. A compressor isentropic efficiency degradation of 5% led to a reduction of about 4.9% in engine power with a corresponding reduction of about 3.3% in engine efficiency. The same amount of degradation in turbine isentropic efficiency (5%) resulted in about 11.6% fall in engine power, and about 11.3% reduction in engine efficiency.

This is due to the fact that, the turbine has two tasks, one to drive the compressor, and the second to produce the engine net power output, and hence the turbine degradation reflection is higher because of the much larger power involved.

A close look at Figs. 2 and 3 will show that, the degradation of 1% in turbine isentropic efficiency has led to approximately 2.5%

reduction in both gas turbine power output and efficiency. These values are in agreement with those described elsewhere, [1].

In the case of a simple Brayton cycle operating in isolation the change in engine exhaust gas properties due to component degradation could be neglected as the gases are usually discharged to the stack. However, in the case of combined cycle, where the bottoming (steam) cycle is dependent on the exhaust gases of the topping (Brayton) cycle, it is very important to know the exhaust gas properties and how they change with different components degradation because there is a strong relationship between gas turbine exhaust properties and steam turbine output in a combined cycle.

The effects of different gas turbine component degradation on engine exhaust mass flow and temperature are illustrated in Figs. 4 and 5, respectively. As Fig. 4 shows, it can be seen clearly that the exhaust mass flow has increased slightly (about 0.3%) with 5% of turbine erosion. On the other hand, in the case of compressor and turbine erosion combination, the process is solely a function of compressor erosion, as the turbine mass flow would follow the compressor inlet mass flow. However, as this figure shows, the exhaust mass flow due to compressor and turbine erosion combination by 5% is higher than that of the compressor degradation (erosion) alone by about 0.28%. This is because of the increased mass flow at the turbine inlet due to erosion. It is also clear from this figure that the compressor and turbine fouling combination has the highest severe effect on exhaust mass flow. The effect of any of the components isentropic efficiencies degradation on gas turbine exhaust mass flow, as can be seen from this figure as well, is approximately negligible.

Figure 5 below shows the variation of gas turbine engine exhaust temperature with different components degradation. Contrary to the effects of components isentropic efficiencies degradation on gas turbine exhaust mass flow, it can be noted that the combination of compressor and turbine isentropic efficiency degradation has the highest effect on exhaust temperature.

As this figure shows, the increase in exhaust temperature was about 4.5%, from original DP, with 5% degradation in both compressor and turbine isentropic efficiencies (notice the small share of the compressor isentropic efficiency degradation). This is due to the fact that for a constant TET, the reduced turbine isentropic efficiency would mean a lower power output, and hence a higher amount of heat content rejected at the exhaust gases, and thus a higher exhaust temperature. This could be favorable to the steam turbine plant, as it would lead to an increase in the steam production, and hence increased steam turbine power output. However, the decreased gas turbine power due to this fault may lead, depending on steam turbine power output production, to an outcome reduction in CCGT power output.

Steam Turbine Performance Deterioration

Given that the focus of the analysis described in this paper is the effect of gas turbine component degradation on the combined cycle, all steam turbine components were assumed to be operating in their clean or new condition, regardless of the situation of the gas turbine plant. The effects of the deteriorated topping (gas turbine) cycle on the steam cycle is discussed here. A more detailed discussion of the effects of degraded components of steam turbine plant have on its and on CCGT power plant performance, as the bottoming cycle of CCGT plant, can be found in [6].

As Fig. 1 shows, the steam section of the plant consists of a single pressure HRSG, one steam turbine, generator, condenser, and a feed pump. This plant has the following specifications at design point conditions:

live steam pressure	= 65.4 bar
live steam temperature	= 537.8°C
steam mass flow	= 67 kg/sec
steam turbine isentropic efficiency	= 89.48%
superheater surface area	= 8424.8 m^2
evaporator surface area	= 29315.6 m^2
economiser surface area	= 38004.1 m^2
condenser surface area	= 3942.9 m^2
HRSG efficiency	= 81.11%
steam turbine plant power output	= 76454.14 kW
steam turbine plant efficiency	= 33.97%

Steam Cycle Performance Simulation

First, the design point performance of the steam turbine plant must be established. Here the components of the steam (bottoming) plant were kept in their original, clean, condition. The exhaust gases, at different deterioration conditions of gas turbine, received from the upper cycle are then passed through bottoming cycle, and the results were observed. By doing so, it was possible to investigate how the bottoming cycle behaves with different types of gas turbine degradations. This is done (as already mentioned) by utilizing a Fortran code produced, at Cranfield University, especially for this purpose (see [6]).

Steam Turbine Operating Performance

As with gas turbine simulation, the most important steam turbine deterioration simulation results are represented graphically in Figs. 6 through 8. It is worth reminding the reader here that the point (0.0) on (Y) axes of all graphs means the design point value.

Figure 6 shows the variation of steam production in the HRSG with different types of gas turbine component degradations. It is clear that all types of gas turbine degradation have led to an increase in the steam mass flow, except compressor degradation.

Although, as Fig. 5 shows, there was an increase in exhaust temperature due to compressor degradation which is expected to result in an increase in steam production in the HRSG, the decrease in gas turbine exhaust mass flow caused by the same fault, as Fig. 4 shows, was predominant, and hence the out come was a decrease in steam mass flow. Another observation from this figure is that the steam production in the HRSG was more sensitive to the faults caused by GT turbine, rather than to those caused by the compressor. This is because (as this figure shows) GT turbine erosion and isentropic efficiency degradations, as individual faults, resulted in the highest amount of steam production. By comparing this with Figs. 4 and 5, it will be observed that the

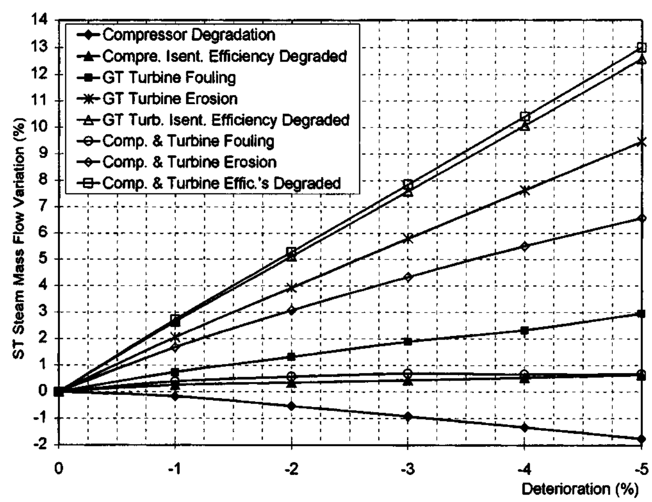

Fig. 6 ST steam mass flow variation with gas turbine component deterioration

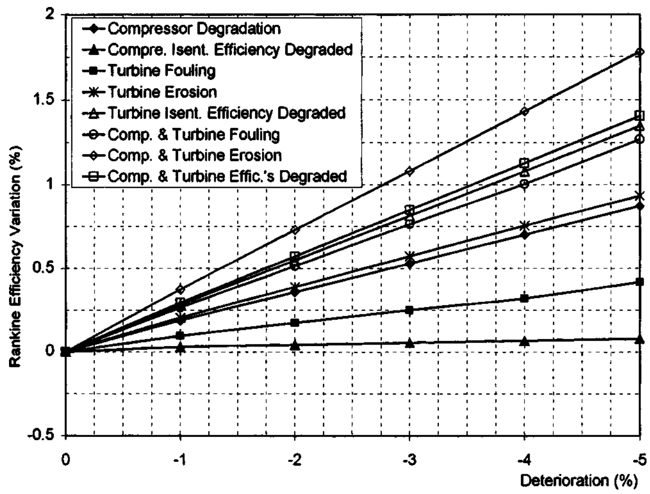

Fig. 7 Rankine efficiency variation with gas turbine component deterioration

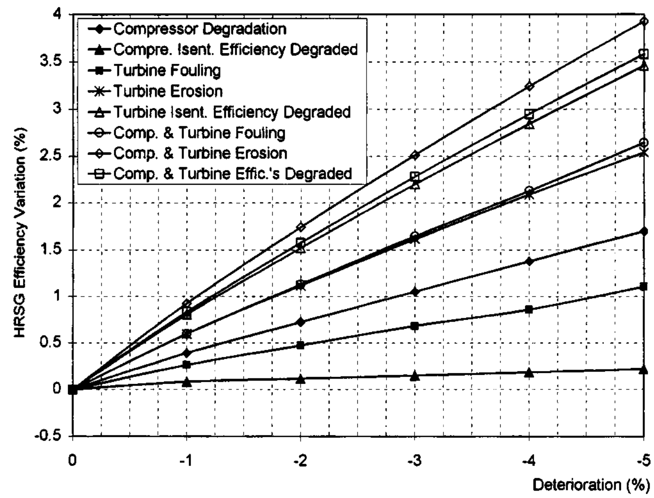

Fig. 9 HRSG efficiency variation with gas turbine component deterioration

exhaust temperature (Fig. 5) was the main contributor to this increase of steam mass flow production, as exhaust mass flow was almost constant with GT turbine erosion and isentropic efficiency degradation (see Fig. 4).

Finally, the increase in steam mass production due to compressor and turbine isentropic efficiencies degradation of 5% together was about 13.0 percent. As Fig. 6 shows, this value is approximately 0.5% above that one caused by the turbine isentropic efficiency degradation. Therefore, this 0.5% increase is mainly the contribution of compressor isentropic efficiency degradation.

The next important performance parameter to discuss here is the steam turbine plant (Rankine) efficiency variation with gas turbine degradation, which is illustrated in Fig. 7 below. The thermal efficiency definition of steam turbine (bottoming) plant is given by

$$\eta_R = \frac{W_{SC}}{Q_{\mathrm{HRSG}}}. \tag{2}$$

This equation shows that, the steam turbine cycle efficiency is a function of steam turbine net power output and the heat transferred in the HRSG. Now by looking at Fig. 9 it will be seen that all types of GT degradations resulted into an increase in the HRSG efficiency. This clearly explains the behavior trend of the graphs representing the steam plant efficiency that appeared on Fig. 7.

Another useful observation is made by comparing Figs. 4 and 5 with Fig. 9. The GT exhaust temperature has a larger effect on steam cycle efficiency, (which by itself is a function of HRSG efficiency), over the exhaust mass flow.

Steam turbine power is a function of steam mass flow and its enthalpy. By comparing Fig. 8 with Fig. 6 it can be noticed that the steam turbine power is more or less following the mass flow behavior. The steam production in the HRSG is more sensitive to the faults of the gas turbine turbine, rather than those of the compressor. Therefore, since steam turbine power is a function of steam mass flow, the steam turbine power is more sensitive to the faults in the GT turbine, rather than to those in the compressor.

Figure 8 shows how steam turbine power varies with different types of gas turbine degradation. The higher exhaust temperature caused by degradation increased the steam turbine power output. As shown in Fig. 8, the maximum effect of gas turbine degradation on steam turbine power output was in the case of compressor and turbine efficiencies degradation. The increase in steam turbine power output was as high as 13.3% with compressor and turbine efficiencies degraded by 5%.

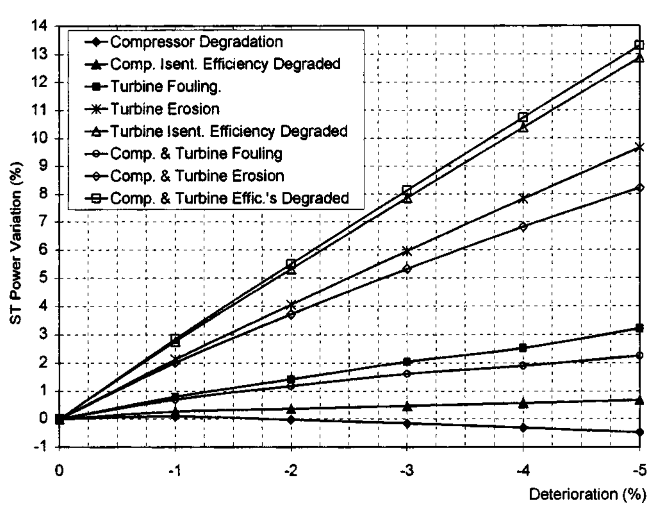

Fig. 8 ST power variation with gas turbine component deterioration

Combined Cycle Degradation Results

A FORTRAN code was specially developed to carry out the degraded performance analysis of the bottoming cycle at design and off design conditions. Since current study deals with a single pressure CCGT plant, only heat transfer paths shown by Fig. 10 (in addition to steam turbine section) were dealt with. The most important performance deterioration simulation results of combined cycle are shown in Figs. 11, 12, and 13. As Eq. (3) below shows, the combined cycle efficiency is a function of gas turbine cycle efficiency, HRSG efficiency, and steam cycle efficiency.

$$\eta_{CC} = \eta_{GT} + (1 - \eta_{GT}) \cdot \eta_{\mathrm{HRSG}} \cdot \eta_{SC} \tag{3}$$

Even though, as Figs. 7 and 9 shows, there was an increase in steam cycle and HRSG efficiencies with some (same) cases of GT component degradation (e.g., GT turbine erosion), still the effect of decreased GT efficiency (see Fig. 2) has a higher effect on CCGT efficiency. This is shown in Fig. 11. The combined cycle efficiency has fallen with all types of GT component degradations, even with steam turbine plant components at their original (DP) conditions. The GT efficiency has a predominant effect on CCGT power plant efficiency over steam cycle and HRSG efficiencies.

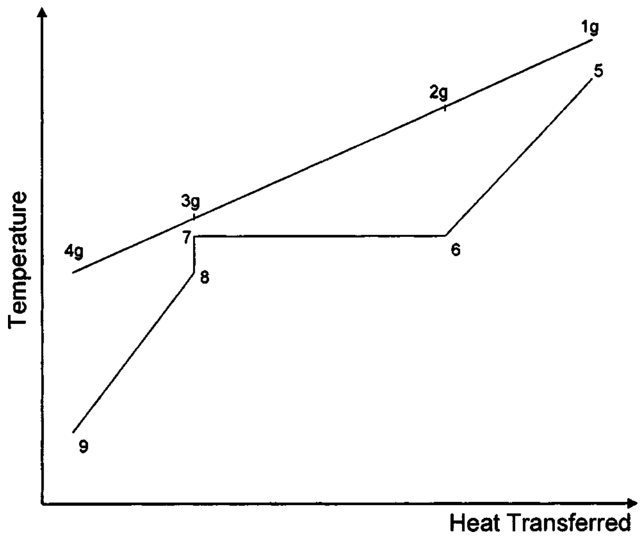

Fig. 10 Heat transfer diagram for a single pressure CCGT power plant

Figure 11 also shows that the CCGT efficiency is more sensitive to the faults that caused by GT turbine, rather than to those caused by the compressor. The GT turbine isentropic efficiency degradation (as an individual fault) gave the highest value in CCGT efficiency reduction. The efficiency deterioration was about 3.5% with 5% degradation in GT turbine isentropic efficiency. When both isentropic efficiencies of GT turbine and compressor are degraded together by 5%, the reduction in CCGT efficiency was about 5.0%.

Given that, typically, the gas turbine output is approximately two thirds of the total output of an unfired CCGT, it is expected to see a stronger influence on CCGT performance arising from changes in the gas turbine than in the steam cycle. Comparing Fig. 12 with Figs. 3 and 8 it may be noticed that the CCGT power output is more or less following the behavior of the gas turbine's power output. This shows, in addition to the conclusion withdrawn on CCGT efficiency above, that the behavior of CCGT plant performance is affected more by gas turbine conditions than by steam turbine conditions. As this figure also shows, although only gas turbine performance was deteriorated while steam turbine was kept at its original DP conditions, the net outcome was a reduction in CCGT plant power output. This reduction was at its

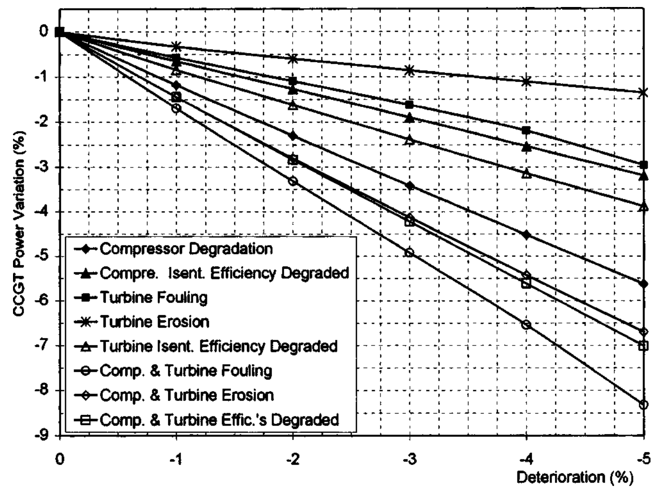

Fig. 12 CCGT power variation with gas turbine component deterioration

highest with the combination of compressor and GT turbine fouling. This was approximately 8.3% with both components fouling together by 5%.

Another important CCGT performance parameter is the stack temperature. This is mainly is a measure of the amount of gas turbine exhaust heat utilisation by the bottoming cycle. By definition, HRSG efficiency is a function of stack temperature and exhaust inlet temperature for a given ambient temperature, Eq. (4).

$$\eta_{\mathrm{HRSG}}=\frac{T_{1g}-T_{4g}}{T_{1g}-T_{\mathrm{amb}}} \tag{4[1]}$$

This equation shows that, for a given inlet exhaust and ambient temperatures, the HRSG efficiency increases with decreased stack temperature (T_{4g}) and vice versa. This can be clearly perceived by comparing Figs. 9 and 13. Now by definition as Eq. (3) shows, CCGT efficiency is a function of HRSG efficiency. Therefore, as Fig. 13 shows, the stack temperature was at its lowest value (about−8.9%) with compressor and turbine erosion by 5%. This reduction in stack temperature, as Fig. 9 shows, was reflected on the HRSG efficiency.

[1]See Fig. 10 for the definition of the notations used in this equation.

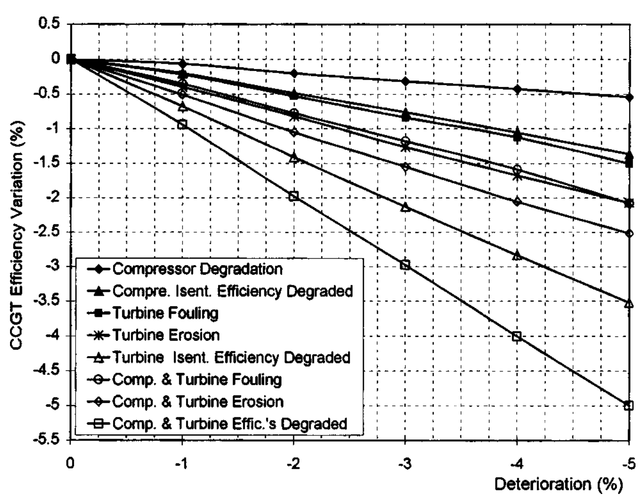

Fig. 11 CCGT efficiency variation with gas turbine component deterioration

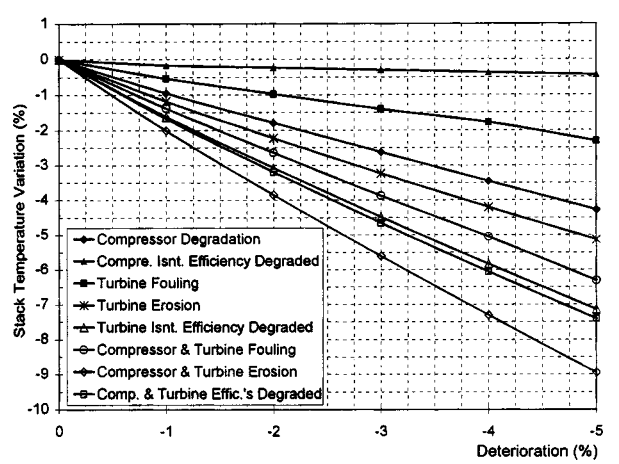

Fig. 13 Stack temperature variation with gas turbine component deterioration

Conclusions

The results obtained from gas turbine simulation Software at Cranfield University were very similar, in fact in many cases are exact, to those quoted elsewhere, [1].

As an individual fault, the GT turbine isentropic efficiency degradation has the most severe effect on gas turbine power and efficiency.

Gas turbine exhaust mass flow is merely a function of flow capacity through the engine; i.e., decreases with fouling, and increases with erosion. In addition, the effect of any of the components isentropic efficiencies degradation on gas turbine exhaust mass flow is almost negligible.

Steam turbine cycle steam mass flow, and hence steam turbine power output are more sensitive to the faults that caused by GT turbine, rather than to those caused by GT compressor.

The GT exhaust temperature has a predominant effect on steam cycle efficiency over the GT exhaust mass flow.

Among the three interrelated CCGT efficiency parameters that shown in Eq. (3), the gas turbine turbine efficiency was the most important parameter on CCGT efficiency.

The behavior of CCGT plant performance is more affected by gas turbine cycle conditions than by steam turbine cycle conditions.

Nomenclature

A = area
CCGT = combined cycle gas turbine plant
DP = design point
FOD = foreign object damage
GT = gas turbine
HRSG = heat recovery steam generator
OD = off-design condition
P = Pressure
Q = Heat Transfer
T = Temperature
TET = turbine entry temperature
$\dot{w}$ = Inlet mass flow
W = power
Γ = nondimensional mass flow
η = efficiency

Subscripts

amb = ambient
c = compressor
CC = combined Cycle
C/T = compressor and/or turbine
g = gas
HRSG = heat recovery steam generator
i = inlet
R = Rankine cycle
SC = steam cycle
T = turbine

References

[1] Diakunchak, I. S., 1992, "Performance Deterioration in Industrial Gas Turbines," ASME J. Eng. Gas Turbines Power, **114**(2).

[2] Lakshminarasimha, A. N., Boyce, M. P., and Meher-Homji, C. B., 1994, "Modelling and Analysis of Gas Turbine Performance Deterioration," ASME J. Eng. Gas Turbines Power, **116**(1).

[3] Tabakoff, W., 1986, "Compressor Erosion and Performance Deterioration," AIAA/ASME 4th Joint Fluid Mechanics, Plasma Dynamics, and Laser Conference, Atlanta, GA, May 12–14.

[4] Tabakoff, W., Lakshminarasimha, A. N., and Pasin, M., 1990, "Simulation of Compressor Performance Deterioration Due to Erosion," ASME J. Eng. Gas Turbines Power, **112**(1).

[5] Escher, P. C., 1995, "Pythia: An Object-Oriented Gas Path Analysis Computer Program for General Applications," Ph.D. thesis, Cranfield University, School of Mechanical Engineering, UK.

[6] Zwebek, A. I., and Pilidis, P., 2001, "Degradation Effects on Combined Cycle Power Plant Performance, Part II: Steam Turbine Cycle Component Degradation Effects," ASME Paper No. 2001-GT-389.